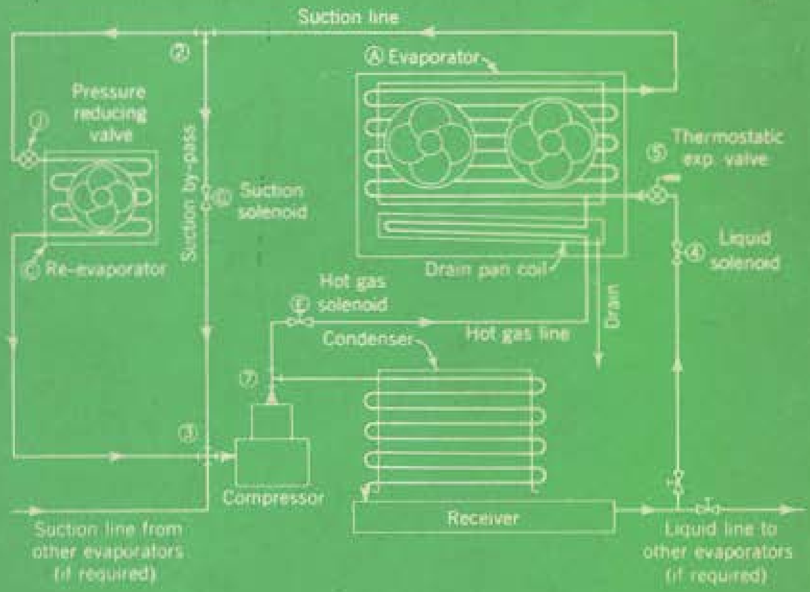
PRINCIPLES OF REFRIGERATION

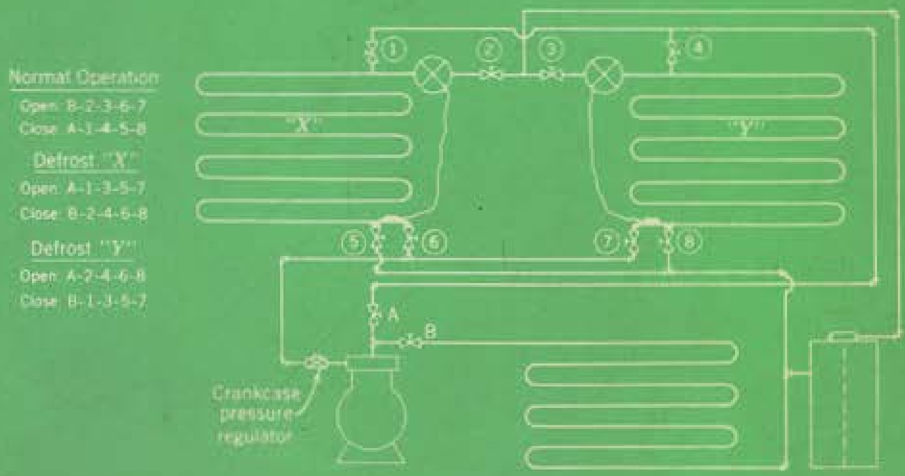
DOSSAT

WILEY
TOPPAN

principles of refrigeration

R. J. DOSSAT





Wiley International Edition

PRINCIPLES OF REFRIGERATION

ROY J. DOSSAT, Associate Professor of Refrigeration
and Air Conditioning, University of Houston, Houston, Texas

JOHN WILEY & SONS, INC.
NEW YORK and LONDON

TOPPAN COMPANY, LTD.
TOKYO, JAPAN

Authorized reprint of the edition published by John Wiley & Sons, Inc., New York and London.

Wiley International Edition

Library of Congress Catalog Card Number: 61-15396

Printed in Singapore by Toppan Printing Co. (S) Pte. Ltd.

Preface

This textbook has been written especially for use in programs where a full curriculum in refrigeration is offered. However, the material covered and the method of presentation are such that the text is also suitable for adult evening classes and for on-the-job training and self-instruction. Furthermore, the material is so arranged and sectionalized that this textbook is readily adaptable to any level of study and to any desired method or sequence of presentation.

Despite a rigorous treatment of the thermodynamics of the cycle, application of the calculus is not required nor is an extensive background in physics and thermodynamics presupposed. The first four chapters deal with the fundamental principles of physics and thermodynamics upon which the refrigeration cycle is based. For those who are already familiar with these fundamentals, the chapters will serve as review or reference material.

Chapter 21 treats electric motors and control circuits as they apply to refrigeration and air conditioning systems. This material is presented from the viewpoint of practical application, the more mathematical approach being left to companion electrical courses.

Throughout this textbook emphasis is placed on the cyclic nature of the refrigeration system, and each part of the system is carefully examined in relation to the whole. Too, care is taken continually to correlate theory and practice through the use of manufacturer's catalog data and many sample problems. To this end, certain pertinent catalog data are included.

ROY J. DOSSAT

July, 1961

Acknowledgments

Most of the material in this textbook is based on information gathered from publications of the American Society of Heating, Refrigerating, and Air Conditioning Engineers and of the following equipment manufacturers:

ACME INDUSTRIES, INC.
ALCO VALVE COMPANY
ANACONDA METAL HOSE DIVISION, THE AMERICAN BRASS COMPANY
BELL & GOSSETT COMPANY
CARRIER CORPORATION
CONTROLS COMPANY OF AMERICA
DEAN PRODUCTS, INC.
DETROIT CONTROLS DIVISION, AMERICAN RADIATOR & STANDARD SANITARY CORPORATION
DETROIT ICE MACHINE COMPANY
DOLE REFRIGERATING COMPANY
DUNHAM-BUSH, INC.
EDWARDS ENGINEERING CORPORATION
E. I. DU PONT DE NEMOURS & COMPANY
FREEZING EQUIPMENT SALES, INC.
FRICK COMPANY
GENERAL CONTROLS COMPANY
GENERAL ELECTRIC COMPANY
HALSTEAD & MITCHELL
INGERSOLL-RAND COMPANY
KENNARD DIVISION, AMERICAN AIR FILTER COMPANY, INC.
KRAMER TRENTON COMPANY
MCQUAY, INC.
THE MARLEY COMPANY
MARSH INSTRUMENT COMPANY
MUELLER BRASS COMPANY
PENN CONTROLS, INC.
RECOLD CORPORATION
SPORLAN VALVE COMPANY
TECUMSEH PRODUCTS COMPANY
TRANTER MANUFACTURING, INC.
TUBULAR EXCHANGER MANUFACTURERS ASSOCIATION, INC.
TYLER REFRIGERATION CORPORATION
THE VILTER MANUFACTURING COMPANY
WORTHINGTON CORPORATION
YORK CORPORATION, SUBSIDIARY OF BORG-WARNER CORPORATION

Appreciation is expressed to all these organizations for their contributions in the form of photographs and other art work, and for granting permission to reproduce proprietary data, without which this textbook would not have been possible.

Contents

1

Pressure, Work, Power, Energy

1-1. Force. A force is defined as a push or a pull. It is anything that has a tendency to set a body in motion, to bring a moving body to rest, or to change the direction of motion. A force may also change the size or shape of a body. That is, the body may be twisted, bent, stretched, compressed, or otherwise distorted by the action of a force.

The most familiar force is weight. The weight of a body is a measure of the force exerted on the body by the gravitational pull of the earth (Fig. 1-1).

There are many forces other than the force of gravity, but all forces are measured in weight units. Although the most commonly used unit of force measure is the pound, any unit of weight measure may be used, and the particular unit used at any time will usually depend on the magnitude of the force to be measured.

1-2. Pressure. Pressure is the force exerted per unit of area. It may be described as a measure of the intensity of a force at any given point on the contact surface. Whenever a force is evenly distributed over a given area, the pressure at any point on the contact surface is the same and can be calculated by dividing the total force exerted by the total area over which the force is applied. This relationship is expressed by the following equation

$$P = \frac{F}{A} \qquad (1\text{-}1)$$

where P = the pressure expressed in units of F per unit of A
F = the total force in any units of force
A = the total area in any units of area

1-3. Measurement of Pressure. As indicated by equation 1-1, pressure is measured in units of force per unit of area. Pressures are most frequently given in pounds per square inch, abbreviated psi. However, pressure, like force, as a matter of convenience and depending on the magnitude of the pressure, may be stated in terms of other units of force and area, such as pounds per square foot, tons per square foot, grams per square centimeter, etc.

Example 1-1. A rectangular tank, measuring 2 ft by 3 ft at the base, is filled with water. If the total weight of the water is 432 lb, determine the pressure exerted by the water on the bottom of the tank in

(*a*) pounds per square foot
(*b*) pounds per square inch

Solution

(*a*) Area of tank base	2×3
	$= 6$ sq ft
Total weight of water	$= 432$ lb
Applying Equation 1-1, P	$\frac{432}{6}$
	$= 72$ psf
(*b*) Area of the tank base	24×36
	$= 864$ sq in.
Total weight of water	$= 432$ lb
Applying Equation 1-1, P	$\frac{432}{864}$
	$= 0.5$ psi

The problem described in Example 1-1 is illustrated in Fig. 1-2. Notice that the pressure on the bottom of the tank in pounds per square foot is equivalent to the downward force exerted by the weight of a column of water having a cross section of one square foot, whereas the pressure in pounds per square inch is equivalent to the downward force exerted by a column of water having a cross section of 1 sq in. Further, since there are 144 sq in. in 1 sq ft, the force exerted per square foot is 144 times as great as the force exerted per square inch.

1-4. Atmospheric Pressure. The earth is surrounded by an envelope of atmosphere or air which extends upward from the surface of the earth to a distance of some 50 mi or more. Air has weight and, because of its weight, exerts a

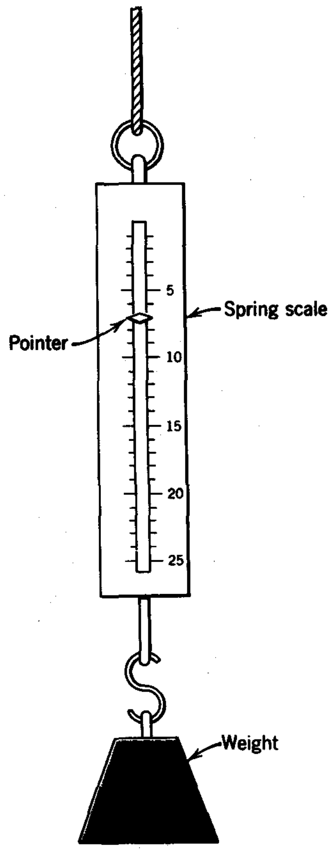


Fig. 1-1. Because of gravity, the suspended weight exerts a downward force of 7 lb.

pressure on the surface of the earth. The pressure exerted by the atmosphere is known as atmospheric pressure.

The weight of a column of air having a cross section of 1 sq in. and extending from the surface of the earth at sea level to the upper limits of the atmosphere is 14.696 lb. Therefore, the pressure on the surface of the earth at sea level resulting from the weight of the atmosphere is 14.696 psi (14.7). This is understood to be the normal or standard atmospheric pressure at sea level and is sometimes referred to as a pressure of one atmosphere. Actually, the pressure of the atmosphere does not remain constant, but will usually vary somewhat from hour to hour depending upon the temperature, water vapor content, and several other factors.

Because of the difference in the height of the column, the weight of a column of air of given cross section will be less when taken at an altitude of one mile above sea level than when taken at sea level. Therefore, it follows that atmospheric pressure decreases as the altitude increases.

1-5. Barometers. Barometers are instruments used to measure the pressure of the atmosphere and are of several types. A simple barometer which measures atmospheric pressure in terms of the height of a column of mercury can be constructed by filling with mercury a hollow glass tube 36 in. or more long and closed at one end. The mercury is held in the tube by placing the index finger over the open end of the tube while the tube is inverted in an open dish of mercury. When the finger is removed from the tube, the level of the mercury in the tube will fall, leaving an almost perfect vacuum at the closed end. The pressure exerted downward by the atmosphere on the open dish of mercury will cause the mercury to stand up in the evacuated tube to a height depending upon the amount of pressure exerted. The height of the mercury column in the tube is a measure of the pressure exerted by the atmosphere and is read in inches of mercury column (abbreviated in. Hg). The normal pressure of the atmosphere at sea level (14.696 psi) pressing down on the dish of mercury will cause the mercury in the tube to rise to a

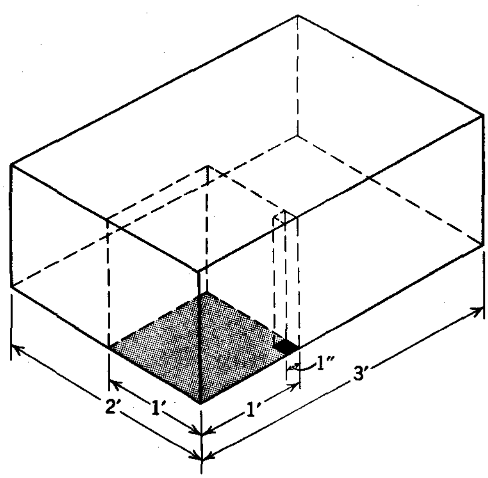


Fig. 1-2. Of the total weight of the water in the container, that part which is exerted on a 1 sq ft area is the pressure in pounds per square foot. Likewise, that part which is exerted on a 1 sq in. area is the pressure in pounds per square inch.

height of 29.921 in. (Fig. 1-3). A column of mercury 29.921 in. high is, then, a measure of a pressure equivalent to 14.696 psi. By dividing 29.921 in. Hg by 14.696 psi, it is determined that a pressure of 1 psi is equivalent to a pressure of 2.036 in. Hg. Therefore, 1 in. Hg equals 1/2.036, or 0.491 psi, and the following equations are established:

$$\text{in. Hg} = \frac{\text{psi}}{0.491} \tag{1-2}$$

and

$$\text{psi} = \text{in. Hg} \times 0.491 \tag{1-3}$$

Example 1-2. What is the pressure of the atmosphere in psi if a barometer reads 30.2 in. Hg?

Solution. Applying Equation 1-3,

$$P = 30.2 \times 0.491$$
$$= 14.83 \text{ psi}$$

Example 1-3. In Fig. 1-3, how high will the mercury stand in the tube when the atmospheric pressure is 14.5 psi?

Solution. Applying Equation 1-2,

$$P = \frac{14.5}{0.491}$$
$$= 29.53 \text{ in. Hg}$$

1-6. Pressure Gages. Pressure gages are instruments used to measure the fluid pressure (either gaseous or liquid) inside a closed vessel. Pressure gages commonly used in the refrigeration industry are of two types: (1) manometer and (2) bourdon tube.

1-7. Manometers. The manometer type gage utilizes a column of liquid to measure the pressure, the height of the column indicating the magnitude of the pressure. The liquid used in manometers is usually either water or mercury. When mercury is used, the instrument is known as a mercury manometer or mercury gage and, when water is used, the instrument is a water manometer or water gage. The simple barometer described previously is a manometer type instrument.

A simple mercury manometer, illustrated in Figs. 1-4*a*, 1-4*b* and 1-4*c*, consists of a U-shaped glass tube open at both ends and partially filled with mercury. When both legs of the U-tube are open to the atmosphere, atmospheric pressure is exerted on the mercury in both sides of the tube and the height of the two mercury columns is the same. The height of the two mercury columns at this position is marked as the zero point of the scale and the scale is calibrated in inches to read the deviation of the mercury columns from the zero condition in either direction (Fig. 1-4*a*).

When in use, one side of the U-tube is connected to the vessel whose pressure is to be measured. The pressure in the vessel, acting on one leg of the tube, is opposed by the atmospheric pressure exerted on the open leg of the tube. If the pressure in the vessel is greater than that of the atmosphere, the level of the mercury on the vessel side of the U-tube is depressed while the level of the mercury on the open side of the tube is raised an equal amount (Fig. 1-4*b*). If the pressure in the vessel is less than that of the atmosphere, the level of the mercury in the open leg of the tube is depressed while the level of the mercury in the leg connected to the vessel is raised by an equal amount (Fig. 1-4*c*). In either case, the difference in the heights of the two mercury columns is a measure of the difference in pressure between the total pressure of the fluid in the vessel and the pressure of the atmosphere.

In Fig. 1-4*b*, the level of the mercury is 2 in.

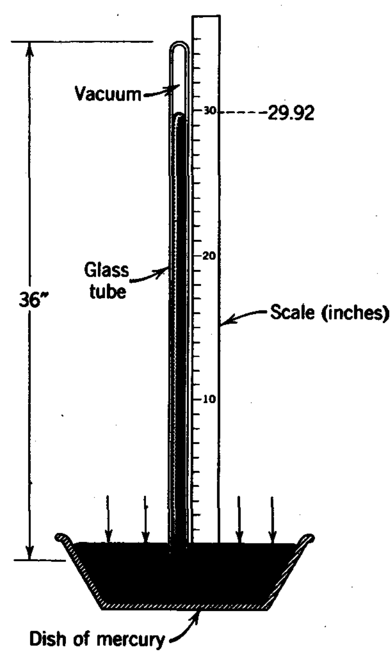


Fig. 1-3. The pressure exerted by the weight of the atmosphere on the open dish of mercury causes the mercury to stand up into the tube. The magnitude of the pressure determines the height of the mercury column.

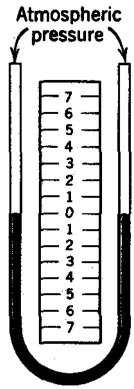


Fig. 1-4a. Simple U-tube manometer. Since both legs of the manometer are open to the atmosphere and are at the same pressure, the level of the mercury is the same in both sides.

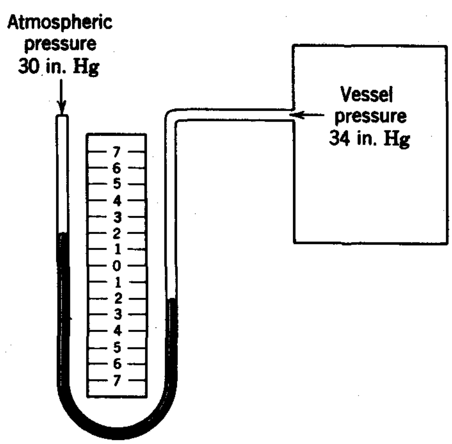


Fig. 1-4b. Simple manometer indicates that the vessel pressure exceeds the atmospheric pressure by 4 in. Hg.

below the zero point in the side of the U-tube connected to the vessel and 2 in. above the zero point in the open side of the tube. This indicates that the pressure in the vessel exceeds the pressure of the atmosphere by 4 in. Hg (1.96 psi). In Fig. 1-4*c*, the level of the mercury is depressed 2 in. in the side of the tube open to the atmosphere and raised 2 in. in the side connected to the vessel, indicating that the pressure in the vessel is 4 in. Hg (1.96 psi) below (less than) atmospheric. Pressures below atmospheric are usually called "vacuum" pressures and may be read as "inches of mercury, vacuum."

Manometers using water as the measuring fluid are particularly useful for measuring very small pressures. Because of the difference in the density of mercury and water, pressures so slight that they will not visibly affect the height of a mercury column will produce easily detectable variations in the height of a water column. Atmospheric pressure, which will support a column of mercury only 29.921 in. high, will lift a column of water to a distance of approximately 34 ft. A pressure of 1 psi will raise a column of water 2.31 ft or 27.7 in. and a

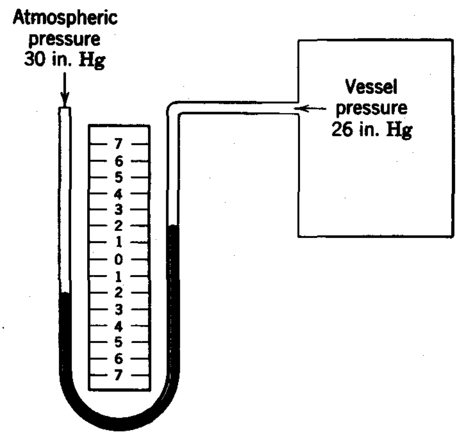


Fig. 1-4c. Manometer indicates that the vessel pressure is 4 in. Hg less than the atmospheric pressure of 30 in. Hg.

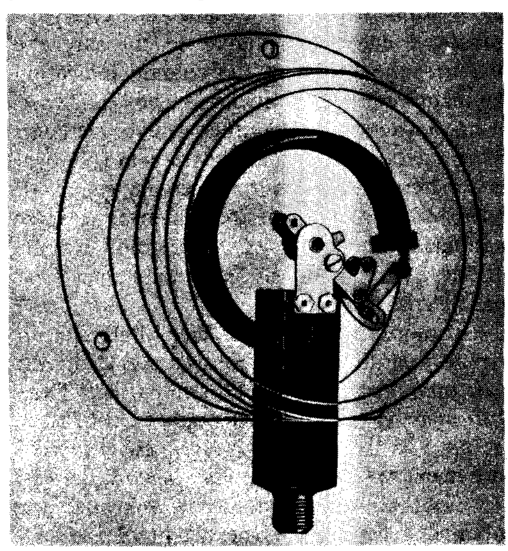


Fig. 1-5. Bourdon tube gage mechanism. (Courtesy Marsh Instrument Company.)

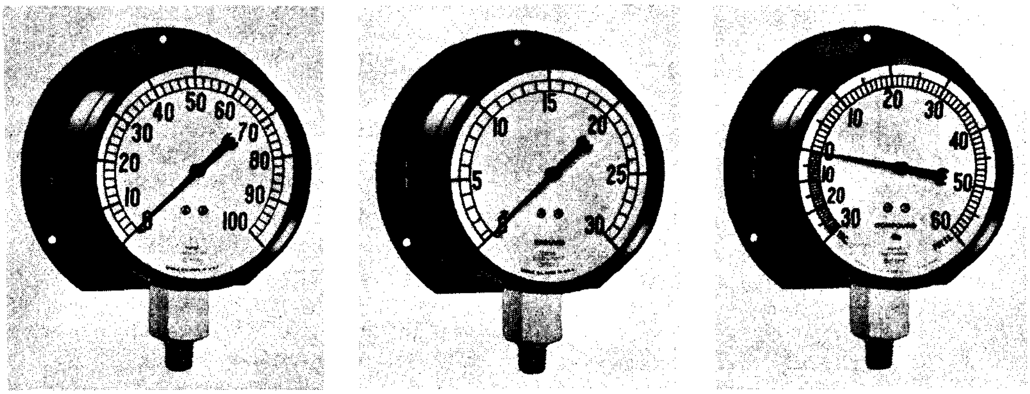
pressure of only 0.036 psi is sufficient to support a column of water 1 in. high. Hence, 1 in. of water column is equivalent to 0.036 psi.

Table 1-1 gives the relationship between the various units of pressure measurement.

1-8. Bourdon Tube Gages. Because of the excessive length of tube required, gages of the manometer type are not practical for measuring pressures above 15 psi and are more or less limited to the measurement of relatively small pressures in air ducts, etc.

Gages of the bourdon tube type are widely used to measure the higher pressures encountered in refrigeration work. The actuating mechanism of the bourdon tube gage is illustrated in Fig. 1-5. The bourdon tube, itself, is a curved, elliptical-shaped, metallic tube which tends to straighten as the fluid pressure in the tube increases and to curl tighter as the pressure decreases. Any change in the curvature of the tube is transmitted through a system of gears to the pointer. The direction and magnitude of the pointer movement depend on the direction and magnitude of the change in the curvature of the tube.

Bourdon tube gages are very rugged and will measure pressures either above or below atmospheric pressure. Those designed to measure pressures above atmospheric are known as "pressure" gages (Fig. 1-6*a*) and are generally calibrated in psi, whereas those designed to read pressures below atmospheric are called "vacuum" gages and are usually calibrated in inches of mercury (Fig. 1-6*b*). In many cases, single gages, known as "compound" gages, are designed to measure pressures both above and below atmospheric (Fig. 1-6*c*). Such gages are calibrated to read in psi above atmospheric and in inches of mercury below atmospheric.



(*a*) (*b*) (*c*)

Fig. 1-6. Typical bourdon tube gages. (*a*) Pressure gage. (*b*) Vacuum gage. (*c*) Compound gage. (Courtesy Marsh Instrument Company.)

1-9. Absolute and Gage Pressures. Absolute pressure is understood to be the "total" or "true" pressure of a fluid, whereas gage pressure is the pressure as indicated by a gage. It is important to understand that gages are calibrated to read zero at atmospheric pressure and that neither the manometer nor the bourdon tube gage measures the "total" or "true" pressure of the fluid in a vessel; both measure only the difference in pressure between the total pressure of the fluid in the vessel and the atmospheric pressure. When the fluid pressure is greater than the atmospheric pressure, the absolute pressure of the fluid in the vessel is determined by adding the atmospheric pressure to the gage pressure, and, when the fluid pressure is less than atmospheric, the absolute pressure of the fluid is found by subtracting the gage pressure from the atmospheric pressure. The relationship between absolute pressure and gage pressure is shown graphically is Fig. 1-7.

Example 1-4. A pressure gage on a refrigerant condenser reads 120 psi. What is the absolute pressure of the refrigerant in the condenser?

Solution. Since the barometer reading is not given, it is assumed that the atmospheric

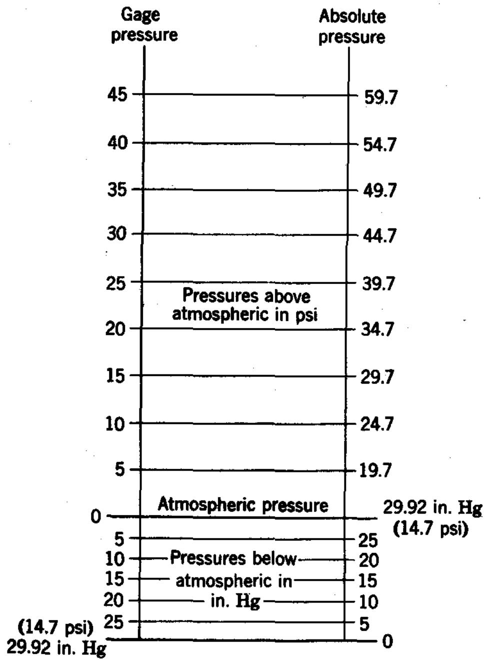


Fig. 1-7. Relationship between absolute and gage pressures.

pressure is normal at sea level, 14.696 psi, and, since the pressure of the refrigerant is above atmospheric, the absolute pressure of the refrigerant is equal to the gage pressure plus the atmospheric pressure.

Gage pressure in psi	= 120
Atmospheric pressure in psi	= 14.696
Absolute pressure of refrigerant	= 134.696 psi

Example 1-5. A compound gage on the suction side of a vapor compressor reads 5 in. Hg, whereas a barometer nearby reads 29.6 in. Hg. Determine the absolute pressure of the vapor entering the compressor.

Solution. Since the pressure of the vapor entering the compressor is less than atmospheric, the absolute pressure of the vapor is computed by subtracting the gage pressure from atmospheric pressure.

Atmospheric pressure in in. Hg	= 29.6
Gage pressure in in. Hg	= 5.0
Absolute pressure in in. Hg	= 24.6 in. Hg
Absolute pressure	24.6 × 0.491
	= 12.08 psi

Example 1-6. During compression the pressure of a vapor is increased from 10 in. Hg gage to 125 psi gage. Calculate the total increase in pressure in psi.

Solution. Since the pressure increases from 10 in. Hg below atmospheric to 125 psi above atmospheric, the total increase in pressure is the sum of the two pressures.

Initial pressure	= 10 in. Hg
Initial pressure in psi below atmospheric	10 × 0.491
	= 4.91 psi
Final pressure in psi above atmospheric	= 125 psi
Total increase in pressure	= 129.91 psi

Absolute pressure in psi is abbreviated psia, whereas gage pressure in psi is abbreviated psig.

1-10. Work. Work is done when a force acting on a body moves the body through a distance. The amount of work done is the product of the force and the distance through which the force acts. This relationship is shown by the following equation:

$$W = F \times 1 \tag{1-4}$$

where F = the force applied in any units of force
1 = the distance through which the force acts in any linear unit
W = the work done expressed in units of force and linear measure

The work done is always expressed in the same unit terms used to express the magnitude of the force and the distance. For instance, if the force is expressed in pounds and the distance in feet, the work done is expressed in foot-pounds. The foot-pound is the most frequently used unit of work measure.

Example 1-7. A ventilating fan weighing 315 lb is hoisted to the roof of a building 200 ft above the level of the ground. How much work is done?

Solution. By applying Equation 1-4, the weight of the fan = 315 lb

Distance through which the fan is hoisted	= 200 ft
Work done	315 × 200
	= 63,000 ft-lb

1-11. Power. Power is the rate of doing work. That is, it is the work done divided by the time required to do the work. The unit of power is the horsepower. One horsepower is defined as the power required to do work at the rate of 33,000 ft-lb per minute or (33,000/60) 550 ft-lb per second. The power required in horsepower may be found by either of the following equations:

$$\text{Hp} = \frac{W}{33{,}000 \times t} \tag{1-5}$$

where Hp = the horsepower
W = the work done in foot-pounds
t = the time in minutes

or

$$\text{Hp} = \frac{W}{550 \times t} \tag{1-6}$$

where t = the time in seconds

Example 1-8. In Example 1-7, if the time required to hoist the fan to the roof of the building is 5 minutes, how much horsepower is required?

Solution. Total work done $= 63{,}000$ ft-lb
Time required to do the work $= 5$ min
Horsepower required $= \dfrac{63{,}000}{33{,}000 \times 5} = 0.382$ hp

1-12. Energy. In order to do work or to cause motion of any kind, energy is required. A body is said to possess energy when it has the capacity for doing work. Hence, energy is described as the ability to do work. The amount of energy required to do a given amount of work is always equal to the amount of work done and the amount of energy a body possesses is equal to the amount of work a body can do in passing from one condition or position to another.

Energy may be possessed by a body in either or both of two basic kinds: (1) kinetic and (2) potential.

1-13. Kinetic Energy. Kinetic energy is the energy a body possesses as a result of its motion or velocity. For instance, a hammer swinging through an arc, a bullet speeding toward a target, and the moving parts of machinery all have kinetic energy by virtue of their motion. The amount of kinetic energy a body possesses is a function of its mass and its velocity and may be determined by the following equation:

$$K = \frac{M \times V^2}{2g} \tag{1-7}$$

where K = the kinetic energy in foot-pounds
M = the weight of the body in pounds
V = the velocity in feet per second (fps)
g = gravitational constant (32.174 ft/sec^2)

Example 1-9. An automobile weighing 3500 lb is moving at the rate of 30 mph. What is its kinetic energy?

Solution. Velocity in fps V $= \dfrac{5280 \text{ ft/mi} \times 30 \text{ mi}}{3600 \text{ sec/hr}} = 44$ fps

Applying Equation 1-7, the kinetic energy K $= \dfrac{3500 \text{ lb} \times (44 \text{ fps})^2}{2 \times 32.174 \text{ ft/sec}^2} = 105{,}302$ ft-lb

1-14. Potential Energy. Potential energy is the energy a body possesses because of its position or configuration. The amount of work a body can do in passing from a given position or condition to some reference position or condition is a measure of the body's potential energy. For example, the driving head of a pile-driver has potential energy of position when raised to some distance above the top of a piling. If released, the driving head can do the work of driving the piling. A compressed steel spring or a stretched rubber band possesses potential energy of configuration. Both the steel spring and the rubber band have the ability to do work because of their tendency to return to their normal condition.

The potential energy of a body may be evaluated by the following equation:

$$P = M \times Z \tag{1-8}$$

where P = the potential energy in foot-pounds
M = the weight of the body in pounds
Z = the vertical distance above some datum or reference

Example 1-10. Ten thousand gallons of water are stored in a tank located 250 ft above the ground. Determine the potential energy of the water in relation to the ground.

Solution. The weight of the water in pounds M $= 10{,}000 \text{ gal} \times 8.33 \text{ lb/gal} = 83{,}300$ lb

Applying Equation 1-8, the potential energy P $= 83{,}300 \text{ lb} \times 250 \text{ ft} = 20{,}825{,}000$ ft-lb

1-15. Energy as Stored Work. Before a body can possess energy, work must be done on the body. The work which is done on a body to give the body its motion, position, or configuration is stored in the body as energy. Hence, energy is stored work. For instance, work must be done to stretch the rubber band, to compress the steel spring, or to raise the driving head of a pile-driver to a position above the piling. In any case, the potential energy stored is equal to the work done.

The amount of energy a body possesses can be ascertained by determining the amount of work done on the body to give the body its motion, position, or configuration. For example, assume that the driving head of a pile-driver weighing 200 lb is raised to a position 6 ft above the top of a piling. The work done in raising the driving head is 1200 ft-lb (200 lb × 6 ft). Therefore, 1200 ft-lb of energy are stored in the driving-head in its raised position and, when released, neglecting friction, the driving-head will do 1200 ft-lb of work on the piling.

1-16. Total External Energy. The total external energy of a body is the sum of its kinetic and potential energies.

Example 1-11. Determine the total external energy of an airplane weighing 10,000 lb and flying 6000 ft above the ground at a speed of 300 mph.

Solution. Applying Equation 1-7, the kinetic energy K

$$= \frac{10{,}000 \text{ lb} \times (440 \text{ fps})^2}{2 \times 32.174 \text{ ft/sec}^2}$$

$$= 30{,}086{,}436 \text{ ft-lb}$$

Applying Equation 1-8, the potential energy P

$$= 10{,}000 \text{ lb} \times 6000 \text{ ft}$$
$$= 60{,}000{,}000 \text{ ft-lb}$$

Adding, the total external energy

$$= 90{,}086{,}436 \text{ ft-lb}$$

1-17. Law of Conservation of Energy. The First Law of Thermodynamics states in effect that the amount of energy is constant. None can be either created or destroyed. Energy is expended only in the sense that it is converted from one form to another.

1-18. Forms of Energy. All energy can be classified as being of either of the two basic kinds, kinetic or potential. However, energy may appear in any one of a number of different forms, such as mechanical energy, electrical energy, chemical energy, heat energy, etc., and is readily converted from one form to another. Electrical energy, for instance, is converted into heat energy in an electric toaster, heater, or range. Electrical energy is converted into mechanical energy in electric motors, solenoids, and other electrically operated mechanical devices. Mechanical energy, chemical energy, and heat energy are converted into electrical energy in the generator, battery, and thermocouple, respectively. Chemical energy is converted into heat energy in chemical reactions such as combustion and oxidation. These are only a few of the countless ways in which the transformation of energy can and does occur. There are many fundamental relationships which exist between the various forms of energy and their transformation, some of which are of particular importance in the study of refrigeration and are discussed in detail later.

PROBLEMS

1. The cooling tower on the roof of a building weighs 1360 lb when filled with water. If the basin of the tower measures 4 ft by 5 ft, what is the pressure exerted on the roof
(*a*) in pounds per square foot? *Ans.* 68 psf.
(*b*) in pounds per square inch? *Ans.* 0.472 psi.

2. If the atmospheric pressure is normal at sea level and a gage on an R-12 condenser reads 130 psi, what is the absolute pressure of the Freon in the condenser in pounds per square inch? *Ans.* 144.7 psia.

3. What is the total force exerted on the top of a piston if the area of the cylinder bore is 5 sq in. and the pressure of the gas in the cylinder is 150 psi? *Ans.* 750 lb.

4. A barometer reads 10 in. Hg. What is the atmospheric pressure in psi? *Ans.* 4.91 psi.

5. A barometer on the wall reads 29.6 in. Hg while a gage on the tank of an air compressor indicates 105 psi. What is the absolute pressure of the air in the tank in pounds per square foot? *Ans.* 119.53 psia.

6. A gage on the suction inlet of a compressor reads 10 in. Hg. Determine the absolute pressure of the suction vapor in psi. *Ans.* 9.79 psia.

7. A gage on the suction side of a refrigeration compressor reads 5 in. Hg. If a gage on the discharge side of the compressor reads 122 psi, what is the increase in pressure during the compression? *Ans.* 124.46 psi.

8. An electric motor weighing 236 lb is hoisted to the roof of a building in 2 min. If the roof is 125 ft above the ground,

(*a*) How much work is done?

Ans. 29,500 ft-lb

(*b*) Neglecting friction and other losses, what is the horsepower required?

Ans. 0.447 hp

9. Compute the kinetic energy of an automobile weighing 3000 lb and moving at a speed of 75 mph. *Ans.* 567,188 ft-lb

10. What is the total external energy of the automobile in Problem 9 if the automobile is traveling along a highway 6000 ft above sea level? *Ans.* 18,000,000 ft-lb

11. What is the total potential energy of 8000 gal of water confined in a tank and located a mean distance of 135 ft above the ground? *Ans.* 8,996,400 ft-lb

12. Water in a river 800 ft above sea level is flowing at the rate of 5 mph. Calculate the sum of the kinetic and potential energies per pound of water in reference to sea level.

Ans. 800.84 ft-lb

13. A water pump delivers 60 gal per minute of water to a water tank located 100 ft above the level of the pump. If water weighs 8.33 lb per gallon and if the friction of the pipe and other losses are neglected,

(*a*) How much work is done?

Ans. 49,980 ft-lb

(*b*) Compute the horsepower required.

Ans. 1.5 hp.

2

Matter, Internal Energy, Heat, Temperature

2-1. Heat. Heat is a form of energy. This is evident from the fact that heat can be converted into other forms of energy and that other forms of energy can be converted into heat. However, there is some confusion as to exactly what energy shall be termed heat energy. Popular usage has made the concept of heat as internal or molecular energy almost universally accepted. Because of this, referring to heat as internal energy is almost unavoidable at times. On the other hand, from a strictly thermodynamic point of view, heat is defined as energy in transition from one body to another as a result of a difference in temperature between the two bodies. Under this concept, all other energy transfers occur as work. Both these concepts of heat will evolve in this and the following chapters. The term heat will be used hereafter in this book in either sense.

2-2. Matter and Molecules. Everything in the universe that has weight or occupies space, all matter, is composed of molecules. Molecules, in turn, are made up of smaller particles called atoms and atoms are composed of still smaller particles known as electrons, protons, neutrons, etc. The study of atoms and subatomic particles is beyond the scope of this book and the discussion will be limited for the most part to the study of molecules and their behavior.

The molecule is the smallest, stable particle of matter into which a particular substance can be subdivided and still retain the identity of the original substance. For example, a grain of table salt (NaCl) may be broken down into individual molecules and each molecule will be a molecule of salt, the original substance. However, all molecules are made up of atoms, so that it is possible to further subdivide a molecule of salt into its component atoms. But, a molecule of salt is made up of one atom of sodium and one atom of chlorine. Hence, if a molecule of salt is divided into its atoms, the atoms will not be atoms of salt, the original substance, but atoms of two entirely different substances, one of sodium and one of chlorine.

There are some substances whose molecules are made up of only one kind of atoms. The molecule of oxygen (O_2), for instance, is composed of two atoms of oxygen. If a molecule of oxygen is divided into its two component atoms, each atom will be an atom of oxygen, the original substance, but the atoms of oxygen will not be stable in this condition. They will not remain as free and separate atoms of oxygen, but, if permitted, will either join with atoms or molecules of another substance to form a new compound or rejoin each other to form again a molecule of oxygen.

It is assumed that the molecules that make up a substance are held together by forces of mutual attraction known as cohesion. These forces of attraction that the molecules have for each other may be likened to the attraction that exists between unlike electrical charges or between unlike magnetic poles. However, despite the mutual attraction that exists between the molecules and the resulting influence that each molecule has upon the others, the molecules are not tightly packed together. There is a certain amount of space between them and they are relatively free to move about. The molecules are further assumed to be in a state of rapid and constant vibration or motion, the rate and extent of the vibration or movement being determined by the amount of energy they possess.

2-3. Internal Energy. It has been previously stated that energy is required to do work or to cause motion of any kind. Molecules, like everything else, can move about only if they possess energy. Hence, a body has internal energy as well as external energy. Whereas a body has external mechanical energy because of its velocity, position, or configuration in relation to some reference condition, it also has internal energy as a result of the velocity, position, and configuration of the molecules of the materials which make up the body.

The molecules of any material may possess energy in both kinds, kinetic and potential. The total internal energy of a material is the sum of its internal kinetic and potential energies. This relationship is shown by the equation

$$U = K + P \tag{2-1}$$

where U = the total internal energy
K = the internal kinetic energy
P = the internal potential energy

2-4. Internal Kinetic Energy. Internal kinetic energy is the energy of molecular motion or velocity. When heat energy flowing into a material increases the internal kinetic energy, the velocity or motion of the molecules is increased. The increase in molecular velocity is always accompanied by an increase in the temperature of the material. Hence, a material's temperature is, in a sense, a measure of the average velocity of the molecules which make up the material. The more kinetic energy the molecules have, the greater is their movement and the faster they move. The more rapid the motion of the molecules, the hotter is the material and the more internal kinetic energy the material has. It follows, then, that if the internal kinetic energy of the material is diminished by the removal of heat, the motion of the molecules will be slowed down or retarded and the temperature of the material will be decreased.

According to the kinetic theory if the removal of heat continues until the internal kinetic energy of the material is reduced to zero, the temperature of the material will drop to Absolute Zero (approximately $-460°$ F) and the motion of the molecules will cease entirely.*

* It is now known that the energy is not zero at Absolute Zero. It is the disorganization (entropy) which diminishes to zero. Heat is sometimes defined as "disorganized energy." Both the energy and the disorganization decrease as the temperature decreases. However, the disorganization decreases faster than the energy and therefore diminishes to zero before the energy reaches zero.

2-5. States of Matter. Matter can exist in three different phases or states of aggregation: solid, liquid, or a vapor or gas. For example, water is a liquid, but this same substance can exist as ice, which is a solid, or as steam, which is a vapor or gas.

2-6. The Effect of Heat on the State of Aggregation. Many materials, under the proper conditions of pressure and temperature, can exist in any and all of the three physical states of matter. It will be shown presently that the amount of energy the molecules of the material have determines not only the temperature of the material but also which of the three physical states the material will assume at any particular time. In other words, the addition or removal of heat can bring about a change in the physical state of the material as well as a change in its temperature.

That heat can bring about a change in the physical state of a material is evident from the fact that many materials, such as metals will become molten when sufficient heat is applied. Furthermore, the phenomenon of melting ice and boiling water is familiar to everyone. Each of these changes in the physical state is brought about by the addition of heat.

2-7. Internal Potential Energy. Internal potential energy is the energy of molecular separation or configuration. It is the energy the molecules have as a result of their position in relation to one another. The greater the degree of molecular separation, the greater is the internal potential energy.

When a material expands or changes its physical state with the addition of energy, a rearrangement of the molecules takes place which increases the distance between them. Inasmuch as the molecules are attracted to one another by forces which tend to pull them together, internal work must be done in order to separate further the molecules against their attractive forces. An amount of energy equal to the amount of internal work done must flow into the material. This energy is set up in the material as an increase in the internal potential energy. It is "stored" energy which is accounted for by the increase in the mean distance between the molecules. The source of this energy is the heat energy supplied.

It is important to understand that in this instance the energy flowing into the material has no effect on molecular velocity (internal kinetic energy); only the degree of molecular separation (the internal potential energy) is affected.

2-8. The Solid State. A material in the solid state has a relatively small amount of internal potential energy. The molecules of the material are rather closely bound together by each other's attractive forces and by the force of gravity. Hence, a material in the solid state has a rather rigid molecular structure in which the position of each molecule is more or less fixed and the motion of the molecules is limited to a vibratory type of movement which, depending upon the amount of internal kinetic energy the molecules possess, may be either slow or rapid.

Because of its rigid molecular structure, a solid tends to retain both its size and its shape. A solid is not compressible and will offer considerable resistance to any effort to change its shape.

2-9. The Liquid State. The molecules of a material in the liquid state have more energy than those of a material in the solid state and they are not so closely bound together. Their greater energy allows them to overcome each other's attractive forces to some extent and to have more freedom to move about. They are free to move over and about one another in such a way that the material is said to "flow." Although a liquid is noncompressible and will retain its size, because of its fluid molecular structure, it will not retain its shape, but will assume the shape of any containing vessel.

2-10. The Vapor or Gaseous State. The molecules of a material in the gaseous state have an even greater amount of energy than those of a material in the liquid state. They have sufficient energy to overcome all restraining forces. They are no longer bound by each other's attractive forces, neither are they bound by the force of gravity. Consequently, they fly about at high velocities, continually colliding with each other and with the walls of the container. For this reason, a gas will retain neither its size nor its shape. It is readily compressible and will completely fill any container regardless of size. Further, if the gas is not stored in a sealed container, it will escape from the container and be diffused into the surrounding air.

2-11. Temperature. Temperature is a property of matter. It is a measure of the level of heat intensity or the thermal pressure of a body. A high temperature indicates a high level of heat intensity or thermal pressure, and the body is said to be hot. Likewise, a low temperature indicates a low level of heat intensity or thermal pressure and the body is said to be cold.

2-12. Thermometers. The most frequently used instrument for measuring temperature is the thermometer. The operation of most thermometers depends upon the property of a liquid to expand or contract as its temperature is increased or decreased, respectively. Because of their low freezing temperatures and relatively constant coefficients of expansion, alcohol and mercury are the liquids most frequently used in thermometers. The mercury thermometer is the more accurate of the two because its coefficient of expansion is more constant through a greater temperature range than is that of alcohol. However, mercury thermometers have the disadvantage of being more expensive and more difficult to read. Alcohol is cheaper and can be colored for easy visibility.

Two temperature scales are in common use today. The Fahrenheit scale is used in English speaking countries, whereas the Centigrade scale is widely used in European countries as well as for scientific purposes.

2-13. Centigrade Scale. The point at which water freezes under atmospheric pressure is taken as the arbitrary zero point on the Centigrade scale, and the point at which water boils is designated as 100. The distance on the scale between these two points is divided into one hundred equal units called degrees, so that the distance between the freezing and boiling points of water on the Centigrade scale is 100°. Water freezes at 0° Centigrade and boils at 100° Centigrade.

2-14. Fahrenheit Scale. Although there is some disagreement as to the actual method used by Fahrenheit in designing the first temperature scale, it was arrived at by means similar to those described in the previous section. On the Fahrenheit scale, the point at which water freezes is marked as 32, and the point at which water boils 212. Thus, there are 180 units between the freezing and boiling points of water. The zero or reference point on the Fahrenheit scale is placed 32 units or degrees below the freezing point of water and is assumed to represent the lowest temperature Fahrenheit could achieve with a mixture of ammonium chloride and snow.

2-15. Temperature Conversion. Temperature readings on one scale can be converted to reading on the other scale by using the appropriate of the following equations:

$$°F = 9/5°C + 32 \qquad (2\text{-}2)$$

$$°C = 5/9(°F - 32) \qquad (2\text{-}3)$$

It should be noted that the difference between the freezing and boiling points of water on the Fahrenheit scale is 180°, whereas the difference between these two points on the Centigrade scale is only 100°. Therefore, 100 Centigrade degrees are equivalent to 180 Fahrenheit degrees. This establishes a relationship such that 1° C equals 9/5° F (1.8° F) and 1° F equals 5/9° C (0.555° C). This is shown graphically in Fig. 2-1. Since 0° on the Fahrenheit scale is 32° F below the freezing point of water, it is necessary to add 32° F to the Fahrenheit equivalent after converting from Centigrade. Likewise, it is necessary to subtract 32° F from a Fahrenheit reading before converting to Centigrade.

Example 2-1. Convert a temperature reading of 50° C to the equivalent Fahrenheit temperature.

Solution. Applying Equation 2-2, ° F   9/5(50° C) + 32
= 122° F

Example 2-2. A thermometer on the wall of a room reads 86° F. What is the room temperature in degrees Centigrade?

Solution. Applying Formula 2-3, the room temperature in ° C   5/9(86-32)
= 30° C

Example 2-3. A thermometer indicates that the temperature of a certain quantity of water is increased 45° F by the addition of heat. Compute the temperature rise in Centigrade degrees.

Solution. Temperature rise in ° F
= 45° F

Temperature rise in ° C 5/9(45° F)
= 25° C

2-16. Absolute Temperature. Temperature readings taken from either the Fahrenheit or Centigrade scales are in respect to arbitrarily selected zero points which, as has been shown, are not even the same for the two scales. When it is desired to know only the change in temperature that occurs during a process or the temperature of a substance in relation to some known reference point, such readings are entirely adequate. However, when temperature readings are to be applied in equations dealing with certain fundamental laws, it is necessary to use temperature readings whose reference point is the true or absolute zero of temperature. Experiment has indicated that such a point, known as Absolute Zero, exists at approximately −460° F or −273° C (Fig. 2-1).

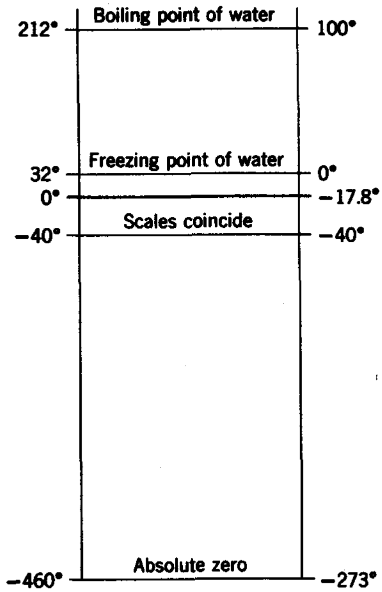


Fig. 2-1. Comparison of Fahrenheit and Centigrade temperature scales.

Temperature readings in reference to Absolute Zero are designated as absolute temperatures and may be in either Fahrenheit or Centigrade degrees. A temperature reading on the Fahrenheit scale can be converted to absolute temperature by adding 460° to the Fahrenheit reading. The resulting temperature is in degrees Rankine (° R).

Likewise, Centigrade temperatures can be converted to absolute temperatures by adding 273° to the Centigrade reading. The resulting temperature is stated in degrees Kelvin (° K).

In converting to and from absolute temperatures, the following equations will apply:

$$T = t + 460 \qquad (2\text{-}4)$$

$$t = T - 460 \qquad (2\text{-}5)$$

$$T = t + 273 \qquad (2\text{-}6)$$

$$t = T - 273 \qquad (2\text{-}7)$$

where T = absolute temperature in degrees Rankine or Kelvin
t = temperature in degrees Fahrenheit or Centigrade

Equations 2-4 and 2-5 apply to the Rankine and Fahrenheit scales, whereas Equations 2-6 and 2-7 apply to the Kelvin and Centigrade scales. Hereafter in this book Rankine and Fahrenheit temperatures are used unless otherwise specified.

Example 2-4. A thermometer on the tank of an air compressor indicates that the temperature of the air in the tank is 95° F. Determine the absolute temperature in degrees Rankine.

Solution. Applying Equation 2-4,

$$T = 95° \text{ F} + 460° = 555° \text{ R}$$

Example 2-5. The temperature of the vapor entering the suction of a refrigeration compressor is −20° F. Compute the temperature of the vapor in degrees Rankine.

Solution. Applying Equation 2-4,

$$T = -20° \text{ F} + 460° = 440° \text{ R}$$

Example 2-6. If the temperature of a gas is 100° C, what is its temperature in degrees Kelvin?

Solution. Applying Equation 2-6,

$$T = 100° \text{ C} + 273° = 373° \text{ K}$$

Example 2-7. The temperature of steam leaving a boiler is 610° R. What is the temperature of the steam on the Fahrenheit scale?

Solution. Applying Equation 2-5,

$$t = 610° \text{ R} - 460° = 150° \text{ F}$$

2-17. Direction and Rate of Heat Flow. Heat will flow from one body to another when, and only when, a difference in temperature exists between the two bodies. If the temperature of the two bodies is the same, there is no transfer of heat.

Heat always flows down the temperature scale from a high temperature to a low temperature, from a hot body to a cold body, and never in the opposite direction. Since heat is energy and cannot be destroyed, if heat is to leave one body of material, it must flow into and be absorbed by another body of material whose temperature is below that of the body being cooled.

The rate of heat transfer between two bodies is always directly proportional to the difference in temperature between the two bodies.

2-18. Methods of Heat Transfer. The transfer of heat from one place to another occurs in three ways: (1) conduction, (2) convection, and (3) radiation.

2-19. Conduction. Heat transfer by conduction occurs when energy is transmitted by direct contact between the molecules of a single body or between the molecules of two or more bodies in good thermal contact with each other. In either case, the heated molecules communicate their energy to the other molecules immediately adjacent to them. The transfer of energy from molecule to molecule by conduction is similar to that which takes place between the balls on a billiard table, wherein all or some part of the energy of motion of one ball is transmitted at the moment of impact to the other balls that are struck.

When one end of a metal rod is heated over a flame, some of the heat energy from the heated end of the rod will flow by conduction from molecule to molecule through the rod to the cooler end. As the molecules at the heated end of the rod absorb energy from the flame, their energy increases and they move faster and through a greater distance. The increased energy of the heated molecules causes them to strike against the molecules immediately adjacent to them. At the time of impact and because of it, the faster moving molecules transmit some of their energy to their slower moving neighbors so that they too begin to move more rapidly. In this manner, energy passes from molecule to molecule from the heated end of the rod to the cooler end. However, in no case would it be possible for the molecules furthest from the heat source to have more energy than those at the heated end.

As heat passes through the metal rod, the air immediately surrounding the rod is also heated by conduction. The rapidly vibrating particles of the heated rod strike against the molecules of the air which are in contact with the rod. The energy so imparted to the air molecules causes them to move about at a higher rate and communicate their energy to other nearby air molecules. Thus, some of the heat supplied to the metal rod is conducted to and carried away by the surrounding air.

If the heat supply to the rod is interrupted, heat will continue to be carried away from the rod by the air surrounding until the temperature of the rod drops to that of the air. When this occurs, there will be no temperature differential, the system will be in equilibrium, and no heat will be transferred.

The rate of heat transfer by conduction, as previously stated, is in direct proportion to the difference in temperature between the high and low temperature parts. However, all materials do not conduct heat at the same rate. Some materials, such as metals, conduct heat very readily, whereas others, such as glass, wood, and cork, offer considerable resistance to the conduction of heat. Therefore, for any given temperature difference, the rate of heat flow by conduction through different materials of the same length and cross section will vary with the particular ability of the various materials to conduct heat. The relative capacity of a material to conduct heat is known as its conductivity. Materials which are good conductors of heat have a high conductivity, whereas materials which are poor conductors have a low conductivity and are used as heat insulators.

In general, solids are better conductors of heat than liquids, and liquids are better conductors than gases. This is accounted for by the difference in the molecular structure. Since the molecules of a gas are widely separated, the transfer of heat by conduction, that is, from molecule to molecule, is difficult.

2-20. Convection. Heat transfer by convection occurs when heat moves from one place to another by means of currents which are set up within some fluid medium. These currents are known as convection currents and result from the change in density which is brought about by the expansion of the heated portion of the fluid.

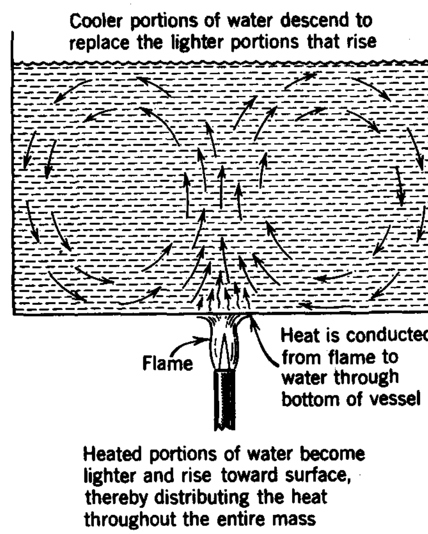


Fig. 2-2. Convection currents set up in a vessel of water when the vessel is heated at bottom center.

When any portion of a fluid is heated, it expands and its volume per unit of weight increases. Thus, the heated portion becomes lighter, rises to the top, and is immediately replaced by a cooler, heavier portion of the fluid. For example, assume that a tank of water is heated on the bottom at the center (Fig. 2-2). The heat from the flame is conducted through the metal bottom of the tank to the water inside. As the water adjacent to the heat source absorbs heat, its temperature increases and it expands. The heated portion of the water, being lighter than the water surrounding, rises to the top and is replaced by cooler, more dense water pushing in from the sides. As this new portion of water becomes heated, it too rises to the top and is replaced by cooler water from the sides. As this sequence continues, the heat is distributed throughout the entire mass of the water by means of the convection currents established within the mass.

Warm air currents, such as those which occur over stoves and other hot bodies, are familiar to everyone. How convection currents are utilized to carry heat to all parts of a heated space is illustrated in Fig. 2-3.

2-21. Radiation. Heat transfer by radiation occurs in the form of a wave motion similar to light waves wherein the energy is transmitted from one body to another without the need for

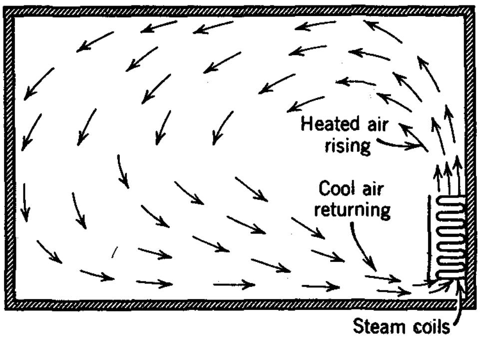


Fig. 2-3. Room heated by natural convection.

intervening matter. Heat energy transmitted by wave motion is called radiant energy.

It is assumed that the molecules of a body are in rapid vibration and that this vibration sets up a wave motion in the ether surrounding the body.* Thus, the internal molecular energy of the body is converted into radiant energy waves. When these energy waves are intercepted by another body of matter, they are absorbed by that body and are converted into its internal molecular energy.

The earth receives heat from the sun by radiation. The energy of the sun's molecular vibration is imparted in the form of radiant energy waves to the ether of interstellar space surrounding the sun. The energy waves travel across billions of miles of space and impress their energy upon the earth and upon any other material bodies which intercept their path. The radiant energy is absorbed and transformed into internal molecular energy, so that the vibratory motion of the hot body (the sun) is reproduced in the cooler body (the earth).

