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Refrigeration Engineering

SECOND EDITION

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Preface

Ever since the publication, exactly thirty years ago, of a series of twenty articles on refrigeration, the name of H. J. Macintire has represented authority in this field of engineering. The present book, which in first edition was published thirteen years ago, is itself a successor to Professor Macintire's *Handbook of Mechanical Refrigeration*. Shortly before his death Professor Macintire discussed plans for a revised and expanded edition and invited me to share in its preparation. Unfortunately illness prevented him from participating actively in writing the new manuscript, but he discussed his ideas and plans for the new edition with me in some detail; hence I hope that this second edition will carry on the thought and the tradition of Professor Macintire's work in a way that he would have found acceptable.

Since this book has been widely used both as a college textbook and as a reference book for practicing engineers, the intent in preparing this second edition has been to increase its effectiveness for both these purposes. As a result of revision, modernization, and expansion, approximately one-half of the material in this edition is entirely new. Eight chapters have been added, over seventy additional illustrative figures are included, and there are over two hundred problems of which one hundred and sixty are new in this edition.

For use as a textbook the book has been strengthened by expansion of the section on reversed cycle theory (from two to eight chapters), by regrouping of the chapters to form four rationally organized sections on theory, load, equipment, and application, and, as already indicated, by inclusion of an unusually large number of representative problems for solution by the student. |

For use as a reference the book has been strengthened by the inclusion of new material in the following fields:

- (1) An extensive discussion (two chapters) of the analysis of transient phenomena in refrigeration is included. Such phenomena include pull-down, subcooling, and delayed heat-up through use of partial load emergency equipment.

- (2) Over thirty full-page graphical solutions are provided giving direct-reading values of the film coefficient of heat transfer for most of the

commonly used refrigerants when being heated or cooled as either a subcooled liquid or a superheated vapor.

(3) The heat pump is treated both from the standpoint of thermodynamic cycle analysis and from the standpoint of transient heat flow problems associated with energy sources and energy sinks.

(4) A chapter has been added covering the influence of radiation effects in determination of equilibrium cold storage temperature, and a detailed design procedure is included for comfort panel cooling systems.

Two of the chapters which have not been appreciably altered are the one on psychrometrics and the one on absorption cycles. Some rewriting of this material was originally contemplated, but many operating engineers indicated a preference for retention of the present treatment; hence no significant changes were made.

Although responsibility for the text must rest entirely with the authors, thanks are due to the following men for helpful comments and suggestions: Professors R. L. Daugherty, N. C. Ebaugh, T. T. Eyre, H. E. Keeler, H. W. Mason, J. W. Medlin, H. L. Nachman, B. F. Raber, H. J. Stoever, L. L. Vaughan. Thanks are also due to Kathleen and to Kay Hutchinson for assistance in many important though intangible ways.

F. W. H.

July, 1950

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Part I
THERMODYNAMICS OF
REVERSED CYCLES

CHAPTER I

ENERGY RELATIONSHIPS IN REFRIGERATION

The science of refrigeration is a branch of engineering thermodynamics and, as such, is largely concerned with energy in storage and in transition. Energy is intangible, but its storage occurs in a tangible body which, in the usual cycles of mechanical refrigeration, is known as the refrigerant. Because it is intangible, energy can neither be weighed nor counted, nor is there any other direct method of determining either the amount present or the availability of that quantity for transfer or for transformation to some other form. Thus a basic need, prerequisite to analysis of refrigeration cycles, is an understanding of the functional relationships among the measurable physical properties of the refrigerant which make possible the evaluation of both the quantity of energy present and its degree of availability.

Once these basic thermodynamic relationships have been recognized they will be observed to have in common the attribute of being point functions, that is, of being single-valued for any given condition or state of the refrigerant; for this reason all such functions can be, and usually are, regarded as "properties" of the refrigerant. From the overall standpoint, the refrigeration engineer makes use of thermodynamic functions for one of two purposes: (1) to evaluate an energy quantity; (2) to investigate the availability of energy for transfer to or from the refrigerant.

These concepts can be more readily visualized by explicit reference to the cycle of mechanical refrigeration. This cycle (see Fig. 1·1) consists essentially of four processes, two of which—those in the evaporator and condenser—are designed to alter the quantity of energy associated with the refrigerant, whereas the other two—in the compressor and the expansion valve—are designed to change the availability of associated energy. Owing to the combined influence of work input and thermal losses, some quantity change does occur in the compressor, but the intent of the process is to change the availability rather than the amount of energy present in the entering refrigerant. The compressor raises the availability of the energy stored in the refrigerant by an amount sufficient to allow it to flow out of the condenser, whereas the expansion valve lowers the availability of energy passing through it

by an amount sufficient to allow additional energy to flow into the evaporator. The cycle is a closed one in so far as the refrigerant is concerned, but although the quantity of energy associated with a given weight of refrigerant is always the same at a given point in the cycle the energy actually present is, of course, not the same energy as was present at any earlier time; thus energy is entering, flowing through, and passing out of the cycle, and the fundamental interest of the engineer in such a

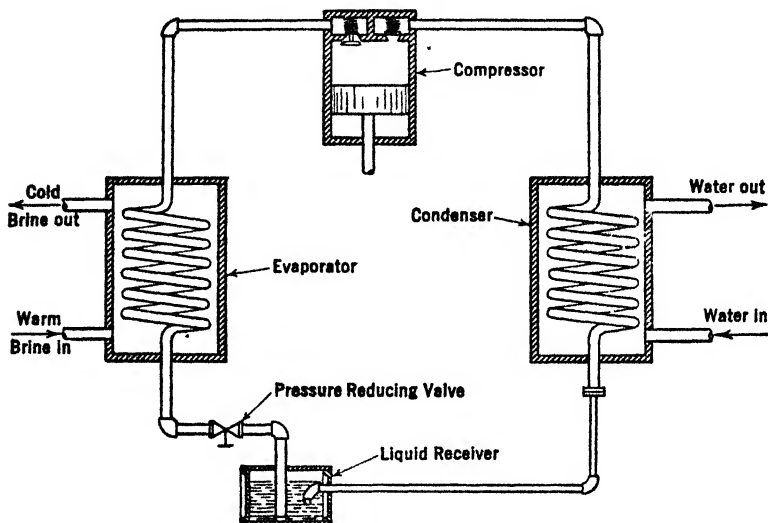


FIG. 1-1. The elementary refrigeration cycle.

cycle concerns the energy rather than the refrigerant. Convenience and practical necessity require that equations be written in terms of the physical properties of the refrigerant, but intelligent formulation and use of such equations requires that they be interpreted in terms of the energy rather than of the fluid.

MEASURABLE PROPERTIES OF THE REFRIGERANT

1-1. Statistical Method. Unlike the physicist, the engineer is primarily interested in the macroscopic rather than the microscopic properties of the materials with which he works. In a very rough sense one may say that his interest is statistical or relative rather than detailed or absolute. The designer of a cooling tower or the specialist in refrigeration compressors is aware that air and refrigerant vapor are, in the absolute sense, non-homogeneous fluids, that neither of them is truly a continuum; yet such knowledge in no way prevents him from

solving most of his practical problems by methods which presuppose homogeneity.

Air is unquestionably made up of individual molecules each of which possesses characteristics which differentiate it from all others; the velocities of no two molecules are exactly the same, either in direction or in magnitude; the specific volume in one microscopic region differs from that in another. Yet in spite of this multiplicity of variations the number of molecules, even in the smallest volume with which the engineer must work, is so great that individual idiosyncrasies are completely unimportant. Consider, for example, that 1 cubic inch of standard air contains not merely 444 billion molecules, but 444 billion billion molecules; if each molecule were the size of a pea this number would be sufficient to form a chain that would go 80 billion times around the world with enough left over to make a few detours to the moon and back again. Under such conditions it is not surprising that statistical methods can be safely used, or that working substances such as air, steam, and refrigerants can be treated as though they were continuous and as though they were continuums.

In the usual applications of refrigeration engineering there are no exceptions to the generalization of the above paragraph, but in scientific work requiring very low temperatures and very high vacuums occasions do arise on which the number of molecules in a small finite volume are reduced to so few that statistical methods, and consequently the macroscopic approach, must be abandoned. Such cases, however, are so few and so remote from the fields of usual refrigeration engineering applications that they need not be given further consideration here.

1.2. Pressure. Of the three measurable properties of a refrigerant, pressure is probably the one of greatest practical importance because it is the simplest to determine under field conditions and with inexpensive instruments. For a fluid at rest, pressure is a measurable property having units of force per unit area; for engineering purposes, gravitational units of pounds per square inch (psi) or pounds per square foot (psf) are ordinarily used. Pressure is characterized by the bursting effect exerted by a fluid on its container or by the collapsing effect exerted by a fluid on a hollow object immersed in it. The pressure exerted by a stationary fluid retained within an enclosure (as a cylinder or a pipe) may be in part of hydrostatic origin and in part due to the action of an external force on one or more of the fluid boundaries. The latter effect is by far the more important for almost all processes that are used in refrigeration, and in most engineering analyses of refrigeration cycles the hydrostatic effect is therefore neglected. Curiously, however, common usage has led to the convention of expressing refrigerant

pressures as though they were due to hydrostatic effects alone and identifying such pressures as equivalent "heads" (vertical columns of fluid) of refrigerant. Thus the normal gage pressure exerted by a refrigerant on the wall of a pipe can be considered equivalent to that which would be exerted on the same wall by a stationary column of fluid rising vertically above it and open, at the top, to the atmosphere.

A direct relationship between the head in feet and the pressure in pounds per square foot can be established only when the density of the fluid column is known and when it is unchanging over the height of the column. For such a case the gage pressure is given by the equation

$$P' = wh_s \text{ psf} \quad (1.1)$$

where P' is in pounds per square foot, w is in pounds per cubic foot, and h_s is the static head expressed in feet. The head produced by the blower in a cold air circulating system may be very small when expressed in pounds per square inch, but larger, relatively, than the head produced by a circulating pump when the comparison is in terms of head, in feet of fluid handled.

When pressure is evaluated by direct measurement the result is usually a gage pressure determined with respect to the atmosphere as zero. Thus the absolute or total pressure will in every case be the sum of the barometric pressure existing at the time and place of the observation and the gage pressure as read from the instrument. Normal barometric pressure (14.696 psia) is the pressure exerted at sea level by a 1-sq-in. column of standard atmosphere rising from the ocean surface through the stratosphere; otherwise considered, barometric pressure is equal to the weight of air in the column responsible for the pressure.

For a fluid at rest, pressure is a measurable property without energy significance. When considered with respect to a fluid in steady and continuous motion, however, it takes on additional importance and requires consideration as an energy term. Dimensionally and numerically pressure as a simple measurable property and pressure as an energy property appear identical, but the distinction is a vital one and misconceptions can lead to practical errors of importance. Much of this possible difficulty can be avoided if the reader will remember that, whereas pressure is always a simple property of the refrigerant, it has energy significance only under conditions for which the fluid is in steady and continuous motion.

Consider a unit volume of fluid flowing without friction through a pipe of cross-section area A and subject to constant absolute pressure P (psfa). The length of pipe occupied by this cylinder of fluid is then $L = 1/A$, and the force exerted on either end of the unit volume is

$F = PA$. The fluid within the pipe is in steady and continuous motion; therefore the work done by the upstream force in pushing the fluid through any cross section of the pipe is equal to the force multiplied by the distance through which it acts:

$$\text{Flow work} = (PA)L = (PA) \frac{1}{A} = P \text{ ft-lb/cu ft or psfa}$$

Thus the absolute pressure (in contrast to gage pressure) is seen to be numerically equal to the work in foot-pounds associated with unit volume of fluid; this is the *flow work* necessary to keep the fluid in steady and continuous motion.

One particular characteristic of pressure as an energy term is especially noteworthy. Although flow work is expressed in terms of foot-pounds per unit volume, one must remember that it is a manifestation of energy in transition; hence it represents energy passing through but *not stored*, even instantaneously, within the fluid. Thus at any cross section of the pipe the flow work there determined, if flow is without friction, is a quantitative statement of the amount of flow work entering the pipe at its upstream end, passing through the section in question, and *simultaneously* leaving the pipe at its point of discharge.

1.3. Specific Volume. Although the easiest of the physical properties of a fluid to visualize, specific volume \bar{V} is the most difficult of the measurable properties to evaluate under field conditions. Hence in most refrigeration analyses the specific volume will be determined indirectly as a dependent variable fixed in terms of known temperature and pressure rather than as a simple experimentally determined property. By definition the specific volume is the space occupied by unit weight of refrigerant, and its units are usually taken as cubic feet per pound; it is the reciprocal of the weight density expressed in pounds per cubic foot.

Neither density nor specific volume need be treated, for the purposes of engineering thermodynamics, as energy properties except in so far as greater density may mean a greater capacity for storing energy. In the field of refrigeration the actual working substance, matter, is something which receives, carries, or discharges energy but is itself not convertible into energy. For practical present-day engineering purposes this concept is always valid, but in the fields of physics and of armaments the respective laws of conservation of mass and conservation of energy do not hold, and mass itself is convertible, under conditions of atomic fission, into energy. Possibly atomic energy may play a future part in providing power for use in the refrigeration cycle; if so, all our present concepts with respect to energy usage will require revision, as the

atomic destruction of one gallon of gasoline would create as much energy as is now released by the combustion of approximately two billion gallons of gasoline. Atomic energy is not, however, of direct concern in refrigeration engineering, and for present purposes we can continue to regard matter as distinct from energy.

1.4. Temperature. As a measurable property, temperature is complex because it cannot be directly expressed in terms of the standard units of mass, length, and time. Many definitions of temperature have been proposed, but, though it is something with which everyone is familiar, its exact description defies simple and satisfying definition. One of the more common of the descriptive statements is that temperature is that property by virtue of a difference in which a transfer of energy will occur through the mechanism of heat. Another such statement is that temperature is the property which must have the same value for any two bodies which are in contact and in equilibrium; if two such bodies are not at the same temperature a transfer of energy will spontaneously occur until the temperatures have been equalized. This same condition has sometimes been expressed by the statement that temperature is indicative of the intensity of energy stored in a body and that temperature is therefore a property which is determined by the relative internal velocities which exist in a tangible body.

For engineering measurements the Fahrenheit temperature scale is used in the United States, but in thermodynamic analyses the absolute temperature ($^{\circ}\text{F} + 460$) must be used; throughout this text the symbol t will refer to $^{\circ}\text{F}$ whereas T will refer to $^{\circ}\text{F}$ absolute.

DEFINED (ENERGY QUANTITY) PROPERTIES OF THE REFRIGERANT

The fluid properties considered under this heading are mathematically defined as functions of one or more of the fundamental measurable properties. The energy quantity properties include: internal energy, enthalpy, specific heat, and latent heat of vaporization. Other energy quantity relationships exist and are important in the field of chemistry and in some branches of engineering, but they are unimportant to the refrigeration engineer.

1.5. Energy Units. Prior to the time when it was recognized that all manifestations of energy are related by a fixed law, various arbitrary energy quantities were selected and used independently of one another. Thus in discussing electrical effects the kilowatt-hour gained acceptance, whereas in dealing with energy manifestations as work (the action of a force through a distance) the common quantity unit was the foot-pound. For thermal effects—manifestations of energy as heat—the accepted unit was the British thermal unit (Btu), which is defined as

$\frac{1}{180}$ of the quantity of energy that must be added to 1 pound of water to change its temperature from 32° F to 212° F, or, with adequate engineering accuracy, the quantity of energy which must be added to 1 pound of water to raise its temperature 1 Fahrenheit degree.

As man's knowledge of energy relationships developed he learned that all the above quantity units were related in fixed ways and that a decrease of the store of energy in thermal form always resulted in a fixed increase in some other form. This observation has been generalized in the first law of thermodynamics, which states that energy can neither be created nor destroyed and, as a corollary, that the transformation from one form to another, when it occurs, is according to an unchanging relationship. Thus the various units of energy quantity can be related in equation form:

$$1 \text{ Btu} = 778 \text{ ft-lb} \quad \text{and} \quad 3413 \text{ Btu} = 1 \text{ kwh}$$

It is therefore evident that there is no genuine need for more than one defined unit of energy. The Btu or the foot-pound or the kilowatt-hour or any other arbitrarily selected unit would adequately suffice for expressing energy in all its forms and manifestations. It is equally as correct to express heat transfer in units of kilowatts as in Btu per unit time; in Europe this convention is commonly used.

1.6. Ton of Refrigeration. In the field of refrigeration a similar "local" energy unit has come into use through custom. One "ton" of refrigeration is, by definition, energy extraction at a rate of 200 Btu/min. Basically, therefore, the ton is a power unit and could equally well be expressed in terms of horsepower or kilowatts. Originally the ton had a physical meaning since it represented cooling at a rate sufficient to produce a ton of 32° F ice in 24 hr from water at an initial temperature of 32° F. Because of its widespread current use the ton will frequently be referred to throughout this text, but the reader should recognize that it is no more than a term from technical folklore and its use therefore represents a reduction in power from the more generalized energy units such as the Btu and the kilowatt-hour.

1.7. Internal Energy. The thermal energy possessed by a tangible body as a result of the velocity and relative positions of the molecules within that body is known as internal energy. The fraction due to molecular velocities is called the kinetic internal energy, and temperature is a quantitative criterion of its magnitude. The more rapidly the molecules are moving, irrespective of the relative insignificance of their mean free paths, the greater will be the temperature and the greater also will be the quantity of kinetic internal energy. For a gas no forces of attraction exist between the molecules; hence the energy due to random

velocities of the molecules constitutes the entire internal energy; for vapors, however, and for other fluids, the molecules possess an energy of position, sometimes called the potential fraction of the internal energy, which is not a straight-line function of temperature but an additive energy term which, together with the internal kinetic energy, constitutes the total internal energy u of 1 pound of the fluid.

The quantitative change of internal energy as a function of temperature is effectively expressed by means of the specific heat at constant volume c_v ; so long as the volume of a stationary fluid does not change, all the energy which is added to it must appear as an increase in the internal energy; thus variation of c_v with temperature is a direct indication of the extent of the departure from a straight line of the internal energy vs. temperature relationship. For most gases whose behavior approaches that of a perfect gas there is no potential fraction of the internal energy; hence for such cases the change of internal energy Δu can be expressed directly as a function of the specific heat at constant volume and of the change in temperature. Thus

$$\Delta u = c_v \Delta t \quad \text{Btu/lb} \quad (1.2)$$

1.8. Latent Heat of Vaporization. For most fluids, and this is true of all fluids used as refrigerants, a large change in the potential fraction of the internal energy occurs during a change of phase, as from the liquid to the vapor state. Thus when a pound of water vaporizes at constant temperature and pressure to a pound of steam the requisite energy input, known as the latent heat of vaporization h_{fg} , consists largely of a change in the quantity of potential energy stored within the system as a result of the different positions of the molecules with respect to one another. Part, however, of the energy required to achieve such a phase change is accounted for by energy supplied to the pound of fluid to do the external work needed to permit expansion against the pressure of the atmosphere. This fraction, frequently referred to as the external latent heat of vaporization, differs from the internal fraction in that it does not represent added energy going to storage within the fluid, but it is an energy quantity which passes through the fluid and goes into subsequent storage (in potential form) in the atmosphere.

Thus the latent heat of vaporization is not an energy quantity term with respect to storage, but it is with respect to transition since it does represent the quantity of energy which must be transferred to unit weight of fluid to achieve its evaporation, and it is similarly equal to the quantity of energy that must be transferred from such a weight of vapor during the process of condensation. The part of the latent heat which goes into storage as internal potential energy can be readily evaluated

from the equation

$$\Delta u_p = h_{fg} - \frac{P(\bar{V}_v - \bar{V}_l)}{778} \quad \text{Btu/lb} \quad (1.3)$$

where P is the absolute pressure (psfa) at which the phase change occurs, \bar{V}_l and \bar{V}_v are the specific volumes of the saturated liquid and saturated vapor, respectively, and h_{fg} is the latent heat of vaporization.

1.9. Specific Heat. In Section 1.7, the specific heat at constant volume was referred to as an energy quantity property directly related to the amount of internal energy stored within the fluid. Other forms of specific heat are also of importance, but none of them is an energy storage property and, except for the specific heat at constant pressure, other specific heats are not properties of the fluid since their values are not independent of path; that is, the generalized specific heat is not fixed for a given state of the fluid hence is not a function having an exact differential and cannot be regarded as a point function. There are an infinite number of process lines that can be followed in going from any fluid state to any other fluid state, and there must be, therefore, an equal number of terms representing specific heats.

For work in the field of refrigeration the specific heats at constant volume and at constant pressure are the only ones of practical importance. Specific heat at constant pressure c_p is somewhat like latent heat as an energy term, since it is equal to the quantity of energy that enters unit weight of fluid to achieve a stated effect (in this case to raise the temperature 1° F while the pressure remains constant), but it is not equal to the increase of internal energy of the fluid. Regardless of the process followed (hence of the particular specific heat applicable to the state change), the specific heat at constant volume is always indicative of the increase in kinetic internal energy and the increase of internal potential energy can always be determined from the equation

$$\Delta u_p = c_x - c_v - AW_x \quad \text{Btu/lb} \quad (1.4)$$

where c_x is the specific heat for any process x and W_x is the work done by 1 lb of the fluid due to expansion during the process x . The work term W_x can be evaluated only in terms of a known functional relationship between pressure and volume during the process in question.

1.10. Enthalpy. A combination fluid property which sometimes is an energy quantity property is obtained by adding to the internal energy the flow work associated with unit weight of fluid. The resultant function, enthalpy h , finds wide use in all thermodynamic analyses and is particularly valuable in treating cases of steady flow such as occur in refrigeration cycles. Since internal energy and flow work are usually

expressed in units of Btu and foot-pounds, respectively, it is necessary to convert one or the other before addition is possible. Thus

$$h = u + AP\bar{V} \quad \text{Btu/lb} \quad (1.5)$$

where A , ($\frac{1}{778}$), is the reciprocal of the mechanical equivalent of heat = $1/J$.

Although both terms of equation 1.5 have the dimensions of Btu, care must be exercised to determine for any particular application whether or not enthalpy is an energy quantity or no more than a numerical property, without energy significance, of the fluid. Since the term $AP\bar{V}$ represents flow work only during steady action of the fluid it follows that enthalpy will have energy significance only for a fluid in steady and continuous motion. Under any other condition the term enthalpy will remain a property of the fluid and will have the dimensions of energy, but it will not represent an energy quantity either stored in or associated with the fluid.

As an example of the above distinction consider 1 lb of fluid moving steadily through a pipe. The fluid is assumed to have an enthalpy of 1000 Btu and an internal energy of 900 Btu. If this fluid were discharged against constant pressure into a receiver and then brought to rest without external loss of energy, its total energy content would, by the first law of thermodynamics, have to remain exactly the same as it had been when in the pipe. Thus the energy which had been associated with the fluid in the pipe as flow work would necessarily be transformed to internal energy after the fluid had come to rest within the receiver. But, if the final receiver pressure equaled the pressure in the pipe and if the fluid density did not change, the $AP\bar{V}$ term, though no longer representing an energy quantity, would have the same numerical value that it had within the pipe and the enthalpy of the fluid within the receiver would therefore be

$$h_r = (u_p + AP_p\bar{V}_p) + AP_r\bar{V}_r = h_p + AP_r\bar{V}_r$$

$$h_r = (900 + 100) + 100 = 1100 \quad \text{Btu/lb}$$

where subscripts p and r refer, respectively, to pipe and to receiver. In this example the *property* enthalpy has undergone an increase of 100 Btu, yet the quantity of energy under consideration (1000 Btu) has not actually been changed.

In another respect, also, enthalpy is somewhat unusual as an energy term. Internal energy is stored within the fluid, but flow work is merely in transition through it, so the sum of stored energy plus energy in transition is something of an anomaly. Thus from a direct examination of the characteristics of the term enthalpy it would appear that it is

not well chosen. Actually, however, experience indicates that there are innumerable thermodynamic and hydrodynamic situations in which this property can be used to achieve substantial simplifications in analytical procedures, and it was with foreknowledge of these specific cases that the term was originally defined.

In the field of refrigeration the significance of enthalpy as an energy quantity term will be found to be of the greatest importance since all four of the basic processes of the mechanical refrigeration cycle can be analyzed in terms of this property. For a constant pressure process (such as occurs in both the evaporator and the condenser of a mechanical refrigeration system) the enthalpy change of the fluid can be shown to be equal to the rate of heat flow to or from the equipment. Thus, for the evaporator,

$$\text{Tons of refrigeration} = \frac{Q_e}{200} = \frac{w(h_e - h_c)}{200} \quad (1.6)$$

where Q_e is the rate of heat transfer to the evaporator expressed in Btu per minute, w is the refrigerant flow rate in pounds per minute, h_e and h_c are the respective enthalpies of unit weight of refrigerant at the exit and entrance of the evaporator. A similar equation can be written for the flow of heat from the condenser. Under certain conditions, which will be discussed later, the enthalpy change during passage of the refrigerant through the compressor is of fundamental thermodynamic importance, whereas for the expansion valve later analysis will show that the process followed is characterized by constancy of the enthalpy property.

Although enthalpy is not a measurable property, values of enthalpy corresponding to given conditions of temperature and pressure are available in tables and charts of the thermodynamic properties of all common refrigerants; hence it is unnecessary for purposes of practical cycle analysis to seek an explicit functional relationship between enthalpy and the measurable properties. Values of the internal energy are not usually given in charts and tables, but they can be calculated readily from a knowledge of the simultaneous values of enthalpy, pressure, and specific volume.

SYSTEM ENERGY QUANTITY PROPERTIES

The energy properties considered in the preceding sections are usually thought of as belonging to the body in question rather than to the more extensive system of which it is a part. Actually, however, they do refer to the system since both flow work and internal energy are measured with respect to differences between the temperature of the body and the surround or to differences between the pressure of the body and the

surround. In each case, however, a change in the energy quantity associated with or stored in the fluid must necessarily result in a change of one or more of the measurable physical properties: pressure, temperature, and specific volume. Two additional energy quantities require consideration, but neither of them can be identified in terms of the measurable properties; hence they are not point functions in the sense of having fixed values corresponding to a given internal state of the fluid, but neither are they characterized by inexact differentials since their respective values corresponding to a given state of the system are fixed and independent of the path by which the system reached that state. In a broad sense these two system properties are the tangible counterparts of the intangible fractions of internal energy. Whereas the sum of *internal* potential and kinetic energies is defined as internal energy, the sum of *external* potential and kinetic energies is defined as mechanical energy.

1.11. External Potential Energy. This fraction of mechanical energy is represented by difference in elevation. Measured with respect to sea level, any object possesses an external potential energy equal in foot-pounds to the product of its weight and the vertical elevation of the center of gravity of the object above the level of the sea. Obviously, however, such an energy quantity will not be of practical significance in the field of refrigeration, at least in absolute terms, because for most cases with which the engineer is concerned there will be no possibility of the object in question (usually a pound of refrigerant) being restored to sea level and hence there will be no opportunity for extracting the external potential energy which it contains.

In a relative sense, however, changes in energy quantity due to differences of elevation do occur in almost every process of interest to the refrigeration engineer; hence it is essential that he determine when and to what extent such changes will be of a magnitude sufficient to influence the practical energy relationships of the system. This condition can be examined by visualizing the extreme case of a refrigeration cycle in which differences in elevation of the piping amount to 77.8 ft; the change in external potential energy associated with such a change of elevation would then be 77.8 foot-pounds per pound of refrigerant or 0.1 Btu/lb. Since the combined changes in internal energy and in enthalpy which occur during the important processes of the cycle are of the order of 10 to 100 or more Btu per pound of refrigerant, it follows that even for this extreme case the effect of change in external potential energy would be of the order of one-tenth of 1 per cent. Thus, in the thermodynamic analysis of refrigeration cycles, changes in the energy of the system due to elevation can always be neglected.

1·12. External Kinetic Energy. Just as the molecules within a given volume of stationary refrigerant vapor are moving with varying velocities and thereby imparting internal kinetic energy to the fluid, so also are small tangible masses of the material moving at varying velocities in the general direction of transport. For any small element of mass, moving uniformly with velocity U (feet per second), the kinetic energy is given by the equation

$$\text{K.E.} = \frac{U^2}{2g} \text{ ft-lb/lb} \quad (1\cdot7)$$

In order to determine the kinetic energy of the entire stream, as for flow of a refrigerant through a pipe, precise information on the velocity distribution over a given cross section normal to the direction of flow is essential; a knowledge of velocity distribution is also needed to permit determination of the velocity gradients and hence of the viscous forces resulting from tangential stresses during flow.

If flow were to occur through a pipe of circular cross section with uniform velocity across the area normal to flow, the kinetic energy would be given by an equation of form similar to that above with the velocity evaluated by the equation

$$U = \frac{V}{A} \text{ ft/sec}$$

where V is the volume of fluid, in cubic feet, passing a cross section of area A in 1 sec. This equation is obviously true only when the velocity is uniform, though in other cases it can be used to determine a mean velocity \bar{U} across the section. When this is done, however, a major difficulty arises since the square of the mean velocity can differ widely from the mean of the squared point velocities. Thus, the kinetic energy of the fluid in the pipe is

$$\text{K.E.} = \sum_A \frac{U^2}{2g} \neq \frac{\bar{U}^2}{2g} \text{ ft-lb/lb}$$

Two possible methods of solution are available. The first and most logical would be to determine accurately a mean value of U^2 and use it in the above equation. The second, and the method most commonly used in practice, is to evaluate the mean U^2 in terms of the mean velocity squared, giving

$$U^2 = K\bar{U}^2 \quad (1\cdot8)$$

where the value of the dimensionless coefficient K will depend on the velocity distribution over the cross section; the value of K for stream-

line flow is 2, whereas for fully developed turbulent flow such as occurs in most refrigerant piping the K value is unity. For flow under conditions of limited turbulence the value of K varies from 2 to 1, and its exact evaluation is difficult. Fortunately, however, flow in this transition region is not important in refrigeration; hence the value of unity can be safely used in all practical problems.

The kinetic energy possessed by unit volume of refrigerant can be readily obtained by dividing both sides of the above equation by the specific volume:

$$\text{K.E.} = \frac{K\bar{U}^2}{2g\bar{V}} \quad \text{ft-lb/cu ft}$$

Then, noting that $g\bar{V}$ is the reciprocal of the mass density ρ , we obtain

$$\text{K.E.} = \frac{1}{2}\rho K\bar{U}^2 \quad \text{ft-lb/cu ft} \quad (1.9)$$

which is in units of foot-pounds per cubic foot but has the dimensions of pounds per square foot; hence it is dimensionally equivalent to pressure. This term is equivalent to pressure in another sense also since it indicates that the unit volume of fluid has stored (with respect to the stationary earth) an amount of kinetic energy sufficient to raise itself, during deceleration, to a vertical height equal to $K\bar{U}^2/2g$ feet. Thus $\frac{1}{2}K\bar{U}^2\rho$ can be defined as a velocity pressure, in pounds per square foot absolute, whereas $K\bar{U}^2/2g$ is a velocity head, in feet of fluid.

1.13. The Bernoulli Equation. One of the most widely used applications of the external kinetic energy is in Bernoulli's equation. Thus, if flow occurs without friction and if no energy exchange takes place between the incompressible fluid and the pipe wall, the sum of the flow work and of the kinetic energy, per unit volume of fluid, must remain constant. This requirement leads to the equation

$$P + \frac{1}{2}K\bar{U}^2\rho = \text{constant} \quad (1.10)$$

from which comes the equation

$$h_s + h_v = h_t \quad (1.10a)$$

which states that the sum of the static head and of the velocity head is equal to the total head.

For most problems in the field of refrigeration the Bernoulli equation is invalid since the conditions of incompressibility and of no external energy exchange do not hold. The requirement of frictionless flow is never fulfilled in practice, and energy is frequently either extracted from or added to the fluid during passage through the pipe. In fact, for the greater number of processes which occur in refrigeration cycles

the velocity-head term of the Bernoulli equation is the least important of all energy terms. As an example, consider a fluid passing through a pipe with the relatively high velocity of 10 fps. The external kinetic energy associated with unit weight of this fluid is $10^2/(2 \times 32.2) = 1.55$ ft-lb/lb or 0.002 Btu/lb, which is so small, compared with other energy quantities with which the refrigeration engineer is concerned, as to be entirely negligible in all cases except that of steam jet systems.

1-14. Mechanical Energy. The sum of external potential and external kinetic energies is frequently referred to as the mechanical energy, that is, the energy of the system of tangible bodies in contrast with the energy of the system of intangible bodies (molecules) which is called internal energy. From the preceding examples of the magnitudes of external potential and external kinetic energies, it is evident that their sum, mechanical energy, will likewise not be of a magnitude sufficient to require consideration in the solving of practical problems in refrigeration.

NON-PROPERTY ENERGY FUNCTIONS

All the energy functions considered in the preceding sections dealt with energy quantities related either to storage or to transition, but under conditions such that the function could mathematically be treated as a property and would possess an exact differential. In all such cases the magnitude of the function was determined by the state of the fluid and was independent of the history of the fluid prior to reaching that state. For manifestations of energy as heat or as work, however, the situation changes, and although energy functions can be established they will not possess exact differentials and will not be point functions, hence cannot be considered properties of the fluid.

For usual engineering purposes the addition or removal of energy in any system must occur—provided that there is not energy transfer by mass movement of the fluid itself—either as some form of work or as some form of heat. In the thermodynamic sense heat and work are external effects resulting from changes within a system; that which causes these effects is called *energy*. Each of these manifestations of energy occurs by various mechanisms, but all except flow work are characterized by the requirement that quantitative evaluation requires knowledge not merely of the initial and final states of the fluid, but also of the path followed during the thermodynamic process which was responsible for the change of state.

1-15. Work. Both work required to maintain steady and continuous flow and work accomplished during constant pressure expansion have received previous consideration. Other forms of work are those repre-

sented by energy transfer through a shaft (as in the case of the refrigeration compressor) and work done during an expansion process in which the pressure is not constant.

Shaft work is the most highly desired of all energy manifestations since it can very readily be transformed to any other form of energy in transition or in storage. Shaft work is added to a fluid passing through a centrifugal fan, a compressor, or an agitated mixing tank and is extracted from fluid during passage through turbines or expansion engines. For convenience in writing equations which include a term for shaft work the symbol W will be used to represent such work in units of foot-pounds, whereas AW will identify shaft work expressed in thermal units.

The general equation for work during a reversible expansion is

$$W_{\text{out}} = \int_{V_1}^{V_2} P dV \quad \text{ft-lb} \quad (1.11)$$

Evaluation of the integral obviously requires knowledge of the path over which the integration is to be made and, specifically, of the functional relationship between P and V which exists along that path. For processes of the kind that occur in refrigerating equipment the only kind of non-isobaric path which is of concern is the so-called *polytropic*, which is defined as a process in which the pressure-volume function is of the form

$$PV^n = \text{constant} \quad (1.12)$$

where the exponent n depends on the conditions under which the process occurs.

In many practical refrigeration problems the value of the polytropic exponent will be known from previous experience with the equipment in question, but for cases where such experience is not available the value of the exponent can be determined by calculation from the defining equation for any polytropic process which occurs between known initial and final fluid states. Then, with known initial values of P_1 and V_1 and with known n , the work during the compression process can be expressed as

$$W_m = -P_1 V_1^n \int_{V_1}^{V_2} \frac{dV}{V^n} = \frac{w(P_2 \bar{V}_2 - P_1 \bar{V}_1)}{n - 1} \quad \text{ft-lb/min} \quad (1.13)$$

If the working substance is a perfect gas, equation 1.13 can be rewritten in terms of temperature as the measurable property by noting that, from the perfect gas law, $PV = wRT$ (where R is the gas constant); hence

$$W_m = \frac{wR}{n - 1} (T_2 - T_1) \quad \text{ft-lb/min} \quad (1.13a)$$

which shows clearly that the work done during a polytropic compression of a perfect gas between the temperature limits T_1 and T_2 must be constant (for a fixed value of n) regardless of the initial pressure of the gas. This fact is of considerable importance and will be referred to again in the subsequent discussion of reversible refrigeration cycles.

An *isentropic* process is defined as one which occurs reversibly and without transfer of heat to or from the refrigerant. This type of process is of very great importance in the analysis of refrigeration cycles since it is approached in practice by the compression process which occurs in a cylinder when heat losses (or gains) between the cylinder and the surround are very small. Furthermore the concept of isentropic compression is helpful since, as will

be shown, the energy required with such a process will usually be in excess of that needed for most actual compression cycles; hence calculations of energy requirements based on the isentropic will be conservatively large. Since analysis of isentropic compression can be carried out more rapidly than that for any other compression process it follows that this process provides a simple, convenient, and conservative method for approximating the power of real compression processes.

Thermodynamic analysis of the isentropic process readily demonstrates that the work of isentropic compression is given by the same equation as that for the polytropic process, but with the exponent fixed at a value k which, for a perfect gas, is numerically equal to the ratio of specific heats at constant pressure and at constant volume.

However, for any real equipment, like a compressor, the work term will include the external energy effects associated with introduction of the refrigerant into the compressor and with its subsequent withdrawal following the compression process. During introduction, the refrigerant is admitted to the cylinder under constant pressure (see Fig. 1·2), but this can hardly be called a thermodynamic process since there is no change of state of the refrigerant but rather a change in the mass of vapor present in the cylinder. The energy associated with admission is therefore related to the mass movement of the refrigerant rather than to a change in either the quantity or quality of energy stored in the vapor.

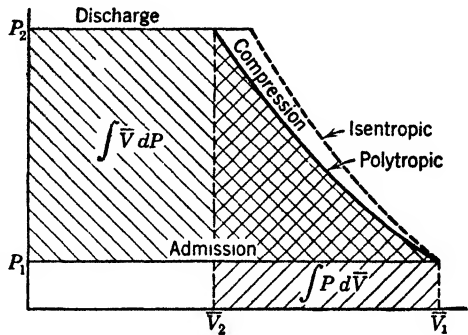


FIG. 1·2.

Admission occurs without change in the numerical value of the enthalpy but with an important change in the significance of the enthalpy term. Prior to admission the enthalpy of the refrigerant is equal to the sum of energies stored in or associated with (excluding mechanical energy) 1 pound of material, but during the admission period the flow work associated with the entering refrigerant is usefully extracted by the piston and thereby transformed to shaft work done by the entering vapor; when the piston reaches the end of the cylinder, and the cylinder is then full of stationary gas, all flow work will have passed out of the cylinder and, although the numerical value of the enthalpy will remain unchanged, this number will now correspond to a non-energy property. The useful work done by 1 pound of entering refrigerant will then be equal to the product of its pressure and its specific volume:

$$\text{Work done during admission} = P_1 \bar{V}_1 \text{ ft-lb/lb} \quad (1.14)$$

where \bar{V}_1 is the specific volume of refrigerant in a zero-clearance cylinder at the time compression starts.

Similarly the work done on the gas during the constant pressure discharge process is $P_2 \bar{V}_2$, where \bar{V}_2 is the specific volume in the cylinder subsequent to compression. Thus the net work of the cycle is the sum of the works done on the gas during compression and discharge less the work returned by the gas during admission, or

$$\text{Work of cycle} = P_2 \bar{V}_2 + \frac{P_2 \bar{V}_2 - P_1 \bar{V}_1}{n - 1} - P_1 \bar{V}_1 \text{ ft-lb/lb} \quad (1.15)$$

Simplifying and grouping terms,

$$\text{Work rate of cycle} = \frac{w(P_2 \bar{V}_2 - P_1 \bar{V}_1)}{n - 1} n \text{ ft-lb/min} \quad (1.16)$$

and, for a perfect gas,

$$\text{Work rate of cycle} = \frac{(T_2 - T_1)wRn}{n - 1} \text{ ft-lb/min} \quad (1.16a)$$

Hence it appears that the work of a polytropic or isentropic compression cycle, for a perfect gas and for fixed value of n , is a function only of the temperature difference and is unaffected by the absolute value of the temperature of either source or receiver.

By examination of Fig. 1.2 it appears that the difference between work of the process and work of the cycle corresponds to the difference between the integrals $\int P dV$ and $\int V dP$; in almost all problems in

refrigerating engineering, excepting only that of dual-effect compression, the work term of interest is that of the cycle rather than of the process.

1-16. Heat. Energy manifestations as heat may occur by different mechanisms. The first of these, *conduction*, depends on variations in the state of molecular agitation within the macroscopic volume of fluid under consideration.

The second mechanism of heat transfer, *radiation*, differs from conduction in that it does not presuppose a substance through which the energy passes. Radiation occurs from the surface of the material in question, and transfer occurs through space at the velocity of light. Radiant transfer is extremely important in many refrigeration problems, particularly those relating to cold storage, and failure to understand the equations which evaluate the radiant transfer rates may lead to serious misconceptions concerning the ability of a given environment to maintain an enclosed object at a specified temperature.

Energy in transition due to interchange of masses of fluid which are at different temperatures is frequently referred to, though the expression is something of a misnomer, as heat transfer by *convection*. Such an interchange is actually characterized by a combination of two processes: mass exchange of fluid, with subsequent leveling of the internal energies of contiguous elements of fluid by the mechanism of conduction.

For purposes of thermodynamic analysis of refrigeration cycles it is necessary to know and use the expressions for evaluating work, but those which evaluate rates of heat transfer are unnecessary because the quantity (in contrast to the rate) of energy transfer as heat can be determined from an overall study of the state changes experienced by the refrigerant during the process change. Thus discussion of heat transfer rates will be postponed to separate chapters (IX to XIII) where the necessary equations will be developed and applied in the design of equipments to carry out specified cycles. To summarize, heat quantity is important in cycle analysis, whereas heat transfer rate is important only in equipment design.

The terms *heat source* and *heat sink* find wide use in the thermodynamic literature and will be used continually in the refrigeration cycle analyses which follow. It is important to recognize that, as used herein, neither of these terms refers to a body containing heat, but rather to a reservoir of internal energy from which heat flow occurs, in the case of the source, or to which it flows, in the case of the receiver. In all cases the reservoir is visualized as possessing a thermal capacity sufficiently great so that the withdrawal or addition of energy in the quantities important for cycle purposes will have no measurable influence on the

temperature; thus it is visualized that no temperature gradient exists within the boundaries of the reservoir, but that the drop in temperature occurs exterior to the reservoir boundary; in some cases—as, for example, the reservoirs used in the Carnot cycle—it is further assumed that no measurable gradient exists exterior to the boundary. In the latter case, with an infinitesimal temperature drop, it is evident that an infinite time would be required for the small thermal potential to drive into or out of the reservoir a finite quantity of heat.

AVAILABILITY; SECOND LAW; ENTROPY

The second law of thermodynamics refers to the observed fact that a transformation of energy, as from heat to work, will occur only in a system that is not in equilibrium. Furthermore the second law provides a basis for investigations of the extent to which a given quantity of thermal energy, as heat, can be transformed to mechanical energy, as work. Some thermodynamic property is needed which will indicate the relative availability of the energy associated with the refrigerant when in a given state, hence will permit evaluation of the tendency of that energy to undergo transformation spontaneously. A property which serves this purpose is defined by the equation

$$\Delta s = \int \frac{dQ}{T} \quad (\text{applicable only to reversible processes})$$

which indicates that a change of s , specific entropy, is related to flow of thermal energy Q and to the intensity of kinetic internal energy as given by the absolute temperature T ; entropy increases during any process in which there is a loss of availability.

1.17. Significance of Entropy. The absolute value of the entropy property is not of practical importance in refrigeration engineering (it does have significance in some chemical analyses connected with absorption refrigeration, but not from the standpoint of practical operation or engineering analysis of the system), but knowledge of changes in entropy will be extremely useful in compressor power calculations. Entropy increase is not merely a qualitative indication of greater unavailability, but it can be used to evaluate numerically the quantity of energy that will be unavailable for any given reversible cycle. Thus, referring to the Carnot cycle (Section 2.3), one can write

$$Q_2 = T_2 \frac{Q_1}{T_1}$$

Since Q_1 is the quantity of heat entering the fluid during the high-temperature isothermal expansion (for heat engine operation), the term

Q_1/T_1 must be equal to the change in entropy of this quantity of energy as it passes through the cycle; therefore

$$Q_2 = wT_2 \Delta s \quad \text{Btu/min} \quad (1.17)$$

which shows that the energy unavailable after completion of the cycle will be equal to the entropy change multiplied by the absolute temperature of the low-temperature heat sink.

In the field of refrigeration the entropy concept finds wide use in three respects:

(1) The isentropic or constant entropy process has already been defined as a polytropic which takes place without gain or loss of heat. Now, in terms of the entropy concept, it is evident that the isentropic is a process in which thermal energy passes between temperature levels without loss of availability. This constancy of entropy during compression provides a simple means of portraying graphically the thermodynamic state changes that occur to the refrigerant as it passes through the compressor and also provides a criterion for use in determining the state of the gas at compressor discharge.

(2) By fortuitous circumstance most polytropic processes that occur in practice can be approximately represented as a straight-line variation of entropy with temperature; this being so, the change in entropy during compression multiplied by the arithmetical average of initial and final process temperatures will be approximately equal to the energy transferred between the cylinder and the surround:

$$\text{Cylinder heat loss} = \frac{1}{2}(T_1 + T_2)(s_1 - s_2)w \quad \text{Btu/min} \quad (1.18)$$

(3) The third important use of entropy is as a coordinate in plotting properties of refrigerants or in graphically representing cycles. When used with temperature a coordinate system is obtained on which the area under any reversible process will correspond to the quantity of heat entering or leaving the system during that process; an isentropic will, of course, be represented by a vertical line showing that no transfer of heat occurs.

PROBLEMS

1. A fluid having a density of 42 lb/cu ft is subjected to a pressure of 80 psig. Determine (a) the head in feet of the fluid and (b) the head in feet of water.

2. A fluid under pressure of 80 psig is flowing through a pipe under conditions of steady and continuous motion. Determine the flow work associated with the fluid, per unit volume.

3. A fluid having a density of 36 lb/cu ft is subjected to a pressure of 30 psia and is flowing under conditions of steady and continuous motion. Determine (a) the flow work in foot-pounds per pound and (b) the flow work in foot-pounds per cubic foot.

4. Heat flow through a wall is at a rate of 10,000 Btu per $1\frac{1}{2}$ hr. Determine (a) the rate of heat loss in horsepower and (b) the rate of heat loss in kilowatts.
5. Calculate the tons of refrigeration needed to freeze 2000 lb of 50° F water to ice at 0° F in 8 hr. (Take the specific heat of ice as 0.5.)
6. A fluid having a specific heat of 0.7 and a latent heat of fusion of 130 Btu/lb is to be cooled from 60° F to its freezing temperature (16° F), frozen, and subcooled to -15° F. The specific heat of the solid is 0.3. Calculate the rate of refrigeration needed to process 1000 lb of this fluid in 3 hr.
7. During the process of freezing, a certain liquid undergoes increase in specific volume from 0.030 to 0.039 cu ft/lb. If freezing occurs at atmospheric pressure and if the latent heat of fusion is 100 Btu/lb determine the change in internal energy. Is the enthalpy change during this process greater or less than the change in internal energy? Why?
8. Determine the change in internal energy of 1 lb of liquid ammonia as it is heated while saturated from 60° F to 100° F. Is the corresponding enthalpy change greater or less? Why?
9. A certain fluid has a specific heat at constant volume of 0.45. This fluid is heated from 60° F to 120° F under constant pressure of 2 psia, and the corresponding average value of c_p is found to be 0.35. Determine the change in specific volume that occurs during the heating process. Is the fluid expanding or contracting during the heat addition?
10. A fluid having $c_v = 0.20$ is heated under conditions such that the average specific heat for the process is 0.27. Calculate the external work done by the fluid as 1 lb is raised in temperature from 15° F to 65° F; express the result in foot-pounds per pound.
11. Determine the enthalpy increase (from Steam Tables) as 1 lb of water is heated from 32° F to 212° F at atmospheric pressure. Determine what per cent of this increase is energy that does *not* go into storage in the water.
12. A fluid having an internal energy of 700 Btu/lb and an enthalpy of 800 Btu/lb is flowing steadily and continuously in a pipe. If this fluid enters a receiver what will be the respective values of its internal energy and enthalpy after it has come to rest? (Assume no change in P or \bar{V} .)
13. Thirty pounds per minute of a refrigerant undergo an enthalpy increase of 20 Btu/lb during passage through an evaporator. Calculate the refrigeration load, in tons, carried by the evaporator.
14. Twenty thousand pounds of a food product having specific heat of 0.75 are to be cooled from 80° F to 40° F in 2 hr. Calculate the resultant load in tons of refrigeration.
15. A fluid flowing turbulently through a pipe has a velocity of 1000 fpm. If flow occurs to a receiver located 20 ft lower than the pipe calculate the part of the resultant internal energy increase which is due to transformation of external potential and external kinetic energies.
16. For a given value of mean fluid velocity is the external kinetic energy greater or less for turbulent than for laminar flow?
17. A fluid under conditions of turbulent flow has a mean velocity of 2000 fpm and a weight density of 50 lb/cu ft. Determine the external kinetic energy in foot-pounds per cubic foot.
18. A vapor at 60° F is compressed polytropically (exponent = 1.28) until its temperature is 130° F. (a) Calculate the work of the *process* in foot-pounds per pound ($R = 30$). (b) Calculate the work of the *cycle* in foot-pounds per pound.

(c) If P_1V_1 is twice as great as P_1V_1 determine the work done by the gas during the admission process, in foot-pounds per pound. (d) If $P_2 = 30$ psia determine the specific volume of the fluid at discharge.

19. Calculate the change of entropy as 1 lb of a liquid at 16° F freezes to a 16° F solid; take the latent heat of fusion as 170 Btu/lb.

20. Twenty pounds per minute of fluid undergo an entropy change of 10 units per pound as the fluid goes through a Carnot cycle which operates between temperatures of 100° F and 300° F. Calculate the flow of heat to the receiver.

CHAPTER II

ENERGY EQUATIONS AND IDEAL CYCLES

2·1. Refrigeration Cycle (Non-flow) Energy Equations. The first law of thermodynamics permits combining the energy-property terms and the energy-in-transition terms to obtain general statements concerning overall energy relationships for any thermodynamic system. Considering the refrigeration cycle as a whole it is possible to write, for example, a simple energy equation stating that the sum of the entering streams of energy must be equal to the rate at which energy is leaving the cycle. Thus

$$Q_e + AW_c = Q_c' + Q_c \quad (2·1)$$

which states mathematically that the heat entering the evaporator plus the thermal equivalent of the work supplied to the compressor must be dissipated as heat from the condenser and the compressor (losses or gains in the piping being neglected).

Equation 2·1 is strictly true only when the system is in equilibrium, hence is undergoing neither a gain nor a loss in stored energy. If, for example, the compressor were equipped with a water jacket, equation 2·1 would be applicable for equilibrium operation either with the jacket in operation or not in operation, but it would not be applicable for the transient interval that would necessarily follow the starting of flow through the jacket or of discontinuing flow through it. An equation of the above form is called a non-flow equation since it deals entirely with a system in which there is neither admission nor discharge of matter; note, however, that, although no refrigerant is flowing, nonetheless each term of the equation refers to a definite flow of energy.

A more general form of the non-flow simple energy equation is one including a term for the increase of stored energy within the system. Thus, if an energy balance is being written on a system containing w pounds of a fluid which undergo a change of internal energy, the new form of equation 2·1 is

$$Q_{in} + AW_{in} = Q_{out} + AW_{out} + w \Delta u \quad (2·2)$$

where $w \Delta u$ is the increase of internal energy of the system. This equation is rarely of importance in refrigeration problems, hence it will not be given further attention.

2.2. Refrigeration Equipment (Flow) Energy Equations. Analysis of the processes carried out in any element of a refrigeration system requires an energy equation which will include terms to account for the energy carried through the equipment in question in the stream of refrigerant. Since all usual refrigeration processes are continuous (exceptions may occur in special applications as in some types of adsorption systems) and since analyses will always be for steady-state conditions, there is no need for inclusion of storage terms, and an energy balance can therefore be written by equating the energy input to the output. Since, previously discussed, mechanical effects are insignificant in refrigeration processes, terms need not be included for external potential or kinetic energy; with this exception the general energy equation for refrigeration applications is the same as the general thermodynamic energy equation and can be written

$$Q_{in} + AW_{in} + wu_{in} + wAP\bar{V}_{in} = Q_{out} + AW_{out} + wu_{out} + wAP\bar{V}_{out} \quad (2.3)$$

By using the enthalpy function and noting that work is never done by a piece of refrigerating equipment (with the exception of the expander used with the air cycle) this equation can be simplified to the form

$$AW_{in} = w(h_{out} - h_{in}) + Q_{out} \quad \text{Btu/min} \quad (2.4)$$

In no equipment is there a simultaneous flow of heat both in and out; hence by including only Q_{out} in this equation the case of an equipment which receives heat (as the evaporator) can be handled by noting that Q_{out} is then negative.

Like all the energy equations, equation 2.4 is merely a statement of the first law of thermodynamics; hence it can be used to determine what overall relationships will exist among various kinds of energy involved in a transformation if the transformation occurs. The question of whether or not a transformation will take place must be answered from the availability relationships that come from a consideration of the second law of thermodynamics. This fact can be visualized most clearly by imagining a process which occurred without flow and with heat transfer in, but not out. Equation 2.4 would then take the form

$$AW_{out} = Q_{in} \quad (2.5)$$

which satisfies the requirements of the first law of thermodynamics but states that heat can be completely converted to work; experience indicates that equation 2.5 could represent a true process only if the thermal transfer were to a receiver at a temperature of absolute zero.

Thus, useful though equation 2·4 is, its application to practical refrigeration problems requires that it be given direction, and this can be done only by applying to it the limitations implicit in the second law of thermodynamics.

Application of equation 2·4 to the process followed in an expansion valve indicates that expansion occurs without change of enthalpy; throughout the field of refrigeration the expansion valve is the only equipment which follows an isenthalpic process.

For a refrigeration system operating without pressure or heat losses (or heat gains) in the connecting piping, the special forms of the general energy equation can be grouped and written in terms of three enthalpies: h_s , h_d , and h_c , which correspond to the respective states of the refrigerant at compressor suction, compressor discharge, and condenser discharge. When these enthalpies are used, the special forms of the general equation are as shown in equations 2·6 through 2·10.

$$\text{Evaporator:} \quad Q_{\text{in}} = w(h_s - h_c) = 200T \text{ Btu/min} \quad (2\cdot6)$$

where w is the flow rate in pounds per minute and T is the load on the system in tons. The enthalpy difference $h_s - h_c$ is sometimes called the refrigerating effect, and for any refrigerating cycle equation 2·6 can obviously be used in solving for the refrigerant flow rate.

For the compressor the energy equation is similar in form to equation 2·4, but it can best be written

$$\text{Work}_{\text{compressor}} = w(h_d - h_s) + Q_{\text{out}} + Q_f \text{ Btu/min} \quad (2\cdot7)$$

where Q_f represents heat leaving the cylinder as a result of mechanical friction. The two terms for heat out are not combined because they have decidedly different effects on the performance of the compressor. Heat dissipation due to reduction in internal energy of the refrigerant (as by water-jacketing) is beneficial and serves to reduce work, whereas heat dissipation due to mechanical friction is always responsible for a unit-for-unit increase in the work of compression. The Q_f term is indicative of mechanical ineffectiveness and cannot be evaluated in terms of the properties of the refrigerant, whereas the Q_{out} term represents a transfer of energy from the refrigerant while it is undergoing a reversible (frictionless) process; hence Q_{out} can be evaluated in terms of fluid properties once the conditions of the process have been established.

From a cursory examination of equation 2·7 one might deduce that the work required would increase as the rate of heat loss increases, but experience shows that this is by no means the case and that the more heat lost from a compressor (that is, the more effective the water-jacketing or other cooling) the less will be the work requirements. Thus experi-

ence indicates that whenever the Q_{out} term on the right side of equation 2·7 is increased there will occur a simultaneous decrease of the enthalpy term and this decrease will be in excess of the change in Q_{out} .

For a polytropic compression the work term of equation 2·7 can be evaluated in terms of the fluid properties and the Q_{out} term then solved for, if Q_f is known or is negligible. A method of approximate solution is to substitute for Q_{out} from equation 1·18:

$$\text{Work}_{\text{compressor}} = w(h_d - h_s) + Q_f + \frac{1}{2}(t_s + t_d + 920)(s_s - s_d) \text{ Btu/min} \quad (2\cdot8)$$

which permits direct and relatively simple computation of the power requirements, without need of solving an exponential equation and from properties of the refrigerant at initial and final states of compression.

For an ideal compressor operating on an isentropic compression process equation 2·8 simplifies to the form

$$\text{Work}_{\text{compressor}} = w(h_d - h_s) \text{ Btu/min} \quad (2\cdot9)$$

and it is in this form that the equation finds greatest application.

The simplified equation for the condenser takes the form

$$Q_{\text{condenser}} = w(h_d - h_c) \text{ Btu/min} \quad (2\cdot10)$$

and if Δt is the temperature rise of the cooling water passing through the condenser the required flow rate of water can then be determined as

$$w' = \frac{w}{\Delta t} (h_d - h_c) \text{ lb/min} \quad (2\cdot11)$$

where w' is in units of pounds per minute and the heat removed by the water is that given up by the refrigerant. Since 1 gal is approximately equal to 8.3 lb it follows that the required cooling water rate expressed in gallons per minute is

$$\text{gpm} = \frac{w}{8.3 \Delta t} (h_d - h_c) \quad (2\cdot11a)$$

No equation need be written for the expansion valve since the process occurs without change in enthalpy.

For the system as a whole a fifth equation indicates that the total energy discharge from the condenser must be equal to the sum of the energies received from the evaporator and the compressor. This equation is not of particular importance in carrying out refrigeration cycle analyses, but it does serve as an effective reminder that the load on the condenser must necessarily exceed that on the evaporator by an amount

equal to the heat equivalent of the shaft work introduced during the isentropic compression cycle.

2.3. **Coefficient of Performance; the Carnot Cycle.** When the Carnot thermodynamic cycle is operated in reverse and visualized as a method of obtaining mechanical refrigeration it consists of four processes which start at point *a* (Fig. 2.1) and follow one another counterclockwise to form the closed cycle *abcd*. Heat enters and leaves the refrigerant along the isothermals *ab* and *cd*, respectively, whereas the isentropic temperature changes occur along the process lines *bc* and

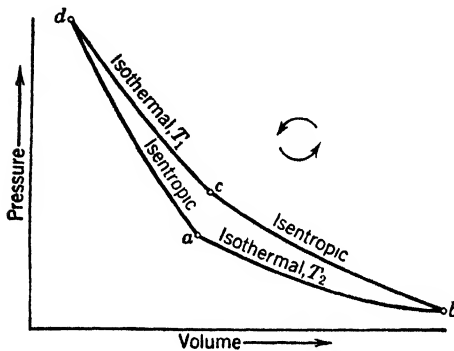


FIG. 2.1.

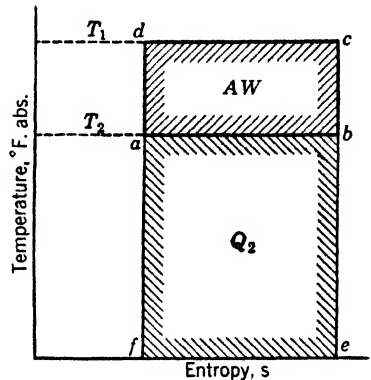


FIG. 2.2.

da. Shaft work must be supplied to the compressor during the compression process, whereas shaft work will leave the compressor (going into kinetic energy storage in the flywheel) during the expansion process *da*. The net result of the cycle will be a transfer of energy from the low-temperature source T_2 to the high-temperature receiver T_1 at the expense of a decrease in system energy as work and an increase in system thermal energy in the higher-temperature receiver. The same cycle is shown plotted to temperature-entropy coordinates in Fig. 2.2.

The effectiveness, or *coefficient of performance*, of any cycle is defined as the ratio of the desired result to the energy required to achieve that result; the coefficient of performance for any refrigeration cycle is therefore equal to the quantity of energy extracted from the body which is to be maintained at a low temperature divided by the shaft work needed to effect that extraction. If Q_2 is the net quantity of energy taken from the cold body, all of this will be discharged to the hot body with the addition of a thermal equivalent of the net shaft work supplied to the system. Thus if Q_1 is the total discharge of thermal energy to the body at T_1 the net shaft work requirement must be $Q_1 - Q_2$, so the

coefficient of performance can be written

$$\text{cop} = \frac{Q_2}{Q_1 - Q_2} \tag{2.12}$$

Numerically, the cop of refrigeration systems is always greater than unity.

In order to use the equation for cop it is first necessary to express the heat quantities in terms of properties of the refrigerant. During the isothermal process ab , shown in Fig. 2.1 for the Carnot cycle, heat flows from the source at T_2 to a perfect gas which is at a temperature infinitesimally less than T_2 . Since this is a non-flow process and occurs without change of temperature, the internal energy of the perfect gas remains constant and the energy equation reduces to a statement of the fact that all the heat added must appear as work of expansion; that is,

$$Q_2 = AW_{ab} = A \int_{V_a}^{V_b} P dV \tag{2.13}$$

But for an isothermal process the equation of a perfect gas gives

$$\frac{P_a V_a}{T_2} = \frac{P_b V_b}{T_2} = \frac{PV}{T_2} \tag{2.14}$$

or



$$P = \frac{P_a V_a}{V}$$

giving

$$Q_2 = AP_a V_a \int_{V_a}^{V_b} \frac{dV}{V} = AP_a V_a \ln \frac{V_b}{V_a} = AwRT_2 \ln \frac{V_b}{V_a} \tag{2.15}$$

In a similar manner the quantity of heat discharged to the hot body is found as

$$Q_1 = -AP_c V_c \ln \frac{V_d}{V_c} = AP_c V_c \ln \frac{V_c}{V_d} \tag{2.16}$$

Hence the net work required for the cycle is

$$Q_1 - Q_2 = \left[\left(P_c V_c \ln \frac{V_c}{V_d} \right) - \left(P_a V_a \ln \frac{V_b}{V_a} \right) \right] A \tag{2.17}$$

But, for any two isentropics between the same two temperatures,

$$\frac{V_c}{V_b} = \frac{V_d}{V_a} \quad \text{or} \quad \frac{V_c}{V_d} = \frac{V_b}{V_a} \tag{2.18}$$

Since for a perfect gas $PV = wRT$, substituting from this equation and

from equation 2·18 into equation 2·17 gives

$$Q_1 - Q_2 = AwR(T_1 - T_2) \ln \frac{V_b}{V_a} \quad (2\cdot19)$$

and, on substitution into equation 2·12 from 2·15 and 2·19,

$$\text{cop} = \frac{Q_2}{Q_1 - Q_2} = \frac{T_2}{T_1 - T_2} \quad (2\cdot20)$$

which thereby relates the effectiveness of the Carnot thermodynamic cycle to the temperature range over which the cycle operates.

Example. A large cold storage room is to be maintained at a fixed temperature of 10° F. Calculation shows that, to maintain this temperature under maximum heat gain, conditions will require extraction of thermal energy by means of refrigeration at a rate of 500,000 Btu/hr. Nearby is a large natural body of water whose maximum temperature at time of maximum load on the building is known to be 60° F. Determine the horsepower requirements of an ideal Carnot cycle refrigeration system to carry this load.

Solution. The required energy extraction Q_2 is 500,000 Btu/hr. Since the coefficient of performance of a reversible system will be $(10 + 460)/(60 - 10) = 470/50 = 9.4$, the power needed would be $500,000/9.4 = 53,200$ Btu/hr or $53,200/2545 = 20.9$ hp. Actually, of course, any real system would require much more energy than this, but the calculated figure is of practical interest in allowing rapid determination of the degree of approach of the actual to the ideal system.

2·4. Air Refrigeration Cycle. Curiously, the use of air as a refrigerant is both older and more recent than the use of liquefiable vapor. The air cycle (Fig. 2·3) uses the same basic equipment as the vapor cycle except that an expanding engine replaces the expansion valve and the two heat transfer equipments now operate without causing the refrigerant to change phase. Developed many years ago, the air cycle was widely used for some time, but eventually was replaced for the great majority of refrigeration installations by vapor systems, because the added first cost of the air expansion engine made this system non-competitive. Recently, however, air systems have again come into prominence in aircraft applications, because in many modern air transports an air compressor is needed for supercharging and for cabin pressurization at high altitudes, but is unnecessary for these purposes at the low altitudes for which air cooling is needed; hence this compressor can serve two purposes when it is made part of an air refrigeration cycle for low-altitude comfort cooling. Figure 2·4 shows diagrammatically the equipment arrangement for a combined supercharger, pressurizer, comfort heating, and cooling installation on a modern four-engine

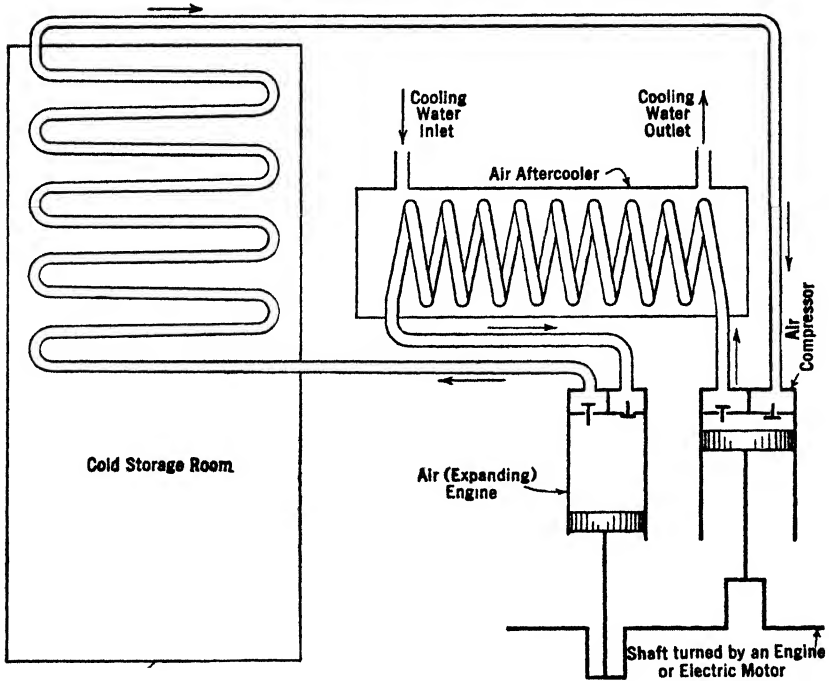


FIG. 2-3. The air refrigeration cycle.

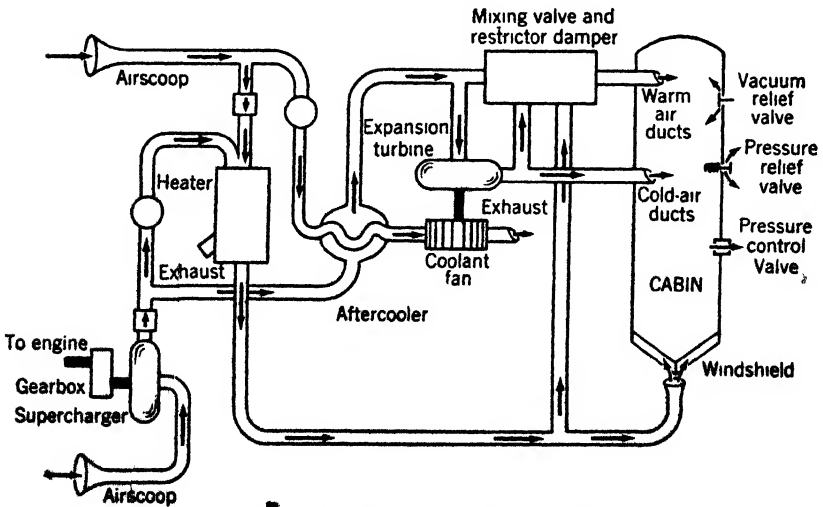


FIG. 2-4. (Courtesy Heating, Piping Air Conditioning.)

transport plane; since weight and space are at a premium for airplane installations, it is particularly interesting to find that a system proving satisfactory on planes is one which was unsatisfactorily large and weighty for stationary service.

2.5. Analysis of Air Cycle. In Fig. 2.5 the cycle $abmna$ on the pressure-volume (PV) coordinate axes represents the series of processes in the compressor, and the cycle $gdep$ represents a similar set of processes

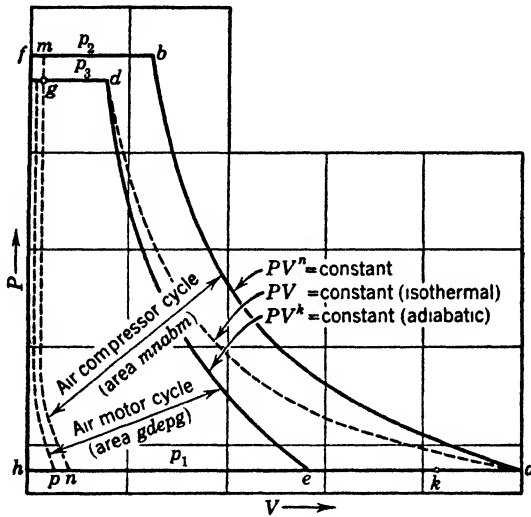


FIG. 2.5.

in the expander, each having an appreciable amount of clearance as seen in the figure. As the area between the path representing any reversible process on the PV plane and the volume axis, $\int P dV$, can be shown to be equal to the external work performed during the process, the two areas $abmn$ and $gdep$ to the scale of the figure represent the work of compression and the work of expansion, respectively, and the difference in the two areas is the amount of work which must be supplied by the engine or the electric motor. The pressure at the beginning of expansion in the expansion engine, d , is taken as less than that at the end of the compression process to allow for friction losses in the piping.

The process during which heat is absorbed by the air at the lower temperature is a constant pressure process from e to an intermediate state k , along the line ek , and the amount of heat so absorbed is

$$Q_{in} = wc_p(t_k - t_e) \text{ Btu/min} \quad (2.21)$$

where w is the weight of air in pounds per minute and $t_k - t_e$ is the rise of temperature of the air; c_p (the specific heat of the air) = 0.24. The rise of temperature from k to a represents the absorption of heat by the air in the suction pipes and ports as well as in the cylinder of the compressor during the suction stroke. In the air or vapor compression cycle $abef$ (Fig. 2·6), the external work performed may be found by considering horizontal strips. The area of the elementary strip is $V\Delta P$,

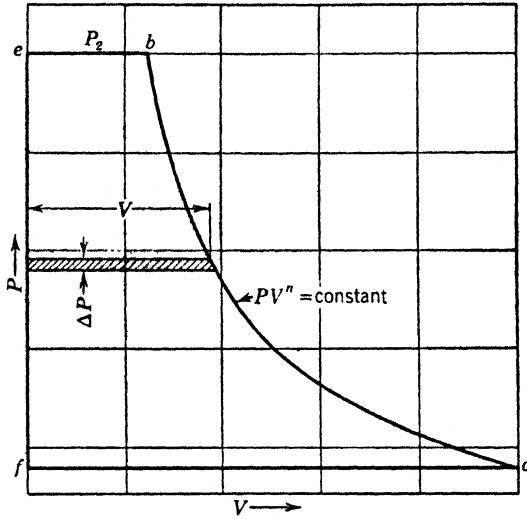


FIG. 2·6.

and that of the entire cycle is $\sum V\Delta P$, which can be integrated as soon as the variation of V with respect to P is known. As the compression line $P_a V_a^n = P V^n$, then the work (putting $P_a = P_1$ and assuming no clearance) becomes (by equation 1·16)

$$\begin{aligned} W_{in} &= \frac{n}{n-1} (P_b V_b - P_a V_a) \\ &= \frac{n}{n-1} wR(T_b - T_a) \quad \text{ft-lb/min} \end{aligned} \quad (2\cdot22)$$

but, from the relation for a perfect gas,

$$c_p = c_v + AR \quad (2\cdot23)$$

so

$$1 = \frac{c_v}{c_p} + \frac{AR}{c_p} = \frac{1}{k} + \frac{AR}{c_p}$$

where $k = c_p/c_v = 1.402$ for air, and R (the gas constant) = 53.34 for air.

Therefore

$$\frac{k-1}{k} = \frac{AR}{c_p} \quad (2.24)$$

from which

$$R = \frac{k-1}{k} \frac{c_p}{A} = Jc_p \frac{k-1}{k} \quad (2.25)$$

and equation 2.22 becomes

$$W_{in} = \frac{n}{n-1} \frac{k-1}{k} Jwc_p(t_b - t_a) \text{ ft-lb/min} \quad (2.26)$$

If the compression can be assumed to be isentropic, the work of compression, W , becomes

$$W_{in} = Jwc_p(t_b - t_a) \text{ ft-lb/min} \quad (2.27)$$

Likewise (assuming no clearance) the work of the expander (Fig. 2.5) is

$$W_{out} = \frac{n}{n-1} \frac{k-1}{k} Jwc_p(T_d - T_e) \text{ ft-lb/min} \quad (2.28)$$

but the action of the expander, which is usually lagged, is such that the exponent of the process $PV^n = c$ is very nearly equal to k , and therefore

$$W_{out} = Jwc_p(t_d - t_e) \text{ ft-lb/min} \quad (2.29)$$

The net work of compression for isentropic expansion and compression becomes

$$W_{in} - W_{out} = Jwc_p(T_b - T_a - T_d + T_e) \text{ ft-lb/min} \quad (2.30)$$

and the coefficient of performance is

$$\text{cop} = \frac{Jwc_p(T_k - T_e)}{Jwc_p(T_b - T_a - T_d + T_e)} = \frac{T_k - T_e}{T_b - T_a - T_d + T_e} \quad (2.31)$$

For the usual case with a water-jacketed compressor and an insulated expansion motor, by use of equations 2.21, 2.26, and 2.29 the coefficient of performance becomes

$$\begin{aligned} \text{cop} &= \frac{Jwc_p(T_k - T_e)}{\frac{n}{n-1} \frac{k-1}{k} Jwc_p(T_b - T_a) - Jwc_p(T_d - T_e)} \\ &= \frac{T_k - T_e}{\frac{n}{n-1} \frac{k-1}{k} (T_b - T_a) - (T_d - T_e)} \end{aligned} \quad (2.32)$$

2.6. The Effect of Clearance. In all commercial engines and compressors some volume of gas V_c (Fig. 2.7) remains in the cylinder when the exhaust valve closes. The volume of the clearance gas at the discharge pressure P_2 expands at the beginning of the suction stroke of the compression cycle until it has a volume V_d at the suction pressure P_1 . The effective stroke, during which a new supply of gas is brought into the cylinder, is represented by the line da . If the volume V_c is given as a certain ratio m of the piston displacement V_p , then $V_c = mV_p$ and

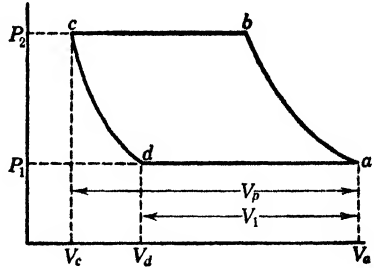


FIG. 2.7.

$$V_1 = V_p + V_c - V_d \quad (2.33)$$

but $V_d = V_c(p_2/p_1)^{1/k}$, assuming isentropic expansion; therefore

$$V_1 = V_p + mV_p - mV_p \left(\frac{p_2}{p_1}\right)^{1/k} = V_p \left[1 + m - m \left(\frac{p_2}{p_1}\right)^{1/k}\right] \quad (2.34)$$

The term $1 + m - m(p_2/p_1)^{1/k}$ is sometimes called the clearance factor. In the case of clearance the net work of compression becomes, from equation 2.26 (taking w_1 pounds per minute of gas during the compression and w_c pounds per minute during the expansion in the clearance volume),

$$W_{in} = w_1 \frac{k-1}{k} \frac{n}{n-1} Jc_p(T_b - T_a) - w_c \frac{k-1}{k} \frac{n}{n-1} Jc_p(T_c - T_d) \text{ ft-lb/min} \quad (2.35)$$

to be supplied by the compressor drive (assuming the same exponent during expansion and compression). However, $T_c = T_b$, as will be seen from the nature of the cycle, and therefore $T_a = T_d$ as the ratio of compression is the same as the ratio of expansion. Then the net work of compression is

$$W = (w_1 - w_c) \frac{k-1}{k} \frac{n}{n-1} Jc_p(T_b - T_a) \text{ ft-lb/min} \quad (2.36)$$

The presence of clearance in a compressor does not change work requirements except for losses due to friction and the fact that the cylinder needs to be larger to obtain the same capacity as one without clearance.

Example. In an air compressor $p_1 = 14$ and $p_2 = 100$ psia. The clearance is 5 per cent. Find the clearance factor if the expansion is represented by $PV^k = c$.

Solution. The clearance factor is

$$1.0 + m - m \left(\frac{p_2}{p_1} \right)^{1/k} = 1.0 + 0.05 - 0.05 \left(\frac{100}{14} \right)^{1/1.402} \\ = 0.846$$

2.7. An Example of Air Refrigeration. An air refrigeration machine is required to absorb 1000 Btu/min from the cold body. The suction pressure is 15 psia, the discharge pressure is 75 psia, and the initial pressure in the expansion cylinder is 70 psia. The compression line is represented by $PV^{1.35}$ due to the water jacket, whereas the expansion in the expansion cylinder is isentropic. Refrigerant is cooled to 70° F prior to admission to the expander. The brine circulated for cooling purposes will be cooled to 30° F, but the air leaves the brine coils at 25° to return to the compressor at 60°. From Fig. 2.5, $t_a = 60^\circ$, $t_k = 25^\circ$, and $t_d = 70^\circ$ F. The power necessary to drive the machine and the coefficient of performance are required. Assume no compressor clearance.

The temperature rise in the suction air temperature allows for the heat absorbed from the atmosphere and the cylinder head and walls during the suction stroke. Since the compression process is isentropic and the fluid obeys the perfect gas law, it can be readily shown that $T_e/T_d = (P_e/P_d)^{(k-1)/k}$; hence

$$T_e = (459.6 + 70) \left(\frac{15}{70} \right)^{(1.4-1.0)/1.4} = 341.1^\circ \text{ F abs} = -118.5^\circ \text{ F}$$

Similarly

$$T_b = (459.6 + 60) \left(\frac{75}{15} \right)^{(1.35-1.0)/1.35} = 788.6^\circ \text{ F abs} = 329.0^\circ \text{ F}$$

The weight of air which must be circulated per minute is

$$Q_{\text{abs}} = w c_p (t_k - t_e) \\ w = \frac{1000}{0.24[25 - (-118.5)]} = 29.03 \text{ lb/min}$$

The work of the compressor is

$$W_{\text{in}} = \frac{n}{n-1} w R (T_b - T_a) \\ = \frac{1.35}{0.35} \times 29.03 \times 53.34 \times (788.6 - 519.6) = 1,606,600 \text{ ft-lb/min}$$

The work of the expansion cylinder is

$$W = \frac{1.40}{1.40 - 1.0} \times 29.03 \times 53.34 \times (70 + 118.5) = 1,021,600 \text{ ft-lb/min}$$

The net work is

$$\begin{aligned} W &= 1,606,600 - 1,021,600 = 585,000 \text{ ft-lb/min} \\ &= 751.4 \text{ Btu/min} \end{aligned}$$

The refrigeration, by statement of the problem, is

$$Q = 1000 \text{ Btu/min}$$

The coefficient of performance is

$$\text{cop} = \frac{1000 \times 778.0}{585,000} = 1.331$$

The heat rejected to the cooling water is $Q_c = wc_p(t_b - t_d)$.

$$Q_c = 29.03 \times 0.24 \times (788.6 - 529.6) = 1805 \text{ Btu/min}$$

The piston displacement per minute is

$$\frac{wRT}{P} = \frac{29.03 \times 53.34 \times 519.6}{144 \times 15} = 372.6 \text{ cfm}$$

or 74.5 cu ft/(ton)(min).

The problem indicates that the temperature of the exhaust is very low (-118.5°F), that the coefficient of performance is also low (1.331), and that the piston displacement is high (74.5 cubic feet per ton of refrigeration). Low exhaust temperature results in operating difficulties such as the freezing of traces of water vapor on the exhaust valves of the expansion engine. The large piston displacement per ton of refrigeration, which results in a relatively large machine, may be reduced by the use of a higher-suction pressure in the compressor which, according to Fig. 2.5, is the same as the exhaust pressure of the air motor. If the exhaust pressure of the air motor is increased, the initial pressure should be increased also in order to obtain a low exhaust temperature. A "dense-air machine" originally meant one in which the suction pressure in the compressor was greater than atmospheric. The term was applied to machines having suction pressures of 40 to 65 psi and discharge pressures of 180 to 230 psi.

Example. If a dense-air refrigerating machine has 60 psia exhaust, and 250 psia initial pressures, with the other conditions as given in the preceding ex-

ample, find the coefficient of performance and the piston displacement per ton of refrigeration per minute, assuming $T = t + 460$.

Solution.

$$T_b = 520 \left(\frac{255}{60} \right)^{0.35/1.35} = 756.6^\circ \text{ F abs} = 297.0^\circ \text{ F}$$

$$T_a = 530 \left(\frac{60}{250} \right)^{0.4/1.4} = 352.6^\circ \text{ F abs} = -107.0^\circ \text{ F}$$

$$\text{Weight of air (pounds)} = \frac{1000}{0.24 \times [25 - (-107.0)]} = 31.56$$

$$\begin{aligned} \text{Compressor work, } W_1 &= \left(\frac{1.35}{0.35} \right) 31.56 \times 53.34 \\ &\quad \times (756.6 - 520.0) = 1,536,500 \text{ ft-lb/min} \end{aligned}$$

$$\begin{aligned} \text{Work of the expansion engine, } W_2 &= \frac{1.4}{0.4} \times 31.56 \\ &\quad \times 53.34 (70 + 107) = 1,045,400 \text{ ft-lb/min} \end{aligned}$$

$$\begin{aligned} \text{New work, } W_1 - W_2 &= 1,536,500 - 1,045,400 &= 491,100 \text{ ft-lb/min} \\ & &= 631.2 \text{ Btu/min} \end{aligned}$$

$$\text{Coefficient of performance} = \frac{1000}{631.2} = 1.58$$

Piston displacement

$$\begin{aligned} \text{per minute} &= \frac{wRT_a}{144P_a} = \frac{31.56 \times 53.34 \times 520.0}{144 \times 60} = 101.3 \text{ cu ft} \\ &= 20.26 \text{ cu ft/ton of refrigeration} \end{aligned}$$

By use of equation 2-32 the value for the coefficient of performance becomes

$$\text{cop} = \frac{25 - (-107)}{\frac{1.35}{0.35} \times \frac{0.4}{1.4} (756.6 - 520) - (530 - 352.6)} = 1.585$$

As no allowance for friction, clearance, or other factors has been made in the two preceding examples the results of the calculations are better than can be obtained in practice. No reliable tests of recent date on air refrigeration machines are available, but the following show what could be expected toward the close of the last century.

T. B. Lightfoot* reports that tests performed on a 27 and 24 by an 18-in. air refrigeration machine operating at 62 rpm and with a final pressure of 65 psi gave the following results:

* Lightfoot, *Proc. Inst. Civil Engrs. (London)*, 1881 and 1886.

Temperature of the air entering the compressor cylinder	52° F
Temperature of the air leaving the compressor cylinder	267° F
Temperature of the air entering the expansion cylinder	70° F
Temperature of the air leaving the expansion cylinder	-82° F

The refrigeration was 50,900 Btu/hr. The compression cylinder indicated 43.1 hp, the expansion cylinder 28.0 hp, and the steam engine required 24.6 hp in order to compress and circulate 1620 pounds of air per hour in addition to overcoming the friction developed. The calculated coefficient of performance was $50,900/(24.6 \times 2543)$ or about 0.8, but later tests, reported by others, were not as good.

PROBLEMS

1. Twenty pounds per minute of refrigerant undergo an enthalpy increase of 10 Btu/lb during passage through a compressor. If the compression process is polytropic and work input amounts to 5 hp, calculate the rate of heat loss from the compressor to the cooling water (neglecting other compressor heat and energy losses).

2. Vapor is admitted to a compressor at 40° F and with a specific entropy of 1.4. The leaving vapor temperature is 140° F and the rate of heat loss from compressor to cooling water is 550 Btu/min. Refrigerant flow rate is 10 lb/min. Calculate, approximately, the specific entropy of the refrigerant at discharge.

3. One pound of a refrigerant undergoes an entropy decrease of 0.05 unit during passage through a compressor. If the entering vapor temperature is 0° F, calculate, approximately, the temperature of the vapor at discharge. Heat loss to cooling water is at a rate of 25 Btu for each pound per minute of refrigerant.

4. A compressor discharges isentropically compressed vapor at a rate of 20 lb/min and with an enthalpy of 700 Btu/lb. If the enthalpy gain of the refrigerant during passage through the evaporator is 200 Btu/lb and if work of compression equals 1 hp lb/min, calculate the enthalpy of the refrigerant as it leaves the condenser.

5. Two gallons per minute of cooling water undergo an 8° F temperature rise during passage through a condenser. If refrigerant flow rate is $\frac{1}{2}$ lb/min and if work of isentropic compression is 60 Btu/lb, determine the capacity, in tons, of the system.

6. A compressor, operating isentropically, requires 2hp to raise $\frac{1}{2}$ lb/min of refrigerant from the evaporator pressure to the condenser pressure. If the rate of heat dissipation from the condenser is 450 Btu/min determine the coefficient of performance of the system.

7. A Carnot refrigeration cycle operates between temperatures of 0° F and 80° F. (a) Determine the cop. (b) If use of undersized evaporator and condenser necessitates an additional 10° F drop in temperature from cold storage room to evaporator and 10° F additional rise in temperature from cooling water to condenser, determine the resulting per cent of decrease in the coefficient of performance of the system.

8. A Carnot cycle operating between T_1 and T_2 compresses vapor from a specific volume of 10 cu ft/lb to one of 5 cu ft/lb. After cooling, the specific volume of vapor at the start of the expansion process is 6 cu ft/lb. Determine the specific volume at the end of the expansion process.

9. At the start of an isothermal process (in a Carnot cycle) the pressure is 80 psia and the specific volume is 10 cu ft/lb. At mid-point in the process the specific volume is 4 cu ft/lb. Determine the mid-point pressure.

10. Investigate the difference in work required to raise 100 Btu/min (on the Carnot cycle) through 100° F from 0° F and through 100° F from -100° F.

11. An air compressor designed for operation between 15 psia and 40 psia handles 800 cfm and has a clearance of 6 per cent. (a) Determine the volume of air, in cubic feet per minute, that could be handled by this compressor if the discharge pressure were raised to 100 psia and if the exponent for polytropic expansion remained fixed at 1.35. (b) Calculate the discharge pressure at which the volume of air flowing through the compressor would become zero.

12. A load of 10 tons is to be carried by an air refrigeration cycle with suction pressure of 15 psia and discharge pressure of 90 psia. The compression process has a polytropic exponent of 1.30 whereas the expansion process exponent is 1.37. Pressure at start of expansion is 83 psia and temperature is 80° F. Air temperature after useful heat addition is 34° F, and the air undergoes a 15° F temperature rise in the piping during return to compressor suction. Clearance is zero. (a) Calculate the cop. (b) Calculate the required piston displacement in cubic feet per minute.

13. An air cycle operates with conditions as in Problem 12 except that suction pressure is 35 psia, discharge pressure is 180 psia, and pressure at the start of expansion is 170 psia. (a) Calculate the cop and compare with the cop of Problem 12. (b) Calculate the piston displacement and compare with the value from Problem 12.

14. A dense air refrigeration machine compresses 50° F air at 50 psia to 100 psia. The initial temperature and pressure in the expander are 70° F and 95 psia respectively, the compression curve is represented by $(P\bar{V})^{1.35}$, the expansion curve is isentropic, and the air is heated in the refrigerating coils to 25° F. Find the cop, assuming that both the compressor and the expander have zero clearance.

CHAPTER III

SIMPLE REFRIGERATION CYCLES

3·1. Refrigeration by Evaporation. Liquefiable vapors possess two very significant advantages which have led to their wide use as working mediums in refrigeration systems. A refrigerant should have a large carrying capacity for internal energy, and it should also receive or release large quantities of heat without undergoing appreciable temperature change. For all single-phase materials these two characteristics are mutually exclusive since the specific heats of suitable liquids do not differ greatly from that of water; hence such liquids necessarily undergo a temperature change of approximately 1 Fahrenheit degree per unit of heat entering or leaving 1 pound of fluid. If then a "carrying capacity" of 50 or 100 Btu/lb were realized it would necessarily be at the expense of a relatively huge (50° to 100° or more) temperature change. But the temperature of the heat source, as a cold storage room, must be held constant; hence the temperature occurring in the energy-receiving equipment would necessarily be less than that of the temperature in the cold storage space, and, for a refrigerant of the type just described, the minimum temperature of the refrigerant would be required to be 50° to 100° F and more below the cold room temperature. Since it can be readily established that the effectiveness of any refrigeration system decreases rapidly with increase of the temperature range over which it must operate, it follows at once that this 50° to 100° F penalty would place any such system at a severe disadvantage. If, however, the temperature change of the fluid were kept low by reducing the energy gain per pound, it then becomes clear that a very large flow rate would have to be maintained with consequent excess pressure losses and with a small working effectiveness of each pound of refrigerant.

Liquefiable vapors avoid these difficulties by accepting and rejecting energy in internal potential rather than internal kinetic form, hence without temperature change. One pound of liquid ammonia, for example, receives 545 Btu during the process of evaporation (change from liquid to vapor) at a constant temperature of 30° F. Since the temperature of the fluid remains constant the equipment in which this energy transfer occurs can be operated at a temperature only sufficiently below

that of the cold storage room as dictated by the need of achieving heat transfer at a rate sufficient to permit use of relatively small transfer surfaces. Similarly, the use of such a fluid as a refrigerant is advantageous in the condenser, since here the heat loss also takes place at fixed temperature and hence with a minimum of necessary temperature difference.

Fundamentally, therefore, the primary advantage of a liquefiable vapor for use as a refrigerant is that it permits maintenance of the lowest possible temperature difference between the low-temperature and

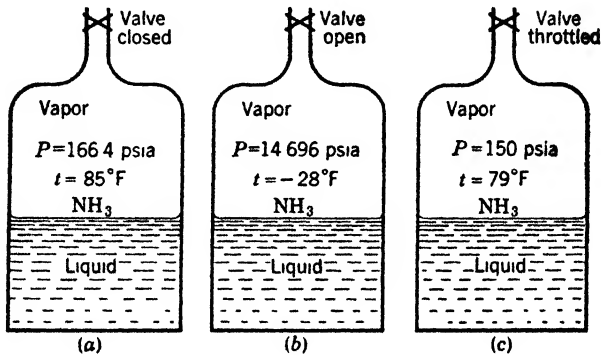


FIG. 3-1.

the high-temperature equipments of the refrigeration cycle. A secondary factor which emphasizes this advantage is that (refer to Chapter IX) the resistance to heat transfer of a fluid during either evaporation or condensation is many times less than the resistance of a liquid or a gas to heat transfer without phase change, hence the temperature difference between the evaporator and the cold storage space, or between the condenser and the cooling water, will be substantially less, with equal area of transfer surface, when a liquefiable vapor is used as the refrigerant.

The simplest visualization of vapor refrigeration can be obtained by considering a sealed receptacle full of liquid ammonia placed in a room which is to be cooled. Let us suppose that room temperature is 85°F and that the outside air temperature is 95°F . Any fluid in a closed container under steady-state conditions must reach equilibrium with its own vapor. Thus ammonia liquid at a temperature of 85° would vaporize into the space above the liquid (see Fig. 3-1) until such time as the vapor pressure of ammonia over the liquid reached that value which, for 85°F , constitutes equilibrium. From a chart or table of the physical properties it can be readily determined that the saturation

pressure of 85° F ammonia is 166.4 psia; hence this is the pressure that, for steady-state conditions, will be exerted by the ammonia on its container. If, now, a pipe connection is run from the opening of the container to the atmosphere (to avoid possible contamination of the room with material flowing from the container) and if the valve on the container is opened, the pressure within the container will necessarily drop to atmospheric (neglecting pressure losses in the pipe). The liquid ammonia will then vaporize rapidly (flash) until the temperature of the residue of liquid will have been dropped, as a result of energy extraction to provide the latent heat of vaporization of the fraction which has evaporated, to that value which, from tables or charts of ammonia properties, is found to represent the saturation temperature corresponding to atmospheric pressure. For 14.696 psia pressure the saturation temperature of ammonia is -28° F; hence the temperature of the fluid remaining within the container will rapidly drop to this value.

But the room temperature is 85° F; hence as soon as the temperature of the liquid ammonia drops below 85° F the container will start to act as a heat sink receiving energy from the room. Because -28° F is the saturation temperature corresponding to atmospheric pressure, it would not be possible for the temperature of liquid in the container to remain above this value, but evaporation would occur and, since the latent heat of vaporization of ammonia under these conditions is 589, it follows that 1 lb of liquid would evaporate for each 589 Btu which enter the container from the room.

A heat sink at -28° F in a room at +85° F represents a large temperature differential. Suppose now that instead of being opened all the way the container valve was merely "cracked" sufficiently to allow flow at a rate corresponding to a container pressure of 150 psia. The saturation temperature at this pressure is 79° F, so again there will be a heat flow from the room to the container and consequent evaporation of the ammonia. Now, however, the latent heat of vaporization of the ammonia is 500 Btu/lb, so that a 20 per cent (approximately) reduction in refrigerating effect has accompanied the increased container pressure. This loss of useful refrigerating effect will later appear as a useless atmospheric cooling at the subsequent point in the system at which the refrigerant is reduced to atmospheric pressure. Hence, for a simple direct evaporation system of the type described, it would be more economical, in terms of the quantity of refrigerant required, to open the container valve wide and operate with the large temperature difference which corresponds to the difference between room temperature and the saturation temperature of the refrigerant at atmospheric pressure. This is contrary to recommended operating procedure for

any *closed* refrigerating cycle, but it is emphasized here to call attention to the fact that loss of availability of energy is important only when interest centers on a series of connected thermodynamic processes and may, indirectly, be desirable rather than otherwise when a single process is under consideration.

If an ample supply of refrigerant were available, refrigeration could obviously be obtained by a system consisting of only an evaporator. Thus, by replacing containers of refrigerant as evaporation occurred, a heat sink could be maintained in the room without further complication. The ordinary icebox represents a refrigeration system of this kind since the ice—the heat sink—is produced elsewhere and introduced into the space to be cooled and then replaced after it has liquefied. Fortunately, water is an inexpensive fluid, and the melted ice, with its load of energy picked up from the cooled space, can economically be wasted. In the case of liquefiable vapors the same situation does not hold because the cost of such fluids is, without exception, so great that they must be recovered for re-use. The term liquefiable vapor, as used in this sense, refers to a material which boils, at atmospheric pressure, at a temperature sufficiently lower than that of the usual surround to permit its use as a heat sink.

If water were costly the liquid resulting from the melting of ice could readily be captured in a container under the icebox and returned to the ice plant to have its internal energy extracted, to be reprocessed and returned again to the consumer in the form of ice. Why cannot the same be done with a liquefiable vapor? The answer is that it could be done, but the volume increase of such a refrigerant during the evaporation process is so great that it would be impractical to provide a container of the requisite size. Thus 64 lb of water occupy approximately 1 cu ft and result from the reception, by ice, of $144 \times 64 = 9216$ Btu (melting ice at 32° F to water at the same temperature). The same refrigerating effect would be obtained from evaporation of $9216/589 = 15.65$ lb of ammonia (from liquid at -28° F to vapor at the same temperature), but the volume occupied by this weight of saturated ammonia vapor would be $15.65 \times 18 = 282$ cu ft; obviously, therefore, the storage of saturated ammonia vapor at atmospheric pressure would be prohibitive in space requirement.

3-2. The Simple Saturation Cycle. For the above reason it is evident that, when liquefiable vapors are used in refrigeration, equipment must be provided at the place of use to permit reprocessing of the vapor as rapidly as it is formed; all equipment used in mechanical refrigeration systems of the vapor type, exclusive of the evaporator, is therefore needed, not to achieve refrigeration, but to reduce the operat-

ing cost by conserving and permitting re-use of the same material. In essence this is accomplished by the elementary equipments shown in Fig. 3-2. Saturated vapor leaving the refrigerant container is not discharged to the atmosphere but is drawn into the suction of a compressor which raises the vapor pressure to a value such that the corresponding saturation temperature exceeds that of whatever cooling medium—water or air—is available. Thus, for the case illustrated, if outside air at 95° F is the only material available for cooling the refrigerant, the compressor would necessarily raise the ammonia vapor pressure to a value at which the saturation temperature would be in

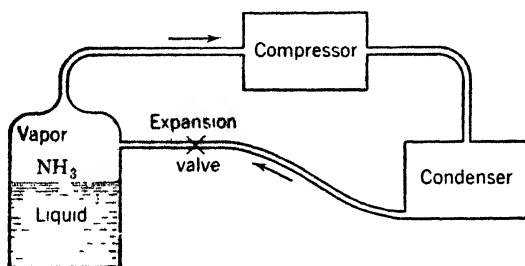


FIG. 3-2.

excess of 95° F. If the condenser were designed for a 5° F temperature drop the resultant temperature of condensation would need to be 100° F and the pressure, from tables or charts of properties of ammonia, would be 212 psia.

This example demonstrates that the pressures maintained in the evaporator and the condenser of a refrigeration system are determined as functions of the desired temperatures of saturated fluid in these two equipments. Thermodynamically, the purpose of a compressor is to raise the boiling temperature of a refrigerant, whereas, thermodynamically, the purpose of an expansion valve is to lower the boiling temperature. Since this temperature change is achieved as a secondary effect due to change of pressure it is customary to think of a refrigeration cycle as operating between certain pressure limits, but in essence it is the temperature change rather than the pressure change that is of importance.

After leaving the compressor, the high-temperature, high-pressure, refrigerant vapor is passed through a condenser in which it first loses superheat and then condenses, leaving the condenser as a liquid at the compressor discharge pressure (assuming an ideal no-loss system) and at the corresponding saturation temperature or, if some subcooling of the liquid occurs, at some lower temperature. The next requirement is

that this liquid refrigerant be reintroduced into the container in which evaporation is occurring. Since the pressure in the condenser is high and that in the evaporator is low the reconditioned refrigerant can be discharged into the evaporator through a simple throttling valve set at a position such that the rate of flow of refrigerant to the evaporator is equal to the rate of evaporation at equilibrium conditions. A valve of this type achieves a pressure reduction with accompanying reduction in the saturation temperature and consequent flash of a portion of the refrigerant; the flash vapor forms during the process of cooling or "refrigerating" the liquid refrigerant during passage through the valve, and, since flash results in a large volume increase, the pressure-reducing valve, when used in such a cycle, is commonly called an expansion valve.

Again, however, an effective understanding of the refrigeration cycle requires recognition of the fact that the purpose of the expansion valve is to achieve a drop in saturation temperature and not to achieve expansion; the fact that expansion occurs is, in reality, a wholly undesirable, though necessary, condition, since expansion is the result of evaporation which, in turn, is due to loss of useful refrigerating effect of part of the refrigerant resulting from necessary cooling of the remainder. If the liquid refrigerant leaving the condenser could be subcooled to a temperature equal to that existing in the evaporator it would then pass through the "expansion" valve without change in either volume or temperature. This assumed case is, of course, impractical since, if a fluid were available for use in subcooling the refrigerant to evaporator temperature, this same fluid could be used directly to extract heat from the evaporator and thereby obviate the need for a refrigerant.

The cycle arrangement of Fig. 3·2 includes the simplest equipment combination that can be used in a mechanical refrigeration system. Many other equipment arrangements are used in practice in attempts to improve the energy requirements of the cycle, but all such complex systems can be considered refinements of the basic cycle which, because of its simplicity, is a standard that is continually referred to. By definition this operating arrangement gives a "simple saturation" cycle; "simple" because of minimum equipment, "saturation" because the vapor entering the compressor is in a saturated state and the liquid entering the expansion valve is likewise saturated.

3·3. Analysis of Basic Refrigeration Cycles. The theoretical vapor refrigeration cycle is shown by the dry compression cycle *abcdfa* (Fig. 3·3) or the wet compression cycles *abefa* and *abghfa*. In dry compression, use is made of a water jacket in the cylinder head and in the compressor cylinder near the head, and the suction vapor is dry or slightly superheated. In wet compression some liquid refrigerant,

indicated by the quality at e or g , is present with the vapor at the beginning of compression. The process ab represents, as well as can be shown on the Ts coordinate plane, the action of the pressure-reducing

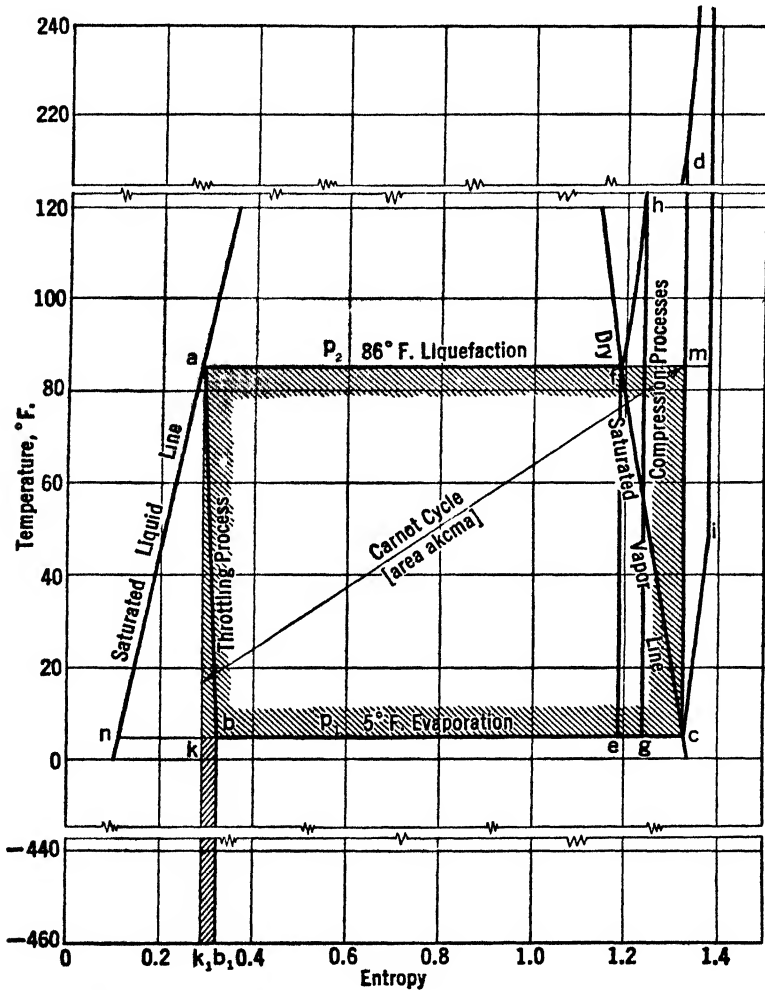


FIG. 3-3. The vapor refrigeration cycle, Ts coordinates. (Note that process $jd hfa$ is at constant pressure.)

valve. The constant entropy, isentropic, process ak indicates the change of state when a reversible, ideal, expanding engine is used. Useful refrigeration is obtained during the evaporation processes be , bg , and bc at the constant pressure p_1 corresponding to the constant tem-

perature t_1 . The theoretical compression lines ef , gh , and cd are shown as isentropic changes of state. The constant pressure (p_2) process dfa represents the rejection of heat in the condenser, first the rejection of superheat along df and then the latent heat of liquefaction along fa . As dfa and bc are both constant pressure processes the heat absorbed or rejected is equal to the difference of the enthalpies at the beginning and the end of the process.

The refrigeration per pound of the refrigerant is therefore $h_c - h_b$. But $h_a = h_b$; therefore the refrigeration per pound becomes $h_c - h_a$. The position of b can be found in the following manner:

$$h_b = h_n + X_b L_{p_1} \quad (3.1)$$

where X_b is the quality of the mixture at state b , L_{p_1} is the latent heat of vaporization at evaporator pressure, and h_n is the enthalpy of the saturated liquid at the state n . Therefore

$$h_a = h_n + X_b L_{p_1} \quad (3.2)$$

and

$$X_b = \frac{h_a - h_n}{L_{p_1}} \quad (3.3)$$

Likewise the heat lost to the cooling water in the condenser by the refrigerant is equal to $h_d - h_a$ or $h_f - h_a$, as the case may be. The work of compression for the entire cycle may be found in the following manner: If the process of the throttling valve may be assumed to be without the gain or loss of heat through the walls of the pipe or valve, and without the performance of external work, then

$$AW_{in} = \text{the heat rejected (to the condenser)} - \text{the heat absorbed (from the evaporator)} \quad (3.4)$$

or

$$\begin{aligned} AW_{in} &= (h_d - h_a) - (h_c - h_b) \\ &= h_d - h_c - (h_a - h_b) \\ &= h_d - h_c \quad \text{Btu/lb} \end{aligned} \quad (3.5)$$

The loss due to the use of the pressure-reducing valve rather than an expansion engine is shown by the hatched area kbb_1k_1 of Fig. 3.3. This area represents the loss of useful refrigeration, the increase in the amount of external work performed, and the increase in the amount of heat required to be removed by the condenser. A similar development of the formula applies to the wet compression cycle indicated by the compression along gh and ef .

3.4. The Mollier Diagram. Following the example of other divisions of mechanical engineering, it is not customary to use the T s diagram in the solution of practical problems not requiring great precision, but rather to use some form of Mollier, as for example the p - h diagram, as shown in Fig. 3.4. The refrigeration cycle, as in Fig. 3.3, is represented by the cycle $abcd$. The absorption of heat during the evaporation of the refrigerant at the pressure p_1 is indicated by the process bc , whereas the loss of superheat and liquefaction at the pressure p_2 is represented by the process dca . As the enthalpies at the initial and final states of the throttling action in the pressure-reducing valve are equal, such a process may be assumed to be shown on this diagram by a vertical line. Finally, the compression process is assumed to be an isentropic as represented on the p - h diagram by a curved line. If no inert gases are present in appreciable amounts in the condenser the pressure of the refrigerant, p_2 , in the condenser is that corresponding to the temperature of liquefaction, which in turn need not be more than 1 Fahrenheit degree greater than the temperature of the cooling water leaving the condenser.

When it is satisfactory to read values of enthalpy directly from the Mollier chart the solution of problems becomes very simple. In the cycle $abcd$ the useful refrigeration per pound of refrigerant is $h_c - h_a$, the work of compression is $h_d - h_c$, and the heat removed from the refrigerant in the condenser is $h_d - h_a$. The coefficient of performance is $(h_c - h_a)/(h_d - h_c)$. In wet compression, if the process ef had been used, the net refrigeration per pound of refrigerant would have been $h_e - h_a$, the work of compression $h_f - h_e$, and the coefficient of performance

$$\frac{h_e - h_a}{h_f - h_e} \quad (3.6)$$

An expression for the relative merit of dry as compared with wet compression can be obtained theoretically from the ratio of coefficients of performance. For example, this becomes

$$\frac{\text{cop}_d}{\text{cop}_w} = \frac{\frac{h_c - h_a}{h_d - h_c}}{\frac{h_e - h_a}{h_f - h_e}} \quad (3.7)$$

There are graphical constructions for securing the best condenser pressure, as well as the best point at which to start compression, but in practice the problem is one of operation, and control is difficult with wet compression. Consequently consideration of the best value of

the coefficient of performance takes second place in considering the convenience of dry compression with little (5° to 10° F) superheat in the suction gas at the compressor. Freon-12 can use more superheat to advantage in certain cases.

The value of h_d can be found by the consideration of constant entropy during the process cd , where in this case s_c is taken as the entropy of dry saturated vapor. The values of h_d , \bar{V}_d , and t_d can be found from suitable tables or diagrams for the superheated refrigerant.

With wet compression the point f is known, for then it is dry and saturated and the value of the entropy can be calculated or found from the tables. The quality of the vapor at e can be found from the relation

$$\begin{aligned} s_f &= s_e \\ &= s_n + X_e \frac{L_{p_1}}{T_e} \end{aligned} \quad (3.8)$$

and the value of

$$h_e = h_n + X_e L_{p_1} \quad (3.9)$$

where s_n is the entropy of the saturated liquid, L_{p_1}/T_e is the increase of entropy during vaporization, X_e is the quality at e , h_f is the enthalpy of the saturated liquid at f , and L_{p_1} is the latent heat of vaporization.

It is evident that the actual commercial refrigeration compressor is subject to the effect of clearance, throttling, or wire drawing (these are terms used to designate internal and skin friction and the effect of the change of direction of flow and sudden change of cross section) of the vapor entering the cylinder as well as the heating effect of the cylinder head and walls, in particular that portion enclosing the clearance volume. Calculations of capacity and the power requirements of compression necessitate information on these different factors, which have been given the common term, volumetric efficiency.

3.5. Volumetric Efficiency. The ratio of the weight of the refrigerant actually delivered to the condenser to the weight theoretically pumped at the suction temperature and pressure, observed just outside the compressor, is called the actual volumetric efficiency e_a (sometimes called the total volumetric efficiency or the charge efficiency). This ratio must be found by test as it includes: (1) the effects of valve and piston ring leakage, although these are expected to be small, and the effect of surface and internal friction e_i ; (2) the effect of superheating the refrigerant, e_s , particularly during the suction stroke resulting from contact with the relatively hot cylinder walls, piston, and cylinder head; and (3) the effect of clearance e_c . The actual volumetric efficiency is

the product of these separate efficiencies, or

$$e_a = e_i \times e_s \times e_c \quad (3 \cdot 10)$$

The volumetric efficiency due to clearance, e_c (also called by some writers the apparent volumetric efficiency), already has been considered, and its effect has been given by equation 2·34. The refrigeration engineer does not consider that its effect is harmful as theoretically it does not increase the work of compression per unit of refrigeration.

The effect of heat-conducting cylinder walls, resulting in the *thermal volumetric efficiency* e_s , is the most serious of the three efficiencies entering into the actual volumetric efficiency. The effect of superheating the vapor during the suction stroke is not shown on the indicator diagram, but the effect of absorbing heat from the cylinder walls results in increasing the specific volume and in decreasing the weight of refrigerant pumped per stroke of the piston. The thermal volumetric efficiency, then, results in a larger piston displacement and an increased work of compression per unit of refrigeration as compared with the compression cylinder constructed with non-conducting cylinder walls.

Actual values for thermal volumetric efficiency have been determined by a number of experimenters.* G. T. Voorhees used the values reported by the York Manufacturing Company as results of tests performed for the Ice Machine Builders Association, 1903, and he found that the following empirical formula gave reasonably good results for ammonia:

$$e_s = 1 - \frac{t_d - t_c}{1330} \quad (3 \cdot 11)$$

where t_d is the temperature obtained from the Mollier diagram, t_c being assumed to be the temperature corresponding to the suction pressure with the machine operated so that there will be little or no superheat when the refrigerant enters the cylinder. Tests are not numerous enough to prove whether an increase of rotative speed increases the value of e_s . It seems reasonable to expect such an increase accompanied with a decrease in the value of e_i as the speed is increased.

Published tests for the actual volumetric efficiency have been confined to ammonia and sulphur dioxide. Other refrigerants have been tested more or less carefully by the manufacturers, who have secured

* Reed and Ambrosius, *Refrig. Eng.*, Vol. 21, No. 3; Phillip and Spreen, *ibid.*, Vol. 18, Nos. 10 and 12, Vol. 14, Nos. 2 and 5; Jenks, *ibid.*, Vol. 12, No. 9; York Manufacturing Company, *Bulletin* 27, October, 1907; *Ice and Refrig.*, August, 1904.

values for the use of their engineers. The following values for thermal volumetric efficiency for carbonic compressors are of such a nature.

Ratio p_2/p_1	e_s
2 0	0 85
3 0	0 80
4 0	0 75
5 0	0 70
6 0	0 64
7.0	0 58

where p_2/p_1 = the ratio of the liquefaction to the evaporation pressure.

From the preceding it is evident that, if a compressor has tight valves and piston rings and is designed with ports and valves of ample size, such essential design calculations as the displacement volume and the work of compression may be made. However, it is necessary that the

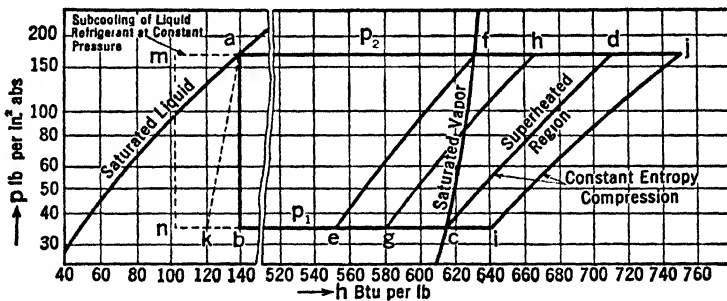


FIG. 3-4. The vapor refrigeration cycle, ph coordinates.

thermal volumetric efficiency be known, as well as the actual clearance volume and the thermodynamic properties of the refrigerant. For design purposes the piston displacement per ton of refrigeration per minute is one of the most important requirements.

3.6. The Piston Displacement per Ton of Refrigeration. It is assumed that the vapor at the end of the suction stroke may be represented by the point c (Fig. 3-4) and that the liquid at the pressure-reducing valve may be indicated by the point a , the weight of refrigerant per ton of refrigeration per minute is $200/(h_c - h_a)$ pounds, and the corresponding swept volume becomes $[200/(h_c - h_a)] \bar{V}_c$. If the volumetric efficiency due to clearance is taken as e_c and the thermal volumetric efficiency is taken as e_s , the total piston displacement per ton per minute is

$$\frac{200}{h_c - h_a} \bar{V}_c \frac{1}{e_s e_c} \text{ cfm/ton} \quad (3.12)$$

Example. Find the piston displacement per ton of refrigeration for standard conditions of operation, namely 86° F liquefaction and 5° F evaporation: (1) with no clearance, and (2) with 2 per cent clearance. Let the refrigerant be ammonia.

Solution. If the temperature at the end of the isentropic compression from the p - h diagram (Fig. 8·3) is taken as 201° F, the value of e_s becomes (by equation 3·11) $1.0 - [(201 - 5)/1330] = 0.853$. The piston displacement for zero clearance becomes

$$\frac{200}{613.3 - 138.9} \times \frac{8.15}{0.853} = 4.03 \text{ cfm/ton}$$

Tables 8·6 and 8·7 being used for the properties of saturated and superheated ammonia. The volumetric efficiency due to clearance, if the expansion process in the clearance volume is represented by $PV^{1.28} = \text{a constant}$, is

$$e_c = 1.0 + 0.02 - \left[0.02 \left(\frac{169.2}{34.27} \right)^{1/1.28} \right] = 0.95$$

The piston displacement per ton of refrigeration per minute is

$$\frac{200}{613.3 - 138.9} \times \frac{8.15}{0.853 \times 0.95} = 4.20 \text{ cfm/ton}$$

The coefficient of performance is the most important means of comparing different cycles for refrigeration purposes as it gives a direct ratio of the useful refrigeration to the work of compression, but sometimes it does not satisfy completely what is desired. For engineering purposes, occasionally, it may be more desirable to make use of the term horsepower per ton of refrigeration.

3·7. Horsepower per Ton of Refrigeration. As has been mentioned, it is assumed that clearance has little effect in the compressor cylinder on the work of compression, and that throttling and other frictional losses may be reduced to a small amount by care in the compressor design; therefore the thermal efficiency e_s is the only volumetric efficiency that has to be considered. Assuming, for calculation purposes, that the compression is a frictionless adiabatic and may be represented by the equation $PV^k = \text{constant}$ but that a multiplying factor $1/e_s$ has to be included in order to rectify for the thermal loss during the suction stroke, the weight of refrigerant compressed per ton of refrigeration per minute is $200/(h_c - h_a)$ and the work of compression is $[200/(h_c - h_a)] [(h_d - h_c)/e_s]$ Btu. Dividing this quantity by 33,000/778.3 (the heat equivalent of a horsepower-minute) gives

$$\begin{aligned} \text{hp/ton of refrigeration} &= \left(\frac{200}{h_c - h_a} \right) \frac{h_d - h_c}{e_s \frac{33,000}{778.3}} \\ &= \frac{4.717}{e_s} \frac{h_d - h_c}{h_c - h_a} \end{aligned} \quad (3·13)$$

Values for h may be taken from the tables, or from the p - h or other form of Mollier chart, or sometimes these values may be calculated.

For the cycle $abefa$ (Fig. 3·5) on the P - V coordinate axes, the work of compression may be found by equation 2·22 after elimination of V_1 and V_2 by noting that for isentropic compression $(P_a/P_b)^{1/k} = V_b/V_a$.

$$W_{in} = \frac{k}{k-1} P_1 V_a \left[\left(\frac{P_2}{P_1} \right)^{(k-1)/k} - 1.0 \right] \text{ ft-lb} \quad (3\cdot14)$$

and the horsepower is found by dividing the work in foot-pounds per minute by 33,000. In the above equation P_2 and P_1 are the discharge and

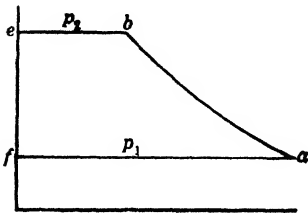


FIG. 3·5.

suction pressures, respectively, in pounds per square foot, V_a is the total piston displacement required per ton of refrigeration per minute corrected for volumetric efficiency. If the value of k for ammonia vapor is taken as 1.285 and the pressures are taken in pounds per square inch, the equation reduces to

$$\text{hp/ton} = 0.0197 p_1 V_a \left[\left(\frac{p_2}{p_1} \right)^{1/4.508} - 1 \right]$$

Example. Find the horsepower per ton of refrigeration in the preceding example.

Solution.

$$hp = \frac{4.717}{e_a} \left(\frac{h_d - h_c}{h_c - h_a} \right) = \left(\frac{4.717}{0.853} \right) \left(\frac{713.0 - 613.3}{613.3 - 138.9} \right) = 1.162$$

Also

$$hp = 0.0197 \times 34.27 \times 4.03 \times \left[\left(\frac{169.2}{34.27} \right)^{1/4.508} - 1 \right] = 1.155$$

About the turn of the century, operation of the ammonia compressor so as to have wet in preference to dry compression, or vice versa, was still a source of vigorous debate, and it is interesting to see what the possibilities of wet compression may be. The best method of comparison makes use of the coefficient of performance.

3·8. Theoretical Cycles. In Fig. 3·4 the cycles $abefa$ and $abgha$ indicate "wet" vapor at the points of compression e and g , respectively, whereas the cycles $abcd$ and $abija$ show dry vapor at c and superheated vapor at i , respectively. A comparison of these four cycles shows the nearest approach to the Carnot to be the cycle $abefa$, for in each of the other cycles the temperature at the end of compression is greater than the temperature of liquefaction, which in the figure is taken as 86° F.

Example. An ammonia compressor operates between the standard temperature limits of 86° and 5° F. The compression process in the compressor is assumed to be isentropic. For the cycles shown in Fig. 3-4, Table 3-1 gives a summary of comparative performance.

TABLE 3-1

Cycle	Useful Refrigeration Btu	Work of Compression Btu	Coefficient of Performance
Carnot	482.6	84.06	$\frac{482.6}{84.06} = 5.74$
<i>abcd</i>	613.3 - 138.9 = 474.4	713.0 - 613.3 = 99.7	$\frac{474.4}{99.7} = 4.74$
<i>abef</i>	550.6 - 138.9 = 411.7	631.5 - 550.6 = 80.9	$\frac{411.7}{80.9} = 5.09$
<i>abgh</i>	573.4 - 138.9 = 434.5	659.5 - 573.4 = 86.1	$\frac{434.5}{86.1} = 5.05$
<i>abija</i>	639.1 - 138.9 = 500.2	750.4 - 639.1 = 111.3	$\frac{500.2}{111.3} = 4.50$
<i>abcija</i>	613.3 - 138.9 = 474.4	750.4 - 639.1 = 111.3	$\frac{474.4^1}{111.3} = 4.28$

¹Refrigeration ceases at *c*.

The single-stage ammonia compressor is not suitable for evaporation at low temperatures, because the compression ratio becomes excessive, the expansion in the clearance volume becomes very large, and the thermal volumetric efficiency becomes seriously low. The temperature of the discharged vapor passes the safety point, resulting in lubrication and other difficulties. For suction pressures of 1 atmosphere or less it is therefore necessary to use two or more stages of compression.

PROBLEMS

1. A simple saturation system using NH₃ as a refrigerant is required to maintain a cold storage room at 10° F when using cooling water at an entering temperature of 60° F. If the temperature drop from storage room to refrigerant in the evaporator is 10° F and from refrigerant in condenser to mean cooling water temperature is 8° F, and if the temperature rise of cooling water is 6° F, calculate the necessary suction and discharge pressures for the system.

2. If carbon dioxide were used instead of ammonia in the system of Problem 1, determine the suction and discharge pressures and compare with the values for ammonia.

3. A simple saturation cycle using ammonia as a refrigerant discharges vapor from the compressor with pressure of 100 psia and temperature of 215° F. Determine (a) the enthalpy of the vapor at discharge from the expansion valve and (b) the specific volume of the vapor at compressor suction.

4. Investigate and discuss the difference in work requirements of a simple saturation cycle which expands ammonia through an expansion valve and one which ex-

pands the same fluid through an isentropic expanding engine. Take suction and discharge pressures as 15 psia and 165 psia respectively.

5. If the conditions of the simple saturation cycle were altered so that vapor would be in a saturated state at discharge from the compressor, would the revised cycle approach the Carnot cycle more or less closely than does the simple saturation cycle?

6. An ammonia simple saturation cycle operates between evaporator and condenser temperatures of 0°F and 70°F . (a) Determine the cop. (b) If wet compression were used and if the state of vapor at discharge from the compressor were saturated, determine the cop. (c) Indicate whether or not, and why, the cop with wet compression would be expected to usually exceed that of the corresponding simple saturation cycle.

7. Repeat Problem 6 with Freon-12 as the refrigerant and compare results.

8. Refrigerant at entrance to an expansion valve has an enthalpy of 580. At the pressure corresponding to expansion valve discharge, the enthalpy of the saturated liquid is 100 and the enthalpy of the saturated vapor is 900. If 10 lb/min of liquid refrigerant leave the expansion valve calculate the total refrigerant flow rate through it.

9. Determine the thermal volumetric efficiency for a simple saturation ammonia cycle operating between suction and discharge pressures of 20 psia and 180 psia.

10. A carbon dioxide compressor operates between pressures of 100 and 800 psia. If the suction pressure were reduced to 80 psia, discharge pressure remaining constant, would you expect the thermal volumetric efficiency to increase or decrease? Explain.

11. An ammonia system operating on the simple saturation cycle has a suction pressure of 30 psia and a discharge pressure of 165 psia. (a) For zero clearance determine the piston displacement per ton of refrigeration. (b) For 6 per cent clearance determine the piston displacement per ton and compare with the results from (a). (c) For the clearance range from zero to 10 per cent plot piston displacement per ton against per cent of clearance, observe the shape of the curve, and discuss. (Assume thermal volumetric efficiency as 100 per cent in all cases.)

12. Calculate the required horsepower per ton of refrigeration for the system in Problem 11, taking clearance as zero.

13. For a simple saturation cycle operating between 0°F in the evaporator and 75°F in the condenser compare the piston displacements per ton of refrigeration when using ammonia, carbon dioxide, or dichlorodifluoromethane as the refrigerant.

14. For the cycle of Problem 13 compare the requirements in horsepower per ton of refrigeration for the three refrigerants indicated.

CHAPTER IV

THERMODYNAMIC REFINEMENTS OF THE SIMPLE REFRIGERATION CYCLE

The analysis of refrigeration cycles can readily be reduced to a matter of grouping into cyclic patterns the relatively few processes and types of equipments that find use in refrigeration. All cycles more complex than the simple compression cycle (Fig. 1-1) must necessarily accomplish the same four thermodynamic objectives, but they may do so by replacing a simple path with a more complex, and usually more efficient, one.

4-1. The Expansion Valve. The most obvious means of improving the simple cycle is to replace the irreversible expansion valve with a reversible expanding engine. All irreversible processes represent failure to use effectively the operation of some type of force—thermal if not

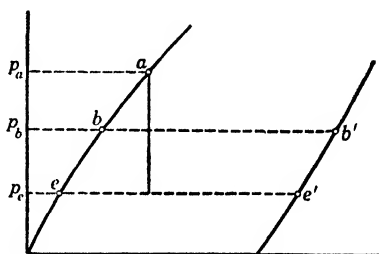


FIG. 4-1.

mechanical—which acts through a spontaneous process. In the case of a constant enthalpy throttling process the opportunity is lost of extracting from the expanding refrigerant some of the work that was supplied to it during compression. If an expanding engine were used and if the pressure drop were thereby caused to occur under isentropic conditions, a substantial fraction of the compressor work requirement would be made up by work done by the expanding vapor on the piston of the expander. In this event the coefficient of performance of the cycle would be materially increased, but, as previously mentioned, at the expense of more costly equipment.

Another method of partially preventing the loss of availability due to the expansion valve process is to carry out the pressure drop in stages and extract, after each stage, that part of the refrigerant which has flashed to a saturated vapor state. In Fig. 4-1, if saturated liquid enters an expansion valve at state a and undergoes a pressure drop from p_a to p_b , the material leaving the valve will consist of a mixture of saturated liquid at state b and of saturated vapor at state b' . The vapor

fraction has already served its full usefulness as a refrigerant and, in fact, contains more energy at state b' than it would if the refrigerant were allowed to continue through the expansion process to final discharge at state e' . Thus if the vapor at b' could be extracted and returned to the compressor two advantages would accrue: (1) the refrigerating effect of the flash would be greater than otherwise; (2) the pressure range through which this fraction of refrigerant must be raised would be reduced from $p_a - p_e$ to the substantially lesser value $p_a - p_b$. The

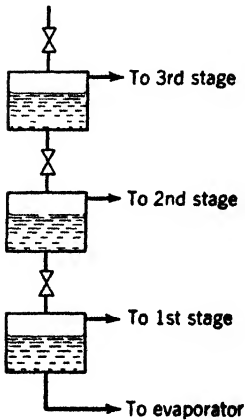


FIG. 4-2.

realization in practice of staged expansion requires, as a corollary, means of carrying out staged compression; this will be discussed in Section 4-5.

To extend the reasoning of the above paragraph, the irreversibility of the throttling process can be reduced by increasing the number of stages through which expansion occurs and "bleeding" the saturated vapor back to the compressor after each stage. In the limit an infinite number of stages would be used, the state of the liquid refrigerant would move continuously down the liquid saturation line, and vapor extraction would occur continuously at a smoothly decreasing pressure from an initial value infinitesimally less than p_a to a final value infinitesimally greater

than p_e . Thus in the ideal case the lowest pressure reached by any small weight of refrigerant would be that value corresponding to which it attained maximum storage of energy; any departure from this ideal necessarily represents a condition in which the refrigerant which has already attained its maximum useful load is allowed to undergo a further, completely useless, pressure reduction. The difficulty, of course, is that, to avoid this useless pressure reduction, additional equipment is needed for the compression processes and the first cost of the system may therefore be substantially greater.

Thus, in summary, the expansion process of the simple saturation cycle can be replaced either with polytropic expansion yielding useful work or with a series of expansion valves separated by flash chambers (Fig. 4-2) from which the saturated vapor formed during each stage expansion is extracted and returned at flash chamber pressure to a compressor.

4-2. The Evaporator. Improvement in effectiveness of the evaporator can be accomplished by reducing the temperature difference between the evaporating refrigerant and the exterior cooled space from which

heat is being transferred. Evaporator problems can most simply be analyzed by dividing them into the classifications of uniform and non-uniform. In a uniform system, such as represented by a cold storage chamber which is held at a fixed temperature by extraction of energy as rapidly as it enters from outside, or is released from within, the temperature of the heat source is fixed and the evaporator will therefore have best economy of operation if it operates at a fixed temperature as small an amount below that of the chamber as the economics of heat transfer surface will permit. If infinite transfer surfaces were available or if the load on the system were infinitesimal, the evaporator temperature could be kept practically equal to that of the cooled space, but in all practical problems a finite, and sometimes large, temperature difference is necessary in order to reduce the first cost of the evaporator. Thus for steady-state systems the only means of improving evaporator operating economy is to increase its size, and this at the expense of greater first cost.

For non-uniform installations the condition is quite different, and the opportunity of developing greater effectiveness by departure from simple saturation cycle conditions is less restricted. Consider, for example, a transient system in which outside air at 60°F is to be cooled as it passes over an evaporator and discharged to a room at a temperature of 40°F . If the evaporator operates at a fixed temperature slightly below that of the leaving air, say at 35°F , the temperature differential with respect to the entering air will be much greater than necessary and, from a consideration of the Carnot efficiency relationship, the work required will be higher than it need be. Suppose now that the single evaporator be replaced by two units, one of which will cool the air through half of the total range—from 60°F to 50°F —whereas the second evaporator will continue the process and reduce the air from 50°F to 40°F . The second evaporator would operate at the same temperature, 35°F , that was originally needed, but the first one could operate at a temperature of 45°F and would be able to carry half of the total load with refrigerant passing through a smaller pressure drop (hence requiring less work for compression) than had previously been the case. Again, however, a system arranged to permit evaporator operation in this way would have to have additional compressor equipments to permit raising the two fractions of the refrigerant through different pressure ranges.

If carried to the limit the above arrangement would lead to use of an infinite number of evaporators each carrying infinitesimal load and each differing in temperature from the mean temperature of the air passing over it by only a very small amount. Visualization of such a system

brings out the striking fact that the use of such series-operated multiple evaporators is no more than a continuation of the idea, already discussed, of using an infinite number of expansion valves. Looking back on the multiple valves we recall that, owing to energy extracted in the process of cooling the remaining liquid, some refrigerant flashes in each expansion; with infinite evaporators exactly the same situation would exist except that a larger fraction of refrigerant would be flashed to provide external refrigerating effect (air cooling) as well as internal refrigerating effect (liquid refrigerant cooling). Thus an expansion valve is, in effect, a non-uniform operating evaporator since flash occurs

continuously as the temperature of the fluid drops.

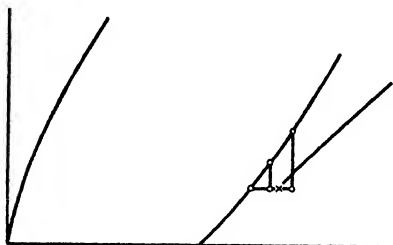


FIG. 4-3.

4.3. Sectionalizing. The concept of multiple evaporators is known technically as sectionalizing, and it finds wide application in systems involving either non-uniform conditions or multiple-load uniform conditions; thus, in a cold storage warehouse with rooms required to

be maintained at different but fixed temperatures, uniformity exists, but sectionalizing will permit carrying the higher-temperature cold storage rooms with higher-temperature evaporators.

In some installations the requirement of maintaining different cold rooms at different temperatures will necessitate maintenance of different temperatures among the evaporators even when the system does not include compression equipment capable of extracting vapor at more than one pressure. In this case all vapor going to the compressor must be at a pressure corresponding to that of the lowest-temperature evaporator. Saturated vapor from other evaporators will then be throttled to the suction pressure and, as an examination of the pressure-enthalpy diagram of Fig. 4-3 shows, the throttling process will impart superheat to the saturated vapor since, although the process occurs at constant temperature, the pressure is reduced below the saturation value for that temperature.

For operation under these conditions the primary advantage of sectionalized evaporators is obviously not attained, since for all evaporators other than that which operates at lowest temperature the evaporator pressure is *not* the lowest cycle pressure of the refrigerant fraction in question. In spite of this, however, operation on such throttling cycles is relatively common since it provides more effective control than could be obtained by allowing all evaporators to operate at the lowest

temperature. Another interesting characteristic of this procedure is that (see Fig. 4-3) the mixed refrigerant entering the compressor has a definite degree of superheat even though all evaporators discharge saturated vapor; superheat at compressor suction is thermodynamically undesirable, but it is usually sought as a means of practical protection of the compressor from hazardous quantities of entrained liquid refrigerant that might be present if the vapor were saturated.

4.4. "Flooded" vs. "Dry" Operation. Another equipment arrangement frequently used is a separating chamber installed between the last expansion valve and the evaporator; saturated vapor is extracted prior to admission of the refrigerant to the evaporator. Thermodynamically this arrangement is identical, for a simple saturation system, with that which allows all the refrigerant to pass through the evaporator, since the state of that fraction of refrigerant which is already in saturated vapor form is unchanged during transit through the evaporator; the only effect resulting from prior extraction of the vapor is the reduction of the flow rate through the evaporator and hence the reduction of the pressure drop for a given size of tube. This may or may not be desirable since in some cases the increased velocity when the vapor is present is sufficient to increase materially the film coefficient of heat transfer and thereby permit reduction in the requisite area of heat transfer surface. Evaporator operation with by-pass of initially saturated vapor is known as "flooded operation," whereas when the vapor-liquid mixture passes through the evaporator the system is said to operate "dry." Flooded operation assures effective distribution of refrigerant to all tubes of the evaporator; the separator in a flooded system serves also to protect the compressor from "slugs" of liquid.

4.5. The Compressor. The thermodynamic improvements over the simple saturation cycle which are attainable through changes in expansion valve and evaporator operation have as prerequisites the availability of compression equipment capable of receiving vapor at more than one suction pressure. Thus, in order to realize the advantages already described, a system of stage compression is essential. However, aside from its value as a means of allowing multiple valve and multiple evaporator operation, staging of the compression process affords additional thermodynamic gains which could still be realized even if simple expansion and evaporation were used.

To clarify the respective gains and indicate their individual significance consider a stage compression system which in all other respects corresponds to a simple saturation cycle; if the compression process is carried out in two isentropic stages and if the connecting pipe between discharge from the first stage and admission to the second stage is

insulated, the combined process will be identical, for ideal compressors, to that of a single-stage compression; hence for the ideal cycle there would be no thermodynamic advantage from simple staging of the compression process.

In real compressors, however, the influence of the temperature change undergone by the refrigerant during the compression process is to cause a heat flow from the warm cylinder wall to the vapor during admission and during the early part of compression followed by a reverse flow of heat during the latter portion of the compression and during the discharge. Thus even in cases where there is no net heat transfer between the compressor cylinder and the surround there will nonetheless be a non-adiabatic condition within the compressor with consequent departure of the compression process from reversibility and a resultant entropy increase of the refrigerant.

The quantity of cyclic heat flow to and from the cylinder walls will depend on the magnitude of the temperature difference between walls and vapor. If, for example, vapor entered at 0°F and left at 100°F the average cylinder wall temperature might well be of the order of 50°F with a consequent maximum temperature differential at the beginning and at the end of the compression process of approximately 50°F . Suppose now that this same vapor were to be compressed in two stages, the vapor temperature change being from 0°F to 50°F in the first stage and from 50°F to 100°F in the second stage; an average temperature of the cylinder wall in the first stage might be 25°F and an average temperature of the second stage 75°F , giving a maximum temperature differential in each of the stages of 25°F in contrast with the value twice that great which was found with single-stage compression.

As a boundary case consider compression as occurring in an infinite number of stages, the temperature rise of the vapor in any one stage then being infinitesimal and the temperature difference between the vapor and the cylinder therefore never attaining a finite value. For this case there would be no finite transfer of energy between the cylinder and the vapor, and the entire compression would therefore be carried out reversibly and hence with minimum (for an adiabatic process) work requirement. Note that the effect of stages in this case is not to improve the ideal thermodynamic efficiency of a simple saturation cycle (since no irreversibility occurs in the ideal cycle), but rather to provide a system which will overcome, at least in part, the difficulties attendant on the operation of actual equipments. The limitation "at least in part" is essential because it obviously would be as impossible in practice to utilize infinite stages of compression as it would be to attain single-stage compression with internal adiabatic conditions.

4-6. Intercooling. When multistage compression is used with intercooling of the vapor between successive stages, the overall compression process can be made to approach in performance the isothermal rather than the isentropic. It has already been shown that extraction of heat during the process of compression leads to a reduction in the required work. The speed of the compression process in most modern compressors is such that relatively little heat extraction can occur, but essentially the same effect can be attained if compression is allowed to occur in small isentropic stages with cooling carried out in special equipments—aftercoolers or intercoolers—located between successive stages.

A boundary case can be visualized as consisting of an infinite number of stages with intercooling at each stage which reduces the temperature of vapor going to the suction of the following compressor to a value nearly as low as that of the available cooling water. Aside from the initial temperature rise of the vapor from the evaporator temperature to the temperature of the cooling water, it is evident that the overall process amounts to the equivalent of isothermal compression from a pressure corresponding to that at which the isentropically compressed vapor from the evaporator reaches the temperature of the cooling water to a pressure equal to slightly more than the saturation value of the refrigerant pressure at a temperature equal to that of the cooling water. In effect the overall compression would then be equivalent, in work requirements, to an initial isentropic compression followed by an isothermal compression, the intermediate pressure at which transition would take place being determined by the temperature of the available cooling water.

The complete isothermal compression of a refrigerant vapor is unthinkable since if the temperature were held at the evaporator value the addition of energy as work would lead to isothermal condensation accompanied by a pressure increase of the liquid; but the purpose of compression in refrigeration is to raise the pressure only sufficiently to permit condensation. If the material could be kept at evaporator temperature during compression there would obviously be no need for any pressure rise; it cannot be kept at evaporator temperature because of the lack of a suitable cooling medium. What can be done, however, is to use liquid refrigerant from the condenser as a coolant in each of the intercoolers, thereby reducing the refrigerant vapor from a superheated to a saturated state. At the limit, such a procedure would be carried out with an infinite number of steps in each of which the vapor would be isentropically compressed over an infinitesimal pressure range then immediately desuperheated to the saturated vapor temperature corresponding to the slightly increased pressure of compressor dis-

charge; the next stage would then follow, and so on, until discharge of the vapor from the last stage at condenser pressure, but with no finite degree of superheat. In effect, such a series of stages would give an overall equivalent compression process such as could be represented by compression from evaporator to condenser pressure along a process line that would superimpose on the saturated vapor line.

Note that in order to carry out infinite stage compression with liquid intercooling there would necessarily be an infinite number of expansion valves as well as stages of compression. Saturated liquid refrigerant would leave the condenser and a continuously decreasing fraction would follow a succession of states down along the saturated liquid line to the evaporator pressure, then evaporate from a saturated liquid to a saturated vapor and be compressed up along the saturated liquid line to final discharge at the condenser pressure.

If, in the cycle described above, the evaporator load were non-uniform and consisted in reducing the temperature of a fluid (as air) from some temperature t_a to a lower value t_b , the cycle could then combine an infinite number of expansion valves, with an infinite number of both evaporators and compressors to give a cycle identical on the P - h chart, but differing thermodynamically in that the actual evaporator loads would be distributed over the pressure range p_a to p_b so that only an infinitesimal quantity of excess refrigerant would ever drop through the entire $p_a - p_b$ pressure reduction. An idealized cycle of this kind is of much greater practical importance than might at first glance be thought since it provides a definite and valuable clue to the best performance that could possibly be realized with any given system. This limiting cyclic efficiency can be calculated readily and used as a criterion for evaluating the performance of the real cycle.

In one other respect the infinite-stage, refrigerant-intercooled, compression cycle is of interest. Basically, in terms both of analytical procedures and of required equipments, this cycle is the exact reverse of the regenerative feed-water steam power plant cycle with an infinite number of water heaters. The two differ only in that the steam cycle uses a reversible feed-water pump as the counterpart of the irreversible expansion valves that find use in the refrigeration cycle. Thus, once again, the processes and cycles of refrigeration evidence themselves as reverse cycle thermodynamics.

An improvement over the use of refrigerant to attain complete desuperheating can be realized by using cooling water in an intercooler ahead of the liquid refrigerant desuperheater. In practice not only are an infinite number of stages of compression impractical, but so also are a finite number greater than, at most, three or four. Thus the temper-

ature rise of the refrigerant during compression is significantly large and, in many cases, cooling water can be used to remove part of the superheat prior to attaining final approach to a saturated liquid condition by means of liquid refrigerant. Each Btu of cooling effect attained by use of refrigerant is costly, since the extracted Btu must be raised in thermal level and later disposed of in the condenser whereas all heat that can be transferred directly to cooling water is immediately discarded from the system.

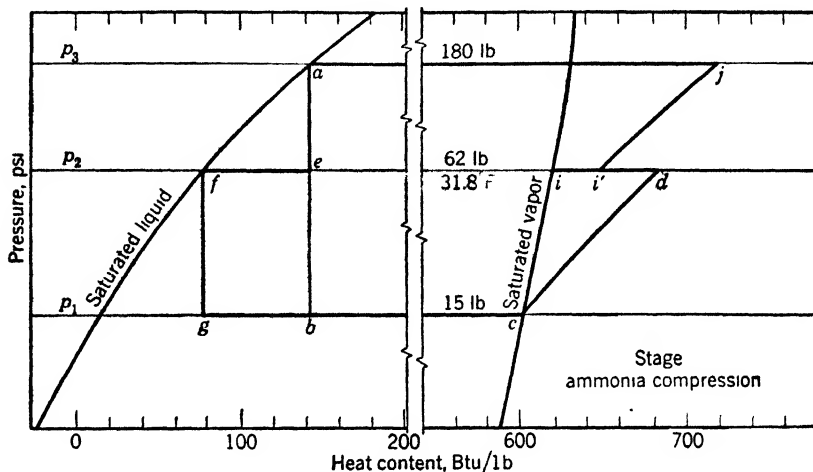


FIG. 4-4.

Example. Using an evaporative pressure of 15 psia and a liquefaction pressure in the condenser of 180 psia, find the useful refrigeration and the work of compression per pound of ammonia leaving the liquid receiver.

Solution. A preliminary investigation indicates that equal work is possible in both cylinders with a receiver pressure of 62 psia. The discharged vapor from the low-pressure cylinder is cooled, by means of water, to 85° F. From Fig. 4-4, the quality after the passage of the first pressure-reducing valve is

$$x_e = \frac{h_a - h_f}{L_{p_2}} = \frac{143.3 - 77.7}{543.2} = 0.1208$$

If the vapor, after compression in the low-pressure cylinder, is to be cooled only to 85° F, the useful refrigeration at 15 lb evaporating pressure becomes (in terms of 1 lb of refrigerant flowing through the condenser)

$$Q_{15 \text{ lb}} = (1 - x_e)(h_c - h_f) = (1.0 - 0.1208)(602.4 - 77.7) = 461.3 \text{ Btu}$$

The work of compression in the low-pressure cylinder (per 1 lb flow through condenser)

$$= 0.8792 (h_d - h_c) = 0.8792 (685.1 - 602.4) = 72.71 \text{ Btu}$$

The cooled discharged vapor, from the low-pressure cylinder, at 62 psia and 85° F, mixes with the vapor separated from its liquid in the accumulator at 62 psia and dry and saturated. The condition of the mixture may be found by considering that the enthalpy of the mixture is equal to the sum of the enthalpies of the two vapors before mixture or

$$(0.1208 \times 620.9) + (1.0 - 0.1208) 653.2 = 649.3 \text{ Btu}$$

Therefore $t = 78.3^\circ \text{ F}$. The work of compression in the high-pressure cylinder is then (using points i' and j as the initial and final states during compression and 1 lb of the refrigerant)

$$AW = 721.5 - 649.3 = 72.2 \text{ Btu/lb}$$

and the coefficient of performance is $461.3/144.9 = 3.183$. If single compression is used the coefficient of performance becomes 2.79.

If cooling of the discharged vapor from the low-pressure cylinder is extended to the temperature of saturation corresponding to the intermediate pressure, the weight of liquid ammonia passing through the second pressure-reducing valve will be smaller by the amount of the liquid required for this cooling. Let this weight be y pounds of the original unit weight of liquid ammonia. Then the amount passing to the evaporator and the low-pressure cylinder will be $1.0 - x_e - y$ pounds. The magnitude of y may be found from the relation

$$(1.0 - x_e - y)(h_{i'} - h_i) = yL_{p_2}$$

or

$$(1.0 - 0.1208 - y)(653.2 - 620.9) = y \times 543.2$$

from which

$$y = 0.0493 \text{ lb}$$

Therefore the useful refrigeration, at 15 psia

$$= (1.0 - 0.1208 - 0.0493)(602.4 - 77.7) = 435.5 \text{ Btu/lb}$$

The work in the low-pressure cylinder

$$= 0.8302 (685.1 - 602.4) = 68.6 \text{ Btu/lb}$$

The work in the high-pressure cylinder

$$= 685.2 - 620.9 = 64.3 \text{ Btu/lb}$$

The total work

$$= 68.6 + 64.3 = 132.9 \text{ Btu/lb}$$

The coefficient of performance,

$$\text{cop} = \frac{435.5}{132.9} = 3.277$$

As a comparison, the value of the coefficient of performance for simple compression, from 15 psia suction to 180 psia discharge pressure, is

$$\text{cop} = \frac{459.1}{766.7 - 602.4} = \frac{459.1}{164.3} = 2.794$$

However, because of the high temperature of discharge, 302° F, the thermal

volumetric efficiency e_v has the value 0.753 (from equation 3-11), which is much less than in stage compression; therefore the advantage of stage compression is even more marked.

Low-temperature refrigeration, from -50° to -100° F or lower, has to be designed for three stages for economical results. Cooling between stages cannot always be secured with the use of water, but by liquid intercooling it is always possible. Table 4-1 (calculated by Tucker) gives a comparison of different methods of operation and their theoretical results.

TABLE 4-1
THEORETICAL CALCULATION FOR STAGE COMPRESSION
(86° F Liquefaction and -100° F Evaporation)

Calculation	Refrigerant	Number of Stages	Compression Ratio per Stage			Volume of Vapor at -100° F	Coefficient of Performance	Remarks
			1	2	3			
I	NH ₃	3	8.4	4.6	3.2	62.4	1.67	Liquid intercooling
II	NH ₃	3	5.0	5.0	5.0	62.2	1.59	Liquid intercooling
III	NH ₃	3	5.0	5.0	5.0	62.1	1.59	Water intercooling
IV	NH ₃ and ethane	2	6.5	6.5	58.1	1.36	NH ₃ used to condense ethane
V	F-12	2	14.1	4.9	80.5	1.23	

4-7. Dual Compression. The principal disadvantage of stage compression is that it greatly increases the first cost of the equipment, since either an added cylinder (if single-acting machines are used) or an added cylinder end (if double-acting machines are used) is required for each stage of compression. One method of reducing the first cost, in systems in which vapor is to be extracted at only two suction pressures, is to use a specially constructed multicompression or dual-compression cylinder. A compressor of this type is of unusual thermodynamic interest since it is the only type of equipment employed in standard mechanical refrigeration systems which makes use of non-flow, in contrast with steady flow, operation. Dual-compression makes use of the fact that flow will occur from a line to a receiver, irreversibly, if the line pressure exceeds the receiver pressure. Thus in the usual type of dual-compression operation, vapor at the pressure existing in the lowest-temperature evaporator flows into a cylinder in the usual way as the piston uncovers the cylinder space.

During entrance of this low-pressure vapor the flow work of the refrigerant is dissipated against the resistance of the moving piston during the constant pressure admission process. Once this vapor has been admitted and the low-pressure suction valve is closed the thermodynamically unusual step occurs of immediate opening of a suction valve connecting the cylinder to a higher-pressure suction line. Because of the pressure differential the high-pressure vapor flows into the cylinder, but as the piston is already on dead center the entering fluid is unopposed by any piston resistance and no useful work is accomplished. For this reason the flow work of the entering suction vapor is not dissipated, but remains present in the cylinder as increased internal energy of the mixture of vapors and hence manifests itself as temperature rise. In effect, therefore, the high-availability flow work of the entering vapor is dissipated as an increase in random motion with consequent loss of availability. Since the process of admission of high-pressure suction vapor occurs irreversibly it cannot be represented as a process line and the final state of the mixture can only be determined by setting up an energy balance on the system. This is the only case in the field of refrigeration cycle analysis where enthalpy loses its energy significance as represented by the fact that the total enthalpy of the mixed vapor at the start of the compression process exceeds the total energy brought into the cylinder by the two streams of entering suction vapors. The explanation of this apparent anomaly resides in the fact that the enthalpy of the mixture is no longer a property having significance as energy.

Dual-effect compression is also of interest because it is one of the unusual cases in which the engineer reverts from a reversible process (a single-stage isentropic compression) to a thermodynamically less desirable set of processes (dual-admission, with irreversibility) in order to gain a thermodynamic advantage which, from the overall standpoint, is more than sufficient to offset the loss of availability associated with the irreversible part of the cycle. Thus dual compression is less desirable than two-stage compression, but it possesses definite thermodynamic advantage over single-stage isentropic compression.

4·8. Dual-Effect Analysis. A thermodynamic analysis of the dual compression cycle* can be made if a few assumptions are accepted. For example, the quality of the vapor at *a* (Fig. 4·5) must be selected, and in the following it is taken to be dry and saturated as an approximation of the combined effect of throttling through the suction valve and connections, the expansion in the clearance volume through a greater pressure range *de* than through the direct compression range *bg* (although

* Sparks, *Refrig. Eng.*, Vol. 30, No. 3, September, 1935.

the action ab has a temperature rise), and the cylinder heating during the suction stroke. The weights of refrigerants follow.

If the pressure at a , at the end of the suction stroke, is taken as P_1 , the weight of the vapor in the cylinder, assumed to be dry and saturated, is

$$w_c + w_1 = \frac{V_a}{\bar{V}_a} \tag{4.1}$$

At b , at the end of the constant volume process ab , the weights are

$$w_c + w_1 + w_2 = \frac{V_b}{\bar{V}_b} = \frac{V_a}{\bar{V}_b} \tag{4.2}$$

where \bar{V}_b is the specific volume of the mixture at the state b , w_1 is the

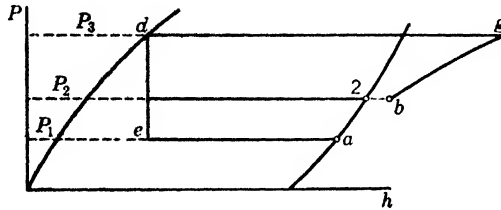


FIG. 4.5.

weight of the vapor entering at the low suction pressure, w_2 is the weight of the vapor entering at the high suction pressure.

The weight of the vapor in the clearance volume is w_c .

If the mixing process is adiabatic,

$$[(w_c + w_1)u_a] + w_2h_2 = (w_c + w_1 + w_2)u_b \tag{4.3}$$

but

$$u_b = h_b - AP_b\bar{V}_b \quad \text{and} \quad w_2 = \frac{V_a}{\bar{V}_b} - (w_c + w_1) \tag{4.4}$$

Substituting in equation 4.3 the corresponding values from equation 4.4 and simplifying gives

$$(h_b - h_2) \frac{\bar{V}_a}{\bar{V}_b} = AP_b\bar{V}_a - (h_2 - u_a) \tag{4.5}$$

As \bar{V}_b and h_b are dependent variables they can be evaluated easily by plotting $(h_b - h_2)/\bar{V}_b$, the numerical value of which is determined by dividing V_a into the right-hand side of equation 4.5, against the assumed values of the temperature taken from the superheat tables for the pressure P_b .

The weight of the clearance vapor, w_c , may be calculated by finding the temperature of the vapor at the end of the compression process and assuming that this is also the temperature at discharge. From the discharge temperature and pressure the specific volume of the vapor may be found in the tables for the superheated refrigerant. The weight of clearance vapor is then equal to the clearance volume divided by the specific volume of the refrigerant.

In the dual compression problem the general energy equation reduces to

$$AW_{in} = w_g h_g - [(w_1 + w_c)h_a] - w_2 h_2 + Q_{out} \quad \text{Btu/min} \quad (4.6)$$

where AW_{in} is the heat equivalent of the work of compression, and $Q_{out} = 0$ for an isentropic process.

Example. Find the coefficient of performance for an ammonia compressor working on the dual compression cycle with 15 and 45 psia (p_1 and p_2) evaporating and 169 psia (p_3) condenser pressure. Assume that the quality of the vapor in the cylinder, at the point of uncovering of the cylinder ports, is unity; that there is no clearance; that the compression is isentropic; and that there is no wire drawing through the ports and valves. The vapor entering the ports at the pressure p_2 may have any reasonable quality, and in this problem it is assumed to be unity.

Solution. Assume that the volume at the end of the suction stroke, a , is 100 cu ft. Then, from equation 4.1, as $w_c = 0$, at point a

$$w_1 = \frac{100}{17.67} = 5.6592 \text{ lb/min}$$

and, from equation 4.2, for the point b where the volume is still 100 cu ft and the compressor is still on dead center

$$w_1 + w_2 = \frac{V_b}{\bar{V}_b} = \frac{V_a}{\bar{V}_b} = \frac{100}{\bar{V}_b}$$

but, from equation 4.5, as \bar{V}_b and h_b are unknown,

$$(h_b - h_2) \frac{V_a}{\bar{V}_b} = AP_b V_a - [w_1(h_2 - u_a)]$$

but as

$$u_a = h_a - 0.1852 p_a V_a$$

$$\begin{aligned} \frac{h_b - 616.9}{\bar{V}_b} &= \frac{144 \times 45}{778} - 0.056592 [616.9 - 602.4 + (0.1852 \times 15 \times 17.67)] \\ &= 8.3290 - 3.5986 \\ &= 4.7304 \end{aligned}$$

As the values of h_b and \bar{V}_b are unknown we may assume several values for the temperature of the mixture and find the corresponding values from the

superheated table for 45 psia. When these answers for the left-hand side of equation 4.5 are compared with the known value in the problem, 4.7304, by interpolation or by plotting, the correct values of the temperature of the mixture are 76.05° F, $h_b = 651.29$, $s_b = 1.37084$, and $h_g = 744.44$. The work performed is found from equation 4.6:

$$\begin{aligned} w_2 &= \frac{100}{V_b} - 5.6592 \\ &= \frac{100}{7.2643} - 5.6592 \\ &= 13.7684 - 5.6592 \\ &= 8.1092 \text{ lb/min} \end{aligned}$$

$$w_3 = 13.7684 \text{ lb/min}$$

$$\begin{aligned} AW &= w_3 h_g - w_1 h_a - w_2 h_2 \\ &= (13.7684 \times 744.44) - (5.6592 \times 602.4) - (8.1092 \times 616.9) \\ &= 1836.2 \text{ Btu/min} \end{aligned}$$

$$\begin{aligned} \text{Refrigeration} &= [8.1092 \times (616.9 - 138.9)] + [5.6592 (602.4 - 138.9)] \\ &= 6499.31 \text{ Btu/min} \end{aligned}$$

$$\text{Coefficient of performance} = \frac{6499.31}{1836.2} = 3.5396$$

4.9. Superheat at Suction. The analysis of compression performance has so far been based on isentropic compression from a saturated state (such as occurs in the simple saturation cycle) as an initially assumed condition which is to be improved upon. Actually, most compressors depart in one definitely undesirable respect from the conditions of this simple process. In order to protect the compressor from possible damage due to the carrying over of liquid refrigerant in the suction line it is customary to operate with some definite amount of superheat in the vapor leaving the evaporator. For such a case it can be readily shown that the power requirement of the compressor, operating between fixed suction and discharge pressures, will increase with the superheat of the entering vapor. Thus it is eminently desirable that the degree of superheat be held to that minimum which may be considered essential for protection, but under no conditions be allowed to exceed this value.

Example. A compressor using ammonia as a refrigerant operates between pressures of 30 psia and 150 psia. The enthalpy of the refrigerant at evaporator entrance (following liquid subcooling and subsequent expansion) is 120 Btu/lb. For a fixed load on the system compare the work requirement of the compressor when suction vapor is saturated and when it possesses 25° F of superheat.

Solution. The enthalpies of saturated vapor at 30 psia and of vapor superheated (at 30 psia) by 35° F are, respectively, 612 and 626 Btu/lb. The respective enthalpies after isentropic compression from these initial states are 712 and 730 Btu/lb; therefore the work required per pound of refrigerant is $712 - 612 = 100$ Btu for the saturated suction vapor and $730 - 626 = 104$ Btu for the superheated vapor. Actually, however, the relative power for the two cycles does not differ in this same proportion because, owing to a somewhat greater refrigerating effect, the required weight rate of refrigerant is less in the case of the superheated initial state. The relative refrigerating effects are $612 - 120 = 492$ Btu/lb and $626 - 120 = 506$ Btu/lb; therefore the refrigerant flow rates are in the ratio of 1.025 to 1 and the power requirements are, respectively, 102.5 Btu and 104 Btu. Thus the effect of superheat is here to increase power by a little over 1 per cent.

From the above example it is evident that the effect on power of a small amount of superheat at suction is not great. When the superheat is large, as it may be when evaporator vapor is blended with vapor discharged from the compression of a lower stage, the effect on power may be of the order of 10 to 20 per cent.

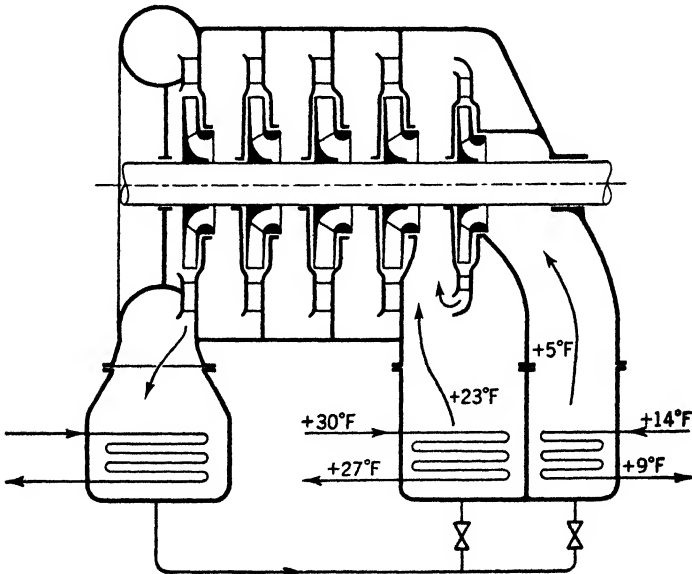


FIG. 4-6.

4-10. Reverse Extraction. One rather unique form of staged compression is that occurring in centrifugal compressors which are constructed to admit suction vapor at more than one pressure. Figure 4-6 shows diagrammatically an arrangement of this kind which is readily

recognizable as being the equivalent of a reverse-acting extraction turbine. Obviously the ratio of vapor weights at the two suction pressures will be fixed as an operating characteristic of the particular compressor, but the system offers interesting possibilities for achieving staging without the use of separate compressors, or compressor cylinders, such as would be necessary if reciprocating equipment were used.

To summarize, the methods by which the effectiveness of compression can be increased are limited to: (1) increasing the rate of heat transfer from the compressor to the surround (or to a cooling water jacket) so that the compression process will depart from isentropic in the direction of isothermal; (2) introducing stage compression with intercooling by means of water or liquid refrigerant or a combination of the two; (3) introducing dual-effect compression to enable a single cylinder to receive vapor at two suction pressures.

4-11. The Condenser. Thermodynamic effectiveness of the condensation process is attained by the inverse of methods used to increase the performance characteristics of evaporation. The condenser should operate at the lowest temperature (hence pressure) that is consistent with the requisite rate of heat transfer to the cooling water.

PROBLEMS

1. A refrigeration system is designed to cool 10,000 lb/hr of water from 60° F to 40° F. Operating conditions necessitate a condenser pressure of 160 psia. (a) If the system operates on the simple saturation cycle with evaporator temperature of 35° F (using ammonia as refrigerant), calculate the required horsepower. (b) If the load is divided evenly between two simple saturation systems, both operating with 165 psia discharge pressure, but one with 35° F evaporator and the other with 45° F evaporator, determine the required horsepower.

2. For the load of Problem 2 assume use of an infinite number of evaporators each carrying an infinitesimal fraction of the load and reducing the water temperature through an infinitesimal drop. By plotting a curve of horsepower against number of evaporators (taking 1, 2, and 4) extrapolate to estimate the horsepower required with infinite evaporators. Investigate the possibility of establishing an analytical evaluation of the horsepower for the system with an infinite number of evaporators.

3. A refrigeration system using ammonia and operating on the simple saturation cycle carries a 30-ton load with evaporator temperature at -50° F and condenser temperature of 60° F. If the load could be so divided that 10 tons could be carried at an evaporator temperature of -20° F, determine the total required horsepower assuming that the compressor receives vapor at the pressure of the -50° F evaporator (vapor from the -20° F evaporator being throttled to the lower pressure after discharge from this higher temperature evaporator).

4. Would the system of Problem 3 be thermodynamically more effective if the evaporator in the higher temperature room were so sized that it could maintain the desired room temperature when operating at the same saturation temperature (-50° F) as is needed for the room with the lower temperature? Explain.

5. Separate simple saturation systems using ammonia as the refrigerant carry

the 20-ton and the 10-ton loads described in Problem 3. Determine the total required horsepower and compare with the horsepower requirements as determined from Problems 3 and 4.

6. A refrigeration system is designed so that there is no possibility of heat loss from the refrigerant in the piping between evaporator and compressor. To prevent the possibility of having entrained liquid leave the evaporator, controls are arranged to assure 10°F of superheat at evaporator discharge. Would the power requirements for a system of this kind be greater for flooded than for dry operation? Explain fully.

7. An ammonia system operating on the simple saturation cycle has an evaporator pressure of 25 psia, a condenser pressure of 160 psia, and operates with a flooded evaporator. (a) Calculate the refrigerant flow rate through the condenser in pounds per minute per ton. (b) Calculate the flow rate through the evaporator in pounds per minute per ton. (c) Calculate the horsepower per ton of refrigeration. (d) If the same system operated with a dry evaporator calculate the flow rate through the evaporator and the horsepower per ton.

8. If the system of Problem 7 operated flooded, but with 15°F superheat at evaporator discharge, determine the number of degrees of superheat in the mixed vapor entering the compressor.

9. An ammonia refrigeration system, operating on the simple saturation cycle, has suction and discharge pressures of 10 psia and 165 psia. (a) Compare the horsepower per ton for isentropic compression with the power that would be needed if the compression process were isothermal. (b) Assume availability of cooling water such that isentropic compression would occur from evaporator temperature to a temperature of 60°F but, from this state on, the process of compression could be made to occur isothermally, and compare the power requirements with those indicated in (a) of this problem.

10. A simple saturation cycle using ammonia between pressures of 10 psia and 165 psia is to be replaced with two stages of compression and water intercooling of vapor from the first stage to 50°F as entering temperature to the second stage. (a) Determine the intermediate pressure corresponding to which the work of both stages will be the same. (b) For the same stroke in both stages what would be the ratio of cylinder diameters of the first stage to the second stage?

11. A system is similar to that of Problem 10 except that liquid refrigerant is used to intercool from 50°F (at discharge of vapor from the water intercooler) to the saturation temperature corresponding to the intermediate pressure. (a) Calculate the intermediate pressure for equal work and compare this value with the one obtained in Problem 10. (b) Compute the ratio of cylinder diameters and compare with the result from Problem 10.

12. For the same evaporator and condenser temperatures as in Problem 10, assume that dichlorodifluoromethane is used as a refrigerant and determine whether or not the advantage of staged compression (expressed as a per cent saving in power) is as great for this refrigerant as it is for ammonia.

13. A quick-freezing plant requires that the evaporator operate at -50°F . If the condenser temperature is 90°F determine the necessary pressure at suction of the second and third stages of compression such that all three stages will have equal work requirements; compression is assumed to be isentropic in all stages and intercooling is to a temperature of 60°F .

14. Investigate the possibility of securing thermodynamic advantage through use of triple-effect compression. When the equations for triple-effect operation have

been written, is their solution more difficult than that of the double-effect equations?

15. For any selected system investigate the relationship between clearance and the ratio of weights of vapor handled in the low-pressure and high-pressure suctions of a double-effect compressor.

16. A refrigerating machine operates at 40° F evaporating and 90° F liquefaction temperature. Calculate the horsepower and the piston displacement per ton of refrigeration required for ammonia, methyl chloride, sulphur dioxide, and Freon-11, 12, 21, and 113. Assume no clearance, and neglect values for e_s .

17. Taking the value of $k = 1.15$ for Freon-12 and using Table 8·10a, find the theoretical piston displacement and the horsepower per ton of refrigeration for 20 psia and 100 psia evaporating and liquefaction pressure respectively. Check the answers by means of the Mollier diagram (Fig. 8·4).

18. A two-stage ammonia compressor is to operate with 10, 60, and 200 psia. The discharge vapor from the low-pressure cylinder is cooled to 85° F with water. Find the theoretical coefficient of performance if the compression is adiabatic.

19. Calculate the theoretical coefficient of performance for a compressor using Freon-12 in three equal pressure stages and three pressure-reducing valves. The compressed vapor is to be cooled, after compression, with water to 85° F if the vapor temperature permits it. The evaporation temperature is -100, and the liquefaction temperature is 80° F.

20. An ammonia refrigerating machine operating at 20° F evaporating and 100° F condensing temperature, and dry saturated vapor in the suction header at the compressor, has had added the equipment in order to operate with dual compression with an ammonia intermediate pressure of 59 psia in order to cool the remaining liquid from the condenser to 42° F. Assuming no losses, find the coefficient of performance for (a) the simple cycle, (b) the dual cycle, (c) the increase in refrigeration at 20° F.

21. Calculate the coefficient of performance for Freon-12, both graphically and analytically, for 0° F evaporation and 100° F liquefaction. Use dry, saturated vapor at the beginning of the compression, which is isentropic.

22. A refrigerating load is at the rate of 100,000 Btu/hr for 8 hr during a 24-hr day. The refrigerating machine operates for 16 hr of the 24-hr day, but it is shut down for 2 hr during the 8-hr period. Find (a) the weight of water to be stored and cooled to 35° F if the allowable rise of temperature of the storage water is to be 10° F; (b) the size of a Freon-12 compressor for evaporation at 30° F, liquefaction at 110° F, the ratio of the stroke to the diameter being 0.8, piston speed 400 fpm, and the thermal volumetric efficiency, e_s , 0.8. Compressor to have twin cylinders.

23. A small cold storage plant has the following load: manufacture of 100 lb of ice, at 220 Btu/lb hr, 5000 Btu/hr heat leakage, 2000 lb of produce cooled from 60° F/hr to 40° F/hr (specific heat to be taken as 0.8). Find the required size of an ammonia compressor of the twin vertical single-acting enclosed type for operation at 200 rpm, 80° F liquefaction and 25° F evaporation, ratio stroke to diameter 1.0, neglecting e_s .

24. An ammonia two-stage cycle operates between the pressure limits of 180 lb and 15 lb with an intermediate pressure of 62 lb, all per square inch absolute as indicated in Fig. 4·4. Find the coefficient of performance if the drop of pressure is from a to b and cooling of the discharged gas from d to i' is obtained by means of water. Compression in the high-pressure cylinder is initially at 62 lb and 85° F.

25. A Freon-12 refrigerating machine operates on the dual cycle between the

temperature limits of 110° liquefaction and 30° F evaporation with an intermediate pressure of 58 psia. The refrigerant evaporating at the intermediate pressure cools to 46.7° , the liquid passing to the 30° F evaporating process. The compressor has no clearance, and the quality of the vapor at the end of the suction stroke and entering through the cylinder ports at 58 lb is just dry and saturated. Find the coefficient of performance for the dual cycle and for the simple cycle without the intermediate evaporator.

26. During the summer operation of an ice manufacture plant using ammonia as the refrigerant and operating at 90° F liquefaction and 16° F evaporation, the water for filling the ice cans is cooled from 85° to 40° F by means of liquid ammonia evaporating at 62 psia. Assume that the heat to be removed by the refrigerating machine, other than that for the cooling of the water to 32.8° F, is 175 Btu/lb of water. Compare, by means of the coefficient of performance, the dual cycle with the usual cycle in ice manufacture.

27. Ammonia operates between the temperature limits, in a refrigerating plant, of 0° F evaporating and 90° F liquefaction. Find the coefficient of performance for adiabatic compression when (a) the discharge vapor from the compressor is just dry and saturated; (b) the discharge vapor is 150° F; and (c) the suction to the compressor is just dry and saturated at 0° F.

28. The compressor in Problem 26 is used to subcool the liquid refrigerant from 90° to 24.4° F by employing an intermediate pressure of 53 psia. Find the coefficient of performance of the dual, as well as the simple, cycle.

29. Water is to be heated by a two-stage ammonia refrigerating machine operating with 0° F evaporation and 120° F liquefaction temperature, and 130 psia intermediate pressure. The discharge vapor from the low-pressure cylinder is cooled to 90° F, and the discharged vapor from the high-pressure cylinder to liquid at 120° F. Find the theoretical coefficient of performance.

CHAPTER V

ANALYSIS OF COMPLETE SYSTEMS

5-1. Comparative Cycle Analysis. The principles developed in the preceding sections can most effectively be visualized if applied to a single complex system. Consider, therefore, a cold storage warehouse which is required to handle the following simultaneous loads:

210 tons with an evaporator temperature of 6°F

105 tons with an evaporator temperature of -9°F

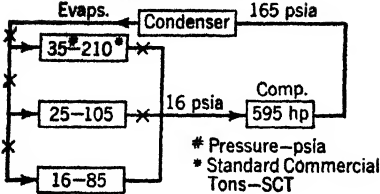
85 tons with an evaporator temperature of -25°F

Ammonia is used as a refrigerant, and the fluid leaving the condenser is subcooled to 60°F ; pressures in the system are 16, 25, 35, 165 psia in the -25°F , -9°F , 6°F evaporators and in the condenser, respectively.

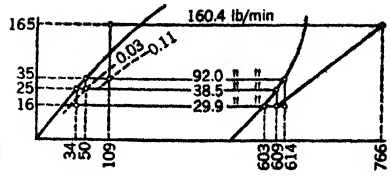
Figure 5-1 shows nine possible equipment arrangements and operating cycles which could be used in this plant. Each of the cases demonstrates one or more of the principles which have been discussed, in connection with individual equipments, in Chapter IV. The nine cases are arranged in the order of decreasing power requirements, hence of increasing operating effectiveness, but it must be remembered that this analysis takes account only of performance; therefore the most economical arrangement, in terms of total cost with allowance for the first cost of the various equipments, is not necessarily the one having the lowest power requirement. In fact a cursory examination of the tabulations will show that, whereas the first case requires more than 36 per cent more power than the sixth case, the saving in power of the ninth over the sixth is only slightly less than 4 per cent; hence a substantial outlay for equipment could be justified to change the cycle from the first to the sixth, but further improvement to the ninth arrangement would not be economical unless the first cost of required extra equipment was small.

CASE 1. The equipment arrangement shown here combines multiple expansion with a throttling system in which vapor from all three evaporators is reduced to the pressure of the lowest to permit entrance to a single compressor cylinder. The throttling arrangement, lacking additional compressors, is a practical procedure which permits maintenance of the requisite evaporator temperatures, but the use of a

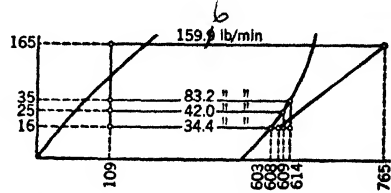
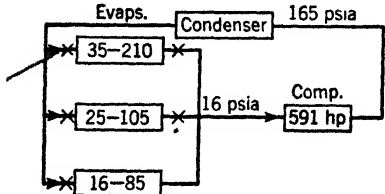
Diagrams of Equipment



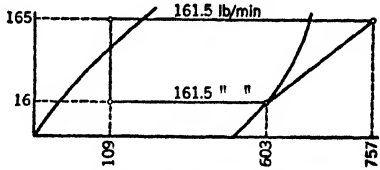
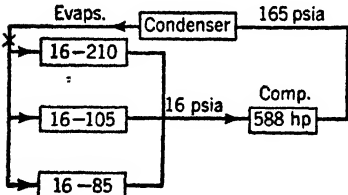
Pressure-Enthalpy Diagrams



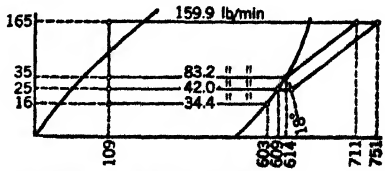
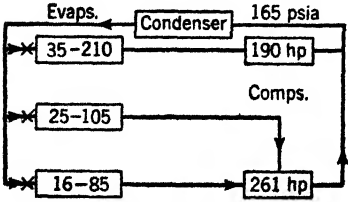
Case 1. Non-sectionalized, throttling operation with multiple expansion valves. (Shows misuse of multiple valves.)
Refrigerant circulated 160.4 lb/min
Total horsepower 595



Case 2. Non-sectionalized, throttling operation with single expansion valves.
Refrigerant circulated 159.9 lb/min
Total horsepower 591

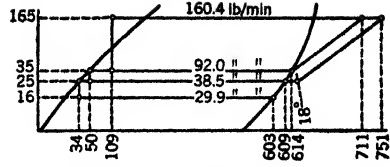
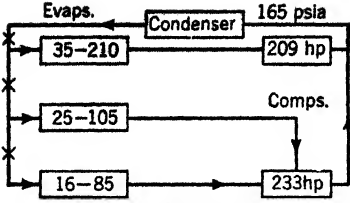


Case 3. Non-sectionalized, non-throttling operation with single expansion valves.
Refrigerant circulated 161.5 lb/min
Total horsepower 588



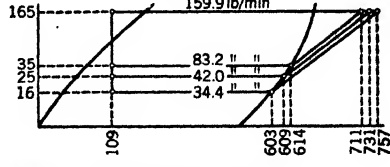
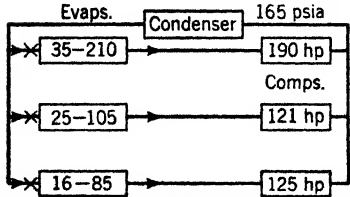
Case 4. Sectionalized, dual compression, with single expansion valves.
Refrigerant circulated 159.9 lb/min
Total horsepower 451

FIG. 5-1. Case 1 to Case 4.



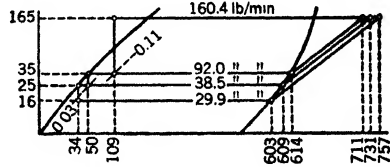
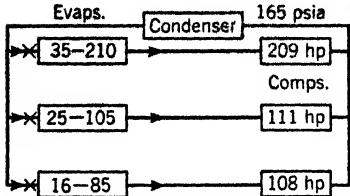
Case 5. Sectionalized, dual compression, multiple expansion valves.

Refrigerant circulated 160.4 lb/min
Total horsepower 442



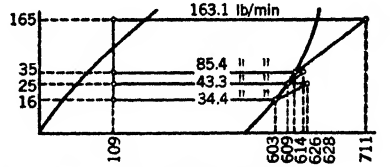
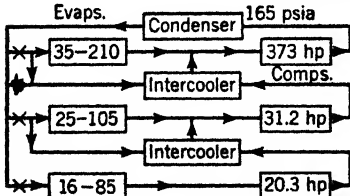
Case 6. Sectionalized, separate compression, single expansion valves.

Refrigerant circulated 159.9 lb/min
Total horsepower 436



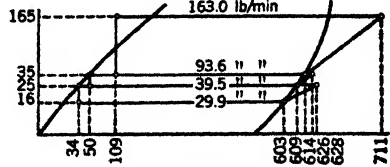
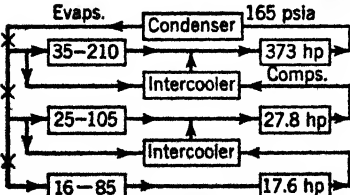
Case 7. Sectionalized, separate compression, multiple expansion valves.

Refrigerant circulated 160.4 lb/min
Total horsepower 428



Case 8. Sectionalized, compound compression, single expansion valves.

Refrigerant circulated 163.1 lb/min
Total horsepower 424.5



Case 9. Sectionalized, compound compression, multiple expansion valves.

Refrigerant circulated 163.0 lb/min
Total horsepower 418.4

FIG. 5-1. Case 5 to Case 9.

multiple expansion valve is thermodynamically undesirable and is responsible for a net loss rather than gain in cycle effectiveness. Thus vapor extracted at the 35 psia pressure does contain more energy than that extracted at 16 psia; hence it provides a greater refrigerating effect which, in itself, is desirable, but before admission to the compressor this vapor is throttled at constant enthalpy, with increasing superheat, to the lowest pressure and therefore increases the superheat of the suction mixture with consequent increase in the required work of compression. This point is one of considerable importance because not infrequently operators familiar with multiple expansion cycles, but unfamiliar with the thermodynamic reason for their use, are likely to misuse such expansions in connection with systems in which they possess only disadvantages. As a general rule it can be unequivocally stated that there is never a thermodynamic advantage in extracting vapor at a higher pressure than that of the lowest evaporator unless means are available to permit recompression of this refrigerant at a suction pressure higher than that of the low-pressure evaporator.

The calculations for analysis of Case 1 are

$$w_{16\text{psia}} = \frac{200 \times 85}{603 - 34} = 29.9 \text{ lb/min}$$

$$w_{25\text{psia}} = \frac{200 \times 105}{609 - 50} + \left(\frac{0.03}{1 - .03} \times 29.9 \right) = 38.5 \text{ lb/min}$$

$$w_{35\text{psia}} = \frac{200 \times 210}{614 - 109} + \left[\frac{0.11}{(1 - .11)} \times (29.9 + 38.5) \right] = 92.0 \text{ lb/min}$$

$$h_m = \frac{(29.9 \times 603) + (38.5 \times 609) + (92.0 \times 614)}{29.9 + 38.5 + 92.0} = 609 \text{ Btu/lb}$$

$$\text{hp} = \frac{(29.9 + 38.5 + 92.0)(766 - 609)}{42.42} = 595$$

CASE 2. This case represents the simplest equipment arrangement which can meet the requirements of this problem. Note that the only difference in arrangement between Case 1 and Case 2 is that with the former the three expansion valves are in series (giving multiple expansion) whereas for the latter the three valves are in parallel. Although the number of valves for the two cases are the same the actual valves used would have to be different since the weight rate of refrigerant flowing through the respective valves is not the same for the two cases.

Calculations are

$$w_{16} = \frac{200 \times 85}{603 - 109} = 34.4 \text{ lb/min}$$

$$w_{25} = \frac{200 \times 105}{609 - 109} = 42.0 \text{ lb/min}$$

$$w_{35} = \frac{200 \times 210}{614 - 109} = 83.2 \text{ lb/min}$$

$$h_m = \frac{(34.4 \times 603) + (42.0 \times 609) + (83.2 \times 614)}{34.4 + 42.0 + 83.2} = 608$$

$$\text{hp} = \frac{(34.4 + 42.0 + 83.2)(765 - 608)}{42.42} = 591$$

CASE 3. This case does not represent a permissible treatment of the problem as it was originally stated, but it is included here to show that, when throttling is the only means of maintaining different evaporator pressures, greater thermodynamic effectiveness is realized by allowing all evaporators to operate at the lowest pressure and thereby maintain a saturated state for vapor at compressor suction. In effect, Case 3 shows the performance of an equivalent simple saturation cycle carrying a total load equal to that indicated as occurring in the plant. Calculations are

$$w = \frac{200 \times 400}{603 - 109} = 161.5$$

$$\text{hp} = \frac{161.5 (757 - 603)}{42.42} = 588$$

CASE 4. The first approach to sectionalizing is to combine two of the evaporator loads in a dual-effect compressor and handle the third evaporator separately. Since irreversibility occurs during the high-pressure admission to the dual compressor this system is not so advantageous as complete sectionalizing would be, but its first cost may be less than that needed to permit three separate compressions. Owing to the irreversible admission the state of the mixture of gases at start of compression is one having greater superheat than did the high-pressure suction vapor prior to mixing; thus the state at start of compression cannot be determined directly from the pressure-enthalpy chart but must be fixed by calculations (refer to Section 4·8). Calculations are

w_{16} , w_{25} , w_{35} the same as in Case 2

Determine state at start of compression by means of equation 4.5 (trial-and-error solution), and from this state ($p = 25$ psia and $t = 18^\circ$ F) follow isentropic to 165 psia where read $h_d = 751$:

$$\begin{aligned} \text{hp} &= \frac{[(42.0+34.4) 751] - (42.0 \times 609) - (34.4 \times 603)}{42.42} + \frac{83.2(711 - 614)}{42.42} \\ &= 451 \end{aligned}$$

CASE 5. The use of multiple expansion in this case gives a slight saving in power over Case 4, but it is doubtful if the saving would be sufficient to justify the increased complexity of the system. Calculations are

w_{16}, w_{26}, w_{36} as in Case 1

State at start of compression as in Case 4

Then

$$\begin{aligned} \text{hp} &= \frac{[(38.5 + 29.9)751] - (38.5 \times 609) - (29.9 \times 603) + [92.0(711 - 614)]}{42.42} \\ &= 442 \end{aligned}$$

CASE 6. If the requisite compressor capacity is available the sectionalized arrangement of this case represents the simplest system which combines good operating economy with an easily controlled flexible equipment arrangement. By comparing power requirements of Case 4 and Case 6 it is evident that the irreversibility of the dual compressor is chargeable with 15 hp—approximately a 4 per cent increase over the reversible sectionalized system. This loss of energy is entirely attributable to the dual-pressure evaporator loads since the compression of vapor from the highest pressure evaporator occurs by the same process for both of these equipment arrangements. Calculations are

w_{16}, w_{26}, w_{36} as in Case 2

$$\text{hp} = \frac{[83.2(711 - 614)] + [42.0(731 - 609)] + [34.4(757 - 603)]}{42.42} = 436$$

CASE 7. The effect of multiple expansion for this system provides an economy of the same order of magnitude as was observed between the systems shown in Case 4 and Case 5. Calculations are

w_{16}, w_{26}, w_{36} as in Case 1

$$\text{hp} = \frac{[92.0(711 - 614)] + [38.5(731 - 609)] + [29.9(757 - 603)]}{42.42} = 428$$

CASE 8. Stage compression with liquid intercoolers provides a slight reduction in power over that needed for sectionalized operation,

but at the expense of greater complexity. It is particularly important to note the degree of unbalance of load that now exists among the three compressors. In the event of failure of the large (third-stage) compressor neither of the others would be able to assist effectively, whereas for the sectionalized system of Case 6 failure of any one compressor could be partially compensated for by either of the others. Calculations are

$$w_{16} = 34.4 \text{ lb/min as in Case 2}$$

$$w_{25} = \frac{(200 \times 105) + [34.4(628 - 609)]}{609 - 109} = 43.3 \text{ lb/min}$$

$$w_{35} = \frac{(200 \times 210) + [(34.4 + 43.3)(626 - 614)]}{614 - 109} = 85.4 \text{ lb/min}$$

$$\begin{aligned} \text{hp} &= \frac{[34.4(628 - 603)] + [(43.3 + 34.4)(626 - 609)]}{42.42} \\ &\quad + \frac{[43.3 + 34.4 + 85.4](711 - 614)}{42.42} \\ &= 424.5 \end{aligned}$$

CASE 9. This case represents the most complex equipment arrangement likely to find use in a real plant. As in Case 8, intercooling is shown as occurring by means of a bubble column, but in practice a water intercooler would almost always be located ahead of the liquid bubble column. Calculations are

$$w_{16} = 29.9 \text{ lb/min as in Case 1}$$

$$\begin{aligned} w_{25} &= \frac{(200 \times 105) + [29.9(628 - 609)]}{609 - 50} + \left[\frac{0.03}{(1 - 0.03)} \times 29.9 \right] \\ &= 39.5 \text{ lb/min} \end{aligned}$$

$$\begin{aligned} w_{35} &= \frac{(200 \times 210) + (29.9 + 39.5)(626 - 614)}{614 - 109} + \left[\frac{(1 - 0.11)}{0.11} \times (29.9 + 39.5) \right] \\ &= 93.6 \text{ lb/min} \end{aligned}$$

$$\begin{aligned} \text{hp} &= \frac{[29.9(628 - 603)] + [(29.9 + 39.5)(626 - 609)]}{42.42} \\ &\quad + \frac{[(29.9 + 39.5 + 93.6)(711 - 614)]}{42.42} \\ &= 418.4 \end{aligned}$$

5.2. The Booster Cycle. With the advent of quick freezing, many existing refrigeration plants were called on to provide freezing temperatures very much lower than those for which the plants had been de-

signed. In order to permit maximum utilization of existing compressors a common procedure has been to install special "booster" compressors which raise the temperature level of the refrigerant from the value corresponding to that of the quick freezer to that of the normal storage rooms. Thus if a given cold storage warehouse were originally provided with mechanical equipment capable of maintaining temperatures down to 0°F , some rooms of this warehouse might readily be adapted for -50°F service by installing a booster compressor which would raise

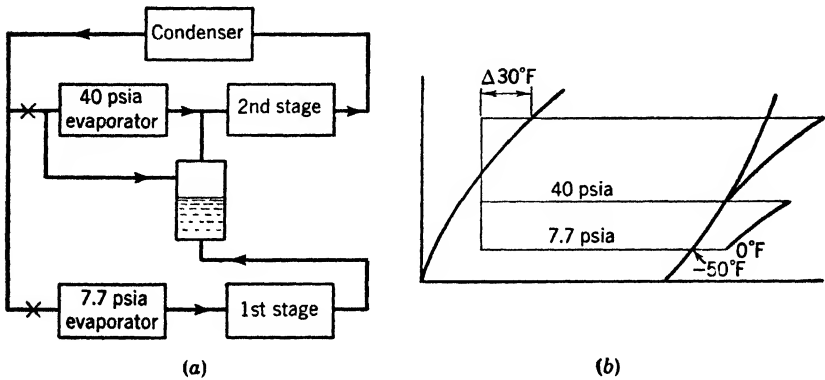


Fig. 5-2.

the entering -50°F vapor to a discharge pressure such that its saturation temperature would be 0°F . Discharge from the booster would then enter the suction line (but usually after desuperheating) of the main compressor.

Example. An ammonia refrigeration plant operates with a quick-freezing load of 50 tons on a -50°F (7.7 psia) evaporator and a 20-ton load on a 12°F (40 psia) evaporator. The cycle (see Fig. 5-2) is based on use of a booster compressor to raise the vapor from the low-temperature evaporator to the pressure existing in the high-pressure evaporator. Saturated vapor at -50°F leaves the low-pressure evaporator and undergoes a 50°F temperature rise in the piping before entering the booster. Booster compression is isentropic, and the superheated material at discharge is bubbled through a liquid desuperheater which reduces it to a saturated state at 40 psia. This vapor then mixes with the saturated vapor leaving the high-pressure evaporator, and the entire quantity of ammonia vapor then enters the main compressor where isentropic compression occurs to a discharge pressure of 180 psia. The overall efficiency of the main compressor is 85 per cent. Fluid leaves the condenser subcooled by 30°F and flows back through parallel expansion valves to the two evaporators and the desuperheating chamber. Determine (a) the horsepower requirements

of the main compressor; (b) the necessary increase in displacement (assuming 100 per cent volumetric efficiency) if the vapor from the booster were not desuperheated prior to mixing with the vapor from the high-pressure evaporator; (c) the power requirement of the main compressor if booster discharge were not desuperheated.

Solution. The enthalpies of the refrigerant at various states of the cycle are determinable from the pressure-enthalpy diagram for ammonia as

- (1) Liquid from condenser at 180 psia with 30° F subcooling, $h_1 = 109$.
- (2) Saturated vapor from low-pressure evaporator at 7.7 psia, $h_2 = 593$.
- (3) Superheated vapor at booster suction at 7.7 psia and 50° F superheat, $h_3 = 620$.
- (4) Discharge from booster after isentropic compression, $h_4 = 727$.
- (5) Saturated 40 psia vapor from desuperheater and from high-pressure evaporator, $h_5 = 616$.
- (6) Discharge from main compressor at 180 psia after isentropic compression, $h_6 = 710$.

(a) The useful refrigerating effect in the 7.7 psia evaporator is $593 - 109 = 484$ Btu/lb; therefore the weight of refrigerant passing through this evaporator must be $(50 \times 200)/484 = 20.7$ lb/min. The required amount of desuperheating in the bubble-type desuperheater is then $20.7(727 - 616) = 2300$ Btu/min, and this is obtained from a refrigerant having a refrigerating effect of $616 - 109 = 507$ Btu/min. Therefore requisite rate of flow of refrigerant through the expansion valve to the desuperheater is $2300/507 = 4.54$ lb/min. The refrigerating effect in the 40 psia evaporator is likewise 507 Btu/lb. Therefore the required flow rate through this evaporator is $(20 \times 200)/507 = 7.91$ lb/min. The total rate of refrigerant flow through the main compressor is then $20.7 + 4.54 + 7.91 = 33.15$ lb/min. The power at the main compressor is $33.15(710 - 616) = 3100$ Btu/min or $(3100 \times 60)/(0.85)(2544) = 86.0$ hp.

(b) The specific volume of 40 psia saturated vapor is 7 cu ft/lb. Therefore the displacement for initial operating conditions is $33.15 \times 7 = 231$ cfm. If vapor from the booster were not desuperheated its specific volume would be 10.5 cu ft/lb, and the total volume of the mixed vapor entering the main compressor would therefore be $20.7 \times 10.5 + 7.91 \times 7 = 218 + 55 = 273$ cfm. Thus the increase in displacement required if the vapor were not desuperheated would be $(273 - 231)/231 = 18$ per cent.