All materials give off and absorb heat in the form of radiant energy. Any time the temperature of a body is greater than that of its surroundings, it will give off more heat by radiation than it absorbs. Therefore, it loses energy to its surroundings and its internal energy decreases. If the temperature of the body is below that of its surroundings, it absorbs more radiant energy than it loses and its internal energy increases. When no temperature difference exists, the energy exchange is in equilibrium and the body neither gains nor loses energy.

* Ether is the name given to that which fills all space unoccupied by matter, such as interstellar space and the space between the molecules of every material.

Heat transfer through a vacuum is impossible by either conduction or convection, since these processes by their very nature require that matter be the transmitting media. Radiant energy, on the other hand, is not dependent upon matter as a medium of transfer and therefore can be transmitted through a vacuum. Furthermore, when radiant energy is transferred from a hot body to a cold body through some intervening media such as air the temperature of the intervening media is unaffected by the passage of the radiant energy. For example, heat is radiated from a "warm" wall to a "cold" wall through the intervening air without having any appreciable effect upon the temperature of the air. Since the molecules of the air are relatively few and widely separated, the waves of radiant energy can easily pass between them so that only a very small part of the radiant energy is intercepted and absorbed by the molecules of the air. By far the greater portion of the radiant energy impinges upon and is absorbed by the solid wall whose molecular structure is much more compact and substantial.

Heat waves are very similar to light waves, differing from them only in length and frequency. Light waves are radiant energy waves of such length as to be visible to the human eye. Thus, light waves are visible heat waves. Whether heat waves are visible or invisible depends upon the temperature of the radiating body. For example, when metal is heated to a sufficiently high temperature, it will "glow," that is, emit visible heat waves (light).

When radiant energy waves, either visible or invisible, strike a material body, they may be reflected, refracted, or absorbed by it, or they may pass through it to some other substance beyond.

The amount of radiant energy which will pass through a material depends upon the degree of transparency. A highly transparent material, such as clear glass or air, will allow most of the radiant energy to pass through to the materials beyond, whereas opaque materials, such as wood, metal, cork, etc., cannot be penetrated by radiant energy waves and none will pass through.

The amount of radiant energy which is either reflected or absorbed by a material depends upon the nature of the material's surface, that is, its texture and its color. Materials having a light-colored, highly polished surface, such as a mirror, reflect a maximum of radiant energy, whereas materials having rough, dull, dark surfaces will absorb the maximum amount of radiant energy.

2-22. British Thermal Unit. It has already been established that a thermometer measures only the intensity of heat and not the quantity. However, in working with heat it is often necessary to determine heat quantities. Obviously, some unit of heat measure is required.

Heat is a form of energy, and as such is intangible and cannot be measured directly. Heat can be measured only by measuring the effects it has on a material, such as the change in temperature, state, color, size, etc.

The most universally used unit of heat measure is the British thermal unit, abbreviated Btu. A Btu is defined as the quantity of heat required to change the temperature of 1 lb of water 1° F. This quantity of heat, if added to 1 lb of water, will raise the temperature of the water 1° F. Likewise, if 1 Btu is removed from 1 lb of water, the temperature of the water will be lowered 1° F.

The quantity of heat required to change the temperature of 1 lb of water 1° F is not a constant amount. It varies slightly with the temperature range at which the change occurs. For this reason, a Btu is more accurately defined as being 1/180th of the quantity of heat required to raise the temperature of 1 lb of water from the freezing point (32° F) to the boiling point (212° F). This is identified as the "mean Btu" and is the exact amount of heat required to raise the temperature of 1 lb of water from 62 to 63° F. If the change in temperature occurs at any other point on the temperature scale, the amount of heat involved is either more or less than the mean Btu, depending upon the particular point on the temperature scale that the change takes place. However, the variation from the mean Btu is so slight that it may be neglected and, regardless of the temperature range, for all practical purposes it is sufficiently accurate to assume that the temperature of 1 lb of water is changed 1° F by the addition or removal of 1 Btu.

2-23. Specific Heat. The specific heat of a material is the quantity of heat required to change the temperature of 1 pound of the material 1° F. For instance, the specific heat of aluminum is 0.226 Btu/lb/°F, whereas that of brass is 0.089 Btu/lb/°F. This means that 0.226 Btu is required to raise the temperature of 1 pound of aluminum 1° F, whereas only 0.089 Btu is necessary to change the temperature of 1 pound of brass 1° F. Note that by the definition of the Btu the specific heat of water is 1 Btu per pound per degree Fahrenheit.

The specific heat of any material, like that of water, varies somewhat throughout the temperature scale. Here again, the variation is so slight that it is sufficiently accurate for most calculations to consider the specific heat to be a constant amount. This is not true, however, as the material passes through a change in physical state. The specific heat of a material in the solid state is approximately one-half that of the same material in the liquid state. For instance, the specific heat of ice is 0.5 Btu, whereas that of water is one. The specific heat values of materials in the gaseous state are discussed in another chapter.

2-24. Calculating Heat Quantity. The quantity of heat which must be added to or removed from any given mass of material in order to bring about a specified change in its temperature can be computed by using the following equation:

$$Q_s = MC(t_2 - t_1) \qquad (2\text{-}8)$$

where Q_s = the quantity of heat either absorbed or rejected by the material
M = the weight of the material in pounds
C = the specific heat of the material
t_1 = the initial temperature
t_2 = the final temperature

Example 2-8. Twenty pounds of water at an initial temperature of 76° F are heated until the temperature is increased to 180° F. How much heat must be supplied?

Solution.
Applying Equation 2-8, $Q_s = 20 \text{ lb} \times 1 \times (180 - 76)$
$= 2080$ Btu

Example 2-9. If water weighs 8.33 lb per gallon, how much heat is rejected by 30 gal of water in cooling from 80° F to 35° F?

Solution.

Weight of water in pounds $= 30 \text{ gal} \times 8.33 \text{ lb/gal}$
$= 250 \text{ lb}$

Applying Equation 2-8, $Q_s = 250 \text{ lb} \times 1 \times (35 - 80)$
$= 250 \text{ lb} \times 1 \times (-45)$
$= 11{,}250 \text{ Btu}$

NOTE: Since the specific heat of a material is given in terms of Btu/lb/° F, the weight of the material must be determined before Equation 2-8 can be applied.

Where t_2 is less than t_1, the answer obtained by applying Equation 2-8 will be negative, indicating that heat is rejected by rather than absorbed by the material. In this type of problem, where the direction of heat flow is obvious, the negative sign can be ignored and the answer assumed to be positive.

Example 2-10. Fifteen pounds of cast iron are cooled from 500° F to 250° F by being immersed in 3 gallons (25 lb) of water whose initial temperature is 78° F. Assuming that the specific heat of the cast iron is 0.101 Btu/lb/° F and that all of the heat given up by the cast iron is absorbed by the water, what is the final temperature of the water?

Solution. By applying Equation 2-8 to compute the total quantity of heat given up by the cast iron,

$$Q_s = 15 \text{ lb} \times 0.101 \times 250 = 378.75 \text{ Btu}$$

By rearranging and applying Equation 2-8 to determine the final temperature of the water after absorbing the heat given up by the cast iron,

$$t_2 = \frac{Q_s}{MC} + t_1 = \frac{378.75}{25 \times 1} + 78° \text{ F} = 15.15° + 78° \text{ F} = 93° \text{ F}$$

2-25. Heat Divided into Two Kinds or Categories. It has been previously stated (Section 2-6) that heat has the ability to bring about a change in the physical state of a material as well as the ability to cause a change in its temperature. Heat is divided into two kinds or categories, depending upon which of these two effects it has on a material which either absorbs or rejects it. The division of heat into several classifications is made only to facilitate and simplify certain necessary calculations and does not stem from any difference in the nature of heat itself.

2-26. Sensible Heat. When heat either absorbed or rejected by a material causes or accompanies a change in the temperature of the material, the heat transferred is identified as sensible heat. The term *sensible* is applied to this particular heat because the change in temperature it causes can be detected with the sense of touch and can, of course, be measured with a thermometer.

2-27. Latent Heat. When heat, either added to or rejected by a material, brings about or accompanies a change in the physical state of the material, the heat is known as latent heat. The name *latent*, a Latin word meaning hidden, is said to have been given to this special kind of heat by Dr. Joseph Black because it apparently disappeared into a material without having any effect on the temperature of the material.

Many materials progressing up the temperature scale will pass through two changes in the state of aggregation: first, from the solid to the liquid phase and then, as the temperature of the liquid is further increased to a certain level beyond which it cannot exist as a liquid, the liquid will change into the vapor state. When the change occurs in either direction between the solid and liquid phases, the heat involved is known as the latent heat of fusion. When the change occurs between the liquid and vapor phases, the heat involved is the latent heat of vaporization.

2-28. Sensible Heat of a Solid. To obtain a better understanding of the concept of molecular energy, consider the progressive effects of heat as it is taken in by a material whose initial thermodynamic condition is such that its energy content is zero. Assume that a solid in an open container is at a temperature of −460° F (Absolute Zero). Theoretically, at this temperature the molecules of the material have no energy and are completely at rest.

When heat energy flows into the solid, the molecules of the solid begin to move slowly and the temperature of the solid begins to climb. The more heat energy taken in by the solid, the faster the molecules vibrate and the warmer the solid becomes. The increase in molecular velocity and in the temperature of the solid continues as more heat is absorbed, until the solid reaches its melting or fusion temperature. The total quantity of heat energy required to bring the temperature of the solid from the original condition of Absolute Zero to the melting or fusion temperature is known as the sensible heat of the solid. As previously shown, the quantity of heat which must be transferred in order to bring about a specified change in the temperature of any given mass of any material can be calculated by applying Equation 2-8.

2-29. The Melting or Fusion Temperature. Upon reaching the fusion temperature, the molecules of the solid are moving as rapidly as is possible within the rigid molecular structure of the solid state. It is not possible to increase further the motion of the molecules or the temperature of the solid beyond this point without first overcoming partially the forces of mutual attraction which exists between the molecules. Hence, the material cannot exist in the solid state at any temperature above its melting or fusion temperature. On reaching the fusion temperature, any additional heat absorbed by the material will cause some part of the solid to revert to the liquid phase.

The exact temperature at which melting or fusion occurs varies with the different materials and with the pressure. For instance, at normal atmospheric pressure, the fusion temperature of lead is approximately 600° F, whereas copper melts at approximately 2000° F and ice at only 32° F. In general, the melting temperature decreases as the pressure increases except for noncrystalline solids, whose melting temperatures increase as the pressure increases.

2-30. Latent Heat of Fusion. When heat is absorbed by a solid at the fusion temperature, the molecules of the solid utilize the energy to overcome partially their attraction for one another. They break away from one another to some extent and become more widely separated. As the molecules flow over and about one another, the material loses the rigidity of the solid state and becomes fluid. It can no longer support itself independently and will assume the shape of any containing vessel.

The attraction which exists between the molecules of a solid is considerable and a relatively large quantity of energy is required to overcome that attraction. The quantity of heat required to melt one pound of a material from the solid phase into the liquid phase is called the latent heat of fusion. The latent heat of fusion, along with other values such as specific heat, fusion temperature, etc., for the different materials has been determined by experiment and may be found in various tables.

It is important at this point to emphasize that the change of phase occurs in either direction at the fusion temperature, that is, the temperature at which the solid will melt into the liquid phase is the same as that at which the liquid will freeze into the solid phase. Further, the quantity of heat that must be rejected by a certain weight of liquid at the fusion temperature in order to freeze into the solid state is exactly equal to the amount of heat that must be absorbed by the same weight of the solid in melting into the liquid state.

None of the heat absorbed or rejected during the change of phase has any effect on molecular velocity. Therefore the temperature of the material remains constant during the phase change, and the temperature of the resulting liquid or solid is the same as the fusion temperature.*

The quantity of heat that is absorbed by a given weight of a solid at the fusion temperature in melting into the liquid phase, or, conversely, the quantity of heat that is rejected by a given weight of liquid at the fusion temperature in freezing or solidifying, can be determined by applying the following equation:

$$Q_L = M \times h_{if} \qquad (2\text{-}9)$$

where Q_L = the quantity of heat in Btu
M = the mass or weight in pounds
h_{if} = the latent heat in Btu per pound

Example 2-11. Calculate the quantity of heat required to melt 12 lb of ice at 32° F into water at 32° F. The latent heat of fusion of water under atmospheric pressure is 144 Btu per pound.

Solution. Applying Equation 2-9, the quantity of heat required to melt 12 lb of ice = 12 lb × 144 Btu/lb
= 1728 Btu

NOTE. Since 12 lb of ice absorb 1728 Btu in melting into water, it follows that 12 lb of water at 32° F will reject 1728 Btu in returning to the solid state.

* This applies with absolute accuracy only to crystalline solids. Noncrystalline solids, such as glass, have indefinite fusion temperatures. That is, the temperature will vary during the change of phase. However, for the purpose of calculating heat quantities, the temperature is assumed to remain constant during the phase change.

Example 2-12. If 50 lb of ice at 32° F absorb 6000 Btu, what part of the ice will be melted?

Solution. By rearranging and applying Equation 2-9, the part of the ice melted, M

$$= \frac{Q}{h_{if}} = \frac{6000 \text{ Btu}}{144 \text{ Btu/lb}} = 41.66 \text{ lb}$$

2-31. Sensible Heat of the Liquid. When a material passes from the solid to the liquid phase, the resulting liquid is at the fusion temperature. The temperature of the liquid may then be increased by the addition of heat. Any heat absorbed by a liquid after the change of state is set up in the liquid as an increase in the internal kinetic energy. Molecular velocity increases and the temperature of the liquid rises. But here again, as in the case of the solid, the temperature of the liquid eventually reaches a point beyond which it cannot be further increased. A liquid cannot exist as a liquid at any temperature above its vaporizing temperature for a given pressure and, upon reaching the vaporizing temperature, if additional heat is taken in by the liquid, some part of the liquid will change to the vapor phase.

The total quantity of heat taken in by a liquid as its temperature is increased from the fusion to the vaporizing temperature is called the sensible heat of the liquid. Here again, Equation 2-8, sometimes known as the "sensible heat equation," can be applied to determine the quantity of heat necessary to change the temperature of any given weight of liquid through any specified temperature range.

2-32. Saturation Temperature. The temperature at which a liquid will change into the vapor phase is called the saturation temperature, sometimes referred to as the "boiling point" or "boiling temperature." A liquid whose temperature has been raised to the saturation temperature is called a saturated liquid.

The saturation temperature, that is, the temperature at which vaporization occurs, is different for each liquid. Iron, for example, vaporizes at 4450° F, copper at 4250° F, and lead 3000° F. Water, of course, boils at 212° F, and alcohol at 170° F. Some liquids boil at extremely low temperatures. A few of these are ammonia, oxygen, and helium, which boil at temperatures of −28° F, −295° F, and −452° F, respectively.

2-33. Latent Heat of Vaporization. Any heat taken in by a liquid after the liquid reaches the saturation temperature is utilized to increase the degree of molecular separation (increases the internal potential energy) and the substance changes from the liquid to the vapor phase.* There is no increase in molecular velocity and, therefore, no change in the internal kinetic energy during the change in phase. Hence, the temperature remains constant during the phase change and the vapor which results is at the vaporizing temperature.

As the material changes state from a liquid to a vapor, the molecules of the material acquire sufficient energy to overcome all restraining forces, including the force of gravity. The amount of energy required to do the internal work necessary to overcome these restraining forces is very great. For this reason, the capacity of a material to absorb heat while undergoing a change from the liquid to the vapor phase is enormous, many times greater even than its capacity to absorb heat in changing from the solid to the liquid phase.

The quantity of heat which 1 lb of a liquid absorbs while changing into the vapor state is known as the latent heat of vaporization. The latent heat of vaporization, like the saturation temperature, is different for each material. It will be shown later that both the latent heat value and the saturation temperature of any particular liquid vary with the pressure over the liquid. When the pressure increases, the saturation temperature increases and the latent heat value decreases.

The quantity of heat required to vaporize any given weight of liquid at the saturation temperature is calculated by the following equation:

$$Q_L = M \times h_{fg} \qquad (2\text{-}10)$$

where Q_L = the quantity of heat in Btu
M = the mass or weight in pounds
h_{fg} = the latent heat of vaporization in Btu/lb

Example 2-13. If the latent heat of vaporization of water is 970 Btu per pound, how

* Some of the energy added to the material leaves the material as external work and has no effect on the internal energy of the material. When the pressure is constant, the amount of external work done is proportional to the change in volume. External work is discussed in detail later.

much heat is required to vaporize 3 gal of water at the saturation temperature of 212° F?

Solution. Total weight of water M	3 gal × 8.33 lb/gal = 25 lb
Applying Equation 2-10, Q_L	25 lb × 970 Btu/lb = 24,250 Btu

Example 2-14. One gallon of water at 200° F in an open container absorbs 1200 Btu. How much water is vaporized?

Solution. Since the saturation temperature of water at atmospheric pressure is 212° F, the entire mass of the water must be raised to this temperature before any water will vaporize

Weight of 1 gal of water	= 8.33 lb
Applying Equation 2-8, the heat required to raise the temperature of the water from 200° F to 212° F, Q_s	8.33 lb × 1 × 12° = 100 Btu
Heat available to vaporize some portion of the water	1200 − 100 = 1100 Btu
Rearranging and applying Equation 2-10, the weight of water vaporized, M	$\frac{1100}{970}$ = 1.135 lb or 0.136 gal

Example 2-15. If 5000 Btu are removed from 8 lb of saturated steam at atmospheric pressure, how much of the steam will condense into water?

Solution. By rearranging and applying Equation 2-10,

$$M = \frac{5000 \text{ Btu}}{970 \text{ Btu/lb}} = 5.15 \text{ lb}$$

2-34. Superheat—the Sensible Heat of a Vapor. Once a liquid has been vaporized, the temperature of the resulting vapor can be further increased by the addition of heat. The heat added to a vapor after vaporization is the sensible heat of the vapor, more commonly called superheat. When the temperature of a vapor has been so increased above the saturation temperature, the vapor is said to be superheated and is called a superheated vapor. Superheated vapors are discussed at length in another chapter.

2-35. Total Heat. The total heat of a material at any particular condition is the sum total of all the sensible and latent heat required to bring it to that condition from an initial condition of Absolute Zero.*

Example 2-16. Compute the total heat content of 1 lb of steam at 212° F.

Solution. The total heat of 1 lb of saturated steam is the sum of the following heat quantities:

(*a*) To raise the temperature of 1 lb of ice from −460° F to 32° F, applying Equation 2-8,

$$Q_s = 1 \times 0.5 \times [32 - (-460)] = 1 \times 0.5 \times 492 = 246 \text{ Btu}$$

(*b*) To melt 1 lb of ice at 32° F into water at 32° F, applying Equation 2-9,

$$Q_L = 1 \times 144 = 144 \text{ Btu}$$

(*c*) To increase temperature of water from 32° F to 212° F, applying Equation 2-8,

$$Q_s = 1 \times 1 \times (212 - 32) = 1 \times 1 \times 180 = 180 \text{ Btu}$$

(*d*) To vaporize 1 lb of water, applying Equation 2-10,

$$Q_L = 1 \times 970 = 970 \text{ Btu}$$

(*e*) Summation:

Sensible heat of the solid	= 246 Btu
Latent heat of fusion	= 144 Btu
Sensible heat of the liquid	= 180 Btu
Latent heat of vaporization	= 970 Btu
Total heat of 1 lb of steam	= 1540 Btu

Through the use of a temperature-heat diagram, the solution to Example 2-15 is shown graphically in Fig. 2-4.

2-36. Mechanical Energy Equivalent. Normally the external energy of a body is expressed in mechanical energy units (work), whereas the internal energy of a body is expressed in heat energy units.

The fact that internal energy is usually expressed in heat energy units gives rise to the definition of heat as molecular or internal energy. As previously stated, from a thermodynamic

* The total heat of a material is commonly known as "enthalpy," and is computed from some arbitrarily selected zero point rather than from Absolute Zero. See Section 4-18.

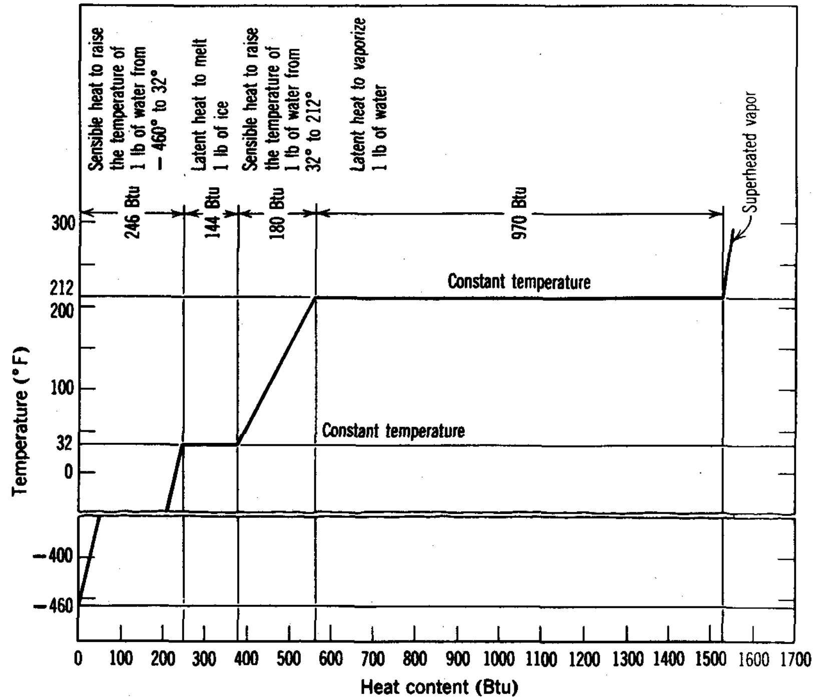


Fig. 2-4. Graphical analysis of the relationship of heat content to the temperature and state of a material.

point of view energy is heat energy only when it is in transition from one body to another because of a difference in temperature between the two bodies. Once the energy flows into a body it becomes "stored" thermal energy. Hence, thermodynamically speaking, internal energy is not heat but thermal energy in storage.

Not all the heat energy flowing into a body is stored in the body as internal energy. In many instances, some or all of the energy flowing into the body passes through or leaves the body as work (mechanical energy). This is made clear in another section.

Furthermore, up to this point it has been assumed that the internal energy of a body is increased only by the addition of heat energy directly, as from a flame or some other heat source. However, this is not the case. The internal or molecular energy of a body may also be increased when work is done on the body. That is, the mechanical energy of the work done on a body may be converted to the internal energy of the body. For example, the head of a nail struck by a hammer will become warm as a part of the mechanical energy of the hammer blow is converted to the internal kinetic energy of the nail head. As the molecules of the metal that make up the nail head are jarred and agitated by the blow of the hammer, their motion or velocity is increased and the temperature of the nail head increases. If a wire is bent rapidly back and forth, the bent portion of the wire becomes hot because of the agitation of the molecules. Also, everyone is familiar with the increase in temperature which is brought about by the friction of two surfaces rubbing together.

Often the external energy of a body is converted to internal energy and vice versa. For example, a bullet speeding toward a target has kinetic energy because of its mass and velocity. At the time of impact with the target, the bullet loses its velocity and a part of its kinetic energy is imparted to the molecules of both the bullet and the target so that the internal energy of each is increased.

Since heat energy is often converted into mechanical energy (work) and vice versa, and since it is often desirable to express both the internal and external energies of a body in terms of the same energy unit, a factor which can be used to convert from one energy unit to the other is useful.

It has been determined by experiment that one Btu of heat energy is equivalent to 778 ft-lb of mechanical energy, that is, one Btu is the amount of heat energy required to do 778 ft-lb of work. This quantity is known as the mechanical energy equivalent and is usually represented in equations by the symbol J.

To convert energy in Btu into energy in foot-pounds, the energy in Btu is multiplied by 778 and, to convert energy in foot-pounds to energy in Btu, the energy in foot-pounds is divided by 778. Expressed as equations, these relationships become

$$Q = \frac{W}{J} \qquad (2\text{-}11)$$

and

$$W = Q \times J \qquad (2\text{-}12)$$

where Q = the quantity of heat energy in Btu
W = mechanical energy or work in foot-pounds
J = the mechanical energy equivalent of heat

Example 2-17. Convert 36,000 ft-lb of mechanical energy into heat energy units.

Solution. Applying Equation 2-11,

$$Q = \frac{36{,}000}{778} = 46.3 \text{ Btu}$$

Example 2-18. Express 12 Btu of heat energy as work in mechanical energy units.

Solution. Applying Equation 2-12,

$$W = 12 \times 778 = 9336 \text{ ft-lb}$$

PROBLEMS

1. A Fahrenheit thermometer reads 85°. What is the temperature in degrees Centigrade? *Ans.* 29.44° C

2. Convert 90° Centigrade to degrees Fahrenheit. *Ans.* 194° F

3. The temperature of a gas is 40° F. What is its temperature on the Rankine scale? *Ans.* 500° R

4. The temperature of the suction vapor entering a refrigeration compressor is −20° F. What is the temperature of the vapor in degrees Rankine? *Ans.* 440° R

5. Thirty gallons of water are heated from 75° F to 180° F. Determine the quantity of heat required? *Ans.* 26,240 Btu

6. In a certain industrial process, 5000 gal of water are cooled from 90° F to 55° F each hour. Determine the quantity of heat which must be removed each hour to produce the required cooling. *Ans.* 1,457,750 Btu

7. Calculate the quantity of heat which must be removed from 60 gal of water in order to cool the water from 42° F and freeze it into ice at 32° F. *Ans.* 77,000 Btu

8. If 12,120 Btu are added to 3 gal of water at 200° F, what fraction of the water in pounds will be vaporized? *Ans.* 9.4 lb or 1.13 gal

9. Twenty-five pounds of ice are placed in 10 gal of water and allowed to melt. Assuming that there is no loss of heat to the surroundings, if the initial temperature of the water is 80° F, to what temperature will the water be cooled by the melting of the ice? *Ans.* 35.7°

10. A gas expanding in a cylinder does 25,000 ft-lb of work on the piston. Determine the quantity of heat required to do the work. *Ans.* 32.13 Btu

3

Thermodynamic Processes

3-1. The Effects of Heat on Volume. When either the velocity of the molecules or the degree of molecular separation is increased by the addition of heat, the mean distance between the molecules is increased and the material expands so that a unit weight of the material occupies a greater volume. This effect is in strict accordance with the theory of increased or decreased molecular activity as described earlier. Hence, when heat is added to or removed from an unconfined material in any of the three physical states, it will expand or contract, respectively. That is, its volume will increase or decrease with the addition or removal of heat.

One of the few exceptions to this rule is water. If water is cooled, its volume will decrease normally until the temperature of the water drops to 39.2° F. At this point, water attains its maximum density and, if further cooled, its volume will again increase. Furthermore, after being cooled to 32° F, it will solidify and the solidification will be accompanied by still further expansion. In fact, 1 cu ft of water will freeze into approximately 1.085 cu ft of ice. This accounts for the tremendous expansive force created during solidification which is sufficient to burst steel pipes or other restraining vessels.

The peculiar behavior of water as it solidifies appears to contradict the general laws governing molecular activity as described previously. However, this is not the case. The unusual behavior of water is explained by the hypothesis that, although the molecules of water are actually closer together in the solid state than they are in the liquid state, they are grouped together to form crystals. It is the relatively large spaces between the crystals of the solid, rather than any increase in the mean distance between the molecules, which accounts for the unusual increase in volume during solidification. This is true also for crystalline solids other than ice.

3-2. Expansion of Solids and Liquids. When a solid or a liquid is heated so that its temperature is increased, it will expand a given amount for each degree of temperature rise. As stated earlier, many temperature measuring devices are based upon this principle. The amount of expansion which a material experiences with each degree of temperature rise is known as its coefficient of expansion. The coefficient of expansion is different for every material, and moreover it will vary for any particular material depending upon the temperature range in which the change occurs.

Since solids and liquids are not readily compressible, if a solid or a liquid is restrained or confined so that its volume is not allowed to change normally with a change in temperature, tremendous pressures are created within the material itself and upon the restraining bodies, which is likely to cause buckling or rupturing of either the material, the restraining bodies, or both. To provide for the normal expansion and contraction occurring with temperature changes, expansion joints are built into highways, bridges, pipelines, etc. Likewise, liquid containers are never completely filled. Space must be allowed for the normal expansion. Otherwise the tremendous expansive forces generated by a temperature increase will cause the containing vessel to rupture, sometimes with explosive force.

3-3. Specific Volume. The specific volume of a material is the volume occupied by a 1 lb mass of the material. Each material has a different specific volume and, because of the change in volume which accompanies a change in temperature, the specific volume of every material varies somewhat with the temperature range. For instance, at 40° F, 1 lb of water has a specific volume of 0.01602 cu ft, whereas the volume occupied by 1 lb of water at 80° F is 0.01608 cu ft.

3-4. Density. The density of a material is the weight in pounds of 1 cu ft of the material. Density is the reciprocal of specific volume, that is, the specific volume divided into one. The density of any material, like specific volume, varies with the temperature, but in the opposite direction. For example, at 40° F, the density of water is 62.434 lb per cubic foot (1/0.01602), whereas water at 80° F has a density of 62.20 lb per cubic foot (1/0.01608). Since density and specific volume are reciprocals of each other, as one increases the other decreases. The density and/or the specific volume of many common materials can be found in various tables.

The relationship between density and specific volume is given by the following equations:

$$\rho = \frac{1}{v} \tag{3-1}$$

$$v = \frac{1}{\rho} \tag{3-2}$$

$$V = M \times v \tag{3-3}$$

$$M = V \times \rho \tag{3-4}$$

where v = the specific volume in cubic feet per pound (cu ft/lb)
ρ = the density in pounds per cubic foot (lb/cu ft)
V = the total volume in cubic feet
M = the total weight in pounds

Example 3-1. If the specific volume of dry saturated steam at 212° F is 26.80 cu ft per pound, what is the density of the steam?

Solution. Applying Equation 3-1, the density $\rho = \frac{1}{26.80}$
$= 0.0373$ lb/cu ft

Example 3-2. The basin of a cooling tower, measuring 5 ft × 4 ft × 1 ft, is filled with water. If the density of the water is 26.8 lb per cubic foot, what is the total weight of the water in the basin?

Solution. The total volume V 5 ft × 4 ft × 1 ft
= 20 cu ft

Applying Equation 3-4, the total weight of water M 20 × 26.8
= 536 lb

3-5. Pressure-Temperature-Volume Relationships of Gases. Because of its loose molecular structure, the change in the volume of a gas as the gas is heated or cooled is much greater than that which occurs in the case of a solid or a liquid. In the following sections, it will be shown that a gas may change its condition in a number of different ways and that certain laws have been formulated which govern the relationship between the pressure, temperature, and volume of the gas during these changes. It should be noted at the outset that in applying the fundamental gas laws it is always necessary to use absolute pressures and absolute temperatures in degrees Rankine. Further, in studying the following sections, it should be remembered that a gas always completely fills any container.

The relationship between the pressure, temperature, and volume of a gas is more easily understood when considered through a series of processes in which the gas passes from some initial condition to some final condition in such a way that only two of these properties vary during any one process, whereas the third property remains unchanged or constant.

3-6. Temperature-Volume Relationship at a Constant Pressure. If a gas is heated under such conditions that its pressure is kept constant, its volume will increase 1/492 of its volume at 32° F for each 1° F increase in its temperature. Likewise, if a gas is cooled at a constant pressure, its volume will decrease 1/492 of its volume at 32° F for each 1° F decrease in its temperature.

In order to better visualize a constant pressure change in condition, assume that a gas is confined in a cylinder equipped with a perfectly fitting, frictionless piston (Fig. 3-1*a*). The pressure of the gas is that which is exerted on the gas by the weight of the piston and by the weight of the atmosphere on top of the piston. Since the piston is free to move up or down in the cylinder, the gas is allowed to expand or contract, that is, to change its volume in such a way that the pressure of the gas remains constant. As the gas is heated, its temperature and volume increase and the piston moves upward in the cylinder. As the gas is cooled, its temperature and volume decrease and the piston moves downward in the cylinder. In either case, the pressure

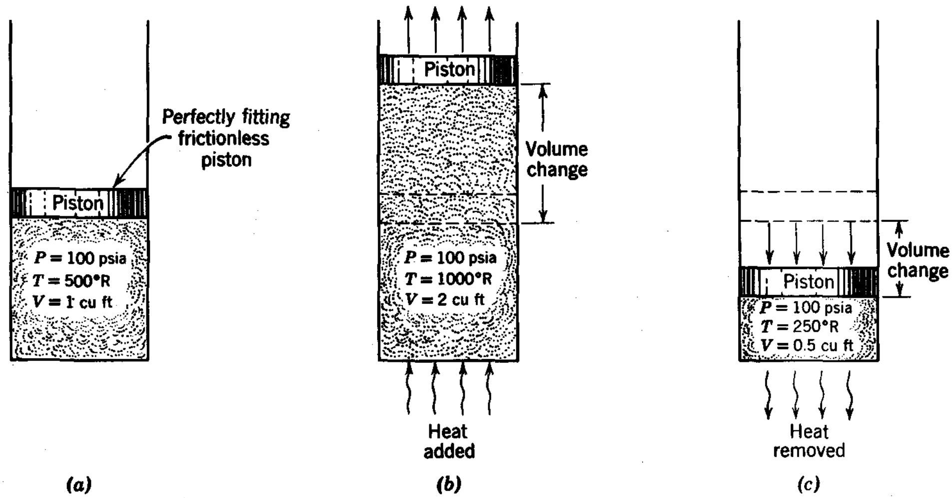


Fig. 3-1. Constant pressure process. (*a*) Gas confined in a cylinder with a perfectly fitting, frictionless piston. (*b*) As gas is heated, both the temperature and the volume of the gas increase. The increase in volume is exactly proportional to the increase in absolute temperature. (*c*) As gas is cooled, both the temperature and the volume of the gas decrease. The decrease in volume is exactly proportional to the decrease in absolute temperature.

of the gas remains the same or unchanged during the heating or cooling processes.

3-7. Charles' Law for a Constant Pressure Process. Charles' law for a constant pressure process states in effect that, when the pressure of the gas remains constant, the volume of the gas varies directly with its absolute temperature. Thus, if the absolute temperature of a gas is doubled while its pressure is kept constant, its volume will also be doubled. Likewise, if the absolute temperature of a gas is reduced by one-half while the pressure is kept constant, its volume will also be reduced by one-half. This relationship is illustrated in Figs. 3-1*b* and 3-1*c*.

Charles' law for a constant pressure process written as an equation is as follows: When the pressure is kept constant,

$$T_1V_2 = T_2V_1 \qquad (3\text{-}5)$$

where T_1 = the initial temperature of the gas in degrees Rankine
T_2 = the final temperature of the gas in degrees Rankine
V_1 = the initial volume of the gas in cubic feet
V_2 = the final volume of the gas in cubic feet

When any three of the preceding values are known, the fourth may be calculated by applying Equation 3-5.

Example 3-3. A gas, whose initial temperature is 520° R and whose initial volume is 5 cu ft, is allowed to expand at a constant pressure until its volume is 10 cu ft. Determine the final temperature of the gas in degrees Rankine.

Solution. By rearranging and applying Equation 3-5, the final temperature of the gas T_2

$$= \frac{T_1V_2}{V_1} = \frac{520 \times 10}{5} = 1040° \text{ R}$$

Example 3-4. A gas, having an initial temperature of 80° F, is cooled at a constant pressure until its temperature is 40° F. If the initial volume of the gas is 8 cu ft, what is its final volume?

Solution. Since the temperatures are given in degrees Fahrenheit, they must be converted to degrees Rankine before being substituted in Equation 3-5.

By rearranging and applying Equation 3-5, the final volume V_2

$$= \frac{T_2V_1}{T_1} = \frac{500 \times 8}{540} = 7.4074 \text{ cu ft}$$

3-8. Pressure-Volume Relationship at a Constant Temperature. When the volume of a gas is increased or decreased under such conditions that the temperature of the gas does not change, the absolute pressure will vary inversely with the volume. Thus, when a gas is compressed (volume decreased) while its temperature remains unchanged, its absolute pressure will increase in proportion to the decrease in volume. Similarly, when a gas is expanded at a constant temperature, its absolute pressure will decrease in proportion to the increase in volume. This is a statement of Boyle's law for a constant temperature process and is illustrated in Figs. 3-2*a*, 3-2*b*, and 3-2*c*.

It has been previously stated that the molecules of a gas are flying about at random and at high velocities and that the molecules of the gas frequently collide with one another and with the walls of the container. The pressure exerted by the gas is a manifestation of these molecular collisions. Billions and billions of gas molecules, traveling at high velocities, strike the walls of the container during each fraction of a second. It is this incessant molecular bombardment which produces the pressure that a gas exerts upon the walls of its container. The magnitude of the pressure exerted depends upon the force and frequency of the molecular impacts upon a given area. The greater the force and frequency of the impacts, the greater is the pressure. The number of molecules confined in a given space and their velocity will, of course, determine the force and the frequency of the impacts. That is, the greater the number of molecules (the greater the quantity of gas) and the higher the velocity of the molecules (the higher the temperature of the gas), the greater is the pressure. The force with which the molecules strike the container walls depends only upon the velocity of the molecules. The higher the velocity the greater is the force of impact. The greater the number of molecules in a given space and the higher the velocity the more often the molecules will strike the walls.

When a gas is compressed at a constant temperature, the velocity of the molecules remains unchanged. The increase in pressure which occurs is accounted for by the fact that the volume of the gas is diminished and a given number of gas molecules are confined in a smaller space so that the frequency of impact is greater. The reverse of this holds true, of course, when the gas is expanded at a constant temperature.

Any thermodynamic process which occurs in such a way that the temperature of the working substance does not change during the process is called an isothermal (constant temperature) process.

Boyle's law for a constant temperature process

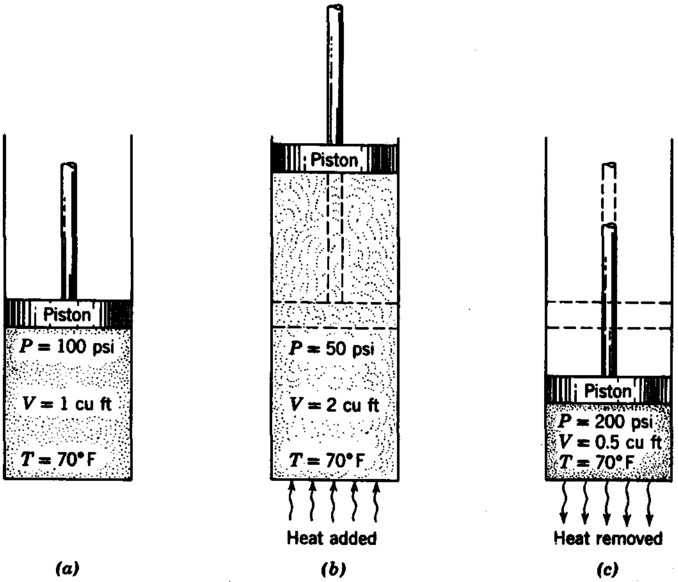


Fig. 3-2. Constant temperature process. (*a*) Initial condition. (*b*) Constant temperature expansion—volume change is inversely proportional to the change in absolute pressure. Heat must be added during expansion to keep temperature constant. (*c*) Constant temperature compression—volume change is inversely proportional to the change in absolute pressure. Heat must be removed during compression to keep temperature constant.

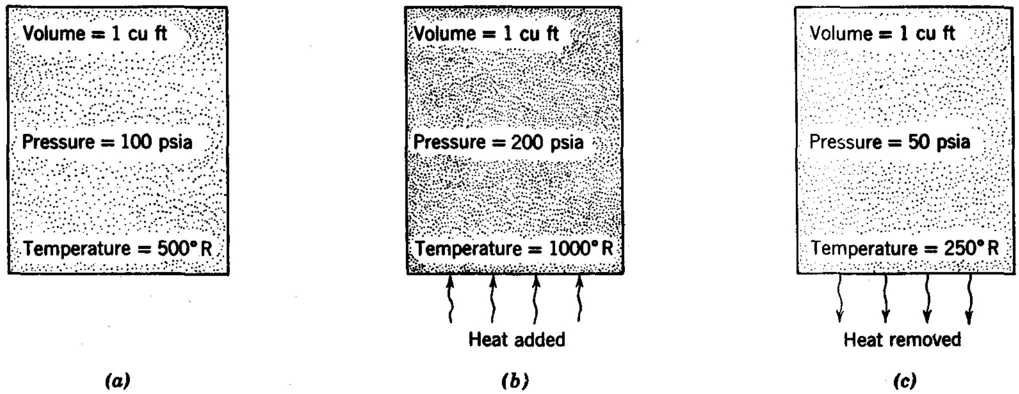


Fig. 3-3. Constant volume process. (*a*) Initial condition. (*b*) The absolute pressure increases in direct proportion to the increase in absolute temperature. (*c*) The absolute pressure decreases in direct proportion to the decrease in absolute temperature.

is represented by the following equation: if the temperature is constant,

$$P_1V_1 = P_2V_2 \tag{3-6}$$

where P_1 = the initial absolute pressure
P_2 = the final absolute pressure
V_1 = the initial volume in cubic feet
V_2 = the final volume in cubic feet

Example 3-5. Five pounds of air are expanded at a constant temperature from an initial volume of 5 cu ft to a final volume of 10 cu ft. If the initial pressure of the air is 20 psia, what is the final pressure in psia?

Solution. By rearranging and applying Equation 3-6, the final pressure P_2

$$= \frac{P_1 \times V_1}{V_2} = \frac{20 \times 5}{10} = 10 \text{ psia}$$

Example 3-6. Four cubic feet of gas are allowed to expand at a constant temperature from an initial pressure of 1500 psfa to a final pressure of 900 psfa. Determine the final volume of the gas.

Solution. By rearranging and applying Equation 3-6, the final volume V_2

$$= \frac{P_1V_1}{P_2} = \frac{1500 \times 4}{900} = 6.67 \text{ cu ft}$$

Example 3-7. A given weight of gas, whose initial volume is 10 cu ft, is compressed isothermally until its volume is 4 cu ft. If the initial pressure of the gas is 3000 psfa, determine the final pressure in psig.

Solution. By rearranging and applying Equation 3-6, the final pressure P_2

$$= \frac{P_1V_1}{V_2} = \frac{3000 \times 10}{4} = 7500 \text{ psfa}$$

Dividing by 144

$$= \frac{7500}{144} = 52.08 \text{ psia}$$

Subtracting the atmospheric pressure

$$= 52.08 - 14.7 = 37.38 \text{ psig}$$

3-9. Pressure-Temperature Relationship at a Constant Volume. Assume that a gas is confined in a closed cylinder so that its volume cannot change as it is heated or cooled (Fig. 3-3*a*). When the temperature of a gas is increased by the addition of heat, the absolute pressure will increase in direct proportion to the increase in absolute temperature (Fig. 3-3*b*). If the gas is cooled, the absolute pressure of the gas will decrease in direct proportion to the decrease in absolute temperature (Fig. 3-3*c*).

Whenever the temperature (velocity of the molecules) of a gas is increased while the volume of the gas (space in which the molecules are confined) remains the same, the magnitude of the pressure (the force and frequency of molecular impacts on the cylinder walls) increases. Likewise, when a gas is cooled at a constant volume, the force and frequency of molecular impingement on the walls of the container diminish and the pressure of the gas will be less than before. The reduction in the force and the frequency of molecular impacts is accounted for by the reduction in molecular velocity.

3-10. Charles' Law for a Constant Volume Process. Charles' law states in effect that when a gas is heated or cooled under such conditions that the volume of the gas remains unchanged or constant, the absolute pressure varies directly with the absolute temperature. Charles' law may be written as the following equation: when the volume is the same,

$$T_1P_2 = T_2P_1 \tag{3-7}$$

where T_1 = the initial temperature in degrees Rankine
T_2 = the final temperature in degrees Rankine
P_1 = the initial pressure in pounds per square inch absolute
P_2 = the final pressure in pounds per square inch absolute

Example 3-8. A certain weight of gas confined in a tank has an initial temperature of 80° F and an initial pressure of 30 psig. If the gas is heated until the final gage pressure is 50 psi, what is the final temperature in degrees Fahrenheit?

Solution. By rearranging and applying Equation 3-7,

$$T_2 = \frac{T_1 \times P_2}{P_1} = \frac{(80 + 460) \times (50 + 14.7)}{(30 + 14.7)} = 782° \text{ R}$$

Converting Rankine to Fahrenheit

$$782 - 460$$
$$t_2 = 322° \text{ F}$$

3-11. The General Gas Law. Combining Charles' and Boyle's laws produces the following equation:

$$\frac{P_1V_1}{T_1} = \frac{P_2V_2}{T_2} \tag{3-8}$$

Equation 3-8 is a statement that for any given weight of a gas, the product of the pressure in psfa and the volume in cubic feet divided by the absolute temperature in degrees Rankine will always be a constant. The constant, of course, will be different for different gases and, for any one gas, will vary with the weight of gas involved. However, if, for any one gas, the weight of 1 lb is used, then V will become the specific volume v, and Equation 3-8 may be written:

$$\frac{Pv}{T} = R \tag{3-9}$$

where R = the gas constant

The gas constant R is different for each gas. The gas constant for most common gases can be found in tables. A few of these are given in Table 3-1.

Multiplying both sides of Equation 3-9 by M produces

$$PMv = MRT$$

but since

$$Mv = V$$

then

$$PV = MRT \tag{3-10}$$

where P = the pressure in psfa
V = the volume in ft^3
M = the mass in pounds
R = the gas constant
T = the temperature in ° R

Equation 3-10 is known as the general gas law and is very useful in the solution of many problems involving gases. Since the value of R for most gases can be found in tables, if any three of the four properties, P, V, M, and T, are known, the fourth property can be determined by Equation 3-10. Notice that the pressure must be in pounds per square foot absolute.

Example 3-9. The tank of an air compressor has a volume of 5 cu ft and is filled with air at a temperature of 100° F. If a gage on the tank reads 151.1 psi, what is the weight of the air in the tank?

Solution. From Table 3-1, R for air $= 53.3$

By rearranging and applying Equation 3-10, the weight of air M

$$= \frac{(151.1 + 14.7) \times 144 \times 5}{53.3 \times (100 + 460)} = \frac{165.8 \times 144 \times 5}{53.3 \times 560} = 4 \text{ lb}$$

Example 3-10. Two pounds of air have a volume of 3 cu ft. If the pressure of the air is 135.3 psig, what is the temperature of the air in degrees Fahrenheit?

Solution. From Table 3-1, R for air $= 53.3$

By rearranging and applying Equation 3-10, the temperature of the air in ° R,

$$T = \frac{PV}{MR}$$

$$= \frac{(135.3 + 14.7) \times 144 \times 3}{2 \times 53.3}$$

$$= \frac{150 \times 144 \times 3}{2 \times 53.3}$$

$$= 607.9° \text{ R}$$

Converting to Fahrenheit,

$$t = 607.9 - 460$$
$$= 147.9° \text{ F}$$

3-12. External Work. Whenever a material undergoes a change in volume, work is done. If the volume of the material increases, work is done *by* the material. If the volume of the material decreases, work is done *on* the material. For example, consider a certain weight of gas confined in a cylinder equipped with a movable piston (Fig. 3-1*a*). As the gas is heated, its temperature increases and it expands, moving the piston upward in the cylinder against the pressure of the atmosphere. Work is done in that the weight of the piston is moved through a distance (Fig. 3-1*b*).* The agency doing the work is the expanding gas.

In order to do work, energy is required (Section 1-12). In Fig. 3-1*b*, the energy required to do the work is supplied to the gas as the gas is heated by an external source. It is possible, however, for a gas to do external work without the addition of energy from an external source. In such cases, the gas does the work at the expense of its own energy. That is, as the gas expands and does work, its internal kinetic energy (temperature) decreases in an amount equal to the amount of energy required to do the work.

When a gas is compressed (its volume decreased), a certain amount of work must be done on the gas in order to compress it. And, an amount of energy equal to the amount of work done will be imparted to the molecules of the gas during the compression. That is, the mechanical energy of the piston motion will be transformed into the internal kinetic energy of the gas (molecular motion) and, unless the gas is cooled during the compression, the temperature of the gas will increase in proportion to the amount of work done. The increase in the temperature of a gas as the gas is compressed is a common phenomenon and may be noted by feeling the valve stem of a tire being filled with a hand pump, or the head of an air compressor, etc.

* Some work is done, also, in overcoming friction and in overcoming the pressure of the atmosphere.

3-13. The General Energy Equation. The law of conservation of energy clearly indicates that the energy transferred to a body must be accounted for in its entirety. It has been shown that some part (or all) of the energy taken in by a material may leave the material as work, and that only that portion of the transferred energy which is not utilized to do external work remains in the body as "stored thermal energy." It is evident then that all of the energy transferred to a body must be accounted for in some one or in some combination of the following three ways: (1) as an increase in the internal kinetic energy, (2) as an increase in the internal potential energy, and (3) as external work done. The general energy equation is a mathematical statement of this concept and may be written:

$$\Delta Q = \Delta K + \Delta P + \Delta W \qquad (3\text{-}11)$$

where ΔQ = the heat energy transferred to the material in Btu
ΔK = that fraction of the transferred energy which increases the internal kinetic energy
ΔP = that fraction of the transferred energy which increases the internal potential energy
ΔW = that fraction of the transferred energy which is utilized to do external work

The Greek letter, Δ (delta), used in front of a term in an equation identifies a change of condition. For instance, where K represents the internal kinetic energy, ΔK represents the change in the internal kinetic energy.

Depending upon the particular process or change in condition that the material undergoes, any of the terms in Equation 3-11 may have any value either positive or negative, or any may be equal to zero. This will be made clear later.

3-14. External Work of a Solid or Liquid. When heat added to a material in either the solid or liquid state increases the temperature of the material, the material expands somewhat and a small amount of work is done. However, the increase in volume and the external work done is so slight that the portion of the transferred energy which is utilized to do external work or to increase the internal potential energy is negligible. For all practical purposes, it can be assumed that all the energy added to a solid or a liquid during a temperature change increases the internal kinetic energy. None leaves the material as work and none is set up as an increase in the internal potential energy. In this instance, both ΔP and ΔW of Equation 3-11 are equal to zero and, therefore, ΔQ is equal to ΔK.

When a solid melts into the liquid phase, the change in volume is again so slight that the external work done may be neglected. Furthermore, since the temperature also remains constant during the phase change, none of the transferred energy increases the internal kinetic energy. All the energy taken in by the melting solid is set up as an increase in the internal potential energy. Therefore, ΔK and ΔW are both equal to zero and ΔQ is equal to ΔP.

This is not true, however, when a liquid changes into the vapor phase. The change in volume that occurs and therefore the external work done as the liquid changes into a vapor is considerable. For example, when 1 lb of water at atmospheric pressure changes into a vapor, its volume increases from 0.01671 cu ft to 26.79 cu ft. Of the 970.4 Btu required to vaporize 1 lb of water, approximately 72 Btu of this energy are required to do the work of expanding against the pressure of the atmosphere. The remainder of the energy is set up in the vapor as an increase in the internal potential energy. In this instance, only ΔK is equal to zero, so that ΔQ is equal to ΔP plus ΔW.

3-15. "Ideal" or "Perfect" Gas. The various laws governing the pressure-volume-temperature relationships of gases as discussed in this chapter apply with absolute accuracy only to a hypothetical "ideal" or "perfect" gas. A "perfect" gas is described as one in such a condition that there is no interaction between the molecules of the gas. The molecules of such a gas are entirely free and independent of each other's attractive forces. Hence, none of the energy transferred either to or from an ideal gas has any effect on the internal potential energy.

The concept of an ideal or perfect gas greatly simplifies the solution of problems concerning the changes in the condition of a gas. Many complex problems in mechanics are made simple by the assumption that no friction exists, the effects of friction being considered separately. The function of an ideal gas is the same as that of the frictionless surface. An ideal gas is assumed to undergo a change of condition without internal friction, that is, without the performance of internal work in the overcoming of internal molecular forces.

The idea of internal friction is not difficult to comprehend. Consider that a liquid such as oil will not flow readily at low temperatures. This is because of the internal friction resulting from strong intermolecular forces within the liquid. However, as the liquid is heated and the molecules gain additional energy, the intermolecular forces are overcome somewhat, internal friction diminishes, and the liquid flows more easily.

Vaporization of the liquid, of course, causes a greater separation of the molecules and brings about a substantial reduction in internal friction, but some interaction between the molecules of the vapor still exists. In the gaseous state, intermolecular forces are greatest when the gas is near the liquid phase and diminish rapidly as the gas is heated and its temperature rises farther and farther above the saturation temperature. A gas approaches the ideal state when it reaches a condition such that the interaction between the molecules and hence, internal friction, is negligible.

Although no such thing as an ideal or perfect gas actually exists, many gases, such as air, nitrogen, hydrogen, helium, etc., so closely approach the ideal condition that any errors which may result from considering them to be ideal are of no consequence for all practical purposes.

Although it is important that the student of refrigeration understand and be able to apply the laws of perfect gases, it should be understood that gases as they normally occur in the mechanical refrigeration cycle are close to the saturation curve, that is, they are vapors, and do not even approximately approach the condition of an ideal or perfect gas.* They follow the gas laws in only a very general way,

* A vapor is sometimes defined as a gas at a condition close enough to the saturation curve so that it does not follow the ideal gas laws even approximately.

and therefore the use of the gas laws to determine the pressure-volume-temperature relationships of such vapors will result in considerable inaccuracy. In working with vapors, it is usually necessary to use values which have been determined experimentally and are tabulated in saturated and superheated vapor tables. These tables are included as a part of this textbook and are discussed later.

3-16. Processes for Ideal Gases. A gas is said to undergo a process when it passes from some initial state or condition to some final state or condition. A change in the condition of a gas may occur in an infinite number of ways, only five of which are of interest. These are the (1) constant pressure (isobaric), (2) constant volume (isometric), (3) constant temperature (isothermal), (4) adiabatic, and (5) polytropic processes.

In describing an ideal gas, it has been said that the molecules of such a gas are so far apart that they have no attraction for one another, and that none of the energy absorbed by an ideal gas has any effect on the internal potential energy. It is evident, then, that heat absorbed by an ideal gas will either increase the internal kinetic energy (temperature) of the gas or it will leave the gas as external work, or both. Since the change in the internal potential energy, ΔP, will always be zero, the general energy equation for an ideal gas may be written:

$$\Delta Q = \Delta K + \Delta W \qquad (3\text{-}12)$$

In order to better understand the energy changes which occur during the various processes, it should be kept in mind that a change in the temperature of the gas indicates a change in the internal kinetic energy of the gas, whereas a change in the volume of the gas indicates work done either by or on the gas.

3-17. Constant Volume Process. When a gas is heated while it is so confined that its volume cannot change, its pressure and temperature will vary according to Charles' law (Fig. 3-3). Since the volume of the gas does not change, no external work is done and ΔW is equal to zero. Therefore, for a constant volume process, indicated by the subscript v,

$$\Delta Q_v = \Delta K_v \qquad (3\text{-}13)$$

Equation 3-13 is a statement that during a constant volume process all of the energy transferred to the gas increases the internal kinetic energy of the gas. None of the energy leaves the gas as work.

When a gas is cooled (heat removed) while its volume remains constant, all the energy removed is effective in reducing the internal kinetic energy of the gas. It should be noted that in Equation 3-12, ΔQ represents heat transferred to the gas, ΔK represents an increase in the internal kinetic energy, and ΔW represents work done by the gas. Therefore, if heat is given up by the gas, ΔQ is negative. Likewise, if the internal kinetic energy of the gas decreases, ΔK is negative and, if work is done on the gas, rather than by it, ΔW is negative. Hence, in Equation 3-13, when the gas is cooled, both ΔQ and ΔK are negative.

3-18. Constant Pressure Process. If the temperature of a gas is increased by the addition of heat while the gas is allowed to expand so that its pressure is kept constant, the volume of the gas will increase in accordance with Charles' law (Fig. 3-1). Since the volume of the gas increases during the process, work is done by the gas at the same time that its internal energy is increased. Hence, while one fraction of the transferred energy increases the store of internal kinetic energy, another fraction of the transferred energy leaves the gas as work. For a constant pressure process, identified by the subscript p, the energy equation may be written

$$\Delta Q_p = \Delta K_p + \Delta W_p \qquad (3\text{-}14)$$

3-19. Specific Heat of Gases. The quantity of heat required to raise the temperature of 1 lb of a gas 1° F while the volume of the gas remains constant is known as the specific heat at a constant volume (C_v). Similarly, the quantity of heat required to raise the temperature of 1 lb of a gas 1° F while the gas expands at a constant pressure is called the specific heat at a constant pressure (C_p). For any particular gas, the specific heat at a constant pressure is always greater than the specific heat at a constant volume. The reason for this is easily explained.

The quantity of energy required to increase the internal kinetic energy of a gas to the extent that the temperature of the gas is increased 1° F is exactly the same for all processes. Since, during a constant volume process, no work is done, the only energy required is that which increases the internal kinetic energy. However, during a constant pressure process, the gas expands a fixed amount for each degree of temperature rise and a certain amount of external work is done. Therefore, during a constant pressure process, energy to do the work that is done must be supplied in addition to that which increases the internal kinetic energy. For example, the specific heat of air at a constant volume is 0.169 Btu per pound, whereas the specific heat of air at a constant pressure is 0.2375 Btu per pound. For either process, the increase in the internal energy of the air per degree of temperature rise is 0.169 Btu per pound. For the constant pressure process, the additional 0.0685 Btu per pound (0.2375 − 0.169) is the energy required to do the work resulting from the volume increase accompanying the temperature rise.

The specific heat of a gas may take any value either positive or negative, depending upon the amount of work that the gas does as it expands.

3-20. The Change in Internal Kinetic Energy. During any process in which the temperature of the gas changes, there will be a change in the internal kinetic energy of the gas. Regardless of the process, when the temperature of a given weight of gas is increased or decreased, the change in the internal kinetic energy can be determined by the equation

$$\Delta K = MC_v(t_2 - t_1) \tag{3-15}$$

where ΔK = the increase in the internal kinetic energy in Btu
M = the weight in pounds
C_v = constant volume specific heat
t_2 = the final temperature
t_1 = the initial temperature

NOTE. The temperature may be in either Fahrenheit or Rankine, since the difference in temperature will be the same in either case as long as the units are consistent.

Example 3-11. The temperature of 5 lb of air is increased by the addition of heat from an initial temperature of 75° F to a final temperature of 140° F. If C_v for air is 0.169 Btu, what is the increase in the internal energy?

Solution. Applying Equation 3-15, the increase in internal kinetic energy ΔK

$$\begin{aligned} &= 5 \times 0.169 \times (140 - 75) \\ &= 5 \times 0.169 \times 65 \\ &= 54.9 \text{ Btu} \end{aligned}$$

Example 3-12. Twelve pounds of air are cooled from an initial temperature of 95° F to a final temperature of 72° F. Compute the increase in the internal kinetic energy.

Solution. Applying Equation 3-15, the increase in internal kinetic energy ΔK

$$\begin{aligned} &= 12 \times 0.169 \times (72 - 95) \\ &= 12 \times 0.169 \times (-23) \\ &= -46.64 \text{ Btu} \end{aligned}$$

In Example 3-12, ΔK is negative, indicating that the gas is cooled and that the internal kinetic energy is decreased rather than increased.

3-21. Heat Transferred during a Constant Volume Process. For a constant volume process, since

$$\Delta Q_v = \Delta K_v$$

then

$$\Delta Q_v = MC_v(t_2 - t_1) \tag{3-16}$$

Example 3-13. If, in Example 3-11, the gas is heated while its volume is kept constant, what is the quantity of heat transferred to the gas during the process?

Solution. Applying Equation 3-16, the heat transferred to the gas,

$$\begin{aligned} \Delta Q_v &= 5 \times 0.169 \times (140 - 75) \\ &= 5 \times 0.169 \times 65 \\ &= 54.9 \text{ Btu} \end{aligned}$$

Alternate Solution. From Example 3-12, $\Delta K_v = 54.9$ Btu

Since $\Delta Q_v = \Delta K_v$, $\Delta Q_v = 54.9$ Btu

Example 3-14. If, in Example 3-12, the air is cooled while its volume remains constant, what is the quantity of heat transferred to the air during the process?

Solution. From Example 3-12, $\Delta K_v = -46.67$ Btu

Since $\Delta Q_v = \Delta K_v$, $\Delta Q_v = -46.67$ Btu

In Example 3-14, notice that since ΔK_v is negative, indicating a decrease in the internal kinetic energy, ΔQ_v must of necessity also be negative, indicating that heat is transferred from the gas rather than to it.

3-22. External Work during a Constant Pressure Process. It will now be shown that the work done during a constant pressure process may be evaluated by the equation:

$$W = P(V_2 - V_1) \tag{3-17}$$

where W = the work done in foot-pounds
P = the pressure in psfa
V_2 = the final volume in cubic feet
V_1 = the initial volume in cubic feet

Assume that the piston in Fig. 3-1*c* has an area of A square feet and that the pressure of the gas in the cylinder is P pounds per square foot. Then, the total force exerted on the top of the piston will be PA pounds, or

$$F = P \times A$$

Assume now that the gas in the cylinder, having an initial volume V_1, is heated and allowed to expand to volume V_2 while its pressure is kept constant. In doing so, the force PA acts through the distance 1 and work is done. Hence,

$$W = P \times A \times 1$$

but since

$$A \times 1 = (V_2 - V_1)$$

then

$$W = P(V_2 - V_1)$$

Example 3-15. One pound of air having an initial volume of 13.34 cu ft and an initial temperature of 70° F is heated and allowed to expand at a constant pressure of 2117 psfa to a final volume of 15 cu ft. Determine the amount of external work in foot-pounds.

Solution. Applying Equation 3-17, the work in foot-pounds W $= 2117 \times (15 - 13.34)$
$= 2117 \times 1.66$
$= 3514$ ft-lb

In Equation 3-12, ΔW is always given in heat energy units. By application of the mechanical energy equivalent (Section 3-16), W in foot-pounds may be expressed as ΔW in Btu. The relationship is

$$\Delta W = \frac{W}{J} \tag{3-18}$$

$$W = \Delta W \times J \tag{3-19}$$

Example 3-16. Express the work done in Example 3-15 in terms of heat energy units.

Solution. Applying Equation 3-18, the work in Btu W $= \dfrac{3514}{778}$
$= 4.52$ Btu

3-23. Heat Transferred during a Constant Pressure Process. According to Equation 3-14, ΔQ_p, the total heat transferred to a gas during a constant pressure process is equal to the sum of ΔK_p, the increase in internal kinetic energy, and ΔW_p, the heat energy equivalent of the work of expansion.

Example 3-17. Compute the total heat energy transferred to the air during the constant pressure process described in Example 3-15.

Solution. Converting 70° F to degrees Rankine, $°R = 70 + 460$
$= 530°$ R

Applying Charles' law, Equation 3-5, to determine the final temperature, $T_2 = \dfrac{T_1 \times V_2}{V_1}$
$= \dfrac{530 \times 15}{13.34}$
$= 596°$ R

Applying Equation 3-15, the increase in internal kinetic energy, $\Delta K = 1 \times 0.169 \times (596 - 530)$
$= 1 \times 0.169 \times 66$
$= 11.154$ Btu

From Example 3-15 and 3-16, $\Delta W_p = 4.52$ Btu

Applying Equation 3-14, $\Delta Q_p = 11.15 + 4.52$
$= 15.67$ Btu

Since the specific heat at a constant pressure C_p takes into account not only the increase in internal energy per pound but also the work done per pound per degree of temperature rise during a constant pressure expansion, for the constant pressure process only, ΔQ_p may be determined by the following equation:

$$\Delta Q_p = MC_p(t_2 - t_1) \tag{3-20}$$

Hence, an alternate solution to Example 3-17 is

Applying Equation 3-20, $\Delta Q_p = 1 \times 0.2375 \times (596 - 530)$
$= 1 \times 0.2375 \times 66$
$= 15.67$ Btu

3-24. Pressure-Volume (PV) Diagram. Equation 3-8 is a statement that the thermodynamic state of a gas is adequately described by any two properties of the gas. Hence, using any two properties of the gas as mathematical coordinates, the thermodynamic state of a gas at any given instant may be shown as a point on a chart. Furthermore, when the conditions under which a gas passes from some initial state to some final state are known, the path that the process follows may be made to appear as a line on the chart.

The graphical representation of a process or cycle is called a process diagram or a cycle diagram, respectively, and is a very useful tool in the analysis and solution of cyclic problems.

Since work is a function of pressure and volume, when it is the work of a process or cycle which is of interest, the properties used as coordinates are usually the pressure and the volume. When the pressure and volume are used as coordinates to diagram a process or cycle, it is called a pressure-volume (PV) diagram.

To illustrate the use of the PV diagram, a pressure-volume diagram of the process described in Example 3-15 is shown in Fig. 3-4. Notice that the pressure in psfa is used as the vertical coordinate, whereas the volume in cubic feet is used as the horizontal coordinate.

In Example 3-15, the initial condition of the gas is such that the pressure is 2117 psfa and the volume is 13.34 cu ft. To establish the initial state of the gas on the PV chart, start at the origin and proceed upward along the vertical pressure axis to the given pressure, 2117 psfa. Draw a dotted line parallel to the base line through this point and across the chart. Next, from the point of origin proceed to the right along the horizontal volume axis to the given volume, 13.34 cu ft. Through this point draw a vertical dotted line across the chart. The intersection of the dotted lines at point 1 establishes the initial thermodynamic state of the gas.

According to Example 3-15, the gas is heated and allowed to expand at a constant pressure until its volume is 15 cu ft. Since the pressure remains the same during the process, the state point representing the final state of the gas must fall somewhere along the line of constant pressure already established. The exact point on the pressure line which represents the final state 2 is determined by the intersection of the line drawn through the point on the volume axis that identifies the final volume.

In passing from the initial state 1 to the final state 2 the air passes through a number of intermediate thermodynamic states, all of which can be represented by points which will fall along line 1 to 2. Line 1 to 2, then, represents the path that the process will follow as the thermodynamic state of the gas changes from 1 to 2, and is the PV diagram of the process described.

The area of a rectangle is the product of its

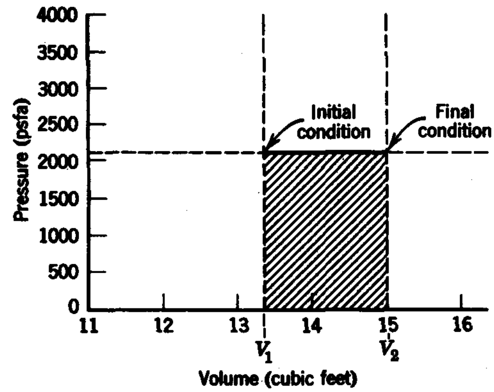


Fig. 3-4. Pressure-volume diagram of constant pressure process. Crosshatched area between process diagram and base line represents external work done during the process.

two dimensions. In Fig. 3-4, the area of the rectangle, 1–2–V_2–V_1 (crosshatched), is the product of its altitude P and its base ($V_2 - V_1$). But according to Equation 3-17, the product $P(V_2 - V_1)$ is the external work done during a constant pressure process. It is evident then that the area between the process diagram and the volume axis is a measure of the external work done during the process in foot-pounds. This area is frequently referred to as "the area under the curve."

Figure 3-5 is a PV diagram of a constant volume process. Assume that the initial condition of the gas at the start of the process is such that the pressure is 2000 psfa and the volume is 4 cu ft. The gas is heated while its volume is kept constant until the pressure increases to 4000 psfa. The process takes place along the constant volume line from the initial condition 1 to the final condition 2.

It has been stated that no work is done during a process unless the volume of the gas changes. Examination of the PV diagram in Fig. 3-5 will show that no work is indicated for the constant volume process. Since a line has only the dimension of length, there is no area between the process diagram and the base or volume axis. Hence, no work is done.

3-25. Constant Temperature Process. According to Boyle's law, when a gas is compressed or expanded at a constant temperature, the pressure will vary inversely with the volume. That is, the pressure increases as the gas is

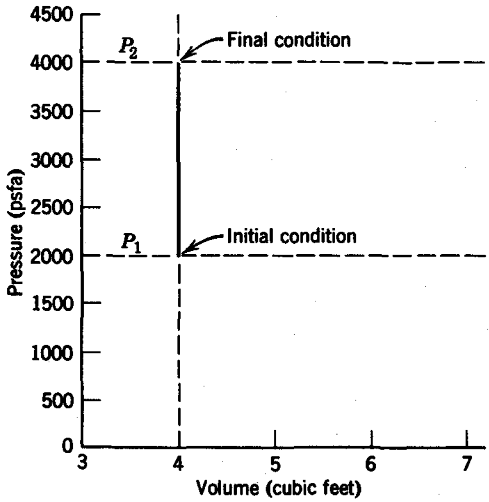


Fig. 3-5. Pressure-volume diagram of constant volume process. Since there is no area between the process diagram and the volume axis, there is no work done during a constant volume process.

compressed and decreases as the gas is expanded. Since the gas will do work as it expands, if the temperature is to remain constant, energy with which to do the work must be absorbed from an external source (Fig. 3-2*b*). However, since the temperature of the gas remains constant, all of the energy absorbed by the gas during the process leaves the gas as work; none is stored in the gas as an increase in the internal energy.

When a gas is compressed, work is done on the gas, and if the gas is not cooled during the compression, the internal energy of the gas will be increased by an amount equal to the work of compression. Therefore, if the temperature of the gas is to remain constant during the compression, the gas must reject to some external body an amount of heat equal to the amount of work done upon it during the compression (Fig. 3-2*c*).

There is no change in the internal kinetic energy during a constant temperature process. Therefore, in Equation 3-12, ΔK is equal to zero and the general energy equation for a constant temperature process may be written

$$\Delta Q_t = \Delta W_t \qquad (3\text{-}21)$$

3-26. Work of an Isothermal Process. A PV diagram of an isothermal expansion is shown in Fig. 3-6. In a constant temperature process the pressure and volume both change in accordance with Boyle's law. The path followed by an isothermal expansion is indicated by line 1 to 2 and the work of the process in foot-pounds is represented by the area 1–2–V_2–V_1. The area, 1–2–V_2–V_1, and therefore the work of the process, may be calculated by the equation

$$W = P_1V_1 \times \ln \frac{V_2}{V_1} \qquad (3\text{-}22)$$

where ln = natural logarithm (log to the base e)

Example 3-18. A certain weight of gas having an initial pressure of 2500 psfa and an initial volume of 2 cu ft is expanded isothermally to a volume of 4 cu ft. Determine:
(*a*) the final pressure of the gas in psfa
(*b*) the work done in heat energy units.

Solution

(*a*) By applying Boyle's law, Equation 3-6, the final pressure P_2
$$= \frac{P_1V_1}{V_2} = \frac{2500 \times 2}{4} = 1250 \text{ psfa}$$

(*b*) By applying Equation 3-22, the external work of the process in foot-pounds W
$$= 2500 \times 2 \times \ln \tfrac{4}{2} = 2500 \times 2 \times \ln 2 = 2500 \times 2 \times 0.693 = 3465 \text{ ft-lb}$$

By Applying Equation 3-18, the work in heat energy units ΔW
$$= \frac{3465}{778} = 4.45 \text{ Btu}$$

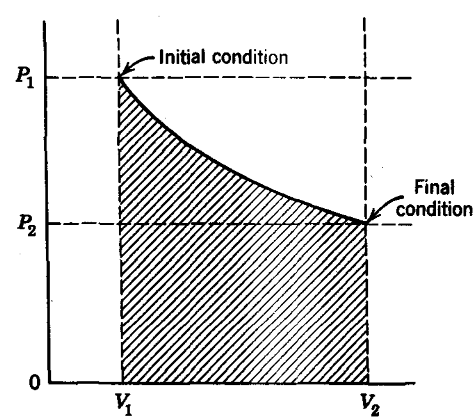


Fig. 3-6. Pressure-volume diagram of constant temperature process. Crosshatched area represents the work of the process.

Example 3-19. A certain weight of gas having an initial pressure of 1250 psfa and an initial volume of 4 cu ft is compressed isothermally to a volume of 2 cu ft. Determine:

(*a*) the final pressure in psfa

(*b*) the work done by the gas in Btu.

Solution

(*a*) By applying Boyle's law, Equation 3-6, the final pressure P_2

$$= \frac{P_1V_1}{V_2} = \frac{1250 \times 4}{2} = 2500 \text{ psfa}$$

(*b*) By Applying Equation 3-22, the external work of the process in foot-pounds W

$$= 2500 \times 2 \times \ln \tfrac{2}{4}$$
$$= 2500 \times 2 \times \ln 0.5$$
$$= 2500 \times 2 \times -0.693$$
$$= -3465 \text{ ft-lb}$$

By applying Equation 3-18, the work in heat energy units ΔW

$$= \frac{-3465}{778} = -4.45 \text{ Btu}$$

Notice that the process in Example 3-19 is the exact reverse of that of Example 3-18. Where the process in Example 3-18 is an expansion, the process in Example 3-19 is a compression. Both processes occur between the same two conditions, except that the initial and final conditions are reversed. Notice also that whereas work is done by the gas during the expansion process, work is done on the gas during the compression process. But since the change of condition takes place between the same limits in both cases, the amount of work done in each case is the same.

3-27. Heat Transferred during a Constant Temperature Process. Since there is no change in the temperature during an isothermal process, there is no change in the internal kinetic energy and ΔK equals zero. According to Equation 3-21, the heat energy transferred during a constant temperature process is exactly equal to the work done in Btu. During an isothermal expansion heat is transferred to the gas to supply the energy to do the work that is done by the gas, whereas during an isothermal compression heat is transferred from the gas so that the internal energy of the gas is not increased by the performance of work on the gas.

Example 3-20. Determine the quantity of heat transferred to the gas during the constant temperature expansion described in Example 3-18.

Solution. From Example 3-18, $\Delta W = 4.45$ Btu

Since, in the isothermal process, ΔW_t equals ΔQ_t, $\Delta Q_t = 4.45$ Btu

Example 3-21. What is the quantity of heat transferred to the gas during the constant temperature process described in Example 3-19.

Solution. From Example 3-19, $\Delta W = -4.45$ Btu

Since ΔW_t equals ΔQ_t, $\Delta Q_t = -4.45$ Btu

Again, notice that a negative amount of heat is transferred to the gas, indicating that heat in this amount is actually given up by the gas during the process.

3-28. Adiabatic Process. An adiabatic process is described as one wherein the gas changes its condition without absorbing or rejecting heat, as such, from or to an external body during the process. Furthermore, the pressure, volume, and temperature of the gas all vary during an adiabatic process, none of them remaining constant.

When a gas expands adiabatically, as in any other expansion, the gas does external work and energy is required to do the work. In the processes previously described, the gas absorbed the energy to do the work from an external source. Since, during an adiabatic process, no heat is absorbed from an external source, the gas must do the external work at the expense of its own energy. An adiabatic expansion is always accompanied by a decrease in the temperature of the gas as the gas gives up its own internal energy to do the work (Fig. 3-7).

When a gas is compressed adiabatically, work is done on the gas by an external body. The energy of the gas is increased in an amount equal to the amount of work done, and since no heat energy is given up by the gas to an external body during the compression, the heat energy equivalent of the work done on the gas is set up as an increase in the internal energy, and the temperature of the gas increases.

Because no heat, as such, is transferred to or from the gas during an adiabatic process, ΔQ_a is always zero and the energy equation for an adiabatic process is written as follows:

$$\Delta K_a + \Delta W_a = 0 \qquad (3\text{-}23)$$

Therefore,

$$\Delta W_a = -\Delta K_a \qquad \text{and} \qquad \Delta K_a = -\Delta W_a$$

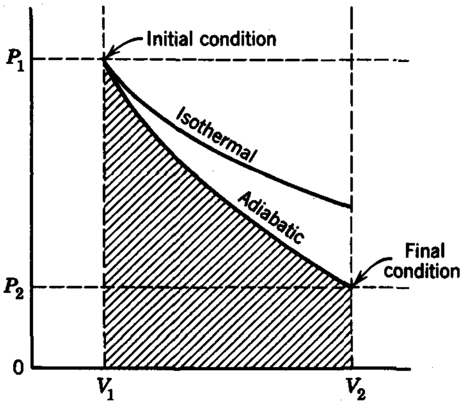


Fig. 3-7. Pressure-volume diagram of adiabatic process. An isothermal curve is drawn in for comparison.

3-29. Work of an Adiabatic Process. The work of an adiabatic process may be evaluated by the following equation:

$$W_a = \frac{P_1V_1 - P_2V_2}{k - 1} \qquad (3\text{-}24)$$

where k = the ratio of the specific heats of the gas in question, C_p/C_v

Example 3-22. A gas having an initial pressure of 2500 psfa and an initial volume of 2 cu ft is expanded adiabatically to a volume of 4 cu ft. If the final pressure is 945 psfa, determine the external work done in heat energy units.

Solution

C_p for air $= 0.2375$ Btu/lb

C_v for air $= 0.169$ Btu/lb

The ratio of the specific heats, k

$$= \frac{C_p}{C_v} = \frac{0.2375}{0.169} = 1.406$$

Applying Equation 3-24, the work of adiabatic expansion in foot-pounds W_a

$$= \frac{(2500 \times 2) - (945 \times 4)}{1.406 - 1} = \frac{5000 - 3780}{0.406} = \frac{1220}{0.406} = 3005 \text{ ft-lb}$$

Applying Equation 3-18, the work in heat energy units ΔW_a

$$= \frac{3005}{778} = 3.86 \text{ Btu}$$

Example 3-23. A gas having an initial pressure of 945 psfa and an initial volume of 4 cu ft is compressed adiabatically to a volume of 2 cu ft. If the final pressure of the air is 2500 psfa, how much work is done in heat energy units?

Solution. From Example 3-22, k for air $= 1.406$

Applying Equation 3-24, the work done in foot-pounds W_a

$$= \frac{(945 \times 4) - (2500 \times 2)}{1.406 - 1} = \frac{3780 - 5000}{0.406} = \frac{-1220}{0.406} = -3005 \text{ ft-lb}$$

Applying Equation 3-18, the work in heat energy units ΔW_a

$$= \frac{-3005}{778} = -3.86 \text{ Btu}$$

3-30. Comparison of the Isothermal and Adiabatic Processes. A comparison of the isothermal and adiabatic processes is of interest. Whenever a gas expands, work is done by the gas, and energy from some source is required to do the work. In an isothermal expansion, all of the energy to do the work is supplied to the gas as heat from an external source. Since the energy is supplied to the gas from an external source at exactly the same rate that the gas is doing work, the internal energy of the gas neither increases nor decreases and the temperature of the gas remains constant during the process. On the other hand, in an adiabatic expansion there is no transfer of heat to the gas during the process and all of the work of expansion must be done at the expense of the internal energy of the gas. Therefore, the internal energy of the gas is always diminished by an amount equal to the amount of work done and the temperature of the gas decreases accordingly.

Consider now isothermal and adiabatic compression processes. In any compression process, work is done on the gas by the compressing member, usually a piston, and an amount of energy equal to the amount of work done on the gas is transferred to the gas as work. During an isothermal compression process, energy is transferred as heat from the gas to an external sink at exactly the same rate that work is being done on the gas. Therefore, the internal energy of the gas neither increases nor decreases during the process and the temperature of the gas remains constant. On the other hand, during an adiabatic compression, there is no transfer of energy as heat from the gas to an external sink. Therefore, an amount of energy equal to the amount of work done on the gas is set up in the gas as an increase in the internal energy, and the temperature of the gas increases accordingly.

3-31. The Polytropic Process. Perhaps the simplest way of defining a polytropic process is by comparison with the isothermal and adiabatic processes. The isothermal expansion, in which the energy to do the work of expansion is supplied entirely from an external source, and the adiabatic expansion, in which the energy to do the work of expansion is supplied entirely by the gas itself, may be thought of as the extreme limits between which all expansion processes will fall. Then, any expansion process in which the energy to do the work of expansion is supplied partly from an external source and partly from the gas itself will follow a path which will fall somewhere between those of the isothermal and adiabatic processes (Fig. 3-8). Such a process is known as a polytropic process. If during a polytropic expansion most of the energy to do the work comes from an external source, the polytropic process will more nearly approach the isothermal. On the other hand, when the greater part of the energy to do the external work comes from the gas itself, the process more nearly approaches the adiabatic.

This is true also for the compression process. When a gas loses heat during a compression process, but not at a rate sufficient to maintain the temperature constant, the compression is polytropic. The greater the loss of heat, the closer the polytropic process approaches the isothermal. The smaller the loss of heat, the closer the polytropic process approaches the adiabatic. Of course, with no heat loss, the process becomes adiabatic.

The actual compression of a gas in a compressor will usually very nearly approach adiabatic compression. This is because the time of compression is normally very short and there is not sufficient time for any significant amount of heat to be transferred from the gas through the cylinder walls to the surroundings. Water jacketing of the cylinder will usually increase the rate of heat rejection and move the path of the compression closer to the isothermal.

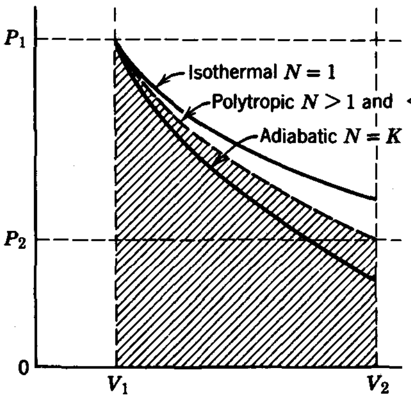


Fig. 3-8. Pressure-volume diagram of a polytropic process. Adiabatic and isothermal curves are drawn in for comparison.

3-32. PVT Relationship during Adiabatic Processes. Since the temperature, pressure, and volume all change during an adiabatic process, they will not vary in accordance with Charles' and Boyle's laws. The relationship between the pressure, temperature, and volume during an adiabatic process may be evaluated by the following equations:

$$T_2 = T_1 \times \frac{V_1^{(k-1)}}{V_2^{(k-1)}} \tag{3-25}$$

$$T_2 = T_1 \times \left(\frac{P_2}{P_1}\right)^{(k-1)/k} \tag{3-26}$$

$$P_2 = P_1 \times \left(\frac{V_1}{V_2}\right)^{k} \tag{3-27}$$

$$P_2 = P_1 \times \left(\frac{T_2}{T_1}\right)^{k/(k-1)} \tag{3-28}$$

$$V_2 = V_1 \times \left(\frac{T_1}{T_2}\right)^{1/(k-1)} \tag{3-29}$$

$$V_2 = V_1 \times \left(\frac{P_1}{P_2}\right)^{1/k} \tag{3-30}$$

Example 3-24. Air is expanded adiabatically from a volume of 2 cu ft to a volume of

4 cu ft. If the initial pressure of the air is 24,000 psfa, what is the final pressure in psfa?

Solution. From Table 3-1, k for air $= 1.406$

Applying Equation 3-27, the final pressure P_2

$$= 24{,}000 \times \left(\frac{2}{4}\right)^{1.406}$$
$$= 24{,}000 \times 0.5^{1.406}$$
$$= 24{,}000 \times 0.378$$
$$= 9072 \text{ psfa}$$

Example 3-25. Air is expanded adiabatically from a volume of 2 cu ft to a volume of 4 cu ft. If the initial temperature of the air is 600° R, what is the final temperature in degrees Rankine?

Solution. From Table 3-1, k for air $= 1.406$

Applying Equation 3-25, the final temperature T_2

$$= 600 \times \frac{(2)^{(1.406-1)}}{(4)^{(1.406-1)}}$$
$$= 600 \times \frac{(2)^{0.406}}{(4)^{0.406}}$$
$$= 600 \times \frac{1.325}{1.756}$$
$$= 600 \times 0.755$$
$$= 453° \text{ R}$$

Example 3-26. Air is expanded adiabatically from an initial pressure of 24,000 psfa to a final pressure of 9072 psfa. If the initial temperature is 600° R, what is the final temperature?

Solution. From Table 3-1, k for air $= 1.406$

Applying Equation 3-26, the final temperature T_2

$$= 600 \times \left(\frac{9072}{24{,}000}\right)^{\frac{1.406-1}{1.406}}$$
$$= 600 \times (0.378)^{(0.289)}$$
$$= 600 \times 0.755$$
$$= 453° \text{ R}$$

3-33. Exponent of Polytropic Expansion and Compression. The presssure-temperature-volume relationships for the polytropic process can be evaluated by Equations 3-25 through 3-30, except that the polytropic expansion or compression exponent n is substituted for k. Too, the work of a polytropic process can be determined by Equation 3-24 if n is substituted for k.

The exponent n will always have a value somewhere between 1 and k for the particular gas undergoing the process.* Usually, the value of n must be determined by actual test of the machine in which the expansion or compression occurs. In some instances average values of n for some of the common gases undergoing changes under more or less standard conditions are given in tables. If the values of two properties are known for both initial and final conditions, the value of n may be calculated. The following sample equation shows the relationship:

$$n = \frac{\log (P_1/P_2)}{\log (V_1/V_2)} \tag{3-31}$$

Example 3-27. Air, having an initial pressure of 24,000 psfa and an initial temperature of 600° R, is expanded polytropically from a volume of 2 cu ft to a volume of 4 cu ft. If the exponent of polytropic expansion is 1.2, determine:

(*a*) The weight of the air in pounds
(*b*) The final pressure in psfa
(*c*) The final temperature in degrees Rankine
(*d*) The work done by the gas in Btu
(*e*) The increase in internal energy
(*f*) The heat transferred to the gas.

Solution

(*a*) From Table 3-1, R for air $= 53.3$

Rearranging and applying Equation 3-10, the weight of air M

$$= \frac{PV}{RT}$$
$$= \frac{24{,}000 \times 2}{53.3 \times 600}$$
$$= 1.5 \text{ lb}$$

* The value of n depends upon the specific heat of the gas during the process. Since the specific heat may take any value, it follows that theoretically n may have any value. In actual machines, however, n will nearly always have some value between 1 and k.

Broadly defined, a polytropic process is any process during which the specific heat remains constant. By this definition, all five processes discussed in this chapter are polytropic processes. It is general practice today to restrict the term polytropic to mean only those processes which follow a path falling between those of the isothermal and adiabatic processes. The exponents of isothermal and adiabatic expansions or compressions are 1 and k, respectively. Hence, the value of n for the polytropic process must fall between 1 and k. The closer the polytropic process approaches the adiabatic, the closer n will approach k.

(*b*) Applying Equation 3-27, the final pressure P_2

$$= 24{,}000 \times \left(\frac{2}{4}\right)^{1.2}$$
$$= 24{,}000 \times (0.5)^{1.2}$$
$$= 24{,}000 \times 0.435$$
$$= 10{,}440 \text{ psfa}$$

(*c*) Applying Equation 3-25, the temperature T_2

$$= 600 \times \frac{(2)^{(1.2-1)}}{(4)^{(1.2-1)}}$$
$$= 600 \times \frac{(2)^{0.2}}{(4)^{0.2}}$$
$$= 600 \times \frac{1.149}{1.32}$$
$$= 522° \text{ R}$$

(*d*) Applying Equation 3-24, the work done W

$$= \frac{(24{,}000 \times 2) - (4 \times 10{,}440)}{1.2 - 1}$$
$$= \frac{48{,}000 - 41{,}760}{0.2}$$
$$= \frac{6240}{0.2}$$
$$= 31{,}200 \text{ ft-lb}$$

Applying Equation 3-18, the work in Btu ΔW

$$= \frac{31{,}200}{778}$$
$$= 40.10 \text{ Btu}$$

(*e*) From Table 3-1, C_v for air

$$= 0.169 \text{ Btu/lb}$$

Applying Equation 3-15, the increase in internal energy ΔK

$$= 1.5 \times 0.169 \times (522 - 600)$$
$$= 1.5 \times 0.169 \times (-78)$$
$$= -19.77 \text{ Btu}$$

(*f*) Applying Equation 3-9, the heat energy transferred to the gas ΔQ

$$= \Delta K + \Delta W$$
$$= -19.77 + 40.10$$
$$= 20.33 \text{ Btu}$$

Notice in Example 3-27 that the work done by the air in the polytropic expansion is equivalent to 40.10 Btu. Of this amount, 20.33 Btu is supplied from an external source, whereas the other portion, 19.77 Btu, is supplied by the gas itself, thereby reducing the internal kinetic energy by this amount.

PROBLEMS

1. Three pounds of air occupy a volume of 24 cu ft. Determine:
   (*a*) The density of the air. *Ans.* 0.125 lb/cu ft
   (*b*) The specific volume. *Ans.* 8 cu ft/lb

2. The volume of a certain weight of air is kept constant while the temperature of the air is increased from 55° F to 100° F. If the initial pressure is 25 psig, what is the final pressure of the air in psig? *Ans.* 28.47 psig

3. A certain weight of air confined in a container is cooled from 150° F to 70° F. If the initial pressure of the air is 36.3 psig, what is the final pressure of the air in psig? *Ans.* 29.6 psig

4. One pound of air at atmospheric pressure has a volume of 13.34 cu ft at a temperature of 70° F. If the air is passed across a heat exchanger and is heated to a temperature of 150° F while its pressure is kept constant, what is the final volume of the air? *Ans.* 15.35 cu ft

5. A cylinder of oxygen has a volume of 5 cu ft. A gage on the cylinder reads 2200 psi. If the temperature of the oxygen is 85° F, what is the weight of the oxygen in the cylinder? *Ans.* 60.6 lb

6. In Problem 4, determine:
   (*a*) The work done by the air during the heating. *Ans.* 4254.8 ft-lb or 5.47 Btu
   (*b*) The increase in the internal kinetic energy. *Ans.* 13.52 Btu
   (*c*) The quantity of heat transferred to the air. *Ans.* 19 Btu

7. A certain weight of air having an initial volume of 0.1334 cu ft and an initial temperature of 70° F is drawn into the suction side of an air compressor. If the air enters the cylinder at standard atmospheric pressure and is compressed isothermally to a final pressure of 150 psia, determine:
   (*a*) The weight of air in the cylinder at the start of the compression stroke. *Ans.* 0.01 lb
   (*b*) The final temperature of the air in degrees Rankine. *Ans.* 530° R
   (*c*) The volume of the air at the end of the compression stroke. *Ans.* 0.0131 cu ft
   (*d*) The work of compression in Btu. *Ans.* 0.843 Btu
   (*e*) The increase or decrease in internal energy. *Ans.* None
   (*f*) The energy transferred to the gas during the compression. *Ans.* −0.843 Btu

8. Assume that the air in Problem 7 is compressed adiabatically rather than isothermally. Compute:
   (*a*) The final temperature of the air in degrees Rankine. *Ans.* 1038° R
   (*b*) The volume of the air at the end of the compression stroke. *Ans.* 0.0256 cu ft
   (*c*) The work of compression in Btu. *Ans.* 0.86 Btu

(*d*) The increase in the internal kinetic energy. *Ans.* 0.86 Btu

(*e*) The heat energy transferred to or from the gas during the compression. *Ans.* None

9. Assuming that the air in Problem 7 is compressed polytropically rather than isothermally. If *n* equals 1.2, compute:

(*a*) The final temperature of the air in degrees Rankine. *Ans.* 779.4° R

(*b*) The volume of the air at the end of the compression stroke. *Ans.* 0.0192 cu ft

(*c*) The work of compression is Btu. *Ans.* 0.85 Btu

(*d*) The decrease in the internal kinetic energy. *Ans.* 0.45 Btu

(*e*) The heat energy transferred from the gas during the compression. *Ans.* 0.40 Btu

10. Compare the results of Problems 7, 8, and 9.

4

Saturated and Superheated Vapors

4-1. Saturation Temperature. When the temperature of a liquid is raised to a point such that any additional heat added to the liquid will cause a part of the liquid to vaporize, the liquid is said to be saturated. Such a liquid is known as a saturated liquid and the temperature of the liquid at that condition is called the saturation temperature (Sections 2-31 and 2-32).

4-2. Saturated Vapor. The vapor ensuing from a vaporizing liquid is called a saturated vapor as long as the temperature and pressure of the vapor are the same as those of the saturated liquid from which it came. A saturated vapor may be described also as a vapor at a temperature such that any further cooling of the vapor will cause a portion of the vapor to condense and thereby resume the molecular structure of the liquid state. It is important to understand that the saturation temperature of the liquid (the temperature at which the liquid will vaporize if heat is applied) and the saturation temperature of the vapor (the temperature at which the vapor will condense if heat is removed) are the same for any given pressure and that the liquid cannot exist as a liquid at any temperature above its saturation temperature, whereas a vapor cannot exist as a vapor at any temperature below its saturation temperature.*

* Under certain conditions it is possible to "supercool" water vapor momentarily below its saturation temperature. However, this is a very unstable condition and cannot be maintained except momentarily.

For example, in Fig. 4-1, the water in the heated vessel is saturated and is vaporizing at 212° F as the latent heat of vaporization is supplied by the burner. The water vapor (steam) rising from the water is saturated and remains at the saturation temperature (212° F) until it reaches the condenser. As the saturated vapor gives up heat to the cooler water in the condenser, it condenses back into the liquid state. Since condensation occurs at a constant temperature, the water resulting from the condensing vapor is also at 212° F. The latent heat of vaporization, absorbed as the water vaporizes into steam, is given up by the steam as the steam condenses back into water.

4-3. Superheated Vapor. A vapor at any temperature above its saturation temperature is a superheated vapor (Section 2-34). If, after vaporization, a vapor is heated so that its temperature is raised above that of the vaporizing liquid, the vapor is said to be superheated. In order to superheat a vapor it is necessary to separate the vapor from the vaporizing liquid as shown in Fig. 4-2. As long as the vapor remains in contact with the liquid it will be saturated. This is because any heat added to a liquid-vapor mixture will merely vaporize more liquid and no superheating will occur.

Before a superheated vapor can be condensed, the vapor must be de-superheated, that is, the vapor must first be cooled to its saturation temperature. Heat removed from a superheated vapor will cause the temperature of the vapor to decrease until the saturation temperature is reached. At this point, any further removal of heat will cause a part of the vapor to condense.

4-4. Subcooled Liquid. If, after condensation, a liquid is cooled so that its temperature is reduced below the saturation temperature, the liquid is said to be subcooled. Thus, a liquid at any temperature below the saturation temperature and above the fusion point is a subcooled liquid.

4-5. The Effect of Pressure on the Saturation Temperature. The saturation temperature of a liquid or a vapor varies with the pressure. Increasing the pressure raises the saturation temperature and decreasing the pressure lowers the saturation temperature. For example, the saturation temperature of water at atmospheric pressure (0 psig or 14.7 psia) is

212° F. If the pressure over the water is increased from 0 psig to 5.3 psig (20 psia), the saturation temperature of the water increases from 212° F to 228° F. On the other hand, if the pressure over the water is reduced from 14.7 psia to 10 psia, the new saturation temperature of the water will be 193.2° F. Figure 4-3 is a graphical representation of the relationship between the pressure and the saturation temperature of water.

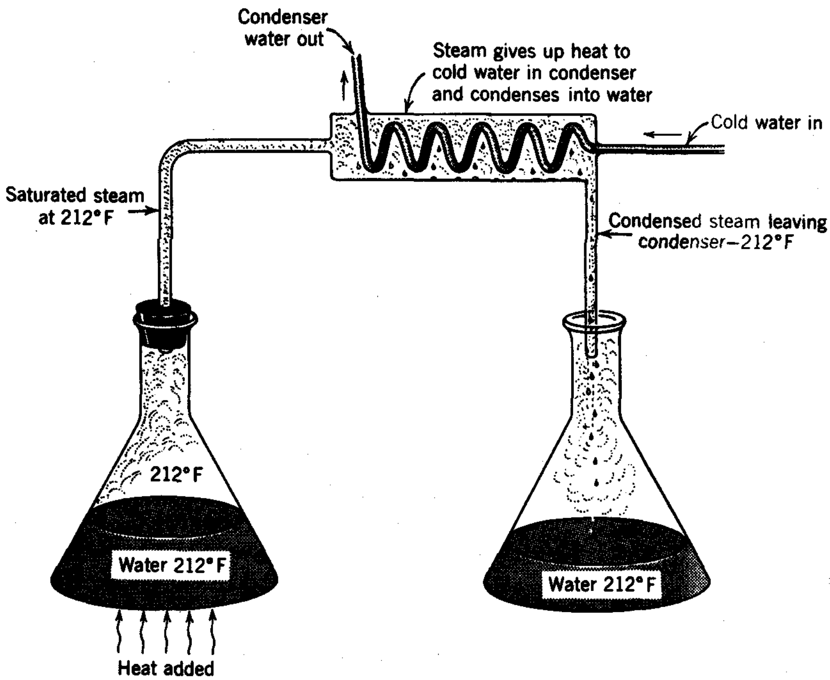


Fig. 4-1. Saturated vapor.

To illustrate the effect of pressure on the saturation temperature of a liquid, assume that water is confined in a closed vessel which is equipped with a throttling valve at the top (Fig. 4-4*a*). A compound gage is used to determine the pressure exerted in the vessel and two thermometers are installed so that one records the temperature of the water and the other the temperature of the vapor over the water. With the throttling valve wide open, the pressure exerted over the water is atmospheric (0 psig or 14.7 psia). Since the saturation temperature of the water at atmospheric pressure is 212° F, the temperature of the water will rise as the water is heated until it reaches 212° F. At this point, if the heating is continued, the water will begin to vaporize. Soon the space above the water will be filled with billions and billions of water vapor molecules darting about at high velocities. Some of the vapor molecules will fall back into the water to become liquid molecules again, whereas others will escape through the opening to the outside and be carried away by air currents. If the opening at the top of the vessel is of sufficient size to allow the vapor to escape freely, the vapor will leave the vessel at the same rate that the liquid is vaporizing. That is, the number of molecules which are leaving the liquid to become vapor molecules will be exactly equal to the number of vapor molecules which are leaving the space, either by escaping to the outside or by falling back into the liquid. Thus, the number of vapor molecules and the density of the vapor above the liquid will

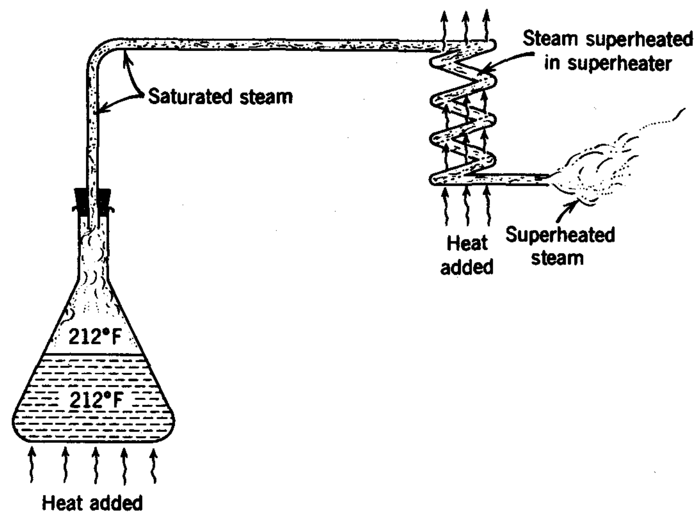


Fig. 4-2. Superheated vapor.

remain constant and the pressure exerted by the vapor will be equal to that of the atmosphere outside of the vessel.

Under this condition the water vapor ensuing from the vaporizing liquid will be saturated, that is, its temperature and pressure will be the same as that of the water, 212° F and 14.7 psia. The density of the water vapor at that temperature and pressure will be 0.0373 lb/cu ft and its specific volume will be 1/0.0373 or 26.8 ft^3/lb.

Regardless of the rate at which the liquid is vaporizing, as long as the vapor is allowed to escape freely to the outside so that the pressure and density of the vapor over the liquid does not change, the liquid will continue to vaporize at 212° F.

Suppose that the throttling valve is partially closed so that the escape of the vapor from the vessel is impeded somewhat. For a time the equilibrium will be disturbed in that the vapor will not be leaving the vessel at the same rate the liquid is vaporizing. The number of vapor molecules in the space above the liquid will increase, thereby increasing the density and the pressure of the vapor over the liquid and raising the saturation temperature.

If it is assumed that the pressure of the vapor increases to 5.3 psig (20 psia) before equilibrium is again established, that is, before the rate at which the vapor is escaping to the outside is exactly equal to the rate at which the liquid is vaporizing, the saturation temperature will be

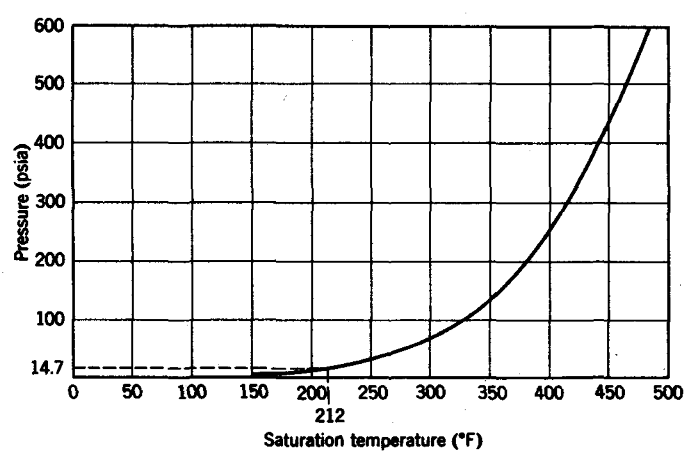


Fig. 4-3. Variation in the saturation temperature of water with changes in pressure.

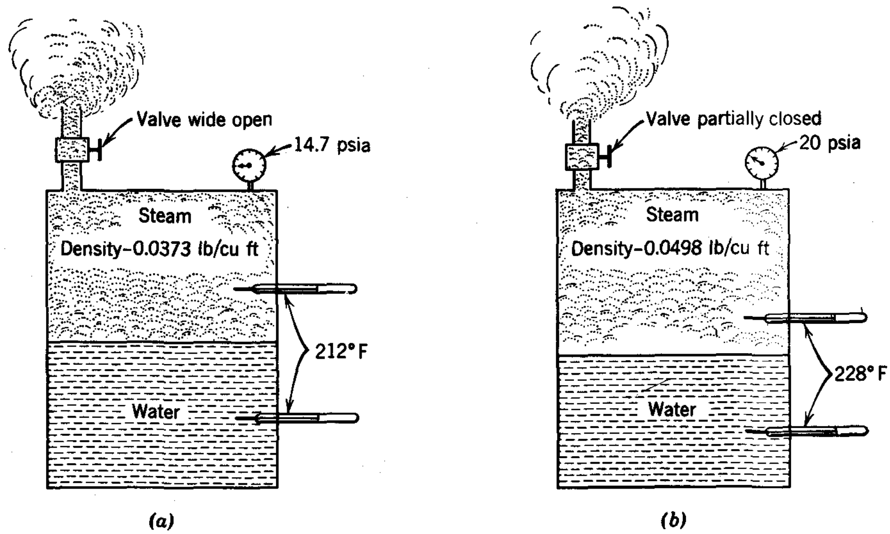


Fig. 4-4.

228° F, the density of the vapor will be 0.0498 lb/cu ft, and 1 lb of vapor will occupy a volume of 20.08 cu ft. This condition is illustrated in Fig. 4-4*b*.

By comparing the condition of the vapor in Fig. 4-4*b* with that of the vapor in Fig. 4-4*a*, it will be noted that the density of the vapor is greater at the higher pressure and saturation temperature. Furthermore, it is evident that the pressure and the saturation temperature of liquid or vapor can be controlled by regulating the rate at which the vapor escapes from over the liquid.

In Fig. 4-4*a*, the rate of vaporization will have little or no effect on the pressure and saturation temperature because the vapor is allowed to escape freely so that the density and pressure of the vapor over the liquid will neither increase nor decrease as the rate of vaporization is changed. On the other hand, in Fig. 4-4*b*, any increase in the rate of vaporization will cause an increase in the density and pressure of the vapor and result in an increase in the saturation temperature. The reason is that any increase in the rate of vaporization will necessitate the escape of a greater quantity of vapor in a given length of time. Since the size of the vapor outlet is fixed by the throttling action of the valve, the pressure of the vapor in the vessel will increase until the pressure difference between the inside and outside of the vessel is sufficient to allow the vapor to escape at a rate equal to that at which the liquid is vaporizing. The increase in pressure, of course, results in an increase in the saturation temperature and in the density of the vapor. Likewise, any decrease in the rate of vaporization will have the opposite effect. The pressure and density of the vapor over the liquid will decrease and the saturation temperature will be lower.

Assume now that the throttling valve on the container is again opened completely, as in Fig. 4-4*a*, so that the vapor is allowed to escape freely and unimpeded from over the liquid. The density and pressure of the vapor will decrease until the pressure of the vapor is again equal to that of the atmosphere outside of the container. Since the saturation temperature of water at atmospheric pressure is 212° F and since a liquid cannot exist as a liquid at any temperature above its saturation temperature corresponding to its pressure, it is evident that the water must cool itself from 228° F to 212° F at the instant that the pressure drops from 20 psia to atmospheric pressure. To accomplish this cooling, a portion of the liquid will "flash" into a vapor. The latent heat necessary to vaporize the portion of the liquid that flashes into the vapor state is supplied by the mass of the liquid and, as a result of supplying the vaporizing heat, the temperature of the mass of the liquid will be reduced to the new saturation temperature. Enough of the liquid will vaporize to provide the required amount of cooling.

4-6. Vaporization. The vaporization of a liquid may occur in two ways: (1) by evaporation and (2) by ebullition or "boiling." The vaporization of a liquid by evaporation occurs only at the free surface of the liquid and may take place at any temperature below the saturation temperature. On the other hand, ebullition or boiling takes place both at the free surface and within the body of the liquid and can occur only at the saturation temperature. Up to this point, only ebullition or boiling has been considered.

4-7. Evaporation. Evaporation is taking place continually and the fact that water evaporates from lakes, rivers, ponds, clothes, etc., is sufficient evidence that evaporation can and does occur at temperatures below the saturation temperature. Any liquid open to the atmosphere, regardless of its temperature, will gradually evaporate and be diffused into the air.

The vaporization of liquids at temperatures below their saturation temperature can be explained in this manner. The molecules of a liquid are in constant and rapid motion, their velocities being determined by the temperature of the liquid. In the course of their movements the molecules are continually colliding with one another and, as a result of these impacts, some of the molecules of the liquid momentarily attain velocities much higher than the average velocity of the other molecules of the mass. Thus, their energy is much greater than the average energy of the mass. If this occurs within the body of the liquid, the high velocity molecules quickly lose their extra energy in subsequent collisions with other molecules. However, if the molecules attaining the higher than normal velocities are near the surface, they may project themselves from the surface of the liquid and escape into the air to become vapor molecules. (Fig. 4-5). The molecules so escaping from the liquid are diffused throughout the air. They occupy the relatively large spaces which exist between the molecules of the air and become a part of the atmospheric air.

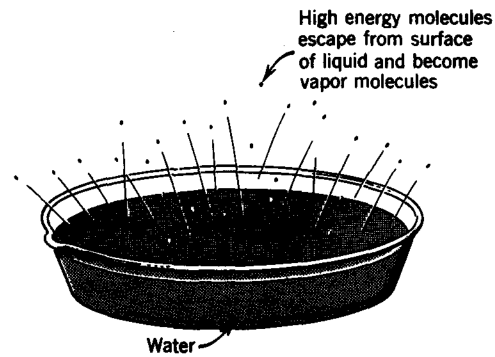


Fig. 4-5. Evaporation from surface of a liquid.

4-8. Rate of Vaporization. For any given temperature, some liquids will evaporate faster than others. Liquids having the lowest "boiling" points, that is, the lowest saturation temperature for a given pressure, evaporate at the highest rate. However, for any particular liquid, the rate of vaporization varies with a number of factors. In general, the rate of vaporization increases as the temperature of the liquid increases and as the pressure over the liquid decreases. Evaporation increases also with the amount of exposed surface. Furthermore, it will be shown later that the rate of evaporation is dependent on the degree of saturation of the vapor which is always adjacent to and above the liquid.

4-9. The Cooling Effect of Evaporation. Since it is the higher velocity molecules (those having the most energy) which escape from the surface of an evaporating liquid, it follows that the average energy of the mass is thereby reduced and the temperature of the mass lowered. Whenever any portion of a liquid vaporizes, an amount of heat equal to the latent heat of vaporization must be absorbed by that portion, either from the mass of the liquid, from the surrounding air, or from adjacent objects. Thus, the energy and temperature of the mass are reduced as it supplies the latent heat of vaporization to that portion of the liquid which vaporizes. The temperature of the mass is reduced to a point slightly below that of the surrounding media and the temperature difference so established causes heat to flow from the surrounding media into the mass of the liquid. The energy lost by the mass during vaporization is thereby replenished and evaporation becomes a continuous process as long as

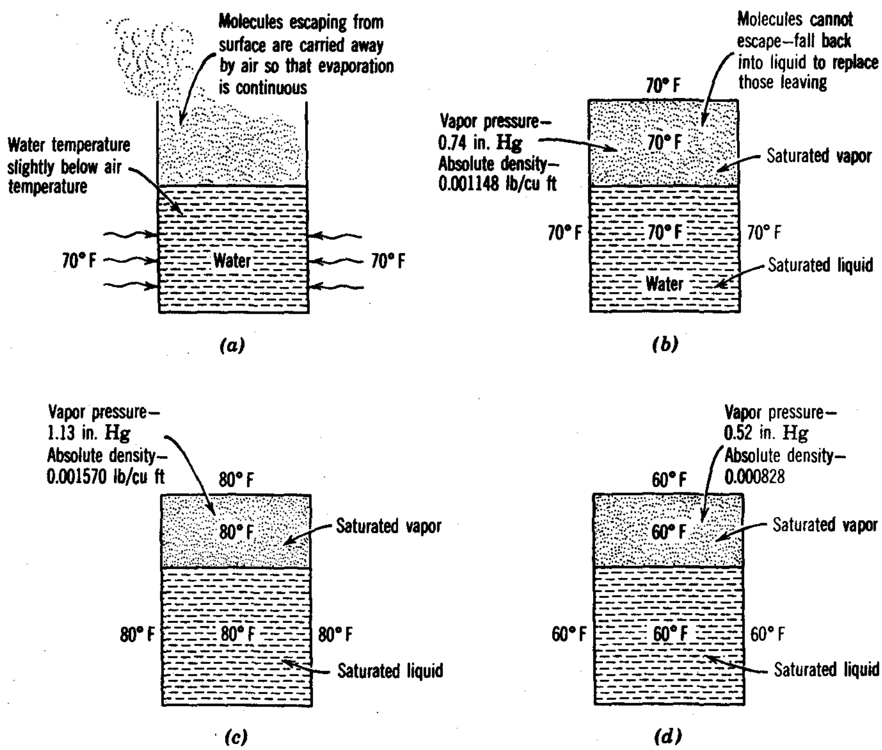


Fig. 4-6.

any of the liquid remains. The vapor resulting from evaporation is diffused into and carried away by the air.

4-10. Confined Liquid-Vapor Mixtures. When a vapor is confined in a container with a portion of its own liquid, both the vapor and the liquid will be saturated. To illustrate, assume that an open container is partially filled with water and is stored where the ambient temperature is 70° F (Fig. 4-6*a*). The water will be evaporating at 70° F and, as described in the previous section, the vapor molecules leaving the liquid will be diffused into the surrounding air so that evaporation will continue until all of the liquid is evaporated. However, if a tightly fitting cover is placed over the container, the vapor molecules will be unable to escape to the outside and they will collect above the liquid. Soon the space above the liquid will be so filled with vapor molecules that there will be as many molecules falling back into the liquid as there are leaving the liquid. A condition of equilibrium will be attained, the vapor will be saturated, and no further evaporation will occur. The energy of the liquid will be increased by the vapor molecules which are returning to the liquid in exactly the same amount that it is diminished by the molecules that are leaving. Since no further cooling will take place by evaporation, the liquid will assume the temperature of the surrounding air and heat transfer will cease. (Fig. 4-6*b*).

If, at this point, the ambient temperature rises to, say, 80° F, heat transfer will again take place between the surrounding air and the liquid. The temperature and average molecular velocity of the liquid will be increased and evaporation will be resumed. The number of molecules leaving the liquid will again be greater than the number returning and the density and pressure of the vapor above the liquid will be increased. As the density and pressure of the vapor increase, the saturation temperature of the liquid increases. Eventually, when the saturation temperature reaches 80° F and is equal to the ambient temperature, no further heat transfer will occur and evaporation will cease. Equilibrium will have again been established. The density and pressure of the vapor will be greater than before, the saturation temperature of the liquid-vapor mixture will be higher, and there will be more vapor and less liquid in the container than previously (Fig. 4-6*c*).

Suppose now that the ambient temperature falls to 60° F. When this occurs, heat will flow from the 80° F liquid-vapor mixture to the cooler surrounding air. As the liquid-vapor mixture loses heat to the surrounding air, its temperature and average molecular velocity will be decreased and many of the vapor molecules, lacking sufficient energy to remain in the vapor state, will fall back into the liquid and resume the molecular arrangement of the liquid state; that is, a part of the vapor will condense. The density and pressure of the vapor will be diminished and the saturation temperature of the mixture will be reduced. When the saturation temperature of the mixture falls to 60° F it will be the same as the ambient temperature and no further heat flow will occur. Equilibrium will have been established and the number of molecules re-entering the liquid will exactly equal those which are leaving. At this new condition, the density and pressure of the vapor will be less than before, the saturation temperature will be lower, and since a part of the vapor condensed into liquid, there will be more liquid and less vapor comprising the mixture than at the previous condition (Fig. 4-6*d*).

4-11. Sublimation. It is possible for a substance to go directly from the solid state to the vapor state without apparently passing through the liquid state. Any solid substance will sublime at any temperature below its fusion temperature. Sublimation takes place in a manner similar to evaporation, although much slower, in that the higher velocity molecules near the surface escape from the mass into the surrounding air and become vapor molecules. One of the most familiar examples of sublimation is that of solid CO_2 (dry ice), which, at normal temperatures and pressures, sublimes directly from the solid to the vapor state. Damp wash frozen on the line in the winter time will sublime dry. During freezing weather ice and snow will sublime from streets and sidewalks, etc.

4-12. Condensation. Condensation of a vapor may be accomplished in several ways: (1) by extracting heat from a saturated vapor, (2) by compressing the vapor while its temperature remains constant, or (3) by some combination of these two methods.

4-13. Condensing by Extracting Heat from a Saturated Vapor. A saturated vapor has been previously described as one at a condition such that any further cooling will cause a part of the vapor to condense. This is because a vapor cannot exist as a vapor at any temperature below its saturation temperature. When the vapor is cooled, the vapor molecules cannot maintain sufficient energy and velocity to overcome the attractive forces of one another and remain as vapor molecules. Some of the molecules, overcome by the attractive forces, will revert to the molecular structure of the liquid state. When condensation occurs while the vapor is confined so that the volume remains constant, the density and pressure of the vapor will decrease so that there is a decrease in the saturation temperature. If, as in a vapor condenser (Fig. 4-1), more vapor is entering the vessel as the vapor condenses and drains from the vessel as a liquid, the density, pressure, and saturation temperature of the vapor will remain constant and condensation will continue as long as heat is continuously extracted from the vapor.