(c) If superheated discharge from the booster were mixed with the saturated vapor from the 40 psia evaporator the enthalpy of the resultant mixture would be $[(20.7 \times 727) + (7.91 \times 616)]/(20.7 + 7.91) = 19,900/28.6 = 700$ Btu/lb. Then, if isentropic compression occurs from a state of 40 psia and 700 Btu/lb to a discharge pressure of 180 psia, the enthalpy at discharge will be 824 Btu/lb and the required power will therefore be $28.6(824 - 700)60/(0.85)(2544) = 99$ hp. The increase in power due to elimination of the desuperheating is therefore $(99 - 86)/86 = 15$ per cent. Thus it is evident that the work of compression increases significantly when the suction vapor is superheated, and the use of refrigerant liquid desuperheaters is economically desirable.

5.3. Deviations of the Actual Cycle from the Ideal.* The fundamental assumptions underlying the ideal cycle are that no pressure change occurs other than in the compressor or the expansion valve and that no gain or loss of heat takes place except during passage of the refrigerant through the condenser and evaporator. A further assumption is that the compression process is isentropic, hence occurs without energy transfer to or from the working fluid other than that introduced into the refrigerant as the heat equivalent of the shaft work. In any actual cycle there are, of course, friction losses in the piping and in the individual pieces of equipment which are responsible for a progressive drop in pressure of the refrigerant from the point of discharge from the compressor through the system to the suction entrance of the compressor. Furthermore the connecting piping inevitably passes through spaces which are at greater or lesser temperatures and which are, therefore, responsible for heat transfer to or from the refrigerant.

The actual compression process is likely to be markedly different from the ideal. During constant pressure admission, the low-temperature suction gases are first subject to wire drawing as they pass through the suction valves of the compressor and then undergo an increase in internal energy as heat flows from the higher temperature cylinder walls to the cold gases. During this period, the internal energy, enthalpy, and entropy of the superheated refrigerant vapor are increasing. During the first part of the actual process of compression, the gases are at a temperature less than that of the cylinder walls; hence, energy continues to flow as heat from walls to refrigerant.

As the compression process continues, the gas temperature increases (owing to the transformation of shaft work to internal energy) until eventually the gas within the cylinder is at the same temperature as the cylinder wall. At this particular instant the rate of energy transfer from wall to gas becomes zero, and from here on the flow of heat is in the opposite direction. As the pressure increases, the temperature difference between gas and wall likewise increases until, at the time the discharge valves open, the rate of energy transfer from gas to wall is a maximum. Throughout the subsequent period of constant pressure discharge, the gas continues to lose heat to the relatively cold cylinder walls; at the same time, wire drawing through the discharge valves serves to reduce the pressure at constant enthalpy so that the actual refrigerant at a point in the line outside the compressor has both a lower pressure and a lower enthalpy than it had within the cylinder at the time the discharge valves opened.

* This section is taken from the article, "Analyzing Refrigeration Cycles," Hutchinson, *Heating, Piping Air Conditioning*, pp. 79-81, May, 1947.

If a compressor cylinder were perfectly insulated there would be no net gain or loss of energy to the surroundings; in practice, however, the cylinder is not merely imperfectly insulated but is in many cases provided with some special means (as a water jacket) for assuring a net transfer of energy as heat from the cylinder wall to the surround. In such cases, and irrespective of the energy added to the refrigerant as a result of shaft work, the heat received by the gas from the cylinder during admission and during the early part of the compression process must inevitably be substantially less than that delivered by the gas to the cylinder during the latter part of compression and during discharge.

At first glance it would appear that this net heat loss would effectively prevent application to the actual compressor of the principle of reversibility, yet it can be shown readily that only when there is a net loss can the unit be expected to perform with reasonable fidelity to the reversible pattern. Paradoxical though this may seem, it occurs because, in a perfectly insulated compressor cylinder, there would necessarily be an increase in entropy of the refrigerant due to the fact that the entropy gain associated with each Btu transferred from wall to gas would be greater than the corresponding entropy loss experienced when a Btu is transferred back (at higher temperature) from the gas to the cylinder wall. Thus, the effect of water-jacketing is to tend to restore the entropy of the refrigerant at discharge from the compressor to the value which it would have had if the compression process (including admission and discharge) had been isentropic.

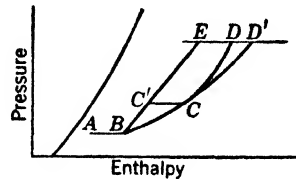


FIG. 5.3.

Figure 5.3 illustrates the energy transfers which occur during a compression process in which there is no net energy transfer from the compressor. During the constant pressure admission the enthalpy increases from h_A to h_B . Subsequently, during the first part of the compression process, energy is received at a constantly decreasing rate until, at point C , the gas is at the same temperature as the cylinder walls and has received an enthalpy increase during the first part of compression amounting to $h_C - h_B$ of which $h_C - h_{C'}$ is a gain in excess of that which would have occurred if the process had followed the isentropic line BC' .

During the latter part of the process, the compression would follow CD' if it were isentropic, but owing to the continuous and increasing heat loss to the cylinder wall the actual enthalpy gain of the gas is reduced from $h_{D'} - h_C$, the isentropic value, to $h_D - h_C$. The added energy gain during the early part of compression, $h_C - h_{C'}$, may be exactly equal to the reduction in gain during the latter part, $h_{D'} - h_D$.

but even so the entropy at point *D* will exceed that at point *B* because the isentropic *CD'* is of flatter slope than the isentropic *BC'*. If water-jacketing were provided, sufficient heat could be removed to reduce the enthalpy of the discharged gas from h_D to h_E , thereby restoring it to the state point which would have existed at discharge if the compression had been reversible and no energy transfer had occurred between the gas and the cylinder wall.

Note, however, that even though the state point has been restored to the isentropic discharge value it does not follow that the work of com-

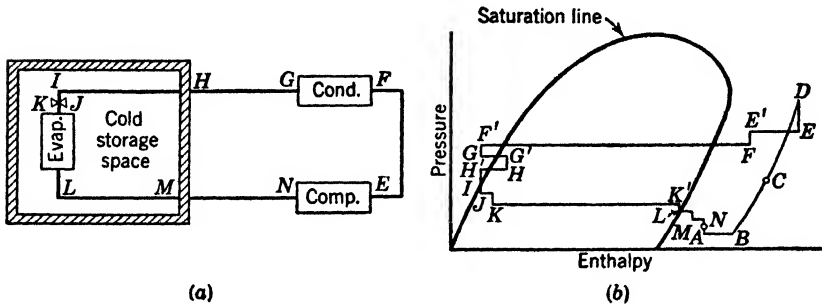


FIG. 5.4.

pression has been reduced to its isentropic magnitude; by the general energy equation

$$(h_E - h_B) + Q_{\text{jacket}} = Wk_{\text{compression}}$$

where Q_{jacket} is the heat loss to the jacket expressed in Btu per pound per minute of refrigerant passing through the compressor, and $Wk_{\text{compression}}$ is the energy input to the compressor (neglecting mechanical losses) expressed in Btu per pound per minute of refrigerant; $(Wk_{\text{compression}} \div 42.42)w$ (where w is the refrigerant flow rate in pounds per minute) is the horsepower required for the system.

Figure 5.4a shows an equipment arrangement corresponding to an actual system while Fig. 5.4b shows the p - h representation of the cycle. Starting with the refrigerant at state *A* just after passage through the suction valves of the compressor, the state changes during admission, compression, and discharge are shown by the path *ABCD*, all as described in detail in the preceding section. From state *D* the refrigerant passes through the discharge valves according to the throttling process *DE*, and its state in the line just outside the compressor is then *E*. From here the gas passes through the line to the condenser and probably undergoes some desuperheating drop due to friction; for purposes of p - h representation these two losses, of heat and of pressure, can be con-

sidered separately, thus permitting representation of the actual complex process as consisting first of the desuperheating constant pressure process EE' followed by the equivalent throttling process $E'F$. Regardless of which of the two equivalent processes is considered as occurring first, the end state of the refrigerant—at entrance to the condenser—must, of course, be the same: F .

During passage through the condenser, the refrigerant continues to lose heat until it reaches a saturated vapor state, then undergoes condensation, and finally may be somewhat subcooled; these three heat dissipation steps can be grouped and represented by the single equivalent constant pressure process FF' . The pressure loss which occurs during passage through the condenser is usually too small to be of practical significance, but, if desired, it can be shown as an equivalent constant enthalpy process $F'G$. Between condenser discharge and entrance to the cold storage room the subcooled liquid may pass through spaces at higher temperature and from which heat will be received. If this is the case the line process will be shown as an equivalent heat gain GG' followed by equivalent line pressure drop $G'H$. Note that reevaporization of the refrigerant, with consequent loss of refrigerating effect and reduction in the coefficient of performance, may then occur.

On entrance to the cold storage room the refrigerant passes through the length of pipe HI which is located ahead of the expansion valve. Since the room is at lower temperature than the refrigerant there will be some cooling effect HH' and the usual line pressure loss $H'I$. Note that, even if the enthalpy loss from H to H' were exactly equal to the enthalpy gain from G to G' , there would be no gain of efficiency since such cooling as occurs along the line HI is at the expense of increased evaporator load. However, if this energy quantity is added to evaporator load in figuring the capacity of the system, it should be subtracted when figuring condenser capacity since the added load does not appear at the condenser. In effect, this energy quantity serves only to increase the required size of the evaporator.

From I to J the fluid passes through the expansion valve. From J to K it picks up heat from the cold storage space and loses pressure as a result of line friction. The evaporator process is shown by the steps KK' and $K'L$, the refrigerant being assumed to be a saturated vapor at discharge from the evaporator. In the piping between the evaporator and the wall of the cold storage room additional useful heat LM is picked up, while between the cold storage room and the compressor such heat gain MN as does occur represents an added compressor and condenser load for which there is no compensating increase in refrigerating effect. For this system, as shown, the actual energy gain

of the refrigerant as it passes through the cold storage space is $h_M - h_I$, but the effective refrigerating effect is $h_M - h_H$.

From state N , in the pipe just ahead of the suction valves of the compressor, the refrigerant goes through the valves with a constant enthalpy wire-drawing loss as shown by NA ; from this state point the cycle is then repeated.

PROBLEMS

1. For the same evaporator and condenser temperatures that were used in Section 5·1, analyze the eight cases assuming use of dichlorodifluoromethane as the refrigerant.

2. Repeat Problem 1 with various other refrigerants and compare the results.

3. An ammonia system operates on the simple saturation cycle between 25 psia and 165 psia. If multiple expansion were used with two expansion valves determine the optimum intermediate pressure at which vapor would be returned to the separate flash-vapor compressor.

4. For the conditions of Problem 3 use two, then three, then four vapor extractions from multiple expansion valves and plot the total power requirement of the various cycles against the number of expansion valves. By extrapolation estimate the power requirement that would correspond to operation with an infinite number of expansion valves. If an infinite number of expansion valves were used, what would be the flow rate of refrigerant through the condenser for 1 lb/min through the evaporator?

5. Refer to a textbook on steam power plants and compare the analysis for the regenerative feed-water heating cycle with infinite feed heaters to the analysis for infinite multiple expansion valves.

6. Investigate analytically the possibility of obtaining a compressor with clearance such that it could handle the weight rates of suction vapor required in the example shown in Case 4.

7. Re-do the example of Section 5·1, but assume that the 400-ton load is evenly distributed over the three evaporators. (The results from this problem permit a more effective comparison of the eight cases.)

8. For the example of Section 5·1 develop an additional case to show power requirements if all three loads were to be carried on one triple-effect compressor.

9. A booster compressor takes saturated ammonia vapor from an evaporator operating at -35°F and discharges it, after isentropic compression, into the suction of the main compressor. The booster load is 15 tons and the main load carried at an evaporator temperature of 6°F , is 30 tons. (a) If vapor leaving the main evaporator is saturated, determine the degrees of superheat possessed by the mixture of vapors at entrance to the main compressor and determine the total horsepower requirements for both compressors. (Take the condenser pressure as 165 psia with saturated liquid at discharge; assume isentropic compression in the main compressor.) (b) Determine the effect on total power of using liquid ammonia to cool the booster discharge down to a saturated vapor state.

10. For fixed compressor speed and clearance determine the increase in capacity of the main compressor in Problem 9 that would result from use of liquid subcooling of the booster discharge.

11. An ammonia system operates between evaporator and condenser pressures of 25 psia and 165 psia. (a) Assuming initial operation on the simple saturation cycle

calculate the increase in power, expressed as per cent, that would occur if a throttling loss of 5 psia should develop in the admission valves of the compressor. (b) Calculate the per cent of power increase over simple saturation if a 5-psia wire-drawing loss were to occur in the discharge valves (assuming no suction valve loss). (c) Calculate the per cent of power increase for a 5-psia loss in both suction and discharge valves.

12. An ammonia compressor receives saturated vapor at -50°F and delivers to a condenser that operates at $+50^{\circ}\text{F}$. The compressor cylinder is so effectively insulated that there is no appreciable transfer of heat from cylinder to room air. When the temperature of the vapor in the cylinder has been raised to 0°F (during the compression process) the entropy of the vapor is 1.47. Assume that the first part of the compression process (with vapor temperature rise from -50°F to 0°F) and the second part of the compression process (with vapor temperature rise from 0°F to the discharge temperature) are both representable as straight lines on a temperature-entropy diagram. (a) Calculate the rate of heat flow from cylinder wall to vapor during the first part of the compression process. (b) Calculate the entropy of the vapor at discharge; note that the rate of heat flow from vapor to cylinder wall, during the second part of the compression process, must be equal to the rate established in (a). (c) Explain the resultant entropy increase during this externally adiabatic compression process.

13. An ammonia compressor receives saturated vapor at -50°F and delivers to a condenser that operates at $+50^{\circ}\text{F}$. The compression process is isentropic, but entering vapor undergoes an entropy increase of 0.2 during the constant pressure admission. (a) For an externally adiabatic system calculate the required rate of heat flow from vapor to cylinder walls during discharge and determine the entropy decrease of the vapor during the constant pressure discharge process. (b) Compare the entropies at discharge for the cases of Problem 12 and Problem 13 and determine, for the same entropy increase, whether heat gain during admission is more undesirable than heat gain during the early part of compression. (c) Explain, thermodynamically, the result of (b).

14. For the conditions of Problem 13 determine the required rate of heat removal during the compression process (by cylinder cooling) necessary to restore the entropy of the vapor at discharge to its value at admission (calculations to be based on the assumption that rates of heat flow during admission and discharge remain the same as they were in Problem 13).

15. Explain the possibility of entropy *increase* of vapor during passage through a compressor which is water cooled.

CHAPTER VI

SPECIAL SYSTEMS; THE HEAT PUMP

6.1. Production of Solid Carbon Dioxide. One of the common methods of producing dry ice is to extract carbon dioxide vapor from the products of combustion of a furnace. This method is thermodynamically interesting because it represents the unusual condition of operating a furnace-boiler system in which the steam is produced as a by-product and primary interest centers on the exhaust gases. Figure 6.1 shows a schematic arrangement of equipments for a dry ice plant of this type. In essence the entire process is divided into three sections: (1) extraction of the carbon dioxide from the exhaust gases; (2) compression and liquefaction of the carbon dioxide; (3) expansion and solidification of the material with formation of dry ice. The first operation is carried out in steps involving the cooling of all exhaust gases and removal of solid impurities by passage through some type of water-cooled packed tower followed by the separation of the carbon dioxide from the other materials (largely nitrogen) that are originally present. The cool, clean gases from the scrubbing tower pass upward through a packed column through which a cool, absorbent fluid rains down; since selective absorption of the carbon dioxide occurs, by the time the gases reach the top of the tower more than 95 per cent of the carbon dioxide has been extracted. The absorbent fluid with its load of carbon dioxide then flows to a generator where heat is applied and the gas driven off; after cooling and removal of any carry-over absorbent the carbon dioxide is ready for compression.

The compression process usually occurs in three stages with water intercooling. In the first stage the pressure rise is from atmospheric to somewhere around 100 psia, whereas in the second stage the pressure is raised to approximately 300 psia. The third stage raises the pressure to 1200 psia. After being condensed the liquid carbon dioxide is expanded, usually in two stages, and finally discharged at atmospheric pressure in the form of a low-density "snow"; a hydraulic press receives the snow and forms it into high-density blocks of a size suitable for commercial purposes.

Example. Figure 6.2 shows a pressure-enthalpy diagram of a typical cycle for the production of carbon dioxide. If the plant output is 1000 lb of solid

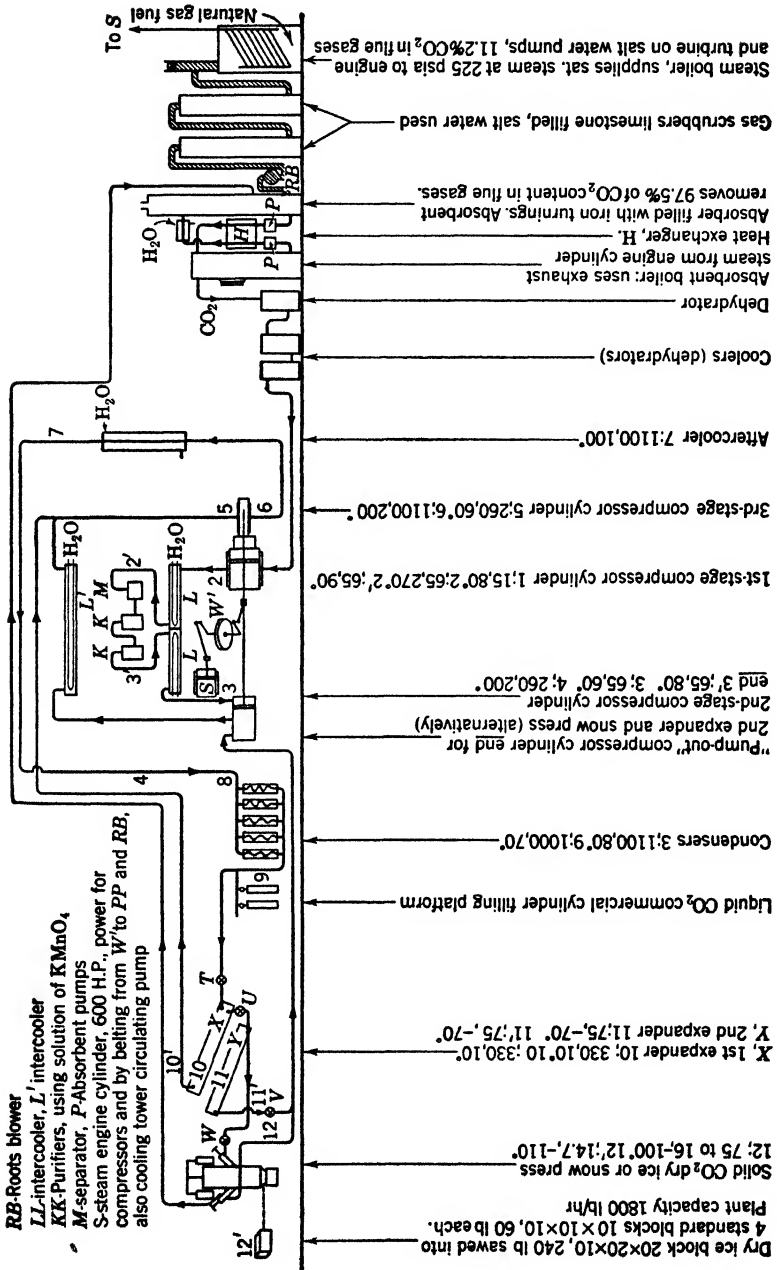


Fig. 6-1.

CO₂ per hour calculate the refrigerant flow rates at all parts of the cycle and determine the horsepower requirements for compression.

Solution. The quality of the snow-vapor mixture at the final state *e* is X_e ; hence the weight of carbon dioxide present at state *d* must be $1000/(1 - X_e) = w_1$ pounds per hour. The quality at point *c* is X_c ; the weight rate of refrigerant expanded from state *b* must be $1000/[(1 - X_e)(1 - X_c)] = w_2$ pounds per hour. The quality at state *a* is X_a ; therefore the refrigerant flow rate through the third stage of compression and through the condenser is $1000/[(1 - X_e)(1 - X_c)(1 - X_a)] = w_3$ pounds per hour. The second compression stage therefore handles w_2 pounds per hour, and the first stage handles w_1 pounds per hour of which $w_1 X_c$ pounds is vapor returned from the expander at a temperature of -110° F, whereas the remaining 1000

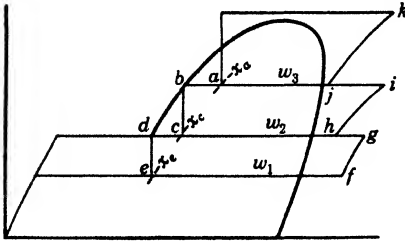


FIG. 6.2.

lb is make-up required to replace the carbon dioxide leaving the system as dry ice. The total work of compression is then

$$hp = \frac{[w_1(h_g - h_f)] + [w_2(h_i - h_h)] + [w_3(h_k - h_j)]}{42.42}$$

6.2. Dual Refrigerants. A two-stage cycle has been proposed (see Fig. 6.3) which uses carbon dioxide for the low-temperature and ammonia for the high-temperature stage, the outstanding feature being the condensation of the carbon dioxide by means of a condenser using liquid ammonia to absorb the heat of liquefaction. The advantages of the combination cycle are the nominal carbonic and ammonia pressures and the moderate-sized carbon dioxide suction piping. The ammonia evaporating coil has to remove heat equal to the useful refrigeration performed by the carbon dioxide at the low temperature plus the heat equivalent of the work of compression in the carbonic compressor. Calculations are subject to some error because of the inadequate knowledge of physical properties of saturated and particularly of superheated carbon dioxide. The triple point at -69.88° F limits the use of carbon dioxide at low temperatures, whereas ammonia has a triple point at -107.8° F.

Example. Assume that useful refrigeration is to be performed at -60° F and that condensing water available permits a liquefaction temperature of 85° F. Carbon dioxide liquefaction temperatures of 5° , 15° , and 25° F will be taken, with evaporation temperatures of the ammonia of -2° , 8° , and 18° F, respectively, thus permitting a temperature difference of 7° in the carbonic condenser. The coefficient of performance is desired in each of the dual refriger-

TABLE 6·1
DUAL REFRIGERANTS

	Temperature of Evaporation, -80° F					Temperature of Liquefaction in the Condenser, +85° F	
	Temperature saturation corresponding to discharge pressure from low-pressure cylinder, °F	Weight in pounds of CO ₂ per ton per minute	Compressor work per pound of CO ₂ in Btu	Heat absorbed by condenser per ton per minute in Btu	Temperature evaporation of NH ₃ in CO ₂ condenser, in °F	Weight NH ₃ per ton of refrigeration at -60° F per minute	Work of NH ₃ compressor per ton of refrigeration in Btu
Dual refrigerants	5	1.735	23.1	240.1	-2.0	0.5073	55.8
	15	1.818	27.9	250.7	8	0.5261	49.2
	25	1.900	32.1	261.0	18	0.5444	42.4
Stage compression (carbon dioxide)	10	3.556	38.3
Stage compression (ammonia)	10	0.4390	88.3

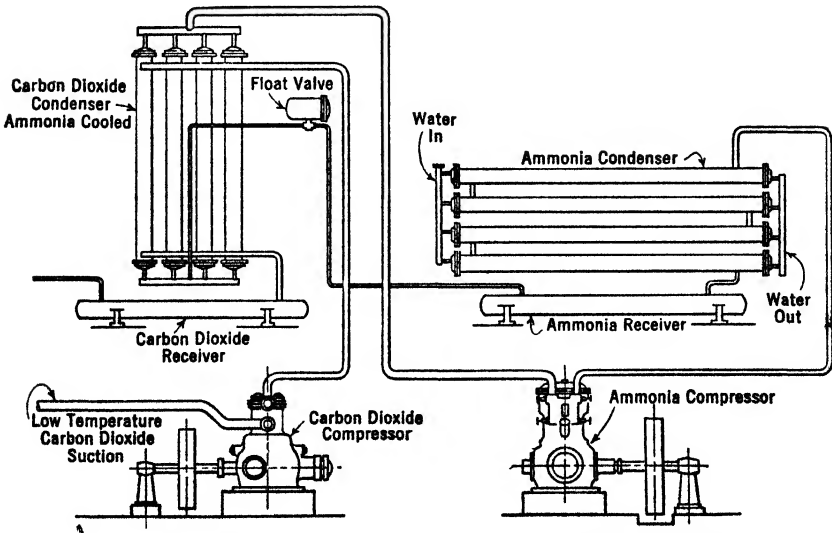


FIG. 6·3. Method of operation when a dual refrigerant is used.

ant cycles as well as the coefficient of performance for stage carbon dioxide and for stage ammonia compression using an intermediate pressure corresponding to 10° F. In the latter stage-compression cycles, use will be made of the accumulator trap and of the intermediate water cooling of the compressed vapor to 85° F. The work of compression for both carbon dioxide and ammonia is to be calculated as the difference in the enthalpies at the beginning and the end of compression, assumed to be isentropic, the values being taken from Tables 8·6, 8·7, and 8·9.

The results of the calculations are shown in Table 6·1. There is, of course, an inherent increase of work of compression in the dual refrigerant cycle because

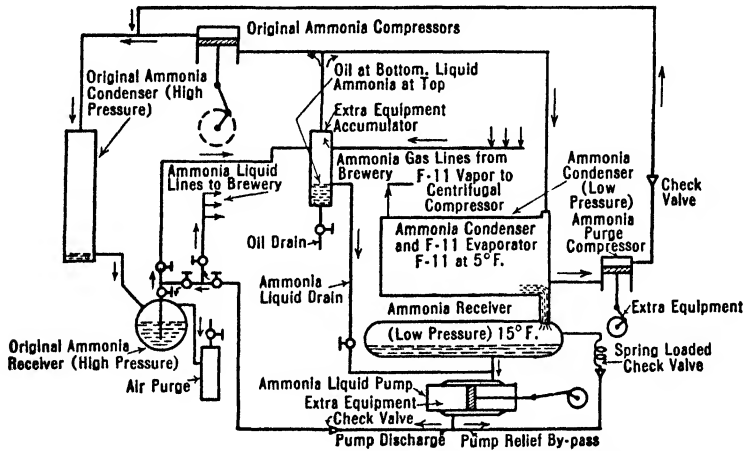


FIG. 6·4. Dual refrigerant cycle using ammonia and Freon-11.

of the necessity of evaporating the ammonia below the temperature of liquefaction of the carbon dioxide and because of the greater refrigeration load in heat units handled by the ammonia than was absorbed at -60° F by the carbon dioxide. Moreover, the calculations indicate by the several values of the coefficient of performance that better results are obtained as the temperature of evaporation of the ammonia is decreased or as the ammonia takes more and more of the temperature range. This fact is further emphasized when it is seen that the maximum coefficient of performance is obtained when ammonia takes the entire temperature range, and likewise the minimum coefficient results with the use of stage carbon dioxide compression. It would seem, then, that, whatever advantages are to be found in the dual refrigerant cycle, they do not include a reduction of the work of compression.

An interesting example of the use of the dual refrigerant cycle is given in Fig. 6·4, where an old ammonia plant was increased in capacity with the addition of a centrifugal compressor using Freon-11. The installation is unique inasmuch as the Freon-11 condensed the ammonia at practically the evaporation pressure of the ammonia.

6.3. The Heat Pump. The use of refrigerating machines as sources of heat for space heating systems has received substantial impetus during recent years owing in part to the narrowing differential between the cost of fuel and the cost of electricity and in part to the increasing number of installations of summer cooling systems. Wherever summer air conditioning is provided there is necessarily some means of artificial cooling, and wherever such cooling is provided by mechanical refrigeration the opportunity exists for utilizing the same equipment to provide some or all of the energy needed for winter heating requirements. In some localities, notably parts of the Pacific Northwest, electrical rates are sufficiently favorable to permit economical use of reverse-cycle heating systems even in structures so located that the summer air conditioning load is negligible. For the most part, however, direct fuel-burning methods will be the more economical in any installation where summer conditioning is not required.

6.4. Space Heating. The concept of space heating by means of the reversed thermodynamic cycle is by no means new; a simple air system of this type was proposed by Lord Kelvin in a paper prepared for the Glasgow Philosophical Society in 1852. Modern heat pumps, however, use a liquefiable vapor rather than air as a working substance for the same reasons which dictate such a selection for refrigerating purposes. Compared with the standard refrigeration cycle the heat pump offers a substantially higher coefficient of performance, over the same temperature range, because the shaft work supplied to the compressor is realized, unit for unit, as part of the resultant heating effect. In terms of the Carnot efficiency equation the added advantage of the heat pump is indicated by the fact that the absolute value of the condenser temperature, rather than the evaporator temperature, appears in the numerator:

$$\text{cop} = \frac{T_c}{T_c - T_e} \quad (6.1)$$

The Carnot efficiency is, of course, substantially higher than the actual coefficient of performance to be attained in a real system operating between the same temperature limits; the difference is due in part to losses in the real system and in part to the fact that the actual refrigeration cycle replaces the expansion engine of the Carnot with an irreversible process in the expansion valve.

With respect to practical operation of the system, units for heat pump service are likely to operate at higher condenser pressures than do most refrigerating machines since the required temperature of discharge to the heating system will normally be many degrees higher than

that of available cooling water; for this reason some types of compressors will be unsatisfactory for use on winter-cycle heat pump service. Another special characteristic of many heat pumps is the use of a secondary working substance (usually air) to carry heat to or from the occupied space. With indirect systems the pump must raise or lower the temperature of the secondary fluid; hence it can neither receive nor dissipate heat at constant temperature.

In such circumstances it obviously becomes necessary to operate the evaporator at a temperature so low, or the condenser at a temperature

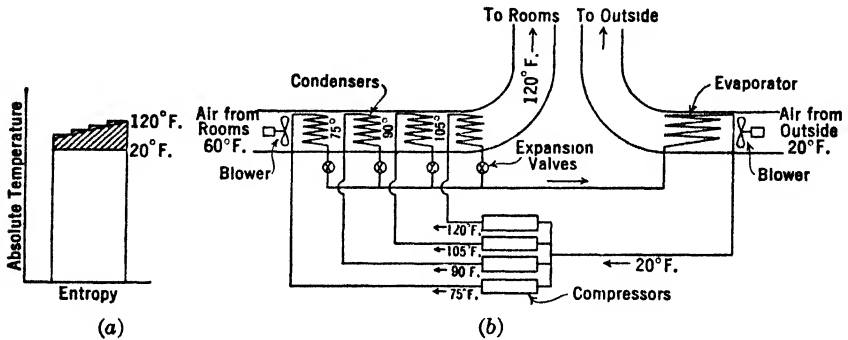


FIG. 6.5. The heating cycle (a) as shown on the Ts coordinates. (b) Diagrammatic sketch using four condensers.

so high, that a large part of the load is carried with an uneconomically low coefficient of performance. Partial alleviation of this condition can be obtained in large installations by using staged evaporation in summer and staged condensation in winter. Thus if in a given installation the secondary heating air were to be raised in temperature from the recirculating value of 70° F to 120° F the heating process could effectively be performed in two steps, the first condenser serving to raise the air temperature through the 70° F to 95° F range and the second condenser operating at a pressure sufficiently high to permit heating of the air to the room supply temperature of 120° F.

Such a heating process in steps has been used in one American installation, employing four enclosed, single-acting compressors with separate air-cooled condensers, designed for a temperature range from 60° to 135° F, to heat a building of 55,200 cu ft capacity. The air changes were designed for 6 volumes per hour of which 1500 cfm were fresh air.* The heating cycle for this installation is shown in Figs. 6.5a and 6.5b; the heating and cooling arrangements are shown in Fig. 6.6.

* Sporn and McLenegan, "An All Electric Heating, Cooling and Air Conditioning System," *Heating, Piping Air Conditioning, Journal Section*, August, 1935.

With outside temperatures of 0° to 55° F the coefficient of performance on test averaged 3.9 without allowance for the work of the water pump in the deep well, and about 3.5 with this allowance.

In the ideal case of staged operation, compression would occur in an infinite number of stages with resultant continuous heating of the air

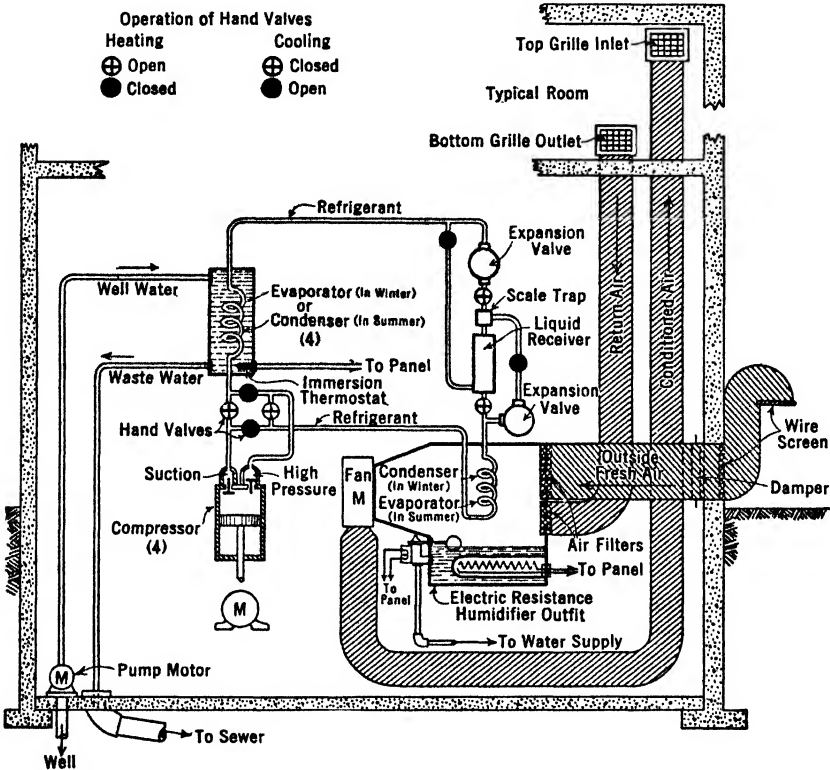


FIG. 6-6. Diagrammatic arrangement for the heating-cooling cycle.

from low to high temperature and with the refrigerant always at a temperature only slightly greater than that of the air itself. In practice the high additional first cost of providing either separate cylinders or separate cylinder ends (in the case of double-acting machines) for each stage makes the use of more than two or three compression stages prohibitively expensive.

Just as with the standard refrigerating cycle, the coefficient of performance of a heat pump increases with decreasing temperature range; hence with systems utilizing a secondary transfer fluid the design objective is to use a large quantity of fluid at a low temperature dif-

ferential rather than a small quantity at large differential. However, this objective must be pursued with caution as injudicious use of too large volumes of heating or cooling air may increase the first cost of distributing the air by more than can be economically justified from the reduction in the operating cost of the pump itself.

A further difficulty arises in that air velocities at grilles and outlets of comfort air conditioning systems must not be allowed to exceed prescribed maximums, since otherwise drafts may be experienced and the noise level in the occupied space may exceed acceptable values. As control of noise is, in any event, a problem of more importance with heat pumps than with most other types of heating systems, it is evident that special attention must be given, in such installations, to the provision of quiet, draft-free distribution. In many of the earlier indirect installations of heat pumps, dissatisfaction with the system could be traced to distribution rather than to operation of the pump itself; obviously, effective design requires careful integration of all such elements of the installation. Where such care is provided, there is no reason why an indirect heat pump cannot equal in adequacy of air distribution any other type of heating system.

A type of indirect system peculiarly well suited for use with the heat pump is the embedded-coil type of radiant heating and cooling. The most significant characteristic of radiant systems is that, through use of large energy-transferring panels, they permit operation with unusually low heated-water temperature in winter and correspondingly high cooled-water temperature in summer. Thus for heat pump service a radiant system would permit higher summer evaporator pressure and lower winter condenser pressure. Probably the first combined heat pump-panel heating system was installed in a residence in Boise, Idaho, in 1938; the distribution system for this particular installation consisted of 6000 lineal feet of $\frac{3}{16}$ -in. copper tube embedded in the exterior walls and part of the ceiling and serving as direct expansion surface equivalent in purpose to a conventional condenser. The capacity of this system is approximately 54,000 Btu/hr, and the realized coefficient of performance has been given as 3.33.

Although an electric motor is the most commonly used drive for a heat pump compressor, other units, like internal combustion engines, can be advantageously used and relatively high coefficients of performance attained through conservation of the heat losses from the engine. Thus, contrary to usual practice in other types of service, not only is it unnecessary to purchase cooling fluid to dissipate waste heat from the driving engine, but also this heat energy can be collected and utilized as effective output of the heat pump. The most serious practical dis-

advantage standing in the way of more widespread utilization of engine-driven pumps is the problem of noise; even with electrically driven compressors the noise problem in residential applications is often troublesome, and with engine drive it becomes even more serious.

In temperate climates or during seasons of the year when the load is moderate and changeable, effective controls are needed for heat pump systems to permit rapid change-over from the heating cycle to the cooling cycle and to prevent overshooting the load with resultant overheating or overcooling. This problem is basically not different from that associated with any other type of year-around air conditioning, but since the usual heat pump is entirely automatic the adequacy or shortcoming of the controls will usually be more readily discernible than with other types of systems. Several methods are in use for varying the output of the pump to meet changing load conditions, but as these do not differ technically from the methods used with usual types of refrigeration systems they will not be discussed in more detail.

Avoidance of overshooting rests more with the system for distributing the energy than with the energy source. Thus any heating system having a large thermal capacity will necessarily be sluggish in its response to load changes and may preclude the possibility of maintaining comfort conditions irrespective of outside temperature changes. Furthermore a thermally heavy system may adversely affect the operating costs since in localities where heating may be desirable in the morning hours (during certain months), whereas cooling is dictated during afternoon hours, a sluggish system may require continuous operation of the heat pump to produce conditions no more comfortable than those which would have existed if nature's own diurnal variations had been left to balance the structure's morning and afternoon lag effects. Thus a building might require 500,000 Btu of morning heating and an equal quantity of afternoon cooling; if neither were provided the two loads would normally cancel each other, whereas with sluggish heat pump operation it might be necessary to pump out of the house, during the afternoon, the energy that was pumped in during the morning.

Based on a survey of 24 installations of heat pumps in commercial and residential structures, Penrod† found an average seasonal coefficient of 3.51. This value cannot be given too much significance, however, because the installations considered were in climates varying from that of Scotland, to Idaho, Pennsylvania, California, and Indiana; furthermore the size of the units ranged from 1 ton to 75 tons, and as compressor efficiencies are decidedly influenced by size it follows that no single

† Penrod, "Development of the Heat Pump," *Univ. Kentucky Eng. Exp. Sta. Bull.*, Vol. L, No. 4, June, 1947.

averaged coefficient of performance can be of direct significance. Possibly of greater interest than the average value is the fact that the range of cop for these 24 installations was from a low of 1.5 for a 15-ton commercial unit installed in Southern California and using 60° F outside air as a design heat source to a maximum of 7.35 for a 10-ton Southern California installation using 63° F outside air as a heat source. The wide variation between two such installations of comparable size and in the same locality is difficult to explain.

In the same report Penrod shows a load factor (defined as the ratio of average yearly rate to maximum rate) of 25.9 per cent for four units in Southern California which use air as both a heat sink in summer and source in winter.

Although comparative performance of heat pump and other types of systems is always of interest it is not necessarily a controlling factor in deciding which type of system is to be used. Intangible advantages of the heat pump as represented by absence of smokestack, greater cleanliness, ability to be installed in any available location (rather than requiring installation adjacent to a convenient source of fuel) are all factors worthy of consideration and are frequently the factors which lead to selection of such an installation.

6.5. Industrial Applications. Aside from its application for space heating purposes the reverse cycle is also of great potential importance in industrial applications involving the heating of a fluid or the thermal "pumping" of a vapor from a low to a higher pressure. Many industrial applications have been made in Europe—notably those of the Brown-Boveri Company—and heat pumps are now in such diverse services as those of water heating for district heating to thermo-compression of evaporated vapor from a unit which serves to condense milk. The latter type of service is of particular interest since it points up the possibility of achieving extremely large savings in many process industries by thermo-compression of vapor from evaporating or distilling units of various kinds.

Sugar refining, for example, makes use of double- and triple-effect evaporators in which the vapor boiled off of one unit is used as the "steam" for heating the following unit. In such a case the possibility of thermo-compressing vapor from the last stage offers attractive possibilities with respect to reduction in the amount of high-pressure steam required at the first stage. An obvious disadvantage of industrial and water heating applications for the heat pump lies in the fact that the required temperatures are likely to be in excess of 140° F; hence the condenser pressure will necessarily be higher than for air conditioning application. The potentialities of industrial application of the heat

pump have barely been touched in the United States, and it now seems more than probable that extensive developments in this field will occur within the next few years.

6.6. Steam Jet Refrigeration. Steam jet refrigeration depends on kinetic energy developed during the flow of steam through nozzles,

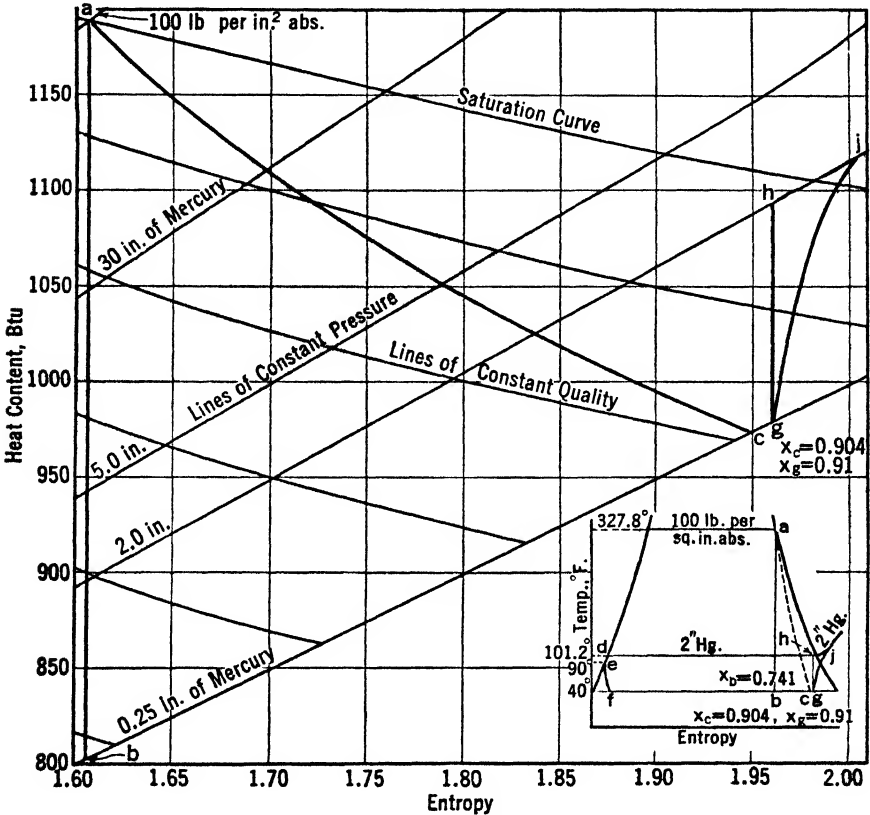


FIG. 6-7. Method of calculation of the steam jet refrigeration cycle.

to maintain a low vacuum in the evaporator (shown in Fig. 16-24). The steam nozzles deliver the high-velocity steam to a combining chamber where it meets the slow-moving water vapor from the evaporator. The mixture then passes through a delivery tube or booster where a large part of the velocity head is changed into pressure head in a manner very similar to the action of an injector or a venturi meter. The pressure in the condenser is about 2 in. of mercury, corresponding to about 101° F liquefaction temperature. This condenser pressure appears to be very low, but with 40° evaporation temperature the ratio

of compression is $2.0/0.248 = 8.1$, which is about the limit for single-stage air compression.

The condensing water usually may be cooled by means of a cooling tower. The make-up water may be taken from other sources, but if it is from the cooling tower it will have a temperature of 90°F as an average. The water vapor leaving the evaporator is not dry unless it is made so by the use of eliminator surfaces, which are not employed extensively because of their resistance to fluid flow, and a value of the steam quality x of about 0.95 is usual.

Example. Find the weight of dry, saturated motive steam at 100 psia for 40° evaporation and 2 in. of mercury condenser pressure per ton of refrigeration per hour, and the heat removed by the condenser. The nozzle efficiency e_n is 0.85; the entrainment efficiency e_e is 0.65,† and the diffuser efficiency e_d is 0.80. Let w_s equal the weight of motive steam per w pounds of mixture leaving the evaporator.

Solution. The hs (Mollier) diagram (Fig. 6·7) shows that although the theoretical expansion of steam is indicated by the line ab the friction losses in the nozzle and in the entrainment result in the actual process ac . The vapor mixture evaporated is usually drier than that at c , and the resulting quality of the mixture may have that indicated at g , which in this example is assumed to be 0.91. The kinetic energy developed must be sufficient to pump the mixture up to the point h theoretically, but (allowing for the diffuser efficiency) to the point j practically. Then

$$(w_s + w) \frac{h_h - h_g}{e_d} = w_s (h_a - h_b) e_n e_e \quad (6.2)$$

Therefore

$$\frac{w_s}{w} = \frac{\frac{h_h - h_g}{e_d}}{[(h_a - h_b)e_n e_e] - \frac{h_h - h_g}{e_d}} = \frac{h_h - h_g}{[(h_a - h_b)e_n e_e e_d] - (h_h - h_g)} \quad (6.3)$$

Assuming that $x_g = 0.91$ and taking values of h from the Mollier chart constructed by Goodenough,

$$\frac{w_s}{w} = \frac{117.0}{[387.0 (0.85 \times 0.65 \times 0.80)] - 117.0} = 2.164 \text{ lb}$$

$$0.91 (2.164 + 1.0) = (2.164 \times 0.904) + (1 \times x)$$

Therefore

$$x = 0.923$$

Assume the make-up water to be at 90°F . Then the refrigeration, per pound, in Btu, is

$$(h_g)_{40^\circ} - [(1 - x)L]_{40^\circ} - (h_f)_{90^\circ} = 1076.8 - (0.077 \times 1068.8) - 58.0 = 936.5$$

† Peter Kalustian, *Refrig. Eng.*, October, 1934; P. Ostertag, *Kalteprozesse*.

Weight of motive steam per ton of refrigeration per hour

$$= \frac{12,000}{936.5} \times 2.164 = 27.72 \text{ lb}$$

Heat removed from steam mixture by the condenser per ton of refrigeration per hour

$$= [(1188.4 - 58.0) 27.72] + 12,000 = 43,335 \text{ Btu}$$

PROBLEMS

1. A thin-walled container having negligible thermal capacity contains 100 lb of liquid carbon dioxide. Room temperature is 70° F. If the pressure within the container is allowed to drop to atmospheric so rapidly that no appreciable heat flow occurs through the walls of the container and if the escaping vapor is considered to be saturated and at atmospheric pressure, determine the weight of solid carbon dioxide that will form in the container.

2. Investigate the variation in pressure in a container of liquid carbon dioxide as a result of diurnal temperature changes in your locality. As a result of such changes what per cent of the material within the container undergoes a cyclic phase change each 24 hours?

3. Carbon dioxide "slush" at 69.88° F contains 10 per cent by weight of solid. Tabulate the various percentages of liquid which this "slush" can contain, assuming that the state of the carbon dioxide prior to expansion to the triple point was at pressure of 200 psia and quality of 50 per cent.

4. A dry ice plant operates between atmospheric pressure and 800 psia with three stages of isentropic compression and liquid intercooling to saturation between stages. Multiple expansion is used, and the intermediate pressures are selected so that the work of all three stages is the same. (a) Calculate the weight of dry ice resulting from a condenser flow rate of 1000 lb/hr. (b) Calculate the power requirements in horsepower per ton of ice. (c) Calculate the displacement in cubic feet per minute per ton of ice for each stage and the required diameter of each cylinder, assuming single-acting cylinders on the same shaft (neglect shaft cross-section area) and with stroke of 10 in. Compressor speed is 300 rpm.

5. A solid carbon dioxide plant is to be designed for one-stage operation between atmospheric pressure and 350 psia. An ammonia system operating between 0° F and 90° F condenses the carbon dioxide in the ammonia evaporator under conditions such that there is no appreciable temperature difference between the condensing carbon dioxide and the evaporating ammonia. Both carbon dioxide and ammonia systems operate on the simple saturation cycle. Calculate the flow rates of the two refrigerants through their respective condensers if the net dry ice output is 1000 lb/hr and if make-up carbon dioxide vapor is precooled from 70° F by passage through the snow chamber.

6. A Carnot cycle is used to raise 100 Btu from 50° F to 150° F. Compare the coefficient of performance of this system when operating as a heat pump with its cop when operating as a refrigerating machine.

7. Compare the heat pump power requirements to raise 100 Btu through 100° F from 0° F with the requirements to raise 100 Btu through 100° F from 50° F. (Note that the 100 Btu is the quantity of heat *delivered* by the heat pump in each case.)

8. A heat pump is to raise 1000 lb/min of water from 60° F to 100° F. The heat source is a large body of water at a temperature such that the evaporator of the

system can operate at 40° F. (a) If a single-stage pump is used with condenser temperature of 110° F (operating on the Carnot cycle) determine the power required. (b) If two pumps are used with condenser temperatures of 90° F and 110° F determine the power requirements. (c) If four pumps were used with condenser temperatures of 80° F, 90° F, 100° F, and 110° F (the water being heated through 10 degrees in each pump) determine the power requirement. (d) Extrapolate from the above results to estimate the power that would be required if the heating process were accomplished through use of an infinite number of heat pumps each serving to heat the water through an infinitesimal temperature rise.

9. Repeat Problem 6, using a simple saturation cycle with ammonia as the refrigerant, and compare the results with those obtained from the Carnot cycle.

10. Repeat Problem 7, using a simple saturation cycle with ammonia as the refrigerant, and compare the results with those obtained from the Carnot cycle.

11. Repeat Problem 8 with ammonia as the refrigerant (simple saturation cycle) and compare the results with those obtained for the Carnot cycle.

12. A heat pump is required to deliver 10000 Btu/hr from a condenser operating at 86° F. Suction temperature (simple saturation cycle) is 10° F. (a) Determine the power requirement if compression occurs in one stage. (b) Determine the power requirement if compression occurs in two stages (equal work per stage) with liquid intercooling and with multiple expansion.

13. A steam jet refrigerating machine has motive steam, assumed dry and saturated, at 100 lb psia, make-up water at 85° F, condenser pressure 2 in. of mercury. Evaporating temperature is 45° F.

$$e_n = 0.85 \quad e_c = 0.65 \quad e_d = 0.80$$

Quality of steam mixture at beginning of compression 0.91. Find weight of motive steam per 1 lb of steam mixture from the evaporator. Find steam required per ton of refrigeration per hour.

14. A tank containing 100 lb of water at 70° F is evacuated until the pressure is so low that the corresponding saturation temperature is 36° F. How many pounds of 36° F water remain in the tank?

15. Assume use of water as a refrigerant in a simple saturation cycle operating between 40° F and 80° F. Calculate the displacement per ton of refrigeration and determine the requisite cylinder size for a reciprocating double-acting compressor with bore equal to stroke and speed of 500 rpm. Discuss.

16. In Fig. 6·3 compare the coefficient of performance for the ammonia cycle with evaporation at 15° as contrasted with a Freon-11 cycle at 6° F and with liquefaction in each case at 80° F.

CHAPTER VII

**ADSORPTION AND ABSORPTION SYSTEMS;
DIFFUSION REFRIGERATION**

One of the oldest refrigerating devices was designed on the absorption principle, using heat directly to drive the refrigerant out of the absorbing medium and into the condenser. The absorbent may be liquid or solid, and the operation may be continuous or intermittent. If operation is continuous a liquid pump may be used, or, with the aid of an inert gas, circulation may be made possible by means of a percolator action as well

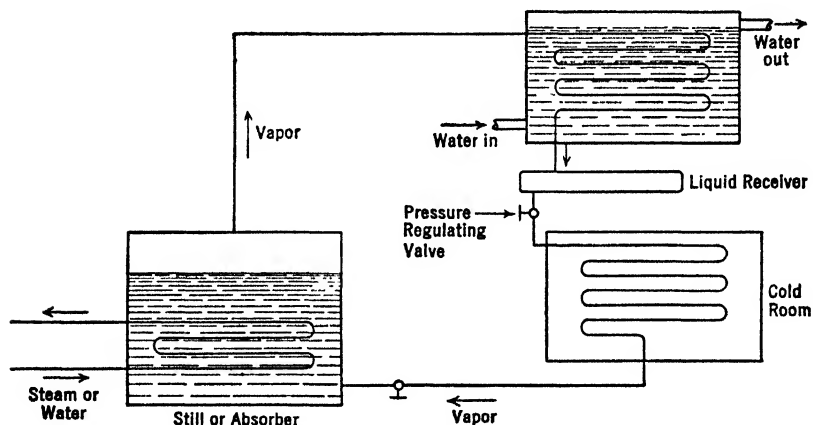


FIG. 7.1. The Carré absorption machine (intermittent).

as by the reduction of the density on account of the rise of temperature due to heating. The machine may have only two fluids, the solvent and the refrigerant, or, again, it may have as many as four or five.

Figure 7.1 represents the Edmund Carré intermittent machine of about 1850 which used sulphuric acid as the absorbing medium; and Fig. 7.2 the more common design using ammonia in solution in water which, although modified in detail, has been in successful operation for a period of more than 60 years. The heat may be applied in any convenient manner, as by a gas burner, electric cartridge, or, in commercial machines, by exhaust or even medium-pressure steam. The action of heat is to cause a distillation process where both water vapor and super-

heated ammonia pass out of the boiler, or *generator*, to the analyzer and then to the *dehydrator* or rectifier. The vapor is cooled to within 10°

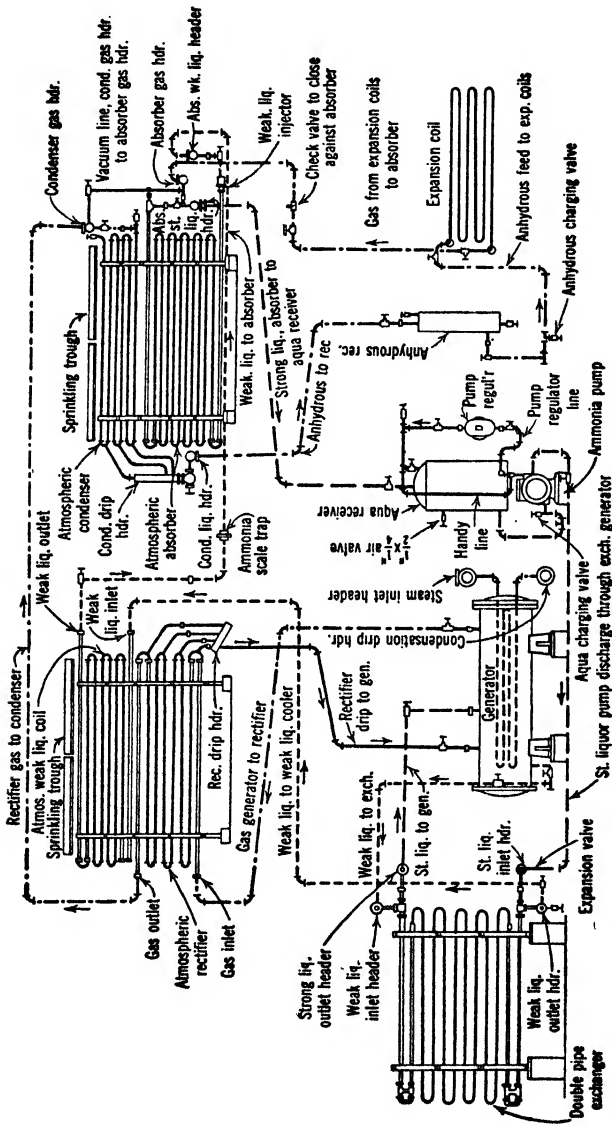


FIG. 7-2. Pipe arrangement for the adsorption refrigerating machine.

or 20° F of the liquefaction temperature in the dehydrator with the condensation of water vapor and the incidental absorption of ammonia so that the mixture entering the condenser has approximately 0.2 per

cent by weight of water vapor present. The condenser, pressure-reducing valve, and evaporator are practically identical with similar parts in the compression refrigerating system. Finally the ammonia vapor enters the absorber where it unites with the cool, weak solution from the generator, and a strong solution is formed. The strong solution is then pumped through a heat exchanger into the analyzer where it tends to cool the distillate being boiled off from the generator. No weak solution cooler is indicated in the figure.

The more common solid absorbents are the chlorides of calcium, barium, and strontium, methyl and ethyl amine, all of which take up ammonia as ammonia of crystallization. Silica gel, made from sodium silicate and sulphuric acid, and activated charcoal are also common solid absorbents which condense the vapor on the surface by means of a surface phenomenon known as *adsorption*. Other inert substances having a large surface could be used, such as activated alumina, ferric hydroxide, titanium oxide, stannous oxide, and other gels. The best solid adsorbent is that which is not affected by the continued presence of the refrigerant, is a good conductor of heat, and is able to hold large quantities of the refrigerant per unit weight. With silica gel, which has the appearance of glass and the composition silicon dioxide, poor thermal conductivity has to be circumvented by having thin layers. Silica gel appears to react with ammonia, so that in refrigeration sulphur dioxide must be chosen as the refrigerant.

Very naturally, if the absorption machine is one using a solid material, as silica gel and sulphur dioxide, the problem of distillation is a very simple one. It is necessary only to volatilize the refrigerant in one vessel and condense the resulting vapor in another. However, if the absorbent is a liquid, like water, especially if the two liquids vaporize at about the same temperatures, the problem becomes much more complicated and the laws governing the process of distillation must be understood.

7-1. Distillation. If a well-mixed liquid is made up of two kinds of substances of low mutual solubility each will give off its own molecules as if occupying the container separately. The total pressure will be the sum of the two partial pressures plus that of any gas present.

If the two substances are mutually soluble the escaping tendency of each substance is reduced by the presence of the other, the result being that each partial pressure also will be reduced. The basic requirement of a separation by distillation of the components of such a mixture is that the composition of the vapor above the liquid must be different from the composition of the liquid mixture from which the vapor was distilled. The action of the distillation can be seen from Fig. 7-3.

Point *c* represents a single liquid of composition *A* which is the more volatile of the two under consideration, whereas point *a* represents a single liquid of composition *B*. Should a mixture of composition *e*, 20 per cent by weight of liquid *A* and 80 per cent by weight of liquid *B*, be heated slowly it will boil at a temperature t_e . The first vapor passing off, or all the vapor passing off if the mixture could be maintained at a fixed composition *e*, will have the composition indicated by the point *f* on the vapor line corresponding to the temperature t_e .

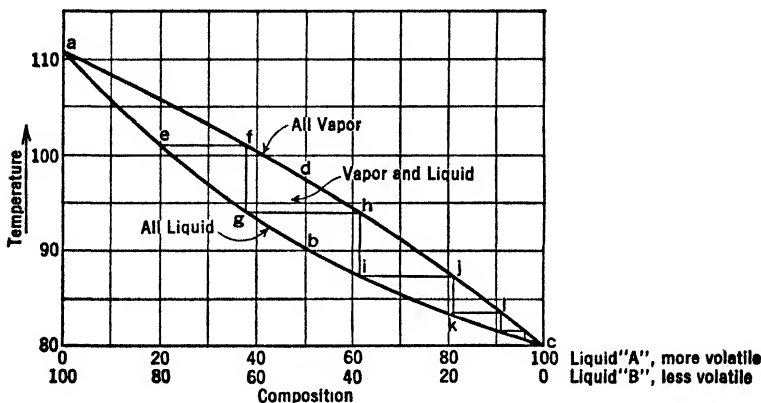


FIG. 7-3. Fractional distillation.

Raoult's law, which can sometimes be used to compute the equilibrium curve from data of the pure components, is stated as follows:

$$P_a = P_a'x \quad (7 \cdot 1)$$

where P_a is the partial pressure of component *A* over a solution in which the mole fraction of *A* is x and P_a' is the vapor pressure of *A* in the pure state at the temperature of the liquid.

Henry's law can also be used at times. This law states that $P_a' = kx$, where P_a' and x have the same definition and k is the Henry law constant.

Wilson* found that a satisfactory equation for expressing the relationship between the concentration of a solution of ammonia in water and the vapor pressure of the water above that solution was not available. However, he considered that the general form of the equation

$$p = (1.0 - \Delta)x_{\text{water}} p_{\text{water}} \quad (7 \cdot 2)$$

could be used, where p is the partial pressure of the water vapor above the solution, Δ is a function of x_{ammonia} , 0.1 x_{ammonia} for values of x_{ammonia}

* Wilson, "The Total and Partial Pressures of Aqueous Ammonia Solutions," *Univ. Illinois Eng. Exp. Sta. Bull.* 146.

below 0.53 and 0.055 for higher values of x_{ammonia} , x is the mole fraction of water or ammonia, as the case may be, p_{water} is the pressure of pure water at the temperature under consideration.

It is convenient to consider the weight in moles rather than in pounds or kilograms. Since the molecular weights of ammonia and water are so nearly alike it is a very simple matter to convert from one to the other. Let y represent the weight of the water present in a solution and z the weight of ammonia. The weight concentration is therefore $z \div (y + z)$, and the molal concentration is $(z/17) \div [(y/18) + (z/17)]$ or

$$k = \frac{0.944 y + z}{y + z} \quad (7.3)$$

where k is the multiplier to change molal to weight concentrations.

7.2. Adsorption. The following explanation may serve to visualize the action of adsorption. The porous material has the ability to reduce the pressure by permitting the molecules of the vapor to enter the pores of the adsorption material and by offering resistance to their exit. The diameter of the pores of silica gel (the internal volume of which is about 50 per cent of the external, from a calculation involving the known reduction in the vapor pressure of the liquid) is about 4×10^{-7} cm, whereas most substances are supposed to have a molecular diameter of 3×10^{-8} cm. Therefore the pores are so small that approximately 10 molecules in line would span each opening in the gel. If the volatile liquid wets the adsorbent material the meniscus will become very prominent and the surface available for releasing the molecules of the liquid to the surrounding space will be small in proportion to the area of opening of the pores. Molecules having velocities greater than the average velocity in the liquid will break through the liquid surface the same as usual, but instead of entering into the space outside the adsorbent a large majority of the molecules will impinge on and enter the liquid meniscus on the other side of the pore. The final result is that if the porous material is kept cool the pores may adsorb as much as 25 per cent of their weight of the vapor. In addition to this action there is also the probable formation of a layer of the adsorbed vapor, one molecule deep, on the surface of the solid combined with a second layer of vapor of variable thickness held by the molecular attraction between the vapor and the solid. Charcoal probably depends on molecular attraction, whereas silica gel depends on capillary adsorption.

In general, vapors from a liquid of a high boiling point are more strongly adsorbed than vapors from a liquid of a low boiling point,† and

† Miller, *Colloid Chemistry*, p. 119.

this tendency decreases with a rise of temperature and increases with the increase of the partial pressure of the vapor being adsorbed.

Certain solids and gases combine to form compounds that may be either loose or very stable. The exact principle of adsorption is still a disputed question, but some evidence seems to indicate that as stated it is due to liquefaction of the gas and to retention by capillary action in the exceedingly fine pores of the adsorbing solid. Whatever the mechanism,

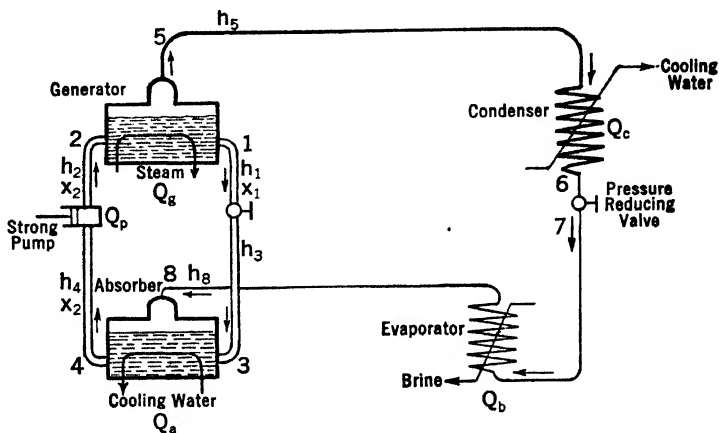


FIG. 7-4. Schematic diagram of the absorption cycle.

the equilibrium between the gas or vapor and the adsorbing solid is usually exponential over certain ranges in concentration and may be expressed by the *Freundlich equation*:

$$x = ap^{1/n}$$

where x is the amount of gas adsorbed per unit quantity of adsorbing solid, p is the partial pressure of the gas in equilibrium with the solid, and a and n are constants but n is usually greater than 1.0. For the adsorption of sulphur dioxide by silica gel the Freundlich equation can be modified to

$$x = a(rs)^{1/n} \quad (7.4)$$

where s is the surface tension and r is the relative humidity or the ratio of the partial to the saturation pressure.

McGavack and Patrick† found that, if x is expressed in cubic centimeters of liquid sulphur dioxide per gram of silica gel, n is 2.24 over a wide range of temperature.

7.3. Theory of the Absorption Machine. Referring to Fig. 7-4, which represents the entire cycle of operations omitting the weak aqueous

† *J. Am. Chem. Soc.*, Vol. 42, p. 946, 1920.

solution cooler, the exchanger, and the dehydrator, and assuming that there are no losses due to radiation and convection, the heat balance becomes

Q_g = heat supplied to generator

Q_p = heat equivalent of pump work

Q_b = heat supplied by brine, etc.

$Q_s = Q_g + Q_p$

Q_a = heat removed by absorber cooling water

Q_c = heat removed by condenser cooling water

x_1 = concentration of weak aqueous solution

x_2 = concentration of strong aqueous solution

n = number of pounds of strong solution per pound of ammonia evaporated

W = weight in pounds of ammonia vaporized

x_g = concentration of vapor leaving generator

h = enthalpy of the substance = $u + AP\bar{V}$

L = vertical lift in feet

$$Q_g + Q_p + Q_b = Q_a + Q_c$$

or

$$Q_s + Q_b = Q_a + Q_c$$

But the work of the strong aqueous pump is small in proportion to Q_g , and therefore $Q_s = Q_g$. Dividing both sides of the equation by W ,

$$q_s + q_b = q_a + q_c \quad (7.5a)$$

During equilibrium

$$nx_2 - (n - 1)x_1 = x_g$$

$$n = \frac{x_g - x_1}{x_2 - x_1} \quad (7.5b)$$

where x_g may be very nearly 1.0 after passing the dehydrator.

7.4. Heat Balance in the Generator. If W pounds of vapor are formed, then during equilibrium

$$\sum W u + \sum A \int P \bar{V} dW + \sum Q + \sum A W L = 0 \quad (7.5c)$$

and, if the pressure is taken as constant,

$$\sum A \int P \bar{V} dW = \sum A P W \Delta \bar{V} \quad \text{or} \quad \sum A P W \bar{V}$$

as the volume of the liquid is small, and, putting $h = u + AP\bar{V}$,

$$\Sigma Wh + \Sigma Q + \Sigma A WL = 0$$

or for 1 pound of the vapor this becomes

$$h + q + AL = 0 \quad (7.5d)$$

Referring to the figure this becomes, for the generator,

$$q_o + nh_2 = h_5 + [(n - 1)h_1] \quad (7.5e)$$

or

$$\begin{aligned} q_o &= h_5 + [(n - 1)h_1] - nh_2 \\ &= h_5 + [n(h_1 - h_2)] - h_1 \end{aligned}$$

As the pump work is $n(h_2 - h_4)$,

$$q_s = q_o + q_p = h_5 + [n(h_1 - h_4)] - h_1 = h_5 - h_0 \quad (7.5f)$$

7.5. Heat Balance in the Absorber. In the absorber the heat balance becomes

$$\begin{aligned} q_a &= [(n - 1)h_3] + h_8 - nh_4 \\ &= h_8 - h_3 + [n(h_3 - h_4)] \end{aligned}$$

but, owing to the throttling effect in the weak solution line $h_1 = h_3$,

$$h_3 - [n(h_3 - h_4)] = h_1 - [n(h_1 - h_4)] = h_0 \quad (7.5g)$$

Therefore

$$q_a = h_8 - h_1 + [n(h_1 - h_4)] = h_8 - h_0 \quad (7.5h)$$

which is an expression for the heat removed by the cooling water in the absorber, if there are no losses. But

$$q_s = h_5 - h_1 + [n(h_1 - h_4)]$$

and therefore

$$q_a - q_s = h_8 - h_s \quad (7.6)$$

7.6. Applications. In the case of the absorber of the absorption machine nearly pure ammonia vapor, with a trace of water vapor, mixes with a larger weight of the weak solution from the generator to form a stronger solution. The weight of the stronger solution, in pounds per pound of ammonia vapor absorbed from the evaporating coils (assuming a quality of the vapor of 1.0), is

$$nx_2 - [(n - 1)x_1] = 1.0$$

$$n = \frac{1.0 - x_1}{x_2 - x_1} \text{ lb} \quad (7.7)$$

and the weight of the weaker solution from the generator, per pound of ammonia absorbed, is $n - 1$ pounds. From Fig. 7·5 the heat balance becomes

$$xW + x_{\partial} \partial w = (x + \partial x)(w + \partial w)$$

and the rate of change of enthalpy with change of concentration while the pressure remains constant will be

$$\left(\frac{\partial h}{\partial x}\right)_p = \frac{h_{\partial} - h - q_{\partial}}{x_{\partial} - x}$$

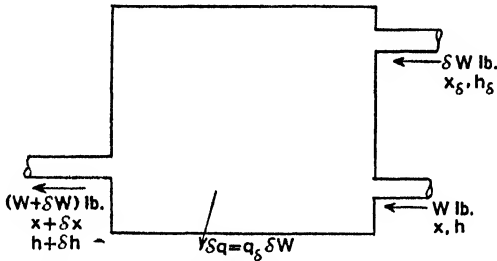


Fig. 7·5. Flow diagram for the absorption machine.

If the amount of the solution is large and the temperature does not change, the equation becomes

$$\left(\frac{\partial h}{\partial x}\right)_{p,t} = - \frac{h' - h - q''}{x} \tag{7·8}$$

The specific heat of liquid ammonia may be closely approximated by the formula

$$c = 1.080 + 0.000488t + 0.0000028t^2 + 0.000000026t^3$$

where t is in degrees Fahrenheit. The specific heat of ammonia solutions may be found from the formula for mixtures:

$$c_m = \frac{\sum wc}{\sum w} = [1.0 (1 - x)] + cx \tag{7·9}$$

where x is the decimal part of 1 pound of mixture of ammonia and water. The heat of solution is given approximately by the formula§

$$h_s = [345 (1 - x)] - 400x^2$$

where h_s is the heat of solution of 1 pound of ammonia liquid in a

§ H. Mollier, "Lösungswärme von Ammoniak in Wasser," *Mitt. Forsch.-Arb.*, Vols. 63 and 64, 1909.

solution of ammonia in water of average concentration $x = (x_1 + x_2)/2$. The heat of solution is zero for $x = 0.59$ and higher values.

The pressure in the absorber is determined by the pressure of equilibrium of the strong solution in the absorber, which in turn may be found by the temperature and concentration. The Goodenough equation for the p, t, x relationship is

$$\frac{\theta}{T} = \frac{1}{1.0 + [070356 (1 - x^2)]}$$

and

$$z = (x + 0.05)^{1/2} (1.347 - 2.9x + 1.77x^2)$$

where x is the mole fraction of ammonia in the solution, θ is the temperature of saturated ammonia vapor, and T is the temperature of saturated water vapor under identical pressures. || These values are shown in Fig. 7·6. Other tabulated values from *Bulletin* 146 are given in Tables 7·1 and 7·2. The heat balance for the absorber can now be calculated as is shown in the following example.

Example. A 35 per cent solution at 80° leaves the absorber and a 30 per cent solution at 120° enters. Find the heat necessary to be removed from the absorber by the cooling water if the entering ammonia is superheated 10°.

Solution. The minimum weight of the entering solution to the absorber is

$$n = \frac{1.0 - x_2}{x_1 - x_2} - 1 = \frac{1.0 - 0.30}{0.35 - 0.30} - 1 = \frac{13.0 \text{ pounds per pound of ammonia gas}}{\text{from the evaporating coils}}$$

Evidently the weak solution must be cooled to 80°. If the average specific heat of the ammonia in the liquid state is taken as 1.175 the specific heat of the mixture will be $(0.7 \times 1.0) + (0.3 \times 1.175) = 1.053$, and the amount of heat that must be removed by the cooling water in the absorber per pound of ammonia entering from the evaporator is

$$Q = 13.0 \times 1.053 \times (120 - 80) = 547.6 \text{ Btu}$$

The pressure, in equilibrium with a concentration of 35 per cent by weight of ammonia and at 80° F, is 20.5 psia. The enthalpy of the entering ammonia at 20.5 lb, and superheated 10°, is 611.9 Btu. It enters into solution where the enthalpy of the liquid ammonia is $h_{80^\circ} = 132.0$ Btu. The heat lost by the ammonia, and gained by the cooling water, is $611.9 - 132.0 = 479.9$ Btu, and the heat evolved during the process of the formation of the solution is given by $[345(1 - x)] - 400x^2$, where $x = (0.35 + 0.30)/2$ or $x = 0.325$. The heat of solution is 190.8; therefore the total heat to be removed by the cooling water is $547.6 + 479.9 + 190.8 = 1218.3$ Btu. It is therefore seen that economical

|| Wilson, "The Total and Partial Pressures of Aqueous Ammonia Solutions," *Univ. Illinois Eng. Exp. Sta. Bull.* 146.

TABLE 7-1a
PARTIAL PRESSURES OF WATER VAPOR ABOVE AQUA AMMONIA
(Pressures in Pounds per Square Inch Absolute)

Temperature, °F	Molal Concentration of Ammonia in the Solutions in Percentages										
	0	5	10	15	20	25	30	35	40	45	50
32	0.09	0.084	0.079	0.074	0.070	0.065	0.060	0.056	0.051	0.047	0.042
40	0.12	0.115	0.108	0.101	0.095	0.089	0.083	0.076	0.070	0.064	0.058
50	0.18	0.17	0.16	0.15	0.14	0.13	0.12	0.11	0.10	0.094	0.085
60	0.26	0.24	0.23	0.21	0.20	0.19	0.17	0.16	0.15	0.13	0.12
70	0.36	0.34	0.32	0.30	0.28	0.26	0.25	0.23	0.21	0.19	0.17
80	0.51	0.48	0.45	0.42	0.40	0.37	0.34	0.32	0.29	0.27	0.24
90	0.70	0.66	0.63	0.58	0.55	0.51	0.47	0.44	0.40	0.37	0.33
100	0.95	0.90	0.85	0.79	0.74	0.69	0.64	0.59	0.55	0.50	0.45
110	1.27	1.20	1.14	1.07	1.00	0.93	0.86	0.80	0.73	0.67	0.60
120	1.69	1.60	1.51	1.42	1.33	1.24	1.15	1.06	0.97	0.89	0.80
130	2.22	2.10	1.98	1.86	1.74	1.62	1.51	1.39	1.28	1.17	1.05
140	2.89	2.73	2.57	2.42	2.26	2.11	1.96	1.81	1.66	1.52	1.37
150	3.72	3.51	3.31	3.11	2.91	2.72	2.52	2.33	2.14	1.95	1.76
160	4.74	4.48	4.22	3.97	3.71	3.46	3.22	2.97	2.73	2.49	2.25
170	5.99	5.66	5.34	5.02	4.70	4.38	4.07	3.75	3.45	3.15	2.84
180	7.51	7.10	6.69	6.30	5.89	5.49	5.10	4.71	4.33	3.94	3.57
190	9.34	8.83	8.32	7.82	7.32	6.83	6.34	5.86	5.38	4.91	4.44
200	11.53	10.90	10.27	9.65	9.04	8.43	7.83	7.23	6.64	6.06	5.48
210	14.12	13.35	12.58	11.82	11.07	10.32	9.59	8.86	8.13	7.42	6.71
220	17.19	16.25	15.32	14.39	13.48	12.57	11.67	10.78	9.90	9.03	8.17
230	20.78	19.64	18.51	17.40	16.29	15.19	14.11	13.03	11.97	10.91	9.87
240	24.97	23.60	22.25	20.91	19.58	18.26	16.95	15.66	14.38	13.12	11.86
250	29.83	28.20	26.58	25.00	23.39	21.82	20.25	18.71	17.18	15.67

TABLE 7-1b
PARTIAL PRESSURES OF AMMONIA ABOVE AQUA AMMONIA
(Pressures in Pounds per Square Inch Absolute)

Temperature, °F	Molal Concentration of Ammonia in the Solutions in Percentages									
	5	10	15	20	25	30	35	40	45	50
32	0.11	0.40	0.90	1.51	2.54	4.14	6.43	8.88	14.13	19.36
40	0.22	0.58	1.14	1.92	3.16	5.13	7.98	11.98	17.14	23.33
50	0.47	0.89	1.51	2.53	4.16	6.63	10.24	15.24	21.56	29.18
60	0.62	1.19	2.00	3.31	5.36	8.48	13.06	19.15	26.92	36.14
70	0.83	1.32	2.69	4.28	6.67	10.76	16.33	23.84	33.20	44.25
80	1.06	1.98	3.34	5.45	8.69	13.52	20.29	29.40	40.69	53.84
90	1.36	2.52	4.25	6.88	10.89	16.75	25.04	35.94	49.45	64.98
100	1.72	3.20	5.34	8.60	13.53	20.68	30.57	43.57	59.49	77.85
110	2.14	4.00	6.65	10.64	16.65	25.21	37.01	52.43	71.20	92.59
120	2.67	4.95	8.21	13.09	20.20	30.54	44.56	62.62	84.44	109.40
130	3.28	6.09	10.05	15.93	24.58	36.74	53.16	74.27	99.69	128.45
140	3.98	7.41	12.21	19.23	29.43	43.77	62.97	87.53	116.72	149.93
150	4.78	8.92	14.70	23.09	35.09	51.91	74.28	102.51	136.15	173.64
160	5.68	10.70	17.57	27.45	41.56	61.03	86.91	119.37	157.71	200.45
170	6.75	12.72	20.85	32.41	48.89	71.48	101.09	138.30	181.95	230.36
180	7.90	14.96	24.56	38.13	57.19	83.07	116.97	159.37	208.66	263.43
190	9.23	17.55	28.78	44.49	66.39	96.22	134.89	182.72	238.39	299.86
200	10.70	20.44	33.39	51.58	76.90	110.85	154.58	208.56	270.94	340.02
210	12.26	23.68	38.76	59.65	88.48	126.83	176.24	236.97	307.08	383.99
220	14.02	27.15	44.61	68.43	101.24	141.74	200.46	268.30	346.07	431.43
230	15.95	31.09	51.06	78.14	115.45	164.17	226.67	302.50	389.20	483.53
240	17.92	35.40	58.00	89.02	130.94	185.79	255.26	339.72	435.78	530.44
250	20.13	40.09	65.74	100.69	147.66	209.37	286.89	380.42	486.73

TABLE 7-2
MOLE PER CENT AQUEOUS CONCENTRATION IN AMMONIA WATER VAPORS

Temperature, °F	Molal Concentration of Ammonia in the Solutions in Percentages										
	0	5	10	15	20	25	30	35	40	45	50
32	100	24.3	13.2	7.63	4.43	2.50	1.43	0.856	0.514	0.335	0.216
40	100	25.3	14.1	8.15	4.73	2.74	1.59	0.943	0.581	0.372	0.248
50	100	26.6	15.2	9.09	5.24	3.03	1.78	1.060	0.652	0.434	0.290
60	100	27.9	16.2	9.50	5.69	3.42	1.97	1.210	0.777	0.481	0.331
70	100	29.1	17.4	10.30	6.14	3.65	2.27	1.390	0.873	0.569	0.383
80	100	31.6	18.5	11.20	6.89	4.08	2.45	1.550	0.978	0.659	0.444
90	100	32.7	20.0	12.00	7.40	4.47	2.73	1.730	1.100	0.742	0.505
100	100	34.4	21.0	12.90	7.92	4.85	3.00	1.890	1.250	0.834	0.574
110	100	35.9	22.2	13.80	8.59	5.29	3.30	2.110	1.370	0.932	0.644
120	100	37.5	23.4	14.70	9.22	5.75	3.63	2.320	1.520	1.044	0.714
130	100	39.0	24.5	15.60	9.85	6.18	3.95	2.550	1.690	1.160	0.811
140	100	40.7	25.8	16.50	10.50	6.69	4.28	2.790	1.860	1.286	0.906
150	100	42.3	27.1	17.50	11.20	7.19	4.63	3.080	2.040	1.410	1.004
160	100	44.1	28.3	18.40	11.90	7.69	5.01	3.300	2.230	1.550	1.110
170	100	45.6	29.6	19.40	12.70	8.22	5.38	3.580	2.430	1.700	1.230
180	100	47.3	30.9	20.40	13.40	8.76	5.78	3.870	2.640	1.850	1.340
190	100	48.7	32.2	21.40	14.10	9.31	6.18	4.160	2.860	2.020	1.460
200	100	50.4	33.4	22.30	14.90	9.88	6.59	4.470	3.080	2.190	1.580
210	100	52.1	34.7	23.40	15.70	10.45	7.03	4.780	3.310	2.360	1.720
220	100	53.7	36.1	24.40	16.40	11.05	7.48	5.100	3.560	2.540	1.860
230	100	55.2	37.3	25.40	17.30	11.63	7.91	5.440	3.810	2.730	2.000
240	100	56.8	38.6	26.50	18.00	12.24	8.36	5.780	4.060	2.920	2.150
250	100	58.4	39.8	27.50	18.80	12.88	8.82	6.120	4.340	3.120

results can be secured, as far as the cooling water to the absorber is concerned, by reducing the temperature of the weak solution entering the absorber by means of an efficient exchanger and the use of a nominal difference in the concentrations of the weak and strong solutions.

7.7. The Generator. To a large extent economical results with the absorption machine depend on the performance of the generator. It is here that heat in the form of steam, gas, or electricity is applied. Heat is required to warm the strong solution coming from the exchanger to the average temperature in the generator, to distill the ammonia out of solution, and to overcome such losses as are incidental to the process.

The older designs of generators included an extension at the top, called an *analyzer*, through which the distilled vapors passed, counter flow, with the stronger, but colder, solution. Such an arrangement had several advantages, the principal one being the cooling of the distilled gas somewhat with the added advantage of condensing some of the water vapor passing out with the superheated ammonia vapor. This may be made clear by referring to the p, t, x diagram (Fig. 7.6).

Taking an example where the total pressure in the generator is 145 psia and the solution temperature 215° F, the concentration of the solution in equilibrium will be 30 per cent molal concentration of ammonia and 7.3 per cent molal concentration of water vapor in the vapors above the solution of ammonia. This water vapor, if permitted, would be condensed in the condenser, and the liquid passing to the evaporator would be a strong solution of ammonia and water which would naturally interfere with continued operation and would make final results problematical. The usual method, therefore, is to use a dehydrator in which the vapor mixture is cooled to 10°, 15°, or 20° F above the temperature of liquefaction in the condenser, which is approximately the temperature corresponding to the condenser pressure from the ammonia tables. In the dehydrator the steam condensed absorbs immediately a relatively large weight of ammonia. If the vapor is cooled to 90°, still at a pressure of 145 lb, the water vapor in equilibrium is 0.1 per cent and the solution formed will have a concentration of 78 per cent. The difference, if all the cooling is done in the dehydrator, will be 7.2 per cent, and the ammonia content will be $[78/(100 - 78)] \times 7.2 = 25.5$ per cent. Not only is considerable cooling in the dehydrator necessary and a considerable loss in capacity caused by the absorption of so much ammonia, but also the heat needed in the generator is increased by the amount required to continue this *eddy* current. As the strong solution is always cooler than the solution temperature in the generator, there will be a substantial advantage in cooling the distilled gas with it. Some of the theory will be made clearer by the following illustrative example.

Example. A strong solution of 35 per cent molal concentration enters the generator at 160° F, 145 psia, and 215° F, as given above, and the distilled vapors leave the analyzer at 192°. Find the heat required to deliver 1 lb of ammonia vapor to the condenser.

Solution. Instead of the 7.3 per cent of water vapor in the above example the water vapor content will be 4.0 per cent and 3.9 per cent will be condensed in the dehydrator. The minimum weight of strong solution to be circulated, as before, will be $(1.0 - 0.30)/0.05 = 14.0$ pounds per pound of ammonia vapor entering the absorber. The heat required to raise the temperature of the strong solution to the temperature of the generator, 215°, will be

$$Q = [14.0 \times 1.12 \times (192 - 160)] + [13.5 \times 1.13 \times (215 - 192)] = 852.6 \text{ Btu}$$

where the value of the specific heat is taken for the average of the temperature range. The action is complicated because distillation will begin at 192°, and when 215° is reached all distillation will be completed.

Q = heat of solution, as before = 190.8 Btu at an average temperature of

$$\frac{215 + 192}{2} = 203.5^\circ$$

Q = heat required to distill off superheated ammonia at an exit temperature of 190° from liquid ammonia at an average temperature of 203.5 and an average partial pressure of $145 - 8.1 = 136.9$ psia.

$$Q = 704.3 - 287.9 = 416.4 \text{ Btu}$$

Finally there is 7.3 per cent of water vapor leaving the generator which would require, per pound of ammonia distilled off, $0.073 \times 969.9 = 70.8$ Btu. The total heat required to be supplied to the generator is therefore $852.6 + 190.8 + 416.4 + 70.8 = 1530.6$ Btu. As the analyzer condenses out some of the steam, only 4.0 per cent reaches the dehydrator, and $4.0 \times (0.645/0.355) = 7.27$ per cent of ammonia will be returned in the dehydrator drips. Therefore only 92.7 per cent of the ammonia will reach the condenser carrying with it 0.1 per cent of water vapor.

Equations 7·5f and 7·6 could be used for the solution of the heat balances in the generator and the absorber if values for the enthalpy of solutions of ammonia and water were known. Calculations of the required quantities have been made by Merkel and Bošnjaković,[¶] and some of these values are given in Table 7·3. In order to show the manner of their use the following example is calculated.

Example. The pressure in the generator and the condenser is 9 atm; that in the evaporator and the absorber is 2 atm. The temperature in the generator

[¶] Merkel and Bošnjaković, *Diagramme und Tabellen zur Berechnung der Absorptions-Kältemaschinen*, 1929.

TABLE 7-3

<i>p</i> = 0.2 atmosphere (The subscript <i>s</i> refers to the condition of boiling liquid; the subscript <i>k</i> refers to the condition of condensing vapor.)						<i>p</i> = 4 atmospheres					
<i>t</i> °F	<i>x_s</i>	<i>z_k</i>	<i>h_s</i> Btu per lb	<i>h_k</i> Btu per lb	<i>k</i>	<i>t</i> °F	<i>x_s</i>	<i>z_k</i>	<i>h_s</i> Btu per lb	<i>h_k</i> Btu per lb	<i>k</i>
-40	0.5185	0.9998	-169.2	524.9	0.000694	32	0.9300	1.0000	-14.4	544.0	0.000125
-22	0.4310	0.9983	-151.2	533.9	0.000828	50	0.6560	0.9994	-51.8	556.6	0.000565
-4	0.3640	0.9958	-127.2	543.2	0.000992	68	0.5470	0.9970	-53.3	567.7	0.000724
+14	0.3065	0.9900	-100.8	554.8	0.001043	86	0.4735	0.9940	-40.7	579.6	0.000840
32	0.2530	0.9764	-72.0	571.0	0.001124	104	0.4145	0.9892	-21.6	592.6	0.000936
50	0.2025	0.9470	-42.5	594.7	0.001171	122	0.3640	0.9808	+1.4	607.0	0.001019
68	0.1550	0.8944	+12.2	630.7	0.001153	140	0.3185	0.9662	25.2	614.6	0.001080
86	0.1100	0.8038	18.7	688.0	0.001037	158	0.2755	0.9425	50.8	647.3	0.001118
104	0.0680	0.6450	50.0	782.3	0.000788	176	0.2345	0.9060	77.4	676.4	0.00112
<i>p</i> = 0.5 atmosphere						<i>p</i> = 8 atmospheres					
-40	0.7410	1.0000	-127.4	524.9	0.000397	68	0.9350	0.9998	+26.6	552.6	0.000123
-22	0.5670	1.0000	-142.2	523.2	0.000641	86	0.6700	0.9980	-10.4	565.4	0.000569
-4	0.4755	0.9986	-133.2	540.4	0.000777	104	0.5600	0.9958	-13.3	578.2	0.000737
+14	0.4065	0.9960	-113.0	560.4	0.000890	122	0.4875	0.9918	-1.8	591.8	0.000850
32	0.3475	0.9924	-88.6	560.9	0.000993	140	0.4295	0.9860	+15.8	606.6	0.000942
50	0.2940	0.9830	-61.9	574.6	0.001083	158	0.3800	0.9754	37.4	614.5	0.001015
68	0.2445	0.9640	-34.6	593.3	0.001146	176	0.3365	0.9588	60.8	642.2	0.001070
86	0.1975	0.9304	-5.4	619.6	0.001172	194	0.2960	0.9326	84.6	665.6	0.001096
						212	0.2575	0.8936	109.8	696.2	0.001085
<i>p</i> = 1 atmosphere						<i>p</i> = 10 atmospheres					
-22	0.8560	1.0000	-86.4	529.2	0.000234	86	0.8240	0.9982	23.0	559.8	0.000324
-4	0.6155	1.0000	-116.3	540.0	0.000586	104	0.6305	0.9970	.1	573.1	0.000693
+14	0.5120	0.9988	-113.4	546.5	0.000738	122	0.5400	0.9942	3.6	586.4	0.000779
32	0.4380	0.9960	-96.1	556.6	0.000855	140	0.4735	0.9894	17.2	600.4	0.000885
50	0.3780	0.9926	-73.8	568.4	0.000958	158	0.4195	0.9816	36.0	615.9	0.000969
58	0.3255	0.9850	-49.0	581.8	0.001045	176	0.3725	0.9690	56.5	633.6	0.001035
86	0.2755	0.9700	-22.0	598.7	0.001119	194	0.3300	0.9488	79.9	655.2	0.001075
104	0.2280	0.9430	+ 6.1	621.7	0.001163	212	0.2900	0.9170	104.4	681.4	0.001087
122	0.1830	0.8992	35.3	655.9	0.001154	230	0.2525	0.8708	129.2	716.0	0.001050
<i>p</i> = 2 atmospheres						<i>p</i> = 12 atmospheres					
+14	0.4830	0.9998	-81.0	545.0	0.000478	104	0.7270	0.9986	21.9	566.6	0.000498
32	0.6215	0.9996	-88.9	553.0	0.000674	122	0.5935	0.9956	12.6	579.9	0.000708
50	0.4830	0.9966	-78.5	562.0	0.000802	140	0.5150	0.9920	20.1	594.3	0.000830
68	0.4185	0.9930	-58.3	574.6	0.000908	158	0.4550	0.9858	34.5	609.8	0.000922
86	0.3630	0.9870	-35.3	588.2	0.001000	176	0.4045	0.9758	54.3	626.7	0.000997
104	0.3140	0.9755	-10.4	603.7	0.001077	194	0.3600	0.9594	76.6	646.9	0.001080
122	0.2690	0.9558	+15.8	622.8	0.001132	212	0.3195	0.9340	100.0	670.6	0.001077
140	0.2250	0.9232	43.2	649.1	0.001153	230	0.2815	0.8964	123.8	700.5	0.001066
						248	0.2450	0.8418	148.6	738.7	0.001011
						266	0.2090	0.7660	174.9	787.6	0.000908
						284	0.1735	0.6720	201.6	843.8	0.000776

is 212° F; that in the absorber and condenser is 68° F; that in the evaporator is 14° F.

Solution. From Table 7·3,

$$\begin{aligned} x_1 &= 0.274 \text{ at } 9 \text{ atm and } 212^\circ \text{ F} & \text{and} & & h_1 &= 106.9 \text{ Btu} \\ x_2 &= 0.419 \text{ at } 2 \text{ atm and } 68^\circ \text{ F} & \text{and} & & h_2 &= -57.3 \text{ Btu} \\ x_g &= 0.983 \text{ at } 9 \text{ atm} \end{aligned}$$

The weight in pounds of the strong solution n per pound of ammonia vapor distilled out of the generator is, as before,

$$n = \frac{0.983 - 0.274}{0.419 - 0.274} = 4.89$$

and the weight of the weak solution leaving the generator for the absorber per pound of ammonia vapor boiled out of the evaporator is $4.89 - 1.0 = 3.89$ lb.

From equation 7·5*g*,

$$h_0 = 106.9 - [4.89 (106.9 + 57.3)] = -699.7 \text{ Btu}$$

From Table 7·3, for a pressure of 9 atmospheres and $x_2 = 0.419$, $h_5 = 612.9$ Btu. From equation 7·5*f*,

$$\begin{aligned} q_g &= h_5 - h_0 = 612.9 + 699.7 \\ &= 1312.6 \text{ Btu} \end{aligned}$$

As the pump work of pumping an incompressible liquid, like the strong solution, into the generator is small, the value 1312.6 Btu is also the value for q_s .

The condensate leaving the condenser, with a quality of $x = 0.983$, has a value of $h_6 = 37.6$ Btu/lb, and this is also the value of h_7 . The heat absorbed by the condensing water is

$$q_c = h_5 - h_6 = 612.9 - 37.6 = 575.3 \text{ Btu}$$

The enthalpy of the mixture at the end of the evaporation process, at 2 atm and 14° F, is given by

$$h_8 = h_g - \left[\frac{1}{k} (x_g - x_f) \right]$$

where the subscripts g and f refer to the vapor and the liquid states, and $k = (h_g - h_f)/(x_g - x_f)$ may be found from Table 7·3.

Therefore $h_8 = 546.8 - 37.6 = 509.2$ Btu. The useful refrigeration is then

$$\begin{aligned} h_8 - h_6 &= 509.2 - 37.6 \\ &= 471.6 \text{ Btu} \end{aligned}$$

The heat absorbed by the cooling water passing through the absorber is

$$h_8 - h_0 = 509.2 - (-699.7) = 1208.9 \text{ Btu}$$

and the coefficient of performance is

$$\frac{471.6}{1312.6} = 0.359$$

7-8. The Absorption Type of Household Machine. In order to apply heat directly to the household refrigerating machine it is necessary to use some sort of absorption apparatus. For simplicity this must be air-cooled, automatic, safe, free of all moving parts, and nominal in first and operating costs. Such a machine would be free of all noise, vibration, and need for lubrication.

The first attempt along these lines appears to have been made by Geppert, who secured a United States patent in 1899. In this device Geppert used a neutral gas, air, which he introduced into the absorber and evaporator in order that the ammonia could exist under a partial pressure. As all throttle valves were omitted, the total pressure was the same throughout the system and no pump was required to return the strong solution to the generator. Geppert failed to secure proper circulation, and his device, although very ingenious, was not successful. By modification of the idea, Baltzar von Platen and Carl G. Munters* of Stockholm, Edmund Altenkirch of Germany, and others have made possible, at least in theory, a machine using three, four, and five fluids.† The devices being developed include ammonia and water with hydrogen as a neutral and methyl chloride in a separate circuit as a cooling agent for the absorber, the lye, or caustic, the propane or butane to replace the hydrogen, sulphuric acid, etc. At present the best-known and most successfully developed machine is the von Platen-Munters, usually called the Electrolux, based on the principle of diffusion.

7-9. The Diffusion Absorption Machine. The law of diffusion states that evaporation is proportional to the difference between the pressure of the vapor above the liquid and the partial pressure of the vapor contained in the inert gas. Therefore the temperature of the liquid ammonia in the evaporator must be sufficient to create a pressure enough greater than the partial pressure of the ammonia in the vapor diffused with the inert gas to be able to carry on the evaporation. The partial pressure of the ammonia is least at the beginning and greatest at the end of the evaporation. In the absorber it is exactly the opposite, the partial pressure of the ammonia being least at the end of the process, and the concentration of the weak aqueous solution determining the minimum partial pressure of the ammonia in the mixture entering the evaporator. It should be remembered that in the ordinary absorption machine the concentration of the strong solution in the absorber determines the pressure of evaporation in the evaporating coils.

As the liquid from the condenser is quite warm it is wise to cool it by

* Von Platen and Munters, *Refrig. Eng.*, November, 1925.

† Taylor, *Refrig. Eng.*, May, 1929.

counter flow with the mixture leaving the evaporator, as well as to cause a counter flow of the inert gases from the absorber with the mixture leaving the evaporator. The weak solution as usual is taken from the bottom of the generator, but it loses its excess heat by the exchanger

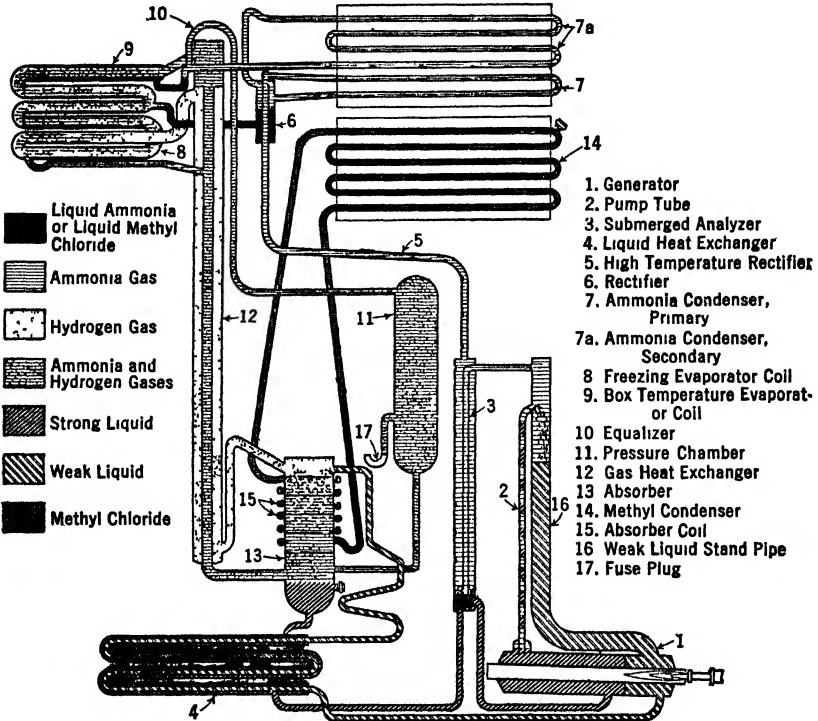


FIG. 7-7. The air-cooled Electrolux, gas heated.

principle above the heating element before it enters exchanger 4 (Fig. 7-7). An example† will make the process in the diffusion machine a little clearer.

Example. Consider the operation of a diffusion absorption machine with air temperatures of 95° to 100° F. The total pressure is 285 psia. Condensation is at 120°, and absorption is at 122° (Fig. 7-8).

Solution. The density of the ammonia leaving the evaporator = $\rho_a = 1/\bar{V}_a$, corresponding to the partial pressure p_a .

The density of the ammonia entering the evaporator = $\rho_0 = 1/\bar{V}_0$, corresponding to the partial pressure p_0 . However, both temperature and partial pressures vary during evaporation; therefore, making corrections, we find that the weight

† Mañuri, *Refrig. Eng.*, August, 1932.

evaporated per cubic foot of gas mixture entering is

$$W_a = \rho_a \left[\left(\frac{p - p_0}{p - p_a} \right) \frac{T_a}{T_0} \right] - \rho_0$$

where p = the total pressure.

If the volume of the gas entering the evaporator per minute is V_0 , the refrigeration per minute is, if h is the net refrigeration per pound,

$$Q_a = W_a h V_0 = h V_0 \left\{ \rho_a \left[\left(\frac{p - p_0}{p - p_a} \right) \frac{T_a}{T_0} \right] - \rho_0 \right\} \quad (7 \cdot 10)$$

Assume that the weak solution is 18 per cent ammonia by weight, the strong solution 33 per cent by weight, and that evaporation at the end of the evaporator

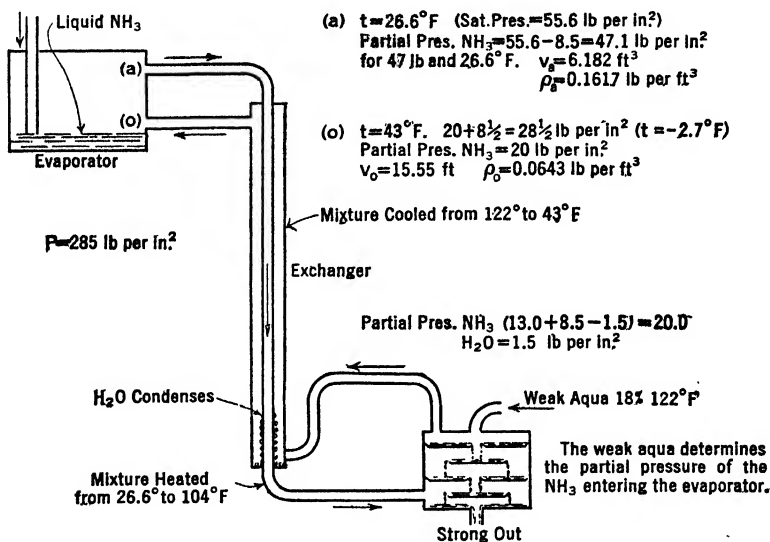


FIG. 7-8. Schematic arrangement of the evaporator and absorber for the diffusion-type absorption machine.

tion coils takes place at 26.6°F . The pressure of equilibrium will be 55.5 psi, but 8.5 psi is required in practice for diffusion. The partial pressure of ammonia at 26.6° is 55.6 psia, and $55.6 - 8.5 = 47.0$ psi, very nearly. For the action in the absorber with a concentration of 18 per cent and a temperature of 122° the pressure is 13 lb. For diffusion from the mixture in the absorber to the weak solution the ammonia pressure in the mixture must be $13 + 8.5 = 21.5$ lb, but 11 per cent of water vapor is present and the partial pressure of the water for 122° is 1.5 psi. Therefore $21.5 - 1.5 = 20.0$ is the partial pressure of the ammonia in the mixture leaving the absorber. During the passage of this mixture through the exchanger, the temperature drops and the greater part of the

water vapor is condensed. The densities (taking the temperature of the gas entering the evaporator as 43°) are

$$\begin{array}{lll} 20 \text{ lb and } 43^\circ & \bar{V}_0 = 15.55 & \rho_0 = 0.0643 \text{ pcf} \\ 47 \text{ lb and } 26.6^\circ & \bar{V}_a = 6.182 & \rho_a = 0.1617 \text{ pcf} \end{array}$$

The ammonia weight evaporated per cubic foot of the mixture entering is

$$W_a = \left[0.1617 \left(\frac{285 - 20}{285 - 47} \right) \left(\frac{486.6}{502.6} \right) \right] - 0.0643 = 0.1099 \text{ lb}$$

The volume of the mixture entering per pound of ammonia evaporated = $1/0.1099 = 9.10$ cu ft. Each cubic foot of the mixture entering contains, at 43°, 0.0643 lb of ammonia and $W = (144 \times 265 \times 1)/(765.9 \times 502.6) = 0.099$ lb of hydrogen.

The specific heat of the mixture is $(3.41 \times 0.099) + (0.0643 \times 0.54) = 0.3722$ and $9.10 \times 0.3722 = 3.39$ Btu °F lb of ammonia evaporated in the evaporator. The gas leaving the evaporator has 0.1617 pound of ammonia per cubic foot, and there is $(144 \times 238 \times 1)/(765.9 \times 486.6) = 0.0916$ pound of hydrogen per cubic foot. The new volume is $9.10 [(264 \times 486.6)/(238 \times 502.6)] = 9.77$ cu ft.

The specific heat of the mixture leaving is $(0.0916 \times 3.41) + (0.1617 \times 0.56) = 0.402$. If the mixture leaving the evaporator reaches a temperature of 104° F in the exchanger, the amount of heat absorbed is $0.402 (104 - 26.6) 9.77 = 305.0$ Btu. This amount of heat can be used to cool the mixture from the absorber and condense the water vapor. The refrigeration required to cool the mixture from the entering temperature of 43° to 26.6°, per pound of ammonia, is $9.10 \times 0.3722 \times (43 - 26.6) = 55.5$ Btu. Refrigeration per pound of liquid ammonia entering the evaporating coils will be, approximately, $615.4 - 55.5 - 179.0 = 380.9$ Btu.

PROBLEM

In the illustrative example on page 126 let the strong solution be 30 per cent and the weak solution 25 per cent. Other conditions are the same. Find the weight of strong solution per pound of ammonia vapor boiled out of the evaporator coils and the total heat required to be removed by the cooling water in the absorber.

CHAPTER VIII

REFRIGERANTS AND PROPERTIES OF SATURATED AND SUPERHEATED VAPORS

It has not been possible, up to the present time, to find a refrigerant that can be called "best" for every kind of refrigeration; hence the choice of the refrigerant to be used is a compromise and depends on the special kind of refrigeration being performed. The following considerations are the principal deciding factors.

(a) **THE VOLUME HANDLED.** The volume of the refrigerant pumped per unit of refrigeration, per unit of time, must be considered although it is the least important factor in making a choice of the refrigerant to be used. For example, carbon dioxide has the least, and water vapor has the largest, displaced volume of the common refrigerants, but carbon dioxide has excessive and water vapor (Table 8·1) the lowest, unit pressures. Where the pressure range is small, as for water vapor where it is less than 1.0 psi and for dichloroethylene and methylene chloride where it is less than 1 atm, the large volume to be handled can be pumped by means of the centrifugal type of compressor, especially in the sizes encountered in comfort cooling. In the household type of machine, however, the statement is made that the larger volume required for sulphur dioxide, as compared with, say, methyl chloride, is an *advantage* because of the small capacity of the compressor.

(b) **PRESSURES.** The pressures encountered in refrigeration are important from the standpoint of the relative amount of material required to withstand bursting, and for the ability to prevent leaks. Leaks are more difficult to prevent as the pressures increase; for example, carbon dioxide as compared with ammonia, and methyl chloride and dichlorodifluoromethane compared with sulphur dioxide.

(c) **INFLAMMABILITY.** Some refrigerants, when mixed with varying amounts of air, have explosive characteristics. Table 8·1, column 14, shows the range of explosibility of the more usual refrigerants when mixed with air. Fire department regulations have ruled out some refrigerants such as ethyl chloride and the hydrocarbons, e.g., butane and propane. Ammonia has caused some bad explosions under unusual conditions such as an arc light in a compressor room with heavy ammonia leaks. As excessive leaks cannot be prevented if material is

TABLE 8-1
COMPARISON OF REFRIGERANTS
Range of Temperature 5° F Evaporation and 86° Liquefaction (No Liquid Subcooling and No Superheat)

Refrigerant	(2) Critical Temperature (°F)	(3) Critical Pressure (psia)	Average Value of $K = C_p/C_v$ (when physical tables in the superheated region are not known)	(4) $(h_p)_{86^\circ F} R(h_c)$ in Fig. 3-3 (Btu/lb)	(5) $(h_p)_{5^\circ F} R(h_c)$ in Fig. 3-3 (Btu/lb)	(6) $(h_p)_{86^\circ F} R(h_c)$ in Fig. 3-3 (Btu/lb)	(7) $(h_p)_{5^\circ F} R(h_c) = h_c - h_a$ (Btu/lb)	Theoretical Specific Work of Compression ($h_d - h_c$) (Fig. 3-3) (Btu/lb)	(9) $(V_p)_{6^\circ F}$ (cu ft/lb)	(10) Coefficient of Perform- ance, Compared with R.74 for Carnot Cycle	Suction Pressure (psia)	Liquefaction Pressure (psia)	(13) Toxicity in % by Volume (kills most animals in a short time)	(14) Remarks
Ammonia (NH ₃)	271.4	1637.0	613.3	138.9	138.9	474.4	99.6	8.15	4.77	34.27	169.2	0.5 to 0.6 (in 30 min)	Easily detected by burning a sulphur stick
Sulphur dioxide (SO ₂)	314.8	1131.5	153.49	42.12	42.12	141.37	29.08	6.42	4.86	11.81	66.45	Tolerable for 0.3 part in 1000	A slight fire hazard Detected in 3 parts per million
Carbon dioxide (CO ₂)	88.0	1089.9	1.30	138.8	83.3	83.3	55.5	21.40	0.266	2.59	332	1043.0	28 to 30 (in 30 to 60 min)	Decreases human effi- ciency in concentration of 2%
Dichlorodifluoromethane (CCl ₂ F ₂) "Freon-12"	233.0	582.0	78.79	27.72	27.72	51.07	10.85	1.485	4.71	26.51	107.9	10% cannot be tolerated over 5 min	Dangerous with 4% and over
Trichloromonofluoro- methane (CCl ₃ F) "Freon-11"	338.4	635.0	92.88	25.34	25.34	67.54	13.40	12.27	5.02	2.93	18.28		20% by volume not per- manently harmful after exposures of 7 to 8 hr
Dichloromonofluoro- methane (CHCl ₂ F) "Freon-21"	353.3	750	119.97	30.5	30.5	89.41	17.61	9.132	5.01	5.243	31.23		
Trichlorotrifluoroethane (CCl ₃ F - CCl ₂ F ₂) "Freon-113"	417.4	495	79.60	25.93	25.93	53.67	10.92	27.04	4.915	0.9802	7.856		
Methyl chloride (CH ₃ Cl)	289.3	389.3	196.92	46.67	46.67	150.25	30.656	4.471	4.901	21.15	94.70	2.0 to 2.5 (2 hr)	Many good features and no bad ones
Butane (C ₄ H ₁₀)	308.0	529.0	1.11	172.2	48.5	48.5	123.7	26.78	9.98	4.61	8.2	41.6	2.4	A slight fire hazard
Propane (C ₃ H ₈)	204.1	661.5	1.15	172.0	51.0	51.0	121.0	27.50	2.48	4.40	42.1	155.3	2.4	A fire hazard
Methylene chloride (CH ₂ Cl ₂)	421.0	670.0	1.185	163.8	29.2	29.2	134.6	28.61	51.3	4.71	2.39*	20.48*	5.1 to 5.3 (30 min)	A fire hazard
Water vapor (H ₂ O)	705.4	3206	1.32	1061.3	64.0	64.0	1007.3	252.6	11.530	3.99	0.0240	0.6152		

* Inches of mercury.

faulty or if accidents occur, the possibility of explosions and fire is frequently a deciding factor.

(d) **CHEMICAL REACTIONS.** Sulphur dioxide in the presence of water will attack all metals; it does not corrode steel and copper if water is entirely absent. Dichlorodifluoromethane is harmless to aluminium, brass, steel, cast or galvanized iron, tin, zinc, lead or lead solder. Ammonia does not react with cast iron or steel even with large amounts of water present, but copper or copper alloys are corroded badly if the ammonia is not anhydrous.

(e) **ACTION WITH OIL.** The Freon refrigerants, methylene chloride (CH_2Cl_2), isobutane (C_4H_{10}), and methyl chloride, are miscible in the mineral oils used for the lubrication of refrigerating machinery, and they raise the evaporating temperature from 1 to 2 Fahrenheit degrees with 10 per cent by weight of oil present. These refrigerants will not permit oil to collect in the evaporating coils, but an oil fog will be formed which will at times return the lubricant to the oil separator. Oil miscibility has the effect of reducing the density of the condensate.

With halide compressors a portion of the oil will eventually pass along from the compressor to the condenser and then to the evaporator. In the smaller evaporators in which the liquid is fed in series there is no oil lagging in the evaporator because the vapor velocity is sufficient to return the oil "fog" to the crankcase. But in the flooded or spray type of evaporator, because of the slow velocity of the vapor the oil is left behind and becomes so concentrated in the evaporator that the cooling efficiency is impaired; periodic purification, by distilling off the refrigerant and allowing the oil to drain into the crankcase, or some special method of returning the oil, has to be resorted to.

(f) **TOXICITY.** Column 13, Table 8·1, gives some idea of the toxicity hazard of the common refrigerants. Ammonia and sulphur dioxide are very irritating, whereas carbon dioxide, methyl chloride and methylene chloride have no marked warning smell. Methyl chloride, dichloroethylene, and ethyl chloride are considered to be anesthetics. Open flames have the effect of forming hydrochloric acid by the decomposition of the chlorine refrigerants and hydrofluoric acid in fluorine compounds, and both of these groups form traces of phosgene (COCl_2) gas on exposure to open flames.

(g) **LEAKS.** Ammonia leaks are easily detected by means of the sulphur-impregnated cotton cord, and sulphur dioxide is indicated by means of ammonia vapor, a white smoke being formed when leaks are present. For methyl chloride, an indicator called acrolein ($\text{CH}_2\cdot\text{CHCHO}$) is mixed with the refrigerant; leaks of such a mixture are highly irritating to the mucous membranes of the nose and throat. Leaks of dichlorodi-

fluoromethane and the other halides are detected by means of the alcohol lamp (the alcohol-burning blow torch) (Fig. 8·1) without an indicator, especially in small installations; large plants use acetylene instead of alcohol for the purpose. Although the alcohol lamp is very sensitive the loss of dichlorodifluoromethane may at times become appreciable if the construction is poor and if testing is not done regularly.

(h) **COST AND MISCELLANEOUS PROPERTIES.** The cost of the refrigerant most certainly has to be considered for commercial installations, not

only on account of the initial charge but also for the replacements occasioned by leaks.

8·1. Refrigerants in Use.

The different refrigerants appear to be used as follows.

AMMONIA is used for general commercial refrigeration. It has medium pressures, and the number of cubic feet of piston displacement per unit of refrigeration is nominal. Ammonia vapor is dangerous to animal life and is a strong irritant, besides being explosive when mixed with air in volumes of 13 to 27 per cent. The cost of ammonia is low, its physical and chemical character-

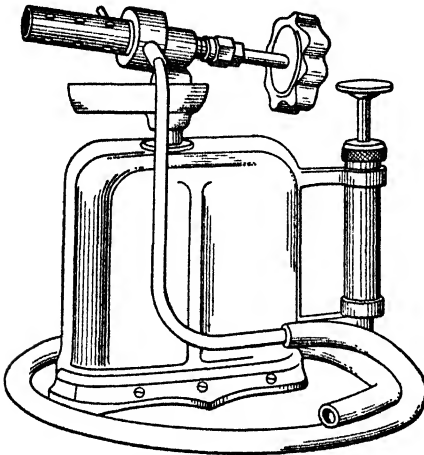


FIG. 8·1. The alcohol-burning blow torch for halides (chlorinated vapors) gives a blue-green flame when halides are present.

istics are well-known, and its dangers are minimized by careful attention to maintenance and repair and by safeguarding against accidents. Ammonia should not be employed in comfort cooling or in installations using direct expansion in hotels, office buildings, theaters, and other places where the public assembles. If ammonia is chosen for such applications it should be self-contained, the indirect system with brine or water being used, with proper isolation of the compressor room.

CARBON DIOXIDE has been used for theater, marine, and certain office refrigeration. Mixed with air it is perfectly safe in nominal amounts unless the exposure is too prolonged or the amount of oxygen is low. It does not have the effect on the blood that carbon monoxide has, and it appears to behave more like an increase in the nitrogen content in the air. It is wasteful of power, as is shown by the small coefficient of performance (see Table 8·1), and its pressures are excessive. Carbon dioxide does not irritate, nor can it support combustion. Leaks are

unusually heavy, and detection of leaks is much more difficult than for ammonia or sulphur dioxide.

SULPHUR DIOXIDE is used in some household machines. It is a non-explosive but it is an irritant. In the liquid phase it has some lubricating properties. The charge of the refrigerant in the usual fractional tonnage machine is small, and it is doubtful whether a leak of sulphur dioxide in the home could possibly be dangerous to human life because of the large volume usually exposed to leaks and the warning due to the irritating nature of the gas. Exposure to the gas increases the tolerance, and sulphur dioxide does not appear to act as a poison.

DICHLORODIFLUOROMETHANE appears to be non-irritating, non-explosive, and non-toxic. If Freon-12 is exposed to open flames, decomposition occurs and some traces of phosgene gas are formed; the same is true of a number of other refrigerants. The Freons in the presence of water do not appear to corrode metals any more than plain water would without the Freons. Dichlorodifluoromethane, with thermodynamic properties somewhat like those of ammonia, but much safer for general use, lends itself to the reciprocating form of compressor, and it is superseding carbon dioxide in comfort-cooling refrigeration. It has found considerable favor with the manufacturers of the household compressor.

METHYL CHLORIDE has been advocated for theater and similar air conditioning work and for apartment and household refrigeration. It is an anesthetic in concentrations of 5 to 10 per cent by volume.

AIR is an ideal refrigerant in many ways, such as its universal presence, its safety in every respect, and its lack of cost, but it is rarely used at the present time because of the bulkiness of the air machine, its small coefficient of performance, and its operating difficulties.

WATER VAPOR is also an ideal refrigerant, particularly for comfort-cooling conditions. It may use ejector nozzles or the centrifugal compressor.

8.2. Operating Cycles. In considering the usual refrigerants it is interesting to study the results in Table 8.2 and to compare the various losses as shown by a reversible cycle in which the thermodynamic properties of the working medium do not enter into the calculations. For example, the throttling process is irreversible, and the compression process, operating on dry compression, is contrary to the ideal cycle because the compression from m to d (Fig. 3.3) and the cooling process df indicate temperatures that are in excess of the upper temperature shown as fa . The result is that certain losses prevail in these two respects, and it may be illuminating to consider the amount of these with different refrigerants, using the temperature limits of 86° and 5° F, respectively.

From Fig. 3.3 the Carnot cycle may be represented by the figure $akcm$, and with dry compression using the throttling process the theoretical refrigerating cycle is $abcd$. The loss in possible refrigeration with the irreversible process ab is shown by the area bb_1k_1k , to the scale of the drawing, and the increase in the power required for compression is a like amount, thereby making the use of the throttle valve a double

TABLE 8.2

CALCULATION OF THROTTLING AND EXCESS WORK OF COMPRESSION
LOSSES FOR THE PRINCIPAL REFRIGERANTS
(5° F Evaporation, 86° F Liquefaction)

Refrigerant	Work of Compression for the Carnot Cycle (area $akma$, Fig. 3.3) (Btu/lb)	Theoretical Work of Compression, $h_d - h_c$ (Fig. 3.3) (Btu/lb)	Loss Due to Throttling Process ab (area kk_1bb_1 , Fig. 3.3) (Btu/lb)	Throttling Loss in Percentage of Work in Carnot Cycle $[(3) \div (1)] \times 100$	Excess Work of Compression ($h_d - h_c$) - area kk_1bb_1 - area $akma$	Excess Work in Per- centage of Work in Carnot Cycle $[(5) \div (1)] \times 100$	Total Losses in Percentage of Carnot Cycle
	(1)	(2)	(3)	(4)	(5)	(6)	(7)
Ammonia (NH ₃)	84.06	99.6	7.7	9.16	7.8	9.28	18.44
Sulphur dioxide (SO ₂)	24.97	29.07	1.84	7.369	2.26	9.05	16.42
Methyl chloride (CH ₂ Cl)	26.62	30.66	2.42	9.09	1.62	6.09	15.18
Dichlorodifluoromethane (CCl ₂ F ₂) Freon-12	9.19	10.85	1.61	17.52	0.05	0.544	18.06
Trichloromonofluoromethane (CCl ₃ F) Freon-11	12.0	13.40	1.30	10.83	0.10	0.83	11.66
Dichloromonofluoromethane (CHCl ₂ F) Freon-21	15.87	17.85	1.61	10.14	0.37	2.33	12.47
Trichlorotrifluoroethane (CCl ₃ F - CClF ₃) Freon-113	9.59	10.92	1.33	13.87	0.00	0.00	13.87

NOTE: In Fig. 3.3 area $akna$ = area kk_1bb_1 = loss due to throttling = $h_a - h_n - T_n(S_a - S_n)$. In column 7, the total losses are greatest for NH₃. The loss, in percentage of the Carnot specific work of compression, is small for F-12 because of its small increase of temperature during compression. For F-113 at the end of compression (d) the vapor is saturated and the loss due to excess work is zero.

loss. The actual compression has to be continued from m to d in order that liquefaction may occur at constant pressure along dfa . The additional work from m to d constitutes a loss. In the Carnot cycle the work of compression per pound of the refrigerant is indicated by the area $akcm$ or is equal to $(s_c - s_a)(t_a - t_c)$. The separate losses are calculated in terms of the work of the ideal cycle.

Refrigerating calculations using the properties of refrigerants are unsatisfactory, especially for the lesser-known refrigerants, because of the meager and only approximate knowledge of the physical properties of their saturated and, even more particularly, of their superheated region. This is especially true of carbon dioxide owing, in part, to its low critical temperature.

TABLE 8-3

COMPARISON OF COEFFICIENTS OF PERFORMANCE AT 70° F
AND 86° F LIQUEFACTION TEMPERATURE

Refrigerant	Enthalpy			Net Refrigeration	Work of Compression		Coefficient of Performance		Liquefaction Pressure	
	(h_f) _{70°}	(h_f) _{86°}	(h_g) _{86°}		At 70°	At 86°	At 70°	At 86°	At 70°	At 86°
Ammonia (NH ₃)	120.5	138.9	613.3	492.8 474.4	78.0	99.6	6.32	4.77	128.8	169.2
Carbon dioxide (CO ₂)	63.7	83.3	138.8	75.1 55.5	17.0	21.40	4.42	2.59	853.4	1043.0
Sulphur dioxide (SO ₂)	36.58	42.12	183.49	148.91 141.37	23.31	29.08	6.30	4.86	49.62	66.45
Dichlorodifluoromethane (CCl ₂ F ₂) Freon-12	23.90	27.72	78.79	54.89 51.07	8.93	10.85	6.15	4.71	84.82	107.9

Tables 8-1 and 8-2 are very instructive in the comparison of refrigerants. In Table 8-1 are given the critical temperatures and the corresponding pressures, the specific volumes, and other physical properties required for the calculation of the coefficients of performance at 5° and

TABLE 8-4

COMPARISON OF CARBONIC REFRIGERATING CYCLES

Cycle	Net Refrigeration	Work of Compression	Coefficient of Performance	Pounds of CO ₂ per Ton of Refrigeration per Minute
<i>abcd</i>	138.9 - 97.0 = 41.9	163.5 - 138.9 = 24.6	$\frac{41.9}{24.6} = 1.70$	$\frac{200}{41.9} = 4.77$
<i>fgcf</i>	138.9 - 79.0 = 59.9	167.5 - 138.9 = 28.6	$\frac{59.9}{28.6} = 2.095$	$\frac{200}{59.9} = 3.34$
<i>fgcdahf</i>	138.9 - 79.0 = 59.9	24.6 + 2.0 = 26.6	$\frac{59.9}{26.6} = 2.25$	$\frac{200}{59.9} = 3.34$

Coefficient of performance of the Carnot cycle = $\frac{461}{94} = 4.85$.

86° F. Table 8-2 gives values for the two losses, namely that due to throttling and that due to compression, with dry suction vapor, to a pressure above that corresponding to the temperature of liquefaction. For example, Freon-12 has a low value for the compression loss but a relatively high throttling loss.

REFRIGERANT PROPERTIES

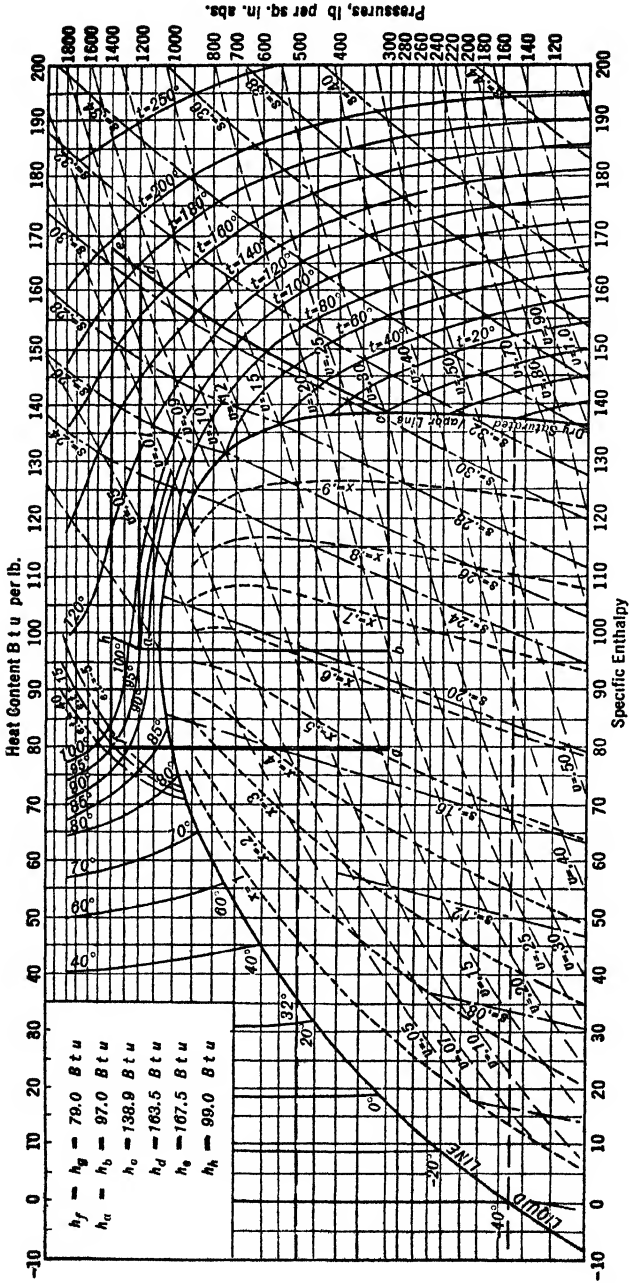


FIG. 8-2. The Mollier diagram for carbon dioxide.

TABLE 8-5a
LIQUID AMMONIA (NH₃)

Temperature °F <i>t</i>	(At Saturation)					Latent Heat of Pressure Variation Btu/lb lb/in. ²	Compressi- bility % change in <i>v</i> lb/in. ³	Temperature °F <i>t</i>
	Pressure (abs) lb/in. ² <i>p</i>	Density lb/ft ³	Specific Heat Btu/lb °F	Total Heat Above -40° Btu/lb <i>h_f</i>	Latent Heat Btu/lb <i>L</i>			
	Triple point	0.88	45.83					
-100	1.24	45.52	1.040				-100	
-95	1.52	45.32	1.042				-95	
-90	1.86	45.12	1.043				-90	
-85	2.27	44.92	1.045				-85	
-80	2.74	44.72	1.046				-80	
-75	3.29	44.52	1.048				-75	
-70	3.94	44.32	1.050				-70	
-65	4.69	44.11	1.052				-65	
-60	5.55	43.91	1.054	-21.18	610.8	-0.0016	0.00044	-60
-55	6.54	43.70	1.056	-15.90	607.5	-0.0016	0.00045	-55
-50	7.67	43.49	1.058	-10.61	604.3	-0.0017	0.00046	-50
-45	8.95	43.28	1.060	- 5.31	600.9	-0.0017	0.00047	-45
-40	10.41	43.08	1.062	0.00	597.6	-0.0018	0.00048	-40
-35	12.05	42.86	1.064	+ 5.32	594.2	-0.0018	0.00050	-35
-30	13.90	42.65	1.066	10.66	590.7	-0.0019	0.00051	-30
-25	15.98	42.44	1.068	16.00	587.2	-0.0019	0.00052	-25
-20	18.30	42.22	1.070	21.36	583.6	-0.0020	0.00054	-20
-15	20.88	42.00	1.073	26.73	580.0	-0.0020	0.00055	-15
-10	23.74	41.78	1.075	32.11	576.4	-0.0021	0.00057	-10
- 5	26.92	41.56	1.078	37.51	572.6	-0.0022	0.00058	- 5
0	30.42	41.34	1.080	42.92	568.9	-0.0022	0.00060	0
5	34.27	41.11	1.083	48.35	565.0	-0.0023	0.00062	5
10	38.51	40.89	1.085	53.79	561.1	-0.0024	0.00064	10
15	43.14	40.66	1.088	59.24	557.1	-0.0025	0.00066	15
20	48.21	40.43	1.091	64.71	553.1	-0.0025	0.00068	20
25	53.73	40.20	1.094	70.20	548.9	-0.0026	0.00070	25
30	59.74	39.96	1.097	75.71	544.8	-0.0027	0.00073	30
35	66.26	39.72	1.100	81.23	540.5	-0.0028	0.00075	35
40	73.32	39.49	1.104	86.77	536.2	-0.0029	0.00078	40
45	80.96	39.24	1.108	92.34	531.8	-0.0030	0.00081	45
50	89.19	39.00	1.112	97.93	527.3	-0.0031	0.00084	50
55	98.06	38.75	1.116	103.54	522.8	-0.0032	0.00088	55
60	107.6	38.50	1.120	109.18	518.1	-0.0033	0.00091	60
65	117.8	38.25	1.125	114.85	513.4	-0.0034	0.00095	65
70	128.8	38.00	1.129	120.54	508.6	-0.0035	0.00100	70
75	140.5	37.74	1.133	126.65	503.7	-0.0037	0.00104	75
80	153.0	37.48	1.138	131.99	498.7	-0.0038	0.00109	80
85	166.4	37.21	1.142	137.75	493.6	-0.0040	0.00114	85
86	169.2	37.16	1.143	138.90	492.6	0.0040	0.00115	86
90	180.6	36.95	1.147	143.54	488.5	-0.0041	0.00120	90
95	195.8	36.67	1.151	149.36	483.2	-0.0043	0.00126	95
100	211.9	36.40	1.156	155.21	477.8	-0.0045	0.00133	100
105	228.9	36.12	1.162	161.09	472.3	-0.0047	0.00141	105
110	247.0	35.84	1.168	167.01	466.7	-0.0049	0.00149	110
115	266.2	35.55	1.176	172.97	460.9	-0.0051	0.00158	115
120	286.4	35.26	1.183	178.98	455.0	-0.0053	0.00167	120
125	307.8	34.96	1.189	185.0	449.0			125
130	330.3	34.66	1.197	191.0	443.0			130
135	354.1	34.35	1.205	197.0	436.0			135
140	379.1	34.04	1.213	203.0	430.0			140
145	405.5	33.72	1.222	210.0	423.0			145
150	433.2	33.39	1.23	216.0	416.0			150
155	462.3	33.06	1.24	222.0	409.0			155
160	492.8	32.72	1.25	229.0	401.0			160
165	524.8	32.37	1.26	235.0	394.0			165
170	558.4	32.01	1.27	241.0	386.0			170
Critical	1,657.0	14.6	∞	438.0	0	-∞	∞	271.4

TABLE 8-5b
AMMONIA, SATURATED, LOW TEMPERATURES (NH₃)

Temperature °F <i>t</i>	Pressure lb/in. ² abs <i>p</i>	Volume ft ³ /lb		Total Heat Btu/lb		Entropy	
		Vapor <i>v_g</i>	Liquid <i>v_f</i>	Liquid <i>h_f</i>	Vapor <i>h_g</i>	Liquid <i>s_f</i>	Vapor <i>s_g</i>
-105	0.995	223.14	0.02188	-68.2	568.6	-0.1717	1.6211
-104	1.04	214.23	.02190	-67.1	569.1	-.1689	1.6174
-103	1.09	205.75	.02191	-66.1	569.6	-.1662	1.6136
-102	1.14	197.70	.02193	-65.1	570.1	-.1635	1.6100
-101	1.19	190.08	.02195	-64.1	570.6	-.1606	1.6062
-100	1.24	182.90	0.02197	-63.0	571.0	-0.1579	1.6025
-99	1.29	175.42	.02199	-61.9	571.4	-.1552	1.5988
-98	1.35	168.48	.02201	-60.9	571.9	-.1525	1.5952
-97	1.41	161.98	.02203	-59.9	572.3	-.1498	1.5915
-96	1.47	155.92	.02205	-58.9	572.7	-.1471	1.5878
-95	1.52	150.30	0.02207	-57.8	573.2	-0.1444	1.5842
-94	1.59	144.68	.02209	-56.7	573.6	-.1417	1.5806
-93	1.66	139.27	.02211	-55.7	574.1	-.1390	1.5771
-92	1.73	134.06	.02213	-54.7	574.5	-.1363	1.5736
-91	1.79	129.06	.02215	-53.7	575.0	-.1336	1.5702
-90	1.86	124.28	0.02217	-52.6	575.4	-0.1309	1.5667
-89	1.94	119.73	.02219	-51.5	575.9	-.1281	1.5634
-88	2.02	115.37	.02221	-50.5	576.3	-.1254	1.5600
-87	2.11	111.21	.02223	-49.5	576.7	-.1227	1.5566
-86	2.18	107.29	.02224	-48.5	577.2	-.1199	1.5533
-85	2.27	103.63	0.02226	-47.4	577.6	-0.1171	1.5499
-84	2.36	99.87	.02228	-46.3	578.0	-.1144	1.5466
-83	2.46	96.28	.02230	-45.3	578.4	-.1117	1.5433
-82	2.55	92.86	.02232	-44.3	578.9	-.1090	1.5401
-81	2.65	89.61	.02234	-43.3	579.7	-.1063	1.5368
-80	2.74	86.54	0.02236	-42.2	580.1	-0.1036	1.5336
-79	2.85	83.50	.02238	-41.1	580.5	-.1009	1.5304
-78	2.96	80.61	.02240	-40.0	581.0	-.0983	1.5273
-77	3.07	77.85	.02242	-39.0	581.4	-.0956	1.5241
-76	3.19	75.25	.02244	-38.0	581.8	-.0930	1.5210
-75	3.29	72.80	0.02246	-36.9	582.2	-0.0903	1.5178
-74	3.43	70.35	.02248	-35.8	582.7	-.0877	1.4148
-73	3.56	68.01	.02250	-34.8	583.1	-.0850	1.5117
-72	3.69	65.78	.02252	-33.8	583.6	-.0824	1.5087
-71	3.82	63.66	.02254	-32.8	584.0	-.0797	1.5056
-70	3.94	61.65	0.02256	-31.7	584.4	-0.0771	1.5026
-69	4.09	59.64	.02258	-30.6	584.8	-.0745	1.4997
-68	4.24	57.64	.02261	-29.6	585.3	-.0719	1.4969
-67	4.39	55.64	.02263	-28.6	585.8	-.0694	1.4940
-66	4.54	54.01	.02265	-27.5	586.2	-.0668	1.4911
-65	4.69	52.24	0.02267	-26.4	586.6	-0.0642	1.4883
-64	4.86	50.70	.02269	-25.3	587.0	-.0616	1.4856
-63	5.03	49.26	.02271	-24.3	587.5	-.0590	1.4829
-62	5.20	47.74	.02273	-23.3	588.0	-.0565	1.4802
-61	5.38	46.23	.02276	-22.3	588.4	-.0539	1.4774
-60	5.55	44.73	0.02278	-21.2	589.6	-0.0517	1.4769

TABLE 8-6
SATURATED AMMONIA (NH₃)

Temperature °F <i>t</i>	Pressure lb/in. ² <i>p</i>	Volume		Total Heat above -40°			Entropy from -40°	
		Liquid ft ³ /lb <i>v_f</i>	Vapor ft ³ /lb <i>v_g</i>	Liquid Btu/lb <i>h_f</i>	Latent Btu/lb <i>L</i>	Vapor Btu/lb <i>h_g</i>	Liquid <i>s_f</i>	Vapor <i>s_g</i>
-60	5.55	0.02278	44.73	-21.2	610.8	589.6	-0.0517	1.4769
-59	5.74	.02280	43.37	-20.1	610.1	590.0	-.0490	1.4741
-58	5.93	.02282	42.05	-19.1	609.5	590.4	-.0464	1.4713
-57	6.13	.02284	40.79	-18.0	608.8	590.8	-.0438	1.4686
-56	6.33	.02286	39.56	-17.0	608.2	591.2	-.0412	1.4658
-55	6.54	0.02288	38.38	-15.9	607.5	591.6	-0.0386	1.4631
-54	6.75	.02291	37.24	-14.8	606.9	592.1	-.0360	1.4604
-53	6.97	.02293	36.15	-13.8	606.2	592.4	-.0334	1.4577
-52	7.20	.02295	35.09	-12.7	605.6	592.9	-.0307	1.4551
-51	7.43	.02297	34.06	-11.7	604.9	593.2	-.0281	1.4524
-50	7.67	0.02299	33.08	-10.6	604.3	593.7	-0.0256	1.4497
-49	7.91	.02301	32.12	-9.6	603.6	594.0	-.0230	1.4471
-48	8.16	.02304	31.20	-8.6	602.9	594.4	-.0204	1.4445
-47	8.42	.02306	30.31	-7.4	602.3	594.9	-.0179	1.4419
-46	8.68	.02308	29.45	-6.4	601.6	595.2	-.0153	1.4393
-45	8.95	0.02310	28.62	-5.3	600.9	595.6	-0.0127	1.4368
-44	9.23	.02313	27.82	-4.3	600.3	596.0	-.0102	1.4342
-43	9.51	.02315	27.04	-3.2	599.6	596.4	-.0076	1.4317
-42	9.81	.02317	26.29	-2.1	598.9	596.8	-.0051	1.4292
-41	10.10	.02319	25.56	-1.1	598.3	597.2	-.0025	1.4267
-40	10.41	0.02322	24.86	0.0	597.6	597.6	0.0000	1.4242
-39	10.72	.02324	24.18	1.1	596.9	598.0	.0025	1.4217
-38	11.04	.02326	23.53	2.1	596.2	598.3	.0051	1.4193
-37	11.37	.02328	22.89	3.2	595.5	598.7	.0076	1.4169
-36	11.71	.02331	22.27	4.3	594.8	599.1	.0101	1.4144
-35	12.05	0.02333	21.68	5.3	594.2	599.5	0.0126	1.4120
-34	12.41	.02335	21.10	6.4	593.5	599.9	.0151	1.4096
-33	12.77	.02338	20.54	7.4	592.8	600.2	.0176	1.4072
-32	13.14	.02340	20.00	8.5	592.1	600.6	.0201	1.4048
-31	13.52	.02342	19.48	9.6	591.4	601.0	.0226	1.4025
-30	13.90	0.02345	18.97	10.7	590.7	601.4	0.0250	1.4001
-29	14.30	.02347	18.48	11.7	590.0	601.7	.0275	1.3978
-28	14.71	.02349	18.00	12.8	589.3	602.1	.0300	1.3955
-27	15.12	.02352	17.54	13.9	588.6	602.5	.0325	1.3932
-26	15.55	.02354	17.09	14.9	587.9	602.8	.0350	1.3909
-25	15.98	0.02357	16.66	16.0	587.2	603.2	0.0374	1.3886
-24	16.42	.02359	16.24	17.1	586.5	603.6	.0399	1.3863
-23	16.88	.02361	15.83	18.1	585.8	603.9	.0423	1.3840
-22	17.34	.02364	15.43	19.2	585.1	604.3	.0448	1.3818
-21	17.81	.02366	15.05	20.3	584.3	604.6	.0472	1.3796
-20	18.30	0.02369	14.68	21.4	583.6	605.0	0.0497	1.3774
-19	18.79	.02371	14.32	22.4	582.9	605.3	.0521	1.3752
-18	19.30	.02374	13.97	23.5	582.2	605.7	.0545	1.3729
-17	19.81	.02376	13.62	24.6	581.5	606.1	.0570	1.3708
-16	20.34	.02378	13.29	25.6	580.8	606.4	.0594	1.3686
-15	20.88	0.02381	12.97	26.7	580.0	606.7	0.0618	1.3664
-14	21.43	.02383	12.66	27.8	579.3	607.1	.0642	1.3643
-13	21.99	.02386	12.36	28.9	578.6	607.5	.0666	1.3621
-12	22.56	.02388	12.06	30.0	577.8	607.8	.0690	1.3600
-11	23.15	.02391	11.78	31.0	577.1	608.1	.0714	1.3579
-10	23.74	0.02393	11.50	32.1	576.4	608.5	0.0738	1.3558
-9	24.35	.02396	11.23	33.2	575.6	608.8	.0762	1.3537
-8	24.97	.02399	10.97	34.3	574.9	609.2	.0786	1.3516
-7	25.61	.02401	10.71	35.4	574.1	609.5	.0809	1.3495
-6	26.26	.02404	10.47	36.4	573.4	609.8	.0833	1.3474
-5	26.92	0.02406	10.23	37.5	572.6	610.1	0.0857	1.3454
-4	27.59	.02409	9.991	38.6	571.9	610.5	.0880	1.3433
-3	28.28	.02411	9.763	39.7	571.1	610.8	.0904	1.3413
-2	28.98	.02414	9.541	40.7	570.4	611.1	.0928	1.3393
-1	29.69	.02417	9.326	41.8	569.6	611.4	.0951	1.3372

TABLE 8-6 (Continued)
SATURATED AMMONIA

Temperature °F <i>t</i>	Pressure	Volume		Total Heat above -40°			Entropy from -40°	
		Abs lb/in. ² <i>p</i>	Liquid ft ³ /lb <i>v_f</i>	Vapor ft ³ /lb <i>v_g</i>	Liquid Btu/lb <i>h_f</i>	Latent Btu/lb <i>L</i>	Vapor Btu/lb <i>h_g</i>	Liquid <i>s_f</i>
0	30.42	0.02419	9.116	42.9	568.9	611.8	0.0975	1.3352
1	31.16	.02422	8.912	44.0	568.1	612.1	.0998	1.3332
2	31.92	.02424	8.714	45.1	567.3	612.4	.1022	1.3312
3	32.69	.02427	8.521	46.2	566.5	612.7	.1045	1.3292
4	33.47	.02430	8.333	47.2	565.8	613.0	.1069	1.3273
5	34.27	0.02432	8.150	48.3	565.0	613.3	0.1092	1.3253
6	35.09	.02435	7.971	49.4	564.2	613.6	.1115	1.3234
7	35.92	.02438	7.798	50.5	563.4	613.9	.1138	1.3214
8	36.77	.02440	7.629	51.6	562.7	614.3	.1162	1.3195
9	37.63	.02443	7.464	52.7	561.9	614.6	.1185	1.3176
10	38.51	0.02446	7.304	53.8	561.1	614.9	0.1208	1.3157
11	39.40	.02449	7.148	54.9	560.3	615.2	.1231	1.3137
12	40.31	.02451	6.996	56.0	559.5	615.5	.1254	1.3118
13	41.24	.02454	6.847	57.1	558.7	615.8	.1277	1.3099
14	42.18	.02457	6.703	58.2	557.9	616.1	.1300	1.3081
15	43.14	0.02460	6.562	59.2	557.1	616.3	0.1323	1.3062
16	44.12	.02462	6.425	60.3	556.3	616.6	.1346	1.3043
17	45.12	.02465	6.291	61.4	555.5	616.9	.1369	1.3025
18	46.13	.02468	6.161	62.5	554.7	617.2	.1392	1.3006
19	47.16	.02471	6.034	63.6	553.9	617.5	.1415	1.2988
20	48.21	0.02474	5.910	64.7	553.1	617.8	0.1437	1.2969
21	49.28	.02476	5.789	65.8	552.2	618.0	.1460	1.2951
22	50.36	.02479	5.671	66.9	551.4	618.3	.1483	1.2933
23	51.47	.02482	5.556	68.0	550.6	618.6	.1505	1.2915
24	52.59	.02485	5.443	69.1	549.8	618.9	.1528	1.2897
25	53.73	0.02488	5.334	70.2	548.9	619.1	0.1551	1.2879
26	54.90	.02491	5.227	71.3	548.1	619.4	.1573	1.2861
27	56.08	.02494	5.123	72.4	547.3	619.7	.1596	1.2843
28	57.28	.02497	5.021	73.5	546.4	619.9	.1618	1.2825
29	58.50	.02500	4.922	74.6	545.6	620.2	.1641	1.2808
30	59.74	0.02503	4.825	75.7	544.8	620.5	0.1663	1.2790
31	61.00	.02505	4.730	76.8	543.9	620.7	.1686	1.2773
32	62.29	.02508	4.637	77.9	543.1	621.0	.1708	1.2755
33	63.59	.02511	4.547	79.0	542.2	621.2	.1730	1.2738
34	64.91	.02514	4.459	80.1	541.4	621.5	.1753	1.2721
35	66.26	0.02518	4.373	82.1	540.5	621.7	0.1775	1.2704
36	67.63	.02521	4.289	82.3	539.7	622.0	.1797	1.2686
37	69.02	.02524	4.207	83.4	538.8	622.2	.1819	1.2669
38	70.43	.02527	4.126	84.6	537.9	622.5	.1841	1.2652
39	71.87	.02530	4.048	85.7	537.0	622.7	.1863	1.2635
40	73.32	.02533	3.971	86.8	536.2	623.0	0.1885	1.2618
41	74.80	.02536	3.897	87.9	535.3	623.2	.1908	1.2602
42	76.31	.02539	3.823	89.0	534.4	623.4	.1930	1.2585
43	77.83	.02542	3.752	90.1	533.6	623.7	.1952	1.2568
44	79.38	.02545	3.682	91.2	532.7	623.9	.1974	1.2552
45	80.96	0.02548	3.614	92.3	531.8	624.1	0.1996	1.2535
46	82.55	.02551	3.547	93.5	530.9	624.4	.2018	1.2519
47	84.18	.02555	3.481	94.6	530.0	624.6	.2040	1.2502
48	85.82	.02558	3.418	95.7	529.1	624.8	.2062	1.2486
49	87.49	.02561	3.356	96.8	528.2	625.0	.2083	1.2469
50	89.19	0.02564	3.294	97.9	527.3	625.2	0.2105	1.2453
51	90.91	.02568	3.234	99.1	526.4	625.5	.2127	1.2437
52	92.66	.02571	3.176	100.2	525.5	625.7	.2149	1.2421
53	94.43	.02574	3.119	101.3	524.6	625.9	.2171	1.2405
54	96.23	.02577	3.063	102.4	523.7	626.1	.2192	1.2389
55	98.06	0.02581	3.008	103.5	522.8	626.3	0.2214	1.2373
56	99.91	.02584	2.954	104.7	521.8	626.5	.2236	1.2357
57	101.8	.02587	2.902	105.8	520.9	626.7	.2257	1.2341
58	103.7	.02490	2.851	106.9	520.0	626.9	.2279	1.2325
59	105.6	.02594	2.800	108.1	519.0	627.1	.2301	1.2310

TABLE 8-6 (Continued)

SATURATED AMMONIA

Temperature °F	Pressure lb/in. ² <i>p</i>	Volume		Total Heat above -40°			Entropy from -40°	
		Liquid ft ³ /lb <i>v_f</i>	Vapor ft ³ /lb <i>v_g</i>	Liquid Btu/lb <i>h_f</i>	Latent Btu/lb <i>L</i>	Vapor Btu/lb <i>h_g</i>	Liquid <i>s_f</i>	Vapor <i>s_g</i>
60	107.6	0.02597	2 751	109.2	518.1	627.3	0.2322	1.2294
61	109.6	.02601	2 703	110.3	517.2	627.5	.2344	1.2278
62	111.6	.02604	2 656	111.5	516.2	627.7	.2365	1.2262
63	113.6	.02608	2 610	112.6	515.3	627.9	.2387	1.2247
64	115.7	.02611	2 565	113.7	514.3	628.0	.2408	1.2231
65	117.8	0.02614	2 520	114.8	513.4	628.2	0.2430	1.2216
66	120.0	.02618	2 477	116.0	512.4	628.4	.2451	1.2201
67	122.1	.02621	2 435	117.1	511.5	628.6	.2473	1.2186
68	124.3	.02625	2 393	118.3	510.5	628.8	.2494	1.2170
69	126.5	.02628	2 352	119.4	509.5	628.9	.2515	1.2155
70	128.8	0.02632	2 312	120.5	508.6	629.1	0.2537	1.2140
71	131.1	.02635	2 273	121.7	507.6	629.3	.2558	1.2125
72	133.4	.02639	2 236	122.8	506.6	629.4	.2579	1.2110
73	135.7	.02643	2 197	124.0	505.6	629.6	.2601	1.2095
74	138.1	.02646	2 161	125.1	504.7	629.8	.2622	1.2080
75	140.5	0.02650	2 125	126.2	503.7	629.9	0.2643	1.2065
76	143.0	.02653	2 089	127.4	502.7	630.1	.2664	1.2050
77	145.4	.02657	2 055	128.5	501.7	630.2	.2685	1.2035
78	147.9	.02661	2 021	129.7	500.7	630.4	.2706	1.2020
79	150.5	.02664	1 988	130.8	499.7	630.5	.2728	1.2006
80	153.0	0.02668	1 955	132.0	498.7	630.7	0.2749	1.1991
81	155.6	.02672	1 923	133.1	497.7	630.8	.2769	1.1976
82	158.3	.02675	1 892	134.3	496.7	631.0	.2791	1.1962
83	161.0	.02679	1 861	135.4	495.7	631.1	.2812	1.1947
84	163.7	.02684	1 831	136.6	494.7	631.3	.2833	1.1933
85	166.4	0.02687	1 801	137.8	493.6	631.4	0.2854	1.1918
86	169.2	.02691	1 772	138.9	492.6	631.5	.2875	1.1904
87	172.0	.02695	1 744	140.1	491.6	631.7	.2895	1.1889
88	174.8	.02699	1 716	141.2	490.6	631.8	.2917	1.1875
89	177.7	.02703	1 688	142.4	489.5	631.9	.2937	1.1860
90	180.6	0.02707	1 661	143.5	488.5	632.0	0.2958	1.1846
91	183.6	.02711	1 635	144.7	487.4	632.1	.2979	1.1832
92	186.6	.02715	1 609	145.8	486.4	632.2	.3000	1.1818
93	189.6	.02719	1 584	147.0	485.3	632.3	.3021	1.1804
94	192.7	.02723	1 559	148.2	484.3	632.5	.3041	1.1789
95	195.8	0.02727	1 534	149.4	483.2	632.6	0.3062	1.1775
96	198.9	.02731	1 510	150.5	482.1	632.6	.3083	1.1761
97	202.1	.02735	1 487	151.7	481.1	632.8	.3104	1.1747
98	205.3	.02739	1 464	152.9	480.0	632.9	.3125	1.1733
99	208.6	.02743	1 441	154.0	478.9	632.9	.3145	1.1719
100	211.9	0.02747	1 419	155.2	477.8	633.0	0.3166	1.1705
101	215.2	.02752	1 397	156.4	476.7	633.1	.3187	1.1691
102	218.6	.02756	1 375	157.6	475.6	633.2	.3207	1.1677
103	222.0	.02760	1 354	158.7	474.6	633.3	.3228	1.1663
104	225.4	.02764	1 334	159.9	473.5	633.4	.3248	1.1649
105	228.9	0.02769	1 313	161.1	472.3	633.4	0.3269	1.1635
106	232.5	.02773	1 293	162.3	471.2	633.5	.3289	1.1621
107	236.0	.02777	1 274	163.5	470.1	633.6	.3310	1.1607
108	239.7	.02782	1 254	164.6	469.0	633.6	.3330	1.1593
109	243.3	.02786	1 235	165.8	467.9	633.7	.3351	1.1580
110	247.0	0.02790	1 217	167.0	466.7	633.7	0.3372	1.1566
112	254.5	.02799	1 180	169.4	464.4	633.8	.3413	1.1538
114	262.2	.02808	1 145	171.8	462.1	633.9	.3453	1.1510
116	270.1	.02817	1 112	174.2	459.8	634.0	.3495	1.1483
118	278.2	.02827	1 079	176.6	457.4	634.0	.3535	1.1455
120	286.4	0.02836	1 047	179.0	455.0	634.0	0.3576	1.1427
122	294.8	.02846	1 017	181.4	452.6	634.0	.3618	1.1400
124	303.4	.02855	0 987	183.9	450.1	634.0	.3659	1.1372
126	312.2	.02865	.958	186.3	447.6	633.9	.3700	1.1344
128	321.2	.02875	.931	188.8	445.1	633.9	.3741	1.1316

TABLE 8-7
SUPERHEATED AMMONIA

Temperature, °F.	5 lb./in. ² (-63.11° F)			10 lb./in. ² (-41.34° F)			15 lb./in. ² (-27.29° F)			20 lb./in. ² (-16.64° F)		
	Volume ft ³ /lb	Total Heat Btu/lb	Entropy Btu/lb °F	Volume ft ³ /lb	Total Heat Btu/lb	Entropy Btu/lb °F	Volume ft ³ /lb	Total Heat Btu/lb	Entropy Btu/lb °F	Volume ft ³ /lb	Total Heat Btu/lb	Entropy Btu/lb °F
(at sat'n)	<i>v</i>	<i>h</i>	<i>s</i>	<i>v</i>	<i>h</i>	<i>s</i>	<i>v</i>	<i>h</i>	<i>s</i>	<i>v</i>	<i>h</i>	<i>s</i>
-50	(49.31)	(698.5)	(1.5025)	(25.81)	(597.1)	(1.4976)	(17.57)	(602.4)	(1.5038)	(13.50)	(606.2)	(1.5700)
-40	51.05	595.2	1.5025									
-30	52.36	600.3	1.5149	25.90	597.8	1.4293	18.01	606.4	1.4031	13.74	610.0	1.3784
-20	53.67	605.4	1.5269	26.58	603.2	1.4420	18.37	611.9	1.4154	14.09	615.5	1.3907
-10	54.97	610.4	1.5385	27.26	608.5	1.4542	19.52	617.2	1.4272	14.44	621.0	1.4025
0	56.26	615.4	1.5498	27.92	613.7	1.4659	19.82	622.5	1.4386	14.78	626.4	1.4138
10	57.55	620.4	1.5608	28.58	618.9	1.4773	20.26	627.8	1.4497	15.11	631.7	1.4248
20	58.84	625.4	1.5716	29.24	624.0	1.4884	20.70	633.0	1.4604	15.45	637.0	1.4356
30	60.12	630.4	1.5821	29.90	629.1	1.4992	21.14	638.2	1.4709	15.78	642.3	1.4460
40	61.41	635.4	1.5925	30.55	634.2	1.5097	21.58	643.4	1.4812	16.12	647.5	1.4562
50	62.69	640.4	1.6026	31.20	639.3	1.5200	22.02	648.5	1.4912	16.45	652.8	1.4662
60	63.96	645.5	1.6125	31.85	644.4	1.5301	22.46	653.7	1.5011	16.78	658.0	1.4760
70	65.24	650.5	1.6223	32.49	649.5	1.5400	22.90	658.9	1.5108	17.10	663.2	1.4856
80	66.51	655.5	1.6319	33.14	654.6	1.5497	23.34	664.0	1.5203	17.43	668.5	1.4950
90	67.78	660.6	1.6413	33.78	659.7	1.5593	23.78	669.2	1.5298	17.76	673.7	1.5042
100	69.06	665.6	1.6506	34.42	664.8	1.5687	24.22	674.4	1.5388	18.08	678.9	1.5133
110	70.33	670.7	1.6598	35.07	670.0	1.5779	24.66	679.6	1.5478	18.41	684.2	1.5223
120	71.60	675.8	1.6689	35.71	675.1	1.5870	25.10	684.8	1.5567	18.73	689.4	1.5312
130	72.87	680.9	1.6778	36.35	680.3	1.5960	25.53	690.0	1.5655	19.07	694.7	1.5399
140	74.14	686.1	1.6865	36.99	685.4	1.6049	25.97	695.3	1.5742	19.37	700.0	1.5485
150	75.41	691.2	1.6952	37.62	690.6	1.6136	26.41	700.5	1.5827	19.70	705.3	1.5569
160	76.68	696.4	1.7038	38.26	695.8	1.6222	26.85	705.8	1.5911	20.02	710.6	1.5653
170	77.95	701.6	1.7122	38.90	701.1	1.6307	27.29	711.1	1.5995	20.34	715.9	1.5738
180	79.21	706.8	1.7206	39.54	706.3	1.6391	27.72	716.4	1.6077	20.66	721.2	1.5817
190	80.48	712.1	1.7289	40.17	711.6	1.6474	28.16	721.7	1.6158	20.98	726.6	1.5898
200				40.81	716.9	1.6556	28.59	727.0	1.6239	21.30	732.0	1.5979
210							29.02	732.4	1.6318	21.62	737.4	1.6059
220							29.44	737.8	1.6397	21.94	742.8	1.6138
230							29.86	743.2	1.6475	22.26	748.3	1.6212
240							30.29	748.6	1.6552			
250							29.71	748.6	1.6552			

TABLE 8-7 (Continued)
SUPERHEATED AMMONIA

Temperature, °F.	25 lb./in. ² (-7.36° F)			30 lb./in. ² (-0.57° F)			35 lb./in. ² (+5.89° F)			40 lb./in. ² (+1.66° F)		
	Volume ft ³ /lb	Total Heat Btu/lb	Entropy Btu/lb °F	Volume ft ³ /lb	Total Heat Btu/lb	Entropy Btu/lb °F	Volume ft ³ /lb	Total Heat Btu/lb	Entropy Btu/lb °F	Volume ft ³ /lb	Total Heat Btu/lb	Entropy Btu/lb °F
(at sat'n)	<i>v</i>	<i>h</i>	<i>s</i>	<i>v</i>	<i>h</i>	<i>s</i>	<i>v</i>	<i>h</i>	<i>s</i>	<i>v</i>	<i>h</i>	<i>s</i>
0	(10.96)	(602.1)	(1.5515)	(9.256)	(611.6)	(1.5264)	(7.921)	(615.6)	(1.5256)	(7.047)	(616.4)	(1.5125)
10	11.19	613.8	1.3616	9.250	611.9	1.3371	8.078	616.6	1.3289	7.203	620.4	1.3231
20	11.47	619.4	1.3738	9.492	617.8	1.3497	8.287	622.0	1.3413	7.387	626.3	1.3353
30	11.75	625.5	1.3855	9.731	623.5	1.3618	8.493	627.7	1.3532	7.568	632.1	1.3470
40	12.03	630.4	1.3967	9.966	629.1	1.3733	8.695	633.4	1.3646	7.746	637.8	1.3583
50	12.30	635.8	1.4077	10.20	634.6	1.3845	8.895	638.9	1.3756	7.922	643.4	1.3692
60	12.57	641.2	1.4183	10.43	640.1	1.3953	9.093	644.4	1.3863	8.096	648.9	1.3797
70	12.84	646.5	1.4287	10.65	645.5	1.4059	9.289	649.9	1.3967	8.268	654.4	1.3900
80	13.11	651.8	1.4388	10.88	650.9	1.4161	9.484	655.3	1.4069	8.439	659.9	1.4008
90	13.37	657.1	1.4487	11.10	656.2	1.4261	9.677	660.7	1.4168	8.609	665.3	1.4119
100	13.64	662.4	1.4584	11.33	661.6	1.4359	9.869	666.1	1.4265	8.777	670.7	1.4238
110	13.90	667.7	1.4679	11.55	666.9	1.4456	10.06	671.5	1.4360	8.945	676.1	1.4381
120	14.17	673.0	1.4772	11.77	672.2	1.4550	10.25	676.8	1.4453	9.112	681.5	1.4471
130	14.43	678.2	1.4864	11.99	677.5	1.4642	10.44	682.2	1.4545	9.278	686.9	1.4561
140	14.69	683.5	1.4954	12.21	682.9	1.4733	10.63	687.6	1.4635	9.444	692.3	1.4648
150	14.95	688.8	1.5043	12.43	688.2	1.4823	10.82	692.9	1.4724	9.609	697.7	1.4735
160	15.21	694.1	1.5131	12.65	693.5	1.4911	11.00	698.3	1.4811	9.774	703.1	1.4820
170	15.47	699.4	1.5217	12.87	698.8	1.5003	11.19	703.7	1.4897	9.938	708.5	1.4904
180	15.73	704.7	1.5303	13.08	704.2	1.5093	11.38	709.1	1.4982	10.10	714.0	1.4987
190	15.99	710.1	1.5387	13.30	709.6	1.5168	11.56	714.9	1.5066	10.27	719.4	1.5069
200	16.25	715.4	1.5470	13.52	714.9	1.5251	11.75	719.9	1.5148	10.45	724.9	1.5150
210	16.50	720.8	1.5552	13.73	720.3	1.5334	11.94	725.3	1.5230	10.63	730.3	1.5230
220	16.76	726.2	1.5633	13.94	725.7	1.5415	12.13	730.7	1.5311	10.82	735.8	1.5309
230	17.02	731.6	1.5713	14.16	731.3	1.5495	12.31	736.2	1.5390	11.01	741.3	1.5389
240	17.27	737.0	1.5792	14.38	736.6	1.5575	12.49	741.7	1.5469	11.20	746.8	1.5468
250	17.53	742.5	1.5870	14.59	742.0	1.5653	12.68	747.2	1.5547	11.39	752.3	1.5541
260	17.79	747.9	1.5948	14.81	747.5	1.5732	12.86	752.7	1.5624	11.58	757.8	1.5619
270	18.04	753.4	1.6025	15.02	753.0	1.5808	13.04	758.2	1.5701	11.77		
280	18.30	758.9	1.6101	15.23	758.5	1.5884						

REFRIGERANT PROPERTIES

TABLE 8-7 (Continued)
SUPERHEATED AMMONIA

Temperature, ° F	60 lb./in. ² (30.21° F)			70 lb./in. ² (37.70° F)			80 lb./in. ² (44.40° F)					
	<i>p</i>	<i>h</i>	<i>s</i>	<i>p</i>	<i>h</i>	<i>s</i>	<i>p</i>	<i>h</i>	<i>s</i>			
	(6.710)	(818.2)	(1.2959)	(4.805)	(620.5)	(1.2737)	(4.151)	(632.4)	(1.2658)	(3.656)	(624.0)	(1.2345)
30	5.838	623.4	1.3046	4.933	626.8	1.2913	4.177	623.9	1.2688	3.712	627.7	1.2619
40	5.988	629.5	1.3169	5.060	632.9	1.3035	4.290	630.4	1.2816	3.812	634.3	1.2745
50	6.135	635.4	1.3286	5.184	639.0	1.3152	4.401	636.6	1.2937	3.909	640.6	1.2866
60	6.283	641.2	1.3399	5.307	644.9	1.3265	4.509	642.7	1.3054	4.005	646.7	1.2981
70	6.564	646.9	1.3508	5.428	650.7	1.3373	4.615	648.7	1.3166	4.098	652.8	1.3092
80	6.704	652.6	1.3613	5.547	656.4	1.3479	4.719	654.6	1.3274	4.190	658.7	1.3199
100	6.843	663.7	1.3816	5.665	662.1	1.3581	4.822	660.4	1.3378	4.281	664.6	1.3303
110	7.050	669.2	1.3914	5.781	667.7	1.3681	4.924	666.1	1.3480	4.371	670.4	1.3404
120	7.177	674.7	1.4009	5.897	673.3	1.3778	5.025	671.8	1.3579	4.460	676.1	1.3502
130	7.252	680.2	1.4103	6.012	678.9	1.3873	5.125	677.5	1.3676	4.548	681.8	1.3598
140	7.357	685.7	1.4195	6.126	684.4	1.3966	5.224	683.1	1.3770	4.635	687.5	1.3692
150	7.521	691.1	1.4296	6.239	689.9	1.4058	5.323	688.7	1.3863	4.722	693.2	1.3784
160	7.655	696.4	1.4374	6.352	695.5	1.4148	5.420	694.3	1.3954	4.808	698.8	1.3874
170	7.758	702.1	1.4452	6.464	701.0	1.4236	5.518	699.0	1.4043	4.893	704.4	1.3963
180	7.921	707.5	1.4535	6.576	706.5	1.4323	5.611	705.5	1.4131	4.978	710.0	1.4050
190	8.053	713.0	1.4633	6.687	712.0	1.4409	5.711	711.0	1.4210	5.063	715.6	1.4138
200	8.185	718.5	1.4716	6.798	717.5	1.4493	5.807	716.6	1.4302	5.147	721.3	1.4220
210	8.317	724.0	1.4799	6.909	723.1	1.4576	5.902	722.2	1.4386	5.231	726.9	1.4304
220	8.448	729.4	1.4880	7.019	728.6	1.4658	6.003	727.7	1.4469	5.315	732.5	1.4386
230	8.579	735.0	1.4961	7.129	734.1	1.4739	6.107	733.3	1.4550	5.398	738.1	1.4467
240	8.710	740.5	1.5040	7.238	739.7	1.4819	6.187	738.9	1.4631	5.482	743.8	1.4547
250	8.840	746.0	1.5119	7.348	745.3	1.4898	6.281	744.5	1.4711	5.565	749.4	1.4626
260	8.970	751.6	1.5197	7.457	750.9	1.4976	6.376	750.1	1.4789	5.647	755.1	1.4704
270	9.100	757.2	1.5274	7.566	756.5	1.5053	6.470	755.8	1.4868	5.730	760.7	1.4781
280	9.230	762.7	1.5350	7.675	762.1	1.5130	6.563	761.4	1.4943	5.812	766.4	1.4857
290	9.360	768.4	1.5425	7.783	767.7	1.5208	6.657	767.1	1.5019	5.894	772.1	1.4933
300	9.489	774.0	1.5500	7.892	773.3	1.5281	6.750	772.7	1.5095	5.976	777.8	1.5008
310	9.618	779.6	1.5574	8.000	779.0	1.5355	6.844	778.4	1.5169			

TABLE 8-7 (Continued)
SUPERHEATED AMMONIA

Temperature, °F.	90 lb./in. ² (60.34° F)			100 lb./in. ² (66.05° F)			110 lb./in. ² (61.21° F)			120 lb./in. ² (66.02° F)		
	<i>v</i>	<i>h</i>	<i>s</i>	<i>v</i>	<i>h</i>	<i>s</i>	<i>v</i>	<i>h</i>	<i>s</i>	<i>v</i>	<i>h</i>	<i>s</i>
(at sat. <i>t_s</i>)	(5.266)	(685.5)	(1.2445)	(2.652)	(686.5)	(1.2556)	(2.692)	(687.5)	(1.2675)	(2.476)	(688.4)	(1.2801)
70	3.442	638.3	1.2695	3.068	636.0	1.2539	2.761	633.7	1.2592	2.505	631.3	1.2355
80	3.529	644.7	1.2814	3.149	642.6	1.2661	2.837	640.5	1.2619	2.576	638.3	1.2386
90	3.614	650.9	1.2928	3.227	649.0	1.2778	2.910	647.0	1.2640	2.645	645.0	1.2510
100	3.698	657.0	1.3088	3.304	655.2	1.2891	2.981	653.4	1.2755	2.712	651.6	1.2628
110	3.780	663.0	1.3144	3.380	661.3	1.2999	3.051	659.7	1.2866	2.778	658.0	1.2741
120	3.862	668.9	1.3247	3.454	667.3	1.3104	3.120	665.8	1.2972	2.842	664.2	1.2850
130	3.942	674.7	1.3347	3.527	673.3	1.3206	3.188	671.9	1.3076	2.905	670.4	1.2956
140	4.021	680.5	1.3444	3.600	679.2	1.3305	3.255	677.8	1.3176	2.967	676.5	1.3058
150	4.100	686.3	1.3539	3.672	685.0	1.3401	3.321	683.7	1.3274	3.029	682.5	1.3157
160	4.178	692.0	1.3633	3.743	690.8	1.3495	3.386	689.7	1.3370	3.089	688.4	1.3254
170	4.255	697.7	1.3724	3.813	696.6	1.3588	3.451	695.4	1.3463	3.149	694.3	1.3348
180	4.332	703.4	1.3813	3.883	702.3	1.3678	3.515	701.2	1.3555	3.209	700.2	1.3441
190	4.408	709.0	1.3901	3.952	708.0	1.3767	3.579	707.0	1.3644	3.268	706.0	1.3531
200	4.484	714.7	1.3988	4.021	713.7	1.3854	3.642	712.8	1.3732	3.326	711.8	1.3620
210	4.560	720.4	1.4073	4.090	719.4	1.3940	3.705	718.5	1.3819	3.385	717.6	1.3707
220	4.635	726.0	1.4157	4.158	725.1	1.4024	3.768	724.3	1.3904	3.442	723.4	1.3793
230	4.710	731.7	1.4239	4.226	730.8	1.4108	3.830	730.0	1.3988	3.500	729.2	1.3877
240	4.785	737.3	1.4321	4.294	736.5	1.4190	3.892	735.7	1.4070	3.557	734.9	1.3960
250	4.859	743.0	1.4401	4.361	742.2	1.4271	3.954	741.5	1.4151	3.614	740.7	1.4042
260	4.933	748.7	1.4481	4.428	747.9	1.4350	4.015	747.2	1.4232	3.671	746.5	1.4123
270	5.007	754.4	1.4559	4.495	753.6	1.4429	4.076	752.9	1.4311	3.727	752.2	1.4202
280	5.081	760.0	1.4637	4.562	759.4	1.4507	4.137	758.7	1.4381	3.783	758.0	1.4281
290	5.155	765.8	1.4713	4.629	765.1	1.4584	4.198	764.5	1.4468	3.839	763.8	1.4359
300	5.228	771.5	1.4789	4.695	770.8	1.4660	4.259	770.2	1.4543	3.895	769.6	1.4435
310	5.301	777.2	1.4864	4.761	776.4	1.4736	4.319	776.0	1.4619	3.951	775.4	1.4511
320	5.374	783.0	1.4938	4.827	782.0	1.4810	4.379	781.8	1.4693	4.006	781.2	1.4586
330				4.893	788.2	1.4884	4.439	787.6	1.4767	4.061	787.0	1.4660
340				4.959	794.0	1.4957	4.500	793.4	1.4841	4.117	792.9	1.4734
350				5.024	799.8	1.5029	4.559	799.3	1.4915	4.172	798.7	1.4807

TABLE 8-7 (Continued)
SUPERHEATED AMMONIA

Temperature, °F	t (sat. vap.)	130 lb./in. ² (70.53° F)			140 lb./in. ² (74.79° F)			150 lb./in. ² (78.81° F)			160 lb./in. ² (82.64° F)		
		v	h	s	v	h	s	v	h	s	v	h	s
90	(8.291)	2.421	643.0	(1.2155) 1.2388	(8.155) 2.228	640.9	(1.2065) 2.061	(8.020) 638.8	(1.2009) 1.2161	(1.874) 1.914	631.1 636.6	(1.1959) 1.2055	
100	2.454	649.7	1.2509	2.288	647.8	1.2396	2.118	645.9	1.2949	1.969	643.9	1.2186	
110	2.546	656.3	1.2625	2.347	654.5	1.2515	2.174	652.8	1.2949	2.023	651.0	1.2311	
120	2.606	662.7	1.2736	2.404	661.1	1.2628	2.228	659.4	1.3047	2.075	657.8	1.2426	
130	2.665	668.9	1.2843	2.460	667.4	1.2738	2.281	665.9	1.3142	2.125	664.4	1.2542	
140	2.724	675.1	1.2947	2.515	673.7	1.2843	2.334	672.3	1.3236	2.175	670.9	1.2652	
150	2.781	681.2	1.3048	2.569	679.9	1.2945	2.385	678.6	1.3327	2.224	677.2	1.2757	
160	2.838	687.2	1.3146	2.622	686.0	1.3045	2.435	684.8	1.3416	2.272	683.5	1.2859	
170	2.894	693.2	1.3241	2.675	692.0	1.3141	2.485	690.9	1.3504	2.319	689.7	1.2958	
180	2.949	699.1	1.3335	2.727	698.0	1.3236	2.534	696.9	1.3590	2.365	695.8	1.3054	
190	3.004	705.0	1.3428	2.779	704.0	1.3328	2.583	702.9	1.3675	2.411	701.9	1.3145	
200	3.059	710.9	1.3516	2.830	709.9	1.3418	2.631	708.9	1.3758	2.457	707.9	1.3240	
210	3.113	716.7	1.3604	2.880	715.8	1.3507	2.679	714.8	1.3840	2.502	713.9	1.3331	
220	3.167	722.5	1.3690	2.931	721.6	1.3594	2.726	720.7	1.3921	2.547	719.9	1.3419	
230	3.220	728.3	1.3775	2.981	727.5	1.3679	2.773	726.6	1.4001	2.591	725.8	1.3506	
240	3.273	734.1	1.3858	3.030	733.3	1.3763	2.820	732.5	1.4079	2.635	731.7	1.3591	
250	3.326	739.9	1.3941	3.080	739.2	1.3846	2.866	738.4	1.4157	2.679	737.6	1.3675	
260	3.379	745.7	1.4022	3.129	744.3	1.3928	2.912	743.5	1.4234	2.723	743.5	1.3757	
270	3.431	751.5	1.4102	3.179	749.8	1.4008	2.958	749.1	1.4310	2.766	749.4	1.3838	
280	3.483	757.3	1.4181	3.227	755.7	1.4088	3.004	756.0	1.4385	2.809	755.3	1.3919	
290	3.535	763.1	1.4259	3.275	761.5	1.4166	3.049	761.8	1.4459	2.852	761.2	1.3998	
300	3.587	769.0	1.4336	3.323	767.3	1.4243	3.095	767.7	1.4532	2.895	767.1	1.4076	
310	3.639	774.8	1.4412	3.371	773.2	1.4320	3.140	773.6	1.4605	2.937	773.0	1.4153	
320	3.691	780.6	1.4487	3.420	779.0	1.4395	3.185	779.4	1.4678	2.980	778.9	1.4229	
330	3.742	786.4	1.4562	3.467	784.8	1.4470	3.230	785.3	1.4750	3.022	784.8	1.4304	
340	3.793	792.3	1.4636	3.515	791.8	1.4544	3.274	791.2	1.4822	3.064	790.7	1.4379	
350	3.844	798.2	1.4709	3.563	797.7	1.4617	3.319	797.1	1.4893	3.106	796.6	1.4452	
360				3.610	803.6	1.4690	3.364	803.0	1.4965	3.148	802.5	1.4525	
370										3.189	808.5	1.4597	
380										3.231	814.5	1.4669	
390										3.273	820.4	1.4740	
400										3.314	826.4	1.4810	

TABLE 8-7 (Continued)
SUPERHEATED AMMONIA

Temperature, °F	170 lb./in. ² (86.29° F)			180 lb./in. ² (89.78° F)			190 lb./in. ² (93.13° F)			200 lb./in. ² (96.34° F)		
	<i>v</i>	<i>h</i>	<i>s</i>	<i>v</i>	<i>h</i>	<i>s</i>	<i>v</i>	<i>h</i>	<i>s</i>	<i>v</i>	<i>h</i>	<i>s</i>
(at sat'g)	(1.764)	(681.6)	(1.1900)	(1.667)	(682.0)	(1.1860)	(1.581)	(682.4)	(1.1808)	(1.508)	(682.7)	(1.1766)
100	1.837	641.9	1.2087	1.720	639.9	1.1992	1.615	637.8	1.1899	1.520	635.0	1.1809
110	1.889	649.1	1.2215	1.770	647.3	1.2123	1.663	645.4	1.2034	1.567	643.4	1.1947
120	1.939	656.1	1.2336	1.818	654.4	1.2247	1.710	652.6	1.2160	1.612	650.9	1.2077
130	1.985	662.3	1.2452	1.865	661.3	1.2363	1.755	659.7	1.2281	1.655	658.1	1.2200
140	2.035	669.4	1.2563	1.910	668.0	1.2477	1.799	666.5	1.2396	1.698	665.0	1.2317
150	2.081	675.9	1.2669	1.955	674.6	1.2586	1.842	673.2	1.2506	1.740	671.8	1.2429
160	2.127	682.3	1.2773	1.999	681.0	1.2691	1.884	679.7	1.2612	1.780	678.4	1.2537
170	2.172	688.5	1.2873	2.042	687.3	1.2792	1.925	686.1	1.2715	1.820	685.3	1.2641
180	2.216	694.7	1.2971	2.084	693.6	1.2891	1.965	692.4	1.2815	1.859	691.7	1.2743
190	2.260	700.8	1.3066	2.126	699.8	1.2987	2.005	698.7	1.2912	1.897	697.7	1.2846
200	2.303	706.9	1.3159	2.167	705.9	1.3081	2.045	704.9	1.3007	1.935	703.9	1.2945
210	2.346	713.0	1.3249	2.208	712.0	1.3172	2.084	711.1	1.3099	1.972	710.1	1.3049
220	2.389	719.0	1.3338	2.248	718.1	1.3262	2.123	717.2	1.3189	2.009	716.2	1.3120
230	2.431	724.9	1.3426	2.288	724.1	1.3350	2.161	723.2	1.3278	2.048	722.4	1.3209
240	2.473	730.9	1.3512	2.328	730.1	1.3438	2.199	729.3	1.3366	2.082	728.4	1.3296
250	2.514	736.8	1.3596	2.367	736.1	1.3521	2.236	735.3	1.3450	2.118	734.5	1.3383
260	2.555	742.8	1.3679	2.407	742.0	1.3605	2.274	741.3	1.3534	2.154	740.5	1.3467
270	2.596	748.7	1.3761	2.446	748.0	1.3687	2.311	747.3	1.3617	2.189	746.5	1.3550
280	2.637	754.6	1.3841	2.484	753.9	1.3768	2.348	753.2	1.3698	2.225	752.5	1.3631
290	2.678	760.5	1.3921	2.523	759.9	1.3847	2.384	759.2	1.3778	2.260	758.5	1.3713
300	2.718	766.4	1.3999	2.561	765.8	1.3926	2.421	765.2	1.3857	2.295	764.5	1.3791
310	2.758	772.3	1.4076	2.599	771.7	1.4004	2.457	771.1	1.3935	2.329	770.5	1.3869
320	2.798	778.3	1.4153	2.637	777.7	1.4081	2.493	777.1	1.4018	2.364	776.5	1.3947
330	2.838	784.2	1.4228	2.675	783.6	1.4156	2.529	783.1	1.4098	2.398	782.5	1.4023
340	2.878	790.1	1.4303	2.713	789.6	1.4231	2.565	789.0	1.4163	2.432	788.5	1.4099
350	2.918	796.2	1.4377	2.750	795.6	1.4305	2.601	795.1	1.4238	2.466	794.5	1.4173
360	2.957	802.0	1.4450	2.788	801.5	1.4379	2.637	801.0	1.4311	2.500	800.5	1.4241
370	2.997	808.0	1.4522	2.825	807.5	1.4451	2.672	807.0	1.4384	2.534	806.5	1.4320
380	3.036	814.0	1.4594	2.863	813.5	1.4523	2.707	813.0	1.4456	2.568	812.5	1.4392
390	3.075	820.0	1.4665	2.900	819.5	1.4594	2.743	819.0	1.4527	2.601	818.0	1.4464
400	3.114	826.0	1.4735	2.937	825.5	1.4665	2.778	825.1	1.4598	2.635	824.6	1.4534

TABLE 8-8a
 SATURATED METHYL CHLORIDE (CH₂Cl)

Temperature °F	Pressure		Volume		Heat Content (Enthalpy) from -40°			Entropy from -40°	
	Abs lb/in. ² <i>p</i>	Vapor ft ³ /lb <i>v_g</i>	Liquid lb/ft ³ <i>l/v_f</i>	Liquid Btu/lb <i>h_f</i>	Latent Btu/lb <i>h_{fg}</i>	Vapor Btu/lb <i>h_g</i>	Liquid Btu/lb °F <i>s_f</i>	Vapor Btu/lb °F <i>s_g</i>	
-80	1.953	41.08	66.98	-13.888	198.64	184.75	-0.0351	0.4882	
-70	2.751	29.84	66.31	-10.521	196.77	186.25	-0.0261	0.4790	
-60	3.799	22.09	65.66	-7.039	194.78	187.74	-0.0172	0.4703	
-50	5.155	16.64	65.02	-3.532	192.72	189.19	-0.0085	0.4620	
-40	6.878	12.72	64.39	0.000	190.66	190.66	0.0000	0.4544	
-30	9.036	9.873	63.78	3.562	188.52	192.08	0.0084	0.4472	
-20	11.71	7.761	63.17	7.146	186.34	193.49	0.0166	0.4405	
-10	14.96	6.176	62.58	10.75	184.11	194.87	0.0247	0.4343	
0	18.90	4.969	62.00	14.39	181.85	196.23	0.0327	0.4284	
2	19.77	4.763	61.88	15.12	181.39	196.51	0.0343	0.4273	
4	20.08	4.568	61.77	15.85	180.93	196.78	0.0359	0.4262	
6	21.15	4.471	61.65	16.21	180.70	196.92	0.0367	0.4257	
8	21.62	4.379	61.54	16.58	180.47	197.05	0.0375	0.4251	
10	22.59	4.206	61.43	17.31	180.01	197.31	0.0390	0.4240	
12	23.60	4.038	61.31	18.04	179.53	197.58	0.0406	0.4229	
14	24.64	3.878	61.20	18.77	179.06	197.83	0.0422	0.4218	
16	25.72	3.726	61.09	19.51	178.58	198.09	0.0437	0.4208	
18	26.83	3.581	60.98	20.25	178.10	198.34	0.0453	0.4198	
20	27.97	3.443	60.83	20.98	177.61	198.59	0.0468	0.4187	
22	29.16	3.312	60.72	21.73	177.11	198.84	0.0484	0.4177	
24	30.38	3.186	60.61	22.47	176.61	199.08	0.0499	0.4166	
26	31.64	3.067	60.46	23.21	176.11	199.32	0.0514	0.4156	
28	32.95	2.952	60.31	23.95	175.61	199.56	0.0530	0.4146	
30	34.29	2.843	60.17	24.70	175.10	199.79	0.0545	0.4136	
32	35.68	2.739	60.06	25.44	174.59	200.03	0.0560	0.4126	
34	37.11	2.640	59.92	26.18	174.08	200.26	0.0575	0.4117	
36	38.58	2.546	59.77	26.93	173.56	200.49	0.0590	0.4107	
38	40.09	2.455	59.63	35.19	173.04	200.71	0.0605	0.4098	
40	41.65	2.369	59.49	28.42	172.53	200.95	0.0621	0.4088	
42	43.25	2.286	59.38	29.17	172.00	201.17	0.0636	0.4079	
44	44.91	2.206	59.24	29.92	171.48	201.40	0.0651	0.4070	
46	46.61	2.130	59.10	30.67	170.95	201.62	0.0665	0.4061	
48	48.35	2.057	58.96	31.42	170.42	201.84	0.0680	0.4052	
50	50.15	1.987	58.82	32.17	169.89	202.06	0.0695	0.4043	
52	51.99	1.920	58.69	32.93	169.35	202.28	0.0710	0.4034	
54	53.88	1.856	58.55	33.68	168.81	202.49	0.0725	0.4025	
56	55.83	1.794	58.41	34.44	168.27	202.71	0.0740	0.4017	
58	57.83	1.735	58.28	35.19	167.72	202.91	0.0754	0.4008	
60	59.88	1.679	58.14	35.95	167.18	203.13	0.0769	0.3999	
62	62.00	1.624	58.00	36.71	166.62	203.33	0.0784	0.3991	
64	64.17	1.572	57.87	37.47	166.07	203.54	0.0798	0.3983	
66	66.39	1.522	57.74	38.23	165.51	203.74	0.0813	0.3974	
68	68.67	1.473	57.60	39.00	164.95	203.95	0.0827	0.3966	
70	71.01	1.427	57.47	39.76	164.39	204.15	0.0842	0.3958	
72	73.41	1.382	57.34	40.52	163.82	204.34	0.0856	0.3950	
74	75.86	1.339	57.21	41.29	163.24	204.53	0.0870	0.3941	
76	78.37	1.298	57.08	42.06	162.66	204.72	0.0885	0.3933	
78	80.94	1.258	56.95	42.82	162.08	204.90	0.0899	0.3925	
80	83.57	1.220	56.82	43.59	161.50	205.09	0.0913	0.3918	
82	86.26	1.183	56.69	44.36	160.91	205.27	0.0928	0.3910	
84	89.01	1.148	56.56	45.13	160.32	205.45	0.0942	0.3902	
86	91.82	1.114	56.40	45.90	159.72	205.62	0.0956	0.3894	
88	94.70	1.081	56.24	46.67	159.13	205.80	0.0970	0.3887	
90	97.64	1.049	56.12	47.44	158.52	205.96	0.0984	0.3879	
92	100.6	1.018	55.99	48.21	157.92	206.13	0.0998	0.3872	
94	103.7	0.9889	55.83	48.99	157.31	206.30	0.1012	0.3865	
96	106.9	0.9603	55.68	49.77	156.69	206.46	0.1026	0.3857	
98	110.1	0.9333	55.56	50.54	156.08	206.62	0.1041	0.3850	
100	113.4	0.9069	55.43	51.32	155.46	206.78	0.1055	0.3843	
102	116.7	0.8814	55.31	52.09	154.85	206.94	0.1069	0.3836	
104	120.1	0.8568	55.15	52.87	154.22	207.09	0.1082	0.3828	
106	123.6	0.8331	55.01	53.65	153.60	207.25	0.1096	0.3822	
108	127.2	0.8105	54.85	54.43	152.97	207.40	0.1110	0.3815	
108	130.8	0.7884	54.70	55.22	152.33	207.55	0.1124	0.3808	
110	134.5	0.7672	54.55	56.00	151.70	207.70	0.1138	0.3801	
120	154.2	0.6710	53.79	59.93	148.46	208.39	0.1206	0.3768	
130	175.9	0.5889	52.99	63.89	145.13	209.02	0.1274	0.3736	
140	199.6	0.5189	52.22	67.87	141.71	209.58	0.1341	0.3705	
150	225.4	0.4586	51.41	71.87	138.23	210.10	0.1407	0.3674	
160	253.5	0.4070	50.56	75.90	134.66	210.56	0.1473	0.3646	
170	283.9	0.3613	49.63	79.97	130.96	210.93	0.1538	0.3618	

TABLE 8-8b
SUPERHEATED METHYL CHLORIDE

Temperature °F	Abs. Press., 10 lb/in. ² Sat'n Temp., -26.1° F			Abs. Press., 15 lb/in. ² Sat'n Temp., -9.9° F			Abs. Press., 20 lb/in. ² Sat'n Temp., 2.5° F			
	<i>t</i>	<i>v</i>	<i>h</i>	<i>s</i>	<i>v</i>	<i>h</i>	<i>s</i>	<i>v</i>	<i>h</i>	<i>s</i>
(at sat'n)	(8.993)	(192.64)	(0.4446)	(6.161)	(194.91)	(0.4342)	(4.710)	(196.58)	(0.4270)	
-20	9.124	193.67	0.4471							
-10	9.346	195.50	0.4512							
0	9.567	197.32	0.4552	6.313	196.67	0.4383				
10	9.788	199.18	0.4591	6.464	198.58	0.4423	4.801	197.95	0.4300	
20	10.01	201.04	0.4630	6.618	200.48	0.4462	4.917	199.90	0.4341	
30	10.23	202.91	0.4669	6.766	202.38	0.4501	5.032	201.82	0.4380	
40	10.45	204.78	0.4707	6.916	204.28	0.4540	5.146	203.75	0.4420	
50	10.67	206.70	0.4745	7.067	206.21	0.4578	5.260	205.71	0.4458	
60	10.89	208.62	0.4782	7.215	208.14	0.4616	5.373	207.66	0.4496	
65	11.00	209.60	0.4800	7.289	209.13	0.4635	5.429	208.66	0.4515	
70	11.11	210.68	0.4819	7.363	210.12	0.4654	5.486	209.66	0.4534	
80	11.33	212.53	0.4856	7.511	212.09	0.4691	5.599	211.65	0.4572	
90	11.55	214.51	0.4892	7.659	214.09	0.4727	5.711	213.67	0.4608	
100	11.77	216.50	0.4928	7.806	216.09	0.4764	5.823	215.69	0.4645	
110	11.99	218.52	0.4964	7.953	218.13	0.4800	5.935	217.75	0.4681	
120	12.21	220.54	0.5000	8.100	220.17	0.4835	6.046	219.80	0.4717	
130	12.43	222.61	0.5035	8.247	222.25	0.4871	6.157	221.90	0.4755	
140	12.65	224.67	0.5069	8.394	224.33	0.4906	6.268	223.99	0.4788	
150	12.86	226.77	0.5104	8.540	226.44	0.4940	6.379	226.12	0.4823	

Temperature °F	Abs. Press., 30 lb/in. ² Sat'n Temp., 21.4° F			Abs. Press., 40 lb/in. ² Sat'n Temp., 35.9° F			Abs. Press., 50 lb/in. ² Sat'n Temp., 47.8° F			
	(at sat'n)	(3.224)	(199.00)	(0.4189)	(2.461)	(200.73)	(0.4097)	(1.992)	(202.09)	(0.4043)
30	3.292	200.62	0.4205							
40	3.373	202.62	0.4245	2.485	201.38	0.4115				
50	3.453	204.66	0.4284	2.547	203.54	0.4156	2.003	202.55	0.4053	
60	3.532	206.70	0.4324	2.608	205.69	0.4196	2.054	204.65	0.4094	
65	3.571	207.71	0.4343	2.638	206.73	0.4216	2.079	205.71	0.4114	
70	3.610	208.73	0.4362	2.668	207.77	0.4236	2.104	206.77	0.4134	
80	3.687	210.77	0.4400	2.729	209.84	0.4275	2.154	208.89	0.4174	
90	3.764	212.82	0.4438	2.788	211.94	0.4313	2.203	211.03	0.4213	
100	3.840	214.87	0.4475	2.847	214.03	0.4351	2.252	213.18	0.4252	
110	3.916	216.96	0.4512	2.906	216.17	0.4389	2.300	215.35	0.4290	
120	3.992	219.06	0.4548	2.964	218.32	0.4426	2.348	217.52	0.4328	
130	4.067	221.18	0.4585	3.022	220.46	0.4463	2.396	219.70	0.4366	
140	4.142	223.30	0.4620	3.080	222.60	0.4499	2.443	221.88	0.4402	
150	4.217	225.45	0.4656	3.138	224.78	0.4535	2.490	224.10	0.4439	
160				3.195	226.96	0.4570	2.537	226.32	0.4475	
170				3.252	229.18	0.4606	2.584	228.55	0.4511	
180				3.309	231.38	0.4641	2.630	230.79	0.4546	
190				3.366	233.62	0.4675	2.676	233.05	0.4581	
200							2.722	235.32	0.4616	

TABLE 8-8b (Continued)

SUPERHEATED METHYL CHLORIDE

Temperature °F	Abs. Press., 60 lb/in. ² Sat'n Temp., 58.1° F			Abs. Press., 70 lb/in. ² Sat'n Temp., 67.1° F			Abs. Press., 80 lb/in. ² Sat'n Temp., 75.3° F			
	<i>t</i>	<i>v</i>	<i>h</i>	<i>s</i>	<i>v</i>	<i>h</i>	<i>s</i>	<i>v</i>	<i>h</i>	<i>s</i>
(at sat'n)	(1.676)	(203.17)	(0.3998)	(1.446)	(204.08)	(0.3960)	(1.272)	(204.84)	(0.3927)	
60	1.684	203.49	0.4008							
65	1.705	204.60	0.4028							
70	1.727	205.71	0.4049	1.458	204.72	0.3974				
80	1.770	207.92	0.4089	1.496	206.93	0.4015	1.289	205.85	0.3949	
90	1.812	210.11	0.4129	1.533	209.17	0.4056	1.323	208.17	0.3991	
100	1.854	212.30	0.4169	1.570	211.41	0.4096	1.356	210.50	0.4032	
110	1.895	214.50	0.4208	1.606	213.64	0.4136	1.389	212.76	0.4073	
120	1.936	216.69	0.4246	1.642	215.86	0.4175	1.421	215.02	0.4113	
130	1.977	218.93	0.4284	1.678	218.14	0.4214	1.453	217.35	0.4152	
140	2.017	221.16	0.4322	1.713	220.43	0.4253	1.485	219.68	0.4191	
150	2.057	223.41	0.4359	1.748	222.71	0.4290	1.516	222.00	0.4229	
160	2.097	225.66	0.4396	1.783	224.99	0.4327	1.547	224.32	0.4267	
170	2.137	227.92	0.4432	1.818	227.28	0.4364	1.578	226.64	0.4304	
180	2.176	230.19	0.4468	1.852	229.57	0.4400	1.608	228.96	0.4341	
190	2.215	232.48	0.4503	1.886	231.89	0.4436	1.638	231.30	0.4377	
200	2.254	234.77	0.4538	1.920	234.21	0.4471	1.668	233.65	0.4413	
Temperature °F	Abs. Press., 100 lb/in. ² Sat'n Temp., 89.6° F			Abs. Press., 120 lb/in. ² Sat'n Temp., 101.9° F			Abs. Press., 140 lb/in. ² Sat'n Temp., 112.9° F			
	(at sat'n)	(1.026)	(206.11)	(0.3872)	(0.858)	(207.10)	(0.3827)	(0.738)	(207.91)	(0.3790)
90	1.026	206.21	0.3877							
100	1.055	208.58	0.3920							
110	1.083	210.96	0.3962	0.878	209.12	0.3867				
120	1.111	213.33	0.4003	0.903	211.61	0.3910	0.754	209.70	0.3825	
130	1.138	215.74	0.4044	0.927	214.09	0.3952	0.776	212.32	0.3868	
140	1.165	218.15	0.4084	0.951	216.57	0.3993	0.796	214.94	0.3911	
150	1.191	220.55	0.4124	0.974	219.04	0.4034	0.818	217.48	0.3954	
160	1.217	222.94	0.4163	0.996	221.51	0.4074	0.838	220.03	0.3995	
170	1.243	225.33	0.4201	1.019	223.97	0.4113	0.858	222.56	0.4036	
180	1.268	227.71	0.4239	1.041	226.42	0.4152	0.877	225.09	0.4076	
190	1.293	230.10	0.4276	1.062	228.88	0.4190	0.896	227.61	0.4115	
200	1.318	232.50	0.4312	1.083	231.33	0.4228	0.915	230.13	0.4154	
210	1.343	234.91	0.4349	1.104	233.79	0.4265	0.934	232.65	0.4192	
220	1.367	237.32	0.4384	1.125	236.25	0.4301	0.953	235.17	0.4229	
230	1.391	239.76	0.4420	1.146	238.74	0.4337	0.971	237.70	0.4266	
240	1.415	242.20	0.4455	1.167	241.22	0.4373	0.989	240.23	0.4302	
Temperature °F	Abs. Press., 160 lb/in. ² Sat'n Temp., 122.8° F			Abs. Press., 200 lb/in. ² Sat'n Temp., 140.3° F			Abs. Press., 240 lb/in. ² Sat'n Temp., 155.3° F			
	(at sat'n)	(0.647)	(208.69)	(0.3757)	(0.517)	(209.60)	(0.3702)	(0.430)	(210.32)	(0.3657)
130	0.661	210.54	0.3793							
140	0.681	213.24	0.3837							
150	0.700	215.87	0.3881	0.533	212.41	0.3749				
160	0.719	218.50	0.3924	0.551	215.30	0.3796	0.437	211.75	0.3683	
170	0.737	221.11	0.3966	0.566	218.09	0.3842	0.451	214.79	0.3732	
180	0.754	223.73	0.4007	0.582	220.87	0.3886	0.466	217.82	0.3779	
190	0.772	226.31	0.4048	0.597	223.61	0.3929	0.480	220.73	0.3824	
200	0.789	228.90	0.4087	0.612	226.35	0.3971	0.493	223.64	0.3868	
210	0.806	231.48	0.4126	0.626	229.05	0.4012	0.506	226.49	0.3911	
220	0.823	234.05	0.4164	0.641	231.75	0.4052	0.519	229.33	0.3953	
230	0.839	236.63	0.4202	0.654	234.43	0.4091	0.531	232.13	0.3994	
240	0.855	239.21	0.4239	0.668	237.12	0.4129	0.543	234.93	0.4034	
250				0.682	239.75	0.4167	0.555	237.67	0.4073	
260				0.695	242.39	0.4204	0.566	240.40	0.4111	
270				0.708	245.02	0.4240	0.578	243.11	0.4149	
280				0.721	247.65	0.4276	0.590	245.82	0.4186	
290				0.734	250.29	0.4311	0.601	248.54	0.4222	

TABLE 8-9
SOLID AND SATURATED CARBON DIOXIDE

Temperature °F <i>t</i>	Pressure lb/in. ² <i>p</i>	Volume		Total Heat from -40°			Entropy from -40° plus 1	
		Solid ft ³ /lb <i>v_f</i>	Vapor ft ³ /lb <i>v_g</i>	Solid Btu/lb <i>h_f</i>	Latent Btu/lb <i>L</i>	Vapor Btu/lb <i>h_g</i>	Solid Btu/lb°F <i>s_f</i>	Vapor Btu/lb°F <i>s_g</i>
-140	3.18	0.01008	24.320	-121.5	250.7	129.2	0.6847	1.4690
-130	5.39	0.01012	14.740	-118.8	249.4	130.6	0.6932	1.4500
-120	8.90	0.01018	9.179	-116.0	248.0	132.0	0.7014	1.4418
-115	11.31	0.01021	7.279	-114.6	247.3	132.7	0.7055	1.4231
-110	14.22	0.01024	5.848	-113.2	246.4	133.2	0.7096	1.4145
-109	14.88	0.01025	5.597	-112.9	246.2	133.3	0.7105	1.4128
-108	15.57	0.01026	5.358	-112.6	246.0	133.4	0.7114	1.4111
-107	16.29	0.01027	5.129	-112.3	245.8	133.5	0.7123	1.4085
-106	17.04	0.01027	4.911	-112.0	245.6	133.6	0.7132	1.4079
-105	17.82	0.01028	4.703	-111.6	245.4	133.8	0.7141	1.4063
-104	18.63	0.01029	4.505	-111.3	245.2	133.9	0.7150	1.4045
-103	19.48	0.01030	4.316	-111.0	245.0	134.0	0.7159	1.4029
-102	20.36	0.01031	4.138	-110.7	244.8	134.1	0.7168	1.4013
-101	21.27	0.01031	3.967	-110.4	244.6	134.2	0.7177	1.3997
-100	22.22	0.01032	3.804	-110.1	244.4	134.3	0.7185	1.3981
-99	23.20	0.01033	3.648	-109.8	244.1	134.3	0.7194	1.3965
-98	24.22	0.01033	3.499	-109.5	243.9	134.4	0.7203	1.3949
-97	25.28	0.01034	3.357	-109.2	243.6	134.4	0.7212	1.3933
-96	26.39	0.01035	3.222	-108.9	243.4	134.5	0.7221	1.3917
-95	27.54	0.01035	3.093	-108.5	243.2	134.7	0.7231	1.3901
-94	28.73	0.01036	2.970	-108.1	242.9	134.8	0.7241	1.3905
-93	29.97	0.01037	2.852	-107.8	242.7	134.9	0.7251	1.3909
-92	31.26	0.01038	2.738	-107.4	242.4	135.0	0.7261	1.3953
-91	32.69	0.01039	2.629	-107.0	242.1	135.1	0.7271	1.3937
-90	33.98	0.01040	2.525	-106.7	241.8	135.1	0.7281	1.3821
-89	35.41	0.01040	2.425	-106.3	241.5	135.2	0.7291	1.3806
-88	36.89	0.01041	2.330	-105.9	241.1	135.2	0.7302	1.3790
-87	38.43	0.01042	2.240	-105.5	240.8	135.3	0.7313	1.3774
-86	40.02	0.01043	2.153	-105.1	240.4	135.3	0.7323	1.3758
-85	41.67	0.01044	2.070	-104.6	240.1	135.5	0.7333	1.3742
-84	43.38	0.01045	1.995	-104.2	239.7	135.5	0.7344	1.3727
-83	45.15	0.01046	1.913	-103.8	239.3	135.5	0.7355	1.3711
-82	46.98	0.01047	1.839	-103.4	238.9	135.5	0.7366	1.3695
-81	48.88	0.01048	1.768	-103.0	238.6	135.6	0.7377	1.3679
-80	50.85	0.01048	1.700	-102.5	238.2	135.7	0.7389	1.3663
-79	52.89	0.01049	1.636	-102.0	237.7	135.7	0.7400	1.3648
-78	55.00	0.01050	1.575	-101.5	237.3	135.8	0.7412	1.3632
-77	57.19	0.01051	1.516	-101.1	236.9	135.8	0.7424	1.3616
-76	59.45	0.01052	1.460	-100.5	236.4	135.8	0.7436	1.3600
-75	61.79	0.01053	1.407	-100.2	236.0	135.8	0.7447	1.3584
-74	64.21	0.01054	1.356	-99.8	235.6	135.8	0.7458	1.3568
-73	66.72	0.01055	1.306	-99.3	235.1	135.8	0.7469	1.3553
-72	69.33	0.01057	1.257	-98.9	234.8	135.9	0.7481	1.3538
-71	72.03	0.01058	1.209	-98.4	234.3	135.9	0.7493	1.3523
-70	74.82	0.01059	1.162	-98.0	233.9	135.9	0.7506	1.3508
-69.9	75.10	0.01059	1.157	-97.9	233.8	135.9	0.7507	1.3506

TABLE 8-9 (Continued)
SOLID AND SATURATED CARBON DIOXIDE

Temperature °F <i>t</i>	Pressure lb/in. ² <i>p</i>	Volume		Total Heat from -40°			Entropy from -40° plus 1	
		Liquid ft ³ /lb <i>v_f</i>	Vapor ft ³ /lb <i>v_g</i>	Liquid Btu/lb <i>h_f</i>	Latent Btu/lb <i>L</i>	Vapor Btu/lb <i>h_g</i>	Liquid Btu/lb°F <i>s_f</i>	Vapor Btu/lb°F <i>s_g</i>
-69.9	75.1	0.01360	1.1570	-13.7	149.6	135.9	0.9677	1.3506
-68.0	78.6	0.01365	1.1080	-12.9	148.9	136.2	0.9688	1.3490
-66	82.4	0.01369	1.0589	-11.9	148.1	136.2	0.9711	1.3474
-64	86.4	0.01374	1.0125	-11.0	147.3	136.0	0.9734	1.3459
-62	90.5	0.01379	0.9686	-10.1	146.6	136.5	0.9757	1.3444
-60	94.7	0.01384	0.9270	-9.2	145.8	136.6	0.9777	1.3429
-58	99.1	0.01389	0.8875	-8.3	145.0	136.7	0.9802	1.3414
-56	103.7	0.01393	0.8502	-7.4	144.2	136.8	0.9824	1.3399
-54	108.4	0.01398	0.8149	-6.5	143.4	136.9	0.9847	1.3384
-52	113.2	0.01404	0.7812	-5.6	142.7	137.1	0.9869	1.3369
-50	118.2	0.01409	0.7492	-4.7	141.9	137.2	0.9892	1.3354
-48	123.4	0.01414	0.7188	-3.8	141.1	137.3	0.9913	1.3340
-46	128.7	0.01420	0.6899	-2.9	140.3	137.4	0.9935	1.3326
-44	134.2	0.01425	0.6624	-2.0	139.5	137.5	0.9957	1.3312
-42	139.9	0.01431	0.6362	-1.0	138.7	137.7	0.9978	1.3298
-40	145.8	0.01437	0.6113	0.00	137.8	137.8	1.0000	1.3285
-38	151.8	0.01442	0.5876	+ .9	136.9	137.8	1.0021	1.3271
-36	158.0	0.01448	0.5649	1.8	136.2	138.0	1.0043	1.3257
-34	164.4	0.01454	0.5433	2.7	135.4	138.1	1.0064	1.3244
-32	171.0	0.01460	0.5227	3.6	134.5	138.1	1.0085	1.3231
-30	177.8	0.01466	0.5029	4.5	133.7	138.2	1.0107	1.3218
-28	184.8	0.01472	0.4841	5.4	132.9	138.3	1.0127	1.3206
-26	192.0	0.01479	0.4661	6.3	132.0	138.3	1.0148	1.3193
-24	199.4	0.01485	0.4489	7.2	131.2	138.4	1.0169	1.3180
-22	207.0	0.01491	0.4325	8.1	130.3	138.4	1.0190	1.3167
-20	214.9	0.01498	0.4168	9.1	129.4	138.5	1.0212	1.3154
-18	223.0	0.01504	0.4022	10.1	128.5	138.6	1.0232	1.3140
-16	231.2	0.01511	0.3872	11.1	127.6	138.7	1.0252	1.3127
-14	239.6	0.01518	0.3666	12.0	126.7	138.7	1.0272	1.3115
-12	248.3	0.01525	0.3600	12.9	125.8	138.7	1.0283	1.3103
-10	257.3	0.01532	0.3472	13.9	124.8	138.7	1.0314	1.3091
-8	266.5	0.01540	0.3349	14.9	123.9	138.8	1.0334	1.3078
-6	275.9	0.01547	0.3231	15.9	122.9	138.8	1.0355	1.3065
-4	285.4	0.01555	0.3118	16.9	122.0	138.9	1.0376	1.3053
-2	295.3	0.01563	0.3009	17.9	121.0	138.9	1.0397	1.3041
0	305.5	0.01570	0.2904	18.8	120.1	138.9	1.0418	1.3029
2	315.9	0.01579	0.2803	19.8	119.0	138.8	1.0439	1.3017
4	326.5	0.01588	0.2707	20.8	118.0	138.8	1.0460	1.3006
6	337.4	0.01596	0.2614	21.8	116.9	138.7	1.0481	1.2994
8	348.7	0.01605	0.2526	22.9	115.8	138.7	1.0502	1.2982
10	360.2	0.01614	0.2437	24.0	114.7	138.7	1.0536	1.2980
12	371.9	0.01623	0.2354	25.0	113.6	138.6	1.0558	1.2967
14	383.9	0.01632	0.2274	26.1	112.5	138.6	1.0580	1.2955
16	396.2	0.01642	0.2197	27.2	111.3	138.5	1.0602	1.2943
18	408.9	0.01652	0.2121	28.3	110.1	138.4	1.0625	1.2931

TABLE 8-9 (Continued)
SOLID AND SATURATED CARBON DIOXIDE

Temperature °F	Pressure lb/in ² <i>p</i>	Volume		Total Heat from -40°			Entropy from -40° plus 1	
		Liquid ft ³ /lb <i>v_f</i>	Vapor ft ³ /lb <i>v_g</i>	Liquid Btu/lb <i>h_f</i>	Latent Btu/lb <i>L</i>	Vapor Btu/lb <i>h_g</i>	Liquid Btu/lb°F <i>s_f</i>	Vapor Btu/lb°F <i>s_g</i>
20	421.8	0.01663	0.2049	29.4	108.9	138.3	1.0648	1.2919
22	434.0	0.01673	0.1979	30.5	107.7	138.2	1.0672	1.2907
24	448.4	0.01684	0.1912	31.7	106.4	138.1	1.0696	1.2595
26	462.2	0.01695	0.1846	32.9	105.1	138.0	1.0720	1.2883
28	476.3	0.01707	0.1783	34.1	103.8	137.9	1.0744	1.2871
30	490.8	0.01719	0.1722	35.4	102.4	137.8	1.0768	1.2859
32	505.5	0.01731	0.1663	36.7	101.0	137.7	1.0792	1.2844
34	502.6	0.01744	0.1603	37.9	99.5	137.4	1.0875	1.2830
36	536.0	0.01759	0.1550	39.1	98.1	137.2	1.0838	1.2816
38	551.7	0.01773	0.1496	40.4	96.5	136.9	1.0873	1.2801
40	567.8	0.01787	0.1444	41.7	95.0	136.7	1.0884	1.2786
42	584.3	0.01801	0.1393	42.9	93.4	136.3	1.0907	1.2771
44	601.1	0.01817	0.1344	44.3	91.8	136.1	1.0932	1.2756
46	618.2	0.01834	0.1297	45.6	90.1	135.7	1.0958	1.2741
48	635.7	0.01851	0.1250	47.0	88.4	135.4	1.0996	1.2725
50	653.6	0.01868	0.1205	48.4	86.6	135.0	1.1010	1.2709
52	671.9	0.01887	0.1161	49.8	84.7	134.5	1.1036	1.2700
54	690.6	0.01906	0.1117	51.2	82.7	133.9	1.1064	1.2674
56	709.5	0.01927	0.1075	52.6	80.8	133.4	1.1075	1.2657
58	728.8	0.01948	0.1034	54.0	78.7	132.7	1.1104	1.2638
60	748.6	0.01970	0.0994	55.5	76.6	132.1	1.1145	1.2618
62	768.9	0.01994	0.0956	57.0	74.4	131.4	1.1174	1.2597
64	789.4	0.02020	0.0918	57.6	72.0	130.6	1.1202	1.2575
66	810.3	0.02048	0.0880	60.2	69.5	129.7	1.1230	1.2552
68	831.6	0.02079	0.08422	61.9	66.8	128.7	1.1260	1.2526
70	853.4	0.02112	0.08040	63.7	63.8	127.5	1.1292	1.2497
72	875.5	0.02150	0.07657	65.5	60.7	126.2	1.1324	1.2466
74	898.2	0.02192	0.07269	67.3	57.2	124.5	1.1360	1.2432
76	921.3	0.02242	0.06875	69.4	53.4	122.8	1.1399	1.2396
78	944.8	0.02300	0.06473	71.6	49.3	120.9	1.1441	1.2357
80	968.7	0.02370	0.06064	73.9	44.8	118.7	1.1486	1.2314
82	993.0	0.02456	0.05648	76.4	40.2	116.6	1.1532	1.2271
84	1017.7	0.02553	0.05223	79.4	34.5	113.9	1.1582	1.2215
86	1043.0	0.02686	0.04789	83.3	27.1	110.4	1.1646	1.2143
87.8	1066.2	0.03454	0.03454	97.0	0.0	97.0	1.1890	1.1890

TABLE 8·10 α
 LIQUID AND SATURATED DICHLORODIFLUOROMETHANE

Temperature °F	Pressure lb/in. ² <i>p</i>	Volume ft ³ /lb <i>v_g</i>	Density Liquid lb/ft ³ <i>l/v_l</i>	Enthalpy from -40°		Entropy from -40°	
				Liquid Btu/lb <i>h_f</i>	Vapor Btu/lb <i>h_g</i>	Liquid Btu/lb °F <i>s_f</i>	⁴ Vapor Btu/lb °F <i>s_g</i>
-150	0.159	173.0	105.8	-10.9	64.8	-0.0545	0.2187
-140	0.266	106.0	104.8	-18.2	64.9	-0.0492	0.2105
-130	0.419	70.0	103.8	-16.5	65.1	-0.0440	0.2033
-120	0.638	47.0	102.8	-14.8	65.3	-0.0390	0.1967
-110	0.970	32.0	101.8	-13.1	66.2	-0.0340	0.1926
-100	1.44	22.1	100.7	-11.3	67.2	-0.0290	0.1891
-90	2.06	15.9	99.7	-9.6	68.1	-0.0243	0.1857
-80	2.87	11.6	98.7	-7.7	69.2	-0.0193	0.1831
-75	3.40	9.93	98.26	-6.81	69.33	-0.01687	0.1818
-70	3.98	8.60	97.77	-5.86	69.95	-0.01446	0.1807
-65	4.63	7.47	97.20	-4.90	70.51	-0.01205	0.1796
-60	5.37	6.51	96.70	-3.94	71.14	-0.00964	0.1785
-55	6.20	5.70	96.20	-2.97	71.70	-0.00723	0.1775
-50	7.13	5.01	95.71	-1.99	72.29	-0.00482	0.1766
-45	8.17	4.42	95.14	-1.00	72.88	-0.00241	0.1758
-40	9.32	3.911	94.58	0	73.50	0	0.17517
-38	9.82	3.727	94.39	0.40	73.74	0.00094	0.17490
-36	10.34	3.553	94.20	0.81	73.98	0.00188	0.17463
-34	10.87	3.389	93.99	1.21	74.22	0.00282	0.17438
-32	11.43	3.234	93.79	1.62	74.46	0.00376	0.17412
-30	12.02	3.088	93.59	2.03	74.70	0.00471	0.17387
-28	12.62	2.950	93.39	2.44	74.94	0.00565	0.17364
-26	13.26	2.820	93.18	2.85	75.18	0.00659	0.17340
-24	13.90	2.698	92.98	3.25	75.41	0.00753	0.17317
-22	14.58	2.583	92.78	3.66	75.64	0.00846	0.17296
-20	15.28	2.474	92.58	4.07	75.87	0.00940	0.17275
-18	16.01	2.370	92.38	4.48	76.11	0.01033	0.17253
-16	16.77	2.271	92.18	4.89	76.34	0.01126	0.17232
-14	17.55	2.177	91.97	5.30	76.57	0.01218	0.17212
-12	18.37	2.088	91.77	5.72	76.81	0.01310	0.17194
-10	19.20	2.003	91.57	6.14	77.05	0.01403	0.17175
-8	20.08	1.922	91.35	6.57	77.29	0.01496	0.17158
-6	20.98	1.845	91.14	6.99	77.52	0.01589	0.17140
-4	21.91	1.772	90.93	7.41	77.75	0.01682	0.17123
-2	22.87	1.703	90.72	7.83	77.98	0.01775	0.17107
0	23.87	1.637	90.52	8.25	78.21	0.01869	0.17091
2	24.89	1.574	90.31	8.67	78.44	0.01961	0.17075
4	25.96	1.514	90.11	9.10	78.67	0.02052	0.17060
5	26.51	1.485	90.00	9.32	78.79	0.02097	0.17052
6	27.05	1.457	89.88	9.53	78.90	0.02143	0.17045
8	28.18	1.403	89.68	9.96	79.13	0.02235	0.17030
10	29.35	1.351	89.45	10.39	79.36	0.02328	0.17015
12	30.56	1.301	89.24	10.82	79.59	0.02419	0.17001
14	31.80	1.253	89.03	11.26	79.82	0.02510	0.16987
16	33.08	1.207	88.81	11.70	80.05	0.02601	0.16974
18	34.40	1.163	88.58	12.12	80.27	0.02692	0.16961
20	35.75	1.121	88.37	12.55	80.49	0.02783	0.16949
22	37.15	1.081	88.13	13.00	80.72	0.02873	0.16938
24	38.58	1.043	87.91	13.44	80.95	0.02963	0.16926
26	40.07	1.007	87.68	13.88	81.17	0.03053	0.16913
28	41.59	0.973	87.47	14.32	81.39	0.03143	0.16900
30	43.16	0.939	87.24	14.76	81.61	0.03233	0.16887
32	44.77	0.908	87.02	15.21	81.83	0.03323	0.16876
34	46.42	0.877	86.78	15.65	82.05	0.03413	0.16865
36	48.13	0.848	86.55	16.10	82.27	0.03502	0.16854
38	49.88	0.819	86.33	16.55	82.49	0.03591	0.16843
40	51.68	0.792	86.10	17.00	82.71	0.03680	0.16833
42	53.51	0.767	85.88	17.46	82.93	0.03770	0.16823
44	55.40	0.742	85.66	17.91	83.15	0.03859	0.16813
46	57.35	0.718	85.43	18.36	83.36	0.03948	0.16803

TABLE 8·10a (Continued)
SATURATED DICHLORODIFLUOROMETHANE

Temperature °F <i>t</i>	Pressure Abs lb/in. ² <i>P</i>	Volume Vapor ft ³ /lb <i>v_g</i>	Density Liquid lb/ft ³ <i>l/v_f</i>	Enthalpy from -40°		Entropy from -40°	
				Liquid Btu/lb <i>h_f</i>	Vapor Btu/lb <i>h_g</i>	Liquid Btu/lb °F <i>s_f</i>	Vapor Btu/lb °F <i>s_g</i>
48	59.35	0.695	85.19	18.82	83.57	0.040371	0.16794
50	61.39	0.673	84.94	19.27	83.78	0.04126	0.16785
52	63.49	0.652	84.71	19.72	83.99	0.04215	0.16776
54	65.63	0.632	84.50	20.18	84.20	0.04304	0.16767
56	67.84	0.612	84.28	20.64	84.41	0.04392	0.16758
58	70.10	0.593	84.04	21.11	84.62	0.04480	0.16749
60	72.41	0.575	83.78	21.57	84.82	0.04568	0.16741
62	74.77	0.557	83.57	22.03	85.02	0.04657	0.16733
64	77.20	0.540	83.34	22.49	85.22	0.04745	0.16725
66	79.67	0.524	83.10	22.95	85.42	0.04833	0.16717
68	82.24	0.508	82.86	23.42	85.62	0.04921	0.16709
70	84.82	0.493	82.60	23.90	85.82	0.05009	0.16701
72	87.50	0.479	82.37	24.37	86.02	0.05097	0.16693
74	90.20	0.464	82.12	24.84	86.22	0.05185	0.16685
76	93.00	0.451	81.87	25.32	86.42	0.05272	0.16677
78	95.85	0.438	81.62	25.80	86.61	0.05359	0.16669
80	98.76	0.425	81.39	26.28	86.80	0.05446	0.16662
82	101.7	0.413	81.12	26.76	86.99	0.05534	0.16655
84	104.8	0.401	80.87	27.24	87.18	0.05621	0.16648
86	107.9	0.389	80.63	27.72	87.37	0.05708	0.16640
88	111.1	0.378	80.37	28.21	87.56	0.05795	0.16632
90	114.3	0.368	80.11	28.70	87.74	0.05882	0.16624
92	117.7	0.357	79.86	29.19	87.92	0.05969	0.16616
94	121.0	0.347	79.60	29.68	88.10	0.06056	0.16608
96	124.5	0.338	79.32	30.18	88.28	0.06143	0.16600
98	128.0	0.328	79.06	30.67	88.45	0.06230	0.16592
100	131.6	0.319	78.80	31.16	88.62	0.06316	0.16584
102	135.3	0.310	78.54	31.65	88.79	0.06403	0.16576
104	139.0	0.302	78.27	32.15	88.95	0.06490	0.16568
106	142.8	0.293	78.00	32.65	89.11	0.06577	0.16560
108	146.8	0.285	77.73	33.15	89.27	0.06663	0.16551
110	150.7	0.277	77.46	33.65	89.43	0.06749	0.16542
112	154.8	0.269	77.18	34.15	89.58	0.06836	0.16533
114	158.9	0.262	76.89	34.65	89.73	0.06922	0.16524
116	163.1	0.254	76.60	35.15	89.87	0.07008	0.16515
118	167.4	0.247	76.32	35.65	90.01	0.07094	0.16505
120	171.8	0.240	76.02	36.16	90.15	0.07180	0.16495
122	176.2	0.233	75.72	36.66	90.28	0.07266	0.16484
124	180.8	0.227	75.40	37.16	90.40	0.07352	0.16473
126	185.4	0.220	75.10	37.67	90.52	0.07437	0.16462
128	190.1	0.214	74.78	38.18	90.64	0.07522	0.16450
130	194.9	0.208	74.46	38.69	90.76	0.07607	0.16438
132	199.8	0.202	74.13	39.19	90.86	0.07691	0.16425
134	204.8	0.196	73.81	39.70	90.96	0.07775	0.16411
136	209.9	0.191	73.46	40.21	91.06	0.07858	0.16396
138	215.0	0.185	73.10	40.72	91.15	0.07941	0.16380
140	220.2	0.180	72.73	41.24	91.24	0.08024	0.16363
150	248.0	0.158	71.04	43.8	91.7		
160	276.0	0.139	69.23	46.75	92.1		
170	310.0	0.121	67.23	49.8	92.4		
180	344.0	0.106	65.11	52.8	92.5		
190	383.0	0.093	62.6	† 56.0	92.4		
200	431.0	0.081	59.9	59.5	92.1		
210	468.0	0.069	56.8	63.75	91.3		
220	515.0	0.057	52.8	68.75	90.0		
230	570.0	0.041	45.4	76.25	87.0		
232.7*	582.0*	0.0288	34.7	82.5	82.5		

* Critical temperature and pressure.
† Extrapolated data — not calculated.

TABLE 8-10b (Continued)
SUPERHEATED DICHLORODIFLUOROMETHANE

Tem- perature °F	Abs. Press., 40 lb./in. ² Sat'n Temp., 25.9° F			Abs. Press., 50 lb./in. ² Sat'n Temp., 33.3° F			Abs. Press., 60 lb./in. ² Sat'n Temp., 43.7° F			Abs. Press., 70 lb./in. ² Sat'n Temp., 57.9° F		
	v	h	s	v	h	s	v	h	s	v	h	s
(at sat'n)	(1.009)	(81.16)	(0.16914)	(0.817)	(82.58)	(0.16841)	(0.698)	(83.65)	(0.16791)	(0.594)	(84.61)	(0.16749)
50	1.070	84.65	0.17612	0.842	84.24	0.17187	0.690	83.83	0.16829			
60	1.218	86.11	0.17896	0.863	85.72	0.17475	0.708	85.33	0.17120	0.507	84.94	0.16810
70	1.120	87.60	0.18178	0.884	87.22	0.17660	0.726	86.84	0.17407	0.612	86.44	0.17087
80	1.144	89.09	0.18455	0.904	88.72	0.18040	0.743	88.35	0.17689	0.628	87.96	0.17352
90	1.169	90.58	0.18731	0.924	90.23	0.18317	0.760	89.87	0.17968	0.643	89.49	0.17665
100	1.194	92.09	0.19004	0.944	91.75	0.18591	0.778	91.41	0.18246	0.658	91.03	0.17943
110	1.218	93.62	0.19272	0.964	93.29	0.18862	0.795	92.96	0.18519	0.673	92.59	0.18219
120	1.242	95.15	0.19538	0.984	94.83	0.19132	0.812	94.51	0.18789	0.689	94.16	0.18493
130	1.267	96.70	0.19803	1.004	96.39	0.19397	0.829	96.07	0.19056	0.704	95.75	0.18763
140	1.291	98.26	0.20066	1.024	97.96	0.19662	0.846	97.65	0.19323	0.719	97.34	0.19030
150	1.315	99.83	0.20325	1.004	99.54	0.19923	0.863	99.24	0.19585	0.733	98.94	0.19293
160	1.340	101.42	0.20583	1.064	101.14	0.20182	0.880	100.84	0.19846	0.748	100.54	0.19555
170	1.364	103.02	0.20838	1.084	102.75	0.20439	0.897	102.45	0.20104	0.763	102.16	0.19814
180	1.388	104.63	0.21092	1.103	104.36	0.20694	0.913	104.07	0.20361	0.777	103.80	0.20071
190	1.412	106.25	0.21343	1.123	105.98	0.20944	0.930	105.71	0.20613	0.792	105.45	0.20325
200	1.435	107.88	0.21592	1.142	107.62	0.21196	0.946	107.36	0.20865	0.806	107.10	0.20579
210	1.459	109.52	0.21840	1.162	109.28	0.21444	0.962	109.02	0.21113	0.820	108.74	0.20828
220	1.482	111.17	0.22085	1.181	110.95	0.21691	0.970	110.69	0.21367	0.833	110.49	0.21079
230	1.506	112.84	0.22329	1.200	112.62	0.21935	0.985	112.37	0.21607	0.849	112.13	0.21325
240	1.530	114.52	0.22572	1.220	114.31	0.22179	1.012	114.06	0.21853	0.863	113.83	0.21570

TABLE 8-10b (Continued)
SUPERHEATED DICHLORODIFLUOROMETHANE

Temperature °F	Abs. Press., 80 lb./in. ² Sat'n Temp., 66.3° F			Abs. Press., 90 lb./in. ² Sat'n Temp., 73.9° F			Abs. Press., 100 lb./in. ² Sat'n Temp., 80.9° F			Abs. Press., 110 lb./in. ² Sat'n Temp., 87.3° F		
	v	h	s	v	h	s	v	h	s	v	h	s
(at sat'n)	(0.581)	(85.4)	(0.16716)	(0.465)	(86.21)	(0.16686)	(0.419)	(86.89)	(0.16669)	(0.382)	(87.50)	(0.16636)
80				0.473	87.18	0.16862				0.385	87.91	0.16711
90				0.486	88.74	0.17149				0.396	89.51	0.17001
100	0.568	90.68	0.17675	0.499	90.31	0.17433	0.442	89.93	0.17210	0.407	91.12	0.17287
110	0.582	92.26	0.17954	0.511	91.89	0.17713	0.454	91.54	0.17493	0.417	92.74	0.17568
120	0.596	93.84	0.18229	0.523	93.48	0.17990	0.465	93.15	0.17773			
130	0.609	95.43	0.18500	0.535	95.08	0.18262	0.477	94.76	0.18049	0.428	94.37	0.17845
140	0.623	97.03	0.18771	0.547	96.69	0.18533	0.488	96.37	0.18321	0.438	96.01	0.18122
150	0.636	98.64	0.19035	0.559	98.31	0.18799	0.499	97.99	0.18590	0.449	97.66	0.18394
160	0.649	100.26	0.19298	0.571	99.94	0.19065	0.510	99.63	0.18856	0.459	99.31	0.18660
170	0.662	101.88	0.19558	0.584	101.58	0.19327	0.521	101.28	0.19120	0.469	100.97	0.18924
180	0.675	103.52	0.19817	0.596	103.23	0.19588	0.531	102.94	0.19381	0.479	102.64	0.19187
190	0.688	105.18	0.20073	0.607	104.89	0.19845	0.542	104.61	0.19638	0.489	104.32	0.19447
200	0.701	106.84	0.20328	0.619	106.56	0.20101	0.553	106.29	0.19894	0.499	106.01	0.19706
210	0.714	108.51	0.20580	0.630	108.24	0.20353	0.563	107.98	0.20148	0.509	107.71	0.19962
220	0.726	110.19	0.20828	0.642	109.93	0.20603	0.574	109.68	0.20401	0.519	109.42	0.20216
230	0.739	111.88	0.21076	0.653	111.63	0.20852	0.585	111.39	0.20650	0.528	111.14	0.20464
240	0.751	113.58	0.21321	0.665	113.35	0.21100	0.595	113.11	0.20899	0.538	112.87	0.20712
250	0.764	115.30	0.21566	0.677	115.06	0.21348	0.606	114.84	0.21145	0.548	114.61	0.20959
260	0.777	117.03	0.21809	0.689	116.81	0.21593	0.616	116.58	0.21389	0.557	116.36	0.21205
270	0.789	118.78	0.22049	0.701	118.58	0.21835	0.626	118.33	0.21631	0.567	118.12	0.21448
280	0.802	120.54	0.22289	0.713	120.34	0.22071	0.636	120.10	0.21870	0.576	119.89	0.21690
290	0.814	122.30	0.22525	0.725	122.10	0.22307	0.646	121.88	0.22108	0.586	121.68	0.21930

TABLE 8-10b (Continued)
SUPERHEATED DICHLORODIFLUOROMETHANE

Temperature °F	Abs. Press., 120 lb./in. ² Sat'n Temp., 93.4° F			Abs. Press., 130 lb./in. ² Sat'n Temp., 99.1° F			Abs. Press., 140 lb./in. ² Sat'n Temp., 104.5° F		
	v	h	s	v	h	s	v	h	s
(at sat'n)	(0.550)	(88.05)	(0.16610)	(0.523)	(88.547)	(0.16688)	(0.498)	(88.99)	(0.16866)
100	0.357	89.13	0.16803	0.324	88.69	0.16615	0.304	89.92	0.16725
110	0.367	90.75	0.17090	0.333	90.33	0.16905	0.314	91.60	0.17021
120	0.377	92.38	0.17374	0.343	91.98	0.17193	0.323	93.28	0.17306
130	0.387	94.01	0.17654	0.353	93.64	0.17476	0.332	94.96	0.17590
140	0.397	95.65	0.17932	0.362	95.30	0.17756	0.341	96.65	0.17868
150	0.407	97.30	0.18207	0.371	96.97	0.18030	0.350	98.34	0.18142
160	0.417	98.96	0.18474	0.380	98.65	0.18392	0.358	100.03	0.18412
170	0.426	100.63	0.18743	0.389	100.34	0.18571	0.366	101.72	0.18678
180	0.436	102.31	0.19011	0.398	102.04	0.18839	0.374	103.42	0.18941
190	0.445	104.00	0.19271	0.407	103.74	0.19102	0.383	105.14	0.19205
200	0.454	105.70	0.19529	0.416	105.45	0.19362	0.391	106.86	0.19466
210	0.463	107.41	0.19785	0.424	107.16	0.19620	0.399	108.59	0.19724
220	0.472	109.13	0.20041	0.433	108.89	0.19877	0.407	110.33	0.19976
230	0.482	110.86	0.20294	0.442	110.62	0.20130	0.415	112.09	0.20229
240	0.491	112.60	0.20545	0.450	112.36	0.20382	0.423	113.85	0.20479
250	0.500	114.35	0.20792	0.458	114.11	0.20629	0.431	115.63	0.20728
260	0.508	116.11	0.21035	0.467	115.87	0.20876	0.439	117.42	0.20974
270	0.517	117.88	0.21279	0.475	117.64	0.21122	0.447	119.22	0.21219
280	0.526	119.66	0.21521	0.483	119.42	0.21364	0.455	121.03	0.21461
290	0.534	121.45	0.21760	0.492	121.21	0.21605			

TABLE 8-11a
 SATURATED SULPHUR DIOXIDE

Temperature °F <i>t</i>	Pressure lb/in. ² <i>p</i>	Volume ft ³ /lb <i>v_g</i>	Density lb/ft ³ <i>l/v_g</i>	Total Heat above -40°			Entropy from -40°		
				Liquid Btu/lb <i>h_f</i>	Latent Btu/lb <i>L</i>	Vapor Btu/lb <i>h_g</i>	Liquid <i>s_f</i>	Evap. $\frac{L}{T}$	Vapor <i>s_g</i>
-40	3.136	22.42	95.79	0.00	178.61	178.61	0.00000	0.42562	0.42562
-30	4.331	16.56	94.94	2.93	176.97	179.90	0.00674	0.41190	0.41864
-20	5.883	12.42	94.10	5.98	175.09	181.07	0.01366	0.39826	0.41192
-10	7.863	9.44	93.27	9.16	172.97	182.13	0.02075	0.38469	0.40544
0	10.35	7.280	92.42	12.44	170.63	183.07	0.02795	0.37122	0.39917
2	10.91	6.923	92.25	13.12	170.13	183.25	0.02941	0.36853	0.39794
4	11.60	6.584	92.08	13.78	169.63	183.41	0.03084	0.36586	0.39670
5	11.81	6.421	92.00	14.11	169.38	183.49	0.03155	0.36454	0.39609
6	12.12	6.266	91.91	14.45	169.12	183.57	0.03228	0.36319	0.39547
8	12.75	5.967	91.74	15.13	168.60	183.73	0.03373	0.36053	0.39426
10	13.42	5.682	91.58	15.80	168.07	183.87	0.03519	0.35787	0.39306
11	13.77	5.5-8	91.49	16.14	167.80	183.94	0.03592	0.35654	0.39246
12	14.12	5.4.7	91.41	16.48	167.53	184.01	0.03664	0.35521	0.39185
13	14.48	5.289	91.33	16.81	167.26	184.07	0.03737	0.35388	0.39125
14	14.84	5.164	91.24	17.15	166.97	184.14	0.03808	0.35257	0.39065
15	15.21	5.042	91.16	17.49	166.72	184.21	0.03880	0.35125	0.39005
16	15.59	4.926	91.07	17.84	166.44	184.28	0.03953	0.34993	0.38946
17	15.98	4.812	90.98	18.18	166.16	184.34	0.04026	0.34861	0.38887
18	16.37	4.701	90.89	18.52	165.88	184.40	0.04098	0.34729	0.38827
19	16.77	4.593	90.80	18.86	165.60	184.46	0.04169	0.34598	0.38767
20	17.18	4.487	90.71	19.20	165.32	184.52	0.04241	0.34466	0.38707
21	17.60	4.386	90.62	19.55	165.03	184.58	0.04313	0.34335	0.38648
22	18.03	4.287	90.53	19.90	164.74	184.64	0.04385	0.34204	0.38589
23	18.46	4.190	90.44	20.24	164.45	184.69	0.04457	0.34073	0.38530
24	18.89	4.096	90.33	20.58	164.16	184.74	0.04528	0.33943	0.38471
25	19.34	3.994	90.24	20.92	163.87	184.79	0.04600	0.33812	0.38412
26	19.80	3.915	90.15	21.26	163.58	184.84	0.04671	0.33683	0.38354
27	20.26	3.829	90.06	21.61	163.28	184.89	0.04743	0.33553	0.38296
28	20.73	3.744	89.96	21.96	162.98	184.94	0.04814	0.33422	0.38237
29	21.21	3.566	89.86	22.30	162.68	184.98	0.04886	0.33292	0.38178
30	21.70	3.581	89.76	22.64	162.38	185.02	0.04956	0.33163	0.38119
31	22.20	3.503	89.67	22.98	162.08	185.06	0.05027	0.33034	0.38061
32	22.71	3.437	89.58	23.33	161.77	185.10	0.05099	0.32904	0.38003
33	23.23	3.355	89.48	23.68	161.46	185.14	0.05171	0.32774	0.37945
34	23.75	3.283	89.39	24.03	161.15	185.18	0.05242	0.32645	0.37887
35	24.28	3.212	89.29	24.38	160.84	185.22	0.05312	0.32517	0.37829
40	27.10	2.887	88.81	26.12	159.25	185.37	0.05668	0.31873	0.37541
45	30.15	2.601	88.34	27.86	157.62	185.48	0.06020	0.31234	0.37264
50	33.45	2.348	87.87	29.61	155.95	185.56	0.06370	0.30599	0.36989
55	37.05	2.124	87.41	31.36	154.24	185.60	0.06715	0.29971	0.36686
60	40.93	1.926	86.95	33.10	152.49	185.59	0.07060	0.29345	0.36405
65	45.13	1.749	86.50	34.84	150.70	185.54	0.07401	0.28724	0.36125
70	49.62	1.590	86.02	36.58	148.88	185.46	0.07736	0.28110	0.35846
75	54.47	1.448	85.52	38.32	147.02	185.34	0.08070	0.27498	0.35568
80	59.68	1.321	85.03	40.05	145.12	185.17	0.08399	0.26897	0.35291
81	60.77	1.297	84.93	40.39	144.74	185.13	0.08462	0.26772	0.35234
82	61.88	1.274	84.84	40.73	144.36	185.09	0.08525	0.26652	0.35177
83	63.01	1.253	84.74	41.08	143.97	185.05	0.08589	0.26532	0.35121
84	64.14	1.229	84.64	41.43	143.58	185.01	0.08653	0.26412	0.35065
85	65.28	1.207	84.54	41.78	143.19	184.97	0.08718	0.26291	0.35009
86	66.45	1.185	84.44	42.12	142.80	184.92	0.08783	0.26171	0.34954
87	67.64	1.164	84.35	42.46	142.41	184.87	0.08847	0.26052	0.34899
88	68.84	1.144	84.25	42.80	142.02	184.82	0.08910	0.25933	0.34843
89	70.04	1.124	84.15	43.15	141.62	184.77	0.08974	0.25813	0.34787
90	71.25	1.104	84.05	43.50	141.22	184.72	0.09038	0.25693	0.34731
95	77.60	1.011	83.57	45.20	139.23	184.43	0.09349	0.25103	0.34452
100	84.52	0.9262	83.07	46.90	137.20	184.10	0.09657	0.24516	0.34173
110	99.76	0.7804	82.03	50.28	133.05	183.31	0.10264	0.23357	0.33611
120	120.93	0.6598	80.90	53.58	128.78	182.36	0.10829	0.22217	0.33046
140	158.61	0.4758	78.61	60.04	119.90	179.94	0.11893	1.19990	0.31888

TABLE 8-11b
SUPERHEATED SULPHUR DIOXIDE

Temperature, °F	Volume ft ³ /lb	Total Heat Btu/lb	Entropy Btu/lb °F	Volume ft ³ /lb	Total Heat Btu/lb	Entropy Btu/lb °F
	<i>v</i>	<i>h</i>	<i>s</i>	<i>v</i>	<i>h</i>	<i>s</i>
	Abs. Press., 10 lb/in. ² Gage Press., 9.6 in. vac. (Sat'n Temp., -1.34° F)			Abs. Press., 15 lb/in. ² Gage Press., 0.30 lb/in. ² (Sat'n Temp., 14.43° F)		
(at sat'n)				(5.110)	(184.17)	(0.39081)
20	7.939	186.7	0.40802	5.192	185.4	0.39270
30	8.030	188.4	0.41159	5.333	187.3	0.39672
40	8.316	190.1	0.41505	5.470	189.2	0.40054
50	8.500	191.8	0.41837	5.604	191.0	0.40424
60	8.681	193.5	0.42161	5.734	192.8	0.40777
70	8.860	195.2	0.42480	5.862	195.6	0.41116
80	9.038	196.9	0.42795	5.988	196.4	0.41443
90	9.214	198.6	0.43104	6.112	198.2	0.41765
100	9.389	200.3	0.43407	6.233	199.9	0.42076
110	9.563	202.0	0.43705	6.353	201.6	0.42383
120	9.736	203.7	0.43997	6.471	203.3	0.42682
130	9.908	205.4	0.44283	6.588	205.6	0.42976
140	10.08	207.1	0.44565	6.705	206.7	0.43264
150	10.25	208.8	0.44842	6.821	208.4	0.43548
160	10.42	210.5	0.45116	6.937	210.1	0.43825
170	10.59	212.2	0.45296	7.052	211.8	0.44097
180	10.76	213.8	0.4551	7.167	213.5	0.44366
190	10.93	215.4	0.45913	7.282	215.2	0.44630
200	11.10	217.0	0.46171	7.396	216.9	0.44889
	Abs. Press., 20 lb/in. ² Gage Press., 5.30 lb/in. ² (Sat'n Temp., 26.44° F)			Abs. Press., 25 lb/in. ² Gage Press., 10.30 lb/in. ² (Sat'n Temp., 36.33° F)		
(at sat'n)	(3.878)	(184.80)	(0.38329)	(3.125)	(185.26)	(0.37764)
40	4.035	187.8	0.38959	3.181	186.1	0.37927
50	4.145	189.8	0.39346	3.273	188.4	0.38372
60	4.251	191.8	0.39719	3.363	190.6	0.38795
70	4.354	193.7	0.40080	3.451	192.7	0.39198
80	4.454	195.6	0.40429	3.536	194.7	0.39582
90	4.552	197.5	0.40758	3.618	196.7	0.39945
100	4.648	199.3	0.41093	3.696	198.6	0.40291
110	4.742	201.1	0.41415	3.772	200.5	0.40625
120	4.834	202.9	0.41726	3.848	202.4	0.40949
130	4.925	204.7	0.42027	3.923	204.2	0.41261
140	5.015	206.5	0.42322	3.998	206.0	0.41568
150	5.104	208.2	0.42613	4.073	207.8	0.41866
160	5.193	209.9	0.42898	4.145	209.6	0.42156
170	5.281	211.6	0.43176	4.216	211.4	0.42439
180	5.369	213.3	0.43449	4.287	213.2	0.42717
190	5.456	215.0	0.43716	4.358	215.0	0.42988
200	5.542	216.7	0.43977	4.428	216.7	0.43253
210	5.629	218.4	0.44234	4.498	218.4	0.43413

TABLE 8-11b (Continued)
SUPERHEATED SULPHUR DIOXIDE

<i>t</i>	Volume ft ³ /lb <i>v</i>	Total Heat Btu/lb <i>h</i>	Entropy Btu/lb °F <i>s</i>	Volume ft ³ /lb <i>v</i>	Total Heat Btu/lb <i>h</i>	Entropy Btu/lb °F <i>s</i>	Volume ft ³ /lb <i>v</i>	Total Heat Btu/lb <i>h</i>	Entropy Btu/lb °F <i>s</i>	Volume ft ³ /lb <i>v</i>	Total Heat Btu/lb <i>h</i>	Entropy Btu/lb °F <i>s</i>
	Abs. Press., 40 lb/in. ² (Sat'n Temp., 58.83°F)			Abs. Press., 50 lb/in. ² (Sat'n Temp., 70.40°F)			Abs. Press., 60 lb/in. ² (Sat'n Temp., 80.29°F)			Abs. Press., 70 lb/in. ² (Sat'n Temp., 83.97°F)		
(sat)	(1.970)	(185.00)	(0.36470)	(1.577)	(185.45)	(0.35353)	(1.5144)	(185.16)	(0.35272)	(1.126)	(184.77)	(0.34789)
100	2.246	196.1	0.38415	1.775	193.9	0.37369	1.238	191.4	0.36403	1.181	187.6	0.35443
110	2.304	198.2	0.38910	1.825	196.4	0.37815	1.246	194.3	0.36906	1.228	191.6	0.36020
120	2.360	200.4	0.39183	1.872	198.5	0.38227	1.493	197.0	0.37375	1.272	194.8	0.36545
130	2.413	202.5	0.39541	1.917	201.1	0.38608	1.499	199.3	0.37810	1.313	197.6	0.37028
140	2.465	204.6	0.39881	1.961	203.4	0.38968	1.514	201.5	0.38217	1.352	200.3	0.37478
160	2.515	206.5	0.40209	2.003	205.4	0.39353	1.563	204.2	0.38603	1.389	202.9	0.37897
180	2.565	208.5	0.40525	2.044	207.5	0.39691	1.608	206.5	0.38963	1.424	205.3	0.38291
170	2.614	210.4	0.40831	2.084	209.6	0.40015	1.650	208.6	0.39310	1.457	207.6	0.38662
180	2.662	212.3	0.41127	2.123	211.6	0.40237	1.689	210.7	0.39626	1.489	209.9	0.39014
190	2.709	214.2	0.41416	2.163	213.4	0.40428	1.726	212.8	0.39906	1.521	212.0	0.39348
200	2.755	216.0	0.41694	2.199	215.4	0.40591	1.751	214.8	0.40200	1.551	214.1	0.39670
210	2.800	217.9	0.41966	2.237	217.3	0.41200	1.785	216.8	0.40554	1.580	216.1	0.39978
220	2.845	219.7	0.42233	2.274	219.2	0.41477	1.819	218.7	0.40839	1.608	218.1	0.40275
230	2.889	221.5	0.42494	2.311	221.0	0.41748	1.853	220.7	0.41118	1.636	220.1	0.40564
240	2.933	223.3	0.42751	2.347	223.0	0.42015	1.885	222.6	0.41391	1.664	222.1	0.40845
250	2.977	225.1	0.43007	2.383	224.9	0.42275	1.917	224.5	0.41657	1.691	224.1	0.41120
260	3.021	227.0	0.43262	2.418	226.7	0.42535	1.948	226.4	0.41917	1.718	226.0	0.41389

TABLE 8-12
METHYLENE CHLORIDE (CARRENE) (CH₂Cl₂)

Temperature °F <i>t</i>	Pressure lb/in. ² <i>p</i>	Volume ft ³ /lb <i>v_g</i>	Heat Content from 0°			Entropy from 0°	
			Liquid Btu/lb <i>h_f</i>	Latent Btu/lb <i>L</i>	Vapor Btu/lb <i>h_g</i>	Liquid Btu/lb °F <i>s_f</i>	Vapor Btu/lb °F <i>s_g</i>
-10	0.69	81.3	-3.4	165.2	161.8	-0.0072	0.3606
0	0.98	58.6	0.0	163.2	163.2	00	.3546
10	1.38	42.55	3.4	161.0	164.4	.0072	.3502
20	1.92	31.40	6.8	158.8	165.6	.0151	.3461
25	2.24	27.0	8.6	157.8	166.4	.0188	.3444
30	2.56	23.90	10.2	156.7	166.9	.0222	.3425
35	2.95	21.10	11.9	155.6	167.5	.0256	.3402
40	3.38	18.60	13.6	154.4	168.0	.0285	.3377
50	4.36	14.68	17.0	152.0	169.0	.0350	.3335
60	5.52	11.68	20.4	149.7	170.1	.0410	.3292
70	7.07	9.38	23.8	147.2	171.0	.0406	.3246
80	8.81	7.50	27.2	144.8	172.0	.0520	.3202
90	10.87	6.20	30.6	142.3	172.9	.0570	.3160
100	13.25	5.14	34.0	139.7	173.7	.0620	.3113
110	16.40	4.31	37.4	137.0	174.4	.0652	.3058
120	19.20	3.65	40.8	134.2	175.0	.0714	.3031
130	22.69	3.10	44.2	131.3	175.5	.0756	.2983
140	26.79	2.69	47.6	128.4	176.0	.0795	.2935

TABLE 8-13a

SATURATED TRICHLOROMONOFUOROMETHANE (CCl₃F)
(Freon-11)

Temperature °F <i>t</i>	Pressure lb/in. ² <i>p</i>	Volume Vapor ft ³ /lb <i>v_g</i>	Density Liquid lb/ft ³ <i>l/v_l</i>	Heat Content (Enthalpy) from -40°			Entropy from -40°	
				Liquid Btu/lb <i>h_l</i>	Latent Btu/lb <i>h_{fg}</i>	Vapor Btu/lb <i>h_g</i>	Liquid Btu/lb °F <i>s_l</i>	Vapor Btu/lb °F <i>s_g</i>
-40	0.7391	44.21	101.25	0.00	87.48	87.48	0.0000	0.2085
-30	1.034	32.33	100.52	1.97	86.70	88.67	0.0046	0.2064
-20	1.420	24.06	99.77	3.94	85.93	89.87	0.0091	0.2046
-10	1.920	18.17	99.03	5.91	85.16	91.07	0.0136	0.2030
0	2.555	13.94	98.27	7.80	84.38	92.27	0.0179	0.2015
2	2.700	13.24	98.11	8.28	84.23	92.51	0.0188	0.2013
4	2.852	12.58	97.96	8.68	84.07	92.75	0.0197	0.2010
5	2.931	12.27	97.88	8.88	84.00	92.88	0.0201	0.2009
6	3.012	11.96	97.81	9.08	83.92	93.00	0.0205	0.2008
8	3.179	11.38	97.65	9.48	83.76	93.24	0.0213	0.2005
10	3.352	10.83	97.50	9.88	83.60	93.48	0.0222	0.2003
12	3.534	10.31	97.34	10.28	83.45	93.72	0.0231	0.2000
14	3.724	9.823	97.19	10.68	83.29	93.97	0.0239	0.1998
16	3.923	9.359	97.03	11.07	83.14	94.21	0.0248	0.1996
18	4.129	8.925	96.88	11.47	82.98	94.45	0.0256	0.1993
20	4.342	8.519	96.72	11.87	82.82	94.69	0.0264	0.1991
22	4.567	8.129	96.57	12.27	82.66	94.94	0.0273	0.1989
24	4.801	7.760	96.41	12.68	82.50	95.18	0.0281	0.1987
26	5.043	7.414	96.25	13.08	82.34	95.42	0.0289	0.1985
28	5.294	7.087	96.10	13.48	82.18	95.66	0.0297	0.1983
30	5.557	6.776	95.94	13.88	82.03	95.91	0.0306	0.1981
32	5.830	6.481	95.78	14.28	81.87	96.15	0.0314	0.1979
34	6.115	6.200	95.62	14.68	81.71	96.39	0.0322	0.1977
36	6.411	5.934	95.46	15.08	81.55	96.63	0.0330	0.1976
38	6.718	5.682	95.30	15.49	81.38	96.87	0.0338	0.1974
40	7.032	5.447	95.14	15.89	81.22	97.11	0.0346	0.1972
50	8.804	4.421	94.34	17.92	80.40	98.32	0.0386	0.1964
60	10.90	3.626	93.53	19.96	79.57	99.53	0.0426	0.1958
70	13.40	2.993	92.71	22.02	78.71	100.73	0.0465	0.1951
80	16.31	2.492	91.88	24.09	77.84	101.93	0.0504	0.1947
90	19.69	2.091	91.04	26.18	76.95	103.12	0.0542	0.1942
100	23.60	1.765	90.19	28.27	76.03	104.30	0.0580	0.1938
110	28.09	1.499	89.34	30.40	75.08	105.47	0.0617	0.1935
120	33.20	1.281	88.47	32.53	74.10	106.63	0.0654	0.1933

TABLE 8-13b
 SUPERHEATED TRICHLOROMONOFUOROMETHANE
 (Freon-11)

Temperature °F	Abs. Press., 0.7 lb/in. ² Sat'n Temp., -41.6° F			Abs. Press., 1.0 lb/in. ² Sat'n Temp., -31.0° F			Abs. Press., 1.4 lb/in. ² Sat'n Temp., -20.5° F		
	<i>v</i>	<i>h</i>	<i>s</i>	<i>v</i>	<i>h</i>	<i>s</i>	<i>v</i>	<i>h</i>	<i>s</i>
(at sat'n)	(46.54)	(87.29)	(0.2090)	(33.37)	(88.55)	(0.2066)	(24.39)	(89.81)	(0.2047)
-40	46.69	87.49	0.2093						
-30	47.81	88.69	0.2121	33.43	88.67	0.2069			
-20	48.93	89.91	0.2149	34.22	89.90	0.2097	24.41	89.87	0.2049
-10	50.06	91.14	0.2177	35.00	91.13	0.2125	24.97	91.10	0.2076
0	51.18	92.38	0.2204	35.79	92.36	0.2153	25.53	92.34	0.2104
10	52.30	93.63	0.2231	36.57	93.61	0.2180	26.09	93.59	0.2131
20	53.42	94.89	0.2258	37.36	94.87	0.2206	26.65	94.85	0.2157
30	54.54	96.16	0.2284	38.14	96.14	0.2232	27.21	96.12	0.2183
40	55.66	97.44	0.2310	38.93	97.42	0.2258	27.78	97.40	0.2209
50	56.77	98.73	0.2335	39.71	98.71	0.2284	28.34	98.69	0.2235
60	57.89	100.03	0.2361	40.50	100.02	0.2309	28.90	100.00	0.2260
70	59.01	101.34	0.2386	41.28	101.33	0.2334	29.46	101.31	0.2285
80	60.13	102.67	0.2411	42.07	102.66	0.2359	30.02	102.64	0.2310
90	61.25	104.00	0.2435	42.85	103.99	0.2383	30.58	103.97	0.2334
Temperature °F	Abs. Press., 1.8 lb/in. ² Sat'n Temp., -12.2° F			Abs. Press., 2.2 lb/in. ² Sat'n Temp., -5.3° F			Abs. Press., 2.4 lb/in. ² Sat'n Temp., -2.2° F		
(at sat'n)	(19.30)	(90.81)	(0.2033)	(16.02)	(91.63)	(0.2023)	(14.77)	(92.00)	(0.2018)
-10	19.39	91.08	0.2040						
0	19.83	92.32	0.2067	16.20	92.29	0.2038	14.84	92.28	0.2025
10	20.27	93.57	0.2094	16.56	93.54	0.2065	15.17	93.53	0.2052
20	20.71	94.83	0.2120	16.92	94.81	0.2091	15.50	94.80	0.2078
30	21.14	96.10	0.2147	17.28	96.08	0.2117	15.83	96.07	0.2105
40	21.58	97.38	0.2173	17.64	97.36	0.2143	16.16	97.35	0.2131
50	22.02	98.67	0.2198	17.99	98.65	0.2169	16.49	98.64	0.2156
60	22.46	99.98	0.2223	18.35	99.96	0.2194	16.82	99.95	0.2181
70	22.89	101.29	0.2248	18.71	101.27	0.2219	17.14	101.26	0.2206
80	23.33	102.62	0.2273	19.07	102.60	0.2244	17.47	102.59	0.2231
90	23.77	103.95	0.2298	19.43	103.93	0.2269	17.80	103.92	0.2256
100	24.20	105.29	0.2322	19.79	105.27	0.2293	18.13	105.27	0.2280
110	24.64	106.65	0.2346	20.14	106.63	0.2317	18.46	106.63	0.2304
120	25.08	108.02	0.2370	20.50	108.00	0.2340	18.79	108.00	0.2328
130	25.51	109.39	0.2393	20.86	109.37	0.2364	19.11	109.37	0.2351
140	25.95	110.77	0.2417	21.22	110.75	0.2387	19.44	110.75	0.2375
Temperature °F	Abs. Press., 3.0 lb/in. ² Sat'n Temp., 5.9° F			Abs. Press., 4.0 lb/in. ² Sat'n Temp., 16.7° F			Abs. Press., 5.0 lb/in. ² Sat'n Temp., 25.6° F		
(at sat'n)	(12.01)	(92.98)	(0.2008)	(9.194)	(94.30)	(0.1995)	(7.475)	(95.38)	(0.1985)
10	12.11	93.50	0.2019						
20	12.38	94.76	0.2046	9.256	94.71	0.2003			
30	12.64	96.04	0.2072	9.455	95.99	0.2030	7.543	95.94	0.1997
40	12.91	97.32	0.2098	9.655	97.27	0.2056	7.703	97.22	0.2023
50	13.17	98.61	0.2123	9.853	98.56	0.2081	7.863	98.51	0.2048
60	13.43	99.92	0.2149	10.05	99.87	0.2107	8.023	99.82	0.2074
70	13.70	101.23	0.2174	10.25	101.18	0.2132	8.182	101.14	0.2099
80	13.96	102.56	0.2199	10.45	102.51	0.2157	8.341	102.47	0.2124
90	14.22	103.90	0.2223	10.65	103.45	0.2181	8.500	103.80	0.2148
100	14.49	105.24	0.2247	10.84	105.20	0.2205	8.659	105.16	0.2173
110	14.75	106.60	0.2271	11.04	106.56	0.2229	8.818	106.51	0.2197
120	15.01	107.97	0.2295	11.24	107.93	0.2253	8.977	107.88	0.2220
130	15.28	109.34	0.2319	11.44	109.30	0.2277	9.135	109.26	0.2244
140	15.54	110.72	0.2342	11.64	110.68	0.2300	9.294	110.65	0.2267
150				11.83	112.08	0.2323	9.452	112.05	0.2290
160				12.03	113.49	0.2346	9.610	113.45	0.2313
170				12.23	114.91	0.2369	9.769	114.87	0.2336
180				12.43	116.34	0.2391	9.927	116.30	0.2359
190				12.62	117.77	0.2414	10.09	117.74	0.2381

TABLE 8-13b (Continued)
 SUPERHEATED TRICHLOROMONOFUOROMETHANE
 (Freon-11)

Temperature °F	Abs. Press., 6.0 lb/in. ² Sat'n Temp., 33.2° F			Abs. Press., 7.0 lb/in. ² Sat'n Temp., 39.8° F			Abs. Press., 8.0 lb/in. ² Sat'n Temp., 45.7° F			
	<i>t</i>	<i>v</i>	<i>h</i>	<i>s</i>	<i>v</i>	<i>h</i>	<i>s</i>	<i>v</i>	<i>h</i>	<i>s</i>
(at sat'n)	(6.313)	(96.89)	(0.1878)	(5.470)	(97.09)	(0.1972)	(4.833)	(97.80)	(0.1968)	
40	6.401	97.16	0.1996	5.472	97.11	0.1973				
50	6.534	98.46	0.2021	5.587	98.41	0.1998	4.875	98.36	0.1979	
60	6.668	99.77	0.2047	5.702	99.72	0.2024	4.976	99.67	0.2004	
70	6.802	101.09	0.2072	5.817	101.04	0.2049	5.077	100.99	0.2029	
80	6.936	102.42	0.2097	5.932	102.37	0.2074	5.178	102.32	0.2054	
90	7.069	103.76	0.2121	6.046	103.71	0.2099	5.279	103.66	0.2079	
100	7.202	105.11	0.2146	6.161	105.06	0.2123	5.380	105.02	0.2103	
110	7.335	106.47	0.2170	6.275	106.42	0.2147	5.480	106.38	0.2127	
120	7.467	107.84	0.2194	6.389	107.79	0.2171	5.580	107.75	0.2151	
130	7.600	109.22	0.2217	6.503	109.17	0.2194	5.680	109.13	0.2175	
140	7.733	110.61	0.2241	6.617	110.56	0.2218	5.780	110.52	0.2198	
150	7.865	112.01	0.2264	6.730	111.97	0.2241	5.880	111.92	0.2221	
160	7.998	113.41	0.2287	6.844	113.38	0.2264	5.980	113.34	0.2244	
170	8.130	114.83	0.2309	6.958	114.80	0.2287	6.081	114.76	0.2267	
180	8.263	116.26	0.2332	7.072	116.23	0.2309	6.181	116.19	0.2290	
190	8.395	117.70	0.2354	7.186	117.67	0.2332	6.281	117.63	0.2312	
200							6.380	119.08	0.2334	
210							6.480	120.55	0.2356	
220							6.579	122.02	0.2378	
230							6.679	123.50	0.2400	
240							6.770	124.99	0.2421	

Temperature °F	Abs. Press., 10.0 lb/in. ² Sat'n Temp., 55.9° F			Abs. Press., 12.0 lb/in. ² Sat'n Temp., 64.6° F			Abs. Press., 14 lb/in. ² Sat'n Temp., 72.2° F			
	(at sat'n)	(3.928)	(99.04)	(0.1960)	(3.317)	(100.08)	(0.1955)	(2.873)	(100.99)	(0.1950)
60	3.961	99.57	0.1970							
70	4.042	100.89	0.1996	3.354	100.80	0.1969				
80	4.123	102.23	0.2021	3.421	102.13	0.1993	2.920	102.04	0.1970	
90	4.204	103.57	0.2045	3.489	103.48	0.2018	2.978	103.38	0.1995	
100	4.285	104.93	0.2070	3.557	104.84	0.2042	3.037	104.74	0.2019	
110	4.366	106.29	0.2094	3.625	106.20	0.2066	3.095	106.11	0.2043	
120	4.447	107.67	0.2118	3.693	107.58	0.2090	3.154	107.49	0.2067	
130	4.528	109.05	0.2141	3.761	109.97	0.2114	3.212	108.88	0.2091	
140	4.609	110.44	0.2165	3.829	110.36	0.2138	3.271	110.28	0.2114	
150	4.690	111.85	0.2188	3.897	111.76	0.2161	3.329	111.69	0.2138	
160	4.771	113.26	0.2211	3.964	113.18	0.2184	3.388	113.11	0.2161	
170	4.851	114.69	0.2234	4.032	114.61	0.2207	3.446	114.54	0.2184	
180	4.932	116.12	0.2257	4.099	116.04	0.2230	3.504	115.97	0.2206	
190	5.012	117.56	0.2279	4.166	117.48	0.2252	3.562	117.41	0.2229	
200	5.092	119.01	0.2301	4.233	118.94	0.2274	3.620	118.87	0.2251	
210	5.171	120.48	0.2323	4.300	120.41	0.2296	3.677	120.34	0.2273	
220	5.251	121.95	0.2345	4.367	121.88	0.2318	3.735	121.81	0.2295	
230	5.331	123.43	0.2367	4.433	123.36	0.2340	3.792	123.29	0.2317	
240	5.411	124.92	0.2388	4.500	124.86	0.2361	3.850	124.79	0.2338	

TABLE 8-13b (Continued)
 SUPERHEATED TRICHLOROMONOFUOROMETHANE
 (Freon-11)

Temperature °F	Abs. Press., 18 lb/in. ² Sat'n Temp., 85.2° F			Abs. Press., 24 lb/in. ² Sat'n Temp., 100.9° F			Abs. Press., 30 lb/in. ² Sat'n Temp., 113.9° F		
	<i>v</i>	<i>h</i>	<i>s</i>	<i>v</i>	<i>h</i>	<i>s</i>	<i>v</i>	<i>h</i>	<i>s</i>
<i>t</i>									
(at sat'n)	(2.274)	(102.55)	(0.1943)	(1.738)	(104.41)	(0.1938)	(1.409)	(105.93)	(0.1935)
90	2.295	103.20	0.1956						
100	2.342	104.56	0.1980						
110	2.388	105.93	0.2005	1.769	105.66	0.1960			
120	2.434	107.32	0.2029	1.805	107.05	0.1984	1.426	106.79	0.1949
130	2.480	108.71	0.2053	1.840	108.45	0.2008	1.455	108.19	0.1973
140	2.526	110.11	0.2076	1.875	109.86	0.2032	1.484	109.60	0.1997
150	2.572	111.53	0.2100	1.910	111.28	0.2055	1.513	111.02	0.2020
160	2.618	112.95	0.2123	1.945	112.70	0.2079	1.541	112.45	0.2044
170	2.663	114.38	0.2146	1.980	114.14	0.2102	1.569	113.90	0.2067
180	2.709	115.82	0.2168	2.015	115.59	0.2124	1.598	115.35	0.2090
190	2.755	117.26	0.2191	2.049	117.04	0.2147	1.626	116.81	0.2112
200	2.801	118.72	0.2213	2.084	118.50	0.2169	1.654	118.27	0.2135
210	2.846	120.19	0.2235	2.119	119.98	0.2192	1.682	119.75	0.2157
220	2.891	121.67	0.2257	2.154	121.46	0.2214	1.711	121.24	0.2179
230	2.936	123.16	0.2279	2.188	122.96	0.2235	1.739	122.74	0.2201
240	2.981	124.66	0.2301	2.222	124.46	0.2257	1.767	124.25	0.2223
250	3.027	126.17	0.2322	2.256	125.97	0.2279	1.794	125.77	0.2244
260	3.072	127.69	0.2343	2.291	127.50	0.2300	1.822	127.30	0.2266
270	3.117	129.22	0.2364	2.325	129.03	0.2321	1.850	128.84	0.2287
280	3.162	130.76	0.2385	2.359	130.58	0.2342	1.878	130.39	0.2308
290	3.207	132.31	0.2406	2.393	132.13	0.2363	1.905	131.95	0.2329

TABLE 8-14a
SATURATED DICHLOROMONOFUOROMETHANE (CHCl₂F)
(Freon-21)

Temperature °F <i>t</i>	Pressure lb/in. ² <i>p</i>	Volume ft ³ /lb <i>v_g</i>	Density lb/ft ³ <i>l/v_f</i>	Heat Content (Enthalpy) from -40°			Entropy from -40°	
				Liquid Btu/lb <i>h_f</i>	Latent Btu/lb <i>h_{fg}</i>	Vapor Btu/lb <i>h_g</i>	Liquid Btu/lb °F <i>s_f</i>	Vapor Btu/lb °F <i>s_g</i>
-40	1.358	32.09	94.52	0.00	114.56	114.56	0.0000	0.2730
-30	1.888	23.61	93.79	2.36	113.40	115.76	0.0055	0.2695
-20	2.578	17.66	93.04	4.71	112.25	116.96	0.0109	0.2663
-10	3.463	13.43	92.28	7.07	111.10	118.17	0.0162	0.2633
0	4.582	10.35	91.52	9.44	109.93	119.37	0.0214	0.2606
2	4.838	9.840	91.36	9.92	109.69	119.61	0.0225	0.2601
4	5.105	9.361	91.21	10.39	109.46	119.85	0.0235	0.2596
5	5.243	9.132	91.13	10.63	109.34	119.97	0.0240	0.2593
6	5.384	8.910	91.05	10.86	109.23	120.09	0.0245	0.2591
8	5.674	8.486	90.90	11.33	109.00	120.33	0.0255	0.2586
10	5.978	8.085	90.74	11.81	108.76	120.57	0.0265	0.2581
12	6.294	7.707	90.59	12.29	108.52	120.81	0.0275	0.2577
14	6.625	7.349	90.43	12.77	108.28	121.05	0.0286	0.2572
16	6.968	7.012	90.27	13.25	108.05	121.30	0.0296	0.2567
18	7.325	6.694	90.11	13.73	107.81	121.54	0.0306	0.2563
20	7.699	6.392	89.96	14.21	107.57	121.78	0.0316	0.2559
22	8.087	6.107	89.80	14.68	107.34	122.02	0.0326	0.2555
24	8.488	5.838	89.64	15.16	107.10	122.26	0.0336	0.2550
26	8.906	5.584	89.48	15.64	106.86	122.50	0.0346	0.2546
28	9.341	5.342	89.32	16.12	106.62	122.74	0.0356	0.2542
30	9.793	5.112	89.16	16.61	106.37	122.98	0.0365	0.2538
32	10.26	4.894	89.00	17.09	106.13	123.22	0.0375	0.2534
34	10.75	4.688	88.84	17.58	105.88	123.46	0.0385	0.2530
36	11.26	4.492	88.68	18.07	105.64	123.71	0.0395	0.2526
38	11.78	4.306	88.52	18.55	105.40	123.95	0.0405	0.2523
40	12.32	4.130	88.35	19.04	105.15	124.19	0.0414	0.2519
50	15.33	3.370	87.54	21.40	103.90	125.39	0.0463	0.2502
60	18.90	2.773	86.71	23.98	102.62	126.60	0.0511	0.2486
70	23.08	2.300	85.87	26.49	101.30	127.79	0.0559	0.2471
80	27.96	1.923	85.03	29.03	99.95	128.98	0.0606	0.2458
90	33.58	1.619	84.17	31.69	98.55	130.14	0.0652	0.2446
100	40.04	1.371	83.31	34.18	97.11	131.29	0.0699	0.2434
110	47.40	1.169	82.43	36.79	95.63	132.42	0.0745	0.2424
120	55.75	1.001	81.54	39.46	94.08	133.53	0.0791	0.2414

TABLE 8-14b

SUPERHEATED DICHLOROMONOFUOROMETHANE
(Freon-21)

Temperature °F	Abs. Press., 1.2 lb/in. ² Sat'n Temp., -43.6° F			Abs. Press., 1.6 lb/in. ² Sat'n Temp., -35.1° F			Abs. Press., 2 lb/in. ² Sat'n Temp., -28.2° F		
	<i>v</i>	<i>h</i>	<i>s</i>	<i>v</i>	<i>h</i>	<i>s</i>	<i>v</i>	<i>h</i>	<i>s</i>
(at sat'n)	(98.02)	(114.13)	(0.2744)	(27.56)	(115.15)	(0.2713)	(22.57)	(115.98)	(0.2689)
-40	36.34	114.57	0.2754						
-30	37.21	115.80	0.2783	27.88	115.78	0.2727			
-20	38.08	117.03	0.2812	28.53	117.01	0.2755	22.80	116.99	0.2712
-10	38.95	118.28	0.2840	29.19	118.26	0.2783	23.33	118.24	0.2740
0	39.83	119.54	0.2867	29.85	119.52	0.2811	23.85	119.49	0.2768
10	40.70	120.80	0.2894	30.50	120.79	0.2839	24.38	120.76	0.2795
20	41.57	122.08	0.2921	31.15	122.07	0.2866	24.90	122.04	0.2822
30	42.44	123.38	0.2948	31.81	123.36	0.2893	25.43	123.34	0.2849
40	43.32	124.69	0.2975	32.46	124.67	0.2919	25.95	124.65	0.2876
50	44.19	126.01	0.3001	33.12	126.00	0.2945	26.47	125.98	0.2902
60	45.06	127.35	0.3027	33.77	127.35	0.2971	27.00	127.33	0.2928
70	45.93	128.71	0.3053	34.43	128.71	0.2997	27.52	128.69	0.2954
80	46.80	130.09	0.3079	35.08	130.09	0.3023	28.05	130.07	0.2979
90	47.67	131.48	0.3104	35.74	131.47	0.3048	28.57	131.45	0.3005
Temperature °F	Abs. Press., 2.6 lb/in. ² Sat'n Temp., -19.7° F			Abs. Press., 3 lb/in. ² Sat'n Temp., -14.9° F			Abs. Press., 3.5 lb/in. ² Sat'n Temp., -9.6° F		
(at sat'n)	(17.52)	(117.00)	(0.2602)	(15.34)	(117.58)	(0.2647)	(13.29)	(118.22)	(0.2632)
-10	17.92	118.21	0.2689	15.51	118.19	0.2661			
0	18.32	119.47	0.2717	15.87	119.45	0.2689	13.58	119.43	0.2659
10	18.73	120.74	0.2744	16.22	120.72	0.2716	13.88	120.70	0.2686
20	19.13	122.02	0.2771	16.57	122.00	0.2743	14.19	121.98	0.2713
30	19.54	123.31	0.2798	16.92	123.30	0.2770	14.49	123.27	0.2740
40	19.94	124.63	0.2825	17.27	124.61	0.2797	14.79	124.59	0.2767
50	20.34	125.96	0.2851	17.62	125.94	0.2823	15.09	125.92	0.2793
60	20.75	127.31	0.2877	17.97	127.29	0.2849	15.39	127.27	0.2819
70	21.15	128.67	0.2903	18.32	128.65	0.2875	15.69	128.63	0.2845
80	21.55	130.05	0.2929	18.67	130.03	0.2901	15.99	130.01	0.2871
90	21.96	131.43	0.2954	19.02	131.41	0.2926	16.29	131.39	0.2896
100				19.37	132.81	0.2951	16.59	132.79	0.2921
110				19.72	134.22	0.2976	16.89	134.20	0.2946
120				20.07	135.65	0.3001	17.19	135.63	0.2971
130				20.42	137.08	0.3026	17.49	137.06	0.2996
140				20.77	138.53	0.3050	17.79	138.51	0.3021
Temperature °F	Abs. Press., 4 lb/in. ² Sat'n Temp., -4.9° F			Abs. Press., 4.5 lb/in. ² Sat'n Temp., -0.7° F			Abs. Press., 5 lb/in. ² Sat'n Temp., +3.2° F		
(at sat'n)	(11.74)	(118.78)	(0.2619)	(10.63)	(119.29)	(0.2608)	(9.543)	(119.75)	(0.2598)
0	11.87	119.40	0.2633	10.54	119.38	0.2610			
10	12.14	120.66	0.2660	10.78	120.64	0.2637	9.687	120.62	0.2616
20	12.40	121.95	0.2687	11.01	121.93	0.2664	9.899	121.91	0.2644
30	12.66	123.25	0.2714	11.25	123.23	0.2691	10.11	123.20	0.2671
40	12.93	124.57	0.2741	11.48	124.55	0.2718	10.32	124.52	0.2697
50	13.19	125.90	0.2767	11.71	125.88	0.2744	10.53	125.86	0.2724
60	13.45	127.25	0.2793	11.95	127.23	0.2770	10.74	127.21	0.2750
70	13.72	128.61	0.2819	12.18	128.59	0.2796	10.96	128.57	0.2776
80	13.98	129.99	0.2845	12.42	129.97	0.2822	11.17	129.95	0.2801
90	14.24	131.37	0.2870	12.65	131.35	0.2847	11.38	131.33	0.2827
100	14.51	132.77	0.2895	12.88	132.75	0.2873	11.59	132.73	0.2852
110	14.77	134.18	0.2920	13.12	134.16	0.2898	11.80	134.14	0.2877
120	15.03	135.61	0.2945	13.35	135.59	0.2922	12.01	135.57	0.2902
130	15.29	137.04	0.2970	13.59	137.03	0.2947	12.22	137.01	0.2926
140	15.56	138.49	0.2994	13.82	138.48	0.2971	12.43	138.46	0.2951

TABLE 8-14b (Continued)

SUPERHEATED DICHLOROMONOFLUOROMETHANE
(Freon-21)

Temperature °F	Abs. Press., 5.5 lb/in. ² Sat'n Temp., 6.8° F			Abs. Press., 6 lb/in. ² Sat'n Temp., 10.1° F			Abs. Press., 7 lb/in. ² Sat'n Temp., 16.2° F			
	<i>t</i>	<i>v</i>	<i>h</i>	<i>s</i>	<i>v</i>	<i>h</i>	<i>s</i>	<i>v</i>	<i>h</i>	<i>s</i>
(at sat'n)	(8.737)	(120.19)	(0.2589)	(8.066)	(120.58)	(0.2581)	(6.983)	(121.32)	(0.2567)	
10	8.796	120.60	0.2598							
20	8.989	121.89	0.2625	8.232	121.86	0.2608	7.041	121.81	0.2578	
30	9.182	123.18	0.2652	8.408	123.16	0.2635	7.193	123.11	0.2605	
40	9.375	124.50	0.2679	8.585	124.48	0.2661	7.345	124.43	0.2631	
50	9.567	125.84	0.2705	8.762	125.81	0.2688	7.497	125.76	0.2658	
60	9.759	127.19	0.2731	8.939	127.17	0.2714	7.648	127.12	0.2684	
70	9.952	128.55	0.2757	9.115	128.53	0.2740	7.800	128.48	0.2710	
80	10.14	129.93	0.2783	9.291	129.91	0.2766	7.951	129.86	0.2735	
90	10.34	131.31	0.2808	9.467	131.29	0.2791	8.102	131.25	0.2761	
100	10.53	132.71	0.2834	9.643	132.69	0.2816	8.254	132.65	0.2786	
110	10.72	134.13	0.2858	9.819	134.11	0.2841	8.405	134.07	0.2811	
120	10.91	135.56	0.2883	9.995	135.54	0.2866	8.556	135.50	0.2836	
130	11.10	136.99	0.2908	10.17	136.97	0.2891	8.707	136.93	0.2861	
140	11.30	138.44	0.2932	10.35	138.42	0.2915	8.858	138.38	0.2885	

Temperature °F	Abs. Press., 8 lb/in. ² Sat'n Temp., 21.6° F			Abs. Press., 9 lb/in. ² Sat'n Temp., 26.4° F			Abs. Press., 10 lb/in. ² Sat'n Temp., 30.9° F			
	(at sat'n)	(6.169)	(121.96)	(0.2556)	(5.589)	(122.55)	(0.2545)	(5.014)	(123.10)	(0.2536)
30	6.281	123.06	0.2578	5.572	123.02	0.2555				
40	6.414	124.39	0.2605	5.691	124.34	0.2582	5.112	124.30	0.2561	
50	6.547	125.73	0.2631	5.809	125.68	0.2608	5.219	125.63	0.2587	
60	6.681	127.08	0.2657	5.928	127.04	0.2634	5.326	126.99	0.2613	
70	6.814	128.44	0.2683	6.046	128.40	0.2660	5.433	128.36	0.2639	
80	6.947	129.82	0.2709	6.165	129.78	0.2686	5.540	129.74	0.2665	
90	7.079	131.21	0.2735	6.283	131.17	0.2711	5.646	131.13	0.2691	
100	7.212	132.61	0.2760	6.402	132.58	0.2737	5.753	132.54	0.2716	
110	7.345	134.03	0.2785	6.520	133.99	0.2762	5.860	133.95	0.2741	
120	7.477	135.46	0.2810	6.638	135.42	0.2787	5.966	135.38	0.2766	
130	7.609	136.90	0.2835	6.755	136.86	0.2811	6.072	136.82	0.2791	
140	7.742	138.35	0.2859	6.873	138.31	0.2836	6.178	138.28	0.2815	

Temperature °F	Abs. Press., 12 lb/in. ² Sat'n Temp., 38.8° F			Abs. Press., 14 lb/in. ² Sat'n Temp., 45.8° F			Abs. Press., 18 lb/in. ² Sat'n Temp., 57.6° F			
	(at sat'n)	(4.232)	(124.04)	(0.2521)	(3.667)	(124.88)	(0.2509)	(2.902)	(126.31)	(0.2490)
40	4.243	124.20	0.2524							
50	4.333	125.54	0.2551	3.700	125.45	0.2520				
60	4.422	126.90	0.2577	3.777	126.82	0.2546	2.916	126.64	0.2496	
70	4.512	128.28	0.2603	3.854	128.19	0.2572	2.977	128.02	0.2522	
80	4.601	129.66	0.2629	3.931	129.58	0.2598	3.038	129.41	0.2548	
90	4.691	131.05	0.2655	4.008	130.97	0.2624	3.098	130.81	0.2574	
100	4.780	132.46	0.2680	4.085	132.38	0.2649	3.159	132.22	0.2599	
110	4.870	133.88	0.2705	4.162	133.80	0.2674	3.219	133.65	0.2624	
120	4.959	135.31	0.2730	4.239	135.23	0.2699	3.279	135.08	0.2649	
130	5.047	136.75	0.2755	4.315	136.68	0.2724	3.339	136.52	0.2674	
140	5.136	138.20	0.2779	4.392	138.13	0.2749	3.399	137.99	0.2699	
150	5.225	139.68	0.2804	4.468	139.60	0.2773	3.459	139.47	0.2723	
160	5.314	141.18	0.2828	4.545	141.11	0.2798	3.519	140.97	0.2748	
170	5.403	142.69	0.2852	4.621	142.62	0.2822	3.579	142.48	0.2772	
180	5.492	144.20	0.2876	4.697	144.13	0.2846	3.639	144.00	0.2796	
190	5.580	145.73	0.2900	4.774	145.66	0.2869	3.699	145.53	0.2820	

TABLE 8·14b (Continued)
 SUPERHEATED DICHLOROMONOFUOROMETHANE
 (Freon-21)

Temperature °F	Abs. Press., 20 lb/in. ² Sat'n Temp., 62.8° F			Abs. Press., 24 lb/in. ² Sat'n Temp., 72.0° F			Abs. Press., 28 lb/in. ² Sat'n Temp., 80.1° F		
	<i>v</i>	<i>h</i>	<i>s</i>	<i>v</i>	<i>h</i>	<i>s</i>	<i>v</i>	<i>h</i>	<i>s</i>
(at sat'n)	(<i>0.630</i>)	(<i>126.94</i>)	(<i>0.2482</i>)	(<i>0.218</i>)	(<i>128.03</i>)	(<i>0.2469</i>)	(<i>1.920</i>)	(<i>128.98</i>)	(<i>0.2458</i>)
70	2.670	127.94	0.2501						
80	2.725	129.33	0.2527	2.256	129.15	0.2489			
90	2.780	130.73	0.2552	2.302	130.55	0.2515	1.900	130.38	0.2483
100	2.834	132.14	0.2578	2.347	131.97	0.2541	2.000	131.81	0.2509
110	2.889	133.56	0.2603	2.393	133.40	0.2566	2.039	133.24	0.2535
120	2.943	135.00	0.2628	2.439	134.85	0.2591	2.079	134.69	0.2560
130	2.998	136.45	0.2653	2.485	136.30	0.2616	2.119	136.14	0.2585
140	3.052	137.91	0.2678	2.530	137.77	0.2641	2.158	137.61	0.2610
150	3.106	139.39	0.2702	2.576	139.26	0.2666	2.197	139.10	0.2634
160	3.160	140.90	0.2727	2.622	140.76	0.2690	2.237	140.61	0.2659
170	3.214	142.41	0.2751	2.667	142.28	0.2714	2.276	142.13	0.2683
180	3.268	143.93	0.2775	2.712	143.80	0.2738	2.315	143.66	0.2707
190	3.322	145.46	0.2799	2.757	145.33	0.2762	2.354	145.20	0.2731
200				2.803	146.88	0.2786	2.393	146.75	0.2755
210				2.848	148.45	0.2809	2.432	148.32	0.2778
220				2.893	150.03	0.2833	2.471	149.91	0.2802
230				2.938	151.62	0.2856	2.510	151.50	0.2825
240				2.983	153.23	0.2879	2.548	153.11	0.2848
Temperature °F	Abs. Press., 32 lb/in. ² Sat'n Temp., 87.3° F			Abs. Press., 36 lb/in. ² Sat'n Temp., 93.9° F			Abs. Press., 40 lb/in. ² Sat'n Temp., 99.9° F		
	(<i>1.694</i>)	(<i>129.83</i>)	(<i>0.2449</i>)	(<i>1.516</i>)	(<i>130.60</i>)	(<i>0.2441</i>)	(<i>1.373</i>)	(<i>131.88</i>)	(<i>0.2435</i>)
90	1.703	130.21	0.2456						
100	1.739	131.64	0.2481	1.535	131.47	0.2457	1.373	131.29	0.2435
110	1.774	133.08	0.2507	1.587	132.91	0.2482	1.402	132.74	0.2460
120	1.809	134.53	0.2532	1.598	134.36	0.2508	1.430	134.20	0.2486
130	1.844	135.99	0.2557	1.630	135.83	0.2533	1.459	135.67	0.2511
140	1.878	137.46	0.2582	1.661	137.30	0.2558	1.487	137.15	0.2536
150	1.913	138.95	0.2607	1.692	138.80	0.2583	1.515	138.65	0.2561
160	1.948	140.46	0.2632	1.723	140.32	0.2607	1.543	140.17	0.2585
170	1.982	141.99	0.2656	1.754	141.85	0.2632	1.572	141.70	0.2610
180	2.017	143.52	0.2680	1.785	143.38	0.2656	1.600	143.24	0.2634
190	2.051	145.06	0.2704	1.816	144.92	0.2680	1.628	144.79	0.2658
200	2.086	146.61	0.2728	1.847	146.48	0.2704	1.656	146.35	0.2682
210	2.120	148.19	0.2751	1.878	148.06	0.2727	1.683	147.93	0.2706
220	2.154	149.78	0.2775	1.908	149.65	0.2751	1.711	149.52	0.2729
230	2.189	151.38	0.2798	1.939	151.25	0.2774	1.739	151.13	0.2753
240	2.223	152.99	0.2821	1.969	152.87	0.2798	1.767	152.75	0.2776
250							1.794	154.38	0.2799
260							1.822	156.01	0.2822
270							1.849	157.66	0.2845
280							1.877	159.33	0.2868
290							1.904	161.02	0.2890

TABLE 8-14b (Continued)
 SUPERHEATED DICHLOROMONOFUOROMETHANE
 (Freon-21)

Temperature °F	Abs. Press., 50 lb/in. ² Sat'n Temp., 113.3° F			Abs. Press., 60 lb/in. ² Sat'n Temp., 124.7° F			Abs. Pressure 80 lb/in. ² Sat'n Temp. 143.8° F		
	<i>v</i>	<i>h</i>	<i>s</i>	<i>v</i>	<i>h</i>	<i>s</i>	<i>v</i>	<i>h</i>	<i>s</i>
(at sat'n)	(1.111)	(132.78)	(0.2421)	(0.9335)	(134.05)	(0.2410)	(0.7069)	(136.06)	(0.2393)
120	1.127	133.78	0.2438						
130	1.150	135.26	0.2463	0.9439	134.83	0.2423
140	1.173	136.75	0.2488	0.9639	136.34	0.2449
150	1.197	138.26	0.2513	0.9836	137.86	0.2474	0.7167	137.01	0.2409
160	1.220	139.79	0.2538	1.003	139.40	0.2490	0.7321	138.58	0.2435
170	1.243	141.33	0.2563	1.023	140.95	0.2524	0.7476	140.16	0.2460
180	1.265	142.88	0.2587	1.042	142.51	0.2548	0.7629	141.74	0.2485
190	1.288	144.43	0.2612	1.062	144.07	0.2573	0.7781	143.33	0.2509
200	1.311	146.00	0.2636	1.081	145.65	0.2597	0.7933	144.93	0.2534
210	1.334	147.59	0.2659	1.100	147.25	0.2621	0.8083	146.54	0.2558
220	1.356	149.19	0.2683	1.120	148.86	0.2645	0.8233	148.17	0.2582
230	1.379	150.80	0.2707	1.139	150.48	0.2668	0.8382	149.81	0.2606
240	1.401	152.42	0.2730	1.158	152.11	0.2692	0.8530	151.46	0.2630
250	1.424	154.06	0.2753	1.177	153.76	0.2715	0.8678	153.12	0.2654
260	1.446	155.71	0.2776	1.196	155.41	0.2738	0.8825	154.78	0.2677
270	1.469	157.37	0.2799	1.215	157.08	0.2761	0.8971	156.46	0.2700
280	1.491	159.05	0.2822	1.234	158.76	0.2784	0.9117	158.16	0.2723
290	1.513	160.74	0.2845	1.252	160.45	0.2807	0.9262	159.87	0.2746

TABLE 8-15

PROPERTIES OF MONOCHLORODIFLUOROMETHANE¹

(Freon-22)

Sat. Temp., ° F	Abs. Press., psi	Volume		Enthalpy and Entropy Taken from - 40° F							
		Liquid	Vapor	Enthalpy		Entropy		50° superheat		100° superheat	
				Liquid	Vapor	Liquid	Vapor	Enthalpy	Entropy	Enthalpy	Entropy
0	38.79	0.01192	1.373	10.63	105.02	0.0240	0.2293	112.35	0.2446	120.00	0.2590
2	40.43	0.01195	1.320	11.17	105.24	0.0251	0.2289	112.59	0.2442	120.26	0.2586
4	42.14	0.01198	1.270	11.70	105.45	0.0262	0.2285	112.83	0.2438	120.52	0.2581
5	43.02	0.01200	1.246	11.97	105.56	0.0268	0.2283	112.95	0.2436	120.65	0.2579
6	43.91	0.01201	1.221	12.23	105.66	0.0274	0.2280	113.07	0.2434	120.78	0.2577
8	45.74	0.01205	1.175	12.76	105.87	0.0285	0.2276	113.31	0.2430	121.04	0.2572
10	47.63	0.01208	1.130	13.29	106.08	0.0296	0.2272	113.55	0.2426	121.30	0.2568
12	49.58	0.01211	1.088	13.82	106.29	0.0307	0.2268	113.79	0.2422	121.56	0.2564
14	51.59	0.01215	1.048	14.36	106.50	0.0319	0.2264	114.02	0.2418	121.82	0.2560
16	53.66	0.01218	1.009	14.90	106.71	0.0330	0.2260	114.25	0.2414	122.08	0.2556
18	55.79	0.01222	0.9721	15.44	106.92	0.0341	0.2257	114.48	0.2410	122.33	0.2552
20	57.98	0.01225	0.9369	15.98	107.13	0.0352	0.2253	114.71	0.2406	122.59	0.2548
22	60.23	0.01229	0.9032	16.52	107.33	0.0364	0.2249	114.94	0.2402	122.84	0.2544
24	62.55	0.01232	0.8707	17.06	107.53	0.0375	0.2246	115.17	0.2398	123.10	0.2540
26	64.94	0.01236	0.8398	17.61	107.73	0.0379	0.2242	115.40	0.2395	123.35	0.2537
28	67.40	0.01239	0.8100	18.17	107.93	0.0398	0.2239	115.62	0.2391	123.60	0.2533
30	69.93	0.01243	0.7816	18.74	108.13	0.0409	0.2235	115.84	0.2387	123.85	0.2529
32	72.53	0.01247	0.7543	19.32	108.33	0.0421	0.2232	116.07	0.2383	124.10	0.2525
34	75.21	0.01250	0.7283	19.90	108.52	0.0433	0.2228	116.29	0.2380	124.35	0.2522
36	77.97	0.01254	0.7032	20.49	108.71	0.0445	0.2225	116.52	0.2376	124.59	0.2518
38	80.81	0.01258	0.6791	21.09	108.90	0.0457	0.2222	116.74	0.2373	124.84	0.2515
40	83.72	0.01262	0.6559	21.70	109.09	0.0469	0.2218	116.96	0.2369	125.08	0.2511
42	86.69	0.01266	0.6339	22.29	109.27	0.0481	0.2215	117.18	0.2366	125.32	0.2508
44	89.74	0.01270	0.6126	22.90	109.45	0.0493	0.2211	117.40	0.2363	125.56	0.2504
46	92.88	0.01274	0.5922	23.50	109.63	0.0505	0.2208	117.61	0.2359	125.80	0.2501
48	96.10	0.01278	0.5726	24.11	109.80	0.0516	0.2205	117.82	0.2356	126.04	0.2497
50	99.40	0.01282	0.5537	24.73	109.98	0.0528	0.2201	118.02	0.2353	126.27	0.2494
52	102.8	0.01286	0.5355	25.34	110.14	0.0540	0.2198	118.22	0.2350	126.50	0.2491
54	106.2	0.01290	0.5184	25.95	110.30	0.0552	0.2194	118.42	0.2347	126.73	0.2488
56	109.8	0.01294	0.5014	26.58	110.47	0.0564	0.2191	118.62	0.2343	126.96	0.2484
58	113.5	0.01299	0.4849	27.22	110.63	0.0576	0.2188	118.82	0.2340	127.19	0.2481
60	117.2	0.01303	0.4695	27.83	110.78	0.0588	0.2185	119.01	0.2337	127.42	0.2478
62	121.0	0.01307	0.4546	28.46	110.93	0.0600	0.2181	119.21	0.2334	127.65	0.2475
64	124.9	0.01312	0.4403	29.09	111.08	0.0612	0.2178	119.40	0.2331	127.87	0.2472
66	128.9	0.01316	0.4264	29.72	111.22	0.0624	0.2175	119.59	0.2327	128.10	0.2469
68	133.0	0.01320	0.4129	30.35	111.35	0.0636	0.2172	119.77	0.2324	128.32	0.2466
70	137.2	0.01325	0.4000	30.99	111.49	0.0648	0.2168	119.96	0.2321	128.54	0.2463
72	141.5	0.01330	0.3875	31.65	111.63	0.0661	0.2165	120.15	0.2318	128.76	0.2460
74	145.9	0.01334	0.3754	32.29	111.75	0.0673	0.2162	120.32	0.2315	128.97	0.2457
76	150.4	0.01339	0.3638	32.94	111.88	0.0684	0.2158	120.50	0.2312	129.19	0.2455
78	155.0	0.01344	0.3526	33.61	112.01	0.0696	0.2155	120.67	0.2309	129.40	0.2452
80	159.7	0.01349	0.3417	34.27	112.13	0.0708	0.2151	120.85	0.2306	129.61	0.2449
82	164.5	0.01353	0.3313	34.92	112.24	0.0720	0.2148	121.02	0.2303	129.82	0.2446
84	169.4	0.01358	0.3212	35.60	112.36	0.0732	0.2144	121.18	0.2300	130.02	0.2443
86	174.5	0.01363	0.3113	36.28	112.47	0.0744	0.2140	121.34	0.2297	130.23	0.2441
88	179.6	0.01368	0.3019	36.94	112.57	0.0756	0.2137	121.50	0.2294	130.43	0.2438
90	184.8	0.01374	0.2928	37.61	112.67	0.0768	0.2133	121.66	0.2291	130.63	0.2435
92	190.1	0.01379	0.2841	38.28	112.76	0.0780	0.2130	121.82	0.2288	130.83	0.2432
94	195.6	0.01384	0.2755	38.97	112.85	0.0792	0.2126	121.97	0.2285	131.03	0.2429
96	201.2	0.01390	0.2672	39.65	112.93	0.0803	0.2122	122.12	0.2282	131.23	0.2427
98	206.8	0.01396	0.2594	40.32	113.00	0.0815	0.2119	122.26	0.2279	131.42	0.2424
100	212.6	0.01402	0.2517	40.98	113.06	0.0827	0.2115	122.40	0.2276	131.61	0.2421
102	218.5	0.01408	0.2443	41.65	113.12	0.0839	0.2111	122.53	0.2273	131.80	0.2418
104	224.6	0.01414	0.2370	42.32	113.16	0.0851	0.2107	122.66	0.2270	131.99	0.2416
106	230.7	0.01420	0.2301	42.98	113.20	0.0862	0.2104	122.79	0.2267	132.17	0.2413
108	237.0	0.01426	0.2233	43.66	113.24	0.0874	0.2100	122.92	0.2264	132.35	0.2411
110	243.4	0.01433	0.2167	44.35	113.29	0.0886	0.2096	123.04	0.2261	132.53	0.2408
112	249.9	0.01440	0.2104	45.04	113.34	0.0898	0.2093	123.16	0.2258	132.71	0.2405
114	256.6	0.01447	0.2043	45.74	113.38	0.0909	0.2089	123.28	0.2255	132.88	0.2403
116	263.4	0.01454	0.1983	46.44	113.42	0.0921	0.2085	123.40	0.2253	133.05	0.2400
118	270.3	0.01461	0.1926	47.14	113.46	0.0933	0.2081	123.51	0.2250	133.22	0.2398
120	277.3	0.01469	0.1871	47.85	113.52	0.0945	0.2078	123.62	0.2247	133.39	0.2395

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TABLE 8-16a
SATURATED TRICHLOROTRIFLUOROETHANE (CCl₂F-CClF₂)
(Freon-113)

Temperature °F <i>t</i>	Pressure lb/in. ² <i>p</i>	Volume ft ³ /lb <i>v_g</i>	Density lb/ft ³ <i>1/v_f</i>	Heat Content (Enthalpy) from -40°			Entropy from -40°	
				Liquid Btu/lb <i>h_f</i>	Latent Btu/lb <i>h_{fg}</i>	Vapor Btu/lb <i>h_g</i>	Liquid Btu/lb °F <i>s_f</i>	Vapor Btu/lb °F <i>s_g</i>
-30	0.2987	82.26	105.64	1.97	72.68	74.65	0.0047	0.1738
-20	0.4288	58.61	104.96	3.96	72.09	76.05	0.0092	0.1732
-10	0.6046	42.48	104.26	5.96	71.51	77.47	0.0137	0.1728
0	0.8377	31.31	103.56	7.98	70.92	78.89	0.0182	0.1725
10	1.142	23.45	102.84	10.00	70.32	80.32	0.0225	0.1723
20	1.534	17.81	102.10	12.03	69.72	81.75	0.0268	0.1722
30	2.031	13.71	101.36	14.08	69.12	83.20	0.0310	0.1722
40	2.655	10.68	100.60	16.16	68.50	84.65	0.0352	0.1723
50	3.427	8.426	99.83	18.24	67.87	86.11	0.0393	0.1725
60	4.374	6.713	99.05	20.35	67.22	87.57	0.0434	0.1728
70	5.523	5.404	98.26	22.48	66.56	89.04	0.0475	0.1731
80	6.902	4.392	97.45	24.63	65.88	90.51	0.0515	0.1736
90	8.545	3.600	96.63	26.80	65.18	91.98	0.0555	0.1741
100	10.48	2.976	95.79	28.99	64.46	93.45	0.0594	0.1746
110	12.76	2.477	94.95	31.22	63.71	94.93	0.0634	0.1752
120	15.40	2.078	94.09	33.48	62.93	96.41	0.0673	0.1758
130	18.45	1.754	93.22	35.75	62.14	97.89	0.0712	0.1765
140	21.93	1.491	92.33	38.05	61.31	99.36	0.0750	0.1773

TABLE 8-166
SUPERHEATED TRICHLOROTRIFLUOROETHANE
(Freon-113)

Temperature °F	Abs. Press., 0.3 lb/in. ² Sat'n Temp., -29.9° F			Abs. Press., 0.5 lb/in. ² Sat'n Temp., -15.6° F			Abs. Press., 0.7 lb/in. ² Sat'n Temp., -5.6° F		
	<i>v</i>	<i>h</i>	<i>s</i>	<i>v</i>	<i>h</i>	<i>s</i>	<i>v</i>	<i>h</i>	<i>s</i>
(at sat'n)	(81.96)	(74.66)	(0.1738)	(60.79)	(76.68)	(0.1730)	(37.06)	(78.10)	(0.1726)
-20	83.83	76.06	0.1770						
-10	85.74	77.48	0.1802	51.43	77.47	0.1748			
0	87.65	78.91	0.1834	52.58	78.90	0.1779	37.49	78.90	0.1743
10	89.55	80.35	0.1865	53.72	80.34	0.1810	38.32	80.34	0.1774
20	91.46	81.80	0.1895	54.87	81.79	0.1841	39.15	81.79	0.1805
30	93.37	83.26	0.1925	56.01	83.26	0.1871	39.98	83.25	0.1835
40	95.27	84.74	0.1955	57.16	84.73	0.1901	40.80	84.72	0.1865
50	97.18	86.23	0.1985	58.31	86.22	0.1931	41.62	86.21	0.1895
60	99.90	87.73	0.2014	59.47	87.72	0.1960	42.43	87.71	0.1924
70	101.0	89.24	0.2043	60.63	89.23	0.1989	43.25	89.22	0.1953
80	102.9	90.76	0.2071	61.78	90.75	0.2017	44.07	90.74	0.1981
90	104.8	92.29	0.2099	62.93	92.28	0.2045	44.89	92.28	0.2009
100	106.7	93.83	0.2127	64.07	93.83	0.2073	45.71	93.82	0.2037
110	108.6	95.40	0.2155	65.21	95.39	0.2101	46.53	95.38	0.2065
120	110.5	96.97	0.2182	66.36	96.96	0.2128	47.35	96.95	0.2092
130	112.4	98.55	0.2209	67.51	98.54	0.2155	48.18	98.53	0.2119
140	114.3	100.14	0.2236	68.65	100.13	0.2182	49.00	100.12	0.2146

Temperature °F	Abs. Press., 1.0 lb/in. ² Sat'n Temp., 5.6° F			Abs. Press., 1.4 lb/in. ² Sat'n Temp., 16.8° F			Abs. Press., 1.8 lb/in. ² Sat'n Temp., 25.6° F		
	<i>v</i>	<i>h</i>	<i>s</i>	<i>v</i>	<i>h</i>	<i>s</i>	<i>v</i>	<i>h</i>	<i>s</i>
(at sat'n)	(26.54)	(79.69)	(0.1723)	(19.40)	(81.30)	(0.1722)	(15.35)	(82.57)	(0.1722)
10	26.79	80.32	0.1737						
20	27.38	81.78	0.1768	19.52	81.76	0.1731			
30	27.96	83.24	0.1798	19.93	83.23	0.1762	15.49	83.21	0.1735
40	28.53	84.71	0.1828	20.35	84.70	0.1792	15.81	84.68	0.1765
50	29.10	86.20	0.1857	20.76	86.19	0.1821	16.13	86.17	0.1794
60	29.67	87.70	0.1886	21.17	87.69	0.1850	16.45	87.67	0.1823
70	30.25	89.21	0.1915	21.58	89.20	0.1879	16.77	89.18	0.1852
80	30.82	90.73	0.1943	21.99	90.72	0.1907	17.09	90.70	0.1881
90	31.39	92.27	0.1972	22.40	92.25	0.1935	17.41	92.24	0.1909
100	31.97	93.81	0.2000	22.81	93.80	0.1963	17.73	93.78	0.1937
110	32.56	95.37	0.2027	23.22	95.36	0.1991	18.05	95.34	0.1964
120	33.14	96.94	0.2054	23.63	96.93	0.2018	18.37	96.91	0.1992
130	33.71	98.52	0.2081	24.04	98.51	0.2045	18.69	98.49	0.2019
140	34.28	100.11	0.2108	24.45	100.10	0.2072	19.01	100.08	0.2045
150				24.86	101.70	0.2099	19.33	101.69	0.2072
160				25.27	103.32	0.2125	19.65	103.31	0.2098
170				25.68	104.94	0.2151	19.97	104.93	0.2124
180				26.10	106.58	0.2177	20.29	106.57	0.2150
190				26.52	108.23	0.2202	20.61	108.22	0.2176

TABLE 8-16b (Continued)
 SUPERHEATED TRICHLOROTRIFLUOROETHANE
 (Freon-113)

Temperature °F	Abs. Press., 2.2 lb/in. ² Sat'n Temp., 32.9° F			Abs. Press., 2.4 lb/in. ² Sat'n Temp., 36.2° F			Abs. Press., 3.0 lb/in. ² Sat'n Temp., 44.7° F		
	<i>v</i>	<i>h</i>	<i>s</i>	<i>v</i>	<i>h</i>	<i>s</i>	<i>v</i>	<i>h</i>	<i>s</i>
(at sat'n)	(12.74)	(83.62)	(0.1722)	(11.74)	(84.10)	(0.1722)	(9.537)	(85.34)	(0.1723)
40	12.92	84.67	0.1743	11.84	84.66	0.1734			
50	13.18	86.16	0.1773	12.08	86.15	0.1763	9.637	86.12	0.1730
60	13.44	87.66	0.1802	12.32	87.65	0.1792	9.831	87.62	0.1760
70	13.70	89.17	0.1831	12.56	89.16	0.1821	10.02	89.13	0.1797
80	13.96	90.69	0.1859	12.80	90.68	0.1850	10.22	90.66	0.1826
90	14.22	92.22	0.1887	13.04	92.21	0.1878	10.41	92.19	0.1854
100	14.48	93.77	0.1915	13.28	93.76	0.1906	10.60	93.74	0.1882
110	14.75	95.33	0.1943	13.52	95.32	0.1934	10.80	95.30	0.1910
120	15.02	96.90	0.1970	13.76	96.89	0.1961	10.99	96.87	0.1937
130	15.28	98.49	0.1997	14.00	98.48	0.1988	11.18	98.46	0.1964
140	15.54	100.08	0.2024	14.24	100.07	0.2015	11.38	100.05	0.1991
150	15.81	101.68	0.2051	14.48	101.68	0.2041	11.57	101.65	0.2017
160	16.07	103.30	0.2077	14.72	103.29	0.2068	11.76	103.27	0.2044
170	16.33	104.93	0.2103	14.96	104.92	0.2094	11.95	104.90	0.2070
180	16.59	106.56	0.2129	15.20	106.55	0.2099	12.15	106.54	0.2095
190	16.85	108.21	0.2154	15.44	108.21	0.2145	12.34	108.19	0.2121

Temperature °F	Abs. Press., 3.6 lb/in. ² Sat'n Temp., 52.0° F			Abs. Press., 4.0 lb/in. ² Sat'n Temp., 56.3° F			Abs. Press., 5.0 lb/in. ² Sat'n Temp., 65.7° F		
	<i>v</i>	<i>h</i>	<i>s</i>	<i>v</i>	<i>h</i>	<i>s</i>	<i>v</i>	<i>h</i>	<i>s</i>
(at sat'n)	(8.047)	(86.40)	(0.1725)	(7.297)	(87.03)	(0.1727)	(5.930)	(88.41)	(0.1730)
60	8.177	87.60	0.1749	7.351	87.58	0.1738			
70	8.338	89.11	0.1778	7.479	89.10	0.1767	5.979	89.06	0.1742
80	8.499	90.63	0.1806	7.642	90.62	0.1795	6.096	90.58	0.1771
90	8.659	92.17	0.1834	7.787	92.15	0.1823	6.213	92.12	0.1799
100	8.819	93.71	0.1862	7.932	93.70	0.1851	6.329	93.67	0.1827
110	8.980	95.27	0.1890	8.077	95.26	0.1879	6.445	95.22	0.1855
120	9.141	96.84	0.1917	8.222	96.83	0.1906	6.562	96.79	0.1882
130	9.305	98.43	0.1944	8.367	98.42	0.1933	6.678	98.38	0.1909
140	9.466	100.03	0.1971	8.522	100.01	0.1960	6.795	99.98	0.1936
150	9.627	101.63	0.1998	8.667	101.62	0.1976	6.912	101.58	0.1962
160	9.788	103.25	0.2024	8.812	103.23	0.2013	7.029	103.20	0.1989
170	9.948	104.88	0.2050	8.957	104.86	0.2039	7.146	104.83	0.2015
180	10.11	106.53	0.2076	9.102	106.51	0.2065	7.263	106.48	0.2041
190	10.27	108.18	0.2102	9.247	108.16	0.2091	7.380	108.13	0.2066
200	10.43	109.84	0.2127	9.387	109.82	0.2116	7.497	109.80	0.2092
210	10.59	111.51	0.2153	9.532	111.50	0.2141	7.614	111.48	0.2117
220	10.75	113.20	0.2178	9.677	113.19	0.2166	7.731	113.16	0.2142
230	10.91	114.90	0.2202	9.822	114.88	0.2191	7.848	114.86	0.2167
240	11.07	116.61	0.2227	9.967	116.59	0.2216	7.965	116.57	0.2192

TABLE 8-16b (Continued)
 SUPERHEATED TRICHLOROTRIFLUOROETHANE
 (Freon-113)

Temperature °F	Abs. Press., 6.0 lb/in. ² Sat'n Temp., 73.7° F			Abs. Press., 7.0 lb/in. ² Sat'n Temp., 80.6° F			Abs. Press., 8.0 lb/in. ² Sat'n Temp., 86.9° F		
	<i>v</i>	<i>h</i>	<i>s</i>	<i>v</i>	<i>h</i>	<i>s</i>	<i>v</i>	<i>h</i>	<i>s</i>
(at sat'n)	(5.004)	(89.58)	(0.1733)	(4.336)	(90.00)	(0.1736)	(3.328)	(91.52)	(0.1739)
80	5.066	90.54	0.1751						
90	5.163	92.08	0.1779	4.414	92.05	0.1763	3.850	92.01	0.1748
100	5.260	93.62	0.1807	4.498	93.60	0.1791	3.924	93.56	0.1776
110	5.357	95.18	0.1835	4.581	95.16	0.1818	3.999	95.12	0.1804
120	5.454	96.76	0.1862	4.666	96.73	0.1845	4.073	96.69	0.1831
130	5.551	98.35	0.1889	4.749	98.32	0.1872	4.147	98.28	0.1858
140	5.649	99.95	0.1916	4.834	99.91	0.1899	4.221	99.88	0.1885
150	5.746	101.56	0.1943	4.919	101.52	0.1926	4.295	101.48	0.1912
160	5.843	103.17	0.1969	5.003	103.14	0.1952	4.36	103.10	0.1938
170	5.940	104.80	0.1995	5.086	104.77	0.1978	4.442	104.74	0.1964
180	6.037	106.45	0.2021	5.169	106.42	0.2004	4.515	106.38	0.1990
190	6.134	108.10	0.2047	5.253	108.07	0.2030	4.588	108.04	0.2016
200	6.233	109.77	0.2072	5.337	109.73	0.2056	4.661	109.71	0.2041
210	6.330	111.44	0.2097	5.420	111.41	0.2081	4.734	111.38	0.2066
220	6.427	113.13	0.2122	5.504	113.10	0.2106	4.807	113.07	0.2091
230	6.525	114.83	0.2147	5.587	114.80	0.2131	4.880	114.77	0.2116
240	6.622	116.54	0.2172	5.671	116.50	0.2156	4.952	116.48	0.2141
250							5.024	118.20	0.2166
260							5.096	119.93	0.2190
270							5.168	121.67	0.2214
280							5.240	123.42	0.2238
290							5.313	125.19	0.2261
Temperature °F	Abs. Press., 10.0 lb/in. ² Sat'n Temp., 97.6° F			Abs. Press., 12 lb/in. ² Sat'n Temp., 106.8° F			Abs. Press., 14 lb/in. ² Sat'n Temp., 114.9° F		
	<i>v</i>	<i>h</i>	<i>s</i>	<i>v</i>	<i>h</i>	<i>s</i>	<i>v</i>	<i>h</i>	<i>s</i>
(at sat'n)	(3.110)	(93.11)	(0.1745)	(2.624)	(94.46)	(0.1750)	(2.271)	(95.66)	(0.1755)
100	3.124	93.48	0.1751						
110	3.183	95.04	0.1779	2.640	94.97	0.1759			
120	3.242	96.62	0.1806	2.689	96.55	0.1786	2.291	96.47	0.1770
130	3.301	98.21	0.1834	2.738	98.14	0.1813	2.334	98.06	0.1797
140	3.361	99.81	0.1860	2.788	99.73	0.1840	2.377	99.66	0.1823
150	3.420	101.41	0.1887	2.838	101.34	0.1867	2.421	101.27	0.1850
160	3.480	103.03	0.1914	2.888	102.97	0.1894	2.465	102.90	0.1876
170	3.539	104.66	0.1940	2.938	104.60	0.1920	2.508	104.53	0.1902
180	3.599	106.31	0.1966	2.987	106.25	0.1946	2.550	106.18	0.1928
190	3.658	107.96	0.1991	3.037	107.90	0.1972	2.593	107.84	0.1954
200	3.716	109.63	0.2017	3.086	109.57	0.1997	2.635	109.51	0.1980
210	3.774	111.32	0.2042	3.135	111.26	0.2022	2.678	111.20	0.2005
220	3.832	113.01	0.2067	3.184	112.95	0.2047	2.720	112.89	0.2030
230	3.891	114.71	0.2092	3.234	114.65	0.2072	2.763	114.59	0.2055
240	3.950	116.42	0.2117	3.283	116.36	0.2097	2.806	116.30	0.2080
250	4.010	118.14	0.2141	3.332	118.08	0.2121	2.848	118.02	0.2105
260	4.068	119.87	0.2166	3.382	119.81	0.2146	2.890	119.76	0.2129
270	4.126	121.61	0.2190	3.431	121.55	0.2170	2.932	121.50	0.2153
280	4.185	123.37	0.2214	3.479	123.31	0.2194	2.975	123.26	0.2177
290	4.243	125.15	0.2237	3.528	125.09	0.2218	3.018	125.04	0.2201

TABLE 8-16b (Continued)
 SUPERHEATED TRICHLOROTRIFLUOROETHANE
 (Freon-113)

Temperature °F	Abs. Press., 18 lb/in. ² Sat'n Temp., 128.6° F			Abs. Press., 20 lb/in. ² Sat'n Temp., 134.6° F			Abs. Press., 25 lb/in. ² Sat'n Temp., 147.8° F			
	<i>t</i>	<i>v</i>	<i>h</i>	<i>s</i>	<i>v</i>	<i>h</i>	<i>s</i>	<i>v</i>	<i>h</i>	<i>s</i>
(at sat'n)	(1.795)	(97.68)	(0.1764)	(1.625)	(98.50)	(0.1769)	(1.317)	(100.50)	(0.1778)	
130	1.799	97.91	0.1769							
140	1.833	99.51	0.1795	1.640	99.43	0.1783				
150	1.867	101.12	0.1822	1.672	101.05	0.1810	1.322	100.87	0.1784	
160	1.901	102.75	0.1848	1.703	102.68	0.1836	1.348	102.50	0.1811	
170	1.935	104.39	0.1875	1.734	104.32	0.1862	1.373	104.14	0.1837	
180	1.968	106.04	0.1901	1.765	105.97	0.1888	1.398	105.80	0.1863	
190	2.002	107.71	0.1926	1.796	107.64	0.1914	1.423	107.47	0.1889	
200	2.036	109.39	0.1952	1.827	109.32	0.1940	1.448	109.15	0.1915	
210	2.069	111.08	0.1977	1.857	111.01	0.1966	1.472	110.84	0.1940	
220	2.103	112.77	0.2003	1.887	112.70	0.1991	1.497	112.54	0.1966	
230	2.136	114.47	0.2028	1.917	114.40	0.2016	1.522	114.25	0.1991	
240	2.169	116.18	0.2052	1.947	116.12	0.2041	1.546	115.96	0.2015	
250	2.203	117.90	0.2077	1.977	117.84	0.2065	1.571	117.69	0.2040	
260	2.236	119.64	0.2101	2.007	119.58	0.2090	1.595	119.44	0.2064	
270	2.269	121.39	0.2125	2.037	121.33	0.2114	1.619	121.19	0.2089	
280	2.303	123.15	0.2149	2.068	123.09	0.2138	1.643	122.95	0.2113	
290	2.336	124.92	0.2173	2.098	124.87	0.2162	1.667	124.73	0.2137	
300	2.370	126.72	0.2197	2.128	126.67	0.2185	1.691	126.53	0.2160	
310	2.403	128.53	0.2220	2.158	128.47	0.2209	1.716	128.34	0.2184	
320	2.436	130.34	0.2244	2.187	130.28	0.2232	1.741	130.16	0.2207	
330	2.469	132.16	0.2267	2.217	132.10	0.2255	1.765	131.98	0.2231	
340	2.502	133.99	0.2290	2.247	133.94	0.2278	1.789	133.82	0.2254	

Temperature °F	Abs. Press., 30 lb/in. ² Sat'n Temp., 159.1° F			Abs. Pressure 35 lb/in. ² Sat'n Temp. 169.0° F			Abs. Pressure 40 lb/in. ² Sat'n Temp. 177.9° F			
	(at sat'n)	(1.109)	(102.15)	(0.1737)	(0.9580)	(103.60)	(0.1795)	(0.8435)	(104.90)	(0.1802)
160	1.111	102.31	0.1790							
170	1.132	103.96	0.1816	0.9594	103.77	0.1797	
180	1.153	105.62	0.1842	0.9780	105.43	0.1824	0.8462	105.26	0.1808	
190	1.173	107.29	0.1868	0.9963	107.10	0.1850	0.8627	106.94	0.1834	
200	1.194	108.97	0.1894	1.015	108.79	0.1876	0.8791	108.62	0.1860	
210	1.215	110.67	0.1919	1.033	110.49	0.1901	0.8953	110.32	0.1885	
220	1.236	112.37	0.1945	1.051	112.20	0.1927	0.9114	112.03	0.1911	
230	1.257	114.08	0.1970	1.069	113.91	0.1952	0.9274	113.74	0.1936	
240	1.278	115.80	0.1995	1.088	115.63	0.1977	0.9435	115.47	0.1961	
250	1.299	117.53	0.2019	1.106	117.37	0.2002	0.9596	117.21	0.1986	
260	1.320	119.28	0.2044	1.124	119.12	0.2026	0.9756	118.97	0.2010	
270	1.341	121.03	0.2068	1.142	120.88	0.2051	0.9915	120.74	0.2035	
280	1.362	122.80	0.2092	1.160	122.66	0.2075	1.008	122.52	0.2059	
290	1.382	124.59	0.2116	1.177	124.45	0.2099	1.023	124.31	0.2083	
300	1.402	126.39	0.2140	1.195	126.26	0.2122	1.038	126.12	0.2107	
310	1.422	128.20	0.2163	1.212	128.07	0.2146	1.054	127.94	0.2131	
320	1.442	130.02	0.2187	1.230	129.89	0.2169	1.070	129.76	0.2154	
330	1.462	131.85	0.2210	1.248	131.72	0.2193	1.085	131.59	0.2178	
340	1.483	133.69	0.2233	1.266	133.56	0.2216	1.101	133.44	0.2201	

TABLE 8-17g
SATURATED STEAM: PRESSURE TABLE

Abs. Press., lb/in. ² <i>p</i>	Temp., °F <i>t</i>	Specific Volume			Heat Content			Entropy			Abs. Press., lb/in. ² <i>p</i>
		Sat. Liquid <i>v_f</i>	Evap. <i>v_{fg}</i>	Sat. Vapor <i>v_g</i>	Sat. Liquid <i>h_f</i>	Evap. <i>L</i>	Sat. Vapor <i>h_g</i>	Sat. Liquid <i>s_f</i>	Evap. <i>s_{fg}</i>	Sat. Vapor <i>s_g</i>	
1/2 Hg	58.83	0.01803	1256.0	26.88	1058.8	1855.7	0.0533	2.0422	2.0955	1/2 Hg	
1 Hg	70.44	0.01805	856.5	35.47	1052.5	1891.0	0.0784	1.9866	2.0609	1 Hg	
1 1/2 Hg	79.06	0.01807	652.7	47.06	1047.8	1904.9	0.0914	1.9451	2.0365	1 1/2 Hg	
2 Hg	91.77	0.01810	445.3	62.52	1040.5	1900.6	0.1147	1.8877	2.0024	2 Hg	
3 Hg	101.17	0.01813	339.5	83.5	1035.7	1904.8	0.1316	1.8468	1.9784	3 Hg	
4 Hg	108.73	0.01816	275.2	107.9	1031.5	1903.1	0.1480	1.8148	1.9588	4 Hg	
5 Hg	115.08	0.01818	231.8	131.8	1027.9	1910.8	0.1561	1.7885	1.9446	5 Hg	
1	101.76	0.01814	333.8	69.69	1035.3	1910.8	0.1326	1.8442	1.9769	1	
5	162.25	0.01841	73.59	73.61	1000.4	1930.6	0.2348	1.6088	1.8435	5	
10	193.21	0.01658	38.44	38.44	981.8	1943.0	0.2834	1.5040	1.7874	10	
14.696	212.00	0.01670	26.80	26.82	970.2	1950.2	0.3119	1.4446	1.7564	14.696	
18	213.03	0.01671	26.31	26.31	969.6	1950.6	0.3154	1.4414	1.7548	18	
30	227.96	0.01682	20.078	20.095	959.9	1956.0	0.3356	1.3960	1.7317	30	
40	250.34	0.01698	13.728	13.745	945.0	1963.7	0.3680	1.3310	1.6990	40	
60	287.24	0.01712	10.480	10.497	933.3	1969.2	0.3919	1.2840	1.6759	60	
80	292.71	0.01724	8.496	8.514	923.5	1973.5	0.4111	1.2469	1.6580	80	
60	292.71	0.01735	7.155	7.172	915.0	1977.0	0.4271	1.2162	1.6434	60	
70	302.92	0.01744	6.186	6.203	907.4	1979.9	0.4410	1.1900	1.6310	70	
80	312.03	0.01754	5.452	5.470	900.5	1982.4	0.4532	1.1670	1.6202	80	
90	320.37	0.01763	4.874	4.892	894.2	1984.6	0.4642	1.1465	1.6107	90	
100	327.83	0.01771	4.408	4.426	888.2	1986.6	0.4742	1.1280	1.6022	100	
110	334.79	0.01779	4.026	4.044	882.7	1988.3	0.4834	1.1111	1.5944	110	
120	341.26	0.01786	3.707	3.725	877.4	1989.8	0.4918	1.0956	1.5874	120	
130	347.31	0.01794	3.433	3.451	872.4	1991.2	0.4996	1.0812	1.5808	130	
140	353.03	0.01801	3.198	3.216	867.7	1992.4	0.5070	1.0677	1.5747	140	
150	358.43	0.01808	2.992	3.010	863.1	1993.5	0.5140	1.0550	1.5690	150	
160	363.55	0.01814	2.812	2.830	858.7	1994.5	0.5205	1.0431	1.5638	160	
170	368.42	0.01821	2.653	2.671	854.4	1995.3	0.5268	1.0318	1.5588	170	
180	373.08	0.01827	2.511	2.529	850.3	1996.3	0.5327	1.0211	1.5538	180	
190	377.55	0.01833	2.383	2.401	846.3	1997.0	0.5384	1.0109	1.5493	190	
200	381.82	0.01839	2.267	2.285	842.4	1997.8	0.5438	1.0012	1.5450	200	
310	385.93	0.01844	2.162	2.180	838.4	1998.4	0.5491	0.9929	1.5409	310	
320	389.89	0.01850	2.066	2.084	835.0	1999.0	0.5549	0.9849	1.5369	320	
330	393.70	0.01856	1.9778	1.9964	831.4	1999.6	0.5603	0.9773	1.5329	330	
340	397.40	0.01861	1.8970	1.9156	827.6	1999.9	0.5653	0.9703	1.5290	340	
350	400.97	0.01867	1.8223	1.8410	824.0	2000.5	0.5699	0.9638	1.5252	350	
360	404.43	0.01872	1.7536	1.7723	820.2	2001.0	0.5732	0.9577	1.5216	360	
370	407.79	0.01877	1.6896	1.7083	816.4	2001.4	0.5765	0.9519	1.5181	370	
380	411.06	0.01882	1.6302	1.6490	812.6	2001.8	0.5795	0.9465	1.5147	380	
390	414.24	0.01887	1.5745	1.5934	808.5	2002.1	0.5825	0.9413	1.5113	390	
400	417.33	0.01892	1.5225	1.5414	804.5	2002.4	0.5853	0.9362	1.5082	400	
450	431.71	0.0191	1.3054	1.3245	793.7	2003.6	0.6061	0.8905	1.4966	450	

TABLE 8-17a (Continued)
SATURATED STEAM: PRESSURE TABLE

Abs. Press., lb/in. ² <i>p</i>	Temp., °F. <i>t</i>	Specific Volume			Heat Content			Entropy			Abs. Press., lb/in. ² <i>p</i>
		Sat. Liquid <i>v_f</i>	Evap. <i>v_{fg}</i>	Sat. Vapor <i>v_g</i>	Sat. Liquid <i>h_f</i>	Evap. <i>L</i>	Sat. Vapor <i>h_g</i>	Sat. Liquid <i>s_f</i>	Evap. <i>s_{fg}</i>	Sat. Vapor <i>s_g</i>	
400	444.53	0.0194	1.1407	1.1601	424.2	779.8	1204.1	0.6218	0.8625	1.4843	400
450	456.27	0.0196	1.0107	1.0303	437.4	766.7	1204.1	0.6361	0.8371	1.4732	450
500	466.99	0.0198	0.9063	0.9261	449.7	754.0	1203.7	0.6493	0.8137	1.4630	500
550	476.91	0.0200	0.8202	0.8402	461.3	741.7	1203.0	0.6616	0.7920	1.4536	550
600	486.17	0.0202	0.7677	0.7877	472.8	729.8	1202.1	0.6731	0.7716	1.4447	600
650	494.86	0.0204	0.7356	0.7600	482.9	719.2	1201.1	0.6840	0.7524	1.4364	650
700	503.04	0.0206	0.7181	0.7427	492.9	709.8	1199.7	0.6943	0.7342	1.4285	700
750	510.80	0.0208	0.7085	0.7311	502.5	701.6	1198.2	0.7042	0.7169	1.4211	750
800	518.18	0.0209	0.7044	0.7253	511.8	694.9	1196.7	0.7135	0.7004	1.4139	800
850	525.21	0.0212	0.7080	0.7292	520.8	674.2	1195.0	0.7225	0.6846	1.4071	850
900	531.95	0.0213	0.7156	0.7369	529.5	663.8	1193.3	0.7311	0.6694	1.4005	900
950	538.40	0.0216	0.7264	0.7480	537.9	653.5	1191.4	0.7394	0.6548	1.3942	950
1000	544.58	0.0217	0.7402	0.7619	546.0	643.5	1189.6	0.7473	0.6408	1.3881	1000
1050	550.53	0.0219	0.7560	0.7780	554.0	633.6	1187.6	0.7550	0.6273	1.3822	1050
1100	556.28	0.0222	0.7738	0.7960	561.7	623.9	1185.6	0.7624	0.6141	1.3765	1100
1150	561.81	0.0224	0.7940	0.8164	569.2	614.3	1183.5	0.7695	0.6014	1.3709	1150
1200	567.14	0.0226	0.8166	0.8386	576.5	604.9	1181.4	0.7764	0.5891	1.3656	1200
1250	572.30	0.0228	0.8415	0.8627	583.6	595.6	1179.2	0.7831	0.5772	1.3603	1250
1300	577.32	0.0230	0.8684	0.8884	590.6	586.3	1177.0	0.7897	0.5654	1.3552	1300
1350	582.21	0.0232	0.8974	0.9159	597.5	577.2	1174.7	0.7962	0.5540	1.3501	1350
1400	586.96	0.0235	0.9288	0.9453	604.3	568.1	1172.4	0.8024	0.5428	1.3452	1400
1450	591.58	0.0237	0.9621	0.9768	611.0	559.1	1170.0	0.8086	0.5318	1.3404	1450
1500	596.08	0.0239	0.9984	1.0107	617.5	550.2	1167.6	0.8146	0.5212	1.3357	1500
1600	604.74	0.0244	1.0644	1.0800	630.2	532.6	1162.7	0.8262	0.5003	1.3265	1600
1700	612.98	0.0249	1.1389	1.1600	642.5	515.0	1157.5	0.8373	0.4801	1.3174	1700
1800	620.86	0.0254	1.2224	1.2500	654.7	497.2	1151.8	0.8482	0.4601	1.3083	1800
1900	628.39	0.0260	1.3144	1.3500	666.8	479.9	1145.7	0.8589	0.4402	1.2990	1900
2000	635.65	0.0265	1.4144	1.4500	679.0	460.0	1139.0	0.8696	0.4200	1.2900	2000
2100	642.6	0.0271	1.5224	1.5500	691.3	444.4	1131.7	0.8804	0.3996	1.2800	2100
2200	649.2	0.0277	1.6384	1.6500	703.7	420.0	1123.8	0.8912	0.3788	1.2700	2200
2300	655.7	0.0284	1.7624	1.7500	716.4	398.7	1115.2	0.9021	0.3575	1.2596	2300
2400	661.9	0.0292	1.8944	1.8500	729.2	376.4	1105.8	0.9133	0.3356	1.2488	2400
2500	668.0	0.0301	2.0344	1.9000	742.8	352.8	1095.6	0.9247	0.3129	1.2375	2500
2600	673.8	0.0310	2.1824	1.9500	756.7	327.8	1084.5	0.9364	0.2892	1.2257	2600
2700	679.5	0.0321	2.3384	2.0000	771.2	301.2	1072.5	0.9487	0.2644	1.2131	2700
2800	684.9	0.0331	2.5024	2.0500	786.7	272.3	1059.9	0.9618	0.2379	1.1996	2800
2900	690.2	0.0343	2.6744	2.1000	803.6	240.0	1043.7	0.9760	0.2088	1.1847	2900
3000	695.2	0.0357	2.8544	2.1500	823.1	202.5	1025.6	0.9922	0.1754	1.1676	3000
3100	700.2	0.0372	3.0424	2.2000	847.2	155.0	1002.2	1.0126	0.1336	1.1462	3100
3200	704.9	0.0388	3.2384	2.2500	877.0	75.9	962.9	1.0461	0.0651	1.1112	3200
3286 ¹	706.1	0.0522	0	0.0522	925.0	0	925.0	1.0785	0	1.0785	3286

¹ Critical pressure

REFRIGERANT PROPERTIES

TABLE 8-176
SUPERHEATED STEAM

Abs. Press., lb/ in. ² (Sat'n Temp.)	Sat'd Water	Sat'd Steam	Temperature — Degrees Fahrenheit												
			130°	200°	230°	240°	260°	280°	300°	400°	500°	600°	700°	800°	900°
SA 1 (101.76)	0.02 69.7 0.1326	333.9 1105.0 1113.3 1.9915	118.24 344.8 1149.8 2.0638	138.24 416.4 1167.8 2.0770	158.24 428.2 1176.9 2.0898	178.24 440.2 1185.9 2.1021	198.24 452.1 1195.0 2.1142	238.24 511.7 1240.7 2.2218	298.24 571.3 1287.2 2.2688	398.24 630.9 1334.6 2.3535	498.24 690.6 1383.0 2.3922	598.24 750.2 1432.6 2.4291	698.24 809.8 1482.9 2.4650	798.24 869.4 1534.3 2.5009	898.24 928.9 1585.7 2.5368
SA 5 (162.25)	0.02 130.1 0.2348	73.61 1130.6 1143.0 1.8435	57.75 80.58 1148.2 1.8712	77.75 82.99 1166.7 1.8984	97.75 85.39 1175.9 1.9114	117.75 87.80 1185.0 1.9239	137.75 90.21 1194.2 1.9361	237.75 102.19 1246.9 2.0443	337.75 114.16 1286.9 2.0443	437.75 126.11 1334.4 2.0614	537.75 138.05 1382.9 2.1351	637.75 149.99 1432.5 2.1761	737.75 161.91 1483.2 2.2149	837.75 173.83 1535.1 2.2517	937.75 185.75 1587.0 2.2885
SA 10 (193.21)	0.02 161.1 0.2834	38.45 1143.0 1150.2 1.7874	26.79 40.10 1155.8 1.8068	46.79 41.31 1174.6 1.8204	66.79 43.75 1183.9 1.8336	86.79 44.98 1193.2 1.8464	106.79 51.01 1239.6 1.8587	206.79 63.01 1286.6 1.9161	306.79 74.96 1334.6 2.0148	406.79 88.99 1382.6 2.0148	506.79 102.82 1432.1 2.0686	606.79 116.67 1482.9 2.0997	706.79 130.53 1534.9 2.1330	806.79 144.40 1587.0 2.1653	906.79 158.27 1639.1 2.1985
SA 14.696 (212.00)	0.02 180.0 0.3119	26.82 1150.2 1156.4 1.7564	8.00 27.16 1154.1 1.7623	28.00 27.98 1173.4 1.7764	48.00 29.67 1182.8 1.7898	68.00 30.52 1192.2 1.8028	88.00 34.65 1239.0 1.8154	188.00 42.83 1286.0 1.8731	288.00 51.01 1333.7 1.9249	388.00 63.01 1382.4 1.9722	488.00 74.96 1432.1 2.0161	588.00 88.99 1482.9 2.0611	688.00 102.82 1534.9 2.1061	788.00 116.67 1587.0 2.1511	888.00 130.53 1639.1 2.1961
SA 18 (213.03)	0.02 181.0 0.3134	26.31 1150.6 1156.8 1.7545	6.97 26.60 1154.0 1.7599	46.97 27.42 1173.3 1.7740	66.97 29.07 1182.7 1.7885	86.97 29.90 1192.2 1.8005	106.97 33.95 1238.9 1.8130	186.97 41.98 1286.0 1.8708	286.97 53.93 1333.7 1.9228	386.97 65.88 1382.4 1.9699	486.97 77.83 1432.1 2.0138	586.97 89.78 1482.9 2.0550	686.97 101.73 1534.9 2.0988	786.97 113.68 1587.0 2.1437	886.97 125.63 1639.1 2.1887
SA 20 (227.96)	0.02 196.1 0.3356	20.10 1156.0 1163.7 1.7317	12.04 20.48 1162.0 1.7403	52.04 21.11 1181.5 1.7543	72.04 22.36 1191.1 1.7675	92.04 25.43 1239.0 1.7802	112.04 28.50 1286.0 1.8134	212.04 31.46 1333.4 1.8667	312.04 43.41 1382.1 1.9312	412.04 55.36 1432.1 1.9819	512.04 67.31 1482.9 2.0322	612.04 79.26 1534.9 2.0827	712.04 91.21 1587.0 2.1332	812.04 103.16 1639.1 2.1837	912.04 115.11 1691.2 2.2342
SA 25 (240.07)	0.017 208.3 0.3533	16.306 1160.2 1171.7 1.7137	19.93 16.82 1170.3 1.7260	39.93 17.33 1180.2 1.7416	59.93 18.68 1190.1 1.7544	79.93 20.03 1239.0 1.7675	99.93 22.36 1286.0 1.7802	159.93 25.43 1333.4 1.8134	259.93 28.50 1382.1 1.8667	359.93 31.46 1432.1 1.9312	459.93 43.41 1482.9 1.9819	559.93 55.36 1534.9 2.0322	659.93 67.31 1587.0 2.0827	759.93 79.26 1639.1 2.1332	859.93 91.21 1691.2 2.1837
SA 30 (250.34)	0.017 218.7 0.3680	13.745 1163.7 1173.7 1.6990	9.66 13.96 1168.7 1.7061	29.56 14.39 1178.9 1.7201	49.56 15.82 1188.2 1.7332	69.56 17.25 1239.0 1.7464	89.56 19.10 1286.0 1.7602	149.56 22.36 1333.4 1.8134	249.56 25.43 1382.1 1.8667	349.56 28.50 1432.1 1.9312	449.56 40.43 1482.9 1.9819	549.56 52.38 1534.9 2.0322	649.56 64.33 1587.0 2.0827	749.56 76.28 1639.1 2.1332	849.56 88.23 1691.2 2.1837

TABLE 8-17b (Continued)
SUPERHEATED STEAM

Abs. Press., lb/in. ² (Sat'n. Temp.)	Sat'd Water	Sat'd Steam	Temperature—Degrees Fahrenheit												
			150°	200°	300°	400°	500°	600°	700°	800°	900°	1000°			
35 (250.28)	S _h	11.898	0.72	40.72	140.72	240.72	340.72	440.72	540.72	640.72	740.72	840.72	940.72	1040.72	
	A	11.913	12.201	12.662	14.452	16.202	17.935	19.657	21.366	23.062	24.747	26.421	28.084	29.736	
	s	1167.0	1177.5	1187.7	1236.3	1284.2	1331.4	1378.1	1424.3	1470.0	1515.3	1560.2	1604.7	1648.8	1692.5
40 (267.24)	S _h	1.6866	1.6871	1.7016	1.7752	1.8276	1.8756	1.9199	1.9612	2.0002	2.0371				
	A	10.497	10.714	11.044	12.623	14.161	15.682	17.190	18.686	20.168	21.637	23.094	24.539	25.973	27.397
	s	1169.2	1176.1	1186.5	1235.6	1283.7	1331.2	1378.2	1424.7	1470.7	1516.3	1561.5	1606.3	1650.8	1695.0
45 (274.45)	S _h	0.017	5.55	25.55	125.55	225.55	325.55	425.55	525.55	625.55	725.55				
	A	9.399	9.485	9.785	11.200	12.574	13.929	15.272	16.602	17.936	19.269	20.600	21.929	23.256	
	s	1171.5	1174.5	1185.2	1234.9	1283.2	1331.7	1380.9	1431.1	1482.2	1534.3	1587.4	1641.5	1696.6	
50 (281.01)	S _h	0.4020	1.6707	1.6348	1.7463	1.7995	1.8475	1.8919	1.9334	1.9724	2.0094				
	A	8.514	8.777	10.061	11.899	13.804	15.774	17.814	19.921	21.099	22.454	23.886	25.296	26.685	
	s	1173.5	1183.9	1234.2	1282.8	1331.4	1380.7	1431.0	1482.1	1534.1	1587.1	1641.1	1696.1	1751.1	
55 (287.07)	S _h	0.4111	1.6718	1.7341	1.7875	1.8357	1.8801	1.9217	1.9608	1.9977					
	A	7.783	7.950	9.130	10.264	11.379	12.481	13.571	14.665	15.757	16.846	17.932	19.016	20.098	
	s	1175.3	1182.6	1233.5	1282.3	1331.0	1380.5	1430.8	1481.9	1534.0	1587.2	1641.4	1696.7	1752.0	
60 (292.71)	S _h	0.4194	1.6589	1.7230	1.7767	1.8249	1.8695	1.9111	1.9502	1.9872					
	A	7.172	7.29	107.29	207.29	307.29	407.29	507.29	607.29	707.29	807.29	907.29	1007.29	1107.29	
	s	1177.0	1181.2	1232.8	1281.9	1330.2	1378.2	1426.6	1475.4	1524.6	1574.3	1624.5	1675.1	1726.2	
65 (297.97)	S _h	0.4271	1.6488	1.7128	1.7667	1.8151	1.8597	1.9014	1.9406	1.9775					
	A	6.652	6.674	7.696	8.363	9.398	10.423	11.455	12.436	13.459	14.440	15.440	16.440	17.440	
	s	1178.5	1179.7	1231.4	1280.2	1328.9	1377.2	1425.9	1475.0	1524.4	1574.1	1624.1	1674.3	1724.6	
70 (302.92)	S _h	0.4343	1.6385	1.7033	1.7576	1.8060	1.8508	1.8926	1.9317	1.9687					
	A	6.203	7.08	197.08	297.08	397.08	497.08	597.08	697.08	797.08	897.08	997.08	1097.08	1197.08	
	s	1179.9	1183.4	1235.0	1283.0	1331.0	1379.7	1428.0	1476.8	1526.0	1575.5	1625.3	1675.4	1725.7	
75 (307.60)	S _h	0.4410	1.6385	1.7033	1.7576	1.8060	1.8508	1.8926	1.9317	1.9687					
	A	5.813	92.40	192.40	292.40	392.40	492.40	592.40	692.40	792.40	892.40	992.40	1092.40	1192.40	
	s	1181.2	1230.4	1278.4	1326.4	1374.4	1422.4	1470.4	1518.4	1566.4	1614.4	1662.4	1710.4	1758.4	
(307.60)	S _h	0.4473	1.6386	1.7410	1.7898	1.8347	1.8752	1.9111	1.9435	1.9724					
	A	5.418	6.444	7.492	8.319	9.000	9.582	10.104	10.576	11.000	11.376	11.704	12.000	12.272	
	s	1182.4	1230.7	1278.7	1326.7	1374.7	1422.7	1470.7	1518.7	1566.7	1614.7	1662.7	1710.7	1758.7	

TABLE 8-17b (Continued)
SUPERHEATED STEAM

Abs. Press. lb./in. ² (Sat'g Temp.)	Sat'd Water	Sat'd Steam	Temperature — Degrees Fahrenheit												
			330°	340°	350°	360°	370°	380°	400°	500°	600°	700°	800°	900°	1000°
80 (312.02)	0.018 281.9 0.4532	5.470 1182.4 1.6202	7.97 5.631 1192.8 1.6262	27.97 5.720 1198.4 1.6405	37.97 5.806 1209.2 1.6473	47.97 5.889 1219.4 1.6538	57.97 5.973 1229.8 1.6602	67.97 6.055 1239.9 1.6665	87.97 6.217 1249.9 1.6785	187.97 7.015 1290.0 1.7336	287.97 7.793 1379.2 1.7824	387.97 8.558 1479.1 1.8274	487.97 9.313 1533.5 1.8694	587.97 10.067 1583.6 1.9086	687.97 10.817 1633.1 1.9456
85 (316.25)	0.018 286.3 0.4589	5.164 1183.6 1.6184	8.75 5.198 1191.6 1.6182	23.75 5.283 1197.3 1.6327	33.75 5.349 1202.8 1.6396	43.75 5.408 1208.2 1.6462	53.75 5.465 1213.5 1.6526	63.75 5.520 1218.8 1.6580	83.75 5.685 1229.2 1.6712	183.75 6.504 1279.5 1.7365	283.75 7.329 1329.0 1.7755	383.75 8.050 1429.8 1.8206	483.75 8.762 1429.8 1.8626	583.75 9.472 1481.1 1.9019	683.75 10.178 1533.4 1.9389
90 (320.27)	0.018 290.4 0.4642	4.892 1184.6 1.6107	9.73 4.973 1190.4 1.6180	19.73 5.055 1196.2 1.6253	29.73 5.132 1201.8 1.6323	39.73 5.208 1207.2 1.6390	49.73 5.283 1212.6 1.6455	59.73 5.357 1218.0 1.6519	79.73 5.504 1228.5 1.6642	179.73 6.220 1279.0 1.7198	279.73 6.916 1328.6 1.7690	379.73 7.599 1429.6 1.8142	479.73 8.272 1429.6 1.8563	579.73 8.943 1481.0 1.8955	679.73 9.610 1533.3 1.9326
95 (324.13)	0.018 294.5 0.4693	4.647 1185.6 1.6064	15.87 4.695 1189.2 1.6107	25.87 4.773 1194.9 1.6182	35.87 4.849 1200.7 1.6252	45.87 4.923 1206.3 1.6320	55.87 5.003 1211.7 1.6386	65.87 5.075 1217.1 1.6451	75.87 5.205 1227.7 1.6575	175.87 5.886 1278.5 1.7135	275.87 6.547 1328.3 1.7623	375.87 7.195 1429.4 1.8081	475.87 7.884 1429.4 1.8502	575.87 8.470 1480.9 1.8895	675.87 9.102 1533.2 1.9266
100 (327.83)	0.018 298.8 0.4742	4.426 1186.6 1.6022	2.17 4.445 1187.9 1.6039	12.17 4.520 1193.9 1.6114	22.17 4.594 1199.7 1.6185	32.17 4.663 1205.3 1.6254	42.17 4.731 1210.8 1.6322	52.17 4.800 1216.2 1.6387	62.17 4.934 1221.9 1.6512	172.17 5.585 1275.0 1.7073	272.17 6.215 1327.9 1.7569	372.17 6.851 1429.2 1.8023	472.17 7.439 1429.2 1.8445	572.17 8.044 1480.7 1.8838	672.17 8.644 1533.1 1.9209
105 (330.27)	0.018 299.8 0.4761	4.244 1186.9 1.6006	0.73 4.352 1187.4 1.6012	10.73 4.426 1193.4 1.6088	20.73 4.498 1199.2 1.6159	30.73 4.566 1204.0 1.6228	40.73 4.634 1210.4 1.6296	50.73 4.701 1215.9 1.6382	60.73 4.833 1226.8 1.6488	170.73 5.473 1277.8 1.7052	270.73 6.101 1327.8 1.7546	370.73 6.703 1429.1 1.8000	470.73 7.289 1429.1 1.8423	570.73 7.865 1481.7 1.8816	670.73 8.474 1533.0 1.9187
104 (330.68)	0.018 301.3 0.4779	4.265 1187.3 1.5990	9.32 4.335 1193.0 1.6061	19.32 4.406 1198.8 1.6133	29.32 4.473 1204.5 1.6203	39.32 4.540 1210.0 1.6271	49.32 4.606 1215.5 1.6337	59.32 4.736 1226.3 1.6463	69.32 4.855 1237.7 1.6592	189.32 5.385 1277.7 1.7029	289.32 5.972 1327.6 1.7524	389.32 6.566 1429.1 1.7978	489.32 7.151 1429.1 1.8402	589.32 7.732 1480.6 1.8794	689.32 8.310 1533.0 1.9196
106 (332.06)	0.018 302.8 0.4798	4.189 1187.6 1.5974	7.92 4.248 1192.5 1.6036	17.92 4.318 1204.1 1.6107	27.92 4.384 1209.7 1.6178	37.92 4.450 1215.2 1.6246	47.92 4.515 1220.8 1.6313	57.92 4.643 1228.0 1.6440	67.92 4.857 1237.5 1.6502	167.92 5.261 1277.5 1.7006	267.92 5.857 1327.5 1.7502	367.92 6.440 1429.0 1.7957	467.92 7.015 1429.0 1.8380	567.92 7.585 1480.8 1.8773	667.92 8.153 1533.3 1.9145

TABLE 8-17b (Continued)
SUPERHEATED STEAM

Abs. Press. (Sat'n Temp.)	Sat'd Water	Sat'd Steam	Temperature—Degrees Fahrenheit															
			330°	330°	340°	350°	360°	370°	380°	400°	500°	600°	700°	800°	900°	1000°		
108 (333.44)	S	0.018	6.52	26.58	36.56	46.56	66.56	169.56	266.56	366.56	466.56	566.56	666.56	766.56	866.56	966.56	1066.56	
	A	304.2	4.163	4.293	4.437	4.591	4.754	4.927	5.101	5.274	5.447	5.620	5.793	5.966	6.139	6.312	6.485	
	h	1192.0	1197.9	1203.7	1209.3	1214.8	1220.7	1226.7	1232.7	1238.7	1244.7	1250.7	1256.7	1262.7	1268.7	1274.7	1280.7	1286.7
	s	1.6010	1.6082	1.6155	1.6222	1.6289	1.6356	1.6416	1.6484	1.6551	1.6618	1.6686	1.6753	1.6820	1.6887	1.6954	1.7021	1.7088
110 (334.70)	S	0.018	5.21	25.21	35.21	45.21	65.21	165.21	265.21	365.21	465.21	565.21	665.21	765.21	865.21	965.21	1065.21	
	A	305.6	4.082	4.151	4.216	4.279	4.343	4.407	4.471	4.535	4.599	4.663	4.727	4.791	4.855	4.919	4.983	5.047
	h	1191.0	1197.0	1203.0	1208.9	1214.5	1220.5	1226.4	1232.4	1238.4	1244.4	1250.4	1256.4	1262.4	1268.4	1274.4	1280.4	1286.4
	s	0.4834	1.5688	1.6058	1.6129	1.6198	1.6265	1.6333	1.6398	1.6463	1.6528	1.6593	1.6658	1.6723	1.6788	1.6853	1.6918	1.6983
113 (336.12)	S	0.018	3.88	13.88	23.88	33.88	53.88	153.88	253.88	353.88	453.88	553.88	653.88	753.88	853.88	953.88	1053.88	
	A	307.0	4.003	4.072	4.135	4.199	4.261	4.323	4.385	4.447	4.509	4.571	4.633	4.695	4.757	4.819	4.881	4.943
	h	1191.0	1197.0	1202.8	1208.5	1214.1	1219.5	1225.1	1230.7	1236.4	1242.0	1247.7	1253.4	1259.1	1264.8	1270.5	1276.2	1281.9
	s	0.4851	1.5861	1.6034	1.6105	1.6174	1.6242	1.6310	1.6377	1.6445	1.6513	1.6581	1.6649	1.6717	1.6785	1.6853	1.6921	1.6989
114 (337.43)	S	0.018	2.57	12.57	22.57	32.57	52.57	152.57	252.57	352.57	452.57	552.57	652.57	752.57	852.57	952.57	1052.57	
	A	308.4	3.928	3.996	4.059	4.120	4.182	4.243	4.304	4.365	4.426	4.487	4.548	4.609	4.670	4.731	4.792	4.853
	h	1190.5	1196.6	1202.4	1208.1	1213.7	1219.7	1224.7	1230.7	1236.7	1242.7	1248.7	1254.7	1260.7	1266.7	1272.7	1278.7	1284.7
	s	0.4868	1.5936	1.6010	1.6082	1.6151	1.6219	1.6288	1.6356	1.6425	1.6493	1.6562	1.6630	1.6699	1.6767	1.6836	1.6904	1.6973
116 (338.72)	S	0.018	1.28	11.28	21.28	31.28	51.28	151.28	251.28	351.28	451.28	551.28	651.28	751.28	851.28	951.28	1051.28	
	A	309.7	3.855	3.922	3.984	4.045	4.106	4.166	4.226	4.286	4.346	4.406	4.466	4.526	4.586	4.646	4.706	4.766
	h	1190.0	1196.1	1201.8	1207.3	1212.4	1217.5	1222.4	1227.5	1232.4	1237.5	1242.4	1247.5	1252.4	1257.5	1262.4	1267.5	1272.4
	s	0.4885	1.5912	1.5986	1.6058	1.6128	1.6196	1.6265	1.6334	1.6403	1.6472	1.6541	1.6610	1.6679	1.6748	1.6817	1.6886	1.6955
118 (340.01)	S	0.018	9.99	19.99	29.99	39.99	59.99	159.99	259.99	359.99	459.99	559.99	659.99	759.99	859.99	959.99	1059.99	
	A	311.0	3.850	3.912	3.973	4.032	4.150	4.213	4.274	4.335	4.396	4.457	4.518	4.579	4.640	4.701	4.762	4.823
	h	1189.5	1195.7	1201.6	1207.4	1213.0	1218.1	1223.1	1228.1	1233.1	1238.1	1243.1	1248.1	1253.1	1258.1	1263.1	1268.1	1273.1
	s	0.4901	1.5887	1.5963	1.6035	1.6106	1.6174	1.6243	1.6312	1.6381	1.6450	1.6519	1.6588	1.6657	1.6726	1.6795	1.6864	1.6933
120 (341.26)	S	0.018	8.74	18.74	28.74	38.74	58.74	158.74	258.74	358.74	458.74	558.74	658.74	758.74	858.74	958.74	1058.74	
	A	312.4	3.781	3.842	3.902	3.961	4.077	4.138	4.198	4.258	4.318	4.378	4.438	4.498	4.558	4.618	4.678	4.738
	h	1189.0	1195.2	1201.0	1206.7	1212.7	1218.4	1224.1	1229.8	1235.8	1241.4	1247.1	1252.8	1258.5	1264.2	1269.9	1275.6	1281.3
	s	0.4918	1.5874	1.5940	1.6013	1.6084	1.6152	1.6223	1.6293	1.6363	1.6433	1.6503	1.6573	1.6643	1.6713	1.6783	1.6853	1.6923
122 (342.50)	S	0.018	7.50	17.50	27.50	37.50	57.50	157.50	257.50	357.50	457.50	557.50	657.50	757.50	857.50	957.50	1057.50	
	A	313.7	3.715	3.775	3.834	3.893	4.007	4.064	4.122	4.179	4.237	4.294	4.352	4.410	4.468	4.526	4.584	4.642
	h	1189.0	1194.7	1200.7	1206.6	1212.3	1218.5	1224.5	1230.5	1236.5	1242.5	1248.5	1254.5	1260.5	1266.5	1272.5	1278.5	1284.5
	s	0.4934	1.5917	1.5990	1.6062	1.6131	1.6202	1.6272	1.6343	1.6413	1.6483	1.6553	1.6623	1.6693	1.6763	1.6833	1.6903	1.6973
124 (343.73)	S	0.018	6.27	16.27	26.27	36.27	56.27	156.27	256.27	356.27	456.27	556.27	656.27	756.27	856.27	956.27	1056.27	
	A	315.0	3.650	3.710	3.769	3.826	3.939	4.000	4.061	4.122	4.183	4.244	4.305	4.366	4.427	4.488	4.549	4.610
	h	1189.0	1194.3	1200.3	1206.2	1212.1	1218.1	1224.1	1230.1	1236.1	1242.1	1248.1	1254.1	1260.1	1266.1	1272.1	1278.1	1284.1
	s	0.4950	1.5895	1.5968	1.6040	1.6110	1.6181	1.6252	1.6323	1.6394	1.6465	1.6536	1.6607	1.6678	1.6749	1.6820	1.6891	1.6962

The calculations, using the physical properties of the refrigerants, are reasonably good for ammonia, sulphur dioxide, methyl chloride, and the Freon group. The properties of carbon dioxide in the superheated region are still unsatisfactory. The physical properties of the other refrigerants, especially where the superheated region is doubtful or unknown, permit only approximate calculations of refrigerating performance.

Figure 8·2 and the three different cycles indicate clearly what can be expected of carbon dioxide where cooling water is in excess of 80° or 85° F. The figure shows the action of the pressure-reducing valve *ab* and *fg* and the resulting large value of *x*, the decimal part by weight of carbon dioxide that is in the vapor phase, at the point where the absorption of heat in the evaporator begins. Cycle *fgcdahf*, devised by R. Plank, has two compressions and two condenser actions. The second compression, *ah*, makes it possible to increase the refrigeration from 42.3 to 60.3 Btu/lb. A similar gain in refrigeration would be possible with the dual compression cycle.

Tables of the physical properties of refrigerants are taken from the following sources: The properties of ammonia to -60° F are taken from *Circular 142* of the Bureau of Standards; the values for ammonia at -60° to -105° were extrapolated from the circular by Bernard E. Oldham of London, England. The properties of the Freon group, except Freon-12, are taken from the publications of Kinetic Chemicals, and those for methyl chloride from the R and H Chemical Department, both of E. I. du Pont de Nemours and Company. The other tables of refrigerants are taken from the *Refrigerating Data Book* published by the American Society of Refrigerating Engineers.

PROBLEMS

1. Ten pounds per minute of ammonia are discharged from a desuperheater as a saturated *vapor* at 100° F and immediately expanded (through a throttling valve) to 15 psia. The vapor leaving the expansion valve immediately enters a compressor and is isentropically compressed to the desuperheater pressure. (a) Determine the coefficient of performance of this system. (b) Note that there is heat flow from, but not to, the system and discuss the thermodynamic significance of this.

2. The constant temperature lines in the superheat region of the pressure-enthalpy chart for ammonia approach verticality at low pressures. Discuss the thermodynamic significance of this in terms of internal potential energy.

3. Discuss the thermodynamic significance of the enthalpy increase that occurs when the pressure on saturated liquid ammonia at -50° F is raised isothermally to 300 psia. If the fluid is at rest, does this enthalpy increase correspond to an increase of energy in the system or is it related to the work needed to raise the pressure? Explain.

4. An inventor proposes to replace the compression process in a standard ammonia

reciprocating compressor with a constant volume process which would bring about the necessary pressure rise. Evaporator pressure is 58 psia and condenser pressure is 100 psia. Vapor enters the compressor (zero clearance) in a saturated state and does work on the piston just as in any other admission process. When the piston reaches dead center, the suction valves close and the pressure is raised at constant volume. Discharge then occurs in the usual way with work done on the high-pressure vapor by the piston. (a) Compare the requirements of shaft work for the proposed cycle with the standard compression cycle. (b) Compare the total energy requirements for the two cycles. (c) Discuss the merits and the deficiencies of the proposed system and explain in thermodynamic terms.

5. The saturated liquid refrigerant entering an evaporator has a lower entropy than the refrigerant entering the expansion valve, yet entropy is said to increase as a result of an irreversible process. Explain this seeming inconsistency.

6. An inventor proposes a refrigeration system which is to use one double-acting cylinder with substantial clearance on both sides. With the piston at the end of its travel to the right, vapor from the evaporator is to be admitted irreversibly on both sides of the piston, and when the cylinder pressure is raised to the evaporator pressure the suction valves on both sides will be closed and heat will be admitted (the piston remaining on dead center) until the pressure of the vapor on each side has been raised by a constant volume process to the condenser pressure. At this point in the cycle the heat addition on the left side of the piston will cease, but heat will continue to be added to the vapor on the right side, allowing it to expand at constant pressure and thereby move the piston to the left discharging vapor from the left side, through the discharge valves, to the condenser. When the piston reaches dead center on the left the vapor space on both sides will already be filled with high-pressure vapor, hence heat addition to the space on the left (with valves closed) will force the piston to the right, discharging vapor from the space on the right side of the piston. Further heat addition on the right side will then force the piston to the left with consequent left-side discharge. Discuss this proposal and investigate other means of accomplishing the inventor's objective, which is compression without need of shaft work.

7. For the proposed cycle of Problem 6 suppose that the high-pressure vapor in the clearance space were irreversibly discharged to the evaporator pressure through a throttling valve and means were provided to "purge" the clearance space and then refill it with low-pressure vapor from the evaporator. Would this system be possessed of any thermodynamic advantage?

Part II
LOAD DETERMINATION

CHAPTER IX

CONDUCTION AND CONVECTION

9·1. Conduction. Energy transfer through solid bodies is due primarily to non-uniform distribution of internal energy within the body; such transfer denotes an unstable state and one which can exist steadily only so long as conditions external to the body remain in non-equilibrium. Thus if a metal rod were heated at one end and cooled at the other a non-uniform distribution of internal energy would exist, and since the availability of the energy at the hot end would be greater than that at the cold end a transfer of energy would occur toward the cold end by the mechanism of conduction. Therefore if the temperature difference within the rod is to be maintained it is necessary that each end of the rod be acted on by an exterior heat source or sink to supply and extract energy as rapidly as the transfer within the rod tends to equalize the temperature. Heat transfer can occur only during non-equilibrium and will be steadily maintained only so long as a dynamic balance, due to the action of fixed non-equilibrium boundary conditions, exists.

For steady-state heat transfer by conduction the classic equation of Fourier can be written in the form

$$q_c = \frac{kA}{L} (t_h - t_c) \quad \text{Btu/hr} \quad (9\cdot1)$$

where q_c is the rate of transfer by conduction, k is the thermal conductivity, A the cross-section area normal to the path of heat flow, L the length of path, and $t_h - t_c$ the difference in temperature between the hot and cold ends of the path. Solving equation 9·1 for the thermal conductivity gives

$$k = \frac{q_c L}{A (t_h - t_c)} \quad (9\cdot2)$$

which shows that k is numerically equal to the number of Btu that would flow through a cube of unit area if unit temperature difference were impressed on opposite faces and if the four remaining sides were perfectly insulated; more generally, k is the heat transfer through any insulated rod having area numerically equal to path length and unit temperature difference impressed across its length. Dimensionally,

k is the number of Btu transferred per hour per square foot per degree temperature drop in unit length of path:

$$k = \text{Btu}/(\text{hr})(\text{sq ft})(^\circ\text{F}/\text{ft})$$

As a matter of practical convenience, values of thermal conductivity are sometimes given in terms of ($^\circ\text{F}/\text{in.}$) rather than ($^\circ\text{F}/\text{ft}$); when these values are used in equation 9.1 the length L is expressed in inches instead of in feet. Conductivity in $\text{Btu}/(\text{hr})(\text{sq ft})(^\circ\text{F}/\text{in.})$ is equal to twelve times conductivity in $\text{Btu}/(\text{hr})(\text{sq ft})(^\circ\text{F}/\text{ft})$.

The variation in values of thermal conductivity among common materials is of the order of 10,000 to 1; hence it is vital that accurate values of k be used in analysis. Cork, for example, has a conductivity of approximately 0.024 as compared with approximately 240 for silver; a 1-ft wall of cork would provide as much insulation as a wall of silver 2 miles thick, yet—ridiculous though this example is—it does point out the very important fact that *all* materials offer a resistance to conductive heat transfer and hence have insulating value. The distinction between a conductor and an insulator is really a distinction between two insulators of relatively small and relatively great effectiveness.

For some problems the area normal to the path of heat transfer varies with the length of path; in such cases the temperature drop will not be a linear function of path length, and equation 9.1 must then be written in differential form

$$q_c = -k \int A \frac{dt}{dL} \quad \text{Btu/hr} \quad (9.3)$$

where the negative sign indicates decreasing temperature with increasing L . Integration of equation 9.3 requires that the area be expressed as a function of path length, and the temperature drop likewise. The thermal conductivity is taken outside the integral because, for the temperature ranges important in refrigeration, the variation of k as a function of temperature is entirely negligible. Integrating equation 9.3 gives

$$q_c = -k \int A \frac{dt}{dL} = \frac{kA_{av} (t_1 - t_2)}{L_2 - L_1} \quad (9.4)$$

where A = area of section normal to heat flow.

For very thick walls of pipes, where r_1 is the inside and r_2 is the outside radius, for a length of pipe N ,

$$q_c = -k2\pi r N \frac{dt}{dr} \quad \text{and} \quad \frac{dr}{r} = -\frac{k2\pi N dt}{q_c}$$

$$\log_e \frac{r_2}{r_1} = \frac{k2\pi N}{q_c} (t_1 - t_2) \quad \text{Btu/hr}$$

and

$$q_c = \frac{k2\pi N (t_1 - t_2)}{\log_e \frac{r_2}{r_1}} = \frac{kA_{av.} (t_1 - t_2)}{L_2 - L_1} \text{ Btu/hr}$$

or

$$A_{av.} = \frac{2\pi N (L_2 - L_1)}{\log_e \frac{r_2}{r_1}} = \frac{A_2 - A_1}{\log_e \frac{A_2}{A_1}} \quad (9.5)$$

However, except where accurate results are desired, the arithmetical mean of the areas, $A_m = \frac{1}{2}(A_1 + A_2)$, is ordinarily good enough in practice as the error in the use of the arithmetical mean is about 1 per cent when the ratio A_2/A_1 is less than 1.5.

9.2. The Mean Temperature Difference. In the flow of heat the loss of heat by one body is exactly equal to the amount gained by another, or

$$dQ_2 = -dQ_1 = w_2c_2 dt_2 = -w_1c_1 dt_1 \quad \text{Btu} \quad (9.6)$$

Therefore

$$\frac{dt_2}{dt_1} = -\frac{w_1c_1}{w_2c_2}$$

and

$$\frac{dt_2 - dt_1}{dt_1} = \frac{d\theta}{dt_1} = -\frac{w_1c_1 - w_2c_2}{w_2c_2}$$

where θ is the temperature difference $t_2 - t_1$ and w is the weight in pounds, c the specific heat.

$$dt_1 = -\frac{d\theta}{\frac{w_1c_1 - w_2c_2}{w_2c_2}} = \frac{dQ_1}{w_1c_1}$$

Therefore

$$\begin{aligned} dQ_1 &= -w_1c_1 \frac{w_2c_2}{w_1c_1 - w_2c_2} d\theta \\ &= -\frac{w_1c_1 \times w_2c_2}{w_1c_1 - w_2c_2} d\theta \\ &= -B d\theta \end{aligned}$$

or

$$\frac{dQ_1}{d\theta} = -B$$

and

$$-Q_2 = Q_1 = -B \int_{\theta}^{\theta_a} d\theta = -B(\theta_a - \theta) \quad (9.7)$$

Let A equal the average area exposed to heat transfer per foot of length of tube or pipe and U equal the overall coefficient of heat transfer.* Then $A dL = \text{area}$, $dQ_1 = A dLU\theta$. Multiplying the reciprocal by $d\theta$,

$$\frac{d\theta}{dQ_1} = \frac{1}{A dLU} \frac{d\theta}{\theta} = \frac{1}{-B}$$

giving

$$dL = \frac{-B d\theta}{AU \theta}$$

from which

$$L = -\frac{B}{AU} \log_e \frac{\theta_a}{\theta}$$

Taking for some average condition θ_m ,

$$Q_1 = ALU\theta_m$$

and remembering that

$$Q_1 = -B(\theta_a - \theta)$$

it follows that

$$\theta_m = \frac{Q_1}{UAL} = \frac{B(\theta_a - \theta)}{UA \frac{B}{AU} \log_e \frac{\theta_a}{\theta}}$$

Therefore

$$\theta_m = \frac{\theta_a - \theta}{\log_e \frac{\theta_a}{\theta}}$$

and when θ becomes θ_b

$$\theta_m = \frac{\theta_a - \theta_b}{\log_e \frac{\theta_a}{\theta_b}} \quad (9.8)$$

If the ratio θ_a/θ_b is equal to 1.5, the arithmetical mean will be 1 per cent in error, and this increases rapidly for larger values of the ratio.

9.3. Conduction through Solids. Conduction of heat through pure metals and alloys (Tables 9.1, 9.2, and 9.3) appears to be affected both by the vibration of the molecules and by the passage of free electrons

* This term is defined in Section 10.1, but its evaluation is unimportant for present purposes as it will be found to cancel out in a later step of the derivation.

TABLE 9-1
CONDUCTIVITY OF CERTAIN METALS AT 68° BTU/(HR)(SQ FT)(°F/IN.)

	<i>k</i>
Aluminum	1300 to 1460
Bearing metal	164
Brass (70% Cu, 30% Zn)	770
Constantan (60% Cu, 40% Ni)	162
Copper	2665
Gold	2050
Iron, gray cast	350
Iron, wrought	420
Lead	241
Mercury	57
Monel metal (70% Ni, 28% Cu, 2% Fe)	241
Nichrome (82% Ni, 12% Cr, 26% Fe)	94
Nickel	405
Platinum	483
Silver	2840
Steel (1% C)	300
Tin	440
Tungsten	1110
Zinc	770

TABLE 9-2
CONDUCTIVITY OF CERTAIN INSULATING MATERIALS (DRY)
[Btu/(hr)(sq ft)(°F/in.)]

	Density	<i>k</i> at 90° F
Aluminum foil (3 air spaces per in.)	0.2	0.29
Balsa wood	8.9	0.38
Balsam wool	2.2	0.27
Celotex	13.2	0.34
Corkboard ¹	10.6	0.30
Dry zero	1.0	0.25
Hair felt	11.0	0.26
Insulite	16.9	0.34
Paper, corrugated	3.5	0.27
Rock wool	10.0	0.27

¹ The conductivity increases with the density and method of manufacture.

TABLE 9·3

CONDUCTIVITY OF CERTAIN INDUSTRIAL MATERIALS AT NORMAL TEMPERATURES
[Btu/(hr)(sq ft)(°F/in.)]

	<i>k</i>
Boiler scale	1.2 to 60
Brick, building	1.5 to 5.0
Brick, refractory	4.0 to 15.0
Concrete	5.0 to 15.0
Cotton	0.3 to 0.5
Earth	1.0 to 4.5
Glass	4.2 to 5.8
Ice at 32° F	14.6 to 15.6
Wood across grain	0.7 to 1.4
Wool	0.24 to 0.26

through the lattice of the molecules. For pure metals thermal conduction decreases slightly with the temperature, whereas for alloys which have lower conductivities than the pure metal conduction tends to increase with temperature except in cast iron and wrought iron. A little arsenic or phosphorus may reduce the conductivity of copper to a point where it is little better than that of steel. The grain structure influences the conductivity especially of alloys, where annealing increases and quenching usually decreases the conductivity.

In insulating materials the conductivity usually increases with the temperature, the amount of moisture and the density, and good insulators are usually porous, cellular, fibrous or laminated material. As a rule insulation is increased by having minute air cells and decreased by large cells where convection may occur.

9·4. Conduction through Liquids. The conductivity of certain important liquids listed in Table 9·4, between 32° and 200° F, with a few exceptions like water, sulphur dioxide, and ammonia, is close to unity, and also as a rule the value of the conductivity *k* decreases as the temperature rises. The variation of the coefficient is given by the linear equation

$$k_t = k_{68}[1 + a(t - 68)] \quad (9\cdot9)$$

for *t* from 32° to 200° F; for *a* see Table 9·4. Under pressures usual in engineering the value of *k* remains practically unchanged, and *k* for a mixture of two or more liquids† may be found approximately by the equation

$$k = \frac{p_1 k_1 + p_2 k_2}{p_1 + p_2} \quad (9\cdot10)$$

where *p*₁ and *p*₂ are parts of each component by volume.

† C. H. Lees, *Phil. Mag.*, Vol. 49, p. 286, 1900.

TABLE 9.4a
 CONDUCTIVITY OF CERTAIN LIQUIDS AT 68° F, k AT 68°
 [Btu/(hr)(sq ft)(°F/in.)]

		a
Acetone.....	1.24	-0.00074
Ammonia.....	4.03
Benzene.....	1.18
Carbon bisulphide.....	1.11	-0.00065
Carbon dioxide.....	1.45
Carbon tetrachloride.....	0.76
Ether.....	0.95	-0.00026
Ethyl alcohol.....	1.26	-0.00039
Glycerine.....	1.98	+0.00029
Kerosene.....	1.05	-0.00081
Methyl alcohol.....	1.44	-0.00029
Oil, lubricating.....	1.16	-0.00023
Oil, transil.....	0.93	-0.00034
Pentane, normal.....	0.94	-0.00057
Sulphur dioxide.....	2.34
Turpentine.....	0.88
Toluene.....	1.05	-0.00080
Vaseline.....	1.27
Water.....	4.10	+0.0010

TABLE 9.4b
 THE THERMAL CONDUCTIVITY OF BRINE, k , IN RELATION TO THAT OF WATER, k_w

	Specific Gravity	$\frac{k}{k_w}$ at 90° F
Sodium chloride	{ 1.09	0.97
	{ 1.19	0.92
Calcium chloride	{ 1.13	0.95
	{ 1.23	0.91

9.5. Conduction through Gases. From the kinetic theory of gases the thermal conductivity of perfect gases is given by the expression

$$k = ec_v\mu \tag{9.11}$$

where μ is the absolute viscosity, c_v is the specific heat at constant

volume, e is a constant depending on the number of atoms in the molecule.

Number of Atoms	e in English Units
1	106,000
2	82,000
3	73,500
More than 3	56,000

Both theory and experiment indicate that the conductivity of gases is independent of the pressure[‡] for moderate pressures and varies according to the Sutherland equation as

$$k_t = k_{492} \frac{492 + c}{T + c} \left(\frac{T}{492} \right)^{1.5} \quad (9 \cdot 12)$$

where c is a constant depending on the gas, T is temperature in degrees F absolute of the gas.

The equation is given approximately as

$$k_t = k_{32}[1 + a(t - 32)] \quad (9 \cdot 13)$$

and it is approximately accurate over small ranges of temperature. Table 9·5 gives values of k for gases and vapors accurate to ± 5 per cent.

Direct use of the conduction equations can very rarely be made, since they are written in terms of surface temperatures and these are not known and usually cannot be determined with adequate accuracy by field methods. Thus, to calculate the conduction transfer into a cold storage chamber through a wall, the inside and outside wall surface temperatures would need to be known. If heat flow is occurring these temperatures will differ from those of the air adjacent to them, and the amount of the difference will be a function of the transfer rate. If thermocouples were available the approximate wall surface temperatures would be obtained, but even under laboratory conditions few thermal measurements are more difficult to make with accuracy than the determination of the temperature of a surface.

9·6. Convection. The restoration of thermal equilibrium in a liquid or gas usually occurs through the mechanism of convection rather than conduction. Actually, convection is a mass transfer leading to mixing of fluid elements having different internal energies and subsequent transfer by conduction from adjacent elements. Because convective transfer is so closely related to fluid movement it must be analyzed in terms of the hydrodynamic as well as the thermodynamic

[‡] Aberdeen and Laby, *Proc. Royal Soc. (London)*, Vol. 113, p. 459, 1926; McAdams, *Heat Transmission*, McGraw-Hill Book Co.

TABLE 9·5a

CONDUCTIVITY OF CERTAIN GASES AND VAPORS AT 32° F BTU/(HR)(SQ FT)(°F/IN.)

	<i>k</i>	<i>a</i>	<i>C</i>
Air	0.163	0.00165	225
Ammonia	0.149	0.0028	
Argon	0.110		
Benzene	0.057		
Carbon dioxide	0.097	0.0021	
Carbon monoxide	0.155	0.0017	281
Chlorine	0.053		
Ethane	0.126	0.0036	
Helium	0.970		59.4
<i>n</i> -Hexane	0.085	0.0032	
Hydrogen	1.130	0.0015	169
Methane	0.210	0.0027	
Neon	0.031		45
Nitrogen	0.163	0.0016	206
Nitrous oxide	0.100		
Oxygen	0.165	0.0016	215
Steam	0.117	0.00217	
Sulphur dioxide	0.056		

TABLE 9·5b

VISCOSITY AND DENSITY

	Density in lb/ft ³		Viscosity, Centipoises	
	5° F Vapor	86° F Liquid	5° F Vapor	68° F Liquid
Ammonia	0.122	37.2	0.0088	0.236
Sulphur dioxide	0.155	84.4	0.0117	0.275
Methylene chloride	0.200	83.3		0.441
Methyl chloride	0.221	55.8	0.0095	0.274
Freon-11	0.0815	91.4	0.00963	0.442
Freon-12	0.6735	80.6	0.01136	0.263
Freon-21	0.1095	84.5	0.0102	0.353
Freon-113	0.0398	96.9	0.00933	0.705
Carbon dioxide	3.74	37.5	0.0163	0.070

TABLE 9-5c
THERMAL CONDUCTIVITY
[Btu/(hr)(sq ft)(°F/ft)]

Vapors	Vapor 32° F	Liquid 68° F	Vapor 212° F
Ammonia	0.0117	0.29	0.0171
Carbon dioxide	0.0081	0.195	0.0111
Methyl chloride	0.00486	0.069 - .0928(1-.00059+)	0.0086
Methylene chloride	0.0036		0.0057
Sulphur dioxide	0.0045	0.195	
Liquids			
Carbon dioxide, 68° F	0.121		
Sulphur dioxide, 68° F	0.195		
Dichloroethylene, 68° F	0.089 - .0928(1-.00059+)		
Methyl chloride, 68° F	0.093 - .1125(1-.00264+)		
Ammonia (14° to 68° F)	0.29		

TABLE 9-5d
VISCOSITY IN CENTIPOISES

Tem- per- ature °F	F-11		F-12		F-21		F-113	
	Liquid at Saturation pressure				Vapor at 14.7 lb/in. ² abs			
	Liquid	Vapor	Liquid	Vapor	Liquid	Vapor	Liquid	Vapor
-40	0.980	0.0088	0.423	0.0106	0.629	0.0090		
-20	0.801	0.0092	0.371	0.0110	0.547	0.0098	1.566	0.0090
0	0.677	0.0095	0.335	0.0113	0.484	0.0101	1.263	0.0093
20	0.587	0.0099	0.300	0.0116	0.436	0.0105	1.043	0.0095
40	0.517	0.0103	0.286	0.0120	0.397	0.0108	0.876	0.0098
60	0.401	0.0106	0.269	0.0123	0.364	0.0111	0.747	0.0101
80	0.417	0.0110	0.255	0.0126	0.337	0.0115	0.646	0.0103
100	0.380	0.0113	0.242	0.0129	0.314	0.0118	0.564	0.0106
120	0.349	0.0116	0.282	0.0130	0.294	0.0121	0.497	0.0109

properties of the fluid. Thus a general equation for heat transfer by convection would necessarily include the area, the temperature difference, and the following physical properties of the fluid: velocity, viscosity, density, specific heat, and thermal conductivity. The interest of the engineer in convection transfer is almost always localized at a solid boundary of the fluid, and the character of this bounding surface with respect to curvature and roughness may likewise be expected to influence the rate of heat transfer by convection to or from the surface. As a convenience in computation it is customary to separate the properties of the system from those of the fluid and write the convection equation in the form

$$q_v = hA(t_h - t_c) \quad \text{Btu/hr} \quad (9.14)$$

where q_v is the transfer rate in Btu per hour, A is the area, t_h and t_c are the hot and cold temperatures, and h is a coefficient of convective heat transfer which must be evaluated in terms of fluid and solid boundary characteristics. For a large body of fluid adjacent to a solid boundary one of the temperatures will be taken on the fluid side of the boundary and the other at a point within the fluid sufficiently far away from the boundary so that this temperature can be considered equal to that of the main body of fluid.

9.7. Approximate Film Coefficients for Air. For some special cases, values of h can be expressed directly or can be computed from simple empirical equations. Thus, for a horizontal cooled surface facing up in air having a velocity not greater than 15 fpm (comparable to conditions in a room with a cooled floor), the value of h can be taken as 0.4 Btu/(hr)(sq ft)(°F). For a similar cooled surface located vertically (as in a wall) the tendency toward mass movement of the air down the wall due to density change would increase the convection coefficient to 0.7, whereas for a cooled surface located in a ceiling the air coming in contact with the surface would drop in temperature, increase in density, and fall away from the surface sufficiently rapidly to cause the coefficient to increase to approximately 1.1 Btu/(hr)(sq ft)(°F). These values are only approximate, but they can be used with adequate engineering accuracy for transfer problems in cold storage rooms which do not have forced air circulation.

For the more general case of air flow past a surface (as wind across an exterior wall) the convective film coefficient of heat transfer can be expressed by empirical equations which take account of both roughness and wind velocity:

$$h = 0.6 + 0.28V \quad (9.15)$$

for smooth surfaces such as glass or polished stone;

$$h = 0.8 + 0.30V \quad (9 \cdot 16)$$

for smooth surfaces such as painted wood or plaster;

$$h = 1.2 + 0.4V \quad (9 \cdot 17)$$

for moderately rough surfaces such as unfinished concrete and smooth brick;

$$h = 1.3 + 0.5V \quad (9 \cdot 18)$$

for rough concrete, rough brick, or stucco. The term V in these equations is the wind velocity expressed in miles per hour, and the equations are approximately applicable irrespective of whether the wind is normal to or parallel with the surface in question. These equations are by no means exact, but for many practical refrigeration problems the air film represents only a very small fraction of the total insulating effect, so that a substantial error in evaluation of the film coefficients is unlikely to have any appreciable influence on the overall transfer rate through a wall.

To permit comparison of various types of thermal structures the American Society of Heating and Ventilating Engineers has standardized the values of h as used in the development of the overall coefficients that appear in the *ASHVE Guide*. For inside surfaces the film coefficient h_i has been arbitrarily taken as 0.85 by convection with an equivalent radiation coefficient of 0.80 added, giving 1.65 Btu/(hr)(sq ft)(°F) as the combined convection-radiation film coefficient for any type of inside surface that is not subjected to a direct blast of air. The standard outside film coefficient allows 5.2 for convective transfer with a "standard" outside wind velocity of 15 mph and 0.8 for radiation, giving a combined coefficient of 6.0 Btu/(hr)(sq ft)(°F).

Convection across air spaces in walls can be roughly evaluated by treating the two solid boundaries separately, but more exact transfer rates can be obtained by using overall convection coefficients which have been experimentally determined for air spaces as a function of both space width and mean temperature. (Refer to Table 10·7.)

9·8. General Equations for Free Convection. In free convection, equations for heat transfer can usually be reduced to the form

$$B = \phi(M)f(N) \quad (9 \cdot 19)$$

where $B = hL/k$, $M = \rho^2 g \beta L^3 t / \mu^2$, and $N = \mu c_p / k$; β = the thermal expansion of the fluid = $(1/\rho)(\delta\rho/\delta t)$ per degree Fahrenheit; h is the surface heat transfer coefficient in Btu per hour per square foot per

degree Fahrenheit; t is the temperature rise of the surface over that of the main body; L is a linear dimension like diameter, in feet; μ is viscosity of fluid = lb/ft-hr; $g = \text{ft}/(\text{hr})^2$.

However, for gases the value of $\mu c_p/k$ varies over a very small range and may be considered a constant, and the equation becomes

$$B = \phi(MN) \quad \text{or} \quad \frac{hL}{k} = \phi \frac{(\rho^2 g \beta c_p L^3 t)}{\mu k} \quad (9.20)$$

where $k = e\mu c_p$. (Values of viscosity and specific heat are given in Tables 9.6 through 9.9.) The function of MN can be found by plotting the values of the two dimensionless quantities $(\rho^2 g \beta c_p / \mu k) \times L^3 t$ against hL/k , and according to King§ the results of several hundred points computed from the data reported by a score of observers give a well-established line, for viscous and turbulent flow, and horizontal as well as vertical surfaces. The slope of this line on log paper is very nearly $1/3$, and if $\rho^2 g \beta c_p / \mu k = \alpha$ then $hL/k = c_1(\alpha L^3 t)^{1/3}$, which reduces to

$$h = 0.13 k \alpha^{1/3} t^{1/3} \quad (9.21)$$

for large surfaces of any shape. However, for a limited range of the variables the slope is nearly constant, and for sizes from $1/4$ to 2 in. and for moderate temperatures the average slope is $1/4$ and the formula becomes

$$h = 0.55 k \alpha^{1/4} t^{1/4} L^{-1/4} = 0.55 k \left[\frac{\alpha t}{L} \right]^{1/4} \quad (9.22)$$

For gases, except for very high or very low pressures, α varies as p^2 . For very small wires and small temperature rises the heat transfer coefficient approaches as a limit

$$h = 0.65 \frac{k}{d} \quad (9.23)$$

where d = the diameter in feet.

9.9. Forced Convection. Within the refrigeration cycle itself interest in convective heat transfer is centered on the refrigerant. In the evaporator and condenser, heat flows to and from the refrigerant while it undergoes a phase change, whereas, in the line from compressor to condenser and in the initial section of the condenser, energy loss is by forced convection from a superheated gas; in the pipe connecting the evaporator to the compressor superheating may occur by forced

§ King, *Mech. Eng.*, May, 1932; Fishenden and Saunders, *The Calculation of Heat Transfer*.

TABLE 9-6

VISCOSITY OF WATER

Temperature in °F Viscosity z in Centipoises¹

t	z	t	z	t	z
32	1.79	80	0.861	120	0.560
40	1.55	86	0.800	130	0.511
50	1.31	90	0.764	140	0.470
60	1.12	100	0.684	150	0.432
70	0.978	110	0.617	160	0.400

t	z	t	z	t	z
170	0.372	220	0.270	270	0.208
180	0.347	230	0.255	280	0.199
190	0.325	240	0.242	290	0.191
200	0.305	250	0.229	300	0.185
210	0.287	260	0.218		

$$^1 z \times 0.000672 = \mu \text{ in } \frac{\text{lb}}{\text{ft-sec}}$$

TABLE 9-7

VISCOSITY OF CERTAIN GASES AND VAPORS

Viscosity in $\frac{\text{lb}}{\text{ft-sec}} \times 10^{-7}$

Temperature, °F	Air	NH ₃	CO ₂	H ₂ O vapor
0	113 × 10 ⁻⁷	59 × 10 ⁻⁷	87 × 10 ⁻⁷	55 × 10 ⁻⁷
20	117	62	90	58
40	120	64.5	94	62
60	123.5	67.5	97	65
80	127	70	101	68
100	130	72.5	105	71.5
120	133.5	75	108.5	75
140	137	78	112	78
160	140	80.5	116	81
180	143.5	83	119.5	84.5
200	147	85.5	123	88

TABLE 9-8
 SPECIFIC HEAT (*c*) FOR LIQUIDS
 (Btu per pound) I.C.T.