4-14. Condensing by Increasing the Pressure at a Constant Temperature. When a vapor is compressed at a constant temperature, its volume diminishes and the density of the vapor increases as the molecules of the vapor are forced into a smaller volume. The saturation temperature of the vapor increases as the pressure increases until a point is reached where the saturation temperature of the vapor is equal to the actual temperature of the vapor. When this occurs, the density of the vapor will be at a maximum value for that condition, and any further compression will cause a part of the vapor to assume the more restrained molecular structure of the liquid state. Thereafter, condensation will continue as long as compression continues so that the density and pressure of the remaining vapor cannot be further increased. If the temperature of the vapor is to remain constant, heat must be removed from the vapor during the compression

(Section 3-25). If heat is not removed from the vapor, the temperature of the vapor will increase and condensation will not occur.

A careful analysis of Sections 4-13 and 4-14 will show that in either case the vapor is brought to a saturated condition before condensation begins and that heat is removed from the vapor in order to bring about condensation. Furthermore, the vapor is saturated in each case only when the saturation temperature and the actual temperature of the vapor are the same.

In Section 4-13, heat is removed from the vapor at a constant pressure until the temperature of the vapor falls to the saturation temperature corresponding to its pressure, whereupon the continued removal of heat causes a part of the vapor to condense.

In Section 4-14, the pressure of the vapor is increased while the temperature of the vapor remains constant until the saturation temperature of the vapor corresponding to the increased pressure is equal to the actual temperature of the vapor. In both cases, since the vapor must give up the latent heat of vaporization in order to condense, heat must be removed from the vapor.

4-15. Critical Temperature. The temperature of a gas may be raised to a point such that it cannot become saturated regardless of the amount of pressure applied. The critical temperature of any gas is the highest temperature the gas can have and still be condensible by the application of pressure. The critical temperature is different for every gas. Some gases have high critical temperatures while the critical temperatures of others are relatively low. For example, the critical temperature of water vapor is 706° F, whereas the critical temperature of air is approximately −225° F.

4-16. Critical Pressure. Critical pressure is the lowest pressure at which a substance can exist in the liquid state at its critical temperature; that is, it is the saturation pressure at the critical temperature.

4-17. Important Properties of Gases and Vapors. Although a gas or vapor has many properties, only six are of particular importance in the study of refrigeration. These are pressure, temperature, volume, enthalpy, internal energy, and entropy. Pressure, temperature, and volume are called measurable properties because they can actually be measured. Enthalpy, internal energy, and entropy cannot be measured. They must be calculated and are therefore known as calculated properties.

Pressure, temperature, volume, and internal energy have already been discussed to some extent. A discussion of enthalpy and entropy follows.

4-18. Enthalpy. Enthalpy is a calculated property of matter which is sometimes loosely defined as "total heat content." More specifically, the enthalpy H of a given mass of material at any specified thermodynamic condition is an expression of the total heat which must be transferred to the material to bring the material to the specified condition from some initial condition arbitrarily taken as the zero point of enthalpy.

Whereas the total enthalpy H represents the enthalpy of M pounds, the specific enthalpy h is the enthalpy of 1 lb. Since it is usually the specific enthalpy rather than the total enthalpy which is of interest, hereafter in this text the term enthalpy shall be used to mean specific enthalpy, h, the enthalpy of 1 lb.

Since little is known about the specific heat or the other properties of materials at low temperatures, it is not possible to determine absolute values for the calculated properties. For this reason, values for the calculated properties must be determined from some arbitrarily selected zero point rather than from absolute zero.* For example, the zero point of enthalpy for water and its vapor, steam, is taken as water at 32° F under atmospheric pressure. The enthalpy of 1 lb of water at 60° F then is the total amount of heat which must be transferred to the water in order to raise the temperature of the water from 32° F to 60° F. According to Equation 2-9, this is 28 Btu (1 × 1 × 28). Hence, based on the assumption that the enthalpy of water is zero at 32° F, the enthalpy of water at 60° F is 28 Btu/lb.

Mathematically, enthalpy is defined as

$$h = u + \frac{Pv}{J} \tag{4-1}$$

* Since it is required to know the change in the enthalpy of the working fluid during a process, rather than the absolute enthalpy at some particular condition, the fact that absolute enthalpy cannot be calculated is of little consequence.

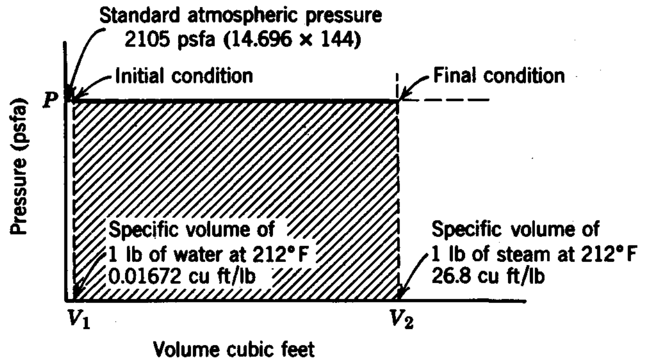


Fig. 4-7. Pressure-volume diagram showing the external work done by fluid expansion as 1 lb of water is vaporized at atmospheric pressure—approximately 59,000 ft-lb.

where h = the specific enthalpy in Btu/lb
u = the specific internal energy in Btu/lb
P = the pressure in psfa
v = the specific volume in cubic feet
J = the mechanical energy equivalent

It has been demonstrated (Section 3-12) that all the heat transferred to a fluid is not necessarily stored in the fluid as an increase in the internal energy of the fluid. In many cases, some part or all or the heat transferred to the fluid passes through the fluid and leaves the fluid as work. In Equation 4-1, that part of the transferred energy which is stored in the fluid as an increase in the internal energy is represented by the term u, whereas that part of the transferred heat which leaves the fluid as work is represented by the term Pv/J. Notice that, although the energy represented by the term Pv/J, does not increase the internal energy of the fluid and is not stored in the fluid, it nevertheless represents energy which must be transferred to the fluid in order to bring the fluid to the specified condition from the initial condition at the arbitrarily selected zero point of enthalpy. Furthermore, even though the external work energy is not stored in the fluid, it must pass back through the fluid and be given up by the fluid as the fluid returns to the initial condition.

Consider, for example, the vaporization of 1 lb of water into steam at 212° F under atmospheric pressure. The volume of 1 lb of water at 212° F is 0.01670 cu ft whereas the volume of 1 lb of steam at 212° F is 26.82 cu ft. Hence, the fluid expands from a volume of 0.0167 cu ft to a volume of 26.82 cu ft during the vaporization thereby doing work in expanding against the pressure of the atmosphere.

The enthalpy of vaporization (latent heat of vaporization) of water at 212° F is 970.4 Btu. Of this amount, only 897.6 Btu actually increases the internal energy and represents energy in storage in the vapor. The other 72.8 Btu leaves the vapor as the work of expansion and is represented by the term Pv/J. A PV diagram of the vaporization process is shown in Fig. 4-7.

4-19. Entropy. Entropy, like enthalpy, is a calculated property of matter. The entropy S of a given mass of material at any specified condition is an expression of the total heat transferred to the material per degree of absolute temperature to bring the material to that condition from some initial condition taken as the zero of entropy.

Since it is not possible to calculate the absolute value of entropy, entropy values, like those of enthalpy, are based on an arbitrarily selected zero point. The zero points of entropy and enthalpy are the same for any one fluid. Hence, for water and its vapor, steam, the zero point of entropy is taken as water at 32° F.

Again, as in the case of enthalpy, it is the specific entropy s rather than the total entropy S which is useful. Therefore, in this book, the term entropy shall be used to mean specific entropy s rather than the total entropy S.

It has been shown (Section 3-22) that the mechanical energy or work of a process can be expressed as the product of the change in volume and the average absolute pressure. Likewise, it is often convenient to express the heat energy transferred during a process as the product of two factors. The concept of entropy makes this possible. The heat energy transferred during a process can be expressed as the product of the change in entropy and the

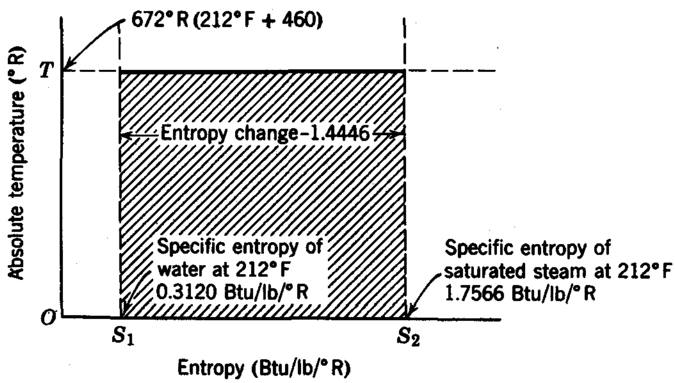


Fig. 4-8

average absolute temperature.* Mathematically the relationship is expressed by the following equations:

$$\Delta Q = \Delta s \times T_m \quad (4\text{-}2)$$

$$\Delta s = \frac{\Delta Q}{T_m} \quad (4\text{-}3)$$

$$T_m = \frac{\Delta Q}{\Delta s} \quad (4\text{-}4)$$

where ΔQ = the heat energy transferred in Btu
Δs = the change in entropy in Btu per pound per °R
T_m = the average absolute temperature in °R

On a pressure-volume diagram (Fig. 4-7), the "area under the curve," which is the product of the change in volume and the average absolute pressure, represents the work of the process. Similarly, on a temperature-entropy diagram (temperature plotted against entropy), the "area under the curve," which is the product of the entropy change and the average absolute temperature, represents the heat transferred during the process (Fig. 4-8).

Although the mathematical treatment of entropy is not required in the study of refrigeration and is beyond the scope of this book, it is important to note that according to Equation 4-2 the entropy changes only when heat is transferred during the process. If there is no heat energy transfer, there is no change in the entropy. The heat energy transfer may occur either to or from an external source or sink or it may take place entirely within the fluid itself as a result of internal friction. However, the entropy of a fluid is not affected by external work done either by or on the fluid. Thus in a frictionless (occurring without either internal or external friction), adiabatic (no heat transfer to or from an external body) compression, as in the ideal compression of the refrigerant vapor in a refrigeration compressor, the entropy of the fluid will remain the same or constant.

* The average absolute temperature is not merely the mean of the initial and final temperatures of the process, but is the average of all of the absolute temperatures through which the process passes.

4-20. Vapor Tables. It has been stated previously that a vapor does not approach the condition of an ideal gas because of the intermolecular forces which exist between the molecules of the vapor. Therefore, internal friction is present whenever a vapor undergoes a change of condition so that the various properties of a vapor at the different conditions cannot be determined by applying the laws of ideal gases.

The properties of vapors at various conditions have been determined by experiment for all common vapors and these data are published in the form of tables. Separate tables are used for saturated and superheated vapors.

4-21. Saturated Vapor Tables. Saturated vapor tables (Fig. 4-9) deal only with saturated liquids and vapors, and usually give values for the following properties: (1) temperature, (2) pressure, (3) specific volume, (4) enthalpy (specific), and (5) entropy (specific). Normally, the temperature in degrees Fahrenheit is listed in the extreme left-hand column. The pressure is given in the second and third columns, followed by the specific volume in cubic feet for both the liquid and the vapor in the fourth and fifth columns, respectively. Some tables list the density in addition to or in place of the

Properties of Saturated Steam

Temp., °F, t (1)	Absolute Pressure: Psi, p (2)	Absolute Pressure: In. Hg, p (3)	Specific Volume: Sat. liquid, v_f (4)	Specific Volume: Evap., v_{fg} (5)	Specific Volume: Sat. vapor, v_g (6)	Enthalpy: Sat. liquid, h_f (7)	Enthalpy: Evap., h_{fg} (8)	Enthalpy: Sat. vapor, h_g (9)	Entropy: Sat. liquid, S_f (10)	Entropy: Evap., S_{fg} (11)	Entropy: Sat. vapor, S_g (12)
200	11.526	23.467	0.01663	33.62	33.64	167.99	977.9	1145.9	0.2938	1.4824	1.7762
202	12.011	24.455	0.01665	32.35	32.37	170.00	976.6	1146.6	0.2969	1.4760	1.7729
204	12.512	25.475	0.01666	31.14	31.15	172.02	975.4	1147.4	0.2999	1.4697	1.7696
206	13.031	26.531	0.01667	29.97	29.99	174.03	974.2	1148.2	0.3029	1.4634	1.7663
208	13.568	27.625	0.01669	28.86	28.88	176.04	972.9	1148.9	0.3059	1.4571	1.7630
210	14.123	28.755	0.01670	27.80	27.82	178.05	971.6	1149.7	0.3090	1.4508	1.7598
212	14.696	29.922	0.01672	26.78	26.80	180.07	970.3	1150.4	0.3120	1.4446	1.7566
214	15.289	31.129	0.01673	25.81	25.83	182.08	969.0	1151.1	0.3149	1.4385	1.7534
216	15.901	32.375	0.01674	24.88	24.90	184.10	967.8	1151.9	0.3179	1.4323	1.7502
218	16.533	33.662	0.01676	23.99	24.01	186.11	966.5	1152.6	0.3209	1.4262	1.7471
220	17.186	34.992	0.01677	23.13	23.15	188.13	965.2	1153.4	0.3239	1.4201	1.7440
222	17.861	36.365	0.01679	22.31	22.33	190.15	963.9	1154.1	0.3268	1.4141	1.7409
224	18.557	37.782	0.01680	21.53	21.55	192.17	962.6	1154.8	0.3298	1.4080	1.7378
226	19.275	39.244	0.01682	20.78	20.79	194.18	961.3	1155.5	0.3328	1.4020	1.7348
228	20.016	40.753	0.01683	20.06	20.07	196.20	960.1	1156.3	0.3357	1.3961	1.7318
230	20.780	42.308	0.01684	19.365	19.382	198.23	958.8	1157.0	0.3387	1.3901	1.7288
240	24.969	50.837	0.01692	16.306	16.323	208.34	952.2	1160.5	0.3531	1.3609	1.7140
250	29.825	60.725	0.01700	13.804	13.821	218.48	945.5	1164.0	0.3675	1.3323	1.6998
260	35.429	72.134	0.01709	11.746	11.763	228.64	938.7	1167.3	0.3817	1.3043	1.6860
270	41.858	85.225	0.01717	10.044	10.061	238.84	931.8	1170.6	0.3958	1.2769	1.6727

Fig. 4-9. Excerpt from typical saturated vapor table.

Reproduced from *Thermodynamic Properties of Steam* by Keenan and Keyes, published by John Wiley and Sons, 1936, with permission.

specific volume. If the density only is given and the specific volume is wanted, the specific volume is determined by dividing the density into one. Likewise, when the specific volume is given and the density is wanted, the density is found by dividing the specific volume into one (Section 3-4).

Three values for enthalpy h are usually given in the saturated vapor tables: (1) the enthalpy of the liquid (h_f), which is the heat required to raise the temperature of the liquid from the temperature at the assumed zero point of enthalpy to the saturation temperature corresponding to the pressure of the liquid; (2) the enthalpy of vaporization (h_{fg}), which is the latent heat of vaporization at the pressure and temperature indicated; and (3) the enthalpy of the vapor (h_g), which is the sum of the enthalpy of the liquid (h_f) and the enthalpy of vaporization (h_{fg}). For example, the enthalpy of the liquid (h_f) for water at 212° F under atmospheric pressure is 180 Btu ($1 \times 1 \times 180$), whereas the enthalpy of the saturated water vapor at 212° F under atmospheric pressure is 1050 Btu, which is the sum of the enthalpy of the liquid (180 Btu) and the enthalpy of vaporization (970 Btu).

Two values of entropy are usually given: s_f, the entropy of the liquid and s_g, the entropy of the vapor, the difference between the two being the change in entropy during vaporization.

Dichlorodifluoromethane (Refrigerant-12)
Properties of Superheated Vapor

Temp. °F	Abs. Pressure 36 lb/in.² Gage Pressure 21.3 lb/in.² (Sat. Temp. 20.4° F)			Abs. Pressure 38 lb/in.² Gage Pressure 23.3 lb/in.² (Sat. Temp. 23.2° F)			Abs. Pressure 40 lb/in.² Gage Pressure 25.3 lb/in.² (Sat. Temp. 25.9° F)			Abs. Pressure 42 lb/in.² Gage Pressure 27.3 lb/in.² (Sat. Temp. 28.5° F)		
t	*V*	*H*	*S*	*V*	*H*	*S*	*V*	*H*	*S*	*V*	*H*	*S*
(at sat'n)	*(1.113)*	*(80.54)*	*(0.16947)*	*(1.058)*	*(80.86)*	*(0.16931)*	*(1.009)*	*(81.16)*	*(0.16914)*	*(0.963)*	*(81.44)*	*(0.16897)*
30	1.140	81.90	0.17227	1.076	81.82	0.17126	1.019	81.76	0.17030	0.967	81.65	0.16939
40	1.168	83.35	0.17518	1.103	83.27	0.17418	1.044	83.20	0.17322	0.991	83.10	0.17231
50	1.196	84.81	0.17806	1.129	84.72	0.17706	1.070	84.65	0.17612	1.016	84.56	0.17521
60	1.223	86.27	0.18089	1.156	86.19	0.17991	1.095	86.11	0.17896	1.040	86.03	0.17806
70	1.250	87.74	0.18369	1.182	87.67	0.18272	1.120	87.60	0.18178	1.063	87.51	0.18086
80	1.278	89.22	0.18647	1.208	89.16	0.18551	1.144	89.09	0.18455	1.087	89.00	0.18365
90	1.305	90.71	0.18921	1.234	90.66	0.18826	1.169	90.58	0.18731	1.110	90.50	0.18640
100	1.332	92.22	0.19193	1.260	92.17	0.19096	1.194	92.09	0.19004	1.134	92.01	0.18913
110	1.359	93.75	0.19462	1.285	93.69	0.19365	1.218	93.62	0.19272	1.158	93.54	0.19184
120	1.386	95.28	0.19729	1.310	95.22	0.19631	1.242	95.15	0.19538	1.181	95.09	0.19451
130	1.412	96.82	0.19991	1.336	96.76	0.19895	1.267	96.70	0.19803	1.204	96.64	0.19714
140	1.439	98.37	0.20254	1.361	98.32	0.20157	1.291	98.26	0.20066	1.227	98.20	0.19979
150	1.465	99.93	0.20512	1.387	99.89	0.20416	1.315	99.83	0.20325	1.250	99.77	0.20237
160	1.492	101.51	0.20770	1.412	101.47	0.20673	1.340	101.42	0.20583	1.274	101.36	0.20496
170	1.518	103.11	0.21024	1.437	103.07	0.20929	1.364	103.02	0.20838	1.297	102.96	0.20751
180	1.545	104.72	0.21278	1.462	104.67	0.21183	1.388	104.63	0.21092	1.320	104.57	0.21005
190	1.571	106.34	0.21528	1.487	106.29	0.21433	1.412	106.25	0.21343	1.343	106.19	0.21256
200	1.597	107.97	0.21778	1.512	107.93	0.21681	1.435	107.88	0.21592	1.365	107.82	0.21505
210	1.623	109.61	0.22024	1.537	109.57	0.21928	1.459	109.52	0.21840	1.388	109.47	0.21754
220	1.650	111.27	0.22270	1.562	111.22	0.22176	1.482	111.17	0.22085	1.411	111.12	0.22000
230	1.676	112.94	0.22513	1.587	112.89	0.22419	1.506	112.84	0.22329	1.434	112.80	0.22244
240	1.702	114.62	0.22756	1.612	114.58	0.22662	1.530	114.52	0.22572	1.457	114.49	0.22486
250	1.728	116.31	0.22996	1.637	116.28	0.22903	1.554	116.21	0.22813	1.480	116.19	0.22728
260	1.754	118.02	0.23235	1.662	117.99	0.23142	1.577	117.92	0.23052	1.502	117.90	0.22967
270	1.780	119.74	0.23472	1.687	119.71	0.23379	1.601	119.65	0.23289	1.524	119.62	0.23204
280	1.807	121.47	0.23708	1.712	121.45	0.23616	1.625	121.40	0.23526	1.547	121.36	0.23441
290	1.833	123.22	0.23942	1.737	123.20	0.23850	1.649	123.15	0.23760	1.570	123.11	0.23675
300	......	......	......	1.762	124.95	0.24083	1.673	124.92	0.23994	1.592	124.87	0.23909

Fig. 4-10. Excerpt from typical superheated vapor table.

It has been stated previously that the condition of a gas or a vapor can be determined when any two of its properties are known. However, for a saturated liquid or vapor at any one pressure, there is only one temperature that the fluid can have and still satisfy the conditions of saturation. This is true also for the other properties of a saturated liquid or vapor. Therefore, if any one property of a saturated liquid or vapor is known, the value of the other properties can be read directly from the saturated vapor table. For instance, assume that the pressure of one pound of dry saturated steam is 20 psia. By locating 20 psia (encircled) in the second column of the abbreviated table in Fig. 4-9 and reading across the table, the values (set off by the heavy lines) for all the other properties of the vapor at this condition can be obtained.

4-22. Superheated Vapor Tables. A superheated vapor table deals with the properties of a superheated vapor rather than those of a saturated vapor, and the arrangement of a superheated vapor table is somewhat different from that of a saturated vapor table. One common form of the superheated vapor table is illustrated in Fig. 4-10.

Before examining the superheated vapor table, it is important to take note of one significant difference between a saturated and a superheated vapor. Whereas, for a saturated vapor at any one pressure there is only one temperature which will satisfy the conditions of saturation, a superheated vapor may have any temperature above the saturation temperature corresponding to its pressure. The specific volume, enthalpy, and entropy of a superheated vapor at any one pressure will vary with the temperature. This does not mean that the properties of a superheated vapor are entirely independent of the pressure of the vapor but only that the properties of the superheated vapor at any one pressure will vary with the temperature. As a matter of fact, superheated vapor tables are based on the pressure of the vapor, and before the properties of a superheated vapor can be determined from a table, the pressure of the vapor or one of the properties of the vapor at saturation must be known. When one of the properties of the vapor at saturation is known, the pressure of the vapor can be found by consulting a saturated vapor table.

In addition to the properties of the superheated vapor at various temperatures above the saturation temperature corresponding to the pressure, superheated vapor tables usually list some or all of the properties of the vapor at the saturation temperature. For example, in Fig. 4-10, the absolute and gage pressures, along with the saturation temperature corresponding to these pressures, are given at the head of the table. The first readings in the body of the table (italicized) lists the specific volume, the enthalpy, and the entropy of the vapor at saturation. The specific volume, enthalpy, and entropy of the superheated vapor at various temperatures above the saturation temperature make up the body of the table. Notice that the temperature of the superheated vapor, given in the extreme left-hand column, is listed in 10° F increments.

Example 4-1. One pound of superheated Refrigerant-12 vapor is at a temperature of 50° F and its pressure is 40 psia. From the abbreviated table in Fig. 4-10, determine:

(*a*) The temperature, volume, enthalpy, and entropy of the vapor at saturation
(*b*) The volume, enthalpy, and entropy of the vapor at the superheated condition
(*c*) The degree of superheat of the vapor in degrees Fahrenheit
(*d*) The amount of superheat in the vapor in Btu
(*e*) The change in the volume during the superheating
(*f*) The change in entropy during the superheating

Solution

(*a*) From the head of the table, the saturation temperature corresponding to 40 psia	= 25.9° F
From the body of the table (first reading, italicized), the specific volume of the vapor at saturation	= 1.009 cu ft/lb
The enthalpy of the vapor at saturation	= 81.16 Btu/lb
The entropy of the vapor at saturation	= 0.16914 Btu/lb/° R
(*b*) From the body of the table, the properties of the vapor superheated to 50° F (offset by heavy lines in Fig. 4-10) the specific volume	= 1.070 cu ft/lb
The enthalpy	= 84.65 Btu/lb
The entropy	= 0.17612 Btu/lb/° R
(*c*) The superheated temperature	= 50.0° F
The temperature at saturation	= 25.9° F
The degree of superheat of the vapor in degrees Fahrenheit	= 24.1° F
(*d*) The enthalpy of the superheated vapor	= 84.65 Btu/lb
The enthalpy of the vapor at saturation	= 81.16 Btu/lb
The amount of superheat in the vapor in Btu	= 3.49 Btu/lb

(*e*) The entropy of the superheated vapor = 0.17612 Btu/lb/° R
The entropy of the vapor at saturation = 0.16914 Btu/lb/° R
The change in entropy during the superheating = 0.00698 Btu/lb/° R

(*f*) The volume of the superheated vapor = 1.070 cu ft/lb
The volume of the vapor at saturation = 1.009 cu ft/lb
The change in volume during the superheating = 0.061 cu ft/lb

5

Psychrometric Properties of Air

5-1. Composition of Air. Air is a mechanical mixture of gases and water vapor. Dry air (air without water vapor) is composed chiefly of nitrogen (approximately 78% by volume) and oxygen (approximately 21%), the remaining 1% being made up of carbon dioxide and minute quantities of other gases, such as hydrogen, helium, neon, argon, etc. With regard to these dry air components, the composition of the air is practically the same everywhere. On the other hand, the amount of water vapor in the air varies greatly with the particular locality and with the weather conditions. Since the water vapor in the air results primarily from the evaporation of water from the surface of various bodies of water, atmospheric humidity (water vapor content) is greatest in regions located near large bodies of water and is less in the more arid regions.

Since all air in the natural state contains a certain amount of water vapor, no such thing as "dry air" actually exists. Nevertheless, the concept of "dry air" is a very useful one in that it greatly simplifies psychrometric calculations. Hereafter in this book the term "dry air" is used to denote air without water vapor, whereas the term "air" is used to mean the natural mixture of "dry air" and water vapor.

5-2. Air Quantities. Air quantities may be stated either in units of volume (cubic feet) or in units of weight (pounds) so that the need for converting air quantities from one unit of measure to the other occurs frequently.

The volume occupied by any given weight of air depends upon the pressure and temperature of the air, and varies inversely with the barometric pressure and directly with the absolute temperature. Air very nearly approaches the condition of an ideal gas and will follow the gas laws with sufficient accuracy for all practical purposes. Therefore, the volume occupied by any given weight of air at any given pressure and temperature can be determined by applying Equation 3-10.

Example 5-1. Determine the volume occupied by 1 lb of air having a temperature of 70° F at standard sea level pressure (14.7 psia).

Solution. Rearranging and applying Equation 3-10,

$$V = \frac{M \times R \times T}{P}$$

$$= \frac{1 \times 53.3 \times (70 + 460)}{14.7 \times 144}$$

$$= 13.34 \text{ cu ft}$$

Example 5-2. Determine the volume of the air in Example 5-1 if the barometric pressure is 12.6 psia.

Solution. Applying Equation 3-10,

$$V = \frac{1 \times 53.3 \times (70 + 460)}{12.6 \times 144}$$

$$= 15.57 \text{ cu ft}$$

Example 5-3. Determine the volume of the air in Example 5-1 if the temperature of the air is 100° F.

Solution. Applying Equation 3-10,

$$V = \frac{1 \times 53.3 \times (100 + 460)}{14.7 \times 144}$$

$$= 14.10 \text{ cu ft}$$

The relationship between the volume and the weight of a given quantity of air at any condition is expressed by the following equations:

$$V = M \times v \tag{5-1}$$

$$M = \frac{V}{v} \tag{5-2}$$

where M = the weight of air in pounds

V = the volume of M pounds of air in cubic feet

v = the specific volume of the air in cubic feet per pound

Example 5-4. Air at a temperature of 95° F is circulated over a cooling coil at the rate of 2000 cu ft/min (cfm). If the specific volume of the air is 14.38 cu ft/lb, determine the weight of air passing over the coil in pounds per hour.

Solution. Applying Equation 5-2, the weight of air passing over the cooling coil
$$M = \frac{2000}{14.38} = 139.2 \text{ lb/min}$$

Multiplying by 60 min
$$M = 139.1 \times 60 = 8346 \text{ lb/hr}$$

5-3. Standard Air. Because of the difference in the volume of any given weight of air at various temperatures and pressures, an air standard has been established for use in the rating of air handling equipment so that all equipment is rated at equal conditions. Dry air having a specific volume of 13.34 cu ft per pound or a density of 0.07496 (0.075) lb per cu ft (1/13.34) is defined as standard air. Air at a temperature of 70° F and at standard sea level pressure has this specific volume and density (see Example 5-1).

A given volume of air at any condition can be converted to an equivalent volume of standard air by applying the following equation:

$$V_s = V_a \times \frac{v_a}{v_s} \quad (5\text{-}3)$$

where V_s = the equivalent volume of standard air
V_a = the actual volume of the air at any given condition
v_a = the specific volume of the air at the given condition
v_s = the specific volume of standard air (13.34 cu ft/lb)

Example 5-5. For the air in Example 5-4, determine the equivalent volume of standard air.

Solution. Applying Equation 5-3, the equivalent volume of standard air V_s
$$= \frac{2000 \times 14.38}{13.34} = 2155 \text{ cfm}$$

5-4. Dalton's Law of Partial Pressure. Dalton's law of partial pressures states in effect that in any mechanical mixture of gases and vapors (those which do not combine chemically): (1) each gas or vapor in the mixture exerts an individual partial pressure which is equal to the pressure that the gas would exert if it occupied the space alone and (2) the total pressure of the gaseous mixture is equal to the sum of the partial pressures exerted by the individual gases or vapors.

Air, being a mechanical mixture of gases and water vapor, obeys Dalton's law. Therefore, the total barometric pressure is always equal to the sum of the partial pressures of the dry gases and the partial pressure of the water vapor. Since psychrometry is the study of the properties of air as affected by the water vapor content, the individual partial pressures exerted by the dry gases are unimportant and, for all practical purposes, the total barometric pressure may be considered to be the sum of only two pressures: (1) the partial pressure exerted by the dry gases and (2) the partial pressure exerted by the water vapor.

5-5. Dew Point Temperature. It is important to recognize that the water vapor in the air is actually steam at low pressure and that this low pressure steam, like high pressure steam will be in a saturated condition when its temperature is the saturation temperature corresponding to its pressure. Since all of the components in a gaseous mixture are at the same temperature, it follows that when air is at any temperature above the saturation temperature corresponding to the partial pressure exerted by the water vapor the water vapor in the air will be superheated. On the other hand, when air is at a temperature equal to the saturation temperature corresponding to the partial pressure of the water vapor, the water vapor in the air is saturated and the air is said to be saturated (actually it is only the water vapor which is saturated). The temperature at which the water vapor in the air is saturated is known as the dew point temperature of the air. Obviously, then, the dew point temperature of the air is always the saturation temperature corresponding to the partial pressure exerted by the water vapor. Hence, when the partial pressure exerted by the water vapor is known, the dew point temperature of the air can be determined from the steam tables. Likewise, when the dew point temperature of the air is known, the partial pressure exerted by the water vapor can be determined from the steam tables.

Example 5-6. Assume that a certain quantity of air has a temperature of 80° F and that the partial pressure exerted by the water vapor in the air is 0.17811 psia. Determine the dew point temperature of the air.

Solution. From Table 4-1, the saturation temperature of steam corresponding to a pressure of 0.17811 psia is 50° F. Therefore, 50° F is the dew point temperature of the air.

Example 5-7. A certain quantity of air has a temperature of 80° F and a dew point temperature of 40° F. Determine the partial pressure exerted by the water vapor in the air.

Solution. From Table 4-1, the saturation pressure corresponding to 40° F is 0.12170 psia and therefore 0.12170 psia is the partial pressure exerted by the water vapor.

It has been shown (Section 4-5) that the pressure exerted by any vapor is directly proportional to the density (weight per unit volume) of the vapor. Since the dew point temperature of the air depends only on the partial pressure exerted by the water vapor, it follows that, for any given volume of air, the dew point temperature of the air depends only upon the weight of water vapor in the air. As long as the weight of water vapor in the air remains unchanged, the dew point temperature of the air will also remain unchanged. If the amount of water vapor in the air is increased or decreased, the dew point temperature of the air will also be increased or decreased, respectively. Increasing the amount of water vapor in the air will increase the pressure exerted by the water vapor and raise the dew point temperature. Likewise, reducing the amount of water vapor in the air will reduce the pressure of the water vapor and lower the dew point temperature.

5-6. Maximum Water Vapor Content. The maximum amount of water vapor that can be contained in any given volume of air depends only upon the temperature of the air. Since the amount of water vapor in the air determines the partial pressure exerted by the water vapor, it is evident that the air will contain the maximum amount of water vapor when the water vapor in the air exerts the maximum possible pressure. Since the maximum pressure that can be exerted by any vapor is the saturation pressure corresponding to its temperature, the air will contain the maximum weight of water vapor when the pressure exerted by the water vapor is equal to the saturation pressure corresponding to the temperature of the air. At this condition the temperature of the air and the dew point temperature will be one and the same and the air will be saturated. It is important to notice that the higher the temperature of the air, the higher is the maximum possible vapor pressure and the greater is the maximum possible water vapor content.

5-7. Absolute Humidity. The water vapor in the air is called humidity. The absolute humidity of the air at any given condition is defined as the actual weight of water vapor contained in 1 cu ft of air at that condition. Since the weight of water vapor contained in the air is relatively small, it is often measured in grains rather than in pounds (7000 grains equal 1 lb).

5-8. The Psychrometric Tables. It was shown in Section 5-5 that the actual weight of water vapor contained in a unit volume of air is solely a function of the dew point temperature of the air. Because of this fixed relationship between the dew point temperature and the absolute humidity of the air, when the value of one is known, the value of the other can be readily computed.

The absolute humidity of air at various dew point temperatures is listed in Tables 5-1 and 5-2. The dew point temperatures are listed in column (1) of the tables, and the absolute humidity corresponding to each of the dew point temperatures is given in columns (4) and (5). The values given in column (4) are in pounds of water vapor per cubic foot of air, whereas the values given in column (5) are in grains of water vapor per cubic foot of air. Too, the partial pressure (saturation pressure) of the vapor corresponding to each dew point temperature is given in inches of mercury in column (2) and in pounds per square inch in column (3).

5-9. Relative Humidity. Relative humidity (RH), expressed in percent, is the ratio of the actual weight of water vapor per cubic foot of air relative to the weight of water vapor contained in a cubic foot of saturated air at the same temperature, viz:

$$\text{Relative humidity} = \frac{\text{Actual weight of water vapor per cubic foot of air}}{\text{Weight of water vapor in 1 cu ft of saturated air at the same temperature}} \times 100 \quad (5\text{-}4)$$

For instance, if air at a certain temperature contains only half as much water vapor per cubic foot of air as the air could contain at that temperature if it were saturated, the relative humidity of the air is 50%. The relative humidity of saturated air, of course, is 100%.

Example 5-8. Air at a temperature of 80° F has a dew point temperature of 50° F. Determine the relative humidity.

Solution. From Table 5-2, absolute humidity corresponding to dew point temperature of 50° F	= 4.106 grains/cu ft
Absolute humidity of saturated air at 80° F	= 11.04 grains/cu ft
Applying Equation 5-4, the relative humidity of the air	$= \frac{4.106}{11.04} \times 100$
	= 37.1%

Example 5-9. Determine the relative humidity of the air in Example 5-8, if the air is cooled to 60° F. (Note: the dew point temperature of the air does not change because the moisture content does not change.)

Solution. From Table 5-2, absolute humidity corresponding to dew point of 50° F	= 4.106 grains/cu ft
Absolute humidity of saturated air at 60° F	= 5.795 grains/cu ft
Applying Equation 5-4, the relative humidity of the air	$= \frac{4.106}{5.795}$
	= 70.8%

5-10. Specific Humidity. The specific humidity is the actual weight of water vapor mixed with 1 lb of dry air and is usually stated in grains per pound, that is, grains of water vapor per pound of dry air. For any given barometric pressure, the specific humidity is a function of the dew point temperature alone. The specific humidity of air at various dew point temperatures is listed in Columns 6 and 7 of Tables 5-1 and 5-2. In Column 6, the specific humidity is given in pounds of water vapor per pound of dry air, whereas in Column 7 the specific humidity is given in grains of water vapor per pound of dry air. Since the specific humidity corresponding to any given dew point temperature varies with the total barometric pressure, the values given in Tables 5-1 and 5-2 apply only to air at standard barometric pressure.

The specific humidity of the air at any given dew point temperature increases as the total barometric pressure decreases and decreases as the total barometric pressure increases. The reason for this is easily explained. It has been shown (Examples 5-1 and 5-3) that the volume occupied by 1 lb of air increases as the total barometric pressure decreases. Since the density of a vapor varies inversely with the volume, it follows that the weight of water vapor required to produce a given vapor density and vapor pressure increases as the volume of the air increases. Likewise, as the volume occupied by 1 lb of air diminishes, the weight of water vapor required to produce a certain vapor density and vapor pressure also diminishes.

5-11. Percentage Humidity. Percentage humidity is defined as the ratio of the actual weight of water vapor in the air per pound of dry air to the weight of water vapor required to saturate completely 1 lb of dry air at the same temperature. Percentage humidity, like relative humidity, is given in percent. Notice, however, that percentage humidity is associated with the weight of water vapor per unit weight of air, whereas relative humidity is associated with the weight of water vapor per unit volume of air. For this reason the percentage humidity varies with the total barometric pressure, whereas relative humidity does not.

Example 5-10. Air at standard sea level pressure has a temperature of 80° F and a dew point temperature of 50° F. Determine the specific humidity and percentage humidity of the air.

Solution. From Table 5-2, the specific humidity of the air in grains per pound corresponding to a 50° F dew point temperature (Column 7)	= 53.38 grains/lb
Specific humidity of saturated air at 80° F (Column 7)	= 155.50 grains/lb
Percentage humidity	$= \frac{53.38}{155.50} \times 100$
	= 34.3%

NOTE. Compare this value with the relative humidity obtained in Example 5-8.

5-12. Dry Bulb and Wet Bulb Temperatures. The dry bulb (DB) temperature of the air is the temperature as measured by an ordinary dry bulb thermometer. When measuring the dry bulb temperature of the air, the bulb of the thermometer should be shaded to reduce the effects of direct radiation.

The wet bulb (WB) temperature of the air is the temperature as measured by a wet bulb thermometer. A wet bulb thermometer is an ordinary thermometer whose bulb is enclosed in a wetted cloth sac or wick. To obtain an accurate reading with a wet bulb thermometer, the wick should be wetted with clean water at approximately the dry bulb temperature of the air and the air velocity around the wick should be maintained between 1000 and 2000 ft per minute. As a practical matter, this velocity can be simulated in still air by whirling the thermometer about on the end of a chain. An instrument especially designed for this purpose is the sling psychrometer (Fig. 5-1). The sling psychrometer is made up of two thermometers, one dry bulb and one wet bulb, mounted side by side in a protective case which is attached to a handle by a swivel connection so that the case can be easily rotated about the hand. After saturating the wick with clean water, the instrument is whirled rapidly in the air for approximately one minute, after which time readings can be taken from both the dry bulb and wet bulb thermometers. The process should be repeated several times to assure that the lowest possible wet bulb temperature has been recorded.

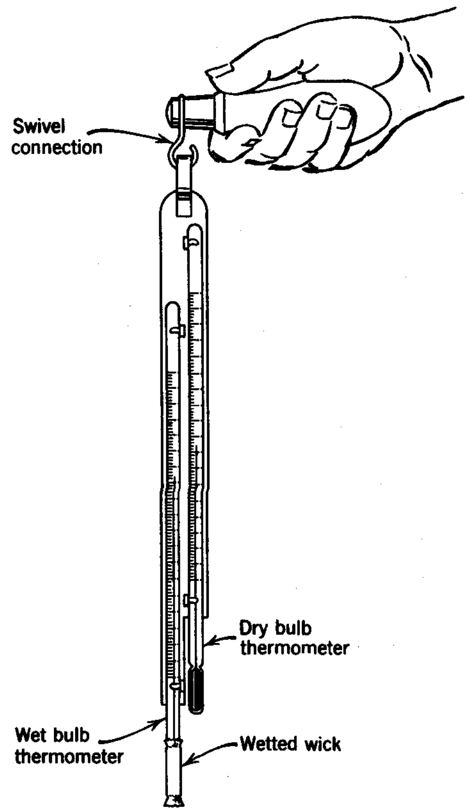


Fig. 5-1. Sling psychrometer.

Unless the air is 100% saturated, in which case the dry bulb, wet bulb, and dew point temperatures of the air will be one and the same, the temperature recorded by a wet bulb thermometer will always be lower than the dry bulb temperature of the air. The amount by which the wet bulb temperature is reduced below the dry bulb temperature depends upon the relative humidity of the air and is called the wet bulb depression.

Whereas a dry bulb thermometer, being unaffected by humidity, measures only the actual temperature of the air, a wet bulb thermometer, because of its wetted wick, is greatly influenced by the moisture in the air; thus a wet bulb temperature is in effect a measure of the relationship between the dry bulb temperature of the air and the moisture content of the air. In general, for any given dry bulb temperature, the lower the moisture content of the air, the lower is the wet bulb temperature. The reason for this is easily explained.

When unsaturated air is brought into contact with water, water will evaporate into the air at a rate proportional to the difference in pressure between the vapor pressure of the water and the vapor pressure of the air. Hence, when a wet bulb thermometer is whirled rapidly about in unsaturated air, water will evaporate from the wick, thereby cooling the water remaining in the wick (and the thermometer bulb) to some temperature below the dry bulb temperature of the air.

It is important to recognize the fact that the wet bulb temperature of the air is a measure of the relationship between the dry bulb and dew point temperatures of the air, and as such it provides a convenient means of determining the dew point temperature of the air when the dry bulb temperature is known. Too, it will be shown later that the wet bulb temperature is also an index of the total heat content of the air.

In order to understand why the wet bulb temperature is a measure of the relationship between the dry bulb and dew point temperatures, a knowledge of the theory of the wet bulb thermometer is required. When water evaporates from the wick of a wet bulb thermometer, heat must be supplied to furnish the latent heat of vaporization. Before the temperature of the water in the wick is reduced below the dry bulb temperature of the air, the source of the heat to vaporize the water is the water itself. Therefore, as water evaporates from the wick, the water remaining in the wick is cooled below the dry bulb temperature of the air. When this occurs, a temperature differential is established and heat begins to flow from the air to the wick. Under this condition, a part of the vaporization heat is being supplied by the air while the other part is supplied by the water in the wick. As the temperature of the wick continues to decrease, the temperature difference between the air and the wick increases progressively so that more and more of the vaporization heat is supplied by the air and less and less is supplied by the water in the wick. When the temperature of the wick is reduced to the point where the temperature difference between the air and the wick is such that the flow of heat from the air is sufficient to supply all of the vaporizing heat, the temperature of the wick will stabilize even though vaporization from the wick continues. The temperature at which the wick stabilizes is called the temperature of adiabatic saturation and is the wet bulb temperature of the air.

Through careful analysis of the foregoing, it can be seen that the wet bulb temperature depends upon both the dry bulb temperature and the amount of water vapor in the air. For example, the lower the relative humidity of the air, the greater is the rate of evaporation from the wick and the greater is the amount of heat required for vaporization. Obviously, the greater the need for heat, the greater is the wet bulb depression below the dry bulb temperature. Too, it follows also that the lower the dry bulb temperature, the lower the wet bulb temperature for any given wet bulb depression.

5-13. The Heat Content or Enthalpy of Air. Air has both sensible and latent heat, and the total heat content of the air at any condition is the sum of the sensible and latent heat contained therein.

The sensible heat of the air is a function of the dry bulb temperature. For any given dry bulb temperature, the sensible heat of the air is taken as the enthalpy of dry air at that temperature as calculated from 0° F. Air sensible heat at various temperatures is given in Btu per pound of dry air in Column 10 of Tables 5-1 and 5-2. With regard to Column 10, the temperatures listed in Column 1 are used as dry bulb temperatures.

Example 5-11. Using Table 5-2, determine the sensible heat in 10 lb of air at 80° F.

Solution. From Table 5-2, the sensible heat of 1 lb of air at 80° F	= 19.19 Btu/lb
For 10 lb of air, the sensible heat at 80° F	= 10 × 19.19 = 191.9 Btu

The quantity of sensible heat added or removed in heating or cooling a given weight of air through a given temperature range may be computed by applying Equation 2-8. The mean specific heat of air at constant pressure is 0.24 Btu/lb. (Although the specific heat of any vapor or gas varies somewhat with the temperature range, the use of a mean specific heat value is sufficiently accurate for all practical purposes.)

Example 5-12. Compute the quantity of sensible heat required to raise the temperature of 10 lb of air from 0° F to 80° F.

Solution. Applying Equation 2-8, Q_s	= 10 × 0.24 × (80 − 0) = 192 Btu
Alternate Solution. From Table 5-2, the sensible heat of 1 lb of air at 80° F	= 19.19 Btu/lb
Sensible heat of 1 lb of air at 0° F	= 0 Btu/lb
For 1 lb of air, Q_s	= 19.19 − 0 = 19.19 Btu/lb
For 10 lb of air, Q_s	= 10 × 19.19 = 191.9 Btu

Since all the components of dry air are noncondensable at normal temperatures and pressures, for all practical purposes the only latent heat in the air is the latent heat of the water vapor in the air. Therefore, the amount of latent heat in any given quantity of air depends upon the weight of water vapor in the air and upon the latent heat of vaporization of water corresponding to the saturation temperature of the water vapor.

Since the saturation temperature of the water vapor is the dew point temperature of the air, the dew point temperature determines not only the weight of water vapor in the air but also the value of the latent heat of vaporization. Hence, the latent heat content of the air is a function of the dew point temperature alone. As long as the dew point temperature of the air remains unchanged, the latent heat content of the air also remains unchanged.

The total heat content of water vapor at various temperatures as computed from 32° F is given in Btu per pound in Column 11 of Tables 5-1 and 5-2. Although the values given in Column 11 include the sensible heat of the liquid above 32° F as well as the latent heat of vaporization at the given temperature, common practice is to treat the entire heat content of the water vapor as latent heat.*

The latent heat content of any given quantity of air can be computed by multiplying the actual weight of water vapor in the air in pounds by the total heat of the water vapor as given in Column 11 of Tables 5-1 and 5-2.

Example 5-13. Compute the latent heat content of the air in Example 5-12, if the dew point temperature of the air is 50° F.

Solution. From Table 5-2, the actual weight of water vapor per pound of dry air (specific humidity) at 50° F DP (Column 6)	= 0.007626 lb
Total heat per pound of saturated water vapor at 50° F (Column 11)	= 1081.7 Btu/lb
Latent heat per pound of dry air at 50° F DP	= 0.007626 × 1081.7
	= 8.25 Btu/lb
For 10 lb of dry air, total latent heat	= 10 × 8.25
	= 82.5 Btu

Since the total heat of the air is the sum of the sensible and latent heat contained therein, the total heat of the air in Examples 5-12 and 5-13 is the sum of the sensible heat of the dry air, as computed in Example 5-12, and the latent heat of the water vapor mixed with the dry air, as computed in Example 5-13, viz:

Sensible heat of 10 lb of dry air at 70° F, from Example 5-12	= 191.9 Btu
Latent heat of water vapor mixed with 10 lb of dry air at 50° F DP, from Example 5-13	= 82.5 Btu
Total heat of 10 lb of air at 70° F DB and 50° F DP*	191.9 + 82.5
	= 274.4 Btu

5-14. Wet Bulb Temperature as a Measure of Total Heat. It has been shown in preceding sections that the sensible heat of the air (the heat content of the dry air) is a function of the dry bulb temperature and that the latent heat of the air (the heat content of the water vapor mixed with the dry air) is a function of the dew point temperature. Since, for any given combination of dry bulb and dew point temperatures, the wet bulb temperature of the air can have only one value, it is evident that the wet bulb temperature is an index of the total heat content of the air. However, it is important to recognize that although there is only one wet bulb temperature that will satisfy any given combination of dry bulb and dew point temperatures, there are many combinations of dry bulb and dew point temperatures which will have the same wet bulb temperature (see Fig. 5-2). This means in effect that different samples of air having the same wet

* Although the total heat of the air at any condition is the sum of the sensible and latent heat contained therein, as a practical matter it is more convenient to consider the total enthalpy of the air as being the sum of the enthalpy of the dry air and the enthalpy of the water vapor mixed with the dry air. Since the amount of sensible heat is comparatively small, the error which accrues from assuming all of the heat of the vapor to be latent heat is of no practical consequence.

* The actual weight of air involved is slightly in excess of 10 lb, being 10 lb of dry air plus the weight of water vapor (0.007626 lb) mixed with the dry air. Too, since the temperature of the water vapor is the same as that of the dry air (70° F), the water vapor contains a certain amount of superheat (50° F to 70° F) which is not included in the total heat. However, since both of these values are very small, the error incurred by neglecting them has no practical significance.

Temperature, ° F			Heat Content, Btu/lb		
Dry Bulb	Dew Point	Wet Bulb	Sensible	Latent	Total
60	60	60	14.39	11.98	26.37
65	57	60	15.59	10.78	26.37
70	53.5	60	16.79	9.58	26.37
75	50	60	17.99	8.38	26.37
80	45.5	60	19.19	7.18	26.37
85	40.5	60	20.39	5.98	26.37
90	34.5	60	21.59	4.78	26.37

Fig. 5-2

bulb temperature have the same total heat, even though the ratio of sensible to latent heat may be different for the different samples.

The values of total heat listed for various temperatures in Column 12 of Tables 5-1 and 5-2 are for 1 lb of saturated air at the temperature shown. However, since all samples of air, saturated or unsaturated, having the same wet bulb temperature have the same total heat, the values given in Column 12 will apply to any sample of air when the temperatures listed in Column 1 are used as wet bulb temperatures.

Example 5-14. If 100 lb of air having an initial wet bulb temperature of 78° F are cooled to a final wet bulb temperature of 60° F, determine the total heat removed from the air during the cooling process.

Solution. From Table 5-2, total heat of 1 lb of air corresponding to 78° F WB (Column 12)	= 63.05 Btu/lb
Total heat per pound of air at 60° F WB (Column 12)	= 26.37 Btu/lb
Total heat removed per pound of air, Q_t	= 63.05 − 26.37 = 36.68 Btu/lb
For 100 lb of air, the total heat removed, Q_t	= 100 × 36.68 = 366.8 Btu

5-15. Specific Volume of Air. It has already been shown that the volume occupied by a given weight of air depends upon the temperature of the air and upon the total barometric pressure. For standard sea level pressure, the volume of 1 lb of dry air at various temperatures is listed in Column 8 of Tables 5-1 and 5-2. The volume of 1 lb of saturated air (1 lb of dry air and the water vapor to saturate it) is listed for various temperatures in Column 9. When the relative humidity of the air is known, the specific volume of partially saturated air at any condition can be computed by applying these values in the following equation:

$$v_a = v_d + [(v_s - v_d) \times \%\text{RH}] \qquad (5\text{-}5)$$

where v_a = the specific volume of partially saturated air

v_d = the specific volume of dry air at the same temperature

v_s = the specific volume of saturated air at the same temperature

Example 5-15. Compute the specific volume of air at 95° F DB and 50% RH.

Solution. From Table 5-2, specific volume of dry air at 95° F (Column 8)	= 13.97 cu ft/lb
Specific volume of saturated air at 95° F (Column 9)	= 14.79 cu ft/lb
Applying Equation 5-5, v_a	= 13.97 + [(14.79 − 13.97) × 0.5] = 14.38 cu ft/lb

5-16. The Psychrometric Chart. Psychrometric charts (Fig. 5-3) are graphical representations of psychrometric data such as those contained in Tables 5-1 and 5-2. The use of psychrometric charts permits graphical analysis of psychrometric data and thereby facilitates the solution of many practical problems dealing with air which would otherwise require tedious mathematical calculation.

Basically, the psychrometric chart shows the relationship between four fundamental properties of air: (1) dry bulb temperature, (2) dew point temperature, (3) wet bulb temperature, and (4) relative humidity. When any two of these four properties are known, the other two can be determined directly from the psychrometric chart without using mathematical calculations.

The skeleton chart in Fig. 5-4 illustrates the general construction of the psychrometric chart which is based primarily upon the relationship that exists between the aforementioned four properties. Notice that the lines of dry bulb temperature are vertical while the lines of dew point temperature are horizontal. The lines of wet bulb temperature run diagonally across the chart as do the lines of constant volume. The curved lines are lines of constant relative humidity. The curved line bounding the chart on the left side is the line of 100% relative humidity and is called the saturation curve. Air at any condition such that its state can be identified by a point falling anywhere along the saturation curve is saturated air. Values for dry bulb, wet bulb, and dew point temperatures are read at the saturation curve. Values for dry bulb temperature are also given at the base of the chart. Notice that the dry bulb, wet bulb, and dew point temperatures for saturated air coincide. Values of specific volume and relative humidity are given along the lines of constant volume and relative humidity, respectively. Values of specific humidity and vapor pressure are given on the right and left margins of the chart. For any given air condition, the specific humidity and vapor pressure corresponding to the dew point temperature can be determined by following the dew point temperature line to the specific humidity and vapor pressure scales. The total heat corresponding to any wet bulb temperature is found by following the wet bulb lines to the total heat scale above the saturation curve. The following example will illustrate the use of the psychrometric chart.

Example 5-16. A certain quantity of air has a dry bulb temperature of 95° F and a wet bulb temperature of 77° F. From the psychrometric chart determine all of the following values: (1) dew point temperature, (2) specific humidity, (3) vapor pressure, (4) specific volume, (5) total heat, and (6) relative humidity.

Solution. Using the two known properties of the air as coordinates the condition of the air can be established as a point on the chart. Once this point has been established, the other properties of the air at this condition can be read directly from the chart as shown in Fig. 5-5, viz:

Dew point temperature	= 70° F
Specific humidity	= 110 grains/lb
Vapor pressure	= 0.37 psia
Specific volume	= 14.33 cu ft/lb
Total heat	= 40.5 Btu/lb
Relative humidity	= 45%

Example 5-17. For the air in Example 5-16, determine: (*a*) the sensible heat per pound of air and (*b*) the latent heat per pound of air.

Solution.

(*a*) From Table 5-2, enthalpy of 1 lb of dry air at 95° F DB, Q_s	= 22.80 Btu/lb
(*b*) From Example 5-16, total heat per pound of air, Q_t	= 40.50 Btu/lb
The latent heat per pound of air, Q_1	$= Q_t - Q_s$
	= 40.50 − 22.80
	= 17.7 Btu/lb

Example 5-18. If the air in Example 5-16 is cooled to 75° F, determine:

(*a*) The final dew point temperature
(*b*) The final wet bulb temperature
(*c*) The final relative humidity
(*d*) The final total heat per pound

Solution. Since the air is not cooled below the initial dew point temperature, no moisture is removed from the air. Therefore, the specific humidity, dew point temperature, and latent heat of the air remain unchanged. Hence, the initial dew point temperature and the new dry bulb temperature can be used as coordinates to locate the new condition of the air on the psychrometric chart (point *B* in Fig. 5-6). The following properties of the air at the new condition are taken from the psychrometric chart as indicated in Fig. 5-6:

(*a*) Wet bulb temperature	= 71.4° F
(*b*) Relative humidity	= 85%
(*c*) Total heat per pound	= 35.69 Btu/lb

Example 5-19. With respect to Fig. 5-6, in cooling the air from condition "A," as described in Example 5-16, to condition "B," as described in Example 5-18, compute:

(*a*) The total heat removed per pound of air
(*b*) The sensible heat removed per pound of air.

Solution.

(*a*) From Example 5-16, air total heat at *A*	= 40.50 Btu/lb
From Example 5-18, air total heat at *B*	= 35.69 Btu/lb
The total heat removed per pound of air in cooling from *A* to *B*	= 40.50 − 35.69
	= 4.81 Btu/lb

(*b*) Since there is no change in the latent heat of the air, the sensible heat removed per pound of air is equal to the total heat removed per pound of air.

Example 5-20. Assume that the air in Example 5-16 is cooled to 40° F and determine:

(*a*) The total heat removed per pound
(*b*) The sensible heat removed per pound
(*c*) The latent heat removed per pound.

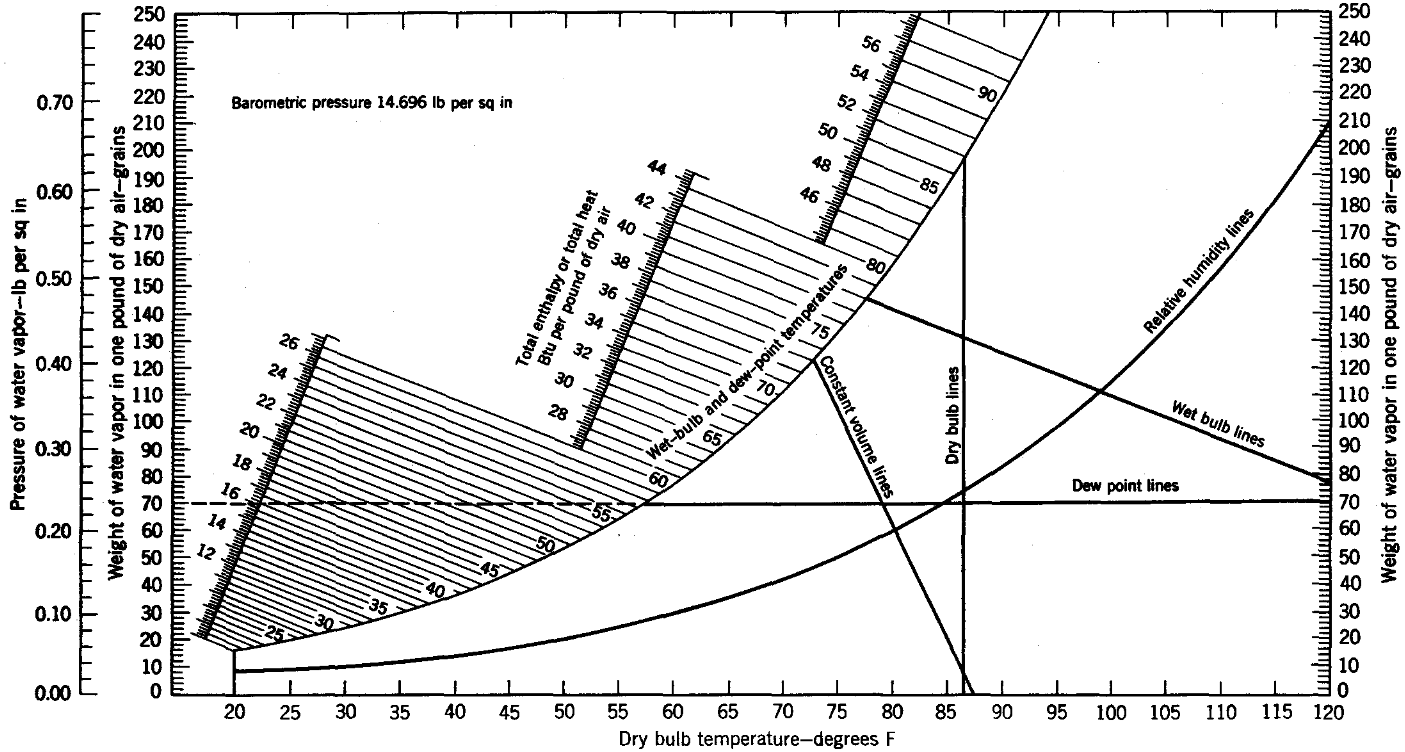


Fig. 5-4. Psychrometric chart. (Copyright 1942 by General Electric Company.)