Substance	Temperature, °F	<i>c</i>
Ammonia, NH ₃	0	1.080
	10	1.085
	20	1.091
	30	1.097
	40	1.104
	50	1.112
	60	1.120
	70	1.129
	80	1.138
	90	1.147
Sulphur dioxide, SO ₂	100	1.156
	20	0.327
	40	0.3416
	60	0.361
Ethyl alcohol, C ₂ H ₅ O.....	80	0.387
	0	0.435
Glycerol, C ₃ H ₈ O.....	50	0.581
	0	0.540
Benzene, C ₆ H ₆	50	0.60
	5	0.3845
	10	0.394
	20	0.406
Toluene, C ₇ H ₈	40	0.423
	0	0.3855
	50	0.4210
	80	0.4465

TABLE 9·9
 c_p FOR GASES AND VAPORS
 Btu per °F at atmospheric pressure
 (Smithsonian Institution)

Gas or Vapor	Temperature, °C	c_p	$\frac{c_p}{c_v}$
Acetone.....	26-110	0.347
Air.....	0-100	0.237
Alcohol.....	108-220	0.453	1.133
Ammonia.....	23-100	0.52	1.317
Argon.....	20-90	0.123	1.667
Benzol.....	34-115	0.299	1.403
Bromine.....	80-230	0.0555	1.293
Carbon dioxide.....	15-100	0.2055
Carbon disulphide.....	86-190	0.1596	1.205
Chlorine.....	13-202	0.1241	1.323
Ether.....	25-111	0.428	1.024
Hydrochloric acid.....	13-100	0.1940	1.389
Hydrogen.....	12-198	3.409
Hydrogen disulphide.....	20-200	0.2451	1.276
Methane.....	18-208	0.5929	1.316
Nitric oxide.....	13-172	0.2317
Nitrogen.....	0-200	0.2438	1.41
Nitrous oxide.....	16-207	0.2262	1.311
Oxygen.....	10-200	0.2175	1.3977
Sulphur dioxide.....	16-200	0.1544	1.256
Water vapor.....	0-86	0.453	1.27

convection, and in the line between condenser and expansion valve some subcooling may likewise occur by forced convection. Thus it is necessary to find means of evaluating the film coefficient of convection heat transfer for a heating or a cooling superheated vapor, for a heating or cooling subcooled liquid, and for an evaporating liquid and a condensing vapor.

The equations for convection coefficients are of different form for streamline than for turbulent flow. Without exception, however, the problems important in refrigeration engineering involve turbulent flow; hence only these equations will be discussed. Furthermore most of these problems are for refrigerant in a pipe or for cooling fluid outside a pipe. The general equation for turbulent transfer to or from a fluid (liquid, vapor, or gas) flowing turbulently within a pipe is attributed to Nusselt and is written in terms of three dimensionless groups of properties of the system

$$\frac{hD}{k} = F \left(\frac{DG}{\mu} \right)^n \left(\frac{c_p \Delta t}{k} \right)^m \mu \quad (9 \cdot 24)$$

The group on the left side of the equation is called the Nusselt number, the first group on the right is Reynolds's number, and the second group on the right is the Prandtl number. The coefficient F and the exponents n and m must be determined experimentally. The mass velocity G is expressed in pounds per hour per square foot of cross section; hence it is equal to the velocity, in feet per hour, multiplied by the weight density w in pounds per cubic foot.

For a fluid heating, McAdams^{||} suggests that F , n , and m be given the respective values of 0.0225, 0.8, and 0.4. Substituting these values in equation 9·24, grouping terms and solving for h gives

$$h = 0.0225 (V^{0.8}) (D^{-0.2}) (w^{0.8} k^{0.6} c_p^{0.4} \mu^{-0.4}) \quad (9 \cdot 25)$$

where the properties grouped in the last parentheses all vary with temperature and all have fixed values, for a given fluid, at any particular temperature. Hence equation 9·25 is of the form

$$h = \phi(V, D, t) \quad (9 \cdot 26)$$

indicating that the film coefficient is fixed in terms of the pipe size, the fluid velocity, and the fluid temperature.

For a fluid cooling inside a pipe McAdams recommends that the same F and n values be used as for heating, but that m be taken as 0.3. The

^{||} McAdams, *Heat Transmission*, McGraw-Hill Book Co., 1933.

resultant equation, comparable in form to equation 9·25, is

$$h = 0.0225(V^{0.8})(D^{-0.2})(w^{0.8}k^{0.7}c_p^{0.3}\mu^{-0.5}) \quad (9\cdot27)$$

which can be represented by

$$h = \phi'(V, D, l) \quad (9\cdot28)$$

Equations 9·26 and 9·28 differ only with respect to the influence of the temperature on the film coefficient; hence a general solution of one can be used with a temperature correction as a solution of the other.

For a liquid outside of a pipe and with a Reynolds number greater than 100 and flowing normally across it, Stoever[†] recommends that F , n , and m be assigned values 0.385, 0.56, and 0.30 irrespective of whether the fluid is heating or cooling. Thus equation 9·24 reduces to

$$h = 0.385(V^{0.56})(D^{-0.44})(w^{0.56}k^{0.7}c_p^{0.3}\mu^{-0.28}) \quad (9\cdot29)$$

and is of the form

$$h = \phi''(V, D, l) \quad (9\cdot30)$$

9·10. Graphical Solutions for Film Coefficients of Subcooled Refrigerants. To assist the refrigeration engineer in evaluating film coefficients for subcooled liquid refrigerants a series of graphical solutions has been prepared providing solutions of equations 9·25, 9·27, and 9·29. Two graphs are available for each refrigerant, one for flow of the refrigerant inside the pipe (giving solutions of both equations 9·25 and 9·27) and the other for flow outside and normal to the pipe (giving solution of equation 9·29). Figures 9·1 through 9·20 inclusive* will give directly the film coefficients for most of the commonly used refrigerants. Use of the graphs is indicated, in each case, by a dashed example line.

Example. Water at 50° F is flowing turbulently through a pipe having an inside diameter of 1 in. The velocity is 7 fps, and the water within the pipe is being heated. Determine the inside film coefficient of heat transfer.

Solution. Enter Fig. 9·1 at the 7 fps on the velocity scale (at the bottom of the figure), then rise (as shown by the dashed example line) to intersection with the horizontal diameter line for 1-in. inside diameter, and from this point follow the directrix up and to the left to intersection with the base scale. From this intersection rise vertically to the solid horizontal for water heating at 50° F, and from this intersection follow a directrix down and to the right to intersection with the base scale at the correct value of the inside film coefficient of heat transfer: 1040 Btu/(hr)(sq ft)(°F).

[†] Stoever, *Applied Heat Transmission*, McGraw-Hill Book Co., 1941.

* These graphical solutions originally appeared in a series of technical articles by Raber and Hutchinson, published in *Heating and Ventilating*; they are used here by permission.

In similar manner the film coefficient for liquid refrigerant heating or cooling within or without a pipe can be obtained from the proper graph. For the most part the construction of all graphs is essentially the same, though minor differences, due to changing importance of the properties of some of the refrigerants as a function of temperature, will be observed. Thus in some cases the value of the film coefficient will be found to increase with temperature, in other cases to decrease, and in the cases of Freon-11 and Freon-14, when outside of a pipe, the coefficient is not appreciably affected by temperature over the range from 0° to 100° F.

9-11. Extensions of Graphical Solutions. In many practical refrigeration problems refrigerant flow occurs under conditions more complex than those represented by the assumptions underlying the graphical constructions. Thus, for a double-pipe heat exchanger, flow will be in the annular space between pipes, whereas in some condensers flow occurs between shell and tubes parallel to the tubes. For all such cases a recommended procedure is to treat the system as though it were a single pipe of diameter such that the ratio of pipe cross-sectional area to pipe circumference is the same as the ratio of the actual cross-sectional area of fluid to the length of boundary across which heat transfer is taking place. Thus for annular flow the equivalent diameter would be $(d_o^2 - d_i^2)/d_i$, where d_o and d_i are, respectively, the inside diameter of the outside pipe and the outside diameter of the inside pipe. For flow parallel to the tubes of a shell-and-tube unit the equivalent d_e is equal to $(d_s^2 - Nd_t^2)/Nd_t$, where d_s is the inside diameter of the shell, d_t is the outside diameter of one tube, and N is the number of tubes within the shell.

For flow through non-circular tubes the principle of the above paragraph can also be used, and an equivalent diameter selected equal to four times the cross-sectional area divided by the circumference. For flow across a tube bundle, Stoever recommends a 30 per cent increase in the chart value of the outside film coefficient based on use of the fluid velocity through the narrowest section of the staggered bundle. For flow across tubes which are in line, opinion is varied, but it is suggested here that uncorrected values of h_o , taken from the graphs for a fluid velocity corresponding to the narrowest section of the bundle, be used. For baffled tubular exchangers, conditions vary from point to point, and specific information on the particular exchanger is needed if an accurate analysis is to be made; if adequate information is lacking it is suggested that the uncorrected value of h_o , taken from the graphs, be used for such cases.

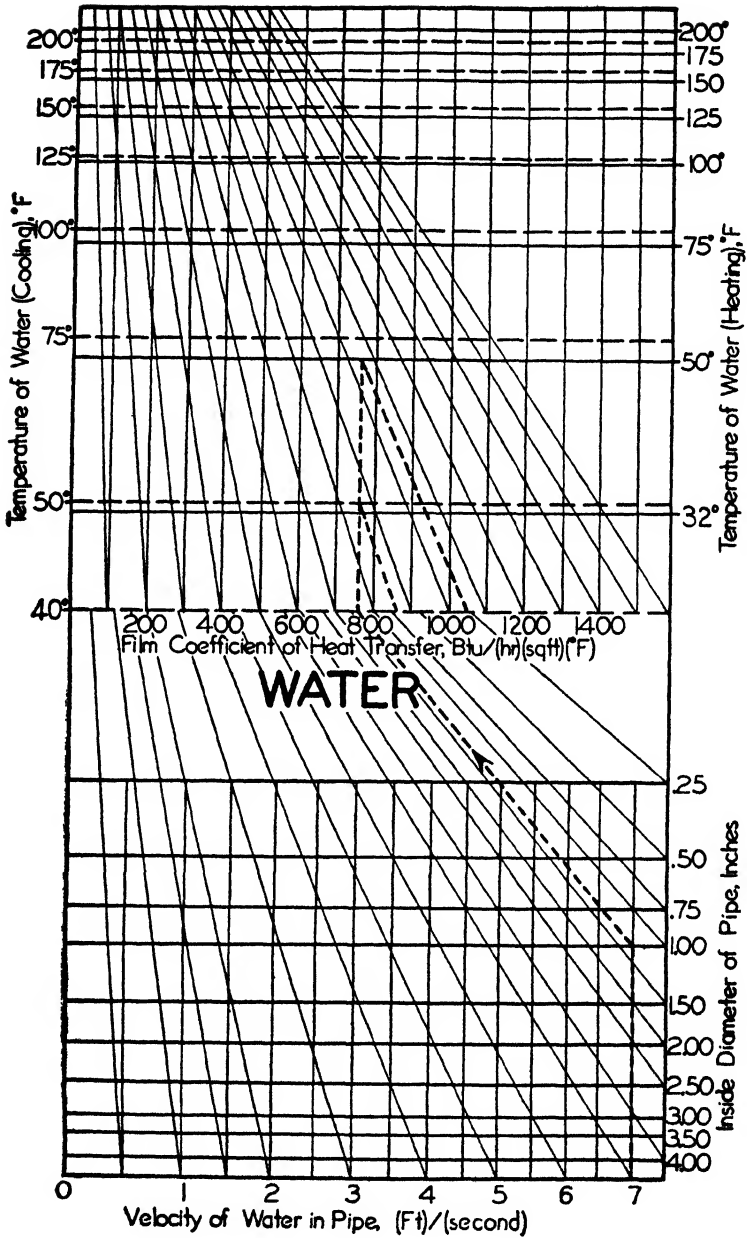


FIG. 9-1.

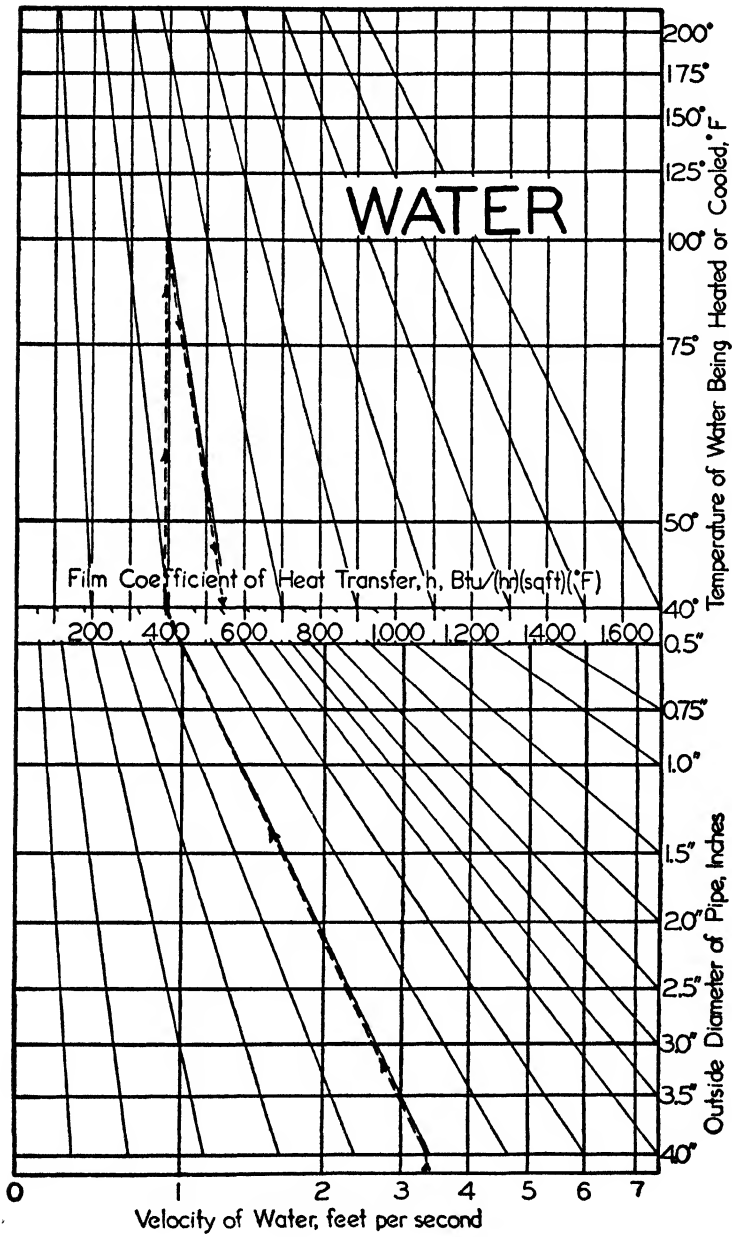


FIG. 9-2.

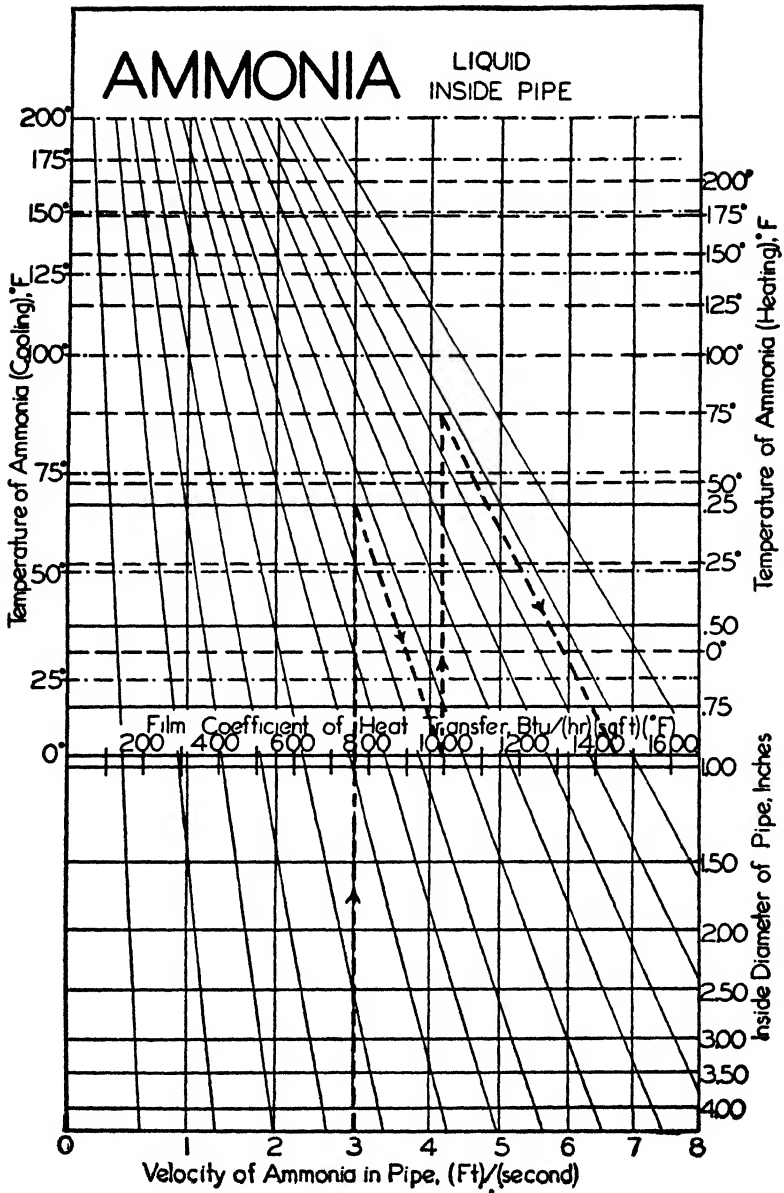


FIG. 9-8.

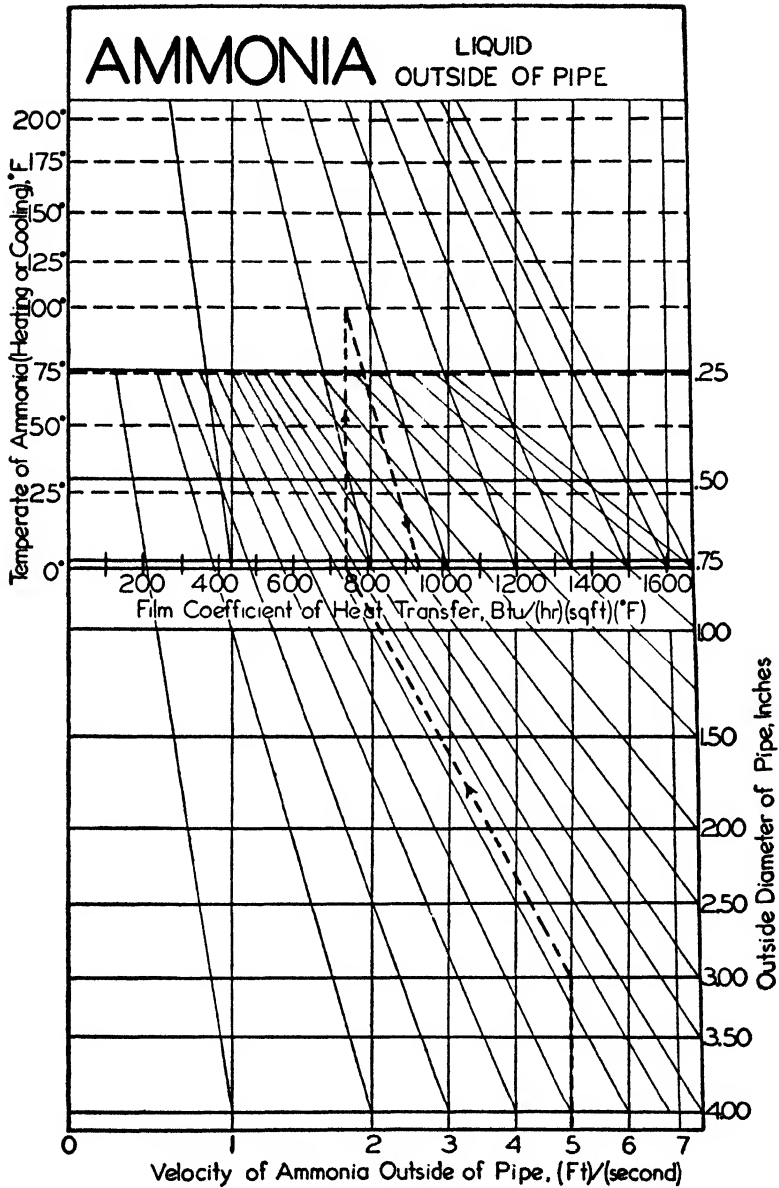


FIG. 9-4.

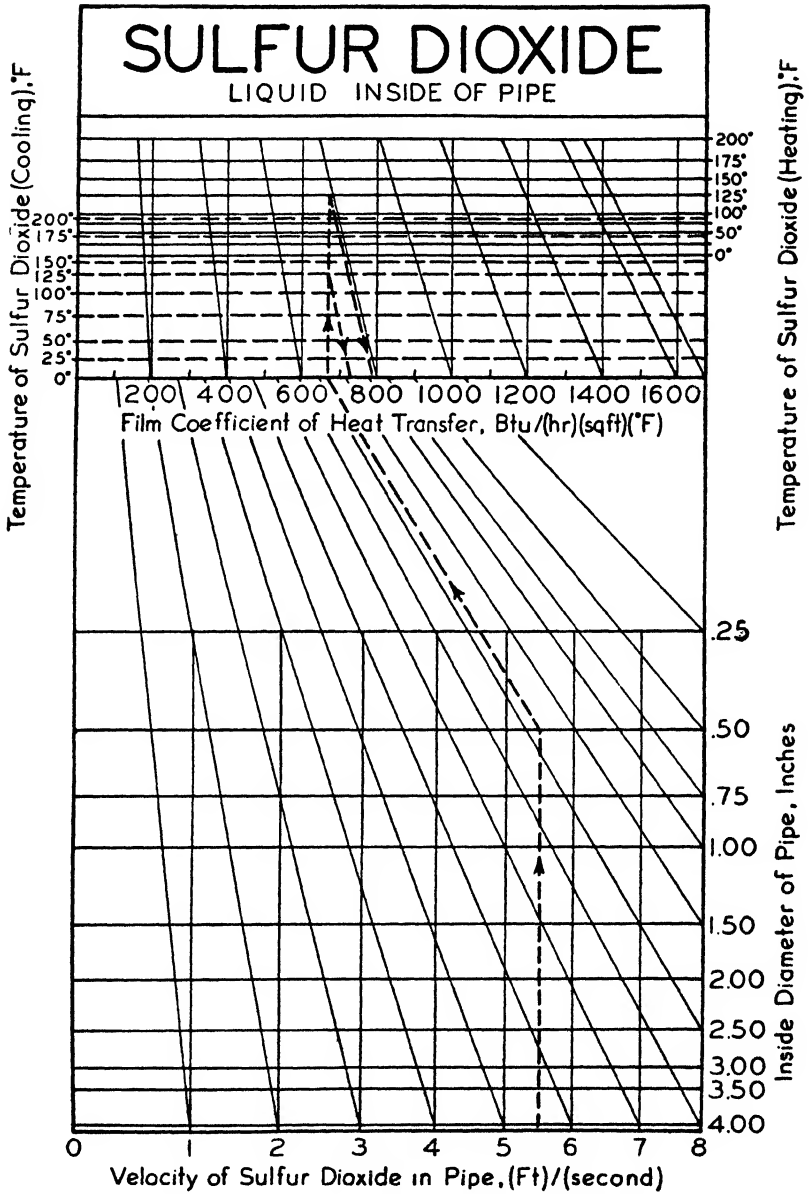


FIG. 9-5.

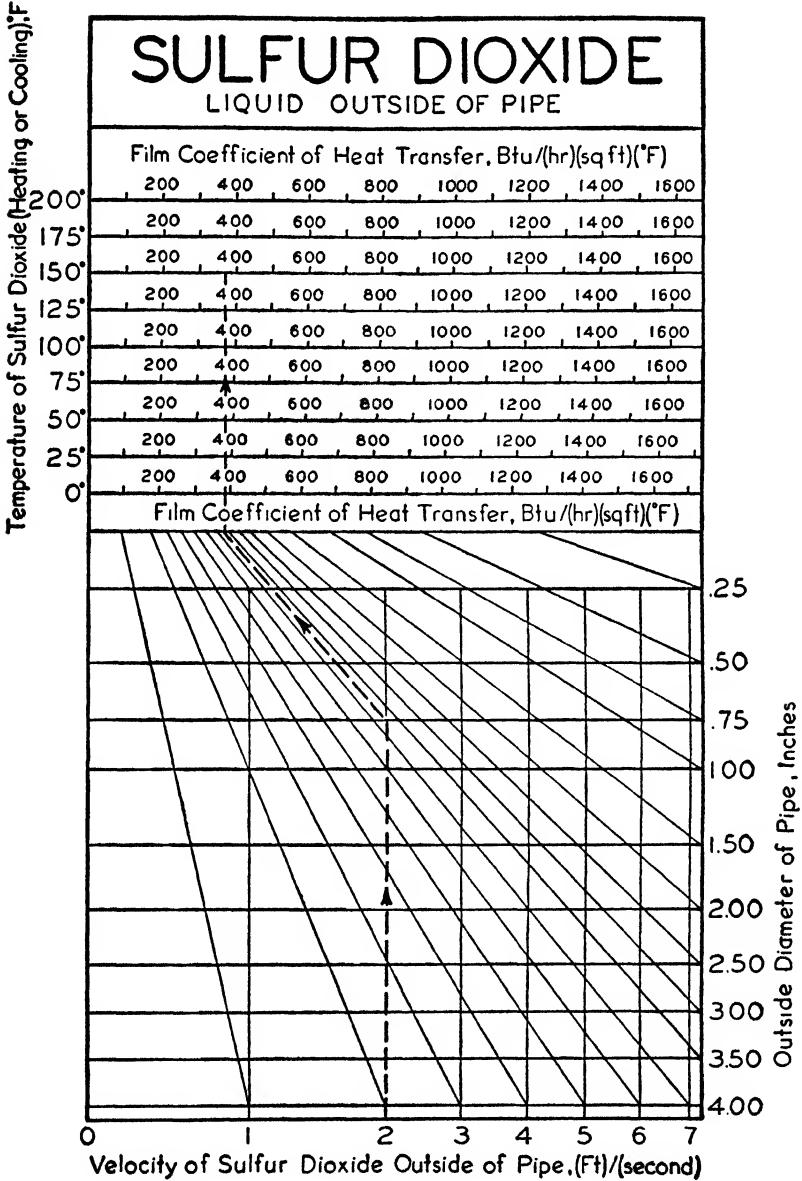


FIG. 9-6.

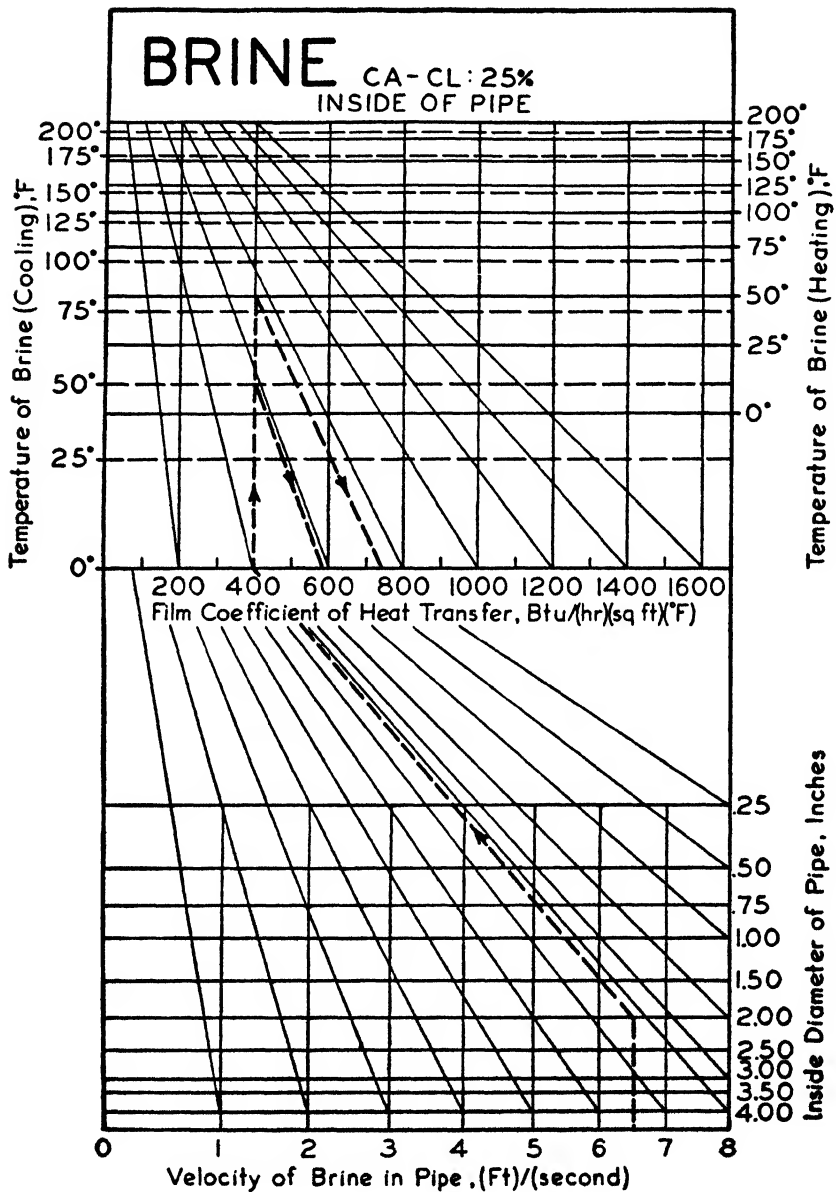


FIG. 9-7.

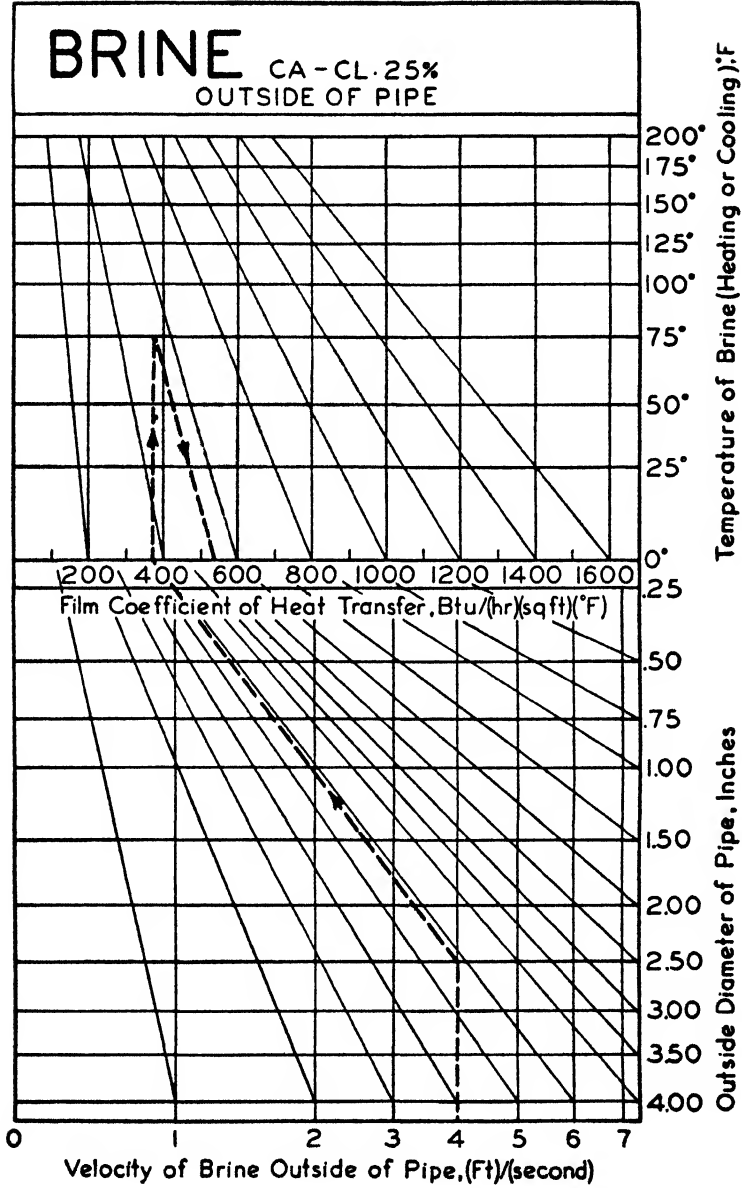


FIG. 9-8.

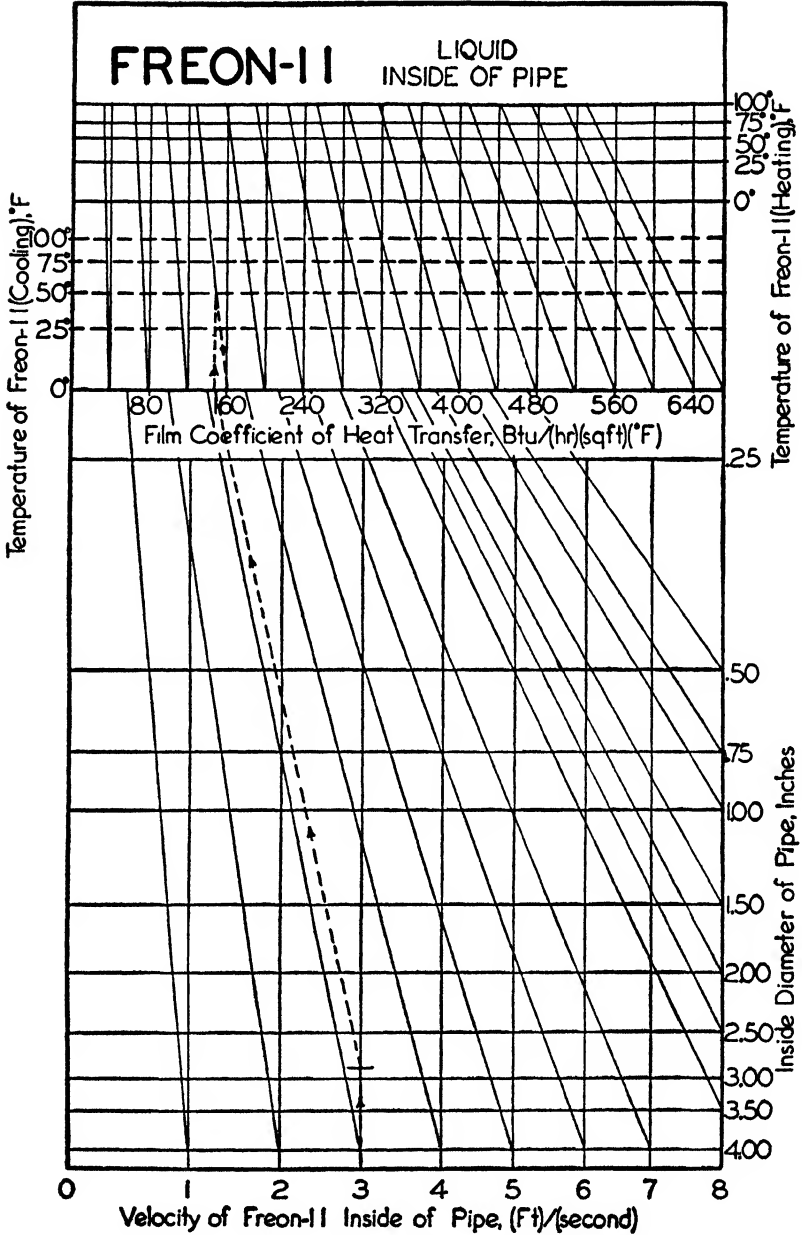


FIG. 9-9.

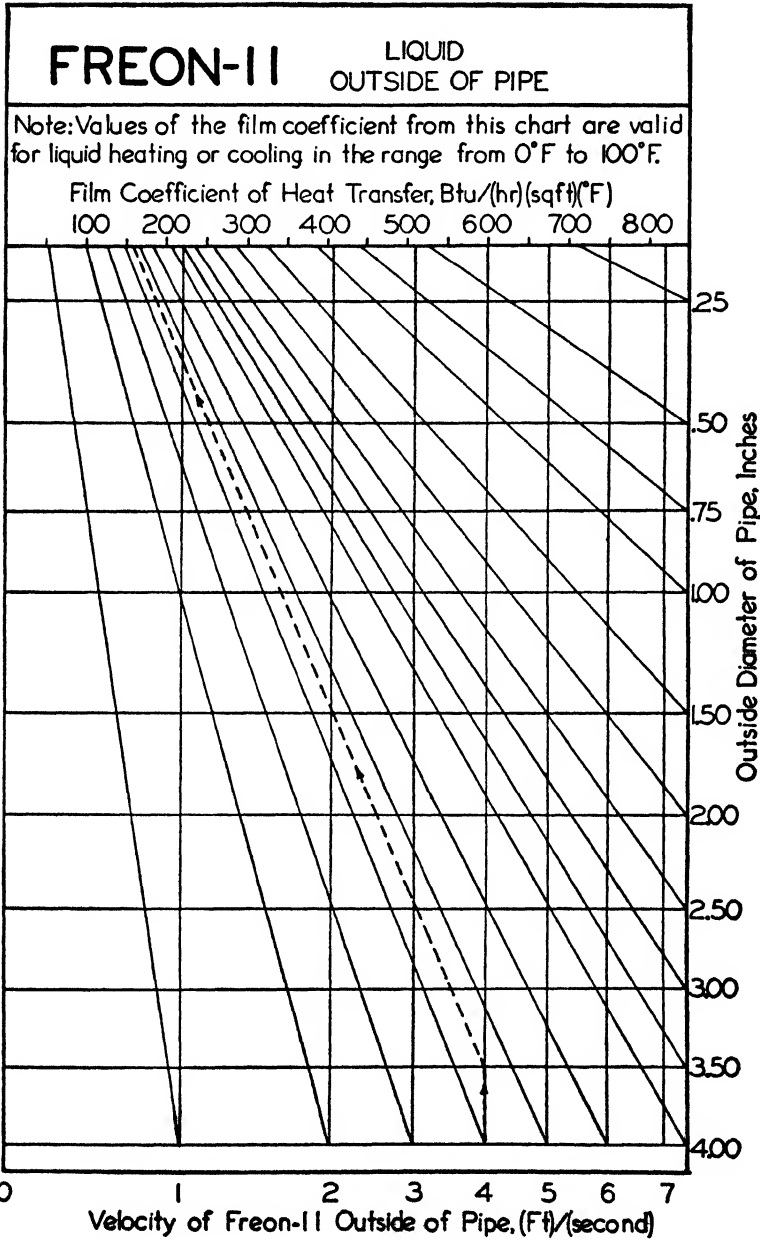


FIG. 9-10.

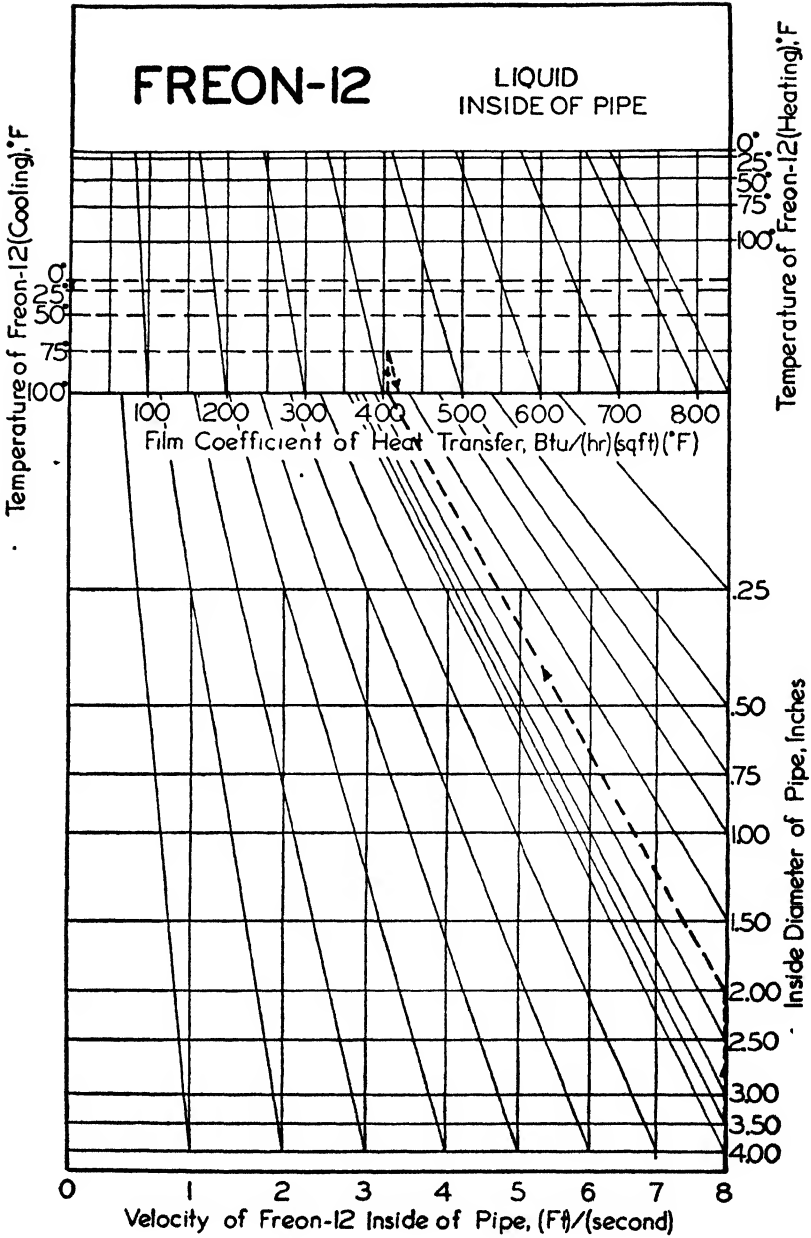


FIG. 9-11.

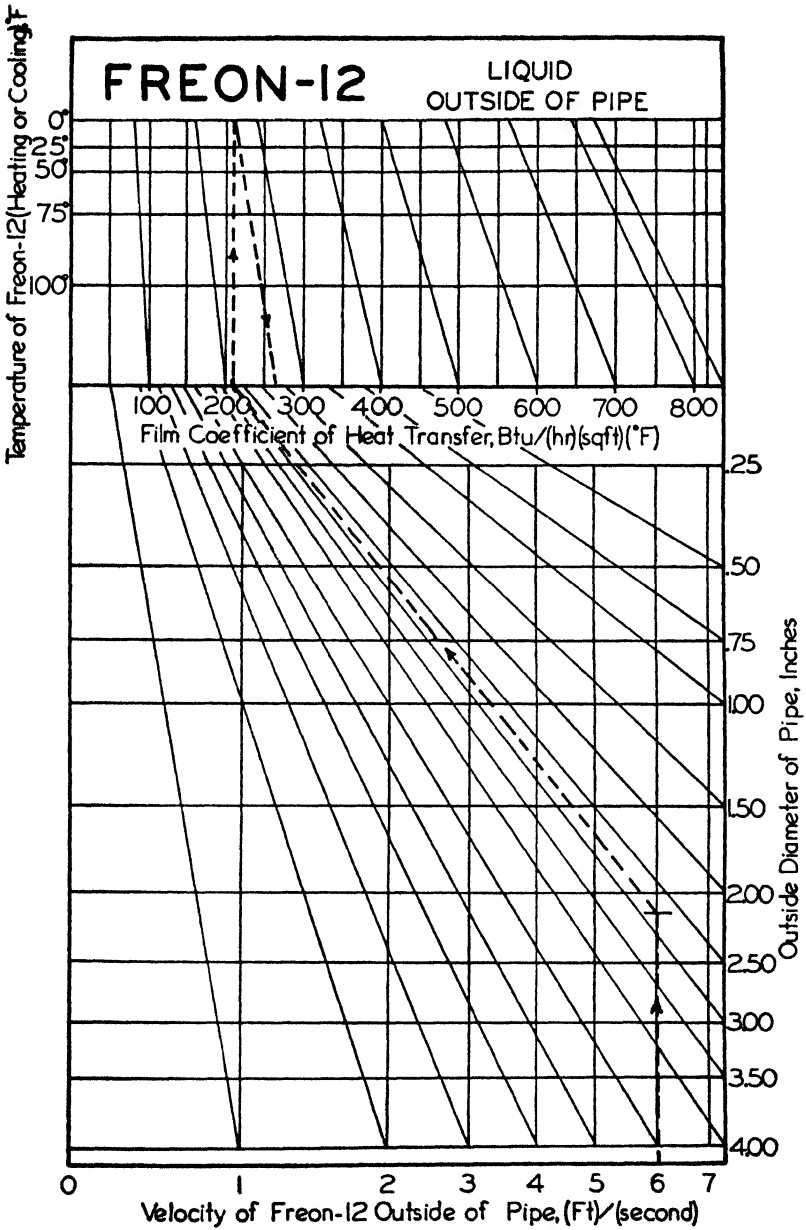


FIG. 9-12.

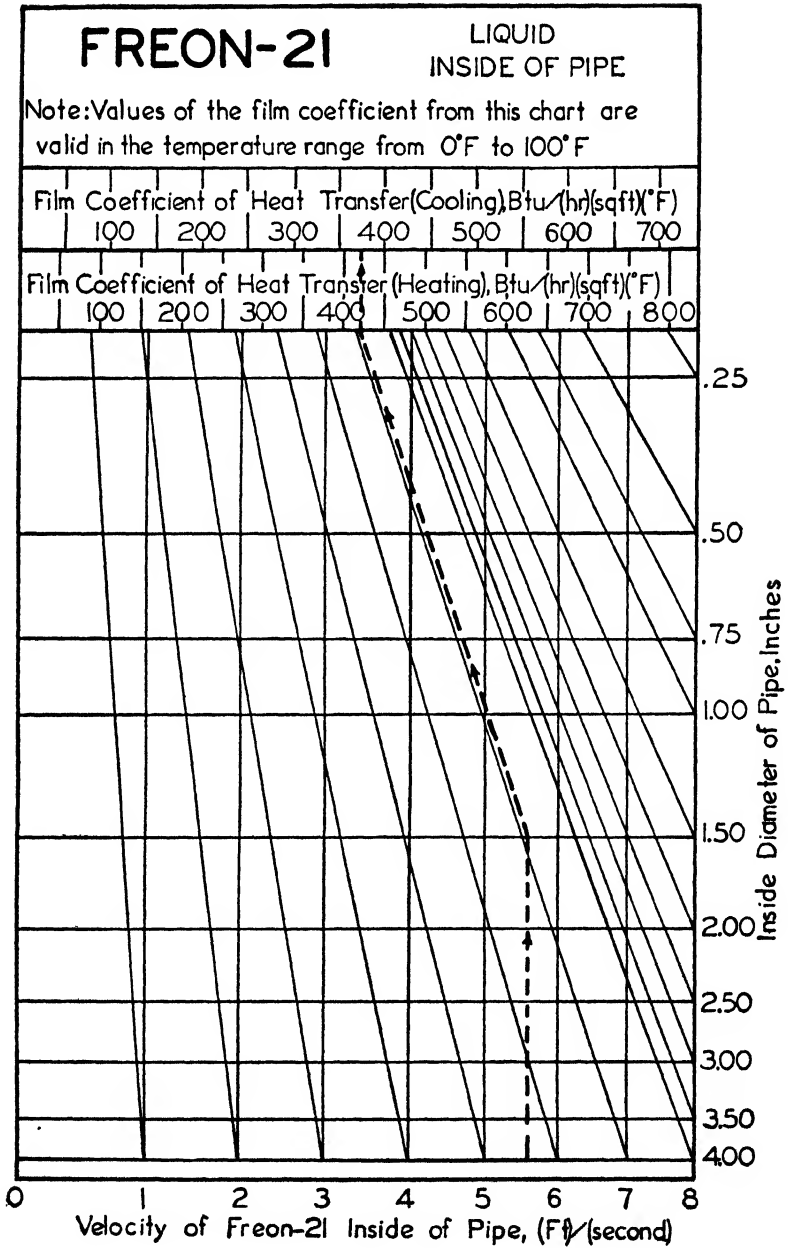


FIG. 9-13.

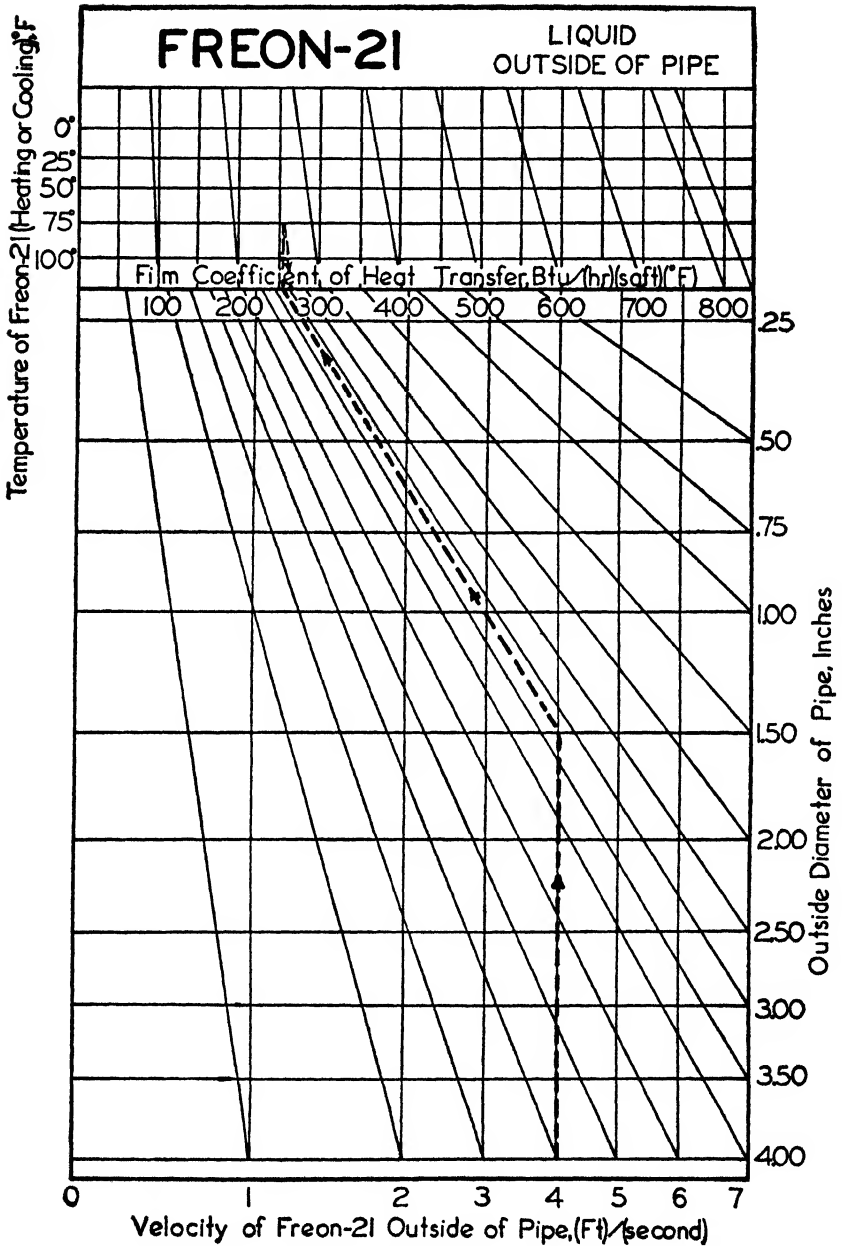


FIG. 9-14.

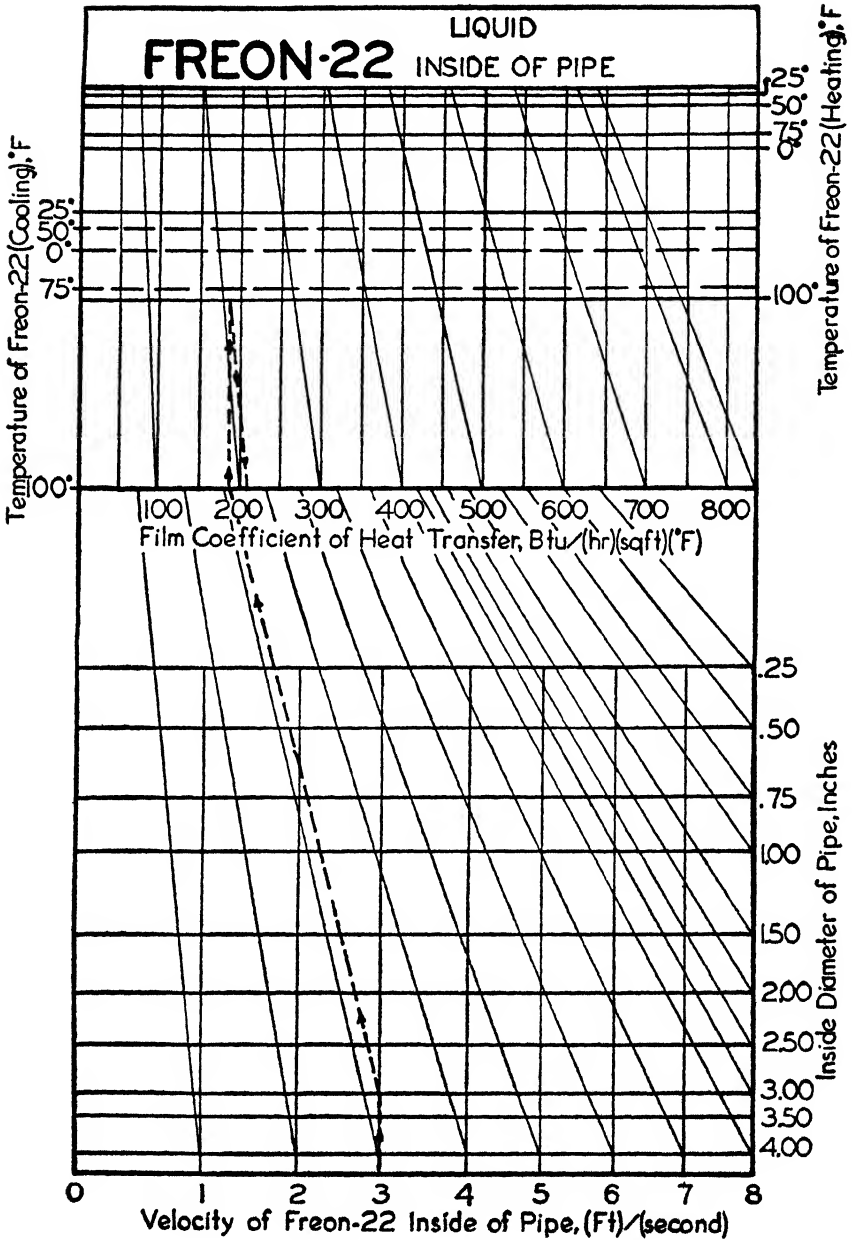


FIG. 9-15.

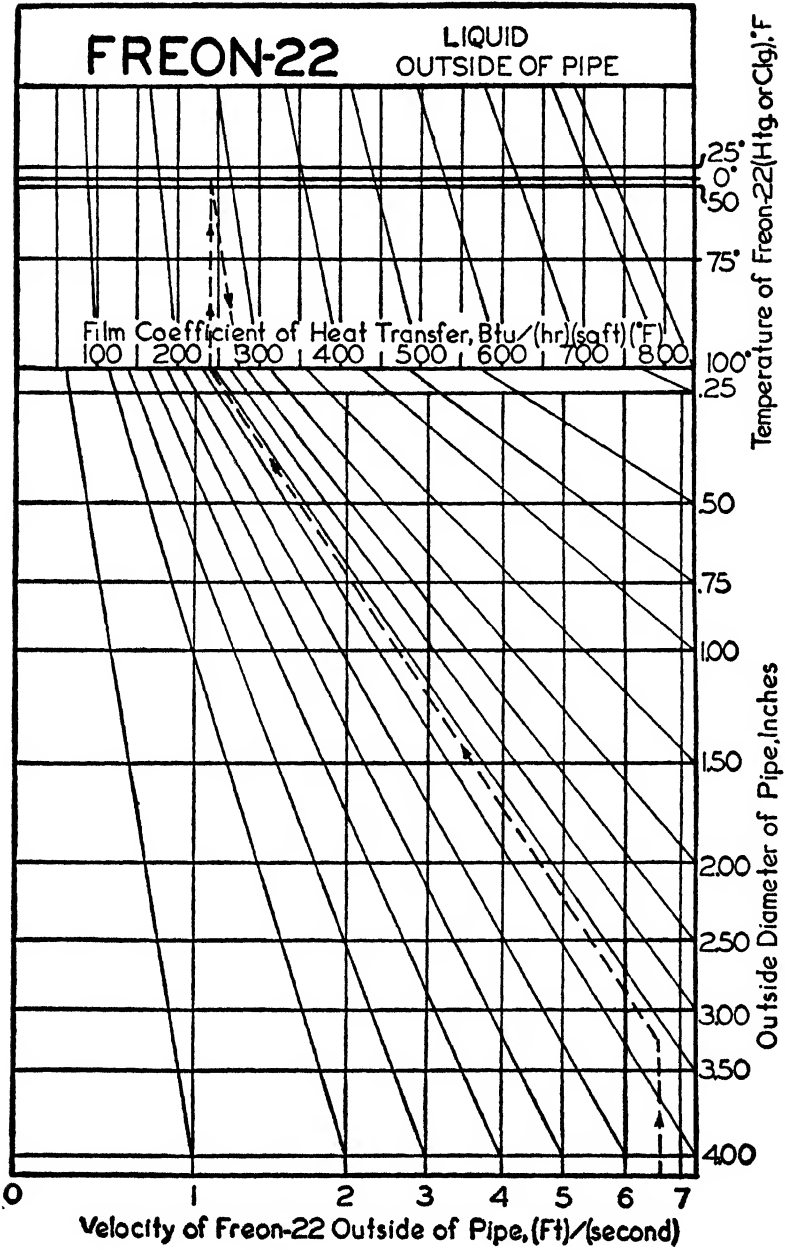


FIG. 9-16.

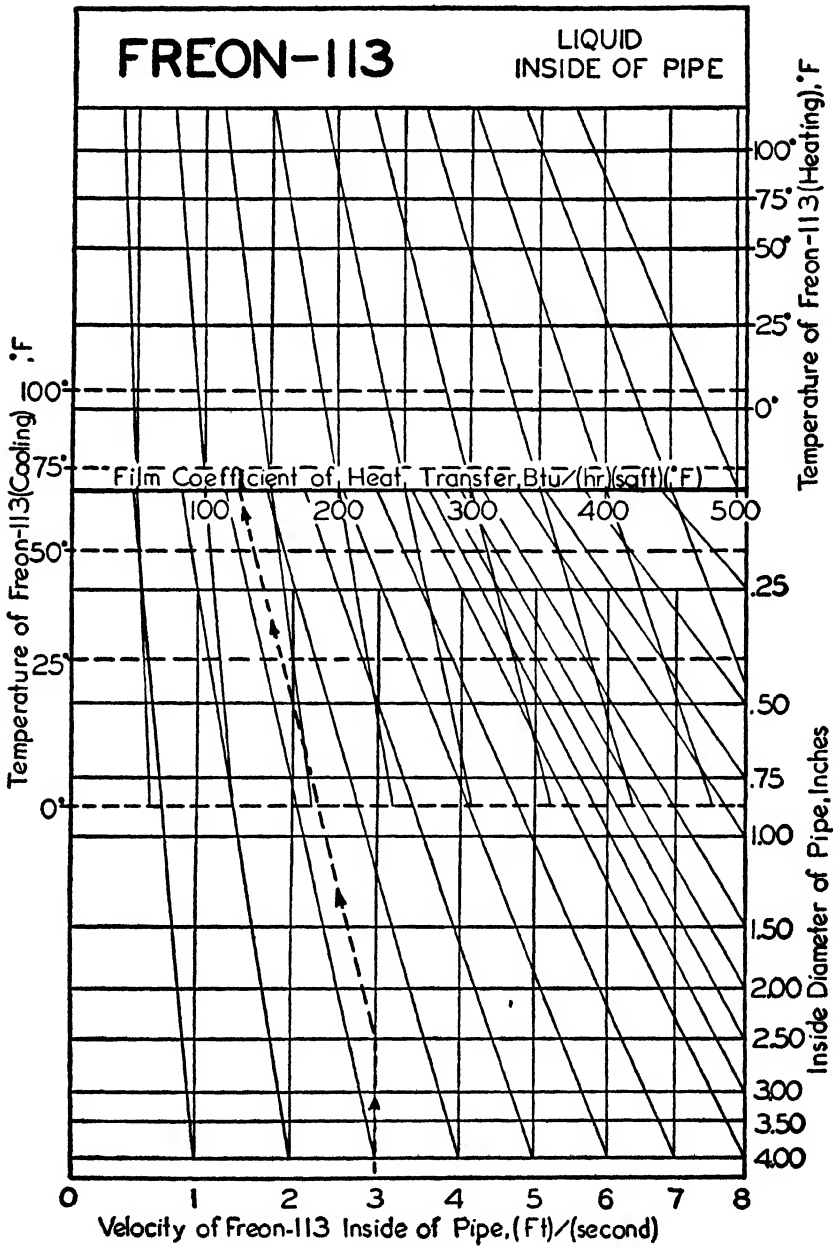


FIG. 9-17.

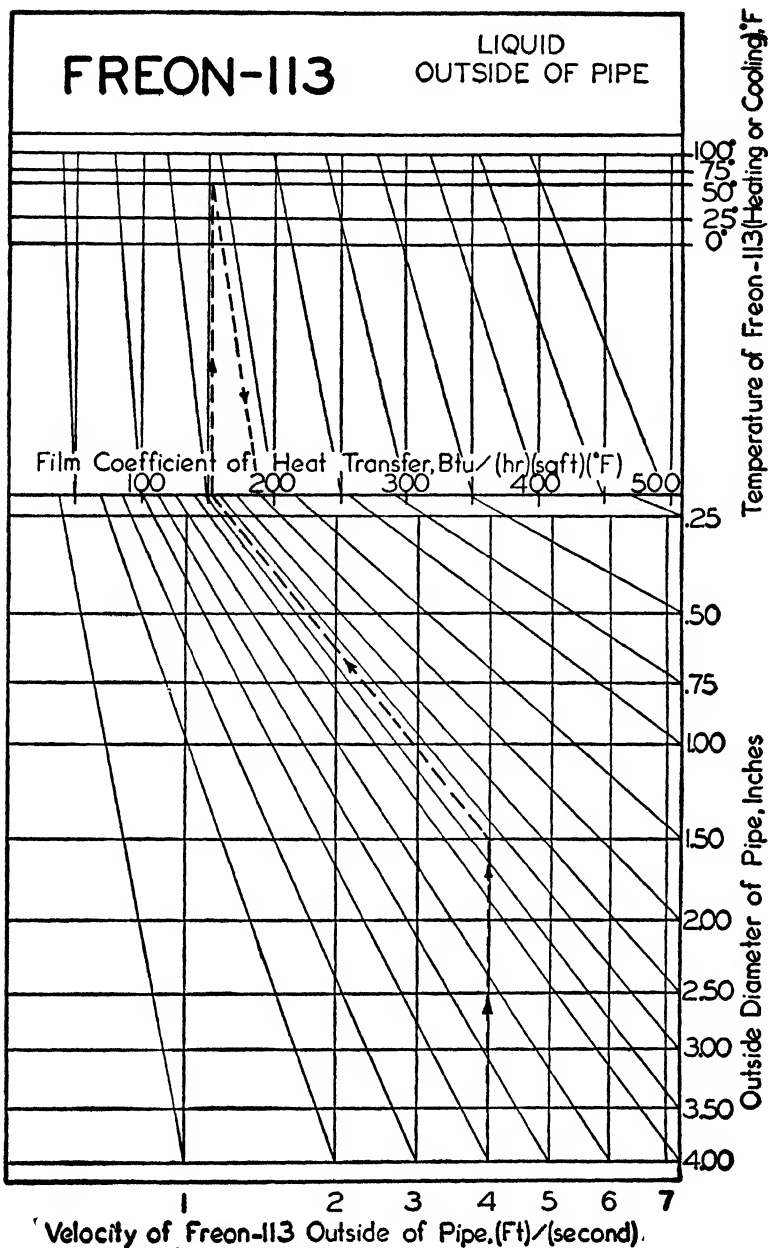


FIG. 9-18.

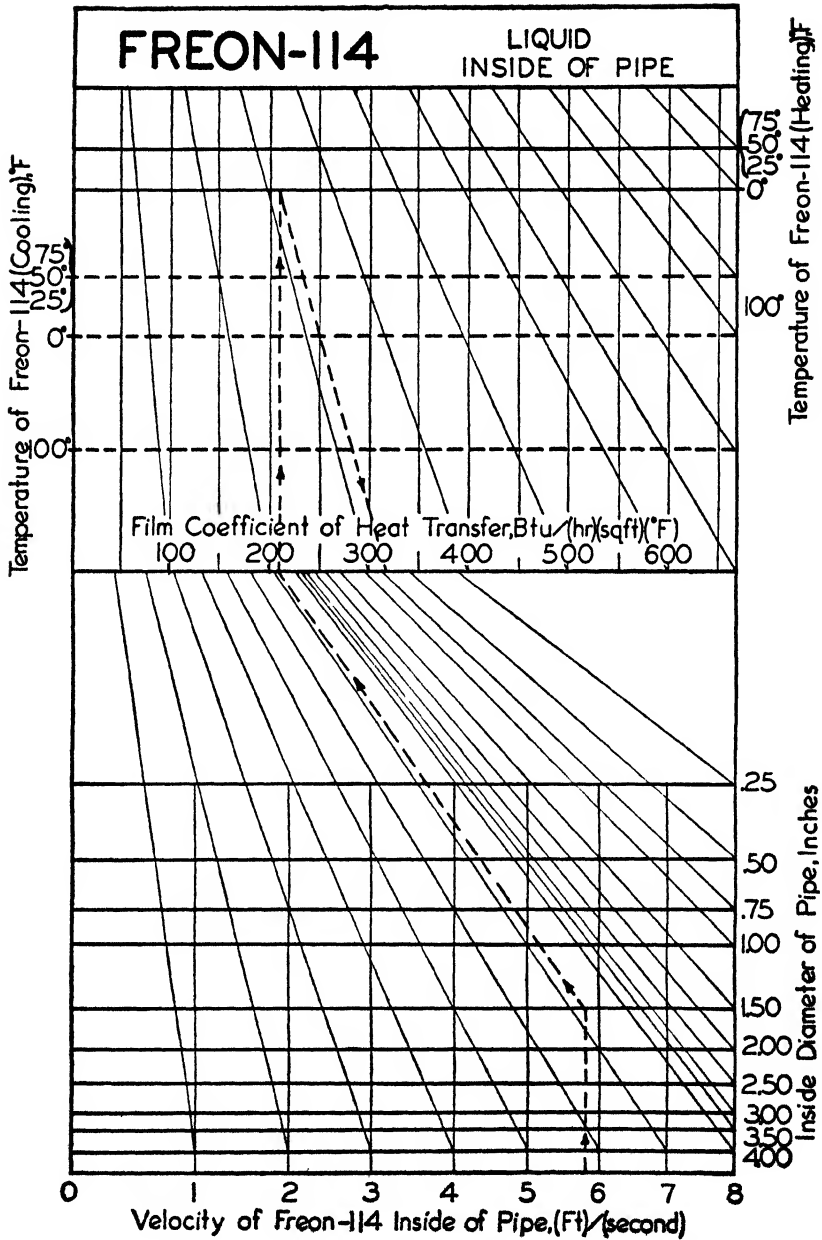


FIG. 9-19.

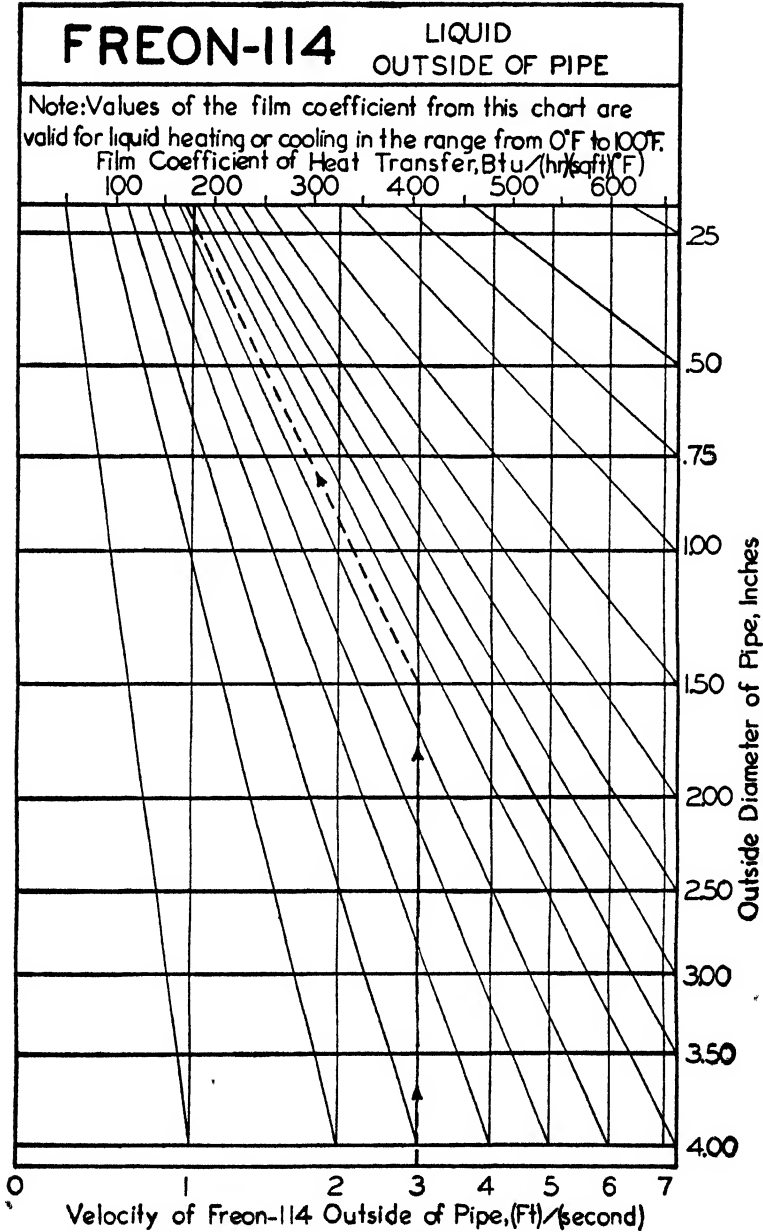


Fig. 9-20.

9·12. Graphical Solutions for Film Coefficients of Superheated Vapors. The graphical solutions presented in this section† (Figs. 9·21 through 9·32) are for non-condensing refrigerant vapor inside, or outside, vertical or horizontal pipes. These solutions complement the graphs which have been presented for subcooled refrigerants. The principal difference between the graphical solutions for subcooled liquid and for superheated vapor is that the latter graphs use weight velocity, rather than fluid velocity, as required basic data; this change is necessitated because of the wide range of variation of vapor density as a function of pressure. The weight density is, however, a very simple term to evaluate since it is equal to the product of fluid velocity (expressed in feet per second) and fluid density (in units of pounds per cubic foot). Values of the density can be readily obtained as the inverse of specific volume as given in standard tables of thermodynamic properties of refrigerants or from pressure-enthalpy diagrams for refrigerants. (All graphs, Figs. 9·1 through 9·32, are solutions of equations 9·25, 9·27, and 9·29.)

Film coefficients for refrigerants in vapor form are useful in calculating the degree of superheat picked up in piping connecting the evaporator with the compressor and also in calculations for the extent of desuperheating effect in piping between the compressor discharge and the condenser. Where highly superheated vapor is supplied to the condenser, design of transfer surfaces within the condenser itself will require consideration of the preliminary section in which the transfer is from dry gas (superheated vapor) rather than from the condensing vapor. Since the film coefficient for cooling of superheated vapor is only a small fraction of that for condensing vapor it follows that failure to take account of the added condenser surface required in the desuperheating section may be responsible for a substantial increase in the condenser pressure (resulting from the necessity of maintaining an average condenser-cooling water temperature difference greatly in excess of the design value). Thus engineering consideration of the film coefficient for desuperheating is essential to maintenance of reasonable operating economy.

From the standpoint of piping insulation, accurate knowledge of the vapor film coefficient is also important since only in terms of the known overall coefficient of heat transfer can the optimum economic thickness of insulation be established. In this connection attention is directed to the fact that refrigerant line velocities are usually designed in terms

† These graphical solutions originally appeared in a series of technical articles by Raber and Hutchinson, published in *Heating and Ventilating*; they are used here by permission.

of an arbitrarily selected pressure drop which, economically speaking, is fixed in terms of the increased compression energy requirements associated with lowered suction pressure; if consideration were also given to the reduced heat transfer rate associated with lowered line velocity it is very probable that the optimum size of piping between evaporator and compressor would be increased and the optimum thickness of insulation, on this piping, decreased.

Example. A refrigeration compressor discharges ammonia vapor at a pressure corresponding to a saturation temperature of 79°F and with 71°F of superheat. The pipe has an inside diameter of $1\frac{1}{4}$ in., and flow occurs at a fluid velocity of 43.2 fps (velocities up to 50 fps are satisfactory for inside film coefficient of heat transfer based on such service). Determine the vapor conditions at the point of discharge from the compressor.

Solution. On Fig. 8-3 read that the saturation pressure of ammonia vapor corresponding to a saturation temperature of 79°F is 150 psia. The actual temperature of the vapor from the condenser is $71 + 79 = 150^{\circ}\text{F}$, and at a state of 150 psia and 150°F the specific volume is read as 2.4 cu ft/lb; therefore the weight density must be the inverse of this or $1/2.4 = 0.417$ lb/cu ft. The weight velocity in the discharge pipe is therefore $43.2 \times 0.417 = 18$ lb/(sec) (sq ft).

Now enter Fig. 9-23 at the known weight velocity of 18 (refer to dotted example line on the figure) and rise to intersection with the horizontal line for $1\frac{1}{4}$ -in. inside diameter. From here follow the sloping directrix lines up to the heat transfer scale, then rise vertically to intersection with the horizontal line marked 150°F . From this intersection move down parallel to the sloping directrix lines to intersect the heat transfer scale at a value of 65 Btu/(hr) (sq ft)($^{\circ}\text{F}$) for the inside film coefficient of heat transfer.

Figure 9-24 is used in exactly the same manner as Fig. 9-23, but with outside diameter instead of inside and with weight velocity based on flow across and normal to the single pipe or tube.

9-13. Thermal Resistance. In most refrigeration problems involving heat transfer the energy passes from a fluid (usually air, water, or a refrigerant) through some type of wall (as an insulated floor, or a pipe, or a metal plate) to a fluid at lower temperature. Thus the heat flow occurs by three convection, conduction, and convection processes in series, with the added possibility of radiant effects occurring in parallel with the convection transfer on both sides of the wall; if the fluid is a liquid, however, radiation does not take place. The visualization of series heat transfer with the obvious requirement, for steady state, that the transfer rate must be the same for all three processes leads to the idea of thermal resistance and the possibility of evaluating the total resistance of any heat flow system by summing the individual resistances of the series processes. If the temperature drop for any part of the

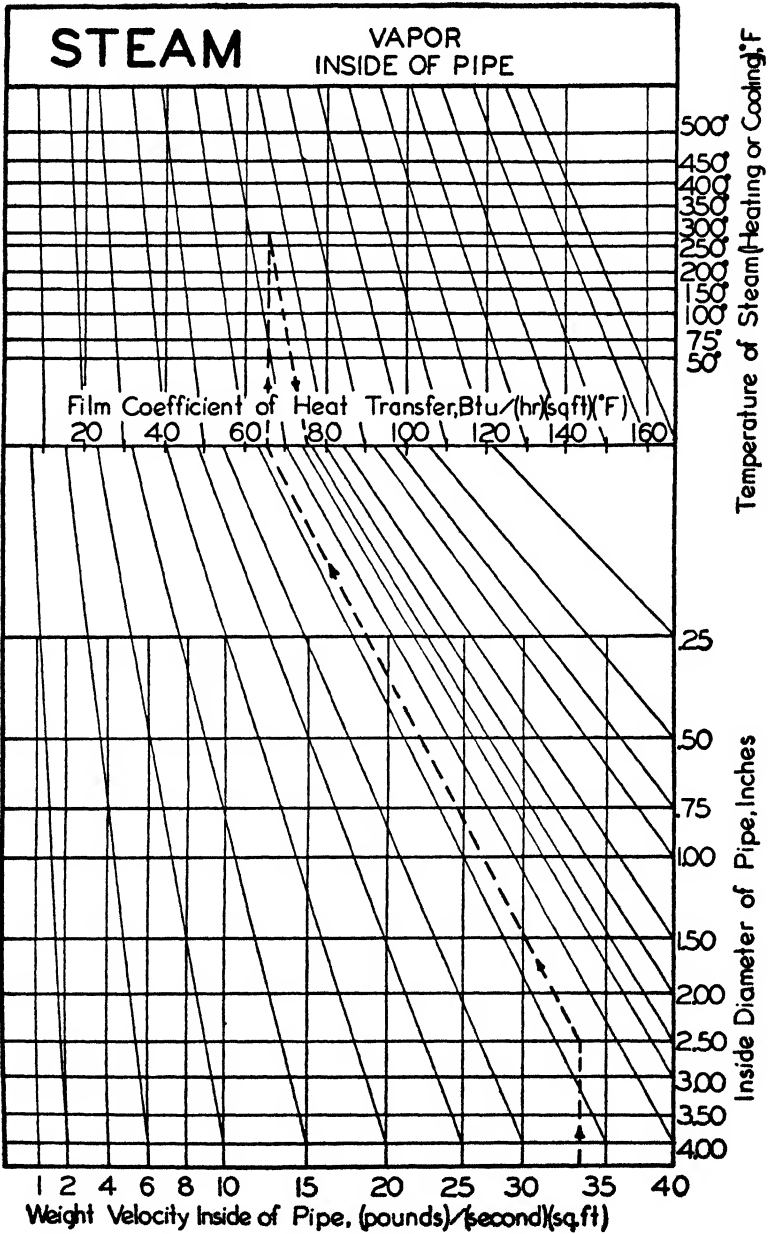


FIG. 9-21.

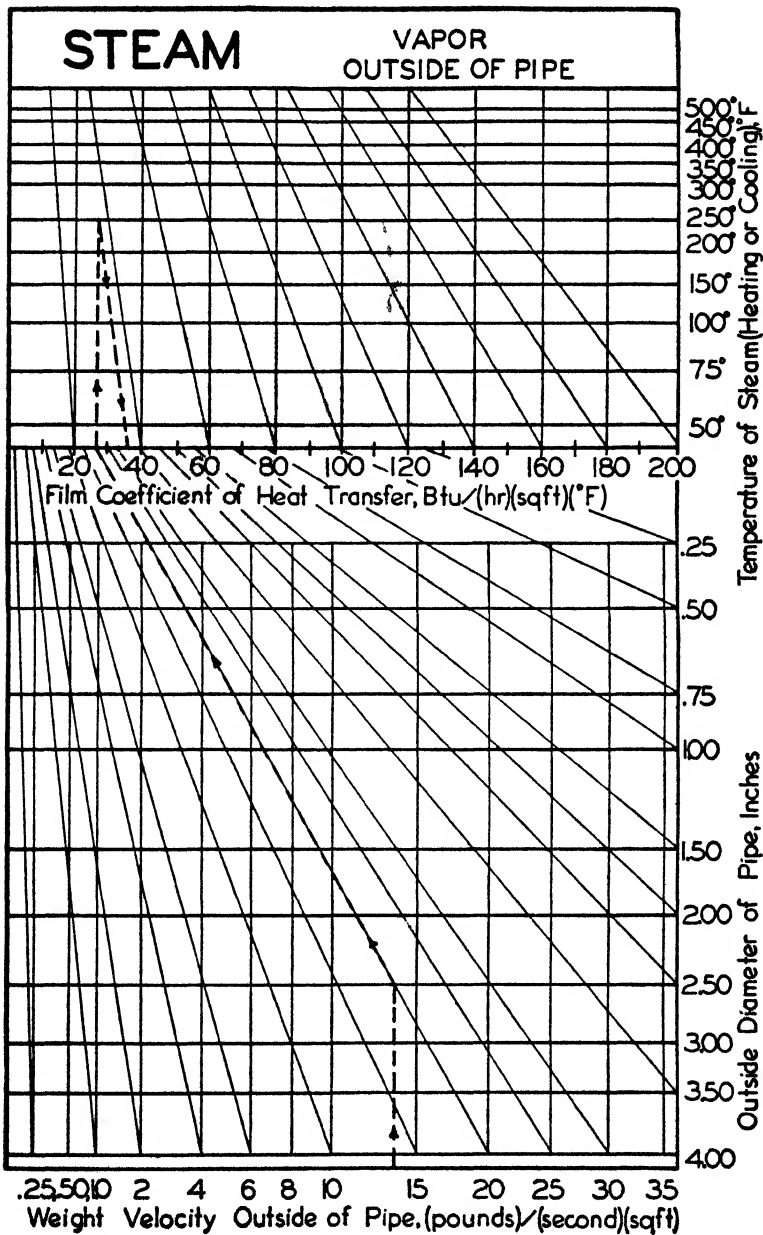


FIG. 9-22.

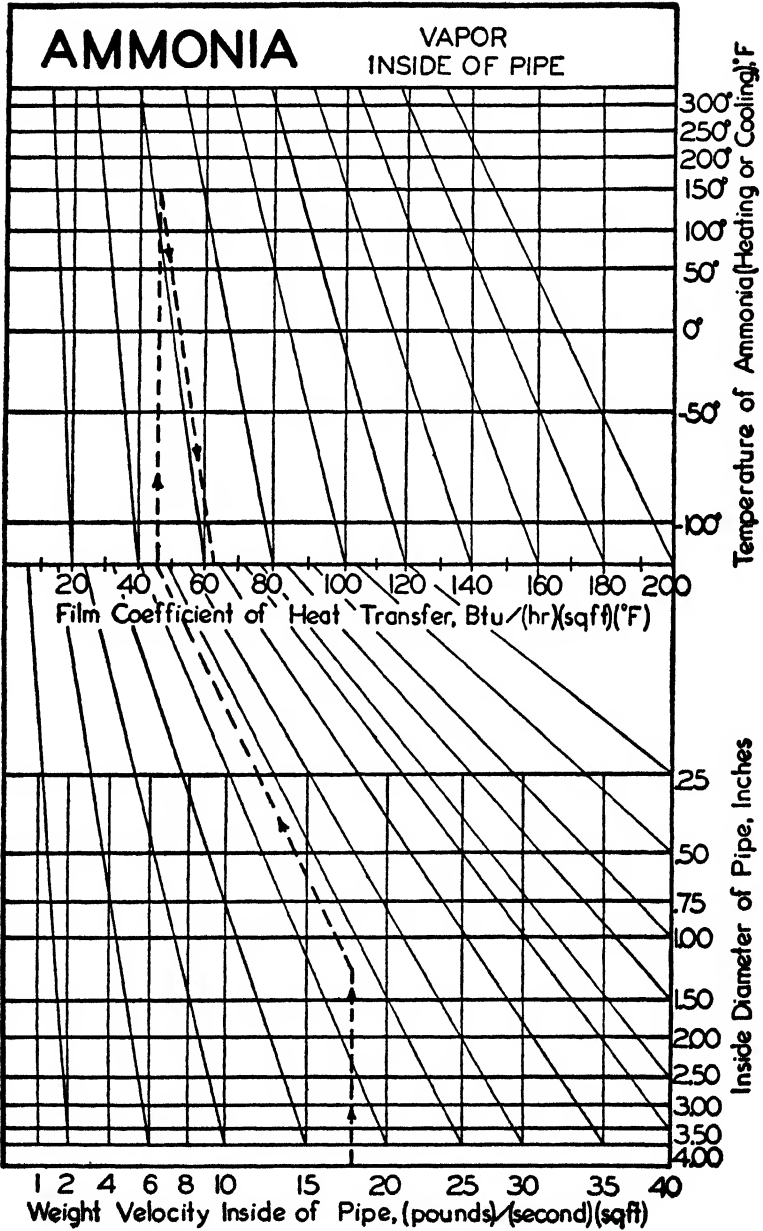


FIG. 9-23.

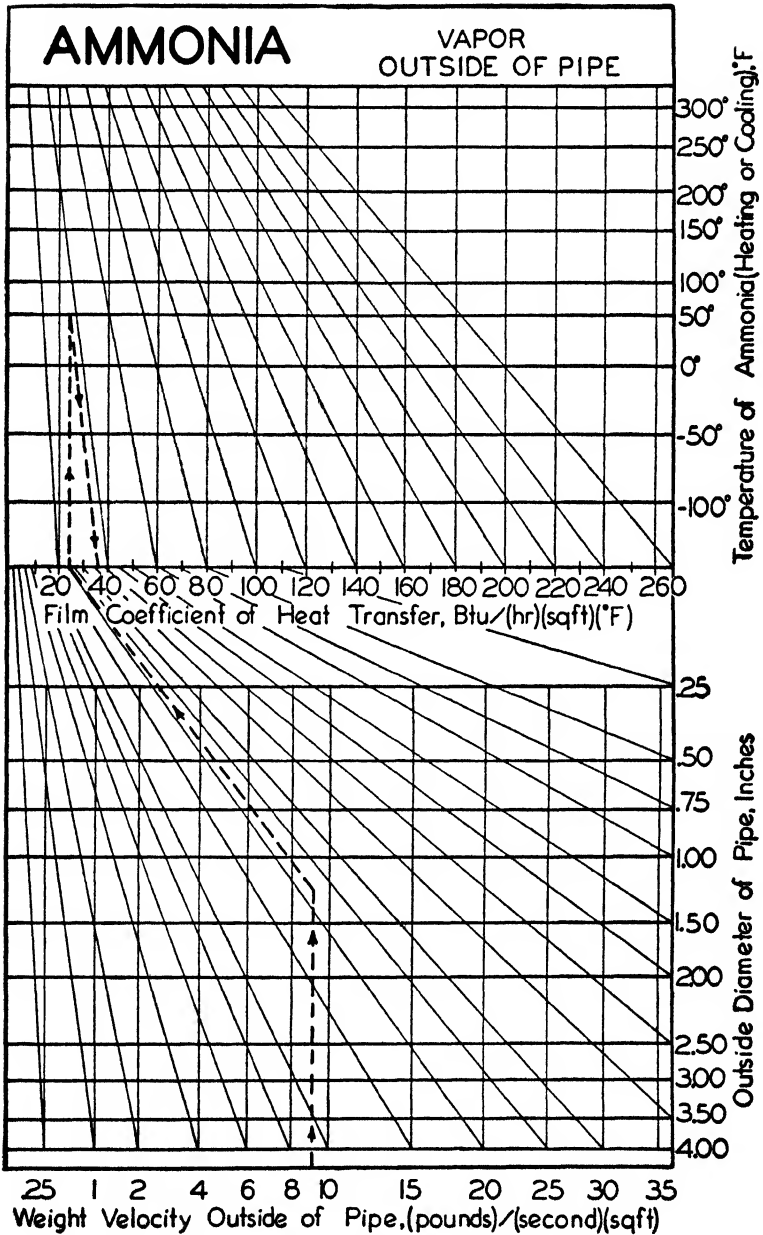


FIG. 9-24.

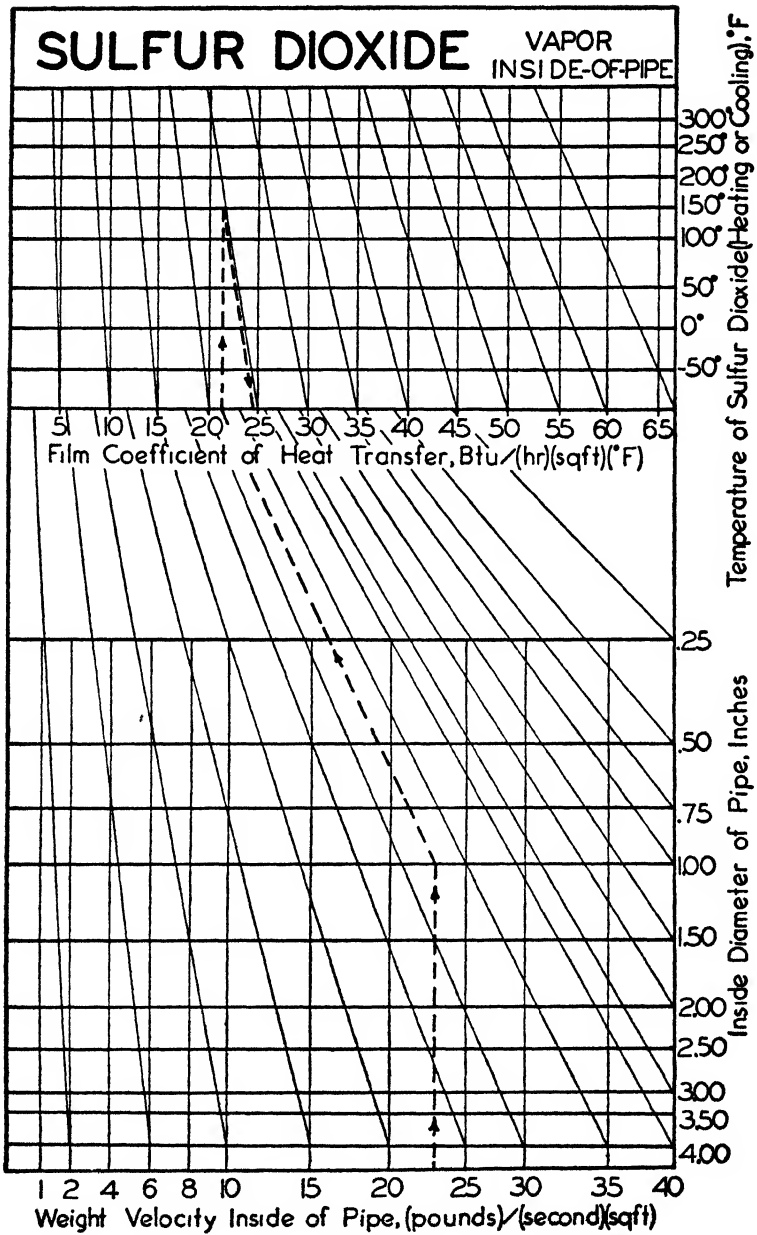


FIG. 9-25.

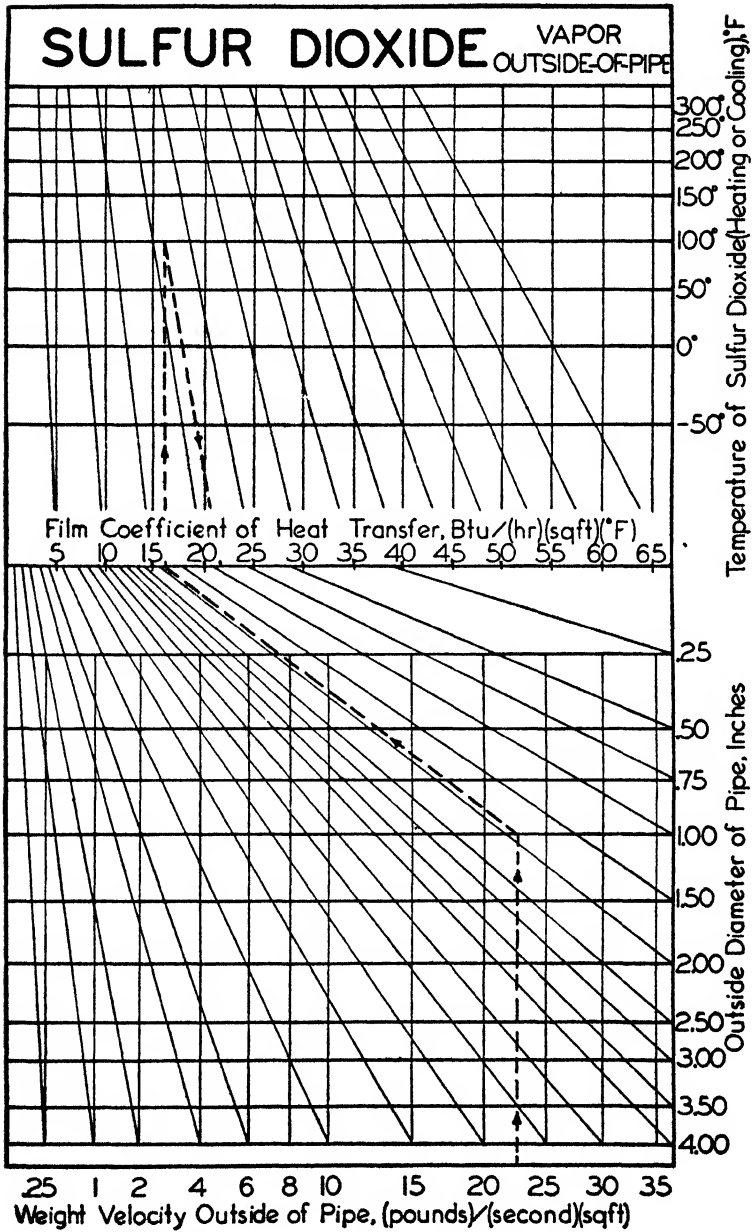


FIG. 9-26.

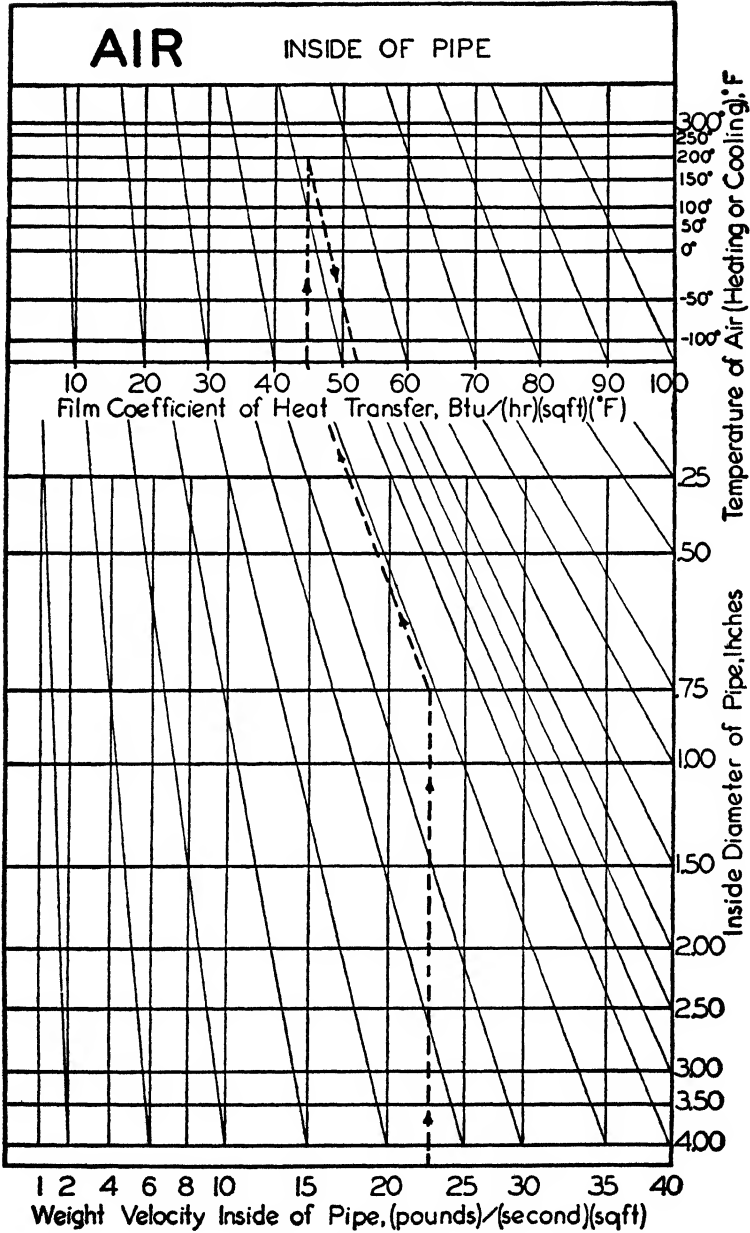


FIG. 9·27.

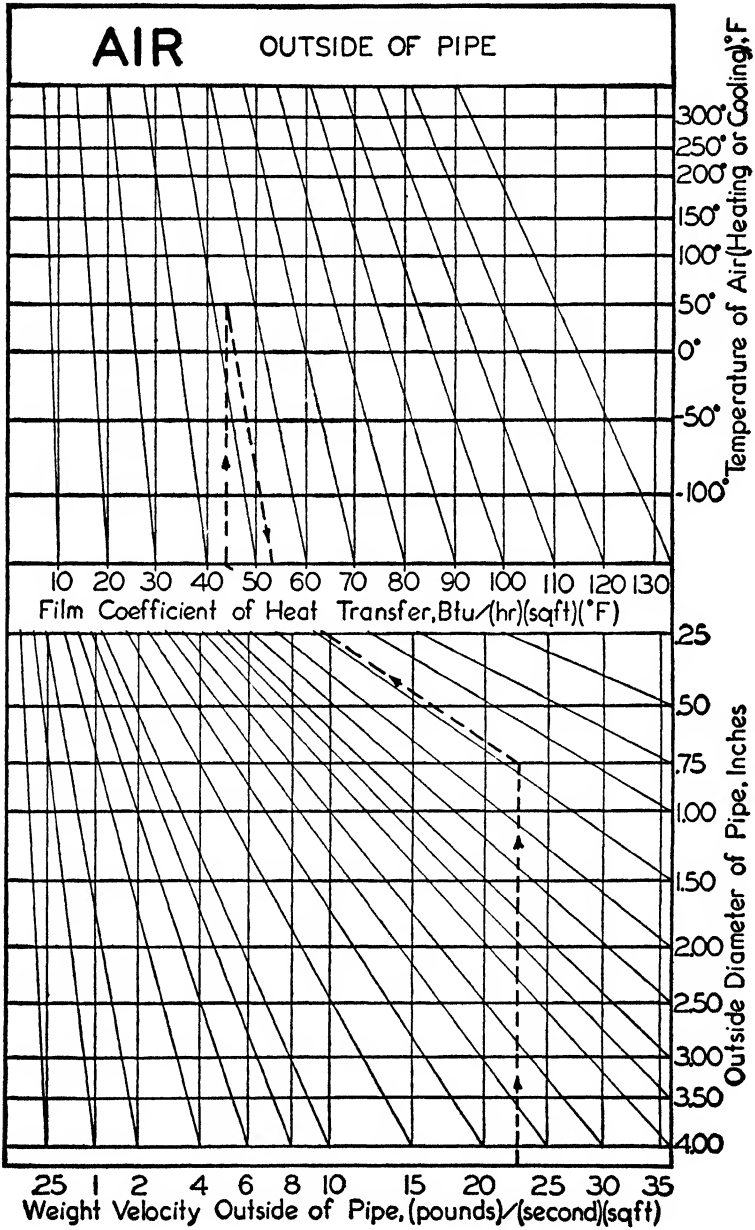


FIG. 9-28.

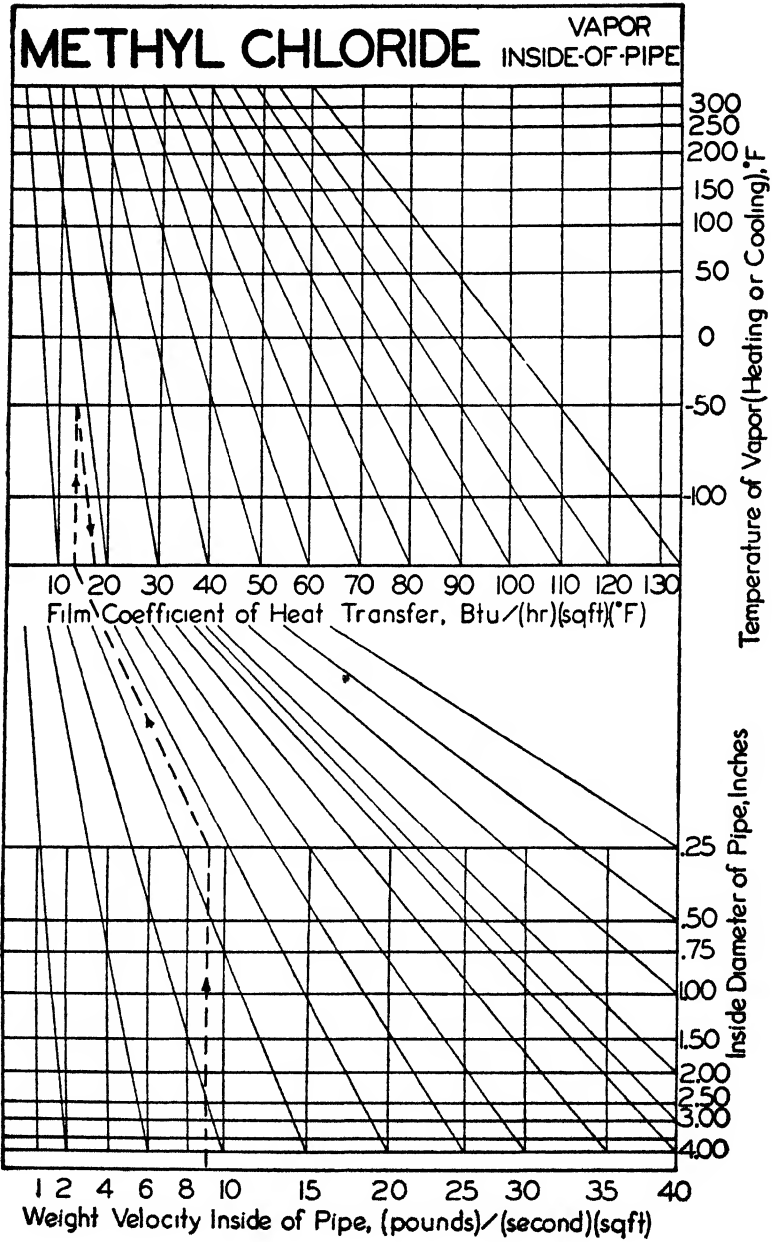


FIG. 9-29.

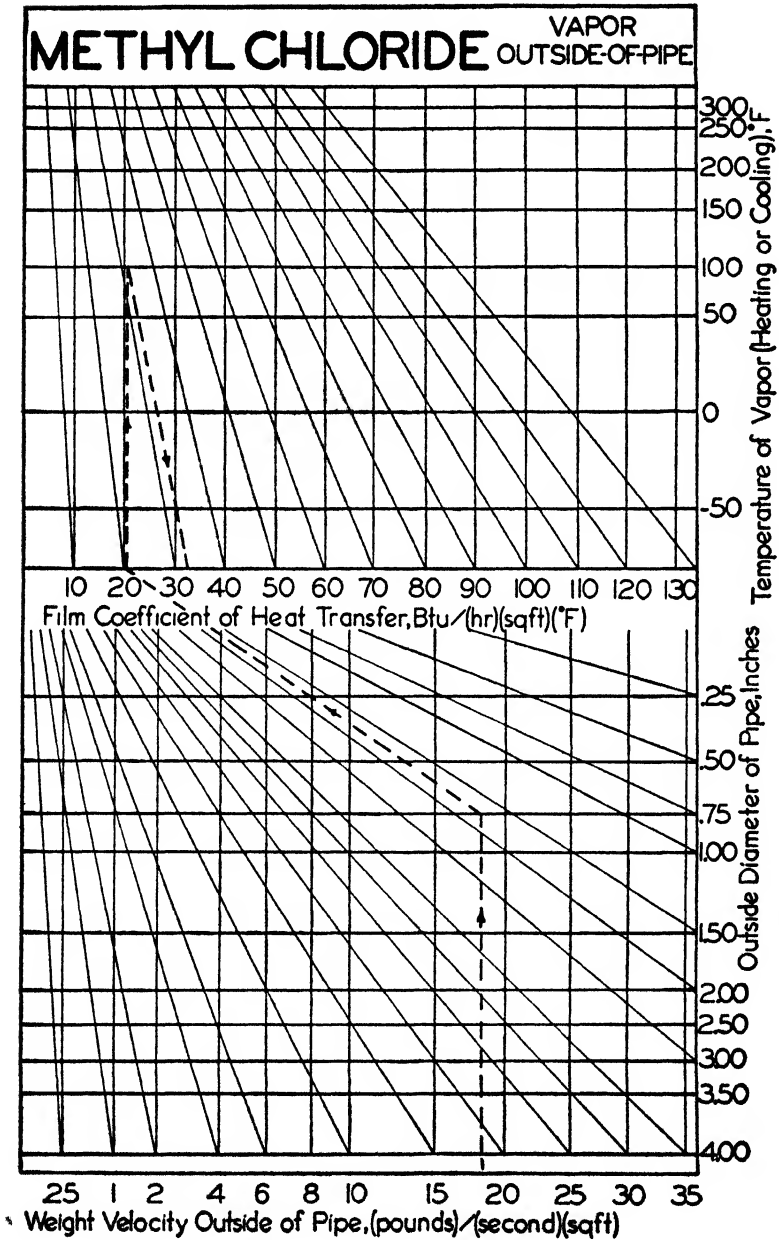


FIG. 9-80.

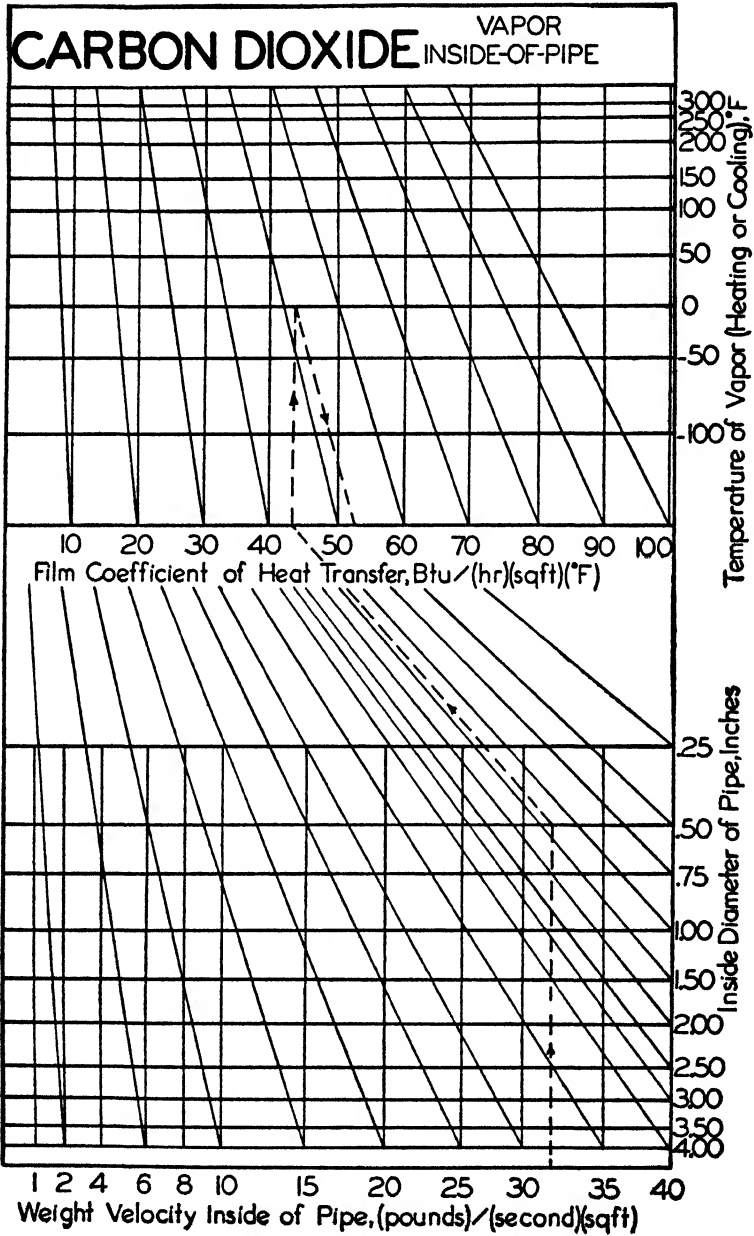


FIG. 9-31.

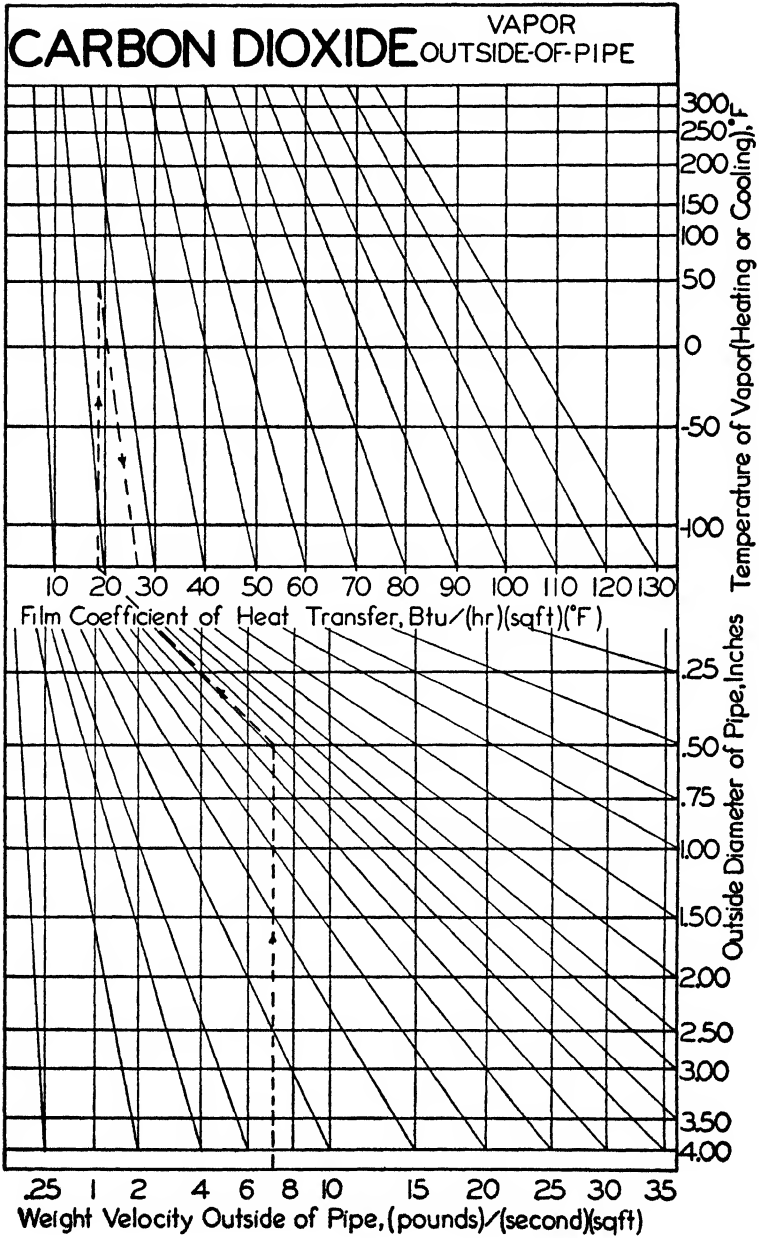


FIG. 9-32.

series is considered the driving potential, the resistance for a conduction process (from equation 9·1) will be L/kA and the resistance for a convection process will be $1/hA$, where the A is in each case the mean area of the path along which the particular transfer process is occurring. Thus for transfer from a fluid through a solid to a fluid the total resistance would be

$$R = r_{v_1} + r_{c_w} + r_{v_2} = \frac{1}{h_1 A_1} + \frac{L}{k A_w} + \frac{1}{h_2 A_2} \quad (9\cdot31)$$

and the total rate of heat transfer would be given by the overall thermal potential divided by the total resistance

$$q = \frac{t_h - t_c}{R} \quad (9\cdot32)$$

The reciprocal of the total resistance is defined as the overall coefficient of heat transfer U and like the film coefficient is expressed in units of Btu/(hr)(sq ft)(°F).

9·14. Pipe Wall Resistance. In many practical problems the designer's principal interest is in the overall rate of heat transfer per lineal foot of pipe. To assist in obtaining this value Tables 9·10 through 9·14 have been prepared.† Each table is for pipe or tube of a different metal and of standard wall thickness (which varies, of course, with the diameter). The resistance to conduction of unit length of any particular size and type of pipe is given near the top of each table, whereas the main body of each table gives the resistances of the inside and outside films per lineal foot of pipe or tube based on values of the film coefficient (per square foot of pipe or tube) listed in the row at the left side. Thus, if the inside film coefficient for a fluid passing through a 2½-in. outside diameter type M copper tube has been determined from the proper graphical solution to be 900 Btu/(hr)(sq ft)(°F), the corresponding inside film resistance per lineal foot r_i can be read from Table 9·13 as 0.0017. Similarly for a known outside film coefficient of say 400 Btu/(hr)(sq ft)(°F) the external film resistance r_o for the same tube would be 0.00364, so the total resistance of the two films would be 0.00534. From the same table the wall resistance of 2½-in. type M tubing is 0.0000431, so the total resistance of the system is 0.00538, giving an overall coefficient of $1/0.00538 = 186$ Btu/(hr)(sq ft)(°F). To distinguish this overall coefficient *per lineal foot of pipe* from the usual overall coefficient per square foot of surface the former will be identified as U_1 in contrast with U for the latter.

† These tables and Tables 9·15 and 9·16 also are reprinted, by permission, from a series of articles by Raber and Hutchinson which originally appeared in *Heating and Ventilating*.

The example just worked shows clearly that the major resistances to heat transfer occur in the films rather than in the pipe wall. If pipe wall resistance had been neglected in the preceding example the resultant value of U_1 would have been 187 Btu/(hr)(sq ft)(°F), which is well within the limit of engineering accuracy. For heat transfer through metallic flat plates Table 9·15 gives the sum of the reciprocals, R , and Table 9·16 the inverse of the two film coefficients, and this value can usually be taken as equivalent to the overall resistance to heat transfer per square foot of surface per degree Fahrenheit temperature difference. For use with pipes and tubes the value of R from Table 9·15 should be used with the inside pipe surface, the smaller area thereby partially offsetting neglect of the wall resistance. Table 9·16 is an abbreviated form of Table 9·15 but it gives the combined conductance instead of combined resistance of the two films.

Example. A Freon-11 heat exchanger consists of a bundle of 3-in. nominal diameter extra heavy wrought iron staggered tubes with liquid flowing in the tubes at a velocity of 3 fps and cooling from an average temperature of 50° F. Outside, and normal to the tubes, is liquid Freon-11 at an average temperature of 35° F and with a velocity through the narrowest cross section between the tubes of 4 fps. Determine the rate of heat transfer per lineal foot of tube and compare it with the heat transfer rate which would exist in an exchanger having the tubes in line and with the same spacing between tubes at the narrowest section. The inside and outside diameters of 3-in. extra heavy wrought iron pipe are 2.9 and 3.5 in., respectively.

Solution. The inside film coefficient is found from Fig. 9·9 by entering the velocity scale at 3 fps, rising to an interpolated horizontal for 2.9 in. I.D., following the directrix to the base scale, rising vertically to the horizontal for 50° F cooling and returning along a directrix to read the film coefficient as 161 Btu/(hr)(sq ft)(°F). Entering Table 9·14 at this value of h , move horizontally over to r_i column for 3-in. pipe and interpolate to find 0.0106 as the inside film resistance per lineal foot of pipe.

The base value of the outside film coefficient (for flow normal to a single pipe) is determined from Fig. 9·10 as 157 Btu/(hr)(sq ft)(°F). The procedure for obtaining this value is shown by the dashed line on the figure, and it will be noted that for the special case of Freon-11 the temperature of the fluid (in the range from 0° to 100° F) does not influence the value of the film coefficient.

For flow normal to a bundle of staggered tubes the base value of h_o can be increased by 30 per cent, as previously discussed; hence, for this problem the corrected value of the outside film coefficient is $1.3 \times 157 = 204$. Entering Table 9·14 at this value and moving over to the column for r_o under 3-in. pipe interpolate to obtain 0.00542 as the outside film

TABLE 9-10
THERMAL RESISTANCES PER LINEAL FOOT OF STANDARD WEIGHT PIPE¹
(ASA Schedule 40-B-36-10)

		Pipe Diameter, Inches																			
		¾		½		¼		1		1½		2		2½		3		3½		4	
Steel W.I.s	h	r _p	r _w	r _p	r _w	r _p	r _w	r _p	r _w	r _p	r _w	r _p	r _w	r _p	r _w	r _p	r _w	r _p	r _w	r _p	r _w
		0 00238	0 00183	0 00178	0 00137	0 00147	0 00110	0 00138	0 00103	0 00101	0 000848	0 000830	0 000834	0 000830	0 000695	0 000691	0 000804	0 000601	0 000733	0 000548	0 000681
50	0 209	0 1415	0 123	0 0910	0 0928	0 0729	0 0581	0 0475	0 0402	0 0369	0 0322	0 0309	0 0265	0 0218	0 0215	0 0191	0 0191	0 0191	0 0191	0 0190	0 0170
100	0 105	0 0707	0 0614	0 0455	0 0463	0 0364	0 0290	0 0237	0 0201	0 0185	0 0161	0 0155	0 0133	0 0124	0 0108	0 00954	0 00948	0 00954	0 00948	0 00948	0 00848
200	0 0525	0 0354	0 0307	0 0227	0 0232	0 0182	0 0145	0 0119	0 0100	0 00923	0 0803	0 0773	0 0664	0 0623	0 0546	0 0477	0 0474	0 0477	0 0474	0 0474	0 0424
300	0 0350	0 0236	0 0205	0 0152	0 0154	0 0121	0 00968	0 00790	0 00670	0 00616	0 00537	0 00516	0 00443	0 00415	0 00364	0 00359	0 00318	0 00359	0 00318	0 00318	0 00283
400	0 0263	0 0177	0 0154	0 0114	0 0116	0 00909	0 00810	0 00725	0 00593	0 00503	0 0462	0 00402	0 00387	0 00334	0 00312	0 00273	0 00269	0 00239	0 00269	0 00237	0 00212
500	0 0210	0 0141	0 0123	0 00909	0 00928	0 00727	0 00728	0 00581	0 00475	0 00369	0 00322	0 00309	0 00266	0 00249	0 00218	0 00215	0 00191	0 00215	0 00191	0 00190	0 00170
600	0 0175	0 0118	0 0102	0 00758	0 00773	0 00606	0 00607	0 00484	0 00395	0 00308	0 00268	0 00258	0 00221	0 00207	0 00182	0 00179	0 00159	0 00179	0 00159	0 00158	0 00141
700	0 0150	0 0101	0 00878	0 00650	0 00662	0 00519	0 00519	0 00413	0 00338	0 00284	0 00230	0 00221	0 00190	0 00178	0 00156	0 00154	0 00136	0 00154	0 00136	0 00135	0 00121
800	0 0131	0 00884	0 00768	0 00568	0 00580	0 00455	0 00456	0 00363	0 00297	0 00251	0 00201	0 00193	0 00166	0 00156	0 00136	0 00130	0 00119	0 00130	0 00119	0 00118	0 00106
900	0 0116	0 00786	0 00682	0 00505	0 00515	0 00404	0 00404	0 00323	0 00264	0 00223	0 00205	0 00179	0 00172	0 00148	0 00139	0 00120	0 00106	0 00120	0 00106	0 00105	0 000842
1000	0 0105	0 00707	0 00614	0 00455	0 00463	0 00364	0 00364	0 00290	0 00237	0 00201	0 00185	0 00155	0 00155	0 00133	0 00124	0 00109	0 00108	0 000854	0 000854	0 000848	0 000848
1100	0 00954	0 00545	0 00568	0 00414	0 00421	0 00331	0 00332	0 00264	0 00215	0 00183	0 00168	0 00141	0 00141	0 00121	0 00113	0 000982	0 000978	0 000868	0 000868	0 000862	0 000771
1200	0 00875	0 00500	0 00512	0 00379	0 00386	0 00303	0 00304	0 00242	0 00198	0 00168	0 00154	0 00129	0 00111	0 00104	0 000909	0 000896	0 000795	0 000790	0 000790	0 000707	0 000707
1300	0 00807	0 00444	0 0472	0 00350	0 00356	0 00279	0 00280	0 00220	0 00182	0 00156	0 00142	0 00119	0 00102	0 000938	0 000839	0 000827	0 000734	0 000734	0 000729	0 000652	0 000652
1400	0 00750	0 00405	0 0439	0 00324	0 00331	0 00260	0 00261	0 00207	0 00169	0 00144	0 00132	0 00111	0 00102	0 000949	0 000859	0 000779	0 000768	0 000682	0 000682	0 000676	0 000608
1500	0 00700	0 00473	0 0409	0 00303	0 00309	0 00242	0 00243	0 00193	0 00158	0 00134	0 00123	0 00103	0 000886	0 000830	0 000727	0 000718	0 000636	0 000636	0 000632	0 000585	0 000585
1600	0 00656	0 00442	0 0384	0 00284	0 00289	0 00227	0 00228	0 00181	0 00148	0 00126	0 00115	0 000967	0 000820	0 000778	0 000682	0 000673	0 000596	0 000596	0 000592	0 000530	0 000530
1700	0 00618	0 00416	0 0361	0 00267	0 00273	0 00214	0 00214	0 00171	0 00139	0 00118	0 00109	0 000910	0 000781	0 000742	0 000641	0 000633	0 000561	0 000561	0 000556	0 000499	0 000499
1800	0 00583	0 00393	0 0341	0 00253	0 00259	0 00202	0 00202	0 00161	0 00132	0 00112	0 0103	0 000894	0 000859	0 000737	0 000692	0 000606	0 000598	0 000598	0 000593	0 000530	0 000530
1900	0 00553	0 00372	0 0323	0 00239	0 00244	0 00191	0 00191	0 00155	0 00125	0 00106	0 000972	0 000814	0 000809	0 000689	0 000655	0 000574	0 000568	0 000568	0 000562	0 000499	0 000499
2000	0 00525	0 00354	0 0307	0 00227	0 00232	0 00182	0 00182	0 00145	0 00119	0 00100	0 000923	0 000804	0 000773	0 000664	0 000622	0 000546	0 000538	0 000538	0 000474	0 000474	0 000474

¹ r_f = inside film resistance; r_p = pipe wall resistance; r_w = outside film resistance; h = film coefficient of heat transfer from charts
s Wrought iron.

TABLE 9-11
THERMAL RESISTANCES PER LINEAL FOOT OF TYPE K STANDARD TUBING¹

Mate- rial	Tubing Diameter, Inches											
	Values of r_p											
	$\frac{1}{8}$	$\frac{1}{4}$	$\frac{3}{8}$	1	1½	2	2½	3	3½	4	r_i	r_o
P.Cu. ²	0.000159	0.000144	0.000137	0.000104	0.0000790	0.0000692	0.0000639	0.0000615	0.0000583	0.0000572		
A	r_i	r_o	r_i	r_o	r_i	r_o	r_i	r_o	r_i	r_o	r_i	r_o
50	0.246	0.145	0.122	0.0766	0.0516	0.0472	0.0390	0.0260	0.0228	0.0211	0.0198	0.0185
100	0.123	0.102	0.0725	0.0383	0.0258	0.0236	0.0195	0.0180	0.0113	0.0105	0.00990	0.00927
200	0.0614	0.0509	0.0363	0.0192	0.0129	0.0118	0.00975	0.00900	0.00564	0.00527	0.00495	0.00463
300	0.0409	0.0336	0.0242	0.0128	0.00860	0.00785	0.00650	0.00600	0.00376	0.00352	0.00330	0.00309
400	0.0307	0.0256	0.0181	0.0113	0.00660	0.00585	0.00450	0.00400	0.00282	0.00264	0.00248	0.00232
500	0.0246	0.0204	0.0145	0.00874	0.00645	0.00589	0.00487	0.00450	0.00294	0.00281	0.00266	0.00250
600	0.0205	0.0169	0.0121	0.00728	0.00516	0.00472	0.00390	0.00360	0.00242	0.00239	0.00226	0.00211
700	0.0175	0.0145	0.0104	0.00671	0.00485	0.00443	0.00363	0.00300	0.00204	0.00204	0.00198	0.00185
800	0.0153	0.0127	0.00906	0.00623	0.00447	0.00408	0.00324	0.00257	0.00188	0.00188	0.00176	0.00165
900	0.0136	0.0113	0.00805	0.00577	0.00405	0.00368	0.00278	0.00225	0.00161	0.00161	0.00151	0.00141
1000	0.0123	0.009725	0.00689	0.005478	0.00392	0.00358	0.00263	0.00200	0.00153	0.00153	0.00141	0.00132
1100	0.0112	0.00925	0.00659	0.00508	0.00358	0.00326	0.00234	0.00180	0.00136	0.00136	0.00125	0.00116
1200	0.0102	0.00848	0.00604	0.00458	0.00319	0.00286	0.00197	0.00163	0.00122	0.00122	0.00113	0.00110
1300	0.00945	0.00733	0.00557	0.00408	0.00294	0.00261	0.00181	0.00150	0.00111	0.00111	0.00105	0.000960
1400	0.00877	0.00726	0.00517	0.00386	0.00273	0.00242	0.00168	0.00138	0.00102	0.00102	0.000940	0.000843
1500	0.00819	0.00679	0.00483	0.00368	0.00260	0.00229	0.00156	0.00128	0.00091	0.00091	0.000829	0.000772
1600	0.00769	0.00635	0.00453	0.00351	0.00242	0.00211	0.00144	0.00118	0.00083	0.00083	0.000754	0.000714
1700	0.00713	0.00599	0.00426	0.00339	0.00230	0.00199	0.00130	0.00104	0.00076	0.00076	0.000687	0.000648
1800	0.00632	0.00565	0.00403	0.00339	0.00225	0.00198	0.00122	0.00090	0.00061	0.00061	0.000532	0.000498
1900	0.00646	0.00535	0.00381	0.00321	0.00212	0.00188	0.00118	0.00082	0.00057	0.00057	0.000498	0.000465
2000	0.00614	0.00509	0.00363	0.00305	0.00201	0.00178	0.00102	0.00069	0.00044	0.00044	0.000363	0.000321

¹ r_i = inside film resistance; r_p = pipe wall resistance; r_o = outside film resistance; A = film coefficient of heat transfer from charts.
² Phosphorized copper.

TABLE 9-12
THERMAL RESISTANCES PER LINEAL FOOT OF TYPE L STANDARD TUBING¹

Mat- rial	Tubing Diameter, Inches											
	Values of r_p											
	$\frac{1}{8}$	$\frac{1}{4}$	$\frac{3}{8}$	1	1½	2	2½	3	3½	4	r_o	r_i
P.Cu. ²	0.000148	0.000116	0.0000923	0.0000760	0.0000653	0.0000580	0.0000535	0.0000505	0.0000483	0.0000467		
A	r_o	r_i	r_o	r_i	r_o	r_i	r_o	r_i	r_o	r_i	r_o	r_i
50	0.242	0.140	0.0874	0.0746	0.0608	0.0384	0.0310	0.0291	0.0259	0.0244	0.0223	0.0211
100	0.121	0.0700	0.0486	0.0373	0.0254	0.0180	0.0155	0.0145	0.0130	0.0122	0.0111	0.0105
200	0.0605	0.0350	0.0243	0.0187	0.0127	0.00962	0.00900	0.00775	0.00649	0.00611	0.00557	0.00527
300	0.0404	0.0236	0.0163	0.0124	0.00847	0.00785	0.00660	0.00517	0.00432	0.00407	0.00372	0.00352
400	0.0303	0.0255	0.0153	0.0109	0.00734	0.00689	0.00560	0.00417	0.00324	0.00306	0.00279	0.00264
500	0.0242	0.0204	0.0122	0.00873	0.00678	0.00641	0.00508	0.00364	0.00259	0.00244	0.00223	0.00211
600	0.0202	0.0169	0.0117	0.00810	0.00622	0.00584	0.00442	0.00300	0.00208	0.00204	0.00186	0.00176
800	0.0173	0.0145	0.0100	0.00713	0.00544	0.00511	0.00363	0.00257	0.00182	0.00182	0.00159	0.00151
900	0.0138	0.0127	0.00875	0.00667	0.00514	0.00485	0.00336	0.00225	0.00155	0.00155	0.00139	0.00132
900	0.0132	0.0113	0.00779	0.00580	0.00445	0.00424	0.00284	0.00200	0.00144	0.00144	0.00124	0.00117
1000	0.0121	0.0102	0.00700	0.00510	0.00377	0.00357	0.00214	0.00150	0.00118	0.00118	0.00101	0.000980
1100	0.0110	0.00925	0.00636	0.00442	0.00313	0.00294	0.00141	0.00098	0.00072	0.00072	0.00059	0.00059
1200	0.0101	0.00848	0.00584	0.00395	0.00264	0.00244	0.00091	0.00063	0.00046	0.00046	0.00032	0.00032
1300	0.00933	0.00773	0.00530	0.00360	0.00231	0.00214	0.00063	0.00046	0.00030	0.00030	0.00017	0.00017
1400	0.00866	0.00725	0.00500	0.00337	0.00212	0.00196	0.00046	0.00030	0.00018	0.00018	0.00008	0.00008
1500	0.00809	0.00719	0.00467	0.00312	0.00195	0.00181	0.00046	0.00030	0.00018	0.00018	0.00008	0.00008
1600	0.00758	0.00655	0.00437	0.00286	0.00181	0.00168	0.00046	0.00030	0.00018	0.00018	0.00008	0.00008
1700	0.00714	0.00599	0.00411	0.00266	0.00169	0.00157	0.00046	0.00030	0.00018	0.00018	0.00008	0.00008
1800	0.00674	0.00565	0.00389	0.00242	0.00158	0.00147	0.00046	0.00030	0.00018	0.00018	0.00008	0.00008
1900	0.00639	0.00532	0.00363	0.00220	0.00149	0.00138	0.00046	0.00030	0.00018	0.00018	0.00008	0.00008
2000	0.00605	0.00500	0.00338	0.00194	0.00141	0.00131	0.00046	0.00030	0.00018	0.00018	0.00008	0.00008

¹ r_o = inside film resistance; r_i = pipe wall resistance; r_o = outside film resistance; A = film coefficient of heat transfer from charts.
² Phosphor-bronze copper.

TABLE 9-13
THERMAL RESISTANCES PER LINEAL FOOT OF TYPE M COPPER TUBING¹

Material	Nominal Outside Diameter of Tube, Inches (Actual O.D. 1/8 in. greater than nominal)																									
	1/4		3/8		1/2		3/4		1		1 1/4		1 1/2		2		2 1/2		3		3 1/2		4			
	r _i	r _o	r _i	r _o	r _i	r _o	r _i	r _o	r _i	r _o	r _i	r _o	r _i	r _o	r _i	r _o	r _i	r _o	r _i	r _o	r _i	r _o	r _i	r _o		
P Cu ³	Values of r _p																									
A	0.0000446																									
50	0.235	0.204	0.1134	0.122	0.0940	0.0874	0.0724	0.0678	0.0500	0.0472	0.0380	0.0360	0.0291	0.0256	0.0244	0.0221	0.0211	0.0211	0.0194	0.0185	0.00927	0.00927	0.00927	0.00927	0.00927	0.00927
100	0.117	0.102	0.0672	0.0610	0.0470	0.0437	0.0362	0.0339	0.0280	0.0236	0.0190	0.0180	0.0145	0.0128	0.0122	0.0110	0.0105	0.0105	0.00973	0.00927	0.00486	0.00486	0.00486	0.00486	0.00486	0.00486
200	0.0688	0.0609	0.0336	0.0306	0.0235	0.0219	0.0181	0.0170	0.0125	0.0118	0.00951	0.00900	0.00765	0.00641	0.00611	0.00552	0.00527	0.00527	0.00486	0.00463	0.00243	0.00243	0.00243	0.00243	0.00243	0.00243
300	0.0394	0.0336	0.0224	0.0203	0.0157	0.0146	0.0121	0.0113	0.00834	0.00785	0.00634	0.00600	0.00510	0.00427	0.00407	0.00368	0.00352	0.00352	0.00324	0.00306	0.00163	0.00163	0.00163	0.00163	0.00163	0.00163
400	0.0294	0.0255	0.0168	0.0153	0.0118	0.0109	0.00906	0.00847	0.00625	0.00589	0.00475	0.00450	0.00383	0.00321	0.00306	0.00276	0.00264	0.00264	0.00243	0.00232	0.00119	0.00119	0.00119	0.00119	0.00119	0.00119
500	0.0235	0.0204	0.0134	0.0123	0.00940	0.00874	0.00724	0.00678	0.00500	0.00472	0.00380	0.00360	0.00306	0.00256	0.00244	0.00221	0.00211	0.00211	0.00194	0.00186	0.00073	0.00073	0.00073	0.00073	0.00073	0.00073
600	0.0196	0.0160	0.0112	0.0102	0.00785	0.00728	0.00603	0.00565	0.00417	0.00392	0.00317	0.00300	0.00257	0.00218	0.00204	0.00184	0.00176	0.00176	0.00163	0.00154	0.00041	0.00041	0.00041	0.00041	0.00041	0.00041
700	0.0168	0.0145	0.00960	0.00871	0.00672	0.00623	0.00517	0.00488	0.00357	0.00336	0.00272	0.00257	0.00218	0.00188	0.00175	0.00158	0.00151	0.00151	0.00139	0.00132	0.00021	0.00021	0.00021	0.00021	0.00021	0.00021
800	0.0147	0.0127	0.00840	0.00763	0.00588	0.00545	0.00452	0.00424	0.00313	0.00284	0.00238	0.00225	0.00188	0.00160	0.00142	0.00128	0.00121	0.00121	0.00108	0.00103	0.00011	0.00011	0.00011	0.00011	0.00011	0.00011
900	0.0131	0.0113	0.00747	0.00677	0.00523	0.00485	0.00402	0.00377	0.00278	0.00262	0.00211	0.00200	0.00170	0.00145	0.00128	0.00116	0.00110	0.00110	0.00097	0.00092	0.00001	0.00001	0.00001	0.00001	0.00001	0.00001
1000	0.0117	0.0102	0.00672	0.00610	0.00470	0.00437	0.00362	0.00339	0.00250	0.00236	0.00190	0.00180	0.00153	0.00128	0.00116	0.00108	0.00103	0.00103	0.00090	0.00085	0.00001	0.00001	0.00001	0.00001	0.00001	0.00001
1100	0.0107	0.00928	0.00611	0.00555	0.00427	0.00395	0.00329	0.00308	0.00228	0.00214	0.00173	0.00163	0.00139	0.00116	0.00108	0.00100	0.00095	0.00095	0.00082	0.00077	0.00001	0.00001	0.00001	0.00001	0.00001	0.00001
1200	0.00960	0.00848	0.00560	0.00508	0.00392	0.00364	0.00302	0.00283	0.00208	0.00196	0.00158	0.00150	0.00127	0.00107	0.00101	0.00092	0.00087	0.00087	0.00074	0.00069	0.00001	0.00001	0.00001	0.00001	0.00001	0.00001
1300	0.00904	0.00733	0.00517	0.00469	0.00362	0.00336	0.00278	0.00261	0.00192	0.00181	0.00146	0.00138	0.00118	0.00109	0.00102	0.00094	0.00089	0.00089	0.00076	0.00071	0.00001	0.00001	0.00001	0.00001	0.00001	0.00001
1400	0.00846	0.00728	0.00490	0.00448	0.00336	0.00312	0.00258	0.00242	0.00178	0.00168	0.00136	0.00128	0.00109	0.00104	0.00096	0.00088	0.00083	0.00083	0.00070	0.00065	0.00001	0.00001	0.00001	0.00001	0.00001	0.00001
1500	0.00784	0.00679	0.00448	0.00408	0.00314	0.00290	0.00241	0.00226	0.00167	0.00157	0.00127	0.00120	0.00100	0.00095	0.00087	0.00079	0.00073	0.00073	0.00060	0.00055	0.00001	0.00001	0.00001	0.00001	0.00001	0.00001
1600	0.00735	0.00635	0.00420	0.00381	0.00294	0.00272	0.00226	0.00212	0.00156	0.00147	0.00119	0.00112	0.00090	0.00085	0.00076	0.00069	0.00063	0.00063	0.00050	0.00045	0.00001	0.00001	0.00001	0.00001	0.00001	0.00001
1700	0.00691	0.00599	0.00396	0.00359	0.00276	0.00256	0.00213	0.00198	0.00147	0.00138	0.00112	0.00106	0.00086	0.00081	0.00072	0.00065	0.00059	0.00059	0.00046	0.00041	0.00001	0.00001	0.00001	0.00001	0.00001	0.00001
1800	0.00653	0.00565	0.00373	0.00339	0.00261	0.00242	0.00201	0.00188	0.00139	0.00131	0.00106	0.00100	0.00080	0.00075	0.00066	0.00060	0.00054	0.00054	0.00041	0.00036	0.00001	0.00001	0.00001	0.00001	0.00001	0.00001
1900	0.00619	0.00535	0.00354	0.00321	0.00248	0.00230	0.00190	0.00178	0.00132	0.00124	0.00100	0.00094	0.00074	0.00069	0.00060	0.00054	0.00048	0.00048	0.00035	0.00030	0.00001	0.00001	0.00001	0.00001	0.00001	0.00001
2000	0.00588	0.00509	0.00336	0.00303	0.00235	0.00219	0.00181	0.00170	0.00125	0.00118	0.00095	0.00090	0.00070	0.00065	0.00056	0.00050	0.00044	0.00044	0.00031	0.00026	0.00001	0.00001	0.00001	0.00001	0.00001	0.00001

¹ r_i = inside film resistance; r_p = pipe wall resistance; r_o = outside film resistance; A = film coefficient of heat transfer from charts.
² Phosphorized copper.

TABLE 9-14
THERMAL RESISTANCES PER LINEAL FOOT OF EXTRA HEAVY IRON OR STEEL PIPE¹

Material	Nominal Diameter, Inches																			
	¾		½		¾		1		1½		2		2½		3		3½		4	
	r_i	r_o	r_i	r_o	r_i	r_o	r_i	r_o	r_i	r_o	r_i	r_o	r_i	r_o	r_i	r_o	r_i	r_o	r_i	r_o
St. ² W.I. ²	0.00346	0.00260	0.00260	0.00194	0.00210	0.00157	0.00103	0.00145	0.00144	0.00108	0.00124	0.000925	0.00129	0.000964	0.00114	0.000856	0.00105	0.000787	0.000992	0.000740
	0.00258	0.00194	0.00157	0.00119	0.00119	0.00088	0.00088	0.00064	0.00064	0.00040	0.00040	0.00025	0.00025	0.00016	0.00016	0.00016	0.00016	0.00016	0.00016	0.00016
A	0.283	0.140	0.103	0.0727	0.0798	0.0581	0.0402	0.0394	0.0322	0.0268	0.0218	0.0191	0.0164	0.0133	0.0109	0.00854	0.00687	0.00548	0.00438	0.00338
	0.126	0.0707	0.0455	0.0364	0.0399	0.0290	0.0255	0.0201	0.0197	0.0161	0.0133	0.0109	0.00854	0.00687	0.00548	0.00438	0.00338	0.00258	0.00194	0.00140
100	0.0631	0.0354	0.0227	0.0258	0.0200	0.0145	0.0127	0.0101	0.0085	0.00684	0.00548	0.00438	0.00338	0.00258	0.00194	0.00140	0.00109	0.000854	0.000687	0.000548
200	0.0421	0.0236	0.0151	0.0172	0.0133	0.00968	0.00839	0.00670	0.00536	0.00442	0.00332	0.00268	0.00218	0.00182	0.00146	0.00119	0.000964	0.000787	0.000631	0.000498
300	0.0316	0.0175	0.0113	0.0124	0.00969	0.00897	0.00726	0.00636	0.00503	0.00493	0.00402	0.00322	0.00268	0.00218	0.00182	0.00146	0.00119	0.000964	0.000787	0.000631
400	0.0253	0.0141	0.00909	0.0103	0.00727	0.00589	0.00509	0.00402	0.00394	0.00322	0.00268	0.00218	0.00182	0.00146	0.00119	0.000964	0.000787	0.000631	0.000498	0.000380
500	0.0210	0.0118	0.00757	0.00858	0.00606	0.00665	0.00484	0.00424	0.00328	0.00268	0.00218	0.00182	0.00146	0.00119	0.000964	0.000787	0.000631	0.000498	0.000380	0.000283
600	0.0180	0.0101	0.00649	0.00736	0.00519	0.00570	0.00415	0.00363	0.00281	0.00230	0.00189	0.00156	0.00121	0.000964	0.000787	0.000631	0.000498	0.000380	0.000283	0.000200
700	0.0158	0.00883	0.00575	0.00668	0.00454	0.00499	0.00363	0.00281	0.00230	0.00189	0.00156	0.00121	0.000964	0.000787	0.000631	0.000498	0.000380	0.000283	0.000200	0.000141
800	0.0140	0.00785	0.00477	0.00569	0.00359	0.00404	0.00281	0.00230	0.00189	0.00156	0.00121	0.000964	0.000787	0.000631	0.000498	0.000380	0.000283	0.000200	0.000141	0.000100
900	0.0126	0.00707	0.00407	0.00493	0.00283	0.00323	0.00223	0.00219	0.00170	0.00132	0.00103	0.00082	0.00064	0.00054	0.00042	0.00032	0.00025	0.00019	0.00014	0.00010
1000	0.0115	0.00643	0.00358	0.00443	0.00244	0.00283	0.00199	0.00197	0.00161	0.00121	0.00096	0.00078	0.00064	0.00054	0.00042	0.00032	0.00025	0.00019	0.00014	0.00010
1100	0.0105	0.00589	0.00303	0.00378	0.00229	0.00268	0.00182	0.00183	0.00146	0.00119	0.00096	0.00078	0.00064	0.00054	0.00042	0.00032	0.00025	0.00019	0.00014	0.00010
1200	0.00972	0.00544	0.00268	0.00349	0.00200	0.00239	0.00168	0.00164	0.00133	0.00109	0.00085	0.00068	0.00054	0.00042	0.00032	0.00025	0.00019	0.00014	0.00010	0.00008
1300	0.00903	0.00505	0.00234	0.00324	0.00182	0.00223	0.00155	0.00152	0.00124	0.00103	0.00082	0.00064	0.00054	0.00042	0.00032	0.00025	0.00019	0.00014	0.00010	0.00007
1400	0.00842	0.00471	0.00207	0.00303	0.00164	0.00206	0.00144	0.00141	0.00115	0.00096	0.00078	0.00064	0.00054	0.00042	0.00032	0.00025	0.00019	0.00014	0.00010	0.00006
1500	0.00790	0.00442	0.00184	0.00284	0.00146	0.00189	0.00134	0.00131	0.00107	0.00088	0.00070	0.00054	0.00042	0.00032	0.00025	0.00019	0.00014	0.00010	0.00006	0.00005
1600	0.00743	0.00416	0.00167	0.00268	0.00133	0.00176	0.00126	0.00123	0.00100	0.00082	0.00064	0.00054	0.00042	0.00032	0.00025	0.00019	0.00014	0.00010	0.00006	0.00005
1700	0.00702	0.00393	0.00151	0.00253	0.00121	0.00164	0.00116	0.00116	0.00094	0.00076	0.00064	0.00054	0.00042	0.00032	0.00025	0.00019	0.00014	0.00010	0.00006	0.00005
1800	0.00665	0.00372	0.00136	0.00239	0.00109	0.00152	0.00104	0.00104	0.00084	0.00066	0.00054	0.00042	0.00032	0.00025	0.00019	0.00014	0.00010	0.00006	0.00005	0.00004
1900	0.00631	0.00354	0.00121	0.00225	0.00103	0.00145	0.00098	0.00098	0.00078	0.00064	0.00054	0.00042	0.00032	0.00025	0.00019	0.00014	0.00010	0.00006	0.00005	0.00004
2000	0.00603	0.00335	0.00107	0.00210	0.00096	0.00133	0.00094	0.00094	0.00076	0.00064	0.00054	0.00042	0.00032	0.00025	0.00019	0.00014	0.00010	0.00006	0.00005	0.00004

¹ r_i = inside film resistance; r_p = pipe wall resistance; r_o = outside film resistance; A = film coefficient of heat transfer from charts.
² Mild Steel.
³ Wrought Iron.

TABLE 9-15c
VALUES OF R FOR INSIDE FILM COEFFICIENTS FROM 200 TO 450

Outside Film Coefficient h_o	Inside Film Coefficient h_i										
	200	225	250	275	300	325	350	375	400	425	450
	Value of R										
1	1.00500	1.00444	1.0040	1.00364	1.00333	1.00308	1.00286	1.00267	1.00250	1.00235	1.00222
2	0.5050	0.5044	0.5040	0.5036	0.5033	0.5031	0.5029	0.5027	0.5025	0.5023	0.5022
4	0.2550	0.2544	0.2540	0.2536	0.2533	0.2531	0.2529	0.2527	0.2525	0.2523	0.2522
6	0.1720	0.1714	0.1710	0.1706	0.1703	0.1701	0.1699	0.1697	0.1695	0.1693	0.1692
8	0.1300	0.1294	0.1290	0.1286	0.1283	0.1281	0.1279	0.1277	0.1275	0.1273	0.1272
10	0.1050	0.1044	0.1040	0.1036	0.1033	0.1031	0.1029	0.1027	0.1025	0.1023	0.1022
15	0.0717	0.0711	0.0707	0.0703	0.0700	0.0698	0.0696	0.0694	0.0692	0.0690	0.0689
20	0.0550	0.0544	0.0540	0.0536	0.0533	0.0531	0.0529	0.0527	0.0525	0.0523	0.0522
25	0.0450	0.0444	0.0440	0.0436	0.0433	0.0431	0.0429	0.0427	0.0425	0.0423	0.0422
30	0.0383	0.0377	0.0373	0.0369	0.0366	0.0364	0.0362	0.0360	0.0358	0.0356	0.0355
35	0.0336	0.0330	0.0326	0.0322	0.0319	0.0317	0.0315	0.0313	0.0311	0.0309	0.0308
40	0.0300	0.0294	0.0290	0.0286	0.0283	0.0281	0.0279	0.0277	0.0275	0.0273	0.0272
45	0.0270	0.0264	0.0260	0.0256	0.0253	0.0251	0.0249	0.0247	0.0245	0.0243	0.0242
50	0.0250	0.0244	0.0240	0.0236	0.0233	0.0231	0.0229	0.0227	0.0225	0.0223	0.0222
60	0.0217	0.0211	0.0207	0.0203	0.0200	0.0198	0.0196	0.0194	0.0192	0.0190	0.0189
70	0.0193	0.0187	0.0183	0.0179	0.0176	0.0174	0.0172	0.0170	0.0168	0.0166	0.0165
80	0.0175	0.0169	0.0165	0.0161	0.0158	0.0156	0.0154	0.0152	0.0150	0.0148	0.0147
90	0.0161	0.0155	0.0151	0.0147	0.0144	0.0142	0.0140	0.0138	0.0136	0.0134	0.0133
100	0.0150	0.0144	0.0140	0.0136	0.0133	0.0131	0.0129	0.0127	0.0125	0.0123	0.0122
120	0.0133	0.0128	0.0123	0.0120	0.0117	0.0115	0.0113	0.0111	0.0109	0.0107	0.0106
140	0.0121	0.0116	0.0111	0.0108	0.0105	0.0103	0.0101	0.00981	0.00964	0.00949	0.00936
160	0.0112	0.0107	0.0102	0.00989	0.00959	0.00933	0.00911	0.00892	0.00875	0.00860	0.00847
180	0.0105	0.00999	0.00953	0.00919	0.00888	0.00863	0.00841	0.00822	0.00805	0.00790	0.00777
200	0.0100	0.00944	0.00900	0.00864	0.00833	0.00808	0.00786	0.00767	0.00750	0.00735	0.00722
225	..	0.00888	0.00844	0.00808	0.00777	0.00752	0.00730	0.00711	0.00694	0.00679	0.00666
250	0.00800	0.00764	0.00733	0.00708	0.00686	0.00667	0.00650	0.00635	0.00622
275	0.00728	0.00697	0.00672	0.00650	0.00631	0.00614	0.00599	0.00586
300	0.00686	0.00641	0.00619	0.00600	0.00583	0.00568	0.00555
325	0.00616	0.00594	0.00575	0.00558	0.00543	0.00530
350	0.00572	0.00553	0.00536	0.00521	0.00508
375	0.00534	0.00517	0.00502	0.00489
400	0.00500	0.00485	0.00472
425	0.00470	0.00457
450	0.00444

TABLE 9·16
 CONDUCTANCE
 [Btu/(hr)(sq ft)(°F)]

Inside Film Coefficient h_i	Outside Film Coefficient h_o									
	100	200	300	400	500	600	700	800	900	1000
100	50	67	75	80	84	86	88	89	90	91
200	67	100	120	133	143	150	155	160	164	167
300	75	120	150	172	188	200	210	218	225	231
400	80	133	172	200	222	240	254	267	277	286
500	84	143	188	222	250	273	292	308	322	333
600	86	150	200	240	273	300	323	343	360	375
700	87	155	210	254	292	323	350	373	394	411
800	89	160	218	267	308	343	373	400	424	445
900	90	164	225	277	322	360	394	424	450	474
1000	91	167	231	286	333	375	411	445	474	500

resistance per lineal foot. From the same table the wall resistance of 3-in. extra heavy, wrought iron pipe is read as 0.000856. The total fluid-to-fluid resistance of the staggered tube exchanger is then

$$R = 0.0106 + 0.000856 + 0.00542 = 0.0169$$

and the rate of heat transfer per lineal foot of tube is

$$\frac{t_h - t_c}{R} = \frac{50 - 35}{0.0169} = 887 \text{ Btu/hr}$$

For the in-line exchanger the value of h_o taken from the graph can be used directly; hence from Table 9·14 for $h = 157$ find $r_o = 0.0078$. Then

$$R = 0.0106 + 0.000856 + 0.0078 = 0.0193$$

and the rate of heat transfer per lineal foot of tube is

$$\frac{50 - 35}{0.0193} = 777 \text{ Btu/hr}$$

9·15. Condensation. By consideration of the average velocity of the liquid refrigerant, passing in viscous flow due to gravity along the metallic wall, and the flow of heat from the vapor to the wall by pure conduction through the film of condensate, the formula for the coefficient of heat transfer at the liquefaction surface becomes, for condensation free of air or other gases,

$$h = 0.725 \left(\frac{k^3 w^2 g L}{d \mu \Delta t} \right)^{1/4} \tag{9·33}$$

for horizontal pipes where the properties of the liquid are taken at the mean film temperature. In this formula

h is the average for the total wall in Btu/(hr)(sq ft)(°F)

k is the conductivity of the condensate Btu/(hr)(sq ft)(°F/ft)

g is in ft/hr² = 4.18×10^8

L is the latent heat of liquefaction of the liquid in Btu per pound

w is the density in pounds per cubic foot

μ is the viscosity in pounds per foot-hour = $2.42 \times z$ (centipoise)

d is the outside diameter in feet

Δt is the difference in temperature between the vapor and the surface wall temperature in Fahrenheit degrees

Example. A 25-ton refrigeration machine operates at 170 psia condensing, 86.3° F, and 25 psia evaporating pressure, using ammonia. The initial superheat in the condenser is 3.7° F (90° F), and there is no subcooling of the condensate. The condenser pipes are horizontal and are of 2-in. normal diameter. Water at 75° F passes inside the tubes, and there is a rise of temperature of 10° F. Find the overall coefficient of heat transfer and the total area required for the condenser.

Solution.

$$\text{Heat removed by the condenser} = \frac{634.4 - 139.3}{609.1 - 139.3} \times 25 \times 200 = 5269 \text{ Btu/min}$$

$$\text{Water required} = \frac{5269}{10} = 526.9 \text{ lb/min}$$

$$\text{Water velocity} = \frac{526.9 \times 144 \times 0.01608}{3.355 \times 60} = 6.061 \text{ fps}$$

h on the water side of the pipe surface

$$= 1078 \text{ (From Fig. 9-1).}$$

For vapors condensing on the outside of horizontal pipes, and taking the physical properties of the condensate at the average film temperature, equation 9-33 becomes

$$h = 0.725 \left[\frac{0.29^3 \times 37.17^2 \times 4.18 \times 10^8 \times 492.3}{0.1979 \times 0.2097 \times 2.42 \times 1.0} \right]^{1/4}$$

$$= 2090 \text{ Btu/(hr)(sq ft)(°F)}$$

The total overall coefficient of heat transfer then becomes

$$U = \frac{1}{2090} + \frac{0.0128}{25.8 \times \frac{2.221}{2.375}} + \frac{1}{h_{\text{water}} \frac{2.067}{2.375}}$$

$$U = \frac{1}{2090} + \frac{0.0128}{24.13} + \frac{1}{938.6} = 482.1 \text{ Btu/(hr)(sq ft)(°F)}$$

The value of U for the shell and tube (multitube, multipass) ammonia condenser was found by Macintire§ to be 460 Btu/hr for the same water velocity.

The area required becomes

$$\Delta = \frac{526.9 \times 60}{482.1 \times 4.624}, \quad \text{where } \Delta t_m = \frac{11.3 - 1.3}{\log_e \frac{11.3}{1.3}} = 4.624^\circ \text{ F}$$

$$= 141.8 \text{ sq ft}$$

9-16. Evaporation. A number of problems are exposed to forced convection on the outside of tubes or pipes, and evaporation of the liquid on the inside of the pipes, similar to the action of a water-tube boiler, or a direct-expansion cooler for the comfort cooling of air.

The Reihel equation for the outside film conductivity becomes

$$\frac{hd}{k_f} = 0.131 \left(\frac{dG_{\max}}{\mu_f} \right)^{0.69} \quad (9-34)$$

where f is the subscript for the film

max is the subscript for maximum

d is the outside diameter in feet

h is the individual coefficient of heat transfer in Btu/(sq ft)(°F)(hr)

μ is the absolute viscosity of the fluid film in pounds per foot-hour

k is the film thermal conductivity in Btu/(sq ft)(°F)(hr/ft)

G is lb/(hr)(sq ft of cross section).

Experimental data for evaporation have been conflicting although some experimenters find that the individual coefficient of heat transfer on the evaporation side of the metallic surface is some power of the difference Δt between the temperature of the wall surface and that of the vapor, and this difference is usually between 1° and 30° F. It has also been shown that the individual coefficient increases rapidly as the boiling temperature is increased.¶ If liquid superheating is not excessive the value of h is usually between 1000 and 2000. Horizontal and vertical pipes or tubes appear to give similar values.

Example. Find the overall value of U , based on the outside surface, for a 4-in. No. 10 B.W.G. boiler tube (0.134-in. wall thickness) with a maximum flue gas velocity of 10 fps. Assume properties of the flue gases to be the same

§ *Univ. Illinois Eng. Exp. Sta. Bull.*, 209, p. 23. The liquefaction surfaces were 2-in. tubes in these tests.

¶ Cryder and Gilliland, "Heat Transmission of Metal Surfaces to Boiling Liquids," *Refrigerating Engineering*, February, 1933.

as for dry air with an average film temperature of 1050° F. Take the value of $h_{\text{for evaporation}} = 2000 \text{ Btu}/(\text{hr})(\text{sq ft})(^\circ\text{F})$; average flue gas temperature = 1717° F; steam temperature = 370° F, and $\Delta t_m = 1347$.

Solution. From equation 9-34,

$$\begin{aligned}
 h &= 0.131 \frac{0.0364}{0.333} \left[\frac{1 \times 10 \times 3600 \times 0.02638}{3 \times 0.091} \right]^{0.69} \\
 &= 3.972 \text{ Btu}/(\text{hr})(\text{sq ft})(^\circ\text{F}) \\
 \frac{1}{U} &= \frac{1}{3.97} + \frac{0.134}{25.8 \times 12 \times \frac{3.866}{4.00}} + \frac{1}{2000 \times \frac{3.722}{4.00}} \\
 U &= 3.95 \text{ Btu}/(\text{hr})(\text{sq ft})(^\circ\text{F})
 \end{aligned}$$

9-17. Approximate Overall Coefficient. Frequently the total overall coefficient of heat transfer U is desired for design purposes, using such a value as will be satisfactory for ordinary operating purposes when the surfaces are kept reasonably clean and the fluid velocities are maintained in the turbulent region. Under these conditions Table 9-17 will be of value, and the methods of calculation to be used may be seen in the following examples.

Example of the Superheat Remover. An ice-making plant of 160 tons of refrigeration capacity makes use of the shell and tube superheat remover. The temperature of discharge is 234° F, and the vapor is cooled to 125° F. Water enters at 70° and is heated during counter flow with the vapor to 150° F. The compressor operates at 200 lb condenser and 35 lb evaporating pressures, in pounds per square inch absolute. Find the required surface.

Solution. The heat to be removed per pound of the refrigerant is, from the Mollier diagram for ammonia (Fig. 8-3), $725 - 655 = 70 \text{ Btu}$. The number of pounds of refrigerant, per hour, is $[200/(613.6 - 150.9)] \times 160 \times 60 = 4152 \text{ lb}$. The amount of heat to be removed per hour is $70 \times 4152 = 290,640 \text{ Btu}$. The weight of water heated per hour is $290640/(150 - 70) = 3633 \text{ lb}$. Taking the average value of U as $20 \text{ Btu}/\text{hr}$ and the log mean difference of temperature as $(84 - 55)/\log_e(84/55) = 68.5$, the required area becomes $290,640 = A \times 20 \times 68.5$, and $A = 212.1 \text{ sq ft}$.

Example. A refrigerating plant of 100 tons refrigerating capacity operates between the limits of 86° F liquefaction and 5° F evaporating temperature and makes use of the old accumulator design of liquid subcooling to 17° F. In this device the condensate from the liquid receiver passes through a coil submerged in the refrigerant under evaporating temperature. Subcooling takes place at condenser pressure. Find the length of $1\frac{1}{4}$ -in. pipe required to do the cooling.

Solution. The heat to be removed is

$$\frac{100 \times 200}{613.3 - 138.9} \times (138.9 - 61.4) = 3267 \text{ Btu}/\text{min}$$

TABLE 9-17
DESIGN VALUES OF *U* FOR METALLIC SURFACES

	Btu/(hr)(ft ²)(°F)
Can ice making piping:	
Old-style feed, non-flooded	12 to 15
Flooded	20 to 30
Ammonia condensers:	
Submerged (obsolete except for CO ₂)	30 to 40
Atmospheric, gas entering at top	60 to 65
Atmospheric, drop or bleeder	125 to 200
Flooded	125 to 150
Shell and tube	150 to 300
Double pipe	150 to 250
Baudalot coolers, counter flow, atmospheric type:	
Milk coolers	75
Cream coolers	60
Oil coolers	10
Water coolers { for direct expansion	60
{ for flooded	80
Brine coolers:	
Shell and tube	90 to 100
Double pipe	150 to 300
Cooling coils:	
Boiling refrigerant to air in unit coolers	4 to 8
Water to air in unit coolers	5 to 9
Brine to unagitated air	2 to 2½
Direct expansion	1½ to 2
Water cooler, shell and coil	15 to 25
Liquid ammonia cooler, shell and coil accumulator	45
Air dehydrator:	
Shell and coil (brine in coil) { 1st coil	5
{ 2d coil	3
Double pipe	6 to 7
Superheat remover, shell and tube	15 to 25

The mean temperature difference is

$$\frac{81 - 12}{\log_e \frac{81}{12}} = 36.1 + ^\circ\text{F}$$

If U (Table 9-17) is taken as 45,

$$\begin{aligned} A &= \frac{60 \times 3267}{45 \times 36.14} = 120.5 \text{ sq ft} \\ &= 277.15 \text{ lin ft} \\ &\quad 1\frac{1}{4}\text{-in. pipe} \end{aligned}$$

The action of forced circulation of the air increases U by an amount varying from $1\frac{1}{2}$ to $2\frac{1}{2}$ times the values for unagitated air, whereas frost decreases the value of U . One inch of frost is considered to decrease the value of U 25 per cent.

9-18. Piping Insulation. The point has already been made that any material, regardless of how high its thermal conductivity may be, provides a resistance to heat transfer and thereby acts as an insulant. One case that appears to be an exception to this rule, and one of very great importance in refrigeration, is represented by a small-diameter pipe to which high conductivity lagging is applied. In such a case, McAdams[¶] has shown, the rate of heat loss may increase with thickness of insulation up to some maximum value beyond which, as the insulation thickness increases, the heat loss gradually drops to below the loss for a bare pipe. This principle is widely utilized in electrical engineering where "lagging" is provided for wire and cable not to prevent heat loss, but to accelerate the rate of loss. Maximum rate of loss can be shown to occur when the outside diameter of the insulated wire or pipe is equal to twice the thermal conductivity of the insulation divided by the combined convection-radiation film coefficient for transfer between the insulation and the surround. This critical value is of interest in that it provides a rapid means of checking the effectiveness of using insulation on a small refrigerant line, but a more important relationship would be one giving the outside diameter for heat loss equal to that of the uninsulated pipe. For this condition to be realized, the resistance of the outside film of the uninsulated pipe would have to be equal to the combined resistance of the insulation and its outside film, or

$$\frac{1}{h_p A_p} = \frac{L}{k A_i} + \frac{1}{h_o A_o} \quad (9-35)$$

where h_p and h_o are the film coefficients on outside of bare pipe and on

[¶] McAdams, *Heat Transmission*, 1st ed., p. 13.

outside of insulation respectively, k is the thermal conductivity of the insulation, and the areas A_p , A_i , A_o refer, respectively, to the outside of the bare pipe, the mean area of the insulation, and the outside surface of the insulation. Thus

$$\begin{aligned} A_p &= \pi L d_p & A_o &= \pi L d_o & A_i &= \frac{A_o - A_p}{\log_e \frac{A_o}{A_p}} \\ & & & & & = \frac{\pi L (d_o - d_p)}{\ln \left(\frac{d_o}{d_p} \right)} \end{aligned} \tag{9.36}$$

and, on substitution of these values into equation 9.35,

$$\frac{1}{h_p d_p} = \frac{(L) \ln \frac{d_o}{d_p}}{(d_o - d_p)k + \frac{1}{h_o d_o}}$$

But

$$L = \frac{d_o - d_p}{2}$$

so

$$\frac{1}{h_p d_p} = \frac{\frac{1}{2} \ln \frac{d_o}{d_p}}{k + \frac{1}{h_o d_o}} \tag{9.37}$$

which is of the form $d_o = \phi(d_p, h_o, h_p, k)$; in many cases the values of h_p and h_o do not vary greatly, and for an insulated pipe in room air (without forced circulation past the pipe) the value of h_o can be taken as approximately constant. Thus the equation is reduced to a form for which a graphical solution can readily be constructed.

PROBLEMS

1. For 1° F temperature difference from surface to passing air a marble wall loses 0.88 Btu/(hr)(sq ft) by convection. For the same wind and temperature conditions what would be the convective loss to air from an unfinished concrete wall?

2. Estimate the total losses—radiation as well as convection—from each of the walls of Problem 1.

3. Liquid sulphur dioxide at 150° F and at a velocity of 2 fps flows outside a pipe of ¾ in. outside diameter. The liquid sulphur dioxide within the pipe is at 125° F and flows with a velocity of 5.5 fps; inside diameter of the pipe is ½ in. Determine the inside and outside film coefficients.

4. Liquid sulphur dioxide is cooling while flowing outside and parallel to tubes of a shell-and-tube exchanger having a 7-in. shell and twenty 1-in. tubes. Fluid temperature is 75° F and velocity is 3 fps. Find the film coefficient.

5. Dichlorodifluoromethane at a temperature of 75° F is flowing at 8 fps in a 2-in. nominal diameter type M copper tube. The liquid to which the tube is losing heat is also dichlorodifluoromethane, but at a temperature of 0° F and flowing normal to the tube at a velocity of 6 fps. (Inside and outside diameters of 2-in. type M tube are 2.009 in. and 2.125 in. respectively.) Determine the rate of heat transfer per lineal foot of tube.

6. A Freon-11 heat exchanger consists of a bundle of 3-in. nominal diameter extra-heavy wrought iron staggered tubes with liquid flowing in the tubes at a velocity of 3 fps and cooling from an average temperature of 50° F. Outside, and flowing normal to the tubes is liquid Freon-11 at an average temperature of 35° F and with a velocity 4 fps through the narrowest cross section between the tubes. Determine the rate of heat transfer per lineal foot of tube and compare it with the heat transfer rate which would exist in an exchanger having the tubes in line and with the same spacing between tubes at the narrowest section. The inside and outside diameters of 3-in. extra-heavy wrought iron pipe are 2.9 in. and 3.5 in. respectively.

7. A refrigerant at 100° F is flowing in a 2-in. standard wrought iron pipe under conditions such that the inside film coefficient is 200. Cooling water at 50° F is outside the pipe and flowing under conditions which give an outside film coefficient of 800. Determine the length of pipe needed to permit a total cooling rate of 200,000 Btu/hr.

8. Heat is flowing through 1 sq ft of pipe surface which has a negligible wall resistance and a negligible wall thickness; diameter of the pipe is 1½ in. Liquid Freon-21 is flowing within the pipe at a rate of 5½ fps and is losing heat from an average temperature of 100° F to liquid Freon-21 which is flowing normal to the outside of the pipe with a velocity of 4 fps and an average temperature of 75° F. Determine the rate of heat transfer between fluids per square foot of surface.

9. Freon-22, in liquid form, is flowing outside and normal to a single pipe at a temperature of 0° F and with a velocity of 6½ fps. The pipe has an inside diameter of 3.00 in. and an outside diameter of 3.25 in. Within the pipe Freon-22 is flowing at a mean temperature of 78° F and with a velocity of 3 fps. Neglecting the thermal resistance of the pipe wall and taking both inside and outside fluid temperatures as mean values, determine the necessary length of pipe to achieve a transfer of energy to the fluid outside the pipe at a rate of 10,000 Btu/hr.

10. A refrigeration compressor discharges ammonia vapor at a pressure corresponding to a saturation temperature of 79° F and with 71° F of superheat. The pipe has an inside diameter of 1¼ in., and flow occurs at a velocity of 43.2 fps. Determine the vapor conditions at the point of compressor discharge and evaluate the inside film coefficient.

11. Superheated sulphur dioxide vapor at 70 psia and 150° F (specific volume of 1.389 cu ft/lb) is cooling as it flows through a very thin-walled copper tube having inside diameter of 1 in.; the velocity of the vapor is 32 fps. A stream of superheated sulphur dioxide at 60 psia and 100° F (specific volume of 1.288 cu ft/lb) is flowing outside and normal to the same tube at a velocity of 29.5 fps. Neglecting the thermal resistance of the tube wall and taking the outside tube diameter as equal to the inside diameter, calculate the rate of heat transfer per lineal foot of tube.

12. Carbon dioxide vapor is being cooled, at a mean temperature of 50° F, within

a thin-walled tube for which both inside and outside diameters can be taken as approximately 0.5 in. The flow rate of the desuperheating vapor is 32 lb/(sec)(sq ft of tube cross section). Cooling effect is provided by a stream of superheating carbon dioxide, at 0° F, which flows normally across the transfer tube at a flow rate of 7 lb/(sec)(sq ft). Determine the inside and outside film coefficients and evaluate the overall coefficient of heat transfer.

13. A $\frac{3}{8}$ -in. outside diameter copper tube is to have its wall thickness increased to an amount such that the rate of heat loss from the thickened tube will equal the rate of heat loss from the outside of the original $\frac{1}{4}$ -in. tube. Determine the outside diameter at which this condition would be attained.

14. A Freon-12 evaporator is designed for 2-in. steel pipe. The water to be cooled enters at 46° and leaves at 40° F with an average velocity of 5.8 fps. Take $h_{\text{evaporation}}$ of 500 Btu/(hr)(sq ft)(°F). Find the overall coefficient of heat transfer (U). If the Freon-12 boils at 30° F find the heat transfer per square foot per hour.

CHAPTER X

STEADY-STATE LOAD CALCULATIONS; INSULATION

With the development of comfort cooling, the refrigerating engineer has been confronted with problems arising out of losses due to infiltration and heat leakage, problems increased to an extent not dreamed of previous to 1920, and in order to handle this kind of work he has been forced to study thoroughly the matter of ordinary building construction. In addition, many problems arise that apply to a specific base or are of a temporary nature, as where the load is a short one, and where the installation of costly construction is not justified. Refrigeration, in addition to the live load, is now required to overcome losses resulting from such a building construction as permits heat leakage, infiltration, and solar radiation.

10·1. Calculation of Heat Losses. Heat leakage through the walls, ceilings, floors, and windows may be found from the equation

$$q = AU\Delta t \quad \text{Btu/hr} \quad (10\cdot1)$$

where q is the heat entering the room or the building through the materials of the walls, ceilings, floors, windows, etc., A is the area of the walls, etc., in square feet; U is the overall coefficient of heat transfer in Btu per square foot per degree F difference in temperature per hour; Δt is the average difference of temperature between the air on the two sides of the transfer surface, in degrees F.

The overall coefficient of heat transfer U for a built-up section may be calculated as follows:

$$U = \frac{1}{\frac{1}{h_1} + \frac{1}{h_2} + \frac{L_1}{k_1} + \frac{L_2}{k_2} + \frac{L_3}{k_3}, \text{ etc.}} \quad (10\cdot2)$$

where h_1 and h_2 are film coefficients such as are found in all solid construction in contact with a fluid, L is the thickness of the material, k is the thermal conductivity, and L/k may be replaced by $1/C$, where C is the conductance, Btu/(hr) (sq ft) ($^{\circ}$ F), of the section in question. The value for h_1 in still air is 1.65, and h_2 for 15-mph wind velocity is 6. The value for h does not vary appreciably with different materials. In fact, with insulated walls the effect of the surface resistance is hardly appreciable and is frequently neglected.

Example. Find the heat leakage per 100 sq ft of wall made of 8-in. brick, $\frac{1}{2}$ -in. cement mortar, 2-in. corkboard, and $\frac{1}{8}$ -in. finish. Take the value of k for the finish as 0.35 Btu/in. thick and obtain conductivities of other materials from Table 10·1, 10·2, or 10·3.

Solution.

$$U = \frac{1}{\frac{1}{6.0} + \frac{1}{1.65} + \frac{8}{5.0} + \frac{2}{0.3} + \frac{0.125}{0.35} + \frac{0.5}{12}} = \frac{1}{9.44} = 0.106 \text{ Btu/(hr)(sq ft)}$$

$$= 10.6 \text{ Btu/(hr)(100 sq ft)}$$

10·2. Infiltration. Infiltration due to wind pressure takes place through all walls, doors, and windows; for example, with a 15-mph wind, $7\frac{1}{2}$ cfm for an $8\frac{1}{2}$ -in. plain brick wall, and 0.13 cfm for a frame wall with lath and plaster, will enter per square foot. According to George* tests for various materials under an air pressure of 40 psf indicated an air infiltration equal to the values given in Table 10·5. However, with good sealing, the amount of infiltration through the walls is negligible. For windows the amount of crack must be *at least* half of the total crackage of the outside wall from any room.

The amount of infiltration under average conditions amounts to *one-half to one complete volume* change per hour.

10·3. Reflective Insulation. According to Gregg† the true conductivity of air is less than that of any commercial insulating material, but radiation and convection are important additional factors. Air spaces $\frac{1}{2}$ in. thick or less separated by paper reduce convection, but radiation is still a large factor. If paper is replaced by bright metal the radiation factor is reduced. All bright, clean, or polished surfaces have low emissivity values (see Chapter XIII), but most metals tarnish, so that value of the emissivity becomes high. However, aluminum appears to be the exception. It does form an oxide coating, but this does not seem to affect the emissivity value. Aluminum foil can be made very thin, in fact as thin as 0.00025 in. Tests by Gregg gave values for conductance as shown in Table 10·6.

10·4. Approximate Window Load. Solar radiation has been found to be a very large factor in many of the applications of comfort cooling. Approximate values for solar radiation through glass in Detroit‡ were found to be 150 Btu/(sq ft)(hr). During tests on two experimental rooms in Pittsburgh, Houghten§ found that the average of a number of

* George, *Refrig. Eng.*, March, 1932.

† Gregg, *Refrigerating Eng.*, May, 1932.

‡ Walker, Sanford, and Wells, "Field Studies of Office Building Cooling," *Heating, Piping, Air Conditioning*, January, 1932.

§ Houghten, Gutberlet, and Blackshaw, "Studies of Solar Radiation," *Heating, Piping, Air Conditioning*, January, 1934.

TABLE 10·1a
THERMAL CONDUCTIVITY OF MATERIALS

(From tests by the Bureau of Standards, materials submitted by manufacturers)

w = weight in pounds per cubic foot

k = thermal conductivity in Btu per hour, square foot, and temperature gradient of 1° F per inch thickness. (The lower the conductivity the greater the insulating values.)

	w	C or k
Soft Flexible Materials in Sheet Form		
Dry Zero Kapok between burlap or paper	1.0	0.24
Dry Zero Kapok between burlap or paper	2.0	0.25
Cabots quilt eel grass between kraft paper	3.4	0.25
Cabots quilt eel grass between kraft paper	4.6	0.26
Hair felt felted cattle hair	11.0	0.26
Hair felt felted cattle hair	13.0	0.26
Balsam wool chemically treated wood fiber	2.2	0.27
Hairinsul 75% jute hair 25%	6.3	0.27
Hairinsul 50% hair 50% jute	6.1	0.26
Linofelt flax fibers between paper	4.9	0.28
Thermofelt jute and asbestos fibers, felted	10.0	0.37
Thermofelt hair and asbestos fibers, felted	7.8	0.28
Loose Materials		
Rock wool, fibrous material made from rock, also made in sheet form, felted and confined with wire netting	6.0	0.26
Rock wool	10.0	0.27
Rock wool	14.0	0.28
Rock wool	18.0	0.29
Glass wool Pyrex glass, curled	4.0	0.29
Glass wool Pyrex glass, curled	10.0	0.29
Sil-O-Cel powdered diatomaceous earth	10.6	0.31
Regranulated cork, fine particles	9.4	0.30
Regranulated cork, about 3.16-in. particles	8.1	0.31
Thermofil gypsum in powdered form	26.0	0.52
Thermofil gypsum in powdered form	34.0	0.60
Sawdust various	12.0	0.41
Sawdust redwood	10.9	0.42
Shavings various, from planer	8.8	0.41
Charcoal from maple, beech and birch, coarse	13.2	0.36
Charcoal from maple, beech and birch, 6 mesh	15.2	0.37
Charcoal from maple, beech and birch, 20 mesh	19.2	0.39
Semi-flexible Materials in Sheet Form		
Flaxlinum flax fiber	13.0	0.31
Fibrofelt flax and rye fiber	13.6	0.32

TABLE 10-1a (Continued)

	<i>w</i>	<i>C</i> or <i>k</i>
Semi-rigid Materials in Board Form		
Corkboard, no added binder; very low density	5.4	0.25
Corkboard, no added binder; low density	7.0	0.27
Corkboard, no added binder; medium density	10.6	0.30
Corkboard, no added binder; high density	14.0	0.34
Eureka Corkboard with asphaltic binder	14.5	0.32
Rock cork rock wool block with binder, also called "Tucork"	16.7	0.37
Lith board containing rock wool, flax and straw pulp	14.3	0.40
Stiff Fibrous Materials in Sheet Form		
Insulite wood pulp	16.2	0.34
Insulite wood pulp	16.9	0.34
Celotex sugar cane fiber	13.2	0.34
Celotex sugar cane fiber	14.8	0.34
Cellular Gypsum		
Insulex or Pyrocell	8	0.35
Insulex or Pyrocell	12	0.44
Insulex or Pyrocell	18	0.59
Insulex or Pyrocell	24	0.77
Insulex or Pyrocell	30	1.00
Woods (across grain)		
Balsa	7.3	0.33
Balsa	8.8	0.38
Balsa	20	0.58
Cypress	29	0.67
White pine	32	0.78
Mahogany	34	0.90
Virginia pine	34	0.98
Oak	38	1.02
Maple	44	1.10
Miscellaneous Building Materials (Data taken from various sources)		
Cinder concrete	2 to 3	<i>k</i>
Building gypsum	About 3	
Plaster	2 to 5	
Building brick	3 to 6	
Glass	5 to 6	
Limestone	4 to 9	
Concrete	6 to 9	
Sandstone	8 to 16	
Marble	14 to 20	
Granite	13 to 28	

TABLE 10-1b

CONDUCTANCE AND INSULATING VALUE OF SHEET MATERIALS IN THICKNESS AS SOLD

 w = weight in pounds per square foot L = thickness in inches C = conductance in Btu per hour, per square foot, and per degree F $R = 1/C$ = resistance or insulating value

	w	L	C	R
Soft Flexible Materials				
Cabots quilt single ply	0.14	0.35	0.72	1.39
Cabots quilt double ply	0.18	0.48	0.54	1.85
Cabots quilt triple ply	0.31	0.67	0.39	2.56
Balsam wool $\frac{1}{2}$ -in. house insulation; smooth paper	0.16	0.55	0.48	2.10
Balsam wool $\frac{1}{2}$ -in. refrigerator insulation, creped paper	0.24	0.66	0.41	2.47
Balsam wool 1-in. refrigerator insulation, creped paper	0.32	1.13	0.25	4.08
Hairinsul 75% hair 25% jute	0.46	0.55	0.49	2.05
Hairinsul 50% hair 50% jute	0.42	0.51	0.51	1.96
Carinsul hairfelt between asbestos paper	0.58	0.60	0.46	2.19
Salamander hairfelt paper, asbestos, and cheesecloth; paper between plies 2 ply	0.54	0.61	0.42	2.40
Salamander hairfelt paper, asbestos, and cheesecloth; paper between plies 3 ply	0.69	0.70	0.36	2.75
Thermofelt jute and asbestos	0.42	0.51	0.72	1.39
Thermofelt hair and asbestos	0.42	0.63	0.45	2.22
Nycinsul hair felt between cheesecloth, the latter treated with magnesite solution	0.97	0.45	0.82	1.21
Linofelt $\frac{1}{2}$ in.	0.41	0.67	0.42	2.40
Resisto similar of Nycinsul, single	0.56	0.40	0.75	1.34
Resisto similar of Nycinsul, double	0.77	0.62	0.49	2.05
Semi-flexible materials				
Flaxlinum	0.61	0.56	0.56	1.80
Fibrofelt	0.66	0.58	0.56	1.80
Stiff Fibrous Materials				
Insulite wall board	0.66	0.49	0.69	1.46
Insulite insulation board	0.80	0.56	0.60	1.67
Celotex building board	0.58	0.47	0.72	1.38
Celotex railroad insulation board	0.64	0.58	0.59	1.71
Plaster and Wall Boards				
Gyplap gypsum between layers of heavy paper	2.23	0.50	2.6	0.38
Sheet rock gypsum mixed with sawdust between layers of heavy paper	1.97	0.39	3.6	0.27

TABLE 10-2

CONDUCTIVITIES AND CONDUCTANCES OF BUILDING MATERIAL
(Values from various sources)

Btu per hr per °F per ft² per in. thick or for the thickness stated

Material	Coefficient (<i>k</i> or <i>C</i>)	Material	Coefficient (<i>k</i> or <i>C</i>)
Brick, common.....	5.00	Plaster board, $\frac{3}{8}$ in.....	3.73
Brick, face.....	9.2	Plaster board, $\frac{1}{2}$ in.....	2.82
Cement mortar.....	12.00	Roofing:	
Cinder concrete.....	5.20	asbestos shingles.....	6.00
Cinder blocks, 8 in.....	0.62	composition roofing.....	6.50
12 in.....	0.51	built-up, $\frac{3}{8}$ in.....	3.53
Concrete blocks, 8 in.....	1.0	slate shingles.....	10.37
12 in.....	0.80	Stone.....	12.50
Concrete, cinder.....	2.4	Stucco.....	12.00
Concrete, stone.....	6.5	Tile or terrazzo.....	12.00
Gypsum fiber concrete.....	1.66	Wood lath and plaster.....	2.50
Hollow clay tile, 4 in.....	1.00	Woods	
6 in.....	0.64	1-in. fir sheathing, build-	
8 in.....	0.60	ing paper and yellow	
10 in.....	0.58	pine lap siding.....	0.50
12 in.....	0.40	1-in. fir sheathing and	
16 in.....	0.31	building paper.....	0.82
Hollow gypsum tile, 4 in....	0.46	yellow pine lap siding...	1.28
Insulations:		yellow pine or fir.....	0.80
corkboard.....	0.30	maple or oak.....	1.15
flexible.....	0.27	shingles, wood.....	1.28
flaked gypsum, 24 lb....	0.48	Air spaces.....	1.10
rigid insulation.....	0.33	Surfaces, still air.....	1.65
rock wool.....	0.30	15 mph.....	6.00
Plaster, gypsum.....	3.3		

tests from 9 A.M. until the sun became ineffective indicated that the east and west exposures gave values of the rate entering through windows of about 70 Btu/(sq ft)(hr), with about one-third of this amount for average outside shades and two-thirds for inside shades, whereas awnings reduced the amount to 20 Btu, and when the window was completely protected from direct sunlight the heat gain was reduced to 4 Btu/(sq ft)(hr).

10-5. Air Spaces. The value for the conductances of air spaces does not change very much with the thickness and temperature,|| and it is shown in Table 10-7.

|| Rowley and Algren, *Trans. ASHVE*, Vol. 35, 1929.

TABLE 10-3

	Thickness, inches	Air by Infiltration, cu ft/sq ft/hr
Corkboard	1.513	432
Beaver board	0.182	0.593
Upsom board	0.196	3.64
Asphalt	one coat	0.0

TABLE 10-4

INFILTRATION IN CUBIC FEET PER HOUR PER FOOT OF CRACK AT 10-MPH WIND VELOCITY

Double-hung wood sash windows (unlocked)	Around frame in masonry wall, not calked	8.2 cu ft
	Around frame in masonry wall, calked	1.5
	Around frame in wood frame construction	6.2
	Total for average window, not weatherstripped, 1/16-in. crack and 3/32-in. clearance, including wood frame leakage	21.4 cu ft
	Same as above, but weatherstripped	15.5
	Total for poorly fitted window, non-weatherstripped, 3/32-in. crack and 3/32-in. clearance, including wood frame leakage	69.0
	Same as the above, but weatherstripped	18.9

TABLE 10-5

INFILTRATION IN CUBIC FEET PER HOUR PER SQUARE FOOT OF WALL

	10 mph	15 mph
8 1/2-in. brick wall, plain	4.20	7.85
plaster	0.037	0.066
13-in. brick wall, plain	3.92	7.48
plaster	0.013	0.025
Frame wall, lath and plaster, painted, sheathing, build- ing paper	0.07	0.13
Frame wall, shingles, paper, sheathing, 3 coats gypsum	0.05	0.10

TABLE 10-6

	Density, pcf	Conduct- ance
Aluminum foil mounted on 16-point cardboard 4-0.5 in. square Masonite strips 24 in. long and 8 in. apart for separators, in air spaces.....	5.1	0.26
Aluminum foil mounted on 138-lb Kraft paper, 8 strips of yellow poplar wood $\frac{1}{2}$ in. high, $\frac{1}{16}$ in. thick and 20 in. long, notched and crossed to form nine $6\frac{3}{8}$ -in. square air spaces.....	3.0	0.26

TABLE 10-7

Tempera- ture, °F	Thickness Air Space, inches				
	0.128	0.250	0.364	0.493	0.713
	Conductances in Btu				
20.....	2.30	1.37	1.18	1.10	1.04
40.....	2.47	1.48	1.29	1.19	1.13
60.....	2.65	1.59	1.39	1.30	1.21
80.....	2.82	1.70	1.49	1.39	1.30
100.....	2.99	1.81	1.60	1.49	1.38

10-6. Climatic Conditions in the United States. In the calculations of the heat gained by heat leakage it is desirable to know what the sustained temperatures are. In Fig. 10-1 the *dotted* lines are the average *daily peak* dry-bulb temperatures for the hottest week of the year over a period of 46 years. The average temperatures for both day and night are found by subtracting the small numerical values on the chart from the estimated dry-bulb temperatures, as indicated by the dotted lines. The solid black lines are the average relative humidity for the hottest month taken at about 8 P.M. for the entire month.

LOAD FACTORS INVOLVED IN AIR CONDITIONING AND COMFORT COOLING

In comfort cooling and air conditioning problems the sources of the load that must be carried by the refrigerating equipment are as follows:

- (a) The sensible heat conducted through the walls, windows, ceilings, and roof.
- (b) The radiant heat received through or by the walls, roof, and windows from the sun.

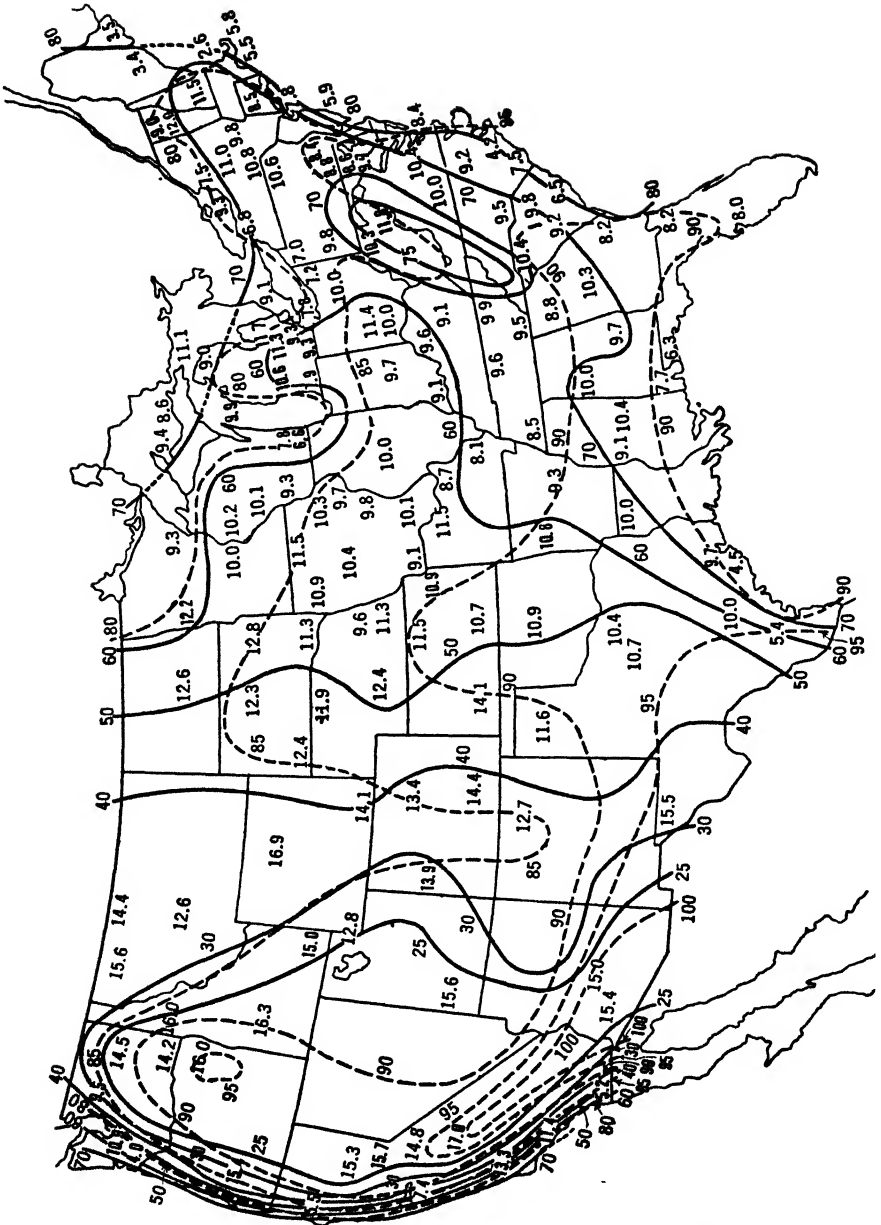


FIG. 10-1. Temperature and humidity chart for the United States. Dotted lines are average daily peak dry-bulb temperature for the hottest week, average for 46 years. For the average temperature for day and night subtract the numbers from the nearest dotted line. The solid black lines are the average relative humidity for hottest month taken at about 8 P.M. for the entire month.

(c) Sensible and latent heat brought into the building by uncontrolled infiltration or on account of controlled fresh air.

(d) Sensible and latent heat liberated by the people present.

(e) Heat generated by various machines and appliances, such as electric lights, machines, and coal, gas, and electric heating appliances.

10-7. Transmission. When an enclosure is subjected to manufactured weather conditions there is a temperature differential on the two sides of the outside walls, or on the two sides of the ceiling or floor. The amount of heat conducted is directly proportional to the area in square feet, the overall coefficient of heat transfer, and the temperature difference in Fahrenheit degrees, and is expressed by the formula

$$q_c = A_w U_w (t_o - t_i) \quad (10\cdot3)$$

where q_c is the heat conducted through the walls, windows, ceilings, or partitions in Btu per hour; A_w is the area of the walls, windows, etc., in square feet; U_w is the overall coefficient of heat transfer in Btu per square foot per Fahrenheit degree difference in temperature per hour; t_o is the design outside temperature in Fahrenheit degrees; t_i is the design inside temperature in Fahrenheit degrees. The value of the design temperature difference ($t_o - t_i$) needs to be chosen with care. The outside air temperature at the time of the maximum cooling load should be selected so that the maximum temperature over a period of years will be included during about 90 per cent of the cooling season. With partitions, where one side is not conditioned, it is possible that the temperature t_o may be considerably greater than that of the atmospheric air, but as a rule it may be 3° to 5° F less than the temperature of the outside air.