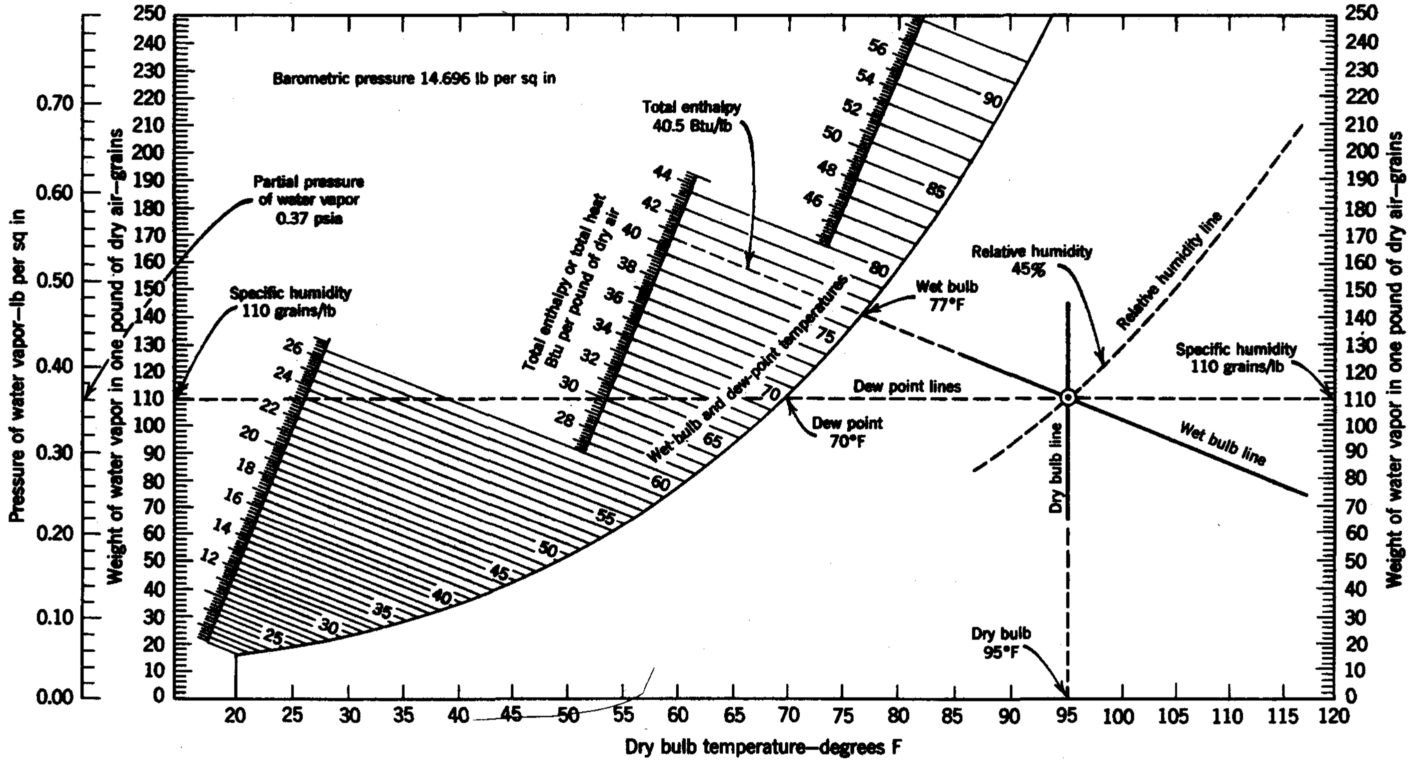


Fig. 5-5. Psychrometric chart. (Copyright 1942, by General Electric Company.)

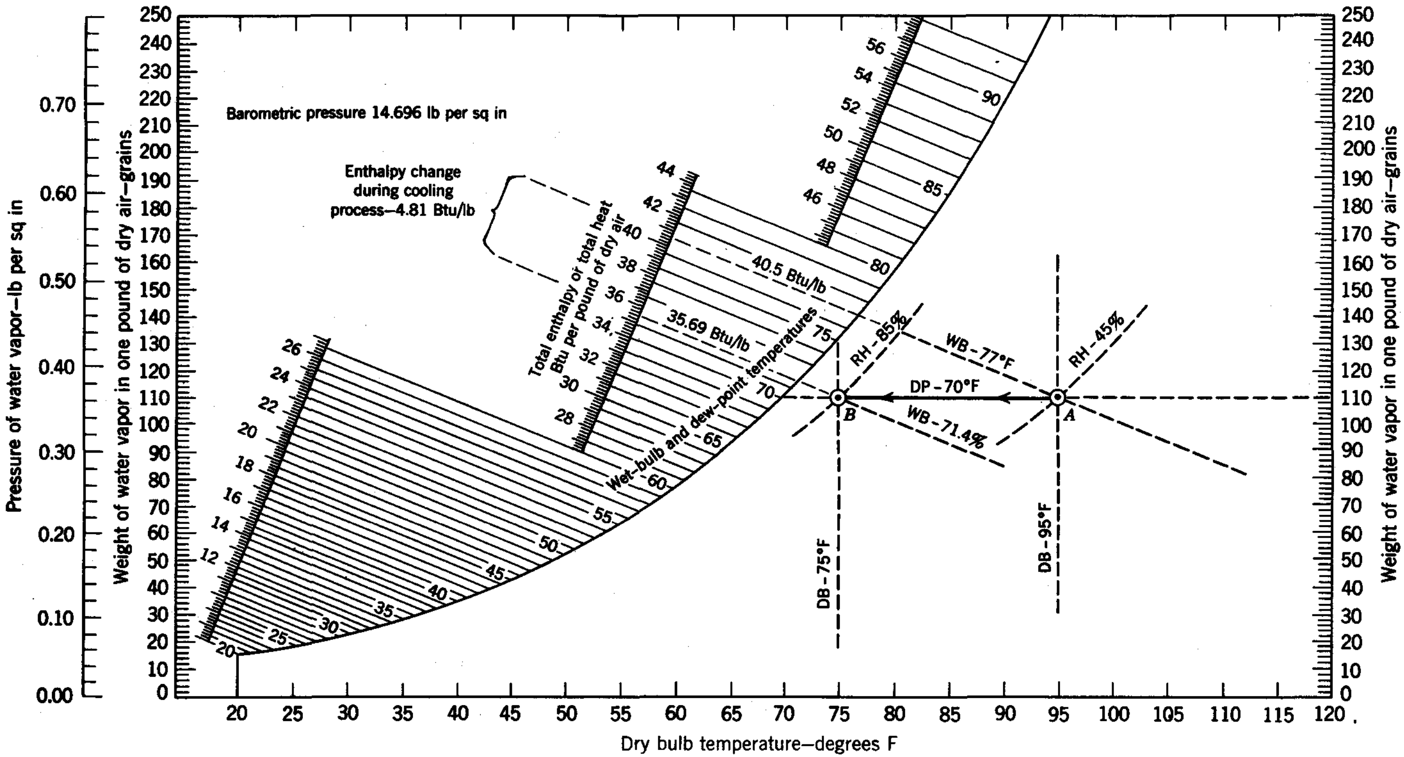


Fig. 5-6. Psychrometric chart. (Copyright 1942, General Electric Company.)

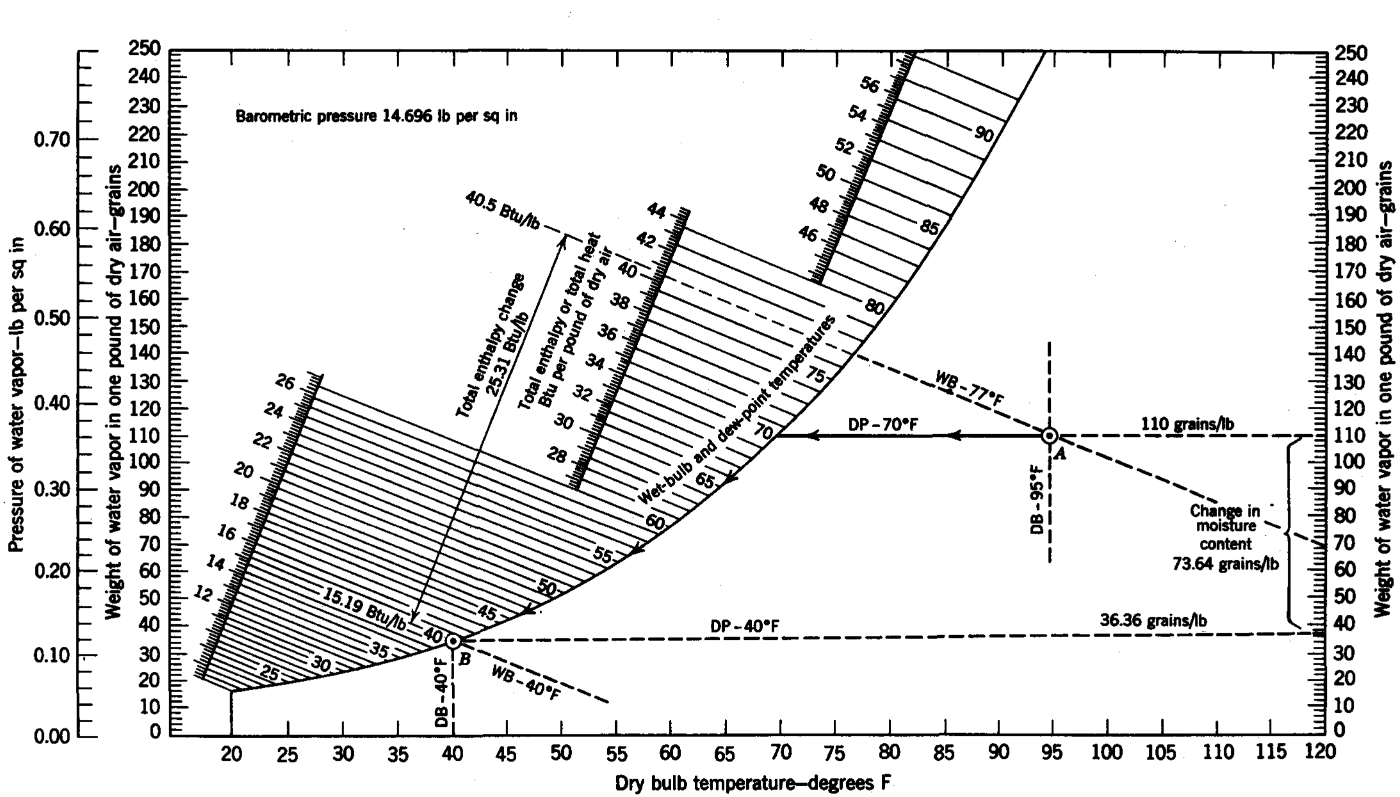


Fig. 5-7. Psychrometric chart. (Copyright 1942, by General Electric Company.)

Solution. Since the air is cooled below the dew point temperature, moisture will be condensed out of the air and the air at the final condition will be saturated. Therefore, the dry bulb temperature, the dew point temperature, and the wet bulb temperature will coincide at 40° F and the relative humidity of the air will be 100%. On the psychrometric chart, the condition of the air falls on the saturation curve at 40° F (point *B* in Fig. 5-7).

(*a*) From the psychrometric chart, the total heat of the air at the initial condition (point *A*)	= 40.50 Btu/lb
The total heat of the air at the final condition (point *B*)	= 15.19 Btu/lb
The total heat removed per pound	40.50 − 15.19 = 25.31 Btu/lb
(*b*) Applying Equation 2-8, the sensible heat removed per pound of air	= 1 × 0.24 × (95 − 40) = 13.2 Btu/lb
(*c*) The latent heat removed per pound of air	= $Q_t - Q_s$ = 25.31 − 13.20 = 12.11 Btu/lb

PROBLEMS

1. Determine the volume occupied by 1 lb of air having a temperature of 80° F at standard sea level pressure. *Ans.* 13.59 cu ft

2. Compute the volume of the air in Problem 1 if the barometric pressure is 13.5 psia. *Ans.* 14.80 cu ft

3. Determine the volume of the air in Problem 1 if the temperature of the air is 120° F. *Ans.* 14.60 cu ft

4. Air at a temperature of 90° F is circulated over a cooling coil at the rate of 1000 cu ft per min (cfm). If the specific volume of the air is 14.10 cu ft/lb, compute the weight of air passing over the coil in pounds per hour. *Ans.* 4255 lb/hr

5. Compute the equivalent volume of standard air for the conditions of Problem 4. *Ans.* 946 cfm

6. Compute the quantity of sensible heat required to raise the temperature of 10 lb of air from a temperature of 35° F to a temperature of 100° F. *Ans.* 156 Btu

7. If 80 lb of air having an initial wet bulb temperature of 80° F are cooled to a final wet bulb temperature of 65° F, determine the total heat removed from the air during the cooling process. *Ans.* 1085.6 Btu

8. A certain quantity of air has a dry bulb temperature of 90° F and a wet bulb temperature of 77° F. From the psychrometric chart determine all of the following values: (*a*) dew point temperature, (*b*) specific humidity, (*c*) vapor pressure, (*d*) specific volume, (*e*) total heat, and (*f*) relative humidity.
Ans. (*a*) 72.3° F; (*b*) 119.5 gpp; (*c*) 0.395 psia; (*d*) 14.23 cu ft/lb; (*e*) 40.5 Btu/lb; (*f*) 58%

9. Assume that the air in Problem 8 is cooled to 75° F and determine:
(*a*) the final dew point temperature *Ans.* 72.3° F
(*b*) the final wet bulb temperature *Ans.* 73° F
(*c*) the final relative humidity *Ans.* 92%
(*d*) the final total heat per pound of air *Ans.* 36.6 Btu/lb

10. Assume that the air in Problem 8 is cooled to 55° F and determine:
(*a*) the total heat removed per pound of air *Ans.* 17.2 Btu/lb
(*b*) the sensible heat removed per pound of air *Ans.* 8.40 Btu/lb
(*c*) the latent heat removed per pound of air *Ans.* 8.8 Btu/lb

6

Refrigeration and the Vapor Compression System

6-1. Refrigeration. In general, refrigeration is defined as any process of heat removal. More specifically, refrigeration is defined as that branch of science which deals with the process of reducing and maintaining the temperature of a space or material below the temperature of the surroundings.

To accomplish this, heat must be removed from the body being refrigerated and transferred to another body whose temperature is below that of the refrigerated body. Since the heat removed from the refrigerated body is transferred to another body, it is evident that refrigerating and heating are actually opposite ends of the same process. Often only the desired result distinguishes one from the other.

6-2. Need for Thermal Insulation. Since heat will always travel from a region of high temperature to a region of lower temperature, there is always a continuous flow of heat into the refrigerated region from the warmer surroundings. To limit the flow of heat into the refrigerated region to some practical minimum, it is usually necessary to isolate the region from its surroundings with a good heat insulating material.

6-3. The Heat Load. The rate at which heat must be removed from the refrigerated space or material in order to produce and maintain the desired temperature conditions is called the heat load. In most refrigerating applications the total heat load on the refrigerating equipment is the sum of the heat that leaks into the refrigerated space through the insulated walls, the heat that enters the space through door openings, and the heat that must be removed from the refrigerated product in order to reduce the temperature of the product to the space or storage conditions. Heat given off by people working in the refrigerated space and by motors, lights, and other electrical equipment also contributes to the load on the refrigerating equipment.

Methods of calculating the heat load are discussed in Chapter 10.

6-4. The Refrigerating Agent. In any refrigerating process the body employed as the heat absorber or cooling agent is called the refrigerant.

All cooling processes may be classified as either sensible or latent according to the effect the absorbed heat has upon the refrigerant. When the absorbed heat causes an increase in the temperature of the refrigerant, the cooling process is said to be sensible, whereas when the absorbed heat causes a change in the physical state of the refrigerant (either melting or vaporizing), the cooling process is said to be latent. With either process, if the refrigerating effect is to be continuous, the temperature of the refrigerating agent must be maintained continuously below that of the space or material being refrigerated.

To illustrate, assume that 1 lb of water at 32° F is placed in an open container inside an insulated space having an initial temperature of 70° F (Fig. 6-1). For a time, heat will flow from the 70° F space into the 32° F water and the temperature of the space will decrease. However, for each one Btu of heat that the water absorbs from the space, the temperature of the water will increase 1° F, so that as the temperature of the space decreases, the temperature of the water increases. Soon the temperatures of the water and the space will be exactly the same and no heat transfer will take place. Refrigeration will not be continuous because the temperature of the refrigerant does not remain below the temperature of the space being refrigerated.

Now assume that 1 lb of ice, also at 32° F, is substituted for the water (Fig. 6-2). This time

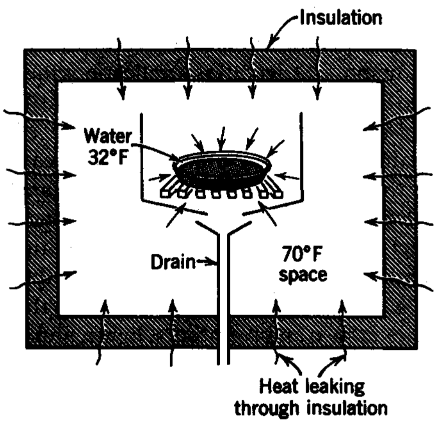


Fig. 6-1. Heat flows from warm space to cold water. Water temperature rises as space temperature decreases. Refrigeration will not be continuous.

the temperature of the refrigerant does not change as it absorbs heat from the space. The ice merely changes from the solid to the liquid state while its temperature remains constant at 32° F. The heat absorbed by the ice leaves the space in the water going out the drain and the refrigerating effect will be continuous until all the ice has melted.

It is both possible and practical to achieve continuous refrigeration with a sensible cooling process provided that the refrigerant is continuously chilled and recirculated through the refrigerated space as shown in Fig. 6-3.

Latent cooling may be accomplished with either solid or liquid refrigerants. The solid refrigerants most frequently employed are ice and solid carbon dioxide (dry ice). Ice, of course, melts into the liquid phase at 32° F, whereas solid carbon dioxide sublimes directly into the vapor phase at a temperature of −109° F under standard atmospheric pressure.

6-5. Ice Refrigeration. Melting ice has been used successfully for many years as a refrigerant. Not too many years ago ice was the only cooling agent available for use in domestic and small commercial refrigerators.

In a typical ice refrigerator (Fig. 6-4) the heat entering the refrigerated space from all the various sources reaches the melting ice primarily by convection currents set up in the air of the refrigerated space. The air in contact with the warm product and walls of the space is heated by heat conducted to it from these materials. As the air is warmed it expands and rises to the top of the space carrying the heat with it to the ice compartment. In passing over the ice the air is cooled as heat is conducted from the air to the ice. On cooling, the air becomes more dense and falls back into the storage space, whereupon it absorbs more heat and the cycling continues. The air in carrying the heat from the warm walls and stored product to the melting ice acts as a heat transfer agent.

To insure adequate air circulation within the refrigerated space, the ice should be located near the top of the refrigerator and proper baffling should be installed to provide direct and unrestricted paths of air flow. A drip pan must be located beneath the ice to collect the water which results from the melting.

Ice has certain disadvantages which tend to limit its usefulness as a refrigerant. For instance, with ice it is not possible to obtain the low temperatures required in many refrigeration applications. Ordinarily, 32° F is the minimum temperature obtainable through the melting of ice alone. In some cases, the melting temperature of the ice can be lowered to approximately 0° F by adding sodium chloride or calcium chloride to produce a freezing mixture.

Some of the other more obvious disadvantages of ice are the necessity of frequently replenishing the supply, a practice which is neither convenient nor economical, and the problem of disposing of the water resulting from the melting.

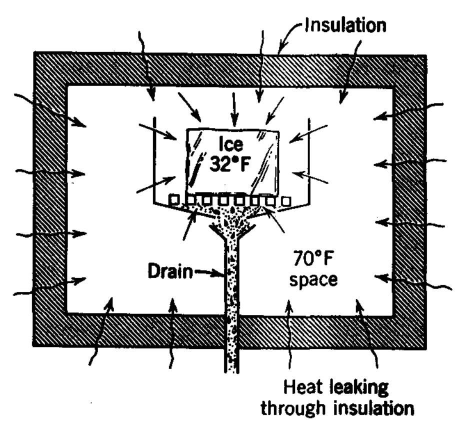


Fig. 6-2. Heat flows from warm space to cold ice. Temperature of space decreases as ice melts. Temperature of ice remains at 32° F. Heat absorbed by ice leaves space in water going out the drain.

Another less obvious, but more important, disadvantage of employing ice as a refrigerant is the difficulty experienced in controlling the rate of refrigeration, which in turn makes it difficult to maintain the desired low temperature level within the refrigerated space. Since the rate at which the ice absorbs heat is directly proportional to the surface area of the ice and to the temperature difference between the space temperature and the melting temperature of the ice, the rate of heat absorption by the ice diminishes as the surface area of the ice is diminished by the melting process. Naturally, when the refrigerating rate diminishes to the point that the heat is not being removed at the same rate that it is accumulating in the space from the various heat sources, the temperature of the space will increase.

Despite its disadvantages, ice is preferable to mechanical refrigeration in some applications. Fresh vegetables, fish, and poultry are often packed and shipped in cracked ice to prevent dehydration and to preserve appearance. Too, ice has tremendous eye appeal and can be used to considerable advantage in the displaying and serving of certain foods such as salads, cocktails, etc., and in chilling beverages.

6-6. Liquid Refrigerants. The ability of liquids to absorb enormous quantities of heat as they vaporize is the basis of the modern mechanical refrigerating system. As refrigerants, vaporizing liquids have a number of advantages over melting solids in that the vaporizing process is more easily controlled, that is, the refrigerating effect can be started and stopped at will, the rate of cooling can be predetermined within small limits, and the vaporizing temperature of the liquid can be governed by controlling the pressure at which the liquid vaporizes. Moreover, the vapor can be readily collected and condensed back into the liquid state so that the same liquid can be used over and over again to provide a continuous supply of liquid for vaporization.

Until now, in discussing the various properties of fluids, water, because of its familiarity, has been used in all examples. However, because of its relatively high saturation temperature, and for other reasons, water is not suitable for use as a refrigerant in the vapor-compression cycle. In order to vaporize at temperatures low enough to satisfy most refrigeration requirements, water would have to vaporize under very low pressures, which are difficult to produce and maintain economically.

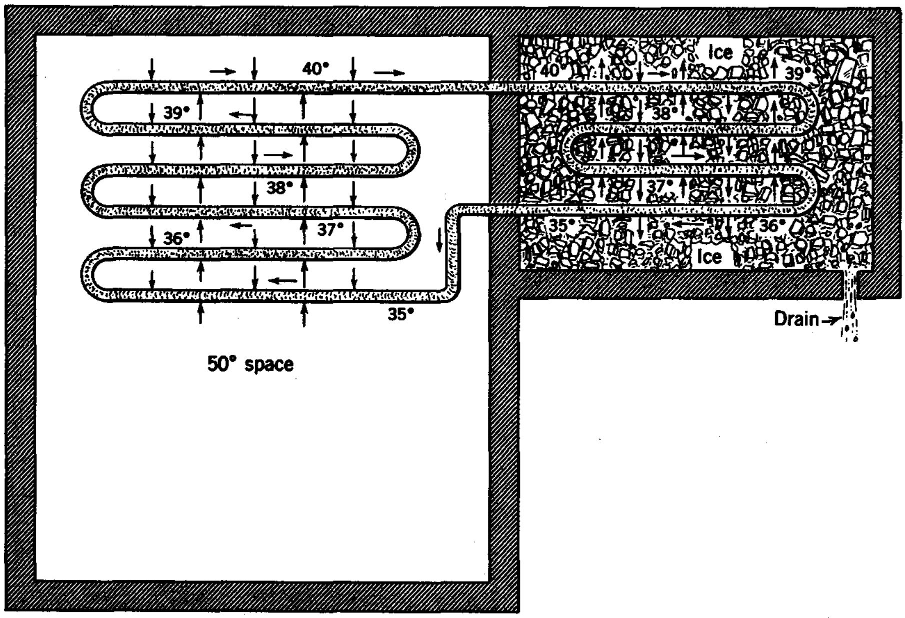


Fig. 6-3. Continuous sensible cooling. Heat taken in by the water in the space is given up to the ice.

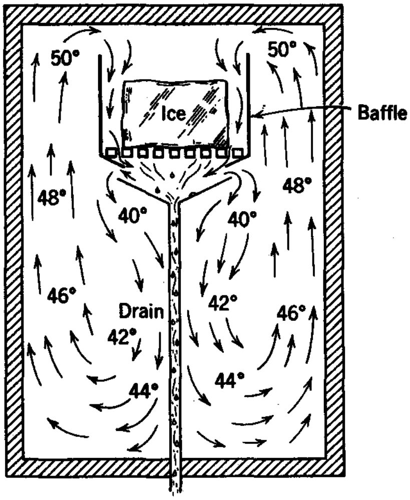


Fig. 6-4. Ice refrigerator. Heat is carried from warm walls and product to the ice by air circulation within the refrigerated space. Air circulation is by gravity.

There are numerous other fluids which have lower saturation temperatures than water at the same pressure. However, many of these fluids have other properties that render them unsuitable for use as refrigerants. Actually, only a relatively few fluids have properties that make them desirable as refrigerants, and most of these have been compounded specially for that purpose.

There is no one refrigerant which is best suited for all the different applications and operating conditions. For any specific application the refrigerant selected should be the one whose properties most closely fit the particular requirements of the application.

Of all of the fluids now in use as refrigerants, the one fluid which most nearly meets all the qualifications of the ideal general-purpose refrigerant is a fluorinated hydrocarbon of the methane series having the chemical name dichlorodifluoromethane (CCl_2F_2). It is one of a group of refrigerants introduced to the industry under the trade name of "Freon," but is now manufactured under several other proprietary designations. To avoid the confusion inherent in the use of proprietory or chemical names, this compound is now referred to as Refrigerant-12. Refrigerant-12 (R-12) has a saturation temperature of −21.6° F at standard atmospheric pressure. For this reason, R-12 can be stored as a liquid at ordinary temperatures only if confined under pressure in heavy steel cylinders.

Table 16-3 is a tabulation of the thermodynamic properties of R-12 saturated liquid and vapor. This table lists, among other things, the saturation temperature of R-12 corresponding to various pressures. Tables 16-4 through 16-6 list the thermodynamic properties of some of the other more commonly used refrigerants. These tables are similar to the saturated liquid and vapor tables previously discussed and are employed in the same manner.

6-7. Vaporizing the Refrigerant. An insulated space can be adequately refrigerated by merely allowing liquid R-12 to vaporize in a container vented to the outside as shown in Fig. 6-5. Since the R-12 is under atmospheric pressure, its saturation temperature is −21.6° F. Vaporizing at this low temperature, the R-12 readily absorbs heat from the 40° F space through the walls of the containing vessel. The heat absorbed by the vaporizing liquid leaves the space in the vapor escaping through the open vent. Since the temperature of the liquid remains constant during the vaporizing process, refrigeration will continue until all the liquid is vaporized.

Any container, such as the one in Fig. 6-5, in

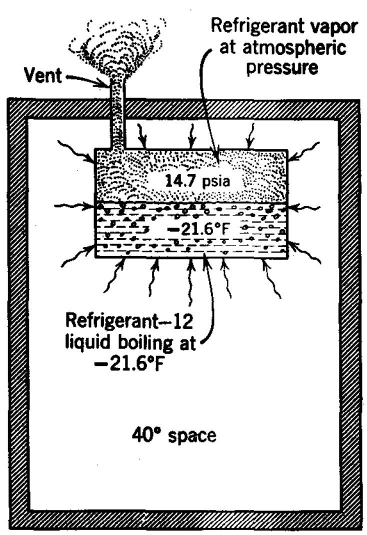


Fig. 6-5. The Refrigerant-12 liquid vaporizes as it takes in heat from the 40° F space. The heat taken in by the refrigerant leaves the space in the vapor escaping through the vent.

which a refrigerant is vaporized during a refrigerating process is called an evaporator and is one of the essential parts of any mechanical refrigerating system.

6-8. Controlling the Vaporizing Temperature. The temperature at which the liquid vaporizes in the evaporator can be controlled by controlling the pressure of the vapor over the liquid, which in turn is governed by regulating the rate at which the vapor escapes from the evaporator (Section 4-5). For example, if a hand valve is installed in the vent line and the vent is partially closed off so that the vapor cannot escape freely from the evaporator, vapor will collect over the liquid causing the pressure in the evaporator to rise with a corresponding increase in the saturation temperature of the refrigerant (Fig. 6-6). By carefully adjusting the vent valve to regulate the flow of vapor from the evaporator, it is possible to control the pressure of the vapor over the liquid and cause the R-12 to vaporize at any desired temperature between −21.6° F and the space temperature. Should the vent valve be completely closed so that no vapor is allowed to escape from the evaporator, the pressure in the evaporator will increase to a point such that the saturation temperature of the liquid will be equal to the space temperature, or 40° F. When this occurs, there will be no temperature differential

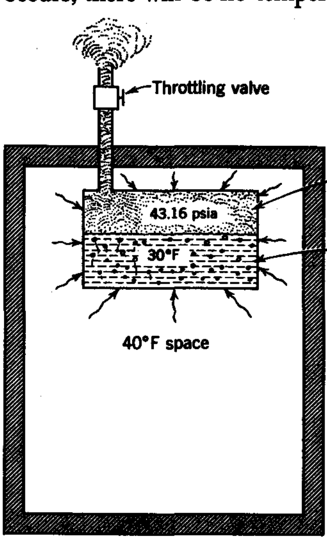


Fig. 6-6. The boiling temperature of the liquid refrigerant in the evaporator is controlled by controlling the pressure of the vapor over the liquid with the throttling valve in the vent.

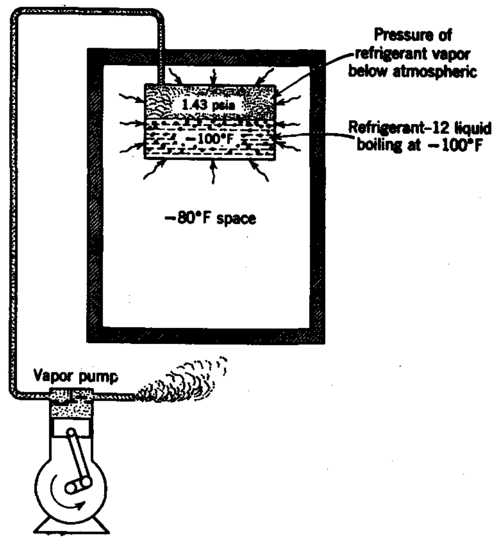


Fig. 6-7. Pressure of refrigerant in evaporator reduced below atmospheric by action of a vapor pump.

and no heat will flow from the space to the refrigerant. Vaporization will cease and no further cooling will take place.

When vaporizing temperatures below −21.6° F are required, it is necessary to reduce the pressure in the evaporator to some pressure below atmospheric. This can be accomplished through the use of a vapor pump as shown in Fig. 6-7. By this method, vaporization of the liquid R-12 can be brought about at very low temperatures in accordance with the pressure–temperature relationships given in Table 16-3.

6-9. Maintaining a Constant Amount of Liquid in the Evaporator. Continuous vaporization of the liquid in the evaporator requires that the supply of liquid be continuously replenished if the amount of liquid in the evaporator is to be maintained constant. One method of replenishing the supply of liquid in the evaporator is through the use of a float valve assembly as illustrated in Fig. 6-8. The action of the float assembly is to maintain a constant level of liquid in the evaporator by allowing liquid to flow into the evaporator from the storage tank or cylinder at exactly the same rate that the supply of liquid in the evaporator is being depleted by vaporization. Any increase in the rate of vaporization causes the liquid level in the evaporator to drop slightly, thereby opening

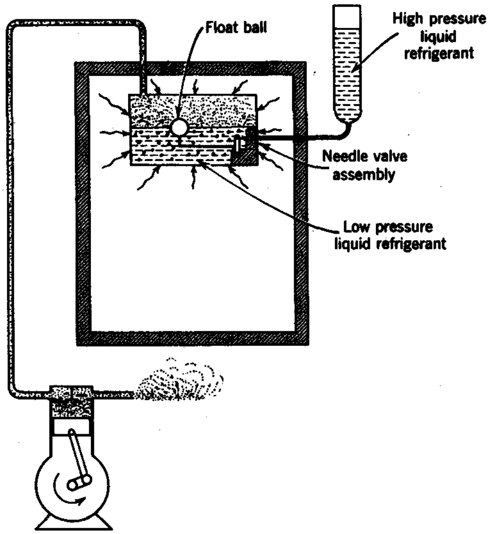


Fig. 6-8. Float valve assembly maintains constant liquid level in evaporator. The pressure of the refrigerant is reduced as the refrigerant passes through the needle valve.

the needle valve wider and allowing liquid to flow into the evaporator at a higher rate. Likewise, any decrease in the rate of vaporization causes the liquid level to rise slightly, thereby moving the needle valve in the closing direction to reduce the flow of liquid into the evaporator. When vaporization ceases entirely, the rising liquid level will close the float valve tightly and stop the flow of liquid completely. When vaporization is resumed, the liquid level will fall allowing the float valve to open and admit liquid to the evaporator.

The liquid refrigerant does not vaporize in the storage cylinder and feed line because the pressure in the cylinder is such that the saturation temperature of the refrigerant is equal to the temperature of the surroundings (see Section 4-10). The high pressure existing in the cylinder forces the liquid to flow through the feed line and the float valve into the lower pressure evaporator. In passing through the float valve, the high pressure refrigerant undergoes a pressure drop which reduces its pressure to the evaporator pressure, thereby permitting the refrigerant liquid to vaporize in the evaporator at the desired low temperature.

Any device, such as the float valve illustrated in Fig. 6-8, used to regulate the flow of liquid refrigerant into the evaporator is called a refrigerant flow control. The refrigerant flow control is an essential part of every mechanical refrigerating system.

There are five different types of refrigerant flow controls, all of which are in use to some extent at the present time. Each of these distinct types is discussed at length in Chapter 17. The float type of control illustrated in Fig. 6-8 has some disadvantages, mainly bulkiness, which tend to limit its use to some few special applications. The most widely used type of refrigerant flow control is the thermostatic expansion valve. A flow diagram illustrating the use of a thermostatic expansion valve to control the flow of refrigerant into a serpentine coil type evaporator is shown in Fig. 6-9.

6-10. Salvaging the Refrigerant. As a matter of convenience and economy it is not practical to permit the refrigerant vapor to escape to the outside and be lost by diffusion into the air. The vapor must be collected continuously and condensed back into the liquid state so that the same refrigerant is used over and over again, thereby eliminating the need for ever replenishing the supply of refrigerant in the system. To

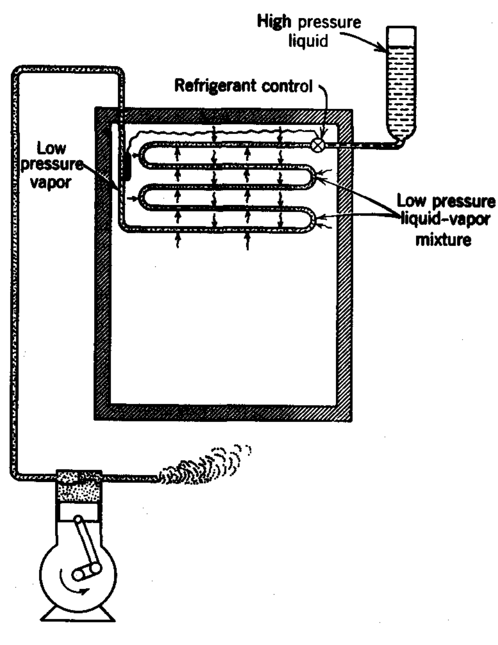


Fig. 6-9. Serpentine coil evaporator with thermostatic expansion valve refrigerant control.

provide some means of condensing the vapor, another piece of equipment, a condenser, must be added to the system (Fig. 6-10).

Since the refrigerant vaporizes in the evaporator because it absorbs the necessary latent heat from the refrigerated space, all that is required in order to condense the vapor back into the liquid state is that the latent heat be caused to flow out of the vapor into another body. The body of material employed to absorb the latent heat from the vapor, thereby causing the vapor to condense, is called the condensing medium. The most common condensing media are air and water. The water used as a condensing medium is usually supplied from the city main or from a cooling tower. The air used as a condensing medium is ordinary outdoor air at normal temperatures.

For heat to flow out of the refrigerant vapor into the condensing medium the temperature of the condensing medium must be below that of the refrigerant vapor. However, since the pressure and temperature of the saturated vapor leaving the evaporator are the same as those of the vaporizing liquid, the temperature of the vapor will always be considerably below that of any normally available condensing medium. Therefore, heat will not flow out of the refrigerant vapor into the air or water used as the condensing medium until the saturation temperature of the refrigerant vapor has been increased by compression to some temperature above the temperature of the condensing medium. The vapor pump or compressor shown in Fig. 6-10 serves this purpose.

Before compression, the refrigerant vapor is at the vaporizing temperature and pressure. Since the pressure of the vapor is low, the corresponding saturation temperature is also low. During compression the pressure of the vapor is increased to a point such that the corresponding saturation temperature is above the temperature of the condensing medium being employed. At the same time, since mechanical work is done on the vapor in compressing it to the higher pressure, the internal energy of the vapor is increased with a corresponding increase in the temperature of the vapor.

After compression, the high-pressure, high-temperature vapor is discharged into the condenser where it gives up heat to the lower temperature condensing medium. Since a vapor

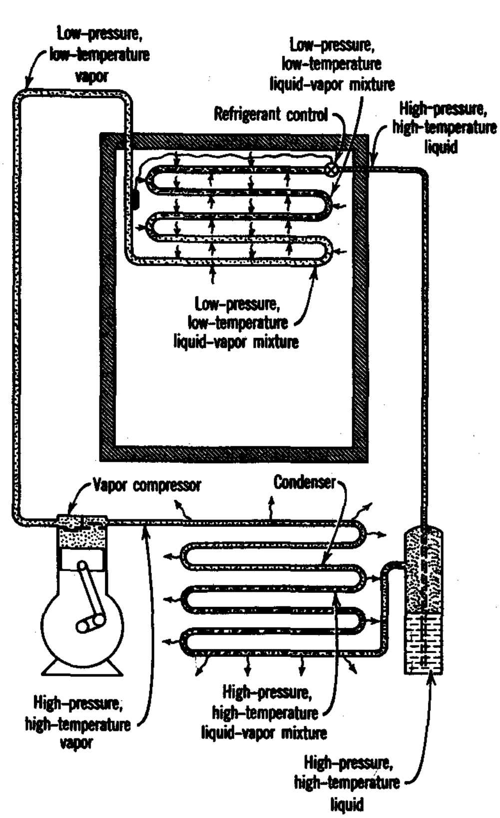


Fig. 6-10. Collecting and condensing the refrigerant vapor. Refrigerant absorbs heat in evaporator and gives off heat in the condenser.

cannot be cooled to a temperature below its saturation temperature, the continuous loss of heat by the refrigerant vapor in the condenser causes the vapor to condense into the liquid state at the new, higher pressure and saturation temperature. The heat given off by the vapor in the condenser is carried away by the condensing medium. The resulting condensed liquid, whose temperature and pressure will be the same as those of the condensing vapor, flows out of the condenser into the liquid storage tank and is then ready to be recirculated to the evaporator.

Notice that the refrigerant, sometimes called the working fluid, is merely a heat transfer agent which carries the heat from the refrigerated space to the outside. The refrigerant absorbs heat from the refrigerated space in the evaporator, carries it out of the space, and rejects it to the condensing medium in the condenser.

6-11. Typical Vapor-Compression System. A flow diagram of a simple vapor-compression system is shown in Fig. 6-11. The principal parts of the system are: (1) an evaporator, whose function it is to provide a heat transfer surface through which heat can pass from the refrigerated space or product into the vaporizing refrigerant; (2) a suction line, which conveys the low pressure vapor from the evaporator to the suction inlet of the compressor; (3) a vapor compressor, whose function it is to remove the vapor from the evaporator, and to raise the temperature and pressure of the vapor to a point such that the vapor can be condensed with normally available condensing media; (4) a "hot-gas" or discharge line which delivers the high-pressure, high-temperature vapor from the discharge of the compressor to the condenser; (5) a condenser, whose purpose it is to provide a heat transfer surface through which heat passes from the hot refrigerant vapor to the condensing medium; (6) a receiver tank, which provides storage for the liquid condenser so that a constant supply of liquid is available to the evaporator as needed; (7) a liquid line, which carries the liquid refrigerant from the receiver tank to the refrigerant flow control; (8) a refrigerant flow control, whose function it is to meter the proper amount of refrigerant to the evaporator and to reduce the pressure of the liquid entering the evaporator so that the liquid will vaporize in the evaporator at the desired low temperature.

6-12. Service Valves. The suction and discharge sides of the compressor and the outlet of the receiver tank are usually equipped with manual shut-off valves for use during service operations. These valves are known as the "suction service valve," the discharge service valve," and the "receiver tank valve," respectively. Receiver tanks on large systems frequently have shut-off valves on both the inlet and the outlet.

6-13. Division of the System. A refrigerating system is divided into two parts according to the pressure exerted by the refrigerant in the two parts. The low pressure part of the system consists of the refrigerant flow control, the evaporator, and the suction line. The pressure exerted by the refrigerant in these parts is the low pressure under which the refrigerant is vaporizing in the evaporator. This pressure is known variously as the "low side pressure," the "evaporator pressure," the "suction pressure," or the "back pressure." During service operations this pressure is usually measured at the compressor by installing a compound gage on the gage port of the suction service valve.

The high pressure side or "high side" of the system consists of the compressor, the discharge or "hot gas" line, the condenser, the receiver tank, and the liquid line. The pressure exerted by the refrigerant in this part of the system is the high pressure under which the refrigerant is condensing in the condenser. This pressure is

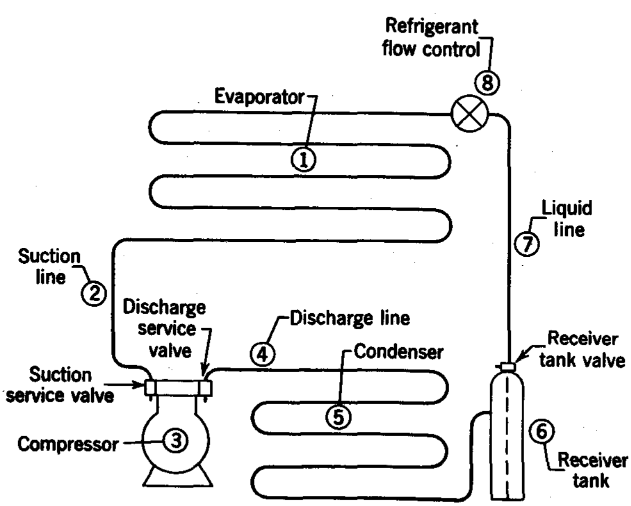


Fig. 6-11. Flow diagram of simple vapor compression system showing the principal parts.

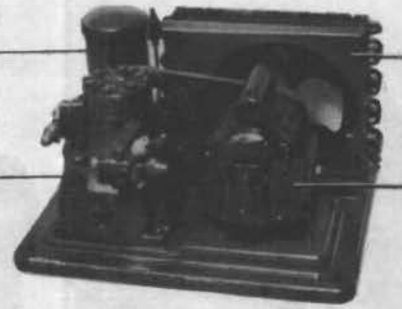


Fig. 6-12. Air-cooled condensing unit. Note fan mounted on motor shaft to circulate air over condenser.

called the "condensing pressure," the "discharge pressure," or, more often, the "head pressure."

The dividing points between the high and low pressure sides of the system are the refrigerant flow control, where the pressure of the refrigerant is reduced from the condensing pressure to the vaporizing pressure, and the discharge valves in the compressor, through which the high pressure vapor is exhausted after compression.* It should be noted that, although the compressor is considered to be a part of the high side of the system, the pressure on the suction side of the compressor and in the crankcase is the low side pressure. The change in pressure, of course, occurs in the cylinder during the compression process.

6-14. Condensing Units. The compressor, hot gas line, condenser, and receiver tank, along with the compressor driver (usually an electric motor), are often combined into one compact unit as shown in Fig. 6-12. Such an assembly is called a condensing unit because its function in the system is to reclaim the vapor and condense it back into the liquid state.

Condensing units are often classified according to condensing medium used to condense the refrigerant. A condensing unit employing air as the condensing medium (Fig. 6-12) is called an air-cooled condensing unit, whereas one employing water as the condensing medium is a water-cooled condensing unit.

6-15. Hermetic Motor-Compressor Assemblies. Condensing units of small horsepower are often equipped with hermetically sealed motor-compressor assemblies. The assembly consists of a direct-driven compressor mounted on a common shaft with the motor rotor and the whole assembly hermetically sealed in a welded steel shell (Fig. 6-13).

Condensing units equipped with hermetically sealed motor-compressor assemblies are known as "hermetic condensing units" and are employed on a number of small commercial refrigerators and on almost all household refrigerators, home freezers, and window air conditioners. For reasons that will be shown later, many hermetic condensing units are not equipped with receiver tanks.

A variation of the hermetic motor-compressor assembly is the "accessible hermetic." It is similar to the full hermetic except that the shell enclosing the assembly is bolted together rather than seam welded. (Fig. 6-14). The bolted construction permits the assemblies to be opened in the field for servicing.

6-16. Definition of a Cycle. As the refrigerant circulates through the system, it passes through a number of changes in state or condition, each of which is called a process. The refrigerant starts at some initial state or condition, passes through a series of processes in a definite sequence, and returns to the initial

* Care should be taken not to confuse the suction and discharge valves in the compressor with the suction and discharge service valves. The suction and discharge valves in a reciprocating compressor perform the same function as the intake and exhaust valves in an automobile engine and are vital to the operation of the compressor, whereas the suction and discharge service valves serve no useful purpose insofar as the operation of the compressor is concerned. The latter valves are used only to facilitate service operations, as their nomenclature implies.

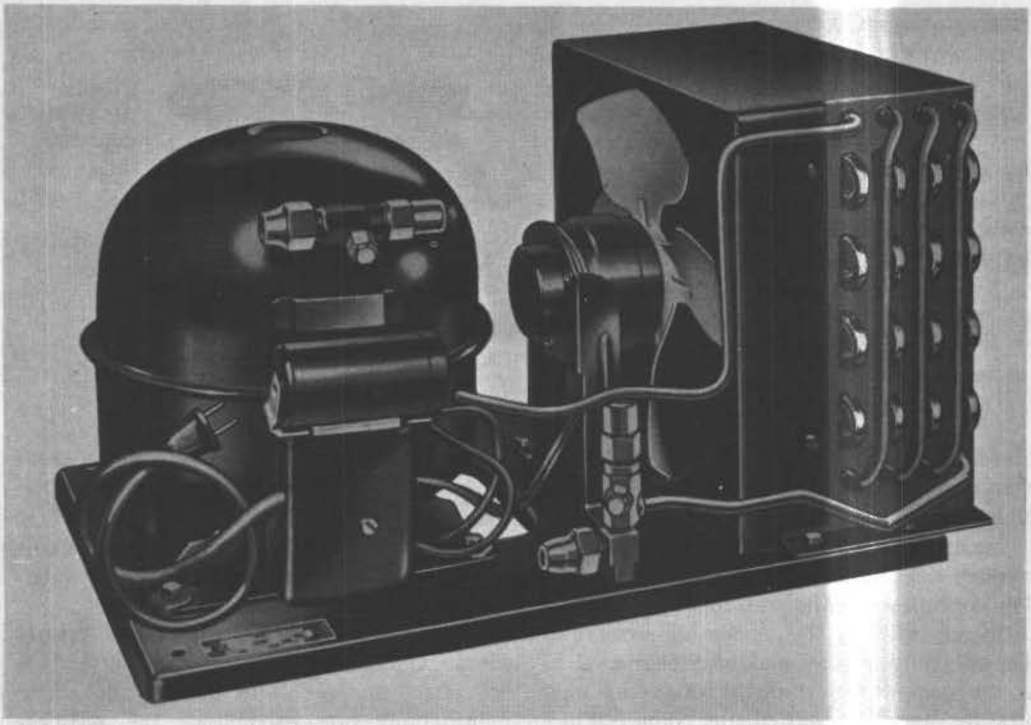


Fig. 6-13. Air-cooled condensing unit employing hermetic motor-compressor. Note separate fan to circulate air over condenser. (Courtesy Tecumseh Products Company.)

condition. This series of processes is called a cycle. The simple vapor-compression refrigeration cycle is made up of four fundamental processes: (1) expansion, (2) vaporization, (3) compression, and (4) condensation.

To understand properly the refrigeration cycle it is necessary to consider each process in the cycle both separately and in relation to the complete cycle. Any change in any one process in the cycle will bring about changes in all the other processes in the cycle.

6-17. Typical Vapor-Compression Cycle. A typical vapor-compression cycle is shown in Fig. 6-15. Starting at the receiver tank, high-temperature, high-pressure liquid refrigerant flows from the receiver tank through the liquid line to the refrigerant flow control. The pressure of the liquid is reduced to the evaporator pressure as the liquid passes through the refrigerant flow control so that the saturation temperature of the refrigerant entering the evaporator will be below the temperature of the refrigerated space. It will be shown later that a part of the liquid vaporizes as it passes through the refrigerant control in order to reduce the temperature of the liquid to the evaporating temperature.

In the evaporator, the liquid vaporizes at a constant pressure and temperature as heat to supply the latent heat of vaporization passes from the refrigerated space through the walls of the evaporator to the vaporizing liquid. By the action of the compressor, the vapor resulting from the vaporization is drawn from the evaporator through the suction line into the suction inlet of the compressor. The vapor leaving the evaporator is saturated and its temperature and pressure are the same as those of the vaporizing liquid. While flowing through the suction line from the evaporator to the compressor, the vapor usually absorbs heat from the air surrounding the suction line and becomes superheated. Although the temperature of the vapor increases somewhat in the suction line as the result of superheating, the pressure of the vapor does not change so that

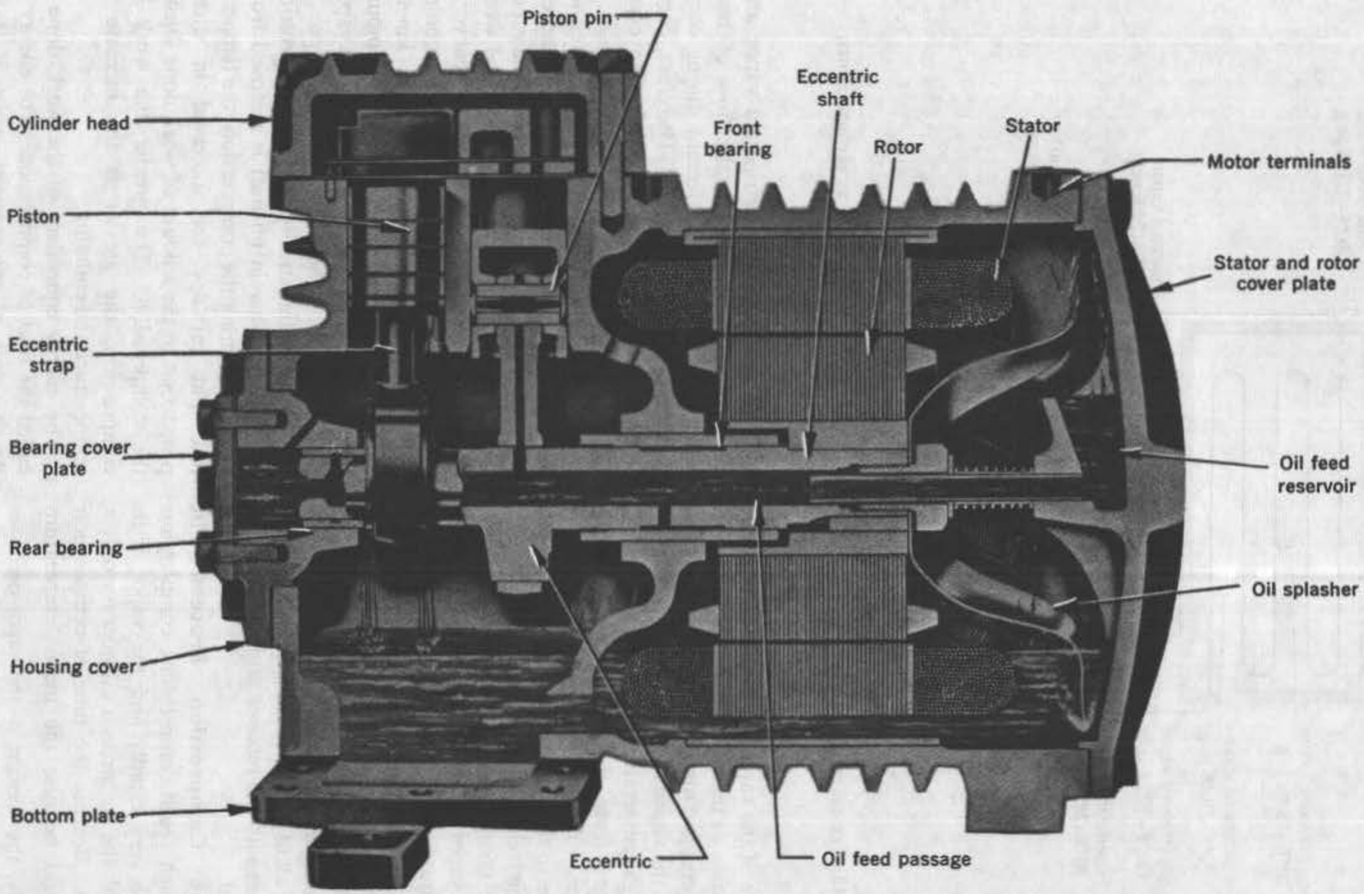


Fig. 6-14. Typical twin cylinder compressor—cutaway view illustrating bolted construction of "accessible hermetic" type motor-compressor unit. (Courtesy Carrier Corporation.)

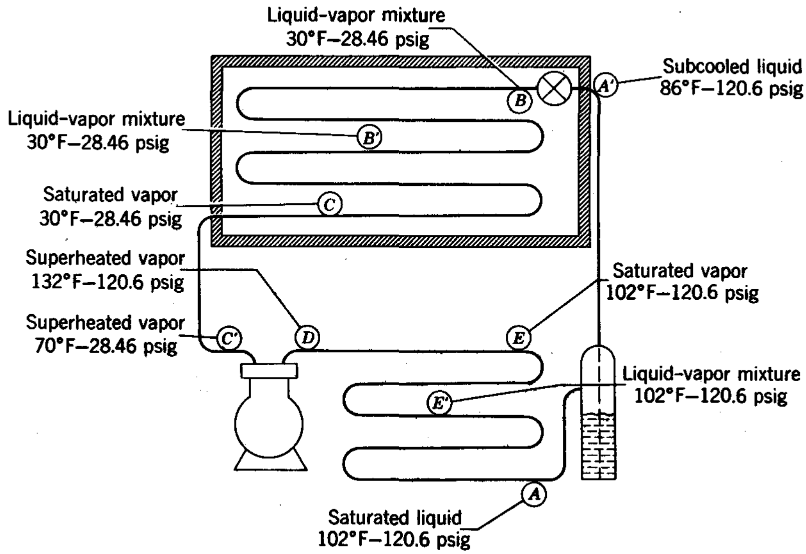


Fig. 6-15. Typical refrigeration system showing the condition of the refrigerant at various points.

the pressure of the vapor entering the compressor is the same as the vaporizing pressure.*

In the compressor, the temperature and pressure of the vapor are raised by compression and the high-temperature, high-pressure vapor is discharged from the compressor into the hot-gas line. The vapor flows through the hot-gas line to the condenser where it gives up heat to the relatively cool air being drawn across the condenser by the condenser fan. As the hot vapor gives off heat to the cooler air, its temperature is reduced to the new saturation temperature corresponding to its new pressure and the vapor condenses back into the liquid state as additional heat is removed. By the time the refrigerant reaches the bottom of the condenser, all of the vapor is condensed and the liquid passes into the receiver tank, ready to be recirculated.

6-18. The Compression Process. In modern, high speed compressors, compression takes place very rapidly and the vapor is in contact with the compressor cylinder for only a short time. Because the time of compression is short and because the mean temperature differential between the refrigerant vapor and the cylinder wall is small, the flow of heat either to or from the refrigerant during compression is usually negligible. Therefore, compression of the vapor in a refrigeration compressor is assumed to occur adiabatically.