In addition to conduction, solar radiation must be taken into consideration whenever the sun's rays actually strike the wall or roof surface. Surfaces subject to solar radiation will absorb or reflect more or less of the radiant energy, depending on the type of surface, as all surfaces are to a degree radiators. Of the solar radiation absorbed by walls or roofs a certain fraction will enter interiors, depending on the overall coefficient of heat transfer.

10-8. Solar Radiation. The amount of heat q_r , absorbed by a wall actually exposed to the sun may be expressed by

$$q_r = A_w F e I \quad \text{Btu/hr} \quad (10\cdot4)$$

where A_w is the area of the wall surface in square feet; e is the decimal part of the incident solar radiation that is absorbed by the wall surface; I is the actual solar radiation striking the wall, allowing for the direction of the rays of the sun; F is the decimal part of the absorbed solar

TABLE 10-8 (Continued)

North Latitude, Degrees	Direction Wall Faces										North Latitude, Degrees	Direction Wall Faces						Horizontal Surface					
	Time of Day	Coef.	NE	E	SE	S	SW	W	NW	Horizontal Surface		Time of Day	Coef.	NE	E	SE	S		SW	W	NW	Horizontal Surface	
35	I, for walls and roofs	6 A.M.	135	145	70	70					30	40	6 A.M.	130	140	65	65				20		
		7	160	200	120	100					85		7	145	185	105	105					70	
		8	150	210	145	135					150		150	8	120	190	135	20					180
		9	110	135	150	30					205		250	10	70	165	140	45					220
		10	60	70	100	55					280		280	11	15	110	125	65					250
		11				75					290		290	Noon		40	40	65	40				255
		Noon				70					280		280	1 P.M.		90	90	65	40	40			250
		1 P.M.				55					260		260	2		125	110	45	125	110	15		220
		2				55					205		205	3		140	165	20	140	165	70		180
		3				30					160		160	4		135	190	20	135	190	120		130
		4				145					85		85	5		105	185	105	105	185	145		70
5				120					200	200	6		165	145	165	165	145	145		20			
6				70					145	135	30		165	140	165	165	140	130		20			
35	I, for walls and roofs	6 A.M.	120	135	55	40					15	45	5 A.M.	70	70	25	25				40		
		7	145	185	100	70					65		6	145	165	85						90	
		8	130	190	125	40					130		130	7	155	205	130	25					145
		9	85	165	125	25					185		185	8	130	210	165	65					195
		10	30	110	105	25					230		230	9	85	180	175	65					230
		11				45					255		255	10	25	135	160	95					260
		Noon				40					265		265	11		70	130	115	30				265
		1 P.M.				70					255		255	Noon		90	125	90	115	30			260
		2				40					230		230	1 P.M.		95	160	135	130	70	25		230
		3				125					185		185	2		175	180	65	175	180	85	25	195
		4				125					190		190	3		165	210	25	165	210	130		145
5				100					185	145	5		130	205	130	130	205	155		95			
6				70					135	120	15		85	165	85	85	165	145		40			
40	I, for walls and roofs	6 A.M.	140	155	80	70					35	40	5 A.M.	65	65	20	20				25		
		7	160	200	125	100					85		6	135	160	70						70	
		8	140	210	155	10					150		150	7	140	180	115						25
		9	100	185	160	45					200		200	8	110	180	135	35					125
		10	40	135	150	75					240		240	9	55	160	140						175
		11				95					270		270	10		140	160	85					210
		Noon				100					280		280	11		40	105	85	10				245
		1 P.M.				70					270		270	Noon		40	55	55	55				240
		2				15					240		240	1 P.M.		10	10	85	105	40	40		245
		3				75					150		135	2		65	140	65	140	110	55		215
		4				45					160		185	3		35	145	150	165	110	125		175
5				10					150	210	4		115	190	110	145	190	140		240			
6				125					200	160	6		115	190	140	115	190	140		240			
				80					155	140	35		170	150	20	170	150	65		25			

TABLE 10·9

Surface Material	Absorption Material	(ϵ)
Very light-colored surfaces	White stone	0.4
	Light-colored cement	
	Light cream or white paint	
Medium dark-colored surfaces	Asbestos shingles	0.7
	Unpainted wood	
	Brown stone	
	Brick or red tile	
	Dark-colored cement	
	Stucco Red, green and gray paint	
Very dark-colored surfaces	Slate roofing	0.9
	Tar roofing material	
	Dark paints	

radiation that is transferred into the inside of the wall. It is approximated by the expression $F = 0.23U_w$.

The amount of heat entering the enclosure, in Btu per hour, is q_r . The value of ϵ depends on the kind of surface material, and it is given in Table 10·9. ¶

At times an approximation of the effect of solar radiation may be obtained by adding 25° F to the outside design temperature for windows and walls subject to direct radiation. As this rule neglects the radiant energy entering, the window area exposed to solar radiation would need to be relatively small.

10·9. Sun Effect through Windows. Transparent windows permit a large percentage of solar radiation to pass through the glass undiminished, approximately 90 per cent when the rays are perpendicular to the glass. However, when the angle of incidence is less than 90° the amount of energy reflected is increased. The heat gain through those windows exposed to the sun may be expressed by

$$q_s = A_g R_g \quad (10·5)$$

where q_s is the amount of solar heat transmitted directly through the glass in Btu per hour, A_g is the area of the glass in square feet, R_g is the amount of solar heat transmitted directly through the glass in Btu per square foot per hour. Values for R_g are given in Table 10·8.

Certain special kinds of glass* have been developed which are both

¶ Faust, Levine, and Urban, *Jour. ASHVE*, August, 1935.

* Shaver, *Heating, Piping, Air Conditioning*, September, 1935.

more heat-absorptive and heat-resistant as regards solar energy, thus reducing the values for R_g in the table as well as the refrigerating load (for direct solar radiation through the glass) by 16 to 24 per cent.

Awnings on the first floor of buildings may be assumed to eliminate 75 per cent, and those on the second and succeeding floors 85 per cent, of the solar radiation[†] that would otherwise pass through the bare window. Inside Venetian blinds with metallic aluminum surfaces, and buff-colored inside shades clean and completely drawn, will eliminate about 50 per cent of the solar radiation that would otherwise pass through the window, but dark shades are not nearly so effective. Results showed also that 5 per cent of the solar radiation that would be expected to pass through a bare window was transferred through a completely shaded window, 22 per cent through an outside Venetian blind, 28 per cent through an awning, 45 per cent through an inside shade, and 58 per cent through an inside Venetian blind. Research indicates that, in general, protection from solar radiation is best obtained by means of some device *outside* the window.

Radiations from sources of heat at temperatures below 500° F are not transmitted by ordinary double-strength window glass,[‡] and only 6 per cent is transmitted at temperatures of 700° F. Window glass therefore constitutes a heat trap permitting solar radiation to enter but not to be reradiated.

10·10. Leakage. Infiltration is a factor so variable that no general rules may be given for its calculation, but the use of storm windows, weather strips, sealing of the window frames with a caulking compound, and tight wall construction all decrease the amount of air entering the building because of wind pressure. Table 10·10a[§] gives a rough approximation of the number of changes of the entire volume of the room that may be expected under conditions of a moderate wind pressure.

If a determination can be made of the air entering from the atmosphere, either that permitted for ventilation or that caused by infiltration or the sum of these two, the sensible and latent heat load on the refrigeration plant may be found by the methods of Chapter XIV. Use of the psychrometric chart will show the difference in the heat contents for the initial and final wet-bulb temperatures, or calculation employing Table 14·1 will give these values for the outside air conditions and the air leaving the spray chamber of the cooling coils after dehumidification.

[†] Houghten, Gutberlet, and Blackshaw, *Jour. ASHVE*, February, 1934.

[‡] Miller and Black, *Heating, Piping, Air Conditioning, Journal Section*, February, 1932.

[§] *ASHVE Guide*, 11th ed.

TABLE 10·10a

AIR CHANGES TAKING PLACE UNDER AVERAGE CONDITIONS, EXCLUSIVE OF AIR PROVIDED FOR VENTILATION

Kind of Room or Building	Number of Air Changes per Hour	
	For Winter	For Summer
Rooms—1 side exposed	1	$\frac{1}{2}$
2 sides exposed	$1\frac{1}{2}$	$\frac{3}{4}$
3 sides exposed	2	1
4 sides exposed	2	1
Rooms with no windows or outside doors	$\frac{1}{2}$ to $\frac{3}{4}$	$\frac{1}{4}$ to $\frac{3}{8}$
Entrance halls	2 to 3	2 to 3
Reception halls	2	1 to 2
Living rooms	1 to 2	
Dining rooms	1 to 2	
Bath rooms	2	2
Drugstores	2 to 3	1 to 2
Clothing stores	1	$\frac{1}{2}$ to 1
Factories, lofts, etc.	$\frac{1}{2}$ to 3	$\frac{1}{2}$ to $1\frac{1}{2}$

10·11. The Sensible and Latent Heats of the People Present.

The heat dissipated by the average person q_h , assumed to have an outside body area of 19.5 sq ft, is practically constant at 400 Btu/hr over a range from 65° to 90° F effective temperature. At normal conditions of temperature and humidity the heat dissipated[¶] is given approximately by

	Per Cent
Radiation, conduction and convection	73
Evaporation of moisture from the skin	14.5
Evaporation of moisture from the lungs	7.2
Warming of inspired air	3.5
Balance	1.8

The sensible heat from the human body, as given by Houghten* for still air and for velocities of the air of 200 fpm, is shown in Table 10·11a.

These figures are given per hour for the person of average size. In addition to these values for conditions at rest, it should be noted that at a medium rate of exertion, as for people who are dancing or for restaurant

¶ Refer to Section 14·11, Fig. 14·3.

* Yaglou *et al.*, "Changes in Ionic Content," *Trans. ASHVE*, Vol. 37, 1931.

* Houghten, "Heat and Moisture Losses from the Human Body," *Trans. ASHVE*, Vol. 35, 1929.

TABLE 10·10b

COOLING LOAD ENTRANCE INFILTRATION LOSSES FOR DOORS IN OPERATION¹

Establishment	Infiltration, CFM per Person in Room					
	72 In. Revolving Door		36 In. Hinged Door ² Single Entrance ³		36 In. Hinged Door ² Cross Draft Entrance ⁴	
	Maximum Small Occupancy	Minimum Large Occupancy	Maximum ⁵	Minimum ⁶	Maximum ⁵	Minimum ⁶
Bank	4.7	2.0	22.0	7.0	30.5	13.4
Barber shop	2.3	2.3	11.0	3.5	15.2	6.7
Broker's office	4.7	3.3	22.0	7.0	30.5	13.4
Candy and soda	3.5	2.0	16.5	5.2	22.7	10.0
Cigar store	14.5	8.0	66.0	20.8	91.0	40.0
Dept. store (small)	6.4	3.0	33.0	10.4	45.5	20.0
Dept. store (large)	4.7	2.0	22.0	7.0	30.5	13.4
Dress shop	1.7	1.6	8.2	2.6	11.4	5.0
Drug store	4.7	3.3	22.0	7.0	30.5	13.4
Furrier	1.7	1.6	8.2	2.6	11.4	5.0
Hospital room	3.5	3.5
Lunch room	3.5	2.0	16.5	5.2	22.7	10.0
Men's shop	2.3	2.1	11.0	3.5	15.2	6.7
Office (private)	2.5	2.5
Office (professional)	3.5	3.5
Office building	1.2	0.5	5.5	1.7	7.7	3.4
Public building	1.7	0.7	8.2	2.6	11.4	5.0
Restaurant	1.7	0.7	8.2	2.6	11.4	5.0
Shoe store	4.7	2.1	11.0	3.5	15.2	6.7

¹ Two doors each 5 by 7 ft open continuously in different outside walls, infiltration loss, maximum 12,500 cfm, minimum 5200 cfm, average 9000 cfm.

One door 5 by 7 ft open continuously in one outside wall only, infiltration loss, maximum 8500 cfm, minimum 3000 cfm, average 5800 cfm. For floor areas of 2500 ft² or less these values may be reduced 50 per cent.

² For vestibule (double bank hinged doors) entrance infiltration is 25 per cent less than value for single bank hinged door.

³ Door in one outside wall only.

⁴ More than one door in different walls.

⁵ Maximum values for exposed location with average wind velocity above 10 mph.

⁶ Minimum values for sheltered location with average wind velocity below 5 mph.

or hotel waiters, the body heat is increased to about 230 Btu of sensible heat and 420 Btu/hr of latent heat per person. See also Table 10·11b.

10·12. Heat Generated by Various Machines and Appliances.

ILLUMINATION. From 50 to 90 per cent† of the total energy consumed by a lighting system goes into heat and becomes a burden on the cooling system. A large part of this heat could be removed by a system of separate ventilation for each fixture, but it is questionable whether the expense and complexity would ever be justified.

† Sturrock and Walker, *Trans. Illum. Eng. Soc. (N.Y.)*, August, 1933.

TABLE 10-11a
HEAT DISSIPATED BY THE AVERAGE PERSON

Dry-bulb Temperature, °F	Sensible Heat from Human Body, Btu per hr			
	For Still Air		For Air of 200 fpm	
60	375		425	
65	340		390	
70	300		340	
75	260		300	
80	215		250	
85	160		200	
90	105		125	
	Latent Heat of Human Body			
	For Still Air		For Air of 200 fpm	
	Btu per hr	Grains	Btu per hr	Grains
60	70	467	62	450
65	75	500	62	450
70	100	700	75	500
75	140	900	100	700
80	175	1200	145	960
85	225	1500	205	1400
90	280	1900	290	2000

MOTORS. It is usual to assume that the entire heat equivalent of the power applied to an electric motor located in the air-conditioned space becomes a burden on the cooling plant. This becomes, for a definite load,

$$q_m = \frac{2545 \text{ hp}}{e} \text{ Btu/hr} \tag{10-6}$$

The value for the motor efficiency, expressed in the formula as a decimal, may vary from 60 per cent for the fractional horsepower to 75 or 80 per cent in the moderate sizes.

In electric toasters, electric irons, electric stoves, etc., the heating effect is

$$q = 3.412 \times \text{watts in Btu/hr}$$

if no separate vent is installed for the purpose of exhausting the heated air into the atmosphere

TABLE 10-11b
RELATION BETWEEN METABOLIC RATE AND ACTIVITY

Activity	Hourly Meta- bolic Rate for Average Person or Total Heat Dissi- pated, Btu per Hour	Hourly Sensible Heat Dis- sipated, Btu per Hour	Hourly Latent Heat Dis- sipated, Btu per Hour	Grains of Moisture Dissi- pated per Hour
Average person seated at rest ¹	384	225	159	1074
Average person standing at rest ¹	431	225	206	1392
Tailor ¹	482	225	257	1737
Office worker moderately active	490	225	265	1791
Clerk, moderately active, standing at counter	600	225	375	2535
Book binder ¹	626	225	401	2711
Shoe maker ¹				
Clerk, very active, standing at counter	661	225	436	2947
Pool player	680	230	450	3042
Walking 2 mph ¹				
Light dancing	761	250	511	3454
Metal worker ¹	862	277	585	3955
Painter of furniture ¹	876	280	596	4029
Restaurant serving, very busy	1000	325	675	4563
Walking, 3 mph ¹	1049	345	704	4759
Walking, 4 mph ¹				
Active dancing, roller skating	1388	450	938	6341
Stone mason ¹	1488	488	1000	6760
Bowling	1500	490	1010	6841
Man sawing wood ¹	1797	587	1210	8180
Slow run ¹	2285			
Walking, 5 mph ¹	2530			
Very severe exercise ¹	2555			
Maximum exertion, different people ¹	3000 to 4800			

¹ Metabolism rates actually determined by test for these activities. Metabolic rates for other activities estimated. Total heat dissipation integrated into latent and sensible rates by actual tests for metabolic rates up to 1250 Btu per hr, and extrapolated above this rate. Values for total heat dissipation apply for all atmospheric conditions in a temperature range from approximately 60° to 90° F dry-bulb. Division of total heat dissipation rates into sensible and latent heat holds only for conditions having a dry-bulb temperature of 79° F. For lower temperatures, sensible heat dissipation in- and latent heat decreases; for higher temperatures the reverse is true.

SPECIAL APPLIANCES. Special appliances, such as gas-heated steam tables and gas-heated hot water tanks, will become either a partial or a complete burden in the matter of the heat evolved in the combustion of the gas. Gas has a calorific value of 600 to 800 Btu/cu ft.

10-13. The Maximum Load. The maximum refrigerating load does not have to occur at any fixed time of the day, and not *all* the heating factors are necessarily additive. Office buildings and residences may have their maximum refrigerating load at 3 or 4 o'clock or even later in the afternoon. Theaters and hotels may have their maximum when the human load is the greatest.

The effect of heat storage in the building material depends on the

building construction. If the building is insulated, probably heat storage can be neglected. Otherwise, and usually this is true of residences, it affects appreciably the time of the maximum load and the amount of refrigeration.

10-14. Cooling of Office Buildings. The metropolitan office building has special features in its construction. As a rule it is tall in proportion to its floor area, its floor area is valuable and limited, the building construction is of the best and its window area is a large proportion of its wall area. Usually the office building is occupied only 8 hours of the day.

In testing the cooling requirements in an office building in Detroit‡ it was found that solar radiation at times accounted for 75 per cent of the total cooling necessary, whereas the sensible heat gained through the walls was only a third or a fourth of what entered through the glass, and both of these were small as compared with the make-up air, the effect of the people present, illumination, and the power required to circulate the air.

The results obtained from testing the cooling requirements of single rooms in a modern office building in Pittsburgh§ appeared somewhat different from those reported by Walker. The transmission of heat through the walls and windows was a larger percentage of the total heat entering the room, the value of U for the walls apparently being a greater contributing factor. One of the outstanding features of these tests was the high heat capacity and conductivity of the floors combined with a film conductance coefficient from the floor surface to the air of considerable amount. This resulted in the absorption of a large percentage of the solar radiation by the material of the floor, from which it passed very slowly to the air in the room.

The Pittsburgh tests included also the effect of a half-opened door. Infiltration increased from a rate of 0.86 air change per hour without an open door to 6.2 air changes per hour with the door opened so as to make an angle of 45°.

10-15. Inside Design Conditions. The tests of relative comfort of some peoples from 20 to 70 years of age who were located in the northeast part of the United States showed a preference for higher temperature in summer than in winter. The preference had to do with the clothing worn at the *time of the test* and the adaptation of the persons to seasonal weather. Young men as a rule prefer colder and women and aged men warmer temperatures. The test results apply to homes, offices,

‡ Walker, Sanford, and Wells, "Field Studies of Office Building Cooling," *Trans. ASHVE, Journal Section*, January, 1932.

§ Houghten, Gutberlet, and Wall, *Trans. ASHVE, Journal Section*, April, 1935.

schools, and other places where people of sedentary habits spend from 3 to 8 hours per day. In theaters and department stores, which are mechanically cooled in warm weather, the contrast between inside and outside air conditions becomes the deciding factor in regard to the temperature and the humidity to be maintained. If the contrast is too great, unpleasant sensations of chill or intense heat combined with mugginess will be experienced on entering or leaving the building. These unpleasant sensations may be avoided by increasing the indoor temperature to a level between the optimum dry-bulb temperature for continuous exposure, 71° to 76° F, and that prevailing out-of-doors, and by

TABLE 10·12

Outside Dry-bulb Temperature, °F	Inside Temperature (Dew-Point Temperature Held Constant at 57° F)		
	Dry-bulb	Wet-bulb	Effective
95	80	65.2	73.4
90	78	64.5	72.2
85	76.5	64	71.1
80	75	63.5	70.2
75	73.5	63	69.3
70	72	62.5	68.2

decreasing the relative humidity to 60 per cent or less in order to keep the skin and clothing dry. Desirable indoor air conditions as given by Yaglou, for sedentary persons for exposures *less than three hours*, are shown in Table 10·12.||

Table 10·13 summarizes suggested inside design air temperatures for summer comfort cooling.

INSULATION

10·16. Moisture in Insulation. The primary driving force tending to send moisture into insulation and building material is the difference in the vapor pressure inside and outside the material. The result is that water will accumulate on the colder surfaces, which in cold storage spaces are those on the inside. The effect of moisture has been very well shown by Miller, ¶ who showed that the presence of water increased

|| Houghten and Gutberlet (*Heating, Piping, Air Conditioning, Journal Section*, October, 1935) state that attempts to apply Table 10·12 have led to chaotic and unsatisfactory results. Further investigations indicate that the average person, within 20 to 40 min after entering any atmospheric condition, will feel comfortable with an effective temperature range from 70° or 71° to 74° or 75° F.

¶ Miller, *Refrig. Eng.*, November, 1927.

TABLE 10-13
INSIDE DESIGN CONDITIONS FOR COMFORT COOLING

Out- side Design Dry Bulb	Class AA Special Application Occupancy over 40 Min for Maximum Comfort			Class A Average Application Occupancy over 40 Min Common Commercial Practice			Class B Occupancy under 40 Min		
	Dry Bulb	Wet Bulb	R.H.%	Dry Bulb	Wet Bulb	R.H.%	Dry Bulb	Wet Bulb	R.H.%
85	73	64	64	74	64	60	75	64	57
	74	62	53	75	62	50	76	63	48
	75	61	45	76	61	43	77	61	41
	76	59	38	77	59	35	78	59	32
90	75	66	65	77	67	62	78	68	60
	76	65	57	78	66	53	79	66	53
	77	63	48	79	65	48	80	65	45
	78	62	41	80	63	40	81	64	40
95	77	67	60	79	68	58	81	69	55
	78	66	54	80	67	51	82	68	49
	79	65	47	81	66	45	83	67	44
	80	64	42	82	65	40	84	66	38
100	78	69	65	80	69	58	82	70	57
	79	68	57	81	68	52	83	69	51
	80	66	50	82	66	44	84	68	45
	81	65	43	83	65	38	85	67	38
105	79	69	63	81	70	58	83	71	57
	80	68	57	82	68	51	84	70	50
	81	67	49	83	67	45	85	69	45
	82	66	43	84	66	39	86	68	40

NOTE: It is recognized that, in certain unusual localities, the outside design dew point is substantially less than the dew point of the tabulated inside design conditions.

the conductivity from 50 to 100 per cent and created a tendency of the insulation to disintegrate, rot, and loosen from the wall as well. The moisture problem is very much a matter of design and protection and particularly a matter of waterproofing. However, as almost every coating will crack with expansion and contraction of the pipe or container or the building, continued replacement of the protecting coating

is required to prevent moisture from entering the insulation. This is especially important in brine pipe lines that are exposed to humid air. Frequently no seal is required to prevent moisture entering from the inside, for example, the cold temperature space. Where materials of low hygroscopic nature permit it, a drying action may be secured by means of the dehydration of the body of the insulation when a low capillarity and low moisture affinity allow it, in which case venting the insulation on the interior results in drying out the insulation if the outside is waterproofed. However, no perfect commercial waterproofing material has so far been discovered.

10·17. Erection. In erecting corkboard against brick or concrete the first process should be to prime the masonry thoroughly with at least one coat of some sort of emulsified asphalt primer. If such a priming coat is not applied to the walls, and the corkboard is erected against the walls in portland cement, the air which seeps through will carry with it moisture which may condense back of the corkboard and

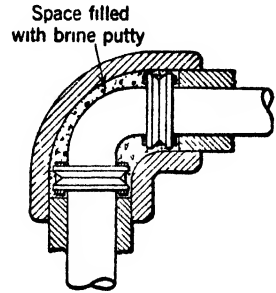


FIG 10·2. Insulation—flanged ell.

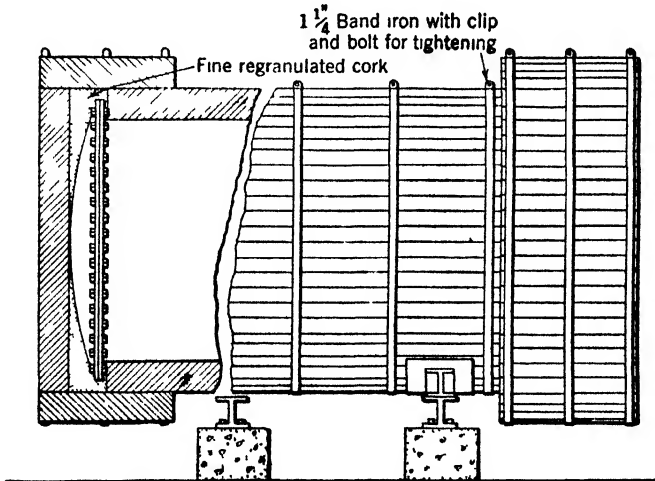


FIG. 10·3. Insulation—horizontal brine cooler.

even enter the interstices between the granules of cork, thereby causing disintegration of the cork.

Whether it is more desirable to use asphalt or cement plaster in cold storage construction is still not definitely settled. Cement gives greater

strength than asphalt, but asphalt is waterproof and costs considerably less than cement mortar. Cement plaster will crack; therefore the plaster finish should be applied in two coats and score marks should be made every 4 ft.

Cement plaster, however, is objectionable, and has proved unsatisfactory in many particulars. In the first place it does not bond well with

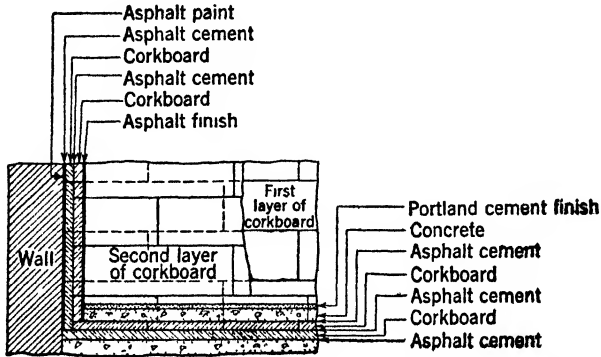


FIG 10 4 Insulation—wall and floor.

cork. Cork is without suction or capillarity, and it is difficult to apply plaster to a surface that lacks this property, with any assurance of good adhesion. There are many instances where plaster has fallen off the cork sheets and has had to be replaced with the addition of wire mesh or expanded metal in order to be held in place.

Not only does plaster develop cracks, but also the very conditions of its use make the development of cracks inevitable. Cork sheets are

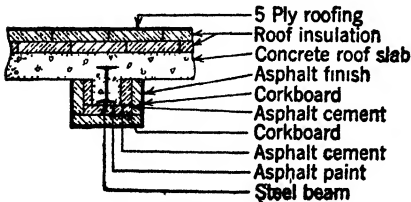


FIG. 10-5. Insulation—concrete beams.

almost always set up in hot asphalt, which is used as a cementing medium, although asphalt is not a solid but is in reality a fluid. When the wall against which the cork is laid expands or contracts, some slight movement of the cork takes place. When this occurs with rigid plaster on the face of

the cork the cement plaster generally yields by cracking. Figures 10-2 to 10-5 show methods of applying cork insulation.

10-18. Selection of Insulants. The choice of an insulating material is not governed entirely by the conductivity of the material but by the cost of erection, the moisture-absorbing properties, the yearly

load factor, the temperature on the cold side as well as the humidity of the air exposed to the insulation, deterioration, ease of erection, and bulkiness. No insulating material is waterproof, for even cork can absorb as much as 20 per cent water vapor by weight. The effect of the absorption of water is to increase the conductivity, to accelerate and to promote disintegration, and to cause the separation of the material from the wall or the lower side of the ceilings, unless it is held by nails to sleepers in the ceiling.

CORK and particularly *corkboard* and molded cork have been favorite insulating materials for a long time and especially for cold storage warehouse insulation. Corkboard comes in a number of thicknesses from $\frac{1}{4}$ to 1 in. by eighths and $1\frac{1}{2}$, 2, 3, 4, and 6 in. and usually in boards 12 by 36 in. Corkboard is erected usually in two or more courses laid in cement mortar or asphalt in order to provide a binder as well as an attempt at both airtightness and waterproofness. If cement mortar is used the initial course should have a uniform thickness of $\frac{1}{2}$ in. Subsequent courses must have their joints broken, and they should be fastened with wooden skewers driven obliquely into the preceding course. The asphalt used should be odorless and must be applied hot mixed with approximately 3 per cent by weight of cork dust. Although no entirely satisfactory method of applying insulation has yet been found some form of asphalt seems at present most desirable, especially where the warm side of the insulation is in contact with a humid atmosphere. Corkboard should have also a finishing layer either applied at the factory or superimposed after erection. When put on after erection it may be an asphalt emulsion or a cement plaster. Finally a coat of aluminum enamel or two coats of white enamel to provide a solid coloring should be applied.

Molded cork covering for pipes and fittings is made up with the desired thickness, and it comes in half sections. These are fastened with copper wire, the joints being carefully filled with corkfiller or cement and the outside painted with an asphalt paint. The important point is to insure at all times that there are no cracks by which water can get into the insulation. At times it is desirable to add to the covering for pipes and molded fittings a waterproofing jacket of two-ply roofing paper or a stock "canvas" strip wound spirally and painted with enough asphalt paint to saturate it. *Hair felt* has been used for pipe line insulation very successfully for a long time. It is erected in 1-in. layers wrapped with twine and having waterproof paper between the layers and a heavy waterproofed roofing felt secured with copper wire on the outside. An example of pipe insulation is shown in Fig. 10·6.

Rock cork is made of a loose mineral wool combined with some water-

proofing binder which is molded into sheets, usually 18 by 36 in., and is then baked. Rock cork does not appear to absorb water readily. As the material is mineral in origin it shows no apparent sign of decay during its normal life, and because of the lack of a tendency to absorb water vapor or water as a liquid the low conductivity is maintained after erection. Moreover, it seems to be possible to seal rock cork thoroughly against infiltration of air.

Truck and refrigerator car manufacturers favor the use of a continuous insulation between the outside sheathing and the inner lining. *Crumbled aluminum* has been used successfully as well as fibrous material in-

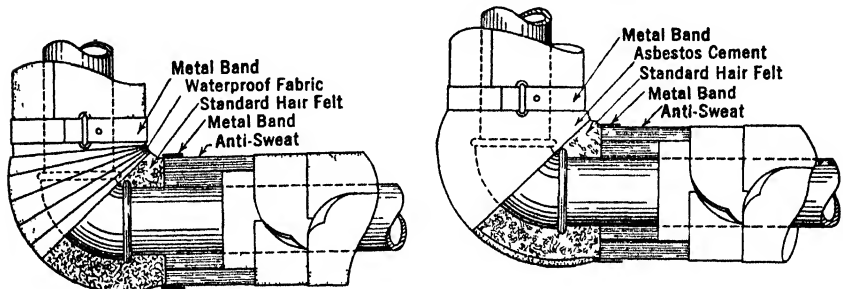


FIG. 10-6. Insulation for ice water piping.

cased in a sealed envelope. The fibers are compressed under controlled pressure, and the mat thus formed is fed by the forming machine into moisture-proof cartons.

Refrigerator boxes of household size are now using very little cork because this material does not lend itself well to mass production and the kind of box construction now in use. First the insulation, heretofore of semi-rigid material, had to be cut exactly to size, although more recently the use of a fibrous material sealed in a moisture-proof envelope has become popular. Materials used include Celotex refrigerator insulation, balsam wool, seed pod fibers from the ceiba tree, expanded rubber, paper, aluminum foil, and other material.

10-19. Economical Thickness. No rigid rule for the thickness of insulation, such as 2 in. of corkboard for a difference of temperature of 10° F, 3 in. for 20° F, 4 in. for 40° F, etc., can be given except as an approximation. The precooler before shipment of fruits and vegetables has a duration of refrigeration of a few weeks only, whereas the terminal cold storage plant may be filled for the greater part of the year. In order to show the factors that should be included in the analysis of the economical thickness of insulation the following costs have to be included:

- (1) The cost of a ton of refrigeration delivered to the room, per 24 hr.

- (2) The cost of the insulation applied, per year.
 (3) The cost of the refrigerating equipment not included in item 1 and particularly the room piping, etc., per year.
 (4) The value of the space occupied by the insulation per year.

The amount of these losses may be expressed in the following manner.

Let

- L = the thickness of the insulation in inches
 k = the conductivity of the insulation, per 1 in. thick, per hour
 F = the yearly load factor
 B = the cost in dollars for the insulation applied, per 1 in. thick, per 1 sq ft of surface
 A = the cost in dollars per ton of refrigeration per 24 hr, delivered
 G = the cost in dollars per ton of refrigeration of the machinery, etc., not included in A
 I = the interest rate, as a percentage, for the insulation investment
 R = the repair cost per year as a percentage of the insulation first cost
 Y = the life of the insulation, in years
 I', R', Y' = similar values applied to machinery, etc., in G
 t_a = the temperature of the outside air, in Fahrenheit degrees, as the average for the period of operations
 t_m = the maximum outside temperature, in Fahrenheit degrees
 t = the temperature of the cold storage room, in Fahrenheit degrees
 t_p = the temperature of the refrigerant in the piping, in Fahrenheit degrees
 S = the value per year of 1 cu ft of space in the cold storage room
 U = the coefficient of heat transfer of the wall for the materials of construction other than the insulation as given by the usual formula:

$$\frac{1}{U} = \frac{1}{h_1} + \frac{1}{h_2} + \frac{L_1}{k_1} + \frac{L_2}{k_2}, \text{ etc.}$$

When these symbols are used, the separate costs are found to be:

(1) The cost per year of the heat leakage through the insulation

$$= \frac{t_a - t}{\frac{1}{U} + \frac{L}{k}} \frac{24 \times 365 \times F \times A}{288,000}$$

(2) The cost of the insulation per year, per square foot, per 1-in. thick

$$= BL \left(\frac{I}{100} + \frac{R}{100} + \frac{1}{Y} \right)$$

(3) The cost per year of the investment required to offset the heat leakage through the insulation

$$= \frac{t_m - t}{\frac{1}{U} + \frac{k}{L}} \frac{24}{288,000} G \left(\frac{I'}{100} + \frac{R'}{100} + \frac{1}{Y'} \right)$$

(4) The cost of the space occupied by the insulation per year = $\frac{SL}{12}$

As a rule the cost of insulation, applied, can be expressed by the formula

$$B = \frac{C'}{L} + B'$$

where C' is the cost of the finish, plaster, nails, labor and overhead per square foot; and B' is the cost of the insulation delivered to the job.

Also, if P is the cost in dollars per square foot of refrigerating piping installed in the cold storage room as the equipment represented by G ,

$$G = \frac{12,000 P}{u(t - t_p)}$$

where u = the overall coefficient of heat transfer for the piping in Btu per hour.

Then, if Z is the total cost per year,

$$Z = \frac{t_a - t}{\frac{1}{U} + \frac{k}{L}} \frac{365 FA}{12,000} + \left(\frac{C'}{L} + B' \right) \left(\frac{I}{100} + \frac{R}{100} + \frac{1}{Y} \right) + \frac{t_m - t}{\frac{1}{U} + \frac{k}{L}} \left(\frac{P}{u(t - t_p)} \right) \left(\frac{I'}{100} + \frac{R'}{100} + \frac{1}{Y'} \right) + \frac{SL}{12}$$

For a minimum, $dZ/dL = 0$; so, differentiating and putting this equal to zero and solving for L gives

$$L = 1.743 \sqrt{\left[\frac{A(t_a - t)F + \frac{0.329P}{u(t - t_p)} \left(I' + R' + \frac{100}{Y'} \right) (t_m - t)}{B' \left(I + R + \frac{100}{Y} \right) + 8.3S} \right]} k - \frac{k}{U} \quad (10.7)$$

TABLE 10-14

RATES OF HEAT TRANSMISSION FOR ANTI-SWEAT MATERIAL

Btu per hour, per degree temperature difference, per linear foot and per square foot of pipe surface.

Insulation	½ in. Thick		¾ in. Thick		1 in. Thick		1¼ in. Thick		2 in. Thick	
	Btu per lin ft	Btu per ft²	Btu per lin ft	Btu per ft²	Btu per lin ft	Btu per ft²	Btu per lin ft	Btu per ft²	Btu per lin ft	Btu per ft²
½	0.212	0.963	0.185	0.841	0.167	0.758	0.143	0.649	0.128	0.581
¾	0.245	0.891	0.212	0.770	0.189	0.687	0.160	0.581	0.142	0.515
1	0.286	0.830	0.243	0.706	0.215	0.625	0.180	0.523	0.158	0.460
1¼	0.338	0.777	0.284	0.653	0.249	0.572	0.206	0.473	0.180	0.413
1½	0.374	0.751	0.312	0.627	0.272	0.546	0.223	0.448	0.194	0.389
2	0.443	0.713	0.367	0.591	0.318	0.512	0.257	0.413	0.221	0.356
2½	0.518	0.688	0.425	0.564	0.364	0.484	0.292	0.388	0.249	0.331
3	0.611	0.666	0.497	0.542	0.423	0.462	0.337	0.367	0.284	0.310
3½	0.685	0.655	0.553	0.528	0.468	0.447	0.369	0.353	0.311	0.297
4	0.757	0.643	0.611	0.518	0.515	0.437	0.404	0.343	0.338	0.287

TABLE 10-15

RATES OF HEAT TRANSMISSION

The rates of heat transmission given below are expressed in Btu per square foot (and also per linear foot) of pipe surface, per hour, per degree temperature difference between fluid in the pipe and air surrounding the pipe. The thickness of 2-layer has been taken as 2 in., of 3-layer as 3 in. and 4-layer as 4 in. Sometimes the thicknesses after application are less than these nominal thicknesses, and for such cases the figures in the table should be modified accordingly.

Pipe Size, inches	2-in. Insulation		3-in. Insulation		4-in. Insulation		5-in. Insulation		6-in. Insulation	
	Btu per lin ft	Btu per ft ²	Btu per lin ft	Btu per ft ²	Btu per lin ft	Btu per ft ²	Btu per lin ft	Btu per ft ²	Btu per lin ft	Btu per ft ²
1/4	0.086	0.390	0.074	0.331	0.065	0.295	0.069	0.252	0.078	0.157
1/2	.095	.345	.079	.288	.070	.255	.072	.210	.078	.139
1	.107	.310	.088	.257	.078	.227	.080	.183	.085	.126
1 1/2	.124	.285	.102	.235	.090	.206	.085	.170	.094	.114
2	.132	.265	.109	.218	.095	.190	.094	.151	.104	.107
2 1/2	.149	.240	.120	.193	.104	.167	.107	.141	.104	.104
3	.167	.222	.133	.177	.114	.151	.114	.125	.112	.112
3 1/2	.192	.210	.148	.162	.128	.140	.124	.119	.120	.107
4	.211	.201	.165	.157	.138	.132	.126	.108	.127	.097
4 1/2	.228	.194	.176	.150	.148	.126	.142	.103	.127	.092
5	.247	.188	.190	.145	.158	.121	.151	.103	.149	.087
5 1/2	.268	.184	.205	.141	.170	.117	.164	.097	.164	.082
6	.307	.177	.233	.134	.191	.111	.184	.092	.178	.079
7	.343	.172	.257	.129	.212	.106	.200	.089	.206	.076
8	.380	.168	.282	.125	.232	.102	.220	.089	.234	.073
9	.415	.165	.307	.122	.242	.099	.234	.083	.266	.070
10	.455	.162	.335	.119	.279	.097	.268	.080		
12	.528	.158	.387	.116	.310	.093				

The ordinary range of temperatures requires the following thicknesses:

Plus 40° F to plus 15° F — 2-in. insulation
 Plus 15° F to minus 5° F — 3-in. insulation
 Minus 5° F to minus 20° F — 4-in. insulation

Minus 20° F to minus 40° F — 5-in. insulation¹
 Minus 40° F to minus 60° F — 6-in. insulation¹

¹ For pipes smaller than 3 1/2 in. at temperatures from minus 20° to minus 60° F, deduct 1-in. insulation

TABLE 10-16
HEAT TRANSMISSION, *U*, THROUGH CORK PIPE INSULATION

Nominal Pipe Size, inches	Ice Water Thickness				Brine Thickness				Heavy Brine Thickness	
	Thickness, inches	Heat Transmission ¹		Thickness, inches	Heat Transmission ¹		Thickness, inches	Heat Transmission ¹		
		Square foot	Linear foot		Square foot	Linear foot		Square foot	Linear foot	
1/4	1.45	0.642	0.091	1.98	0.571	0.081	2.97	0.494	0.070	
1/2	1.40	.576	.102	1.91	.511	.091	2.90	.436	.077	
3/4	1.35	.525	.116	1.83	.460	.101	2.82	.385	.085	
1	1.47	.454	.125	1.97	.396	.109	2.72	.345	.095	
1 1/4	1.59	.394	.135	2.09	.345	.119	2.84	.299	.103	
1 1/2	1.42	.378	.164	2.42	.289	.126	3.16	.254	.111	
2	1.52	.348	.174	2.55	.264	.132	3.05	.241	.120	
2 1/2	1.55	.317	.197	2.31	.253	.157	3.31	.208	.129	
3	1.31	.330	.248	2.56	.221	.165	3.06	.200	.151	
3 1/2	1.50	.287	.263	2.75	.197	.181	3.24	.179	.164	
4	1.50	.278	.292	3.00	.179	.187	3.50	.163	.171	
4 1/2	1.75	.244	.286	2.75	.182	.214	3.25	.163	.192	
5	2.00	.216	.284	3.00	.166	.218	4.25	.133	.174	
6	1.72	.231	.336	2.72	.170	.248	3.97	.133	.193	
7	1.68	.227	.393	2.94	.154	.267	3.92	.127	.220	
8	1.69	.223	.445	2.94	.149	.298	4.00	.121	.242	
9	1.94	.197	.444	2.94	.144	.324	4.00	.117	.264	
10	1.94	.194	.488	3.19	.133	.335	4.00	.113	.284	
11	1.87	.195	.549	3.12	.133	.375	4.00	.110	.310	
12	1.62	.213	.709	3.12	.127	.423	4.00	.106	.348	
14	3.12	.125	.460	4.00	.103	.378	
16	3.12	.123	.515	4.00	.101	.432	

¹Btu per square foot of pipe surface, and per linear foot of pipe, per degree temperature difference, per hour.

TABLE 10-16 (Continued)

Insulation	Space between Parallel Pipes, inches	Space between Pipe and Wall, inches
Ice Water Thickness		
Up to 6-in. pipe — screwed fittings	6	4
Larger than 8-in. pipe — screwed fittings	10	5
All pipe sizes — flanged fittings	10	5
Standard Brine Thickness		
Up to 6-in. pipe — screwed fittings	8	6
Larger than 6-in. pipe — screwed fittings	14	8
All pipe sizes — flanged fittings	14	8
Heavy Brine Thickness		
Up to 3-in. pipe — screwed fittings	10	8
Larger than 3-in. pipe — screwed fittings	18	12
All pipe sizes — flanged fittings	18	12

Example. Assume the following values: wall = 12 in. brick, $\frac{1}{2}$ in. cement mortar; $I = 6$ per cent; $I' = 6$ per cent; $R = 3$ per cent; $R' = 3$ per cent; $Y = 15$ yr; $Y' = 8$ yr; $u = 1.5$ Btu/hr; $K = 0.35$ Btu/hr; $t_a = 50^\circ$ F; $t = 0^\circ$ F; $t_p = 95^\circ$ F; $t - t_m = 10^\circ$ F; $F = 1.0$; $A = 1.90$; $S = \$0.15$ per cu ft; $t_m - t = 95^\circ$ F; $P = \$0.50$, the cost of piping installed, plus all accessories per cu ft³; $t_m - t = 95^\circ$ F; $P = \$0.50$, the cost of piping installed, plus all accessories.

Solution.

$$B = \left(\frac{0.16}{L} + 0.065 \right) \text{ in dollars} \quad B' = 0.065 \quad \frac{1}{U} = 3.22$$

Therefore, from equation 10-7, $L = 6.28$ in.

For extremely low temperatures it is generally advisable, in order to prevent condensation on the surface of the insulation, to use lags over heavy thicknesses of cork. Between -25° and 40° F, lags $1\frac{1}{2}$ in. thick are recommended; below -40° F, lags 2 in. thick. The lags are waterproofed with asphalt.

Where finished with a coat of asphalt paint over the original mineral rubber finish, the cork pipe insulation presents a pleasing appearance, making it unnecessary to apply canvas or other material.

If there is objection to the dark finish, the surface can be painted with special enamel or covered with canvas and painted the desired color.

PROBLEMS

1. Using standard values for the inside and outside film coefficients, calculate the overall coefficient of heat transfer for a wall consisting of 8 in. of concrete with

6 in. of medium density corkboard applied on the inside and in direct contact with the concrete.

2. An insulated wall has an overall coefficient of 0.05 when outside wind velocity is 15 mph. If the outside surface is smooth determine the per cent of change in load that would occur if the wind velocity increased to 60 mph.

3. A room 20 ft by 20 ft has a 12-ft ceiling. If walls, floor, and ceiling have an overall coefficient of heat transfer of 0.10 and if they are all subject to outside exposure, determine the rate of heat loss from the room when inside air temperature is 0° F and outside is 90° F.

4. If a 4-ft by 8-ft double-glazed window (zero infiltration) were installed in each of two walls of the room described in Problem 3, calculate the resultant room heat loss and compare with the result from Problem 3.

5. If the windows in Problem 4 were not weatherstripped and had $\frac{3}{32}$ -in. crack and $\frac{3}{32}$ -in. clearance determine the ventilation load on the room due to infiltration. Discuss the significance of infiltration in an insulated structure.

6. Plot the conductance of an air space as a function of thickness at fixed temperature.

7. Plot the conductance of an air space as a function of temperature at fixed thickness.

8. The statement is made in the text that the fraction of absorbed solar radiation which is transferred into the cooled space is approximately equal to 0.23 times the overall coefficient. Would you not expect the factor 0.23 to change as a function of U ? Would you expect the variation to be direct or inverse?

9. Plot R_p values from Table 10-8 for a wall facing south and for each hour of the day as a function of latitude and discuss the resultant curves.

10. Plot the ratio of sensible to latent body heat loss as a function of metabolic rate and discuss the resultant curve.

11. Find the effect at 2 P.M. of solar radiation striking a 12-in. brick wall located at approximately 42.5° north latitude and having a southwest exposure, and also for vertical windows per square foot per hour.

12. Derive an equation for the conductance of a wall from inside surface to outside air.

13. Discuss the relative importance of thermal conductivity of a thin-walled metal pipe when used to separate two low-velocity gases which are exchanging heat and, when used to separate two high-velocity liquids, which are exchanging heat.

14. As the thickness of a homogeneous wall is increased (for fixed inside and outside air temperatures) would the temperature at the mid-point be expected to increase, decrease, or stay the same? Explain.

CHAPTER XI

TRANSIENT LOADS: PULL-DOWN; SUBCOOLING

11.1. Unsteady-State Heat Flow. The equations which have so far been developed are for evaluation of heat transfer rates for systems which have reached a condition of stable non-equilibrium. Although equilibrium never exists when energy is in transition the fact does remain that under certain circumstances, and these of greatest importance in practical refrigeration load analysis, the system does not vary with time. In other cases, however, change of state does occur as a function of time, and in this event the simple heat transfer equations which have so far been developed can no longer be applied. When material is placed in cold storage a transient interval occurs during which the temperature of the product is pulled down to the box temperature, and, during this interval, the load on the box differs from the normal "holding" load, varying from a maximum immediately after introduction of the product to a normal carrying value at some later time. The engineer must be able to estimate the maximum load, the variation in load, and the time interval required for a return of the system to stability.

Paradoxically, the influence of transient conditions in cold storage is such that the hazard of an equipment breakdown necessitating temporary lack of refrigeration capacity is least at a time when the cold storage chambers are filled with maximum weight of stored product. This is evident when one considers that if power failure occurred in a cold box containing no stored product the temperature within the box would rise relatively rapidly, since the thermal capacity of the room air is small and only the walls would have to receive appreciable quantities of energy in storage. If, however, the box were almost filled with a product in storage the thermal capacity of the product would retard the rate of temperature rise and a longer time interval would pass before a hazardously high temperature would be realized. Thus methods must be developed to permit anticipation of the heating curve (time-temperature) of a cold storage space in the event of power or other mechanical failure, the cooling (pull-down) time-temperature curve, and the influence on such transient phenomena of the thermal capacity of both the space and its contents.

11.2. Cooling Period.* Exact determination of the time-temperature relationship for a refrigerated space during the cooling period is a complex problem in transient heat flow. When the stored material is in geometrically similar, homogeneous units of approximately uniform size a precise analytical solution by means of differential equations is usually possible if circulation of air in the space is sufficient to maintain uniformity of space air temperature. In many practical problems, however, the exact solution is either inapplicable (owing to lack of adequate knowledge of boundary conditions) or too complex and time-consuming to be of value for routine field calculations. Furthermore, many cases occur in practice where available design data are insufficient to warrant more than approximate treatment; for problems of this type less precise methods can be used to advantage.

By considering that the transient effects occur slowly and that, at any particular time during the cooling period, the equivalent of a steady-state temperature gradient exists, a very great simplification of the analysis is possible. Holme† proposed such a method for determining the heat-up time of intermittently heated structures, and its use for this same purpose has been extended and developed by others.‡ The same general procedure can be followed in adapting the Holme method to the reverse problem of a cold storage chamber.

The three basic factors which influence cooling rate are the thermal capacity of the system (including both material in storage and in the structure), the rate at which heat appears within the space (either due to inflow through walls or in ventilation air, or due to released heat due to work within the room, or due to chemical energy released by the stored material), and the energy-absorbing capacity of the refrigerating equipment. The first of these factors, heat capacity, is divisible into two parts each of which is subject to different treatment: The stored product, at equilibrium, is at the same temperature as the air within the cold storage space and can therefore be regarded as undergoing a temperature change equal to that of the air; thus for each 1° F drop in equilibrium inside air temperature the material in storage liberates an amount of heat equal to

$$Q_p = fVw_p c_p \quad \text{Btu} \quad (11.1)$$

* The material of this and the two following sections is largely taken from or adapted from "Refrigerating Design Problems in Marine Installations," Hutchinson, *J. Am. Soc. of Refrig. Engrs.*, March, 1944.

† Holme, *Warne*, October 17, 1931.

‡ Giesecke, *Heating, Piping, Air Conditioning*, August, 1938; Hutchinson, *Proc. Natl. Distr. Heating Assoc.*, Vol. 32, 1941.

where f = fraction of gross cold storage volume occupied by the stored material

V = gross volume of the refrigerated space

w_p = weight density of the stored produce in pounds per cubic foot

c_p = specific heat of the stored product in Btu per pound per Fahrenheit degree

For the conditions of analysis, cooling will be assumed to start from an initial equilibrium inside temperature of t_{i1} and proceed to a final equilibrium inside temperature t_{i2} . At the beginning and end of the cooling period the stored material will be considered to be at the same temperature as the inside air; during cooling, outside conditions are taken as constant and corresponding to the design conditions for which the refrigerating unit was selected. The walls, ceiling, and floor of the refrigerated space may adjoin spaces which are maintained at different temperatures; therefore the actual design air-to-air temperature difference to be used in calculating the load may vary widely for different surfaces of the room. To simplify the equations, without loss of accuracy, an equivalent outside design temperature t_d can be selected and an equivalent overall coefficient of heat transfer U_w then calculated from the equation

$$U_w = \frac{U_1 A_1 \Delta t_1 + U_2 A_2 \Delta t_2 + \cdots + U_n A_n \Delta t_n}{(A_1 + A_2 + \cdots + A_n)(t_d - t_i)} = \frac{\sum U_n A_n \Delta t_n}{A_w(t_d - t_i)} \quad (11.2)$$

where U_n , A_n , Δt_n are the respective overall coefficients, inside surface areas, and temperature differences of each of the n sections of refrigerated space which differ in construction or in exposure. Based on the equivalent design temperature t_d and the calculated equivalent overall coefficient U_w , the equilibrium rate of heat flow into the space, for the inside temperature t_i , is

$$q = U_w A_w (t_d - t_i) \quad \text{Btu/hr} \quad (11.3)$$

If outside air is introduced into the space for ventilation purposes an additive term on the right side of equation 11.3 will take account of the increased heat gain.

For each differently constructed or exposed section of the refrigerated space the mean structural temperature t_{mn} of wall, floor, or ceiling is calculable from the equation

$$\frac{t_{dn} - t_{mn}}{t_{dn} - t_i} = \frac{r_n}{R_n} = r_n U_n \quad (11.4a)$$

or

$$t_{mn} = t_{dn} - r_n U_n (t_{dn} - t_i) \quad (11.4b)$$

where t_{dn} is the temperature of the air exterior to section n for which r_n and R_n are the resistances from the outside air to, respectively, the mid-point of the construction and the inside air. The reduction in mean wall temperature Δt_{mn} accompanying a 1°F change of inside air temperature (when outside temperature remains constant) is then

$$\Delta t_{mn} = r_n U_n \quad (11.5)$$

and the total heat leaving storage in the structure is

$$Q_w = \sum_{n=1}^{n=n} r_n U_n A_n \frac{L_n''}{12} w_n c_n \quad \text{Btu} \quad (11.6)$$

where L_n'' , w_n , c_n are, respectively, the thickness (in inches), weight density, and specific heat of a material making up a section n of the cold storage room. Separate terms should, strictly speaking, be included in the summation for each type of material which, in series, constitutes a wall section, but, with insulated walls of the usual type, adequate accuracy can be attained by using fixed values of w and c corresponding to the properties of the insulation. Wall thickness is then taken at its overall value, and the storage characteristics of such non-insulating structural layers as sheathing, etc., are considered the same as for the insulant. Except in the case of walls with air spaces this simplification is usually permissible, and it leads to the equation

$$Q_w = w_w c_w \sum_{n=1}^{n=n} r_n U_n A_n \frac{L_n''}{12} \quad \text{Btu} \quad (11.7)$$

For most refrigeration problems the resistance of the films is relatively small in comparison with that of the insulating material; for such cases, when wall sections are approximately symmetrical with respect to the center plane of the insulant, the resistance r_n is practically equal to $0.5R_n$, so equation 11.7 simplifies to

$$Q_w = 0.5 w_w c_w \sum_{n=1}^{n=n} \frac{A_n L_n''}{12} = 0.5 w_w c_w V_w \quad (11.8)$$

where V_w is the gross volume, in cubic feet, of the walls, floor, and ceiling structure.

The small quantity of heat which leaves storage in the inside air being neglected, the total heat released from storage for each 1°F reduction of inside air temperature is (from equations 11.1 and 11.8)

$$Q = Q_p + Q_w = f V w_p c_p + 0.5 V_w w_w c_w \quad (11.9)$$

For a given compressor and condenser the capacity R in Btu per hour

is known as a function of the cooling water rate and of the temperature of refrigerant in the evaporator and cooling water in the condenser. For design cooling water rate and temperature, the capacity can be plotted as a function of evaporator temperature. If subscript f designates design (and therefore known conditions) the relationship between cooling capacity and temperature drop from room air to refrigerant is given by

$$\frac{t_i - t_r}{t_{if} - t_{rf}} = \frac{R}{R_f} \quad (11 \cdot 10)$$

or

$$t_r = t_i - \left[R \left(\frac{t_{if} - t_{rf}}{R_f} \right) \right] \quad (11 \cdot 11)$$

where the parentheses enclose a design constant of the system, t_i is arbitrarily selected, and t_r and R are dependent variables related by the capacity curve for the unit. Thus, to determine the capacity at any particular inside air temperature, the procedure is to assume a value of t_r , determine the corresponding value of R from rating tables, and test the assumed value by then evaluating the right-hand side of equation 11·11. In this way the capacity can be determined as a function of the inside air temperature.

When, as for the conditions of this analysis, equilibrium exists at an inside temperature t_i , corresponding to which the refrigerating system is not operating at full capacity, there will be a very rapid, but small, drop of inside air temperature as soon as the maximum capacity is applied. Usually the effect on the time-temperature curve of neglecting this drop is too small to be of significance, and for this reason the initial and final equilibrium temperatures, t_{i1} and t_{i2} , will be used as approximations to the actual inside air temperature at start and finish of the cooling period.

The initial and final rates of heat extraction from storage based on the above approximation are

$$R_1 - [U_w A_w (t_d - t_{i1})] = \text{initial heat extraction rate} \quad (11 \cdot 12a)$$

and

$$R_2 - [U_w A_w (t_d - t_{i2})] = \text{final heat extraction rate} \quad (11 \cdot 12b)$$

where R_1 and R_2 , corresponding to t_{i1} and t_{i2} , are evaluated from equation 11·11. The time in hours, θ_c , needed to pull down the temperature of the cold storage compartment from t_{i1} to t_{i2} is then equal to the total heat leaving storage divided by the log mean average rate of heat ex-

traction. Thus, from equations 11·9 and 11·12,

$$\theta_c = \left\{ \frac{(fVw_p c_p + 0.5V_w w_w c_w)(t_{i1} - t_{i2})}{R_1 - R_2 + [U_w A_w (t_{i1} - t_{i2})]} \right\} \left\{ \log_e \frac{R_1 - [U_w A_w (t_d - t_{i1})]}{R_2 - [U_w A_w (t_d - t_{i2})]} \right\} \quad (11 \cdot 13a)$$

When the ratio of initial rate to final rate of heat extraction is less than 2, equation 11·13a can be simplified by using the arithmetic average instead of the log mean rate:

$$\theta_c = \frac{2(fVw_p c_p + 0.5V_w w_w c_w)(t_{i1} - t_{i2})}{R_1 + R_2 - [f_w A_w (2t_d - t_{i1} - t_{i2})]} \quad (11 \cdot 13b)$$

By use of equation 11·13a or 11·13b sets of cooling curves can be readily constructed from which the operator can read directly the time needed for cooling a given load ($fVw_p c_p$ as the criterion of load) through any temperature range ($t_{i1} - t_{i2}$) when outside temperature is t_d and condenser cooling water has the value t_w (fixing corresponding values of R_1 and R_2). Further simplification is possible by breaking the load term, $fVw_p c_p$, into three parts such that: (1) f is a storage coefficient determined by loading conditions or amount of cargo available; (2) V is a constant and can be implicitly included in the curves; (3) $w_p c_p$ is a property of the stored material. On this basis the cooling curves would be constructed so that the operator, knowing the kind of material in storage, storage coefficient, cooling water temperature, initial inside temperature, and desired temperature reduction, could read directly the time in hours to reach equilibrium at the desired lower inside temperature.

The required cooling time corresponding to a given temperature drop from an initial temperature t_i will be a maximum when outside conditions and condenser cooling water are at their design values. Thus a set of cooling curves constructed for design conditions will give conservative results. Where the use of the curves is intended primarily to inform the operator of a possible hazard to the product due to too slow cooling, a single set of design condition curves having storage coefficient, material in storage, initial equilibrium temperature, and desired temperature reduction as the variables would suffice.

Accuracy of the cooling curves is limited by the assumption (upon which the analysis is based) that each instantaneous temperature gradient through the walls of the structure has the steady-state value and that the stored material remains at a uniform, but inconstant, temperature. Actually, the effect of the thermal diffusivity is to increase

the wall gradient at the inside surface and decrease it (from equilibrium value) at the outside surface, thereby establishing a curved gradient for which the mean wall temperature is higher than it would be for the equivalent steady-state condition. Thus for each 1° F drop in inside temperature, somewhat less than the calculated quantity of heat would actually leave storage in the walls, and the actual time required for the temperature change would therefore be less than that calculated from equation 11-13. Similarly, extraction of heat from the stored product necessitates a temperature gradient from the interior of the product to the air of the room; thus the mean product temperature lags the inside air temperature, and the time required for air temperature reduction is therefore less than that value given by the equations. For both of these reasons the cooling curves give conservative results.

Worthy of consideration, however, is the lag of mean product temperature. If the air temperature within the refrigerated space is thermostatically controlled, the cooling unit may cut off long before the mean product temperature has been reduced to its proper holding value. To offset the hazard to the product represented by this condition, a temporary lower inside air temperature setting is frequently used to maintain the maximum possible cooling rate until the mean temperature of the stored material is lowered to the necessary value. When this has occurred the thermostat setting can be raised, as the transfer of heat from the center to the subcooled exterior parts of the stored material will suffice to complete the cooling process.

The extent of the lag of product temperature depends, of course, on the method of storage and on the surface-volume ratio of the material. For thin slabs with ample air circulation around them the lag will be insignificantly small, whereas for large, thick packages of material having low thermal diffusivity, k/cw , the lag is likely to be great. Critical examination of the conditions of any particular problem will usually permit a reasonably accurate estimation of the importance of lag effects and of the adjustment of end conditions needed to correct for them; for many practical cases such effects can be entirely neglected without serious discrepancy in results.

11.3. Heating Period. Less frequently used than cooling curves, but of value in estimating the possible hazard associated with either failure or temporary shut-down of refrigerating equipment, is the time-temperature relationship during the heating period. Another very significant application of this same analysis to special temporary storage problems will be discussed in the next section. Basically the heating curve for a refrigerated space is the inverse of the cooling curve for an intermittently heated structure. During the heating period, energy

enters storage at a rate equal to the heat transmitted through the structure (plus such heat gain as may be due to introduction of outside air for purposes of ventilation). The heating time is therefore determined entirely from considerations of structural heat resistance and the stored product heat capacity; rating of the refrigerating equipment does not enter into the analysis. (An exception to the latter statement occurs when a subcooled material is introduced into a refrigerated space and the minimum holding temperature, determined by maximum capacity of cooling, exceeds the initial temperature of the product; for this case the rate of temperature rise would, of course, be influenced by cooling capacity.)

The equations for the heating-up period can be written by analogy with equations 11·13a and 11·13b:

$$\begin{aligned}\theta_h &= \frac{(fVw_p c_p + 0.5V_w w_w c_w)(t_{i2}' - t_{i1}')}{U_w A_w [(t_d - t_{i1}') - (t_d - t_{i2}')] } \log_e \frac{U_w A_w (t_d - t_{i1}')}{U_w A_w (t_d - t_{i2}')} \\ &= \frac{(fVw_p c_p + 0.5V_w w_w c_w)(t_{i2}' - t_{i1}')}{U_w A_w (t_{i2}' - t_{i1}')} \log_e \frac{t_d - t_{i1}'}{t_d - t_{i2}'}\end{aligned}\quad (11\cdot14a)$$

where t_{i1}' and t_{i2}' are initial and final inside air temperatures, respectively, for the heating period. If $(t_d - t_{i1}')/(t_d - t_{i2}')$ is less than 2, cooling time can be calculated with 96 per cent accuracy by using the arithmetic average rate of heat flow of storage:

$$\theta_h = \frac{2(fVw_p c_p + 0.5V_w w_w c_w)(t_{i2}' - t_{i1}')}{U_w A_w (2t_d - t_{i1}' - t_{i2}')} \quad (11\cdot14b)$$

A problem of importance for emergency conditions is determination of the maximum time during which the inside temperature will remain below a predetermined danger value, t_{i2}' , if cooling is provided only from an inadequate auxiliary unit for which initial and final capacities are R_1' and R_2' . By analogy with equations 11·14a and 11·14b,

$$\theta_h' = \left\{ \frac{(fVw_p c_p + 0.5V_w w_w c_w)(t_{i2}' - t_{i1}')}{[U_w A_w (t_{i2}' - t_{i1}')] - R_1' + R_2'} \right\} \left\{ \log_e \frac{[U_w A_w (t_d - t_{i1}')] - R_1'}{[U_w A_w (t_d - t_{i2}')] - R_2'} \right\} \quad (11\cdot15a)$$

and

$$\theta_h' = \frac{2(fVw_p c_p + 0.5V_w w_w c_w)(t_{i2}' - t_{i1}')}{[U_w A_w (2t_d - t_{i1}' - t_{i2}')] - R_1' - R_2'} \quad (11\cdot15b)$$

A frequent use for one of the forms of equation 11·15 is in determination of the required average cooling capacity needed to extend the heating time over a known number of hours.

11.4. Subcooling. The heating and cooling equations developed in preceding sections are directly applicable to the special problem of analyzing subcooling of stored material as a means of reducing or eliminating refrigeration needed for transportation of food products. For short trips, subcooled cargo can, in some cases, be safely carried in uninsulated holds of ships having neither cooling equipment nor ventilation, and, for longer voyages (or on slower ships), subcooled cargo can often be carried in uninsulated and uncooled holds provided temporary insulation in the form of quilts or blankets is either draped from the battens or wrapped around the piles of cargo. By such methods the effective carrying capacity of unrefrigerated trucks, railroad cars, or vessels can be partially utilized in safely transporting certain kinds of perishable goods.

Thermodynamically, subcooling is basically inefficient as it requires operation of the refrigerating system over a pressure range much greater than that indicated by the maximum storage temperature of the product. Economically, however, the penalty of lowered coefficient of performance is in some cases, and particularly in marine transportation, likely to be offset by other factors such as the greater effectiveness of one large stationary subcooling refrigerating plant over that of a number of small mobile holding plants.

The effectiveness of subcooling increases with the volume-surface ratio of the storage space and is therefore much greater for large than for small compartments of the same geometrical shape. But for large compartments the thermal capacity of the walls, floor, and ceiling is likely to be insignificantly small with respect to that of the cargo; the numerator of equation 11.15*b* then becomes (from equation 11.9)

$$2Q(t_{i2}' - t_{i1}') = 2(Q_p + Q_w)(t_{i2}' - t_{i1}') \\ \doteq 2Q_p(t_{i2}' - t_{i1}') = 2fVw_p c_p(t_{i2}' - t_{i1}') \quad (11.16)$$

and the time for a cargo temperature rise from t_{i1}' to t_{i2}' is

$$\theta_h = \frac{2f \frac{V}{A_w} w_p c_p (t_{i2}' - t_{i1}')}{[U_w(2t_d - t_{i1}' - t_{i2}')] - R_a} = 24D \quad (11.17)$$

where $V/A_w = V' =$ volume-surface ratio of cargo space

$R_a =$ average capacity of refrigerating equipment expressed in Btu per hour per square foot of compartment space and evaluated between inside temperatures t_{i1}' and t_{i2}'

$D =$ number of days during transit

Then

$$12D\{[U_w(2t_d - 2t_{i2}')] + [U_w(t_{i2}' - t_{i1}') - R_a]\} = fV'w_p c_p (t_{i2}' - t_{i1}') \quad (11 \cdot 18)$$

or

$${}^\circ P = t_{i2}' - t_{i1}' = \frac{24D \left[U_w(t_d - t_{i2}') - \frac{R_a}{2} \right]}{fV'w_p c_p - 12DU_w} = \frac{(t_d - t_{i2}') - \frac{R_a}{2U_w}}{\frac{fV'w_p c_p}{24DU_w} - 0.5} \quad (11 \cdot 19)$$

where ${}^\circ P$ is the number of degrees of subcooling required to prevent the mean cargo temperature from exceeding t_{i2}' after D days with inadequate refrigeration. Thus the necessary subcooling can be calculated in terms of the final difference $t_d - t_{i2}'$, the thermal capacity of the cargo $w_p c_p$, the shape and size of the storage space V' , the heat transfer characteristic of the space U_w , the length of time with inadequate refrigeration D , and a factor representing the effective capacity of the available cooling equipment $R_a/2U_w$.

When the characteristics of the stored material establish a maximum permissible value of ${}^\circ P$, equation 11·19 can be used to solve for the required auxiliary refrigerating capacity, or, for a fixed value of this capacity, the necessary amount of insulation. For problems of the latter type, equation 11·19 can more effectively be written in terms of U_w as the unknown:

$$U_w = \frac{fV'w_p c_p {}^\circ P - 12DR_a}{12D \{ [2(t_d - t_{i2}')] + {}^\circ P \}} \quad (11 \cdot 20)$$

With particular reference to marine applications the premium gained by using a large compartment having a high volume-surface ratio plus the need for carrying more cargo than can be stowed in between-decks freezer rooms has recently been reflected in a trend toward use of deep holds for refrigerated cargo. Nickson§ describes the method used in insulating the hatches of small freighters for use in carrying frozen meat, whereas Cook and Steeves|| discuss the preparation of an uninsulated hold for use in carrying bacon; they point out that No. 2 hold (the second from the bow of the ship) is usually the best for this purpose (if there is another compartment, or insulation, between it and the engine room) because it is sufficiently amidships to have a high volume-surface ratio, yet is not subject, as are the aft hatches, to heat gain from the shaft alley. The volume-surface ratio is further improved if the lower hold is used.

§ Nickson, *Refrig. Eng.*, November, 1942.

|| Cook and Steeves, *Refrig. Eng.*, December, 1942.

For parallelepiped compartments of any size the volume-surface ratio, expressed in terms of mean length, width, and height, L , W , and D , respectively is given by the equation

$$V' = \frac{LWH}{2(LW + LH + WH)} \quad (11 \cdot 21)$$

When a hatch is used the length and width are frequently nearly the same, in which case equation 11·21 reduces to

$$V' = \frac{HB}{(4H + 2B)} \quad (11 \cdot 22)$$

where B is the molded beam of the ship. When the lower hold is used the height can frequently be taken as $\frac{1}{2}B$, in which case

$$V' = \frac{B}{8} \quad (11 \cdot 23)$$

For a lower hold having a volume-surface ratio as given by equation 11·23 and for use with subcooled cargo as the only available source of refrigerating effect, equations 11·19 and 11·20 take the simplified forms

$${}^{\circ}P = \frac{t_d - t_{i2}'}{\frac{fBw_p c_p}{192DU_w} - 0.5} \quad (11 \cdot 24)$$

$$U_w = \frac{fBw_p c_p {}^{\circ}P}{192D[(t_d - t_{i2}') + 0.5{}^{\circ}P]} = \frac{fBw_p c_p}{192D \left[\frac{(t_d - t_{i2}')}{{}^{\circ}P} + 0.5 \right]} \quad (11 \cdot 25)$$

Equation 11·24 gives the number of degrees of subcooling necessary to permit transportation of a cargo in a hold of fixed transmission characteristics, whereas equation 11·25 determines the required equivalent overall coefficient of heat transfer for a hold which is to transport a cargo having a known amount of subcooling. Both equations are based on the assumptions that the thermal capacity of hold walls is very small with respect to that of the cargo and that subcooled material is of such shape and size, and is so stowed, that the temperature rise occurs uniformly throughout the entire mass.

Example. A cold storage cargo is to be carried for 10 days with a design outside temperature of 65° and a maximum hold temperature of 35° . The material weighs 70 pcf, has a specific heat of 0.8, does not have any latent heat of fusion, and is stowed so that it occupies 80 per cent of the hold space. For an uninsulated hold having an equivalent $U_w = 1.3$, calculate the required

subcooling and determine the amount of insulation required to permit transportation of this cargo with 15° subcooling. The ship has a molded beam of 60 ft.

Solution. Substituting in equation 11·24,

$$^{\circ}P = \frac{65 - 35}{\frac{0.8 \times 60 \times 70 \times 0.8}{192 \times 10 \times 1.3} - 0.5} = \frac{30}{1.08 - 0.5} = \frac{30}{0.58} = 52^{\circ} \text{ F}$$

which shows that use of the uninsulated hold would be impractical for this case. Now substituting in equation 11·25 to determine the overall coefficient necessary if °P is to be limited to 15° F,

$$U_w = \frac{0.8 \times 60 \times 70 \times 0.8 \times 15}{192 \times 10[(65 - 35) + (0.5 \times 15)]} = 0.56$$

The requisite coefficient could be attained by placing ½-in. blankets of hair felt, eel grass, kapok, or similar material either as a lining around the hold (against battens, etc.) or as a covering over the stored cargo.

In cases where heat gain of the uninsulated hold does not exceed the permissible value by more than 25 per cent the need for reduction of the overall transmission coefficient can sometimes be met by painting the inside surface of the plates and the underside of the hatch with aluminum paint. Because the principal resistance to heat flow of the shell is represented by the combined (radiation and convection) inside film coefficient, use of a low-emissivity inside surfacing material is very much more effective for hold insulation than it usually is in shore installations. Additional advantages of such paint are that it can be applied with less effort and in less time than can customary insulating materials and it represents neither a possible damage problem (as do quilts) nor a weight and volume problem. Effective use of such materials is limited, however, to reductions of not much more than 25 per cent in the overall uninsulated plate coefficient; if greater insulating effect is necessary, blankets or other bulk insulating material must be used.

11·5. Subcooling Limitations. The most serious limitation to the subcooling analysis is the assumption, implicit in the method, that there is no resistance to the flow of heat into or out of the stored material and that its temperature therefore does not vary as a function of position. For a cargo having high thermal diffusivity and stowed in small packages, separated by means of dunnage, in a way which would permit forced air circulation through the pile, the assumption is justified, and results obtained from equations 11·24 and 11·25 can be used with confidence. Usually, however, forced air circulation within the pile is impractical, and a temperature gradient will therefore develop through the cargo from warm ambient air to cold material at the center of the

pile. Paradoxically, this gradient is likely to be both an advantage and a disadvantage.

The disadvantage arises from the fact that the outer section of cargo undergoes a temperature rise greater than the amount of subcooling and is therefore likely to be damaged. This hazard can be overcome by stowing near the shell plates some other material which either has a higher critical carrying temperature or which need not be refrigerated. By use of such "buffer" cargo the rapid heat gain of the outer sections can be absorbed without hazard to the large central core of perishable cold storage goods.

The advantage of the temperature gradient is that the rapid rise of surface temperature reduces the rate of heat gain and thus greatly reduces the degree of subcooling, based on *mean* cargo temperature, necessary for the voyage. By using buffer material and taking advantage of the lag in temperature rise of the central core it becomes evident that a large part of the cargo can actually be stowed *without* subcooling. This application is possible only when the cargo consists of a dense material stowed closely (preferably without dunnage between vertical layers) so as to restrict circulation through the interstices. When such stowage is possible the hold should be lined with a 2- or 3-foot layer of subcooled buffer material followed by a similar wall of subcooled basic cargo; the entire central section of the hold can then be filled with basic cargo stowed at its carrying temperature. The transient condition resulting from a hold packed in this way will be reflected in an initial temperature drop of the central core as heat flows from this core to the subcooled buffer section, followed at a later time by heat flow back to the core as the temperature of the buffer layer increases.

Exact analysis of the time-space-temperature changes within the cargo is usually impossible owing to lack of homogeneity or the presence of variables (as interstice size and free convection between and around units of the stored material) which are not amenable to precise evaluation. Even in cases where the nature of the cargo permits rigorous treatment (as with bulk shipments) boundary conditions usually are so complicated that a detailed analysis would be both tedious and complex. Fortunately, these difficulties are not of great significance in practice because the actual temperature-time distribution throughout the cargo will have a value between the limits represented by conditions of bulk or separated package stowage. The subcooling is therefore an accurate evaluation of one limiting condition, and from it can be determined the maximum *quantity* of cooling effect which must be stored in the cargo. At the other limit, approximate data are needed on the temperature gradient corresponding to homogeneous solid pack; from

such data the necessary *distribution* of the cooling effect (thickness of buffer layer) can then be calculated.

A reasonably conservative analysis for solid pack conditions is possible if the hold is treated as a wall of infinite thickness, subject to uni-dimensional heat flow and surrounded by an ambient fluid at design outside temperature having an equivalent film coefficient numerically equal to U_w . The use of U_w in place of the film transfer factor is usually justified in ship work because the thermal storage capacity of the uninsulated hold (including such insulating blankets as may be used around the cargo) is so small in comparison with that of the cargo itself as to be negligible. If, for a particular problem, conditions are such that this reasoning is not valid, the problem can then be solved as one of heat flow through a composite wall.¶

Restatement and idealization of the hold problem, as above, permits determination of the time-space-temperature relationship as for a homogeneous semi-infinite solid at uniform temperature and suddenly immersed in an ambient fluid at t_d . The solution of this problem is available in analytical expressions* by use of sets of curves,† or by graphical solution‡ using the Schmidt method of finite differences. All these procedures are treated in detail in the literature on transient heat flow and can be used without alteration in obtaining an approximate boundary condition for the marine subcooling problem. Subcooling of a buffer layer made up of cargo not requiring refrigeration may be economically justifiable since it permits transportation of the main mass of cold storage material with little or no required subcooling.

In many practical cases, loading problems, unavailability of suitable cargo, inadequate control of circulation, difficulty in obtaining or installing adequate insulation, and a multitude of other unforeseeable factors may restrict or prevent use of subcooling systems of the type described in this chapter. In all cases subcooling, even when applicable, is less economical and less satisfactory than would be transportation of cold storage cargo at standard carrying temperatures. But the method does offer attractive possibilities for handling some types of refrigerated cargo under emergency conditions and in transportation equipment not designed for refrigerated service. In such cases the

¶ Raber and Hutchinson, *Heating, Piping, Air Conditioning*, June, 1942.

* Boelter, Cherry, and Johnson, *Heat Transfer Notes*, University of California Press, 1942.

† McAdams, *Heat Transmission*, McGraw-Hill Book Co.; Schack, Goldschmidt, and Partridge, *Industrial Heat Transfer*, John Wiley & Sons, 1933.

‡ Schmidt, *Foppls Festschrift*, Springer (Berlin); Nessi and Nisolle, *Méthodes graphiques pour l'étude des installations de chauffage*, Dunod (Paris); Raber and Hutchinson, *Heating, Piping, Air Conditioning*, May, 1942.

opportunity of carrying more cargo in a greater number of trucks, railroad cars, or ships, and with less specialized mechanical equipment may outweigh all the disadvantages, difficulties, and possibly greater refrigerating costs which use of subcooling may entail.

PROBLEMS

1. A room with all surfaces equally exposed has an overall coefficient of walls of 0.18, a floor overall coefficient of 0.25, and a ceiling overall coefficient of 0.10. The room is 20 ft by 30 ft by 14 ft in height and it has no windows. Determine the equivalent overall coefficient of heat transfer for this room.

2. For the room of Problem 1 calculate the mid-point temperature of the wall, assuming that inside and outside air temperatures are 0°F and 80°F and inside and outside film coefficients are $1.0\text{ Btu}/(\text{hr})(\text{sq ft})(^{\circ}\text{F})$ and $10\text{ Btu}/(\text{hr})(\text{sq ft})(^{\circ}\text{F})$.

3. A cold storage room 20 ft by 20 ft by 10 ft is three-fourths full of a product which has a density of 50 lb/cu ft and a specific heat of $0.8\text{ Btu}/(\text{lb})(^{\circ}\text{F})$. The walls, floor, and ceiling of the room have a combined weight 75,000 lb and a specific heat of 0.3. The equivalent overall coefficient of heat transfer is 0.14. When the outside air temperature remains constant at 80°F , the inside temperature of the room and contents is to be brought down from 80°F to 0°F . (a) Calculate the required constant rate of refrigeration needed to accomplish the required cooling in 16 hr. (b) Determine the required rate of refrigeration needed to maintain the room at the 0°F temperature. (c) Obtain the necessary data and plot cooling time as a function of the refrigerating capacity (assuming that refrigerating capacity does not change with inside temperature).

4. If the refrigeration system in Problem 3 failed, determine the time for the inside air temperature to rise from 0°F to 20°F . Plot the heating curve.

5. For the conditions of Problem 3 determine the refrigeration rate needed to extend the heat-up time to 30 hr for a temperature rise from 0°F to 20°F .

6. As a means of evaluating the requirements for emergency refrigeration equipment plot a curve of refrigerating capacity versus heat-up time for the conditions of Problem 3 and for a temperature rise from 0°F to 20°F .

CHAPTER XII

TRANSIENT LOADS: HEAT PUMP SOURCES AND SINKS

The most important single factor governing the effectiveness of a reverse cycle heating system is the adequacy and availability of the source from which heat is to be extracted. Great care must be exercised in selection of a heat source and the full ingenuity of the designer is often required to provide a type and size of heat transfer surface which will assure continuous transfer throughout all load conditions. For summer operation the same problem occurs in reverse since it is then necessary to have an adequate heat receiver, or sink, with thermally effective connection to the heat pump.

12-1. Steady (Class I) Sources and Sinks. Sources and sinks are divisible into two major classifications, the first including all types in which mass transfer assists in taking heat to or from the region of the transfer surface, whereas the second classification embraces stationary, solid systems in which the entire collection or rejection of heat is determined solely by conduction. Systems which come under the first classification include those using well water, river water, or atmospheric air; in all such systems thermal currents within the fluid equalize the temperature. Hence, in effect, transfer is from a large mass of fluid at practically uniform temperature. In the case of rivers, lakes, or atmospheric air the heat pump has negligible effect on the temperature of the source or sink; hence for such cases transfer surface is designed to operate at steady-state conditions as defined by the load and by the source or sink temperature.

For all types of Class I systems the characteristic design feature is stability with respect to time of the temperature difference between the transfer surface and the source or sink; the thermal capacity of the source or sink determines its temperature change (if any), whereas the load requirements of the structure dictate the necessary transfer rate, hence indirectly determine the temperature of the transfer surface. Once the temperature difference has been evaluated it is subject to change only as a function of load changes, or as a function of the slow change in uniform temperature of the sink or source. Thus if a large reverse cycle system were used to extract heat from a small lake, the mean temperature of the lake might perceptibly decrease, but this

change would occur very slowly and under conditions such that the temperature of water in the vicinity of the evaporator would never become appreciably lower than the mean temperature of the entire water mass.

From the above discussion it appears that Class I sources and sinks are characterized by steady-state transfer and that even in cases where significant temperature change of the main mass does occur the effect of this variation on effectiveness can be very readily anticipated by use of the simple equation for thermal gain:

$$Q = wc(t_2 - t_1) \text{ Btu/cu ft} \quad (12\cdot1)$$

As an example consider a small residential heat pump which is required to extract 50,000 Btu/hr from a mass consisting of 100,000 pounds of water. The specific heat of the water is unity, so it follows directly that the mean temperature of the water will undergo reduction at the rate of $\frac{1}{2}^\circ$ F per hour; at any time after start-up of this system it would be a simple matter to determine the instantaneous water temperature and the resultant necessary value of the temperature difference. For such a case the transfer surface would be designed in accordance with the usual steady-state equation:

$$q = UA(t_s - t_p) \text{ Btu/hr} \quad (12\cdot2)$$

where t_s and t_p are, respectively, the temperatures, at a given time, of the source and the transfer surface. For summer operation the temperature difference in the above equation would be $t_p - t_k$, where t_p is as before and t_k is the sink temperature.

12-2. Transient (Class II) Sources and Sinks. Sources and sinks of the second class offer very much more difficult problems of design because temperature gradients are set up within the mass and it is no longer possible to evaluate transfer to or from the mass in terms of the mean temperature. In such cases the design approach based on steady-state analysis must be abandoned and resort made to the more complex method of transient analysis. Perhaps the greatest danger inherent in Class II systems is the hazard of failure through erroneous design based on the fallacy of steady state. The importance of the problem can be demonstrated from another point of view by noting that with Class I systems the variation in source-to-plate temperature difference—where source temperature does vary—is expressible, at constant load, in terms of the quantity of heat which has been extracted from the source; the temperature difference decreases as a direct function of energy extraction. With Class II systems this relationship no longer

holds, and marked reduction in the temperature difference may occur even though but a small quantity of energy has been extracted. In this case the variation in temperature difference is directly related not to the quantity of energy, but to the rate of heat removal and to the thermal diffusivity of the main mass.

Thus when a mass has low thermal conductivity and high thermal capacity (defined as the product of density and specific heat) the thermal diffusivity (k/wc) will be low and there will be a rapid change of mass temperature adjacent to the transfer plate, but a very slow change in the mean temperature of the mass and a correspondingly long time interval before the influence of the heat extraction is evidenced at any distance from the transfer plate. In effect such a situation leads to rapid development of a flattened temperature gradient adjacent to the transfer surface with consequent falling off of the transfer rate.

Most early heat pump installations used Class I sources and sinks, but around 1945 a trend was established toward use of transfer surface embedded in the earth; since 1945 many such systems have been installed and earth sources are now widely used for small heat pump installations. A substantial number of the earlier Class II installations led to operating difficulties which could be traced back to faulty design based on misuse of the steady-state concept. One common form of difficulty was that of excess condenser pressure due to flattened temperature gradient near the transfer surface.

In many cases earth-embedded surfaces act in part as Class I units due to flow of ground water across the surface or, where such flow does not occur but where the earth is moist or wet, due to mass transfer out through the earth of vapor resulting from evaporation of water from the region near the transfer surface. Any tendency toward Class I operation is, of course, highly desirable, but the cautious designer will nonetheless assure himself that his installed transfer surfaces are of a capacity adequate to handle the load under the closest approach to Class II conditions that can be expected to occur.

Existing data on performance of Class II units are meager and incomplete; tests on this type of surface are costly and time-consuming as they must be carried through a transient period of the same order of magnitude as that which occurs in practice; thus for reverse cycle heating the test period would need to be of the order of 7 months, whereas for reverse cycle cooling the period would approach 4 to 5 months. Many short-term tests have been conducted, but more often than not the results are misleading and cannot safely be extrapolated over the necessary time range. Adding to the complexity of the situation is the fact that in order for a test to have significance for the actual installation

the load applied to the transfer surface would have to vary with time in the same way as the real load on the actual system varies seasonally.

12.3. Idealization of Transient Systems. As approximations to the actual conditions existing in Class II systems various idealizations have been established which can be used as the basis of analytical methods of approach. Thus if a tube or plate were buried in the earth below the level at which seasonal temperature variations are experienced, the early stages of the transient heat flow pattern would correspond to those that would exist in an infinite solid. One mathematically simple idealization is to visualize the transfer surface as a very large flat plate buried so deeply that earth surface effects are not experienced; initially the surrounding earth is assumed to be at uniform temperature and the plate is assumed at some instant of time to be raised or lowered to some other temperature value and subsequently maintained at this new value irrespective of the resultant time variation of rate of heat transfer. Heat would then flow to or from both sides of the plate, but if end effects were neglected (that is, if the plate were very large) the thermal system on each side would correspond to the idealization of a semi-infinite solid of uniform temperature bounded on the end by a plane at which a temperature other than that of the main mass was suddenly impressed. By methods common to mathematical analysis it can be readily shown* that at any time θ hours after the boundary temperature changes, the temperature at any parallel plane x feet in from the boundary will have changed from the initial temperature t_e to an instantaneous temperature t_p in accordance with the equation

$$t_x = t_p + (t_e - t_p)\phi\left(\frac{x}{2} \times \sqrt{\frac{wc}{k\theta}}\right) \quad (12.3)$$

where k , w , and c are, respectively, the thermal conductivity of the earth, the density of the earth, and the specific heat of the earth, all three being assumed to have constant values throughout the region to which the analysis is to be applied; the function ϕ is the Gauss or probability function and is evaluated in standard mathematical references (as for example the Pierce table of integrals).

For the same system Schack derives the equation for the instantaneous rate of heat transfer from one side of the plate θ hours after start of transfer and subject to the assumption that the plate temperature has remained constant throughout this period; the equation is

$$q_p = 0.567\sqrt{wck} \frac{t_e - t_p}{\sqrt{\theta}} \quad (12.4)$$

* Schack, Goldschmidt, and Partridge, *Industrial Heat Transfer*, John Wiley & Sons, 1933.

The average rate of heat transfer from one side of the plate from start-up until θ hours after start-up is given by Schack as

$$q_p' = 2q_p \quad (12.5)$$

Hence the total quantity of heat transferred from one side of the plate during the first θ hours of operation is

$$Q_t = 2\theta q_p \quad (12.6)$$

12.4. Earth-Embedded Surface. A GRAPHICAL SOLUTION. A general graphical solution of equations 12.4 and 12.6 is given in Fig. 12.1. As an indication of its use consider the following example.

Example. A very large flat plate is embedded in earth having a density of 100 pcf, a thermal conductivity of 0.5 Btu/(hr)(sq ft)(°F/ft), and a specific heat of 0.2 Btu/(lb)(°F). The earth is initially at a uniform temperature of 50° F, and the plate is suddenly heated to 85° F and maintained at this temperature for 3 hr. At the end of this time it is desired to determine the instantaneous rate of heat loss from one side of the plate and to evaluate the total quantity of heat which has left during the 3-hr period.

Solution. Enter Fig. 12.1 at the lower left at a density of 100 and rise (see dotted example line) to intersection with the 0.5 conductivity line, then move right to intersection with the 0.2 specific heat line and rise to intersect the curved reference line in the upper left quadrant. From this intersection move left to meet the diagonal reference line, and then drop vertically to the radial line corresponding to a time interval of 3 hr. From here move right to the upper right-hand quadrant, and intersect the radial line for an initial temperature difference between plate and earth of $85 - 50 = 35^\circ$ F, and from this point drop to the instantaneous heat transfer scale and read the rate of heat loss at the end of 3 hr as 36 Btu/(hr)(sq ft) from *one* side of the plate. The transfer rate for the plate as a whole (assuming loss from both sides) is then 72 Btu/(hr)(sq ft). The total quantity of heat lost during this period is now found by dropping from the instantaneous scale down to intersection with the radial line for time equal to 3, then moving right to intersect the total heat scale where the answer is read as 216 Btu/sq ft from one side or 432 Btu/sq ft from both sides of the plate.

In using Fig. 12.1 note that the time lines in the upper left quadrant are valid for either 1 to 80 hr, or 100 to 1000 hr. When the time is less than 100 hr the upper scale for instantaneous transfer rate (upper right quadrant) is used, whereas for time between 100 and 1000 hr the lower scale is used. The radial time lines in the lower right quadrant are valid for any time interval up to 10,000; thus the line marked 6 can be used for 6 hr or 60 hr or 600 or 6000 hr, and the corresponding answer would be determined from the total heat transfer scale with a multiplying factor as shown in the notes below the scale.

In using Fig. 12·1 it must be remembered that the equations on which the graphical solution is based are valid only for earth which is sufficiently dry so that no appreciable thermal influence is experienced from either flow of water or transmission of vapor. Furthermore the equations assume a plate so large that end effects are negligible and a depth of bury so great that earth surface effects are not experienced.

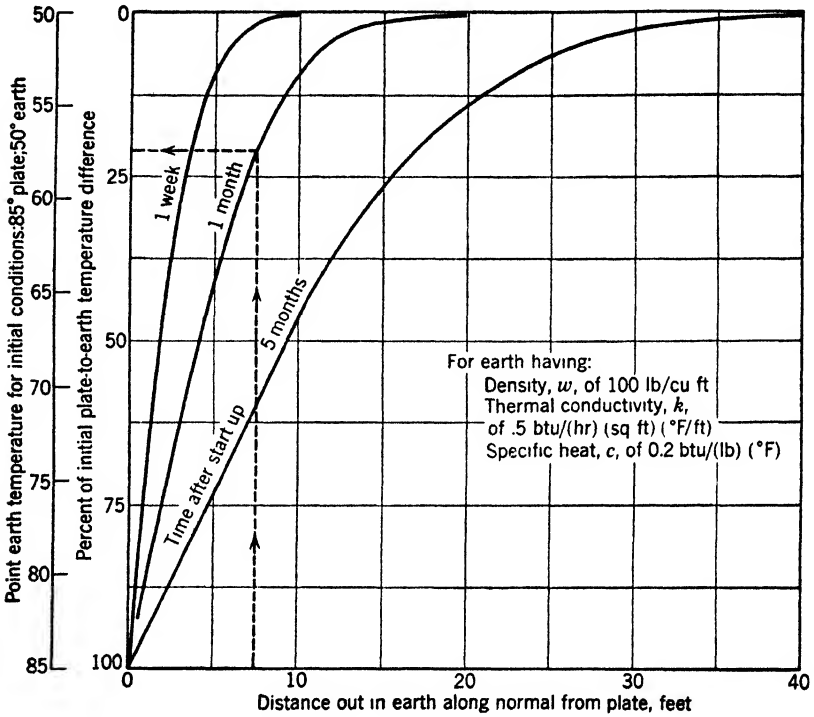


FIG. 12·2. (From Raber, Boester, and Hutchinson, courtesy *Heating, Piping Air Conditioning*.)

In order to show the seriousness of the transient flow problem two additional graphical solutions have been prepared (Figs. 12·2 and 12·3) for a special case. It must be emphasized that these two solutions do not possess the generality of Figure 12·1 but are intended merely to show, for a particular installation, the range of variation in transfer rate and in total energy transfer that must be expected from Class II installations. Figure 12·2 is a solution of equation 12·3 for earth of 100 pcf density, specific heat of 0.2, and thermal conductivity of 0.5 Btu/(hr)(sq ft)(°F/ft). Examination of this figure will show that the influence of the plate temperature is very slow in its movement out

through the earth; after one week the effect of a 35° F plate-to-earth temperature difference is experienced only within 10 ft of the plate, while even at the end of a full 3600-hr cooling season there would be no effect on the earth at a distance greater than 40 ft from the plate.

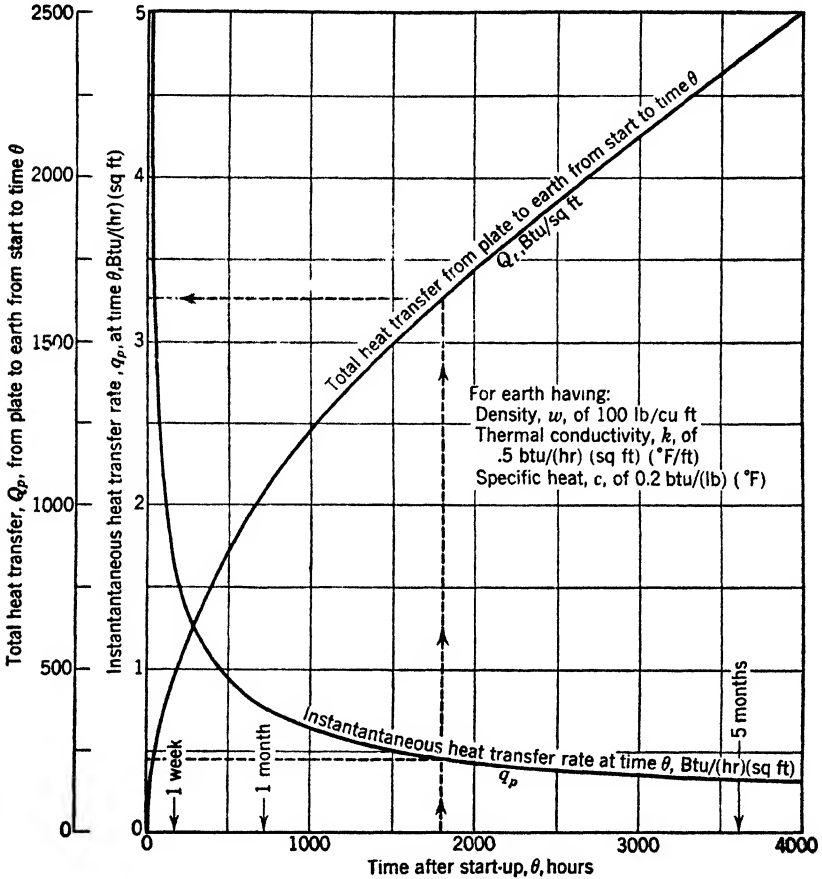


FIG. 12-3. (From Raber, Boester, and Hutchinson, courtesy *Heating, Piping Air Conditioning*.)

Evaluation of quantity of heat transferred can be made from Fig. 12-2 by noting that the area between the ordinate for zero distance and the curve for a given time is proportional to the quantity of heat entering or leaving storage in the earth. Examination of the areas of Fig. 12-2 will show clearly that the quantity of energy going to or from storage between the first and fourth weeks is not appreciably greater than

that entering storage during the first week; hence it is evident that there is a very rapid falling off of the transfer between the earth and the plate.

Figure 12·3, developed for the same earth conditions that were used in Fig. 12·2, shows the rapid falling off of the instantaneous transfer rate q_p as a function of time. Thus for one week of operation the final and the average rates of transfer would be 1.5 and 3.0 Btu/(hr)(sq ft), respectively, whereas over a full 5-month heating season the corresponding values of instantaneous and average transfer rates would have dropped to 0.33 and 0.66 Btu/(hr)(sq ft). This example clearly demonstrates the need for reporting heat pump test results in terms of seasonal performance rather than for brief test periods, since for the case of the example if the area of transfer surface were designed on the basis of one-week performance data it is evident that the surface might constitute less than one-fifth as much area as actually needed for seasonal operation.

In using the average value of the transfer rate for any time period of θ hours, the designer must bear in mind that surface of sufficient extent to give the required total seasonal transfer will give an instantaneous transfer rate at season's end which will be only 50 per cent of the average rate. Since the load on the system normally falls off near the end of the season this fact is not of great importance, but what is vitally significant is that a transfer rate many times that of average must be maintained at the beginning of the season if the total seasonal transfer is to be achieved with the design value of surface area. But at the start of the season, just as at the end, the load on the system is normally low; hence high transfer rates cannot be justified and, in practice, the temperature of the embedded plate (acting as evaporator or condenser) will necessarily be maintained at a value closer to that of the earth than it will be later in the season. For this reason the required surface area for seasonal operation will be greater than the average seasonal transfer rates would indicate. Hence the idealization of a large embedded flat plate is useful in giving a conservative evaluation of overall conditions for seasonal operation.

Another limitation of the flat plate concept is that it assumes unidirectional heat flow. For very large plates this would be a reasonable approximation, but for small plates the end effect will be responsible for a greater transfer rate than that shown by the equations or the graphs. However, departure of the real from the fictitious system is in the direction of greater capacity, so use of the infinite plate concept will give a conservative approximation.

At the expense of a more complex analysis idealizations can be established which more closely approach the conditions usually existing in

actual heat pump installations. For a heated or cooled hollow cylinder immersed in material (as earth) of uniform temperature, Perry and Berggren† developed a method of solution based on the graphical solutions of Nessi and Nisolle‡ which could be applied to a condenser or evaporator tube of a heat pump by considering that the tubular transfer surface was “insulated” with a cylindrical layer of earth. For the more complex case of a tube embedded in the earth, but so close to the surface that at equilibrium there would be an appreciable rate of transfer across the earth’s surface, equations are given in the Boelter-Cherry-Johnson notes,§ which can be used for determination of the equilibrium value of the rate of heat transfer.

At best, however, idealizations of Class II sources and sinks are but approximations and have value only in assisting the designer in establishing boundary values of the rates of heat transfer and conservative values (often unduly conservative) of the required surface area. Most actual installations necessarily involve interference effects between and among the individual tubes of the condenser; hence analysis of transfer rates from a single unobstructed tube would be non-conservative.

12.5. Combined Systems. Because of the difficulty in securing practicable transfer rates in systems using the earth directly as a source or sink many attempts are being made to develop methods of utilizing the thermal storage advantages of the earth’s mass in indirect systems which will have transfer rates more in line with those obtainable from Class I systems. One such method utilizes a series of water-storage tanks buried in the earth. Each tank, in turn, is connected to the heat pump and supplies or receives energy as in Class I until its temperature has undergone a maximum prescribed change, at which time it is disconnected from the heat pump and allowed to “regenerate” by transfer, over a longer time period, to or from the earth. Thus each tank operates as a Class I unit when connected to the pump, but as a Class II unit during the disconnected regenerative period.

Another system consists in burying the transfer surface in a “prepared” earth fill consisting of a special clay which retains a substantial amount of bound water and has a correspondingly high thermal diffusivity.

Many systems have been developed which use well water as a second-

† Perry and Berggren, “Transient Heat Conduction in Hollow Cylinders after Sudden Change of Inter-Surface Temperature,” *Univ. Calif. (Berkeley) Publ. Eng.*, Vol. 5, No. 3, pp. 59–88, 1944.

‡ Nessi and Nisolle, “Méthodes graphiques pour l’étude des installations de chauffage et de réfrigération en régime discontinu,” Dunod (Paris), 1929.

§ Boelter, Cherry, and Johnson, *Heat Transmission Notes*, University of California Press, 1948.

ary fluid, but except in localities where flow occurs across the region in which the well is located the basic problem of heat transfer by conduction through the soil adjacent to the well casing will remain one of paramount importance. In terms of elementary heat transfer the problem is one of overcoming the thermal resistance of the earth and of achieving the rapid diffusion, specifically by conduction, of heat through the earth mass adjacent to the transfer surface; whether the transfer surface is between earth and water or between earth and metal plate the problem is fundamentally the same.

12-6. Typical Installations. An interesting earth-embedded heat pump installation was designed and installed by Carl F. Boester in a small residence in Lafayette, Indiana. The reverse cycle is alone of sufficient capacity to handle both maximum heating and cooling loads, but to permit an engineering survey of the relative effectiveness of various types of transfer surfaces a number of full-capacity entirely independent evaporator and condenser surfaces were installed. Thus on one operating scheme a network of coils was embedded around and just outside of the foundations of the house; with this arrangement it was expected that heat lost from the basement of the house would be captured as it traveled out through the earth and pumped back into the structure. A second independent transfer surface was located in a septic tank where it was expected to pick up heat liberated by waste matter within the tank during the process of oxygenation. This installation successfully completed its first full year of operation during 1949.

The largest heat pump installation in the United States (as of 1948) was located in the Equitable Building in Portland, Oregon. This building consists of 12 rental stores with basement and two penthouse levels; it is 200 by 100 ft at ground level reducing to 200 by 62 ft at the third floor and above. Total floor space is 212,000 sq ft and volume 2,275,000 cu ft. A minimum of approximately 80 tons of refrigeration are required for parts of the structure even during the coldest (10° F) weather, and the designers have made excellent use of this need by realizing low-cost, by-product heating from the condenser discharge for minimum cooling. Thus, in effect, an 80-ton heat pump operates throughout the winter to provide necessary cooling from its evaporator and simultaneously to provide necessary heating from its condenser. Needless to say, such operation is highly desirable and serves to more than double the coefficient of performance that would be attained with cooling only. Since year-around cooling is a requisite in many internal spaces of commercial and industrial buildings it follows that the idea of combining the cooling load of one zone with the heating load of another

zone affords unusually attractive possibilities for achieving great economy of heat pump operation.

The total heat pump capacity in the Equitable Building is 540 tons made up of two 200-ton units and two 70-ton units. Maximum load occurs during the cooling season and is so much greater than the maximum heating needs that the heat pump, unlike most commercial installations, is designed to carry the full heating load without any auxiliary heat supply. The heat source for the system is well water which is pumped from two wells of 150-ft depth. One well supplies 195 gpm of $64\frac{1}{2}^{\circ}$ F water, whereas the other supplies 450 gpm of $62\frac{1}{2}^{\circ}$ F water; disposal is to a 510-ft well with 57° F water. The installed cost of the system has been given|| as \$0.29 per cubic foot or approximately \$1250 per rated ton.

A 15-ton heat pump has been in use for many years¶ in the Westinghouse Building in Emeryville, California. This is an air-to-air system, two streams of outside air being directed over the evaporator and the condenser, respectively, and one of the streams then being distributed through the working space, whereas the other is discharged outside. The change-over from summer to winter conditions is accomplished by adjustment of dampers which determine which of the two air streams is to be admitted to the building. Outside air conditions in Emeryville during the winter are mild; hence the coefficient of performance for the system is higher than for most winter localities (cop of approximately 5). When the outside air temperature approaches freezing, a preheating coil, located ahead of the evaporator, automatically goes into operation with city water, at approximately 60° F, as the heating medium; this precludes the possibility of frosting the evaporator transfer surfaces.

Detailed descriptions of residential* and of commercial† heat pump installations are available in the literature, and a comprehensive bibliography of some 215 references up to 1948 is also available.*

12·7. Summary. Reverse cycle heating systems use a small quantity of energy possessing a high degree of availability to raise the temperature level of a large quantity of energy which has low availability. When electrical energy (or any other energy possessing the availability of shaft work) is used to operate the pump it is frequently possible to deliver from 3 to 10 units of heat energy for each unit of

|| Kroeker and Chewning, "A Heat Pump in an Office Building," *Heating, Piping, Air Conditioning, ASHVE Journal Section*, March, 1948.

¶ "Heat Pump Serves New Coast Building," *Power*, April, 1939.

* Penrod, "The Development of the Heat Pump," *Univ. Kentucky Eng. Exp. Sta. Bull.*, Vol. 1, No. 4, June, 1947.

† *A Review of Commercial Heat Pump Installations for Southeastern Electric Exchange*, Southern Research Institute, Birmingham, Alabama, October 15, 1946.

required work. This does not necessarily mean, however, that the heat pump would result in an overall saving in fuel since if the electrical energy is supplied from a steam power plant it will require more energy units of fuel supply to the combustion chamber than will be obtained from the generator. Penrod* estimates that a heat pump having a coefficient of performance of 3.33 will deliver approximately as much heat per pound of coal consumed at the central steam plant as would a stoker-fired furnace for each pound of coal consumed in conventional heating of the structure.

PROBLEMS

1. A small residence has thermal characteristics such that the average heat loss amounts to 40,000 Btu/hr for the 5-month heating season. This structure is to be heated by means of a heat pump which extracts energy from a 20-ft by 20-ft by 20-ft insulated and buried tank of water. (a) If the water temperature is 60° F at the start of the heating season what will the temperature be at the season's end? (b) Determine the coefficient of performance of this heat pump (assuming Carnot cycle) at the beginning of the heating season and compare this value with the cop at the end of the season. (Assume that the condenser operates at 110° F.)

2. For the conditions of Problem 1, but with earth as the heat source ($w = 40$ lb/cu ft, and $c = 0.30$), assume that the earth is so wet that the temperature remains uniform throughout the mass and determine how deep the required earth core would be if its horizontal cross section is 20 ft by 20 ft.

3. Earth under a large concrete slab is at a uniform temperature of 40° F. If the slab temperature were rapidly (assume instantaneously) raised to 100° F determine (a) the temperature 5 ft below the slab after 6 hr, (b) the rate of heat flow into the earth after 6 hr, (c) the total heat transferred to the earth in the first 6 hr.

* Penrod, "The Development of the Heat Pump," *Univ. Kentucky Eng. Exp. Sta. Bull.*, Vol. 1, No. 4, June, 1947.

CHAPTER XIII

RADIATION; PANEL COOLING

The transfer of energy by radiation differs from other mechanisms of heat transmission in that it does not require the presence of a tangible medium; radiant energy moves from place to place at the speed of light (186,000 miles per second) and without aid of a working substance. In refrigeration, radiant transfer finds only limited direct application in cooling problems, but the radiant effect must be evaluated in many poorly insulated storage spaces or in other locations where a cooled object immersed in cold air may "see" surfaces at a higher temperature.

13-1. Radiation Effect on Equilibrium Air Temperature. In low-temperature work, radiation effects may be very serious. Consider, for example, a small insulated box cooled to an inside air temperature of approximately -110° F by means of solid carbon dioxide. If the outside air temperature is 70° the temperature of the inside surface of the box (assuming 4 in. of cork insulation) will be of the order of -102° F and the resultant equilibrium temperature of a small object placed within this box will be of the order of 5° greater than the inside air temperature.

Radiant cooling is likewise important in agricultural operations since the equivalent temperature of a clear night sky is of the order of -60° F and the energy loss by nocturnal radiation of growing fruits and vegetables may be responsible for frost effects at times when the actual air temperature is higher than 32° F. Thus a bunched head of lettuce, insulated from the ground by a heavy, high-density stem, may lose heat to a clear night sky at a rate sufficient to depress its equilibrium temperature substantially below the air temperature; for an air temperature only slightly above 32° the lettuce might partially freeze.

Of more direct concern to refrigeration engineers is the basic concept that effective cold storage of any object requires that the equilibrium temperature of the object be below the required maximum rather than that the air temperature of the storage space be held at a predetermined value. For a poorly insulated storage space the equilibrium temperature will necessarily vary as a function of outside conditions since load will determine, for a fixed inside air temperature, the actual temperature of the internal surfaces of the storage space. As load increases, the equilibrium temperature will likewise increase, so that a product maintained

safely in air at 10° F under conditions of light load might commence to spoil at the same air temperature under conditions of greater load. Consider, for example, a showcase which is shown in simplified cross section in Fig. 13·1. An object, as a piece of meat, is placed on the bottom of the case and is in air at temperature t , which is considered the maximum temperature at which the meat can be safely stored. The glass window reaches an inside surface temperature of, say, Δt° above t , so by inspection of the symmetry of the system it is evident that the

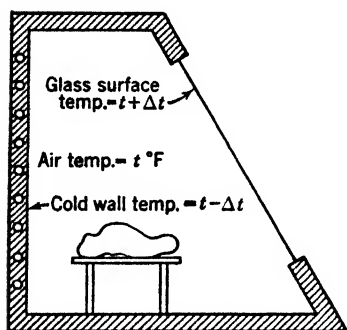


FIG. 13·1.

radiant gain due to the warmer window must be offset by an equivalent radiant loss to the cooled surfaces which make up the other half of the showcase, and which must therefore be maintained at a surface temperature of not more than $t - \Delta t$. The actual temperature of operation of the cold surface might be greater or less than the calculated $(t - \Delta t)$ since the air temperature inside the showcase is itself determined by the cold surface temperature; hence exact solution of the problem of maintaining an equivalent temperature t would require

solving a set of heat balance equations to determine both the air temperature and the cold surface temperature. For a showcase in which the warm and cold surfaces are not of equal area and are not symmetrically spaced, the analysis of radiation effects becomes more complicated; to permit solutions of problems of this kind some of the basic radiation equations will be developed.

13·2. Basic Radiation Equations. The emissive power E of any surface is defined as the rate of energy emission by radiation expressed in Btu per hour per square foot of surface. For all surfaces used in modern refrigeration applications the radiant characteristic can be taken as that of a thermally "gray" body; that is, the energy emitted at a particular wave length varies with wave length on a spectro-radiometric curve similar to that for a thermally "black" body (a perfect radiator), but with a monochromatic intensity that is a fixed fraction of the corresponding intensity of a black body. This gray characteristic of practical surfaces is a fortunate one and permits evaluation of the emissive power by the equation

$$E = 0.172e \left(\frac{T}{100} \right)^4 \text{ Btu/(hr)(sq ft)} \quad (13\cdot1)$$

in which the coefficient 0.172 is the Stefan-Boltzmann constant and the coefficient e is defined as the emissivity of the body or the ratio of its emissive power to the emissive power of a black body.

The energy leaving any flat area A by radiation during unit time is then

$$q = EA \quad \text{Btu/(hr)} \quad (13.2)$$

and for all surfaces important in refrigeration applications the emission can be considered to occur diffusely and with a directional intensity (per unit solid angle) which is constant when based on the projected area of A in a plane normal to the direction in question; the numerical value of the intensity (based on projected area) can be shown to be E/π . Radiant transfer between two surfaces depends on their positions with respect to one another and on their emissivities. It can be shown that the emissivity of any surface is numerically equal to its absorptivity (for radiant energy); hence for surfaces of the usual type which do not transmit radiation (glass being the only exception) the energy which is not absorbed must be reflected, and the reflectivity of such a surface will necessarily be equal to its emissivity subtracted from unity. For surfaces having a high reflectivity, as polished aluminum or copper, the reflection characteristics may appreciably alter the exchange rate between surfaces since the reflected energy will be partially returned from receiver to source, re-reflected, and continued in motion until complete absorption has finally occurred. Thus in a room completely finished with a particular kind of reflective surface the emissive power of the surface will be lower than it would be for a thermally black finish, but the radiant flux existing in the room will be exactly the same as though the room were finished in black. If this were not the case, net radiant transfer could be realized from a thermally black body to a non-black body at the same temperature, or, as a boundary case, to a non-black body at a slightly higher temperature. Such a condition is contrary to the second law of thermodynamics, and all experience shows that it will not occur.

Reflective surfaces do not play an important part in refrigeration or cold storage systems (for which usual surfaces have emissivities in excess of 90 per cent), and the complexity of analysis resulting from reflective energy transfers need not be given detailed consideration here. References are available* from which equations can be obtained, if needed, for solving problems in reflective systems. The entire problem

* McAdams, *Heat Transmission*, McGraw-Hill Book Co., Chapter III, 1942; Raber and Hutchinson, *Panel Heating and Cooling Analysis*, John Wiley & Sons, Chapter V, 1947.

can therefore be greatly simplified for practical refrigeration purposes by noting that the emissivity term F_e for exchange between any two surfaces having emissivities greater than 90 per cent can be taken as equal to the product of emissivities for the two surfaces ($F_e = e_1 \times e_2$) except in the case of a small body entirely enclosed, in which case the emissivity of the enclosure can be neglected and the emissivity term taken as equal to the emissivity of the small body. Table 13·1 gives values of the emissivity of various representative materials.

TABLE 13·1
EMISSIVITIES OF TYPICAL BUILDING MATERIALS IN TEMPERATURE
RANGE FROM 0° F TO 120° F

Material	e (average)
Clean metal plate (aluminum, copper, brass), average.....	0.10
Galvanized sheet iron (average condition).....	0.25
Aluminum paint.....	0.50
Glass.....	0.80
Linoleum.....	0.85
Stone (average).....	0.87
Wood (unpolished).....	0.89
Brick.....	0.90
Paper.....	0.92
Plaster.....	0.95
Asbestos board.....	0.96

13·3. Net Radiant Transfer. Then for the general case the radiant exchange between two surfaces can be evaluated from the equation

$$q = 0.172F_e F_{A_2 A_1} A_1 \left[\left(\frac{T_1}{100} \right)^4 - \left(\frac{T_2}{100} \right)^4 \right] \quad (13 \cdot 3)$$

in which the F_A term is defined as the shape factor (sometimes called angle factor or geometry factor) of surface A_2 with respect to energy received from A_1 ; this term is numerically equal to the fraction of energy striking A_2 of that emitted hemispherically by A_1 . If the intensity of radiation leaving a surface were uniform the shape factor would equal the solid angle subtended by the receiver divided by the solid angle equivalent of a hemisphere, but since the intensity is uniform only with respect to the projected area of the source it follows that the shape factor may be either greater or less than the ratio of solid angles. Figure 13·2 is a graphical solution of equation 13·3; use of the graph is shown by the dotted example line.

Values of the F term are given in the literature for the common cases

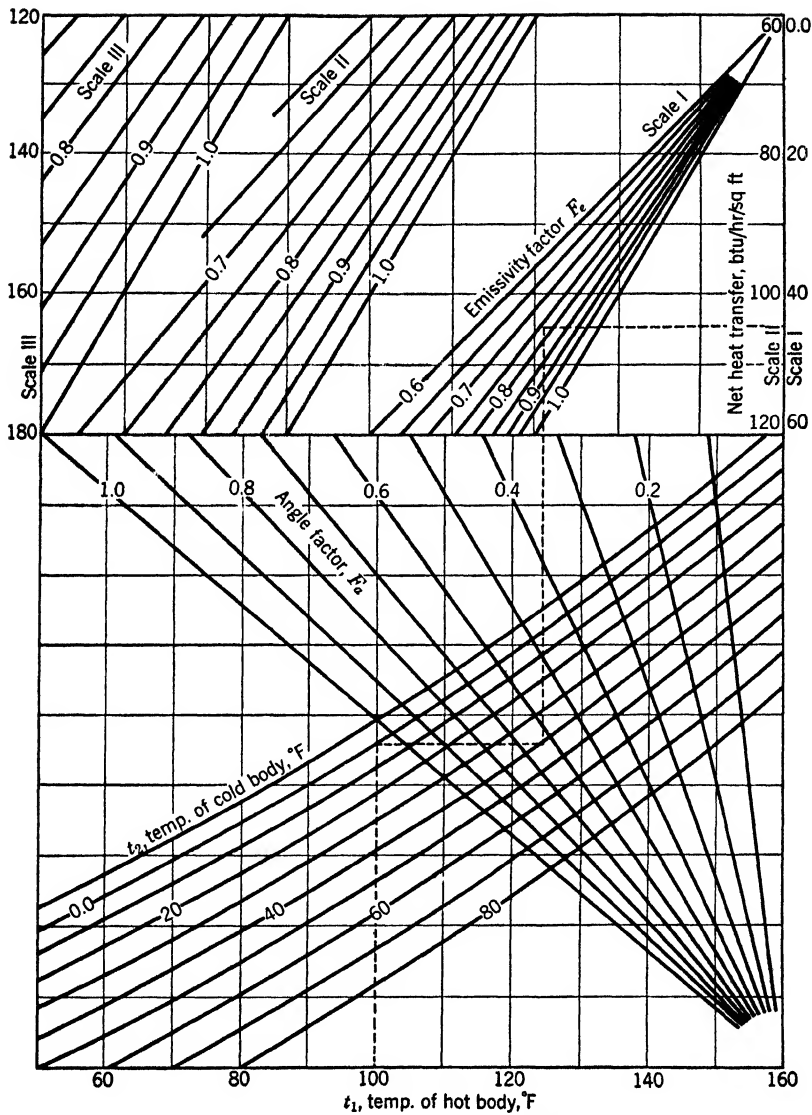


FIG. 13-2. (From Raber and Hutchinson, courtesy *Heating, Piping Air Conditioning*.)

of two surfaces in parallel or in normal planes, but for unusual configurations the shape factor must be determined either by integration (often a difficult task) or by use of some approximation. One convenient simplified method of evaluating F for surfaces that are small in area

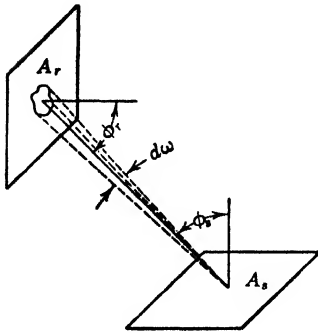


FIG. 13·3.

relative to their separating distance is to make use of the fact that the shape factor varies inversely as the square of the distance between the two surfaces and directly as the product of the cosines between normals to the surfaces and the line connecting them, this product to be divided by π . For surfaces of finite area the separating distance will obviously vary as the points on the two surfaces which it connects are allowed to move; so also will the two angles vary, but when the areas are small with respect to separating distance the variation will be negli-

gible, and a rough approximation to an average value for both the distance and the angles can be readily selected. In such cases an approximate shape factor can be evaluated from the equation

$$F_a = \left(\frac{\cos \phi_s'}{\pi} \right) (\cos \phi_r') \frac{A_r}{r^2} \tag{13·4}$$

where ϕ_s' and ϕ_r' are average values of the angles ϕ_s and ϕ_r which are shown on Fig. 13·3. For the special case of two areas in parallel planes the angles ϕ_s' and ϕ_r' are equal, and equation 13·4 simplifies to the form

$$F_{a(\text{parallel planes})} = \frac{\cos^2 \phi}{\pi} \frac{A_r}{r^2} = K_1 \frac{A_r}{r^2} \tag{13·5}$$

Values of K_1 are given in Table 13·2. For the special case of two areas in normal planes the sum of angles ϕ_s' and ϕ_r' must equal 90° ; hence equation 13·4 simplifies to

$$F_{a(\text{normal planes})} = \frac{\cos \phi [\cos (90 - \phi)]}{\pi} \frac{A_r}{r^2} = K_2 \frac{A_r}{r^2} \tag{13·6}$$

Values of K_2 are given in Table 13·2.

Example. Determine the net transfer by radiation from a small heating panel (plaster surface, operating at 100°F) located in a wall directly opposite an 8-sq-ft window (inside surface temperature 0°F) in a room 20 ft wide.

Solution. Since the areas are small with respect to the separating distance

the angle ϕ will be close to 0° . Entering Table 13-2 at $\phi = 0$, read $K_1 = 0.3182$; substitute this value in equation 13-5 to obtain

$$F_a = 0.3182 \times \frac{8}{20^2} = 0.00637$$

From Table 13-1 the emissivities of plaster and of glass are, respectively, 0.95 and 0.80; therefore $F_e = 0.95 \times 0.80 = 0.76$. Then, by equation 13-3,

$$q = 0.172 \times 0.76 \times 0.00637 \left[\left(\frac{460 + 100}{100} \right)^4 - \left(\frac{460 + 0}{100} \right)^4 \right]$$

$$= 0.44 \text{ Btu/(hr)(sq ft)}$$

[Rather than use equation 13-3, the problem can be solved from the graph of Fig. 13-2. The method of solution is shown by the dotted line; since the actual value of F_a is $\frac{1}{100}$ of the graph value the actual answer is $\frac{1}{100}$ of the answer given on the graph, or $0.01 \times 44 = 0.44 \text{ Btu/(hr)(sq ft)}$.]

TABLE 13-2

ϕ	K_1	K_2
0	0.3182	0.0000
5	0.3170	0.0276
10	0.3080	0.0545
15	0.2960	0.0797
20	0.2810	0.1020
25	0.2610	0.1220
30	0.2380	0.1375
35	0.2140	0.1495
40	0.1865	0.1568
45	0.1590	0.1590
50	0.1395	0.1568
55	0.1047	0.1495
60	0.0795	0.1375
65	0.0570	0.1220
70	0.0372	0.1020
75	0.0213	0.0797
80	0.0096	0.0545
85	0.0024	0.0276
90	0.0000	0.0000

13-4. Equivalent Radiation Coefficient. Most refrigeration problems are of a form involving radiant transfer between a small body and a surround (as in the case of a stored object in a cold storage compart-

ment), hence can be evaluated by the simple equation

$$q = 0.172e_1 \left[\left(\frac{T_1}{100} \right)^4 - \left(\frac{T_2}{100} \right)^4 \right] A_1 \quad (13.7)$$

with the shape factor equal to unity. Further simplification can be obtained by noting that the above equation takes the form of the equation for convective heat transfer if the radiation effect is expressed in terms of a first-power temperature differential:

$$h_r = \frac{0.172 \left[\left(\frac{T_1}{100} \right)^4 - \left(\frac{T_2}{100} \right)^4 \right]}{t_1 - t_2} \quad (13.8)$$

Equation 13.7 then simplifies to the form

$$q = e_1 h_r A_1 (t_1 - t_2) \quad (13.9)$$

For radiant exchange between surfaces at temperatures of approximately 70° F, the value of h_r is approximately equal to unity. For transfer between two surfaces at approximately 32° F, h_r is 0.85. For any special problem in the solution of which accuracy is essential, resort may be made to equation 13.8, but for most practical work in refrigeration an approximate h_r value can be obtained by interpolation or extrapolation from the values for 70° F and 32° F.

13.5. The Heat Balance. The great majority of heat transfer problems that occur in refrigerating engineering have to do with the determination of equilibrium conditions. Heat gains through the walls of a cold storage chamber, equilibrium temperatures of stored materials exposed to surfaces at temperatures other than that of the ambient air, heat transfer to or from the refrigerant—all of these focus interest on the system after it has reached a steady state, that is, has come to equilibrium and exists under conditions such that the temperature at any point in the system does not change with time. For all such problems, solutions are obtained by using the thermodynamic general energy equation with all storage and work terms deleted, leaving only the simple statement that the rate of energy transfer to the system as heat must be equal to the rate of transfer from the system as heat; thus

$$Q_{in} = Q_{out} \quad (13.10)$$

Heat transfer can occur by the mechanisms of conduction, convection, and radiation; hence an explicit statement of equation 13.10 would be

$$(q_c + q_v + q_r)_{in} = (q_c + q_v + q_r)_{out} \quad (13.11)$$

From a knowledge of existing thermal conditions on both sides of the

system in question sufficient data are usually available to permit determination from equation 13·11 of the temperature at any point in the system (as a wall) through which heat is flowing. For transient and periodic heat transfer, equation 13·11 must be rewritten to include a storage term, and for such cases the practical solution of the equation becomes of much greater difficulty. Exact analysis of transient problems is often very difficult, and a detailed mathematical treatment of such cases is beyond the scope of this text. For most practical problems, however, approximate methods which give conservative answers (see Chapter XI) are entirely adequate since the refrigeration engineer can rarely afford to operate close to the limiting temperature conditions, but must allow a factor of safety sufficiently great to assure successful operation of the plant under unforeseeable adverse conditions.

Example. A food substance is packed in cases and kept under cold storage at a temperature of 0° F. Transport from the cold storage warehouse to refrigerated vessels must occur by railroad flat car through air at a temperature of 80° (summer semi-tropical conditions) on clear nights. There is no wind, but the train moves at a speed of 15 mph. Calculate the rate of heat gain at the top surface of the packing cases at the start of the trip: (1) if they are painted black; (2) if they are painted with aluminum paint with a reflectivity of 0.5.

Solution. At the start of the voyage the top of the packing cases will be receiving energy by convection from the air and losing it by radiation to the clear night sky. For unit area of a case at a temperature of 0° F the gain by convection will be (since the film coefficient corresponding to a 15-mph wind is 6.0)

$$q_{in} = 6(80 - 0) = 480 \text{ Btu/hr}$$

whereas the loss by radiation (based on an equivalent sky temperature of -60° F) will be given by equation 13·7:

$$q_{out} = 0.172 \times 1.0 \left[\left(\frac{0 + 460}{100} \right)^4 - \left(\frac{-60 + 460}{100} \right)^4 \right] = 31 \text{ Btu/(hr)(sq ft)}$$

giving a net gain of 449 Btu/hr.

If, on the other hand, the tops of the boxes were painted with aluminum paint the convection gain would be unchanged, but the radiant loss to the night sky would be reduced to $0.5 \times 31 = 17.5$, giving a net gain of 462 Btu/(hr)(sq ft) and indicating very clearly that the tops of the storage cases should be painted black rather than with a reflective paint. This conclusion is in direct and startling contrast to that which would be realized if the transfer were to take place by day rather than by night, since in the former case the radiant effect would be a gain due to solar and diffuse sky radiation and the use of a reflective paint would serve to minimize it.

13·6. Panel Cooling; Comfort Equation. The maintenance of comfort conditions within an occupied space during warm weather can

be achieved by control of environment rather than by control merely of the inside air temperature. The most common method of environmental control involves use of large cooled flat room surfaces and is commonly known as panel cooling. Systems of this type are based on the comfort equation, which states that an average person will experience comfort in any room in which the sum of the air temperature t_a and average surface temperature ast (more specifically, the mean radiant temperature) is equal to the same value as the sum of t_a and ast for conditions under which comfort is realized when these two temperatures are equal to one another. Thus for average winter conditions a sensation of comfort is experienced by an average adult in a room with air and surfaces at 70° F; hence the winter comfort equation is of the form

$$t_a + ast = 140 \quad (13 \cdot 12)$$

Owing to the effects of acclimatization and of shock it would be undesirable to maintain summer conditions at the same base values. Agreement has not yet been realized as to what constitute optimum summer inside conditions, but many designers now base warm weather designs on maintenance of an inside environment equivalent to that represented by 76° F air and 76° F surfaces. This would correspond to an environment defined by the equation

$$t_a + ast = 152 \quad (13 \cdot 13)$$

In order to establish environmental conditions meeting the requirements of equation 13·13 it is necessary to cool large surface areas of the occupied enclosure. Because of the hazard of condensation, particularly in localities where the outside humidity is high, it is also necessary to limit the minimum panel surface temperature to a value substantially in excess of the maximum outside dew-point temperature. If the panel surface temperature were at any time to be lower than the dew point it is immediately evident that there would be every opportunity for an indoor rainstorm with disastrous results to both the occupants and the furnishings of the rooms. Serious though this hazard undoubtedly is, it is not of practical significance if adequate controls are installed and if the thermal capacity of the panels is sufficiently low to assure their ability to respond quickly to sudden or rapid load changes.

13·7. Panel Location. In most cases panel cooling is used as an adjunct to panel heating since where heating is already installed the cooling can use the distribution units of the heating system with the addition of a source of cooled water. Thus the designer usually does not have control over the location of the panels but must utilize those

which were installed for purposes of winter operation. A simple analysis, however, will show the relative merits of the various positions.

Assume, for example, that panels have been installed for ceiling operation during the heating season and that the area of panel is sufficient to maintain comfort with a maximum surface temperature of 120°F when the outside air temperature reaches the design minimum of 0°F . Under these conditions the convective output of the ceiling panel can be taken as $0.4\text{ Btu}/(\text{hr})(\text{sq ft})(^{\circ}\text{F})$ and the radiant output as $1.0\text{ Btu}/(\text{hr})(\text{sq ft})(^{\circ}\text{F})$, or the total energy dissipation rate as $1.4\text{ Btu}/(\text{hr})(\text{sq ft})(^{\circ}\text{F})$ or $1.4 \times (120 - 70) = 70\text{ Btu}/(\text{hr})(\text{sq ft})$. Now suppose that the same panel is to operate during the summer with outside design air temperature of 100°F and equivalent inside air temperature of 76°F . Under these conditions the convective reception of the ceiling surface would be $1.1\text{ Btu}/(\text{hr})(\text{sq ft})(^{\circ}\text{F})$ and the radiant reception would remain $1.0\text{ Btu}/(\text{hr})(\text{sq ft})(^{\circ}\text{F})$, so the total energy reception would be $2.1\text{ Btu}/(\text{hr})(\text{sq ft})(^{\circ}\text{F})$. The ratio of summer transmission load to winter transmission load is approximately as the ratio of inside-to-outside temperature differences, or $(100 - 76)/(70 - 0) = 24/70$; therefore the required summer load on the panel is $(24/70) \times 70 = 24\text{ Btu}/\text{hr}$, and the temperature difference from room air to panel surface must therefore be a maximum of $24/2.1 = 11.4^{\circ}\text{F}$. (This calculation is only an approximation because solar and internal loads would change the ratio.) Thus in a locality where a 120°F ceiling panel would maintain comfort during 0° winter weather the same panel in summer would permit realization of environmental conditions approximately equivalent to those represented by 76°F air and 76°F surfaces if the minimum panel surface temperature during 100°F weather were 64.6°F .

Thus it is evident that the use of ceiling-type panels for summer operation is particularly advantageous because maximum energy reception is realized and consequently only a minimum temperature depression of panel surface below room air needs to be maintained. In the case of wall panels the coefficient for reception of energy in summer is exactly equal to the coefficient for dissipation of energy during the winter; hence if heating panels are designed to operate at 120°F (a 50°F differential from room temperature) the same panel will necessarily operate at a correspondingly large temperature difference (or a correspondingly low absolute surface temperature) during the summer. Thus, if 120°F panel temperature were adequate for 0°F outside air (and 70°F equivalent inside conditions), the surface temperature required for 104°F outside air (with 76°F equivalent inside conditions) would be $[(104 - 76)/70]50 = 20^{\circ}\text{F}$ below inside conditions, or 56°F .

Obviously this temperature is too low to be practicable, because of the condensation hazard, in any but unusually dry climates.

The situation with respect to floor-type cooling panels is somewhat less unfavorable. During the heating season a floor panel dissipates 1.1 Btu/(hr)(sq ft)(°F) by convection and 1.0 Btu/(hr)(sq ft)(°F) by convection, or a total of 2.1 Btu/(hr)(sq ft)(°F). For summer operation the convection reception of a floor panel drops off to that of convection dissipation from a ceiling-type heating panel, or 0.4 Btu/(hr)(sq ft)(°F) with total reception of 1.4 Btu/(hr)(sq ft)(°F). Since the maximum surface temperature of a floor panel is 85° F the winter total dissipation is $(85 - 70) \times 2.1 = 31.5$ Btu/(hr)(sq ft). If the panel area is such that this dissipation rate is sufficient to provide inside comfort during 0° weather the required summer reception rate during 104° F weather (based on 76° F inside) would be $(28/70) \times 31.5 = 12.6$ Btu/(hr)(sq ft), necessitating a room air-to-panel surface temperature of $12.6/1.4 = 9^\circ$ F. In general, however, cooled floors are less desirable than cooled ceilings; hence the ceiling-type installation is almost always to be preferred.

13-8. Performance Characteristics. In one major respect a radiant panel cooling system differs from convection cooling; with the latter type of system the influence of warm walls will always act to make the existing inside air temperature seem warmer than it actually is, or, to put it differently, the inside air temperature corresponding to a given sensation of comfort will necessarily be lower than would be needed if walls were at the same temperature as the air. With panel cooling the opposite condition exists since the average inside surface temperature is less than the air temperature; hence the optimum inside air temperature corresponding to a given sensation of warmth will be lower than would be the case if the walls were at the same temperature as the air. Thus when panel cooling is used a comparable degree of comfort is realized with a lowered air temperature; hence the effect of shock is reduced and the cooling load represented by cooling of ventilation air is lessened. For this reason the operating cost of a panel cooling system can be expected to be somewhat less than that of an equally effective convective cooling system, though the magnitude of the saving will vary with ventilation rate and will usually not be of practical significance in structures which are ventilated only by infiltration.

13-9. Rational Design Procedure. In view of the fact that the optimum inside air temperature corresponding to any selected basis of inside comfort varies with the particular installation, it follows that arbitrary selection of a design value of the inside air temperature would not lead to a satisfactory design. Rather than selecting the inside air

temperature a more logical procedure is to write heat balance equations on a section of uncooled surface within the cooled space, on a unit volume of ventilation air as it passes through the space, and on an occupant. But most rooms are not homogeneous, hence many types of uncooled surface exist. To evade this difficulty it is convenient to replace the actual room with an ideal room which is homogeneous and for which the calculated heat gain by transmission is the same as the transmission gains of the actual room. The overall coefficient of heat transfer of the equivalent room will be defined as U_e by the equation

$$U_e = \frac{U_f A_f + U_g A_g + U_w A_w + U_c A_c}{A_f + A_g + A_w + A_c + A_i} \quad (13 \cdot 14)$$

where subscripts f, g, w, c, i refer to floor, glass, exterior wall, ceiling, and interior partition. The equivalent conductance of the uncooled room, based on transfer of energy from the outside air through to the inside surface, is then

$$C_e = \frac{1}{\frac{1}{U_e} - \frac{1}{h_i}} \quad (13 \cdot 15)$$

where h_i is the inside film coefficient of heat transfer and is usually taken equal to 1.65 Btu/(hr)(sq ft)(°F).

Now, letting u equal the fraction of total room surface that is cooled and $v = 1 - u$ equal the uncooled fraction, a heat balance can be set up on the uncooled surface by equating the net gain of energy from the outside air to the loss of energy by radiation to the cooled surface and by convection to the room air; thus

$$C_e v (t_o - t_e) = h_r u (t_e - t_p) + h_c v (t_e - t_a) \quad (13 \cdot 16)$$

where the subscripts e, o, p, a represent equivalent uncooled room surface, outside air, cooling panel, and inside air, respectively, and the term h_c indicates the film coefficient for convective transfer from the uncooled room surface to the room air. In average practice the value of h_c can be taken with adequate engineering accuracy as 0.8, whereas the values of h_r will be approximately 1.0 for the average room and panel temperatures.

Examination of equation 13·16 shows that $t_o, t_p,$ and C_e might be considered arbitrarily selected design constants, in which case the panel area (indicated by the fraction u), the room air temperature, and the inside surface temperature t_e would all be unknown. Under other circumstances the panel area might be arbitrarily selected, and the panel surface temperature t_p might then be considered an unknown. In

either event it is evident that the equation contains at least three unknowns; hence eventual solution will depend on the setting up of two additional simultaneous equations.

A second simultaneous equation can be written from the comfort equation (13·13) by substituting for the average surface temperature to get

$$t_a + ut_p + vt_e = 152 \quad (13\cdot17)$$

The third simultaneous equation can be obtained by establishing a heat balance on the ventilation air. Letting V_c be the cubic feet of outside ventilation air passing each square foot of total inside room surface per hour, it follows that V_c is related to the room air changes V by the equation

$$V_c = \frac{V(LWH)}{2(LW + WH + LH)} \quad (13\cdot18)$$

where L , W , and H are the length, width, and height of the room.

A heat balance on the ventilation air gives

$$0.018V_c(t_o - t_a) = [uh_p(t_p - t_a)] + [hcv(t_e - t_a)] \quad (13\cdot19)$$

where the constant 0.018 is the quotient of specific heat (0.24) divided by specific volume (13·5).

Solving equation 13·16 for t_e gives

$$t_e = \frac{ut_p + 0.8vt_a + C_evt_o}{u + 0.8v + C_e v} \quad (13\cdot20)$$

and solving equation 13·19 for t_a gives

$$t_a = \frac{h_p ut_p + 0.8vt_e - 0.018V_c t_o}{h_p u + 0.8v - 0.018V_c} \quad (13\cdot21)$$

where h_p has the values 0.4, 0.7, and 1.1 for floor, wall, and ceiling locations, respectively, of the cooling panels. The comfort equation (13·17) can be rewritten

$$t_a = 152 - ut_p - vt_e \quad (13\cdot22)$$

Solving the above three equations simultaneously permits elimination of t_e and t_a with consequent determination of panel area (if t_p is selected) or of panel temperature (if u is selected). Graphical solutions of these three equations are available† in the literature permitting direct design for most of the temperature and area ranges that occur in common practice.

† Hutchinson, *A Graphical Design Procedure for Radiant Panel Cooling*, Revere Copper and Brass Incorporated, New York, 1950.

Once the design value of the panel temperature and the panel area are known the total cooling load can be approximately evaluated by the equations:

- (1) For cooling panels installed in the ceiling,

$$q = 2.0A_p(76 - t_p) \quad (13.23)$$

- (2) For cooling panels installed in the walls,

$$q_p = 1.7A_p(76 - t_p) \quad (13.24)$$

- (3) For cooling panels installed in the floor,

$$q_p = 1.4A_p(76 - t_p) \quad (13.25)$$

The above equations are slightly non-conservative since they assume the environment (air and surfaces) around the panel to be at an equivalent temperature of 76° F, whereas actually the value will be somewhat warmer than this because the panel area is lower than 76° F; for purposes of approximate load analysis this difference is not, however, of practical importance.

13.10. Energizing the Panel. Up to this point the procedure for panel cooling design has been concerned solely with the determination of how much cooling surface is required and at what design temperature it must operate. The results of this analysis will obviously be the same irrespective of the method used to carry energy away from the panel once it has been received at the surface. Thus cooled air can be used as a working substance for maintaining the panel surface temperature at the prescribed reduced value, or a refrigerated liquid, like water, can be circulated through tubes embedded in the mass of the panel. The latter method is by far the more common and is the only one which will be considered here.

When panel cooling is achieved by means of cooled water circulating through embedded pipes or tubes, the designer's problem is to determine the required size and spacing of the tubes and the required temperature and flow rate of the cooling medium. Experimental work at the University of California‡ has shown that over the range from $\frac{3}{8}$ in. through $\frac{3}{4}$ in. pipe or tube the diameter has very little influence on heat output; similarly, for pipe or tube spacing of 4 in., 6 in., or 9 in. the output per lineal foot is not appreciably affected by spacing. As a conservative approximation it is suggested that the known panel rating in Btu/(hr)(sq ft), be divided by 0.77 to obtain a number which will be equal to the product of the surface-to-water temperature difference

‡ Raber and Hutchinson, "Experimental Studies on Panel Heating Tube Spacing," *Trans. ASHVE*, Vol. 53, 1947.

and the required lineal feet of tube per square foot of panel; either the mean water temperature can be arbitrarily fixed and the tube spacing calculated or vice versa.

PROBLEMS

1. Determine the emissive power of the human body, taking skin temperature as 86°F and emissivity as unity.

2. A translucent sheet of glass has an emissivity of 0.5 and a transmissivity of 0.3. (a) What is the absorptivity? (b) Determine the reflectivity.

3. An average sedentary adult having clothed surface area of 21 sq ft loses approximately 140 Btu/hr by convection to air at 70°F when clothing surface temperature is 84°F . (a) Calculate the equivalent film coefficient for convective heat transfer and compare with that for a flat plate. (b) Does the human body more closely approach the thermal characteristics of a vertical plate or of a horizontal one?

4. A uniformly constructed and exposed refrigerated box has an inside air temperature of 50°F when the surface temperature of the smooth interior is 70°F . Would the "standard" inside film coefficient of $1.65\text{ Btu}/(\text{hr})(\text{sq ft})(^{\circ}\text{F})$ be applicable in this case? Why?

5. For the box described in Problem 4 determine the equilibrium temperature of a small lump of material suspended in the air within the box. Would this equilibrium temperature be changed if the box were lined with a material having a reflectivity of 90 per cent (assuming that the inside surface temperature remains at 70°F)? Why?

6. Calculate the equilibrium temperature of a blackened flat plate exposed on one side to a clear night sky and surrounded by air at 40°F . Consider air movement as negligible and the rear of the plate as insulated.

7. For the conditions of Problem 6 determine the equilibrium temperature if the wind velocity past the smooth surface of the plate is 15 mph.

8. Determine the net radiant transfer between two areas, each 1 sq ft, in parallel planes 15 ft apart; the distance between the normals to the centers of the two areas is 7 ft. One area is at 120°F , the other at 60°F .

9. The shape factor of a standing occupant with respect to energy received from a 10 sq ft cooling panel located in the ceiling is 0.04. If the panel has a surface temperature of 45°F and an absorptivity of 0.85, calculate the net radiant transfer from occupant to panel.

10. A small object having a uniform surface temperature of 40°F is in a room in which all surfaces are at 60°F . Determine the equivalent radiation coefficient h_r .

11. For the case described in Problem 10 evaluate h_r if the object is at 80°F and the temperature difference from room surfaces to object remains 20°F .

12. A room is to be maintained at 72°F air temperature when the outside air temperature is zero. For conditions of winter comfort calculate the required value of the equivalent overall coefficient of heat transfer.

13. A convection-type cooling system is to establish summer comfort in a room which has an equivalent overall coefficient of heat transfer of 0.70. Calculate the optimum inside air temperature when the outside air temperature is 90°F .

14. For the room of Problem 13 plot a curve of optimum inside air temperature as a function of outside air temperature.

15. A room having U_c of 0.15 is to be cooled by means of a ceiling panel which operates at design surface temperature of 60° F when outside air temperature is 100° F. If outside air is introduced into the cooled space at a rate of 3 cu ft/sq ft of room surface determine the fraction of room surface which will have to be cooled.

16. If the ventilation rate in Problem 15 were 7 cu ft/sq ft what fraction of the room surface would have to be cooled?

17. For the room of Problem 15 assume zero ventilation rate and calculate the fraction of room surface to be cooled.

18. For the room of Problem 15 calculate the refrigeration load and compare it with the refrigeration load that would be required if the room were to be cooled by a convective system to the corresponding optimum air temperature.

CHAPTER XIV

PSYCHROMETRIC LOADS; AIR CONDITIONING METHODS

The control of the humidity and temperature of air has now become one of the foremost considerations of the refrigerating engineer both in connection with cold storage and for the maintenance of manufactured weather in industry and in comfort cooling. The subject involves the laws governing *mixtures of air and water vapor*, the enthalpy of the mixture both at standard atmospheric pressure and at pressures above and below this point, factors entering into the requirements of comfort cooling, the calculations of the refrigerating load, and the means whereby temperature, humidity, and air mixing may be controlled. Numerous illustrative examples indicate the recommended method of calculation.

THERMODYNAMIC LAWS OF MIXTURES

14-1. Specific Volume. According to *Avogadro's law* equal volumes of all gases at the same temperature and pressure have the same number of molecules. *Dalton's law* states that in a mixture of two or more perfect gases each gas exerts the same pressure that it would if it occupied the entire volume by itself. Therefore the total pressure will be the sum of the partial pressures caused by the bombardment on the sides of the containing vessel produced by the molecules of each individual gas. In mixtures of dry air and water vapor it is customary to use Avogadro's and Dalton's laws for, although the water vapor is not a permanent gas and much less a perfect one, the discrepancy is not great at the low partial pressures of water vapor usual in comfort cooling and cold storage problems. Therefore, according to Dalton's law of partial pressures, if the vapor pressure of the water vapor in pounds per square inch is p_w , the pressure exerted by the molecules of the air alone is $14.7 - p_w$, where the total pressure, or the sum of the partial pressures, is normal atmospheric or approximately 14.7 psia. The *volume in cubic feet of 1 pound of dry air saturated with water vapor for a temperature of t degrees F* is, from the characteristic equation of a perfect gas,

$$P\bar{V} = RT$$

$$\bar{V} = \frac{53.34T}{144(14.7 - p_w)} = \frac{0.3704T}{14.7 - p_w} \text{ cu ft} \quad (14.1)$$

where R is the gas constant or 53.34 for air, and T is the absolute temperature or $t + 459.6^\circ \text{F}$; by Avogadro's law this is also the volume of the water vapor.

14.2. Absolute Humidity. If the volume of 1 pound of dry, saturated steam, i.e., the specific volume, is \bar{V}_g , the *absolute humidity* in pounds of water vapor per pound of dry air, if the mixture is saturated with water vapor, is

$$H = \frac{0.3704T}{(14.7 - p_w)\bar{V}_g} \text{ lb/lb} \quad (14.2)$$

The specific volume of the steam must be found from the partial pressure of the steam, p_w , or the temperature of the mixture. The perfect gas laws may be applied approximately to steam, when the characteristic equation becomes, per pound of water vapor,

$$P_w \bar{V}_w = R_w T$$

$$\bar{V}_w = \frac{85.7T}{144p_w} \text{ cu ft}$$

where the gas constant R_w for the water vapor for temperatures below 100°F is approximately 85.7.

Another formula for the absolute humidity may be found by a very different process. As the ratio of the partial pressures is also the ratio of the number of molecules the ratio of the number of molecules of the steam and air is given by the expression $p_w/(14.7 - p_w)$, and the ratio of their weights, i.e., the ratio of the weight in pounds of water vapor per one pound of dry air, is

$$H = \frac{p_w}{14.7 - p_w} \times \frac{18.016}{28.967} \text{ lb/lb} \quad (14.3)$$

where 18.016 and 28.967 are the molecular weights or the weight in pounds per *mole* of water vapor and dry air, respectively. A mole is the weight in grams or pounds numerically equal to the molecular weight. This formula for the absolute humidity H would be correct if the gas laws held for water vapor, but they do not hold exactly. Therefore a correction is necessary,* so that the formula becomes

$$H = S \frac{p'_w}{29.92 - p'_w} \text{ lb/lb}$$

where S is the specific weight of water vapor compared with air as unity

* Carrier, *Trans. A.S.M.E.*, Vol. 33, 1911.

and is given by the expression

$$S = 0.6219 + 0.001815p'_w{}^{1/2} + 0.0000051p'_w{}^{3/2}$$

p' = the pressure of water vapor in inches of mercury

If H_1 is the weight of the water vapor in *grains*, $H_1 = 7000H$ and therefore

$$H_1 = 7000 \frac{Sp'_w}{29.92 - p'_w}$$

$$= \frac{7000(0.6219p'_w + 0.001815p'_w{}^{3/2} + 0.0000051p'_w{}^{5/2})}{29.92 - p'_w} \text{ grains/lb}$$

or using

$$H_1 = \frac{7000 \times 53.34T}{144(14.7 - p_w)\bar{V}_g} = \frac{2593T}{(14.7 - p_w)\bar{V}_g} \text{ grains/lb} \quad (14.4)$$

Example. If air at 70° F is saturated with water vapor its partial pressure p is 0.3628 psia, and its specific volume is 869 cu ft/lb. Therefore

Solution.

$$H_1 = \frac{2593T}{(14.70 - 0.363) 869} = 110.1 \text{ grains per pound of dry air}$$

There are several different methods of designating the amount of water vapor in the air. The absolute humidity has already been defined as the weight of water vapor, in pounds or grains, present in the air per pound of dry air. *Percentage humidity* is the weight of water vapor in a unit weight of dry air divided by the weight of water vapor that could be contained in the same unit weight of dry air if it were saturated with water vapor. For example, air at 70° F and 1 atmosphere has 110.1 grains of water vapor per pound of dry air when saturated, so that if 1 lb of dry air at 70° F contains 75 grains the percentage humidity is $75/110.1 = 68.1$ per cent. If the percentage humidity is 50 per cent and the dry-bulb temperature is 70° the weight of water vapor present will be $0.5 \times 110.1 = 55.05$ grains per pound of dry air. As the pressure exerted is proportional to the number of molecules, this percentage is proportional approximately to the ratio of the actual pressure due to the water vapor p to the pressure that would be exerted if the vapor were in a saturated condition p_s .† The number of grains or

† Relative humidity defined as the ratio of the pressures is given as $100(p/p_s)$,

whereas percentage humidity is given as $100 \frac{\frac{p}{14.7 - p}}{\frac{p_s}{14.7 - p_s}}$.

pounds of water vapor per cubic foot of the mixture is also used in psychrometric calculations.

If an unsaturated mixture of air and water vapor is cooled at constant pressure the volume of the mixture will decrease, as the perfect gas formula $V_1/T_1 = V_2/T_2$ indicates, and the relative humidity will increase. This increase of the relative humidity will continue with further lowering of the temperature until the humidity becomes 100 per cent when the *dew-point temperature* is reached. A still further lowering of the temperature will cause condensation of water vapor. Figure 14·1 shows the variation of temperature and humidity in the lower range of air conditioning problems.

14·3. Adiabatic Saturation. If an unsaturated mixture of air and steam is passed through an insulated chamber, equipped for the recirculation of the spray water or for recirculation of water to wet sponges or tile, as explained by Carrier,‡ the process is one of *adiabatic saturation*. Depending on the initial temperature of the water, the spray water will give up heat to or receive heat from the air until *dynamic equilibrium* is established. At this temperature, which is approximately the *wet-bulb temperature*, evaporation of some water will occur, so that, if the chamber is large enough, the air will leave saturated with steam and at the wet-bulb temperature of the air entering the sprays. As no heat is understood to enter through the walls the loss of sensible heat of the initial air-steam mixture is numerically equal to the gain in latent heat.

For example, if dry air at 108.7° F is passed through an insulated chamber loosely packed with wet sponges dynamic equilibrium will be established at 60° if the wetted surfaces are great enough in amount. As all the water which evaporates does so at the wet-bulb temperature of 60°, the steam evaporated during the process does not affect the specific heat of the air, which is 0.24.

A very careful investigation was performed on adiabatic saturation by Carrier and Lindsay,§ who reported that under most practical conditions the temperature of evaporation of a free water surface approximates the theoretical temperature of adiabatic saturation, and that the variations observed were insignificant, from the engineering standpoint, for velocities of the air from a minimum of 200 fpm. The percentage deviation of the wet-bulb depression found by experiment compared with the true wet-bulb depression was only a fraction of 1 per cent.

Another method of analysis, first devised by Regnault|| and amplified

‡ Carrier, *Trans. A.S.M.E.*, Vol. 33, p. 1005, 1911.

§ Carrier and Lindsay, *Trans. A.S.M.E.*, Vol. 46, p. 739, 1924.

|| *Mémoires de l'académie des sciences de l'institute de France*, Vol. 21, 1847.

by W. K. Lewis,[¶] analyzes the rate of heat flow due to the inward rate of diffusion caused by the difference in the partial pressures. Heat passes from the air to the drop of water by diffusion through what is practically a stationary film of air.

The rate of vapor diffusion per unit area of the liquid surface per unit of time is

$$-\frac{dH}{A d\theta} = k'(p_w - p) \quad (14.5)$$

where

t_w, p_w, H_w = quantities representing the condition of liquid water in contact with the air

$(c_p)_m$ = specific heat of the mixture

H = absolute humidity of gas

A = area of liquid in contact with air

h = surface conductivity between gas and liquid in Btu/(sq ft) (°F)

k = coefficient of diffusion, parts by weight of vapor diffused per unit time per unit area per unit vapor pressure difference

k' = coefficient of diffusion, parts by weight of vapor diffused per unit time per unit area per unit absolute humidity difference

p = partial pressure of vapor

p'' = barometric pressure

θ = time

L_w = latent heat of vaporization

H_e, t_e = end conditions for the processes in question

whereas the rate of heat diffusion from the atmospheric air to the liquid surface is

$$\frac{dQ}{A d\theta} = h(t - t_w) = \frac{-L_w dH}{A d\theta}$$

$$-\frac{dH}{A d\theta} = \frac{h}{L_w} (t - t_w)$$

Therefore

$$p_w - p = (t - t_w) \frac{h}{k'L_w}$$

[¶] Lewis, "The Evaporation of a Liquid into a Gas," *Trans. A.S.M.E.*, Vol. 44, 1922,

From Avogadro's law,

$$\frac{\frac{H}{18.02}}{\frac{1}{29.02}} = \frac{p}{p'' - p} \quad \text{or} \quad p = p'' \frac{\frac{H}{18.02}}{\frac{H}{18.02} + \frac{1.0}{29.0}}$$

Therefore

$$p'' \frac{k' L_w}{h} \frac{\frac{H_w}{18.02}}{\frac{H_w}{18.02} + \frac{1.0}{29.0}} - \frac{\frac{H}{18.02}}{\frac{H}{18.02} + \frac{1}{29.0}} = t - t_w$$

Where H is small, as it usually is below 150°F , $H_w/18.02$ and $H/18.02$ are negligible when compared with the quantity $1/29.0$, and the expression simplifies to

$$p'' \frac{k' L_w 29.0}{18.02 h} (H_w - H) = t - t_w$$

and if

$$k = \frac{29.0}{18.02} k' p''$$

then

$$t - t_w = \frac{k L_w}{h} (H_w - H) \tag{14.6}$$

but

$$-L_w dH = (c_p)_m dt$$

or

$$H = -\frac{(c_p)_m}{L_w} t + \text{a constant}$$

or

$$H_e - H = \frac{(c_p)_m}{L_w} (t - t_e)$$

and therefore

$$t - t_e = \frac{L_w}{(c_p)_m} (H_e - H) \tag{14.7}$$

Equations 14.6 and 14.7 must be identical expressions if the process is extended indefinitely; therefore

$$H_w = H_e, \text{ a constant} \quad t_w = t_e, \text{ also a constant}$$

Therefore

$$\frac{L_w}{(c_p)_m} = \frac{kL_w}{h} \quad \text{or} \quad (c_p)_m = \frac{h}{k}$$

From these equations Lewis concludes that the statement made by Carrier (i.e., that during adiabatic saturation the wet-bulb temperature remains constant) would appear to represent only a limiting case, but that the variation is negligible in almost all engineering work.

14.4. The Specific Heat of the Mixture. According to the International Critical Tables, the specific heat of dry air at constant pressure is approximately constant from 0° to 100° F and may be taken as 0.24. The specific heat of steam at the low partial pressures usually encountered in air conditioning may also be taken as a constant at 0.453. The specific heat of a mixture may be found from the formula

$$c_m = \frac{\sum wc}{\sum w}$$

For the air-steam mixture this becomes, for 1 lb dry air plus H pounds of water vapor,

$$(c_p)_m = 0.24 + 0.453H \quad (14.8)$$

where H is the weight in pounds of steam per 1 lb of dry air.

14.5. The Wet-Bulb Depression. The expression for adiabatic saturation during the evaporative cooling of air may be stated mathematically by

$$-c_{p_m} dt = L_w dH$$

or

$$\frac{dH}{dt} = -\frac{(c_p)_m}{L_w} \quad (14.9)$$

where L is the latent heat of evaporation at the wet-bulb temperature, which gives the slope of the line representing the process if the absolute humidity and the dry-bulb temperature are the chosen coordinates. As the value of c_{p_m} varies with the amount of water vapor present in the *initial* mixture of the air and water vapor the slope of the adiabatic saturation process will vary. However, for conditions of cold storage and comfort cooling, the deviations are slight in the range of calculations and need not be considered. Integrating this expression gives:

$$H_2 - H_1 = \frac{(c_p)_m}{L_w} (t_1 - t_2)$$

from which it is possible to get the wet-bulb depression:

$$t_1 - t_2 = \frac{L_w(H_1 - H_2)}{(c_p)_a + H_1(c_p)_v}$$

$$L_w(H_2 - H_1) = [(c_p)_a(t_1 - t_2)] + [(c_p)_v(t_1 - t_2)H_1]$$

$$H_1 = \frac{H_2 L_w - [0.24(t_1 - t_2)]}{L_w + [0.453(t_1 - t_2)]} \quad (14 \cdot 10)$$

where H_2 is the absolute humidity at the wet-bulb temperature t_2 in pounds of water vapor per pound of dry air. This formula gives a means of calculating the initial absolute humidity if the wet- and dry-bulb temperatures are known.

Another formula connecting points along the line of adiabatic saturation follows:

$$p_w = p'_w - \frac{(t - t')(p'' - p'_w)}{2827 - 1.276t'} \quad (14 \cdot 11)$$

where p'_w is the partial pressure of the steam in inches of mercury at the temperature of the wet bulb t' , t is the dry-bulb temperature, p_w is the partial pressure of the steam at the dew-point temperature, and p'' is the barometric pressure in inches of mercury.

Example. Find the dry-bulb temperature where the dew-point pressure p_w is zero and the wet-bulb temperature is 60° F.

Solution.

$$0 = 0.52 - \frac{(29.92 - 0.52)(t - 60)}{2827 - (1.276 \times 60)}$$

$$t = 108.6^\circ \text{ F}$$

Air leaving the washer is seldom at the wet-bulb temperature, and it is sometimes convenient to know how efficient the washer is. In such a case the humidifying efficiency may be defined as the ratio of the actual wet-bulb depression to the maximum wet-bulb depression and may be expressed by

$$e = 1.0 - \frac{t_2 - t_3}{t_1 - t_3} \quad (14 \cdot 12)$$

where t_1 is the dry-bulb temperature of the entering air, t_2 that of the air leaving the washer, and t_3 the constant wet-bulb temperature.

Example. If the initial dry-bulb temperature is 80° and the wet-bulb temperature is 60° F find the initial absolute humidity.

Solution. In this case $H_2 = 0.01103$ lb, the absolute humidity at 60° . Therefore

$$H_1 = \frac{H_2 L - [0.24 (t_1 - t_2)]}{L + [0.453 (t_1 - t_2)]} = \frac{(0.01103 \times 1058.1) - (0.24 \times 20)}{1058.1 + (0.453 \times 20)}$$

$$= 0.00643 \text{ lb} = 45.0 \text{ grains/lb}$$

14·6. Enthalpy of a Mixture of Dry Air and Water Vapor. There are two methods of calculation of the enthalpy of the air-steam mixture. The first method is to consider all the steam to be evaporated *from and at* the wet-bulb temperature. This was the procedure used by Goodenough and Carrier, and such values did not include the heat content of the liquid. The calculations, based on the principle of adiabatic saturation, would be according to the formula

$$h = c_{pa}t + HL_t \quad (14·13)$$

where h is the heat content of the air-steam mixture at temperature t , c_{pa} is the average specific heat of the air, H is the absolute humidity in pounds at the wet-bulb temperature t , and L is the latent heat of vaporization at the wet-bulb temperature.

The preferred method of calculation is to assume that the enthalpy of the air-steam mixture is the sum of the enthalpies of the separate weights of air and steam. This becomes

$$h_m = c_{pa}t + [H(1059.0 + 0.453t)] \quad (14·14)$$

where t is in degrees Fahrenheit, where the weight of dry air is 1 lb, and the steam may be superheated or dry, saturated. The formula is nearly exact from 0° to 100° F.

14·7. Heat Involved in Conditioning Air. The general energy equation applies to the cooling of air, and may be expressed as follows:

$$h_1 + Q + [(H_2 - H_1)h_f] - h_2 - R = 0 \quad (14·15)$$

Therefore

$$Q = h_2 - h_1 + [(H_1 - H_2)h_f] \quad \text{if } R = 0 \quad (14·16)$$

where h_1 = the enthalpy of the entering mixture in Btu per pound of dry air, h_2 = the enthalpy of the exit mixture in Btu per pound of dry air, $H_2 - H_1$ = the change in absolute humidity in pounds per pound of dry air, h_f = the enthalpy of the water supplied (or removed) in Btu per pound of dry air, Q = the total amount of heat supplied in Btu per pound of dry air, R = the radiation and other losses in Btu per pound of dry air.

The actual amount of heat involved in cooling or heating a mixture of air and water vapor in order to permit a change from one condition to

another at constant pressure may be calculated with reasonable accuracy by means of three different methods. The amount of heat so involved is equal (1) to the difference in the enthalpies at the initial and final conditions and (2) the difference in the enthalpies for the initial and the final wet-bulb temperature taken from the values given in Table 14·1; finally (3) the enthalpy for any condition not saturated may be calculated by adding to the enthalpy at the dew-point temperature the heat involved in the superheating process by the use of the expression

$$Q = Wc_{p_m} \Delta t$$

Example. Air at 78° F dry-bulb and 64.5° F wet-bulb ($H = 0.00994$) is to be cooled to 53.55° F. It is then to be heated to 68° F at a constant dew-point temperature. Find the amount of heat involved per pound of dry air. The dew-point temperature for the initial condition is 57.2°.

Solution. Several methods may be used, as just listed.

(1) The initial enthalpy is

$$\begin{aligned} h &= 18.720 + \frac{0.00984}{0.02076} \times 22.718 \\ &= 29.51 \end{aligned}$$

The final enthalpy, at 53.55° F, $h = 22.296$ Btu, and the refrigeration (neglecting the heat involved in the condensate) is $29.60 - 22.30 = 7.30$ Btu. The heat required to raise the temperature from 53.55° to 68° F is: the enthalpy at 68° is $h = 16.320 + [(0.008716/0.014680) \times 16.001] = 25.820$, and the heat required to make the change is $25.820 - 22.30 = 3.52$ Btu.

(2) The initial enthalpy, at the wet-bulb temperature, is 29.60 Btu.

(3) The enthalpy at the dew-point temperature of 56.90° F is 24.354 Btu, and the heat required to superheat to 78° is $(78 - 56.9) 0.2444 = 5.157$, making a total of 29.51 Btu. The refrigeration and the heat required to reheat are found in a similar manner. For example, the reheat is at constant dew-point temperature, therefore the value of $c_{p_m} = 0.24 + [0.008716 \times 0.453] = 0.24395$. The heat involved in the reheating Q_r is

$$\begin{aligned} Q_r &= 0.24395 \times (68 - 53.55) \\ &= 3.525 \text{ Btu} \end{aligned}$$

Example. An air-steam mixture under a pressure of 1 atmosphere at 80° dry-bulb and 60° wet-bulb is cooled in a suitable spray chamber to 40°.

Solution. From Table 14·1

$$H_1 = 0.00643 \text{ lb} \quad H_2 = 0.00519 \text{ lb} \quad h_1 = 26.44 \text{ Btu} \quad h_2 = 15.19 \text{ Btu}$$

$$Q = -26.44 + 15.19 - [(0.005193 - 0.00643) (40 - 32)] = 11.24 \text{ Btu}$$

If the Carrier and Goodenough method is used the dew-point temperature corresponding to the initial condition is 45.46° F. The total heat at 45.46° F* is

* See Goodenough, *Properties of Steam and Ammonia*, John Wiley & Sons, Table 6.

TABLE 14-1
MIXTURES OF AIR AND SATURATED WATER VAPOR

Temperature, °F	Pressure, psia	Weight of Saturated Vapor per lb of Dry Air		Volume in Cubic Feet		Specific Heat of Saturated Mixture	Heat Content in Btu					
		Pounds	Grains	of 1 lb of Dry Air	of 1 lb of Dry Air + Vapor to Saturate It		Increase Due to Presence of Vapor	1 lb of Vapor	Vapor to Saturate 1 lb of Air	1 lb of Dry Air	1 lb of Dry Air + Vapor to Saturate It	Water Solid or Liquid
0	0.0185	.000784	5.488	11.584	11.599	0.2403	1059.00	0.831	0.000	0.831	-158.9	
1	0.0195	.000828	5.796	11.600	11.625	0.015	1059.45	0.879	0.240	1.119	158.4	
2	0.0206	.000874	6.118	11.625	11.663	0.017	1059.91	0.926	0.480	1.406	157.9	
3	0.0217	.000920	6.440	11.660	11.715	0.018	1060.36	0.956	0.720	1.676	157.5	
4	0.0228	.000967	6.769	11.685	11.770	0.019	1060.81	0.986	0.960	1.936	157.0	
5	0.0240	.001018	7.126	11.710	11.820	0.020	1061.27	1.083	1.200	2.283	156.5	
6	0.0253	.001073	7.511	11.735	11.875	0.021	1061.72	1.140	1.440	2.580	156.0	
7	0.0266	.001138	7.896	11.760	11.930	0.022	1062.17	1.202	1.680	2.880	155.6	
8	0.0280	.001198	8.310	11.786	11.989	0.023	1062.62	1.261	1.920	3.183	155.1	
9	0.0294	.001268	8.736	11.811	11.835	0.024	1063.08	1.331	2.160	3.491	154.6	
10	0.0310	.001315	9.205	11.837	11.862	0.025	1063.53	1.399	2.400	3.799	154.1	
11	0.0326	.001332	9.681	11.862	11.888	0.026	1063.98	1.472	2.640	4.112	153.7	
12	0.0342	.001349	10.164	11.887	11.915	0.028	1064.44	1.545	2.880	4.425	153.2	
13	0.0359	.001355	10.675	11.941	11.941	0.030	1064.89	1.620	3.120	4.740	152.7	
14	0.0377	.001500	11.200	11.937	11.969	0.032	1065.34	1.705	3.360	5.065	152.2	
15	0.0397	.001655	11.795	11.962	11.995	0.033	1065.80	1.797	3.600	5.397	151.8	
16	0.0417	.001771	12.399	11.987	12.022	0.035	1066.25	1.888	3.840	5.728	151.3	
17	0.0437	.001837	12.999	12.012	12.049	0.037	1066.70	1.934	4.080	6.012	150.8	
18	0.0459	.001950	13.650	12.038	12.076	0.038	1067.15	2.081	4.320	6.401	150.3	
19	0.0481	.002044	14.308	12.063	12.103	0.040	1067.61	2.186	4.560	6.746	149.8	
20	0.0505	.002146	15.022	12.088	12.131	0.043	1068.06	2.282	4.800	7.092	149.3	
21	0.0530	.002254	15.748	12.113	12.158	0.045	1068.51	2.410	5.040	7.450	148.9	
22	0.0556	.002364	16.548	12.139	12.185	0.046	1068.97	2.527	5.280	7.807	148.4	
23	0.0583	.002480	17.360	12.164	12.212	0.048	1069.42	2.654	5.520	8.174	147.9	
24	0.0611	.002598	18.186	12.190	12.240	0.050	1069.87	2.780	5.760	8.540	147.4	
25	0.0640	.002724	19.068	12.215	12.268	0.053	1070.33	2.920	6.000	8.920	146.9	
26	0.0671	.002855	19.985	12.240	12.296	0.056	1070.78	3.057	6.240	9.297	146.4	
27	0.0704	.002996	20.972	12.265	12.324	0.059	1071.23	3.209	6.480	9.680	146.0	
28	0.0737	.003137	21.959	12.290	12.352	0.062	1071.68	3.362	6.720	10.082	145.5	
29	0.0772	.003287	23.000	12.315	12.380	0.065	1072.13	3.529	6.960	10.489	145.0	
30	0.0809	.003445	24.115	12.340	12.409	0.069	1072.59	3.695	7.200	10.895	144.5	

TABLE 14-1 (Continued)

31	0.0848	0.03612	25.284	12.365	12.437	0.072	0.2416	1073.04	3.877	7.440	11.317	144.0
32	0.0887	0.03777	26.456	12.391	12.466	0.075	0.2417	1073.49	4.054	7.680	11.724	-133.5
33	0.0923	0.03935	27.545	12.416	12.495	0.079	0.2417	1073.94	4.226	7.920	12.146	+1.01
34	0.0961	0.04094	28.658	12.442	12.522	0.082	0.2418	1074.40	4.398	8.160	12.558	-2.01
35	0.1001	0.04266	29.862	12.467	12.552	0.085	0.2419	1074.86	4.585	8.400	12.985	3.02
36	0.1044	0.04438	31.066	12.492	12.580	0.088	0.2420	1075.31	4.772	8.640	13.412	4.03
37	0.1083	0.04620	32.340	12.517	12.610	0.093	0.2420	1075.76	4.970	8.880	13.850	5.03
38	0.1126	0.04802	33.614	12.542	12.639	0.097	0.2421	1076.21	5.168	9.120	14.288	6.04
39	0.1171	0.04998	34.986	12.567	12.669	0.102	0.2422	1076.68	5.381	9.360	14.741	7.04
40	0.1217	0.05193	36.351	12.593	12.698	0.106	0.2423	1077.13	5.594	9.600	15.194	8.05
41	0.1265	0.05394	37.828	12.618	12.728	0.110	0.2424	1077.58	5.823	9.840	15.663	9.05
42	0.1315	0.05615	39.305	12.643	12.757	0.114	0.2425	1078.04	6.053	10.080	16.131	10.05
43	0.1367	0.05845	40.915	12.668	12.787	0.119	0.2426	1078.49	6.301	10.320	16.620	11.05
44	0.1420	0.06069	42.483	12.693	12.818	0.125	0.2427	1078.94	6.548	10.560	17.108	12.05
45	0.1475	0.06311	44.177	12.718	12.848	0.130	0.2428	1079.39	6.812	10.800	17.612	13.05
46	0.1532	0.06553	45.871	12.744	12.878	0.134	0.2429	1079.84	7.076	11.040	18.116	14.06
47	0.1591	0.06823	47.761	12.769	12.909	0.140	0.2430	1080.29	7.360	11.280	18.640	15.06
48	0.1652	0.07072	49.504	12.794	12.940	0.146	0.2432	1080.75	7.643	11.520	19.163	16.06
49	0.1715	0.07350	51.450	12.819	12.971	0.152	0.2433	1081.20	7.947	11.760	19.707	17.06
50	0.1780	0.07637	53.389	12.844	13.002	0.158	0.2434	1081.65	8.250	12.000	20.250	18.06
51	0.1848	0.07938	55.504	12.869	13.034	0.165	0.2435	1082.10	8.579	12.240	20.819	19.06
52	0.1918	0.08258	57.504	12.894	13.066	0.172	0.2437	1082.59	8.907	12.480	21.387	20.06
53	0.1989	0.08584	59.808	12.918	13.098	0.179	0.2438	1083.01	9.254	12.720	21.974	21.06
54	0.2063	0.08887	61.999	12.945	13.130	0.185	0.2440	1083.46	9.600	12.960	22.560	22.06
55	0.2149	0.09197	64.379	12.970	13.163	0.193	0.2441	1083.92	9.972	13.200	23.172	23.06
56	0.2219	0.09557	66.759	12.995	13.195	0.200	0.2443	1084.37	10.343	13.440	23.780	24.06
57	0.2300	0.09854	69.048	13.020	13.228	0.208	0.2444	1084.83	10.733	13.680	24.363	25.06
58	0.2384	0.1026	71.820	13.046	13.261	0.215	0.2446	1085.28	11.133	13.920	24.952	26.05
59	0.2471	0.10662	74.340	13.071	13.295	0.224	0.2448	1085.73	11.559	14.160	25.519	27.05
60	0.2564	0.1103	77.210	13.096	13.329	0.233	0.2450	1086.18	11.985	14.400	26.436	28.05
61	0.2654	0.1145	80.150	13.121	13.364	0.243	0.2451	1086.63	12.440	14.640	27.080	29.05
62	0.2749	0.1186	83.020	13.147	13.398	0.251	0.2453	1087.09	12.860	14.880	27.773	30.05
63	0.2848	0.1230	86.100	13.172	13.433	0.261	0.2455	1087.55	13.378	15.120	28.498	31.05
64	0.2949	0.1274	89.180	13.197	13.468	0.271	0.2457	1088.00	13.862	15.360	29.222	32.04
65	0.3054	0.1320	92.400	13.222	13.504	0.282	0.2459	1088.45	14.377	15.600	29.977	33.04
66	0.3162	0.1368	95.760	13.248	13.540	0.291	0.2462	1088.90	14.892	15.840	30.732	34.04
67	0.3273	0.1410	98.700	13.273	13.575	0.303	0.2464	1089.36	15.442	16.080	31.526	35.04
68	0.3388	0.1468	102.76	13.298	13.612	0.314	0.2466	1089.81	16.001	16.320	32.321	36.03
69	0.3506	0.1521	106.47	13.323	13.650	0.327	0.2469	1090.26	16.581	16.560	33.141	37.03

TABLE 14-1 (Continued)
MIXTURES OF AIR AND SATURATED WATER VAPOR

Temperature, °F	Pressure, psia	Weight of Saturated Vapor per lb of Dry Air		Volume in Cubic Feet		Specific Heat of Saturated Mixture	Heat Content in Btu					
		Pounds	Grains	of 1 lb of Dry Air	of 1 lb of Dry Air + Vapor to Saturate It		Increase Due to Presence of Vapor	1 lb of Vapor	Vapor to Saturate 1 lb of Air	1 lb of Dry Air	1 lb of Dry Air + Vapor to Saturate It	Water Solid or Liquid
70	0.3628	.01573	110.11	13.349	13.688	0.339	1090.71	17.160	16.800	33.960	38.03	
71	0.3754	.01631	114.17	13.374	13.735	0.351	1091.16	17.794	17.040	34.834	39.03	
72	0.3883	.01680	118.23	13.399	13.782	0.363	1091.62	18.433	17.280	35.713	40.03	
73	0.4016	.01745	122.31	13.424	13.831	0.377	1092.08	19.053	17.520	36.573	41.03	
74	0.4153	.01810	126.70	13.450	13.880	0.390	1092.53	19.771	17.760	37.433	42.03	
75	0.4295	.01875	131.25	13.475	13.890	0.405	1092.98	20.498	18.000	38.498	43.01	
76	0.4440	.01939	136.72	13.500	13.917	0.417	1093.43	21.205	18.240	39.445	44.01	
77	0.4590	.02008	142.55	13.525	13.960	0.435	1093.88	21.962	18.480	40.442	45.01	
78	0.4744	.02076	148.32	13.559	14.022	0.452	1094.33	22.718	18.720	41.438	46.00	
79	0.4903	.02149	150.43	13.575	14.044	0.469	1094.79	23.520	18.960	42.480	47.00	
80	0.5067	.02222	155.54	13.601	14.087	0.486	1095.24	24.336	19.200	43.536	48.00	
81	0.5236	.02300	161.00	13.626	14.130	0.504	1095.69	25.204	19.440	44.640	49.00	
82	0.5409	.02378	166.46	13.651	14.173	0.522	1096.15	26.071	19.680	45.751	50.00	
83	0.5588	.02461	172.27	13.676	14.217	0.541	1096.60	26.993	19.920	46.913	51.00	
84	0.5772	.02544	178.08	13.701	14.261	0.560	1097.06	27.914	20.160	48.074	52.00	
85	0.5960	.02632	184.24	13.726	14.307	0.581	1097.51	28.889	20.400	49.289	53.00	
86	0.6152	.02720	190.40	13.752	14.353	0.602	1097.96	29.862	20.640	50.502	54.00	
87	0.6352	.02807	196.49	13.777	14.401	0.624	1098.42	30.908	20.880	51.788	55.00	
88	0.6555	.02896	203.42	13.802	14.447	0.645	1098.87	31.937	21.120	53.037	56.00	
89	0.6765	.03005	210.35	13.827	14.493	0.666	1099.32	33.039	21.360	54.399	57.00	
90	0.6980	.03104	217.28	13.853	14.544	0.691	1099.77	34.141	21.600	55.741	58.00	
91	0.7201	.03209	224.63	13.878	14.594	0.716	1100.22	35.312	21.840	57.152	59.00	
92	0.7429	.03314	231.98	13.903	14.644	0.741	1100.68	36.482	22.080	58.562	59.98	
93	0.7662	.03425	239.75	13.928	14.694	0.766	1101.04	37.735	22.320	60.065	60.98	
94	0.7902	.03540	247.80	13.954	14.744	0.790	1101.49	38.987	22.560	61.547	61.97	
95	0.8149	.03658	256.06	13.979	14.798	0.819	1102.04	40.311	22.800	63.111	62.96	
96	0.8403	.03776	264.32	14.004	14.852	0.848	1102.49	41.635	23.040	64.675	63.96	
97	0.8663	.03903	273.21	14.029	14.907	0.878	1102.95	43.045	23.280	66.325	64.95	
98	0.8930	.04029	282.03	14.054	14.963	0.909	1103.40	44.455	23.520	67.975	65.94	
99	0.9205	.04164	291.48	14.079	15.021	0.942	1103.85	45.961	23.760	69.721	66.94	
100	0.9487	.04298	300.86	14.105	15.078	0.973	1104.30	47.467	24.000	71.467	67.93	

17.82 Btu, and at 40° F it is 15.21. The specific heat of the air-steam mixture is $0.24 + [0.453 \times 0.00643] = 0.243$, and the heat removed in cooling down to the dew-point temperature is $0.243 (80 - 45.46) = 8.38$ Btu. The total cooling is $8.38 + 17.82 - 15.21 = 10.99$ Btu. The problem can be done more easily another way. The total heat at 60° F wet-bulb is 26.18 and at 40° F wet-bulb it is 15.21 Btu, so that the heat removed is $26.18 - 15.21 = 10.97$ Btu.

14·8. Elevations above Sea Level. Should it be desired to make calculations for mixtures of dry air and the amount of water vapor necessary to saturate the volume occupied by the air at other than standard atmospheric pressure, as for example at the altitude of Denver, Colorado, or of Mexico City, the principal change will be in the volume of 1 lb of dry air when mixed with steam. The formula will be the same as before, namely,

$$\bar{V}_1 = \frac{RT}{144(p - p_w)} \quad \text{cu ft for air saturated with water vapor}$$

and

$$\bar{V}_2 = \frac{RT}{144p} \quad \text{cu ft for 1 lb of dry air}$$

where p is in pounds per square inch, T is the temperature in degrees Fahrenheit absolute, and R is 53.34 for air. The value of p_w will be found for the pressure of saturated steam at the temperature t from the steam tables. The weight of water vapor in pounds for a standard condition at the temperature t is the value of \bar{V}/\bar{V}_g for steam at the temperature t .

The weight of water vapor, the value of c_p , for 1 lb of dry air saturated with water vapor, and the heat content will vary.

Example. Calculate values for the heat content, with the zero at 0° F, c_p for 1 lb of dry air with enough water vapor to saturate it and the specific volumes of dry and humid air for 40°, 60°, 80° and 100° F if the barometer reads 25 in. of mercury.

Solution. The atmospheric pressure is $25 \times 0.491 = 12.275$ psia.

The volume of 1 lb of dry air is given by

$$\bar{V}_1 = \frac{53.34}{144 \times 12.275} T$$

and the volume of 1 lb of dry air, saturated with water vapor, is

$$\bar{V}_2 = \frac{53.34}{144(12.275 - p_w)} T$$

The weight of water vapor, per pound of dry air, is \bar{V}_2/\bar{V}_g , where \bar{V}_g is the specific volume of dry saturated steam at the desired temperature, as seen in Table 8·17.

14.9. Elevations below Sea Level. At times it is desired to cool and condition air at pressures greater than 1 atmosphere. If any particular pressure is desired it is a simple matter to calculate a table similar to Table 14.1 and if desired to construct a psychrometric chart for this condition. The formulas become

$$P_a \bar{V}_a = R_a T \quad \text{or} \quad \bar{V}_a = \frac{53.34T}{144(p - p_w)}$$

$$H = \frac{\bar{V}_a}{\bar{V}_g} \text{ pounds}$$

$$(c_{p_m}) \text{ (for humid air)} = 0.24 + 0.453H$$

where \bar{V}_a = the volume in cubic feet of 1 lb of dry air, p = the total pressure in pounds per square inch, p_w = the partial pressure of saturated steam in pounds per square inch, \bar{V}_g = the specific volume in cubic feet of dry saturated steam at temperature t .

The amount of sensible and latent heat at any particular dew-point temperature may be calculated in a manner similar to that of Table 14.1. Sometimes, however, the principal objective is to control the humidity and not to reduce the temperature more than a small amount. For example, if air is saturated with water vapor at a temperature of 60° F and 1 atmosphere pressure it will contain 77.2 grains of water vapor per pound of dry air. If the pressure is now increased to 2 atmospheres and the temperature held at 60° F the ratio of the air pressures will be

$$\frac{14.7 - 0.26}{29.4 - 0.26} = \frac{14.44}{29.14} = 0.4955$$

and the volume will be decreased in like amount. If the volume at 60° and 1 atmosphere is 13.33 cu ft, at 2 atmospheres it will be $(14.44/29.14) \times 13.33 = 6.61$ cu ft, and at 3 atmospheres it will be $(14.44/43.84) \times 13.33 = 4.39$ cu ft. The amount of water vapor contained in the air will be reduced in proportion to the decrease in volume, and this will be $(6.61/13.33) \times 77.2 = 38.25$ grains and $(4.39/13.33) \times 77.2 = 25.43$ grains, respectively. To get 25.43 grains at 1 atmosphere would require a temperature of 31.9° F, and 38.25 grains and 1 atmosphere would require 41.3° F. It is evident then that air when under pressure can be dehumidified by less depression of the temperature than would be required at 1 atmosphere, and if the air is to be compressed as well as dehumidified it would be better to dehumidify after compression.

14.10. The Psychrometric Chart. A suitable psychrometric chart will make the processes in the conditioning of air much easier to under-

TABLE 14-2
MIXTURES OF AIR AND SATURATED WATER VAPOR

Temperature of °F	Pressure, psia		Weights of Saturated Vapor per lb of Dry Air		Volume in Cubic Feet		Specific Heat of Saturated Mixture	Heat Content in Btu				
	Pounds	Grains	of 1 lb of Dry Air	of 1 lb of Dry Air + Vapor to Saturate It	Increase Due to Pressure of Vapor	1 lb of Vapor		Vapor to Saturate 1 lb of Air	1 lb of Dry Air	1 lb of Dry Air + Vapor to Saturate It	Water Solid or Liquid	
0	0.0185	0.009390	6.573	13.864	0.021	13.885	24042	1059.00	0.9945	0.000	0.9945	-158.9
5	0.0240	0.01219	8.533	14.014	0.028	14.042	24057	1061.27	1.2925	1.200	2.4955	-166.5
10	0.0310	0.01585	11.095	14.165	0.036	14.201	24073	1063.53	1.6755	2.600	4.2755	-194.1
15	0.0397	0.02019	14.133	14.317	0.047	14.364	24091	1065.80	2.1515	3.600	5.7515	-191.3
20	0.0505	0.02570	17.990	14.467	0.059	14.526	24116	1068.06	2.7450	4.800	7.5450	-149.3
25	0.0640	0.03261	22.827	14.617	0.077	14.694	24145	1070.33	3.4901	6.000	9.4901	-146.9
30	0.0809	0.04034	28.238	14.768	0.098	14.866	24183	1072.59	4.4272	7.200	11.6272	-142.5
35	0.1001	0.05105	35.735	14.919	0.123	15.042	24232	1074.86	5.4757	8.400	13.8757	-137.0
40	0.1217	0.066225	43.575	15.070	0.141	15.221	24282	1077.13	6.7057	9.600	16.3057	8.05
45	0.1475	0.085563	52.941	15.221	0.164	15.405	24342	1079.39	8.1630	10.800	18.9630	13.05
50	0.1780	0.099147	64.029	15.372	0.222	15.594	24414	1081.65	9.8940	12.000	21.8940	18.06
55	0.2149	0.11094	77.280	15.523	0.276	15.799	24502	1083.92	11.971	13.200	25.171	23.06
60	0.2561	0.1325	92.750	15.674	0.334	16.008	24600	1086.18	14.364	14.400	28.794	28.05
65	0.3054	0.1587	111.090	15.824	0.403	16.227	24718	1088.45	17.271	15.600	32.871	33.04
70	0.3628	0.1982	132.440	15.975	0.487	16.452	24858	1090.76	20.638	16.800	37.438	38.03
75	0.4295	0.2253	157.710	16.126	0.584	16.710	25021	1092.98	24.623	18.000	42.623	43.01
80	0.5067	0.2679	187.530	16.277	0.700	16.977	25212	1095.24	29.338	19.200	48.538	48.00
85	0.5960	0.3175	222.250	16.428	0.838	17.266	25440	1097.51	34.846	20.400	55.246	53.00
90	0.6980	0.3752	262.640	16.579	1.000	17.579	25700	1099.77	41.266	21.600	62.866	58.00
95	0.8149	0.4425	309.750	16.729	1.290	17.919	26008	1102.04	48.770	22.800	71.570	62.96
100	0.9487	0.5215	365.050	16.880	1.414	18.294	26362	1104.30	57.564	24.000	81.564	67.93
0	0.0185	0.009390	5.8730	12.353	0.017	12.400	24038	1059.00	0.5891	0.000	0.5891	-158.9
10	0.0310	0.01439	10.073	12.682	0.029	12.709	24053	1061.27	1.5808	2.400	3.9808	-154.1
20	0.0505	0.02295	16.065	12.922	0.048	12.970	24069	1063.53	2.4609	4.800	7.2609	-149.3
30	0.0809	0.03693	25.851	13.191	0.079	13.270	24117	1065.80	3.9608	7.200	11.1608	-144.5
40	0.1217	0.05554	38.878	13.461	0.119	13.580	24252	1071.13	5.9827	9.600	15.5827	8.05
50	0.1780	0.08160	57.120	13.730	0.180	13.910	24370	1081.65	8.8958	12.000	20.8958	18.06
60	0.2561	0.1181	82.670	14.000	0.265	14.265	24523	1086.18	12.872	14.400	27.272	23.06
70	0.3628	0.1689	118.230	14.269	0.387	14.656	24765	1090.77	17.271	16.800	34.071	28.05
80	0.5067	0.2382	166.740	14.539	0.536	15.095	25080	1095.24	26.681	18.000	44.681	33.04
90	0.6980	0.3300	231.000	14.808	0.892	15.600	25494	1099.77	36.681	21.600	58.281	38.03
100	0.9487	0.4617	323.190	15.078	1.117	16.195	26094	1104.30	50.980	24.000	74.980	43.01

P = 28 in. of Mercury

P = 28 in. of Mercury

stand, and the values obtained from it are usually sufficiently accurate for all engineering problems. There are a number of such charts, each with special advantages. The Carrier† chart follows closely one by Grosvenor‡ and another proposed by Tyler§ in 1902 in grains of water vapor as ordinate and the dry-bulb temperature as abscissa. Figure 14·1 is similar to these but is designed to be used in conjunction with Table 14·1.

In constructing a psychrometric chart as shown in the figure, the 100 per cent humidity line is first plotted by using the two coordinates of absolute humidity and dry-bulb temperature. The adiabatic saturation lines, sloping upward to the left, are next drawn in, the slope increasing slightly as the initial absolute humidity increases. As the adiabatic saturation line is also one of constant wet-bulb, it provides a graphical means of determining the wet-bulb temperature of any point on the chart in the superheated region. Percentage humidity lines are drawn in order to facilitate the location of points in the unsaturated region. Curves showing the values of the specific heat of the mixture of air and steam, $c_{p,m}$, and enthalpies of saturated mixtures for dew-point temperatures from 0° to 100° F are also drawn. The method of using the chart may be seen from the solution of the following example.

Example. Fresh air supplied to a building at 90° F, with a percentage humidity of 49 per cent, is cooled to the dew-point temperature corresponding to 80° F and 35 per cent humidity. Find the refrigeration required per 1000 cu ft delivered at the dew-point temperature.

Solution. In this case: $H_1 = 0.01521$ lb, $H_2 = 0.00778$ lb, $h_1 = 38.33$ Btu, $h_2 = 20.54$ Btu, $\bar{V} = 13.02$ cu ft, $t_2 = 50.5^\circ$ F.

From equation 14·16,

$$\begin{aligned} Q &= 20.54 - 38.33 + [(0.01521 - 0.00778) (50.5 - 32)] \\ &= -17.65 \text{ Btu/lb of dry air} \\ &= -\frac{1000 \times 17.65}{13.02} = -1356 \text{ Btu/min} \\ &= 6.78 \text{ tons} \end{aligned}$$

14·11. The Physical Effect of Air Mixing. In most air conditioning projects there are three separate factors to be given attention, the control of the dry-bulb temperature, of the moisture content in the air, and of the amount of air movement. If it is cold air that is to be conditioned it may be heated at constant water vapor content, and then either enough steam is added to secure the desired absolute humidity and

† Carrier, *Trans. A.S.M.E.*, Vol. 33, 1911.

‡ Grosvenor, *Trans. Am. Inst. Chem. Engrs.*, Vol. 1, 1908.

§ Tyler, *J. State Med.*, Vol. 42, No. 2.

temperature or use is made of a spray chamber with recirculating water in order to cool and humidify the air to the amounts desired. An automatic method of securing the same results is to control the dry-bulb temperature by means of a thermostat set to regulate the tempering coils and a hydrostat designed to adjust the amount of the water vapor by means of the temperature of the water going to the sprays.

When air is by-passed or fresh air from the outside is admitted and mixed with the recirculated air it is necessary to know at times something of the condition of the final mixture. The best way is to consider the weights involved.

Example. If 7500 cu ft of recirculated air at 65° F and 60 per cent relative humidity are mixed with 2500 cu ft of fresh air at 85° and 50 per cent relative humidity, what are the characteristics of the mixture?

Solution. The weight of the recirculated air at 65° is $7500/13.39 = 560.1$ lb. The weight of the fresh air at 85° is $2500/14.02 = 178.3$ lb. The heat capacity of each can be taken as the value of c_p of the mixture for the dew-point temperature, which is 0.2436 for the recirculated air and 0.2460 for the fresh air. The temperature of the mixture becomes

$$560.1 \times [0.2436(t_x - 65)] = 178.3 \times [0.2460(85 - t_x)]$$

from which $t_x = 69.9^\circ$ F. The percentage humidity of the mixture can be found from the consideration of the initial water content.

The weight of the water vapor in the recirculated air is $560.1 \times 55.44 = 31,052$ grains, and that in the fresh air is $178.3 \times 92.12 = 16,425$ grains, thus making a total of 47,480 grains. The weight of the water vapor per pound of the mixture is $47,480/(560.1 + 178.3) = 64.3$ grains. The percentage humidity of the mixture is therefore $64.3/109.75 = 58.6$ per cent.