Although no heat as such is transferred either to or from the refrigerant during the compression, the temperature and enthalpy of the vapor are increased because of the mechanical work done on the vapor by the piston. Whenever a vapor is compressed, unless the vapor is cooled during the compression, the internal kinetic energy of the vapor is increased by an amount equal to the amount of work done on the vapor (Section 3-12). Therefore, when a vapor is compressed adiabatically, as in a refrigeration compressor, wherein no heat is removed from the vapor during the compression, the temperature and enthalpy are increased in direct proportion to the amount of work done during the compression. The greater the work of compression, the greater is the increase in temperature and enthalpy.

The energy equivalent of the work done is called the heat of compression. The energy to do the work of compression, which is transferred to the vapor during the compression process, is supplied by the compressor driver,

* Actually, the pressure of the vapor decreases slightly between the evaporator and compressor because of the friction loss in the suction line resulting from the flow.

usually an electric motor. It will be shown later that the theoretical horsepower required to drive the compressor can be calculated from the heat of compression.

6-19. Discharge Temperature. Care should be taken not to confuse discharge temperature with condensing temperature. The discharge temperature is that at which the vapor is discharged from the compressor, whereas the condensing temperature is that at which the vapor condenses in the condenser and is the saturation temperature of the vapor corresponding to the pressure in the condenser. Because the vapor is usually superheated as it enters the compressor and because it contains the heat of compression, the vapor discharged from the compressor is highly superheated and its temperature is considerably above the saturation temperature corresponding to its pressure. The discharge vapor is cooled to the condensing temperature as it flows through the hot-gas line and through the upper part of the condenser, whereupon the further removal of heat from the vapor causes the vapor to condense at the saturation temperature corresponding to the pressure in the condenser.

6-20. Condensing Temperature. To provide a continuous refrigerating effect the refrigerant vapor must be condensed in the condenser at the same rate that the refrigerant liquid is vaporized in the evaporator. This means that heat must leave the system at the condenser at the same rate that heat is taken into the system in the evaporator and suction line, and in the compressor as a result of the work of compression. Obviously, any increase in the rate of vaporization will increase the required rate of heat transfer at the condenser.

The rate at which heat will flow through the walls of the condenser from the refrigerant vapor to the condensing medium is the function of three factors: (1) the area of the condensing surface, (2) the coefficient of conductance of the condenser walls, and (3) the temperature difference between the refrigerant vapor and the condensing medium. For any given condenser, the area of the condensing surface and the coefficient of conductance are fixed so that the rate of heat transfer through the condenser walls depends only on the temperature difference between the refrigerant vapor and the condensing medium.

Since the condensing temperature is always equal to the temperature of the condensing medium plus the temperature difference between the condensing refrigerant and the condensing medium, it follows that the condensing temperature varies directly with the temperature of the condensing medium and with the required rate of heat transfer at the condenser.

6-21. Condensing Pressure. The condensing pressure is always the saturation pressure corresponding to the temperature of the liquid-vapor mixture in the condenser.

When the compressor is not running, the temperature of the refrigerant mixture in the condenser will be the same as that of the surrounding air, and the corresponding saturation pressure will be relatively low. Consequently, when the compressor is started, the vapor pumped over into the condenser will not begin to condense immediately because there is no temperature differential between the refrigerant and the condensing medium, and therefore no heat transfer between the two. Because of the throttling action of the refrigerant control, the condenser may be visualized as a closed container, and as more and more vapor is pumped into the condenser without condensing, the pressure in the condenser increases to a point where the saturation temperature of vapor is sufficiently high to permit the required rate of heat transfer between the refrigerant and the condensing medium. When the required rate of heat transfer is reached, the vapor will condense as fast as it is pumped into the condenser, whereupon the pressure in the condenser will stabilize and remain more or less constant during the balance of the running cycle.

6-22. Refrigerating Effect. The quantity of heat that each pound of refrigerant absorbs from the refrigerated space is known as the refrigerating effect. For example, when 1 lb of ice melts it will absorb from the surrounding air and from adjacent objects an amount of heat equal to its latent heat of fusion. If the ice melts at 32° F it will absorb 144 Btu per pound, so that the refrigerating effect of 1 lb of ice is 144 Btu.

Likewise, when a liquid refrigerant vaporizes as it flows through the evaporator it will absorb an amount of heat equal to that required to vaporize it; thus the refrigerating effect of 1 lb of liquid refrigerant is potentially equal to its latent heat of vaporization. If the temperature of the liquid entering the refrigerant control from the liquid line is exactly equal to the vaporizing temperature in the evaporator the entire pound of liquid will vaporize in the evaporator and produce useful cooling, in which case the refrigerating effect per pound of refrigerant circulated will be equal to the total latent heat of vaporization. However, in an actual cycle the temperature of the liquid entering the refrigerant control is always considerably higher than the vaporizing temperature in the evaporator, and must first be reduced to the evaporator temperature before the liquid can vaporize in the evaporator and absorb heat from the refrigerated space. For this reason, only a part of each pound of liquid actually vaporizes in the evaporator and produces useful cooling. Therefore, the refrigerating effect per pound of liquid circulated is always less than the total latent heat of vaporization.

With reference to Fig. 6-15, the pressure of the vapor condensing in the condenser is 120 psig and the condensing temperature (saturation temperature) of the R-12 vapor corresponding to this pressure is 102° F. Since condensation occurs at a constant temperature, the temperature of the liquid resulting from the condensation is also 102° F. After condensation, as the liquid flows through the lower part of the condenser it continues to give up heat to the cooler condensing medium, so that before the liquid leaves the condenser its temperature is usually reduced somewhat below the temperature at which it condensed. The liquid is then said to be subcooled. The temperature at which the liquid leaves the condenser depends upon the temperature of the condensing medium and upon how long the liquid remains in contact with the condensing medium after condensation.

The liquid may be further subcooled in the receiver tank and in the liquid line by surrendering heat to the surrounding air. In any case, because of the heat exchange between the refrigerant in the liquid line and the surrounding air, the temperature of the liquid approaching the refrigerant control is likely to be fairly close to the temperature of the air surrounding the liquid line. In Fig. 6-15, the liquid approaches the refrigerant control at a temperature of 86° F, whereas its pressure is still the same as the condensing pressure, 120.6 psig. Since the saturation temperature corresponding to 120.6 psig is 102° F, the 86° F liquid at the refrigerant control is subcooled 16° F (102 − 86) below its saturation temperature.

Since the saturation pressure corresponding to 86° F is 93.2 psig, the R-12 can exist in the liquid state as long as its pressure is not reduced below 93.2 psig. However, as the liquid passes through the refrigerant control its pressure is reduced from 120.6 psig to 28.46 psig, the saturation pressure corresponding to the 30° vaporizing temperature of the refrigerant in the evaporator. Since the R-12 cannot exist as a liquid at any temperature above the saturation temperature of 30° F when its pressure is 28.46 psig, the liquid must surrender enough heat to cool itself from 86° F to 30° F at the instant that its pressure is reduced in passing through the refrigerant control.

From Table 16-3, the enthalpy of liquid at 86° F and at 30° F is 27.73 Btu per pound and 14.76 Btu per pound, respectively, so that each pound of liquid must surrender 12.97 Btu (27.73 − 14.76) in order to cool from 86° F to 30° F. Because the liquid expands through the refrigerant control so rapidly, the liquid is not in contact with the control for a sufficient length of time to permit this amount of heat to be transferred from the refrigerant to the control. Therefore, a portion of each pound of liquid vaporizes as the liquid passes through the control, and the heat to supply the latent heat of vaporization for the portion that vaporizes is drawn from the body of the liquid, thereby reducing the temperature of the refrigerant to the evaporator temperature. In this instance, enough of each pound of liquid vaporizes while passing through the refrigerant control to absorb exactly the 12.97 Btu of sensible heat that each pound of liquid must surrender in order to cool from 86° F to 30° F and the refrigerant is discharged from the refrigerant control into the evaporator as a liquid-vapor mixture. Obviously, only the liquid portion of the liquid-vapor mixture will vaporize in the evaporator and produce useful cooling. That portion of each pound of liquid circulated which vaporizes in the refrigerant control produces no useful cooling and represents a loss of refrigerating effect. It follows, then, that the refrigerating effect per pound of liquid circulated is equal to the total latent heat of vaporization less the amount of heat absorbed by that part of each pound that vaporizes in the control to reduce the temperature of the liquid to the vaporizing temperature.

From Table 16-3, the latent heat of vaporization of R-12 at 30° F is 66.85 Btu per pound. Since the loss of refrigerating effect is 12.97 Btu per pound, the refrigerating effect in this instance is (66.85 − 12.97) 53.88 Btu per pound.

The percentage of each pound of refrigerant that vaporizes in the refrigerant control can be determined by dividing the total latent heat of vaporization into the heat absorbed by that part of the pound that vaporizes in the control. In this instance the percentage of each pound vaporizing in the control is (12.97/66.85 × 100) 19.4%. Only 80.6% of each pound circulated actually vaporizes in the evaporator and produces useful cooling (66.85 × 0.806 = 53.88 Btu/lb).

Even though a portion of each pound circulated vaporizes as it passes through the refrigerant control, the enthalpy of the refrigerant does not change in the control. That is, since there is no heat transfer between the refrigerant and the control, the enthalpy of the liquid-vapor mixture discharged from the control into the evaporator is exactly the same as the enthalpy of the liquid approaching the control. Therefore, the difference between the enthalpy of the refrigerant vapor leaving the evaporator and the enthalpy of the liquid approaching the control is only the amount of heat absorbed by the refrigerant in the evaporator, which is, of course, the refrigerating effect. Hence, for any given conditions the refrigerating effect per pound can be easily determined by subtracting the enthalpy of the liquid refrigerant entering the control from the enthalpy of the saturated vapor leaving the evaporator.

Example 6-1. Determine the refrigerating effect per pound if the temperature of the liquid R-12 approaching the refrigerant control is 86° F and the temperature of the saturated vapor leaving the evaporator is 30° F.

Solution. From Table 16-3, enthalpy of R-12 saturated vapor at 30° F	= 81.61 Btu/lb
Enthalpy of R-12 liquid at 86° F	= 27.73 Btu/lb
Refrigerating effect per pound	= 53.88 Btu/lb

Example 6-2. If, in Example 6-1, the temperature of the liquid entering the refrigerant control is 60° F rather than 86° F, determine the refrigerating effect.

Solution. From Table 16-3, enthalpy of R-12 saturated vapor at 30° F	= 81.61 Btu/lb
Enthalpy of R-12 liquid at 60° F	= 21.57 Btu/lb
Refrigerating effect	= 60.04 Btu/lb

Example 6-3. If, in Example 6-1, the pressure in the evaporator is 21.05 psig, and the liquid reaching the refrigerant control is 86° F, what is the refrigerating effect?

Solution. From Table 16-3, the saturation temperature of R-12 corresponding to 21.05 psig is 20° F and the enthalpy of R-12 saturated vapor at that temperature	= 80.49 Btu/lb
Enthalpy of R-12 liquid at 86° F	= 27.73 Btu/lb
Refrigerating effect	= 52.77 Btu/lb

A comparison of Examples 6-1 and 6-2 indicates that the refrigerating effect increases as the temperature of the liquid approaching the refrigerant control decreases, whereas a comparison of Example 6-1 and 6-3 shows that the refrigerating effect decreases as the vaporizing temperature decreases. Therefore, it is evident that the refrigerating effect per pound of liquid circulated depends upon two factors: (1) the evaporating temperature and (2) the temperature at which the liquid refrigerant enters the refrigerant control. The higher the evaporating temperature and the lower the temperature of the liquid entering the refrigerant control, the greater will be the refrigerating effect.

6-23. System Capacity. The capacity of any refrigerating system is the rate at which it will remove heat from the refrigerated space and is usually stated in Btu per hour or in terms of its ice-melting equivalent.

Before the era of mechanical refrigeration, ice was widely used as a cooling medium. With the development of mechanical refrigeration, it was only natural that the cooling capacity of mechanical refrigerators should be compared with an ice-melting equivalent.

When one ton of ice melts it will absorb 288,000 Btu (2000 lb × 144 Btu/lb). If one ton of ice melts in one day (24 hr), it will absorb heat at the rate of 12,000 Btu/hr (288,000 Btu/24 hr) or 200 Btu/min (12,000 Btu/hr/60). Therefore, a mechanical refrigerating system having the capacity of absorbing heat from the refrigerated space at the rate of 200 Btu/min (12,000 Btu/hr) is cooling at a rate equivalent to the melting of one ton of ice in 24 hr and is said to have a capacity of one ton.

The capacity of a mechanical refrigeration system, that is, the rate at which the system will remove heat from the refrigerated space, depends upon two factors: (1) the weight of refrigerant circulated per unit of time and (2) the refrigerating effect of each pound circulated.

Example 6-4. A mechanical refrigerating system is operating under conditions such that the vaporizing temperature is 30° F while the temperature of the liquid approaching the refrigerant control is 86° F. If R-12 is circulated through the system at the rate of 5 lb/min, determine:

(*a*) the refrigerating capacity of the system in Btu per hour.
(*b*) the refrigerating capacity of the system in tons.

Solution

(*a*) From Example 6-1, refrigerating effect	= 53.88 Btu/lb
Weight of refrigerant circulated per minute	= 5 lb
Refrigerating capacity in Btu per minute	= 5 × 53.88
	= 269.40 Btu/min
Refrigerating capacity in Btu per hour	= 269.40 × 60
	= 16,164 Btu/hr
(*b*) Refrigerating capacity in tons	$= \frac{269.40}{200}$
	= 1.347 tons

6-24. Weight of Refrigerant Circulated per Minute per Ton. The weight of refrigerant which must be circulated per minute per ton of refrigerating capacity for any given operating conditions is found by dividing the refrigerating effect per pound at the given conditions into 200.

Example 6-5. An R-12 system is operating at conditions such that the vaporizing temperature is 20° F and the condensing temperature is 100° F. If it is assumed that no subcooling of the liquid occurs so that the temperature of the liquid at the refrigerant control is also 100° F, find:

(*a*) The refrigerating effect per pound
(*b*) The weight of refrigerant circulated per minute per ton
(*c*) The weight of refrigerant circulated per minute for a 10-ton system.

Solution

(*a*) From Table 16-3, enthalpy of R-12 saturated vapor at 20° F	= 80.49 Btu/lb
Enthalpy of R-12 liquid at 100° F	= 31.16 Btu/lb
Refrigerating effect	49.33 Btu/lb
(*b*) Weight of refrigerant circulated per minute per ton	$= \frac{200}{49.33}$
	= 4.05 lb
(*c*) Weight of refrigerant circulated per minute for a 10-ton system	= 10 × 4.05
	= 40.5 lb

Example 6-6. If, in Example 6-5, the liquid is subcooled from 100° F to 80° F before it reaches the refrigerant control, calculate:

(*a*) the refrigerating effect
(*b*) the weight of refrigerant circulated per minute per ton

Solution

(*a*) From Table 16-3, enthalpy of R-12 saturated vapor at 20° F	= 80.49 Btu/lb
Enthalpy of R-12 liquid at 80° F	= 26.28 Btu/lb
Refrigerating effect	= 54.21 Btu/lb
(*b*) Weight of refrigerant circulated per minute per ton	$= \frac{200}{54.21}$
	= 3.69 lb

In comparing Examples 6-5 and 6-6, it is apparent that the weight of refrigerant which must be circulated per minute per ton of refrigerating capacity varies with the refrigerating effect and depends upon the operating conditions of the system. As the refrigerating effect per pound increases, the weight of refrigerant circulated per minute per ton decreases.

6-25. Volume of Vapor Displaced per Minute per Ton. When 1 lb of liquid refrigerant vaporizes, the volume of vapor which results depends upon the vaporizing temperature. The lower the vaporizing temperature and pressure, the greater is the volume of the vapor which is produced. When the vaporizing temperature is known, the specific volume of the saturated vapor which results from the vaporization can be found in the saturated vapor tables. For instance, from Table 16-3, the specific volume of R-12 saturated vapor at 10° F is 1.351 cu ft per pound. This means that each pound of R-12 that vaporizes at 10° F produces 1.351 cu ft of vapor. Therefore, if 10 lb of R-12 are vaporized at 10° F in an evaporator each minute, saturated vapor will be produced at the rate of 13.51 cu ft per minute (10 × 1.351).

In order to produce one ton of refrigerating capacity, a definite weight of refrigerant must be vaporized each minute. The volume of vapor which must be removed from the evaporator each minute can be calculated by multiplying the weight of refrigerant circulated per minute by the specific volume of the saturated vapor at the vaporizing temperature.

Example 6-7. Determine the volume of vapor to be removed from the evaporator per minute per ton of refrigerating capacity for the system described in Example 6-5.

Solution. From Table 16-3, specific volume of R-12 saturated vapor at 20° F	= 1.121 cu ft/lb
From Example 6-5, weight of refrigerant circulated per minute per ton	= 4.05 lb/min/ton
Volume of vapor displaced per minute per ton	= 4.05 × 1.121 = 4.55 cu ft/min/ton

6-26. Compressor Capacity. In any mechanical refrigerating system the capacity of the compressor must be such that vapor is drawn from the evaporator at the same rate that vapor is produced by the boiling action of the liquid refrigerant. If the refrigerant vaporizes faster than the compressor is able to remove the vapor, the excess vapor will accumulate in the evaporator and cause the pressure in the evaporator to increase, which in turn will result in raising the boiling temperature of the liquid. On the other hand, if the capacity of the compressor is such that the compressor removes the vapor from the evaporator too rapidly, the pressure in the evaporator will decrease and result in a decrease in the boiling temperature of the liquid. In either case, design conditions will not be maintained and the refrigerating system will be unsatisfactory.

The maintenance of design conditions and therefore good refrigeration depends upon the selection of a compressor whose capacity is such that the compressor will displace in any given interval of time a volume of vapor that is equal to the volume occupied by the weight of refrigerant which must be vaporized during the same time interval in order to produce the required refrigerating capacity at the design conditions.

For instance, in Example 6-7, 4.05 lb of R-12 must be vaporized each minute at 20° F for each one ton of refrigerating capacity desired. In vaporizing, the 4.05 lb of R-12 produce 4.55 cu ft of vapor (4.05 × 1.121). If the evaporator pressure and the boiling temperature of the liquid in the evaporator are to remain constant, this volume of vapor must be removed from the evaporator each minute for each one ton of refrigerating capacity. Hence, the compressor selected for a system operating at the conditions of Example 6-7 should have a capacity such that it will remove vapor from the evaporator at the rate of 4.55 cu ft per minute for each ton of refrigerating capacity required. For a 10 ton system, the compressor would have to remove vapor from the evaporator at the rate of 45.50 cu ft per minute (10 × 4.55).

PROBLEMS

1. The temperature of liquid R-12 entering the refrigerant control is 86° F and the vaporizing temperature 30° F. Determine:

(*a*) The refrigerating effect per pound of refrigerant circulated. *Ans.* 53.89 Btu/lb

(*b*) The loss of refrigerating effect per pound. *Ans.* 12.96 Btu/lb

(*c*) The weight of refrigerant circulated per minute per ton. *Ans.* 3.71 lb/min/ton

(*d*) The volume of vapor displaced per minute per ton. *Ans.* 3.48 cu ft/min/ton

2. If saturated R-12 liquid reaches the refrigerant control at a pressure of 136 psig and the vaporizing pressure in the evaporator is 30.07 psig, determine:

(*a*) The refrigerating effect per pound. *Ans.* 48.18 Btu/lb

(*b*) The weight of refrigerant circulated per minute per ton. *Ans.* 4.15 lb/min/ton
(*c*) The volume of vapor displaced per minute per ton. *Ans.* 3.77 cu ft/min/ton

3. If the liquid approaching the refrigerant control in Problem 2 is subcooled to 70° F, determine:
(*a*) The refrigerating effect. *Ans.* 57.93 Btu/lb
(*b*) The weight of refrigerant circulated per minute per ton. *Ans.* 3.45 cu ft/min/ton
(*c*) The volume of vapor to be displaced per minute per ton. *Ans.* 3.13 cu ft/min/ton

4. If, in Problem 2, the liquid is subcooled to 70° F and the evaporating pressure is lowered to 16.35 psig, determine
(*a*) The refrigerating effect. *Ans.* 45.25 Btu/lb
(*b*) The weight of refrigerant circulated per minute per ton. *Ans.* 4.42 lb/min/ton
(*c*) The volume of vapor displaced per minute per ton. *Ans.* 6.44 cu ft/min/ton

7

Cycle Diagrams and the Simple Saturated Cycle

7-1. Cycle Diagrams. A good knowledge of the vapor-compression cycle requires an intensive study not only of the individual processes that make up the cycle but also of the relationships that exist between the several processes and of the effects that changes in any one process in the cycle have on all the other processes in the cycle. This is greatly simplified by the use of charts and diagrams upon which the complete cycle may be shown graphically. Graphical representation of the refrigeration cycle permits the desired simultaneous consideration of all the various changes in the condition of the refrigerant which occur during the cycle and the effect that these changes have on the cycle without the necessity of holding in mind all the different numerical values involved in cyclic problems.

The diagrams frequently used in the analysis of the refrigeration cycle are the pressure-enthalpy (*Ph*) diagram and the temperature-entropy (*Ts*) diagram. Of the two, the pressure-enthalpy diagram seems to be the most useful and is the one which is emphasized in the following sections. The temperature-entropy diagram has already been introduced (Section 4-19) and its application to the refrigeration cycle will be discussed to some extent in this chapter.

7-2. The Pressure-Enthalpy Diagram. A pressure-enthalpy chart for R-12 is shown in Fig. 7-1.* The condition of the refrigerant in any thermodynamic state can be represented as a point on the *Ph* chart. The point on the *Ph* chart which represents the condition of the refrigerant in any one particular thermodynamic state may be located if any two properties of the refrigerant at that state are known. Once the state point has been located on the chart, all the other properties of the refrigerant for that state can be determined directly from the chart.

As shown by the skeleton *Ph* chart in Fig. 7-2, the chart is divided into three areas which are separated from each other by the saturated liquid and saturated vapor curves. The area on the chart to the left of the saturated liquid curve is called the subcooled region. At any point in the subcooled region the refrigerant is in the liquid state and its temperature is below the saturation temperature corresponding to its pressure. The area to the right of the saturated vapor curve is the superheated region and the refrigerant is in the form of a superheated vapor. The center section of the chart, between the saturated liquid and saturated vapor curves, represents the change in phase of the refrigerant between the liquid and vapor states. At any point between the two curves the refrigerant is in the form of a liquid-vapor mixture. The distance between the two curves along any constant pressure line, as read on the enthalpy scale at the bottom of the chart, is the latent heat of vaporization of the refrigerant at that pressure. The saturated liquid and saturated vapor curves are not exactly parallel to each other because the latent heat of vaporization of the refrigerant varies with the pressure at which the change in phase occurs.

On the chart, the change in phase from the liquid to the vapor phase takes place progressively from left to right, whereas the change in phase from the vapor to the liquid phase occurs from right to left. Close to the saturated liquid curve the liquid-vapor mixture is nearly all liquid, whereas close to the saturated vapor curve the liquid-vapor mixture is almost all vapor. The lines of constant quality (Fig. 7-3), extending from top to bottom through the center section of the chart and approximately parallel to the saturated liquid and vapor

* The pressure-enthalpy chart for each refrigerant is different, depending upon the properties of the particular refrigerant.

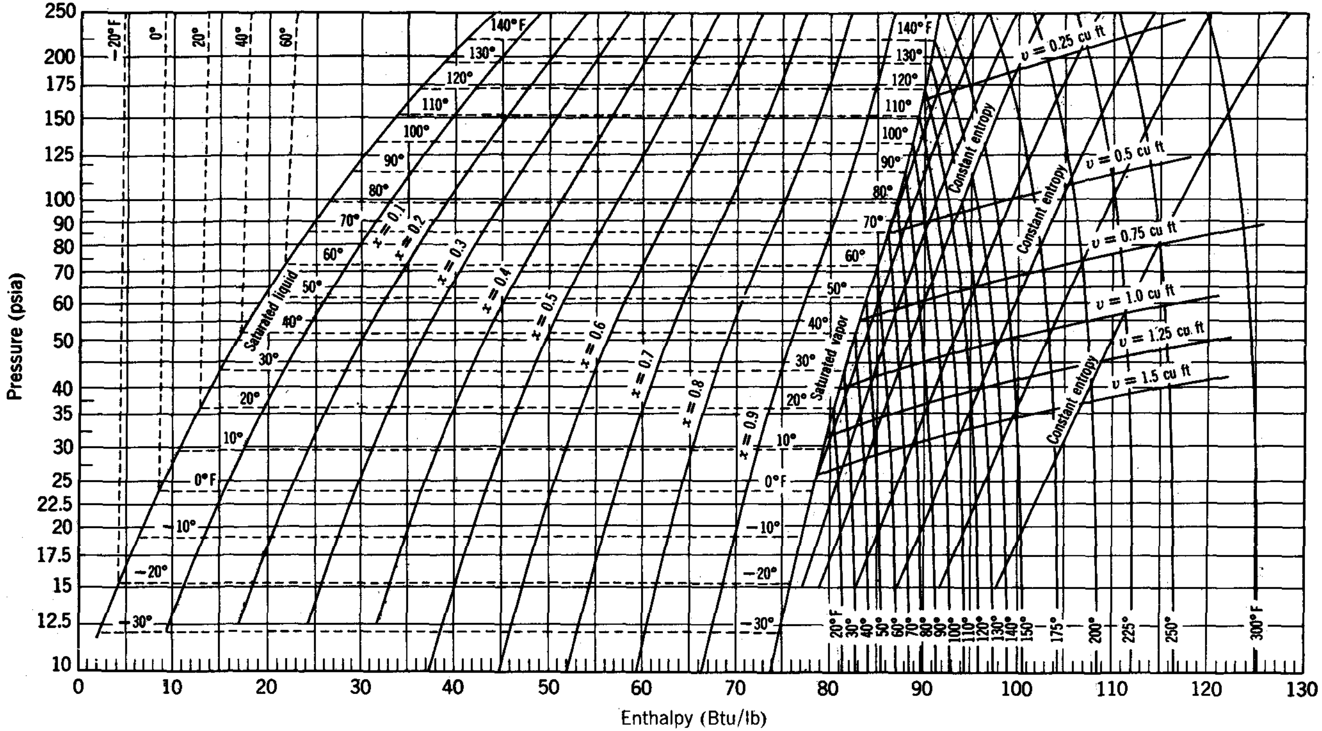


Fig. 7-1. Pressure-enthalpy diagram for Refrigerant-12.

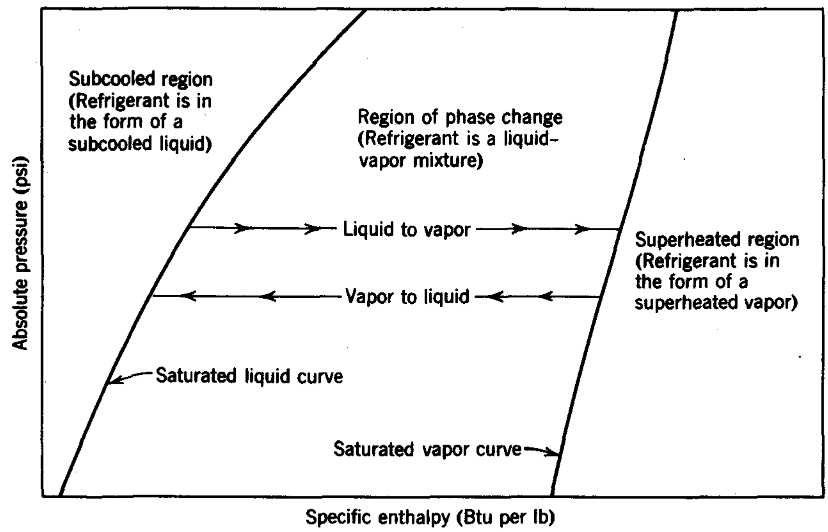


Fig. 7-2. Skeleton *Ph* chart illustrating the three regions of the chart and the direction of phase changing.

curves, indicate the percentage of vapor in the mixture in increments of 10%. For example, at any point on the constant quality line closest to the saturated liquid curve the quality of the liquid-vapor mixture is 10%, which means that 10% (by weight) of the mixture is vapor. Similarly, the indicated quality of the mixture at any point along the constant quality line closest to the saturated vapor curve is 90% and the amount of vapor in the liquid-vapor mixture is 90%. At any point on the saturated liquid curve the refrigerant is a saturated liquid and at any point along the saturated vapor curve the refrigerant is a saturated vapor.

The pressure is plotted along the vertical axis, and the enthalpy is plotted along the horizontal axis. Hence, the horizontal lines extending across the chart are lines of constant pressure and the vertical lines are lines of constant enthalpy.

The lines of constant temperature in the subcooled region are almost vertical on the chart and paralled to the lines of constant enthalpy. In the center section, since the refrigerant changes state at a constant temperature and pressure, the lines of constant temperature run horizontally across the chart and parallel to the lines of constant pressure. At the saturated vapor curve the lines of constant temperature change direction again and, in the superheated vapor region, fall off sharply toward the bottom of the chart.

The straight lines which extend diagonally and almost vertically across the superheated vapor region are lines of constant entropy. The curved, nearly horizontal lines crossing the superheated vapor region are lines of constant volume.

The values of any of the various properties of the refrigerant which are of importance in the refrigerating cycle may be read directly from the *Ph* chart at any point where the value of that particular property is significant to the process occurring at that point. To simplify the chart, the number of lines on the chart is kept to a minimum. For this reason, the value of those properties of the refrigerant which have no real significance at some points in the cycle are omitted from the chart at these points. For example, in the liquid region and in the region of phase change (center section) the values of entropy and volume are of no particular interest and are therefore omitted from the chart in these sections.

Since the *Ph* chart is based on a 1 lb mass of the refrigerant, the volume given is the specific volume, the enthalpy is in Btu per pound, and the entropy is in Btu per pound per degree of absolute temperature. Enthalpy values are found on the horizontal scale at the bottom of the chart and the values of entropy and volume are given adjacent to the entropy and volume lines, respectively. The values of both enthalpy and entropy are based on the arbitrarily selected zero point of −40° F.

The magnitude of the pressure in psia is read on the vertical scale at the left side of the chart. Temperature values in degrees Fahrenheit are found adjacent to the constant temperature lines in the subcooled and superheated regions of the chart and on both the saturated liquid and saturated vapor curves.

7-3. The Simple Saturated Refrigerating Cycle. A simple saturated refrigerating cycle is a theoretical cycle wherein it is assumed that the refrigerant vapor leaves the evaporator and enters the compressor as a saturated vapor (at the vaporizing temperature and pressure) and the liquid leaves the condenser and enters the refrigerant control as a saturated liquid (at the condensing temperature and pressure). Although the refrigerating cycle of an actual refrigerating machine will usually deviate somewhat from the simple saturated cycle, the analysis of a simple saturated cycle is nonetheless worthwhile. In such a cycle, the fundamental processes which are the basis of every actual vapor compression refrigerating cycle are easily identified and understood. Furthermore, by using the simple saturated cycle as a standard against which actual cycles may be compared, the relative efficiency of actual refrigerating cycles at various operating conditions can be readily determined.

A simple saturated cycle for a R-12 system is plotted on a *Ph* chart in Fig. 7-4. The system is assumed to be operating under such conditions that the vaporizing pressure in the evaporator is 35.75 psia and the condensing pressure in the condenser is 131.6 psia. The points *A, B, C, D,* and *E* on the *Ph* diagram correspond to points in the refrigerating system as shown on the flow diagram in Fig. 7-5.

At point *A*, the refrigerant is a saturated liquid in the condenser at the condensing pressure and temperature, and its properties, as given in Table 16-3, are:

$$p = 131.6 \text{ psia} \qquad t = 100° \text{ F}$$
$$h = 31.16 \text{ Btu/lb} \qquad s = 0.06316 \text{ Btu/lb/° F}$$
$$v = 0.0127 \text{ cu ft/lb}$$

At point *A*, the values of *p*, *t*, and *h* may be read directly from the *Ph* chart. Since the refrigerant is always a saturated liquid at point *A*, point *A* will always fall somewhere along the saturated liquid curve and can be located on the *Ph* chart if either *p*, *t*, or *h* is known. Usually

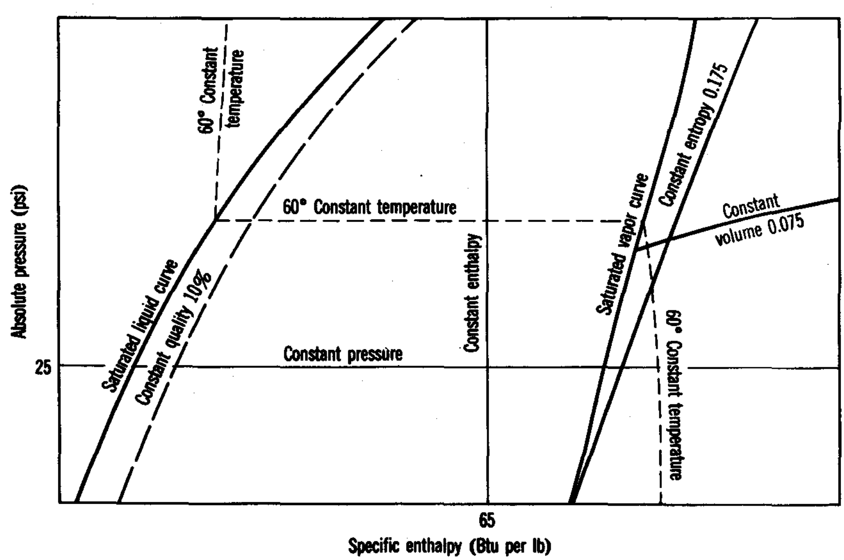


Fig. 7-3. Skeleton *Ph* chart showing paths of constant pressure, constant temperature, constant volume, constant quality, constant enthalpy, and constant entropy. (Refrigerant-12.)

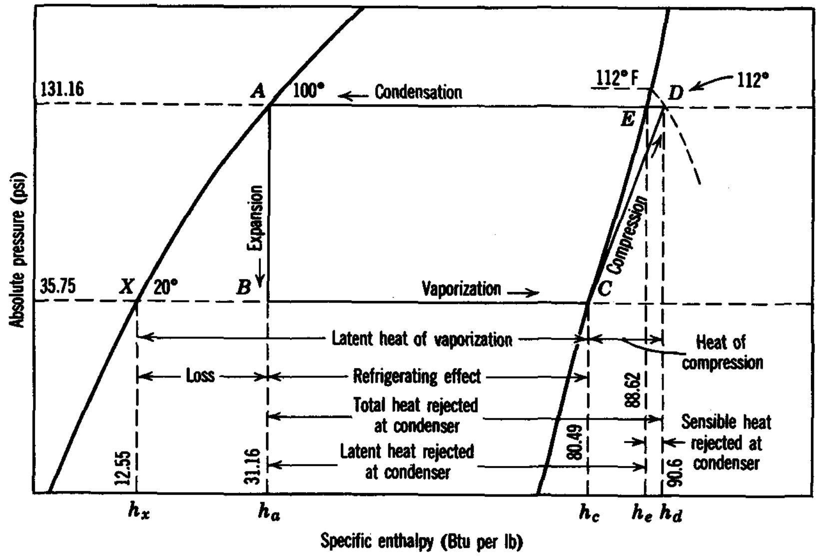


Fig. 7-4. Pressure-enthalpy diagram of a simple saturated cycle operating at a vaporizing temperature of 20° F and a condensing temperature of 100° F. (Refrigerant-12.)

in actual practice, either p, t, or both will be measurable.

7-4. The Expansion Process. In the simple saturated cycle there is assumed to be no change in the properties (condition) of the refrigerant liquid as it flows through the liquid line from the condenser to the refrigerant control and the condition of the liquid approaching the refrigerant control is the same as its condition at point A. The process described by the initial and final state points A–B occurs in the refrigerant control when the pressure of the liquid is reduced from the condensing pressure to the evaporating pressure as the liquid passes through the control.* When the liquid is expanded into the evaporator through the orifice of the control, the temperature of the liquid is reduced from the condensing temperature to the evaporating temperature by the flashing into vapor of a small portion of the liquid.

Process A–B is a throttling type of adiabatic expansion, frequently called "wire-drawing," in which the enthalpy of the working fluid does not change during the process. This type of expansion occurs whenever a fluid is expanded through an orifice from a high pressure to a lower pressure. It is assumed to take place without the gain or loss of heat through the piping or valves and without the performance of work.†

Since the enthalpy of the refrigerant does not change during process A–B, point B is located on the Ph chart by following the line of constant enthalpy from point A to the point where the constant enthalpy line intersects the line of constant pressure corresponding to the evaporating pressure. To locate point B on the Ph chart, the evaporating pressure or temperature must be known.

As a result of the partial vaporization of the liquid refrigerant during process A–B, the

* Process A–B is an irreversible adiabatic expansion during which the refrigerant passes through a series of state points in such a way that there is no uniform distribution of any of the properties. Hence, no true path can be drawn for the process and line A-B merely represents a process which begins at state point A and terminates at state point B.

† Actually, a certain amount of work is done by the fluid in projecting itself through the orifice of the control. However, since the heat equivalent of the work done in overcoming the friction of the orifice merely heats the orifice and is subsequently reabsorbed by the fluid, the assumption that the enthalpy of the fluid does not change during the process is not in error.

refrigerant at point B is a liquid-vapor mixture whose properties are:

p = 35.75 psia
t = 20° F
h = 31.16 Btu/lb (same as at point A)
v = 0.1520 cu ft/lb
s = 0.06316 Btu/lb/° F

NOTE. The change in entropy during the process A–B results from a transfer of heat energy which takes place within the refrigerant itself because of internal friction. A transfer of energy which occurs entirely within the working fluid does not affect the enthalpy of the fluid, only the entropy changes.

At point B, in addition to the values of p, t, and h, the approximate quality of the vapor can be determined from the Ph chart by interpolating between the lines of constant quality. In this instance, the quality of the vapor as determined from the Ph chart is approximately 27%.

Since the refrigerant at point B is a liquid-vapor mixture, only the values of p and t can be read directly from Table 16-3. However, because the enthalpy of the refrigerant at points A and B is the same, the enthalpy at point B may be read from Table 16-3 as the enthalpy at the conditions of point A. The quality of the vapor at point B can be determined as in Section 6-22, using enthalpy values taken either from Table 16-3 or from the Ph chart directly.

The values of s and v at point B are usually of no interest and are not given either on the Ph chart or in the vapor tables. If the values of s and v are desired, they must be calculated.

7-5. The Vaporizing Process. The process B–C is the vaporization of the refrigerant in the evaporator. Since vaporization takes place at a constant temperature and pressure, B–C is both isothermal and isobaric. Therefore, point C is located on the Ph chart by following the lines of constant pressure and constant temperature from point B to the point where they intersect the saturated vapor curve. At point C the refrigerant is completely vaporized and is a saturated vapor at the vaporizing temperature and pressure. The properties of the refrigerant at point C, as given in Table 16-3 or as read from the Ph chart, are:

p = 35.75 psia (same as at point B)
t = 20° F (same as at point B)
h = 80.49 Btu/lb
v = 1.121 cu ft/lb
s = 0.16949 Btu/lb/° F

The enthalpy of the refrigerant increases during process B–C as the refrigerant flows through the evaporator and absorbs heat from the refrigerated space. The quantity of heat absorbed by the refrigerant in the evaporator (refrigerating effect) is the difference between the enthalpy of the refrigerant at points B and C. Thus, if h_a, h_b, h_c, h_d, h_e, and h_x represent the

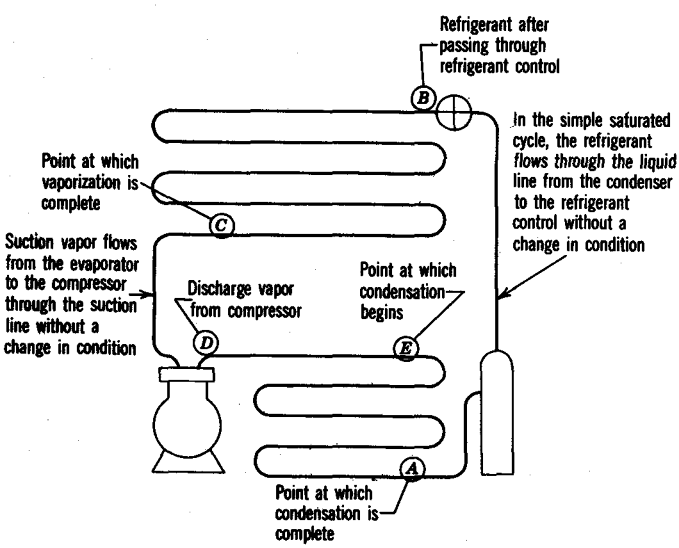


Fig. 7-5. Flow diagram of a simple saturated cycle.

enthalpies of the refrigerant at points A, B, C, D, E, and X, respectively, then

$$q_1 = h_c - h_b \tag{7-1}$$

where q_1 = the refrigerating effect in Btu/lb.

But since h_b is equal to h_a, then

$$q_1 = h_c - h_a \tag{7-2}$$

When we substitute the appropriate values in Equation 7-2 for the example in question,

$$\begin{aligned} q_1 &= 80.49 - 31.16 \\ &= 49.33 \text{ Btu/lb} \end{aligned}$$

On the Ph diagram, the distance between point X and point C represents the total latent heat of vaporization of 1 lb of R-12 at the vaporizing pressure of 35.75 psia (h_{fg} in Table 16-3). Therefore, since the distance B–C is the useful refrigerating effect, the difference between X–C and B–C, which is the distance X–B, is the loss of refrigerating effect.

7-6. The Compression Process. In the simple saturated cycle, the refrigerant undergoes no change in condition while flowing through the suction line from the evaporator to the compressor. Process C–D takes place in the compressor as the pressure of the vapor is increased by compression from the vaporizing pressure to the condensing pressure. For the simple saturated cycle, the compression process, C–D, is assumed to be isentropic.* An isentropic compression is a special type of adiabatic process which takes place without friction.† It is sometimes described as a "frictionless-adiabatic" or "constant-entropy" compression.

According to Equation 4-3, Section 4-19, the change in entropy (Δs) during any process is equal to the transferred heat (ΔQ) divided by the average absolute temperature (°R). In any frictionless-adiabatic process, such as the compression process C–D, wherein no heat, as such, is transferred either internally (within the vapor itself) or externally (to or from an external source) ΔQ will always be equal to zero. If ΔQ is equal to zero, then Δs must also be equal to zero. Hence, there is no change in the entropy of the vapor during a frictionless-adiabatic (isentropic) compression.

Since there is no change in the entropy of the vapor during process C–D, the entropy of the refrigerant at point D is the same as at point C. Therefore, point D can be located on the Ph chart by following the line of constant entropy from point C to the point where the constant entropy line intersects the line of constant pressure corresponding to the condensing pressure.

At point D, the refrigerant is a superheated vapor whose properties are:

p = 131.6 psia
t = 112° F (approximate)
h = 90.6 Btu/lb (approximate)
v = 0.330 cu ft/lb (approximate)
s = 0.16949 Btu/lb/° F (same as at point C)

All of the properties of the refrigerant at the condition of point D are taken from the Ph chart. Since the values of t, h, and v require interpolation, they are only approximations. The properties of the superheated refrigerant vapor cannot usually be read accurately from the vapor table unless the pressure of the vapor in question corresponds exactly to one of the pressures listed in the table. This is seldom the case, particularly at the higher pressures where the pressure listings in the table are in 10 lb increments.

Work is done on the vapor during the compression process, C–D, and the enthalpy of the refrigerant is increased by an amount equal to the heat energy equivalent of the mechanical work done on the vapor. The heat energy equivalent of the work done during the compression is often referred to as the heat of compression and is equal to the difference in the enthalpy of the refrigerant at points D and C. Thus, where q_2 is the heat of compression per pound of refrigerant circulated,

$$q_2 = h_d - h_c \tag{7-3}$$

For the example in question,

$$\begin{aligned} q_2 &= 90.60 - 80.49 \\ &= 10.11 \text{ Btu/lb} \end{aligned}$$

* It will be shown later that compression of the vapor in an actual refrigerating compressor usually deviates somewhat from true isentropic compression. As a general rule, compression is polytropic.

† The term, adiabatic, is used to describe any number of processes which take place without the transfer of energy as heat to or from the working substance during the process. Thus, an isentropic process is only one of a number of different processes which may be termed adiabatic. For example, compare process C–D with process A–B. Both are adiabatic, but C–D is frictionless, whereas A–B is a throttling type of process which involves friction.

The mechanical work done on the vapor by the piston during the compression may be calculated from the heat of compression. If w is the work done in foot-pounds per pound of refrigerant circulated and J is the mechanical energy equivalent of heat, then

$$w = q_2 \times J \tag{7-4}$$

or

$$w = J(h_d - h_c) \tag{7-5}$$

when we substitute in Equation 7-4,

$$\begin{aligned} w &= 10.11 \times 778 \\ &= 7865.58 \text{ ft-lb} \end{aligned}$$

As a result of absorbing the heat of compression, the hot vapor discharged from the compressor is in a superheated condition, that is, its temperature is greater than the saturation temperature corresponding to its pressure. In this instance, the vapor leaves the compressor at a temperature of 112° F, whereas the saturation temperature corresponding to its pressure of 131.6 psia is 100° F. Thus, before the vapor can be condensed, the superheat must be removed and the temperature of the vapor lowered from the discharge temperature to the saturation temperature corresponding to its pressure.

7-7. The Condensing Process. Usually, both processes D–E and E–A take place in the condenser as the hot gas discharged from the compressor is cooled to the condensing temperature and condensed. Process D–E occurs in the upper part of the condenser and to some extent in the hot gas line. It represents the cooling of the vapor from the discharge temperature to the condensing temperature as the vapor rejects heat to the condensing medium. During process D–E, the pressure of the vapor remains constant and point E is located on the Ph chart by following a line of constant pressure from point D to the point where the constant pressure line intersects the saturated vapor curve.

At point E, the refrigerant is a saturated vapor at the condensing temperature and pressure. Its properties, as read from either the Ph chart or from Table 16-3, are:

p = 131.6 psia (same as at point D)
t = 100° F
h = 88.62 Btu/lb
s = 0.16584 Btu/lb/° F
v = 0.319 cu ft/lb

The quantity of sensible heat (superheat) removed from 1 lb of vapor in the condenser in cooling the vapor from the discharge temperature to the condensing temperature is the difference between the enthalpy of the refrigerant at point D and the enthalpy at point E ($h_d - h_e$).

Process E–A is the condensation of the vapor in the condenser. Since condensation takes place at a constant temperature and pressure, process E–A follows along lines of constant pressure and temperature from point E to point A. The heat rejected to the condensing medium during process E–A is the difference between the enthalpy of the refrigerant at points E and A ($h_e - h_a$).

On returning to point A, the refrigerant has completed one cycle and its properties are the same as those previously described for point A.

Since both processes D–E and E–A occur in the condenser, the total amount of heat rejected by the refrigerant to the condensing medium in the condenser is the sum of the heat quantities rejected during processes D–E and D–A. The total heat given up by the refrigerant at the condenser is the difference between the enthalpy of the superheated vapor at point D and the saturated liquid at point A. Hence,

$$q_3 = h_d - h_a \tag{7-6}$$

where q_3 = the heat rejected at the condenser per pound of refrigerant circulated.

In this instance,

$$\begin{aligned} q_3 &= 90.60 - 31.16 \\ &= 59.44 \text{ Btu/lb} \end{aligned}$$

If the refrigerant is to reach point A at the end of the cycle in the same condition as it left point A at the beginning of the cycle, the total heat rejected by the refrigerant to the condensing medium in the condenser must be exactly equal to the heat absorbed by the refrigerant at all other points in the cycle. In a simple saturated cycle, the refrigerant is heated at only two points in the cycle: (1) in the evaporator by absorbing heat from the refrigerated space (q_1) and in the compressor by the heat of compression (q_2).

Therefore,

$$q_3 = q_1 + q_2 \tag{7-7}$$

In this instance,

$$\begin{aligned} q_3 &= 49.33 + 10.11 \\ &= 59.44 \text{ Btu/lb} \end{aligned}$$

Where m is the weight of refrigerant to be circulated per minute per ton,

$$m = \frac{200 \text{ Btu/min}}{q_1} \tag{7-8}$$

For the cycle in question,

$$m = \frac{200}{49.33}$$
$$= 4.05 \text{ lb/min/ton}$$

Then, where Q_3 is the total quantity of heat rejected at the condenser per minute per ton,

$$Q_3 = m(q_3) \tag{7-9}$$

or

$$Q_3 = m(h_d - h_a) \tag{7-10}$$

For the cycle in question,

$$Q_3 = 4.05 \times 59.44$$
$$= 240.93 \text{ Btu/min/ton}$$

Where Q_2 is the heat of compression per minute per ton of refrigerating capacity,

$$Q_2 = m(q_2) \tag{7-11}$$

or

$$Q_2 = m(h_d - h_c) \tag{7-12}$$

Substituting,

$$Q_2 = 4.05 \times 10.11$$
$$= 40.95 \text{ Btu/min/ton}$$

Where W is the work of compression done on the vapor per minute per ton of refrigerating capacity,

$$W = m(w) \tag{7-13}$$

or, since w equals $J(q_2)$ or $J(h_d - h_c)$,

$$W = Jm(q_2) \tag{7-14}$$

or

$$W = Jm(h_d - h_c) \tag{7-15}$$

or

$$W = J(Q_2) \tag{7-16}$$

When we substitute in Equation 7-15,

$$W = 778 \times 4.05 \times (90.60 - 80.49)$$
$$= 31{,}856 \text{ ft-lb/min/ton}$$

7-8. Theoretical Horsepower. The theoretical horsepower required to drive the compressor per ton of refrigerating capacity may be found by applying Equation 1-5 (Section 1-11):

$$\text{hp} = \frac{31{,}856}{33{,}000 \times 1}$$
$$= 0.965 \text{ hp/ton}$$

A more convenient method of determining the theoretical horsepower per ton is produced by combining Equations 1-5 and 7-15:

$$\text{hp} = \frac{m(h_d - h_c)}{42.42} \tag{7-17}$$

The compressor horsepower as calculated above represents only the horsepower required to compress the vapor. That is, it is the theoretical power which would be required per ton of refrigerating capacity by a 100% efficient system. It does not take into account the power required to overcome friction in the compressor and other power losses. The actual (brake) horsepower required per ton of refrigeration will usually be from 30% to 50% more than the theoretical horsepower calculated, depending upon the efficiency of the compressor. The factors governing compressor efficiency are discussed later.

7-9. Coefficient of Performance. The coefficient of performance of a refrigerating cycle is an expression of the cycle efficiency and is stated as the ratio of the heat absorbed in the refrigerated space to the heat energy equivalent of the energy supplied to the compressor, that is,

$$\text{Coefficient of performance} = \frac{\text{Heat absorbed from the refrigerated space}}{\text{Heat energy equivalent of the energy supplied to the compressor}}$$

For the theoretical simple saturated cycle, this may be written as

$$\text{c.o.p.} = \frac{\text{Refrigerating effect}}{\text{Heat of compression}} \tag{7-18}$$
$$= \frac{(h_c - h_a)}{(h_d - h_c)}$$
$$= \frac{(q_1)}{(q_2)}$$

Hence, for the cycle in question,

$$\text{c.o.p.} = \frac{49.33}{10.11}$$
$$= 4.88$$

7-10. Effect of Suction Temperature on Cycle Efficiency. The efficiency of the vapor-compression refrigerating cycle varies considerably with both the vaporizing and condensing

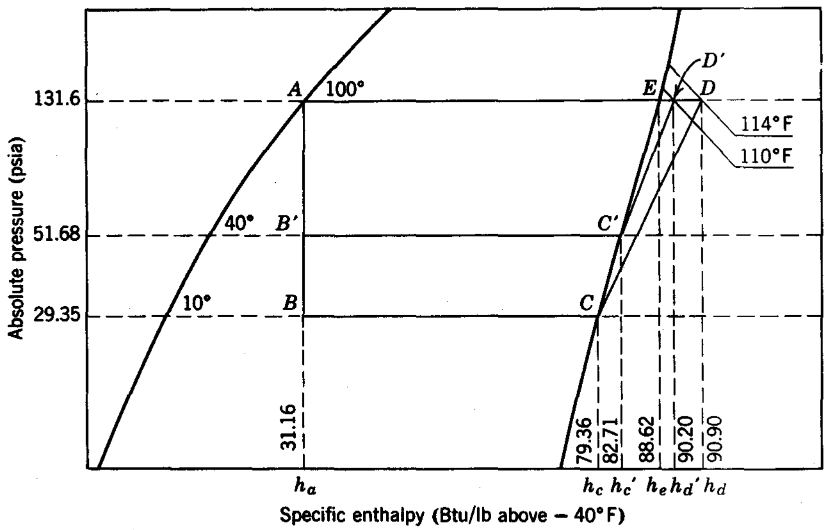


Fig. 7-6. Comparison of two simple saturated cycles operating at different vaporizing temperatures (figure distorted). (Refrigerant-12.)

temperatures. Of the two, the vaporizing temperature has by far the greater effect.

To illustrate the effect that varying the suction temperature has on cycle efficiency, cycle diagrams of two simple saturated cycles operating at different suction temperatures are drawn on the *Ph* chart in Fig. 7-6. One cycle, identified by the points *A*, *B*, *C*, *D*, and *E*, is operating at a vaporizing temperature of 10° F and a condensing temperature of 100° F. A similar cycle having the same condensing temperature but operating at a vaporizing temperature of 40° F is set off by the points *A*, *B'*, *C'*, *D'*, and *E*.

To facilitate a comparison of the two cycles, the following values have been determined from the *Ph* chart:

(*a*) For the 10° F cycle,

$q_1 = h_c - h_a = 79.36 - 31.16 = 48.20$ Btu/lb
$q_2 = h_d - h_c = 90.90 - 79.36 = 11.54$ Btu/lb
$q_3 = h_d - h_a = 90.90 - 31.16 = 59.74$ Btu/lb

(*b*) For the 40° F cycle,

$q_1 = h_{c'} - h_a = 82.71 - 31.16 = 51.55$ Btu/lb
$q_2 = h_{d'} - h_{c'} = 90.20 - 82.71 = 7.49$ Btu/lb
$q_3 = h_{d'} - h_a = 90.20 - 31.16 = 59.04$ Btu/lb

In comparing the two cycles, note that the refrigerating effect per pound of refrigerant circulated is greater for the cycle having the higher vaporizing temperature. The refrigerating effect for the cycle having the 10° F vaporizing temperature is 48.20 Btu/lb. When the vaporizing temperature of the cycle is raised to 40° F, the refrigerating effect increases to 51.55 Btu/lb. This represents an increase in the refrigerating effect per pound of

$$\frac{(h_{c'} - h_a) - (h_c - h_a)}{h_c - h_a} \times 100$$

$$= \frac{51.55 - 48.20}{48.20} \times 100$$

$$= 6.95\%$$

The greater refrigerating effect per pound of refrigerant circulated obtained at the higher vaporizing temperature is accounted for by the fact that there is a smaller temperature differential between the vaporizing temperature and the temperature of the liquid approaching the refrigerant control. Hence, at the higher suction temperature, a smaller fraction of the refrigerant vaporizes in the control and a greater portion vaporizes in the evaporator and produces useful cooling.

Since the refrigerating effect per pound is greater, the weight of refrigerant which must be circulated per minute per ton of refrigerating capacity is less at the higher suction temperature than at the lower suction temperature. Whereas the weight of refrigerant circulated per minute per ton for the 10° F cycle is

$$\frac{200}{h_c - h_a} = \frac{200}{48.20} = 4.15 \text{ lb/min}$$

The weight of refrigerant circulated per minute per ton for the 40° F cycle is only

$$\frac{200}{h_{c'} - h_a} = \frac{200}{51.55} = 3.88 \text{ lb/min}$$

The decrease in the weight of refrigerant circulated at the higher suction temperature is

$$\frac{4.15 - 3.88}{4.15} \times 100 = 6.5\%$$

Since the difference between the vaporizing and condensing pressures is smaller for the cycle having the higher suction temperature, the work of compression per pound required to compress the vapor from the vaporizing pressure to the condensing pressure is less for the higher temperature cycle than for the lower temperature cycle. It follows then that the heat of compression per pound for the cycle having the higher vaporizing temperature is also less than that for the cycle having the lower vaporizing temperature. The heat of compression per pound for the 10° F cycle is 11.54 Btu, whereas the heat of compression for the 40° F cycle is only 7.49 Btu. This represents a decrease in the heat of compression per pound of

$$\frac{(h_d - h_c) - (h_{d'} - h_{c'})}{h_d - h_c} = \frac{11.54 - 7.49}{11.54} \times 100 = 35.1\%$$

Because both the work of compression per pound and the weight of refrigerant circulated per minute per ton are less at the higher suction temperature, the work of compression per ton and therefore the theoretical horsepower required per ton will be smaller at the higher suction temperature. The theoretical horsepower required per ton of refrigerating capacity for the 10° F cycle is

$$\frac{m(h_d - h_c)}{42.42} = \frac{4.15 \times (90.90 - 79.36)}{42.42} = 1.13$$

For the 40° F cycle, the theoretical horsepower required per ton is

$$\frac{m(h_{d'} - h_{c'})}{42.42} = \frac{3.88 \times (90.20 - 82.71)}{42.42} = 0.683$$

In this instance, increasing the vaporizing temperature of the cycle from 10° F to 40° F reduces the theoretical horsepower required per ton by

$$\frac{1.13 - 0.683}{1.13} \times 100 = 39.5\%$$

Later, when the efficiency of the compressor is taken into consideration, it will be shown that the difference in the actual horsepower required per ton at the various suction temperatures is even greater than that indicated by theoretical computations.