As the values of c_p do not vary very much in these mixture problems, consideration of the values of the specific heat may be omitted without much error. If in the problem a final condition of 65° F and a percentage of 60 is desired it is clear that either of two methods may be pursued: first, to cool the entire mixture to the dew-point temperature of 51.0° F, at which condition the weight of water vapor is 55.44 grains per pound of dry air and, second, to cool the fresh air to such a temperature that a mixture with the recirculated air will have the water-vapor content desired. The first condition may be secured by the use of an efficient spray chamber and water with an exit temperature from the sprays of about 50°. There would have to be sufficient heat in the building from one source or another to heat the air from 51° to 65°; heating coils may be used if necessary. In the second case the water vapor in the mixture has to be $738.4 \times 55.44 = 40,937$ grains. The weight of water vapor in the recirculated air is 31,050 grains, leaving the difference of 9885 grains to be supplied by the fresh air, or 55.44 grains per pound of dry

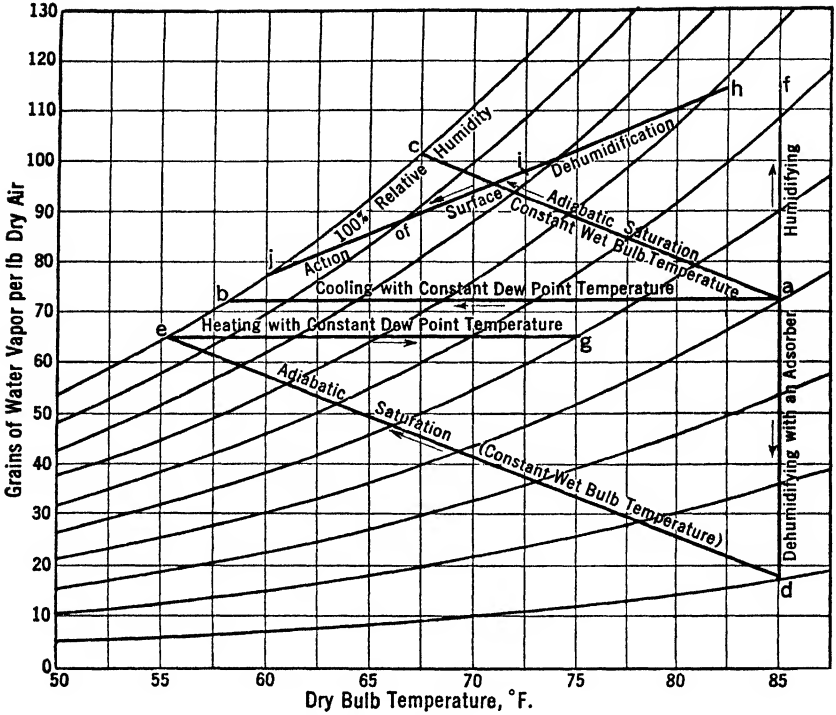


FIG. 14-2. Various air conditioning processes indicated on the psychrometric chart.

air. This amount corresponds to a dew-point temperature of 51.0° F, approximately. The temperature of the mixture would be, as before.

$$560.1 \times [0.2436 (65.0 - t_x)] = 178.3 \times [0.2436 (t_x - 51.0)]$$

$$t_x = 61.6^\circ \text{ F}$$

as compared with a dew-point temperature of 51.0° F for the first case where all the air, recirculated and fresh, goes immediately to the spray chamber. If volumes had been used, with the assumption that the two specific heats were the same, the final temperature would be

$$7500(65 - t_x) = 2500(t_x - 51)$$

from which $t_x = 61.5^\circ \text{ F}$.

The air may be cooled by means of an air washer, the principle of adiabatic saturation being used. The water in the washer is continuously recirculated until evaporated, and the air in an efficient washer can be cooled to within about 1.0° of the wet-bulb temperature although it will be saturated very nearly 100 per cent with water vapor. Many of the processes are illustrated in Fig. 14-2.

Example. Outside air is supplied to the spray chamber of an auditorium at 90° F dry-bulb and 50 per cent relative humidity. This condition corresponds to a wet-bulb temperature of about 75.2°; hence the air may be cooled to approximately 76.0° by the use of recirculated water in a spray chamber. If this

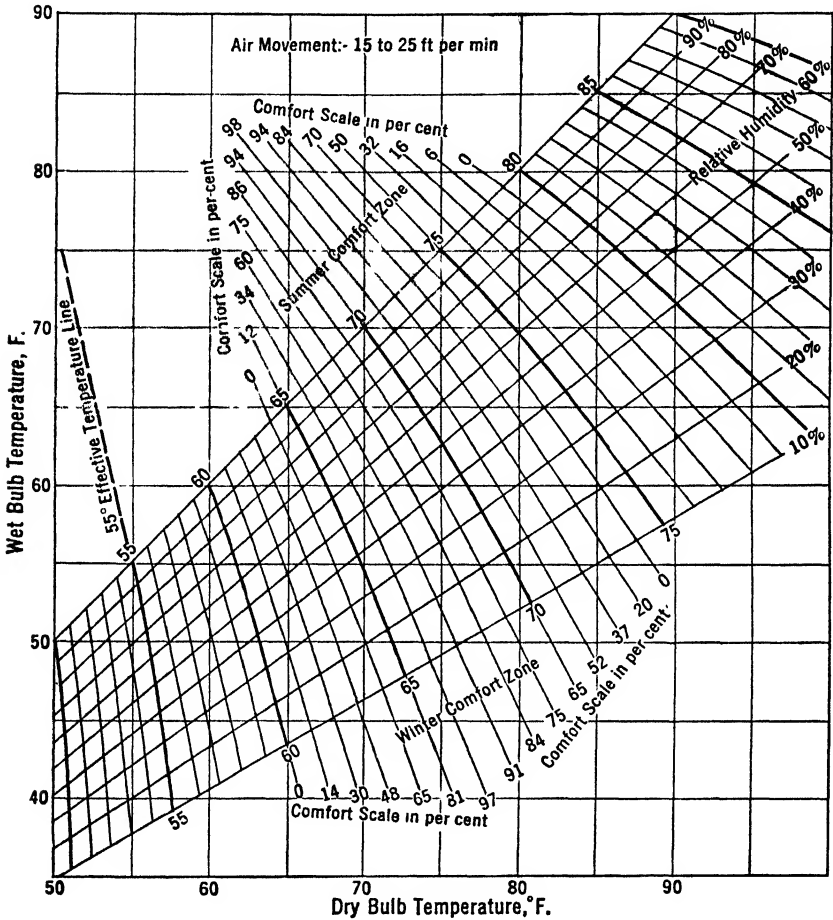


FIG. 14-3. Effective temperature chart.

air is sent to the auditorium for ventilation and for the purpose of absorption of heat and moisture the temperature may be allowed to increase to 80.0, thus giving an effective temperature of 78.1° (Fig. 14-3) instead of an effective temperature of 81.4° for the outside air, and, with the same rise of temperature of 4° F in the auditorium, the effective temperature will be about 82.9° without the use of the washer. An effective temperature of 78.1° F would be comfortable, but 82.9° would not be, according to Fig. 14-3.

METHODS EMPLOYED IN SECURING AIR CONDITIONING REQUIREMENTS

14.12. Comfort Cooling with Ice. One of the greatest applications of refrigeration, especially during the last three decades, has been in the field of *air conditioning* or "manufactured weather." Certain factories may require a temperature of 65° and 60 per cent relative humidity, whereas an instrument-manufacturing plant may require only a low humidity to prevent tarnishing of the steel during handling. One of the earliest and usually the cheapest method of removing water vapor from the air has been to condense it out by lowering the temperature, and for such a process the need of refrigeration is plainly evident.

During the last decade the term *comfort cooling* has taken on a new and more popular significance. Whereas at first it was applied only to theaters, hotels, and office buildings, the term now includes the manufacture of comfortable air conditions in the home. Air conditioning applied to comfort cooling means more than simply the control of the humidity and of the dry-bulb temperature, as the *rate of air movement* has to be considered also. The air must be moved vigorously but not sufficiently to cause drafts.

Comfort may be secured by the use of ice or of a machine. The advantage of ice is that it can take a very heavy load. Whereas any machine, as for example a heat engine, has a fixed capacity, the capacity of the ice-cooled plant depends on the melting of the ice and the absorption of heat from the air by the ice water. Ice has been found valuable for comfort cooling where the load is at once heavy and of short duration.

Both natural and manufactured ice may be advantageously employed in *comfort cooling*, as shown in Fig. 14.4, where two fluid circuits are indicated. In the first of these, ice water is circulated by means of a pump from the bottom of the ice tank to the atomizing sprays in the spray chamber or air washer. In the second circuit a motor-driven fan circulates air through the air washer and then to the space to be cooled. When needed, temperatures colder than 32° F may be secured by means of solid sodium chloride and other solid salts, mixed with crushed ice. Table 14.3 gives examples of various mixtures and their resulting temperatures. For example, a 2:1 mixture by weight of granulated calcium chloride and snow will give a temperature of -43.6° F. The use of ice or of ice and salt is justified at times, especially if the cost of ice is nominal and if the period of operation of the plant is relatively small. This is the method employed at the present time for railroad refrigerator cars. The same is true of the use of ice for household refrigerators which require a temperature of only 45° to 50° F, and for commercial

iceboxes, although the improved, automatic, small mechanical unit has also replaced a considerable proportion of commercial ice refrigeration. Since 1930 ice for the comfort cooling of homes, churches, and even theaters frequently has been employed with satisfactory and economical results, especially when only occasional cooling is required.

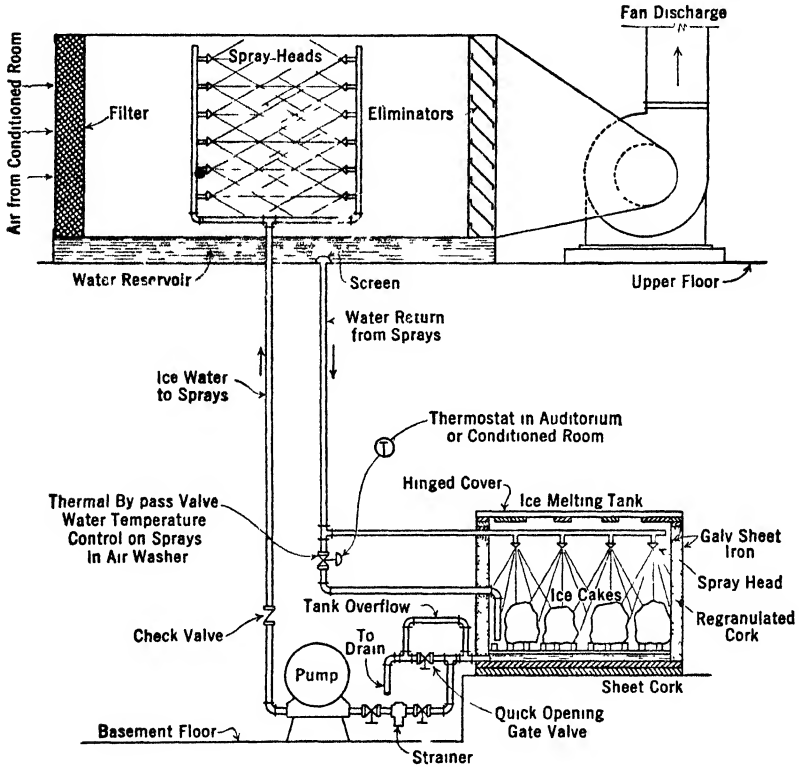


FIG. 14-4 An automatic method of conditioning air by means of ice water in a spray chamber.

14-13. Cooling by a Change of State. An example of the absorption of heat by the melting of a solid is to be found in the cooling of air caused by the melting of snow on snow-capped mountains. Another instance is the cooling of ice refrigerators and of refrigerator cars where the latent heat of fusion requires the absorption of considerable heat at 32° F during the change of state from solid ice to liquid water.

In localities where the air is naturally very dry, as in Arizona and other western states and in parts of Mexico, Egypt, and India, water in a porous jug has been cooled through the evaporation of some of the water

TABLE 14·3
FREEZING MIXTURES

	Parts by Weight	Resulting Temperature, °F
NaCl	1	-1.6
Snow	3	
CaCl ₂	1½	-27.6
Snow	1	
Dilute HNO ₃	1	-31.0
Snow	1	
KOH	4	-38.3
Snow	3	
NaCl	1	-0.4
Snow	1	
CaCl ₂	2	-43.6
Snow	1	
Salt peter	1	-11.2
Sal ammoniac	1	
Water	1	
H ₂ SO ₄	Trace	-40
HNO ₃	1	
Snow	1	

on the surface of the jug, the evaporation being occasioned by the action either of a natural wind or of an artificial air movement made by swinging the jug. The physical law in this case is that the evaporation of the surface water takes place at a temperature corresponding to the comparatively low vapor pressure of the water. This may be as much as 10°, 20°, or even 40° F below that of the air (refer to Section 14·3).

Egyptian frescoes indicate that *fanning porous jars*, filled with water, in order to cool the liquid was practiced as far back in history as 2500 B.C. According to Parks' *Wandering of a Pilgrim* it was an old custom in regions along the Indus, in Calcutta and other parts of India to manufacture ice by rapid evaporation of water into the atmosphere during the night. The water was placed in shallow pans (Fig. 14·5) resting on dry straw about a foot thick, and with night air temperatures as high

even as 40° F it was possible to freeze the water into ice from 1 to 1½ in. thick before sunrise the next morning. In this freezing of the water the air would need to be quite dry, and there would need to be considerable radiation into a cloudless sky in order to get the freezing temperature required for the rapid formation of the ice.



FIG. 14·5. An early method of making ice.

14·14. Principles of Air Washing. Air washing depends on breaking air strata so as to insure air contact with a wet solid surface. Scrubber and eliminator surfaces are arranged vertically about an inch apart with sharp bends and projecting edges, and the scrubbers are flooded with water at the top. The spray chamber (Fig 14·6) is designed for not less than 300 and usually from 500 to 600 fpm air velocity uniformly distributed across the section and is supplied with water spray nozzles directed usually toward the eliminator surfaces, although some spray chambers are designed with two banks of sprays opposed to one another. The spray nozzles are designed for 20 to 25 psig, and each at about 2½ gpm capacity. The nozzles are arranged to cover completely the section of the chamber, arranged usually with about 0.8 cu ft of section per nozzle, with a finely divided water spray in either one or two banks of sprays, uniformly spaced about 10½ in. on centers. The flooding nozzles on the scrubbing surfaces are designed for about 5 psig and for 1 to 1.5 gpm capacity per 1000 cfm. They are arranged on about 3-in. centers across the top of the scrubber plates. Three banks of nozzles, two upstream and one downstream, have approximately 100 per cent

saturation efficiency, and they use less than 15 gal of water per 1000 cu ft of air. The air tends to become saturated at the temperature of the exit water which usually has a rise of temperature of 6° to 10° F. The resistance to the air flow is usually designed for about 0.55 in. of water although it may vary from 0.25 to 0.6 in. of water pressure. The amount of water used varies from 5 gpm for the spray and 1.0 for the flooding nozzles, with a humidifying effect of 0.7, to 10 gpm for sprays

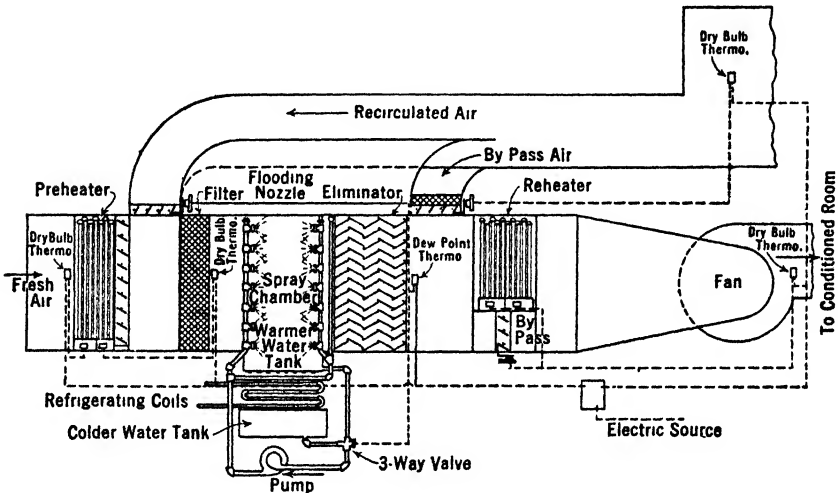


FIG. 14-6. The spray chamber, showing methods of thermostatic control.

and 1.3 for the flooding nozzles with a humidifying effect of 0.9 per 1000 cfm supplied to the spray chamber. In a well-designed spray chamber the air can be cooled within 1.0° F of the wet-bulb temperature.

14-15. Temperature and Humidity Control. Thermostats are of two general designs, the self-contained instruments which have enough power to operate the control mechanism without assistance and the pilot thermostat which initiates the movement, such as with an air pressure of about 15 psi or an electric switch. The pilot type may be graduated for slow movement or quick action. For quick action, as with opening or closing by an electric switch, the control valve also is either open or closed. Figure 14-6 gives one design of a thermostatic control.

Most thermostats are based on the expansion of a solid or of a volatile liquid. The volatile liquid, as sulphur dioxide, may be contained in a metallic bulb placed in the location where control is desired with a flexible tube extending to the diaphragm or siphon. The change of pressure Δp occasioned by a change of temperature Δt gives the necessary force to operate the valve which may start a machine such as the com-

pressor, or regulate the amount or temperature of the water, or, perhaps, steam, in the sprays or coils as the case may be.

The relative humidity is controlled by adjusting both the dry-bulb and the dew-point temperatures. As a rule, the dry-bulb temperature is determined by other considerations, necessitating the varying of the

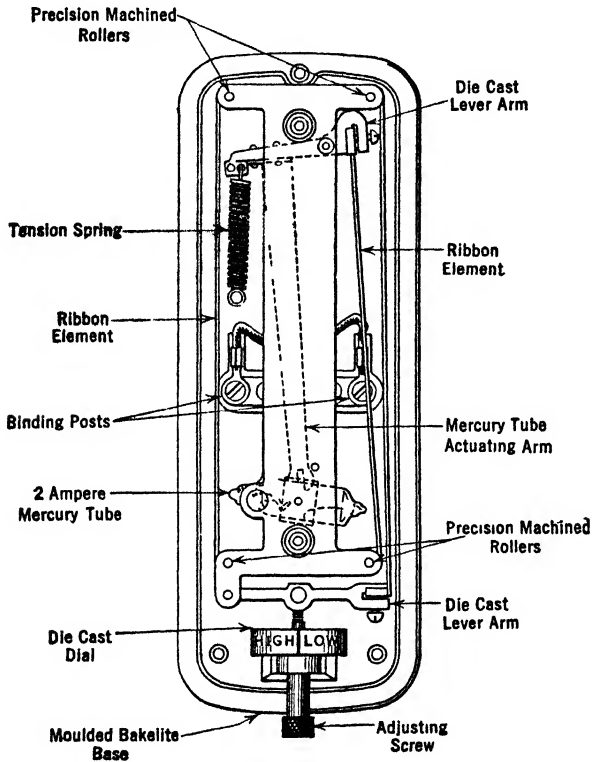


FIG 14 7. Humidity control

absolute humidity to suit the conditions imposed on the plant. If a spray chamber is used, the temperature of the water will vary the dew-point temperature of the air leaving the sprays, thus increasing or decreasing the absolute humidity in direct proportion to the air temperature as shown in Table 14·1. The dew-point temperature may be properly maintained by means of a humidostat placed in the conditioned space or on the suction side of the fan. The humidostat (Fig. 14·7) is an instrument composed in part of some hygroscopic material such as human hair, certain varieties of wood elements, and certain cellulose materials which react quickly to a change in the water vapor content in the air.

In addition to the separate control of the dry-bulb temperature and the absolute humidity the percentage humidity may be held constant by the application of the principle of the wet and dry temperature. The conditioned room or the return duct may have a wet- and a dry-bulb thermostat set for the conditions desired, one arranged to control the water on the sprays or in the finned coils with or without atomizing sprays, and the other to maintain at the desired point the dry-bulb temperature of the air entering the conditioned room.

If it may be assumed that the air leaving the spray chamber is at the dew-point temperature a thermostat placed at this point may be made to control the spray water or the water temperature entering the coils, and a thermostat in the conditioned room may be arranged to regulate the amount of by-pass air or the heat supplied in order to bring about the desired temperature.

14·16. Air Cleaning Devices. It has been found essential that not only the fresh air but also the recirculated air (Fig. 14·6) should be made to pass through air filters which are usually classified as dry filters or viscous coated filters. The dry filter may be a fine felt pad which should have a low resistance to the flow of air but a high dust-resisting power. In the viscous type of filter some of the dust may be screened out, but it is expected that the viscous coated surface will retain the dust particles. In both types a high dust-holding capacity is required, as well as economy in first cost and upkeep.

14·17. Entrance and Exit. The entrances and exits for ventilation and cooling must be such as to make the ventilation effective with thorough mixing of the air in all parts of the room and yet with velocities which will not cause objectionable drafts. This may be accomplished in relatively small rooms by the entrance of the air near the ceiling or at least 7 ft above the floor with the initial direction parallel to the ceiling. Local recirculation methods cool and dehumidify a small portion of the air which is ejected at relatively high velocities up to about 1200 fpm by means of special nozzles designed to reduce noise.

The air can be distributed in large rooms like theater auditoriums in two ways. In the first method it may be blown down through a deflector panel slightly below the ceiling level. In the other method, called the ejector system, the air is blown through a series of nozzles placed near the ceiling in the rear and directed toward the stage. The colder air is expected to carry by an inductive effect some three or four times as much of the warmer air, and to cause a reversed, uniformly distributed current at the level of the auditorium floor.

The *unit cooler* (Fig. 14·8) which is a good example of air conditioning principles, has been applied to a large number of commercial applica-

tions, especially where space is limited or where elaborate changes in the existing building are not justified in order to install a duct system. It is a self-contained unit with coils or sprays or both, eliminator surfaces, fan and motor and adjustable distribution outlets. The air is delivered horizontally near the ceiling at velocities up to 1200 fpm, and the return air enters at or near the floor. Either dry pipes or sprays over the coils may be used, but the sprays have the advantage where humidity control is important and where frost on the coils has to be prevented. Water is used for the sprays for temperatures above 32° F, and brine for temperatures of the air at and below 32° F. The unit cooler is compact and easily moved, and the circulation is positive and uniform. The coils may be designed for direct expansion, for brine or water (Fig. 14·9).

14·18. Spray Chambers or Coils.

The spray chamber is not a perfect cleansing medium as it will never remove oily or greasy dust particles, and therefore filters are necessary whether

or not sprays or coils are used. The cross-sectional area is the same for both as it is determined by the maximum permissible velocity of the air. The spray chamber has to be about 9 ft long as compared with 9 in. for coils or 4½ ft if coils and humidifier are used.

The life of an air washer is frequently found to be a short one, the maintenance and repairs are high, and the spray nozzles are continually becoming clogged. The eliminators may have to be renewed in 8 or 10 years unless they are made of a non-ferrous metal. The pump horsepower is about 25 per cent less with coils, and if they are properly installed the air can be cooled to a lower temperature than by means of sprays. Dehumidification can be obtained without reducing the temperature of the main stream to the dew point, and under certain sets of conditions the relative humidity of the exit air mixture can be controlled without the by-pass method or the necessity of heating the air after dehumidification. The spray units are a little more noisy than the dehumidifier using coils.

14·19. Dehumidification with Extended Surfaces. For some time it has been known that dehumidification may be accomplished by means of pipe surfaces without cooling the entire volume of air to the

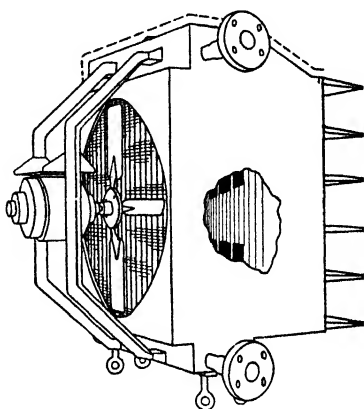


FIG. 14·8. The unit cooler, ceiling type.

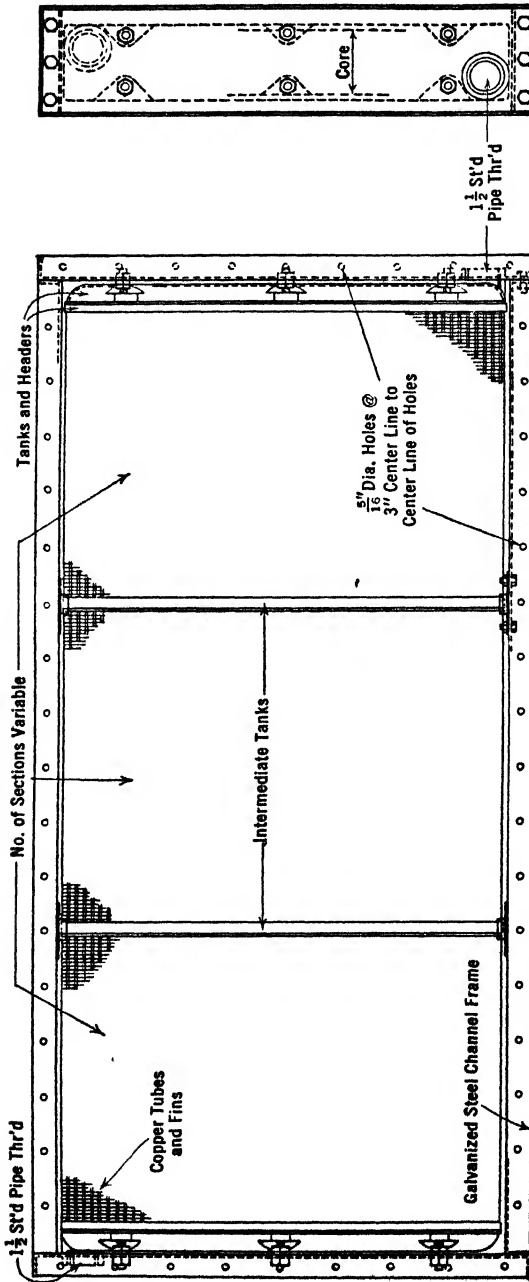


FIG. 14-9. Finned coils for the use of cold water.

dew-point temperature and without recourse to the by-pass system. Carrier and Busey^{||} state that "the air in the surface film will always be saturated at the temperature of this film, while the main body of the air passing through the cooler may be considerably above the saturation point, even after considerable moisture has been condensed out of it, the final state of the air being simply the result of a mixture of air saturated at the mean film temperature with the main body of the air." This means that the air at the pipe surface is in viscous flow while the main body of the air is in turbulent flow. If the pipe surface is held at or below the dew-point temperature, condensation of water vapor will occur, owing to the reduction of the dew-point temperature of the air at the surface, and therefore the mixture will have a reduced dew-point temperature and a reduced absolute humidity in *the same proportion*.

Keevil and Lewis[¶] and W. L. Knaus* have developed a theory on the dehumidification of air which is essentially as follows.

In Fig. 14·2 let the temperature and the humidity of the main stream be indicated by h , the pipe surface temperature by j , and the final main stream after mixture by i . Then the flow of sensible heat through the air film at the pipe surface is indicated by

$$Ws dt = hA(t_h - t_i) \tag{14·17}$$

and the diffusion of water vapor is

$$W dH = k'A(H_h - H_i) \tag{14·18}$$

Therefore

$$\frac{dH}{dt} = \frac{sk'(H_h - H_i)}{h(t_h - t_i)}$$

or

$$\frac{dH}{H_h - H_i} = \frac{sk'}{h} \frac{dt}{t_h - t_i} \tag{14·19}$$

In the above, W is the weight of dry air, in pounds; s is specific heat of the mixture $c_{p,m}$; h is the coefficient of sensible heat transfer; k' is the coefficient of vapor diffusion; A is the area of the pipe surface exposed to the air; H is the absolute humidity.

The relationship of the coefficients has been studied by Arnold,[†] Merkel,[‡] and others, and the conclusion is that for most engineering purposes $k'/h = s$ and that the line hij (Fig. 14·2) may be assumed to be

^{||} Carrier and Busey, *Trans. A.S.M.E.*, p. 1096, 1911.

[¶] Keevil and Lewis, *Ind. Eng. Chem.*, Vol. 20, No. 10, 1928.

* Knaus, *Refrig. Eng.*, Vol. 29, Nos. 1 and 2.

[†] Arnold, *Physics*, Vol. 4.

[‡] Merkel, *Verdunstungs-Kuehling*.

straight. Then

$$\log \frac{H_h - H_j}{H_i - H_j} = \log \frac{t_h - t_j}{t_i - t_j} \quad \text{or} \quad \frac{H_h - H_j}{H_i - H_j} = \frac{t_h - t_j}{t_i - t_j} \quad (14.20)$$

From equation 14.20 it is clear that the control of the dry-bulb temperature and of absolute humidity, obtained by the other methods of dehumidification, is lost in the present method inasmuch as the temperature and humidity of the mixture must remain in the same proportion as in the initial condition. There is no independent control of temperature or of absolute humidity. In air conditioning problems where water vapor and sensible heat have to be removed the pipe cooler, or preferably the extended pipe surface cooler, using the cooling water in counter flow to the direction of the air movement, will probably be the most satisfactory solution.

Example. Air at 80.8° dry-bulb and 68° wet-bulb ($H_1 = 0.0116$) is brought into contact with refrigerated surfaces with a surface temperature of 52.6° F corresponding to an absolute humidity of 0.0084 lb/lb of dry air. If the dry-bulb temperature of the mixture is 65.3°, what will be the absolute and relative humidity of the mixture?

Solution.

$$\frac{0.0116 - 0.0084}{80.8 - 52.6} = \frac{0.0116 - H_b}{80.8 - 65.3} \quad H_b = 0.0098$$

from which the relative humidity becomes 73.4 per cent. Tests on air cooling reported by Knaus gave results of the absolute humidity of 0.0097, which would indicate that the line *abc* is in reality slightly concave.

If a lower humidity is desired, for example 71 per cent, with the same dry-bulb temperature of the mixture a lower surface temperature of the pipes will be required. By drawing a line in Fig. 14.1 similar to the line *hij* in Fig. 14.2, this will be seen to be 40° F, and it will be observed also that a reduction of the surface temperature to 35° will not improve the relative humidity appreciably.

14.20. Dehumidification with Calcium Chloride. At times water vapor may be removed from the air satisfactorily by means of solid or liquid solutions of calcium chloride. The theory of such absorption of water vapor is identified with the vapor pressures of calcium chloride, e.g.,

- CaCl₂·H₂O has less than 1.0 mm of mercury vapor pressure
- CaCl₂·2H₂O has less than 1.0 mm of mercury vapor pressure
- CaCl₂·4H₂O has less than 3.4 mm of mercury vapor pressure
- CaCl₂·6H₂O has less than 5.1 mm of mercury vapor pressure
- CaCl₂ saturated solution 6.5

If air with a greater water vapor partial pressure is brought into contact with solid or liquid solutions of calcium chloride there will be a tendency for equilibrium of the vapor pressures, which will result in the absorption of water vapor from the air. To do this, heat will be evolved, equal to the latent heat of liquefaction and the heat of solution, which, if not removed by a cooling water device, will need to be absorbed by the air.

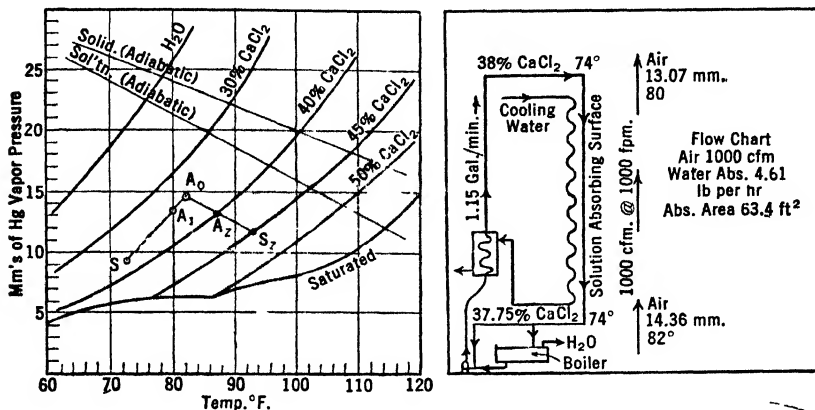


FIG. 14-10.

Tables of the properties of calcium chloride may be used, or a suitable chart§ as seen in Fig. 14-10. Two problems will make the process of dehumidification clear.

Example. Calcium chloride solution of 38 per cent (Fig. 14-10) is cooled with water to maintain a constant temperature of 74°. Air enters with an initial temperature of 82° and a vapor pressure of 14.36 mm of mercury. The final condition is 80° and 13.07 mm of mercury, whereas the brine solution has a vapor pressure of 9.4 mm mercury.

Solution.

$$\Delta p = \frac{14.36 + 13.07}{2} - 9.4 = 4.32$$

The water absorbed per 1000 cfm is given by

$$W = 3.57 \Delta p \quad \text{or} \quad W = 3.57 \times (14.36 - 13.07) = 4.61 \text{ lb/hr}$$

The area exposed to the air must be

$$A = \frac{59.8 \times 4.61}{4.32} = 63.8 \text{ sq ft}$$

ADIABATIC ABSORPTION OF WATER VAPOR. In this case the solution will be maintained at 45 per cent with a partial pressure of 11.6 mm of mercury water

§ *Dehumidification of Air*, The Dow Chemical Company.

TABLE 14·4

SUGGESTED TEMPERATURE AND RELATIVE HUMIDITY FOR INDUSTRIAL PROCESSES

Product	Process	Temperature °F	Relative Humidity, per cent
Cotton.....	Carding	68-73	50
	Combing	68-73	60-65
	Roving	68-73	50-60
	Spinning	68-73	60-65
	Spooling, twisting	68-73	65
	Warping	68-73	65
Wool.....	Weaving	68-73	75-80
	Carding	73-77	65-70
	Spinning	73-77	55-60
	Weaving	68-73	50-55
Silk.....	Storage for shipping	68-73	55-60
	Dressing	69-77	60-65
	Spinning	69-77	65-70
	Throwing	69-77	65-70
Rayon.....	Weaving	69-77	60-70
	Filament mfg.	—	—
	Winding	70-80	40-50
Confectionery... ..	Weaving	80-85	55-60
	Chocolate enrobing	64	55
	Hard candy making	69	50
Tobacco.....	Storage	30-59	70-55
	Softening	84	85
	Cigar and cigarette making	69-73	55-70
Printing.....	Lithographing	69	45
	Relief and offset	77	45
	Folding	77	65
Baking.....	Binding	69	45
	Dough — fermentation	80	65
	Proofing	89-95	80-90
Electrical cable.....	Loaf cooling	69	65
Cellulose lacquers.....	Winding insulation	104	5
Munitions.....	Application	75	20
Cereals.....	Fuse loading	69	55
	Seal packing		
	prepared crisp cereals	73	45.5

pressure and a temperature of 92.8°, as a more concentrated solution may clog the pipes. The air leaving will be at 86.7°, and the vapor pressure will be 13.07 mm. The log difference in the partial pressures Δp is 1.99 mm, and the water absorbed per 1000 cfm is $3.57 \times 1.29 = 4.61$ lb/hr. The area of the absorbing surface is

$$A = \frac{59.8 \times 4.61}{1.99} = 138.5 \text{ sq ft}$$

14-21. Ventilation Standards. Industrial processes have individual conditions of operation with regard to dry-bulb temperature and

TABLE 14-5
VENTILATION STANDARDS

Application	Cfm per Person	
	Preferred ¹	Minimum ¹
Apartment ²	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 ²	15	10
Hotel room	15	10
Office, general	15	10
Office, private ²	30	15
Restaurant	15	12
Shop, retail	10	7½
Theater	7½	5
Notes: For general application: Each person, not smoking	7½	5
Each person, smoking:	30	25

Design occupancy:

1. The design load calculations shall be based on the stated occupancy of the building during the time of maximum design conditions.
2. The heat given off by each occupant shall be calculated as not less than that in Tables 10-11a and 10-11b.

¹ Whether the preferred or minimum values are used, the outside air ducts shall be of a size to admit at least 50 per cent more than the preferred values.

² 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.

relative humidity. In addition there are certain ventilation standards for the preferred volume of air to be delivered and the minimum requirements for auditorium and office work. These values are given in Tables 14-4 and 14-5.

14.22. Typical Air Conditioning Design Examples. The following examples illustrate the methods used in solving air conditioning and comfort cooling calculations. These consist of a consideration of a factory, a restaurant, a theater, and finally a detached residence.

Example. A room in an industrial plant is to be maintained at a temperature of 65° F and 60 per cent relative humidity. Its cubical content is 204,300 cu ft; it has 1248 sq ft of glass and 908 sq ft of wall surface exposed to the sun's rays, and 2154 sq ft glass, 1863 sq ft of wall, 2728 sq ft of partition, and 12,770 sq ft of floor area subject to heat leakage. The outside temperature will be taken as 95° maximum, with a wet-bulb temperature of 76° F. The infiltration loss will be assumed as a complete volume change of the air in the room calculated on the basis of the inside air temperature and water-vapor content, e.g., 65° F and 60 per cent relative humidity, corresponding to a dew-point temperature of 51.0° F in 5 hr. The maximum number of operators will be taken as 500 persons at any one time and the illumination as thirty-nine 250-watt lamps. It is required to find the total refrigeration and the fan capacity necessary to circulate the air.

Solution. A certain amount of water vapor will enter the room. This is approximately the water vapor contained in the air entering by infiltration together with that exhaled by the operators. If the latter amount is taken as 3.0 Btu of latent heat per minute the total weight will be (allowing 10 per cent to be exhausted and the latent heat of liquefaction being taken at 49° F)

$$\frac{3.0 \times 7000}{1064} = 19.74 \text{ grains per minute per person}$$

or

$$19.74 \times 500 \times 0.9 = 8883 \text{ grains total} \\ = 1.269 \text{ lb}$$

The water vapor in the outside air entering by infiltration (and allowing for 10 per cent to be exhausted) is

$$\frac{204,300 \times (0.015 - 0.0080) \times 0.9}{5 \times 60 \times 13.39} = 0.3204 \text{ lb/min}$$

Hence the total is 1.589 lb/min.

The sensible heat, per hour, is:

Heat due to direct solar radiation

For the glass 1248 × 30.0 = 37,440 Btu

For the wall 908 × 6.0 = 5,450

Operators 500 × 320 = 160,000

Conduction

For the glass 2154 × [0.46(95 - 65)] = 29,710

For the wall 1863 × [0.075(95 - 65)] = 4,190

For the partition 2728 × [0.09(95 - 65)] = 7,370

For the floor 12,770 × [0.08(95 - 65)] = 30,650

Infiltration	$\frac{204,300 \times [0.2465(95 - 65)]}{5 \times 13.39}$	= 22,570
Illumination	$\frac{39 \times 250 \times 3600 \times 0.9485}{1000}$	= 33,290
	Total	330,670 Btu/hr
	Add 5 per cent, for the unforeseen	16,533
	Estimate, 5-hp fan motor	12,720
	Total	
	Per hour	359,923
	Per minute	5998.7 Btu

The sensible heat, 5999 Btu/min, is the total heat entering the room from all the sources tending to raise the temperature of the air in the room, and this is the only factor affecting the capacity of the fan. As refrigeration is to be accomplished by means of water sprays in a spray chamber it is necessary to move enough air through the chamber so that heat will be absorbed by the water at the rate of 5999 Btu/min. The air so circulated leaves the room at 65° and re-enters it from the sprays at 47.0°, thereby permitting a rise of temperature of 18° F.

If an amount of water vapor amounting to 1.589 lb or 11,126 grains must be removed per minute it is evident that some temperature must be chosen below the dew-point temperature in the room of 51.0° F. Evidently there will be only one temperature of the air leaving the spray chamber, or cooling coils, that will remove both the sensible and latent heats. If the temperature of the air leaving the spray chamber is t , the weight of dry air in circulation will be

$$\frac{Q}{c_{pm} \Delta t} = \frac{5999}{0.2434 (65 - 47)} = 1369 \text{ lb/min}$$

The water vapor removed per pound of dry air will be $11,126/1369 = 8.13$ grains.

As the absolute humidity of 55.4 grains per pound of dry air, corresponding to 65° and 60 per cent relative humidity, is identified with a dew-point temperature of 51.0° F, the required absolute humidity in the spray chamber will be $55.4 - 8.1 = 47.3$ grains. Referring to Table 14.1, the absolute humidity at 47° is seen to be 47.76 grains. The value of c_{pm} is taken as 0.2434 as a compromise (remembering that the cooling from 65° to 51.0° F is at a constant absolute humidity and with a value of $c_p = 0.2435$).

REFRIGERATION REQUIRED. The refrigerating load consists of

Cooling the fresh air, 10 per cent of 1369 lb, from a wet-bulb temperature of 76° F to a dew-point temperature of 47.0.	
Total heat, enthalpy, at 76°, from Table 14.1	39.445 Btu
Total heat, enthalpy, at 47.0	18.640 Btu
Refrigeration per pound of dry air	
$39.445 - 18.640 - 15 (0.015 - 0.00682)$	= 20.682 Btu
Total refrigeration of the fresh air	$136.9 \times 20.684 = 2831.6$ Btu
Cooling of the recirculated air	$0.9 \times 5999 = 5399$

The water vapor entering by infiltration is 0.3204 lb/min, which will need to be condensed at approximately 49° F. If the latent heat is taken as 1064 Btu the refrigeration required will be

	$0.3240 \times 1064 =$	341 Btu
To these must be added;		
Latent heat of the operators	$0.9 \times 3.0 \times 500 =$	1350 Btu
Estimated heat equivalent of the work of the water pump	$=$	<u>200</u>
	Total	10,122
Add 5 per cent for safety		506
	Total	= 10,628 Btu
		= 53.14 tons
		of refrigeration

Example. In a restaurant the sensible heat entering is 200,000 Btu and the latent heat is 373,000 grains of water vapor per hour. Outside air is to be taken at 95° F dry-bulb and 75° F wet-bulb, and the restaurant is to be held at 80° dry-bulb and 65° wet-bulb, corresponding to 56.7° F dew point and 67.9 grains (0.00970 lb) of water vapor per pound of dry air, and the conditioned air will enter the room at 68° F. As the relative humidity is to be kept at approximately 44 per cent the air from the spray chamber must be able to absorb the water vapor and still not exceed the requirement of 67.9 grains per pound of dry air.

Taking the value of $c_{p,m} = 0.2440$, the heat absorbed per pound of air delivered to the restaurant at 68° F will be $12 \times 0.2440 = 2.928$ Btu and the number of pounds to be circulated per minute is $200,000 / (2.928 \times 60) = 1138.4$ lb. This 1138.4 pounds of air per minute will absorb $373,000 / 60 = 6217$ grains of water vapor per minute, or each pound will absorb 5.4611 grains. The dew-point temperature in the spray chamber, for a water vapor content of $67.9 - 5.5 = 62.4$ grains, is 54.17° F. If the air is reheated to 68° the volume handled by the fan will be $1138.4 \times 13.49 = 15,357$ cu ft. If 10 per cent of the fan capacity is fresh air, 113.8 lb/min must be cooled from the outside conditions of 95° dry-bulb and 75° wet-bulb temperature to the dew-point temperature in the spray chamber. This will require

$$113.8\{38.5 - [22.66 + (0.0141 - 0.0089) 22]\} = 1788.9 \text{ Btu/min}$$

and the recirculated air will give up to the water in the sprays

$$[1024.6(80 - 54.17)0.2444] + \frac{0.9 \times 373,000 \times 1060.5}{60 \times 7000} = 6468 + 848 \text{ Btu per min}$$

making a total of 9100 Btu/min or 45.5 tons of refrigeration.

USING THE BY-PASS. It is not necessary to pass all the air through the spray chamber, but if a part is by-passed a calculation of the amount is necessary. If x is the decimal part by weight of the recirculated air and y is the decimal part by weight that is conditioned, then $x + y = 1.0$. Let t represent the tempera-

ture of the conditioned air leaving the air washer. Then

$$xc_{p1}(80 - 68) = yc_{p2}(68 - t)$$

but $c_{p1} = c_{p2}$, very nearly. Therefore

$$80x + yt = 68$$

The weight of water vapor in the recirculated air plus the amount in the conditioned air must be equal to the amount in the mixture. For *small* changes of temperature the water vapor content is approximately directly proportional to the temperature in Fahrenheit degrees. Therefore we can say, without appreciable error (taking the respective dew-point temperatures) that

$$56.7x + yt = 54.17$$

and, solving these equations for x , y , and t ,

$$x = 0.594$$

$$y = 0.406$$

$$t = 50.5^\circ \text{F}$$

where the temperature of 50.5°F is the dew-point temperature for the fraction that is to be conditioned. Note that this is lower than the dew-point temperature in the first calculation, which was 54.17° and which has as the water vapor content required in the problem 62.4 grains per pound of dry air.

In the first calculation the fan circulated 1138.4 lb of air per minute, so that the amount by-passed will be $1138.4 \times 0.594 = 676.2$ lb/min, and the amount conditioned will be $1138.4 \times 0.406 = 462.2$ lb. Of the 462.2 lb a weight of 113.8 lb of fresh air will be brought in as before, so that the amount drawn in from the restaurant will be $462.2 - 113.8 = 348.4$ lb. The difference, or 113.8 lb, will be exhausted or permitted to escape through the windows or doors into the atmosphere because of the slight positive pressure in the restaurant. The refrigeration becomes

$$\begin{aligned} 113.84 (h_{75^\circ} - h_{50.49}) &= 113.84 [38.50 - 20.53 - (0.014 - 0.0078)19] \\ &= 113.8 \times 17.85 = 2032 \text{ Btu,} \end{aligned}$$

$$\begin{aligned} 348.4 (h_{65^\circ} - h_{50.49}) &= 348.4 [29.977 - 20.53 - (0.0019 \times 19)] \\ &= 348.4 \times 9.41 = 3278 \text{ Btu} \end{aligned}$$

The total refrigeration becomes 5310 Btu or 26.55 tons of refrigeration, and there will be no need of reheating the air. Figure 14.11 shows how comfort cooling may be applied to an office building. A cooling tower is indicated on the roof, but the air ducts are not shown.

14.23. Theater Cooling. Theater or auditorium cooling is a special problem in air conditioning. It is a control not only of the dry-bulb temperature and the humidity but also of the air velocity, with the added stipulation that there must not be any drafts. The entering air must be tempered before reaching the people present, so that there will not be any sensation of chill.

PSYCHROMETRIC LOADS

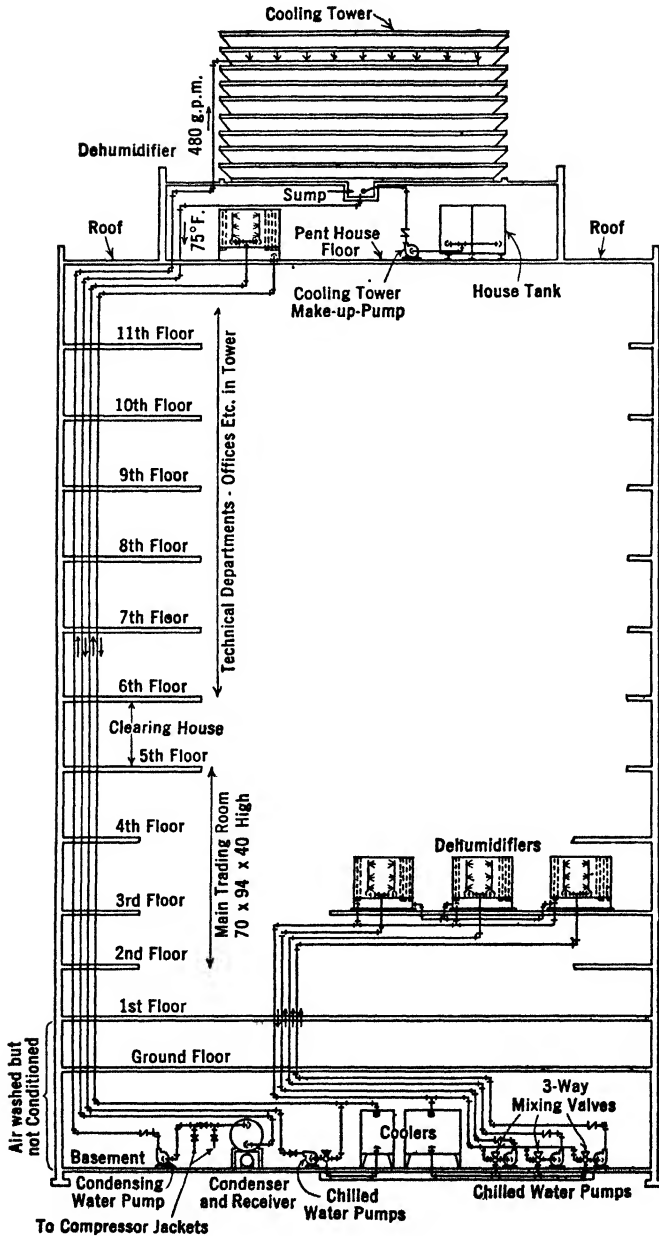


FIG. 14-11. The Los Angeles Board of Trade Building.

The amount of air supplied per person, in the absence of city ordinances, varies considerably, but apparently with equally satisfactory results, from 12 to 30 cfm per person, of which approximately 5 cu ft are fresh air. The difference in the temperature of the air entering and leaving may vary from 10° to 20° F, depending on the amount circulated per person. As a rule the patron load is about 75 per cent of the total refrigeration required, and the other 25 per cent consists of heat leakage, the heating effect of illumination, machinery, infiltration, etc. The tonnage should not be calculated on the maximum number of people and the maximum dry-bulb temperature on record for there is always a chance, unless forbidden by city ordinance, of supplying a smaller amount of fresh air and reducing the load in consequence.

Ice has been used for theater cooling with good results where the load factor is small and the ice is nominal in cost. One especially good feature of the use of ice is that it will take a heavy overload easily, and the first cost of the cooling equipment is less than that for machinery. In the theater air conditioning problem the amount of air to be supplied and the required refrigeration can be calculated from the following typical example.

Example. A theater (Fig. 14-12) is to be designed for air conditioning for 3000 persons. Outside air is to be taken at 90° F dry-bulb and 50 per cent relative humidity, while the inside dry-bulb temperature is to be maintained at 78° and the wet-bulb temperature is to be 64.5°. The sensible heat load per person is 233 Btu/hr, and the latent heat is 1080 grains of water vapor per person per hour. All other sources of sensible heat amount to 300,000 Btu/hr, and 25 per cent of the air circulated is to be fresh air. The air will enter the theater at 68°. As it is likely that there is a slight positive pressure in the auditorium the infiltration of air is assumed to be zero. Find the refrigeration required when all the air passes through the spray chamber, and also when the by-pass method is used. Find the weight of exhaust steam required for heating, the tonnage when no fresh air and when the maximum fresh air is supplied, the size of the compressor for a twin cylinder (single-acting), and the horsepower required when dichlorodifluoromethane is used with the by-pass method of operation.

Solution. From equation 14-10 (and referring to Fig. 14-13)

$$\begin{aligned}
 H_c &= \frac{H_d L_d - [0.24 (t_c - t_d)]}{L_d + [0.453 (t_c - t_d)]} \text{ pounds per pound of dry air} \\
 &= \frac{(0.01297 \times 1055.7) - (0.24 \times 13.5)}{1055.7 + (0.453 \times 13.5)} = 0.00984 \text{ lb}
 \end{aligned}$$

If the value of c_{pm} at f is 0.2442

$$\begin{aligned}
 \text{Fan capacity} &= \frac{999,000}{60 \times 10 \times 0.2442} \\
 &= 6818 \text{ lb}
 \end{aligned}$$

The water vapor absorbed by each pound of air at

$$f = \frac{3000 \times 1080}{60 \times 6818 \times 7000} = 0.001131 \text{ lb}$$

Therefore the temperature $t_f = 53.55^\circ \text{ F}$ and the fan capacity at the temperature $t_f = 6818 \times 13.111 = 89,391 \text{ cu ft}$.

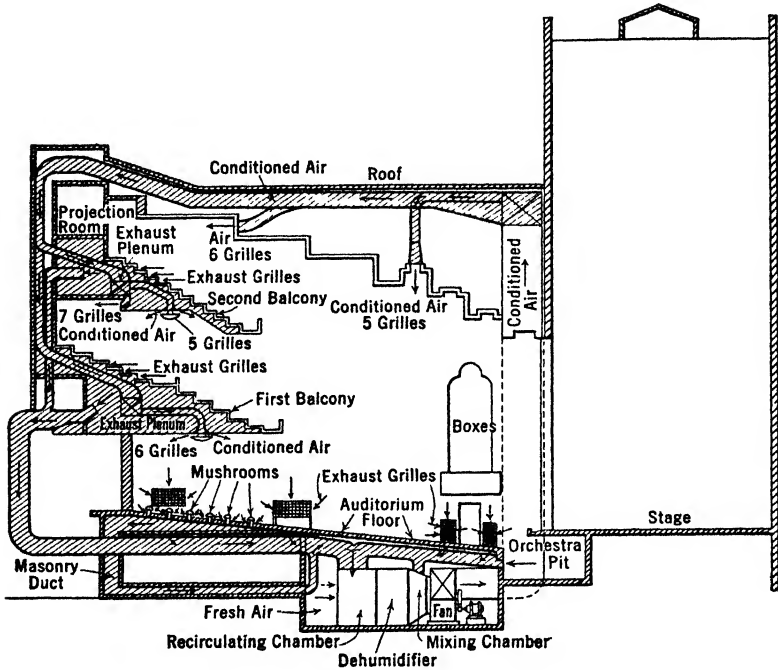


FIG. 14-12. Section of a modern theater, showing air circulation.

The heat supplied to raise the air temperature from 53.55° to 68° F is

$$6818 \times 0.2439 \times 14.45 = 24,029 \text{ Btu/min}$$

The weight of steam at atmospheric pressure required for this heating is

$$\frac{24,029}{970.2} = 24.76 \text{ lb/min}$$

If a counter flow exchanger is used, the temperature of the air entering the exchanger from the auditorium being 78° , and of that leaving the spray chamber for the exchanger being 53.55° F , the mean temperature difference will be approximately 10.0° F . Therefore

$$60 \times 6818 \times 0.2445 \times 14.4 = A \times 3.0 \times 10.0$$

$$A = 48,010 \text{ sq ft}$$

which is absurd, indicating that in this example the exchanger principle is not practical.

The refrigeration is

$$Q = 0.75 \times 6818 \times [(78 - 56.9) 0.2444 + 24.354 - 22.296 - (0.00113 \times 21.55)] = 36,770 \text{ Btu}$$

$$Q = 0.25 \times 6818 [(90 - 69.6) 0.2470 + 33.632 - 22.296 - (0.00681 \times 21.55)] = 27,661$$

$$\text{Total} = 64,430 \text{ Btu/min} = 322.2 \text{ tons}$$

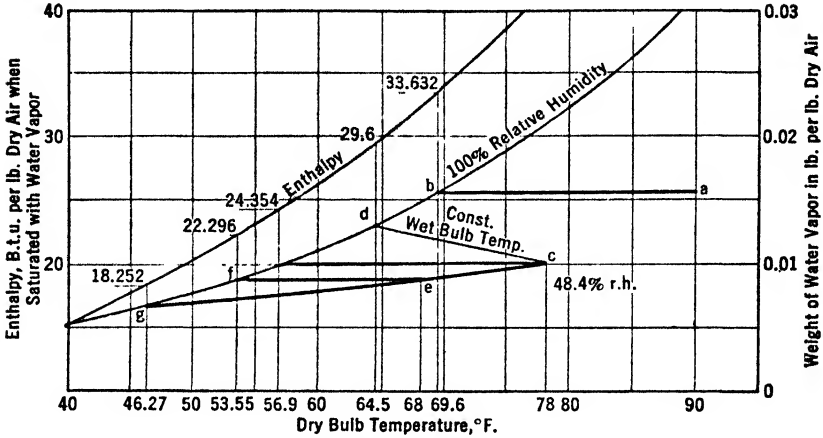


FIG. 14-13. Problem in comfort cooling illustrated on the psychrometric chart.

With the by-pass, let x = the decimal part by weight passing through the sprays; y = the decimal part by weight recirculated. Then

$$x(68 - t) = y(78 - 68) \quad \text{and} \quad xt + 56.9y = 53.55$$

and

$$x = 0.3152 \quad y = 0.6848 \quad \text{and} \quad t = 46.27^\circ \text{ F}$$

The refrigeration becomes

$$Q = 0.25 \times 6818 [5.039 + 33.632 - 18.255 - (0.01552 - 0.00663)14.27] = 34,584 \text{ Btu/min for the fresh air}$$

$$Q = [(0.315 - 0.25) 6818] \times [(24.354 - 18.255) - (0.00321 \times 14.27) + 5.157] = 4967.9 \text{ for the recirculated air}$$

$$\text{Total} = 39,552 \text{ Btu} = 197.8 \text{ tons}$$

The weight of steam required for heating becomes

$$\frac{6818 (68 - 53.55) 0.244}{970.2} = 24.78 \text{ lb/min}$$

The piston displacement of the compressor (for 40° evaporation and 90° F

liquefaction and 90 per cent actual volumetric efficiency) is

$$\frac{200 \times 0.792}{(82.71 - 28.70) 0.9} \times 197.79 = 644.52 \text{ cfm} = 1,113,730 \text{ cu in./min}$$

If the rpm is 200, $d = 12.10$ in., if two single-acting twin vertical compressors are used with the stroke equal to the diameter.

The hp of the compressor is

$$\frac{200 \times 6.18 \times 197.8}{[(82.71 - 28.70) 0.9] \times 42.44} = 118.5$$

The tonnage (using no fresh air) is

$$6818 \times 0.315 \times 11.21 = 24,075.4 \text{ Btu} = 120.4 \text{ tons}$$

The tonnage with the maximum fresh air is

$$0.315 \times 6818 \times 20.29 = 43,576 \text{ Btu} = 217.8 \text{ tons}$$

It is quite evident from the preceding examples that the refrigeration involved in the condensate removed at the temperature of the coils, or the exit temperature from the spray chamber, is small and may be omitted without appreciable error. Also the method of calculating the enthalpy of an unsaturated mixture, shown in the examples of Section 14·7, appears to be a shorter one than that involving the specific heat of the mixture, and it may be advantageous at times to use the first method.

14·24. Comfort Cooling Research House. The principal features of comfort cooling of dwellings can be brought out best by reviewing the results of tests on a research residence built and equipped especially for testing purposes.

The research residence is a detached frame building of common construction involving clapboard, building paper, 2- by 6-in. studs, wood lath, and plaster with a rough sand finish. At the time of the tests no insulation, weather strips, or storm windows were being used. For summer cooling, the sun porch and the attic were shut off and the kitchen and laundry were not put to their ordinary use. A preliminary set of tests indicated that the air change by infiltration was equal to a total volume of the house in 50 to 60 min, an amount usual with the ordinary house having a similar construction.

The total space cooled consisted of three rooms on the first and three on the second floor, with the two interconnecting halls, making a total of 14,170 cu ft. The warm air circulating system for heating was used with slight changes for the summer cooling operations, and these

included two sections of finned cooling coils of 473 cu ft total area, a fan of 1475 cfm capacity, and a dry-bulb thermostatic control from a room on the first floor to a modulating by-pass damper. The actual cooling was obtained, for test purposes, by means of ice water at about 35° F which could be circulated at the rate of 396 gph. The air temperature entering the rooms was lowered to a minimum of 60° at the register faces.

During the tests|| the indoor dry-bulb temperature was held at 75° to 81° F, depending on the relative humidity, and the effective temperature was held at about 72°, while the relative humidity in the residence was controlled at 45 to 50 per cent.

A complete summary of tests performed during the summer of 1932 is shown in Table 14·6. The basement correction consisted mostly of the electrical load imposed on the fan and pump motors and the basement lights. The tests indicated that warm air ducts and registers can be adapted successfully to comfort cooling, and with air temperatures of 60° and higher the air in the house is well diffused. The research men did not appear to be affected unfavorably by living in the house. The use of awnings resulted in a reduction of 20 to 30 per cent of the cooling load. The maximum calculated load occurred at about 4 P.M., but the actual maximum load in the house was not found until 8 P.M., an indication of the time lag due to the materials in the building construction. The result of such a time lag makes calculations on the temperature differential basis of doubtful value. The actual load during any two seasons of a 10-year period may vary as much as 1:7½. It is not very clear just what values for the temperature difference should be used, especially for the southeast, south, and southwest sides of the house, and undoubtedly larger values than 10° F are justified at times. The second-floor ceiling, or the ceiling below the attic, acts as a large panel radiator as the attic can easily acquire temperatures of 100° to 115° F or more during that part of the day from 12 M. to 8 P.M., and a differential of 25° may be justified for the latitude of Urbana, Illinois.

It is quite evident that comfort cooling of a whole house involving the use of about 42 tons of ice costing about \$4.00 per ton during the cooling season of approximately 62 days, which would be increased by the actual use of the kitchen even when it is separately vented, would be impracticable for the vast majority of people, and the alternative would be to use comfort cooling applied to the bedrooms or the bedrooms and the living room. However, other methods can be used.

14·25. The Use of Deep Well Water. Where deep well water is available at reasonable cost at temperatures of 60° or less the cheapest

|| Willard and Kratz, *Refrig. Eng.*, Vol. 26, No. 2, 1933.

PSYCHROMETRIC LOADS

TABLE 14-6
COMPLETE SUMMARY OF TESTS

Start of Test	Length of Test, hr	Outdoor Temperature			Indoor Temperature Average	Average Temperature Difference In-Out	Average Humidity		Ice Meltage		Dehumidification Total, lb	Hourly Heat Loads				
		Maximum During Test	Minimum During Day	Average During Test			Outdoors During Test	Indoors	Equivalent Total, lb	Corrected for Basement Loss, lb		Uncorrected Total	Sensible	Corrected for Basement Total	Sensible	Corrected for Basement Total
10:45A	8.0	88.0	73.5	83.8	77.7	6.1	63.4	1060	701	32.19	19,225	15,000	12,625	8,400	2070	1377
1:15P	10.0	87.0	63.5	81.3	75.5	5.7	58.1	1459	1001	38.24	21,000	16,980	14,400	10,380	2526	1821
1:55P	4.0	90.5	70.5	86.3	77.2	9.1	65.6	472	289	20.51	16,975	11,590	10,375	4,990	1140	648
2:00P	6.0	85.0	66.5	81.5	78.2	3.3	47.4	907	632	27.82	22,885	18,010	16,285	11,410	4937	3456
2:30P	7.5	93.5	59.5	86.4	77.0	9.4	51.9	1318	975	41.40	25,500	19,500	18,700	12,900	1989	1372
9:30A	13.5	99.0	70.0	91.0	78.4	12.6	50.8	2642	2024	97.96	28,180	20,560	21,580	13,960	1712	1108
11:40A	14.25	99.0	75.0	89.4	81.1	8.3	62.8	3050	2397	113.23	30,820	22,470	24,220	15,870	2917	1912
10:00A	24.0	103.0	73.0	88.8	80.8	8.0	59.9	4243	3144	111.56	25,450	20,400	18,850	13,800	2356	1725
10:00A	24.0	97.0	77.5	83.2	80.0	3.2	59.1	2713	1614	76.48	16,280	12,930	9,680	6,330	3026	1978
10:00A	12.0	90.5	72.5	85.8	80.7	5.1	37.3	1261	711	30.31	15,130	12,480	8,530	5,880	1673	1153
9:00P	24.0	98.0	76.0	86.7	80.0	6.7	60.5	3530	2431	109.04	21,170	16,410	14,570	9,810	2174	1464
9:00P	24.0	97.5	74.0	83.9	79.2	4.7	55.4	3320	2221	88.94	19,920	16,030	13,320	9,430	2853	2005
9:30A	12.0	91.0	67.0	83.1	76.4	6.7	73.4	1761	1211	61.11	21,120	15,775	14,920	9,175	2166	1369
9:30A	10.5	87.0	63.0	81.8	75.6	6.2	71.5	1238	767	35.49	16,975	13,425	10,375	6,825	1674	1101
9:30A	13.5	90.0	72.0	84.0	77.1	6.9	76.3	2025	1407	60.71	21,600	16,880	15,000	10,280	2173	1489
10:00A	11.0	87.5	57.5	80.9	74.9	6.0	44.4	1838	1334	40.64	24,060	20,180	17,460	13,580	2968	2282
10:30A	11.0	90.0	60.0	83.5	75.1	8.4	52.3	1464	960	35.01	19,170	15,830	12,570	9,230	2406	1698
9:00A	13.0	89.5	60.0	83.6	77.3	6.3	63.5	1578	983	45.01	17,470	13,885	10,870	7,235	1725	1148
9:00A	12.0	87.0	67.5	80.8	76.6	4.2	69.0	1513	963	41.44	18,155	14,525	11,555	7,925	2761	1886
7:30A	24.0	94.0	75.0	82.7	78.0	4.7	74.2	3680	2581	104.77	22,090	17,500	15,490	10,900	3295	2318
7:00A	24.0	95.0	73.0	81.9	77.9	4.0	72.2	3752	2653	109.39	22,500	17,720	15,900	11,120	3975	2780
7:30A	24.0	93.0	71.5	79.0	77.1	1.9	74.1	3237	2138	87.15	19,420	15,600	12,820	9,000	6750	4737

Notes: Basement loss approximately 6600 Btu per hr.

method of residence cooling is to use water. The rate for city water at 55° in Urbana, Illinois, is

For the first	500 gal per 24 hr	\$0.37 per 1000 gal
second	500 gal per 24 hr	0.30 per 1000 gal
third	500 gal per 24 hr	0.22 per 1000 gal
fourth	500 gal per 24 hr	0.12 per 1000 gal
fifth	500 gal per 24 hr	0.09 per 1000 gal

On the basis of a cooling load of 21,400 Btu/hr, requiring 6900 gal of water, the research residence could have been cooled at a cost of \$0.95 per day, and this would have been the most severe day during the test period of the summer of 1932.

However, the general use of deep water for comfort cooling is hardly practicable because of the heavy drain on the city water system. The mechanical system involved in deep water cooling would be the simplest of any of the comfort cooling systems and the easiest to control. The apparatus would be confined to the sprays or to finned cooling coils, the circulating fan, and the thermostatic controls. However, there is little control of the humidity with water greater than 50° F, which limits the use of deep well water to latitudes north of Boston, Buffalo, Detroit, and Milwaukee, except in the Rocky Mountains, unless dehumidification can be obtained by a separate method.

14·26. The Use of the Electric Refrigerating Machine. Comfort cooling ordinarily involves a short period of operation, which means that the cheapest first cost is usually desirable. As the length of time of operation increases, the first cost can be increased with an idea of decreasing the operating cost. If some sort of rotary compressor were available (of low first cost, of nominal replacement cost, noiseless when starting, stopping, and during operation, and employing an air-cooled condenser), it would be a boon to comfort cooling.

During the summer of 1933 a series of tests was made in the research residence at the University of Illinois on the effect of night air, opening the attic and the basement windows, and at first using a fan to pump air into the building and later a natural chimney draft. The house was kept closed except during the cooling period from about 9 P.M. to 6 A.M. It was found that natural ventilation was about as good as any other method, and that the actual refrigeration was reduced to about 0.25 of the cooling load expected. It now seems that residence comfort cooling is practicable, for cooling loads comparable with Urbana, Illinois, at a reasonable cost, if the house construction permits proper chimney effects and ice can be secured at a moderate price. Insulation of the walls will be advantageous in both winter and summer, and a generous use of awnings will assist materially.

PROBLEMS

1. If the wet-bulb temperature is 70° and the dry-bulb temperature is 80° F, find the absolute humidity of the air by calculation and check this answer by the use of the psychrometric chart.

2. If the wet-bulb temperature is 70° and the dry-bulb temperature is 80° F, find the actual slope of the adiabatic saturation line for a scale of the ordinate of 1 in. = 20 grains and of the abscissa of 1 in. = 5° F.

3. Check one row of values given in Table 14.2 for, say, 60° F.

4. Calculate the enthalpy, volume, and specific heat of 1 lb of dry air saturated with water vapor at 60° F and at a total pressure of 34 in. of mercury.

5. If 4000 cu ft of air at 50° F and 98 per cent relative humidity mix with 8000 cu ft of air at 80° F and 70 per cent relative humidity, find the temperature and relative humidity of the mixture.

6. Air at 80° F and 50 per cent relative humidity is passed through bunker coils having surface temperatures of 45° . The temperature of the exit air is reduced to 70° F. What are the final absolute humidity and the relative humidity of the exit air?

7. If air is cooled to a dew-point temperature of 50° F and a pressure of 35 psia what will be the dew-point temperature at 14.7 psia?

8. If the barometer stands at 24 in. of mercury (11.78 psia), if the temperature of the air mixture is 70° F, and if the air is saturated with water vapor, find (a) the volume of 1 lb of air, (b) the weight of water vapor, (c) c_p of the mixture, and (d) h calculated from 0° F for the mixture. Find similar answers for 80° F.

9. A portion of a factory floor is to be maintained at 60° F and 65 per cent humidity. It is 240 ft long, 120 ft wide, and 12 ft high, and a partition separates the floor from another part of the factory. One-third of the outside wall area is glass. The temperature outside of the conditioned room is to be taken as 95° F and 70 per cent humidity. Values of U may be taken as 0.44 for the glass, 0.062 for the wall, 0.08 for the partition, and 0.07 for the floor and the ceiling. The infiltration is a complete volume in 3 hr, calculated on the basis of the inside conditions. The number of operators is 400, and fifty 250-watt lamps are in use. Ten horsepower of small motors are connected. The solar radiation may be taken as 0.5 and 0.1 Btu/(sq ft)(min) for the glass and the wall area respectively, calculated for one long wall only. The space above and below the room is to be taken at 95° F. Take the average heat given up by each workman at 60° F at 450 Btu of sensible heat and 200 Btu of latent heat per hour. Note that this is an *air conditioning* problem and that the air may be returned to the work room at any reasonable temperature below 60° F. Find (a) the fan capacity in cubic feet per minute after allowing for a fan horsepower necessary to circulate 27,000 cfm (estimate) at $\frac{3}{4}$ in. of water static pressure (refer to equation 15.10); (b) the refrigeration capacity required, if 10 per cent of the calculated fan capacity is fresh air, plus 5 per cent for safety.

10. Ten thousand cubic feet of air per minute at 90° dry-bulb and 80° wet-bulb temperature are to be cooled to 60° F. Find the size of fan required to circulate this air, after cooling, the refrigeration required, the probable horsepower of the compressor, and the amount of water necessary in the spray chamber. The water-temperature rise may be taken as 5° F. Let $e_s = 0.9$ for F-12.

11. A theater is designed to seat 2000 people. Outside conditions are 95° F dry-bulb and 75° F wet-bulb temperature. Inside conditions are 80° F dry-bulb and 65° F wet-bulb. Assume water vapor exhaled per person to be 1280 grains per hour,

the sensible heat 230 Btu per person per hour, sensible heat inflow from all sources (heat leakage, lights, electrical power, etc.) 320,000 Btu/hr. Rise of temperature, above inlet to fan, is 12°. The cooling system will use the by-pass method. Plot the calculated tonnage, with a varying amount of fresh air all of which passes through the washer, using at least three calculated points.

12. A restaurant is to be air conditioned. The sensible heat entering from all sources is estimated to be 200,000 Btu/hr and the latent heat is 373,000 grains of water vapor per hour. Outside air temperature assumed to be 95° F dry-bulb and 75° F wet-bulb temperature. Inside the restaurant the air is to be held at 80° F dry-bulb and 65° F wet-bulb temperature. The conditioned air enters the dining room at 70° F. Fresh air will be 10 per cent of the fan capacity. Find (a) the fan capacity and the refrigeration required when (1) all the air passes through the spray chamber and (2) the by-pass method is used; (b) the steam used in (1) at 5 psig to heat the air up to 70° F; (c) the area of a counter-flow exchanger for use in (1) if the minimum heating is to be used, taking a value of U of 3.0. Note that this area will be large even when the exhausted air is passed through the exchanger.

13. A theater is to be comfort cooled for 3500 seats. Outside air is 95° F and 60 per cent humidity. Inside air is to be 85° F and 45 per cent humidity. The sensible heat per person is 225 Btu/hr, and the latent heat is 1100 grains of water vapor per person per hour. All other sources of heat amount to 400,000 Btu/hr. Twenty-five per cent of the fan capacity is to be fresh air. The air enters the conditioned room at 75° F. Find (a) the tonnage when all the air is conditioned; (b) the tonnage when the by-pass method is used; (c) the tonnage when the maximum amount and the minimum amount of fresh air is used in (b); (d) the piston displacement of a Freon-12 compressor when liquefaction of the refrigerant takes place at 90° F and evaporation at 50° F.

14. Twenty-five thousand cubic feet of air per minute at 14.7 psia 90° F and 70 per cent humidity, are compressed to 35 psia and cooled by the means of water cooling coils to 80° F and then with brine-refrigerated piping, with the brine initially at 20° and leaving at 25° F passing in counter flow with the air. Find the refrigeration in Btu per minute and the area of the heat-transfer surface, selecting a suitable value of U as suggested in Table 9-17. Note that reversal of direction of flow eliminates consideration of ice formation. Air to be cooled to 30° F.

15. Seven hundred eighty cubic feet of air per minute at 14.7 psia, 90° F dry-bulb, 65° F wet-bulb, are compressed to 30 psig. After cooling to 75° F with water, the compressed air is refrigerated by means of brine to 25° F. If the initial brine temperature is 15° F and the final temperature is 20° F, find (a) the total heat removed during the cooling by the use of brine; (b) the area of the heat-transfer surface needed in the counter-flow dehydrator if the value of U is 4.0 (see Table 9-17).

CHAPTER XV

FLUID FLOW; DUCT DESIGN; NOISE CONTROL

15·1. The Critical Velocity. When the motion of each particle of fluid is parallel to the walls of the conduit, and no cross currents occur, the flow is said to be stream line or *viscous*, and the average velocity of flow over the entire cross section is equal to 0.5 of the maximum velocity. All fluids below certain critical velocities flow in straight-line motion as, for example, air at 70° F and 14.7 psia in a 2-in. pipe below 1.947 fps, water at 70° below 0.122 fps, and California crude oil at 70° below 271 fps. The loss of pressure Δp due to flow at velocities below the critical is given by the Poiseuille formula:

$$\Delta p = \frac{32\mu Lw}{gd^2} \text{ psf} \quad (15\cdot 1)$$

where Δp = loss of pressure in pounds per square foot, L = length of straight pipe in feet plus the equivalent length due to fittings, w = the average velocity of the fluid in feet per second, μ = the absolute viscosity of the fluid in pounds per foot-second, d = the diameter of the conduit in feet.

If the fluid motion is increased above the critical velocity there is a sudden change in the nature of the flow.* There are now innumerable eddies in the stream, and the fluid is said to be *turbulent*. The average velocity over the entire cross section is now approximately 0.8 of the maximum. The drop of pressure is given by the Darcy formula

$$\Delta p = \frac{\rho f L w^2}{2gm} \text{ psf} \quad (15\cdot 2)$$

where f = the friction factor, ρ = the density of the fluid in pounds per cubic foot, m = the hydraulic radius = $d/4$ for circular sections. The critical velocity can be found by equating formulas 15·1 and 15·2, so that

$$w_{crit} = \frac{16\mu}{f_c \rho d} \quad (15\cdot 3)$$

As the friction factor at the critical velocity f is practically the same

* Osborne Reynolds, *Trans. Royal Soc. (London)*, Vol. 174, p. 935, 1883.

for cast iron and for steel pipe for all fluids and is equal to 0.008, equation 15·3 becomes

$$w_{\text{crit}} = \frac{2000\mu}{\rho d} = \frac{2000\nu}{d} \tag{15·4}$$

where $\nu = \mu/\rho =$ the kinematic viscosity. The same critical velocity is to be found always at the boundary film where the creeping fluid changes into turbulent flow.

15·2. Dimensional Analysis. In studying the principles of fluid flow it is advantageous at times to use the theory of dimensional analysis. According to this theory *all* the terms of any equation, having a physical significance, must have *identical* dimensions. For example, in the equation for motion $s = \frac{1}{2}at^2$; taking the dimensions of a as feet per second squared, L/T^2 , the equation becomes $s = LT^2/T^2 = L$, from which one concludes that the dimensions of s are in feet. Other fundamental dimensions are

$$\begin{aligned} \text{Mass (weight) in pounds} &= M\uparrow \\ \text{Force} &= Ma = MLT^{-2} \\ \text{Velocity} &= LT^{-1} \\ \text{Density} &= ML^{-3} \end{aligned}$$

If a *pendulum* vibrates in a vacuum the only factors that can be conceived to influence the time of vibration are the length and weight of the pendulum and the acceleration due to gravity. Therefore $t = cL^xM^y g^z$, or written with dimensions, the equation becomes

$$T^1 = L^x M^y L^z T^{-2z}$$

As all the terms must have identical dimensions the equation may be rewritten

$$L^0 M^0 T^1 = L^{x+z} M^y T^{-2z}$$

and, equating identical exponents,

$$\begin{aligned} 0 &= x + z \\ 0 &= y \\ 1 &= -2z \end{aligned}$$

and therefore

$$\begin{aligned} z &= -\frac{1}{2} \\ x &= \frac{1}{2} \end{aligned}$$

in consequence the equation becomes

$$t = cL^{1/2}M^0g^{-1/2} \quad \text{and} \quad t = c\left(\frac{L}{g}\right)^{1/2}$$

† There is no confusion in regard to the dimensions of mass if *consistent units* have been selected. See *Instruments*, May, 1934, p. 89.

so that it would appear that the time of vibration is independent of the weight of the pendulum, and that the length and the value of g vary in identical manner. The value of c is a constant, found by experiment to be equal to 2π .

As another example of dimensional analysis, suppose that a particle has an initial velocity of w feet per second and that a constant force is acting on it. Then the total distance s passed through in the time t will be given by the expression

$$s = cf(w, a, t) \dagger$$

or

$$s = cw^x a^y t^z$$

and, substituting the dimensions,

$$L^1 M^0 T^0 = c L^x T^{-x} L^y T^{-2y} T^z$$

and, equating identical exponents,

$$1 = x + y$$

$$0 = -x - 2y + z$$

therefore

$$z = 1 + y$$

Substituting

$$s = cw^{1-y} a^y t^{1+y}$$

$$= cwt \left(\frac{at}{w} \right)^y$$

or from experiment the value of the constant c and the exponent y gives

$$s = wt \left(1 + \frac{1}{2} \frac{at}{w} \right)$$

which is the familiar formula. Thus it is seen that it is possible to get the general expression of the required formula if the correct assumptions are made about the proper factors influencing the problem. However, the great objection to this procedure is that the general expression derived by dimensional analysis does not throw any light on what are the important and the minor factors in the formula, and there is sometimes a doubt that all the terms are included in the expression.

15.3. Viscosity. Viscosity is the internal resistance offered to the motion of a fluid. It is the tangential force on a unit area of either of two horizontal planes at a unit distance apart required to move one plane at unit velocity with reference to the other plane, the space be-

† Glazebrook, *Dictionary of Applied Physics*, Vol. 1.

tween the two planes being filled with the viscous fluid. If one horizontal plane has a velocity w relative to the other and the two planes are a distance s apart, the rate of shear is w/s . The coefficient of viscosity is the tangential force per unit area divided by the rate of shear. As it is possible always to replace a force by the effect of a force, as $f = Ma$, the dimensions of the coefficient of viscosity μ are

$$\frac{MLT^{-2}L^{-2}}{LT^{-1}L^{-1}} = ML^{-1}T^{-1}$$

The units of the coefficient of viscosity are grams per centimeter-second in the cgs system and pounds per foot-second in the English system. The multiplying constant to convert cgs to the English units of viscosity is

$$\frac{30.5}{454} = 0.0672$$

The unit of absolute viscosity is the *poise* in grams per centimeter-second. The centipoise, 0.01 poise, is the absolute viscosity of water at 68° F. A centipoise is equal to $0.01 \times 0.0672 = 0.000672$ lb/ft-sec unit in the English system. Within reasonable limits the viscosity of both gases and liquids is independent of the pressure but increases with the temperature in gases and drops as the temperature rises in liquids.

15.4. The Derived Flow Formula. By experiment it has been shown that the pressure drop Δp per unit area between two points in a conduit L feet apart due to the flow of a fluid varies as the diameter d , the length L , the mean velocity w , the viscosity μ , and the fluid density ρ . This may be put in the form of an equation as

$$\Delta p \propto f(d, L, w, \mu, \rho)$$

or

$$\Delta p = kd^x L^m w^n \mu^y \rho^z$$

where k is some unknown constant. Experiment also shows that the value of the exponent m is 1.0, and, substituting the dimensions in the formula, it becomes

$$M^1 L^1 T^{-2} L^{-2} = L^x L^1 L^n T^{-n} M^y L^{-y} T^{-y} M^z L^{-3z}$$

so that

$$z = n - 1 \quad y = 2 - n \quad x = n - 3$$

These values being substituted for x , y , and z , the equation becomes

$$\Delta p = kd^{n-3} L w^n \mu^{2-n} \rho^{n-1} = k \frac{\rho L w^2}{d} \left(\frac{dw\rho}{\mu} \right)^{n-2}$$

but, as $p = h_f \rho$, where h_f is the loss of head in terms of a column of the fluid flowing and $\mu/\rho = \nu$, the kinematic viscosity, then

$$\Delta h_f = \frac{k_0 L w^2}{2dg} \left(\frac{dw}{\nu} \right)^{n-2} \quad (15.5)$$

The dimensions of dw/ν or $R\delta$ are

$$\frac{L \times L \times M \times LT}{T \times L^3 \times M} = 1.0$$

and the quantity is therefore dimensionless. It has been given the name *Reynolds' number* because of the early mention of the expression by Osborne Reynolds. As with all other dimensionless quantities the numerical value is the same in any set of consistent units. Some writers use the expression dws/z , where the quantity s is the specific gravity, the quantity z is the viscosity in centipoises, and d is the diameter in inches. These units are not consistent, but as the density of water is understood in the specific gravity the expression may be converted into consistent values by multiplying by the constant 7742. ¶

Equation 15.5 may be written

$$\frac{2}{k_0} h_f \frac{dg}{Lw^2} = \phi \left(\frac{dw}{\nu} \right)$$

which will give dimensionless quantities on both sides of the equation, and this provides a very convenient means of plotting, as shown in Fig. 15.1. In the Darcy formula (15.2) the expression f which is called the *friction factor* is equal to the value of the product $h_f(2dg/k_0Lw^2)$, provided the constant k_0 is given the value of 4.0.

Another dimensionless quantity used as ordinate is the value of $S/\rho w^2$ where S is the *surface friction*. This surface friction is an imaginary resistance per unit area on the inside surface of the conduit. Mathematically it can be expressed as follows: The total pressure tending to cause flow along the pipe of length L is $(\pi d^2/4) \Delta p$, and the resistance to flow is $S\pi dL$, so that $S = \Delta p d/4L$. The dimensions of S are

$$\frac{MLT^{-2}L}{L^2L} = ML^{-1}T^{-2}$$

§ Re is also used to indicate Reynolds' number.

¶ Osborne Reynolds, "Water—Direct and Sinuous," *Phil. Trans. Royal Soc. (London)*, Vol. 174, pp. 935–982.

$$\text{¶ } 7742 = \frac{62.4}{12 \times 0.000672}$$

Reference to Fig. 15·1 will show that the left-hand side is in the region of viscous or straight-line flow and that the critical velocity is at the point where $R =$ about 2000. In Reynolds' experiments the difference between viscous and turbulent flow was very well shown by injecting a colored liquid which continued to flow as a colored streak until the critical velocity was reached and a complete mixing of the color occurred.

Equation 15·5 may be written

$$h_f = k_0 \frac{L}{d} \frac{w^2}{2g} \left(\frac{\mu}{dw\rho} \right)^{2-n} = k_0 \frac{Lw^2}{2dg} (\nu)^{2-n} \frac{1}{d^{2-n}} \frac{1}{w^{2-n}}$$

or

$$h_f = k_0 \frac{Lw^n}{2g} (\nu)^{2-n} \frac{1}{d^{2-n}}$$

For identical diameters and length of pipe this becomes

$$\frac{h_1}{h_2} = \left(\frac{v_1}{v_2} \right)^{2-n} \left(\frac{w_1}{w_2} \right)^n$$

and for identical velocities

$$\frac{h_1}{h_2} = \left(\frac{v_1}{v_2} \right)^{2-n}$$

where h_1 is the unknown loss of head of the fluid being investigated, in feet of the fluid flowing; and h_2 is the loss of head, in feet of water flowing in the same pipe and at the same velocity. If this formula is correct and the value of n is known it will be possible to calculate a multiplier that will give the loss of head in terms of that of water. Gibson* performed experiments to verify this fact and found values of n of about 1.79 for certain solutions of calcium and sodium chloride in water. If in these tests, as with 2-in. galvanized iron pipe, the value of n was found to be 2.0, the formula would indicate that the ratio of the loss of head h_1/h_2 becomes 1.0, which shows that the loss of head is in proportion to the ratio of the two densities.

An extension of this analysis† indicates (Fig. 15·2) that the pressure loss of one well-known fluid like water can be used to calculate the pressure loss of any fluid like steam, air, or mercury. The kinematic viscosity ν will be required, and an assumption of the value of n , which may vary slightly.

* Gibson, *Inst. Mech. Eng.*, February, 1914.

† Macintire and Edwards, "Pressure Losses of One Fluid as a Criterion of the Pressure Losses of Any Fluid," *Refrig. Eng.*, October, 1933.

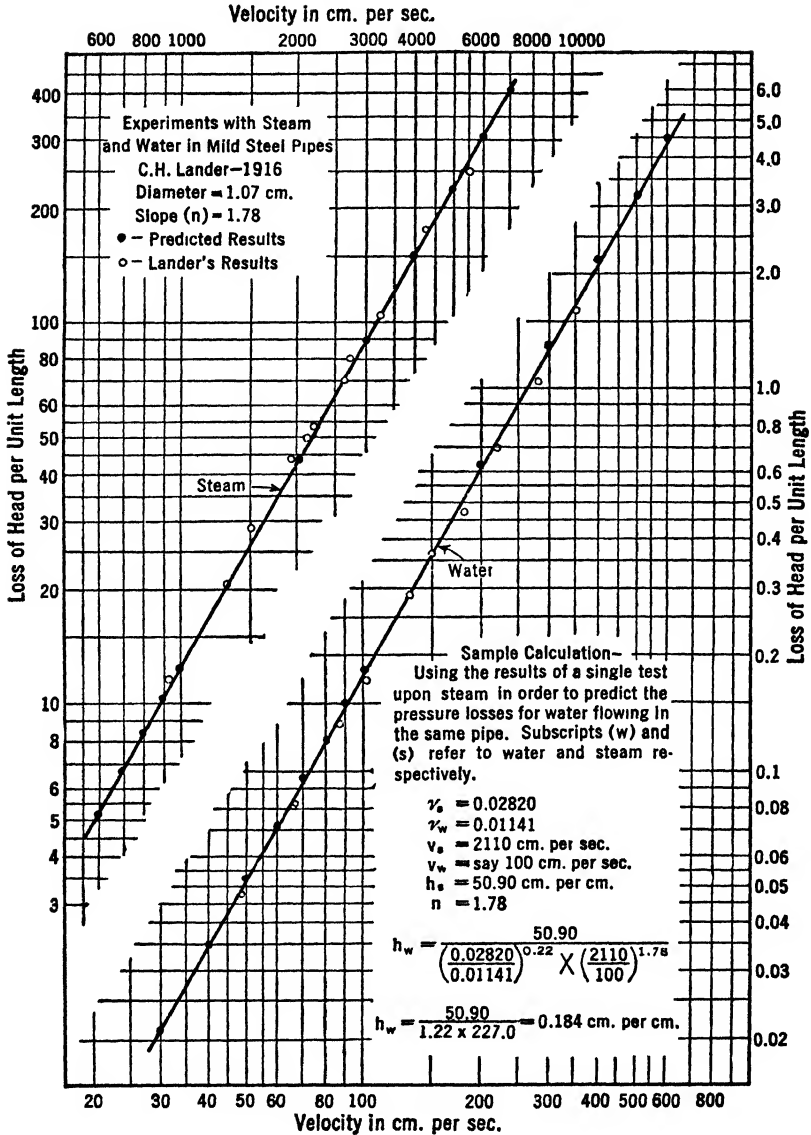


FIG. 15-2. Comparison of observed and calculated loss of head.

Example. A calcium chloride brine of sp gr 1.134 and a temperature of 20.7° F is to flow through a 1¼-in. pipe, 1.38 in. internal diameter, at an average velocity of 5.055 fps. Find the loss of head per 100 ft. of length.

Solution. The value of the reciprocal of the kinematic viscosity $1/\nu$ is found, from Fig. 15-3, to be 31,000 for commercial calcium chloride brine, and the value

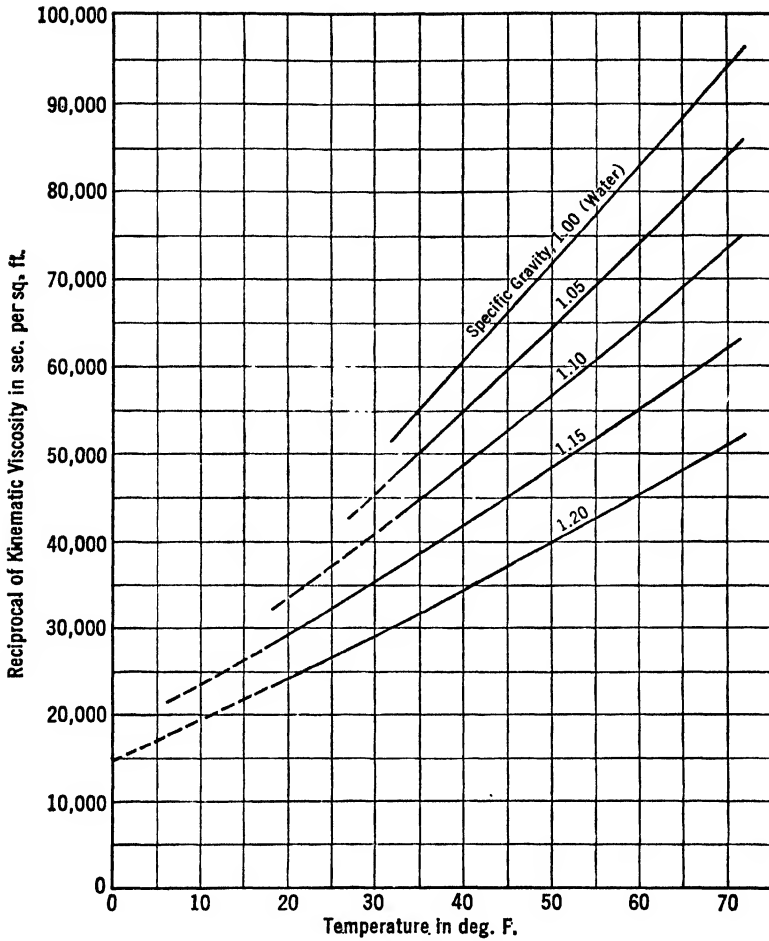


FIG. 15.3. Relation between the reciprocal of the kinematic viscosity of calcium chloride and the temperature.

of Reynolds' number R is

$$R = \frac{dw}{\nu} = \frac{1.38}{12} 5.055 \times 31,000 = 18,020$$

The value of the friction factor† (Fig. 15.1) is 0.00759 for this value of R ,

† The friction factor, from Fig. 15.1, calculated for drawn brass pipe, is less than that found during tests for wrought iron or steel pipe where the wetted surface is rougher.

and the loss of head, in feet of the fluid flowing, is

$$h = f \frac{4Lw^2}{2dg} = 0.00759 \frac{4 \times 100 \times \overline{5.055}^2}{\frac{1.38}{12} \times 2 \times 32.16} = 10.48 \text{ ft}$$

$$= \frac{10.48}{2.035} = 5.15 \text{ psi}$$

Example. The loss of head may be found in another manner. If the velocity of the brine is 2.022 fps, if its temperature is 40°, if the sp gr is 1.133, and if there are 108 ft of 1¼-in. pipe, the loss of head is

$$\frac{h_b}{h_w} = \left(\frac{v_b}{v_w} \right)^{2.0-1.78}$$

where h_w = loss of head of water in feet.

$$h_b = h_w \left(\frac{v_b}{v_w} \right)^{0.22} = 1.91 \left(\frac{0.0000227}{0.0000167} \right)^{0.22}$$

$$= 2.04 \text{ ft of brine}$$

According to test § the loss of head was found to be 2.06 ft of brine.

It must be kept in mind, however, that when the *similarity of motion* is under discussion, such motion exists at constant values of Reynolds' number R only when the *surfaces of the conduit are geometrically similar, and this similarity must even extend to the irregularities of the wetted surface.* To get true similarity it would be necessary to have the velocity distribution similar. However, it is sometimes possible to get *approximately comparable* conditions, as for example where the fluid passes through an annular section, as in double pipe apparatus.

In the annular section there is no particular diameter that can be selected for the calculation of Reynolds' number R , but the diameter of a single pipe can be taken such that the *hydraulic radius* will be the same for both. As the hydraulic radius is defined as the net cross-sectional area divided by the surface wet by the fluid this becomes, for the annular section,

$$m = \frac{\frac{\pi}{4} (d_1^2 - d_2^2)}{\pi (d_1 + d_2)} = \frac{d_1 - d_2}{4}$$

15.5. Friction Losses in Fittings. Various methods are in use for estimating the loss of head due to fluid flow through fittings. This loss is sometimes stated in terms of a fraction of velocity head or is expressed

directly as a loss of static head. A convenient method is to express fitting loss in terms of equivalent length of straight pipe. Table 15·1 gives the equivalent length of straight pipe to be added to a circuit in order to establish resistance equal to that of a standard elbow; the table gives equivalent length as a function of the diameter of the pipe in which the elbow is installed.

TABLE 15·1
EQUIVALENT LENGTH OF ELBOWS

Line Size, O.D., in.	Equivalent Length of Pipe Added for Elbows, ft	
	Per elbow	For 6 elbows
$\frac{3}{8}$	0.788	4.725
$\frac{5}{8}$	1.363	8.175
$\frac{7}{8}$	1.963	11.775
$1\frac{1}{8}$	2.563	15.375
$1\frac{3}{8}$	3.163	18.975
$1\frac{5}{8}$	3.763	22.575
$2\frac{1}{8}$	4.963	29.775
$2\frac{5}{8}$	6.163	36.975
$3\frac{1}{8}$	7.363	44.175
$3\frac{5}{8}$	8.563	51.375
$4\frac{1}{8}$	9.763	58.575
$5\frac{1}{8}$	12.188	73.125

15·6. The Size of Pipe. As a rule the pressure drop due to the flow of the gaseous refrigerant limits the *average velocity* to less than 100 fps, although some higher velocities are possible under special conditions, as when the suction and discharge lines are short. Carbon dioxide vapor, being very heavy, acts more like a liquid than a vapor, so that high velocities pound and shake the pipes severely. Average velocities for carbon dioxide should be limited to 1000 to 1200 fpm. Sulphur dioxide suction tube connections are designed for 1000 to 2000 fpm and the discharge tubes for slightly higher velocities, so that the discharge line is commonly one size smaller than the suction line. Methyl chloride operates satisfactorily with the same average velocities as sulphur dioxide and has a maximum velocity of 2000 fpm, particularly in the small tube sizes. As the suction pressure drop in ammonia installation frequently has to be kept at a minimum the average vapor velocity is limited to less than 50 fps whenever possible. Piping should be kept as straight as possible and free from obstructions. The liquid line should be chosen with an idea of limiting the velocity to 3 to 5 fps. As it is

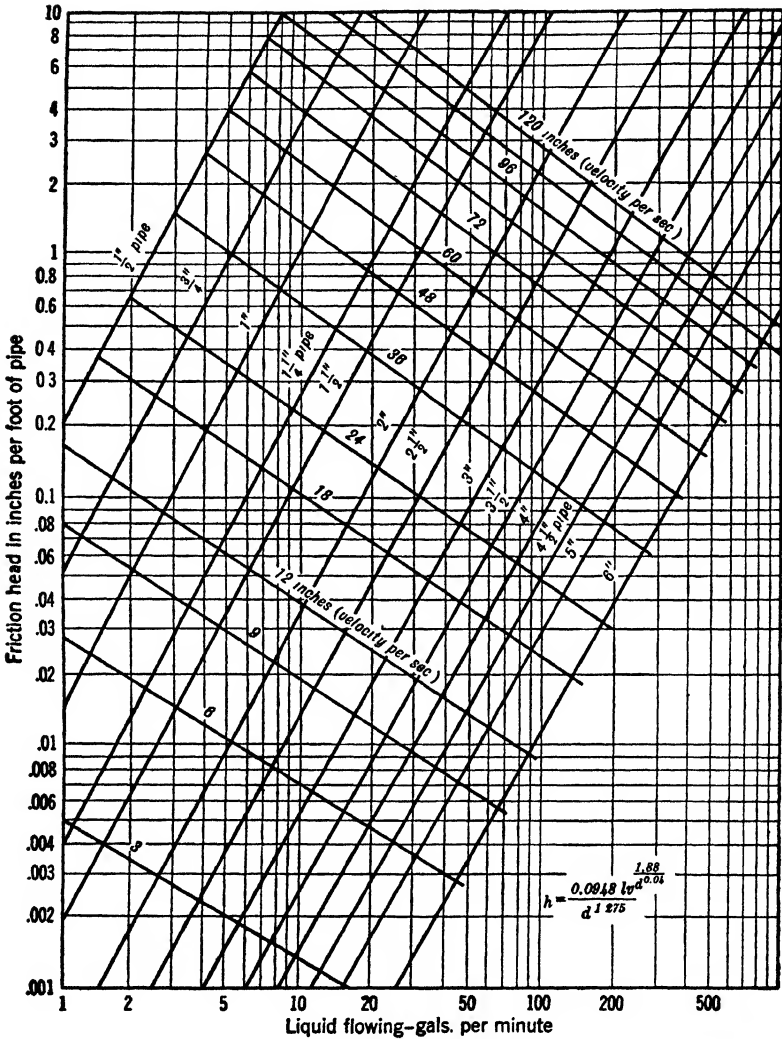


FIG. 15·4. Resistance to flow of water in pipes.

important to keep the liquid temperature from rising above that of liquefaction, the pipe line should be insulated where it has to pass through regions materially hotter than the condensing water.

As an example of the method of determining the pressure drop due to flow the following example may be of value.

Example. The suction line of an ammonia compressor is 2 in. in diameter, the length is 300 ft, and it is designed for a gas velocity of 5000 fpm when

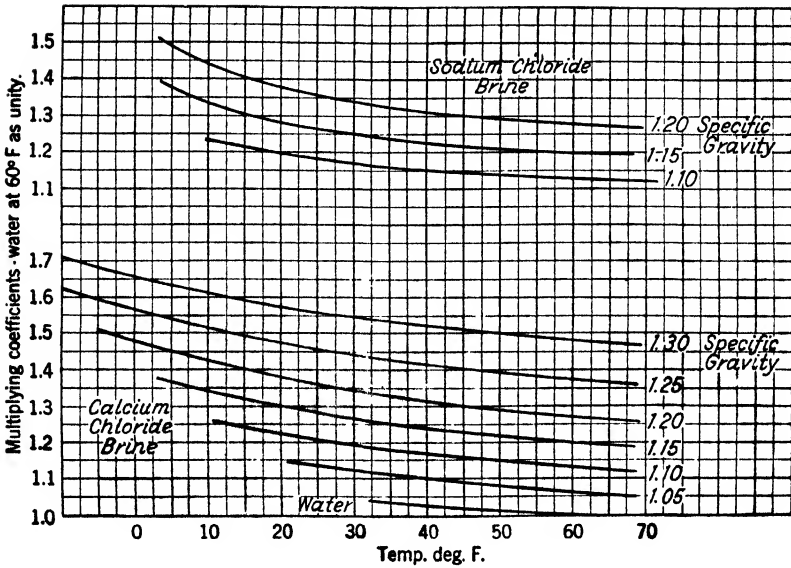


FIG. 15-5. Gibson's multiplying factors for brine.

operating at 20 psig. Find the pressure loss. Reynolds' number R is obtained from the formula, $R = d v \rho / \mu$.

Solution.

$$R = \frac{2.067 \times 5000}{12 \times 60 \times 8.06 \times 60 \times 10^{-7}} = 296,800$$

and the friction factor for this value of R in Fig. 15-1 is 0.0051. From Darcy's formula

$$h_f = f \frac{4Lw^2}{2dg} \text{ ft of the fluid flowing}$$

and this becomes

$$h_f = 0.0051 \frac{4 \times 300 (83.3)^2}{\frac{2.067}{12} \times 64.32} = 3830 \text{ ft of ammonia}$$

and

$$h = \frac{3830}{144 \times 8.06} = 3.31 \text{ psi}$$

This calculation makes no allowance for loss of head due to fittings. The loss of head due to the flow of brine may be found in a similar manner, or by multiplying the loss of head of water flowing in pipe of the same size at the same velocity by the ratio of the kinematic viscosities raised to the $2n$ power, where n is approximately 1.77 as shown previously. The loss of head due to the flow of water is shown in Fig. 15-4, and the multiplying factors for brine are given in Fig. 15-5.

15.7. Duct Design. In duct design the air should be conveyed as directly as possible with reasonable velocities, all sharp changes of direction being avoided, and the sections used should be with sides as nearly equal as possible. Elimination of noises in public buildings requires velocities from 900 to 1200 fpm. In public buildings the standard velocities are

(a) Through intakes	1000 fpm
(b) To and from the conditioner	1000 to 1200 fpm
(c) Main discharge duct	900 to 1200 fpm
(d) Through branch ducts	600 to 1000 fpm
(e) Through registers and grills	200 fpm at the breathing line 400 fpm 10 or more feet above the floor
(f) The velocity through the fan outlet	1500 to 2500 fpm

In industrial plants (c) and (d) may be increased from 50 to 100 per cent as the lack of noise is not an important factor. The discharge from the fan can be distributed either by means of branches from one or two main trunk lines (the preferred method in industrial applications) or by the use of a *plenum* chamber from which the separate ducts are taken. The sizes of all main and branch ducts can be calculated by selecting a uniform friction pressure drop and calculating the size of the ducts at each section on that basis or arbitrarily fixing the velocity of the different sections, basing the amounts on design experience. A fan must be selected that will deliver the quantity of air required and at a total pressure sufficient to overcome all losses due to friction, loss of head in changes of direction, and cross section, and will provide the final residual velocity. Usually, but not always, the longest duct has the greatest resistance, the other branches being damped in order to give equal resistances, and the total fan head should be calculated from this value. If d is the diameter of the duct the radius of the elbows should not be less than $1.5d$, in which case the loss due to the change of direction will be nominal; it is usually given as approximately 17 per cent of the velocity pressure for round and 8 per cent for square ducts. Another method is to allow 10 diameters of straight pipe as the equivalent loss in a long-radius elbow. Table 15.2 is also used in terms of the loss of velocity pressure VP without the use of modern guide vanes.

The fans used are the multiblade type, of low peripheral speed and large capacity, which are particularly quiet in operation but are not adapted to parallel operation. However, the compound curved blades, sloping slightly backward at the tip, have some of the characteristics of the forward and some of the backward curved blades, and these may

be used in parallel. The compound curved blade is quiet and has been used for years for ventilation work. For selection of a fan the most economical operating conditions and therefore the deciding ones are indicated in tables. If other conditions are used, the quantity of air in cubic feet varies approximately as the first, the static head as the second and the power required to move the air as the third power of the

TABLE 15-2

<i>r</i> = radius of curvature	
Ells	
Round section, <i>r</i> = <i>d</i> loss of head	0.25VP
<i>r</i> = 2 <i>d</i> loss of head	0.15VP
Square section, <i>r</i> = width loss of head	0.18VP
<i>r</i> = 2 × width loss of head	0.08VP
Right-angled turn	
For round section, loss of head	0.85VP
For square section, loss of head	1.25VP
Tee connection	
Branches in opposite directions, for right-angled turn	1.0 VP
Rounded with radius, equal to 1.5 × diameter of branch	0.15VP
Branch off of duct	
Making 30 with main duct, loss of head	0.17VP
Making 45 with main duct, loss of head	0.22VP
Entrance to pipe, without provision to reduce eddies	0.85VP
Coned entrance to pipe	0.2 VP
Enlargement of pipe area, cone with 10 per cent slope	0.23VP
	in smaller pipe
Discharge to plenum chamber	
For cone connection with 10 per cent slope	0.48VP in pipe
Without cone connection	1.0 VP in pipe

number of revolutions of the fan per minute. A noisy fan indicates frequently that the fan is operating considerably beyond the maximum efficiency.

The connection between the head in feet of the fluid flowing and the velocity of the air in feet per second is shown by the expression

$$v = (2gh)^{1/2}$$

where *h* is the head in feet of the fluid flowing, which in this case is air, but as a rule all tables involving fan capacity and heads are in terms of inches of water. The connection between these two can be found by the expression

$$h_a \rho_a = \frac{h_w'' \rho_w}{12}$$

TABLE 15-3

Velocity, feet per minute	Velocity Head of Air, inches of water
750	0.035
900	0.0505
1050	0.0686
1200	0.090
1350	0.1135
1500	0.140
1650	0.175
1800	0.202
1950	0.237
2100	0.275
2400	0.360

from which

$$h_a = \frac{h_w'' \rho_w}{12\rho_a}$$

but

$$v = (2gh)^{1/2} = 64.32^{1/2} h_a^{1/2}$$

and putting

$$\rho_w = 62.4 \text{ pcf at } 70^\circ\text{F}$$

and

$$\rho_a = 0.075 \text{ pcf at } 70^\circ\text{F}$$

$$\begin{aligned} v &= \left(\frac{62.4 \times 2g}{12 \times 0.075} \right)^{1/2} (h_w'')^{1/2} \\ &= 66.75 (h_w'')^{1/2} \text{ fps} \end{aligned} \quad (15.7)$$

The *velocity head* for air at 70°F, from this formula, gives the values shown in Table 15-3.

The Darcy formula for the loss of head can be rewritten as

$$h_a = f \frac{LPw^2}{A2g} \text{ in feet of the fluid flowing}$$

where L is the length of the conduit in feet; P is the perimeter of the conduit in feet; A is the area of the conduit in feet; w is the average fluid velocity in feet per second; f is the so-called coefficient of friction; Q is the volume of the fluid passing through the conduit, in feet per minute; ρ is the density of the fluid.

The loss of head, in inches of water, is then

$$h = f \frac{12\rho_a LPw^2}{\rho_w A 2g}$$

Using a value of f of 0.0050 for an average value according to Fritsche and Stanton and Pannell for air, for values of Reynolds' number well above the critical velocity, the length of conduit being 100 ft,

$$Q = Aw \quad \rho_a = 0.075 \quad \rho_w = 62.4$$

$$Q = 5666 (h)^{1/2} \left(\frac{A^3}{P} \right)^{1/2}$$

and, if the conduit is round,

$$\begin{aligned} Q &= 2225.5 (hd^5)^{1/2} \quad \text{when the diameter is in feet} \\ &= 4.451 (hd^5)^{1/2} \quad \text{when the diameter is in inches} \end{aligned} \quad (15.8)$$

As rectangular conduits frequently are necessary because of the construction of the building and the necessity of using the least amount of

TABLE 15.4

a/b	a/d	a/b	a/d
1.0	0.9075	6.0	2.387
1.5	1.116	6.5	2.499
2.0	1.320	7.0	2.608
2.5	1.464	7.5	2.714
3.0	1.617	8.0	2.817
3.5	1.760	8.5	2.917
4.0	1.898	9.0	3.015
4.5	2.028	9.5	3.110
5.0	2.153	10.0	3.206
5.5	2.272		

head room, it is convenient to have the formula for capacity in terms of a and b , where a and b are the longer and the shorter sides of the rectangular section, respectively. Making these substitutions and equating equal capacities gives

$$2225.5 (h)^{1/2} d^{2.5} = 5666 (h)^{1/2} \left(\frac{a^3 b^3}{2(a+b)} \right)^{1/2}$$

and this simplifies to

$$\frac{a}{d} = 0.79 \left[\left(\frac{a}{b} \right)^2 + \left(\frac{a}{b} \right)^3 \right]^{0.2} \quad (15.9)$$

Table 15.4 gives values of a/d for even increments of a/b .

Table 15.5a gives the manufacturer's rating for a type of multiblade fan. The ratio of the static to the total pressure varies from 85 to 90 per cent at maximum efficiency. As already mentioned, for other than the tabulated values corresponding to small changes of speed, the capacity of the fan in feet, the static pressure at the fan outlet, and the required

TABLE 15-5c
CAPACITIES OF BUFFALO NIAGARA CONOIDAL FANS (TYPE N) UNDER AVERAGE WORKING CONDITIONS
70° F and 29.92-in. Barometer

Size	Diameter of Blast Wheel, in.	Area of Outlet, ft ²	½-in. Static Pressure = 0.288 oz			¾-in. Static Pressure = 0.433 oz			1-in. Static Pressure = 0.577 oz			1½-in. Static Pressure = 0.865 oz		
			Rpm	Volume of cubic feet per minute	Horse-power	Rpm	Volume of cubic feet per minute	Horse-power	Rpm	Volume of cubic feet per minute	Horse-power	Rpm	Volume of cubic feet per minute	Horse-power
3	15½	1.31	544	1,945	0.28	668	2,380	0.51	770	2,750	0.78	943	3,365	1.45
3½	18½	1.79	465	2,642	0.38	572	3,240	0.69	660	3,740	1.06	809	4,580	1.97
4	20½	2.33	408	3,459	0.50	500	4,230	0.90	577	4,895	1.39	709	5,980	2.58
4½	23½	2.95	362	4,375	0.63	445	5,350	1.14	514	6,195	1.75	630	7,575	3.26
5	26½	3.64	326	5,400	0.77	400	6,610	1.41	462	7,645	2.16	566	9,350	4.03
5½	28½	4.41	296	6,540	0.94	364	8,000	1.71	420	9,250	2.62	515	11,320	4.87
6	31½	5.25	272	7,780	1.11	334	9,525	2.03	386	11,000	3.12	472	13,450	5.80
7	36½	7.14	233	10,590	1.52	286	12,950	2.77	330	14,980	4.24	405	18,330	7.90
8	42	9.33	204	13,820	1.98	250	16,910	3.61	289	19,550	5.54	354	23,950	10.30
9	47	11.81	181	17,500	2.51	222	21,400	4.57	256	24,750	7.01	314	30,300	13.05
10	52	14.58	103	21,600	3.09	200	26,450	5.65	231	30,550	8.65	283	37,400	16.10
11	58	17.64	148	26,150	3.74	182	32,000	6.85	210	37,000	10.48	257	45,250	19.48
12	63	21.00	136	31,100	4.45	167	38,100	8.15	193	44,050	12.48	236	53,900	23.20
13	68	24.65	125	36,500	5.22	154	44,700	9.56	178	51,650	14.62	217	63,200	27.20
14	73	28.68	116	42,350	6.06	143	51,900	11.08	165	60,000	16.96	202	73,200	31.55
15	78	32.80	109	48,550	6.95	133	59,500	12.70	154	68,850	19.49	189	84,100	36.25
16	84	37.32	102	55,300	7.91	125	67,750	14.46	144	78,300	22.15	177	95,750	41.20
17	89	42.14	96	62,500	8.95	118	76,500	16.32	136	88,400	25.00	167	108,000	46.50
18	94	42.24	91	70,000	10.01	111	85,600	18.30	128	99,100	28.05	157	121,200	52.15
19	99	52.63	86	78,000	11.15	105	95,500	20.40	122	110,200	31.25	149	135,000	58.05
20	105	58.32	82	86,450	12.36	100	105,850	22.60	116	122,200	34.65	142	149,500	64.45

TABLE 15-5a (Continued)

Size	Diam-eter of Blast Wheel, in.	Area of Outlet, ft ²	2-in. Static Pressure = 1.154 oz			2½-in. Static Pressure = 1.442 oz			3-in. Static Pressure = 1.734 oz			3½-in. Static Pressure = 2.019 oz		
			Rpm	Volume of cubic feet per minute	Horse-power	Rpm	Volume of cubic feet per minute	Horse-power	Rpm	Volume of cubic feet per minute	Horse-power	Rpm	Volume of cubic feet per minute	Horse-power
3	15½	1.31	1088	3,890	2.21	1215	4,350	3.08	1332	4,770	4.05	1443	5,150	5.13
3½	18½	1.79	934	5,300	3.01	1010	5,930	4.19	1141	6,495	5.53	1238	7,010	6.98
4	20½	2.33	817	6,920	3.93	912	7,730	5.47	1000	8,480	7.22	1082	9,160	9.12
4½	23½	2.95	726	8,750	4.97	810	9,795	6.93	890	10,740	9.14	964	11,590	11.55
5	26½	3.64	655	10,820	6.15	730	12,070	8.55	800	13,250	11.26	868	14,300	14.25
5½	28½	4.41	595	13,100	7.43	664	14,600	10.35	728	16,030	13.62	789	17,300	17.25
6	31½	5.25	545	15,550	8.85	609	17,390	12.30	667	19,090	16.22	723	20,600	20.55
7	36½	7.14	468	21,200	12.02	522	23,650	16.75	572	26,000	22.10	620	28,050	27.95
8	42	9.33	409	27,650	15.70	456	30,900	21.90	500	33,950	28.85	542	36,600	36.50
9	47	11.81	364	35,050	19.90	405	39,100	27.70	445	42,950	36.55	482	46,350	46.20
10	52	14.58	327	43,250	24.55	365	48,300	34.20	400	53,000	45.15	433	57,200	57.00
11	58	17.64	297	52,300	29.70	332	58,450	41.45	364	64,100	54.60	394	69,300	69.00
12	63	21.00	272	62,300	35.50	304	69,550	49.25	334	76,400	65.00	361	82,500	82.15
13	68	24.65	252	73,050	41.50	280	81,600	57.80	308	89,550	76.30	334	96,750	96.45
14	73	28.68	234	84,900	48.15	261	94,600	67.05	286	103,900	88.70	310	112,050	111.90
15	78	32.80	218	97,250	55.25	243	108,700	77.00	267	119,200	101.50	289	128,800	128.20
16	84	37.32	204	110,750	62.85	228	123,600	87.50	250	135,800	115.50	271	146,400	146.00
17	89	42.14	192	125,000	71.00	214	139,500	99.00	235	153,100	130.30	255	165,300	164.80
18	94	47.24	182	140,000	79.50	203	156,500	110.80	222	171,800	146.00	241	185,300	184.60
19	99	52.63	172	156,000	88.55	192	174,200	123.40	211	191,200	162.80	228	206,200	206.00
20	105	58.32	164	173,000	98.25	183	193,000	136.80	200	212,000	180.30	217	229,000	228.00

Total pressure is 127.4 per cent of the rated static pressure.

TABLE 15-5b
CAPACITIES OF BUFFALO TURBO CONOIDAL FANS (TYPE T) UNDER AVERAGE WORKING CONDITIONS
70° F and 29.92-in. Barometer

Size	Diam-eter of Blast Wheel, in.	Area of Outlet, ft ²	¼-in. Static Pressure = 0.288 oz			¾-in. Static Pressure = 0.433 oz			1-in. Static Pressure = 0.577 oz			1½-in. Static Pressure = 0.865 oz		
			Rpm	Volume of cubic feet per minute	Horse-power	Rpm	Volume of cubic feet per minute	Horse-power	Rpm	Volume of cubic feet per minute	Horse-power	Rpm	Volume of cubic feet per minute	Horse-power
2½	14½	0.91	1115	1,230	0.20	1368	1,500	0.36	1580	1,740	0.56	1935	2,120	1.03
3	17½	1.31	930	1,770	0.28	1140	2,160	0.52	1315	2,500	0.81	1610	3,060	1.48
3½	20	1.79	797	2,410	0.39	976	2,940	0.71	1130	3,410	1.10	1380	4,160	2.02
4	22½	2.33	697	3,140	0.51	855	3,850	0.93	987	4,450	1.44	1208	5,440	2.64
4½	25½	2.95	620	3,980	0.64	760	4,860	1.18	879	5,640	1.82	1075	6,890	3.34
5	28½	3.64	558	4,910	0.79	684	6,000	1.45	790	6,950	2.25	966	8,500	4.12
5½	31½	4.41	507	5,950	0.96	621	7,270	1.76	719	8,400	2.72	880	10,300	5.00
6	34½	5.25	465	7,070	1.14	570	8,650	2.09	658	10,000	3.24	806	12,230	5.77
6½	36½	6.16	430	8,300	1.33	526	10,200	2.46	608	11,750	3.80	743	14,350	6.96
7	39½	7.14	398	9,630	1.55	488	11,780	2.85	565	13,610	5.40	690	16,650	8.09
7½	42½	8.19	372	11,050	1.78	456	13,500	3.27	526	15,610	5.05	645	19,100	9.27
8	45½	9.33	349	12,590	2.02	428	15,370	3.72	495	17,800	5.75	604	21,750	10.55
8½	48	10.53	328	14,200	2.28	402	17,380	4.21	465	20,100	6.50	569	24,600	11.90
9	51½	11.81	310	15,900	2.56	380	19,450	4.71	440	22,500	7.29	536	27,500	13.35
10	56½	14.58	279	19,650	3.16	342	24,050	5.82	395	27,800	9.00	483	34,000	16.50
11	62½	17.64	254	23,800	3.82	311	29,100	7.05	359	33,700	10.90	439	41,100	19.95
12	68	21.00	232	28,300	4.55	286	34,600	8.40	329	40,100	12.95	402	49,000	23.80
13	73½	24.65	214	33,200	5.34	263	40,600	9.85	304	47,000	15.20	372	57,500	27.90
14	79	28.68	198	38,500	6.20	244	47,100	11.40	282	54,500	17.62	345	66,700	32.35
15	84½	32.80	186	44,200	7.11	228	54,050	13.08	264	62,600	20.20	322	76,500	37.15
16	90½	37.32	174	50,300	8.09	214	61,500	14.90	247	71,200	23.00	302	87,100	42.25

TABLE 15-5b (Continued)
CAPACITIES OF BUFFALO TURBO NIAGARA CONOIDAL FANS (TYPE I) UNDER AVERAGE WORKING CONDITIONS
70° F and 29.92-in. Barometer

Size	Diameter of Blast Wheel, in.	Area of Outlet, ft. ²	2-in. Static Pressure = 1.154 oz			2½-in. Static Pressure = 1.442 oz			3-in. Static Pressure = 1.734 oz			3½-in. Static Pressure = 2.019 oz		
			Rpm	Volume of cubic feet per minute	Horse-power	Rpm	Volume of cubic feet per minute	Horse-power	Rpm	Volume of cubic feet per minute	Horse-power	Rpm	Volume of cubic feet per minute	Horse-power
2½	14½	0.91	2325	2,455	1.59	2490	2,750	2.22	2740	3,010	2.94	2958	3,250	3.69
3	17½	1.31	1860	3,540	2.29	2075	3,960	3.19	2282	4,330	4.23	2463	4,680	5.30
3½	20	1.79	1595	4,800	3.12	1780	5,390	4.35	1958	5,890	5.75	2115	6,360	7.22
4	22½	2.33	1395	6,270	4.08	1559	7,050	5.68	1713	7,700	7.52	1850	8,320	9.45
4½	25½	2.95	1240	7,950	5.16	1385	8,920	7.19	1522	9,740	9.52	1645	10,550	11.95
5	28½	3.64	1117	9,800	6.37	1249	11,000	8.87	1370	12,000	11.75	1480	13,000	14.75
5½	31½	4.41	1015	11,880	7.72	1133	13,300	10.75	1245	14,550	14.23	1345	15,750	17.85
6	34½	5.25	932	14,120	9.18	1040	15,800	12.78	1141	17,300	16.92	1232	18,700	21.25
6½	36½	6.16	860	15,690	10.76	960	18,600	15.00	1054	20,300	19.85	1139	22,950	24.90
7	39½	7.14	799	19,250	12.50	891	21,550	17.40	978	23,550	23.05	1056	25,450	28.90
7½	42½	8.19	745	22,100	14.32	831	24,750	19.95	914	27,050	26.40	987	29,200	33.20
8	45½	9.33	700	25,100	16.30	780	28,150	22.70	856	30,800	30.10	925	33,300	37.75
8½	48	10.53	657	28,400	18.40	736	31,800	25.60	807	34,750	33.95	870	37,550	42.25
9	51½	11.81	621	31,800	20.65	693	35,600	28.75	761	38,950	38.05	822	41,050	47.80
10	56½	14.58	559	39,300	25.50	625	44,000	35.50	685	48,100	47.00	740	52,000	59.00
11	62½	17.64	507	47,450	30.85	567	53,250	42.95	623	58,150	56.90	673	62,000	71.45
12	68	21.00	465	56,500	36.75	520	63,500	51.10	570	69,250	67.70	616	74,950	85.00
13	73½	24.65	430	66,200	43.05	480	74,400	60.00	527	81,300	79.40	569	87,900	99.60
14	79	28.68	399	76,800	50.00	445	86,300	69.55	489	94,300	92.10	528	101,900	115.60
15	84½	32.80	373	88,500	57.40	415	99,000	80.00	456	108,000	105.70	493	117,000	133.00
16	90½	37.32	349	100,500	65.30	390	112,500	90.90	428	123,000	120.50	462	133,000	151.00

Total pressure is 122.7 per cent of the rated static pressure.

air horsepower vary as the first, the second, and the third powers of the fan speed. For example, if a certain fan requires 3.95 hp at 354 rpm in order to deliver 18,000 cu ft of air at 1.0 in. of water static pressure, what will be required at a capacity of 20,000 cu ft? The new rpm will be $354 (20,000/18,000) = 393$, the new static pressure will be $1.0 (20,000/18,000)^2 = 1.236$ and the new horsepower will be $3.95 (20,000/18,000)^3 = 5.42$.

Tables and diagrams for the flow of air and for fan performance are generally calculated with a density of the air corresponding to dry air at 70° F. For *any other condition* at constant speed of the fan the static pressure and the horsepower vary directly as the density of the air. For any other condition at constant pressure the speed, capacity, and power vary inversely as the square root of the ratio of the densities. For example, if the speed of the fan in the above problem is increased sufficiently to keep the static pressure constant at 1.0 in. of water with an air temperature of 200° F,

$$(a) \quad \text{The speed will have to be } 354 \left(\frac{0.075}{0.0602} \right)^{1/2} = 395 \text{ rpm.}$$

$$(b) \quad \text{The capacity will become } 18,000 \left(\frac{0.075}{0.0602} \right)^{1/2} = 20,040 \text{ cu ft.}$$

$$(c) \quad \text{The horsepower will become } 3.95 \left(\frac{0.075}{0.0602} \right)^{3/2} = 4.4.$$

The loss of head due to air flow in ducts is directly proportional to the density of the air. If the static pressure loss in a certain duct amounts to 1.0 in. of water calculated for the standard temperature of 70° F, what will be the loss at 40° F? As the density of air at 40° F is 0.07945 the loss will be $1.0 \times (0.07945/0.07495) = 1.06$ in. of water.

Example. As an example of the preceding principles the flow of 1624 lb of dry air at 48.2° F dew-point temperature through a very simple single duct will be taken, involving two ells, a single Y-connection, and sixteen 14-in. outlets.

Solution. The volume of the air at 48.2° F entering the fan from the spray chamber is $1624 \times 12.92 = 21,030$ cfm, or 21,000 cu ft for design purposes. If 16 outlets are used the volume per outlet will be about 1312 cfm, but as the air absorbs heat and is heated by friction in the fan and duct the volume passing through the outlets is somewhat greater than 1312 cfm.

As already stated, there are two general methods for designing of ducts. Since the volume of air passing each cross section is known, either the velocity can be selected, which will vary according to engineer-

ing practice, or the pressure drop per foot or per 100 ft of duct length can be chosen from established engineering practice and be kept constant for the entire length of the duct. These two factors determine the size of the duct.

Example. Let the friction drop be taken as 0.14 in. of water per 100 ft of length.

Solution. Referring to Fig. 15-6, the diameter and velocity at the different sections will be as shown in Table 15-6.

TABLE 15-6

Section	Volume of Air in Section, fpm	Diameter of Duct, in.	Equivalent Rectangular Duct for $\frac{a}{b} = 3$	Velocity of Air in Duct, fpm round section, (from Fig. 15-6)	Length of Section, ft	Allowances for Changes in Direction
a	2,625	19	30.7 × 10.3+	1350	22	Residual velocity Loss in branch, 30° with main duct = 0.17 × VP in branch
b	5,250	25	40.4 × 10.1+	1520	22	
c	7,875	29.6	47.9 × 16.0-	1640	22	
d	10,500	33.0	53.4 × 17.8	1770	22	Loss in rounded Y = 0.15 × VP
e and f	21,000	43.6	70.5 × 23.5	2100	60	Loss in 2L = 2 (0.15 × VP)

The tabulation makes an allowance for the loss of head due to two ells near the fan, the Y-connection, the branch outlet, and the straight duct. The fan must develop enough head to deliver the required volume of air at the remotest outlet, so that the nearest outlets have to be provided with dampers. As the two duct branches out of the Y are in parallel and are symmetrical it will be necessary to calculate the resistance for one branch only. The air must pass out of the branches with a velocity of 1225 fpm, which of itself corresponds to a static pressure head of 0.09+ in. of water. In this simple design the air will work back to the spray chamber, although a similar duct could be used for the return. The loss of static head in the spray chamber will be taken as 0.55 in. of water. The total resistance to flow, which must be developed by the circulating fan, is

$$h = \left(\frac{148}{100} \times 0.14\right) + [(0.15 \times 0.275) 2] + (0.15 \times 0.202) + (0.17 \times 0.09) + (0.09 + 0.55) = 0.975 \text{ in. of water}$$

In calculating the diagram (Fig. 15-6) for losses of head due to friction the density of the air was taken as 0.075; for any other temperature the multiplying factor can be found by calculating the value for the inverse ratio of the absolute temperatures. For example, the factor for 40° F will be 530/500 = 1.06, and for 50° F it will be 530/510 = 1.04. The total head therefore becomes 0.975 × 1.04 = 1.014.

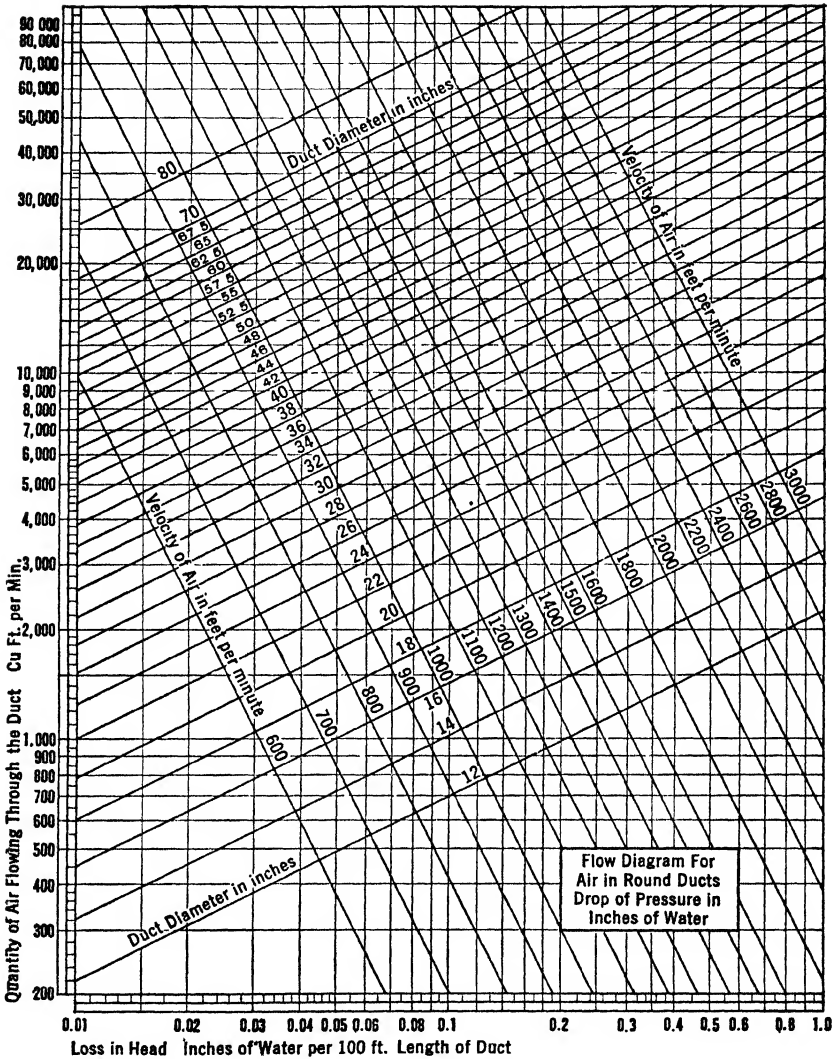


FIG. 15·6. Flow diagram for air in round conduits.

A fan should be selected to provide 21,000 cfm at a total head of 1.02 in. of water. Figure 15·7 presents a good example of duct design.

15·8. Air and Water Horsepower. The power required to circulate air can be found from the following considerations.

$$\begin{aligned} \text{Work performed} &= \text{weight of air in pounds} \times \text{total lift in feet} \\ &= Q\rho a h_a \text{ ft-lb} \end{aligned}$$

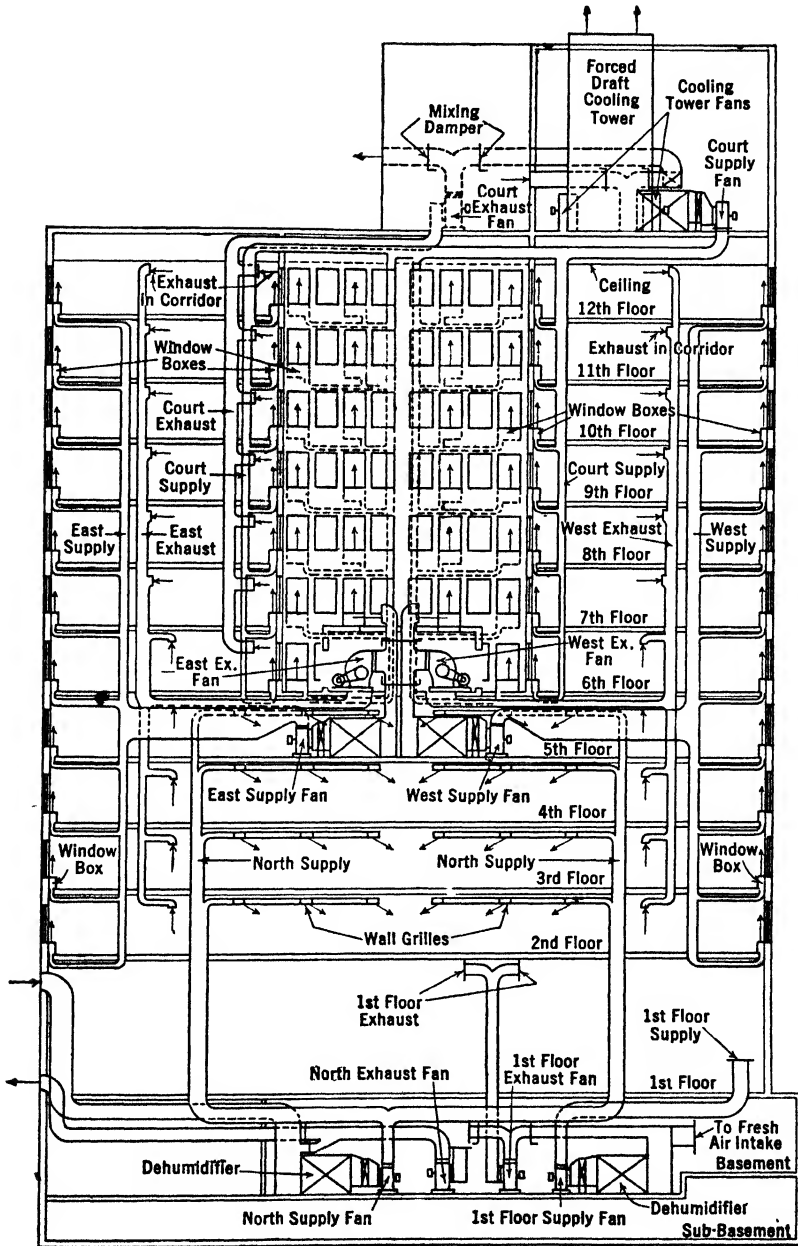


FIG. 15-7. The banks—Huntley Building, Los Angeles.

where Q = volume of air in cubic feet; ρ_a = the density of the air pounds per cubic foot; h_a = the pumping head, feet of air; but as fans and blowers are always reported in inches of water static or total head we have from hydraulics

$$h_a \rho_a = \frac{h_{\text{in of water}}}{12} \rho_{\text{of water}}$$

Therefore

$$h_a = \frac{h_u \text{ in inches} \times \rho_w}{12 \rho_a}$$

or the horsepower

$$= \frac{5.2Qh_w}{33,000e} \quad (15 \cdot 10)$$

where h is the total head, inches of water; e is the efficiency of the fan or blower, varying from 0.5 to 0.7; ρ_w is the density of water.

The work of pumping water is found in a similar manner and becomes

$$\text{hp} = \frac{8.33Qh}{33,000e} \quad (15 \cdot 11)$$

where Q = the number of gallons of water per minute; h is the total lift of the pump in feet; e is the efficiency of the pump, usually about 0.5.

15.9. Sound Control. As air conditioning also involves the control of sound the methods used in reducing noises due to machinery and the flow of air must be considered. The *intensity* in acoustics is given as the quantity of work performed per unit area or

$$I = \frac{p^2}{\rho w 10^7} \quad (15 \cdot 12)$$

where p = the sound pressure in dynes per square centimeter, w is the sound velocity in centimeters per second, ρ is the air density in grams per cubic centimeter. The relationship of sensation (loudness) as compared to the ratio of physical sound intensity is approximately as the logarithm of the intensities. Thus an increase of 1000 times in the amount of physical noise gives only twice the amount of intensity. The increase in noise caused by equipment over that not caused by the equipment varies as the sum of the physical sound energy existent in the room and the noise energy produced by the apparatus. For example, let

I_1 = the initial room intensity above zero

I_2 = the apparatus intensity above zero

I_3 = the final room intensity above zero

Then $I_3 = I_1 + I_2$, or

$$\text{Decibels} = 10 \log \frac{I_3}{I_1}$$

The relation of physical intensity to sensation, bel, is given by

$$\text{Sensation} = \log \frac{I_2}{I_1}$$

The least perceptible change of sound detected by the human ear approximates 0.1 bel or a decibel (db); therefore

$$\text{db} = 10 \log \frac{I_2}{I_1} \quad (15 \cdot 13)$$

The acceptable noise levels are from 8 to 10 db in hospitals, 12 to 24 in theaters, and 20 to 30 in offices.

15·10. Sound Insulation of Machinery. It is a mistake to assume that a pad of cork or felt under a machine will sound-insulate the machine from the building. It will be an insulation if the machine is sufficiently heavy and the cork or felt is sufficiently resilient so that the natural frequency of the machine on the pad is low compared with the frequency generated by the equipment; otherwise the pad is of no value. Elastic supports (springs) may be used successfully, but the design of these supports should be the result of calculation. For example, if the principal vibrations are of the order of approximately 100 vibrations per second and the natural frequency of the machine mounted on its elastic pad is 50 or preferably 25 to 20, the noise of the machine will be isolated. The same thing will be true if the frequency of the mounting is increased from a machine with a forced frequency of 10 to one of a natural frequency of 20 to 25.

15·11. Control of Noise Transmission through Ducts. The reduction, in decibels, of noise transmitted through a duct is approximately

- (a) Directly as the length.
- (b) Directly as the perimeter of the duct.
- (c) Inversely as the cross-sectional area.
- (d) Approximately as the coefficient of sound absorption of the material lining the internal area of the duct.

It is therefore apparent that narrow ducts lined with highly absorptive material will absorb a large percentage of the machinery noises. For example, small ducts, say 4 by 6 in., made of a material having a coefficient of sound absorption of 0.50, will provide a noise reduction of slightly more than 1 decibel per lineal foot.

PROBLEMS

1. Steam, initially dry, saturated, at 200 psia flows at the rate of 600 lb/min through a 6-in. steel pipe. Find the drop in pressure per 100 ft of pipe.
2. Gasoline flows through a steel pipe at the rate of 740 gpm. The specific gravity is 0.75. Find the loss of pressure in 8 miles of 8-in. pipe.
3. Find the loss of head, due to the flow of both ammonia and Freon-12, per 100 ft of length in a 1-in. diameter steel pipe for the suction and discharge pipe line. The pressures correspond to liquefaction at 90° and evaporation at 30° F. The vapor in the suction pipe will be superheated 10° F and will have a velocity of 3000 fpm, and the vapor in the discharge pipe will be superheated 60° F and will have a velocity of 5000 fpm. What is the conclusion regarding Freon-12 velocities in pipes or valves?

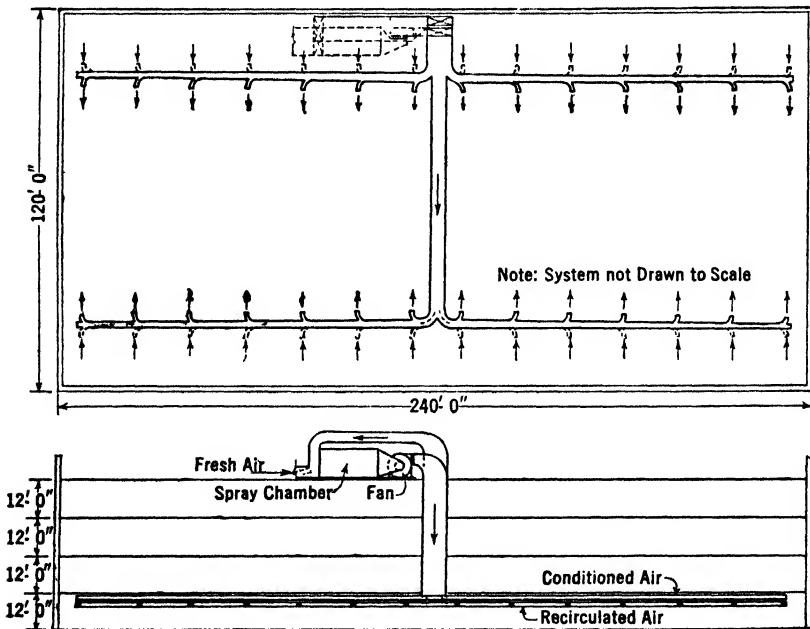


FIG. 15-8. Problem in duct design.

4. Design a duct system for the following conditions: 30,000 cu ft of air per minute at 40.5° F. Use 28 outlets as indicated in Fig. 15-8. Arrange a tabulated form showing the diameters and velocities of the air for the separate duct sections, on the basis of a static pressure loss of 0.12 in. of water per 100 ft of length. Show also the sizes of equivalent rectangular sections, with width to breadth of 3.0. Allow for additional resistances due to ells, Y, branch outlets, inlets to duct, residual and initial velocity heads, spray chamber, and inlet into spray chamber. Take static to total head ratio equal to 0.85. Allow for reduced temperature of air. Use the arrangement of ducts shown. Find the horsepower and the total head of the fan. Select a suitable fan for the conditions of operation and state the required rpm.

Part III
REFRIGERATION EQUIPMENT

CHAPTER XVI

COMPRESSION REFRIGERATING MACHINES

✓ 16.1. History and Development of Refrigerating Machines.

The first types of apparatus for the mechanical production of cold were constructed on the principle of the vacuum machine, wherein a vacuum, obtained by means of the machine, permitted the liquid refrigerant to boil at a sufficiently low temperature to secure the results required.)

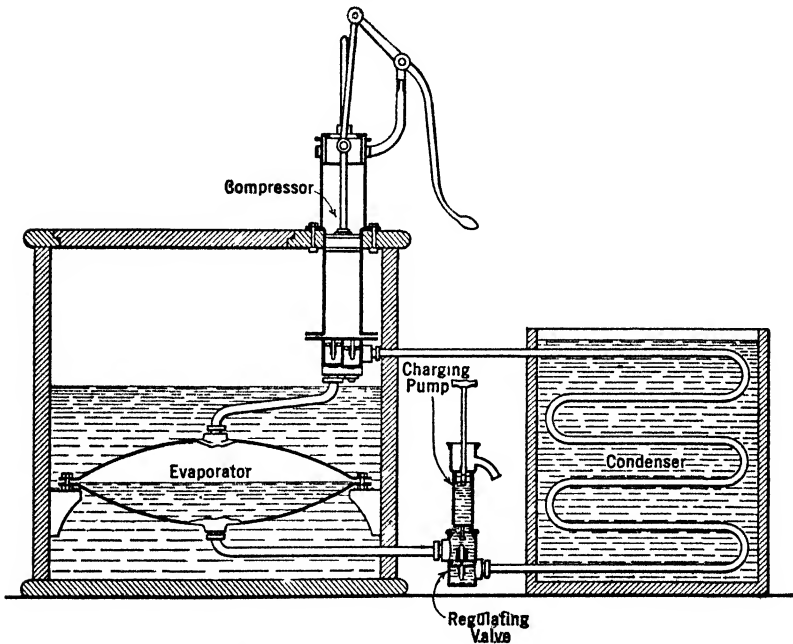


FIG. 16-1. The first practical refrigerating machine using the compression cycle, built by Jacob Perkins in 1834.

These machines included that of William Cullen (1755) which used water under a high vacuum, Vallance's machine of 1824, and Edmund C. Garré's device of 1850, the last two using sulphuric acid to absorb the water vapor and thereby permitting less severe operating conditions than were necessary with the air pump. In or about 1845 Dr. John Gorrie developed the cold air machine by using a closed cycle and an

expansion cylinder, and his device was improved by Kirk in 1861, by Postle in 1868, and by Windhausen and Nehrlich in 1869. (The air refrigerating machine was perfected by James Coleman and John and Henry Bell in 1877 and subsequent years, with the result that the Bell-Coleman compressor became very well known and mechanical refrigeration gained a substantial impetus.) (The present-day reciprocating compressor using vapor is based, however, on the designs of Linde, Ferguson, and John de la Vergne.)

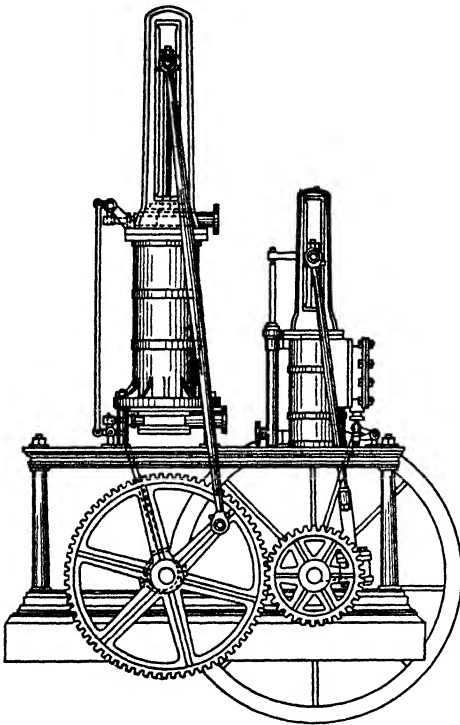


FIG. 16-2. The James Harrison sulphuric ether machine, with steam engine, 1859.

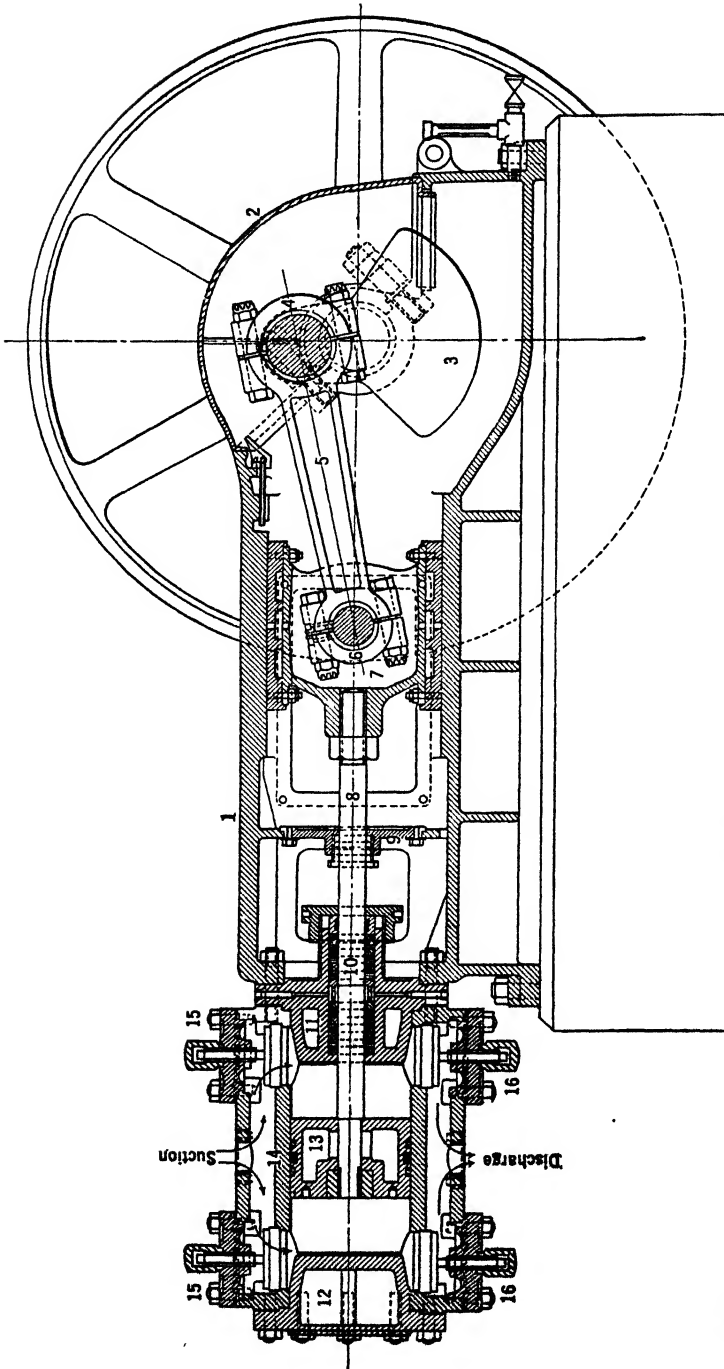
The first *refrigerant* other than air or water was that used by Jacob Perkins, whose compression machine (Fig. 16-1) was invented in 1834. This machine was intended to employ sulphuric ether, $(C_2H_5)_2O$, but it never went beyond the experimental stage. The Perkins machine was revived by Dr. James Harrison in 1857, who also used sulphuric ether, and he was quite successful in producing refrigeration for breweries and in perfecting machines (Fig. 16-2) for the cooling of meats and other perishable products. Finally Dr. Carl Linde, in 1873, introduced the ammonia compressor, and in 1876 Raoul Pictet devised the sulphur dioxide compressor using water to cool the piston and piston rod as well as the cylinder. In the eighties, mixtures of different refrigerants and volatile liquids like ethyl and methyl chloride, ethylic ether, acetylene, ethylsulphurous dioxide, naphtha, gasoline, cyrogene, etc., were given attention. In 1850 E. Carré introduced the small intermittent absorption machine employing water and sulphuric acid, and this was improved by Ferdinand Carré in 1859, using ammonia and water, by Windhausen in 1878, Pontifex and Wood in 1876, and others, also using ammonia and water. For a while by a system of elimination the practical refrigerants were narrowed, so far as Great Britain and the United

States were concerned, to ammonia, sulphur dioxide, and carbon dioxide; but more recently interest has been directed toward other refrigerants such as methyl chloride (CH_3Cl), dichlorodifluoromethane (CCl_2F_2), methylene chloride (CH_2Cl_2), dichloroethylene ($\text{C}_2\text{H}_2\text{Cl}_2$), etc.

16.2. Early Compressor Design. The design of the compressor used in the refrigerating industry has largely paralleled that of the air compressor and to some extent the development of the steam engine. The first compressors were very slow, operating at 50 to 55 rpm. All were steam-driven, most of them with Corliss engines, manufactured by de la Vergne, Frick, and York with vertical compressor cylinders, or by Vilter, Ball, and Wolf with the horizontal double-acting compressor of the Linde design using liquid ammonia injection in order to keep the cylinder cool. The first de la Vergne machines were vertical double-acting, having the suction valves in pockets for safety and injecting oil during the suction stroke to improve the volumetric efficiency (according to the theory at that time). More than 1000 de la Vergne machines of this design were installed in the eighties and the nineties, and they gave good service.

About 1880 to 1885 a number of the present-day well-known plants for refrigerating equipment came into existence, developing a line of large compressors for breweries, packing plants, and the manufacture of ice. In 1896 a vertical, single-acting, twin, open-frame compressor with the flywheel placed between the columns was installed in Kansas City with cylinders 27 in. in diameter with 48-in. stroke, and during the next decade even larger machines were built, one, for the Quincy Market and Warehouse Company, having a capacity of 1000 tons of refrigeration. The features of these designs were economical operation; they used the most economical steam engine, including, in the last part of the period, the Lentz and the uniflow types of steam engines, and later, with higher rotative speeds, the oil engine of the Diesel and the semi-Diesel design. The ammonia valves were of the poppet type exclusively, both suction and discharge valves being cushioned, and just before the turn of the century the suction valve was placed in the piston and balanced by a coil spring so as just to compensate for its weight, thereby making it act as an inertia valve. It was made large enough in net opening to reduce the wire drawing of the suction gas to a minimum. As the suction gas entered below the piston the stuffing box was subjected to suction pressure only.

During the period from 1890 to 1900 the demand for small capacities developed, and compressors as small as about 2 tons of refrigeration were put into operation; but shortly after 1900 the electric motor drive was applied, materially changing the whole design. Next there ap-



- | | | | |
|------------------------------|-------------------------|--------------------------------|--------------------|
| 1 Frame | 5 Connecting rod | 9 Oil guard with wiper rings | 13 Piston |
| 2 Crankcase cover | 6 Crosshead pin bearing | 10 Stuffing box for piston rod | 14 Cylinder |
| 3 Counterbalanced crankshaft | 7 Crosshead | 11 Front cylinder head | 15 Suction valve |
| 4 Crank pin bearing | 8 Piston rod | 12 Back cylinder head | 16 Discharge valve |

Fig. 16-3. A typical center crank ammonia compressor.

peared the *enclosed* type of compressor with splash and, later, forced-feed oil lubrication, high rotative speeds, and, finally, an entirely automatic type of compressor.

16.3. The Horizontal Double-Acting Compressor. The *horizontal double-acting compressor* (Fig. 16.3) has changed but little since the first Linde compressor was built. However, since 1920 the design has been modified by increasing the rotative speed, by the use of: (a) plate or ribbon valves for both suction and discharge, (b) metallic rod packing, (c) water jackets rather than wet compression, and (d) forced feed for oil lubrication. Figure 16.4 shows a design of a horizontal double-acting ammonia compressor using plate valves, the discharge valves being at the lower part of the cylinder. This machine is designed for piston speeds of approximately 600 fpm, and varying from 200 to about 140 rpm. Forced-feed lubrication is used throughout. The enclosed compressor is generally considered in three separate groups, the fractional tonnage, the size from 1 to 8 tons of refrigeration, and the industrial type which is greater than 8 tons.

16.4. The Vertical Single-Acting Compressor. Most vertical single-acting compressors (Figs. 16.5 and 16.10) have two cylinders, although often in British designs there are four. The two cylinders, the crankcase, and the outboard bearing may be in one casting, or with a separate crankcase which may carry the outboard bearing. In some designs the lubrication is forced-feed for all sizes, whereas in others splash lubrication is found in the smaller sizes. The water jacket usually cools part of the head and the upper part of the cylinder barrel. The piston and crosshead are combined to form a trunk piston, the upper and lower part being connected by columns or the middle portion being cut away for the passage of the suction gas, allowing free passage to the suction valve for all positions of the piston. The upper part of the piston usually has three or four, and the lower part two or three rings. The tolerance of the piston increases with the diameter, being approximately 0.001 or 0.002 in. for pistons up to 4 in. in diameter, 0.002 to 0.003 for 4- to 6-in. diameter, and 0.003 to 0.004 for 8-in. diameter and over.

The materials of construction are usually close-grained cast iron for cylinders calculated on the basis of about 1000 psi fiber stress, although some manufacturers use a special steel casting or even a nickel steel. The cylinders should be tested for 400 psi water and 300 lb air pressure. The wrist pin should be of hardened steel with a working fiber stress of 6000 psi. The connecting rod is generally of forged steel, the I-beam section being calculated with a maximum fiber stress in tension and also in compression of 6000 lb psi. Some manufacturers are using steel cast-

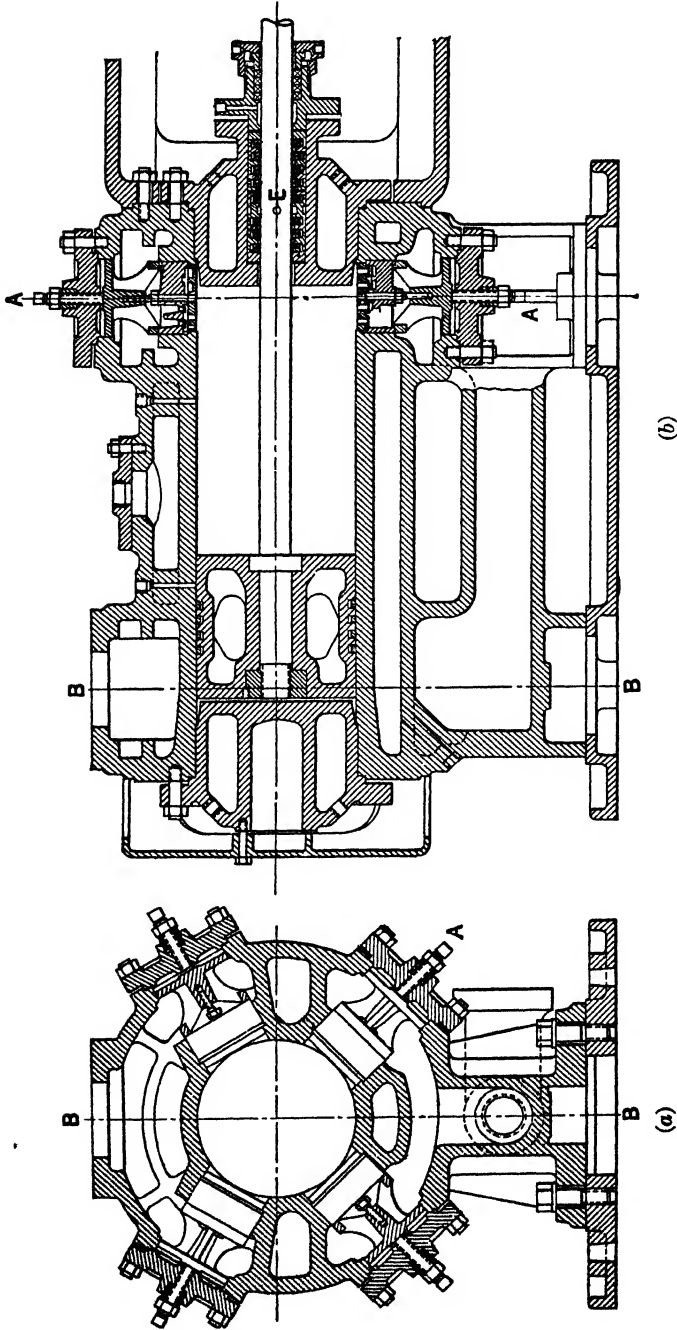


FIG. 16-4. The horizontal double-acting ammonia compressor. (a) Cross section through the valves. (b) Sectional elevation.

ings also for connecting rods. The piston commonly is made of the same material as the cylinder. The wrist pins are of low-carbon steel, casehardened and ground. The crankshaft is always a steel forging, heat-treated and carefully finished. Figure 16·6 shows one design for variable capacity at constant rotative speed.

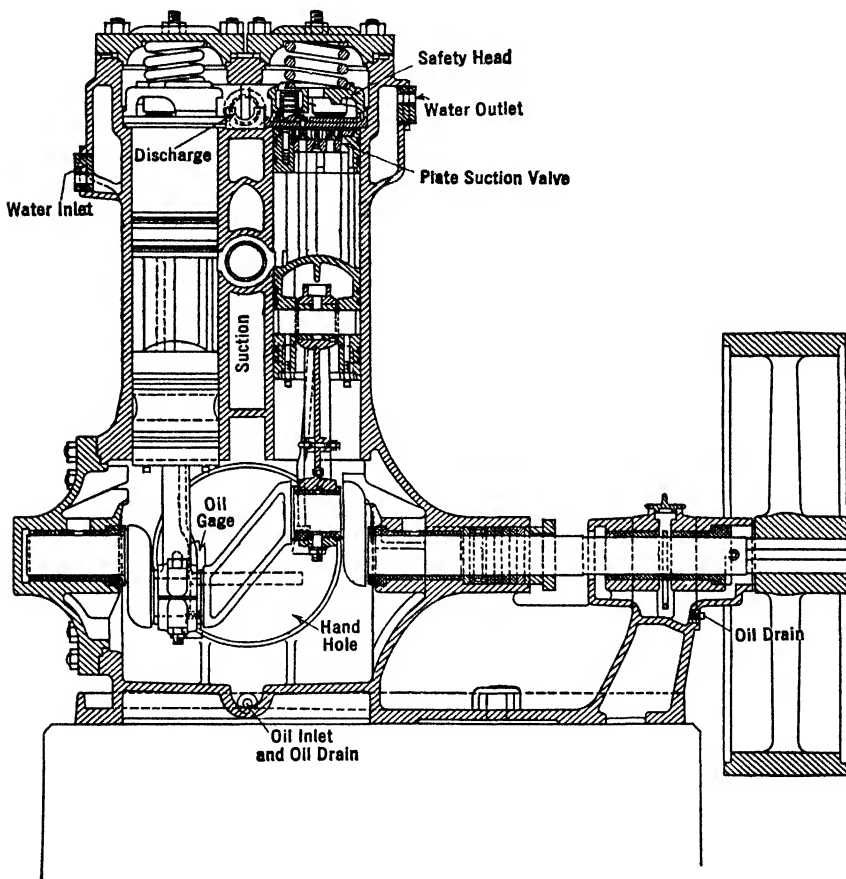


FIG. 16·5. The enclosed type of twin vertical single-acting ammonia compressor for splash lubrication.

The striking clearance in this form of ammonia compressor is very small, frequently being only $\frac{1}{64}$ in., so that the volume in the clearance is only about 1 per cent of the piston displacement compared with 3, 4, and more per cent in the horizontal double-acting compressor. Owing to the long-established fear of rupture the cylinder is still made with a safety head (Figs. 16·10 and 16·5), this head being held in place by

means of a heavy coil spring and containing the discharge valve or valves of the lightweight poppet or of the ring-plate type. Both the suction and discharge poppet valves, even when they are of light weight,

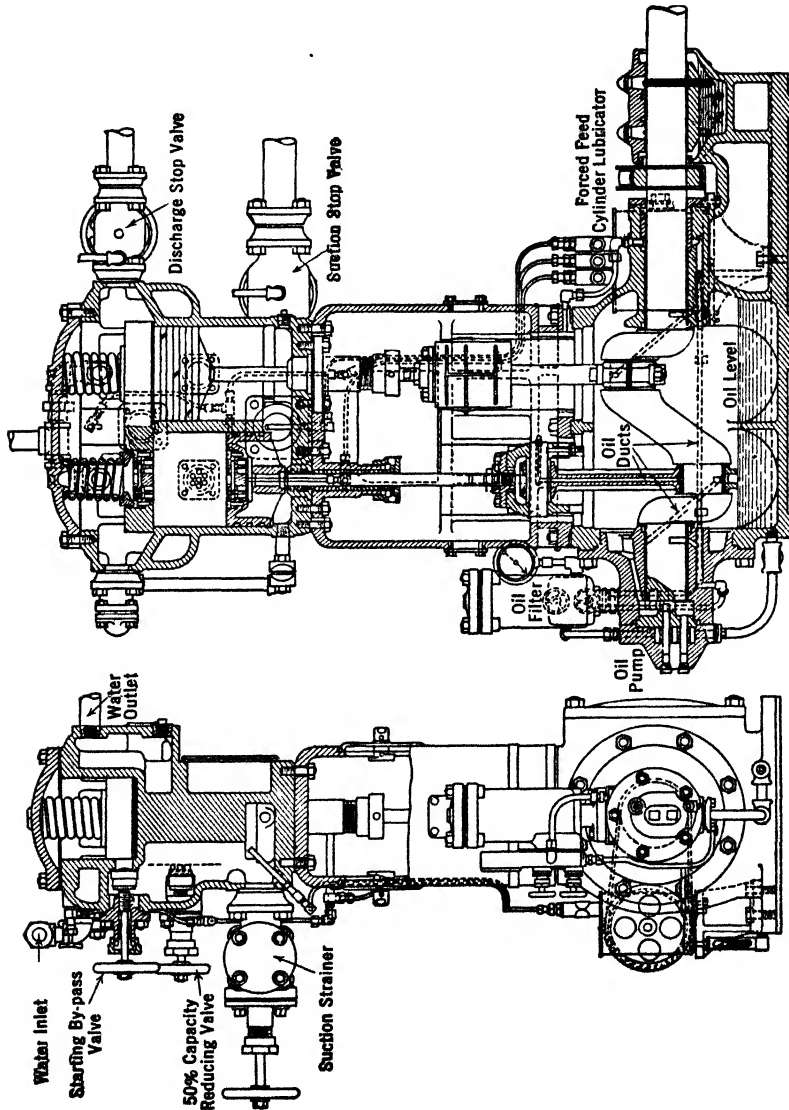


Fig. 16-6. Ammonia compressor, designed for forced-feed lubrication and variable capacity.

require cushioning, especially as the rotative speeds are increased so that the speeds of compressors with poppet valves are limited. Because of the variation of the suction as well as the condenser pressure, the

mechanically operated valve is not used in refrigeration except in one or two special designs, so that the valve is opened and closed by means of a difference of pressure on the two sides of the valve. The ring plate (Fig. 16·7) and the ribbon valve appear to have no reasonable limitation as regards speed because the inertia is comparatively small. Rotative speeds higher than 800 have been employed in some of the smaller

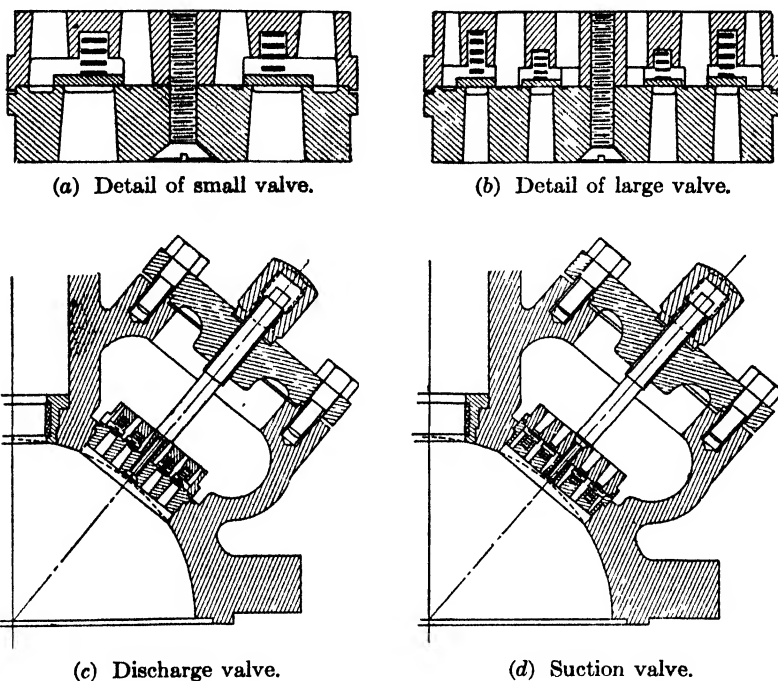


FIG. 16-7. The plate type of suction and discharge valve.

compressors, but, as refrigerating machines may sometimes operate for 6 months without stopping, designers, with safety in mind, have been conservative and have been inclined to keep the speed below 600 rpm. The piston speed also has been maintained from 600 to 700 fpm, under which conditions lubrication can be satisfactory.

The present-day valve of the ring-plate or the ribbon type requires a low entrance gas velocity and minimum valve lift. For speeds of 300 rpm and above, the lift of the valve should be $\frac{1}{16}$ in. or less, and the width of the seating surface should be approximately $\frac{1}{16}$ in., but not more than $\frac{3}{32}$ in. The average gas velocity should be held to 5000 fpm for the discharge and 4000 fpm for the suction with 6000 and 5000 fpm, re-

spectively, as the maxima. The valves must be kept from fluttering, and the springs must be designed with that idea in mind. These valves are assembled in cages and may be changed from a suction to a discharge valve by reversing the cage (Fig. 16-7).

16-5. Carbonic Refrigeration. Carbon dioxide refrigeration was one of the first types to be developed, and soon, in spite of certain inherent objectionable features, became very successful because it ap-

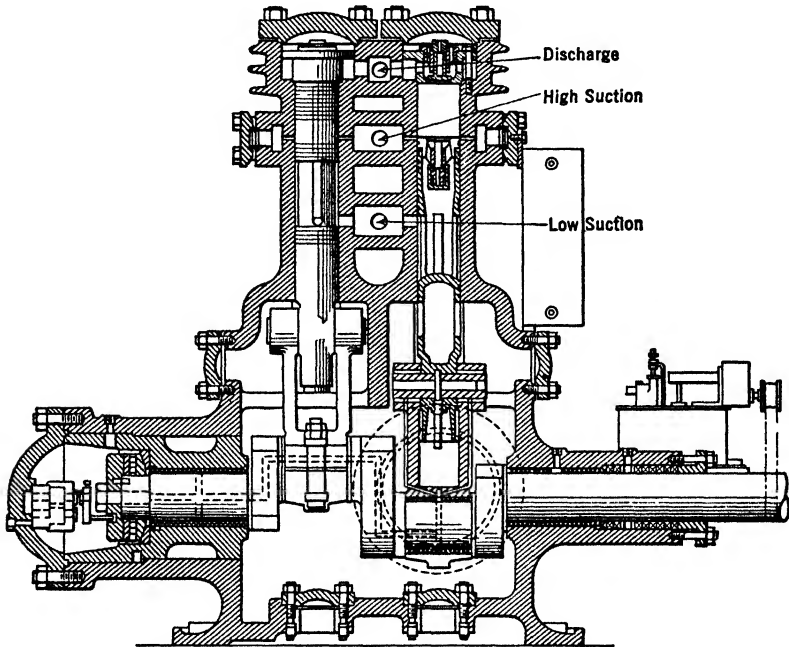


FIG. 16-8. The enclosed type of carbonic compressor, designed for dual compression.

proximated absolute safety from toxic effects. Certain kinds of work demanded particularly such safe refrigeration, as, for example, marine, theater, hotel, and office building refrigeration. Nevertheless, two main objections to the use of carbon dioxide as a refrigerant must be mentioned. The pressures are very high, usually from 300 to 450 psi evaporation and from 900 to 1100 psi liquefaction pressure, a fact which makes it difficult to obtain efficient operation even if the gas leaks are kept small. The low critical temperature of carbon dioxide is detrimental to a large coefficient of performance, especially as the latent heat of vaporization is small. It is always necessary to have a liquid seal on the pressure-reducing valve; otherwise non-condensed gas will pass through the condenser into the evaporating coils. In order to insure

that the liquid seal is always present it is essential that the liquid receiver be adequate and that some positive indicator show the level of the liquid in the receiver.

If the temperature of the refrigerant leaving the condenser is 75° F or higher the decimal part by weight which becomes vaporized during the throttling process in the pressure-reducing valve is 0.4 or more. Sometimes if the condensing water is high in temperature the refrigerant is not condensed but reaches the pressure-reducing valve as a dense vapor. However, during the reduction of pressure there is a change of state, first to all liquid and then to a liquid and a vapor.

The carbonic compressor is made both horizontal double-acting and vertical single-acting, the latter being either of the open-frame or the enclosed type. The enclosed machine is shown in Fig. 16·8. Its outstanding features are small ratio of cylinder diameter to stroke, heavy-walled cylinder and crankcase (which may be required to withstand pressures of 1000 to 1200 psi), good-sized crank pins, forced-feed lubrication, liberal number of piston rings, metallic rod packing, and finned cylinders for air cooling of the cylinder. The cylinder and crankcase are made of special semi-steel, and the remaining parts of the compressor follow ammonia practice. The open-frame type is being discarded because of operating difficulties such as gas leaks.

The horizontal double-acting compressor (Fig. 16·9) follows similarly standard practice except that it has a long stroke in order to reduce the cylinder diameter because of the heavy pressures imposed upon it. The cylinder is made of special semi-steel, the piston of a steel forging with a bull ring and piston rings of cast iron. It has metallic packing, connecting rod and crosshead of steel casting, and plate valves made of a special spring steel.

16·6. The Dichlorodifluoromethane Compressor. There is no material difference in the design of the dichlorodifluoromethane compressor, as seen in Fig. 16·10, from that of the ammonia or the methyl chloride compressor. However, the vapor density of this particular halide is three times that of methyl chloride and five times that of ammonia at 5° F evaporating temperature; at standard operating conditions it has theoretically about 69 per cent greater piston displacement than ammonia. The heavy vapor densities require relatively low vapor velocities in the suction and discharge pipelines, ports, and valves, and care in making changes in flow direction. Like methyl chloride, dichlorodifluoromethane is miscible in oil, and the evaporator should be so arranged as to assist in permitting the oil to return to the compressor where its separation must be arranged for. In comfort cooling, the temperature of discharge (see the Mollier diagram, Fig. 8·4) is usually low

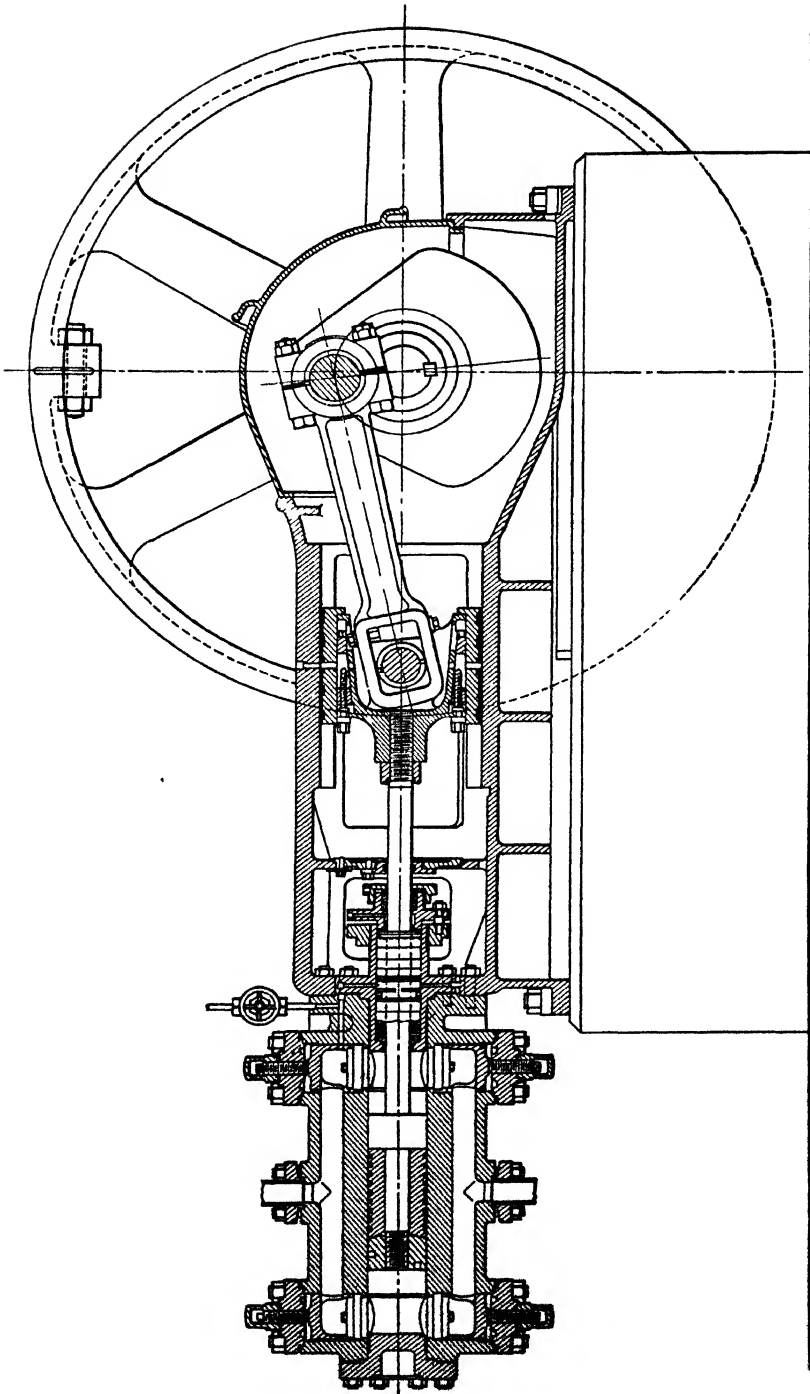


Fig. 16-9. Longitudinal section through a double-acting carbon dioxide compressor.

enough to permit the use of the air-cooled compressor cylinder, shown in Fig. 16·10 by fins on the cylinder.

16·7. Mechanical Features of Compressors. Figures 16·11 and 16·12 show, respectively, a typical small-size modern compressor and a

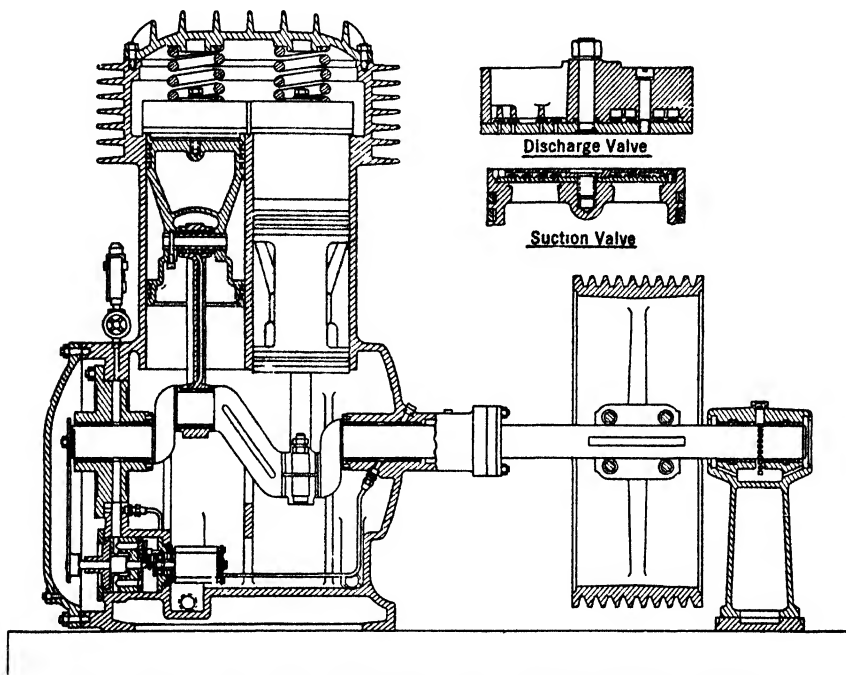


FIG. 16·10. Compressor designed for use of dichlorodifluoromethane.

typical unit of larger capacity. In the smaller unit (Fig. 16·11) the design features highlighted by the manufacturer are indicated by numbers shown on the figure and are identified as follows:

- A. Factors assisting in maintenance of high volumetric efficiency
 1. Large gas passages
 2. Wide-opening ring-type valves of Swedish steel
 3. Thin valve plate
 4. Small clearance between piston and valve plate
 5. Close tolerance in machining cylinders and pistons
 6. Constant oil supply to cylinder walls
- B. Pressure-controlled forced-feed lubrication
 7. Oil reservoir
 8. Strainer
 9. Rotary oil pump—an eccentric integral with crankshaft
 10. Drilled oil passages

11. Oil pressure regulator
 12. Sight glass for visual check on oil circulation
- C. Mechanical features
13. Patented valve backer
 14. Plated-copper gaskets
 15. Connecting rods of manganese-bronze alloy
 16. Casehardened-steel wrist pins
 17. Counterbalanced crankshaft
- D. Rotating balanced-bellows type of shaft seal
18. Rotating nose piece
 19. Stationary Nitroloy collar
 20. Neoprene washer

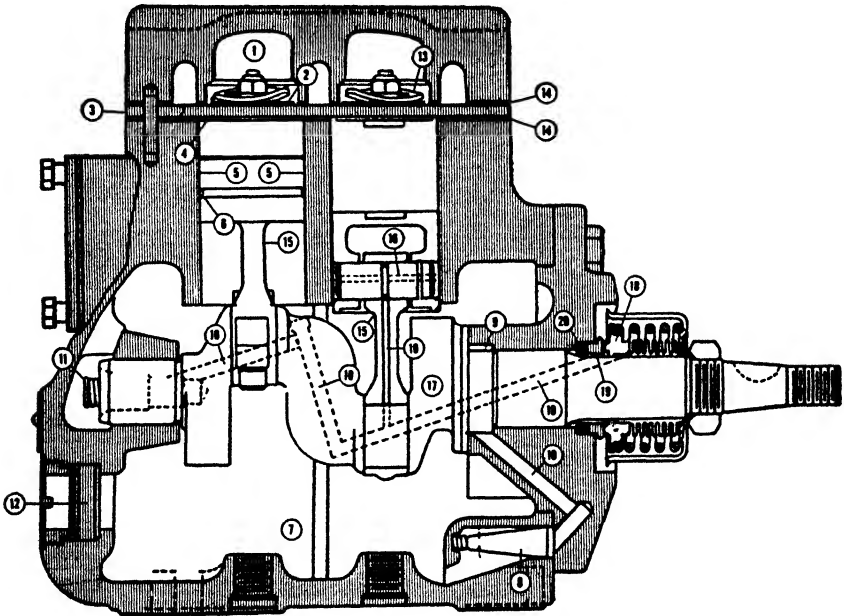


FIG. 16-11. (Courtesy General Electric Co.)

The major points of the larger unit (Fig. 16-12) are likewise numbered on the figure and are identified as follows:

1. Discharge valves of the ring-plate type, constructed with light moving parts; quiet, tight, and reliable.
2. Cylinder heads, held down by springs, lift in case slugs of liquid enter cylinder; wasteful clearance space eliminated.
3. Capacity control furnished on each cylinder; capacity of machine can be varied in steps of 100, $87\frac{1}{2}$, 75, $62\frac{1}{2}$, and 50 per cent.
4. Top of each piston carries three alloy-inserted compression rings, while the bottom has two oil-scraper rings.

5. Manifold and by-pass fabricated of steel pipe; special manifolds made for every purpose.
6. The stuffing box is made double length, and it holds pressure with the packing gland nuts only finger-tight.
7. Safety relief valve prevents injury to crankcase from excessive pressure.
8. Nine-point force-feed lubricator supplies oil to wall of each cylinder at two places and to the suction of the compressor.
9. Standard drive is through synchronous motor or V-belt wheel; direct-connected steam, gas, or oil engine, or any special drive, readily applied.

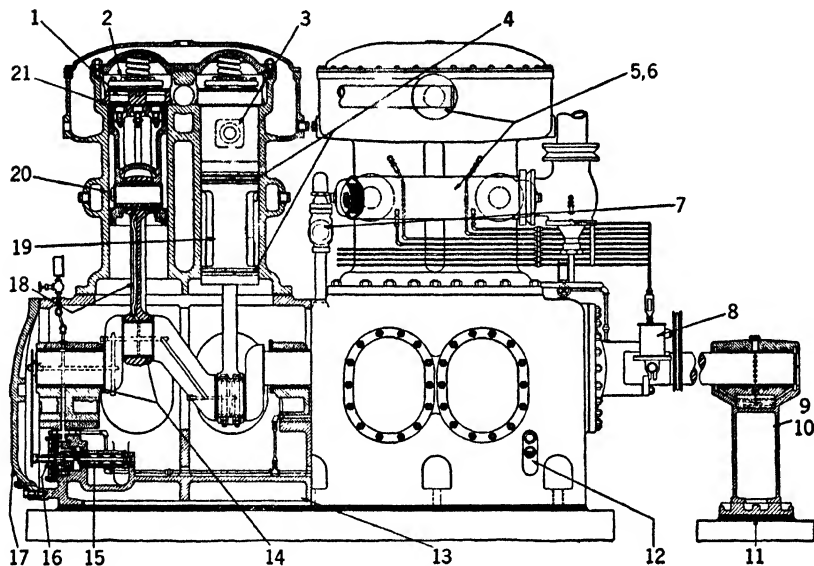


FIG. 16-12. (Courtesy Frick Co.)

10. Chain-oiled outboard bearing supports weight of motor, or flywheel and pull of belt; bearing lined with heavy-duty babbitt (chain or two oil rings provided).
11. Separate base plate under outboard bearing pedestal permits removal of bearing and wheel without disturbing concrete.
12. Two oil sight glasses allow checking of the oil level while the compressor is in operation.
13. Crankcase is a heavy one-piece casting of semi-steel, reinforced with stout ribs, assuring permanent alignment.
14. Crank bearings are of improved type, with shim adjustments; shaft bearings are die-cast of heavy-duty babbitt; split sleeves allow renewal without removal of shaft.
15. Mechanically driven self-cleaning filter purifies oil; filter plates are cleaned every 30 sec.

16. Internal oil pump, driven by roller chain, provides force-feed lubrication to all bearings and to shaft seal.
17. End cover plate encloses oil pump and filter; crankshaft is admitted through either end of the machine.
18. Connecting rods are of drop-forged steel, H-section, and are made long to lessen side thrust on the pistons.
19. Pistons are made of semi-steel, with ample babbitted thrust surfaces accurately ground to size.
20. Separate suction ports to each cylinder permit free gas passage; dual suction connections, one on each side, when desired.
21. Suction valves are of the multiple-poppet type and are floated by springs; the motion is also cushioned by dash pots (15-in. by 10-in. and $17\frac{3}{4}$ -in. by 12-in. machines use ring plate suction valves).

16·8. Clearance, Clearance Pockets, and Capacity Control.

Clearance was considered by the early designers of ammonia compressors to be something to avoid at all costs, and in the vertical single-acting type it has been kept down to less than 1 per cent of the piston displacement. The striking clearance of only $\frac{1}{64}$ in. was made safe from operating difficulties by introducing a false head held in place by heavy coil springs. This type of construction is still adhered to, although the plate type of valve now so common makes it difficult to limit the clearance to less than 1 per cent. However, it is now very generally understood that a clearance of 3 or 4 per cent of the piston displacement is not a disadvantage and does not represent a loss of power except with large ratios of condenser to suction pressures, where the loss of capacity may become excessive because of the re-expansion of the clearance gas during the drop of pressure at the beginning of the suction stroke.

The modern compressor is operated at practically constant speed. The result is that unless the design consists of an automatic control involving some temperature range the compressor invariably has to operate inefficiently; that is, the machine must have an excess capacity, and during operation the suction pressure will automatically drop until the product of the weight of refrigerant boiled per unit of time and its specific volume at the suction pressure will be equal to the piston displacement for the same unit of time, and this will result in a higher cost of operation. It would be much better to decrease the pumping capacity in some manner.

The most commonly used means of decreasing the pumping capacity, under conditions of constant rotative speed are:

- (1) Installing a by-pass valve between the cylinder barrel and the suction (Fig. 16·13) permitting return of vapor to the suction during part of the compression stroke; the device reduces the effective stroke

and thereby reduces capacity. In Fig. 16·13 the capacity reduction is 50 per cent.

(2) Utilizing one or more clearance pockets (Fig. 16·14) which are subject to manual adjustment by the operator. By turning the hand-

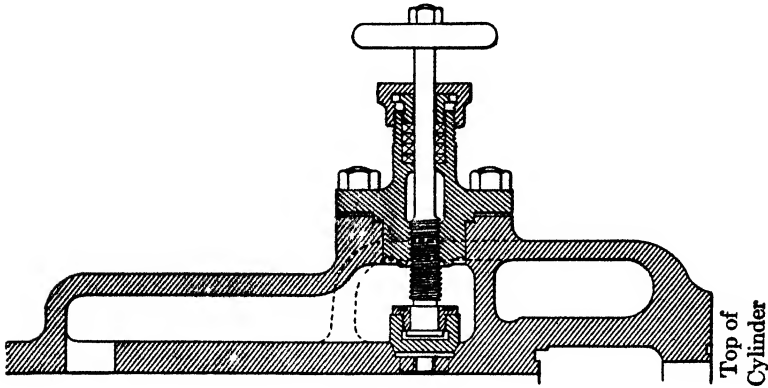


FIG. 16·13.

wheel the operator fixes the position of the clearance piston and thereby controls the clearance volume. With increased clearance the re-expansion of gas from the clearance space reduces the effective stroke

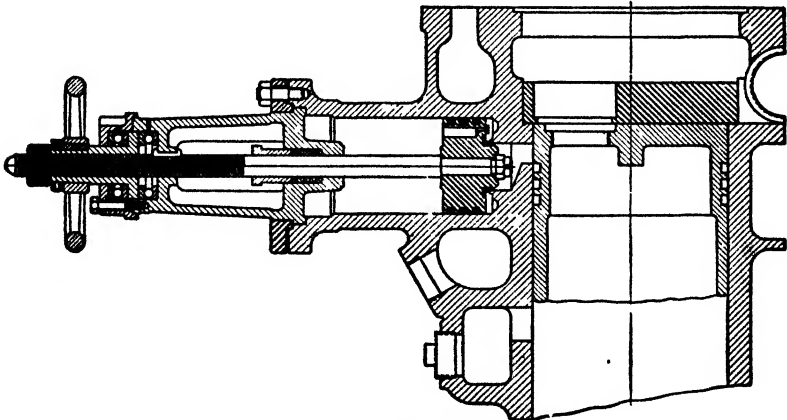


FIG. 16·14. (Courtesy Frick Co.)

and thereby decreases the capacity. The capacity effect of a given amount of clearance will necessarily vary with both suction and discharge pressure since each of these pressures influences the point at which the suction valve opens.

(3) Suction valves can be equipped with “unloaders” operated either manually (Fig. 16·15) or automatically by means of a refrigerant-actuated power diaphragm (Fig. 16·16). Unloaders hold the suction valve open and thereby permit free flow of gas to and from the cylinder without compression and with zero discharge. By installing an unloader on each cylinder of a multicylinder compressor the capacity can be reduced by various fixed percentages.

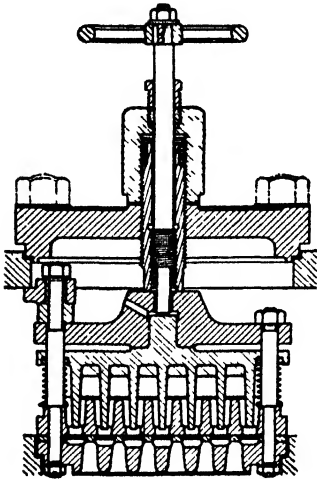


FIG. 16·15.

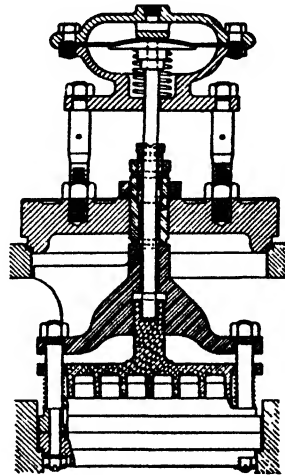


FIG. 16·16.

(Courtesy Worthington Pump and Machinery Co.)

16·9. Lubrication. Lubrication of the old slow-speed, horizontal double-acting or the vertical single-acting compressor of the open-frame type with speeds of approximately 60 to 75 rpm was a simple matter and consisted, as far as the compressor was concerned, of an occasional “shot” of lubricating oil to the rod packing and the suction pipe. With the increase in the rotative speeds and the introduction of the enclosed compressor a decided change of design was made which involved a copious supply of oil and a positive feed at all times.

Figure 16·5 shows a method of lubrication still followed in some of the smaller-sized compressors. It is the “splash” system, and it has been reasonably successful for medium speeds where the action of the crank and connecting rod causes a spray of oil sufficient to fill the oil-cup openings. However, insufficient oil in the crankcase or too high a speed may result in a lack of oil and the freezing of the wrist pin or excessive wear of the crankshaft and bearings. In one design the wrist pin has a force feed by means of a scoop pump with a ball check valve assembled at the lower end of the connecting rod. Figures 16·6 and 16·10 show

a forced-feed system now used in all high-speed compressors of all makes in sizes above approximately the 5-in. by 5-in., including the oil pump, which may be connected to the end of the crankshaft (Fig. 16·17) or belt-driven from the shaft or both. The oil is filtered and distributed by pipings to the bearings and then by drilled holes through the crankshaft and connecting rods to the crank and crosshead or wrist pins.