Since the coefficient of performance is an index of the power required per unit of refrigerating capacity and, as such, is an indication of cycle efficiency, the relative efficiency of the two cycles can be determined by comparing their coefficients of performance. The coefficient of performance for the 10° F cycle is

$$\frac{h_c - h_a}{h_d - h_c} = \frac{48.20}{11.54} = 4.17$$

and the coefficient of performance for the 40° F cycle is

$$\frac{h_{c'} - h_a}{h_{d'} - h_{c'}}$$
$$= \frac{51.55}{7.49}$$
$$= 6.88$$

It is evident that the coefficient of performance, and hence the efficiency of the cycle, improves considerably as the vaporizing temperature increases. In this instance, increasing the suction temperature from 10° F to 40° F increases the efficiency of the cycle by

$$\frac{6.88 - 4.17}{4.17} \times 100 = 65\%$$

Although the difference in the weight of refrigerant which must be circulated per minute per ton of refrigerating capacity at the various suction temperatures is usually relatively small, the volume of vapor that the compressor must handle per minute per ton varies greatly with changes in the suction temperature. This is probably one of the most important factors influencing the capacity and efficiency of a vapor-compression refrigerating system and is the one which is the most likely to be overlooked by the student when studying cycle diagrams. The difference in the volume of vapor to be displaced per minute per ton at the various suction temperatures can be clearly illustrated by a comparison of the two cycles in question.

For the 10° F cycle, the volume of vapor to be displaced per minute per ton is

$$m(v) = 4.15 \times 1.351 = 5.6 \text{ cu ft}$$

whereas, at the 40° F suction temperature, the volume of vapor to be displaced per minute per ton is

$$m(v) = 3.88 \times 0.792 = 3.075 \text{ cu ft}$$

It is of interest to note that, whereas the decrease in the weight of refrigerant circulated per minute per ton at the higher suction temperature is only 6.5%, the decrease in the volume of vapor handled by the compressor per minute per ton is

$$\frac{5.6 - 3.075}{5.6} \times 100 = 45\%$$

Obviously, then, the smaller weight of refrigerant circulated per minute per ton accounts for only a very small part of the reduction in the volume of vapor displaced per minute per ton at the higher suction temperature. To a far greater extent, the decrease in the volume of vapor displaced per minute per ton is a result of the lower specific volume of the suction vapor which is coincident with the higher suction temperature (0.792 cu ft/lb at 40° F as compared to 1.351 cu ft/lb at 10° F). This aspect of system capacity and efficiency in relation to suction temperature will be further investigated in conjunction with compressor performance in Chapter 12.

The quantity of heat to be rejected at the condenser per minute per ton is much smaller for the cycle having the higher suction temperature. This is true even though the quantity of heat rejected at the condenser per pound of refrigerant circulated is nearly the same for both cycles. For the 10° F cycle, the quantity of heat rejected at the condenser per minute per ton is

$$m(h_d - h_a) = 4.15 \times 59.74 = 247.92$$

whereas for the 40° F cycle the heat rejected at the condenser per minute per ton is only

$$m(h_{d'} - h_a) = 3.88 \times 59.04 = 229.08 \text{ Btu}$$

The quantity of heat rejected per minute per ton at the condenser is less for the higher suction temperature because of (1) the smaller weight of refrigerant circulated per minute per ton and (2) the smaller heat of compression per pound.

It has been shown previously that the heat rejected at the condenser per pound of refrigerant circulated is the sum of the heat absorbed in the evaporator per pound (refrigerating effect) and the heat of compression per pound. Since increasing the vaporizing temperature of the cycle brings about an increase in the refrigerating effect as well as a decrease in the heat of compression, the quantity of heat rejected at the condenser per pound remains very nearly the same for both cycles (59.74 at 10° F as compared to 59.04 at 40° F). In general, this is true for all suction temperatures because any increase or decrease in the heat of compression is usually accompanied by an offsetting increase or decrease in the refrigerating effect.

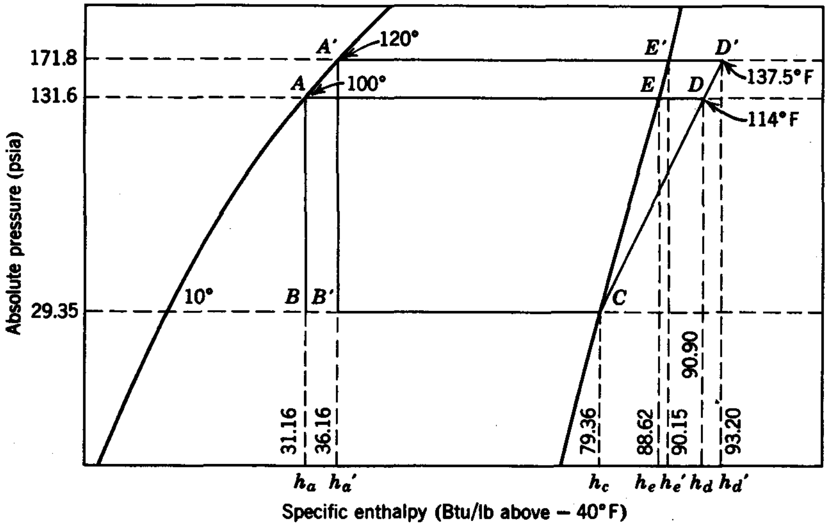


Fig. 7-7. Comparison of two simple saturated cycles operating at different condensing temperatures (figure distorted). (Refrigerant-12.)

7-11. Effect of Condensing Temperature on Cycle Efficiency. Although the variations in cycle efficiency with changes in the condensing temperature are not as great as those brought about by changes in the vaporizing temperature, they are nonetheless important. In general, if the vaporizing temperature remains constant, the efficiency of the cycle decreases as the condensing temperature increases, and increases as the condensing temperature decreases.

To illustrate the effect of condensing temperature on cycle efficiency, cycle diagrams of two saturated cycles operating at different condensing temperatures are drawn on the *Ph* chart in Fig. 7-7. One cycle, *A*, *B*, *C*, *D*, and *E*, has a condensing temperature of 100° F, whereas the other cycle, *A'*, *B'*, *C*, *D'*, and *E'*, is operating at a condensing temperature of 120° F. The evaporating temperature of both cycles is 10° F. Values for cycle *A–B–C–D–E* have been determined in the previous section. Values for cycle *A'–B'–C–D'–E'* are as follows:

From the *Ph* diagram,

$$q_1 = h_c - h_{a'} = 79.36 - 36.16 = 43.20 \text{ Btu/lb}$$
$$q_2 = h_{d'} - h_c = 93.20 - 79.36 = 13.84 \text{ Btu/lb}$$
$$q_3 = h_{d'} - h_{a'} = 93.20 - 36.16 = 57.04 \text{ Btu/lb}$$

In a simple saturated cycle the liquid refrigerant reaches the refrigerant control at the condensing temperature. Therefore, as the condensing temperature is increased, the temperature of the liquid approaching the refrigerant control is increased and the refrigerating effect per pound is reduced. In this instance, the refrigerating effect is reduced from 48.20 Btu/lb to 43.20 Btu/lb when the condensing temperature is increased from 100° F to 120° F. This is a reduction of

$$\frac{48.20 - 43.20}{48.20} \times 100$$
$$= 10.37\%$$

Because the refrigerating effect per pound is less for the cycle having the higher condensing temperature, the weight of refrigerant to be circulated per minute per ton must be greater. For the cycle having the 100° F condensing temperature the weight of refrigerant to be circulated per minute per ton is 4.15 lb. When the condensing temperature is increased to 120° F, the weight of refrigerant which must be circulated per minute per ton increases to

$$\frac{200}{43.20} = 4.63 \text{ lb}$$

This is an increase of

$$\frac{4.63 - 4.15}{4.15} \times 100 = 11.57\%$$

Since the weight of refrigerant which must be circulated per minute per ton is greater at the higher condensing temperature, it follows that the volume of vapor to be compressed per minute per ton must also be greater. In a simple saturated cycle the specific volume of the suction vapor varies only with the vaporizing temperature. As the vaporizing temperature is the same for both cycles, the specific volume of the vapor leaving the evaporator is also the same for both cycles and therefore the difference in the volume of vapor to be compressed per minute per ton is in direct proportion to the difference in the weight of refrigerant circulated per minute per ton. At the 100° F condensing temperature the volume of vapor to be compressed per minute per ton is 5.6 cu ft, whereas at the 120° F condensing temperature the volume of vapor compressed per minute per ton increases to

$$4.63 \times 1.351 = 6.25 \text{ cu ft}$$

This represents an increase in the volume of vapor compressed per minute per ton of

$$\frac{6.25 - 5.6}{5.6} \times 100 = 11.57\%$$

Note that the percent increase in the volume of vapor handled by the compressor is exactly equal to the percent increase in the weight of refrigerant circulated. Contrast this with what occurs when the vaporizing temperature is varied.

Since the difference between the vaporizing and condensing pressures is greater, the work of compression per pound of refrigerant circulated required to raise the pressure of the vapor from the vaporizing to the condensing pressure is also greater for the cycle having the higher condensing temperature. In this instance, the heat of compression increases from 11.54 Btu/lb for the 100° F condensing temperature to 13.84 Btu/lb for the 120° F condensing temperature. This is an increase of

$$\frac{13.84 - 11.54}{11.54} \times 100 = 20\%$$

As a result of the greater work of compression per pound and the greater weight of refrigerant circulated per minute per ton, the theoretical horsepower required per ton of refrigerating capacity increases as the condensing temperature increases. Whereas the theoretical horsepower required per ton at the 100° F condensing temperature is only 1.13 hp when the condensing temperature is increased to 120° F, the theoretical horsepower per ton increases to

$$\frac{4.63 \times 13.84}{42.42} = 1.52 \text{ hp}$$

This is an increase in the power required per ton of

$$\frac{1.52 - 1.13}{1.13} \times 100 = 34.6\%$$

Note that the increase in the horsepower required per ton at the higher condensing temperature is greater than the increase in the work of compression per pound. This is accounted for by the fact that, in addition to the 20% increase in the work of compression per pound, there is also a 6.5% increase in the weight of refrigerant circulated per minute per ton.

The coefficient of performance of the cycle at the 100° F condensing temperature is 4.17. When the condensing temperature is raised to 120° F, the coefficient of performance drops to

$$\frac{43.20}{13.84} = 3.12$$

Since the coefficient of performance is an index of the refrigerating capacity per unit of power, the decrease in refrigerating capacity per unit of power in this instance is

$$\frac{4.17 - 3.12}{3.12} \times 100 = 33.7\%$$

Obviously, the effect of raising the condensing temperature on cycle efficiency is the exact opposite of that of raising the evaporating temperature. Whereas raising the evaporating temperature increases the refrigerating effect per pound and reduces the work of compression so that the refrigerating capacity per unit of power increases, raising the condensing temperature reduces the refrigerating effect per pound and increases the work of compression so that the refrigerating capacity per unit of power decreases.

Although the quantity of heat rejected at the condenser per pound of refrigerant circulated varies only slightly with changes in the condensing temperature because any change in the heat of compression is accompanied by an offsetting change in the refrigerating effect per pound,

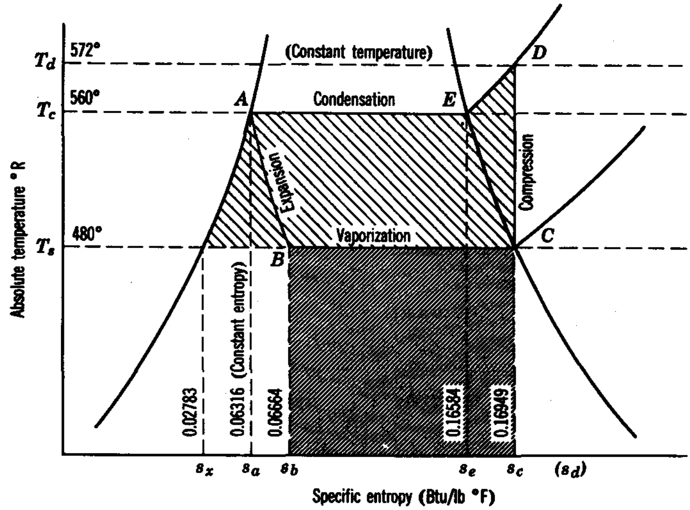


Fig. 7-8. Temperature-entropy diagram of simple saturated cycle on skeleton *Ts* chart (figure distorted). (Refrigerant-12.)

the total heat rejected at the condenser per minute per ton varies considerably with changes in the condensing temperature because of the difference in the weight of refrigerant circulated per minute per ton. It was shown in Section 7-7 that the total heat rejected at the condenser per minute per ton (Q_3) is always the sum of the heat absorbed in the evaporator per minute per ton (Q_1) and the total heat of compression per ton (Q_2). Since Q_1 is always 200 Btu/min/ton, then Q_3 will vary only with Q_2, the heat of compression per minute per ton. Furthermore, since Q_2 always increases as the condensing temperature increases, it follows then that Q_3 also increases as the condensing temperature increases.

For the two cycles in question, the heat rejected at the condenser per minute per ton at the 100° F condensing temperature is 218.75 Btu. For the 120° F condensing temperature, the quantity of heat rejected at the condenser per minute per ton increases to

$$4.63 \times (43.20 + 13.84) = 310.4 \text{ Btu}$$

The percent increase is

$$\frac{310.40 - 218.75}{218.75} \times 100 = 41.8\%$$

It is interesting to note also that the amount of sensible heat rejected at the condenser increases considerably at the higher condensing temperature, whereas the amount of latent heat rejected diminishes slightly. This indicates that, at the higher condensing temperature, a greater portion of the condenser surface is being used merely to reduce the temperature of the discharge vapor to the condensing temperature.

7-12. The Temperature-Entropy Diagram. Although the author is partial to the pressure-enthalpy diagram, there are many who prefer to use the temperature-entropy diagram to analyze the refrigeration cycle. To acquaint the student with the use of *Ts* diagrams in cycle analysis, a diagram of the simple saturated cycle described in the foregoing sections is drawn on *Ts* coordinates in Fig. 7-8. The state points *A*, *B*, *C*, *D*, and *E* represent the points in the cycle as shown by the flow diagram in Fig. 7-5. The state point *X* represents saturated liquid at the vaporizing temperature. Process *A–B* is the irreversible adiabatic expansion through the refrigerant control.* Process *B–C* is the isobaric and isothermal vaporization in the evaporator. Process *C–D* is the reversible adiabatic (isentropic) compression in the compressor and processes *D–E* and *E–A* are the desuperheating and condensing processes in the condenser. T_s, T_c, and T_d are the absolute suction, absolute condensing, and absolute discharge temperatures, respectively, whereas s_a, s_b, s_c, s_d, s_e, and s_x are the specific entropies of the refrigerant at the various state points.

The principal advantage of the *Ts* diagram is that the areas shown on the chart represent actual heat quantities. In Fig. 7-8, the area

* The line *A–B* does not necessarily follow the actual path of process *A–B*. See Section 7-4.

Condensing Temperature, 100° F   Condensing Pressure, 136.16 Psia

Suction temperature	50°	40°	30°	20°	10°	0°	−10°	−20°	−30°	−40°
Absolute suction pressure	61.39	51.68	43.16	35.75	29.35	23.87	19.20	15.28	12.02	9.32
Refrigerating effect per pound	52.62	51.55	50.45	49.33	48.20	47.05	45.89	44.71	43.54	42.34
Weight of refrigerant circulated per minute per ton	3.80	3.88	3.97	4.05	4.15	4.25	4.36	4.48	4.59	4.73
Specific volume of suction vapor	0.673	0.792	0.939	1.121	1.351	1.637	2.00	2.47	3.09	3.91
Volume of vapor compressed per minute per ton	2.56	3.08	3.77	4.55	5.60	6.96	8.72	11.10	14.20	18.50
Heat of compression per pound	6.01	7.49	8.79	10.11	11.54	13.29	14.85	16.73	18.50	20.40
Theoretical horsepower per ton	0.539	0.683	0.818	0.965	1.13	1.35	1.54	1.78	2.00	2.26
Coefficient of performance	8.76	6.88	5.74	4.88	4.17	3.54	3.09	2.67	2.36	2.07

Fig. 7-9

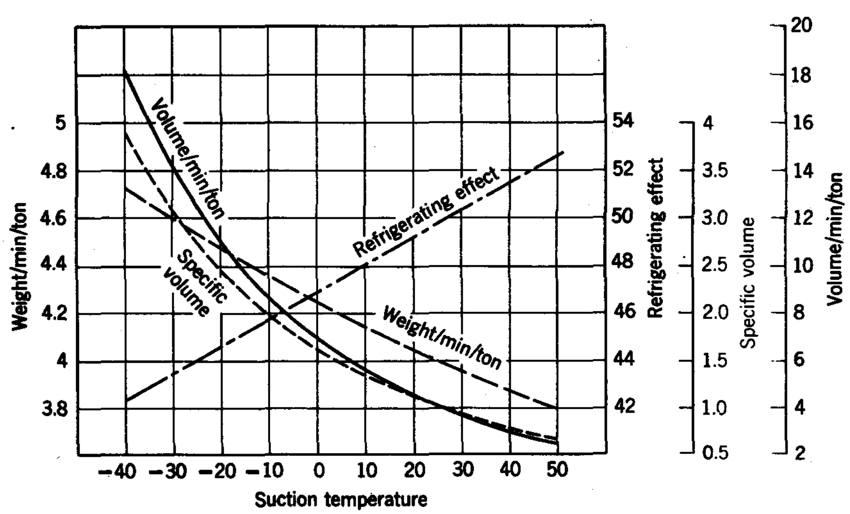


Fig. 7-10a. For Refrigerant-12, the refrigerating effect per pound, the weight of refrigerant circulated per minute per ton, the specific volume of the suction vapor, and the volume of vapor compressed per minute per ton are each plotted against suction temperature. Condensing temperature is constant at 100° F.

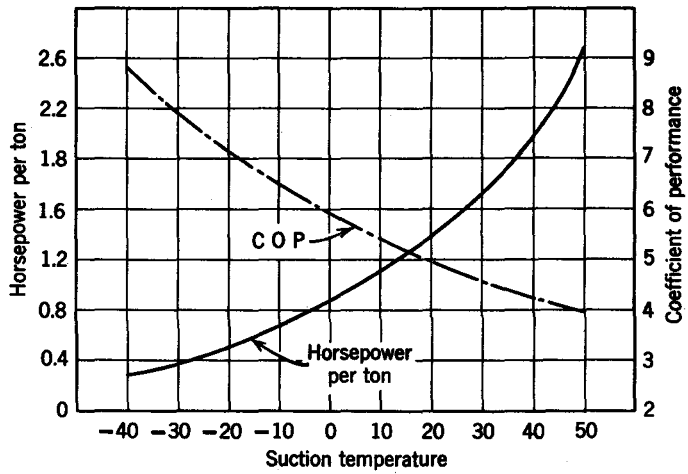


Fig. 7-10b. For Refrigerant-12, the coefficient of performance and the horsepower per ton are plotted against suction temperature. Condensing temperature is constant at 100° F.

A–X–C–D–E–A represents the heat energy equivalent of the work of adiabatic compression and, since the distance between the base line and T_s, foreshortened in the figure, represents the absolute vaporizing temperature of the liquid, area B–C–s_c–s_b–B represents the refrigerating effect per pound. The sum of the areas *A–X–C–D–E–A* and B–C–s_c–s_a–B, of course, represents the heat rejected at the condenser per pound. As in the case of the *Ph* diagram, it can be readily seen on the *Ts* chart that either lowering the vaporizing temperature or raising the condensing temperature tends to increase the work of compression, reduce the refrigerating effect per pound, and lower the efficiency of the cycle.

7-13. Summary. Regardless of the method used to analyze the cycle, it is evident that the capacity and efficiency of a refrigerating system improve as the vaporizing temperature increases and as the condensing temperature decreases. Obviously, then, a refrigerating system should always be designed to operate at the highest possible vaporizing temperature and the lowest possible condensing temperature commensurate with the requirements of the application. This will nearly always permit the most effective use of the smallest possible equipment and thereby effect a savings not only in the initial cost of the equipment but also in the operating expenses.

In any event, the influence of the vaporizing and condensing temperatures on cycle efficiency is of sufficient importance to warrant a more intensive study. To aid the student in this, the relationship between the refrigerating effect per pound, the weight of refrigerant circulated per minute per ton, the specific volume of the suction vapor, the volume of vapor compressed per minute per ton, the horsepower required per ton, and the coefficient of performance of the cycle has been calculated for various suction temperatures. These values are given in tabular form in Fig. 7-9 and are illustrated graphically in Figs. 7-10*a* and 7-10*b*. In addition, the effect of condensing temperature on the horsepower required per ton of refrigerating capacity is shown for several condensing temperatures in Fig. 7-11.

Since the properties of the refrigerant at point *D* on the cycle diagram cannot ordinarily be obtained from the refrigerant tables and since these properties are difficult to read accurately from the *Ph* chart because of the size of the chart, the approximate isentropic discharge

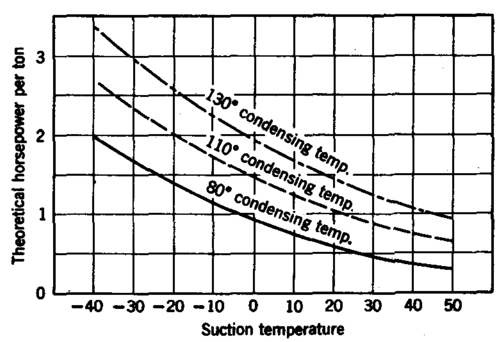


Fig. 7-11. The effect of condensing temperature on the horsepower per ton.

temperatures and the approximate enthalpy of the refrigerant vapor at point *D* have been compiled for a variety of vaporizing and condensing temperatures and are given in Table 7-1 to aid the student in arriving at more accurate solutions to the problems at the end of the chapter.

PROBLEMS

1. A Refrigerant-12 system operating on a simple saturated cycle has an evaporating temperature of 0° F and a condensing temperature of 110° F. Determine:

A. (1) The refrigerating effect per pound of refrigerant circulated.
Ans. 44.56 Btu/lb

(2) The weight of refrigerant circulated per minute per ton.
Ans. 4.49 lb/min/ton

(3) The volume of vapor compressed per minute per ton.
Ans. 7.35 cu ft/min/ton

B. (1) The heat of compression per pound of refrigerant circulated.
Ans. 14.39 Btu/lb

(2) The heat of compression per minute per ton of refrigeration.
Ans. 64.61 Btu/min/ton

(3) The work of compression per minute per ton in foot-pounds.
Ans. 50.267 lb/min/ton

(4) The theoretical horsepower per ton.
Ans. 1.52 hp/ton

(5) The coefficient of performance.
Ans. 3.1

C. (1) The heat rejected per minute per ton at the condenser.
Ans. 264.61 Btu/min/ton

8

Actual Refrigerating Cycles

8-1. Deviation from the Simple Saturated Cycle. Actual refrigerating cycles deviate somewhat from the simple saturated cycle. The reason for this is that certain assumptions are made for the simple saturated cycle which do not hold true for actual cycles. For example, in the simple saturated cycle, the drop in pressure in the lines and across the evaporator, condenser, etc., resulting from the flow of the refrigerant through these parts is neglected. Furthermore, the effects of subcooling the liquid and of superheating the suction vapor are not considered. Too, compression in the compressor is assumed to be true isentropic compression. In the following sections all these things are taken into account and their effect on the cycle is studied in detail.*

8-2. The Effect of Superheating the Suction Vapor. In the simple saturated cycle, the suction vapor is assumed to reach the suction inlet of the compressor as a saturated vapor at the vaporizing temperature and pressure. In actual practice, this is rarely true. After the liquid refrigerant has completely vaporized in the evaporator, the cold, saturated vapor will usually continue to absorb heat and thereby become superheated before it reaches the compressor (Fig. 8-1).

* The departure from true isentropic compression and the effect that it has on the cycle are discussed in Chapter 12.

On the *Ph* diagram in Fig. 8-2, a simple saturated cycle is compared to one in which the suction vapor is superheated from 20° F to 70° F. Points *A*, *B*, *C*, *D*, and *E* mark the saturated cycle, and points *A*, *B*, *C′*, *D′*, and *E* indicate the superheated cycle.

If the slight pressure drop resulting from the flow of the vapor in the suction piping is neglected, it may be assumed that the pressure of the suction vapor remains constant during the superheating. That is, after the superheating, the pressure of the vapor at the suction inlet of the compressor is still the same as the vaporizing pressure in the evaporator. With this assumption, point *C′* can be located on the *Ph* chart by following a line of constant pressure from point *C* to the point where the line of constant pressure intersects the 70° F constant temperature line. Point *D′* is found by following a line of constant entropy from point *C′* to the line of constant pressure corresponding to the condensing pressure.

In Fig. 8-2, the properties of the superheated vapor at points *C′* and *D′*, as read from the *Ph* chart, are as follows:

At point *C′*,

$$p = 35.75 \text{ psia} \qquad t = 70^\circ \text{ F}$$
$$v = 1.260 \text{ cu ft/lb} \qquad h = 88.6 \text{ Btu/lb}$$
$$s = 0.1840 \text{ Btu/lb/}^\circ \text{ R}$$

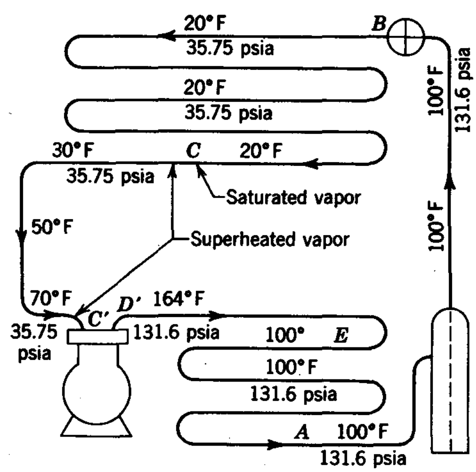


Fig. 8-1. Flow diagram of superheated cycle. Liquid completely vaporized at point *C*—saturated vapor continues to absorb heat while flowing from *C* to *C′*—vapor reaches compressor in superheated condition. Notice the high discharge temperature. (Refrigerant-12.)

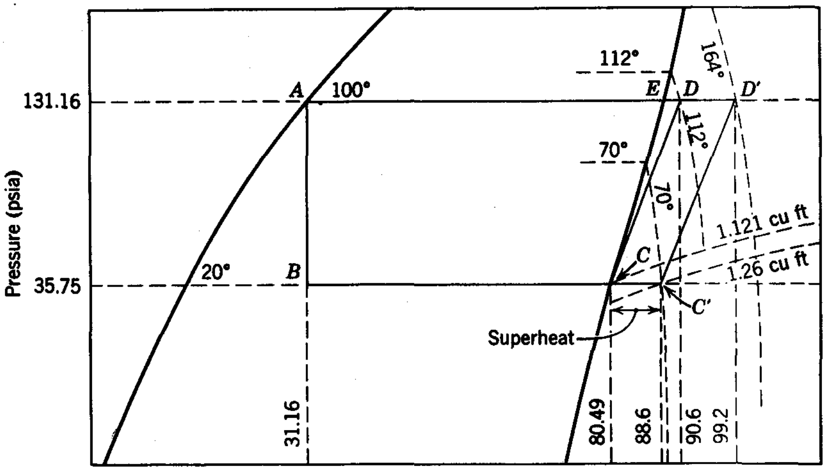


Fig. 8-2. *Ph* **diagram comparing simple saturated cycle to the superheated cycle. (Refrigerant-12).**

At point D',

$$p = 131.6 \text{ psia} \qquad t = 164° \text{ F}$$
$$v = 0.380 \text{ cu ft/lb} \qquad h = 99.2 \text{ Btu/lb}$$
$$s = 0.1840 \text{ Btu/lb/° R}$$

On the *Ph* chart, process C–C' represents the superheating of the suction vapor from 20° F to 70° F at the vaporizing pressure, and the difference between the enthalpy of the vapor at these points is the amount of heat required to superheat each pound of refrigerant. In comparing the two cycles, the following observations are of interest:

1. The heat of compression per pound for the superheated cycle is slightly greater than that for the saturated cycle. For the superheated cycle, the heat of compression is

$$h_{d'} - h_{c'} = 99.2 - 88.6 = 10.6 \text{ Btu/lb}$$

whereas for the saturated cycle the heat of compression is

$$h_d - h_c = 90.6 - 80.49 = 10.11 \text{ Btu/lb}$$

In this instance, the heat of compression per pound is greater for the superheated cycle by

$$\frac{10.6 - 10.11}{10.11} \times 100 = 4.84\%$$

2. For the same condensing temperature and pressure, the temperature of the discharge vapor leaving the head of the compressor is considerably higher for the superheated cycle than for the saturated cycle—in this case, 164° F for the superheated cycle as compared to 112° F for the saturated cycle.

3. For the superheated cycle, a greater quantity of heat must be dissipated at the condenser per pound than for the saturated cycle. This is because of the additional heat absorbed by the vapor in becoming superheated and because of the small increase in the heat of compression per pound. For the superheated cycle, the heat dissipated at the condenser per pound is

$$h_{d'} - h_a = 99.2 - 31.16 = 68.04 \text{ Btu/lb}$$

and for the saturated cycle the heat dissipated at the condenser per pound is

$$h_d - h_a = 90.6 - 31.16 = 59.44 \text{ Btu/lb}$$

The percent increase in the heat dissipated at the condenser per pound for the superheated cycle is

$$\frac{68.04 - 59.44}{59.44} \times 100 = 14.4\%$$

Note that the additional heat which must be dissipated per pound at the condenser in the superheated cycle is all sensible heat. The amount of latent heat dissipated per pound is the same for both cycles. This means that in the superheated cycle a greater amount of sensible heat must be given up to the condensing medium before condensation begins and that a greater portion of the condenser will be used in cooling the discharge vapor to its saturation temperature.

Notice also that, since the pressure of the suction vapor remains constant during the superheating, the volume of the vapor increases with the temperature approximately in accordance with Charles' law.* Therefore, a pound of superheated vapor will always occupy a greater volume than a pound of saturated vapor at the same pressure. For example, in Fig. 8-2, the specific volume of the suction vapor increases from 1.121 cu ft per pound at saturation to 1.260 cu ft per pound when superheated to 70° F. This means that for each pound of refrigerant circulated, the compressor must compress a greater volume of vapor if the vapor is superheated than if the vapor is saturated. For this reason, in every instance where the vapor is allowed to become superheated before it reaches the compressor, the weight of refrigerant circulated by a compressor of any given displacement will always be less than when the suction vapor reaches the compressor in a saturated condition, provided the pressure is the same.

The effect that superheating of the suction vapor has on the capacity of the system and on the coefficient of performance depends entirely upon where and how the superheating of the vapor occurs and upon whether or not the heat absorbed by the vapor in becoming superheated produces useful cooling.*

8-3. Superheating without Useful Cooling. Assume first that the superheating of the suction vapor occurs in such a way that no useful cooling results. When this is true, the refrigerating effect per pound of refrigerant circulated is the same for the superheated cycle as for a saturated cycle operating at the same vaporizing and condensing temperatures, and therefore the weight of refrigerant circulated per minute per ton will also be the same for both the superheated and saturated cycles. Then, for both cycles illustrated in Fig. 8-2,

The weight of refrigerant circulated per minute per ton m

$$= \frac{200}{h_c - h_a} = \frac{200}{49.33} = 4.05 \text{ lb/min/ton}$$

Since the weight of refrigerant circulated is the same for both the superheated and saturated cycles and since the specific volume of the vapor at the compressor inlet is greater for the superheated cycle than for the saturated cycle, it follows that the volume of vapor that the compressor must handle per minute per ton of refrigerating capacity is greater for the superheated cycle than for the saturated cycle.

For the saturated cycle, the specific volume of the suction vapor v_c $= 1.121$ cu ft/lb

The volume of vapor compressed per minute per ton V

$$= m \times v = 4.05 \times 1.121 = 4.55 \text{ cu ft/min/ton}$$

For the superheated cycle, the specific volume of the suction vapor $v_{c'}$ $= 1.260$ cu ft/lb

The volume of vapor compressed per minute per ton V

$$= m \times v = 4.05 \times 1.260 = 5.02 \text{ cu ft/min/ton}$$

In regard to percentage, the increase in the volume of vapor which must be handled by a compressor operating on the superheated cycle is

$$\frac{5.02 - 4.55}{4.55} \times 100 = 10.3\%$$

This means, of course, that a compressor operating on the superheated cycle must be 10.3% larger than the one required for the saturated cycle.

Again, since the weight of refrigerant circulated per minute per ton is the same for both cycles and since the heat of compression per pound is greater for the superheated cycle than for the saturated cycle, it is evident that the horsepower per ton is greater for the superheated cycle and the coefficient of performance is less.

* The temperature and volume of the vapor do not vary exactly in accordance with Charles' law because the refrigerant vapor is not a perfect gas.

* The effects of superheating depend also upon the refrigerant used. The discussion in this chapter is limited to systems using R-12. The effects of superheating on systems using other refrigerants are discussed later.

For the saturated cycle, the horsepower per ton

$$= \frac{m(h_d - h_c)}{42.42} = \frac{4.05 \times 10.11}{42.42} = 0.965 \text{ hp/ton}$$

The coefficient of performance

$$= \frac{h_c - h_a}{h_d - h_c} = \frac{49.33}{10.11} = 4.88$$

For the superheated cycle, the horsepower per ton

$$= \frac{m(h_{d'} - h_{c'})}{42.42} = \frac{4.05 \times 10.6}{42.42} = 1.01 \text{ hp/ton}$$

The coefficient of performance

$$= \frac{h_c - h_a}{h_{d'} - h_{c'}} = \frac{49.33}{10.60} = 4.65$$

In summary, when superheating of the vapor occurs without producing useful cooling, the volume of vapor compressed per minute per ton, the horsepower per ton, and the quantity of heat given up in the condenser per minute per ton are all greater for the superheated cycle than for the saturated cycle. This means that the compressor, the compressor driver, and the condenser must all be larger for the superheated cycle than for the saturated cycle.

8-4. Superheating That Produces Useful Cooling. Assume, now, that all of the heat taken in by the suction vapor produces useful cooling. When this is true, the refrigerating effect per pound is greater by an amount equal to the amount of superheat. In Fig. 8-2, assuming that the superheating produces useful cooling, the refrigerating effect per pound for the superheated cycle is equal to

$$h_{c'} - h_a = 88.60 - 31.16 = 57.44 \text{ Btu/lb}$$

Since the refrigerating effect per pound is greater for the superheated cycle than for the saturated cycle, the weight of refrigerant circulated per minute per ton is less for the superheated cycle than for the saturated cycle. Whereas the weight of refrigerant circulated per minute per ton for the saturated cycle is 4.05, the weight of refrigerant circulated per minute per ton for the superheated cycle is

$$\frac{200}{h_{c'} - h_a} = \frac{200}{57.44} = 3.48 \text{ lb/min/ton}$$

Notice that, even though the specific volume of the suction vapor and the heat of compression per pound are both greater for the superheated than for the saturated cycle, the volume of vapor compressed per minute per ton and the horsepower per ton are less for the superheated cycle than for the saturated cycle. This is because of the reduction in the weight of refrigerant circulated. The volume of vapor compressed per minute per ton and the horsepower per ton for the saturated cycle are 4.55 cu ft and 0.965 hp, respectively, whereas for the superheated cycle

The volume of vapor compressed per minute per ton V

$$= m \times v_{c'} = 3.48 \times 1.260 = 4.38 \text{ cu ft/min/ton}$$

The horsepower per ton

$$= \frac{m(h_{d'} - h_{c'})}{42.42} = \frac{3.48 \times 10.60}{42.42} = 0.870 \text{ hp/ton}$$

For the superheated cycle, both the refrigerating effect per pound and the heat of compression per pound are greater than for the saturated cycle. However, since the increase in the refrigerating effect is greater proportionally than the increase in the heat of compression, the coefficient of performance for the superheated cycle is higher than that of the saturated cycle. For the saturated cycle, the coefficient of performance is 4.69, whereas for the superheated cycle

The coefficient of performance

$$= \frac{(h_{c'} - h_a)}{(h_{d'} - h_{c'})} = \frac{57.44}{10.60} = 5.42$$

It will be shown in the following sections that the superheating of the suction vapor in an actual cycle usually occurs in such a way that a part of the heat taken in by the vapor in becoming superheated is absorbed from the refrigerated space and produces useful cooling, whereas another part is absorbed by the vapor after the vapor leaves the refrigerated space and therefore produces no useful cooling. The portion of the superheat which produces useful cooling will depend upon the individual application, and the effect of the superheating on the cycle will vary approximately in proportion to the useful cooling accomplished.

Regardless of the effect on capacity, except in some few special cases, a certain amount of superheating is nearly always necessary and, in most cases, desirable. When the suction vapor is drawn directly from the evaporator into the suction inlet of the compressor without at least a small amount of superheating, there is a good possibility that small particles of unvaporized liquid will be entrained in the vapor. Such a vapor is called a "wet" vapor. It will be shown later that "wet" suction vapor drawn into the cylinder of the compressor adversely affects the capacity of the compressor. Furthermore, since refrigeration compressors are designed as vapor pumps, if any appreciable amount of unvaporized liquid is allowed to enter the compressor from the suction line, serious mechanical damage to the compressor may result. Since superheating the suction vapor eliminates the possibility of "wet" suction vapor reaching the compressor inlet, a certain amount of superheating is usually desirable. Again, the extent to which the suction vapor should be allowed to become superheated in any particular instance depends upon where and how the superheating occurs and upon the refrigerant used.

Superheating of the suction vapor may take place in any one or in any combination of the following places:

1. In the end of the evaporator
2. In the suction piping installed inside the refrigerated space (usually referred to as a "drier loop")
3. In the suction piping located outside of the refrigerated space
4. In a liquid-suction heat exchanger.

8-5. Superheating in Suction Piping outside the Refrigerated Space. When the cool refrigerant vapor from the evaporator is allowed to become superheated while flowing through suction piping located outside of the refrigerated space, the heat taken in by the vapor is absorbed from the surrounding air and no useful cooling results. It has already been demonstrated that superheating of the suction vapor which produces no useful cooling adversely affects the efficiency of the cycle. Obviously, then, superheating of the vapor in the suction line outside of the refrigerated space should be eliminated whenever practical.

Superheating of the suction vapor in the suction line can be prevented for the most part by insulating the suction line. Whether or not the loss of cycle efficiency in any particular application is sufficient to warrant the additional expense of insulating the suction line depends primarily on the size of the system and on the operating suction temperature.

When the suction temperature is relatively high (35° F or 40° F), the amount of superheating will usually be small and the effect on the efficiency of the cycle will be negligible. The reverse is true, however, when the suction temperature is low. The amount of superheating is apt to be quite large.

Too, at low suction temperatures, when the efficiency of the cycle is already very low, each degree of superheat will cause a greater reduction in cycle efficiency percentagewise than when the suction temperature is high. It becomes immediately apparent that any appreciable amount of superheating in the suction line of systems operating at low suction temperatures will seriously reduce the efficiency of the cycle and that, under these conditions, insulating of the suction line is not only desirable but absolutely necessary if the efficiency of the cycle is to be maintained at a reasonable level.

Aside from any considerations of capacity, even at the higher suction temperatures, insulating of the suction line is often required to prevent frosting or sweating of the suction line. In flowing through the suction piping, the cold suction vapor will usually lower the temperature of the piping below the dew point temperature of the surrounding air so that moisture will condense out of the air onto the surface of the piping, causing the suction piping to either frost or sweat, depending upon whether or not the temperature of the piping is below the freezing temperature of water. In any event, frosting or sweating of the suction piping is usually undesirable and should be eliminated by insulating the piping.

8-6. Superheating the Vapor inside the Refrigerated Space. Superheating of the suction vapor inside the refrigerated space can take place either in the end of the evaporator or in suction piping located inside the refrigerated space, or both.

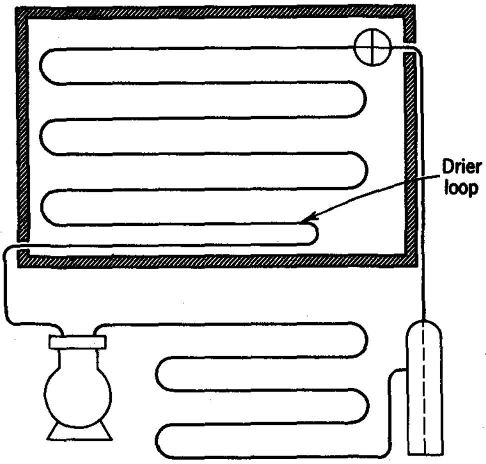


Fig. 8-3. Flow diagram showing drier loop for superheating suction vapor inside refrigerated space.

To assure the proper operation of the refrigerant control and to prevent liquid refrigerant from overflowing the evaporator and being carried back to the compressor, when certain types of refrigerant controls are used, it is necessary to adjust the control so that the liquid is completely evaporated before it reaches the end of the evaporator. In such cases, the cold vapor will continue to absorb heat and become superheated as it flows through the latter portion of the evaporator. Since the heat to superheat the vapor is drawn from the refrigerated space, useful cooling results and the refrigerating effect of each pound of refrigerant is increased by an amount equal to the amount of heat absorbed in the superheating.

It has been shown that when the superheating of the suction vapor produces useful cooling the efficiency of the cycle is improved somewhat.* However, in spite of the increase in cycle efficiency, it must be emphasized that superheating the suction vapor in the evaporator is not economical and should always be limited to only that amount which is necessary to the proper operation of the refrigerant control. Since the transfer of heat through the walls of the evaporator per degree of temperature difference is not as great to a vapor as to a liquid, the capacity of the evaporator is always reduced in any portion of the evaporator where only vapor exists. Therefore, excessive superheating of the suction vapor in the evaporator will reduce the capacity of the evaporator unnecessarily and will require either that the evaporator be operated at a lower vaporizing temperature or that a larger evaporator be used in order to provide the desired evaporator capacity. Neither of these is desirable nor practical. Since the space available for evaporator installation is often limited and since evaporator surface is expensive, the use of a larger evaporator is not practical. Because of the effect on cycle efficiency, the undesirability of lowering the vaporizing temperature is obvious.

Often, a certain amount of suction piping, usually called a drier loop, is installed inside the refrigerated space for the express purpose of superheating the suction vapor (Fig. 8-3). Use of a drier loop permits more complete flooding of the evaporator with liquid refrigerant without the danger of the liquid overflowing into the suction line and being drawn into the compressor. This not only provides a means of superheating the suction vapor inside the refrigerated space so that the efficiency of the cycle is increased without the sacrifice of expensive evaporator surface, but it actually makes possible more effective use of the existing evaporator surface. Also, in some instances, particularly where the suction temperature is high and the relative humidity of the outside air is reasonably low, superheating of the suction vapor inside the refrigerated space will raise the temperature of the suction piping and prevent the formation of moisture, thereby eliminating the need for suction line insulation. It should be noted, however, that the extent to which the suction vapor can be superheated inside the refrigerated space is limited by the space temperature. Ordinarily, if sufficient piping is used, the suction vapor can be heated to within 4° F to 5° F of the space temperature. Thus, for a 40° F space temperature, the suction vapor may be superheated to approximately 35° F.

8-7. The Effects of Subcooling the Liquid. On the *Ph* diagram in Fig. 8-4, a simple saturated cycle is compared to one in which the liquid is subcooled from 100° F to 80° F before it reaches the refrigerant control. Points *A*, *B*, *C*, *D*, and *E* designate the simple saturated cycle, whereas

* Although this is true for systems using R-12 as a refrigerant, it will be shown later that this is not true for all refrigerants.

points A', B', C, D, and E designate the subcooled cycle.

It has been shown (Section 6-28) that when the liquid is subcooled before it reaches the refrigerant control the refrigerating effect per pound is increased. In Fig. 8-4, the increase in the refrigerating effect per pound resulting from the subcooling is the difference between h_b and $h_{b'}$, and is exactly equal to the difference between h_a and $h_{a'}$, which represents the heat removed from the liquid per pound during the subcooling.

For the saturated cycle, the refrigerating effect per pound, q_1	$= h_c - h_a$ $= 80.49 - 31.16$ $= 49.33$ Btu/lb
For the subcooled cycle, the refrigerating effect per pound, q_1	$= h_c - h_{a'}$ $= 80.49 - 26.28$ $= 54.21$ Btu/lb

Because of the greater refrigerating effect per pound, the weight of refrigerant circulated per minute per ton is less for the subcooled cycle than for the saturated cycle.

For the saturated cycle, the weight of refrigerant circulated per minute per ton m	$= \dfrac{200}{49.33}$ $= 4.05$ lb
For the subcooled cycle, the weight of refrigerant circulated per minute per ton m	$= \dfrac{200}{54.21}$ $= 3.69$ lb

Notice that the condition of the refrigerant vapor entering the suction inlet of the compressor is the same for both cycles. For this reason, the specific volume of the vapor entering the compressor will be the same for both the saturated and subcooled cycles and, since the weight of refrigerant circulated per minute per ton is less for the subcooled cycle than for the saturated cycle, it follows that the volume of vapor which the compressor must handle per minute per ton will also be less for the subcooled cycle than for the saturated cycle.

For the saturated cycle, the specific volume of the suction vapor v_c	$= 1.121$ cu ft/lb
The volume of vapor compressed per minute per ton V	$= m \times v_c$ $= 4.05 \times 1.121$ $= 4.55$ cu ft/min
For the subcooled cycle, the specific volume of the suction vapor v_c	$= 1.121$ cu ft/lb
The volume of vapor compressed per minute per ton V	$= m \times v_c$ $= 3.69 \times 1.121$ $= 4.15$ cu ft/min

Because the volume of vapor compressed per minute per ton is less for the subcooled cycle, the compressor displacement required for the subcooled cycle is less than that required for the saturated cycle.

Notice also that the heat of compression per pound and therefore the work of compression per pound is the same for both the saturated and subcooled cycles. This means that the refrigerating effect per pound resulting from the subcooling is accomplished without increasing the energy input to the compressor. Any change in the refrigerating cycle which increases the quantity of heat absorbed in the refrigerated space without causing an increase in the energy input to the compressor will increase the c.o.p. of the cycle and reduce the horsepower required per ton.

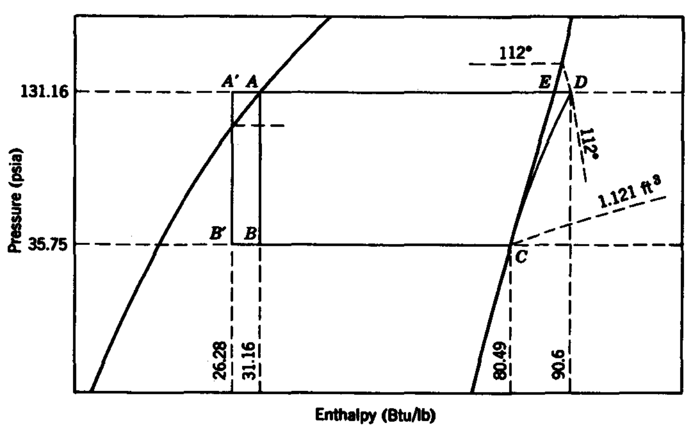


Fig. 8-4. *Ph* diagrams comparing the subcooled cycle to the simple saturated cycle. (Refrigerant-12.)

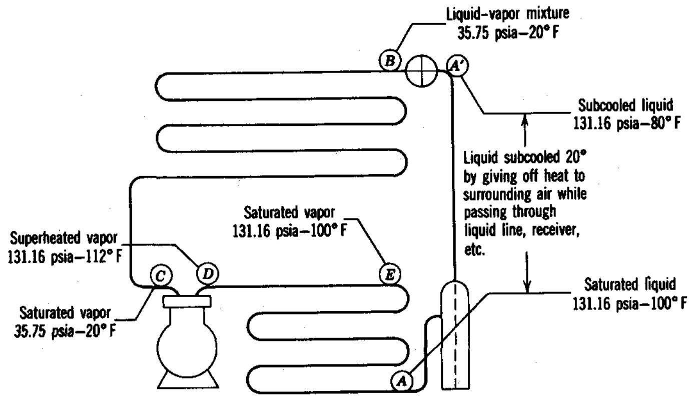


Fig. 8-5. Flow diagram illustrating subcooling of the liquid in the liquid line. (Refrigerant-12.)

For the saturated cycle, the coefficient of performance

$$= \frac{h_c - h_a}{h_d - h_c} = \frac{80.49 - 31.16}{90.60 - 80.49} = 4.88$$

The horsepower per ton

$$= \frac{m(h_d - h_c)}{42.42} = 4.05 \times 10.51 = 0.965 \text{ hp/ton}$$

For the subcooled cycle, the coefficient of performance

$$= \frac{h_c - h_{a'}}{h_d - h_c} = \frac{80.49 - 26.28}{90.60 - 80.49} = \frac{54.21}{10.51} = 5.16$$

The horsepower per ton

$$= \frac{m(h_d - h_c)}{42.42} = \frac{3.69 \times 10.51}{42.42} = 0.914 \text{ hp/ton}$$

In this instance, the c.o.p. of the subcooled cycle is greater than that of the saturated cycle by

$$\frac{5.16 - 4.88}{4.88} \times 100 = 5.7\%$$

Subcooling of the liquid refrigerant can and does occur in several places and in several ways. Very often the liquid refrigerant becomes subcooled while stored in the liquid receiver tank or while passing through the liquid line by giving off heat to the surrounding air (Fig. 8-5). In some cases where water is used as the condensing medium, a special liquid subcooler is used to subcool the liquid (Fig. 8-6). The gain in system capacity and efficiency resulting from the liquid subcooling is very often more than sufficient to offset the additional cost of the subcooler, particularly for low temperature applications.

The liquid subcooler may be piped either in series or in parallel with the condenser. When the subcooler is piped in series with the condenser, the cooling water passes through the subcooler first and then through the condenser, thereby bringing the coldest water into contact with the liquid being subcooled (Fig. 8-7). There is some doubt about the value of a subcooler piped in series with the condenser. Since the cooling water is warmed by the heat absorbed in the subcooler, it reaches the condenser at a higher temperature and the condensing temperature of the cycle is increased. Hence the increase in system efficiency resulting from the subcooling is offset to some extent by the rise in the condensing temperature.

When the subcooler is piped in parallel with the condenser (Fig. 8-6), the temperature of the water reaching the condenser is not affected by the subcooler. However, for either series or parallel piping, the size of the condenser water pump must be increased when a subcooler is added. If this is not done, the quantity of water circulated through the condenser will be diminished by the addition of the subcooler and the condensing temperature of the cycle will be increased, thus nullifying any benefit accruing from the subcooling.

Notice that in each case discussed so far, the heat given up by the liquid in becoming subcooled is given up to some medium external to the system.

8-8. Liquid-Suction Heat Exchangers. Another method of subcooling the liquid is to bring about an exchange of heat between the liquid and the cold suction vapor going back to the compressor. In a liquid-suction heat exchanger, the cold suction vapor is piped through the heat exchanger in counterflow to the warm liquid refrigerant flowing through the liquid line to the refrigerant control (Fig. 8-8). In flowing through the heat exchanger the cold suction vapor absorbs heat from the warm liquid so that the liquid is subcooled as the vapor is superheated, and, since the heat absorbed by the vapor in becoming superheated is drawn from the liquid, the heat of the liquid is diminished by an amount equal to the amount of heat taken in by the vapor. In each of the methods of subcooling discussed thus far, the heat given up by the liquid in becoming subcooled is given up to some medium external to the system and the heat then leaves the system. When a liquid-suction heat exchanger is used, the heat given up by the liquid in becoming subcooled is absorbed by the suction vapor and remains in the system.

On the *Ph* diagram in Fig. 8-9, a simple saturated cycle is compared to one in which a liquid-suction heat exchanger is employed. Points *A*, *B*, *C*, *D*, and *E* identify the saturated cycle and points *A′*, *B′*, *C′*, *D′*, *E* identify the in which the heat exchanger is used. In the cycle latter cycle, it is assumed that the suction vapor is superheated from 20° F to 60° F in the heat exchanger.

The heat absorbed per pound of vapor in the heat exchanger is

$$h_{c'} - h_c = 86.20 - 80.49 = 5.71 \text{ Btu/lb}$$

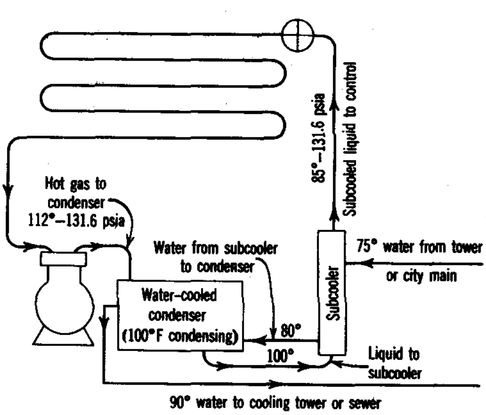


Fig. 8-6. Flow diagram illustrating subcooler piped in series with condenser.

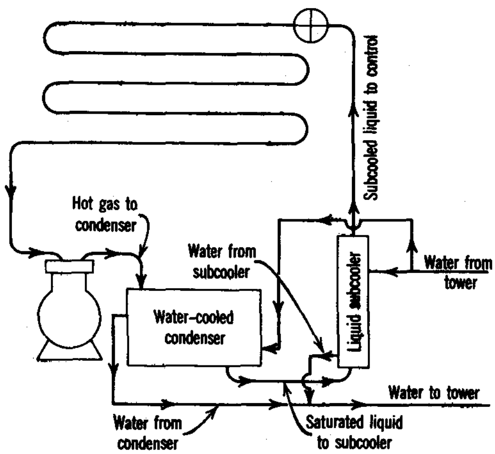


Fig. 8-7. Flow diagram showing parallel piping for condenser and subcooler.

Since the heat given up by the liquid in the heat exchanger in becoming subcooled is exactly equal to the heat absorbed by the vapor in becoming superheated, $h_a - h_{a'}$ is equal to $h_{c'} - h_c$ and therefore is also equal to 5.71 Btu/lb. Since $h_a - h_{a'}$ represents an increase in the refrigerating effect, the refrigerating effect per pound for the heat exchanger cycle is

$$h_c - h_{a'} = 80.49 - 25.45 = 55.04$$

The heat of compression per pound for the heat exchanger cycle is

$$h_{d'} - h_{c'} = 97.60 - 86.20 = 11.40$$

Therefore, the coefficient of performance is

$$\frac{h_c - h_{a'}}{h_{d'} - h_{c'}} = \frac{55.04}{11.40} = 4.91$$

The coefficient of performance of the saturated cycle is 4.88. Therefore, the coefficient of performance of the heat exchanger cycle is greater than that of the saturated cycle by only

$$\frac{4.91 - 4.88}{4.88} \times 100 = 0.5\%$$

Depending upon the particular case, the coefficient of performance of a cycle employing a heat exchanger may be either greater than, less than, or the same as that of a saturated cycle operating between the same pressure limits. In any event, the difference is negligible, and it is evident that the advantages accruing

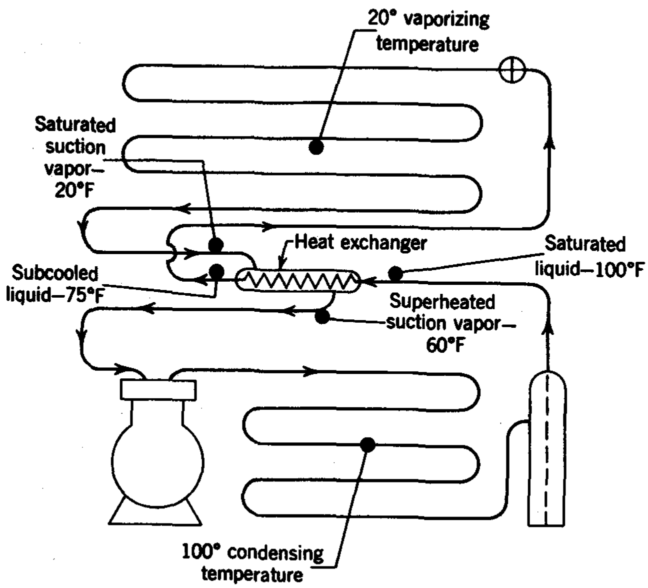


Fig. 8-8. Flow diagram of refrigeration cycle illustrating the use of a liquid-suction heat exchanger.

from the subcooling of the liquid in the heat exchanger are approximately offset by the disadvantages of superheating the vapor. Theoretically, then, the use of a heat exchanger cannot be justified on the basis of an increase in system capacity and efficiency. However, since in actual practice a refrigerating system does not (cannot) operate on a simple saturated cycle, this does not represent a true appraisal of the practical value of the heat exchanger.

In an actual cycle, the suction vapor will always become superheated before the compression process begins because nothing can be done to prevent it. This is true even if no superheating takes place either in the evaporator or in the suction line and the vapor reaches the inlet of the compressor at the vaporizing temperature. As the cold suction vapor flows into the compressor, it will become superheated by absorbing heat from the hot cylinder walls. Since the superheating in the compressor cylinder will occur before the compression process

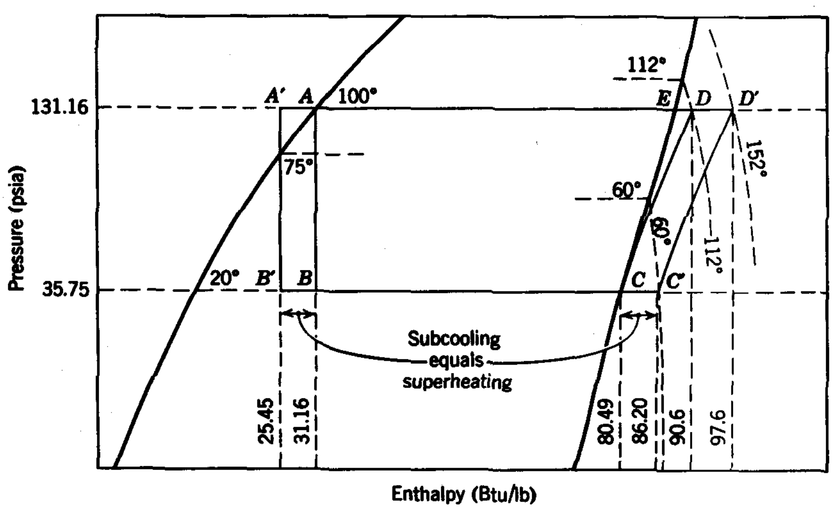


Fig. 8-9. *Ph* **diagrams comparing simple saturated cycle to cycle employing a liquid-suction heat exchanger. The amount of subcooling is equal to the amount of superheating. (Refrigerant-12.)**

begins, the effect of the superheating on cycle efficiency will be approximately the same as if the superheating occurred in the suction line without producing useful cooling.*

The disadvantages resulting from allowing the suction to become superheated without producing useful cooling have already been pointed out. Obviously, then, since superheating of the suction vapor is unavoidable in an actual cycle, whether or not a heat exchanger is used, any practical means of causing the vapor to become superheated in such a way that useful cooling results are worthwhile. Hence, the value of a heat exchanger lies in the fact that it provides a method of superheating the vapor so that useful cooling results. For this reason, the effect of a heat exchanger on cycle efficiency can be evaluated only by comparing the heat exchanger cycle to one in which the vapor is superheated without producing useful cooling.