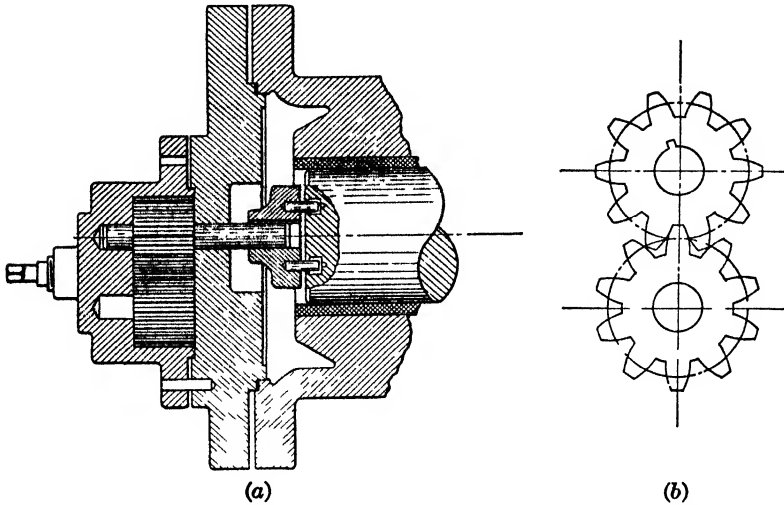


FIG. 16-17. The gear type of oil pump, direct-connected to the compressor shaft.

A pressure gage indicates the oil pressure. Forced-feed lubrication has made the refrigerating machine automatic and successful at medium and high speeds.

16·10. Cylinder Cooling. The use of wet compression has been discarded by manufacturers in the United States except when excessive condenser pressures are necessary; thus the water jacket is universally found except in certain types such as the carbon dioxide compressor (Fig. 16·8) which does not have high discharge temperatures. The only portions of the compressor cylinder where effective cooling is possible are at the heads and at the cylinder barrel near the heads. In the vertical single-acting ammonia compressor the jacket is usually confined to the upper third of the barrel and portions of the head where a jacket can be arranged for. The horizontal double-acting ammonia compressor is jacketed in the cylinder barrel and the two heads.

16·11. Valve Manifold. The vertical single-acting compressor, except in the sizes above 10 in. by 10 in., is usually supplied with valve manifolds made of a single casting (Fig. 16·18). The manifold is usually arranged for a full-sized by-pass for starting, suction, and dis-

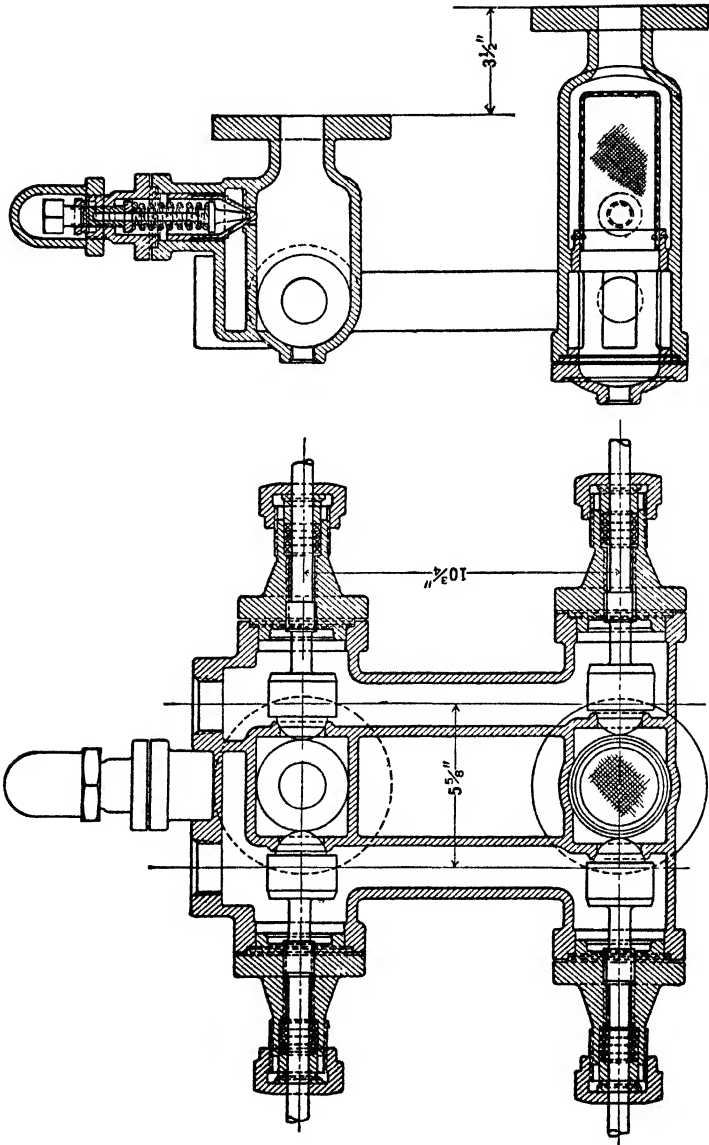


Fig. 16-18. The valve manifold.

charge stop valves and cross connections for pumping out the condenser system to discharge it into the evaporating coils. Usually it is convenient to include also a pop safety valve so erected as to discharge into the suction line, and a suction trap with a suitable suction screen designed for ready cleaning. Under any condition the suction trap should be placed in the suction line as near the compressor as possible. In addition a sight feed lubricator connection is made to the suction of the larger machines, and either plugs or valves are placed in the manifold so that the compressor may pump a vacuum or an air pressure on the entire system at the end of erection or on parts of the system after repairs have been made.

16·12. The Piston. The vertical single-acting enclosed type of compressor has a short connecting rod, usually $2\frac{1}{2}$ to 3 times the cylinder diameter, and no crosshead in the real sense of the word. However, the lower part of the so-called trunk or slipper piston is in fact a crosshead and is designed to take the side thrust of the connecting rod. The piston is made amply long in order to keep the suction ports open during the entire suction stroke. In some designs the single balanced-piston type poppet valve has been replaced by three or more smaller balanced valves or by ring plate valves. The lower part of the piston is provided with special oil rings with oil grooves to collect and return to the crankcase the excess oil on the cylinder wall. In some designs lightweight, automobile-type pistons without valves are used. Some modern compressor cylinders are also equipped with removal alloy-steel sleeves, or liners, which reduce maintenance and facilitate repair or replacement.

16·13. Stuffing Box. The stuffing box for the enclosed compressor is subject to nominal pressure and has rotation only, so that scoring of the rod is not so serious as in the horizontal double-acting compressor. The packing is usually metallic or semi-metallic and is fitted with a modified lantern for connection to the lubricator. The horizontal double-acting compressor has much more of a problem, as the stuffing box has to withstand the discharge pressure. In consequence the stuffing box is much longer than in the enclosed compressor and includes metallic or semi-metallic packing, a lantern connected to the suction line, and an outer fiber packing. The stuffing box is usually surrounded by the water jacket in the cylinder head.

16·14. Rotary Compression. The piston displacement per minute per unit of refrigeration depends principally on the refrigerant and the suction pressure. With ammonia, as soon as the suction pressure drops to 1 atmosphere or lower, the piston displacement becomes excessive. For example, at 25 psia evaporating and 170 lb condenser pressure the piston displacement per ton of refrigeration per minute is 5.8 cu ft

approximately, at 20 lb it is 7.3 cu ft, at 15 lb it is 9.9 cu ft, and at 10 lb it is 15.7 cu ft. With a reciprocating compressor the necessary displacement becomes a big factor, especially as the number of revolutions per minute is kept low except in the smaller compressors. The result is a desire on the part of the designers to secure the same results by means of a rotary compressor, which may be made in four different types, the centrifugal, the eccentric, the gear and the blade types.

16·15. The Blade-Type Compressor. The first successful rotary compressor was of the type that used a rotor eccentrically located in the stator with one or more blades that pressed against the sides of the casing because of the centrifugal force developed. Such a design always included a discharge valve, but a suction valve is not particularly needed. The action is positive, and a large displacement is easily obtained. However, two difficulties arise. As the blade is difficult to lubricate, wear of the blade and casing becomes excessive and the side plates can be kept tight only with difficulty. Unless some means can be found for good lubrication of the blade there does not appear to be any chance for success with this form of design.

16·16. The Gear-Type Compressor. The gear type of pump or compressor has been used for a considerable time. It was an early form of water pump, and as an air blower for pressures under 2 atmospheres it has had a wide application; in refrigeration it has been applied to the compression of sulphur dioxide and the compression of methyl chloride. In the latter, the compressor is limited to two gears, using suction connections at the bottom, with direct connection to electric motor drive at about 1150 rpm. Glycerin is used for lubrication and as a seal at the end plates and in the gears. In order to prevent back flow after operation stops there are only check valves in the suction and the discharge lines. The machine appears to be reasonably noiseless but seems to take an excessive amount of power. Tests* performed on four 120-ton duplex rotary compressors direct-connected to synchronous motors at about 1000 rpm required 2.57 hp of the compressor per ton of refrigeration. The suction pressure was 23.2, and the discharge pressure 94.9 psig.

16·17. The Pendulum-Type Compressor. The pendulum design of compressor, having a small clearance with the casing and a tongue sliding in a swivel, has been brought out in the United States and Germany. The principle is old, and in the improved designs it appears to be satisfactory under test. However, with the small clearances required, the oil and gas filters must be very efficient or a lapping process will break down the oil seal in the clearance space. A satisfactory discharge

* Doolittle, *Power*, Sept. 8, 1931.

valve is necessary, and the side plates are likely to cause trouble unless they are tight.

One form of eccentric rotary (Fig. 16·19) appears to have considerable chance for success. This type of rotary has been applied extensively

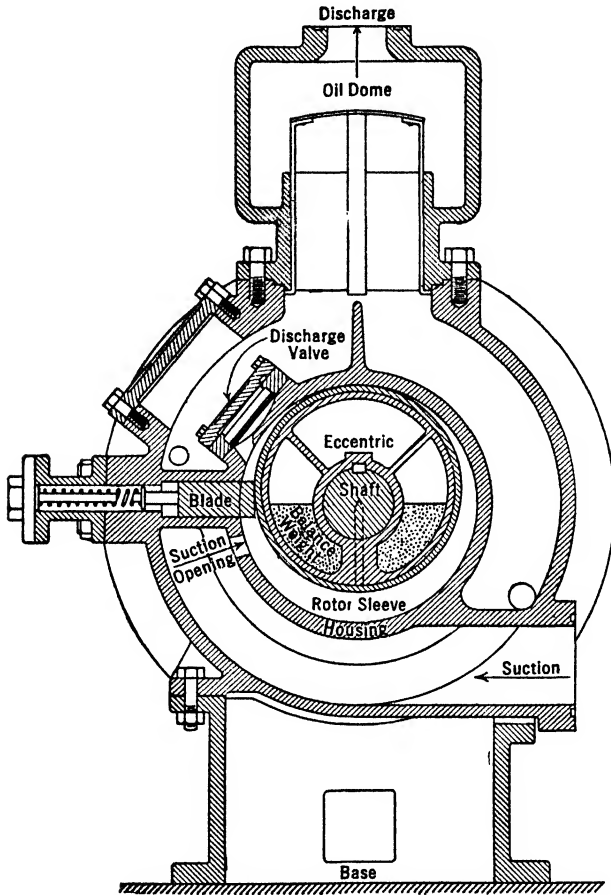


FIG. 16·19. Cross section of a rotary compressor with a sleeve rotor.

to household service (Fig. 17·11). The rotor sleeve (Fig. 16·19) of semi-steel is fitted over a cast iron eccentric, and the two surfaces are copiously supplied with oil. The rotor sleeve rolls on the eccentric and revolves slightly so that the wear will be uniform.

16·18. The Centrifugal Compressor. The basic elements of centrifugal compressors are the impeller and the diffuser. The impeller is the rotating element, which develops some static pressure as a result of

centrifugal force† but serves primarily to impart kinetic energy to the refrigerant vapor and thereby increase its velocity pressure. The ratio of static to velocity pressure imparted by the impeller varies with a number of factors but is usually greater for backward-curved impeller blades than for radial blades. Impellers are cast, machined from forgings, or built up by either welding or riveting.

The function of the stationary diffuser is to transform the high velocity pressure of the gas as it leaves the impeller to static pressure. This is done by using a volute, vortex, or vaned diffuser which achieves gradual reduction in velocity with regain of static pressure; in the ideal case the total pressure would not change during passage through the diffuser, but the kinetic fraction would be transformed to flow work and evidence itself as an increase in static pressure. The detailed analysis of pressure characteristics of a centrifugal compressor is similar to that for a centrifugal fan.

Centrifugal compressors are of more recent design than reciprocating units, one of the first installations in the United States being a unit designed by W. H. Carrier and installed in 1922. Figure 16·20 shows an early type of centrifugal compressor, and Fig. 16·21 gives test results on such a machine. Modern centrifugal compressors are manu-

† In a centrifugal compressor if we consider a thin layer of thickness dx between two vanes of the impeller at a distance x and an area F the weight of the fluid is $F dx\rho$. The centrifugal force at a speed of w radians per second will be

$$\frac{F dx\rho}{g} w^2 x \text{ lb}$$

if the dimensions are in feet, and the unit pressure is

$$\frac{dx\rho}{g} w^2 x$$

in pounds per square foot.

The head, in feet of the fluid flowing, will then be

$$\frac{dx\rho}{g} w^2 x \div \rho = \frac{dxw^2x}{g}$$

and the total head, in feet of the fluid flowing, will be

$$\frac{w^2}{g} \int_{x=r_1}^{x=r_2} x dx = \frac{w^2}{g} \left(\frac{r_2^2 - r_1^2}{2} \right)$$

If the entire column from the axis of rotation to the top of the blades can be utilized for developing centrifugal force the equation becomes

$$h = \frac{w^2 r_2^2}{2g}$$

and as the tip velocity v is equal to $w r_2$ the centrifugal force is $v^2/2g$.

factured in the United States by a number of companies and can be obtained in single units (though multistage) in sizes up to more than 2600 tons and for evaporation temperatures as low as -100°F . Since centrifugal units do not have positive displacement they can be easily controlled by throttling either the suction or discharge, thereby permitting simple variation in capacity at fixed speed; this characteristic

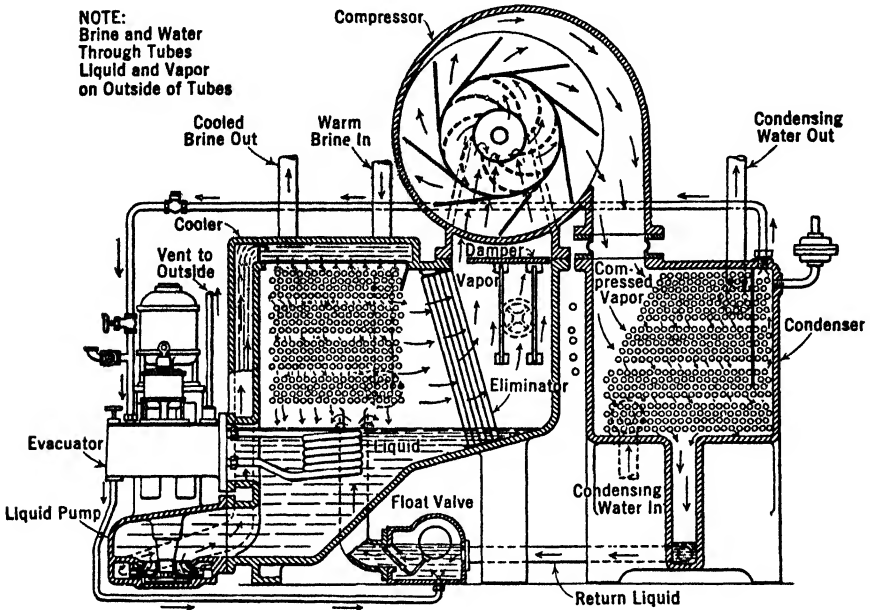


FIG. 16·20. Typical diagram of Carrier centrifugal refrigeration, using methylene chloride (design of 1924).

represents a marked advance over capacity regulation of reciprocating units (see Section 16·8) since the latter, at best, permit change of capacity by increments rather than by continuous modulation. A centrifugal compressor also permits delivery of a variable displacement at constant speed and with an approximately constant ratio of compression.

Figure 16·22 shows diagrammatically a typical flow diagram for multistage operation as recommended by one manufacturer. Liquid from the condenser discharges through multiple expansion valves which permit the return of flash vapor to the third stage of the centrifugal compressor.

16·19. Purging Centrifugal Compressors. Purging equipment is provided with centrifugal compressors to permit extraction of the non-condensable gases and subsequent discharge from the system. Since

the presence of such gases in the condenser would necessarily raise the condenser pressure above the saturation value for the operating temperature it follows that purging is essential to economy of operation. Figure 16·23 shows one arrangement of purging equipment.

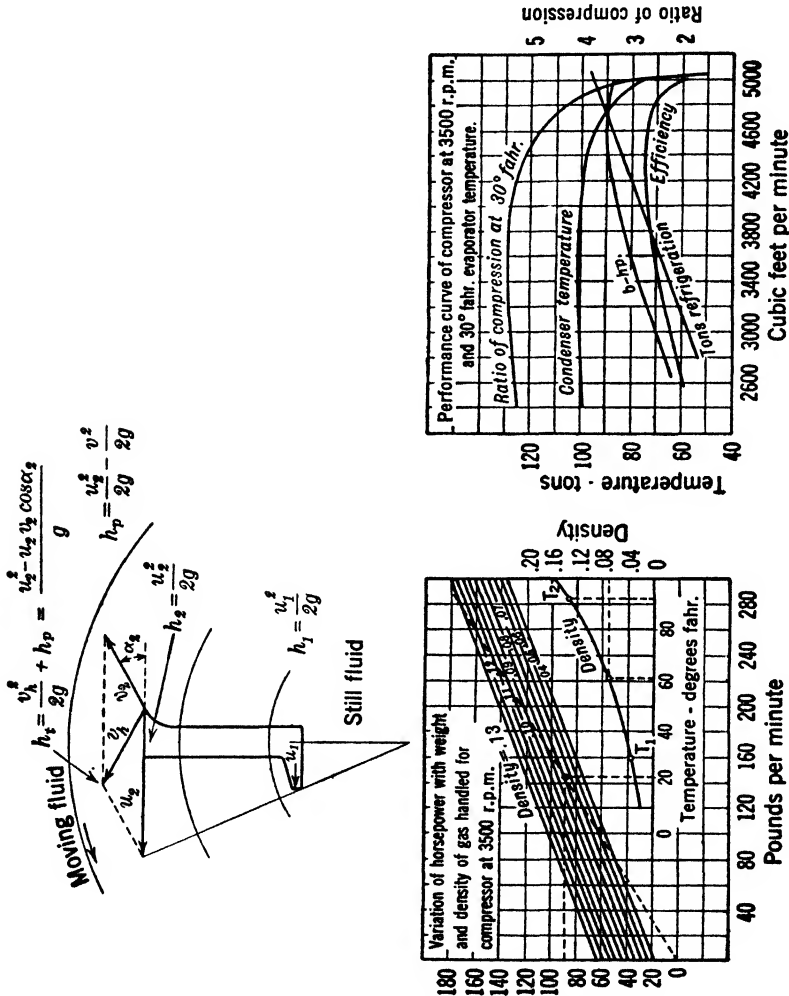


Fig. 16-21. Test results on a compressor of the type shown in Fig. 16-20.

Non-condensable gases, together with some refrigerant vapor and some water vapor, are withdrawn from the top section of the condenser, and they flow into the suction of a 1/4-hp motor-driven compressor. To protect the compressor against condensation of vapor within it during the "off" period a solenoid-actuated valve is located in the suction and is closed when the purge system is not operating. The vapor and gas

discharged from the purge compressor flows successively through an oil separator, an air-cooled condenser, a settling chamber, and a refrigerant-cooled, strip condenser, resulting in separation of the refrigerant from the oil, air, and water.

16-20. Steam Jet Refrigeration. The only really safe refrigerants are water and air, and it is just possible that the desire for safety may force the use of these less efficient refrigerants for comfort cooling. There is a strong possibility that steam jet refrigeration may find a field of operation for itself in special applications, as for instance in comfort

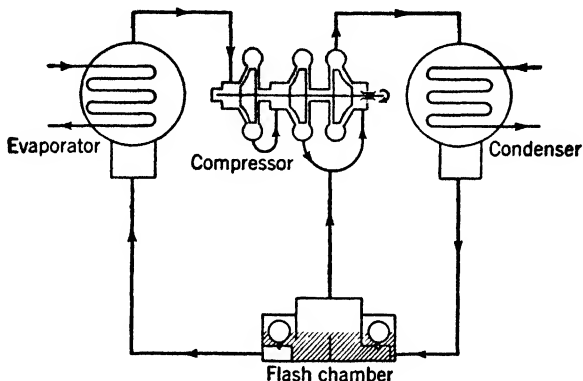


FIG. 16-22. (Courtesy Worthington Pump and Machinery Co.)

cooling the coaches of passenger railroad trains and in industries that require cool water for process work and have a plentiful supply of medium-pressure steam. An example of steam jet refrigeration is to be found in a 180-ton installation, with an ultimate capacity of 250 tons, using a forced-draft basement cooling tower for a 40-story New York office building. This building is provided with steam from a district heating plant, and the particular specification is the elimination of vibration. A low rate for steam in the summer made the installation of steam jet refrigeration the cheapest one in operating cost. In such systems steam ejector nozzles provide the compression necessary because of the extremely low vacuum, incidental with the large specific volumes, required at the temperatures of 40°, 45°, and 50° usually specified for comfort cooling and similar applications of refrigeration. Steam jet refrigeration for air conditioning has a particular value inasmuch as it can utilize high-temperature cooling and can adjust quickly to variations in the load.

The steam ejector refrigerating system, of the Maurice Leblanc and Sir Charles Parsons type, was first used successfully during the first

decade of the century. Steam was made to acquire a high velocity by being passed through the usual steam nozzle and then directed through ejector cones which were connected to the "evaporator" maintained at a pressure of about 1 in. of mercury. The air was removed by means

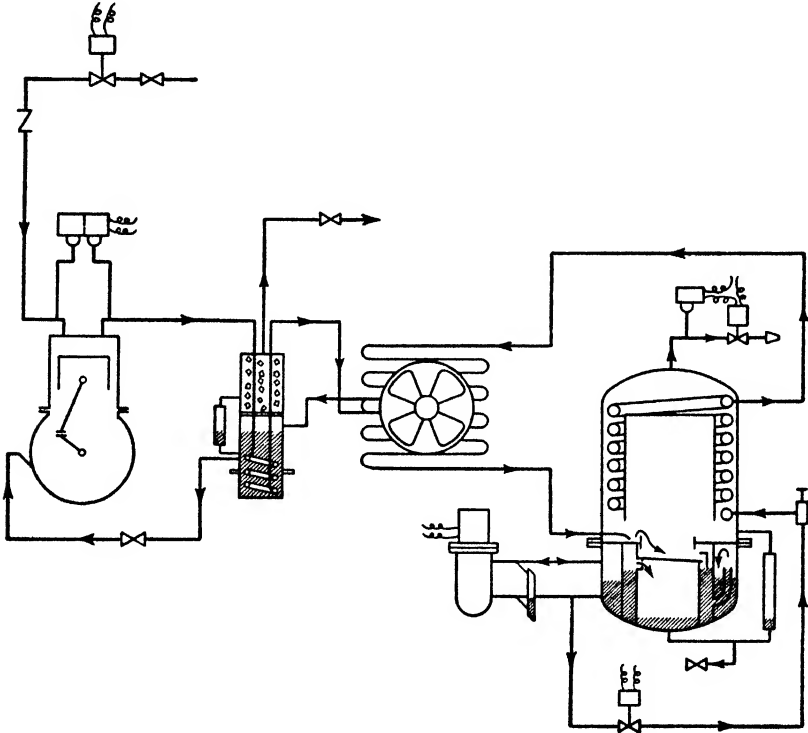


FIG. 16·23. (Courtesy Worthington Pump and Machinery Co.)

of the Leblanc air pump, and the steam was condensed by means of the low-head jet condenser. The steam required as well as the power used by the pumps was excessive.

In the later designs[‡] the tendency appears to be to use two or more sets of nozzles as shown in Fig. 16·24. A large number of variations of steam pressures, evaporating pressures, and ratios of compression, as well as the number of nozzles, are possible. Incomplete tests would seem to indicate that from 30 to 40 lb of steam at 100 psig pressure are required per hour per ton of refrigeration. To this must be added the power required to operate the condensate and the condenser water pump. The advantages are safety and the relative simplicity of the

[‡] Everetts, *Refrig. Eng.*, December, 1932; Dean, *ibid.*, August, 1932.

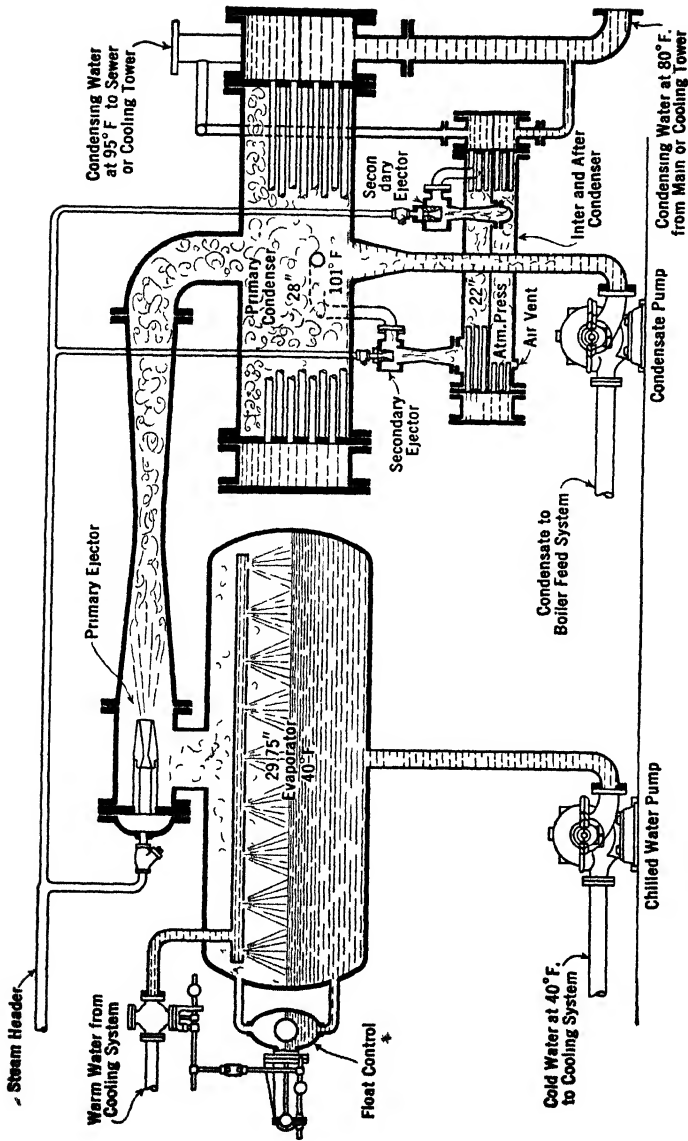


Fig. 16-24. Flow diagram for a steam jet refrigerating unit.

design as well as lack of vibration. A liberal supply of inexpensive steam as well as cheap water for condensing purposes is necessary. The machine is compact and occupies less space than the compressor type, especially as the "evaporator" is only a small spray chamber. In first cost it is the lowest of all types for comfort cooling temperatures, but it is not adapted to refrigeration needing temperatures much below 40° F.

It will be seen in the figure that the primary condenser operates with a condensate of about 100° F and a vacuum of 28 in. of mercury, under a

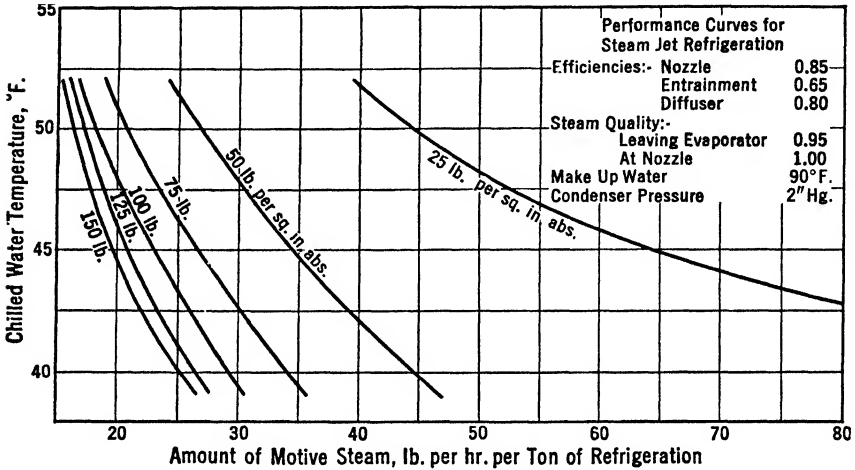


FIG. 16·25. Calculated weight of steam required in steam jet refrigeration.

barometer of 29.92 in., at which conditions the ejector pump has a pumping head of 1¾ in. of mercury and the condensing water leaves at about 95°. This will require approximately 3 gal of 70° water per ton of refrigeration per minute. A little calculation will show that apparently, if a 28½ in. of mercury vacuum is maintained with a liquefaction temperature of 92°, nearly 50 per cent more water would be required, but as a matter of fact as the steam consumption of the nozzles is so much less the amount of water needed is practically the same. The secondary and atmospheric condensers are for the removal of air, and they use approximately 10 per cent of the total amount of steam required for producing refrigerating effect. The use of two or more surface condensers rather than one jet condenser, with its much increased amount of air to be removed, is the principal change over the earlier designs. The amount of steam required per ton of refrigeration per hour at various operating conditions is shown in Fig. 16·25, and a typical calculation is indicated in Section 6·6. Figure 16·26 shows a typical barometric-type steam jet unit in contrast with the surface-type unit of Fig. 16·25.

16.21. Stage Compression. The standard conditions of operation, according to the American Society of Refrigerating Engineers code, are

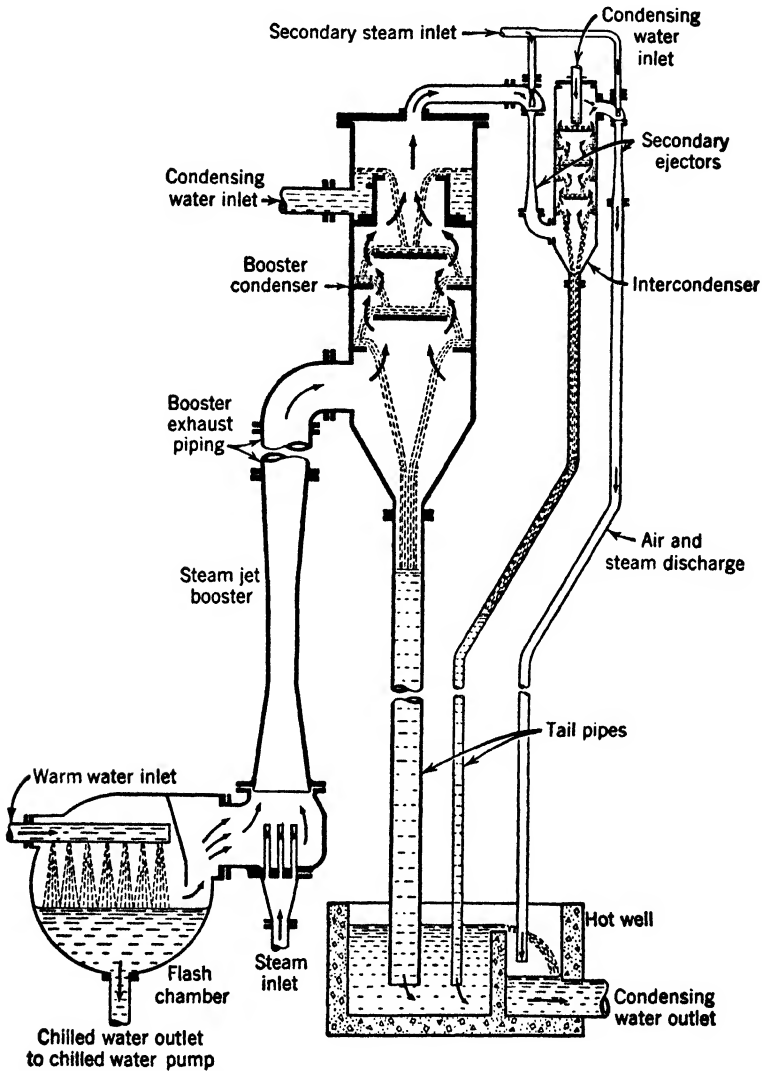


FIG. 16.26. (Courtesy Worthington Pump and Machinery Co.)

5° F evaporation and 86° F liquefaction temperature. However, there is no limitation in practice. The operating suction pressure must be low enough so that the required temperature of evaporation may be obtained,

whereas the temperature of liquefaction fluctuates with the temperature of the atmosphere and with the amount of water that can be used on the condenser; therefore liquefaction may vary from 60° to 100° F. As a comparison the usual air compressor operation is much more uniform in

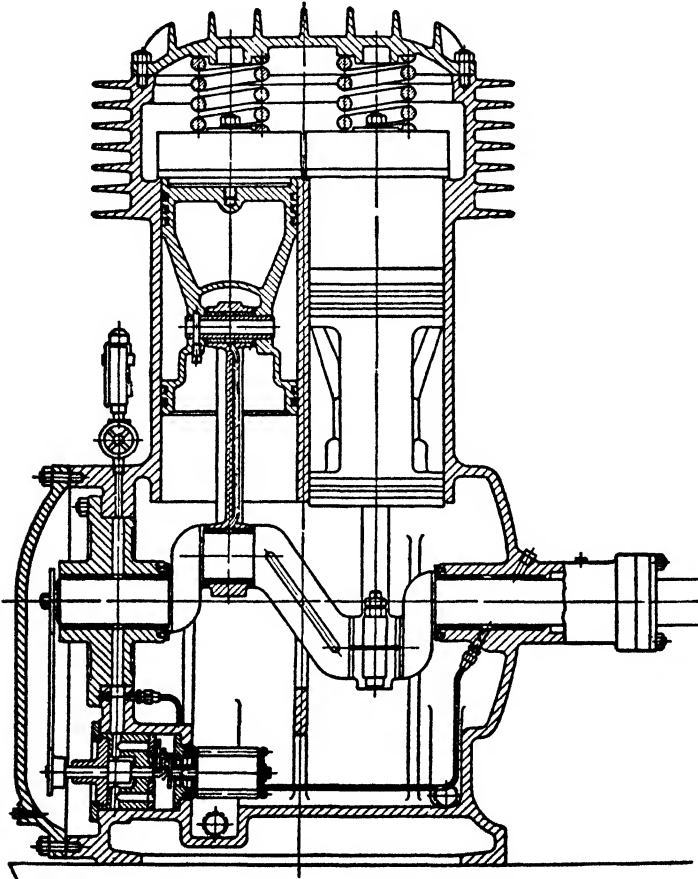


FIG. 16-27. (Courtesy Frick Co.)

its suction and discharge pressure although the suction pressure decreases as the elevation above sea level increases. However, it seems agreed that, for refrigeration at temperatures of about -20° F or lower, stage compression in one form or another is an advantage if not a necessity.

The advantages of stage compression are decreased temperature of discharge, higher volumetric efficiency, decreased power input per unit of refrigeration, and greater refrigerating capacity per cubic foot of piston displacement.

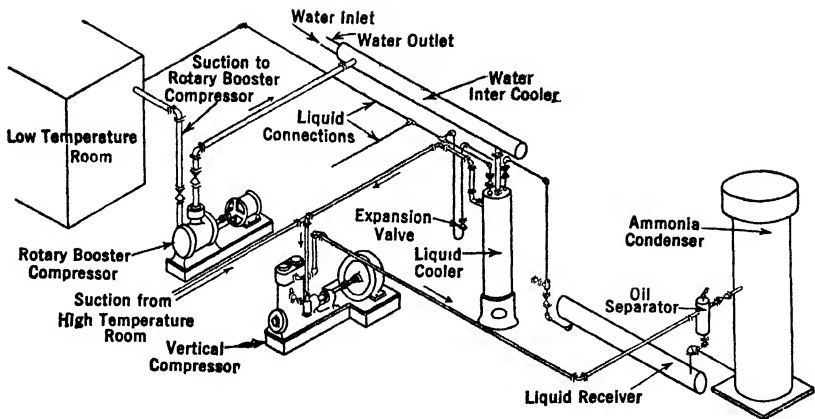


FIG. 16-28. Isometric piping diagram for a two-stage ammonia compression plant.

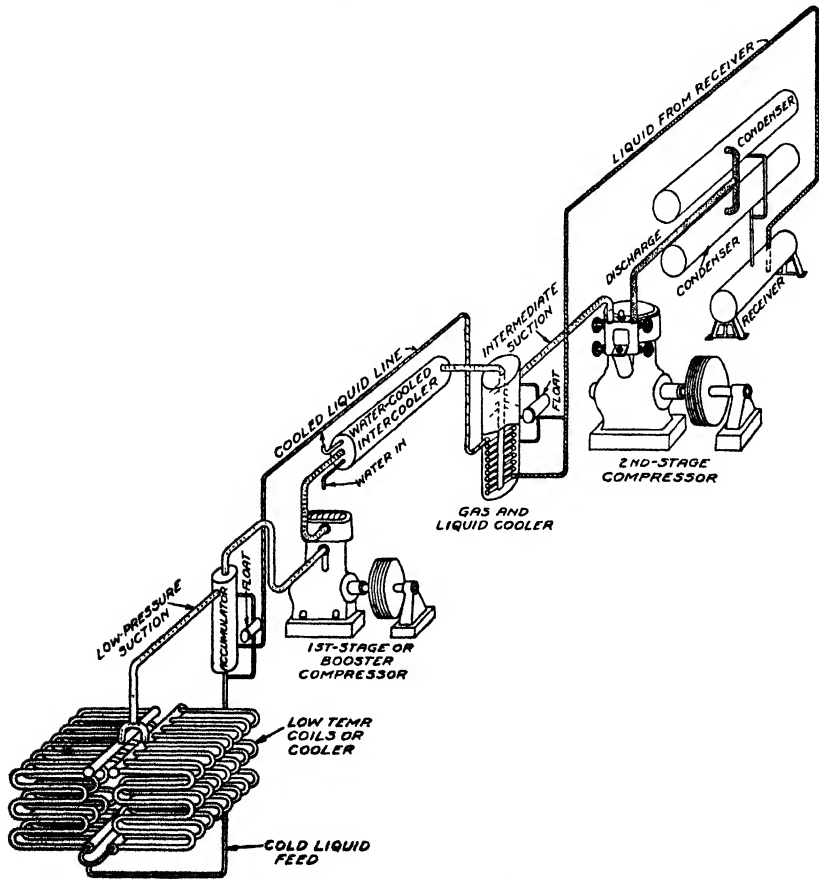


FIG. 16-29. (Courtesy Frick Co.)

In stage compression it is desirable to make the work of compression the same in the two cylinders if they are on the same shaft. Equalization of the load is not always possible, because of the need at times of fixing the intermediate pressure and because of the difficulty in maintain-

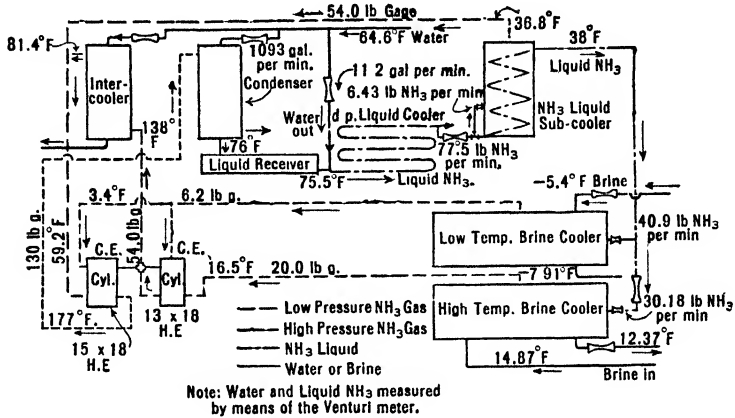


FIG. 16-30. Test results for a two-stage compression plant.

ing constant the suction pressure of the low-pressure cylinder under varying loads without elaborate capacity adjustments. In addition the condenser pressure varies with the load and the temperature of the water used and so cannot be kept constant in practice.

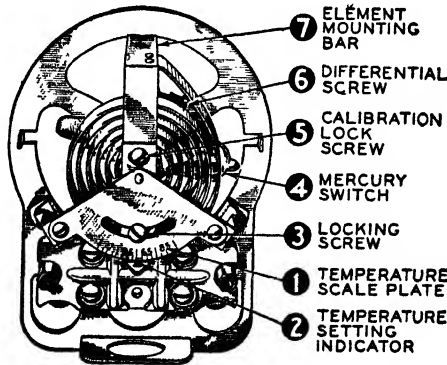


FIG. 16-31. (Courtesy Minneapolis Honeywell Regulator Co.)

At the present time the booster compressor (Fig. 16-27) makes a desirable low-pressure compressor because of the comparative ease with which the large volumes are handled. The standard reciprocating com-

pressor and the booster, or two compressors with suitable difference in diameters and a booster, have been found to be successful arrangements. Stage compression is used in the refining of lubricating oils, the quick freezing of meats, ice cream manufacture, and a number of special processes connected more or less with chemical engineering, like the manufacture of photographic films and plates. Figures 16·28 and 16·29 show typical plant layouts, Table 16·1 indicates results of tests on a two-stage refrigerating machine, and Fig. 16·30 is a flow diagram giving an idea of the piping arrangements for the test.

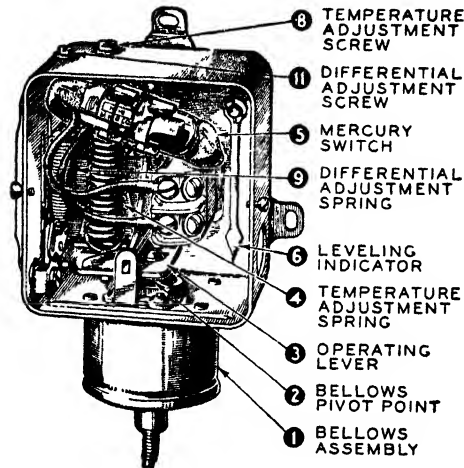


Fig. 16·32. (Courtesy Minneapolis Honeywell Regulator Co.)

16·22. Controls. Many types of control methods are in use with refrigeration systems. The following representative controls are presented merely to indicate a few of the typical functions and to show the equipments used in carrying them out.

(1) For many types of commercial or industrial applications a heavy-duty thermostat is needed which will operate dependably in moist, dusty, or corrosive atmospheres. Figure 16·31 shows a cut-away view of such a unit. The non-corrosive bi-metal actuating element transmits its motion to a mercury switch which can either control the refrigeration valve or motor directly or (for heavy loads or polyphase motors) through a magnetic switch.

(2) Figure 16·32 shows the element of a heavy-duty temperature controller which cycles refrigeration equipment in response to a temperature change "sensed" by a remote bulb. The mercury switch is

TABLE 16·1

TEST¹ ON A 13 AND 15 BY 18 YORK DUPLEX

Two-stage, double-inlet, horizontal, double-acting ammonia compressor at 200 rpm direct connected to a 350-hp synchronous motor.

Suction to low-pressure cylinder	6.2 lb per in. ³ abs
Discharge from low-pressure cylinder	68.7 lb per in. ³ abs
Discharge from high-pressure cylinder . . .	144.7 lb per in. ³ abs

1st stage clearance 4.42 per cent

2d stage clearance 5.66 per cent

(The machine operated with intermediate-pressure gas cooling with water, and liquid sub-cooling using water and then ammonia.)

		Actual i-hp	Theoretical i-hp	Efficiency
1st stage	13-in. cylinder, C.E.	31.83	66.90	87.7
	15-in. cylinder, C.E.	45.21		
2d stage	13-in. cylinder, H.E.	36.70	28.04	76.3
	15-in. cylinder, H.E.	106.79	79.60	75.3
	Total i-hp =	220.53		

(All fluid measurements were made by means of the Venturi meter.)

Heat Balance, Btu per hour

Heat absorbed		Heat rejected	
In evaporator	2,270,500	In intermediate vapor cooler	67,680
In compressors, heat equivalent of work	561,600	High-pressure discharge pipe, by radiation and convection	115,500
		In ammonia condensers . .	2,688,000
		In liquid cooler	30,480
	2,832,100		2,901,660

The coefficient of performance is $\frac{2,270,500}{561,600} = 4.05$.

¹G. A. Horne and A. W. Oakley, *Refrig. Eng.*, June, 1930.

actuated by a vapor-filled bellows which expands as the temperature at the location of the remote bulb increases.

(3) A combination remote-bulb temperature controller and high-pressure cut-out is shown in Fig. 16-33. The remote bulb operates as

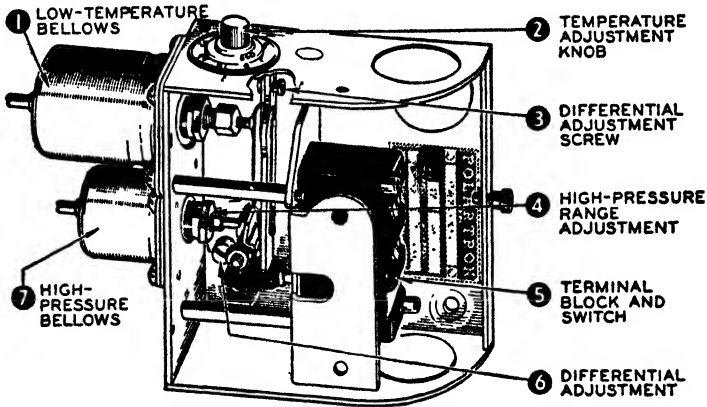


Fig. 16-33. (Courtesy Minneapolis Honeywell Regulator Co.)

in (2), whereas the high-pressure control protects the system against high condenser pressure such as would occur in the event of failure or interruption in the flow of condenser cooling water.

PROBLEMS

1. An ammonia compressor receives saturated vapor at 10°F and discharges against a condenser pressure of 170 psia; minimum clearance is 2 per cent. Determine the required value of maximum clearance if clearance pockets are to be provided which will permit load variation down to 70 per cent of the maximum load.

2. If the discharge pressure of the compressor in Problem 1 were to be reduced to 150 psia, what per cent change would occur in maximum load?

3. For a discharge pressure of 150 psia and saturated suction temperature of 6°F , what maximum clearance would be required to permit operation at 70 per cent of the load corresponding to 2 per cent clearance?

4. Compare and analyze the results from the three above problems and deduce generalizations concerning the influence of minimum clearance and of discharge pressure on the effective operating range of a given set of clearance pockets.

CHAPTER XVII

AUTOMATIC REFRIGERATING MACHINES; HOUSEHOLD MACHINES

The automatic household and commercial refrigerating machines have been made possible by the perfection of certain automatic devices, principally for regulation, which—in some form—all new refrigerating plants are now using. The time-honored, hand-operated, pressure-reducing or throttle valve has to a large extent been replaced by float valves in ice-manufacturing plants or in other installations where the piping is suitable for the flooded system of operation as with brine coolers. Automatic shut-off valves and other safety devices are becoming more or less common. In general, the completely automatic refrigerating unit requires regulation of the flow of the liquid into the evaporating portion of the circuit, a thermostatic control either on the liquid stop valve or on the relay (Fig. 17·1*b*) that starts and stops the compressor, a control on the water supplied to the condenser (determined by the condenser pressure or the temperature of the water leaving the condenser), and certain other controls such as low-voltage release and high-pressure cutout, both of which result in shutting down the compressor.

/17·1. The Expansion and Float Valve. Nearly all automatic pressure-reducing valves, more usually called *expansion valves* (Figs. 17·1*a* and 17·2), have a spring-loaded diaphragm or bellows connected by means of a steel stem to the pin valve. The opening required for the passage of the liquid refrigerant does not have to be large, but just sufficient weight of liquid must pass the valve to absorb the cooling load. The filter placed on the inlet side and adjacent to the valve must be so efficient as to remove not only solid substances of material size but even very fine solid particles, and provision must be made for replacing and cleaning the filter. With the drop of pressure in the valve there is an evaporation of a small fraction of the liquid; the resultant vapor is likely to cut the valve seats and make the valve leak when it is supposed to be closed, and it is especially important that a liquid seal be maintained on the valve at all times. The drop of pressure is accompanied by a drop of temperature which with certain refrigerants like methyl chloride is ordinarily so great as to freeze water present in

the system on the expansion valve, thus materially interfering with the action of this valve. Moreover, if oil passes to the valve, as it may, especially when the filter is imperfect, it is likely to form a viscous mass. This also interferes with good operation. Numerous attempts at

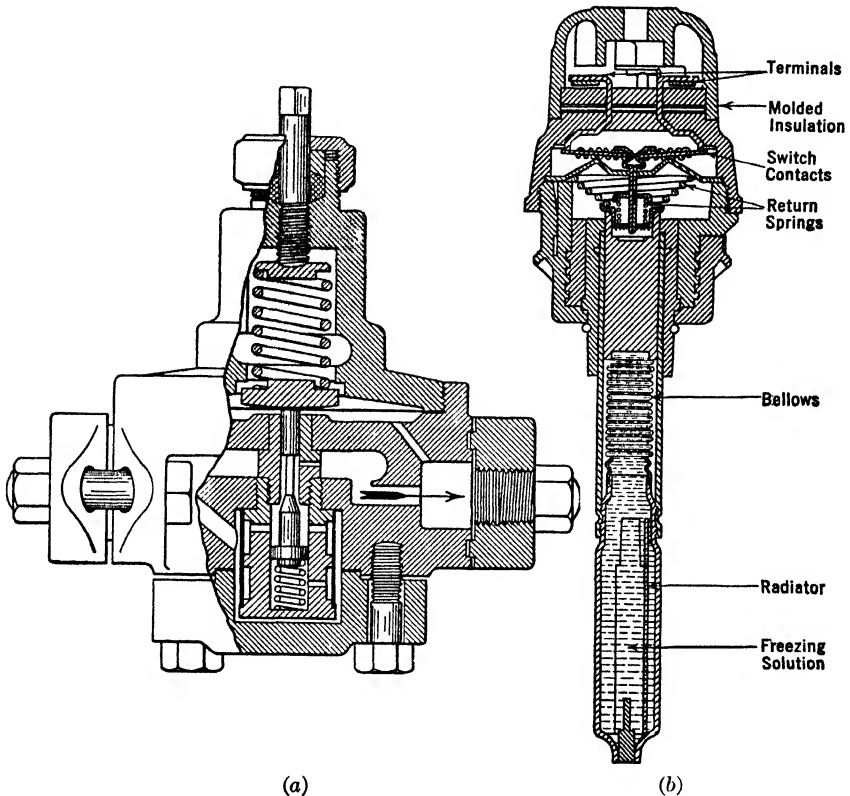


FIG. 17-1. Refrigerant control. (a) The usual design of pressure-reducing valve. (b) Device using a freezing solution.

designs intended to keep the valve and the seat tight and free to act have been made more or less successfully. Figure 17-3 shows a design where the pressure drop takes place gradually until the point *A* is reached, with the idea that the usual wire drawing in the ordinary throttle valve will be eliminated. —

The *thermostatic expansion valve* has an auxiliary feature whereby the spring-actuated diaphragm is replaced by the pressure exerted by a volatile liquid which in turn is varied by the temperature of the suction

vapor returned to the compressor (Fig. 17·3). Evidently, if the refrigerant in the vapor phase passing to the compressor is superheated and if this vapor is in metal contact with the liquid in the bulb of the thermostatic expansion valve the vapor pressure in the bulb will increase with the increase in the amount of superheat. As the fluid pressure increases on the top of the diaphragm the valve will be made to open, resulting in more refrigerant passing the valve and a subsequent reduction in the superheat until the reduced vapor pressure closes the valve.

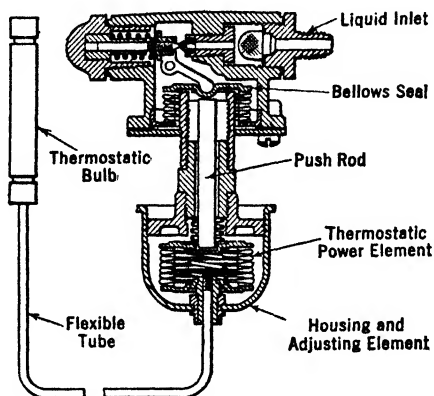


Fig. 17·2. Pressure-reducing valve with thermostatic bulb.

The *float valve* (Fig. 17·4) may be for high-pressure or low-pressure liquid level control. In the high-pressure type the liquid is fed to the evaporating coils at the same rate at which it is evaporated; therefore the coils contain practically the entire charge of the refrigerant. In the low-pressure type of float valve the float is assembled to maintain a particular level in the evaporator. In neither type will the pressure remain constant for, with constant piston displacement, the pressure will vary with the temperature difference and therefore with the weight of the refrigerant boiled per unit of time. On starting up with a heavy demand the compressor will take the maximum load. Then as the temperature difference decreases the boiling action decreases, and the capacity drops with a resulting reduced evaporating pressure and temperature. The high-pressure float valve can feed only one set of coils or piping arranged in series; it cannot be used for multiple systems, nor can several high-pressure float valves be employed in the same system.

17·2. Magnetic Valves. The constant-pressure expansion valve will keep the suction pressure reasonably constant during normal operation of the unit although it cannot be depended on to stop the flow of liquid

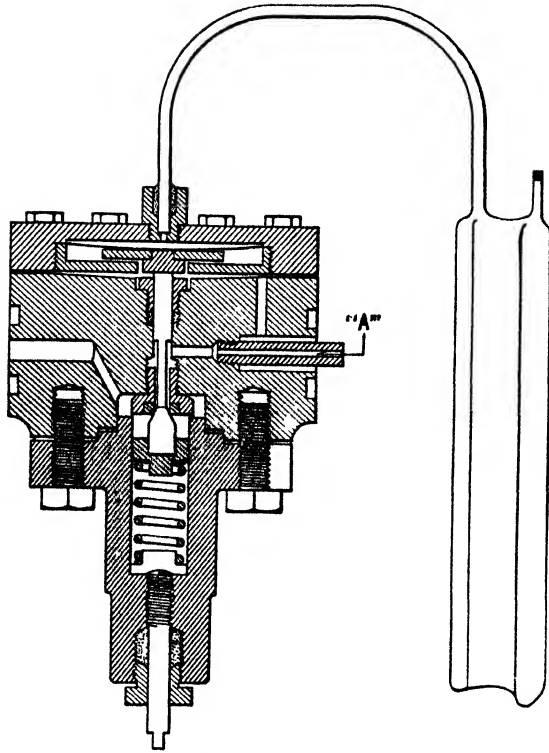


FIG. 17-3. A thermostatic bulb type of pressure-reducing valve.

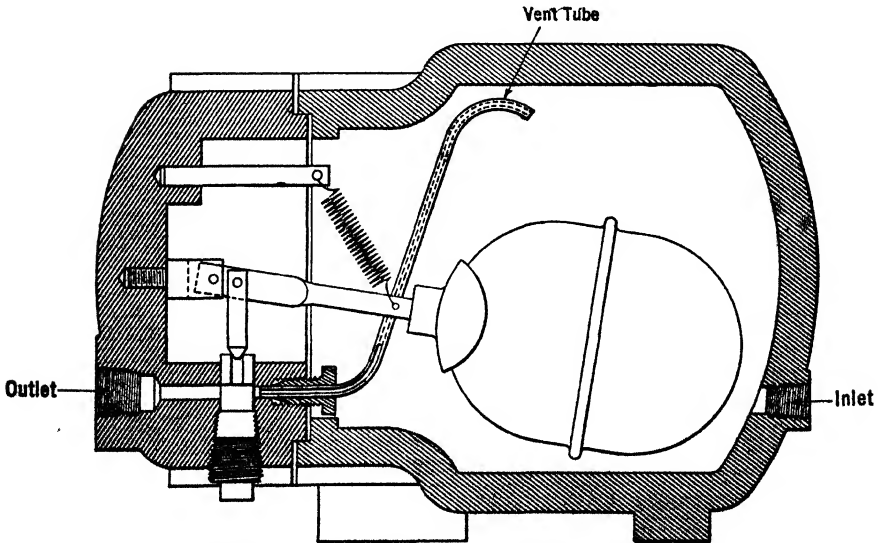


FIG. 17-4. The high-pressure float valve.

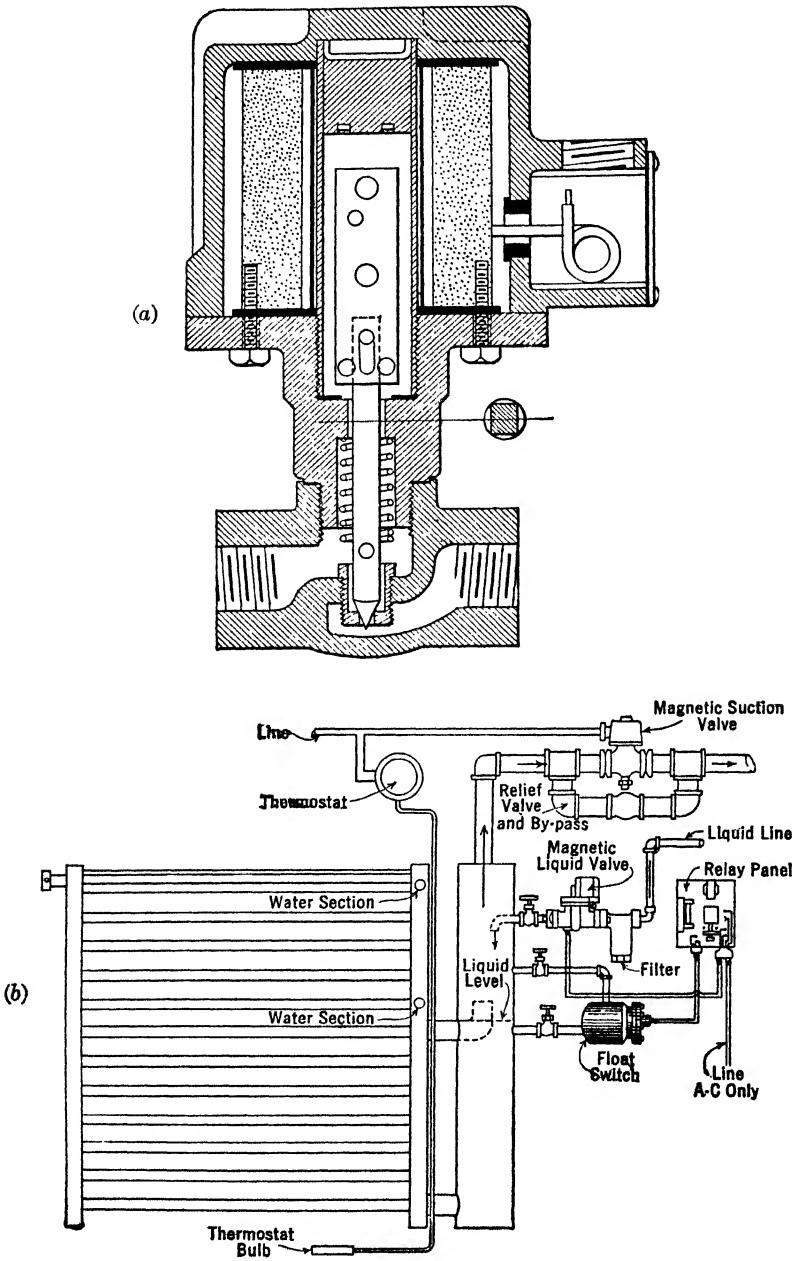


FIG. 17·5. Magnetic valves. (a) Detail of valve. (b) Arrangement of valve in a refrigerating system.

during the shutdown period. The thermostatic expansion valve, however, will provide a tight closure against a flow of liquid if the temperature difference between the medium to be cooled and the boiling refrigerant is at least 15° F. If this difference is not permissible a solenoid stop valve (Fig. 17·5), operated on a thermostat inside the room or

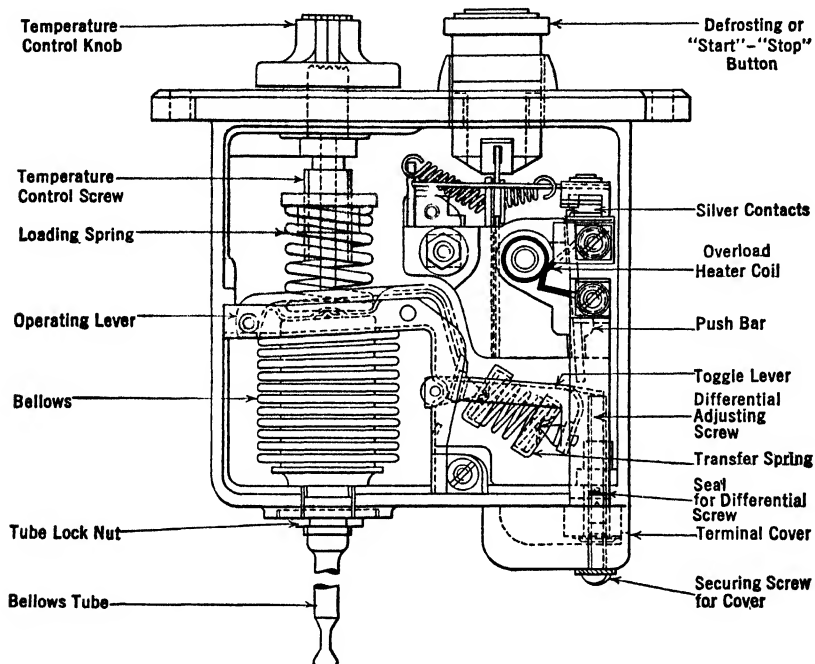


FIG. 17-6. Controls applied to household refrigeration.

exposed to the medium to be cooled, should be placed ahead of the thermostatic expansion valve.

If two room temperatures, for example 0° and 36° F, are to be carried by the same compressor system a *magnetic suction* stop valve, connected to the same thermostat as the magnetic liquid stop valve, is advised for use in the 36° room to prevent continued evaporation after the liquid valve has been closed. Either room can start the compressor, and when both rooms are down to the desired temperature the machine will stop.

17-3. The Thermostatic Switch. The compressor is started and stopped by the expansion and contraction of a metal or vapor in order to make an electric contact direct or through an auxiliary circuit. The bellows type of switch (Fig. 17-6) is one of the most successful of these

devices. The mercury tube is actuated by the pressure developed in a bulb in contact with the fluid to be held at constant temperature. The bulb contains a volatile liquid, frequently sulphur dioxide. The tilting of the mercury tube makes or breaks the power circuit.

Figure 17·7 shows how automatic controls may be assembled so as to include a whole plant.

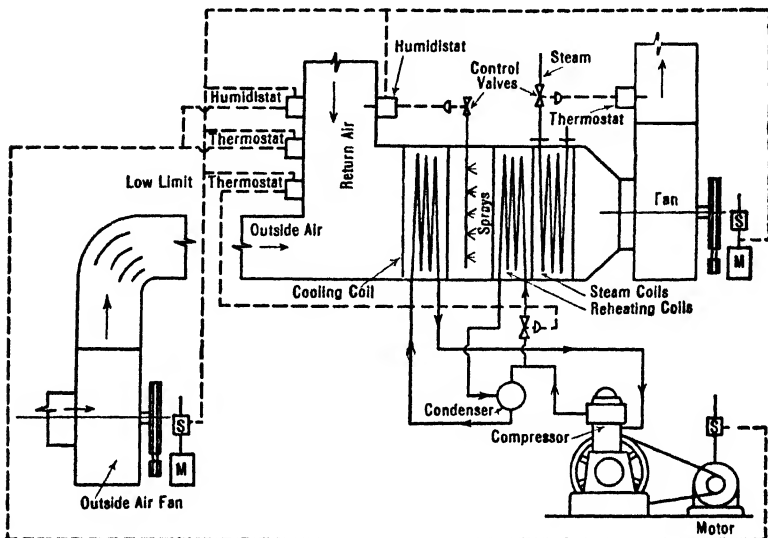


FIG. 17·7. Assemble of automatic controls for a whole plant.

THE FRACTIONAL-TONNAGE REFRIGERATING MACHINE

The perfected, automatic refrigerating machine has made possible the fractional unit which must be completely automatic, nominal in first cost, dependable, and thereby free of maintenance costs or need for expert attendance. It must be simple in construction. This means the use of the air-cooled condenser and a tendency toward the hermetic sealing of motor and compressor. Moreover, non-irritant refrigerants are favored, such as methyl formate, $C_2H_4O_2$, methylene chloride, CH_2Cl_2 , dichlorotetrafluoromethane, CCl_2F_4 , and dichlorodifluoromethane, CCl_2F_2 (and the last has found considerable favor with the manufacturers), even in the small units which have only two or three pounds of refrigerant per unit. As the cabinet or refrigerating box is an integral part of the fractional tonnage the operating cost is always in terms of the machine and box, for example *A* kilowatt-hours per month, the result being that the construction of the refrigerator box is much

better than that of ice refrigerator boxes before the days of the automatic machine. All fractional tonnage, as well as the small machines of only a few tons, are designed for intermittent operation, with the actual running time from a third to a half of the total time. At present there are three types of units: the conventional or open type, those with compressor and motor hermetically sealed, and the absorption-machine types.

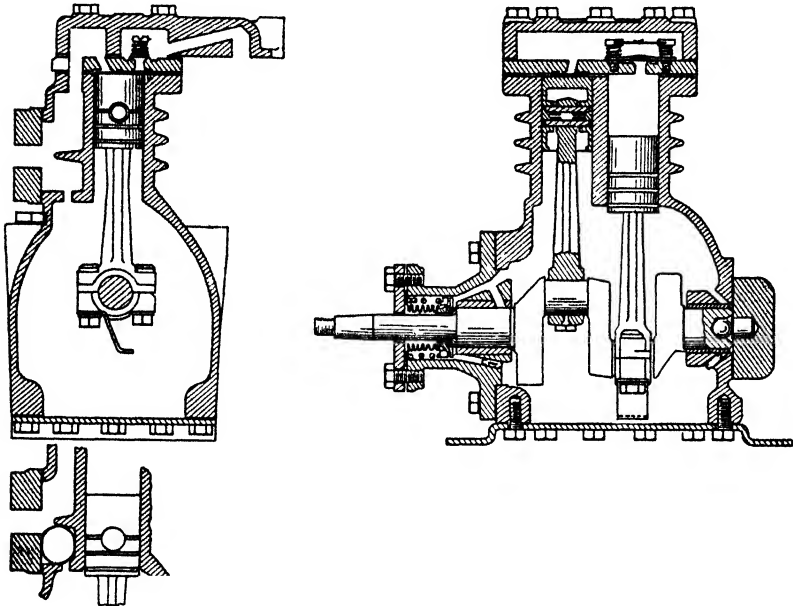


FIG. 17·8. The compressor of an electrically driven unit.

17·4. The Open Compression Type. The open compression type (Fig. 17·8) is similar to the commercial automatic design. The compressor has to be air-cooled, and it is designed with fins cast on the barrel; it is vertical single-acting, frequently taking the suction vapor into the crankcase as in the larger, commercial machines, and under these conditions it has the suction valve in the piston. The compressor operates at speeds from 300 to 600 rpm and is driven from an electric motor by means of a V-belt. Lubrication is usually by means of the splash system. The open compression system has one very serious drawback in that the shaft stuffing box is difficult to keep tight against oil and gas leaks. A loss of either oil or gas results in excessive wear, noisy operation, or loss of capacity, or all three. The design now followed for many household machines is a form of *sylphon bellows* with an outside coil

spring (Fig. 17·9) having a thrust collar which presses against a shoulder on the shaft. The spring may be designed to compensate partly for the internal pressure, thus decreasing the friction and the amount of wear on the shaft. As valves are likely to leak around the valve stem, the packless valve (Fig. 17·10) has decided advantages.

In addition to the open-type reciprocating compressor there is also the eccentric type of rotary compressor shown in Fig. 17·11. This unit,

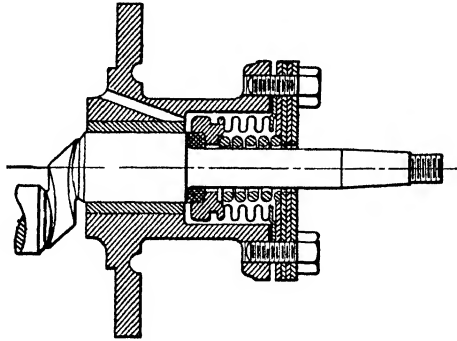


FIG. 17·9. A typical shaft seal.

which is driven by means of a V-belt from the motor, has to use a suitable shaft seal also. A fan on the motor shaft gives positive flow of air for the condenser and the motor.

✓ **17·5. The Hermetically Sealed Compressor.** With the success, in the early days of mechanical refrigeration, of the Audiffern-Singrun compressor which was hermetically sealed in units up to 10-ton capacity it is not surprising that some makes of household machines are designed to be self-contained so that leaks due to shaft seals are not possible. Such a unit (Fig. 17·12) must necessarily be arranged so that the motor can drive the compressor directly, and the motor will have to be exposed to the refrigerant so that the heat due to the motor windings and friction will be carried away by the vapor passing to the condenser. In the figure, which shows a motor of about 1760 rpm, the lubrication is by means of a spiral groove in the vertical shaft, as indicated, so that oil will be supplied copiously to the bearings.

✓ **17·6. The Evaporator.** The evaporator of most household refrigerating machines is simple and compact. Attempts are being made to improve the efficiency of the heat transfer apparatus by entering the liquid so as to give forced circulation, to prevent superheating of the liquid by the use of an ebullator, and to return dry vapor to the compressor by having a drum or header construction at the top of the

evaporator (Figs. 17·11 and 17·12). It is made of copper or stainless steel.

The necessary pressure reduction between the condenser and the evaporator is accomplished by means of throttling either by a float valve (Fig. 17·12) and expansion valve (Fig. 17·1) or by means of an orifice with a constant amount of opening. The latter, in one design called a constrictor, is a triangular opening in the form of a helical groove. It is

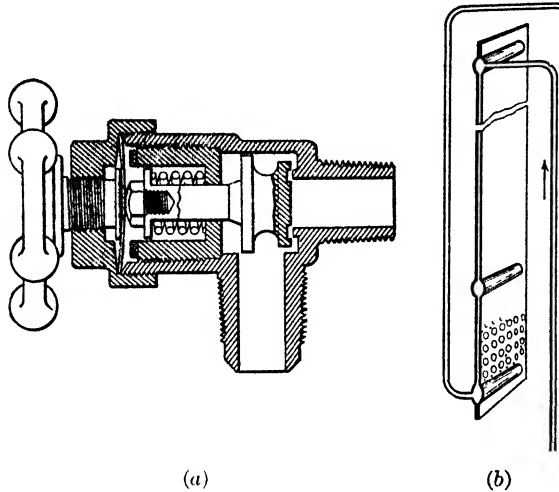


FIG. 17-10. Fractional tonnage details. (a) A packless valve. (b) An air-cooled condenser.

protected by means of a fine mesh, and it automatically equalizes the pressure when the compressor is stopped.

The float valve may be arranged for high-pressure control of the liquid (Fig. 17·12) or for operation with high-pressure control of the flow of the liquid and an intermediate-pressure-regulating valve. This expansion valve simply regulates the pressure of evaporation and has little control of the amount of liquid passing through the valve unless a thermostatic element, as in Fig. 17·2, is attached to the suction return line of the compressor.

17-7. The Condenser. The condenser, air-cooled in nearly all the fractional tonnage units, may be made of the radiator, coil (Figs. 17·11 and 17·13), or plate types (Fig. 17·12). The last is spot-welded sufficiently to prevent distortion under pressure. The air may be circulated by a fan, or it can have natural convection and radiation, but in the latter event, the heat transfer being less than for forced circulation, the

cooling surface provided must be increased. The condenser may be made of steel or copper, depending on its design and the refrigerant chosen.

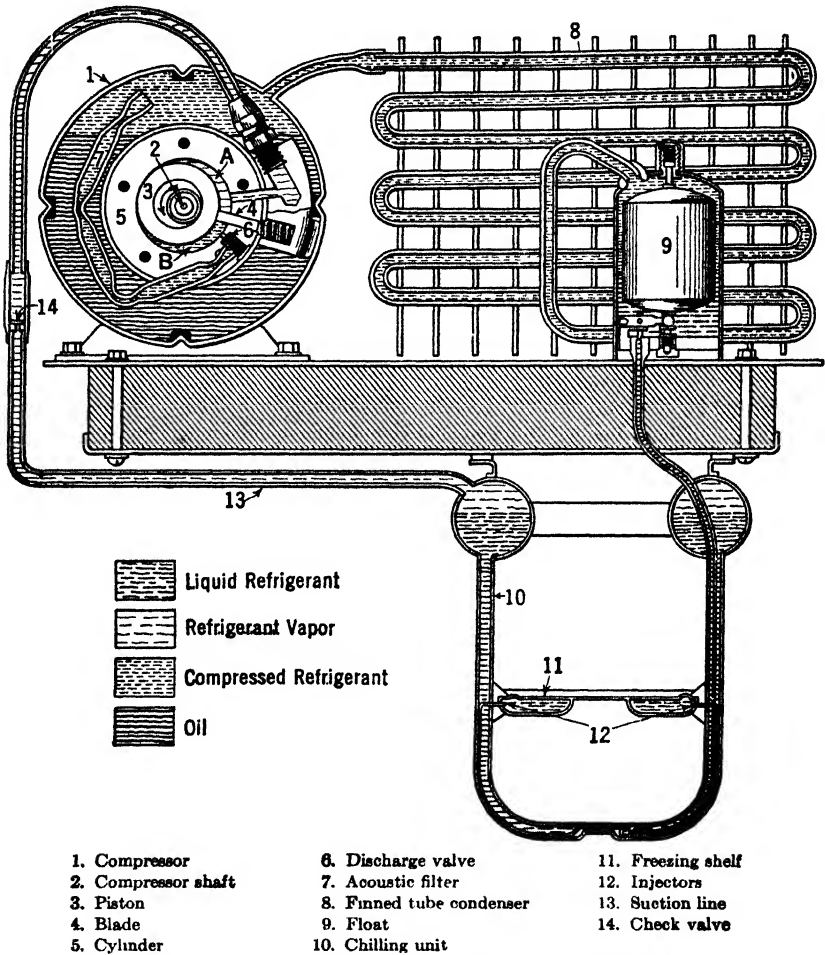


FIG. 17·11. The rotator type of household refrigerating unit.

17·8. Lubrication and Motors. In general a highly refined mineral oil, highly dehydrated, has to be used with a viscosity and pour test best suited to the refrigerant. Methyl chloride needs a viscous oil, as does the rotary compressor, since it depends on the oil for the seal between the rotor and the casing.

Both repulsion-induction and condenser types of motors are possible except in the hermetically sealed designs, which find the commutator

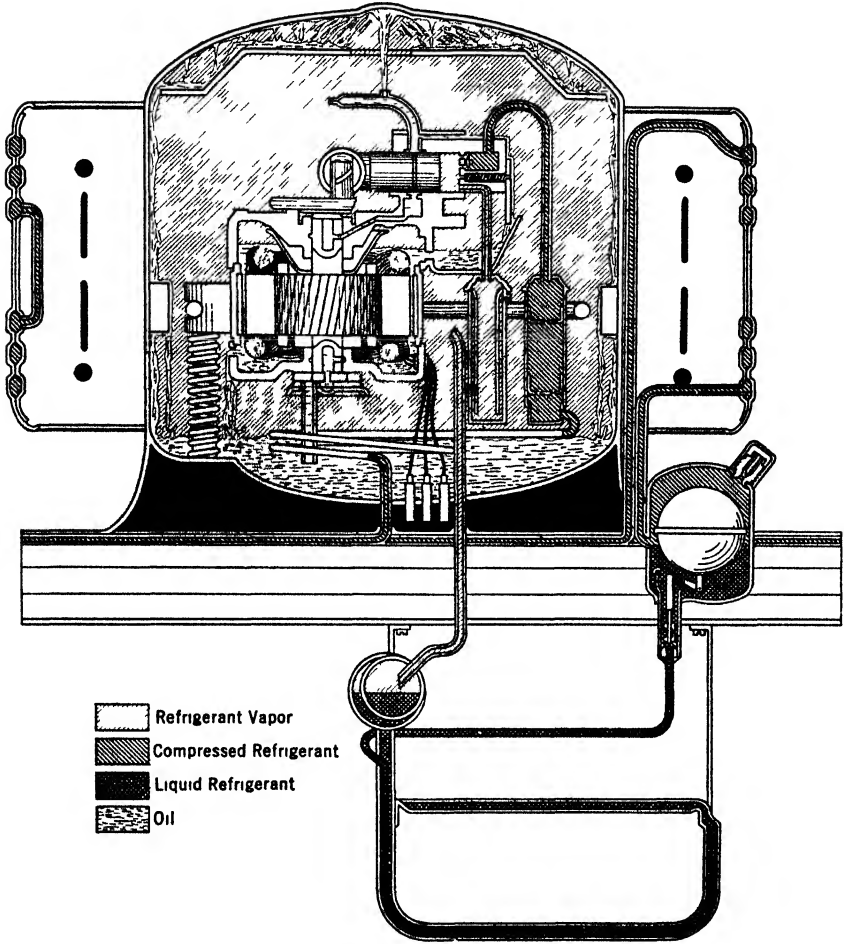


FIG. 17-12. The hermetically sealed type of household refrigerating machine.

type of motor unsuitable, but they can use the split phase or condenser type and also, with modification, the resistance-start induction motor.

17-9. The Absorption Machine. For the household absorption machine, gas, oil, or electricity may serve as the source of heat, and it may operate under thermostatic control with either gas, or electric resistance, or intermittent operation with oil. The air-cooled diffusion machine has already been described in Chapter VII.

In the oil-heated intermittent absorption machine (Fig. 17-13) the container *A* acts as a generator when heat drives the ammonia vapor out of solution and later as an absorber when the evaporator *K* begins to function. The condenser *E* is submerged in water, but the water is not

AUTOMATIC REFRIGERATING MACHINES

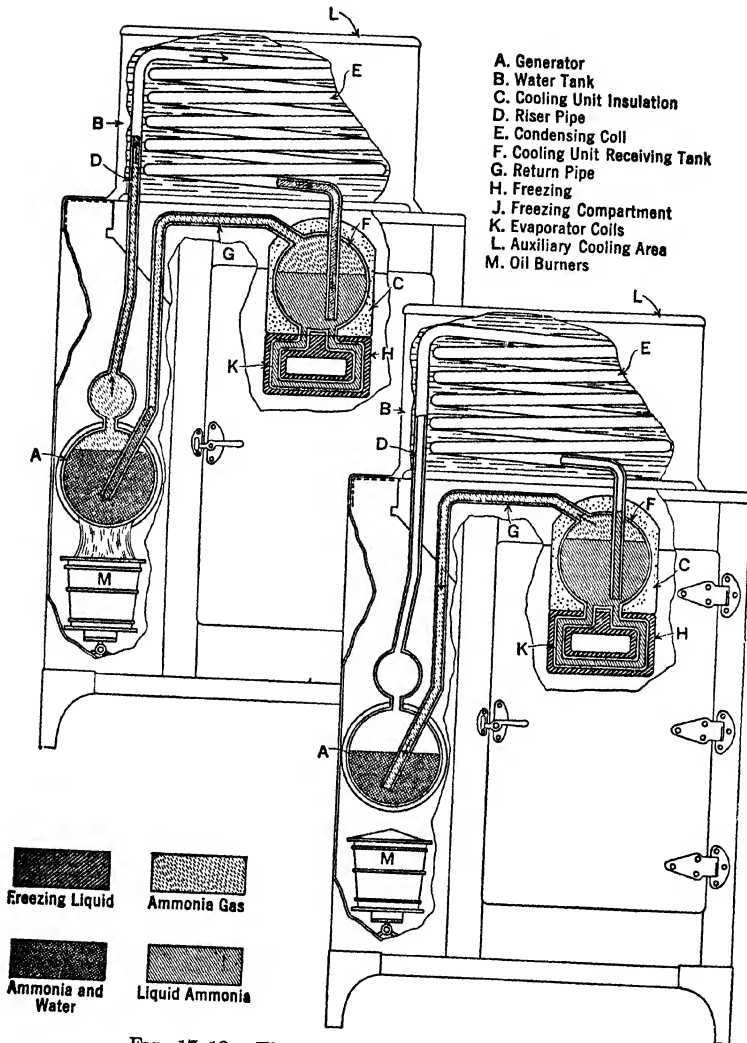


FIG. 17-13. The intermittent absorption machine.

continually renewed. Being intermittent the machine can be designed for the heating portion of the cycle to take place once in 24 hr. One of the essential features is a syphon (not shown in the figure) which operates for a few minutes after lighting the oil burner in order to return to the generator the condensed steam carried over with the ammonia during the heating period.

CHAPTER XVIII

ERECTION, OPERATION, AND TESTING

The refrigerating plant is subject to difficulties peculiar to the cycle. In the commercial sizes it has a tendency to accumulate air and water. It may carry over into the evaporator large amounts of oil which, with ammonia as the refrigerant, will remain there until removed. Loss of oil and of the refrigerant are two serious results of leaks and must be prevented if possible. The commercial plant is subject also to the loss of capacity and to excessive condenser pressure. Whereas the 1000-ton refrigeration compressor will probably be an uneconomical unit in the future, the emphasis now being on smaller units of about 10-in. by 10-in. maximum size, some consideration of the foundations required for the medium and large-sized machine, of the anchor bolts, the leveling of the machine, and the squaring of the pipe lines is still important.

18-1. Template and Anchor Bolts. A good mixture of concrete should be used, either a 1:2½:5 or a 1:2:4 mixture by volume of cement, clean sand, and crushed stone. Sufficient time should be allowed for hardening before any weight is placed on the foundation, usually from 36 to 48 hr. Before pouring, a template of the anchor bolts with the bolts and anchor plates attached should be set in place and rigidly fastened so that they will not float out of place. Care must be taken to insure the proper level, and the proper alignment with the building wall. The larger machines usually have pockets in the foundation so that access may be had to the anchor plates at any time. In any event, ferrules made of tin, old pipe, or thin lumber are so placed that the anchor bolt will hang central and will have a clearance of ¾ to 1 in. all around it. While the concrete is green these ferrules can be removed. The anchor bolts are screwed up to the middle of the threads on the lower part of the bolt. The pouring of the concrete should stop at a point from ½ to 1 in. below the base level in order to permit grouting under the machine.

18-2. Leveling and Grouting. When the foundation is sufficiently hard to take the weight, the compressor may be moved into place and leveled. Wedges made of wood or iron should be placed near the anchor bolts, and the machine should be leveled in both directions. In a

horizontal machine the center of the shaft must be accurately placed and can be located at right angles to the compressor line of centers by means of a triangular template made with distances between the apexes in the ratio of 3:4:5. The anchor-bolt holes should be filled first with a thin grout of sand and cement in the ratio of 1:1 by volume. Then the machine should be filled in underneath with grouting thin enough to float but not to show water at the top, the grout being carefully worked in under all parts of the compressor bed. In a few hours the wedges can be removed and the foundation given a smooth-finished surface. The foundation bolts must not be tightened until the grout is thoroughly dried, a matter of 2 or 3 days as a rule. Small compressors can be erected sometimes by drilling out the concrete and using an expansion bolt. These holes are filled with grouting as before.

18-3. Kind of Joint. The essential of good pipe work is clean, accurately cut pipe threads and the use of sharp dies with Briggs standard threads (for work in the United States). These threads should be cleaned thoroughly before making up, by brushing them with gasoline and wiping clean the pipe threads and the flanges or screwed fittings. Making up the joint so that it will be hot is no indication that it is tight, as the heat indicates friction only, which, of course, could be developed by dirt or scale as well as by tightness.

Both soldered and litharge joints are used for ammonia, although the tendency appears to be toward the elimination of the former except in condenser work where the soldering is done at the factory. The litharge joint is as easy to make up as a first-class steam-fitting connection.

The solder joint is made by cleaning the fittings and the pipe until the metal is bright. The pipe and the fittings are then dropped into molten solder, made of half-and-half tin and lead, and the threads are evenly tinned over with soldering acid made of muriatic acid cut with zinc. The fitting is made up while both are hot, and the recess at the back of the fitting is filled with solder. In the larger pipe sizes the solder in this recess should be calked.

The litharge joint is made by applying a thin layer of the mixture to the threads of both the pipe and the fittings. The mixture should be made in small amounts as needed, in the proportion of $1\frac{1}{4}$ parts of litharge to 1 part of glycerin by volume, and it should be mixed thoroughly. Care should be taken not to have excessive litharge lest the effective area of the pipe be decreased, especially in small pipe.

Much future trouble may be eliminated by care in the initial cleaning of the inside of any pipe to be erected so as to get rid of any loose mill scale or pipe threads, and it is advised that at the very least the pipe be examined on the inside and that it be rapped with a light hammer when

held vertically. Bent pipe should be tested for obstructions by rolling a steel ball through it.

18·4. Erection of Piping. The principal considerations in piping erection are proper support and the proper slope of the pipe lines so that

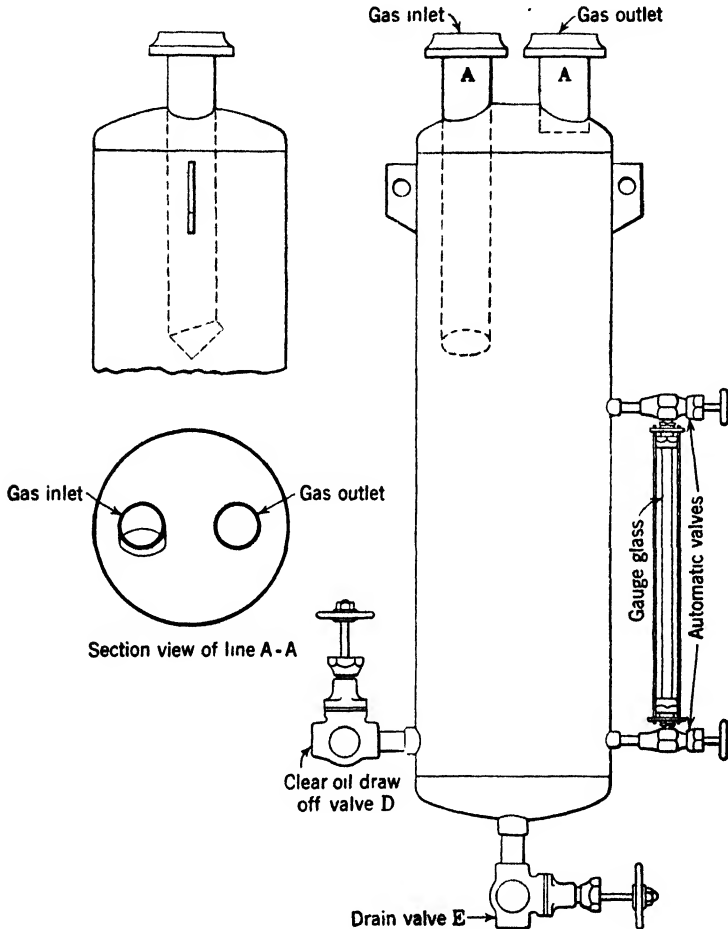


FIG. 18-1. The oil separator.

drainage of the water pipes and prevention of liquid seal in the ammonia pipes will be provided for, and the proper clearance allowed for so that erection and repairs will be as simple as possible.

The pipe should be very carefully erected, and it should be remembered that most cold storage piping has a heavy accumulation of frost and that it may be filled with liquid ammonia. Low points, where liquid may

trap in the piping, should be eliminated as far as possible, unless connection is made at every low point to suitable traps and to the regenerator. Pipe vibration must be kept at a minimum, or leaks may occur at any time. Flange joints may be made up with either lead or oilproof-rubber gaskets, and it is wise to chalk-mark the flanges as the gaskets are put in place, as a check. The difficulty with lead gaskets is that lead has no elasticity and that it will not give good service when exposed to the hot discharge gases from the compressor. For the ammonia discharge lines it is better to make use of a high-grade asbestos sheet gasket material which takes without trouble the highest temperatures found in refrigeration and has some elasticity. In making up, the bolts must be pulled up evenly all around, and special care must be taken to see that the flanged surfaces are free from dirt and scale.

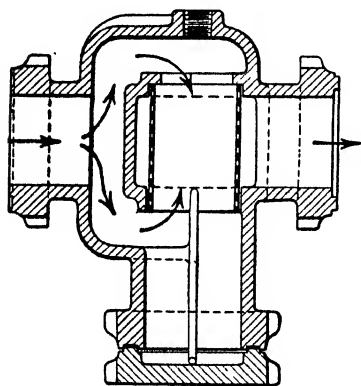


FIG. 18·2. Scale trap.

The *oil separator* (Fig. 18·1) can separate the oil only when it is in a liquid condition. In consequence the separator should be placed as near the condenser as practicable in order that the gas passing through may become as cool

as conditions will permit. At times it may be wise to install a precooling just ahead of the separator. The precooling is sometimes used in ice-manufacturing plants for the purpose of heating water for the dip tank.

The *scale trap* (Fig. 18·2) is placed in the line especially to protect the compressor. Even with the best of care in erection, mill scale, sand, and metal cuttings from the dies will circulate with the ammonia or other refrigerant. The scale trap should be as close to the compressor as possible, and it should be of the size and design required to make it function well. At first the need of such a device is very great, but this need decreases as time elapses. Means should be provided for easy cleaning of the trap with as little air as possible entering the system during each such cleaning.

As soon as the piping has been erected and water and steam or electric connections made, the plant is ready to be tested for leaks, with air under pressure. The compressor is used for pumping this air pressure, care being taken not to pump up too rapidly at any time for fear of explosions should too high a temperature be generated and too much oil enter the discharge line. The compressor should be checked for mechanical difficulties and the proper grade of lubricating oil. The splash type of

oiling is not used very extensively at the present time, but, if used, the oil level must be kept up to the mark on the crankcase. The oil should have a freeze test of -20° F or lower.

18-5. Starting up. All modern refrigerating compressors of nominal size and larger have by-pass connections (Fig. 18-3), from the suction to the discharge and from the discharge to the suction lines so that the normal action of the compressor may be reversed. There is also a means of opening the suction as well as the discharge to the atmosphere.

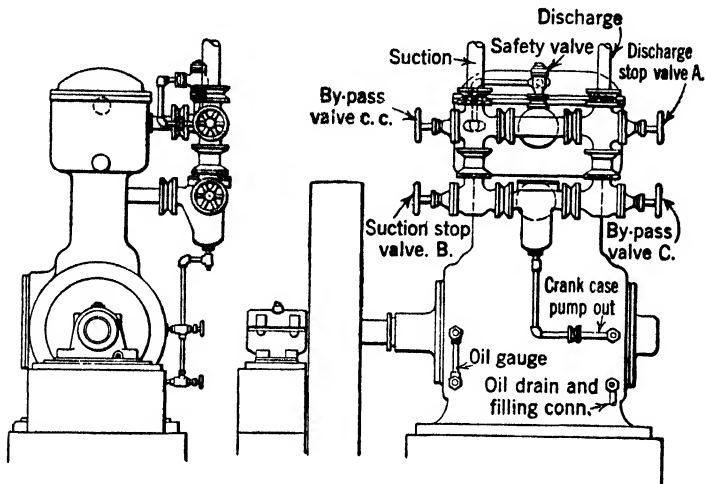


FIG. 18-3. By-pass valves for the vertical single-acting compressor.

In order to pump an air pressure on the refrigerating system, open the connection to the atmosphere on the suction side of the compressor and have the suction stop valve near the compressor closed. Separate parts of the system may be put under pressure by closing the appropriate stop valves if the compressed air can be made to reach the proper section of piping.

With water passing through the compressor jackets operate the compressor slowly, or intermittently, and raise the pressure to about 150 lb unless large leaks are evident before then, as indicated by the noise of escaping air. In a new plant the first things to look for are split pipe, flanges with defective gaskets or even with none at all, leaky valve bonnets, valve stems, etc. At first the escape of the air can be heard, but, when this is no longer true, painting with soap and water must serve. This method is necessary only for small leaks, for all other leaks can be heard if there are no other noises in the compressor room. The lather should be put on the pipe and fittings freely, and the pressure raised to

200 lb at this point for the entire system and the low-pressure gage should be safeguarded by shutting it off.

18-6. Blowing out the Coils. The piping is usually designed to permit the blowing down of each set of coils, thus freeing the pipes of any loose dirt, steel chips, etc., which may still be in the line. For this purpose it is usual to increase the air pressure to 250 lb and open freely and quickly to the atmosphere so that the velocity of flow will sweep out the loose material. After the blowing-down process it is wise to examine the compressor to make sure that no dirt has passed the suction trap, where there would be danger of injuring the cylinder or valves.

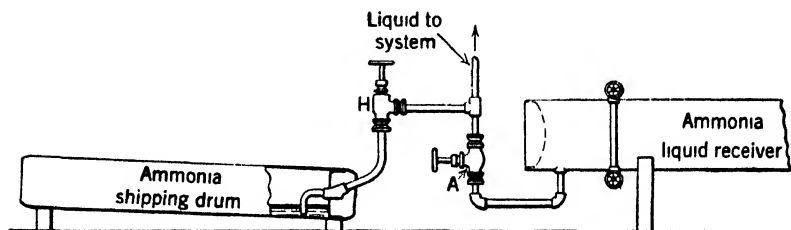


FIG. 18-4. Charging connections.

18-7. Test with Ammonia. Pump a vacuum on the entire system by closing the discharge stop valve, with the entire system open to the suction and the atmospheric discharge valve open. It is quite essential to produce as good a vacuum as the compressor will permit since the air has to be removed by one method or another.

Attach the ammonia charging drum (Fig. 18-4) to the charging connection and permit liquid ammonia to flow slowly into the low-pressure and the high-pressure side of the plant. Tests should be conducted with a burning sulphur stick, made by passing cotton cord through liquid sulphur. The indication of a leak, the chemical reaction of sulphur dioxide and ammonia gas, is a white smoke. Small leaks can be repaired by calking the metal, but for large leaks the pressure must be removed and the defective part replaced. The test for leaks of the refrigerants of the halide group is the alcohol lamp; see Chapter VIII.

18-8. To Charge the Plant. To charge, the expansion, pressure-reducing valve on the line must be opened full, and the valve on the shipping drum serves as the expansion valve. If the evaporator is a brine cooler, the brine must be continuously circulated and the compressor and the condenser must be kept in operation. The first indication of the emptying of the shipping drum is the appearance of frost on the drum and on the connections to the expansion valve, but the only

way to be positive that the drum is empty is to weigh the drum and to compare this weight with the tare on the shipping tag.

18·9. The Amount of Ammonia Charge. A number of approximate rules for charging are suggested, among them the following: Consider that the liquid receiver is three-fourths full of liquid and the condenser one-fourth full of liquid, with the remainder of the volume up to the compressor filled with refrigerant vapor at condenser pressure. If a brine cooler is used consider that it is three-fourths full of liquid, the liquid line up to the expansion valve completely full of liquid, and the remainder of the system to the compressor filled with suction pressure vapor. If so-called direct expansion piping is used the calculation is problematical because of the variation in operation with the design. The liquid may occupy a quarter to a half of the volume of the piping.

18·10. Operating the Expansion Valve. The function of the pressure-reducing (expansion) valve is to feed the evaporating surfaces with liquid at the proper rate. If the load is constant and the compressor is operating at constant speed, conditions will automatically adjust themselves after a time with any setting of this valve, as the suction pressure will be raised or lowered until the weight of refrigerant being drawn into the compressor and condensed in the condenser is equal to the weight passing through the expansion valve.

However, each plant has its own operating pressure dependent on the duty to be performed. This is such an operating pressure as will give 3° to 5°, 10°, or even 15° difference between the evaporating temperature of the refrigerant and the temperature of the fluid surrounding the surfaces. The low-pressure piping is laid out in such a way as to assure this particular difference of temperature, so that the plant, to do its particular duty, must be operated under the conditions for which it was planned.

18·11. Operation. If air or non-condensable gases enter the refrigerating system the only practicable manner of removal after charging the system is to use some form of *purger* (see Fig. 19·1), which is simply a device for the condensation of the refrigerant in the presence of the inert gas. To operate, the vapor mixture from the top of the receiver is bled into the purger, as shown in the figure. Should the liquid need to be removed the method used in Fig. 18·5 may be employed.

After a period of operation the larger plants should have some device by which oil and water may be removed from the low-pressure side of the plant. The low point of the system is connected to a *regenerator* which, applying sufficient heat to evaporate the ammonia, forces it to pass through the suction line of the compressor, leaving the oil and water to be drained to the waste. It may be necessary, in order to make

the accumulation in the evaporator move into the regenerator, to bypass hot gas from the compressor, thereby forcing the evaporating coils to act momentarily as a condenser.

The accumulation of frost on the direct-expansion piping retards heat transfer; there are many ways of removal, the most favored being to pass hot gases from the compressor into the evaporating piping. This also permits some freeing of oil, etc., if a trap is installed at the low point.

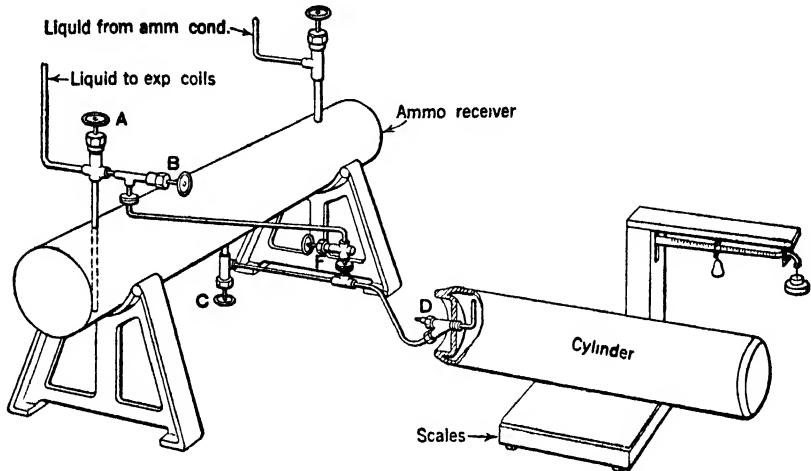


FIG. 18-5. Withdrawing the ammonia charge.

The refrigerating system is a closed one, using the refrigerant over and over; but its operation has inherent troubles. Scum, dirt, and scale from impure water may accumulate on the water side of the condenser; frost and oil may cover the outside and the inside, respectively, of the evaporator surfaces; leaky valves, piston rings, or scored cylinder walls may result in poor pumping ability of the compressor; obstructed circulation of either the water or the ammonia may put the condenser out of commission or at least decrease its working ability; and faulty action of the brine cooler or the direct-expansion piping may result in poor heat transfer. Examples of improvement in operation follow.

(a) Removal of a long accumulation of ice and frost permitted the machine to be operated at 15-lb suction pressure gage instead of 3-lb gage with equally good room temperatures.

(b) Removal of a long accumulation of oil in the piping permitted the suction pressure to be further increased to 17-lb gage.

(c) By processes (a) and (b) the piping had now been returned to normal conditions. This particular plant was under-piped, and by an increase of the piping the proper room temperature was obtained by

operating at 22-lb gage. It can be shown that with the same condenser pressure a 22-lb gage suction pressure will result in 2.5 times the refrigerating capacity of the compressor over a suction pressure of 3-lb gage.

18-12. Causes of Excess Pressure. Excess pressure in the condenser may be caused by: (a) too small a condenser, hot cooling water, or too small an amount of water for the temperature of the water used; (b) inert gases in the condenser, mostly air; (c) poor heat transfer due to dirt or scale from the cooling water and oil and dirt on the ammonia side of the condenser; (d) condensers partly filled with liquid ammonia as the result of an overcharge or of improper drainage of the condensate caused by partly closed valves or an obstruction.

18-13. Causes of Loss of Capacity. Loss of capacity in the plant may be caused by: (a) insufficient piping surface, or frosting on the outside or oil and scale on the inside; (b) leaking valves, safety head, or piston rings, or scored cylinders in the compressor, or slow closing of the valves due to a number of causes; (c) obstructions in the suction lines or broken valves preventing full opening; (d) too small a charge of ammonia, permitting gas to pass from the compressor through the expansion valve into the evaporating coils; (e) the presence of water in the evaporating coils, the action of which is to require a lower pressure to be carried in order to obtain the temperature expected of anhydrous ammonia.

18-14. The Testing of Refrigerating Plants. The refrigerating engineer is interested in securing information on the *capacity* of the plant, on the *power required* to produce a unit of refrigeration, and on *heat transfer*, but he is no longer interested in a number of the details of test procedure so important at the beginning of the century. The almost universal adoption of the electric motor has stimulated the use of light valves and of medium rotative speeds as well as excellent design and machine work. The indicator diagram is of value as a means of showing valve action, but it is no longer of primary interest. Testing refrigerating machines is similar to other mechanical engineering laboratory testing except where the necessity of measuring the amount of the refrigerant used per unit of time makes the problem different.

18-15. Measuring the Refrigerating Effect. The *weight* of ammonia may be determined in three different ways: (a) by weighing the condensate in drums under condenser pressure (the drums rest on platform scales connected with the condenser and the liquid line to the expansion valve by means of suitable flexible pipes); (b) by the use of calibrated tanks, the density of the liquid ammonia being found from the tables corresponding to the liquid temperature in the drum; and (c) by the use of calibrated liquid meters.

With methyl chloride, dichlorodifluoromethane, and methylene chloride, each of which is miscible with oil, none of these methods is accurate. With these refrigerants the only possible method is to calculate the refrigerating effect. For boiling temperatures of the refrigerant above 32° F, distilled water may be used in an evaporator; the test readings consist of the water temperatures in and out of the evaporator and the weight of the water determined by means of a calibrated flat plate orifice or a venturi meter. Care would be required to prevent liquid refrigerant passing over to the compressor with the vapor and oil *mist*. This could probably be assured by means of an insulated separator above the evaporator.

For temperatures at or below 32° F a brine of suitable density will have to be used, complicating as it does the determination of the refrigerating load because of the variation of the specific heat of the brine with the density and temperature. The specific heat will need to be found accurately.

CHAPTER XIX

CONDENSERS; COOLING WATER SUPPLY

The film theory propounds the idea that, when molecules escaping from the surface of a solid or liquid pass into air or other inert gas within a container, there first results a condition of equilibrium upon the inner surface of the relatively stationary gas film surrounding the solid or liquid. The molecules pass through this film by diffusion, and then into the surrounding space by both diffusion and convection until equilibrium is established throughout the entire container. The total pressure exerted on the walls of the containing vessel is the sum of the vapor pressure of the liquid (which would exist independent of the presence of any other gas or vapor), and the pressure that the gas would exert if it occupied the entire volume of the container by itself. (See also Chapter XIV for mixtures of air and water vapor.) If vaporization is to continue at a constant temperature it is necessary that the tendency for dynamic equilibrium be destroyed. This is achieved by constantly removing the vapor molecules and by decreasing the temperature and pressure, thus tending to decrease the number and the velocity of the molecules, a process performed in the condenser. It is very evident that the action of liquefaction is the reverse of evaporation, and that the one may be made to replace the other if we exchange, for the addition of heat to the liquid, the removal of heat from the vapor.

The *vapor pressure* during the process of liquefaction in the refrigeration cycle is determined by what occurs in the condenser. Such a temperature of liquefaction is established as will permit the mean temperature difference to make it possible for the water or air at its particular initial temperature, and with the available effective area of heat-absorbing surface, to remove the amount of heat coming to the condenser per unit of time. A condition of equilibrium is therefore established, and the *pressure in the condenser* will be the sum of the vapor pressures of the refrigerant, an amount corresponding to the temperature of liquefaction, and the pressure of the inert gas, if any appreciable amount is present, according to Dalton's law of partial pressures.

It is not possible to free refrigerating systems of all *inert gases* even when the purge device shown in Fig. 19·1 is used. These gases, which may be either decomposed ammonia and air or simply and, usually, the

latter, are carried along with the ammonia during the cycle of operations and finally reach the condenser and the liquid receiver, where they remain. Although diffusion would be likely to keep the inert gases uniformly mixed during operation there is a tendency for a film of very nearly pure inert gas to collect around the heat transfer surfaces, thereby assisting to insulate the surface.* Therefore, both from the standpoint

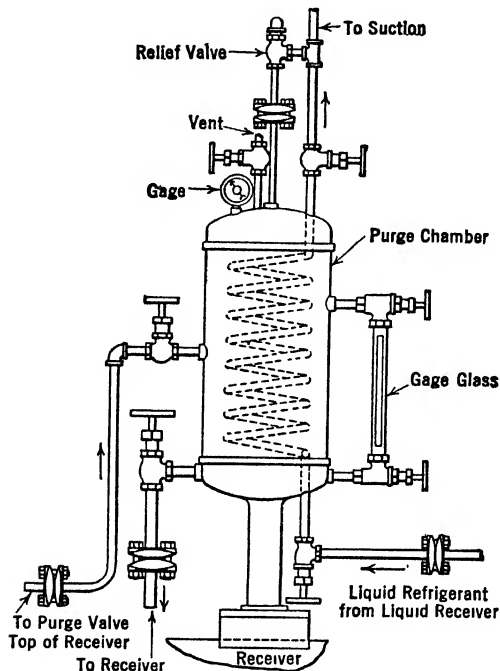


FIG. 19-1. A method of condensing ammonia out of the gases bled off from the liquid receiver.

of an increase of work done in the compressor and of a reduction of efficiency in the condenser surface, the inert gas in the refrigerating system should be eliminated.

19-1. Designs of Condensers. A large number of *designs of condensers* have been placed in operation. Condensers for ammonia and carbon dioxide are made of steel and wrought iron, made up with fittings, welded, or the tubes expanded into the tube sheets; condensers for sulphur dioxide, methyl chloride, and the halides are made of copper

* Orrok found that air in a steam surface condenser affected heat transfer to the fifth power of the richness ratio p_s/p , where p_s was the partial pressure of the steam and p the total pressure in the condenser.

and compositions. Condensers were designed in the early days with the principal objective of safety, but more recently economy of heat transfer surface as well as compactness and freedom from splash and humidity have also been carefully considered. The problem is different from that of steam engineering inasmuch as the pressures are very much greater with most refrigerants (see Chapter VIII), and it is imperative to avoid loss of the refrigerant through leaks. As the temperature range is much greater than with the modern steam surface condenser a much harder scale is likely to be found on the surfaces of condensers for refrigerating plants, especially where deep-well or other condensing water containing carbonates and sulphates in solution is used, although in less amount than would be found on the tubes of the smaller boiler plants. It is quite evident that the design of condensers for refrigerants must be such as will permit easy access to the surfaces for cleaning.

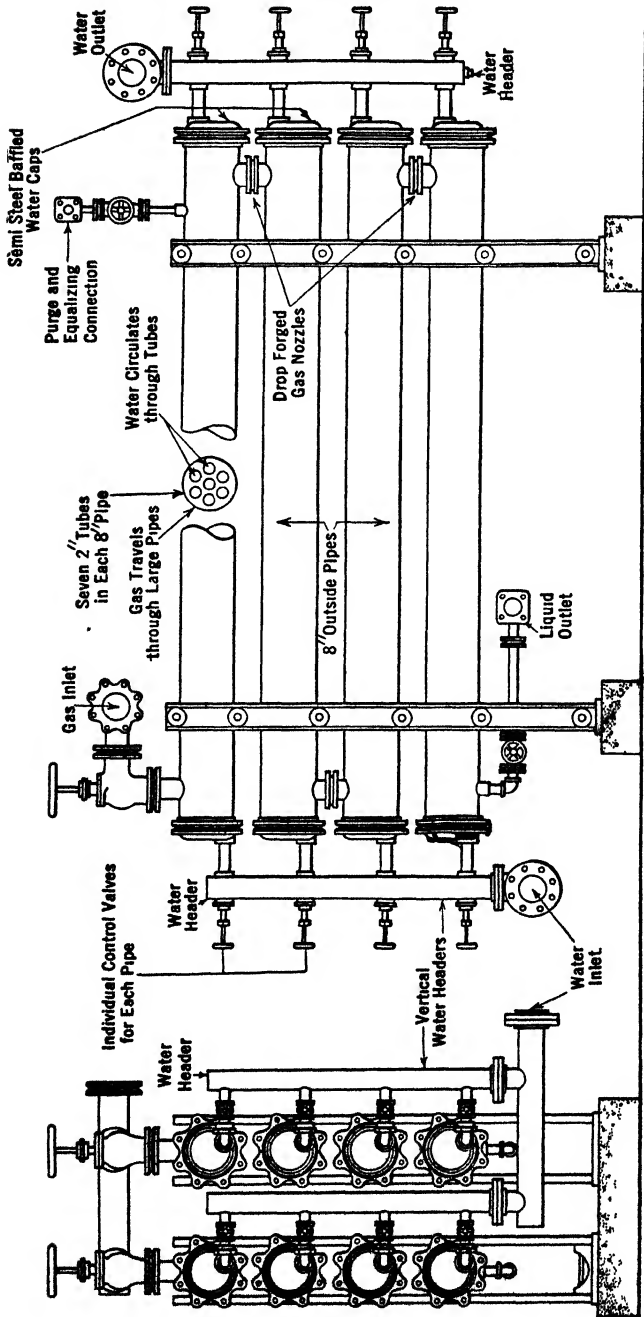
Research has proved that the best steam surface condenser is the one that permits the vapor to reach the cooling surfaces with the least possible resistance and removes the condensate quickly. In other words, the purpose of the condenser is to bring the cooling surface into contact with the vapor and keep it continuously so exposed.

It would seem, therefore, that certain of the early condensers, the so-called *pipe condensers*, which were designed principally for safety, could not be very efficient in the economical use of the pipe surface. These include the atmospheric, bleeder, and double-pipe designs where the refrigerant enters at one end of a continuous pipe construction and travels some 220 ft or more before it finally leaves in the liquid state. In the double-pipe condenser using *counter flow* of the refrigerant and the water, with the condensing water entering at the bottom of the stand, it is evident that if any of the refrigerant is condensed in the upper pipes it will have to flow with the uncondensed vapor through the remainder of the pipes of the condenser before it can get out. Tests performed by Macintire† indicate that from 50 to 70 per cent of all liquefaction occurs in the last two pipes, which in this case are the two bottom ones. In the bleeder atmospheric condenser, with the vapor entering at the bottom and the coldest water at the top, from 50 to 70 per cent of all liquefaction occurs in the two top pipes, where the coldest water is and where the temperature difference between the ammonia and the water is the greatest.

According to the theory of vapors free from non-condensable gases there cannot be any *subcooling of the condensate* except when the pipe becomes filled with liquid to shut the liquid materially away from its vapor, or when a separate liquid cooler is used that functions in the same

† *Univ. Illinois Eng. Exp. Sta. Bull.* 171.

CONDENSERS; COOLING WATER SUPPLY



(a)

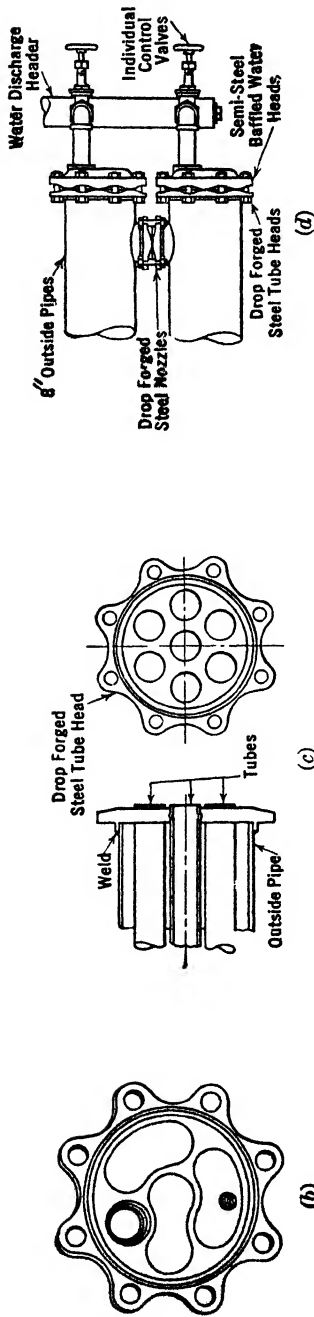


FIG. 19-2. Design of a seven-pass multitube ammonia condenser.

manner. At all other times the temperature of liquefaction will need to remain constant whether the condenser is called a counter- or a parallel-flow condenser, although the term is meaningless as far as liquefaction

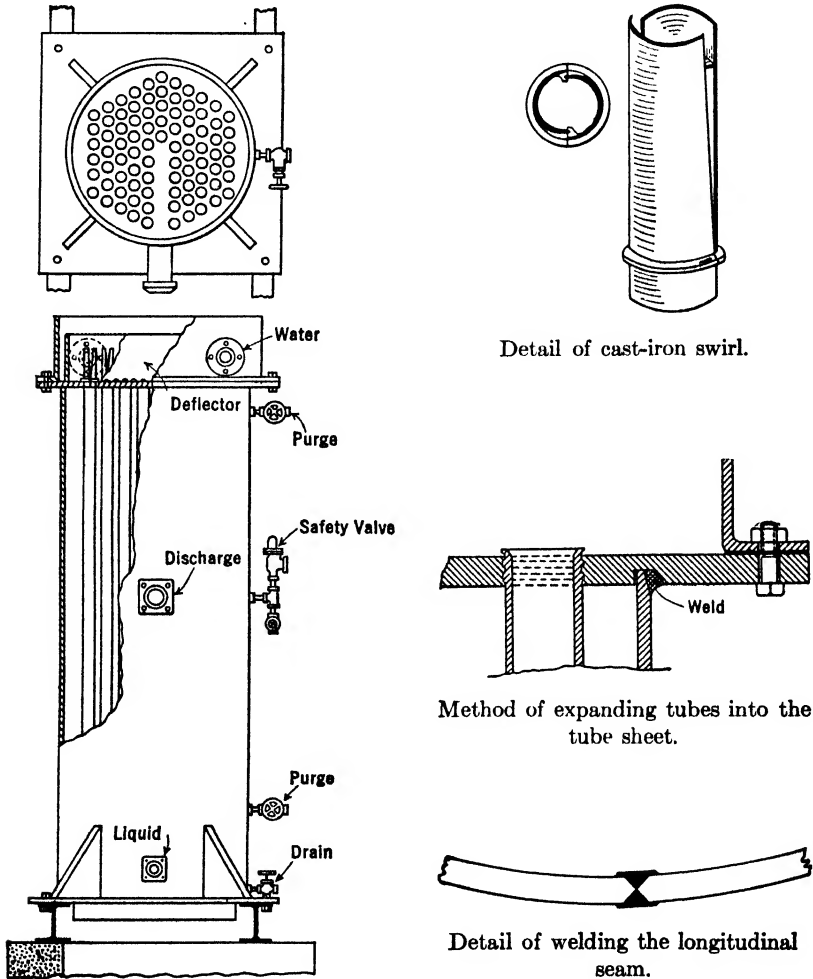


FIG. 19-3. The vertical shell and tube ammonia condenser.

is concerned. For the removal of the superheat the problem is different, and at times a separate superheat remover, using the water leaving the liquefaction condenser and passing counter flow with the vapor, will be advantageous. If inert gases are present in the condenser it will seem that subcooling of the condensate is taking place because the temperature of liquefaction always corresponds to the *partial pressure* of the

vapor. If no inert gases are present a condenser operating at or near its designed capacity will have a liquefaction temperature of the refrigerant within 1.0° F of the exit temperature of the condensing water. If the condenser is overloaded the excess capacity is obtained by increasing the average temperature difference between the water and the refrigerant, and the rule will not apply.

The so-called *flooded-type condenser* was designed with the idea that, as better heat transfer is possible with wet surfaces than where superheated vapors are present, an advantage was to be found by showering the condensing surfaces with liquid condensate. This was accomplished by means of an ejector which mixed liquid ammonia with the hot compressed gas from the compressor. High values of heat transfer were obtained with the flooded condenser, but at a sacrifice due to the increase in the condenser pressure resulting from the decrease in the cross-sectional area of the pipe and the excessive gas velocity resulting therefrom. The large values of heat transfer were due to the high gas velocity, in spite of the reduced area of condenser surface in contact with the vapor. The criterion is to remove the condensate as quickly as possible, as it is well established that the presence of the liquid is a detriment to heat transfer.

As a result of more confidence on the part of the manufacturers, two new types, the *horizontal shell and tube* and the *vertical shell and tube* condenser, have been developed where the tubes have been expanded into the tube sheets. In the vertical condenser the water flows inside the tubes in a water film holding to the surface by adhesion and to some extent by the initial swirl where the flow is caused by gravity with one laminar layer flowing past another in viscous flow. In the horizontal multipass condenser the pipes are filled with water, which usually has six or more passes before the condensing water leaves the condenser, and as the flow is controlled by a head of water or a pump the water flow is usually turbulent. Figures 19.2 and 19.3 show these more recent types of condensers.

19.2. The Evaporative Condenser. Owing to the rapid development of air conditioning the demand for the conservation of water has occasioned a need for a modified condenser, namely, the evaporative condenser. This apparatus is not simply a cooling tower added to the usual condenser, but a complete combination condenser which includes the condenser, the cooling tower, and the water and air circulation systems. The design usually includes one or more fans, water spray nozzles, and condenser surface made up of standard pipe or finned tubes or pipes, as well as, frequently, eliminator surfaces to prevent excessive loss of water in the discharge air from the installation. As the

water evaporates on the condenser surface at the wet-bulb temperature of the entering air it absorbs approximately 1000 Btu/lb of water, as compared with about 15 Btu absorbed owing to temperature rise by a pound of condensing water in the conventional condenser. The evaporative condenser requires theoretically from 1.5 to 2.0 per cent of the water employed in the usual condenser, but the actual consumption is from 5 to 10 per cent of the amount required for air conditioning units. The overall coefficient of heat transfer† U , varies from 12 to 50 Btu/(sq ft)(hr)(°F).

19.3. The Subcooler. As the usual condenser cannot subcool the condensate, is a separate subcooler justified and how much better will it make the operation? It may be assumed that the subcooler is designed for counter flow with the coldest water available, that the condenser is free from non-condensable gases, and that the operation is such that the condensing pressure corresponds to the temperature of the exit water plus 1° F.

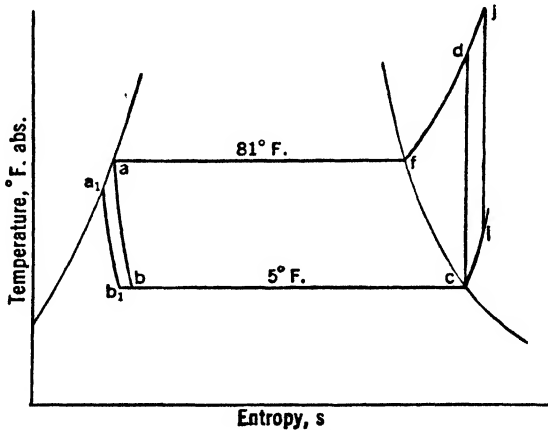


FIG. 19.4. The T - s diagram showing subcooling of the refrigerant.

Example. Assume 70° F initial temperature of the water and 10° F rise of the water temperature. Assume that the temperature of evaporation is 5° F. Show the theoretical advantage by the use of a liquid subcooler that is designed to cool the condensate to 72° F.

Solution. According to Fig. 19.4 the amount of water used, per pound of ammonia, is for the cycle $abcda$

$$W = \frac{h_d - h_a}{10} = \frac{707.3 - 133.1}{10} = 57.4 \text{ lb} \\ = 6.88 \text{ gal}$$

† James, *Refrig. Eng.*, January, 1937.

The coefficient of performance for the cycle $abcd a$ is

$$\frac{613.3 - 133.1}{707.3 - 613.3} = 5.11$$

and for the cycle $aa_1b_1cdf a$ it is

$$\frac{613.3 - 122.8}{707.3 - 613.3} = 5.22$$

The amount of water required to subcool the liquid from 81° to 72° F (using a 9.0° F rise of water temperature) is

$$\frac{h_a - h_{a_1}}{9.0} = \left(\frac{133.1 - 122.8}{9.0} \right) = 1.14 \text{ lb}$$

If, on the other hand, all the water had been put through the liquefaction condenser the rise of temperature would not have been 10° and the temperature of liquefaction would not have been 81° F. The water temperature rise would be, approximately,

$$\frac{h_d - h_a}{57.4 + 1.14} = \frac{707.3 - 133.1}{58.5} = 9.82^\circ$$

This corresponds to a temperature of liquefaction of $70^\circ + 9.8^\circ + 1.0^\circ = 80.8^\circ$ and a pressure of 155.1 lb, resulting in a new value of $h_d = 706.0$ Btu and a new value of $h_a = 132.9$. The new coefficient of performance is

$$\frac{613.3 - 132.9}{706.0 - 613.3} = 5.18$$

From a comparison of the different values of the coefficient of performance it appears that there is a slight theoretical advantage in the use of the liquid subcooler; however, this advantage is soon lost if the liquid line to the pressure-reducing or float valve is not insulated against the absorption of heat from the surrounding air.

19·4. Heat Removed by the Condenser. The heat removed by the condenser is usually considered to be the amount of useful refrigeration plus the work of compression. Using Fig. 19·4 this becomes, in Btu per pound of refrigerant, $h_d - h_a$, and the amount of heat removed per ton of refrigeration per minute is

$$Q_c = 200 \left(\frac{h_d - h_a}{h_c - h_a} \right) \quad (19\cdot1)$$

The expression for the actual volumetric efficiency, given in Section 3·5, should not enter into this formula because the transient heat absorbed by the cylinder as a skin effect during the compression is lost to the suction vapor during that part of the cycle when the refrigerant has a lower temperature than that of the metal composing the cylinder.

Equation 19·1 gives the maximum amount of heat to be removed by the condenser, for in practice the loss of heat to the jacket or the atmosphere will tend to lower the value of h_d . However, some friction is developed in the cylinder, tending to increase the transient heat, with the result that for moderate-speed compressors having dry compression, where the condenser is reasonably close to the compressor, as it is in most refrigerating plants, equation 19·1 applies.

The amount of water in gallons per minute per ton of refrigeration required by the condenser (using t_d to represent the rise of temperature of the water) is

$$G = \frac{Q_c}{8\frac{1}{3}t_d} = \frac{200 \left(\frac{h_d - h_a}{h_c - h_a} \right)}{8\frac{1}{3}t_d} \quad (19\cdot2)$$

If a superheat remover or a separate liquid subcooler is used, the amount of heat absorbed by the water per ton of refrigeration per minute is

$$\text{For the superheat remover, } Q_1 = 200 \frac{h_d - h_f}{h_c - h_{a_1}}$$

$$\text{For the liquid subcooler, } Q_2 = 200 \frac{h_a - h_{a_1}}{h_c - h_{a_1}}$$

$$\text{For liquefaction, } Q_3 = 200 \frac{h_f - h_a}{h_c - h_{a_1}}$$

$$\text{Total heat removed, } Q_c' = Q_1 + Q_2 + Q_3$$

where h_{a_1} and h_a will coincide if there is no liquid subcooler. However, if a counter-flow superheat remover is used and the initial water temperature in the superheat remover is below the temperature of liquefaction there will be a tendency for some condensation to occur even in the presence of superheated vapor. If t_s , t_l , and t_a are the average temperature differences between the refrigerant and the water during superheat removal, liquefaction, and liquid subcooling, respectively, the average temperature difference t_m is

$$t_m = \frac{Q_1 t_s + Q_2 t_a + Q_3 t_l}{Q_c}$$

19·5. Optimum Water Rate. As the temperature of the exit condensing water is approximately the temperature of liquefaction the rise in temperature of the water is an important consideration in the study of the economical production of refrigeration. If the water costs a fixed amount of \$*B* per 1000 gal, the power costs \$*A* per hp-hr, and the

increase of power m per 1.0° F rise of the temperature of liquefaction per ton of refrigeration, then the total cost of power for a temperature increment t_d is Amt_d/e_1 , the cost of the condenser water per ton of refrigeration per hour is $(60/1000)B(h_c/8.33t_d) = (0.0072Bh_c)/t_d$, and the total operating cost, as far as the power and water are concerned, is

$$C = \frac{Amt_d}{e_1} + \frac{0.0072Bh_c}{t_d}$$

where e_1 is the combined efficiencies of the compressor and the motor and h_c is the heat removed from the refrigerant per ton of refrigeration per minute. The value of m has to be found by the use of equation 3·13.

For a minimum cost put the first derivative equal to zero:

$$\frac{d}{dt}C = \frac{Am}{e_1} - \frac{0.0072Bh_c}{t_d^2} = 0$$

or

$$t_d = 0.0848 \left(\frac{Be_1}{A} \right)^{1/2} \left(\frac{h_c}{m} \right)^{1/2} \quad (19\cdot3)$$

Example. Taking the cost of power at \$0.02 per hp-hr and the cost of water at \$0.10 per 1000 gal, find the most economical water rate per ton of refrigeration per minute for standard operating conditions and with a value of e_1 of 0.85.

Solution.

$$t_d = 0.0848 \left(\frac{0.10 \times 0.85}{0.02} \right)^{1/2} \left(\frac{239.9}{0.023} \right)^{1/2}$$

$$G = \frac{239.9}{8.33 \times 17.9} = 1.61 \text{ gpm}$$

If sprayed, cooling tower, or deep-well water is used the cost of the water is mostly the power cost, and this is equal to the weight of the water in pounds per unit of time times the total pumping head in feet. The total head is equal to the vertical lift plus the friction head, which usually varies as the 1.8 power of the average velocity in feet per second in the pipe.

The power increase per 1.0° F rise of liquefaction temperature is, as before, mt_d/e_1 ; the power required for pumping is $8.33GL/33,000e_p = h_cL/33,000t_d e_p$. The total cost of pumping, per ton of refrigeration per hour, is

$$C = A \left(\frac{mt_d}{e_1} + \frac{h_cL}{33,000t_d e_p} \right)$$

where L is the total head of water on, and e_p is the efficiency of, the pump.

For a condition of minimum cost (putting the first derivative equal to zero),

$$\frac{d}{dt}C = A \left(\frac{m}{e_1} - \frac{h_cL}{33,000e_p t_d^2} \right) = 0$$

Therefore

$$m - \frac{h_c L e_1}{t_d^2 e_p 33,000} = 0 \quad \text{and} \quad t_d = \left(\frac{h_c L e_1}{33,000 e_p m} \right)^{1/2} \quad (19.4)$$

Example. If, as in the preceding example, $h_c = 239.9$, $m = 0.023$, $e_p = 0.5$, $e_1 = 0.85$, and $L = 100$ ft,

$$\begin{aligned} t_d &= \left(\frac{239.9 \times 100 \times 0.85}{0.023 \times 0.5 \times 33,000} \right)^{1/2} \\ &= 7.32^\circ \text{F} \end{aligned}$$

19.6. The Water Supply. Commercial refrigeration has been considered, up to the present, in terms of the *cooling water* as regards its temperature, its cost, and its amount. Usually the most satisfactory source of condensing water is from deep wells,§ which (Fig. 19.5) give water temperatures that are approximately the average air temperature for the entire year for each locality. In addition, Fig. 19.6 gives the average surface temperatures for the three summer months, which are, ordinarily, the approximate air temperatures for the same period.

If sprays or cooling towers are used, the lowest temperature obtainable is some 2° or 3° F above the wet-bulb temperature. This method of securing condensing water for commercial plants is the only practicable one if the water supply is limited or costly.

19.7. Spray Ponds and Cooling Towers. An example of the principles of adiabatic saturation, combined with some cooling due to the air, is to be found in the cooling of condensing water by the use of spray ponds and towers. The amount of cooling can be expressed by the formula

$$c_w w (t_1 - t_2) = [W_a c_{p_m} (t_3 - t_2)] + W_w L \quad (19.5)$$

where w is the weight of the water leaving the tower, W_a is the weight of the air, W_w is the weight of the water evaporated, $t_1 - t_2$ is the decrease in temperature of the water, $t_3 - t_2$ is the change in temperature of the air, and L is the latent heat of evaporation of the water. The rate of flow of heat is proportional to the temperature difference between the liquid and the gas, whereas the rate of diffusion of water vapor is proportional to the difference between the vapor pressure of the water and the partial pressure of the water vapor in the air, or the amount of heat flowing from the water to the air per unit of time is

$$W_s dT = h_a A dx (T - t) \quad (19.6)$$

§ *U. S. Geol. Survey*, No. 520-F. Other charts and tables of interest are: Fig. 10.1, Temperature and Relative Humidity Chart for the Hottest Week, and Table 21.1, Climatic Conditions.

|| *Robinson, Trans. A.S.M.E.*, Vol. 44, p. 669, 1922.

where s is the specific heat of the air-water vapor mixture, h is the coefficient of heat transfer per unit area, a is the square feet of cooling

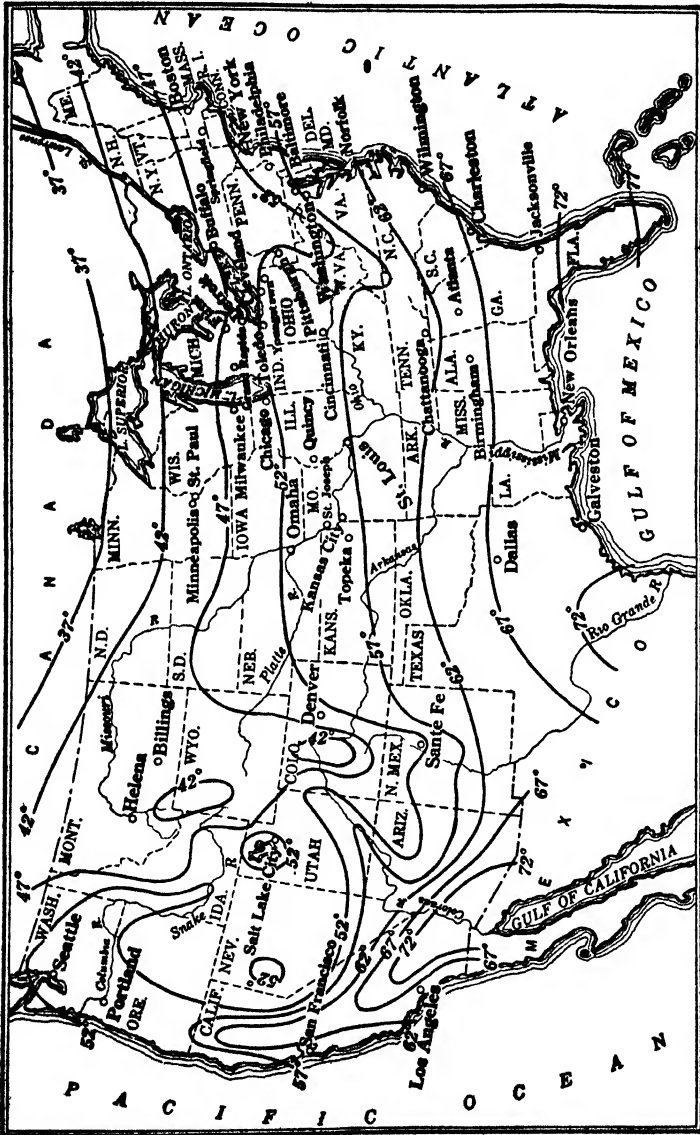


Fig. 19-5. Temperature of non-thermal wells.

surface per cubic feet of volume of the tower, and x is the height of the tower.

As a rule it is satisfactory in the design of a piece of apparatus subject to much change in operating conditions, such as the atmospheric dry-

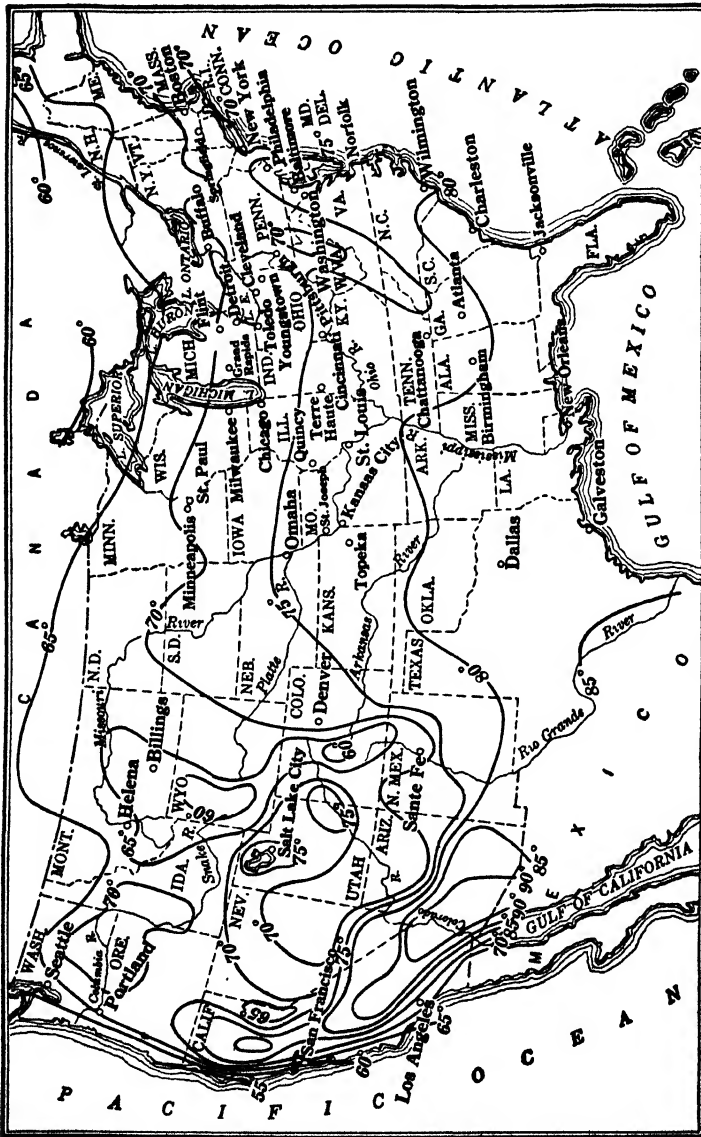


Fig. 19-6. Temperature of surface water during summer months.

and wet-bulb temperature, to use empirical formulas which have been found workable. In any cooling device the rate of cooling depends on

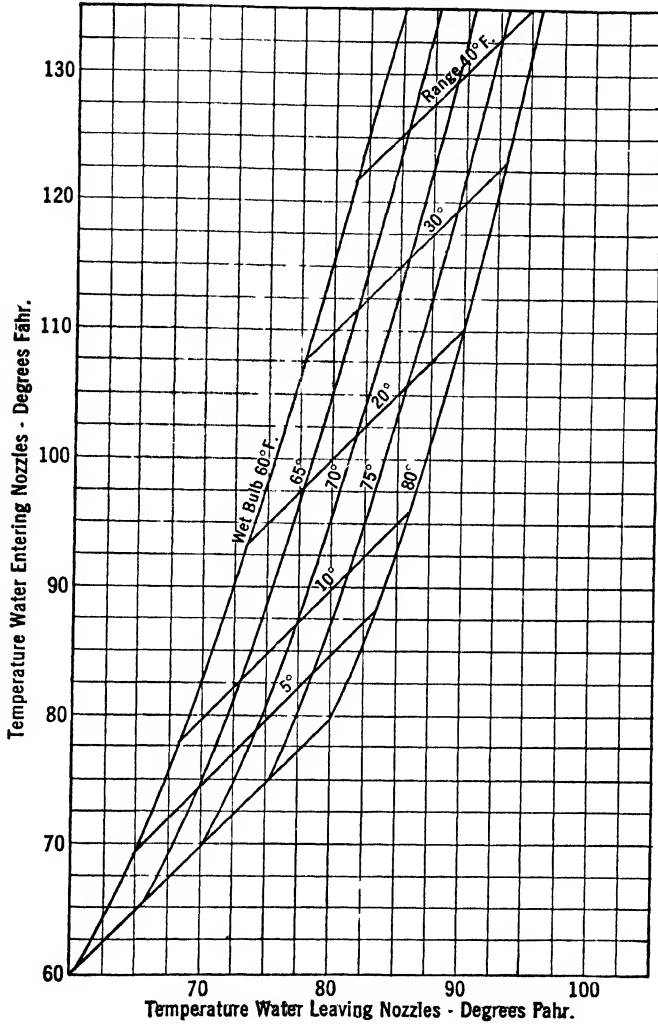


FIG. 19-7. Chart showing spray water cooling. (Courtesy Cooling Tower Co.)

the water surface exposed to the atmosphere, the air velocity, the dry- and the wet-bulb temperature, and the length of time the water is exposed to the air. In refrigeration little of the cooling is caused by radiation and convection. To complicate the situation, with spray and tower cooling the time interval is so short that very incomplete cooling is possible. Figure 19-7 shows that with 90° water entering the nozzles a temperature of 81° would be expected with a 75° wet-bulb temperature,

whereas with sufficient time the wet-bulb temperature itself could be approximated. The loss of water due to evaporation is approximately

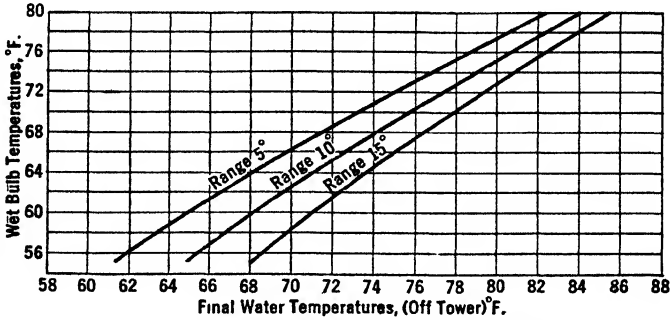


FIG. 19-8. Curves for a mechanical-draft cooling tower, based on 2.6 gpm per sq ft loading and an air velocity of 700 fpm. (Courtesy Cooling Tower Co.)

1 per cent, and windage may account for an additional 1 per cent or considerably more.

Cooling towers may be of the atmospheric type with louvers on the sides to decrease the drift, or of the forced-draft type. The atmospheric

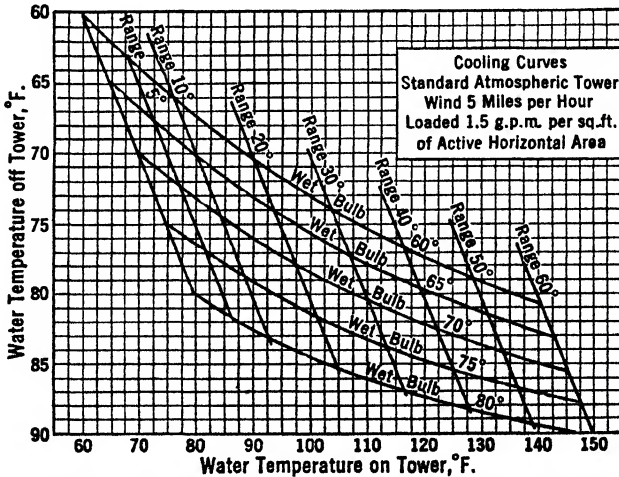


FIG. 19-9. Cooling tower water cooling for natural draft. (Courtesy Cooling Tower Co.)

type depends on the wind, and it requires an unobstructed location where the drift will not be objectionable. The forced-draft tower is becoming more popular for refrigeration as the drift is absent, the air circulation is constant, and the power consumed is usually nominal.

As a rule the cross-sectional area of the tower is found by allowing $1\frac{1}{2}$ gpm/sq ft of active cross-sectional area and 3 gpm/sq ft for the forced-draft type. The nozzle pressure is usually about 2 psi. Figure 19·9 may be used in the solution of natural-draft cooling tower problems.

Example. A forced-draft tower is to operate at 75° F wet-bulb and with a 10° range.

Solution. From Fig. 19·8 the off temperature is 80°, and the on temperature would be 80° + 10° = 90° F. The net cross-sectional area for the design is on the basis of 2.58 gpm/sq ft. The tower consists of slats or wire mesh which prevents free fall and exposes a large wetted surface to the air. The maximum drop of temperature would be 90° - 75° = 15°, and with a well-designed tower an efficiency of 73½ per cent can be attained so that the outgoing air will have a dry-bulb temperature of 75° + 11° = 86° F. The "total heat" on the psychrometric chart will be 38.5 Btu at 75°, and at 86° and 92 per cent relative humidity the total heat will be 48.1 Btu. If the plant is designed for 500 tons of refrigeration and 3 gal/ton/min the weight of air required per minute will be

$$W = \frac{1500 \times 8\frac{1}{2} \times (90 - 80)}{48.1 - 38.5} = 13,050 \text{ lb}$$

Taking the specific volume of the entering air as 13.88 sq ft and the velocity of the air in the tower as 700 fpm, and assuming that 0.5 of the area of the tower is free area, the cross-sectional area is

$$\frac{13,050 \times 13.88 \times 2}{700} = 518 \text{ sq ft}$$

Example. In Fig. 20·1 is there any advantage in subcooling the liquid refrigerant by means of liquid refrigerant in the accumulator at the evaporation temperature? Would there be an advantage in cooling this liquid by means of a limited amount of cold water? If liquefaction occurs at 90° F and evaporation is at 15° F (using ammonia as the refrigerant and dry saturated vapor at the beginning of compression), what per cent improvement will be obtained by the use of a subcooler to cool the liquid to 60° F by deep-well water?

Answer. 7.18 per cent.

Example. A refrigeration load near Omaha, Nebraska, used deep-well water, and it was decided to allow the water to rise 25° F in passing through the condenser. Find the weight of water required per hour, the size of a twin vertical single-acting ammonia compressor operating at 300 rpm, and the probable horsepower of the compressor motor, allowing for the thermal volumetric efficiency. The refrigerating load is 100,000 Btu/hr at 0° F.

Answer. 5¼ in. by 5¼ in., 9.56 hp, 1.16 gpm/ton.

Example. Ten thousand feet of 2-in. pipe are arranged in 10 coils in parallel containing brine which enters at 0° and leaves at 4° F. Find the minimum loss of head of the brine if the total loss is twice that of the 2-in. pipe coils and if the loss due to fittings is 20 per cent of that of the pipes. Find the horsepower of the brine pump. The room temperature is 10° F.

Answer. 3.84 ft, 0.14 hp, allowing for pump and motor losses.

TABLE 19-1
 SPRAY POND AND COOLING TOWER PERFORMANCE

Temperature Water Leaving Spray Pond at Various Wet-Bulb Temperatures and Ranges

Cooling Range, °F	Wet-Bulb Temperature, °F				
	60°	65°	70°	75°	80°
5	65.0	70.0	74.0	78.5	83.0
6	65.5	70.5	74.5	79.0	83.5
7	66.0	71.0	75.0	79.5	84.0

Temperature Water off Cooling Tower at Various Wet-Bulb Temperatures and Ranges

1½ gpm/sq ft. Active tower surface					2¼ gpm/sq ft. Active tower surface					
Wet-bulb temperature, °F					Wet-bulb temperature, °F					
Cooling range, °F	60°	65°	70°	75°	80°	60°	65°	70°	75°	80°
10	66.0	70.2	74.4	78.6	83.0	70.7	74.3	77.9	81.6	85.3
9	65.6	69.8	74.1	78.4	82.8	70.1	73.7	77.3	81.1	84.9
8	65.0	69.4	73.7	78.1	82.5	69.3	73.0	76.7	80.6	84.5
7	64.4	68.9	73.2	77.7	82.1	68.4	72.2	76.0	80.0	84.0

PROBLEMS

1. Cooling water enters a subcooler at 70° F and leaves at 74° F; it then enters the condenser and undergoes a temperature rise to 80° F. If the saturation temperature in the condenser is 5° F greater than the mean temperature of the condenser cooling water, determine the condenser pressure (ammonia as refrigerant). Assume saturated vapor going to the compressor and evaluate the cop.

2. For the system of Problem 1 assume that the subcooler is disconnected and the same flow rate of water enters the condenser with initial temperature of 70° F. (a) If the temperature drop from condenser to mean water temperature remains constant at 5° F determine the condenser pressure. (b) Evaluate the cop of the system.

3. Compare the results from Problems 1 and 2 with the example given in the text and discuss.

4. Taking the cost of power at \$0.02 per hp-hr, plot the most economical water rate per ton of refrigeration (for standard conditions and for $e_1 = 0.85$) as a function of water cost over the range from \$0.03 per 1000 gal to \$0.20 per 1000 gal.

5. Taking the cost of water at \$0.10 per 1000 gal, plot the most economical water rate per ton of refrigeration (for standard conditions and $e_1 = 0.85$) as a function of power cost over the range from \$0.005 per hp-hr to \$0.04 per hp-hr.

CHAPTER XX

EVAPORATORS; BRINE FLOW; PIPING

When a volatile liquid and its vapor are in thermodynamic equilibrium the system comprises one component, which is the pure liquid, and two phases, liquid and vapor. With such a substance in static equilibrium, in an isothermic condition, the number of liquid molecules which escape from the liquid and enter the vapor phase during a certain period of time is just balanced by an equal number of molecules which return to the liquid phase. If, on account of certain conditions, the molecules leaving the liquid are more numerous than those that return to the liquid, evaporation is said to occur, and if sufficient heat is supplied to maintain constant the average kinetic energy of the molecules a constant rate of evaporation will take place. When the average kinetic energy is constant, the liquid is said to be in a condition of *dynamic equilibrium* and the temperature of the liquid may be considerably greater than the liquid temperature for static equilibrium under the same vapor pressure. It is obvious that the attractive forces between the molecules of a liquid are responsible for the relatively small volume occupied by the liquid. In the absence of such an attractive force the pressure required for such a compression is called the internal pressure.

Jakob and Fritz* show that at a given temperature the vapor pressure for a concave liquid surface, corresponding to the surface of a steam bubble in water, is less than the vapor pressure for a flat liquid surface. In consequence, for a given vapor pressure a liquid must be hotter to evaporate into a small bubble of vapor than to evaporate into the vapor space above the surface of the liquid, and this vapor pressure decreases as the size of the bubble decreases. Hence it would seem that ebullition is retarded until some favorable factor occurs, such as rough surfaces, the action of the expansion valve incidental with high liquid velocity and initial superheat, or the presence on the surface of something which stimulates the formation of a bubble.

If a liquid is gradually heated from the outside it is frequently possible to *raise the temperature* considerably above the boiling temperature, as much as 200° F in the case of water, according to Washburn.† Such a

* Jakob and Fritz, *Forsch. Gebiete Ing.*, Vol. 434, pp. 2, 282-288, 1931.

† Washburn, *Principles of Physical Chemistry*, 2nd ed., p. 77.

condition, where the liquid is *superheated*, is one of very unstable equilibrium which may occasion sudden vaporization with explosive violence known as "bumping." Superheating may be materially reduced or even prevented by having some of the vapor phase in contact with the liquid at the point where the heat is applied. This can be accomplished by means of an *ebullator* of porous material, like porous porcelain or wood fiber.‡ In such a case the small capillaries become filled with vapor, and boiling proceeds quietly from the point of contact of the liquid with the vapor. In general, when any phase, whether liquid, gas or solid, reaches a condition where it ought normally to change into another state of aggregation or another phase this change does not take place immediately but may be preceded by supercooling or superheating.

The second step in ebullition is the growth of the vapor film into a bubble large enough to break away from the solid, and this increases with the ease with which the liquid wets the surface of the solid and the resulting smaller area of contact. Superheating of the vapor in a bubble is always present, but it is reduced with decreased size of the bubble. Bošnjaković§ has worked out a theory for dynamic evaporation in which he shows that the amount of superheat is given by

$$\Delta t = \frac{3600Lw}{hv} \frac{dq}{dL} \quad (20 \cdot 1)$$

where L is the latent heat of vaporization, h is the coefficient of heat transfer from liquid to vapor at the surface of the metal, w is the velocity of rise of the bubble, v is the specific volume of dry saturated vapor, and dq/dL is the rate of increase of bubble size with distance from the heating surface. Thus it would appear that the amount of superheat varies directly as the latent heat of vaporization and inversely as the specific volume.

20·1. Evaporators. In evaporation the operation of heat transfer is very different from that in all other cases. When boiling starts, the action is such as to clear the surface and the volatile liquid film momentarily disappears incidentally with a high rate of heat transfer. The bubbles formed are not uniform but appear to rise from separate favored points on the surface, and these points appear to shift back and forth with the passing of time. In refrigeration with the use of pipe surfaces, heat transfer is improved, probably by the scouring action of bubbles, which, unless the pipes are too large, act as alternate pistons of liquid and vapor. However, it is very evident that the design of the heat transfer surface is an extremely important matter, as one sees on reviewing the change

‡ Phillip and Tiffany, "Ebullition of Refrigerants," *Refrig. Eng.*, March, 1933.

§ F. Bošnjaković, *Forsch. Gebiete Ing.*, Vol. 3, No. 135, p. 270, 1932.

from the long trombone piping so often used in ice-making tanks and in cold storage to the use of the shell-and-tube brine cooler, the York trunk system with V-shaped connecting pipes, and the so-called "boilers" used by the designers of household refrigerators.

In the case of a single (e.h. $1\frac{1}{4}$ -in. pipe about 8 ft long) inclined evaporator under a constant liquid head of ammonia|| the average value

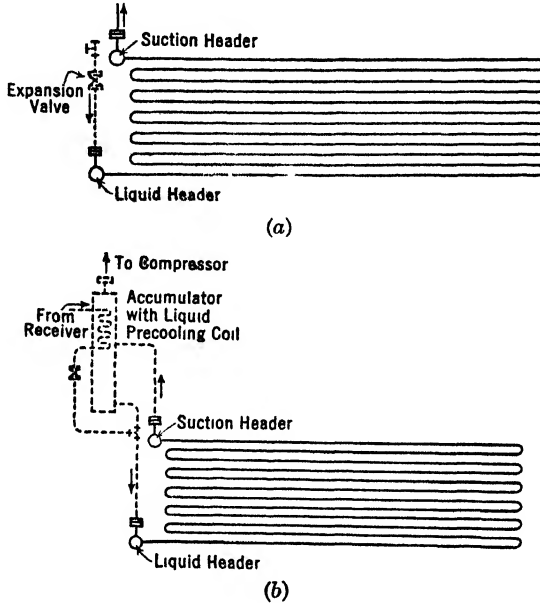


FIG. 20-1. Old evaporator design. (a) Simple coil arrangement. (b) Evaporator using the accumulator for liquid subcooling.

of U varied from 224 to 688 Btu per square foot per degree difference per hour. The average water velocity varied from 1.01 to 9.45 fps. With vigorous boiling the ammonia film resistance is practically zero.

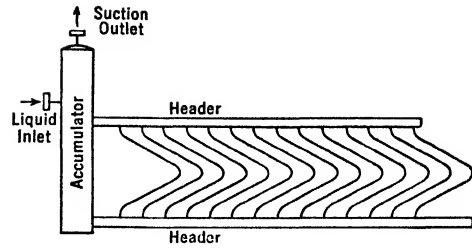
The early designers of evaporators neglected to consider the factors that determine heat transfer. These include practically stagnant fluid films, either liquid or gas, on the two sides of the metal surface and other resistances due to scale, oil, or dirt accumulations. They also neglected especially that feature in the design of the evaporator which would permit a ready removal of the vapor after the evaporation of the liquid. The fact that the absorption of heat by the refrigerant is due to the evaporation only has been ignored too frequently. It is also true that the vapor must be removed as quickly as it is formed so as to permit more liquid to evaporate, and this fact likewise has not been given

|| Graduate thesis, University of Illinois, 1933.

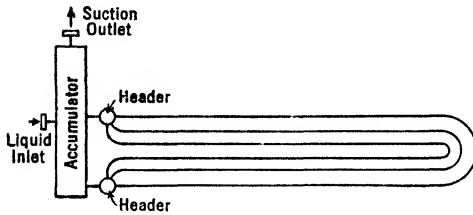
proper consideration. For example, the standard ice tank construction in 1920 was the trombone pipe, which was specified to be 14 pipes high on 5-in. return bends, collapsed, and 50 ft long. The vapor formed by evaporation at the end near the expansion valve would travel 700 ft before it got out (Fig. 20·1). If 10 per cent by weight of the liquid refrigerant is vaporized during the throttling action at the expansion valve, the volume of the gas after the pressure drop from 169 to $34\frac{1}{4}$ lb will be about 350 times the volume of the liquid from which it was vaporized. All this vapor will have to travel through the entire length of the pipe before it gets out and will absorb no heat during the process. To make matters worse, it was frequently the custom to feed the liquid in at the top with the suction at the bottom, and to feed both the top and the bottom with so little liquid as to keep a large part of the piping non-effective as a result of the absence of the liquid.

The first step toward an economical design came with the so-called *flooded system*, which meant that the liquid feed was heavy enough to permit liquid to cover all the heat transfer surface. In order to protect the compressor, a vapor separator, or accumulator, was placed in the return header with a means of evaporating the trapped liquid in a suitable manner. The next step was to make the connection between the liquid header and the suction header as short as possible; good examples of which are shown in Figs. 20·2, 20·3, and 20·4. The whole matter can be compared with advantage to the modern water tube boiler construction as regards the tubes and the manner of securing dry vapor. A prompt removal of the vapor, and velocity of the liquid on the two sides sufficient to remove appreciably the two films and the presence of clean metal, are the requirements for high values of the coefficient of heat transfer.

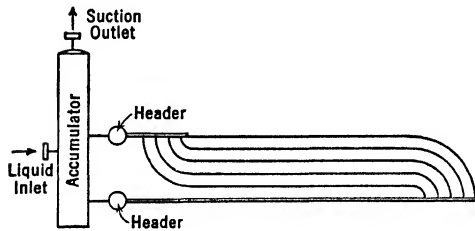
20·2. Brine Cooling. For general cooling of brine both the double pipe, with the refrigerant in the annular space, and the tube brine cooler, with the brine in the tubes, are used, but the double-pipe brine cooler is no longer popular except for carbonic refrigeration. The shell and tube cooler, with the tubes expanded into the tube sheets, very similar to the construction of the horizontal return tubular boiler, is a good example of a correct design for an evaporator (Figs. 20·5a and 20·5b). The gas has a direct path out to the suction header, and there is enough *vapor-releasing surface* to insure freedom from heavy priming in the return to the compressor. It is always well to keep in mind the effect of static head, and in this case the pressure at the bottom may be 2 or more pounds greater than at the top, and a higher boiling temperature of the refrigerant results. If more than one brine pass is used the brine should be brought in at the bottom.



(a)



(b)



(c)

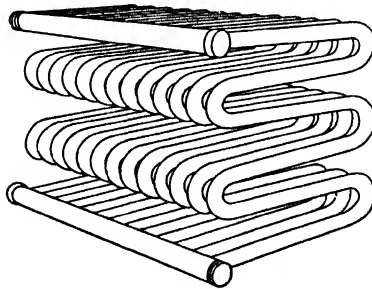


FIG. 20·2. Recent evaporator design using short connections between headers. (a) The herringbone coil used with high brine velocities. (c) Design with short pipe connection between headers.

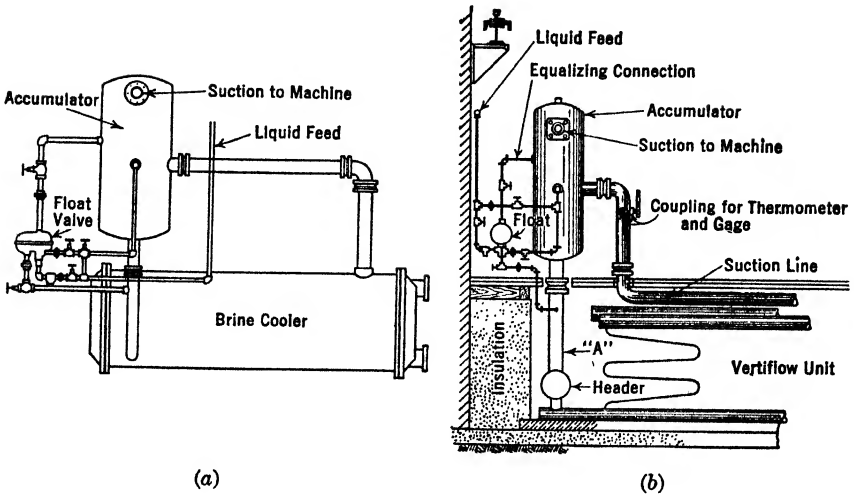


FIG. 20-3. Float valve control on the evaporator. (a) Liquid feed to a brine cooler. (b) Liquid feed to an ice tank evaporator. (See Fig. 17-4 for the construction of the float valve.)

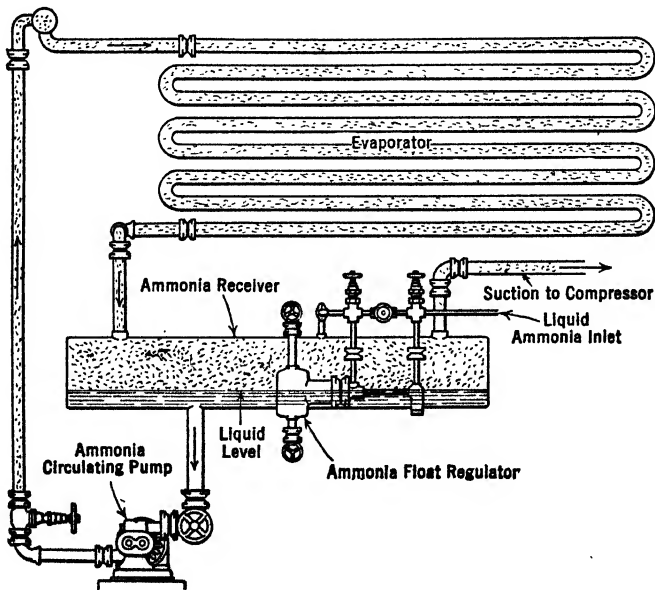


FIG. 20-4. Downward feed through the evaporator. (See Fig. 16-20 for a similar arrangement using a pump to circulate the refrigerant as here.)

Attempts have been made to use cast radiator sections for ammonia with direct expansion. The main difficulty has been in regard to the cost of securing tight sections and the proper arrangement of headers for feeding the liquid and removing the vapor. There must be pro-

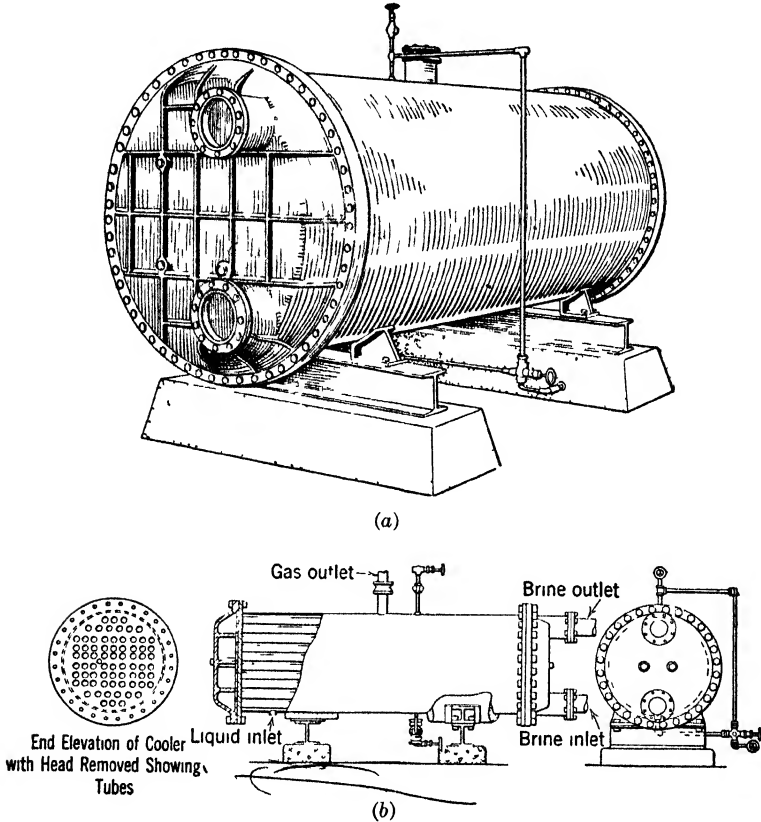


FIG. 20-5. The shell-and-tube brine cooler. (a) Isometric view. (b) Section of a cooler designed for eight passes.

vision for expansion and end contraction, and the assembly must be carefully executed because there is no appreciable flexibility while the flanges are being bolted up.

Whether to use *brine* or *ammonia* direct expansion has to be decided in many cases. For district cooling, brine has the preference, as losses of brine can be safeguarded to an extent impossible with ammonia. In cold storage warehouses, brine has the preference because of the decreased chance of damage to the goods in storage in case of leaks.

TABLE 20-1
SODIUM CHLORIDE — NaCl
Specific Heat¹ (Temperature Table)
Jesup, "Properties of Refrigerating Brines," *Refrigerating Engineering*, December, 1925

Degrees		Specific Heats of Various Per Cent NaCl Solutions at Various Temperatures																					
F	C	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24	25	
-4	-20																			0.784	0.784	0.779	
+14	-10																			0.780	0.784	0.783	
32	0																			0.793	0.788	0.783	
50	+10																			0.796	0.791	0.786	
68	20																			0.803	0.798	0.793	
86	30																			0.804	0.799	0.794	
Per Cent of NaCl.....		5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24	25	
Specific gravity ²		1.036	1.044	1.051	1.058	1.066	1.073	1.081	1.088	1.096	1.104	1.112	1.119	1.127	1.135	1.143	1.151	1.159	1.167	1.176	1.184	1.192	
Degrees Baumé.....		5.08	6.06	7.04	8.00	8.96	9.91	10.85	11.78	12.71	13.63	14.54	15.45	16.35	17.25	18.14	19.02	19.90	20.78	21.65	22.51	23.37	
Degrees F., Freezing Pt., Degrees F.....		28.7	25.5	24.2	22.9	21.6	20.2	18.8	17.3	15.7	14.1	12.4	10.6	8.7	6.7	4.6	2.4	0.0	-2.5	-5.2	+1.4	+13.3	
Per Cent of Sodium Chloride and Corresponding Freezing Points (Specific Gravity Table)																							
Specific gravity ²		1.03	1.04	1.05	1.06	1.07	1.08	1.10	1.11	1.12	1.13	1.14	1.15	1.16	1.17	1.18	1.19	1.20					
Per cent NaCl.....		4.14	5.51	6.86	8.21	9.56	10.88	12.20	13.51	14.81	16.09	17.36	18.63	19.87	21.11	22.33	23.54	24.74	25.93				
Degrees F., Freezing Pt., Degrees F.....		27.6	26.0	24.4	22.7	20.8	19.0	17.0	14.9	12.7	10.4	8.0	5.4	2.7	-0.3	-3.5	-9.0	+9.0	+27.0				
Per Cent of Sodium Chloride and Corresponding Freezing Points (Baumé Density Table)																							
Degrees Baumé.....		5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23			
Per cent NaCl.....		4.92	5.94	6.96	8.00	9.04	10.10	11.16	12.23	13.31	14.40	15.50	16.61	17.72	18.84	19.97	21.11	22.26	23.41	24.57			
Degrees F.—F.....		26.7	25.5	24.3	22.9	21.6	20.1	18.6	16.9	15.2	13.4	11.5	9.4	7.3	5.0	2.4	-0.3	-3.2	-5.0	+8.0			

¹ Specific heat expressed in 20° Cal.-g., degrees C.
² Specific gravity, based on 60° F. water and 60° F. brine; degrees B₄, Baumé density; degrees F.—F. P., freezing point.

TABLE 20·2
CALCIUM CHLORIDE — CaCl₂ (PURE)
Specific Heat¹ (Temperature Table)
J Jessup, "Properties of Refrigerating Brines," *Refrigerating Engineering*, December, 1925

Degrees F	Degrees C	Specific Heats of Various Per Cent Calcium Chloride Solutions at Various Temperatures																														
-40	-40	0.882	0.867	0.853	0.839	0.825	0.812	0.799	0.787	0.775	0.763	0.752	0.741	0.731	0.721	0.711	0.702	0.693	0.683	0.673	0.670	0.663	0.656	0.649	0.642	0.635	0.631					
-22	-20																															
-4	-10																															
14	0																															
32	0																															
50	10	0.887	0.872	0.858	0.844	0.831	0.818	0.805	0.793	0.781	0.770	0.759	0.748	0.738	0.728	0.718	0.709	0.700	0.692	0.683	0.676	0.669	0.662	0.655	0.648	0.641	0.634	0.627	0.620	0.613	0.606	
68	20	0.892	0.877	0.863	0.849	0.835	0.823	0.811	0.799	0.787	0.775	0.764	0.754	0.744	0.734	0.724	0.715	0.706	0.698	0.690	0.682	0.675	0.668	0.661	0.654	0.647	0.640	0.633	0.626	0.619	0.612	
86	30	0.897	0.882	0.868	0.855	0.842	0.829	0.817	0.805	0.792	0.780	0.769	0.759	0.749	0.739	0.729	0.721	0.712	0.704	0.696	0.689	0.682	0.675	0.668	0.661	0.654	0.647	0.640	0.633	0.626	0.619	
Per cent of CaCl ₂ ,.....		8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24	25	26	27	28	29	30								
Specific gravity ² ,.....		1.069	1.078	1.087	1.096	1.105	1.114	1.121	1.133	1.143	1.152	1.162	1.172	1.182	1.192	1.202	1.212	1.223	1.233	1.244	1.254	1.265	1.276	1.287								
Degrees Baumé,.....		9.33	10.45	11.57	12.68	13.78	14.86	15.96	17.04	18.12	19.20	20.24	21.28	22.32	23.35	24.38	25.38	26.41	27.41	28.41	29.41	30.39	31.37	32.34								
Freezing point,.....		24.2	22.8	21.4	19.8	18.2	16.3	14.4	12.2	9.9	7.4	4.7	1.9	-1.0	-4.0	-7.3	-10.6	-14.1	-18.0	-22.0	-27.0	-32.0	-39.0	-46.0								

Per Cent of Calcium Chloride and Corresponding Freezing Points (Specific Gravity Table)		Per Cent of Calcium Chloride and Corresponding Freezing Points (Baumé Density Table)																					
Specific gravity ² ,.....	1.07	1.08	1.09	1.10	1.11	1.12	1.13	1.14	1.15	1.16	1.17	1.18	1.19	1.20	1.21	1.22	1.23	1.24	1.25	1.26	1.27	1.28	1.29
Per cent of CaCl ₂ ,.....	8.14	9.26	10.36	11.46	12.53	13.60	14.66	15.71	16.75	17.78	18.80	19.81	20.81	21.79	22.78	23.74	24.70	25.65	26.59	27.52	28.44	29.35	30.26
Freezing point, degrees F....	24.0	22.5	20.8	19.1	17.2	15.1	13.0	10.6	8.1	5.4	2.6	-0.4	-3.4	-6.4	-9.7	-13.2	-16.8	-20.7	-25.0	-30.0	-35.0	-41.0	-48.0

Per Cent of Calcium Chloride and Corresponding Freezing Points (Baumé Density Table)		Per Cent of Calcium Chloride and Corresponding Freezing Points (Baumé Density Table)																					
Degrees Baumé,.....	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24	25	26	27	28	29	30	31	32
Per cent of CaCl ₂ ,.....	8.59	9.49	10.39	11.29	12.20	13.11	14.04	14.96	15.89	16.83	17.78	18.73	19.69	20.66	21.63	22.62	23.60	24.59	25.59	26.59	27.61	28.62	29.64
Freezing point, degrees F....	23.4	22.1	20.8	19.3	17.8	16.1	14.3	12.3	10.1	7.8	5.3	2.7	-0.1	-3.0	-6.1	-9.3	-12.7	-16.4	-20.4	-25.0	-30.0	-36.0	-43.0

¹ Specific heat expressed in 20° Cal-g., degrees C.
² Specific gravity, based on 60° F water and 60° F brine; degrees B_e, Baumé density; degrees F — F. P., freezing point.

The temperature can be maintained more uniformly with brine, and the dripping due to melting frostation is unlikely where suitable spare circulating pumps are installed. Certain installations, like the cooler rooms off the killing floor of packing plants, require brine sprays.

Brine lines have to be insulated to an amount depending on the temperature being carried, whereas frequently the liquid and occasionally the suction return ammonia lines are not insulated (although this neglect is not to be recommended). The first cost of brine systems is frequently greater than that of ammonia systems because of the duplication of process: first to cool the brine and then to have it perform useful cooling. Brine lines are usually laid out for 5 to 7 fps, and ammonia return lines for 40 to 70 fps velocity of the fluid, both depending on the length and size of the pipe. With brine the cost of pumping may assume large values.

Whether to use calcium or sodium chloride brine depends on the type of work being handled. Calcium chloride can be used for temperatures below 0° F, and it is used in nearly all cases, except that sodium brine sprays are employed in packing plants. Both sodium and calcium brine should be free of other salts in quantities greater than a fraction of 1 per cent. To prevent corrosion the brine should be slightly alkaline. Tables 20·1, 20·2, and 20·3 are taken from the publications of the Bureau of Standards.

Quick calculations for the loss of head of brine flowing through commercial pipe lines may be accomplished through use of Figs. 15·4 and 15·5. Figure 15·5 gives multiplying factors in terms of the loss of head for water flowing through the same pipes at the same velocity. The loss of head for the brine is the product of the loss of head of water (using Fig. 15·4), and the multiplying factor given in Fig. 15·5.

Example. A brine coil of 10,000 lineal feet of 2-in. pipe is located on the tenth floor of a warehouse. The brine circuit consists of the following: The refrigerating piping includes 10,000 lineal feet of 2-in. pipe in 10 coils, the distance from the brine cooling coils to the riser is 108 lineal feet, the length of riser, 121 lineal feet, and the length of pipe from the brine cooler to the riser, 75 lineal feet, the supply pipes being a total of 608 ft of 3-in. pipe.

The cooler is to be maintained at 32° F, and the brine will enter at 18° and leave at 22° F. The value for U for the coils will be taken as 1.7 Btu/hr. The amount of brine to be circulated, the friction head loss, and the horsepower of the pump (assuming an overall efficiency of 50 per cent) are desired.

Solution. The amount of brine to be circulated may be found by the amount of cooling surface. The heat absorbed by the brine is

$$Q = \frac{10,000}{1.6} \times 1.7 \left(32 - \frac{22 + 18}{2} \right) = 127,500 \text{ Btu/hr}$$

where the factor 1.6 is the number of feet of 2-in. pipe per 1 sq ft of outside

surface. Referring to the tables for the properties of calcium chloride brine,

$$\begin{aligned} \text{Weight of brine} &= \frac{127,500}{0.76 \times 4} = 41,940 \text{ lb} \\ &= \frac{62.4 \times 1.15}{41,940} = 584.4 \text{ cu ft/hr} \\ &= 72.85 \text{ gpm} \end{aligned}$$

From the chart, Fig. 15·4, it will be seen that at a velocity of 3.3 fps a 3-in. pipe will have a pressure drop, per 100 ft of pipe, of 1.6 ft of water, and the friction head loss in the cooling coils (7.3 gpm in the 2-in. pipe) will be 0.17 ft per 100 ft. The total friction head loss is then, using the multiplying factor of 1.3 from Fig. 15·5,

$$\begin{aligned} 0.17 \times 1.3 \times 10 &= 2.21 \text{ ft of water} \\ 1.6 \times 1.3 \times 6.08 &= 12.65 \text{ ft of water} \\ \text{Total} &= 14.86 \end{aligned}$$

Adding 20 per cent for fittings = 17.83 ft

$$\text{Work performed} = \frac{41,940}{60} \times 17.83 = 12,463 \text{ ft-lb/min}$$

$$\begin{aligned} \frac{12,463}{33,000} &= 0.3776 \text{ hp for circulating brine} \\ &= 0.755 \text{ hp (assuming 50 per cent efficiency} \\ &\quad \text{of the pump)} \end{aligned}$$

20·3. Brine Piping. In *brine piping* nothing is to be gained by long coils. The velocity should be great enough to avoid viscous flow, and the rise of temperature should be kept less than 5° F. With values of U of 2 to 2½ Btu/per (sq ft) (° F) it is easy to calculate the minimum rate of brine flow. As a rule the brine velocity should be less than 5 fps from the consideration of the friction involved.

A brine pipe line should be arranged so that all the points that may trap air can be vented by means of pet cocks. It should be assembled with stop, throttling, and drain valves. The brine line system may be designed for two- and three-pipe main systems.

The two-pipe system, one for supply and the other for return, is suitable for small installations and particularly for low-pressure heads, say for not more than 3 or 4 stories. It is a simple pipe system.

The three-pipe main (Fig. 20·6) is by far the better for high buildings. The separate pipe coils on each floor receive brine from the supply pipe and deliver to the second pipe which in turn continues upward to the upper floor that is refrigerated, where connection is made to a balance tank. The third pipe is a common return. The advantages of the three-pipe system are: (a) Air is vented through the balance tank; (b) it gives opportunity for expansion and contraction with the change of

temperature; (c) it gives a balanced system, whereby the only load on the brine-circulating pump is that due to the velocity head and friction, regardless of height; (d) the loss of brine is easily determined; (e) the temperature of the brine supplied to each coil is uniform.

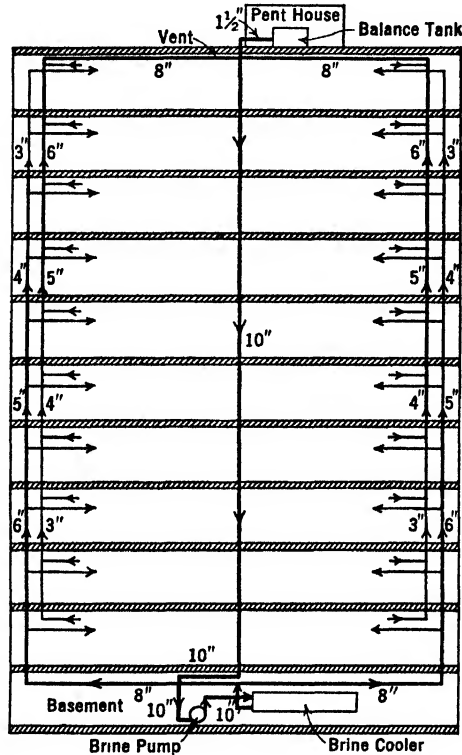


FIG. 20-6. The three-pipe main system for brine.

Where two-temperature cold storage rooms are maintained, and when only one brine system is desired, the *cascade* method whereby the brine leaving the freezer is sent to the cooler coils, is feasible. Automatic by-pass valves would be required to supply brine to the cooler coils should the freezer coils be throttled or shut down.

PIPING AND FITTINGS

Because of the nature of the refrigeration cycle, piping is a very essential part of the installation. Low-temperature piping is designed both for direct expansion of the refrigerant and for brine. Refrigeration piping must be laid out with regard to the laws of the flow of fluids

(Chapter XV); the pipes must be tight against even traces of leaks (Chapter XVIII); and they must be arranged properly as regards: (a) the elimination of gas pockets and slugs of liquid returning to the compressor, (b) the necessary amount of heat transfer surface, (c) the removal of oil and scale, and (d) the variation of length with the change of temperature.

20·4. Kind of Piping. For different kinds of work in refrigeration engineering choice must be made of the appropriate piping material—wrought iron, steel, brass, or copper. *Wrought iron* and *steel* pipe are

TABLE 20·4
STEEL PIPE BENDS, HAIRPIN AND COILS—LIMITING DIMENSIONS
(Figures are distance on centers across U-bend, in inches.)

Nominal pipe size ¹ (Standard or extra heavy)	¼		⅜		½		¾		1		1¼		2	
Usual diameter, in.	2½	2	3	2½	3	2½	3½	3	4	3	5	4	12	8
Limiting diameter, in.	2	1¾	2½	2¼	2½	2¼	3	2¾	3½	2¾	5	4	6	6
Nominal pipe size ²	2½		3		3½		4		4½		5		6	
Usual diameter, in.	20		32		40		44		48		54		60	
Limiting diameter, in.	10		16		20		24		28		32		40	

¹ Usually cold rolled; sometimes hot or requiring filling with sand.
² Usually hot rolled; Bessemer steel used mainly.

identified with the use of ammonia, whereas copper and compositions are possible with a number of refrigerants in the fractional-tonnage machines where it generally is important to have flexible pipe connections. Wrought iron has been used extensively, and is still favored by some refrigerating engineers, but the tendency is toward full-weight, black, steel pipe which is cheaper in first cost than wrought iron and can also be bent cold in pipe-bending machines in the smaller sizes of pipe. The minimum recommended diameters for pipe bends are given in Table 20·4. Butt-welded pipe is usual in the smaller sizes, as shown in Tables 20·5 and 20·6, but lap-welded pipe is preferred for receivers, traps, accumulators, and shell condensers and coolers. The strength of pipe under bursting pressure is given by the usual formula for thin-walled cylinders:

$$p = \frac{2ft}{d} \text{ pounds per square inch}$$

where *d* is the outside diameter in inches; *t* is the thickness of the wall in inches; *f* is the allowable tensile strength, about 10,000 psi.

Seamless copper tubing is used for the fractional-tonnage as well as for the small commercial machines, for sulphur dioxide, methyl chloride, and the halide refrigerants for evaporators, condensers and tubing connections. The tube wall is not less than 0.035 in. thick if the pipe connection is made by flaring the tube ends and using flared tube fittings. Tables 20·8 and 20·9 give dimensions of copper tubing.

TABLE 20·5
STANDARD WROUGHT PIPE
(Welded Wrought Iron and Steel, and Seamless Steel)

Diameter			Thick- ness, in.	Trans- verse Internal Area, in. ²	Length of Pipe per Ft ²		Weight per Foot Length, lb	Length of Pipe Containing 1 Ft ³	No. Threads per Inch
Nomi- nal, in.	External, in.	Internal, in.			External Surface, ft	Internal Surface, ft			
½	0.405	0.269	0.068	0.057	9.431	14.15	0.25	2513.0	27
⅝	0.540	0.364	0.088	0.104	7.073	10.49	0.43	1383.3	18
¾	0.675	0.493	0.091	0.191	5.658	7.73	0.57	751.2	18
⅞	0.840	0.622	0.109	0.304	4.547	6.13	0.85	472.4	14
1	1.050	0.824	0.113	0.533	3.637	4.635	1.13	270.0	14
1 ¼	1.315	1.049	0.133	0.864	2.904	3.645	1.68	168.9	11½
1 ½	1.660	1.380	0.140	1.495	2.301	2.768	2.28	98.25	11½
1 ¾	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
2 ¼	2.875	2.469	0.203	4.788	1.328	1.547	5.82	30.10	8
2 ½	3.500	3.068	0.216	7.393	1.091	1.245	7.62	19.50	8
3	4.000	3.548	0.226	9.886	0.954	1.077	9.20	14.57	8
3 ¼	4.500	4.026	0.237	12.730	0.848	0.949	10.89	11.31	8
3 ½	5.000	4.506	0.247	15.947	0.763	0.848	12.64	9.02	8
4	5.563	5.047	0.258	20.006	0.686	0.757	14.81	7.20	8
4 ½	6.625	6.065	0.280	28.891	0.576	0.630	19.19	4.98	8
5	7.625	7.023	0.301	38.738	0.500	0.543	23.77	3.72	8
5 ½	8.625	8.071	0.277	51.161	0.442	0.473	25.00	2.81	8
6	8.625	7.981	0.322	50.027	0.442	0.478	28.81	2.88	8
6 ½	9.625	8.941	0.342	62.786	0.396	0.427	34.19	2.29	8
7	10.750	10.192	0.279	81.585	0.355	0.374	32.00	1.76	8
7 ½	10.750	10.136	0.307	80.691	0.355	0.376	35.00	1.78	8
8	10.750	10.020	0.365	78.855	0.355	0.381	41.13	1.82	8
8 ½	11.750	11.000	0.375	95.033	0.325	0.347	46.25	1.51	8
9	12.750	12.090	0.330	114.800	0.299	0.315	45.00	1.25	8
9 ½	12.750	12.000	0.375	113.097	0.299	0.318	50.71	1.27	8

¹All weights subject to 5 per cent variation. Mill test pressures, butt welded up to 2 in., 700 lb per in.²; lap welded 1½ to 9 in., 1000; 10-12 in., 600 to 800.

20·5. Length of Piping per Coil. In general, a direct expansion system should have the radiator type of design. During the evaporator process the liquid should be continually and immediately separated from its vapor as is seen in the shell-and-tube brine cooler and the ice-tank details (Figs. 20·4 and 20·7, which show the trunk, the verti-flow, and the supercirculation coil systems). In the condenser the opposite action is desired, namely, to remove the liquid condensate as quickly and as effectively as possible. As the *coefficient of expansion* of wrought steel is approximately 0.000075, in the range of temperature usually found in refrigeration the increase of length of the pipe line may be found by the

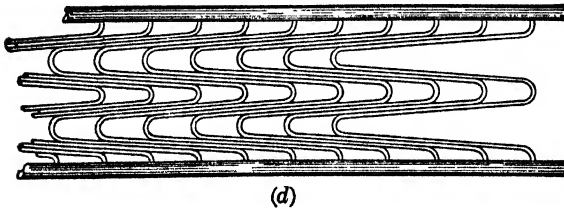
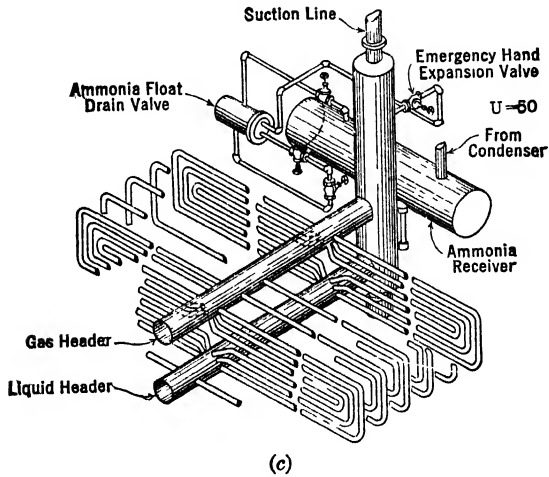
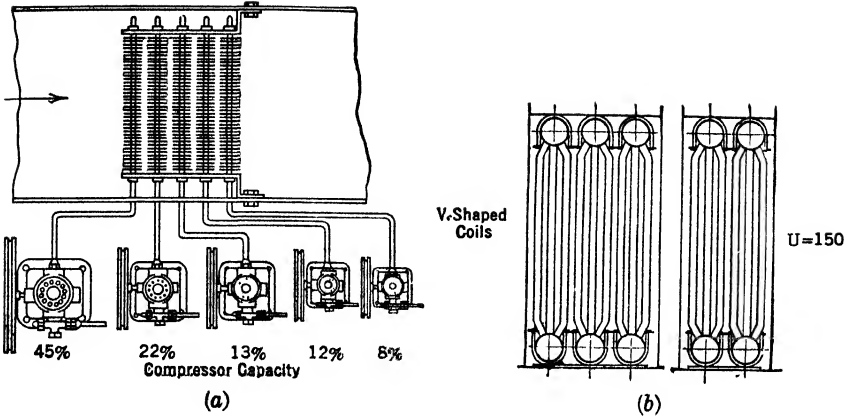


FIG. 20-7. Modern evaporators.

formula

$$\Delta L = 0.0000075L \Delta t$$

where L is the length of the pipe in inches, ΔL is the change in length of the pipe in inches, and Δt is the difference in pipe temperature.

TABLE 20-6
EXTRA STRONG PIPE
(Welded Wrought Iron and Steel and Seamless Steel)

Diameter			Thickness, in.	Transverse Internal Area, in. ²	Length of Pipe per Ft ²		Weight per Foot Length, lb	No. Threads per Inch
Nomi- nal, in.	External, in.	Internal in.			External Surface, ft	Internal Surface, ft		
1/8	0.405	0.215	0.095	0.036	9.431	18.632	0.31	27
1/8	0.540	0.302	0.119	0.072	7.073	12.986	0.54	18
1/8	0.675	0.423	0.126	0.141	5.658	9.070	0.74	18
1/8	0.840	0.546	0.147	0.234	4.547	7.048	1.09	14
1/8	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	11 1/2
1 1/8	1.660	1.278	0.191	1.283	2.301	3.003	3.00	11 1/2
1 1/8	1.900	1.500	0.200	1.767	2.010	2.556	3.63	11 1/2
2	2.375	1.939	0.218	2.953	1.608	1.975	5.02	11 1/2
2 1/2	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
3 1/2	4.000	3.364	0.318	8.888	0.954	1.137	12.51	8
4	4.500	3.826	0.337	11.497	0.843	1.000	14.98	8
4 1/2	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

¹ Weights subject to variations of 5 per cent. 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.

Where direct expansion of ammonia is employed it has been customary to use, at 0° F evaporating temperature, per expansion valve:

- 430 ft of 1/2-in. pipe
- 640 ft of 3/4-in. pipe
- 850 ft of 1-in. pipe
- 1210 ft of 1 1/4-in. pipe
- 1450 ft of 1 1/2-in. pipe
- 1950 ft of 2-in. pipe

The difficulty in all such pipe evaporators is that the vapor formed when the pressure is reduced during the throttling action, or as heat is absorbed, occupies considerable volume, in fact as much as 375 times an equal weight of liquid at 0° F. The presence of vapor in the pipe coils,

TABLE 20-7
LARGE SIZE O.D. LAP-WELDED STEEL PIPE IN VARIOUS SHELL THICKNESSES¹

O.D. in.	Thickness of Shell, inches												
	$\frac{1}{4}$	$\frac{3}{8}$	$\frac{1}{2}$	$\frac{5}{8}$	$\frac{3}{4}$	$\frac{7}{8}$	1	$1\frac{1}{8}$	$1\frac{1}{4}$	$1\frac{3}{8}$	$1\frac{1}{2}$		
14	Wt. per foot, plain ends Test pressure in lb	36.71 500	45.68 650	54.57 750	63.37 900	72.09 1000	80.73 1150	89.28 1250	97.75 1275	106.13 1300	122.65 1400	138.84 1500	154.70 1600
15	Wt. per foot, plain ends Test pressure in lb	39.38 500	49.02 600	58.57 700	68.04 800	77.43 950	86.73 1100	95.95 1200	106.09 1250	114.14 1300	132.00 1400	149.52 1500	166.71 1600
16	Wt. per foot, plain ends Test pressure in lb	42.05 450	52.36 550	62.58 700	72.72 800	82.77 950	92.74 1100	102.63 1200	112.43 1250	122.15 1300	141.85 1400	160.20 1500	178.73 1600
17	Wt. per foot, plain ends Test pressure in lb	44.72 400	55.70 500	66.58 650	77.39 750	88.11 850	98.75 950	109.30 1100	119.78 1200	130.16 1300	150.69 1400	170.88 1500	190.74 1600
18	Wt. per foot, plain ends Test pressure in lb	47.39 400	59.03 500	70.56 600	82.06 700	93.45 800	104.76 950	115.98 1050	127.12 1150	138.17 1250	160.04 1400	181.56 1500	202.76 1600
20	Wt. per foot, plain ends Test pressure in lb	65.71 450	81.41 500	97.60 550	114.11 600	130.13 650	146.77 700	163.33 750	179.83 800	196.19 850	222.73 900	249.92 950	277.11 1000
21	Wt. per foot, plain ends Test pressure in lb	69.05 450	86.08 500	103.60 550	121.10 600	138.47 650	155.78 700	173.01 750	190.15 800	207.20 850	234.35 900	261.50 950	288.65 1000
22	Wt. per foot, plain ends Test pressure in lb	72.38 450	90.71 500	109.61 550	128.51 600	147.41 650	166.31 700	185.21 750	204.11 800	223.01 850	251.91 900	280.81 950	309.71 1000
24	Wt. per foot, plain ends Test pressure in lb	94.62 500	116.10 550	137.58 600	159.06 650	180.54 700	202.02 750	223.50 800	244.98 850	266.46 900	297.94 950	329.42 1000	361.90 1050
26	Wt. per foot, plain ends Test pressure in lb	102.63 450	126.11 500	149.59 550	173.07 600	196.55 650	220.03 700	243.51 750	266.99 800	290.47 850	313.95 900	337.43 950	360.91 1000
28	Wt. per foot, plain ends Test pressure in lb	128.79 475	157.49 500	186.19 550	214.89 600	243.59 650	272.29 700	300.99 750	329.69 800	358.39 850	387.09 900	415.79 950	444.49 1000
30	Wt. per foot, plain ends Test pressure in lb	138.13 450	170.11 500	202.09 550	234.07 600	266.05 650	298.03 700	330.01 750	361.99 800	393.97 850	425.95 900	457.93 950	490.91 1000

¹ All weights indicated are subject to a variation of ± 5 per cent.

TABLE 20-8
SEAMLESS DRAWN BRASS AND COPPER TUBING
(Standard Pipe Sizes)

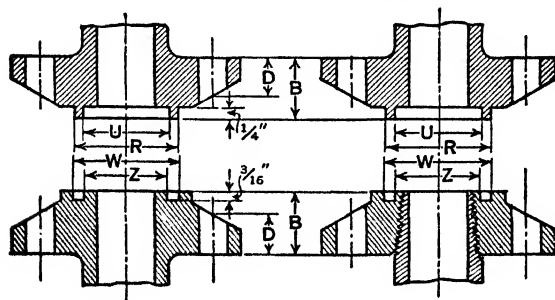
Nominal, in.	Diameter		Thick- ness, in.	Trans- verse Internal Area, in. ²	Length of Pipe per Ft ² Internal Surface, ft	Weight of Brass per Linear Foot, lb	Weight of Copper per Linear Foot, lb	No. Thrds. per Inch
	External, in.	Internal, in.						
1/8	0.405	0.281	0.062	0.062	13.60	0.246	0.259	27
1/4	0.540	0.375	0.082	0.111	10.19	0.437	0.459	18
3/8	0.675	0.494	0.090	0.192	7.73	0.612	0.644	18
1/2	0.840	0.625	0.107	0.307	6.11	0.911	0.958	14
5/8	1.050	0.822	0.114	0.532	4.65	1.235	1.298	14
1	1.315	1.062	0.126	0.888	3.60	1.740	1.829	11 1/2
1 1/4	1.660	1.368	0.146	1.472	2.790	2.557	2.689	11 1/2
1 1/2	1.900	1.600	0.150	2.020	2.388	3.037	3.193	11 1/2
2	2.375	2.062	0.156	3.350	1.854	4.017	4.224	11 1/2
2 1/2	2.875	2.500	0.187	4.928	1.529	5.830	6.130	8
3	3.500	3.062	0.219	7.378	1.247	8.314	8.741	8
3 1/2	4.000	3.500	0.250	9.640	1.091	10.85	11.41	8
4	4.500	4.000	0.250	12.566	0.956	12.29	12.93	8
4 1/2	5.000	4.500	0.250	15.950	0.849	13.74	14.44	8
5	5.563	5.062	0.250	20.200	0.755	15.40	16.19	8
6	6.625	6.125	0.250	29.498	0.624	18.44	19.39	8
7	7.625	7.062	0.281	39.271	0.541	23.92	25.15	8
8	8.625	8.000	0.312	50.265	0.478	30.05	31.60	8
9	9.625	8.937	0.344	62.918	0.427	36.94	38.84	8
10	10.750	10.019	0.366	79.020	0.337	43.91	46.17	8

TABLE 20-9
SEAMLESS COPPER TUBING

Diameter		Wt./Ft, lb	Lin Ft per Ft ² , ft	Ft ² per Lin Ft, ft ²
Ext., in.	Int., in.			
0.250	0.18	0.091	15.3	0.0654
0.375	0.305	0.143	10.2	0.0982
0.500	0.403	0.196	7.64	0.1309
0.625	0.555	0.249	6.13	0.1636
0.750	0.666	0.359	5.10	0.1955
0.875	0.777	0.489	4.37	0.2200
1.00	0.888	0.649	3.82	0.2610

TABLE 20-10

AMMONIA FLANGED FITTINGS AND COMPANION FLANGES

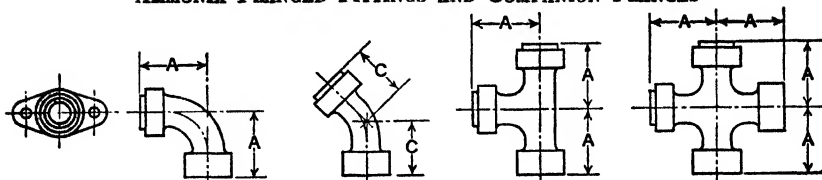


FACING DIMENSIONS

Nominal Pipe Size	Tongue			Groove			Flange Thickness, Min	
	Height	Outside Diameter E	Inside Diameter U	Depth	Outside Diameter W	Inside Diameter Z	D	B
OVAL								
1/4	1/4	1 3/8	1	3/16	1 7/16	1 5/16	3/8	1 3/16
3/8	1/4	1 3/8	1	3/16	1 7/16	1 5/16	9/16	1 5/16
1/2	1/4	1 3/8	1	3/16	1 7/16	1 5/16	5/8	1
3/4	1/4	1 11/16	1 5/16	3/16	1 3/4	1 1/4	1 1/16	1 1/8
1	1/4	1 7/8	1 1/2	3/16	1 13/16	1 7/16	1 3/16	1 1/4
1 1/4	1/4	2 1/4	1 7/8	3/16	2 5/16	1 13/16	7/8	1 3/8
SQUARE								
3/4	1/4	1 11/16	1 5/16	3/16	1 3/4	1 1/4		1 3/16
1	1/4	1 7/8	1 1/2	3/16	1 13/16	1 7/16		1 3/16
1 1/4	1/4	2 1/4	1 7/8	3/16	2 5/16	1 13/16		1 3/16
1 1/2	1/4	2 1/2	2 3/8	3/16	2 9/16	2 1/16		1 3/8
2	1/4	3 1/4	2 7/8	3/16	3 5/16	2 13/16		1 3/4
2 1/2	1/4	3 3/4	3 3/8	3/16	3 13/16	3 3/16		1 3/4
3	1/4	4 9/16	4 1/4	3/16	4 1/4	4 9/16		1 3/4
3 1/2	1/4	5 1/8	4 3/4	3/16	5 1/8	4 1/16		1 3/4
4	1/4	5 11/16	5 3/16	3/16	5 5/8	5 5/8		1 3/4
ROUND								
5	1/4	6 13/16	6 5/16	3/16	6 7/8	6 1/4		1 3/4
6	1/4	8	7 1/2	3/16	8 1/16	7 3/16		2 1/4
8	1/4	10	9 3/4	3/16	10 1/16	9 5/16		2 1/2
10	1/4	12	11 3/4	3/16	12 1/16	11 3/16		2 3/4
12	1/4	14 1/4	13 3/4	3/16	14 9/16	13 7/16		3
SPECIAL SQUARE DROP-FORGED FLANGES								
1 1/4	1/4	2 1/4	1 7/8	3/16	2 9/16	1 13/16		1
2	1/4	3 1/4	2 7/8	3/16	3 9/16	2 13/16		1 1/4

All dimensions given in inches.

TABLE 20-11
AMMONIA FLANGED FITTINGS AND COMPANION FLANGES

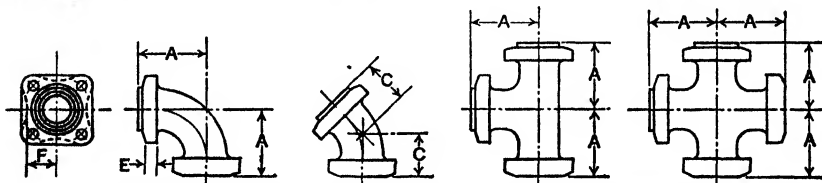


DIMENSIONS FOR OVAL FLANGED FITTINGS AND COMPANION FLANGES

Nominal Pipe Size	Inside Diameter of Fitting	Metal Thickness of Fitting, Min	Long Diameter of Flange	Short Diameter of Flange	Thickness of Flange, Min	Thickness of Flange at Bolt Circle	Center ¹ to Flange Edge, Elbow Tee and Cross	Center ¹ to Flange Edge 45° Elbow	Diameter of Bolt Circle	Number of Bolts	Size of Bolts
					B	D	A	C			
1/4	3/8	1/4	3 7/8	1 3/4	1 3/8	1/2	2 1/4	2 1/4	2 1/4	2	7/8
3/8	1/2	1/4	3 1/2	1 3/8	1 3/8	9/8	2 1/4	2 1/4	2 1/2	2	7/8
1/2	5/8	3/4	3 7/8	1 3/4	1	5/8	2 3/4	2 1/2	2 3/4	2	1 1/2
3/4	1 1/8	3/4	4 1/4	2 1/4	1 1/2	1 1/8	3	2 1/2	3	2	1 1/2
1	1 1/2	3/4	4 3/8	2 1/2	1 1/2	1 1/4	3 3/4	3	3 3/8	2	5/8
1 1/4	1 3/8	5/8	5	2 1/4	1 3/8	7/8	3 3/8	3 1/2	3 3/4	2	5/8

¹ The 1/4-in. tongue is included in the "thickness of flange, minimum," and "center to flange edge" dimensions.

The companion flanges have the same external dimensions, including the thickness, as the flanges on the fitting.



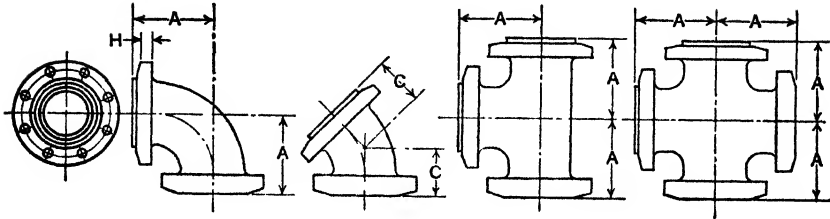
DIMENSIONS FOR SQUARE FLANGED FITTINGS AND COMPANION FLANGES

Nominal Pipe Size	Inside Diameter of Fitting	Metal Thickness of Fitting, Min	Length and Width of Flange	Thickness of Flange, Min	Thickness of Cast Iron and Forged Flanges at Edge	Center Line to Side Contour of Forged Flange	Center to Flange Edge, Elbow Tee and Cross	Center to Flange Edge 45° Elbow	Diameter of Bolt Circle	Number of Bolts	Size of Bolts
				B	E	F	A	C			
3/8	1/2	1/4	2 3/4	1 1/8	7/8	1 1/8	2 1/2	2 1/4	2 1/4	4	7/8
1/2	5/8	1/4	3	1 1/4	1 1/2	1 1/4	2 3/4	2 1/2	2 1/2	4	7/8
3/4	1 1/8	1/4	3 1/4	1 3/8	1 1/2	1 1/8	3	2 1/2	2 1/2	4	7/8
1	1 1/4	3/4	3 3/4	1 3/8	9/8	1 3/8	3 3/8	3	3 1/4	4	1 1/2
1 1/4	1 3/8	5/8	3 7/8	1 1/2	5/4	1 1/2	3 1/2	3 1/4	3 3/4	4	1 1/2
1 1/2	1 1/2	5/8	4 1/4	1 1/2	1 1/4	2	4 1/4	3 1/4	4	4	1 1/2
2	2 1/8	3/4	5	1 1/4	1 3/4	2 1/4	4 3/4	3 1/2	4 3/4	4	1 1/2
2 1/2	2 1/2	3/4	5 1/2	1 3/8	7/4	2 1/8	5 3/8	3 1/2	5 1/2	4	3/4
3	3	7/8	6 1/4	1 1/2	1 3/4	2 3/4	5 1/2	4	6 1/4	4	3/4
3 1/2	3 1/2	7/8	6 1/4	1 3/8	1	2 7/8	6 1/4	4	6 3/4	4	3/4
4	4	7/8	7 1/4	1 3/4	1 1/4	3 1/4	6 1/2	4 1/4	7 1/4	4	3/4

All dimensions given in inches.

¹ The 1/4-in. tongue is included in the "thickness of flange, minimum," and "center to flange edge" dimensions.

TABLE 20.12
AMMONIA FLANGED FITTINGS AND COMPANION FLANGES



DIMENSIONS FOR ROUND FLANGED FITTINGS AND COMPANION FLANGES

Nominal Pipe Size	Inside Diameter of Fitting	Metal Thickness of Fittings, Min	Outside Diameter of Flange	Thickness of Flange, Min B	Thickness of Flange at Edge H	Center ¹ to Flange Edge, Elbow, Tee and Cross A	Center ¹ to Flange Edge, 45° Elbow C	Diameter of Bolt Circle	Number of Bolts	Size of Bolts
5	5	1/2	10 1/4	1 13/16	1	7 3/8	5	8 1/2	8	3/4
6	6	1/2	11 3/4	2 1/4	1 1/8	9	5 1/2	9 3/4	8	7/8
8	8	5/8	14 1/4	2 1/2	1 3/8	10 3/4	6	12 1/4	12	7/8
10	10	3/4	16 1/2	2 3/4	1 9/16	12 1/8	7 1/2	14 1/2	16	7/8
12	12	1 1/16	19 1/4	3	1 3/4	13 3/4	8 1/2	17	16	1

All dimensions given in inches.

¹ The 1/4-in. tongue is included in the "thickness of flange, minimum," and "center to flange edge" dimensions.

DIMENSIONS FOR SPECIAL DROP-FORGED SQUARE FLANGES

Nominal Pipe Size	Length and Width of Flange	Thickness of Flange, Min B	Center ¹ Line to Side Contour of Flange F	Diameter of Bolt Circle	Number of Bolts	Size of Bolts
1 1/4	3 1/2	1	1 3/8	3 1/2	4	1/2
2	4 1/2	1 1/4	1 7/8	4 3/8	4	5/8

All dimensions given in inches.

¹ The 1/4-in. tongue is included in the "thickness of flange, minimum," dimension.

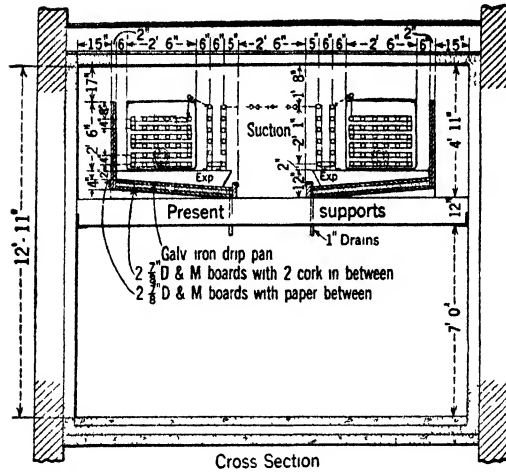
with an average travel of 500 to 1000 ft before it can get out and into the suction line, is not conducive to high values of heat transfer, for it should be remembered that the liquid must be kept in contact with the metal surface. Without some form of flooded system the last portion of the coil is likely to be non-effective because of the absence of liquid. With the flooded system some form of trap or accumulator is necessary in order to prevent slugs of the liquid from returning through the suction line to the compressor. Long-pipe evaporating coils for ammonia are obsolete.

20·6. Fittings. Standard ammonia fittings (Tables 20·10, 20·11, and 20·12) have recently been approved by the leading manufacturers. The materials chosen have been close-grained cast iron, cast semi-steel, and forged steel. Threaded pipe is made up with fittings for ammonia with the use of litharge and glycerin, although soldered joints are specified for special work fabricated in the shop. The fittings specified in the tables are designed for tongue-and-groove flange joints using asbestos and lead sheet packing gaskets cut to fit the groove in the flanges. The oval flanges, for $1\frac{1}{4}$ -in. pipe and smaller, are designed for two bolts.

20·7. Welded Pipe. Pipe welding has not been generally accepted except for special work, as for carbon dioxide piping, concealed piping like that laid in the concrete floors of skating rinks, piping in the floors or walls of tall buildings, or a special header construction as in ice-tank evaporators.

The forge-welded longitudinal joint for receivers, condensers, etc., has been attempted but has not been universally successful, nor does there appear to be any real reason for it as pipes of sufficient diameter can be secured from the steel mills. Flame welding for longitudinal joints has not been any more successful.

20·8. Piping Examples. Figures 20·8, 20·9, and 20·10 show typical arrangements of direct expansion and brine piping. In addition, Figs. 20·8 and 20·9 show methods of maintaining a cold storage space during the night by means of brine storage.



Cross Section

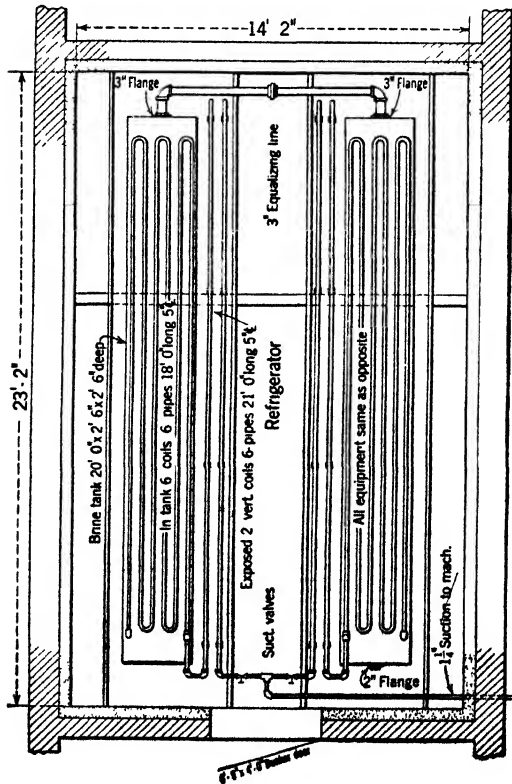


FIG. 20·9. Piping—direct expansion and brine tank.

Part IV
SPECIAL APPLICATIONS

CHAPTER XXI

COLD STORAGE; TRANSPORTATION; MISCELLANEOUS STORAGE

Cold storage includes not only such service as the terminal cold storage warehouse, where eatables are preserved frozen or at a temperature slightly above the freezing point until marketing conditions permit distribution to the retail trade, but also other kinds of service identified with space cooling. Goods having to be shipped long distances at times need marine space cooling, refrigerator cars, or refrigerator trucks. As the goods are frequently delivered for shipment warm from the fields or from exposure to summer-time conditions, precooling has to be provided before shipment, and where this applies to meats the packing plant has to remove the animal sensible heat quickly and sometimes freeze the meat. Also, at times, a commission house, retail market, and terminal cold storage warehouse district is congested to a point where a street pipe-line piping system from a central refrigerating plant can handle the separate space cooling projects to advantage. Finally the more recent development of unit coolers makes the problem of the actual cooling of the air considerably different from that which prevailed during the first quarter of the century. Since the principle in the cooling of passenger trains is very similar to that applied to refrigerator cars (although quite different in details), both these applications may with propriety be included in this chapter.

COLD STORAGE WAREHOUSE

21-1. General. Large cold storage houses (Fig. 21-1) have approximately 25 to 40 per cent freezer space, some of which should be arranged to be used as coolers. The freezer space should be so arranged as to be above or beside the coolers. If a freezer room is below a cooler, the cooler space should have an especially well-insulated floor to prevent freezing of goods sorted in the cooler. Insulation is necessary also to protect cooler goods kept adjacent to a freezer wall. Freezers should never be placed in basements because of the ever-present danger of the ground freezing and the consequent upheaval of footings and foundations.

The allowance for receiving and shipping floors including platforms is from 4 to 5 per cent of the total cold storage floor area. Usually receiv-

ing and shipping floors are located between two cold storage floors, as, for example, a basement cooler and a second-floor cooler or freezer.

The usual rule is to provide one elevator for each 30,000 sq ft of cold storage floor area. The minimum platform area must be sufficient to allow for two 30- by 60-in. trucks and two storage men, hence a size $7\frac{1}{2}$ by $7\frac{1}{2}$ ft is satisfactory, with a design which will take a load of $1\frac{1}{2}$ to 2 tons at a lifting speed of 75 to 250 fpm. For this purpose, both the plain elevators having vestibules at each floor and the vestibule-type elevator are in common use.

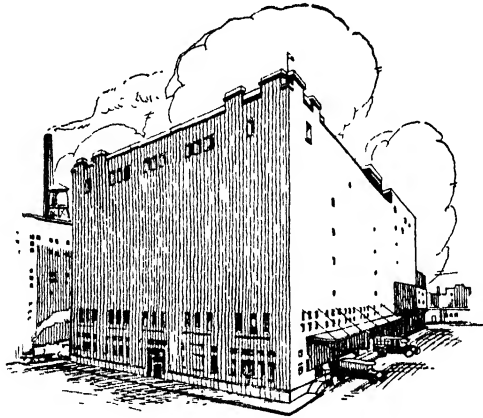


FIG. 21-1. Cold storage warehouse.

21-2. Construction. The construction of cold storage warehouses in the large cities is fireproof throughout in order to reduce the insurance rate; they usually are of reinforced concrete, sometimes faced with brick, of either the curtain wall or the insulated warehouse design. In addition, fire walls have to be provided as well as cut-offs between sections of the building (Fig. 21-2).

The exterior columns of curtain walls are split so that the entire wall is carried up independent of the rest of the building, whereas the floors, roofs, and interior columns form a structure which may expand or contract independently of the outer shell. The insulation would need to be as nearly continuous as possible, and it should be applied to the inner surface of the outer shell. The inner surface of the curtain wall should be thoroughly waterproofed. The insulation is usually corkboard, but it may be regranulated cork if provision is made for filling in at the top as settlement occurs. Corkboard must have at frequent intervals a brick or concrete shelf, or rails of wood or steel, fastened to the curtain wall to assist in preventing the loosening of the insulation

from the walls. Corkboard should not be placed on the underside of ceilings unless, in addition to the asphalt or mortar bond, it is securely fastened to the ceiling. When the insulated warehouse construction is used, independent movement of the inner and the outer portions of the building is impossible, so that undesirable stresses are developed which ultimately result in the cracking of the insulation and the accumulation

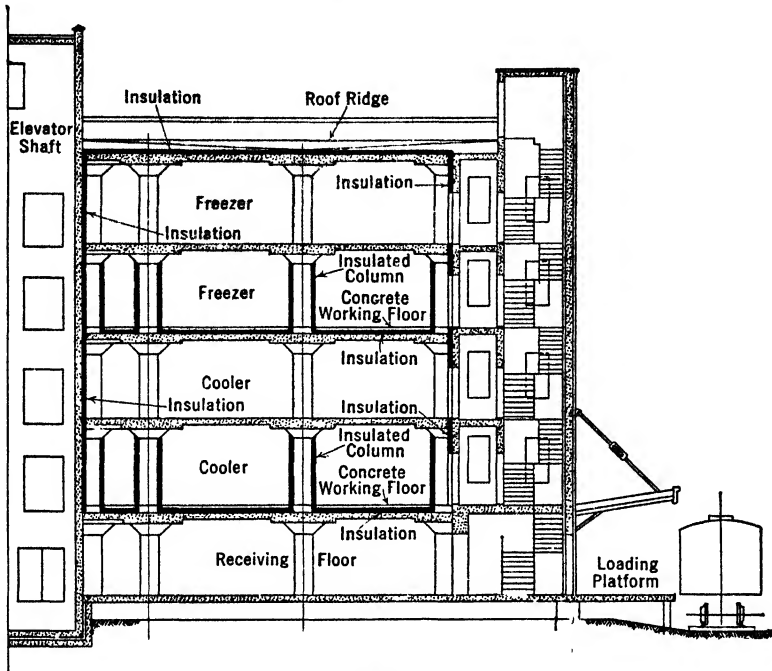


Fig. 21.2. Section of a cold storage building, curtain wall construction.

of moisture and ice. Approximately 50 per cent more insulation is required with the insulated warehouse construction as compared with the curtain wall design.

21.3. Insulation Thickness. Thickness of insulation for standard construction is taken as approximately 1 in. of corkboard per 10° F differential temperature on the two sides of the insulated wall. Cooler walls are usually 4 or 5 in. and freezer walls are usually 6 to 7 in. thick, depending on the amount of exposure to the sun, whereas sharp freezers have as much as 8 to 10 in. of corkboard. The roofs over freezers usually have 8 in., and over cooler space 6 in., of corkboard.

21.4. Floors. Cold storage construction is based on a live load of 250 lb/sq ft of floor area, but the average maximum is from 150 to 200 lb/sq ft. The height of the ceiling from the floor is from 10 ft without

sprinklers to about 11 ft with sprinklers. The flat slab and beam and girder floor constructions are used, but the latter interferes with free air movement and may result in moisture and snow deposits on the ceiling between the girders. For floors subject to much trucking, a wearing surface poured monolithic and containing some hardener may be advantageous. The concrete poured over cork insulation should be at least 3 in. thick.

21-5. Fire Protection. Sprinkler systems protect the goods in storage and therefore reduce the insurance rate. The pipes may be filled with air, under sufficient pressure to prevent the flow of water from the water tank for fire protection, or with a non-freezing solution of calcium chloride brine. After the sprinkler head melts and flow of water is started, the city water supply is fed into the line.

21-6. Piping. The three-pipe brine system (see Chapter XX) has the advantage of equal pressure drop in each circuit and more efficient air removal as compared with the two-pipe system, which is considerably lower in first cost. In passing from floor to floor the pipe shaft method is better although more space is required and the shaft has to be insulated. Brine velocities of 5 to 7 fps are allowed unless the pipe circuit is too long. The brine concentration should be such as to freeze at a temperature at least 10° below the minimum required brine temperature. Except for "quick freeze" refrigeration a brine temperature of less than -20° F is seldom required, and for coolers a brine temperature of 10° to 20° F is usual.

The advantages of the use of brine are several. The danger due to the escape of ammonia through a rupture in the pipe is a serious one, both to people and to the commodity in storage. This system suffers, however, less trouble from dripping pipes such as would occur during occasional short-time shutdown of a compressor system for minor repairs or adjustments. The room temperature is usually more easily regulated, and sometimes it is possible to secure a storage supply of refrigeration by means of a moderate-sized brine tank. The brine refrigerating system appears to be the favorite; it is used in the majority of cold storage warehouses and in district cooling refrigerating plants.

The *amount of piping* depends on the size of the room, the temperature carried, the insulation and the kind of service. In the packing plant the cooler load is heavy, being a peak load in the same manner as the pre-cooler for apples, lettuce, celery, and grapes. The load can be figured satisfactorily if the details are accurately known. Rough estimates may be taken from Figs. 21-3 and 21-4 or by using piping ratios such as 1 ft of 2-in. pipe per 8 cu ft of freezer space or 1 ft of 1¼-in. pipe per cu ft of sharp freezer. The freezer usually has piping placed directly in the

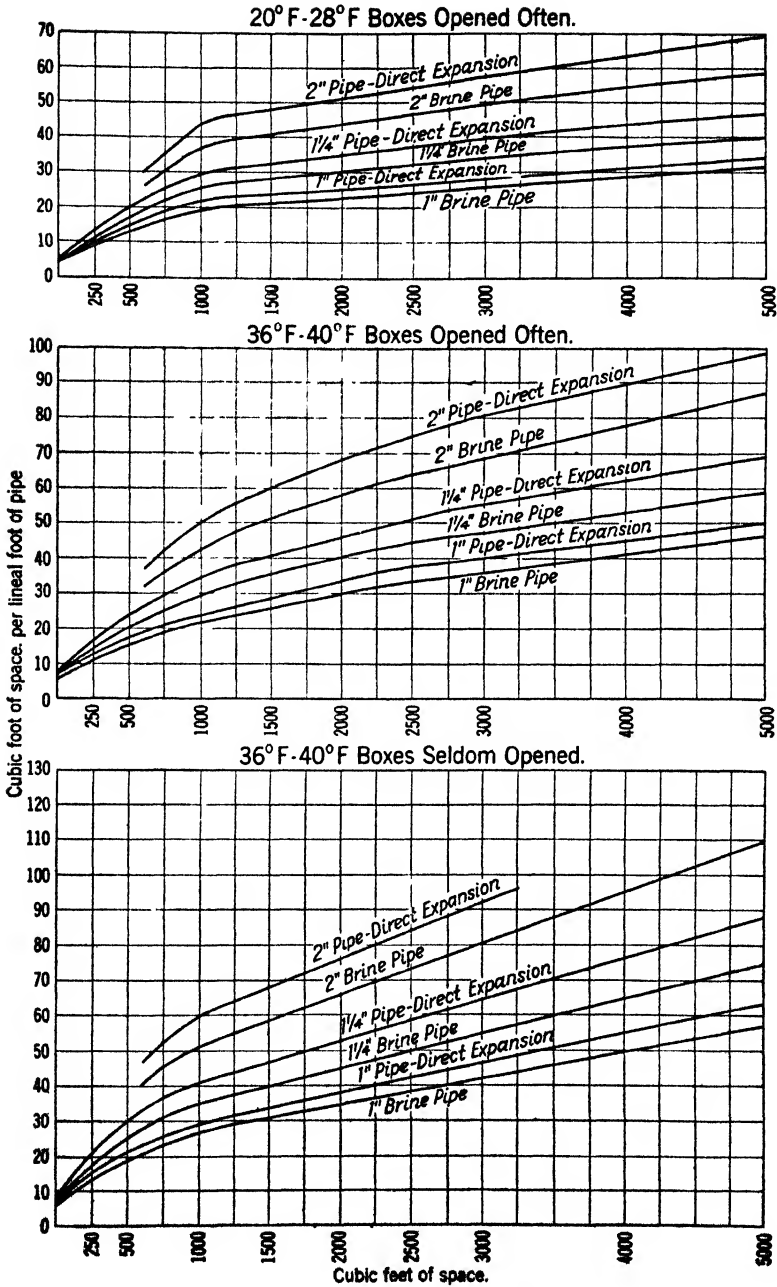


FIG. 21-3. Piping for small cold storage boxes.

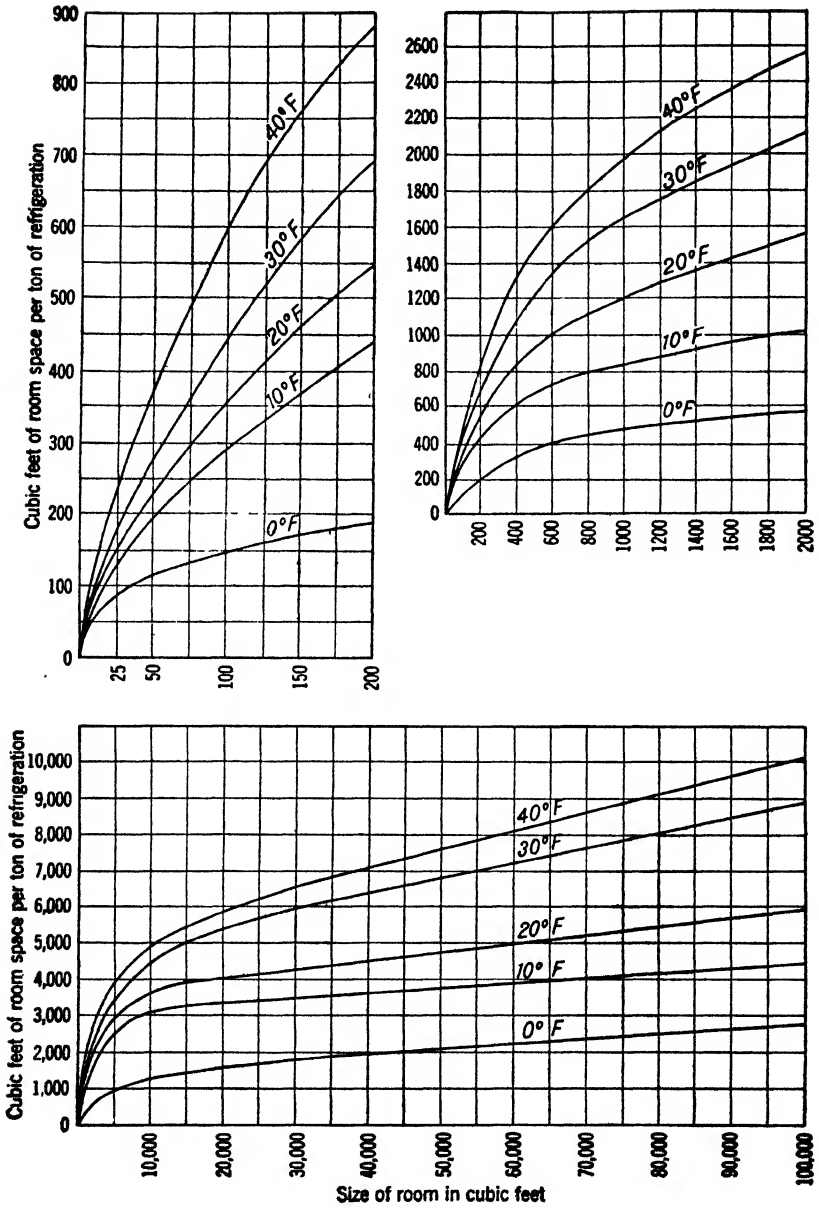


Fig. 21·4. Piping for cold storage rooms with usual insulation.

room, whereas the cooler may either have a similar arrangement or have a "loft" with brine sprays or piping or a separate bunker room supplied with ventilating fans. In the last arrangement, as in egg storage requiring a constant temperature, the fan capacity is from six to ten volumes of the refrigerating space per hour. The average temperature between the air and the brine or direct-expansion evaporating temperature is from 6° to 8° F in freezers and 10° to 15° in coolers. The tendency has been to make this difference small so as to keep the suction pressure of the compressor as high as possible, thereby decreasing the required piston displacement as well as the power required to operate the compressor. Tables 21·1 to 21·4 may be used to advantage in calculating the capacities required or in selecting operating temperatures.

21·7. Hold-Over Tanks and Pipes. Small plants, not designed for automatic control, find it uneconomical to operate at times more than 10 to 12 hours per day. Such plants, if considerable temperature rise is to be prevented, under these conditions either require brine operation and a brine tank of sufficient storage capacity to carry the load during periods of shutdown of the compressor, or they must provide some auxiliary means to absorb the heat leakage during this part of the day. One of the accepted methods is to use brine storage, or congealing, tanks or tubes (Fig. 21·5). In either, a certain amount of brine is cooled during the normal operation of the compressor, at the same time that the room is cooled by the usual piping; and piping arrangements, by means of direct-expansion piping submerged in the brine, are so located as to take the place of the direct-expansion piping. The scheme is seldom better than a compromise, but it has certain advantages. At times 6-in., 7-in., or 8-in. pipe can be used to better advantage than a tank, in which case the pipes are filled with brine. The following example shows the method of computation.

Example. During a period of 12 hr a refrigerating load of 30,240 Btu is expected. The room is to be held at 33° F, and the brine is to be cooled to 13° F with the expectation that the brine may rise to 28° during the period of shutdown. Find the number of cubic feet required. See Table 20·1.

Solution. The weight of brine required is

$$W = \frac{30,240}{0.80 \times 15} = 2520 \text{ lb}$$

and the volume of brine (see Chapter XX), using a specific gravity of 1.16,

$$\frac{2520}{62.3 \times 1.16} = 34.87 \text{ cu ft}$$

TABLE 21-1
CLIMATIC CONDITIONS
(U. S. Weather Bureau)

State	City	Altitude	Population in 1930 Thousands	Dry-bulb Temperature, °F					Average Daily Max. Wet-bulb Temp., July		Wind, Miles per hour		Approx. Min. Relative Humidity during Hot Weather
				Highest on Record	Lowest on Record	Average for Whole Year	Average Oct. 1 to May 1	Average Daily Max. July	Average Velocity, Dec. 1 to Mar. 1	Prevailing Direction, Dec. 1 to Mar. 1			
1	2	3	4	5	6	7	8	9	10	11	12	13	
Alabama	Mobile Birmingham	12 597	68 258	103 104	-1 -10	65 61	58 54	90 89	78 75	8 9	N N	29 24	
Arizona	Phoenix Flagstaff	1085 6900	49 4	119	16	69	59	104	72	4	E	10	
Arkansas	Fort Smith Little Rock	433 275	31 82	108 106	-25 -15 -12	60 61	35 49 52	81 91 90	61 76 76	7 8 10	SW E NW	23	
California	San Francisco Los Angeles	50 291	637 1234	101 109	29 28	55 60	54 59	64 82	57 67		N NE	18	
Colorado	Denver Grand Junction	5227 4573	288 10	105 104	-29 -20	49 51	39 39	86 92	64 64	7	S SE		
Connecticut	New Haven	21	163	101	-14	49	38	80	69	9	N	26	
District of Columbia	Washington	80	487	106	-15	54	43	86	73	7	NW	40	
Florida	Jacksonville	8	130	104	10	67	62	90	77	8	NE	23	
Georgia	Atlanta Savannah	1082 21	267 85	102 105	-8	59 64	51 58	87 90	74 77	12 8	NW NW	32	
Idaho	Lewiston Pocatello	738 4461	9 16	102	-13 -20	47	42 36	89 86	65 61	5	E	36	
Illinois	Chicago Springfield	609 602	3375 72	103 107	-23 -24	49 52	36 40	80 80	69 71	17 10	SW NW	34	

CLIMATIC CONDITIONS

1	2	3	4	5	6	7	8	9	10	11	12	13
Indiana	Indianapolis Evansville	718 360	364 103	106 107	-25 -16	52	40 44	85 88	71 73	12 8	S S	26
Iowa	Des Moines Dubuque Sioux City	790 612 1101	142 42 79	110 106 107	-30 -32 -35	49 47 47	34 32 32	86 85 84	71 70 68	6 12	NW NW NW	28 31
Kansas	Concordia Dodge City	1370 2480	6 10	110 108	-25 -26	56 52	39 40	89 90	72 71	7 10	N NW	18 13
Kentucky	Louisville	436	308	107	-20	56	45	88	72	9	SW	25
Louisiana	New Orleans Shreveport	5 210	455 77	102 110	7 -5	67 64	61 56	89 92	77 77	10 8	N SE	31 18
Maine	Eastport Portland	76 16	3 70	100 103	-23 -21	41 44	31 34	69 76	69 65	14 10	W NW	38 28
Maryland	Baltimore	97	802	105	-7	54	44	86	72	7	NW	23
Massachusetts	Boston	48	787	104	-14	49	39	50	68	12	W	29
Michigan	Alpena Detroit Marquette	591 594 618	12 1574 15	101 104 108	-27 -24 -27	42 47 41	29 35 23	75 81 74	65 68 63	11 13 11	W SW NW	30 28
Minnesota	Duluth Minneapolis	611 825	101 465	99 102	-41 -33	44 44	25 30	75 82	63 63	11 12	SW NW	42 31
Mississippi	Vicksburg	186	23	104	-1	64	56	90	77	8	SE	20
Missouri	Kansas City St. Joseph St. Louis Springfield	750 830 446 1260	393 81 822 58	108 107 106	-22 -24 -22 -29	54 47 55 54	40 40 43 43	86 90 87 85	72 73 72	9 12 11	NW NW SE	28 29 36
Montana	Billings Havre	3119 2480	16	112 108	-49 -57	41	35 28	88 82	63	9	W SW	

TABLE 21-1 (Continued)
CLIMATIC CONDITIONS
(U. S. Weather Bureau)

State	City	Altitude	Population in 1930, Thousands	Dry-bulb Temperature, °F					Average Daily Max. Wet-bulb Temp., July	Average Daily Max. Temp., July	Average Wet-bulb Temp., July	Wind		Approx. Min. Relative Humidity during Hot Weather
				Highest on Record	Lowest on Record	Average for Whole Year	Average Oct. 1 to May 1	Average Daily Max. July				Miles per hour		
												Average Velocity, Dec. 1 to Mar. 1	Prevailing Direction, Dec. 1 to Mar. 1	
1	2	3	4	5	6	7	8	9	10	11	12	13		
Nebraska	Lincoln North Platte	1148 2803	78 12	110 107	-20 -35	50 47	37 35	87 86	71 69	11 9	N W	41 31		
Nevada	Reno Tonopah Winnemucca	4490 4824	18 2	102 106 104	-19 -28	49 50 48	40 40 38	88 88 88	62 59 61	10 10	SE NE	32 47 31		
New Hampshire	Concord	244	25	102	-35	33	33	81	68	6	NW	41		
New Jersey	Atlantic City	10	66	104	-7	52	42	79	71	11	NW	31		
New Mexico	Santa Fé	6951	11	97	-13	47	38	81	60	7	NE	31		
New York	Albany Buffalo New York	15 589 29	127 873 6982	104 95 102	-24 -14 -13	47 47 51	35 35 40	82 76 82	70 67 70	8 18 13	S W NW	32 47 31		
North Carolina	Charlotte Raleigh Wilmington	736 539 25	83 37 82	103 103 103	-5 -2 5	58 58 61	48 50 53	88 88 87	74 74 76	7 9	SW SW W	23 32 34		
North Dakota	Bismarck Devil's Lake	1649 1465	11 6	108 106	-45 -44	40 36	24 19	82 79	66 66	11	NW W	25		
Ohio	Cleveland Columbus	918 784	900 289	100 104	-17 -20	49 51	37 40	79 85	68 71	15 9	SW SW	35		
Oklahoma	Oklahoma City	1197	185	108	-17	58	48	90	73	12	N	21		
Oregon	Baker Portland	3437 66	302	102 104	-24 -2	45 53	34 46	78 78	63	6 7	SE S	31		

CLIMATIC CONDITIONS

1	2	3	4	5	6	7	8	9	10	11	12	13
Pennsylvania	Philadelphia Pittsburgh	149 742	1964 670	106 103	-6 -20	53 52	42 41	85 85	71 70	11 14	NW NW	28 34
Rhode Island	Providence	11	252	100	-12		38	81		15	NW	34
South Carolina	Charleston Columbia	13 239	62 50	104 106	7 -2	6* 61	57 54	88 90	77 75	11 8	N NE	34 31
South Dakota	Huron Rapid City	1285 3217	11 10	108 106	-43 -34	43 46	28 32	84 84	68 64	12 8	NW W	23
Tennessee	Knoxville Memphis	868 222	106 252	100 104	-16 -9	57 61	47 51	87 89	73 76	7 10	SW NW	27 29
Texas	El Paso Fort Worth San Antonio	3703 589 658	102 161 255	105 112 107	-5 -8 4	62 64 67	53 55 61	94 92 94	68 73 75	11 11 8	NW NW N	7 19 21
Utah	Modena Salt Lake City	5479 4269	140	99 103	-24 -20	47 47	38 40	87 88	60 62	9 5	W SE	
Vermont	Burlington	112	25	100	-28		29	79	66	13	S	46
Virginia	Norfolk Lynchburg Richmond	12 689 71	128 41 183	105 105 107	2 -7 -3	58 55	49 45 47	87 87 88	75 74 72	9 5 7	N NW S	32 27
Washington	Seattle Spokane	15 1899	366 116	98 104	3 -30	51 48	45 37	73 83	61 60	9	SE SW	
West Virginia	Elkins Parkersburg	1940 602	7 30	99 106	-28 -27	46 52	39 42	82 85	70 72	5 7	W S	27 29
Wisconsin	Green Bay La Crosse Milwaukee	590 654 600	37 39 573	101 104 102	-36 -43 -25	43 39 46	29 31 33	80 83 78	67 69 67	13 6 12	SW NW W	28 32 31
Wyoming	Sheridan Lauder	3729 5372	2	105 99	-45 -36	42 41	31 29	86 83	64 61	5 3	NW NE	

TABLE 21-2

Product	Handling Factors		Warehouse Requirements			
	Pre-storage Handling and Condition when Placed in Storage	Varieties and Grades	A — Construction	B — Temperature	C — Humidity	D — Stowing
Apples	Apples should be picked when well matured but not over-ripe. In all the operations of picking, packing and handling they should be so handled as to avoid bruising, pin, punctures, and other mechanical injuries, and they should be so graded as to be practically free from insect damage. If insects, diseases or mechanical injuries are found, they should be handled promptly from the orchard to the storage room and cooled quickly.	Only varieties which have a recognized storage period of three months or more should be considered. Common storage houses should be sufficiently insulated to prevent freezing, and should be provided with the necessary inlet and outlet vents to permit adequate ventilation and temperature regulation.	Cold-storage houses should be so constructed and equipped as to maintain practically uniform temperature and humidity conditions throughout the storage season.	Cold-storage temperature range should be 31° to 32° F for the storage of apples. Common storage temperature should be maintained at from 31° to 36° F after the initial cooling of the fruit.	Humidity range, 80 to 90 per cent.	Apples should be stowed with sufficient spacing to permit of free air circulation, and to render each lot readily accessible for inspection and withdrawal.
Potatoes	Potatoes should be well matured, and graded to conform to the specifications of the United States standard grades. Stock should be certified by a competent inspector.	All varieties harvested in autumn keep well in storage.	The building should be so constructed and insulated as to prevent fluctuations in temperature. Means for ample ventilation should be provided, and all unnecessary light should be excluded.	Temperature range from 35° to 40° F.	Humidity range, 80 to 85 per cent.	When stowed in bags, boxes or crates, potatoes should be so piled as to permit free air circulation. Bulk potatoes should not be stored to a greater depth than 6 ft nor more than 60,000 lb in a single compartment. They should be carefully handled to avoid unnecessary injuries. When stored in bins, the potatoes should be carefully poured from basket or crate into the bin. To allow free circulation of air, the bins should have slatted sides and floor, and at least 4 in. of air space on all sides. The bin floor should be raised 2 in. or more above the house floor. When stowed in crates, baskets or hampers, the containers should be stacked so as to allow circulation of air, and to avoid the crushing or breaking of the packages and the bruising of their contents.
Potatoes (sweet)	Sweet potatoes should be well developed, carefully handled to avoid bruising, and should be practically free from damage caused by disease, insect or mechanical injury. They should not be allowed to become chilled or frosted, and when placed in storage the surface should be dry and practically clean.	All varieties grown on a commercial scale.	The building should be so constructed that all light is excluded and moderate changes in outside temperature will not outside affect inside temperature. Wood construction is preferable, and ample means for ventilation control should be provided.	While the potatoes are being stored, and for a period of ten days to two weeks thereafter, until the potatoes are cured, a temperature of 80° to 90° F should be maintained. Thereafter, a temperature of as low as 55° F, as is practicable, should be maintained. Ventilation and artificial heat are necessary to control temperature and moisture.	The percentage of humidity should not be so high that moisture is deposited on the walls of the storage house.	To allow free circulation of air, the bins should have slatted sides and floor, and at least 4 in. of air space on all sides. The bin floor should be raised 2 in. or more above the house floor. When stowed in crates, baskets or hampers, the containers should be stacked so as to allow circulation of air, and to avoid the crushing or breaking of the packages and the bruising of their contents.
Onions	The onions should be well ripened, dry and thoroughly air-cured when stored. Onions intended for storage should be practically free from damage caused by disease, insects, or mechanical injury and from other stock commercially known as culls.	All common varieties of onions, except those of the Bermuda type.	The building should be so constructed and insulated as to prevent fluctuations in temperature, and means for ample ventilation should be provided.	In cold storage the temperature should be from 32° to 36° F. In common storage the same range of temperature should be maintained as nearly as possible, but with the proper ventilation, onions will keep well in common storage at higher temperatures.	Low humidity is desirable.	Onions should be stowed in suitable "caskets," as indicated by the "Onions," and should be stacked in such a way as to permit of free air circulation throughout the lot.

Cabbage	Cabbage must be of solid heads, practically free from injuries caused by insects and diseases. Heads should be cut with but few if any loose leaves adhering and carefully handled from field to storage house. Special care should be used to avoid bruising and other mechanical injuries.	Well-ventilated, frost-proof root cellar or warehouse type of construction, with ample intake and outlet vents for quick cooling and ventilation, and equipped with slatted shelves supported on starting so that the heads may be stored one layer deep, with at least 15 to 18 in. clear space around the walls of the building. The ceiling should be so constructed as to prevent drip on the product.	Temperature range, 32° to 36° F.	The humidity should be maintained as high as possible without actual deposition of moisture on the product.	Cabbage should be stored on flat shelves in single layers. The height of the staging and the number of shelves will be determined by convenience and dimensions of the building.
Eggs	Eggs should be moved quickly from the producer to the warehouse. They should be carefully sorted and candled, so that none showing mechanical defects or noticeable deterioration is included in the storage stocks. No washed eggs should be stored.	The grades should conform to those generally adopted by the wholesale trade, until United States standards are promulgated.	Temperature range, 29° to 32° F.	Humidity range, 82 to 85 per cent.	Egg cases should be stored so that separate lots may be easily inspected, and with 4-in. to 1-in. dunnage between the cases to insure space for free air circulation.
Frozen eggs	Eggs should be removed from shell in chilled sanitary surroundings and frozen immediately on fish-bait sharp freezers.	One grade for food. One grade for manufacturing purposes.	Temperature range, 0° or below to 10° F above.	The usual humidity at the temperature of storage.	Protect eggs from heat leakage at doors and elevator shafts.
Poultry	Poultry should be dry pickled, dry cooled, and dry packed at temperatures ranging from 30° to 35° F for 18 to 24 hours, then frozen at 6° F or below.	The classes and grades should conform to those generally adopted by the wholesale trade, until United States standards are promulgated.	Preferred temperature, 0° to 10° F. Admissible temperature, 12° to 14° F.	Same as frozen eggs.	Poultry should be so stored that separate lots may be inspected easily, and protected from injury by heat leakage at doors and elevator shafts.
Butter	Butter should be placed in cold storage within ten days after it is manufactured. When storage facilities are not available during this period, the product should be held at a temperature below 40° F.	The grades should conform to those generally adopted by the wholesale trade, until United States standards are promulgated.	Temperature, 2° F or below.	Same as frozen eggs.	Packages of butter should be so stored as to permit a free air circulation beneath the pile, and so stacked that separate lots may be inspected easily. Cube and box packages should be separated by 1-in. dunnage.
Fish	Fish should be placed in storage in a fresh condition, as indicated by their physical appearance.	Practically all kinds used for food.	Hard frozen and glazed at temperature of -5° F or below, and stored at 0° F or below to 10° F, depending on the kind. For holding less than six months it is admissible to store at 12° F.	Same as frozen eggs.	Fish should be stored as compactly as possible.

TABLE 21-2 (Continued)

Product	Warehouse Requirements		Storage Period	Other Considerations		Remarks
	E — Containers	F — Inspection		Shrinkage		
Apples	Containers shall be clean, strongly built barrels, boxes or crates, and when packed for market shall be plainly marked with the grade variety and the grower's or packer's name.	All lots of apples should be inspected when received for storage by a qualified inspector. Subsequent inspections of representative packages of all lots should be made at intervals of 15 to 30 days, depending on the variety and condition of the fruit as indicated by previous inspections.	The usual cold-storage period for winter varieties of apples is from three to six months, depending upon the variety and condition of the fruit when stored.	The shrinkage in cold storage is from 2 per cent to 5 per cent. In common storage the shrinkage is variable.	Attention is directed to the fact that a delay of one or more weeks between the picking and storing of apples greatly reduces the storage period of the product and results in early deterioration. The successful storage of apples is as much dependent upon the treatment they receive before being placed in cold storage as the conditions under which they are held in storage. See <i>U. S. Department of Agriculture Bulletin 5877</i> .	
Potatoes	Potatoes may be stored in clean burlap bags, barrels, boxes or crates; or when in bulk they should be stored in ventilated bins.	Potatoes should be inspected by a qualified inspector when received for storage, and again within 30 days. The frequency of the inspections thereafter will depend upon the condition of the potatoes as determined by previous inspections.	The usual storage period is from three to six months, depending upon the section of the country in which the storage is located, the type of the storage house and the condition of the stock. Allowing for a high percentage of deterioration, potatoes may be held in storage for a much longer period.	When potatoes are stored, in containers or in bulk, as specified in Column 6, the shrinkage may amount to about 7 per cent, although it varies greatly.	Potatoes are usually stored in cellars and common storages, but are sometimes held in cold storages. See <i>Farmers' Bulletin 847</i> .	
Potatoes (sweet)	Sweet potatoes are usually stored in bins, but may be stored satisfactorily in substantial crates, baskets or hampers which permit of a free air circulation.	The potatoes should be thoroughly inspected by a qualified inspector at the time they are put in the storage house, within 15 days after the beginning of the storage period and from 15 to 30 days thereafter.	The safe storage period is about four months. Under the most favorable conditions and good management sweet potatoes may be kept six months.	The shrinkage from loss of moisture is from 8 per cent to 10 per cent in bins and somewhat higher in packages. An additional shrinkage of 5 per cent should be allowed for decay.	It is recommended that sweet potatoes be not considered properly stored until they have passed through the curing period. See <i>Farmers' Bulletin 970</i> .	
Onions	The best containers are slatted crates, although baskets, hampers and bags are used successfully.	Thorough inspection should be made when the onions are placed in storage, and at intervals not exceeding 30 days. The frequency of the inspections thereafter will depend upon the condition at the previous inspection.	The usual storage period of onions with proper ventilation is six months.	The shrinkage should not exceed 10 per cent or 12 per cent.	See <i>Farmers' Bulletin 354</i> .	

<p>Cabbage;</p>	<p>Containers are not generally used.</p>	<p>Cabbage should be inspected at intervals of 15 to 30 days.</p>	<p>The storage period for cabbage extends from November to April—5 or 6 months.</p>	<p>The shrinkage in cabbage is quite variable.</p>	<p>Stores should be provided in common storages to prevent freezing in cold periods. See <i>Farmers' Bulletin</i> 433.</p>
<p>Eggs</p>	<p>Eggs should be packed in clean, odorless, wood cases. Fillers should be of new No. 1 or medium straw or wood pulpboard with straw over top and under bottom. Paddling must be kin-dried extender, cork shavings or corrugated straw or wood pulpboard on top and bottom of the case. No glue or caseoil should be used. The cases should be plainly marked with the grade.</p>	<p>Inspection of eggs should be at intervals of 15 to 30 days, and the storage house should have daily attention from a competent warehouseman skilled in the handling of such structures and commodities.</p>	<p>The storage period for eggs should not exceed 12 months.</p>	<p>The shrinkage depends upon the humidity, and should not be more than 5.5 per cent. Shrinkage should be calculated from net weight of produce.</p>	<p>Rooms must be clean and odorless. See <i>Bureau of Chemistry Circular</i> 64.</p>
<p>Frozen eggs</p>	<p>Thirty-pound tin buckets are most common. The use of smaller tin cans is now increasing, owing to wider use of this product.</p>	<p>Inspections of frozen eggs should be made about every 30 days.</p>	<p>No change in composition up to 24 months. After 12 months egg thickens slightly. White near top of can may become pink, due to iron under tin. Egg not injured as food thereby.</p>	<p>The shrinkage is not of commercial importance.</p>	<p>See <i>U. S. Department of Agriculture Bulletin</i> 51. See <i>U. S. Department of Agriculture Bulletin</i> 224. See <i>Bureau of Chemistry Circular</i> 98.</p>
<p>Poultry</p>	<p>All poultry should be packed in clean, strongly built, odorless boxes, lined with parchment or other suitable paper and should be plainly marked to indicate the grades and classes. Barrels are still admissible, especially for turkeys, but are less desirable than boxes.</p>	<p>All lots of poultry should be inspected by a qualified inspector when received for storage, and at intervals of 50 days or longer, depending upon the conditions found at the previous inspections.</p>	<p>The storage period for poultry should not exceed 12 months.</p>	<p>The shrinkage varies from 1 per cent to 3 per cent.</p>	<p>Water-cooled or ice-packed poultry should not be stored for long periods. Scalded birds deteriorate more rapidly than dry-picked. Drawn poultry should never be stored. See <i>Bureau of Chemistry Circulars</i> 64 and 70.</p>
<p>Butter</p>	<p>Packages should conform to the regular commercial styles, including 63-lb tubs, 63- to 75-lb casks and standard boxes of 1-lb prints.</p>	<p>All lots of butter should be inspected by a qualified inspector when received for storage, and at intervals of 30 days or more, depending upon the quality and condition of the lots at previous inspection.</p>	<p>The storage period for butter should not exceed 12 months.</p>	<p>In general the shrinkage will run from 4 per cent to 1 per cent.</p>	<p>See <i>Bureau of Animal Industry Bulletins</i> 84 and 148.</p>
<p>Fish</p>	<p>Fish are stored in boxes and in bulk.</p>	<p>Inspection of fish should be made at intervals of 30 days or more by a qualified inspector.</p>	<p>The storage period for fish should not exceed 12 months.</p>	<p>The shrinkage is not of commercial importance.</p>	<p>Boxed fish should be reglased in three to six months. Slacked fish should be spray-glassed every three months or more frequently. See <i>U. S. Department of Agriculture Bulletin</i> 635.</p>

TABLE 21-3
PROPERTIES OF FISH AND MEAT

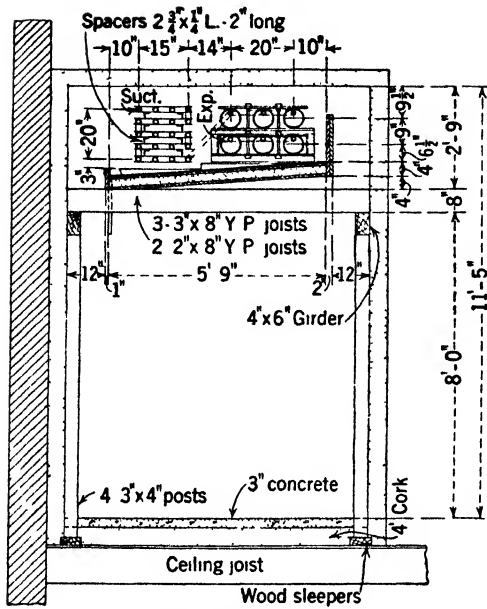
	Fresh Fish				Fresh Meat		
	Water Content		Fat in Edible Portion, %		Water Content		Fat in Edible Portion, %
	As Purchased, %	In Edible Portion, %			As Purchased, %	In Edible Portion, %	
Bass, striped.....	35.1	77.7	2.8	Brisket, medium fat.....	41.6	54.6	28.5
Blackfish.....	31.4	79.1	1.3	Chuck, incl. shoulder, very lean.....	60.2	73.8	3.9
Bluefish.....	..	78.5	1.2	Chuck, incl. shoulder, lean... ..	57.4	71.3	8.2
Butter-fish.....	42.8	70.0	11.0	Chuck, incl. shoulder, med. fat.....	57.9	68.3	11.9
Catfish.....	51.5	64.1	20.6	Chuck, incl. shoulder, fat... ..	53.3	62.3	18.8
Cod.....	38.7	82.6	0.4	Chuck, incl. shoulder, very fat.....	40.8	53.2	29.0
Eels (salt water).....	..	71.6	9.1	Fat, clear.....	13.4
Flounder.....	32.6	84.2	0.6	Ribs, very lean.....	52.4	70.9	3.5
Haddock.....	..	81.7	0.3	Ribs, lean.....	52.6	67.9	12.0
Hake.....	..	83.1	0.7	Ribs, med. fat.....	48.8	55.5	26.6
Halibut (steaks).....	..	75.4	5.2	Ribs, fat.....	39.6	48.5	35.6
Herring.....	41.7	72.5	7.1	Ribs, very fat.....	42.9	45.9	38.7
Kingfish.....	34.4	79.2	0.9	Round, very lean.....	65.9	73.6	2.8
Lamprey.....	38.5	71.1	13.3	Round, lean.....	64.4	70.0	7.9
Mackerel.....	40.4	73.4	7.1	Round, med. fat.....	60.7	65.5	13.6
Mullet.....	31.5	74.9	4.6	Round, fat.....	54.0	60.4	19.5
Muskellunge.....	38.7	76.3	2.5	Rump, very lean.....	60.9	71.2	5.1
Perch (white).....	28.4	75.7	4.0	Rump, lean.....	56.6	65.7	13.7
Pickrel (pike).....	42.2	79.8	0.5	Rump, medium.....	45.0	56.7	25.5
Pike (gray).....	29.7	80.8	0.8	Rump, fat.....	36.2	47.1	35.7
Pollock.....	54.3	76.0	0.8	Rump, very fat.....	33.7	40.2	44.3
Pompano.....	39.5	72.8	7.5	Lamb, fresh.....
Porgy.....	29.9	75	5.1	Leg, hind, medium fat.....	52.9	63.9	16.5
Redgrouper.....	..	79.5	0.6	Leg, hind, fat.....	47.3	54.6	27.4
Redsnapper.....	42.0	78.5	1.0	Leg, hind, very fat.....	48.2	51.8	30.1
Salmon.....	40.9	64.6	12.8	Pork, fresh.....	46.1	60.3	19.0
Salmon (landlocked).....	42.3	77.7	3.3	Loin (chops), lean.....	46.1	60.3	19.0
Shad.....	35.2	70.6	9.5	Loin (chops), med. fat.....	41.8	52.0	30.1
Shad roe.....	..	71.2	3.8	Loin (chops), fat.....	34.8	41.8	44.4
Sheepshead.....	26.9	75.6	3.7	Loin, tenderloin.....	66.5
Skate, lobe of body.....	40.2	82.2	1.4	Shoulder.....	44.9	51.2	34.2
Smelt.....	46.1	79.2	1.8	Poultry (dressed)
Spanish mackerel.....	44.5	68.1	9.4	Chickens, broilers.....	43.7	74.8	2.5
Sturgeon, anterior sections.....	67.4	78.7	1.9	Fowl.....	47.1	63.7	16.3
Tomead.....	32.7	81.5	0.4	Geese.....	38.5	46.7	36.2
Trout, brook.....	40.4	77.8	2.1	Turkey.....	42.4	55.5	22.9
Trout, salmon or lake... ..	36.6	70.8	10.3	Shellfish
Turbot.....	37.3	71.4	14.4	Clams.....	28.0	36.2	0.1
Weakfish.....	38.0	79.0	2.4	Lobsters.....	30.7	79.2	1.8
Whitefish.....	32.5	69.8	6.5	Oysters.....	16.1	36.9	0.2

TABLE 21-4
VEGETABLES — WATER CONTENT AND FREEZING POINT

	Freezing Point, °F	Water Content, %		Freezing Point, °F	Water Content, %
Asparagus	94.0	Cucumbers	31.46	94.5
Beans (snap)	29.74	68.5	Eggplant	30.41	92.2
Beans (green)	31.28	58.9	Kohlrabi	30.02	91.1
Beets	31.10	88.7	Lettuce	31.20	94.4
Cabbage (Early Jersey Wakefield)	31.18	91.5	Onions (dry)	30.09	87.6
Carrots:		88.2	Onion sets (Yellow Globe)	29.50
Danvers	29.61	Peas (green)	30.03	74.6
Chantenay	29.53	Potatoes	28.92
Average	29.57	Potatoes, sweet	28.44	78.3
Cauliflower	30.08	92.3	Squash	31.28	88.3
Celery	29.73	94.5	Tomatoes (ripe)	30.38	94.8
Corn, sweet:		75.4	Tomatoes (green)	30.40
Crosby	29.07	Turnips	30.23
Country Gentleman	29.11			
Howling Mob	28.00			
Golden Bantam	29.61			
Average	28.95			

21-8. Quick Freezing of Foods. Both plant and animal tissues are composed of a multitude of small cells between which are solutions of many complex chemical salts. With a slow lowering of the temperature the water element freezes out first, leaving, as the freezing process progresses, a more and more concentrated liquid behind. This does not freeze until the temperature is much lower still. The slower the process of freezing the larger the ice crystals become, resulting in discoloration, oxidation, evaporation, and loss of volatile flavoring elements even though the thawing is conducted slowly so that the damaged tissue may reabsorb the water. Quick freezing, however, prevents the formation of large crystals, and in general the quicker the freeze the better the product, although there appears to be less disturbance in animal than in plant structure. Quick-frozen products do not require slow thawing, because of the minute ice crystals, and both plant and animal material may be cooked in an initially frozen condition. It is claimed that a quick-frozen food has the same nutritive value and is as palatable as the unfrozen article.

As a rule the temperature zone of maximum ice crystal formation is between 31° and 25° F, but freezing continues to -60° F or below in many cases, and the object of quick freezing is to pass through the maximum crystal zone as quickly as possible. The amount of heat



Cross Section C-D

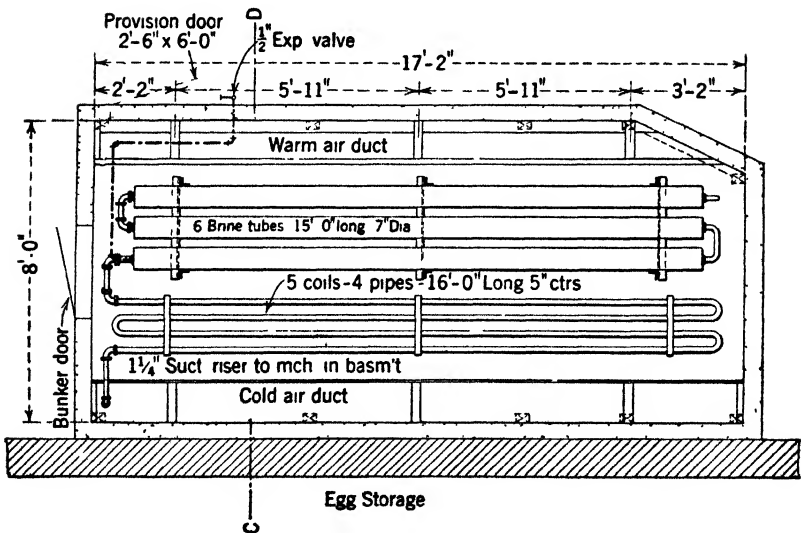


FIG. 21-5. Brine tubes and ammonia piping in pipe deck for small cold storage rooms.

removed in such quick freezing depends largely on the water content, which varies from less than 40 per cent in lean meats and poultry to 90 per cent or more in rhubarb, asparagus, and strawberries. It ranges from 74 Btu/lb for geese to 157 to 174 Btu/lb for codfish and soft fruits and vegetables which are cooled from 60° to -5° F. However, as a rule, the minimum temperature employed in quick freezing varies from -20° to -50° F.

The most common methods of quick freezing are:

(a) Direct immersion in brine of low temperature. This is likely to injure the container, or if no container is used the product is likely to be discolored, to form irregular shapes, and to need washing before the ice glaze is applied.

(b) Indirect freezing by placing the commodity in metal packages which are immersed in very cold brine.

(c) The double-plate belt conveyor and the stationary multiplate with proper side plates, in either case designed so that the commodity will be subjected to a metal contact with a spray of brine.

Still other methods of quick freezing use "diving bells" immersed in brine or a blast of very cold air. Storage should be at -5° to -10° F, transportation at 10° or lower, and the commodity should be delivered to the consumer in a hard-frozen condition.

Low-temperature freezing is applied* to sliced peaches and ice cream; freezing tunnels are used, and a loft arrangement containing direct-expansion piping laid out with liquid and gas headers and short V-shaped connecting columns, the entire unit being steel-welded. Forced-air circulation is provided at about 1500 fpm; for peaches the temperature is from -15° to -35° F, and for pint packages of ice cream it is still lower. With these packages a drop of temperature from 23° to 0° F was secured in 65 min with -33° air moving at 710 fpm past the packages. A loft over the piping provides a brine spray for the removal of the frostation. The piping is approximately 1.45 linear feet of 1¼-in. pipe per cubic foot of refrigerated space. The insulation is 10 in. of corkboard for the sides exposed to atmospheric air. A room 41 ft long by 13 ft wide and 13 ft high is designed for 17,300 cu ft of fan capacity per minute.

TRANSPORTATION

21·9. Refrigerator Car. The refrigerator car, which is likely to be 33 ft 2¾ in. long, 8 ft 2¾ in. wide, and 7 ft 5¼ in. high with two center doors, has been difficult to refrigerate properly near the center

* Zumbro, *Refrig. Eng.*, April, 1931.

of the car even with floor racks arranged so that cold air may pass along the floor from the end compartments. The service is very difficult, the center doors frequently permitting excessive air infiltration and the insulation being poor at best. Where ice, or ice and salt for temperatures below 35°, is used, as much as 5 tons of ice may be required per 24 hr. The two end ice baskets may have a capacity of 3 to 5 tons of ice, but usually in the latest designs of cars the capacity is 5½ tons, and in spite of the advances in the development of mechanical refrigeration ice-refrigerated cars are still the favorite because of the low first cost of the equipment.

The gastight insulated container for solid carbon dioxide, designed for 3000 lb of dry ice and thermostatically controlled, gives a successful arrangement and is less objected to by the railroad maintenance-of-way department than the use of an ice and salt brine unless tanks are provided to catch the brine until it can be released without objection. Some piping is required to direct the air and gas flow. However, it would seem that the initial cost of dry ice would have to be reduced considerably before any general use would be warranted in refrigerator cars. Where water ice is used, properly designed icing stations are required. These embrace an ice-manufacture plant, unless natural ice is cheap and can be depended on, ice storage, and an approved arrangement of car-icing platforms with a capacity either of a few cars or of an entire train. Usually a speed of one car per minute for re-icing can be secured by means of a conveyor on an elevated platform running both ways from a central ice-storage building.

A number of devices for mechanically operated units have been suggested. The lack of flexibility prevents the general use of the single head-end system, so that the mechanical unit has to be supplied to each car. Among those that have been attempted in the United States is the silica gel adsorption machine using sulphur dioxide and gas heat. The complete unit requires two units for the gel, an air-cooled condenser, and a pipe evaporator, with drip pans, with the necessary thermostat and actuating valves. There are also some cars supplied with compressors receiving power directly from the driving shaft.

The insulation should be of the flexible blanket type, 2 in. thick, placed inside the frame and protected by an inner lining of 1¾-in. wood running horizontally. The car is usually made with a steel underframe and with wooden bodies, using 2-in. wood frames on the sides, floor, and ends. Outside the frame is a wood insulation ½ in. thick covered with paper and a sealing compound, and this in turn is protected by a 1¾-in. vertical wood sheathing. The ceiling usually has 2½-in. flexible insulation and an air space. With a mechanical unit it is usual to have

a steel superstructure and better insulation in which case the heat leakage is estimated to be from 90 to 100 Btu/hr per degree difference of temperature between the inside and outside air temperatures. The heat leakage in ice refrigerator cars is about 160 Btu/hr per degree difference of temperature. The infiltration loss is not accurately known, but it must be considerable.

21-10. Passenger Cars. Refrigeration applied to freight cars and ships is comparatively old, but extensive comfort cooling for passenger cars is recent. With passenger cars in the United States the load factor is small, and therefore the tendency at first was to look with favor on the low-first-cost, high-operation-cost installations using ice. Railroad car air conditioning requires† special consideration inasmuch as the apparatus must be compact and free from leaks; it must be light, reliable, easily accessible, quiet in operation, and low in first cost, and it must cool the car without drafts.

The capacity requirements of passenger car machines are approximately 4½ tons for the Pullman, 5½ tons for the diner, and 7½ tons for the coach, and the power requirements are approximately 4 kw for ice cooling, 12 to 15 kw for mechanical refrigeration with electric drive, 6 kw for steam jet, and 4 kw with direct drive from the shaft. With ice cooling, the dead weight of the insulated box, etc., varies from 3300 to 5500 lb and the ice charge varies from 1500 to 4500 lb. Although this method demands a bulky design, and a system that is awkward to re-ice in transit, it has advantages for small load factors. Steam jet refrigeration for short trains probably has a future as it is very reliable and requires little attention and maintenance.

The direct drive to the compressor from the shaft, or to a generator of sufficient capacity for the compressor, involves a number of difficult problems, such as the belt or magnetic clutch and means of providing power when the train is still or operating under 20 mph. The head-end compressor system is satisfactory in the streamlined trains, but is not flexible and is awkward to operate in the baggage car. The compressor unit requires dichlorodifluoromethane or some other non-toxic refrigerant.

Cost for passenger car cooling (from a study by the Association of American Railroads)

	Gross Installation Cost	Operating Costs
Ice-cooled	\$3980	\$ 9.48 per 1000 train-hours
Electric-driven	6480	12.63 per 1000 train-hours
Steam jets	8480	14.25 per 1000 train-hours
Power from shaft	8520	14.49 per 1000 train-hours

† Sahlmann, *Refrig. Eng.*, May, 1934.

Operating costs are based on a 3-month operating season. These costs, fixed charges, operation and maintenance costs, are calculated on the basis of an average speed of 50 mph and a total mileage of 150,000 car-miles per year.

21·11. Refrigerator Trucks. Although truck bodies, constructed with full post and sheathing and insulated with cork, are used, the pliable blanket insulation is coming into more general favor. Insulation with blankets 2 in. thick corresponds to 45° F, and two 2-in. blankets to 10° F inside temperatures. Especial care must be taken to waterproof the outer surface of the insulation.

Refrigeration may be secured by means of ice or ice and salt, frozen brine, solid carbon dioxide, or a refrigerating machine. Frozen brine may be placed in a lead-coated container usually holding 10 lb of brine and capable of absorbing approximately 1000 Btu at 6° F. Dry ice is utilized by permitting the gas to circulate through the double walls of the compartment, the finned metal ceiling plate, or through special pipe coils on the side of the truck.

A refrigerating machine for a truck may be driven in a variety of ways. The compressor and condenser fan may be belt-driven from a gasoline engine, or the engine may be connected by a belt drive to an electric generator which furnishes electric power to a motor, or an electric motor may be included in the equipment to be used in the garage or any other place where electric power is available. The truck engine may be utilized by means of a power take-off from the truck to the generator, provided that an automatic regulator keeps the voltage delivered by the generator constant, or through a special speed device to keep the rotative speed of the compressor below the maximum. In this case an electric motor is provided for operation when the truck is in a garage. The evaporation surfaces are usually finned coils on the walls, protected by wire mesh, with drip pans if needed.

21·12. Marine Refrigeration. Fundamentally, marine refrigeration design problems do not differ from the problems associated with shore refrigerating establishments. Emphasis, however, is likely to be placed on performance or operating characteristics which reflect the very specialized conditions that exist on shipboard. One such specialized characteristic is the relatively short time (in comparison with a cold storage warehouse) of storage and the consequent greater interest in the transient performance characteristic of the refrigerating system during the initial "pull-down" period. Of equal interest is the heating-up period which occurs during breakdown, during operation at reduced capacity, or in the interval accompanying removal of a low-temperature cargo and replacement with material having a higher carrying tempera-

ture. A variation of the latter problem occurs for the special case of emergency operation of a freezer box with auxiliary equipment incapable of handling the heat gain associated with the desired inside air temperature. In this case the temperature of inside air must rise, but the rate is dependent on both the thermal capacity and the thermal diffusivity of the stored material; transient analysis is useful in determining the maximum permissible time for such operation prior to realization of the critical inside air temperature.

The methods employed for heat-up and pull-down analyses of ships are those which have been developed in Chapter XI; simplified forms of these general equations can be developed, however, for the special case of transient flow which occurs when subcooling is used as a means of storing refrigerating effect. The concept of subcooling as a means of providing a temporary refrigerating capacity is not limited to marine service, but it is of special interest in ship work because it affords a simple means of making up for inadequate insulation or for refrigerating equipment. Subcooling is not recommended as a substitute for adequate refrigerating capacity or for effective insulation, but where either or both of these are lacking it does have possibilities as a rapid, relatively inexpensive, remedial measure and as a means of increasing cargo capacity and utilizing otherwise ineffective compartments for cargo which requires cold storage.

Refrigerator ships (Fig. 21·6) are handicapped because space is so valuable that the use of insulation is limited. Moreover, the insulation (Fig. 21·7) has to be applied so that panels may be removed for inspection and repair. The cargo varies greatly in amount of perishables, temperature requirements, and relative proportions. Frequently the owners have to take cargoes not requiring refrigeration. Danger of breakdown on the high seas has to be minimized, even by the alternative of adding considerably to the dead weight. Steam-driven compressors have been used considerably, but more recently the electric-motor-driven compressor has gained in popularity. Carbonic compressors have been widely favored, because of both their lack of danger from leaks and their small piston displacement; but the loss in capacity in localities where the sea water is warm, the heavy pressures, and the resulting loss of the refrigerant make their value debatable if other types are available. At times ammonia has been used when the compressor is located in a separate compressor room, separately vented, or on an upper deck. In modern design, however, dichlorodifluoromethane or other non-irritating refrigerants are being used in preference to either ammonia or carbonic compressors.

The British have led in the equipment of cargo refrigerator ships,

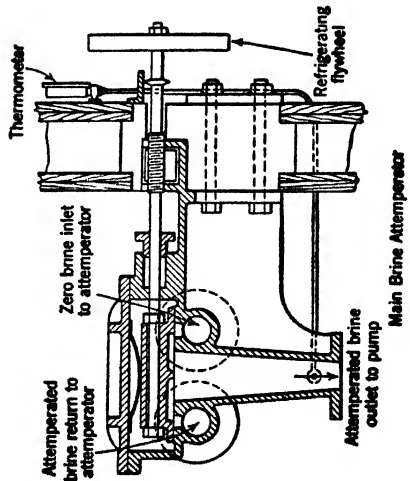
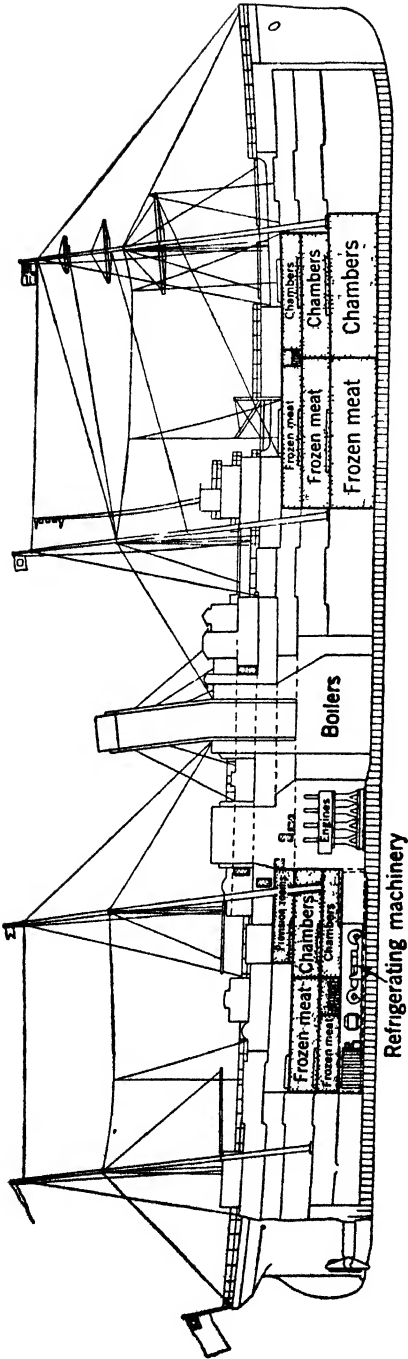


Diagram Arrangement of Attemperated Brine System.

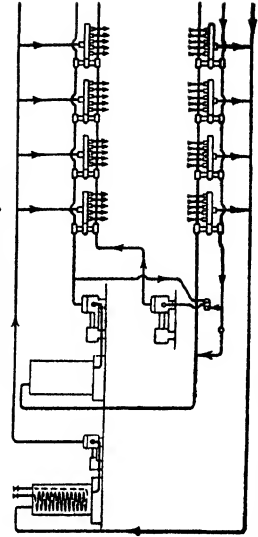


FIG. 21-6. Marine refrigeration details.

and they have more or less standardized an elaborate brine circulating system using † “a tangle of piping generally rather appalling to those accustomed to land practice. On one ship there are 54 cargo spaces including 24 hatch trunks and 3 provision rooms with a total of 560,900 cu ft. These spaces are piped with 195,637 ft of cooler pipe and 46,250 ft of service pipe; including the evaporator and condenser items there are 60 miles of refrigerating piping.” The tendency in American vessels

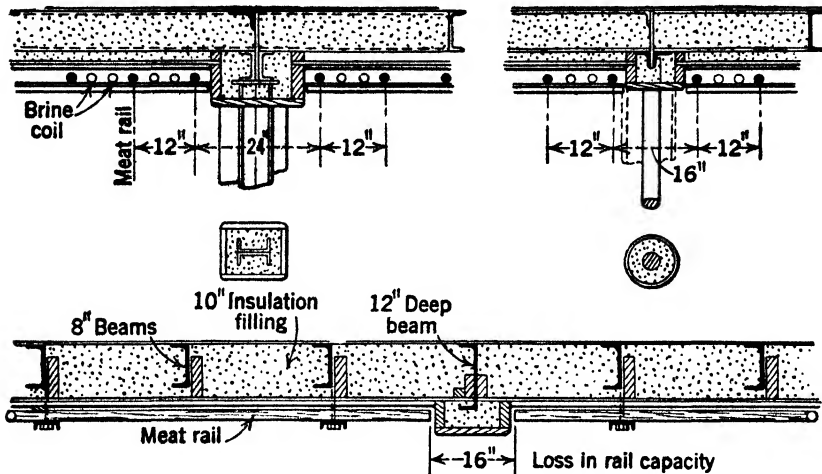


FIG. 21-7. Marine refrigeration—piping, insulation, and meat rails.

is to use the modern land practice of standardized evaporators, with forced air or some modification of the brine spray chamber, and unit coolers.

21-13. Precooling. In the shipment of fruits and vegetables, usually harvested at temperatures from 70° to 80°, it is essential that cooling be accomplished with the utmost speed. To do so in the refrigerator car by means of natural convection from the end ice bunkers is very nearly impossible, especially near the center of the car, and to have the produce wait at the port of departure for loading in a warm condition would result in considerable spoilage. It is very necessary to bring the temperature down to the vicinity of 32° promptly, thus slowing up the physiological process of ripening and decay and also checking the bacterial activity. Precooling may be done in a number of ways:

(a) The preferred method in the United States is to use air with a uniform distribution and rapid circulation. The boxes and crates must be constructed so that the air may enter into the interior readily, other-

† McKenzie, “Refrigeration on Board Ship,” *Refrig. Eng.*, January, 1932.

wise the quality of the center of the crates will suffer. The air may be cooled by brine or water sprays or by means of finned coils.

(b) With a few exceptional products, like grapes, celery, and lettuce, a cold water spray thrown on the crates has been successful. The cooling action is rapid, and the thorough wetting not only does not damage the lading but frequently improves it. In some cases crushed ice may be placed directly on the crates (Fig. 21·8).

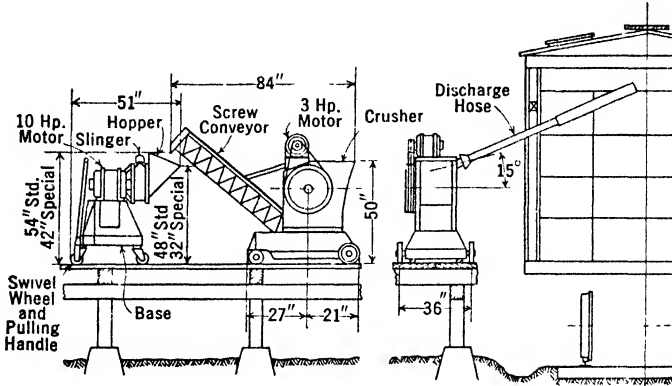


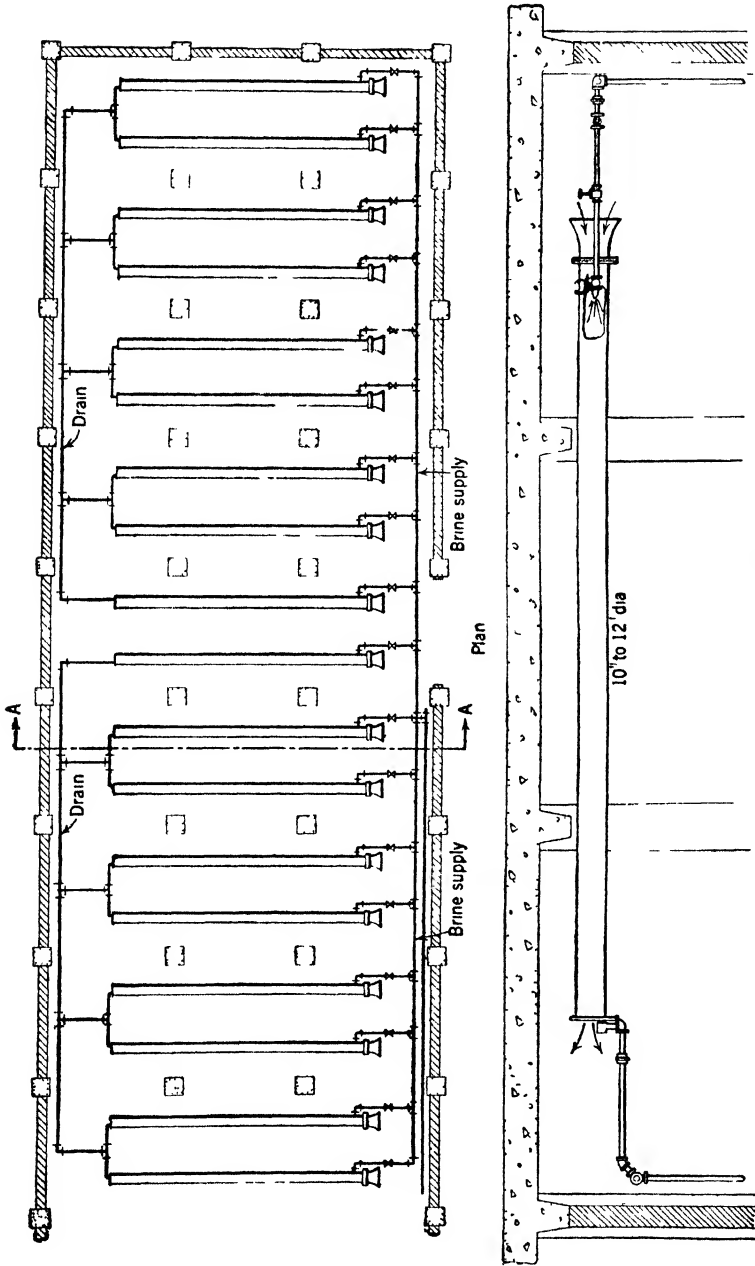
FIG. 21·8. The portable ice slinger with crusher and inclined conveyor.

(c) If the crates are loaded on a freight car a special arrangement of forced circulation of cold air through the middle (loading) door into the car has been successful in quick lowering of the temperature to the desired point. Then the ice baskets are expected to maintain this temperature.

MISCELLANEOUS STORAGE

21·14. The Packing House. Packing house refrigeration differs from cold storage installations mainly in the heavy live loads required in the cooler space and in the manufacturing processes called for. The packing plant has coolers and chill rooms held at 30° to 31° carrying freezers at -10° to 0° and sharp freezers at -15° to -10°. In addition, refrigeration is needed for making ice and for the manufacture of lard and compound.

The overhead bunker type of chill room, using both brine sprays and pipe coils (Fig. 21·9), has been popular with packing plant engineers for a considerable time; if the bunker is properly designed the circulation of the air is satisfactory. Pipe coils tend to dehumidify the air and increase the shrinkage of the meats being chilled. Meat shrinkage



Section A-A

FIG. 21 10. Air cooling with brine sprays

is decreased by having sodium chloride brine of just sufficient concentration to prevent freezing at the brine temperatures being carried, which may be slightly below 32°. Brine-spray nozzles may be $\frac{3}{8}$ in., $\frac{1}{2}$ in. or $\frac{3}{4}$ in. in diameter spaced on centers from 1 to 5 ft, but usually not more than 24 in. The spray pressure depends on the size of nozzle and the quantity of brine desired, and it varies from 6 to 15 psia. Owing to its contact with the air the brine tends to become acid; therefore it should be kept alkaline with calcium hydrate. Typical spray and piping systems are shown in Figs 21·10 and 21·11.

Example. Find the weight of brine required to chill 50 beefs at 750 lb each from a temperature of 90° to 38° in 12 hr, using brine at 32° initial and 35° final temperature.

Solution. To cool the meat there will be required

$$\begin{aligned} 50 \times 750 \times [0.77 (90 - 38)] &= 1,501,500 \text{ Btu} \\ \text{Allowing 20 per cent for losses} & \quad 300,300 \text{ Btu} \\ \text{Total} & \quad 1,801,800 \text{ Btu} = 12.51 \text{ tons} \end{aligned}$$

The amount of brine required (see properties of NaCl brine, Chapter XX) will be

$$\begin{aligned} W \times 0.913 \times 3 &= 12.51 \times 200 \\ W &= 913 \text{ lb/min} \\ &= \frac{913}{8.33 \times 1.051} = 104.3 \text{ gpm} \end{aligned}$$

21·15. Lard Chilling. A number of manufactured products require quick chilling at the end of the cooking or rendering process. Such a method has been used in lard and compound cooling in packing plants, where a large refrigerated cylindrical roll, about 4 ft in diameter and 6 ft long, revolving at 10 to 14 rpm, congeals the lard on its surface. The roll revolves partly submerged in the liquid fat whose initial temperature is about 80° F. When the lard is solidified a long knife scrapes the product off the surface into a trough. The refrigeration involves the cooling of the liquid fat and the removal of the latent heat of fusion of the substance.

Example. Taking the specific heat of the liquid and solid as 0.6, and the latent heat of fusion as 90 Btu/lb, find the required refrigeration if 10,000 lb of lard are to be cooled in 4 hr. Find the piston displacement of the ammonia compressor required to do this cooling.

Solution. The total cooling Q , if the lard is cooled to 30° F, is

$$\begin{aligned} 10,000 [0.6(80 - 30) + 90] &= 1,200,000 \text{ Btu in 4 hr} \\ &= 300,000 \text{ Btu per hr} \\ &= 25 \text{ tons of refrigeration} \end{aligned}$$

If 10 per cent has to be added for losses the tonnage developed by the compressor will be 27.5 tons. As the compressor has to cool the brine to 10° F or

less the boiling temperature of the ammonia in the brine cooler may be assumed to be 0° F, corresponding to a boiling pressure of 15 psig. With such a pressure there will be required 4.63 cu ft piston displacement per ton of refrigeration per minute (using a volumetric efficiency of 0.83); therefore the total piston displacement for 27.5 tons will be 128 cfm. The method of calculation for piston displacement is shown in equation 3·12.

21·16. Fur Storage. Furs, carpets, and valuable material of a similar nature, according to the U. S. Department of Agriculture, can be expected to be fully protected against clothes moth larvae or worms if the necessary holding conditions are maintained. The recommended procedure is to bring the temperature of the stored goods to 10° F for a few days, then for a short time to 50° in order to hatch the larvae and finally return the temperature to 10° F again, the change of temperature being expected to destroy the larvae. The permanent temperature is from 40° to 45°.

21·17. Unit Space Coolers. The outstanding differences between air cooling and conditioning by means of unit space coolers and the cumbersome duct system are that the unit cooler is located in or adjacent to the space to be cooled; that it is comparatively small and self-contained; and that greater control is possible, especially in multiple installations. In such installations it is possible to have as many different temperatures as there are rooms. The unit can be moved at slight expense if moving is desired. The first cost is usually much less than in other methods of cooling, and the air temperatures are uniformly equal. If the operation is made automatic by the use of magnetic stop valves and float valves or direct expansion (Chapter XVII) full flooded condition of operation is possible.

The unit cooler (Fig. 21·12) consists always of cooling surfaces, a motor-driven fan, or several fans on the same shaft, and some sort of deflector or directional outlets. The air is usually directed horizontally at velocities of about 1000 fpm, near the ceiling and above the product, and it is found to be well diffused throughout the room. The suction return for the air is placed at the floor level for the floor-mounted types and near the ceiling for the suspended types. The coils may be designed for direct expansion or for brine. The unit cooler naturally divides itself into three classes, for cooling below 33° F, above 33°, and for comfort cooling.

For temperatures below 33° F or where excessive moisture is given off by the products it is desirable to install a brine spray unit with eliminators. The unit consists of a brine tank and pump, coils, spray headers and nozzles, eliminators, and a fan. The value for the coefficient of heat transfer for the cooling surfaces of 140 and more is obtained.

For temperatures above 33° F the surfaces may be defrosted by a shutdown of the machine or by the use of two or more cooling surfaces equipped with by-pass defrosting valves permitting one section to be defrosted without shutting down the other. The brine spray system may be used for this temperature range also, if desired, thereby decreasing the amount of cooling surface necessary. By the use of brine spray, or water spray where operating conditions permit, the humidity is easily controlled by raising or lowering the temperature of the spray. Where

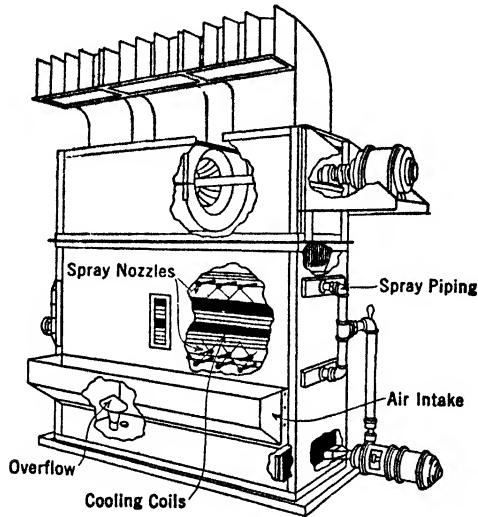


FIG. 21-12. The unit cooler.

brine spray is used the coils may be designed for direct expansion or for brine. In any event the brine has to be reconcentrated either by the addition of more salt or in some other way. The cooling coils, fans, scrolls and casing should be galvanized or painted with a moisture-proof paint.

The refrigerating machine as a rule is located at some convenient point outside the room to be cooled. The capacities of unit coolers vary from $\frac{1}{2}$ to 20 or more tons, and the condensers therefore are water-cooled. The connected power load on account of the fan and the sprays may be one-third of that of the compressor.

21-18. Pipe-Line Refrigeration. Pipe-line refrigeration has to be restricted to the warehouse, commission and retail districts where the use of refrigeration is concentrated. As a rule the pipe line once installed does not permit much extension, and additional contracts can be taken only by adding new pipe lines. The load can be carried successfully by

means of both direct expansion and brine piping. In spite of the fact that the brine system costs from 40 to 60 per cent more than direct expansion very few large installations in the United States use the direct ammonia system.

In the *direct-expansion ammonia system* no attempt is made to return cold vapor to the compressor; therefore neither the feed nor the return is insulated. In addition to some 50° to 60° F of superheat there is a drop in pressure of the suction vapor of 10 to 15 psi. Owing to the danger of slugs of liquid ammonia returning to the compressors it is essential that an extra large suction trap be installed in the compressor room. The pipe line installed in the street consists of a liquid, a gas, and a vacuum or pump-out line. The pump-out lines are connected to both the liquid and the vapor pipes in order to make any necessary repairs. All pipes are extra heavy and are welded whenever possible. The actual cooling may be done by means of direct expansion of the ammonia or brine or both and, in air conditioning, of cold water sprays.

In the *brine system* special care has to be taken to prevent the loss of brine. By an arrangement of indicating and control devices the operator can detect the loss of half a barrel of brine during a short time period and shut the main stop valve in that particular division by means of motor-driven valves. The cost of pumping depends on the installation.

Brine pipes should be full-weight steel or wrought iron pipe with flanged joints and ring gaskets. In some pipes the alternate joint is made of heavy, cast iron sleeves and is calked with lead, or it may be of cast iron using the bell-and-spigot, the flanged-and-screw-thread, or the gland-end types. In some systems a dual system is employed so that in the event of a leak the other system will carry the load during the period of repair. It is necessary to place all piping below the frost line, from 3 to 7 ft below the surface, and when possible to confine all connections to the manholes which preferably are located at intersecting streets. All pipes between manholes should be welded. Expansion joints should be provided every 500 ft at a maximum and every 300 ft where heavy distributing pipe lines are taken off the mains. These expansion joints should be located at the manholes and may be of the slip-joint, the corrugated, or the U-type, and the anchor should be located somewhere midway between to reduce the actual movement to the lowest amount possible. If a slip joint is used it should be kept smeared with grease to prevent tearing of the packing of the joint.

Valves not placed in the manhole should have an extension valve stem. Service branches should not come up into the frost line but should take off from the top of the main, using a nipple of sufficient

length and an ell to give flexibility of movement so as to conform with the expansion and contraction of the main. All coils in the boxes should have stop valves in the supply and the return lines and a lock valve to regulate the supply of the brine. The brine coils should have air valves at the high points. Large mains should be carried in tunnels.

PROBLEMS

1. A two-room cold storage warehouse, designed for an army camp, had one room for meats which received 24,000 lb per day. The other room received daily 3000 lb of apples, 9000 lb of potatoes, 630 lb of butter, and 1200 lb of eggs. The butter and meat were delivered at 60° F and the other commodities at 80° F, and both rooms were to be kept at 33° F. The specific heats are: beef 0.77, apples 0.92, potatoes 0.80, butter 0.60, and eggs 0.76. The heat leakage was 6600 Btu/hr for the meat room and 4650 Btu/hr for the vegetable room, allowing for door losses. The entire load was designed to be handled in 18-hr operation of the compressor. Find (a) the capacity of the refrigerating machine in tons; (b) the required piping for the meat room, in order to cool the commodity to 33° F in 18-hr operation. Two-inch piping is to be used. Brine rises from 21° to 25° F.

2. A simple cold storage plant has the following load: (1) the heat leakage in the cold storage room is 200,000 Btu/hr; (2) the milk cooling of 1000 gal/per hour from 80 to 40° F, the specific heat is 0.9, specific gravity is 1.03; (3) the manufacture of 1000 lb of ice per hour. (Assume that 220 Btu/lb of water frozen is required of the refrigerating machine.) The brine is carried at 15° F, and liquefaction of the refrigerant is at 80° F. Find (a) the size of a single-acting twin-vertical ammonia compressor at 200 rpm, (b) the horsepower required to drive the compressor, allowing 10 per cent of the theoretical power for friction. Evaporation temperature 10° F.

3. A small cold storage plant has the following load: 100 lb of ice manufactured per hour taken as 220 Btu/lb; 5000 Btu/hr for heat leakage; 2000 lb of produce per hour cooled from 60° to 40° F, specific heat. Find the size of a twin-vertical, single-acting ammonia compressor at 200 rpm operating at 80° F liquefaction and 25° F evaporation with ratio of diameter to stroke of 1.0.

CHAPTER XXII

ICE MANUFACTURE; DRY ICE

22.1. Introduction. The use of natural ice has been replaced, to a large extent, by that of manufactured ice since the beginning of the twentieth century, because of the uncertainty of the natural supply due to an occasional open winter and also because of the cost of transportation to the retail trade. In general, natural ice cannot compare in appearance with the crystal cake that can be manufactured and that is now generally demanded by the householder in the United States, nor is the natural ice likely to be as clean.

The *manufacture of ice* is still one of the principal needs of refrigeration engineering, and undoubtedly it will continue to be for some time to come as ice is the cheapest means for short-time preservation of food if the amount used is nominal and if the locality can furnish it at a reasonable price. As the cost of ice is an important factor a vast amount of research has been put into developing an inexpensive means of manufacture. There has, however, existed a tradition in the United States that ice must be crystal in appearance, for which reason the method of manufacture has vacillated from the production of the so-called raw-water plate ice to that of distilled water can ice and, more recently, to that of ordinary drinking water can ice with elaborate water treatment and agitation. Although the retail purchaser has seemed to feel that an opaque cake of ice is unhygienic and that a transparent cake is pure, there is no justification for this feeling, and it now seems possible that white ice will be accepted by the retail purchaser in the near future.

With the steam-engine-driven compressor and the steam-heated absorption machine, in the early days of manufactured ice it was convenient to use the condensed steam from the steam engine for freezing, and *distilled water ice* became a slogan. This condensed steam was filtered more or less perfectly to remove traces of cylinder oil and was then placed in galvanized iron tapered cans immersed in a cold sodium or calcium chloride brine. Where work was done carefully, this ice had a good appearance and not more than a trace of oil that could be detected by taste. At this same early period a flat coil or a hollow plate was immersed in a large tank of tap or raw water, as it was then

called, for freezing cakes of 3 to 5 tons apiece. The process took from 6 to 7 days even though the temperature carried in the plate or coil was 0° F or less. Coincidentally with the advent of the oil engine and particularly the electric-motor-driven compressor, deep-well or city water had to be used in the ice freezing cans (the plate system had been abandoned because it could not compete with the can system), and additional refinements had to be made in order to eliminate impurities

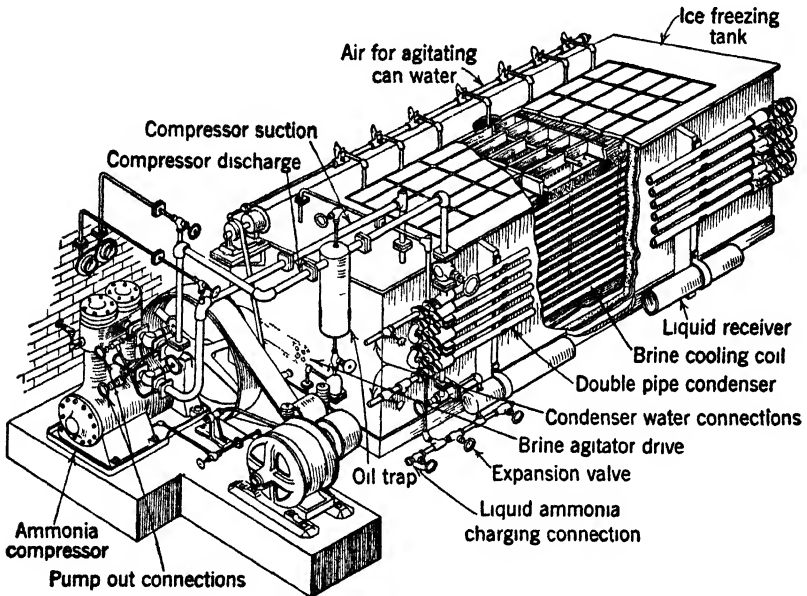


FIG. 22-1. Typical ice plant.

in the water and to reduce the power and labor costs of manufacture. Ice, being a heavy commodity, has to be manufactured near where it is to be used. For the retail trade the appearance is a large factor, and a preliminary chemical treatment and filtration of the water, agitation of the water during freezing to insure transparency, core sucking to remove impurities in solution in the raw water supply, and scoring of the ice cakes in sizes required for the trade are features of the process. On the side of expense the cost of power and labor has to be given careful attention; Fig. 22-1 shows diagrammatically a complete refrigeration plant for the manufacture of can ice.

22.2. The Building. Unless the cost of land is high the buildings should consist of one story on one level so arranged as to be completely visible to the engineer from some central point. The desirability of ice

storage is debatable, but the plant should have sufficient day storage for emergencies. For 60- to 130-ton daily capacity ice plants* the day storage should be from 800 to 1400 tons arranged for stacking in 4 to 7 tiers. The walls and roof must be insulated with granulated or sheet cork unless a cheap construction involving 12 in. of dry mill shavings on the walls and 24 in. on the roof is desired. The floor can be made with 36 to 42 in. of well-tamped soft coal cinders and an oak floor with $\frac{1}{4}$ -in. spacings nailed to 2- by 3-in. screeds.

22·3. The Ice Tank. The ice tank (Fig. 22·2) should be of $\frac{1}{4}$ -in. riveted or welded steel and consist of a single tank up to 100 tons† daily capacity; it should be 54 in. deep for the 300-lb. can. Cans of this size are commonly used; they are 11 by 22 in. in top measurement, with a 1-in. taper, and 53 in. long, and they have number 14 gage sides and number 12 bottoms welded and galvanized. A can of this size has a capacity of about 315 lb, and the tank is designed for 11 cans per ton of daily ice capacity although the tendency now appears to be to make the amount slightly less. The brine temperature should not vary more than $\frac{1}{2}^{\circ}$ F throughout the tank, and the brine velocity should be from 18 to 30 fpm. As a result there will occur about $1\frac{1}{4}$ -in. drop of brine level in 60 ft. Some plants have been designed to bleed brine through the bulkhead to stimulate brine flow, but this should be permitted only under exceptional conditions. Care should be taken to prevent excessive bypassing of the brine under the cans and near the sides of the tank. The brine level should be about $1\frac{1}{2}$ in. above the top of the ice when the freezing process is over, as it has been found that otherwise an excessively long time is required to complete the freezing. If the tank rests on unexcavated ground the ground should be well drained, and from 12 to 24 in. of well-tamped steam cinders should be supplied as a base for the corkboard insulation and the tank. The insulation should be at least 5 in. of corkboard on the bottom of the ice tank and from 12 to 18 in. of granulated cork on the sides. Freezing of the ground by the brine tanks may cause buckling of the tank and other more serious troubles.

The cans (Fig. 22·3) should be assembled in *grids* to pull a complete row at a time in the medium-sized tanks, with the 60-ton daily capacity as the minimum size, and from this capacity the half row is the economical number to lift at a time. The grids should be galvanized and should weigh at least 50 to 55 lb per can to prevent floating of the can. If the grid is designed for 24 cans it should have four lifting points. The cans should be spaced so that lifting a full row will not drop the brine level more than 1 in. The modern plant does not use wooden framing, but

* Brizzolara, *Refrig. Eng.*, February, 1930.

† Neeson, *Refrig. Eng.*, November, 1931.

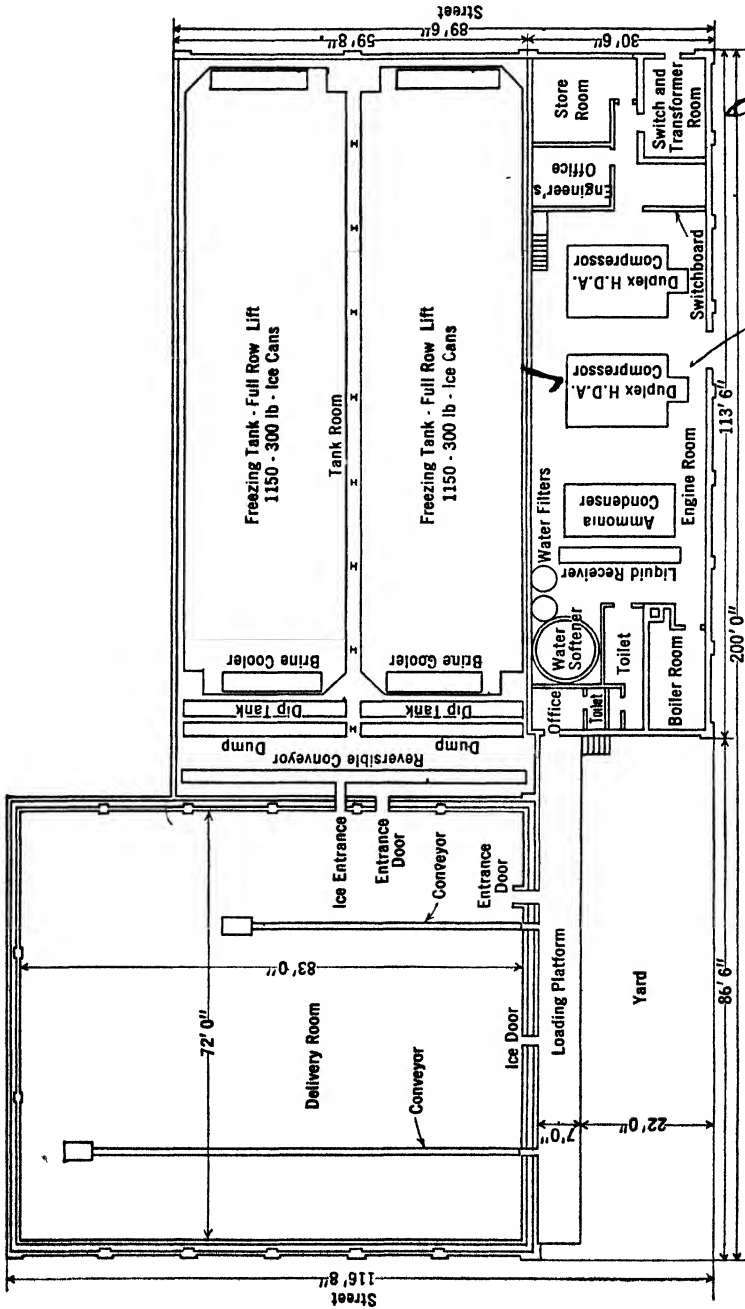
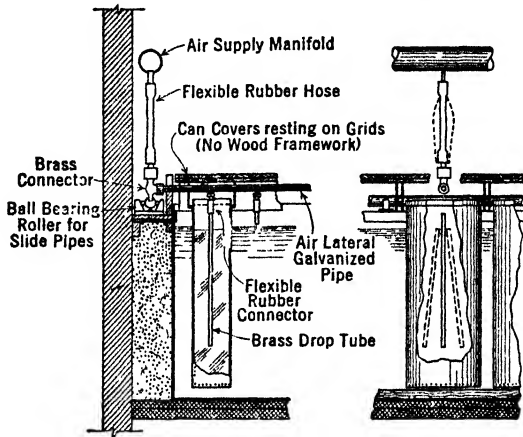
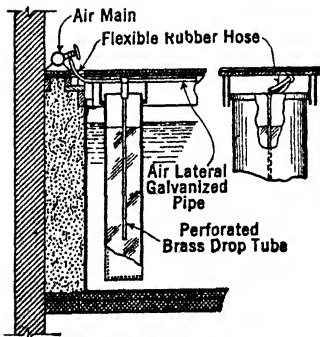


Fig. 22-2. Plan view of a modern can ice-manufacturing installation.

the wooden tank covers, which serve as an insulation and a working floor, are placed directly on the grids.



(a)



(b)

FIG. 22-3. Ice manufacture. (a) Drop tube designed to oscillate. (b) Perforated drop tube.

22.4. The Evaporator. The long trombone type of evaporating coil is no longer used, but the single-pass, shell-and-tube submerged brine cooler, the trunk and the brine-race types (Fig. 22·4), are economical of space and eliminate piping between the cans (see Chapter XX for a discussion of evaporators). Where piping is used it is short and is tied in with large headers to the suction pipe. The principle underlying evaporator design is to keep all the evaporator surface flooded with liquid ammonia, to remove the vapor formed as quickly as possible, and to have

a short suction line with low gas velocity back to the compressor. This permits high values of U for the evaporator surface, about 80 Btu/sq ft for the brine cooler and from 90 to 120 or more for the trunk or race design, and cooling of the brine to within 3° to 4° F of the evaporation temperature. In the shell-and-tube brine cooler some form of separator should be placed in the suction line in order to return any liquid ammonia

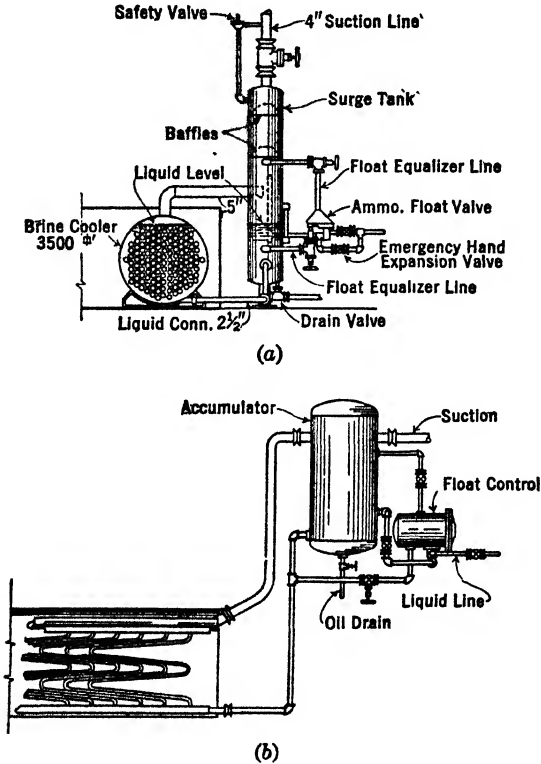


FIG. 22-4. Ice manufacture. (a) Shell and tube cooler. (b) Headers and short connecting pipes.

which tends to pass out of the evaporator as a result of priming. It is generally satisfactory to use one brine cooler up to 50 tons and two, one at each end, for 50 to 100 tons daily capacity. The brine velocity is about 2 fps in the tubes and about $2\frac{1}{2}$ to $2\frac{3}{4}$ fps in the race (Fig. 22-5) or trunk. The low-pressure float valve has been found to be much more satisfactory than the expansion valve from the viewpoint both of uniformity and of decrease of care in operation.

22.5. Dual-Effect Precooling and Subcooling. Since the water used in ice making is ordinarily admitted to the system at the temperature of the city water supply (possibly 60° F or more) a substantial fraction of the load on the refrigeration system will be represented by the need of cooling this water to its saturation temperature prior to freezing. If this precooling is accomplished with the refrigerant at the temperature required for the ice tank it is evident that a large temperature difference will exist, and this part of the load will be carried with an uneconomically low evaporator pressure. To overcome this inefficiency

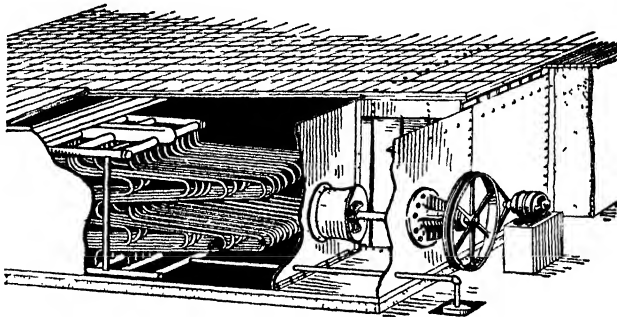


FIG. 22.5. (Courtesy Frick Co.)

a separate evaporator, operating at a higher pressure, can be used to precool water going to the ice tank. This method will also permit operation with multiple expansion valves and consequent extraction of flash vapor and recompression from the pressure of the intermediate evaporator.

Operation with water precooling requires, of course, either stage compression or use of a dual-effect compressor (refer to Section 4.7). Figure 22.6 shows a typical flow diagram for a modern dual-pressure ice plant operating with multiple-effect compression and with expansion valves in series (multiple expansion). Under normal operating conditions a plant of this kind will maintain from 35 to 40 lb gage pressure on the high-pressure suction of the dual-effect compressor.

22.6. Brine Corrosion. Both sodium and calcium chloride are satisfactory for use in the ice tank. Brines should be treated with sodium dichromate in amounts of approximately 125 lb for every 1000 cu ft of calcium and 200 lb per 1000 cu ft of sodium chloride. The dichromate is not injurious to the tankmen if care is taken about personal cleanliness. The exact concentration of the dichromate varies slightly with different plants, but according to G. V. Thompson† the

† Thompson, *Refrig. Eng.*, March, 1930.

best condition is obtained with the chemical indicator for a pH § of 7.0, when the solution is just neutral. An alkaline solution having a higher pH value has a tendency to remove the galvanizing and to cause the growth of sediment on the cans which acts as an insulation and tends to carry brine out and bring fresh water back to the tank during the dipping process. A strong alkaline solution also interferes in the test for ammonia leaks, making an extremely corrosive mixture where there is a tendency toward electrolytic action. However, the almost total removal of piping in the more recent plants assists in decreasing this tendency. There appears to be less corrosive tendency with calcium brine, probably because of more difficulty in introducing air into the brine. Under no circumstances should the brine be permitted to have a free fall or have any other opportunity for mixing with air.

22·7. Agitation. The brine agitator or propeller should have a capacity of 60 to 70 gal/ton/min. It may be either horizontal or vertical, although the vertical is more convenient when removal is required for repair. The agitator should be of medium speed and of large diameter, and the brine passages should be carefully designed in order to prevent undue loss of head. With tanks of 900 cans two 2-hp agitators are usually sufficient, or 1 hp per 15 to 20 tons of ice daily capacity. Excessive brine head results in increased kilowatt-hours per ton which may not be justified by local conditions.

22·8. Manufacturing Factors. For agitation of the water in the can, both *high-pressure air*, resulting in white tops, and *low-pressure air*, with cupped tops, are used, the ice being of approximately the same quality in each case, where the water is reasonably low in dissolved salts, but as a rule the high-pressure air requires more power and has to be dehumidified. With low-pressure air agitation (Fig. 22·7) approximately 0.5 cu ft of free air per minute is supplied at about 2 psi for the 300-lb can and 2.5 psi for the 400-lb one, whereas the high-pressure air is supplied at the rate of about 0.2 cu ft of free air per minute per can. In low-pressure agitation the air passes through a tube which may be allowed to swing or to be fastened in a central position. Water for ice manufacture varies in chemical composition so much that the need of

§ It is convenient to have a scale of measurement for alkalinity or acidity, and this is expressed as the pH value, which is the logarithm of the reciprocal of the positive hydrogen ions in grams per liter of water; this amounts to 10^{-7} gram for pure water or 10^{-14} for the sum of the hydrogen- and the hydroxyl-ion concentrations. A pH value of less than 7 indicates *acidity*, and greater than 7 *alkalinity*. Certain indicators give a means of determining what the pH value may be. For example, phenol red is yellow in a solution having a pH value of 6.8, and as the acidity becomes less it changes to an orange, and finally, with an alkaline solution with a pH value of 8.4, to a deep red.

operation of each plant must be determined by itself. Burks^{||} found that any commercial water, even up to 1300 parts by weight of solids per million, could be used successfully for the manufacture of merchantable ice with brine temperatures of 16° if the air agitation was effective and sufficient core removal was resorted to. The tube used in the low-

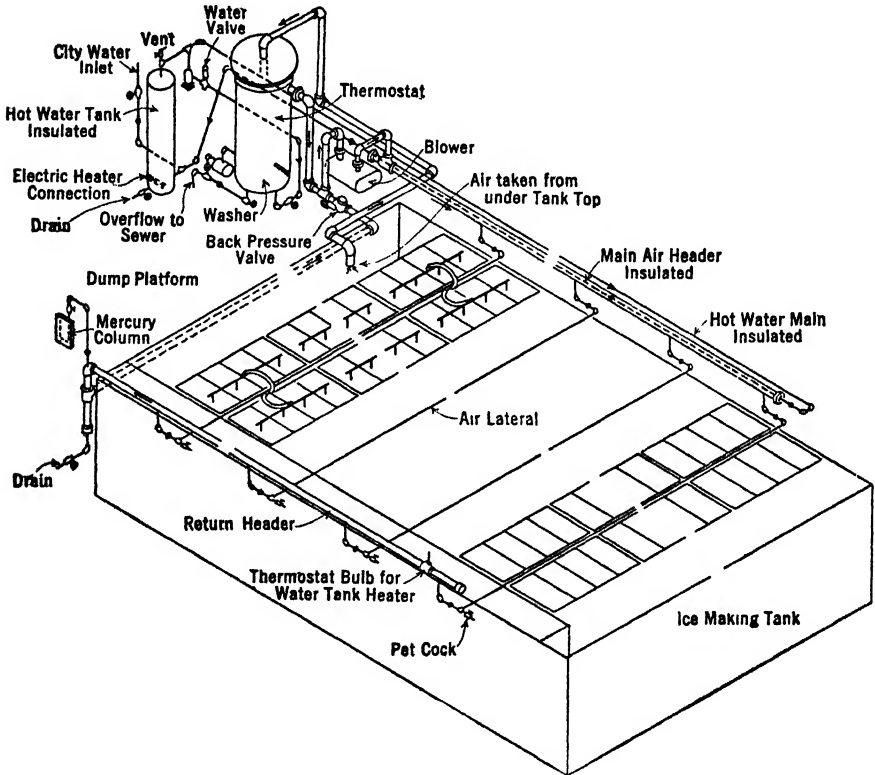


FIG. 22-7. Diagram illustrating the low-pressure air humidifier.

pressure air agitation may be permitted to freeze in the ice cake, or it may be removed near the end of the freezing process. In the usual high-pressure air system the air tube is soldered permanently to the can and the air is dehumidified to a point where the dew-point temperature of the air cannot be reached with the brine temperature in use. For this reason the dehumidification may be at a pressure considerably higher than that used in the laterals.

^{||} Burks, "The Treatment of Water for Ice Manufacture," *Univ. Illinois Eng. Exp. Sta. Bull.* 219.

The *time of freezing* of can ice has been found to be a function of the thickness of the ice, the brine temperature, and the amount of brine agitation, and it is given as

$$T_{\text{in hr}} = c \frac{x^2}{32 - t}$$

where c = a constant varying from $5\frac{3}{4}$ to $6\frac{1}{2}$, depending on the brine agitation; x is the thickness of the ice in inches; and t is the temperature of the brine in Fahrenheit degrees.

The *temperature of the brine* has been an important consideration for some time, especially in the Middle West of the United States, where the water used in the cans is heavy in solutions of sodium chloride, sodium sulphate, magnesium sulphate, calcium bicarbonate, and magnesium bicarbonate. At times agitation of the water in the cans has to be started before the cans are immersed in the brine. The presence of salts materially affects the tendency of the cakes to crack from strains set up during the process of freezing. Burks[¶] found that the cracking tendency was materially increased by the presence of chlorides and decreased by the presence of ammonium ions. It was found that, with care in annealing the ice and with the use of ammonium ions, cracking was prevented if the concentration of the dissolved salts was 500 parts by weight per million when the ratio of the chloride to the sulphate ions was 2.37, and 916 parts per million when the ratio was 0.097. However, the tests indicated that as a rule the control of the ammonium-ion concentration alone was sufficient to prevent cracking at a brine temperature of 6° F.

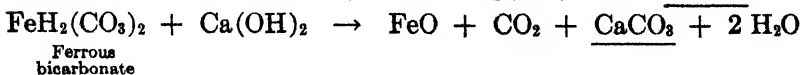
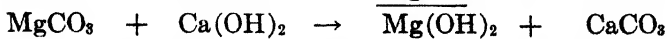
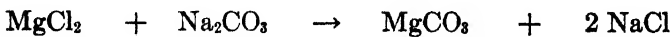
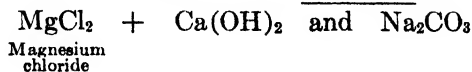
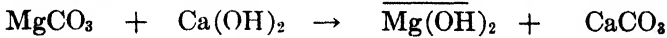
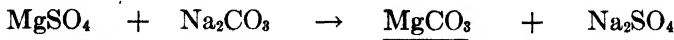
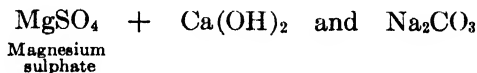
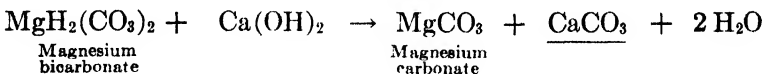
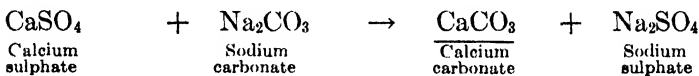
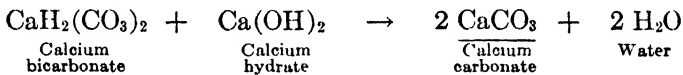
In general, the manufacture of ice is influenced by the rate at which heat is absorbed by the brine from the solution of water and salts and by the *normal rate of diffusion* of the salts in the aqueous solution. If the ice is formed rapidly (in modern practice within the first few hours after immersion) there is a tendency for the formation of a strong concentration at the surface of the ice, and diffusion must be augmented by *mechanical agitation* if opacity is to be prevented (Fig. 22·7). As the ice thickens, the rate of conduction of heat through the ice and the metal walls decreases and the freezing temperature of the core water decreases, so that the need of mechanical agitation decreases, and opacity will result only when the unfrozen water has reached the critical concentration. However, the separation of a eutectic mixture does not always produce opacity, as proved by solutions that have been lime-treated and are saturated with either calcium carbonate or magnesium

¶ Burks, "The Production of Manufactured Ice at Low Temperatures," *Univ. Illinois Eng. Exp. Sta. Bull.* 254.

hydrate, or both, where the solubility increases with a decrease of temperature. The pumping of a medium-sized core assists in the removal of the accumulated salts and in the raising of the freezing temperature, thereby promoting an early closing of the top of the cake of ice.

Water that has more than a trace of solids in solution is found to be improved by having a *special chemical treatment* before it is placed in the cans to be frozen. However, the treatment is one that takes place cold where the reaction progresses slowly and incompletely. The usual manner is to permit the water and the chemical, calcium hydrate or sodium carbonate or both (and sometimes alum as a coagulator), to stand for 4 hr in the water-softener settling tank. During the sedimentation period the chemicals sink to the bottom, where they are drained off as a thin sludge. The modern tendency is to allow more time for sedimentation, thus permitting more complete chemical action and settling. The maximum flow rate is 3 gal/sq ft/min of cross-sectional area and four times this rate for backwash.

The chemical reactions in the filter are



22-9. Dry Ice. The use of solid carbon dioxide, i.e., dry ice, had a remarkable increase during the decade beginning with the year 1925; in that year about 270 tons were produced, as compared with approximately 50,000 tons in 1931. Dry ice has been used particularly in the retailing of ice cream in place of water ice and salt. The refrigerating effect of dry ice is dependent on the final temperature of the carbon dioxide, and under one atmosphere this varies from 266 Btu/lb at -10° to 283 Btu/lb at 75° F; at 32° it is 275 Btu. Dry ice passes directly from the solid to the vapor phase, and, as the process at 1 atmosphere takes place at -110° F, it has the advantage of low temperature, dryness, and compactness. By increasing the pressure, solid carbon dioxide can be made to absorb heat in refrigerating coils at a higher temperature.

The principal *sources of carbon dioxide* are in certain fermentation processes, as in the manufacture of beer and industrial alcohol, the treatment of carbonates such as limestone, dolomite, and magnesite with acids, and the absorption of carbon dioxide from flue gases. In the last process (Section 6-1) the flue gases are washed and then brought into absorption towers where potassium carbonate or sodium carbonate lye or triestanolamine absorbs out the carbon dioxide. By the addition of heat the carbon dioxide gas is then driven off. After purification to remove water and odors the carbon dioxide gas is ready for the compression process, which is necessary before the solid phase can be obtained

The *solid phase* is produced by throttling from 70 or 75 atmospheres to 1 atmosphere, and the efficiency of the operation depends on the details of the cycle. As a rule the three-stage compressor is used, and, according to Stickney,* who took the amount possible with the Carnot cycle as 100 per cent and 80° liquid condensate at the condenser, the efficiency of snow production varies from 26 per cent in the simplest to nearly 60 per cent in the so-called bleeder-precooling cycle with isentropic compression. The power required varies from 250 in the simplest to about 140 kw hr in the bleeder-precooling cycle per ton of snow produced, with 80° F liquefaction in the carbon dioxide condenser; but in practice these values can usually be reached only approximately.

The throttling process is common to all the different cycles. In the simplest, the vapor caused by the constant h process is returned immediately to the compressor. In the second cycle the liquid passing to the expansion valve is pre-cooled by being brought into metal contact with the cold carbon dioxide gas after the throttling process. Still further economy is obtained by having three throttling processes and a separation of the vapor formed at each stage of the process so that it may

* Stickney, "The Thermodynamics of CO_2 Cycles," *Refrig. Eng.*, December, 1932.

enter the suction and be compressed in the corresponding pressure stage of the three-stage compressor (similar to the two-stage ammonia cycle described in Section 4.6). The result of this bleeder-precooler cycle is to reduce the power and piston displacement for the low-pressure and the intermediate-pressure cylinders.

Undoubtedly the operating cycle for the production of solid carbon dioxide will be improved in the future, and the cost of manufacture will

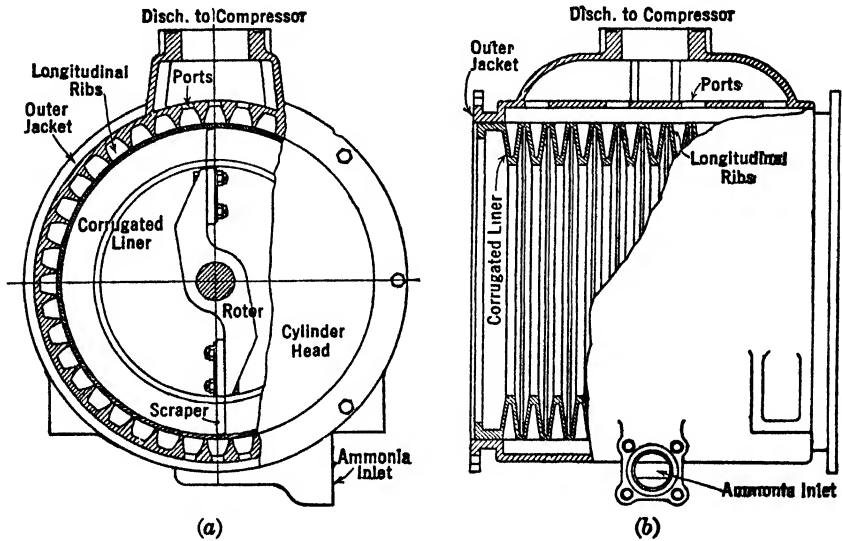


FIG. 22-8. Ice manufacture, Pakice.

be decreased. Dowling,[†] on the basis of an existing plant, gives an analysis of the cost of manufacture and fixed and delivery costs for a plant of 10-ton capacity per 24 hr, using the coke-burning recovery process. On the basis of an annual load factor of 55.6 per cent the cost per pound was 1.45 cents and the initial cost of the plant was about \$135,000.

22-10. "Flakice" and "Pakice." The plate ice process was objectionable particularly because of the long time required to complete the freezing. This fault is found to a lesser extent in can ice. Years ago the Holden process attempted to remove this objection by having a hollow refrigerated cylinder revolving with a portion at least of the outside surface in contact with the water to be frozen and a long knife adjusted so as to scrape off the film of ice as it was frozen. Later the snow was compressed into blocks of opaque ice. The ice was produced

[†] Dowling, "Dry Ice—Some Operating Data," *Refrig. Eng.*, September, 1933.

rapidly, as the heat transfer was exceedingly high through the thin film, but the machine was never perfected to the point required in commercial plants.

More recently the “Pakice” machine (Fig. 22·8) has been developed with a stationary corrugated cylinder, arranged for direct expansion in the surrounding jacket, and a rotating element with an adjustable cutting edge for the removal of the film. The snow ice is washed out

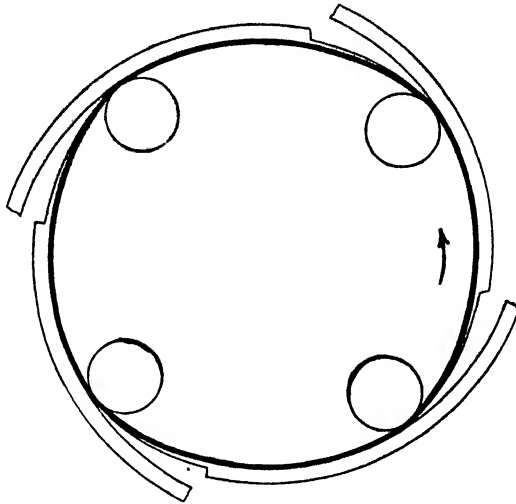


FIG. 22·9. Figure showing peeling of Flakice from the cylinders as they revolve.

of the cylinder and is filtered out of the water, which can be returned to the cylinder. The snow can be compressed into any convenient shape.

To overcome the inherent difficulty of a knife as used in the Holden machine the “Flakice”‡ process (Fig. 22·9) has a flexible cylinder assembled to rotate slowly while totally submerged in the water to be frozen. The refrigeration is applied from the interior, the mechanism including a number of rollers which can be moved radially outward and inward in order to distort the cylinder barrel sufficiently to break the flakes away from the metal surface after the film has reached a thickness of 0.12 to 0.25 in., depending on the setting of the machine. The rotation is slow; it takes about 2 min to secure a thickness of about 0.1 in. with an average evaporating temperature. The Flakice can be separated from the water by screening.

If in can agitation the compressed air is admitted to the bottom of the can by means of a tube soldered to the interior surface of the can,

‡ Field, *Refrig. Eng.*, October, 1931.

initial dehumidification has to be performed, for otherwise water vapor would condense in the tube and freeze. An example will show how this can be calculated for the refrigeration required.

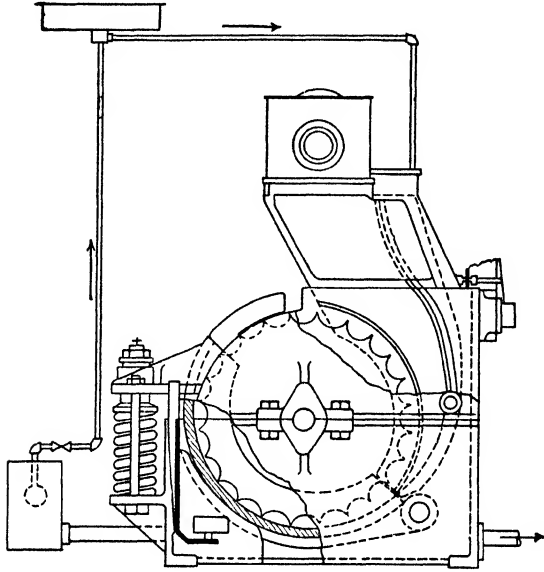


FIG. 22·10. The Pakice briquette press.

Example. At 1 atmosphere and 90° F dry-bulb and 75° F wet-bulb, 500 cu ft of air are compressed to 20 psig. After cooling to 80° F in an aftercooler the mixture of compressed air and water vapor is further cooled by means of brine coils in two shell-and-coil dehydrators to 20° F by counter flow of the brine, which enters at 15° and leaves at 20° F. Find the total length of 1¼-in. pipe if the dehydrator is fitted with four-way valves for reversing the flow of the air and the brine.

Solution. Weight of dry air,

$$\frac{500}{14.20} = 35.20 \text{ lb/min}$$

Volume of mixture at 34.7 lb and 80° F,

$$V = \frac{35.2 \times 53.34 \times 540}{144 (34.70 - 0.51)} = 205.9 \text{ cu ft}$$

Weight of water vapor at 34.7 lb and 80° F,

$$W = \frac{205.9}{632.9} = 0.3254 \text{ lb per 35.2 lb of dry air}$$

Volume of dry air at 34.7 lb and 20° F,

$$V = \frac{35.2 \times 53.34 \times 480}{144 (34.7 - 0.05)} = 180.6 \text{ cu ft}$$

Weight of water vapor at 34.7 lb and 20° F,

$$W = \frac{180.6}{5670} = 0.032 \text{ lb per 35.2 lb of dry air}$$

Latent heat removed,

$$(0.325 - 0.032) 1063.5 = 311.6 \text{ Btu}$$

c_p of the mixture at 80° F,

$$0.24 + \frac{0.325}{35.2} \times 0.453 = 0.2442$$

c_p of the mixture at 20° F,

$$0.24 + \frac{0.032}{32.5} \times 0.453 = 0.2404$$

Average value of c_p ,

$$= 0.2423$$

Sensible heat removed,

$$Q_s = 35.2 \times 0.2423 \times 60 = 512 \text{ Btu}$$

$$\text{Total heat removed, } Q = 824 \text{ Btu}$$

Average temperature difference,

$$\Delta t_m = \frac{(80 - 20) - (20 - 15)}{\log_e \frac{60}{5}} = 22.15^\circ \text{ F}$$

Area of heat-transfer surface, taking $U = 4.0$ (see Table 9-17),

$$A = \frac{824 \times 60}{22.15 \times 4} = 558 \text{ sq ft}$$

$$= 1285 \text{ ft } 1\frac{1}{4}\text{-in. pipe}$$

22-11. The Automatic Ice Plant. The small ice-manufacturing plant of 10 to 20 tons daily capacity can be designed for automatic operation for 15 to 20 hr of the day, in order to decrease the labor cost and to permit the attendant at the plant in the daytime to handle the office and sales. Such a plant needs to be electrically operated and to have an oversized ice tank. The cans need to be arranged in baskets so as to permit the ice pulling to be a quick and easy process, frequently of only 4-hr duration divided equally between morning and afternoon.

The brine temperature varies over a wide range from 12° to 15° because of filling the cans with water at approximately atmospheric temperature during the short period available. The compressors, selected to cover the range of load in the best manner and driven by oversized electric motors, have to be automatic, involving thermostatic control from the brine to the compressor motor, excess pressure cut-out, low-voltage release, float-valve operation to the shell-and-tube brine cooler, trunk or brine race, cut-out in case of decrease in brine flow for any reason whatever, etc. Where water treatment is required the water tanks will need to be oversized or a storage tank will have to be installed to receive the treated water. Such a plant could not utilize water requiring core sucking unless the freezing time was adjusted to include this process.

Typical of the best performance now being obtained from semi-automatic ice plants is the record of a one-man plant in Indianapolis which operates on a 10-hour work day; average daily output for the first 485 days of continuous operation (as of 1946) was over 40 tons per day.

PROBLEM

1. For a brine temperature of 5° F calculate the approximate cooling time to freeze a 2-in. thickness of can ice, assume excellent brine circulation

CHAPTER XXIII

SPECIAL APPLICATIONS OF REFRIGERATION ENGINEERING

For a large number of the special applications of refrigeration (other than the cooling and conditioning of air, the cooling and freezing of foodstuffs, and the manufacture of ice) the engineer needs only a knowledge of elementary physics, and he can calculate the amount of refrigeration needed by extremely simple methods. He would have to know among other fundamental physical facts the specific heat, the latent heat, and the heat of solution of different materials and the temperature range required in different processes. As examples of calculation methods the following are listed. *Cast iron* at about 1600° F will develop a surface hardness by being quenched in calcium chloride brine held at 45° F. The refrigeration required is the product of the weight of the cast iron, its specific heat, and its drop of temperature. During certain steps in its manufacture *rubber* is flattened by being made to pass between heavy metal rolls. Considerable friction occurs by which the rubber is heated. This heat, which may be 90 per cent of the heat equivalent of the power supplied to the rolls, must be removed by a refrigerating machine if cool water of sufficient amount cannot be secured in any other manner. Still another process requiring refrigeration is the freezing method of securing *potassium chlorate* crystals from a potassium chlorate solution in water. The aqueous solution is cooled to a desirable temperature by means of suitable refrigerated pipe coils. The necessary amount of refrigeration may be calculated when the specific heat of the potassium chlorate solution and the heat involved in the precipitation are known. The actual amount of refrigeration is the sum of the number of heat units, Btu, required to cool the solution and of the heat units required to cause the crystals to form. Some of the better-known uses of refrigeration follow.

23-1. Milk. Milk has a certain number of bacteria, and this number is materially increased by unsanitary methods of milking. Some kinds of bacteria double their number in 30 min when held at temperatures of 70° to 100° F, but the bacterial growth decreases rapidly as the temperature is reduced below 70° and practically ceases at approximately 32° F, although the bacilli are not killed even with temperatures con-

siderably below this. Certain ferments and fungi and rennet-forming udder cocci thrive at low temperatures, so that freezing of the milk does not preserve it, and in addition the structure of the milk is materially altered.

In consequence, milk should be cooled immediately after being drawn from the cow to temperatures as near 40° as practicable. Only a small fraction of the bacilli are dangerous to humans, but because of the danger that does exist most milk retailed in towns and cities is pasteurized. In

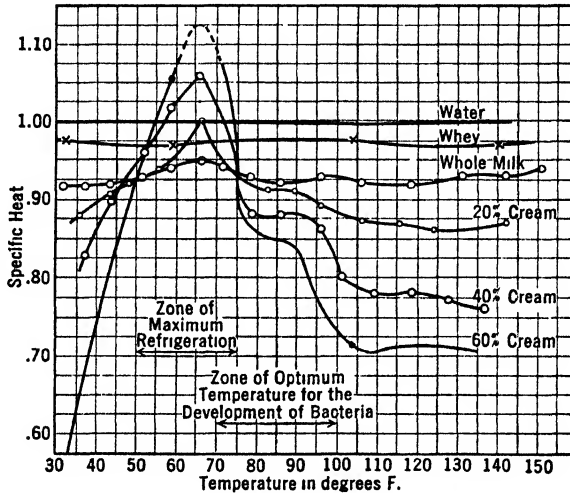


FIG. 23-1. Specific heats of dairy products.

pasteurizing, the temperature of the milk is raised to approximately 140° for a period of 30 min, after which it is cooled with water to about 80° and then to approximately 40° with cold brine or direct-expansion ammonia piping. The refrigeration is therefore this last cooling, and because of the short duration it is a peak load on the plant.

Example. Find the refrigeration required to cool 5000 gal of milk in 3 hr from 75° to 40° F by the use of brine. Also find the size of brine storage required to enable the compressor to be operated for 10 hr with the pasteurizer to be in use daily for 3 hr only.

Solution. The refrigeration (Fig. 23-1 and Table 23-1) is

$$5000 \times 8.33 \times 1.032 \times 0.9 \times (75 - 40) = 1,355,000 \text{ Btu}$$

$$\text{Adding 10 per cent for losses, heat leakage, etc.,} = 135,500$$

$$\text{Total} = 1,490,500 \text{ Btu}$$

The capacity of the compressor, for 10-hr operation, is therefore

$$\frac{1,490,500}{10 \times 12,000} = 12.4 \text{ tons}$$

TABLE 23-1
 SPECIFIC GRAVITY OF MILK AND CREAM
 (Products containing various percentages of butterfat, at 68° F)

Per-centage of Fat	Spe-cific Gravity	Per-centage of Fat	Spe-cific Gravity	Per-centage of Fat	Spe-cific Gravity
0.025	1.037			27	1.007
1	1.036	14	1.019	28	1.006
2	1.035	15	1.018	29	1.005
3	1.034	16	1.017	30	1.004
4	1.032	17	1.016	31	1.003
5	1.031	18	1.015	32	1.002
6	1.030	19	1.014	33	1.001
7	1.029	20	1.013	34	1.000
8	1.027	21	1.012	35	0.999
9	1.026	22	1.011	36	0.999
10	1.025	23	1.010	37	0.998
11	1.024	24	1.009	38	0.997
12	1.022	25	1.008	39	0.996
13	1.020	26	1.008	40	0.995

If a compressor of such size is in use the brine storage would have to carry what the machine could not deliver during the 3 hr that the milk is being pasteurized, or

$$1,490,500 - (12.4 \times 3 \times 12,000) = 1,044,000 \text{ Btu}$$

If brine at 0° is carried at the time that the milk cooling begins and if it is permitted to rise to 20° F, the average specific heat of calcium chloride brine of 1.2 specific gravity would be 0.71. The volume of the brine is therefore

$$\text{Weight of storage brine} = \frac{1,044,000}{0.71 \times 20} = 73,500 \text{ lb}$$

$$\text{Volume of storage brine} = \frac{73,500}{62.4 \times 1.2} = 981 \text{ cu ft}$$

In order to store this volume of brine there will be required a cylindrical tank 11 ft in diameter and 13 ft 1 in. high or the equivalent, although a somewhat taller tank would be supplied. Without the storage tank a refrigerating machine of 41.4 tons capacity would be required, but in order to cool brine at 0° the piston displacement of the compressor would be greater than for the higher evaporating temperature without brine storage, and the power to drive the compressor would be greater. The use of brine storage is a subject that must always be decided on the merits of the individual problem.

23-2. Ice Cream. Refrigeration is required in the manufacture of ice cream in cooling the mix after pasteurization, in holding the mix in cold storage during the ripening process, in freezing and in hardening the

frozen cream. In addition, cold storage is required for the storage of materials and, at times, for the making of ice. A homogenizer is used after pasteurizing to improve the final texture of the cream, and the mix is then cooled quickly to about 35° or 40° for the aging process of 24 to 36 hr.

The mix should enter the freezer at about 40°, and freezing usually starts at about 29° F. The increase of volume, the over-run, during freezing is from 80 to 100 per cent, depending on the material in the mix; it is due usually to the whipping of air into the mass by the freezer dasher, where it is retained in tiny cells. The batch freezers may be refrigerated by means of *direct expansion* or *brine*. If brine is used the rise of temperature of the brine should be limited to 5° and the initial temperature should be from -5° to +5° F. A good product requires a constant flow of brine. *Direct-expansion freezers* are gaining in favor because of the saving in space and lower initial and operating costs. Quick freezing, with evaporating temperatures from -15° to -20°, also appears to be gaining in favor because of the decreased size of the ice crystals and therefore the increased smoothness of the product.

The *continuous direct-expansion* freezer is also gaining in favor. A pump delivers the mix and the required amount of air in a thin layer at a high velocity over the freezing surface. The result is a smooth-textured product.

The ice cream leaves the freezer at 23° to 27° F with about 50 per cent of the water content of the mix in a frozen condition, and it is placed in containers initially colder than the cream to prevent melting on the sides of the container. These containers are then placed promptly in the hardening rooms held at -15° to +10°, although -10° is the preferred temperature. Nearly always still air is used in the hardening rooms, the direct-expansion piping being arranged on close centers to form the shelves required for the containers of the size used in the factory. The hardening room is piped as heavily as possible. Not all the water content of the mix is ever frozen, but the cream will usually reach the room temperature about 24 hr after entering storage. An anteroom should be supplied piped for about 30° and insulated with at least 4 in. of corkboard. The hardening room should have 8-in. corkboard for -10° and 10-in. for -20°.

Example. One thousand gallons of ice cream are to be manufactured per day. The over-run will be assumed, for calculation purposes, to be 60 per cent (Fig. 23·2). The mix enters the freezers at 40°, and the sharp freezer is held at zero.

Solution. If 1000 gal is the amount of the finished product the amount of the mix is $1000/1.6 = 625$ gal. If the specific gravity of the mix can be taken at

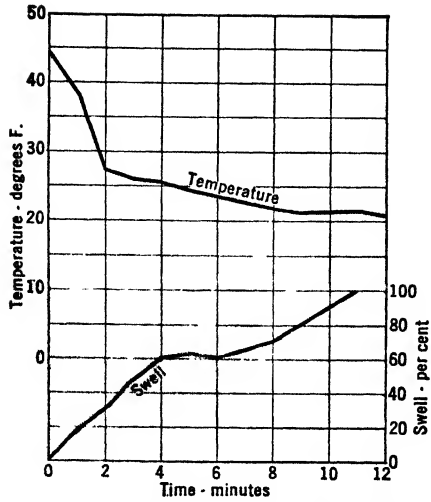


FIG. 23-2. Typical ice cream freezing curves.

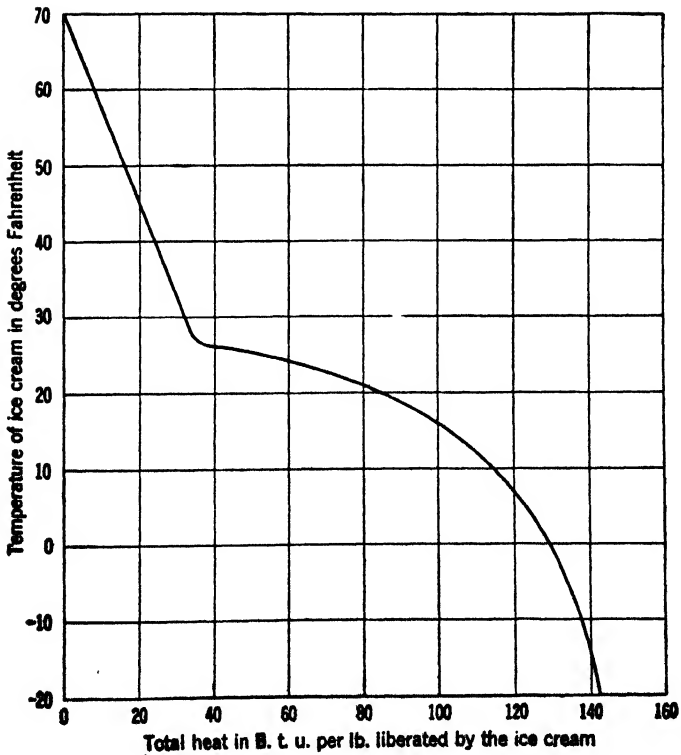


FIG. 23-3. Total heat of ice cream.

1.10, the weight per gallon will be 5.73 lb and the total weight will be 5725 lb. The refrigeration required (Fig. 23·3) if the cream is cooled to 25° is $5725 \times (60 - 24) = 206,100$ Btu. If an allowance of 10 per cent is made for losses the total refrigeration will be 226,710 Btu or 2.36 tons if the work is done in 8 hr. The refrigeration in the hardening room may be found by adding to the live load 70 Btu/lb (from Fig. 23·3), the heat leakage, infiltration losses from the anteroom, and the refrigeration necessary to cool the containers.

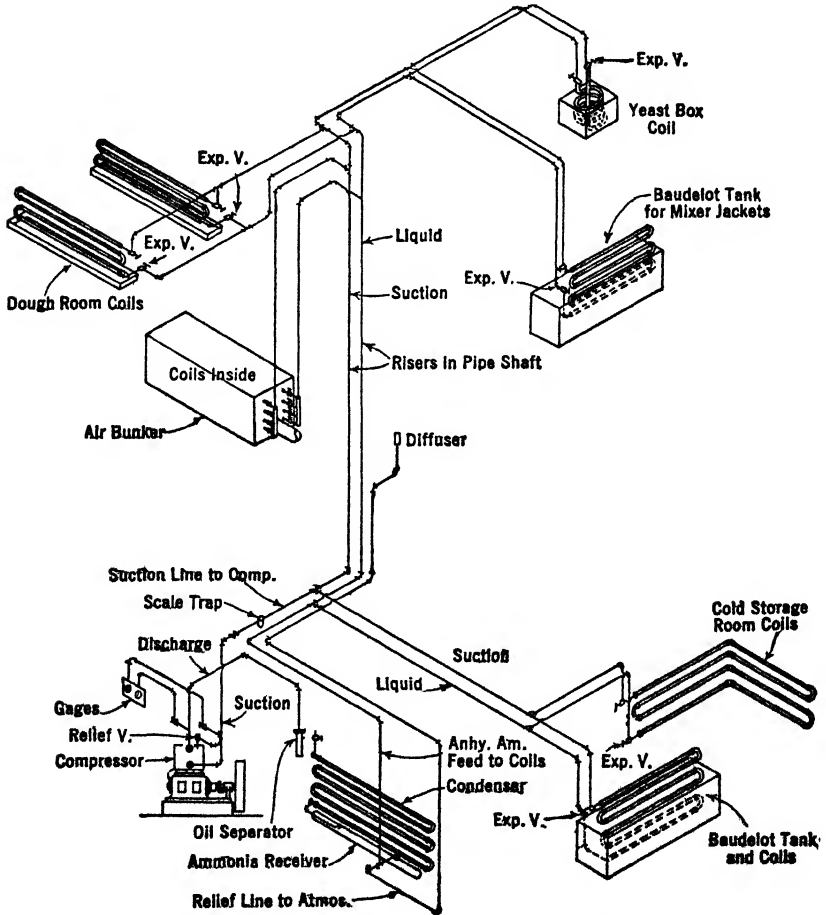


FIG. 23·4. Refrigeration piping for a modern bakery.

23·3. Bakery Refrigeration. The bakery (Fig. 23·4) is a good example of the application of refrigeration and air conditioning to an industry manufacturing foodstuffs where the production is on a very

close time schedule, where the margin of profits is small, where temperature and humidity control need not be exact but is within moderate limits, and where cleanliness is the first consideration. The problem involves: (a) cold storage rooms for yeast, flour, milk, shortening, and waxed paper; (b) mixing the dough; (c) fermentation; (d) make-up department; (e) proofing.

The *materials used* by the baker consist of flour, salt, sugar, malt, shortening, yeast, and waxed paper. The ideal storage conditions for flour are 78° and 75 per cent relative humidity, and if held at this condition flour will retain its normal moisture content of 13½ per cent and will not gain weight or shrink during the normal period of 4 weeks of storage. High humidities tend to deteriorate the quality of the bread. The perishable materials such as yeast, milk, malt, and shortening usually are best stored at 40°, whereas waxed paper may be held at 50° to 60° F.

As about 3 hp per 100 lb of dough are used in mixing, the heat equivalent of the work performed will cause the temperature of the dough to rise to a point where it will be burned unless some form of refrigeration is supplied. This is accomplished by the addition of ice water to the mix at 33° to 35° and by the circulation of water at the same temperature in the jackets of the mixer bowl. The substitution of chilled air for ice water appears to be undesirable although a small amount of air at 40° makes a finer texture and a whiter loaf. Crushed ice has been used in the mix successfully, especially in small commercial bakeries of 20,000 loaves per week. The finished mix should be at 78° to 80° F.

The *fermentation room* requires an air temperature of 78° to 80° and 75 per cent relative humidity, held so by uniform distribution of the conditioned air from a carefully proportioned duct system. The quality, character, and flavor of the loaf are affected by the condition maintained in the fermentation room. If the proper conditions are carried, there will be no crusting of the sponges and straight doughs, which results in bad lumps and streaks in the finished loaf, and the shop schedule will be very much more regular and lower labor cost will result. Air conditioning* permits a clear saving of approximately 0.7 per cent on the cost of bread making. Air conditioning should also be used in the make-up department, although it is frequently omitted. The dough reaches the dividers at 80°, and it should be held at this temperature and at a relative humidity of 60 to 70 per cent during the interval of make-up and the passage to the proofing room.

Proofing, the rising process before baking, should be at 95° and 85 per cent relative humidity. In cooling, the bread requires from 80 to 90

* Reece, *Chicago Section, Am. Soc. Refrig. Engrs.*, February, 1934.

min in 70° to 75° air and rather high relative humidity, unless loss of weight is to result.

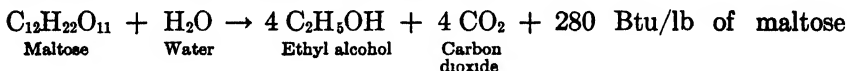
23·4. Brewery Refrigeration. The brewery (Fig. 23·5) is a special problem in air conditioning applied to a process involving the cooling of the wort passing from the kettle to the fermentation room from about 72° to about 48° F, and in maintaining a uniform temperature of fermentation. The storage and process rooms have to be held at temperatures from 32° to 48°.

The hot wort from the brew kettle is cooled to the temperature required for starting fermentation (40° to 45° for lager beer, 50° to 55° for ale) in three stages by means of Baudelot open-type, or double-pipe closed-type, exchangers utilizing the counter-flow principle as far as possible. The first stage uses water for the second brew; this stage uses the coldest cooling water, and the final stage employs refrigeration. The refrigeration per barrel is found from the expression

$$\begin{aligned} Q &= 259 \, d \, c \, \Delta t \\ &= 259 \times 1.053 \times [0.909 (72 - 48)] \\ &= 5940 \text{ Btu per barrel} \end{aligned}$$

where d is the density of the wort, taken as 1.053 in the example corresponding to 13 on the saccharimeter scale; c is the specific heat, taken as 0.909 for the example; Δt is the drop of temperature during cooling.

The heat removed to maintain a constant temperature during fermentation is shown by the chemical reaction



so that 1 lb of maltose forms 0.538 lb of ethyl alcohol and 0.515 lb of carbon dioxide.

The refrigeration per barrel is, approximately,

$$Q = 280 \times [0.281 (w_1 - w_2)] \text{ Btu per barrel}$$

where w_1 and w_2 are the weights of the solids for the initial and final saccharimeter readings, per barrel.

The brewery industry is still tied rather tightly to tradition, and the air conditioning engineer has not had much opportunity to modernize these plants. There would seem to be an excellent opening for introducing unit coolers and filters for the incoming air, for better removal of carbon dioxide, and, in general, for a better control of air movement.

23·5. Skating Rinks. For ice hockey the ice skating rink must be at least equal to the standard dimensions of 80 by 185 ft. If the ice area is larger, provision must be made to secure the standard size by an

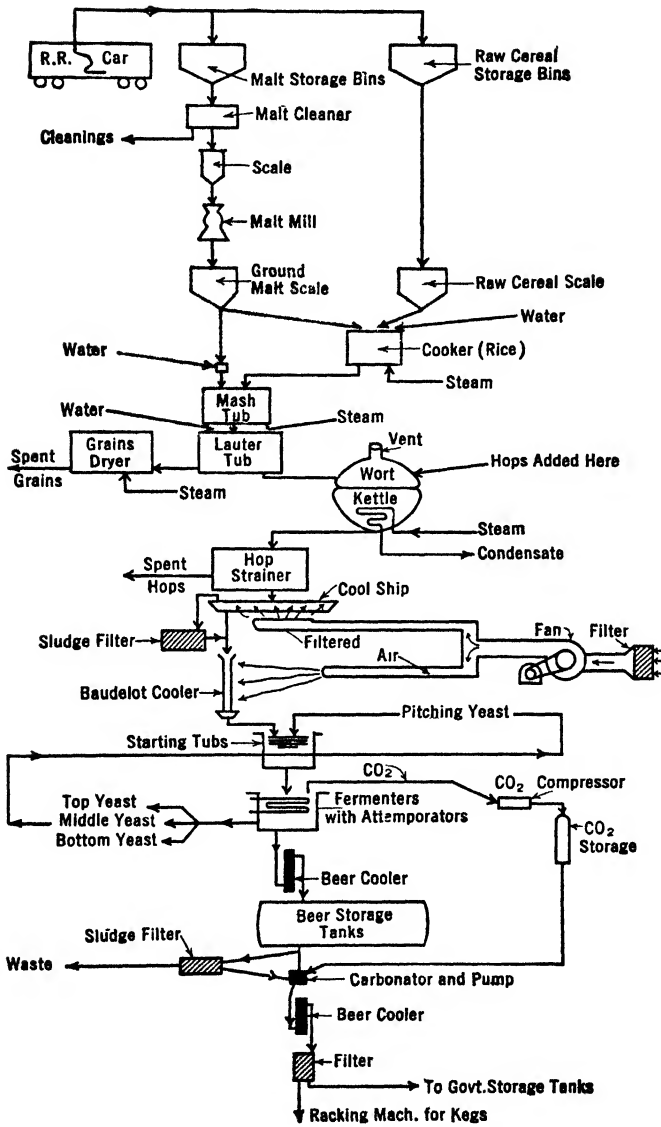


FIG. 23-5. Flow diagram for the manufacture of beer.

adjustment of the barriers. Up to the present time there has been nothing standard in the design of ice skating rinks. The piping has been fastened to the sleepers laid on sand, on floors of concrete, or the pipes have been embedded in concrete. The pipes have been designed for the use of brine or for direct expansion, and they have been spaced from 3 in. to 6 in. on centers. The fact that the skating season is limited to the late fall or the winter months, usually from October 15 to March 15, requires that commercial plants be used for other purposes when the skating season is over, and provision for this change be made. Lately rinks have been placed in auditoriums, necessitating both quick removal and replacement of the ice within 8 to 12 hr so that the floor may be used for various sports, shows, and expositions. The present practice is described below.

THE FLOOR. If the pipes are laid on the ground the only provisions are that the ground may be drained properly and that the pipes are level. If the pipes are embedded in concrete they should be fastened to metal chairs which in turn are secured to screeds. The screeds should rest on a firm foundation, with sand between them flush with their top surface, and if the arena is unexcavated the lower concrete floor should be resting either on 12 to 16 in. of cinders or 2 to 4 in. of corkboard laid in waterproof asphalt. The upper concrete slab is usually about $3\frac{1}{2}$ to $3\frac{3}{4}$ in. thick. If the space below the rink is excavated at least 4 in. of corkboard must be used if it is desired that the time of the removal and the replacement of the ice be limited to 8 to 12 hr. The floor must be separated from the building proper in order to permit freedom in expansion and contraction independent of the building.

PIPING. The amount of piping to be used depends somewhat on the kind of service imposed on the rink. One-inch pipe is still employed on 4-in. centers, but $1\frac{1}{4}$ -in. pipe is preferred on both 4- and $4\frac{1}{2}$ -in. centers, the former when rapid changes from skating conditions to some other utilization of the arena are desired. If the pipes are embedded only standard steel pipes with welded joints are satisfactory. Bare pipes should be galvanized and have galvanized heavy-steel screw couplings. The pipes should be laid parallel and level and should be placed to run across the arena (Fig. 23·6).

The upper slab containing the brine pipes may be laid either monolithic or with expansion joints running parallel with the pipes, and about 6 ft apart. The upper slab, in addition to the piping, should be reinforced.

BRINE DISTRIBUTION. It is found that about 0.4 Btu/min/sq ft of ice surface has to be absorbed by the brine in order to maintain a good skating surface. Given a temperature rise of the brine of 2° F this

pipes connected directly to the headers. A sufficiently long vertical nipple is required to take with the elbow the expansion and contraction incidental to change of temperature of the pipes and headers. Reversing valves have also been inserted in the brine mains, but these are not necessary. Brine is controlled by means of the valves in the vertical line between the headers and the hairpin coils. Air is automatically purged by running the air pipeline, which connects with the upper part of one end of the hairpin coil, up to the roof. A brine heater, employing steam, is essential for the ice, which should never be allowed to melt on the concrete slab.

THE REFRIGERATION REQUIRED. The tonnage is too variable to give in terms of the floor area. The brine temperature varies from about 18° F for bare pipe to 10° to 12° F for embedded pipe. The tonnage varies from one ton per 100 sq ft of ice surface in the Chicago Stadium to one ton per 160 sq ft in the Cleveland and the Toronto rinks; see Table 23·2. If the ice is to be removed one day and replaced the next, nearly double the refrigeration is required as compared with the rink that remains frozen the entire season.

Brine tanks are always used, but there is no method of calculation of their proper size. A brine tank will store up refrigeration, which will be useful during a short peak load or when a light load occurs, but the amount of storage per unit volume is very small. As the machine capacity installed is much more than the ordinary running requirements it is doubtful whether a brine tank of more than medium size is justified. All exposed brine piping, as well as the brine tank and the brine cooler, should be properly insulated. Insulation in the brine trench, or tunnel, should be so erected as to offer no difficulties in the operation or repair of the piping within the trench. The liquid receiver should be large enough to hold, possibly with the assistance of the condensers, the entire ammonia charge.

Table 23·2 presents the main details of the more recent skating rinks. In addition some consideration should be given to the latitude of the city where the rink is located.

23·6. Apartment Refrigeration. In general, for buildings designed for 45 apartments or more, with 5 to 7 cu ft capacity boxes, the cost of power necessary to operate the brine system will be about \$1.50 for each apartment per month, whereas the individual unit will cost \$0.75 and the multiple unit \$0.50. The brine piping has to be insulated where it runs exposed to the atmosphere, as must also the brine tank and the brine pump. In fact the cost of insulation is such a great proportion of the total cost that there is a marked tendency to reduce the size of the brine pipe to a point where the cost of operation becomes excessive.

TABLE 23-2
TYPICAL SKATING RINK DETAILS

Location.....	Toronto	Detroit	Philadelphia	Rye, N.Y.	West Point
Dimensions, ice surface.....	190 × 80	242 × 110	80 × 180	208 × 80	230 × 90
No., size and speed, machines.....	2-9 × 9-180	3-12 × 12-200	2-10 × 10-300	1-9 × 9 1-10 × 10-300	2-9 × 9-300
Ft ³ per ton.....	260	104	113	169	251
Ft pipe per ft ³ ice ..	2.74	3.13	3.14	3.15	2.5
Size pipe, in.....	1½	1½	1½	1	1
Spacing on centers..	4.5	4.0	4.0	4.0	5.0
No. headers.....	4	2	2	2	2
Location.....	2 ends	2 sides	1 side	2 sides	2 sides
Flow.....	Counter	One way	Hairpin	One way	One way

Location.....	Hershey, Pa.	Chicago	Champaign, Ill.	Tulsa, Okla.
Dimensions, ice surface.....	175 × 60	145 × 245	126 × 193	100 × 280
No., size and speed, machines.....	2-8 × 8-300	2-140 ton cent.	2-10 × 10-300	2-9 × 9-225
Ft ³ per ton.....	164	100	187	176
Ft pipe per ft ³ ice ..	2.9	3.0	2.75	2.31
Size pipe, in.....	1½	1½	1½	1½
Spacing on centers..	4.0	4.0	4.5	4.0
No. headers.....	4	2	2	2
Location.....	2 ends	1 side	1 side	1 side
Flow.....	Counter	Hairpin	Hairpin	Hairpin

However, the maintenance cost of the brine pipe is relatively small. With direct expansion the multiple unit can be installed with bare pipe, including even the suction pipe. With the individual unit there is no unusual expense except for service.

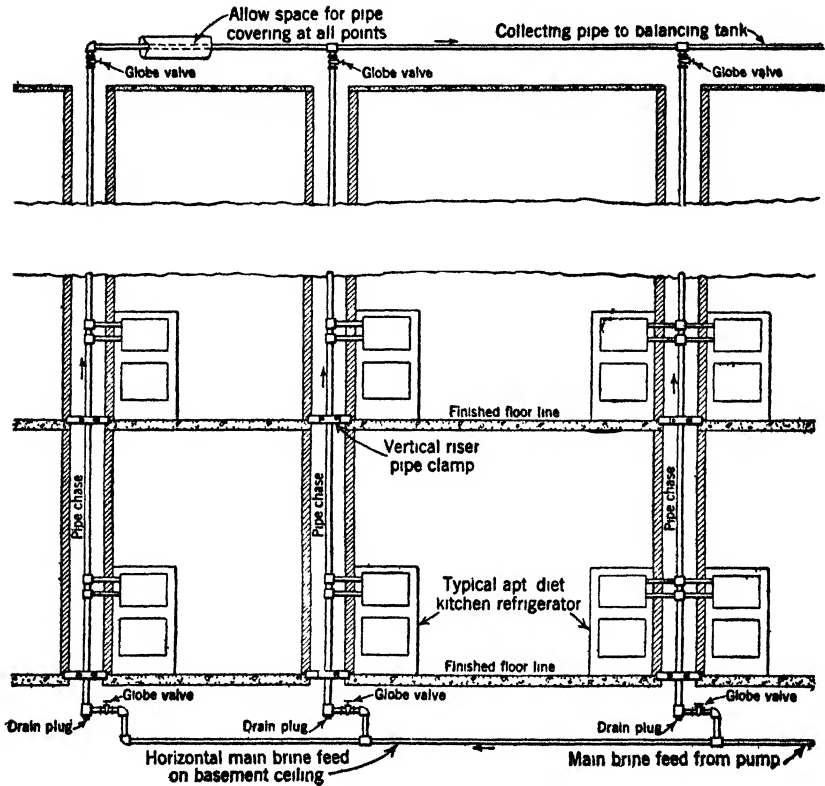


FIG 23·7. Apartment piping

PIPING. Brine pipe risers (Fig. 23·7) for single boxes should be $\frac{1}{2}$ in. for 2 to 3 stories, $\frac{3}{4}$ in. up to 6 stories, and 1 in. up to 12 stories. Risers for double boxes should be increased to the next size of pipe. For methyl chloride and sulphur dioxide all liquid mains and risers up to 20 apartments may be designed for $\frac{1}{2}$ -in. pipe; up to 50 refrigerators it can be 1 in., and up to 100 it can be $1\frac{1}{4}$ in. Small pipes and a gas receiver in the suction line near the compressor permit a long shutdown period where the compressor is operated by means of the suction pressure which starts and stops with a rise and fall of 5 lb above and below the average pressure.

Buildings taller than about 14 stories designed for the multiple direct-

expansion system cannot have the compressor and the condenser located in the basement, although the head (condenser) pressure can be increased by using a smaller condenser. When the building height becomes excessive a second installation should be placed at the highest point and fed downward to about the middle floor.

The safety code for mechanical refrigeration requires that no refrigerant shall be placed in elevator, dumbwaiter, or other shaft containing a moving object or one that has openings to living quarters or main exit hallways. The safety code has the following specifications which apply to apartment house refrigeration:

(a) Protected seamless copper tubing may be used in systems charged with 100 lb of refrigerant or less if the tubes are properly enclosed in steel pipes or tubes and if the conduit is sealed in the upper end. Flexible tubing not over 6 ft long may be used at the terminals or at bends. All valves and fittings except service valves and those in the machine room must be arranged in a suitable metal box.

(b) A refrigerant harmful to health and not apparent to human sense must have an odorant added which will make the gas detectable.

(c) Twenty pounds of an irritant and inflammable refrigerant may use full-weight or extra-heavy butt-welded pipe, depending on the pressure carried. If copper tubing is used it must be protected.

(d) Fifty pounds of an irritant and inflammable refrigerant may be used if the compressor, condenser, liquid receiver, and liquid distribution header are located in a separate machinery room with not more than 20 lb of refrigerant per section.

(e) One hundred pounds of an irritant and inflammable refrigerant may be used if the compressor, condenser, liquid receiver, and distribution headers are located in a separate machinery room, if the compressor is tested to twice the minimum test pressure, and if the conduit or pipe enclosing the copper tubing is sufficiently tight to require 2 hr for an air pressure of 2 psig to drop to 1.0 psig. The conduit system must enclose all the copper tubing between the machinery room and the evaporators and must be vented to the roof. A loss of not more than 10 lb of refrigerant must automatically stop the compressor.

Example. An apartment has 50 separate refrigerator boxes of a daily capacity of 100 lb of ice and uses calcium chloride brine with 20° F entering and 22° F leaving temperatures. Find the length of 1¼-in. pipe to be used if the box is to be held normally at 33° F. Find the total capacity of the compressor, allowing 10 per cent for line transmission losses. Take U (Table 9-17) as 2.0.

Solution. One hundred pounds of ice capacity is

$$\begin{aligned} 100 \times 144 &= 14,400 \text{ Btu per box per 24 hr} \\ &= 600 \text{ Btu per hr per box} \end{aligned}$$

The average temperature difference is

$$33 - 21 = 12^{\circ} \text{ F}$$

The area of the box coils,

$$\frac{600}{2.0 \times 12} = 25$$

$$25 \times 2.3 = 57.5 \text{ lineal feet of } 1\frac{1}{4}\text{-in. pipe}$$

Total capacity of the refrigerating machine,

$$600 \times 50 = 30,000 \text{ Btu/hr}$$

$$30,000 \times 1.1 = 33,000 \text{ Btu} = 2\frac{3}{4} \text{ tons}$$

23·7. Drinking Water. Drinking water systems in which the water is maintained cooled at about 50° F are now installed in hotels, offices, public buildings, stores, and factories. The water may be filtered or, in some installations, sterilized. The piping system is always a closed one with a return line back to the circulating pump and the water cooler. As usually constructed in the larger systems a balance tank is installed at the high point as shown in Fig. 23·8. This tank insures that the pipelines will be filled with water all the time, and it makes the system balanced; hence the only pumping load is that necessary to overcome the friction due to flow and to maintain a velocity head. The make-up water is brought in from the house tank or the balance tank. Where the house tank and the cooler are incorporated together, the make-up water is brought in on the suction side of the circulating pump and an automatic air vent is placed at the highest point of the loop. In order to save insulation in the multiple system the return to the pump can be arranged for by means of a double-pipe installation whereby the suction to the pump is made to pass through the inner pipe suitably centralized by means of an adapter. The water cooler may be a Baudelot cooler (where the water passes by gravity on the outside of a vertical stand of piping), or simply a coil in a tank, or in fact any device that will not allow injury to the piping should the water freeze in the cooler.

It is customary to circulate more water than is expected to be used and wasted during the extreme conditions of operation, in order to keep the pipes always full of water and also to control the rise of the water temperature on account of heat leakage into the system. The usual method is to circulate the amount of water required to absorb the heat leakage in the piping, figured from the estimated size and length of the piping, the extreme sustained temperature of the air surrounding the pipes, and the kind and thickness of the insulation used. An amount of water, in addition to the amount used and wasted, which will absorb this heat leakage with a rise of temperature of not more than 4½° F must be circulated by the water pump.

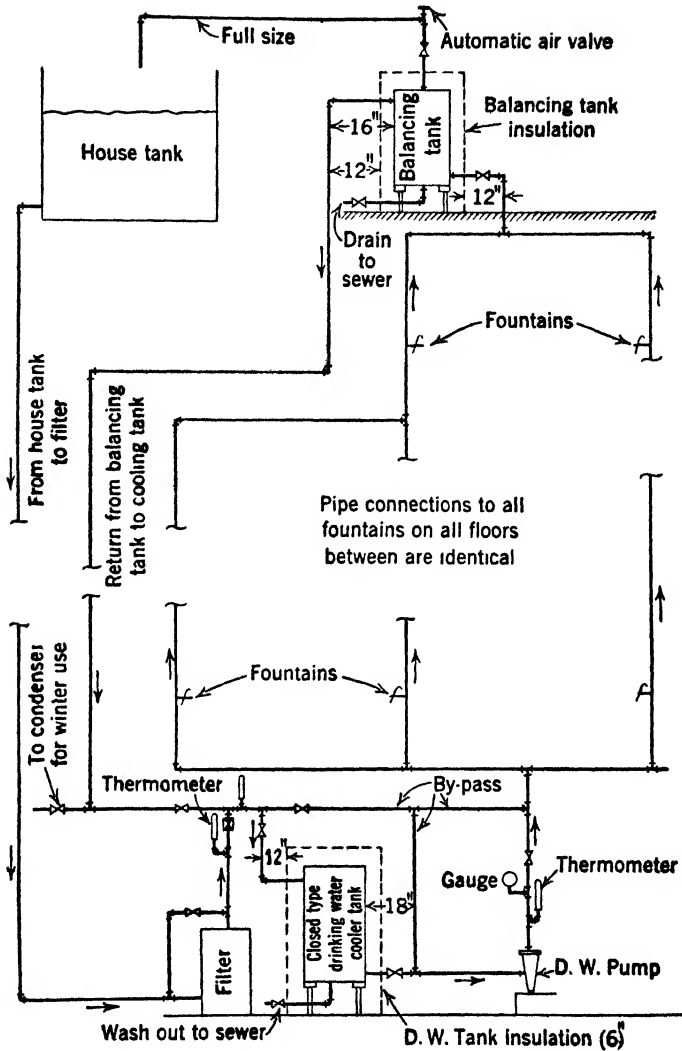


FIG. 23-8. Drinking water system.

The pipeline must not be too long; 2000 ft is usually the limit, although 5000 ft have been used successfully. Too long a circuit results in over-great friction head and therefore an excessive pumping cost. It is usual to consider that all the work performed in pumping goes into the heating of the water at the rate of approximately 0.2 ton of refrigeration per horsepower of the pump. The fountains should be arranged sym-

metrically if possible, and the run from the main to the fountains should be very short as, of course, this water is stagnant, being a dead end, and will heat up unless the flow from the fountain is frequent.

The amount of water used and wasted can be found only approximately. Hotels and office buildings having an outlet for each room should be designed on the basis of 1 gal per person likely to use the room per day. In all likelihood $\frac{1}{4}$ gal per faucet per hour will be liberal although systems have been reported in which six times this amount

TABLE 23·3
COEFFICIENT OF HEAT TRANSFER, PER LINEAL FOOT

Size of Pipe, in.	Union Lith	Hair Felt (Two Lavers)	Corkboard
$\frac{1}{2}$	0 157	0 086	0 160
$\frac{3}{4}$	0 170	0 095	0 167
1	0 180	0 107	0 178
$1\frac{1}{4}$	0 213	0 124	0 199
$1\frac{1}{2}$	0 217	0 132	0 220
2	0 269	0 149	0 245
$2\frac{1}{2}$	0 272	0 167	0 291
3	0 294	0 192	0 304
$3\frac{1}{2}$	0 355	0 211	0 330
4	0 367	0 228	0 345
$4\frac{1}{2}$	0 376	0 247	0 385
5	0 466	0 268	0 410
6	0 558	0 307	0 437

has been provided. Fountains installed in the aisles of stores or the halls of hotels may be in continuous use. In factories it is good practice to use $\frac{1}{2}$ gal per person per hour, and in offices $\frac{1}{4}$ gal per person per hour.

In estimating the heat leakage with the formula

$$Q = LUt_m$$

where L is the length of the pipe in feet, U is the coefficient of heat transfer in Btu per foot of length per hour, the value of t_m is taken as the average difference of temperature for sustained summer temperatures in the tunnel or pipe opening. The maximum air temperature may be as much as 85° to 90° F, and if the average water temperature is 47.5° F the value of t_m becomes 37.5° or 42.5°. Values for U per linear foot per hour are given in Table 23·3. The fountains should be located with care so they will be of the greatest convenience possible, but it is necessary to bring the cold water up to the fountains and to

carry the waste away in the usual manner. It is customary to allow 100 operators in the shop and 50 operators in the office per fountain. The maximum number of persons can be calculated on the basis of 70 sq ft per person in an office building and 100 sq ft per person in a factory.

In office buildings, hotels, and large factories it is usual to have a central plant and distribution by means of insulated pipelines. In smaller installations single bubblers may be installed direct-connected to their own coolers and the supply water may be delivered directly to the coolers.

Example. This example illustrates the manner of calculation of the size of a drinking water system. An office building 160 ft by 80 ft and 10 stories high is to have a drinking water system with the water to be delivered at a temperature of 50° F. The floors are 13 ft high, the make-up water is at a temperature of 80°, and 3.0 gal are to be provided per person per day of 8 hr.

The floor area is 12,800 sq ft, and with an allowance of 1 person per 70 sq ft of floor area there will be 183 persons on the floor at a maximum. Using one fountain per 50 persons, 4 fountains per floor or 40 fountains in all will be required. The piping has, therefore, four circuits in parallel. Assume that the risers will be ½-in. pipe, each riser using 225 ft, the subheaders ¾-in. pipe, requiring 80 ft, and that 220 ft of 1-in. pipe will be required as a common return. The heat absorbed (using an average temperature of the water of 47.5° F, 95° outside temperature, and cork covering) follows.

$$\begin{aligned} \frac{1}{2}\text{-in. line } Q &= 225 \times 0.160 \times (95 - 47.5) = 1710 \text{ Btu (per riser) per hr} \\ &= 6840 \text{ Btu (total)} \\ \frac{3}{4}\text{-in. line } Q &= 160 \times 0.167 \times (95 - 47.5) = 1270 \text{ Btu} \\ 1\text{-in. line } Q &= 220 \times 0.178 \times (95 - 47.5) = 1860 \text{ Btu} \\ &= 9970 \text{ Btu total per hr} \end{aligned}$$

The amount of water required to absorb this heat (assuming a rise of temperature of the water 4½° F) is

$$\begin{aligned} \frac{9970}{4.5} &= 2216 \text{ lb/hr} \\ &= 4.43 \text{ gpm total} \\ &= 1.11 \text{ gpm per circuit} \end{aligned}$$

The make-up water required (neglecting the effect of friction heating) is

$$\begin{aligned} 3.0 \times 10 \times 183 &= 5490 \text{ gal per 8 hr} \\ &= 686 \text{ gph} = 11.45 \text{ gpm} \end{aligned}$$

$$\begin{aligned} \text{The total water required} &= 2.88 \text{ gpm per riser} \\ &= 15.88 \text{ gpm} \end{aligned}$$

Refrigeration Required.

$$60 \times 8.33 \times 11.45 \times (80 - 45) + 9970 = 210,400 \text{ Btu} = 17.5 \text{ tons}$$

The friction developed in the water circuit is, from Fig. 15-4,

$\frac{1}{2}$ -in. line (3.97 gpm) = 2.6 in. of water head per foot of pipe

$\frac{3}{4}$ -in. line (7.94 gpm) = 2.3 in. of water head per foot of pipe

1-in. line (15.88 gpm) = 2.3 in. of water head per foot of pipe

Total friction is

$$(2.6 \times 225) + (2.3 \times 80) + (2.3 \times 220) = 1275 \text{ in. of water} \\ = 106\frac{1}{4} \text{ ft of water head}$$

The heating effect of the friction head is

$$15.88 \times 8.33 \times 106\frac{1}{4} = 14,050 \text{ ft-lb/min} \\ \frac{14,050}{33,000} \times 42.4 = 18.1 \text{ Btu/min}$$

The size of the motor will be

$$\frac{0.425}{0.5} = 0.964 = 1.00 \text{ hp}$$

allowing for 50 per cent motor efficiency in the small size called for in this example.

23-8. Oil Refining. The oil refineries manufacturing a low cold test oil require a procedure that will permit the wax to congeal. This requirement demands the cooling of the oil to varying degrees of temperature and some means of separation of the solid wax from the liquid oil.

THE WAX PROCESS. In certain grades of oil the required cold test is obtained by cooling the oil to some temperature near 0° F and then separating the resulting congealed paraffin wax from the oil by pumping through a filter press. The problem involves the cooling of brine, as direct expansion is seldom used, and a suitable device for the chilling of the oil. A scraping attachment of some sort must be provided in order that as the wax congeals and collects on the pipe surface it may be scraped away and pushed along in the direction of flow of the chilled oil. One such design uses a double-pipe cooler with a screw conveyor in the inner pipe (Fig. 23-9), so constructed as to give a positive flow and an efficient clearing of the pipe surface at all times in order to obtain a good value for U .

The amount of wax and the physical constants of distillate vary with the different fields, but the average is taken as 0.5 for the specific heat of the oil, and 0.87 for the specific gravity. The latent heat of fusion of the paraffin is 125 Btu/lb, and it begins to separate out at 60° F. As an illustration the following example includes the essential points.

Example. A thousand gallons per hour of distillate are to be cooled from 80° to 20° F, and 10 per cent of wax by weight is to be removed. Assume the specific

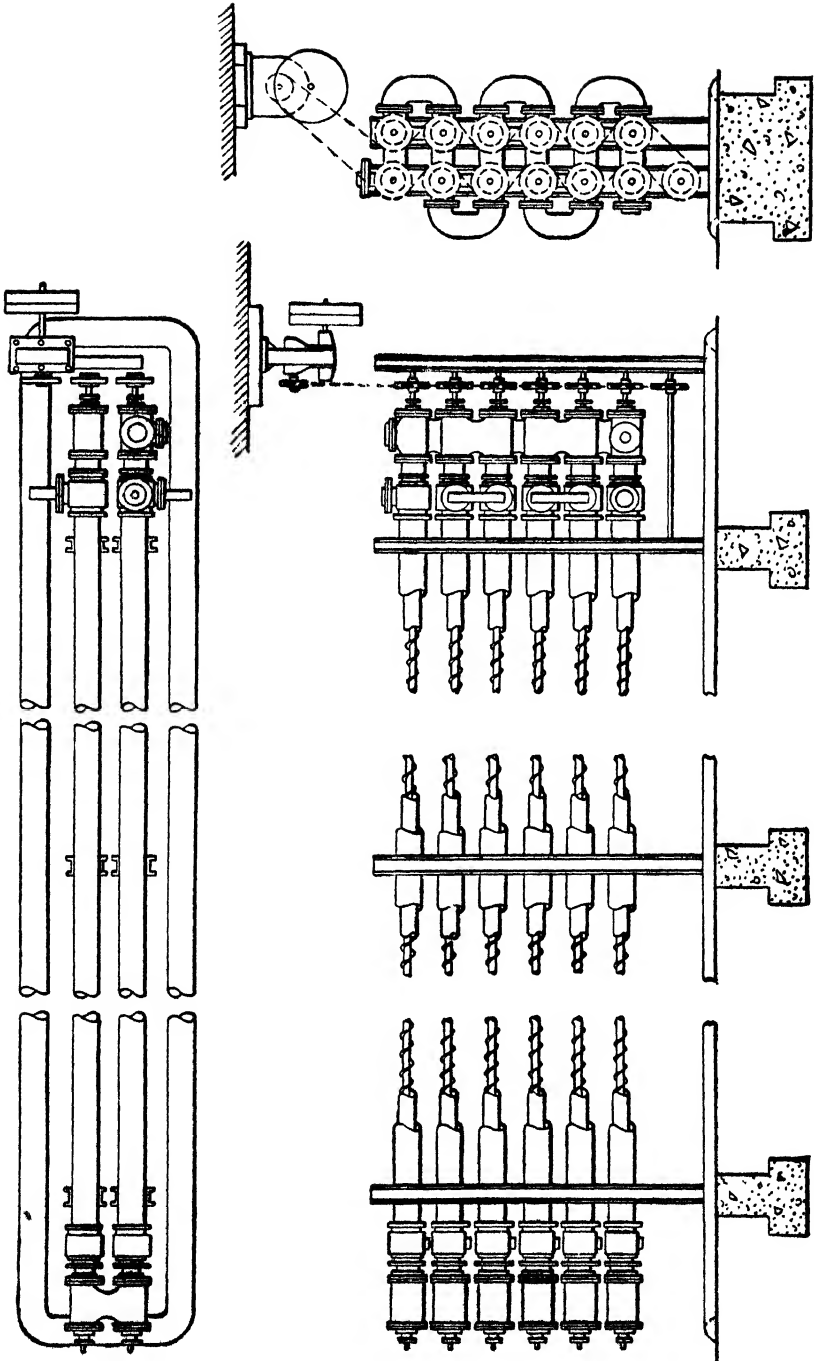


Fig. 23-9. The wax chilling machine.

heat of the solid wax as 0.6. Find the size of the compressor required, allowing 10 per cent for losses by heat leakage, etc., and find the size of the double-pipe cooler required or the cooling of the oil.

Solution. The weight of the oil per hour is

$$1000 \times 8.33 \times 0.87 = 7,250 \text{ lb}$$

(a) To cool the oil to 60° F:	$0.5 \times 7250 \times (80 - 60) =$	72,500 Btu
(b) To freeze the paraffin:	$0.1 \times 7250 \times 125 =$	90,600 Btu
(c) To cool the oil to 20° F	$0.9 \times 7250 \times 0.5 \times (60 - 20) =$	130,500 Btu
(d) To cool the wax:	$0.1 \times 7250 \times 0.6 \times (60 - 20) =$	17,400 Btu
		Total = 311,000 Btu

If an allowance of 10 per cent is made for losses the total tonnage becomes 28.5.

The resulting clarified oil, being at 20°, could be used to decrease the refrigerating load by pumping it back to the counter-flow double-pipe cooler and permitting it to warm up to 40° F or more by the absorption of heat from the distillate initially at 80°. If such an arrangement is used the refrigeration saved could be expected to be

$$0.9 \times 7250 \times (40 - 20) \times 0.5 = 65,250 \text{ Btu}$$

and the tonnage becomes

$$\frac{1.1 \times 245,750}{12,000} = 22.5 \text{ tons}$$

The coefficient of heat transfer U , however, will be less with oil on both sides of the pipe than with brine and oil.

If the brine enters at 0° and leaves the cooler at 5° F the mean temperature difference t_m is

$$t_m = \frac{(80 - 5) - (20 - 0)}{\log_e \frac{80 - 5}{20 - 0}} = 41.6^\circ$$

If the value of U may be taken as 25,

$$311,000 = A \times 25 \times 41.6$$

$$A = 298 \text{ sq ft} = 478 \text{ lineal feet of 2-in. pipe}$$

PROBLEMS

1. One thousand gallons per hour of distillate are to be cooled from 70° to 10° F, and 12 per cent of wax by weight is separated out at 60° F. The specific heat of the oil is 0.5 and the specific gravity 0.87, the specific heat of the wax is 0.6 and the latent heat of fusion is 125 Btu/lb. Allow 10 per cent for losses. Find (a) the cylinder size of an ammonia compressor if it is a vertical single-acting twin cylinder at 160 rpm, and the ratio $l/d = 1.0$; (b) the length of 2-in. pipe for the oil cooler if the brine enters at 0° and leaves at 5° F; (c) using the principle of the exchanger find what refrigeration could be saved by the warming of the chilled oil from 10° to 40° F. Liquefaction is at 86° F.

2. Sixty beefs of 700 lb each are to be chilled from 90° to 35° F in 14 hr. Sodium

chloride brine is sprayed at 29° F which rises to 33° F. The construction is similar to that shown in Fig. 21·9. Find (a) the tonnage required, the number of gallons per minute of brine, and the head of brine on the nozzles from a constant level brine tank if the brine requires 5 psi pressure at the nozzles; (b) the size of shell and tube brine cooler required, using a value of U of 200 Btu/(hr) (sq ft) (°F); (c) the probable horsepower of the centrifugal pump required to return the brine to the tank. Take the distance from the nozzles to the floor as 15 ft 0 in. Evaporation at 25° F. Add 20 per cent for losses.

3. An apartment house has 60 separate refrigerator boxes, having a daily ice capacity of 120 lb; calcium chloride brine enters at 20° F and leaves at 24° F; a three-pipe system is employed. If the box temperature is held at 33° F find the area of the brine-cooling surface. Find the piston displacement of the compressor, allowing 10 per cent for line transmission losses. Find the size of a suitable Freon-12 compressor, assuming 80° F liquefaction. *Suggestion:* Evaporating temperature of the refrigerant has to be below 20° F, say 15° F. Assume the volumetric efficiency of the compressor to be 0.9 and 200 rpm.

4. A central cold storage warehouse has, on one floor, ten 1200-ft 2-in. coils in parallel. The room is held at 6° F with brine in at 0° and out at 2° F. Assuming that the total loss of head is double the amount in the coils, corrected for fittings, find the minimum horsepower required to circulate the brine. Select a suitable brine concentration from Table 20·1, and the friction constant from Fig. 15·5.

5. A simple refrigerating plant has the following load: (a) heat leakage and live load in a cold storage room estimated as 500,000 Btu/hr; (b) milk cooling, 2000 gph, from 80° to 40° F with specific heat of 0.9, specific gravity of 1.03; (c) manufacture of ice estimated as 1.0 ton/hr. (Assume 220 Btu per pound of water frozen, for freezing, cooling, allowance for heat leakage and the agitation of the brine and the water.) The brine is carried at 15° F; liquefaction of the refrigerant is at 80° F. Find (a) the size of a single-acting, vertical, enclosed ammonia compressor for 200 rpm with $l/d = 1.0$; (b) the horsepower required to drive the compressor, allowing 10 per cent for friction, 12° F evaporation, and twin cylinders.

6. Design a drinking water system for a factory loft 12 stories high, excluding the basement and attic. Each floor is 300 ft by 80 ft, and the height of the stories is 12 ft. Take make-up water as 80° F, the average drinking water temperature as 45° F, and the air temperature as 85° F. Assume that risers are $\frac{3}{4}$ -in., the return pipe $1\frac{1}{4}$ -in., and the intermediate headers 1-in. pipe. Allow one person per 75 sq ft of floor and 50 persons per fountain. Allow $\frac{1}{2}$ gph per person for make-up water. Assume coefficient of heat transfer per lineal foot of pipe through the cold water insulation as 0.167 for $\frac{3}{4}$ -in., 0.178 for 1-in., and 0.199 for $1\frac{1}{4}$ -in. pipe. Use a symmetrical arrangement of piping, with pipe shafts in the outside walls. Find (a) proper arrangement of piping; (b) amount of water required to absorb the heat leakage, assuming a 5° F rise of temperatures; (c) amount of make-up water; (d) total refrigeration; (e) power required to circulate the water.

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