The maximum amount of heat exchange which can take place between the liquid and the vapor in the heat exchanger depends on the initial temperatures of the liquid and the vapor as they enter the heat exchanger and on the length of time they are in contact with each other.

The greater the difference in temperature, the greater is the exchange of heat for any given period of contact. Thus, the lower the vaporizing temperature and the higher the condensing temperature, the greater is the possible heat exchange. Theoretically, if the two fluids remained in contact for a sufficient length of time, they would leave the heat exchanger at the same temperature. In actual practice, this is not possible. However, the longer the two fluids stay in contact, the more nearly the two temperatures will approach one another. Since the specific heat of the vapor is less than that of the liquid, the rise in the temperature of the vapor is always greater than the reduction in the temperature of the liquid. For instance, the specific heat of R-12 liquid is approximately 0.24 Btu per pound, whereas the specific heat of the vapor is 0.15 Btu per pound. This means that the temperature reduction of the liquid will be approximately 62% (0.15/0.24) of the rise in the temperature of the vapor, or that for each 24° F rise in the temperature of the vapor, the temperature of the liquid will be reduced 15° F.

For the heat exchanger cycle in Fig. 8-9, the vapor absorbs 5.71 Btu per pound in superheating from 20° F to 60° F. Assuming that all of the superheating takes place in the heat exchanger, the heat given up by the liquid is 5.71 Btu, so that the temperature of the liquid is reduced 23.8° F (5.71/0.24) as the liquid passes through the heat exchanger.

8-9. The Effect of Pressure Losses Resulting from Friction. In overcoming friction, both internal (within the fluid) and external (surface), the refrigerant experiences a drop in pressure while flowing through the piping, evaporator, condenser, receiver, and through the valves and passages of the compressor (Fig. 8-10).

A *Ph* diagram of an actual cycle, illustrating the loss in pressure occurring in the various parts of the system, is shown in Fig. 8-11. To simplify the diagram, no superheating or subcooling is shown and a simple saturated cycle is drawn in for comparison.

Line B'-C' represents the vaporizing process in the evaporator during which the refrigerant undergoes a drop in pressure of 5.5 psi. Whereas the pressure and saturation temperature of the liquid-vapor mixture at the evaporator inlet is 38.58 psia and 24° F, respectively, the pressure of the saturated vapor leaving the evaporator is 33.08 psia, corresponding to a saturation temperature of 16° F. The average vaporizing temperature in the evaporator is 20° F, the same as that of the saturated cycle.

As a result of the drop in pressure in the evaporator, the vapor leaves the evaporator at a lower pressure and saturation temperature and with a greater specific volume than if no drop in pressure occurred.

The refrigerating effect per pound and the weight of refrigerant circulated per minute per ton are approximately the same for both cycles, but because of the greater specific volume the volume of vapor handled by the compressor per

* It will be shown later that some advantages accrue from superheating which takes place in the compressor: (1) When the suction vapor absorbs heat from the cylinder walls, the cylinder wall temperature is lowered somewhat and this brings about a desirable change in the path of the compression process. However, the change is slight and is difficult to evaluate. (2) When hermetic motor-compressor assemblies are used, the suction vapor should reach the compressor at a relatively low temperature in order to help cool the motor windings.

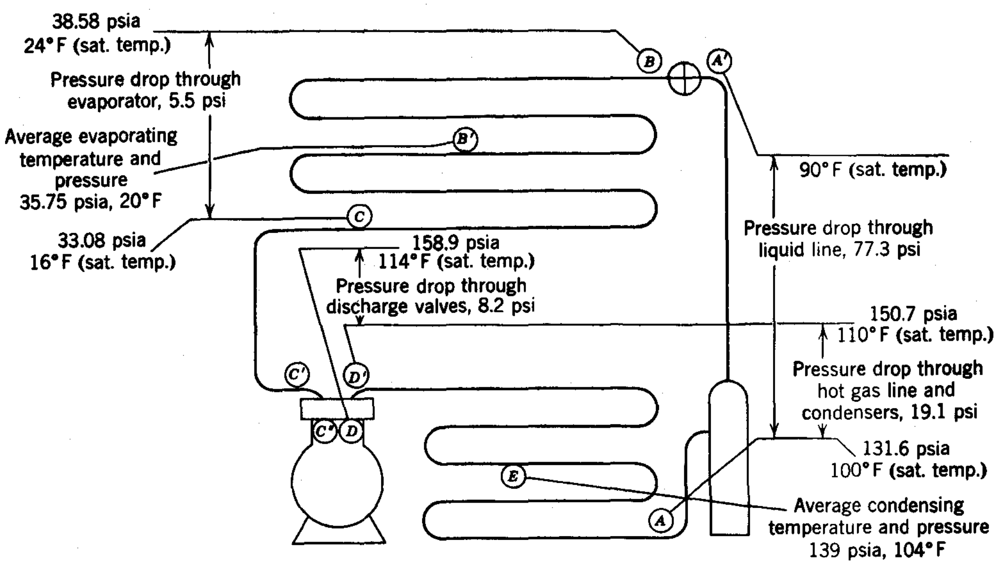


Fig. 8-10. Flow diagram illustrating the effect of pressure drop in various parts of the system. Pressure drops are exaggerated for clarity. (Refrigerant-12.)

minute per ton is greater for the cycle experiencing the pressure drop. Too, because of the lower pressure of the vapor leaving the evaporator, the vapor must be compressed through a greater pressure range during the compression process, so that the horsepower per ton is also greater for the cycle undergoing the drop in pressure.

Line C'–C'' represents the drop in pressure experienced by the suction vapor in flowing through the suction line from the evaporator to the compressor inlet. Like pressure drop in the evaporator, pressure drop in the suction line causes the suction vapor to reach the compressor at a lower pressure and in an expanded condition so that the volume of vapor compressed per minute per ton and the horsepower per ton are both increased.

It is evident that the drop in pressure both in the evaporator and in the suction line should be

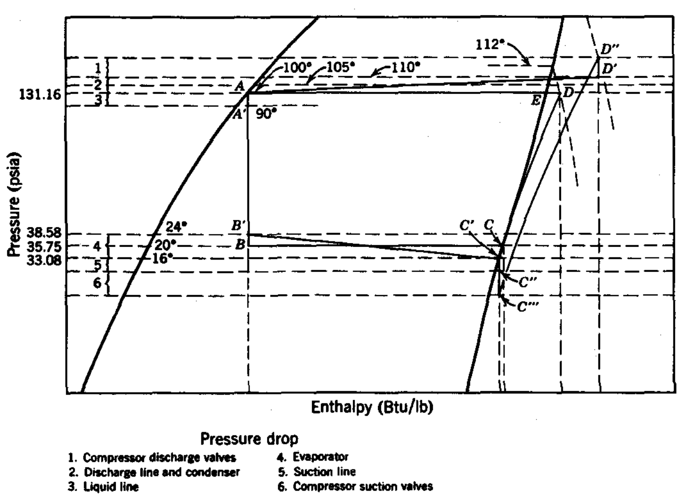


Fig. 8-11. *Ph* diagram of refrigeration cycle illustrating the effect of pressure losses in the various parts of the system. A simple saturated cycle is drawn in for comparison. (Refrigerant-12.)

kept to an absolute minimum in order to obtain the best possible cycle efficiency. This applies also to heat exchangers or any other auxiliary device intended for installation in the suction line.

In Fig. 8-11, the pressure drops are exaggerated for clarity. Ordinarily, good evaporator design limits the pressure drop across the evaporator to 2 or 3 psi. Ideally, the suction line should be designed so that the pressure drop is between 1 and 2 psi.

against the spring-loading and to force the vapor out through the discharge valves and passages of the compressor into the discharge line.

Line D'–A represents the drop in pressure resulting from the flow of the refrigerant through the discharge line and condenser. That part of line D'–A which represents the flow through the discharge line will vary with the particular case, since the discharge line may be either quite long or very short, depending upon the application. In any event, the result of the pressure drop will

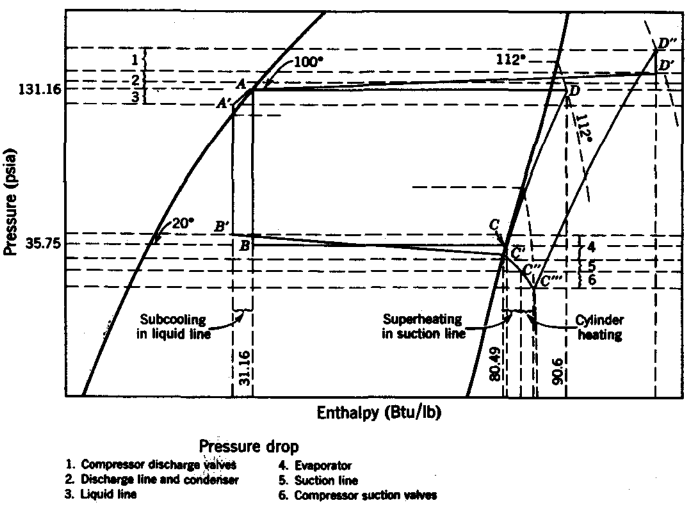


Fig. 8-12. *Ph* diagram of actual refrigeration cycle illustrating effects of subcooling, superheating, and losses in pressure. A simple saturated cycle is drawn in for comparison. (Refrigerant-12).

Line C''–C''' represents the drop in pressure that the suction vapor undergoes in flowing through the suction valves and passages of the compressor into the cylinder. The result of the drop in pressure through the valves and passages on the suction side of the compressor is the same as if the drop occurred in the suction line, and the effect on cycle efficiency is the same. Here again, good design requires that the drop in pressure be kept to a practical minimum.

Line C'''–D'' represents the compression process for the cycle undergoing the pressure drops. Notice that the vapor in the cylinder is compressed to a pressure considerably above the average condensing pressure. It is shown later that this is necessary in order to force the vapor out of the cylinder through the discharge valves against the condensing pressure and against the additional pressure occasioned by the spring-loading of the discharge valves.

Line D''–D' represents the drop in pressure required to force the discharge valves open be the same. Any drop in pressure occurring on the discharge side of the compressor (in the discharge valves and passages, in the discharge line, and in the condenser) will have the effect of raising the discharge pressure and thereby increasing the work of compression and the horsepower per ton.

Line A–A' represents the pressure drop resulting from the flow of the refrigerant through the receiver tank and liquid line. Since the refrigerant at A' is a saturated liquid, the temperature of the liquid must decrease as the pressure decreases. If the liquid is not subcooled by giving up heat to an external sink as its pressure drops, a portion of the liquid must flash into a vapor in the liquid line in order to provide the required cooling of the liquid. Notice that point A'' lies in the region of phase-change, indicating that a portion of the refrigerant is a vapor at this point.

Despite the flashing of the liquid and the drop in temperature coincident with the drop in pressure in the liquid line, the drop in pressure in the liquid line has no effect on cycle efficiency. The pressure and temperature of the liquid must be reduced to the vaporizing condition before it enters the evaporator in any case. The fact that a part of this takes place in the liquid line rather than in the refrigerant control has no direct effect on the efficiency of the system. It does, however, reduce the capacity of both the liquid line and the refrigerant control. Furthermore, passage of vapor through the refrigerant control will eventually cause damage to the refrigerant control by eroding the valve needle and seat.

Ordinarily, even without the use of a heat exchanger, sufficient subcooling of the liquid will occur in the liquid line to prevent the flashing of the liquid if the drop in pressure in the line is not excessive. Flashing of the liquid in the liquid line will usually not take place when the drop in the line does not exceed 5 psi.

The effect of pressure drop in the lines and in the other parts of the system is discussed more fully later in the appropriate chapters.

A *Ph* diagram of a typical refrigeration cycle, which illustrated the combined effects of pressure drop, subcooling, and superheating, is compared to the *Ph* diagram of the simple saturated cycle in Fig. 8-12.

PROBLEMS

1. The vaporizing and condensing temperature of a Refrigerant-12 system are 40° F and 110° F, respectively. The suction vapor is superheated to 70° F in the suction line, whereas the liquid is subcooled to 90° F by giving off heat to the ambient air. Determine:

(*a*) The refrigerating effect per pound. *Ans.* 54.01 Btu/lb

(*b*) The weight of refrigerant circulated per minute per ton. *Ans.* 3.70 lb/min/ton

(*c*) The volume of vapor compressed per minute per ton. *Ans.* 2.93 cu ft/min/ton

(*d*) The loss of refrigerating effect per pound in the refrigerant control. *Ans.* 11.7 Btu/lb

(*e*) The quantity of superheat in the suction vapor. *Ans.* 4.39 Btu/lb

(*f*) The gain in refrigerating effect per pound resulting from the liquid subcooling. *Ans.* 4.93 Btu/lb

(*g*) The adiabatic discharge temperature. *Ans.* 138.5° F

(*h*) The heat of compression per pound. *Ans.* 9 Btu/lb

(*i*) The heat of compression per minute per ton. *Ans.* 33.3 Btu/min/ton

(*j*) The work of compression per minute per ton. *Ans.* 25.907 lb/min/ton

(*k*) The theoretical horsepower per ton. *Ans.* 0.755 hp/ton

(*l*) The heat rejected at the condenser per pound. *Ans.* 67.4 Btu/lb

(*m*) The heat rejected at the condenser per ton. *Ans.* 249.38 Btu/min/ton

(*n*) The coefficient of performance. *Ans.* 6

NOTE: Some of the properties of the refrigerant at various points in the cycle must be determined from the *Ph* chart in Fig. 7-1.

9

Survey of Refrigeration Applications

9-1. History and Scope of the Industry. In the early days of mechanical refrigeration, the equipment available was bulky, expensive, and not too efficient. Also it was of such a nature as to require that a mechanic or operating engineer be on duty at all times. This limited the use of mechanical refrigeration to a few large applications such as ice plants, meat packing plants, and large storage warehouses.

In the span of only a few decades refrigeration has grown into the giant and rapidly expanding industry that it is today. This explosive growth came about as the result of several factors. First, with the development of precision manufacturing methods, it became possible to produce smaller, more efficient equipment. This, along with the development of "safe" refrigerants and the invention of the fractional horsepower electric motor, made possible the small refrigerating unit which is so widely used at the present time in such applications as domestic refrigerators and freezers, small air conditioners, and commercial fixtures. Today, there are few homes or business establishments in the United States that cannot boast of one or more mechanical refrigeration units of some sort.

Few people outside of those directly connected with the industry are aware of the significant part that refrigeration has played in the development of the highly technical society that America is today, nor do they realize the extent to which such a society is dependent upon mechanical refrigeration for its very existence. It would not be possible, for instance, to preserve food in sufficient quantities to feed the growing urban population without mechanical refrigeration. Too, many of the large buildings which house much of the nation's business and industry would become untenable in the summer months because of the heat if they were not air conditioned with mechanical refrigerating equipment.

In addition to the better known applications of refrigeration, such as comfort air conditioning and the processing, freezing, storage, transportation, and display of perishable products, mechanical refrigeration is used in the processing or manufacturing of almost every article or commodity on the market today. The list of processes or products made possible or improved through the use of mechanical refrigeration is almost endless. For example, refrigeration has made possible the building of huge dams which are vital to large-scale reclamation and hydroelectric projects. It has made possible the construction of roads and tunnels and the sinking of foundation and mining shafts through and across unstable ground formations. It has made possible the production of plastics, synthetic rubber, and many other new and useful materials and products. Because of mechanical refrigeration, bakers can get more loaves of bread from a barrel of flour, textile and paper manufacturers can speed up their machines and get more production, and better methods of hardening steels for machine tools are available. These represent only a few of the hundreds of ways in which mechanical refrigeration is now being used and many new uses are being found each year. In fact, the only thing slowing the growth of the refrigeration industry at the present time is the lack of an adequate supply of trained technical manpower.

9-2. Classification of Applications. For convenience of study, refrigeration applications may be grouped into six general categories: (1) domestic refrigeration, (2) commercial refrigeration, (3) industrial refrigeration, (4) marine and transportation refrigeration, (5) comfort air conditioning, and (6) industrial air conditioning. It will be apparent in the discussion which follows that the exact limits of these areas are not precisely defined and that there is considerable overlapping between the several areas.

9-3. Domestic Refrigeration. Domestic refrigeration is rather limited in scope, being concerned primarily with household refrigerators and home freezers. However, because the number of units in service is quite large, domestic refrigeration represents a significant portion of the refrigeration industry.

Domestic units are usually small in size, having horsepower ratings of between $\frac{1}{20}$ and $\frac{1}{2}$ hp, and are of the hermetically sealed type. Since these applications are familiar to everyone, they will not be described further here. However, the problems encountered in the design and maintenance of these units are discussed in appropriate places in the chapters which follow.

9-4. Commercial Refrigeration. Commercial refrigeration is concerned with the designing, installation, and maintenance of refrigerated fixtures of the type used by retail stores, restaurants, hotels, and institutions for the storing, displaying, processing, and dispensing of perishable commodities of all types. Commercial refrigeration fixtures are described in more detail later in this chapter.

9-5. Industrial Refrigeration. Industrial refrigeration is often confused with commercial refrigeration because the division between these two areas is not clearly defined. As a general rule, industrial applications are larger in size than commercial applications and have the distinguishing feature of requiring an attendant on duty, usually a licensed operating engineer. Typical industrial applications are ice plants, large food-packing plants (meat, fish, poultry, frozen foods, etc.), breweries, creameries, and industrial plants, such as oil refineries, chemical plants, rubber plants, etc. Industrial refrigeration includes also those applications concerned with the construction industry as described in Section 9-1.

9-6. Marine and Transportation Refrigeration. Applications falling into this category could be listed partly under commercial refrigeration and partly under industrial refrigeration. However, both these areas of specialization have grown to sufficient size to warrant special mention.

Marine refrigeration, of course, refers to refrigeration aboard marine vessels and includes, for example, refrigeration for fishing boats and for vessels transporting perishable cargo as well as refrigeration for the ship's stores on vessels of all kinds.

Transportation refrigeration is concerned with refrigeration equipment as it is applied to trucks, both long distance transports and local delivery, and to refrigerated railway cars. Typical refrigerated truck bodies are shown in Fig. 11-8.

9-7. Air Conditioning. As the name implies, air conditioning is concerned with the condition of the air in some designated area or space. This usually involves control not only of the space temperature but also of space humidity and air motion, along with the filtering and cleaning of the air.

Air conditioning applications are of two types, either comfort or industrial, according to their purpose. Any air conditioning which has as its primary function the conditioning of air for human comfort is called comfort air conditioning. Typical installations of comfort air conditioning are in homes, schools, offices, churches, hotels, retail stores, public buildings, factories, automobiles, buses, trains, planes, ships, etc.

On the other hand, any air conditioning which does not have as its primary purpose the conditioning of air for human comfort is called industrial air conditioning. This does not necessarily mean that industrial air conditioning systems cannot serve as comfort air conditioning coincidentally with their primary function. Often this is the case, although not always so.

The applications of industrial air conditioning are almost without limit both in number and in variety. Generally speaking, the functions of industrial air conditioning are to; (1) control the moisture content of hydroscopic materials; (2) govern the rate of chemical and biochemical reactions; (3) limit the variations in the size of precision manufactured articles because of thermal expansion and contraction; and (4) provide clean, filtered air which is often essential to trouble-free operation and to the production of quality products.

9-8. Food Preservation. The preservation of perishable commodities, particularly foodstuffs, is one of the most common uses of mechanical refrigeration. As such, it is a subject which should be given consideration in any comprehensive study of refrigeration.

At the present time, food preservation is more important than ever before in man's history. Today's large urban populations require tremendous quantities of food, which for the most part must be produced and processed in outlying areas. Naturally, these foodstuffs must be kept in a preserved condition during transit and subsequent storage until they are finally consumed. This may be a matter of hours, days, weeks, months, or even years in some cases. Too, many products, particularly fruit and vegetables, are seasonal. Since they are produced only during certain seasons of the year, they must be stored and preserved if they are to be made available the year round.

As a matter of life or death, the preservation of food has long been one of man's most pressing problems. Almost from the very beginning of man's existence on earth, it became necessary for him to find ways of preserving food during seasons of abundance in order to live through seasons of scarcity. It is only natural, then, that man should discover and develop such methods of food preservation as drying, smoking, pickling, and salting long before he had any knowledge of the causes of food spoilage. These rather primitive methods are still widely used today, not only in backward societies where no other means are available but also in the most modern societies where they serve to supplement the more modern methods of food preservation. For instance, millions of pounds of dehydrated (dried) fruit, milk, eggs, fish, meat, potatoes, etc., are consumed in the United States each year, along with huge quantities of smoked, pickled, and salted products, such as ham, bacon, and sausage, to name only a few. However, although these older methods are entirely adequate for the preservation of certain types of food, and often produce very unusual and tasty products which would not otherwise be available, they nonetheless have inherent disadvantages which limit their usefulness. Since by their very nature they bring about severe changes in appearance, taste, and odor, which in many cases are objectionable, they are not universally adaptable for the preservation of all types of food products. Furthermore, the keeping qualities of food preserved by such methods are definitely limited as to time. Therefore, where a product is to be preserved indefinitely or for a long period of time, some other means of preservation must be utilized.

The invention of the microscope and the subsequent discovery of microorganisms as a major cause of food spoilage led to the development of canning in France during the time of Napoleon. With the invention of canning, man found a way to preserve food of all kinds in large quantities and for indefinite periods of time. Canned foods have the advantage of being entirely imperishable, easily processed, and convenient to handle and store. Today, more food is preserved by canning than by all other methods combined. The one big disadvantage of canning is that canned foods must be heat-sterilized, which frequently results in overcooking. Hence, although canned foods often have a distinctive and delicious flavor all their own, they usually differ greatly from the original fresh product.

The only means of preserving food in its original fresh state is by refrigeration. This, of course, is the principal advantage that refrigeration has over other methods of food preservation. However, refrigeration too has its disadvantages. For instance, when food is to be preserved by refrigeration, the refrigerating process must begin very soon after harvesting or killing and must be continuous until the food is finally consumed. Since this requires relatively expensive and bulky equipment, it is often both inconvenient and uneconomical.

Obviously, then, there is no one method of food preservation which is best in all cases and the particular method used in any one case will depend upon a number of factors, such as the type of product, the length of time the product is to be preserved, the purpose for which the product is to be used, the availability of transportation and storage equipment, etc. Very often it is necessary to employ several methods simultaneously in order to obtain the desired results.

9-9. Deterioration and Spoilage. Since the preservation of food is simply a matter of preventing or retarding deterioration and spoilage regardless of the method used, a good knowledge of the causes of deterioration and spoilage is a prerequisite to the study of preservation methods.

It should be recognized at the outset that there are degrees of quality and that all perishable foods pass through various stages of deterioration before becoming unfit for consumption. In most cases, the objective in the preservation of food is not only to preserve the foodstuff in an edible condition but also to preserve it as nearly as possible at the peak of its quality with respect to appearance, odor, taste, and vitamin content. Except for a few processed foods, this usually means maintaining the foodstuff as nearly as possible in its original fresh state.

Any deterioration sufficient to cause a detectable change in the appearance, odor, or taste of fresh foods immediately reduces the commercial value of the product and thereby represents an economic loss. Consider, for example, wilted vegetables or overripe fruit. Although their edibility is little impaired, an undesirable change in their appearance has been brought about which usually requires that they be disposed of at a reduced price. Too, since they are well on their way to eventual spoilage, their keeping qualities are greatly reduced and they must be consumed or processed immediately or become a total loss.

For obvious reasons, maintaining the vitamin content at the highest possible level is always an important factor in the processing and/or preservation of all food products. In fact, many food processors, such as bakers and dairymen, are now adding vitamins to their product to replace those which are lost during processing. Fresh vegetables, fruit, and fruit juices are some of the food products which suffer heavy losses in vitamin content very quickly if they are not handled and protected properly. Although the loss of vitamin content is not something which in itself is apparent, in many fresh foods it is usually accompanied by recognizable changes in appearance, odor, or taste, such as, for instance, wilting in leafy, green vegetables.

For the most part, the deterioration and eventual spoilage of perishable food are caused by a series of complex chemical changes which take place in the foodstuff after harvesting or killing. These chemical changes are brought about by both internal and external agents. The former are the natural enzymes which are inherent in all organic materials, whereas the latter are microorganisms which grow in and on the surface of the foodstuff. Although either agent alone is capable of bringing about the total destruction of a food product, both agents are involved in most cases of food spoilage. In any event, the activity of both of these spoilage agents must be either eliminated or effectively controlled if the foodstuff is to be adequately preserved.

9-10. Enzymes. Enzymes are complex, protein-like, chemical substances. Not yet fully understood, they are probably best described as chemical catalytic agents which are capable of bringing about chemical changes in organic materials. There are many different kinds of enzymes and each one is specialized in that it produces only one specific chemical reaction. In general, enzymes are identified either by the substance upon which they act or by the result of their action. For instance, the enzyme, lactase, is so known because it acts to convert lactose (milk sugar) to lactic acid. This particular process is called lactic acid fermentation and is the one principally responsible for the "souring" of milk. Enzymes associated with the various types of fermentation are sometimes called ferments.

Essential in the chemistry of all living processes, enzymes are normally present in all organic materials (the cell tissue of all plants and animals, both living and dead). They are manufactured by all living cells to help carry on the various living activities of the cell, such as respiration, digestion, growth, and reproduction, and they play an important part in such things as the sprouting of seeds, the growth of plants and animals, the ripening of fruit, and the digestive processes of animals, including man. However, enzymes are catabolic as well as anabolic. That is, they act to destroy dead cell tissue as well as to maintain live cell tissue. In fact, enzymes are the agents primarily responsible for the decay and decomposition of all organic materials, as, for example, the putrification of meat and fish and the rotting of fruit and vegetables.

Whether their action is catabolic or anabolic, enzymes are nearly always destructive to perishable foods. Therefore, except in those few special cases where fermentation or putrification is a part of the processing, enzymic action must be either eliminated entirely or severely inhibited if the product is to be preserved in good condition. Fortunately, enzymes are sensitive to the conditions of the surrounding media, particularly with regard to the temperature and the degree of acidity or alkalinity, which provides a practical means of controlling enzymic activity.

Enzymes are completely destroyed by high temperatures that alter the composition of the organic material in which they exist. Since most enzymes are eliminated at temperatures above 160° F, cooking a food substance completely destroys the enzymes contained therein. On the other hand, enzymes are very resistant to low temperatures and their activity may continue at a slow rate even at temperatures below 0° F. However, it is a well-known fact that the rate of chemical reaction decreases as the temperature decreases. Hence, although the enzymes are not destroyed, their activity is greatly reduced at low temperatures, particularly temperatures below the freezing point of water.

Enzymic action is greatest in the presence of free oxygen (as in the air) and decreases as the oxygen supply diminishes.

With regard to the degree of acidity or alkalinity, some enzymes require acid surroundings, whereas others prefer neutral or alkaline environments. Those requiring acidity are destroyed by alkalinity and those requiring alkalinity are likewise destroyed by acidity.

Although an organic substance can be completely destroyed and decomposed solely by the action of its own natural enzymes, a process known as autolysis (self-destruction), this seldom occurs. More often, the natural enzymes are aided in their destructive action by enzymes secreted by microorganisms.

9-11. Microorganisms. The term microorganism is used to cover a whole group of minute plants and animals of microscopic and submicroscopic size, of which only the following three are of particular interest in the study of food preservation: (1) bacteria, (2) yeasts, and (3) molds. These tiny organisms are found in large numbers everywhere—in the air, in the ground, in water, in and on the bodies of plants and animals, and in every other place where conditions are such that living organisms can survive.

Because they secrete enzymes which attack the organic materials upon which they grow, microorganisms are agents of fermentation, putrification, and decay. As such, they are both beneficial and harmful to mankind. Their growth in and on the surface of perishable foods causes complex chemical changes in the food substance which usually results in undesirable alterations in the taste, odor, and appearance of the food and which, if allowed to continue for any length of time, will render the food unfit for consumption. Too, some microorganisms secrete poisonous substances (toxins) which are extremely dangerous to health, causing poisoning, disease, and often death.

On the other hand, microorganisms have many useful and necessary functions. As a matter of fact, if it were not for the work of microorganisms, life of any kind would not be possible. Since decay and decomposition of all dead animal tissue are essential to make space available for new life and growth, the decaying action of microorganisms is indispensable to the life cycle.

Of all living things, only green plants (those containing chlorophyll) are capable of using inorganic materials as food for building their cell tissue. Through a process called photosynthesis, green plants are able to utilize the radiant energy of the sun to combine carbon dioxide from the air with water and mineral salts from the soil and thereby manufacture from inorganic materials the organic compounds which make up their cell tissue.

Conversely, all animals and all plants without chlorophyll (fungi) require organic materials (those containing carbon) for food to carry on their life activities. Consequently, they must of necessity feed upon the cell tissue of other plants and animals (either living or dead) and are, therefore, dependent either directly or indirectly on green plants as a source of the organic materials they need for life and growth.

It is evident, then, that should the supply of inorganic materials in the soil, which serve as food for green plants, ever become exhausted, all life would soon disappear from the earth. This is not likely to happen, however, since microorganisms, as a part of their own living process, are continuously replenishing the supply of inorganic materials in the soil.

With the exception of a few types of soil bacteria, all microorganisms need organic materials as food to carry on the living process. In most cases, they obtain these materials by decomposing animal wastes and the tissue of dead animals and plants. In the process of decomposition, the complex organic compounds which make up the tissue of animals and plants are broken down step by step and are eventually reduced to simple inorganic materials which are returned to the soil to be used as food by the green plants.

In addition to the important part they play in the "food chain" by helping to keep essential materials in circulation, microorganisms are necessary in the processing of certain fermented foods and other commodities. For example, bacteria are responsible for the lactic acid fermentation required in the processing of pickles, olives, cocoa, coffee, sauerkraut, ensilage, and certain sour milk products, such as butter, cheese, buttermilk, yogurt, etc., and for the acetic acid fermentation necessary in the production of vinegar from various alcohols. Bacterial action is useful also in the processing of certain other commodities such as leather, linen, hemp, and tobacco, and in the treatment of industrial wastes of organic composition.

Yeasts, because of their ability to produce alcoholic fermentation, are of immeasurable value to the brewing and wine-making industries and to the production of alcohols of all kinds. Too, everyone is aware of the importance of yeast in the baking industry.

The chief commercial uses of molds are in the processing of certain types of cheeses and, more important, in the production of antibiotics, such as penicillin and aureomycin.

Despite their many useful and necessary functions, the fact remains that microorganisms are destructive to perishable foods. Hence, their activity, like that of the natural enzymes, must be effectively controlled if deterioration and spoilage of the food substance are to be avoided.

Since each type of microorganism differs somewhat in both nature and behavior, it is worthwhile to examine each type separately.

9-12. Bacteria. Bacteria are a very simple form of plant life, being made up of one single living cell. Reproduction is accomplished by cell division. On reaching maturity, the bacterium divides into two separate and equal cells, each of which in turn grows to maturity and divides into two cells. Bacteria grow and reproduce at an enormous rate. Under ideal conditions, a bacterium can grow into maturity and reproduce in as little as 20 to 30 min. At this rate a single bacterium is capable of producing as many as 34,000,000,000,000 descendants in a 24-hr period. Fortunately, however, the life cycle of bacteria is relatively short, being a matter of minutes or hours, so that even under ideal conditions they cannot multiply at anywhere near this rate.

The rate at which bacteria and other microorganisms grow and reproduce depends upon such environmental conditions as temperature, light, and the degree of acidity or alkalinity, and upon the availability of oxygen, moisture, and an adequate supply of soluble food. However, there are many species of bacteria and they differ greatly both in their choice of environment and in the effect they have on their environment. Like the higher forms of plant life, all species of bacteria are not equally hardy with respect to surviving adverse conditions of environment, nor do all species thrive equally well under the same environmental conditions. Some species prefer conditions which are entirely fatal to others. Too, some bacteria are spore-formers. The spore is formed within the bacteria cell and is protected by a heavy covering or wall. In the spore state, which is actually a resting or dormant phase of the organism, bacteria are extremely resistant to unfavorable conditions of environment and can survive in this state almost indefinitely. The spore will usually germinate whenever conditions become favorable for the organism to carry on its living activities.

Most bacteria are saprophytes. That is, they are "free living" and feed only on animal wastes and on the dead tissue of animals and plants. Some, however, are parasites and require a living host. Most pathogenic bacteria (those causing infection and disease) are of the parasitic type. In the absence of a living host, some parasitic bacteria can live as saprophytes. Likewise, some saprophytes can live as parasites when the need arises.

Since bacteria are not able to digest insoluble food substances, they require food in a soluble form. For this reason, most bacteria secrete enzymes which are capable of rendering insoluble compounds into a soluble state, thereby making these materials available to the bacteria as food. The deterioration of perishable foods by bacteria growth is a direct result of the action of these bacterial enzymes.

Bacteria, like all other living things, require moisture as well as food to carry on their life activities. As in other things, bacteria vary considerably in their ability to resist drought. Although most species are readily destroyed by drying and will succumb within a few hours, the more hardy species are able to resist drought for several days. Bacterial spores can withstand

The Growth of Bacteria in Milk in Various Periods

Temp., °F	Time, hours 24	48	96	168
32	2,400	2,100	1,850	1,400
39	2,500	3,600	218,000	4,200,000
46	3,100	12,000	1,480,000	
50	11,600	540,000		
60	180,000	28,000,000		
86	1,400,000,000			

Fig. 9-1. From *ASRE Data Book*, Applications Volume, 1956–57. Reproduced by permission of the American Society of Heating, Refrigerating, and Air-Conditioning Engineers.

drought almost indefinitely, but will remain dormant in the absence of moisture.

In their need for oxygen, bacteria fall into two groups: (1) those which require free oxygen (air) and (2) those which can exist without free oxygen. Some species, although having a preference for one condition or the other, can live in the presence of free oxygen or in the absence of it. Those bacteria living without free oxygen obtain the needed oxygen through chemical reaction which reduces one compound while oxidizing another. Decomposition which occurs in the presence of free oxygen is known as decay, whereas decomposition which takes place in the absence of free oxygen is called putrification. One of the products of putrification is hydrogen sulfide, a foul-smelling gas which is frequently noted arising from decomposing animal carcasses.

Bacteria are very sensitive to acidity or alkalinity and cannot survive in an either highly acid or highly alkaline environment. Most bacteria require either neutral or slightly alkaline surroundings, although some species prefer slightly acid conditions. Because bacteria prefer neutral or slightly alkaline surroundings, nonacid vegetables are especially subject to bacterial attack.

Light, particularly direct sunlight, is harmful to all bacteria. Whereas visible light only inhibits their growth, ultraviolet light is actually fatal to bacteria. Since light rays, ultraviolet or otherwise, have no power of penetration, they are effective only in controlling surface bacteria. However, ultraviolet radiation (usually from direct sunlight), when combined with drying, provides an excellent means of controlling bacteria growth.

For each species of bacteria there is an optimum temperature at which the bacteria will grow at the highest rate. Too, for each species there is a maximum and a minimum temperature which will permit growth. At temperatures above the maximum, the bacteria are destroyed. At temperatures below the minimum, the bacteria are rendered inactive or dormant. The optimum temperature for most saprophytes is usually between 75° F and 85° F, whereas the optimum temperature for parasites is around 99° F or 100° F. A few species grow best at temperatures near the boiling point of water, whereas a few other types thrive best at temperatures near the freezing point. However, most species are either killed off or severely inhibited at these temperatures. The effect of temperature on the growth rate of bacteria is illustrated by the chart in Fig. 9-1 which shows the growth rate of bacteria in milk at various temperatures. In general, the growth rate of bacteria is considerably reduced by lowering the temperature.

9-13. Yeasts. Yeasts are simple, one-cell plants of the fungus family. Of microscopic size, yeast cells are somewhat larger and more complex than the bacteria cells. Although a few yeasts reproduce by fission or by sexual process, reproduction is usually by budding. Starting as a small protrusion of the mature cell, the bud enlarges and finally separates from the mother cell. Under ideal conditions, budding is frequently so rapid that new buds are formed before separation occurs so that yeast clusters are formed.

Like bacteria, yeasts are agents of fermentation and decay. They secrete enzymes that bring about chemical changes in the food upon which they grow. Yeasts are noted for their ability to transform sugars into alcohol and carbon dioxide. Although destructive to fresh foods, particularly fruits and berries and their juices, the alcoholic fermentation produced by yeasts is essential to the baking, brewing, and wine-making industries.

Yeasts are spore-formers, with as many as eight spores being formed within a single yeast cell. Yeasts are widespread in nature and yeast spores are invariably found in the air and on the skin of fruit and berries, for which they have a particular affinity. They usually spend the winter in the soil and are carried to the new fruit in the spring by insects or by the wind.

Like bacteria, yeasts require air, food, and moisture for growth, and are sensitive to temperature and the degree of acidity or alkalinity in the environment. For the most part, yeasts prefer moderate temperatures and slight acidity. In general, yeasts are not as resistant to unfavorable conditions as are bacteria, although they can grow in acid surroundings which inhibit most bacteria. Yeast spores, like those of bacteria, are extremely hardy and can survive for long periods under adverse conditions.

9-14. Molds. Molds, like yeasts, are simple plants of the fungi family. However, they are much more complex in structure than either bacteria or yeasts. Whereas the individual bacteria or yeast plants consist of one single cell, an individual mold plant is made up of a number of cells which are positioned end to end to form long, branching, threadlike fibers called hypha. The network which is formed by a mass of these threadlike fibers is called the mycelium and is easily visible to the naked eye. The hyphae of the mold plant are of two general types. Some are vegetative fibers which grow under the surface and act as roots to gather food for the plant, whereas others, called aerial hyphae, grow on the surface and produce the fruiting bodies.

Molds reproduce by spore formation. The spores develop in three different ways, depending on the type of mold: (1) as round clusters within the fibrous hyphae network, (2) as a mass enclosed in a sac and attached to the end of aerial hyphae, and (3) as chainlike clusters on the end of aerial hyphae. In any case, a single mold plant is capable of producing thousands of spores which break free from the mother plant and float away with the slightest air motion.

Mold spores are actually seeds and, under the proper conditions, will germinate and produce mold growth on any food substance with which they come in contact. Since they are carried about by air currents, mold spores are found almost everywhere and are particularly abundant in the air.

Although molds are less resistant to high temperatures than are bacteria, they are more tolerant to low temperatures, growing freely at temperatures close to the freezing point of water. Mold growth is inhibited by temperatures below 32° F, more from the lack of free moisture than from the effect of low temperature. All mold growth ceases at temperatures of 10° F and below.

Molds flourish in dark, damp surroundings, particularly in still air. An abundant supply of oxygen is essential to mold growth, although a very few species can grow in the absence of oxygen. Conditions inside cold-storage rooms are often ideal for mold growth, especially in the wintertime. This problem can be overcome somewhat by maintaining good air circulation in the storage room, by the use of germicidal paints, and ultraviolet radiation, and by frequent scrubbing.

Unlike bacteria, molds can thrive on foods containing relatively large amounts of sugars or acids. They are often found growing on acid fruits and on the surface of pickling vats, and are the most common cause of spoilage in apples and citrus fruits.

9-15. Control of Spoilage Agents. Despite complications arising from the differences in the reaction of the various types of spoilage agents to specific conditions in the environment, controlling these conditions provides the only means of controlling these spoilage agents. Thus, all methods of food preservation must of necessity involve manipulation of the environment in and around the preserved product in order to produce one or more conditions unfavorable to the continued activity of the spoilage agents. When the product is to be preserved for any length of time, the unfavorable conditions produced must be of sufficient severity to eliminate the spoilage agents entirely or at least render them ineffective or dormant.

All types of spoilage agents are destroyed when subjected to high temperatures over a period of time. This principle is used in the preservation of food by canning. The temperature of the product is raised to a level fatal to all spoilage agents and is maintained at this level until they are all destroyed. The product is then sealed in sterilized, air-tight containers to prevent recontamination. A product so processed will remain in a preserved state indefinitely.

The exposure time required for the destruction of all spoilage agents depends upon the temperature level. The higher the temperature level, the shorter is the exposure period required. In this regard, moist heat is more effective than dry heat because of its greater penetrating powers. When moist heat is used, the temperature level required is lower and the processing period is shorter. Enzymes and all living microorganisms are destroyed when exposed to the temperature of boiling water for approximately five minutes, but the more resistant bacteria spores may survive at this condition for several hours before succumbing. For this reason, some food products, particularly meats and nonacid vegetables, require long processing periods which frequently result in overcooking of the product. These products are usually processed under pressure so that the processing temperature is increased and the processing time shortened.

Another method of curtailing the activity of spoilage agents is to deprive them of the moisture and/or food which is necessary for their continued activity. Both enzymes and microorganisms require moisture to carry on their activities. Hence, removal of the free moisture from a product will severely limit their activities. The process of moisture removal is called drying (dehydration) and is one of the oldest methods of preserving foods. Drying is accomplished either naturally in the sun and air or artificially in ovens. Dried products which are stored in a cool, dry place will remain in good condition for long periods.

Pickling is essentially a fermentation process, the end result of which is the exhaustion of the substances which serve as food for yeasts and bacteria. The product to be preserved by pickling is immersed in a salt brine solution and fermentation is allowed to take place, during which the sugars contained in the food product are converted to lactic acid, primarily through the action of lactic acid bacteria.

Smoked products are preserved partially by the drying effect of the smoke and partially by antiseptics (primarily creosote) which are absorbed from the smoke.

Too, some products are "cured" with sugar or salt which act as preservatives in that they create conditions unfavorable to the activity of spoilage agents. Some other frequently used preservatives are vinegar, borax, saltpeter, bonzoate of soda, and various spices. A few of the products preserved in this manner are sugar-cured hams, salt pork, spiced fruits, certain beverages, jellies, jams, and preserves.

9-16. Preservation by Refrigeration. The preservation of perishables by refrigeration involves the use of low temperature as a means of eliminating or retarding the activity of spoilage agents. Although low temperatures are not as effective in bringing about the destruction of spoilage agents as are high temperatures, the storage of perishables at low temperatures greatly reduces the activity of both enzymes and microorganisms and thereby provides a practical means of preserving perishables in their original fresh state for varying periods of time. The degree of low temperature required for adequate preservation varies with the type of product stored and with the length of time the product is to be kept in storage.

For purposes of preservation, food products can be grouped into two general categories: (1) those which are alive at the time of distribution and storage and (2) those which are not. Nonliving food substances, such as meat, poultry, and fish, are much more susceptible to microbial contamination and spoilage than are living food substances, and they usually require more stringent preservation methods.

With nonliving food substances, the problem of preservation is one of protecting dead tissue from all the forces of putrification and decay, both enzymic and microbial. In the case of living food substances, such as fruit and vegetables, the fact of life itself affords considerable protection against microbial invasion, and the preservation problem is chiefly one of keeping the food substance alive while at the same time retarding natural enzymic activity in order to slow the rate of maturation or ripening.

Vegetables and fruit are as much alive after harvesting as they are during the growing period. Previous to harvesting they receive a continuous supply of food substances from the growing plant, some of which is stored in the vegetable or fruit. After harvesting, when the vegetable or fruit is cut off from its normal supply of food, the living processes continue through utilization of the previously stored food substances. This causes the vegetable or fruit to undergo changes which will eventually result in deterioration and complete decay of the product. The primary purpose of placing such products under refrigeration is to slow the living processes by retarding enzymic activity, thereby keeping the product in a preserved condition for a longer period.

Animal products (nonliving food substances) are also affected by the activity of natural enzymes. The enzymes causing the most trouble are those which catalyze hydrolysis and oxidation and are associated with the breakdown of animal fats. The principal factor limiting the storage life of animal products, in both the frozen and unfrozen states, is rancidity. Rancidity is caused by oxidation of animal fats and, since some types of animal fats are less stable than others, the storage life of animal products depends in part on fat composition. For example, because of the relative stability of beef fat, the storage life of beef is considerably greater than that of pork or fish whose fatty tissues are much less stable.

Oxidation and hydrolysis are controlled by placing the product under refrigeration so that the activity of the natural enzymes is reduced. The rate of oxidation can be further reduced in the case of animal products by packaging the products in tight-fitting, gas-proof containers which prevent air (oxygen) from reaching the surface of the product. The packaging of fruit and vegetables in gas-proof containers, when stored in the unfrozen state, is not practical. Since these products are alive, packaging in gas-proof containers will cause suffocation and death. A dead fruit or vegetable decays very quickly.

As a general rule, the lower the storage temperature, the longer is the storage life of the product.

9-17. Refrigerated Storage. Refrigerated storage may be divided into three general categories: (1) short-term or temporary storage, (2) long-term storage, and (3) frozen storage. For short- and long-term storage, the product is chilled and stored at some temperature above its freezing point, whereas frozen storage requires freezing of the product and storage at some temperature between 10° F and −10° F, with 0° F being the temperature most frequently employed.

Short-term or temporary storage is usually associated with retail establishments where rapid turnover of the product is normally expected. Depending upon the product, short-term storage periods range from one or two days in some cases to a week or more in others, but seldom for more than fifteen days.

Long-term storage is usually carried out by wholesalers and commercial storage warehouses. Again, the storage period depends on the type of product stored and upon the condition of the product on entering storage. Maximum storage periods for long-term storage range from seven to ten days for some sensitive products, such as ripe tomatoes, cantaloupes, and broccoli, and up to six or eight months for the more durable products, such as onions and some smoked meats. When perishable foods are to be stored for longer periods, they should be frozen and placed in frozen storage. Some fresh foods, however, such as tomatoes, are damaged by the freezing process and therefore cannot be successfully frozen. When such products are to be preserved for long periods, some other method of preservation should be used.

9-18. Storage Conditions. The optimum storage conditions for a product held in either short- or long-term storage depends upon the nature of the individual product, the length of time the product is to be held in storage, and whether the product is packaged or unpackaged. In general, the conditions required for short-term storage are more flexible than those required for long-term storage and, ordinarily, higher storage temperatures are permissible. Recommended storage conditions for both short- and long-term storage and the approximate storage life for various products are listed in Tables 10-10 through 10-13, along with other product data. These data are the result of both experiment and experience and should be followed closely, particularly for long-term storage, if product quality is to be maintained at a high level during the storage period.

9-19. Storage Temperature. Examination of the tables will show that the optimum storage temperature for most products is one slightly above the freezing point of the product. There are, however, notable exceptions.

Although the effect of incorrect storage temperatures generally is to lower product quality and shorten storage life, some fruits and vegetables are particularly sensitive to storage temperature and are susceptible to so-called cold storage diseases when stored at temperatures above or below their critical storage temperatures. For example, citrus fruits frequently develop rind pitting when stored at relatively high temperatures. On the other hand, they are subject to scald (browning of the rind) and watery breakdown when stored at temperatures below their critical temperature. Bananas suffer peel injury when stored below 56° F, whereas celery undergoes soggy breakdown when stored at temperatures above 34° F. Although onions tend to sprout at temperatures above 32° F, Irish potatoes tend to become sweet at storage temperatures below 40° F. Squash, green beans, and peppers develop pits on their surface when stored at or near 32° F. Too, whereas the best storage temperature for most varieties of apples is 30° F to 32° F, some varieties are subject to soft scald and soggy breakdown when stored below 35° F. Others develop brown core at temperatures below 36° F, and still others develop internal browning when stored below 40° F.

9-20. Humidity and Air Motion. The storage of all perishables in their natural state (unpackaged) requires close control not only of the space temperature but also of space humidity and air motion. One of the chief causes of the deterioration of unpackaged fresh foods, such as meat, poultry, fish, fruit, vegetables, cheese, eggs, etc., is the loss of moisture from the surface of the product by evaporation into the surrounding air. This process is known as desiccation or dehydration. In fruit and vegetables, desiccation is accompanied by shriveling and wilting and the product undergoes a considerable loss in both weight and vitamin content. In meats, cheese, etc., desiccation causes discoloration, shrinkage, and heavy trim losses. It also increases the rate of oxidation. Eggs lose moisture through the porous shell, with a resulting loss of weight and general downgrading of the egg.

Desiccation will occur whenever the vapor pressure of the product is greater than the vapor pressure of the surrounding air, the rate of moisture loss from the product being proportional to the difference in the vapor pressures and to the amount of exposed product surface.

The difference in vapor pressure between the product and the air is primarily a function of the relative humidity and the velocity of the air in the storage space. In general, the lower the relative humidity and the higher the air velocity, the greater will be the vapor pressure differential and the greater the rate of moisture loss from the product. Conversely, minimum moisture losses are experienced when the humidity in the storage space is maintained at a high level with low air velocity. Hence, 100% relative humidity and stagnant air are ideal conditions for preventing dehydration of the stored product. Unfortunately, these conditions are also conducive to rapid mold growth and the formation of slime (bacterial) on meats. Too, good circulation of the air in the refrigerated space and around the product is necessary for adequate refrigeration of the product. For these reasons, space humidity must be maintained at somewhat less than 100% and air velocities must be sufficient to provide adequate air circulation. The relative humidities and air velocities recommended for the storage of various products are listed in Tables 10-10 through 10-13.

When the product is stored in vapor-proof containers, space humidity and air velocity are not critical. Some products, such as dried fruits, tend to be hydroscopic and therefore require storage at low relative humidities.

9-21. Mixed Storage. Although the maintenance of optimum storage conditions requires separate storage facilities for most products, this is not usually economically feasible. Therefore, except when large quantities of product are involved, practical considerations often demand that a number of refrigerated products be placed in common storage. Naturally, the difference in the storage conditions required by the various products raises a problem with regard to the conditions to be maintained in a space designed for common storage.

As a general rule, storage conditions in such spaces represent a compromise and usually prescribe a storage temperature somewhat above the optimum for some of the products held in mixed storage. The higher storage temperatures are used in mixed storage in order to minimize the chances of damaging the more sensitive products which are subject to the aforementioned "cold storage diseases" when stored at temperatures below their critical temperature.

Although higher storage temperatures tend to shorten the storage life of some of the products held in mixed storage, this is not ordinarily a serious problem when the products are stored only for short periods as in temporary storage.

For long-term storage, most of the larger wholesale and commercial storage warehouses have a number of separate storage spaces available. General practice in such cases is to group the various products for storage, and only those products requiring approximately the same storage conditions are placed together in common storage.

Another problem associated with mixed storage is that of odor and flavor absorption. Some products absorb and/or give off odors while in storage. Care should be taken not to store such products together even for short periods. Dairy products in particular are very sensitive with regard to absorbing odors and flavors from other products held in mixed storage. On the other hand, potatoes are probably the worst offenders in imparting off-flavors to other products in storage and should never be stored with fruit, eggs, dairy products, or nuts.

9-22. Product Condition on Entering Storage. One of the principal factors determining the storage life of a refrigerated product is the condition of the product on entering storage. It must be recognized that refrigeration merely arrests or retards the natural processes of deterioration and therefore cannot restore to good condition a product which has already deteriorated. Neither can it make a high quality product out of one of initial poor quality. Hence, only vegetables and fruit in good condition should be accepted for storage. Those that have been bruised or otherwise damaged, particularly if the skin has been broken, have lost much of their natural protection against microbial invasion and are therefore subject to rapid spoilage by these agents. Too, as a general rule, since maturation and ripening continue after harvesting, vegetables and fruit intended for storage should be harvested before they are fully mature. The storage life of fully mature or damaged fruit and vegetables is extremely short even under the best storage conditions, and such products should be sent directly to market to avoid excessive losses.

Since a food product begins to deteriorate very quickly after harvesting or killing, it is imperative that preservation measures be taken immediately. To assure maximum storage life with minimum loss of quality, the product should be chilled to the storage temperature as soon as possible after harvesting or killing. When products are to be shipped over long distances to storage, they should be precooled and shipped by refrigerated transport.

9-23. Product Chilling. Product chilling is distinguished from product storage in that the product enters the chilling room at a high temperature (usually either harvesting or killing temperature) and is chilled as rapidly as possible to the storage temperature, whereupon it is normally removed from the chilling room and placed in a holding cooler for storage. The handling of the product during the chilling period has a marked influence on the ultimate quality and storage life of the product.

The recommended conditions for product chilling rooms are given in Tables 10-10 through 10-13. Before the hot product is loaded into the chilling room, the chilling room temperature should be at the "chill finish" temperature. During loading and during the early part of the chilling period, the temperature and vapor pressure differential between the product and the chill room air will be quite large and the product will give off heat and moisture at a high rate. At this time, the temperature and humidity in the chill room will rise to a peak as indicated by the "chill start" conditions in the tables.* At the end of the cycle, the chill room temperature will again drop to the "chill finish" conditions. It is very important that the refrigerating equipment have sufficient capacity

* The temperatures listed in the tables as chill start temperatures are average values and are intended for use in selecting the refrigerating equipment. Actual temperatures in the chilling room during the peak chilling period are usually 3° F to 4° F higher than those listed.

to prevent the chill room temperature from rising excessively during the peak chilling period.

9-24. Relative Humidity and Air Velocity in Chill Rooms. The importance of relative humidity in chilling rooms depends upon the product being chilled, particularly upon whether the product is packaged or not. Naturally, when the product is chilled in vapor-proof containers, the humidity in the chilling room is relatively unimportant. However, during loading and during the initial stages of chilling, chilling room humidity will be high if the containers are wet, but will drop rapidly once the free moisture has been evaporated.

Products chilled in their natural state (unpackaged) lose moisture very rapidly, often producing fog in the chilling room during the early stages of chilling when the product temperature and vapor pressure are high. Rapid chilling and high air velocity are desirable during this time so that the temperature and vapor pressure of the product are lowered as quickly as possible in order to avoid excessive moisture loss and shrinkage. High air velocity is needed also in order to carry away the vapor and thereby prevent condensation of moisture on the surface of the product.

Although high air velocity tends to increase the rate of evaporation of moisture from the product, it also greatly accelerates the chilling rate and results in a more rapid reduction in product temperature and vapor pressure. Since the reduction in vapor pressure resulting from the higher chilling rate more than offsets the increase in the rate of evaporation occasioned by the higher air velocity, the net effect of the higher air velocity during the early stages of chilling is to reduce the over-all loss of moisture from the product. However, during the final stages of chilling, when the temperature and vapor pressure of the product are considerably lower, the effect of high air velocity in the chilling room is to increase the rate of moisture loss from the product. Therefore, the air velocity in the chilling room should be reduced during the final stages of chilling.

As a general rule, the humidity should be kept at a high level when products subject to dehydration are being chilled. Some extremely sensitive products, such as poultry and fish, are frequently chilled in ice slush to reduce moisture losses during chilling. For the same reason, eggs are sometimes dipped in a light mineral oil before chilling and storage. Too, poultry, fish, and some vegetables are often packed in ice for chilling and storage. When products packed in ice are placed in refrigerated storage, the slowly melting ice keeps the surface of the product moist and prevents excessive dehydration.

9-25. Freezing and Frozen Storage. When a product is to be preserved in its original fresh state for relatively long periods, it is usually frozen and stored at approximately 0° F or below. The list of food products commonly frozen includes not only those which are preserved in their fresh state, such as vegetables, fruit, fruit juices, berries, meat, poultry, sea foods, and eggs (not in shell), but also many prepared foods, such as breads, pastries, ice cream, and a wide variety of specially prepared and precooked food products, including full dinners.

The factors governing the ultimate quality and storage life of any frozen product are:

1. The nature and composition of the product to be frozen
2. The care used in selecting, handling, and preparing the product for freezing
3. The freezing method
4. The storage conditions.

Only high quality products in good condition should be frozen. With vegetables and fruit, selecting the proper variety for freezing is very important. Some varieties are not suitable for freezing and will result in a low quality product or in one with limited keeping qualities.

Vegetables and fruit to be frozen should be harvested at the peak of maturity and should be processed and frozen as quickly as possible after harvesting to avoid undesirable chemical changes through enzymic and microbial action.

Both vegetables and fruit require considerable processing before freezing. After cleaning and washing to remove foreign materials—leaves, dirt, insects, juices, etc.—from their surfaces, vegetables are "blanched" in hot water or steam at 212° F in order to destroy the natural enzymes. It will be remembered that enzymes are not destroyed by low temperature and, although greatly reduced, their activity continues at a slow rate even in food stored at 0° F and below. Hence, blanching, which destroys

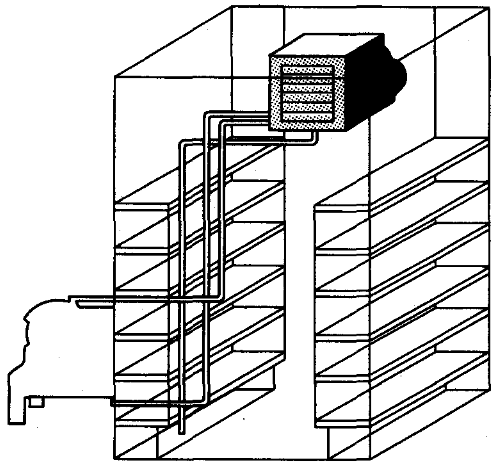
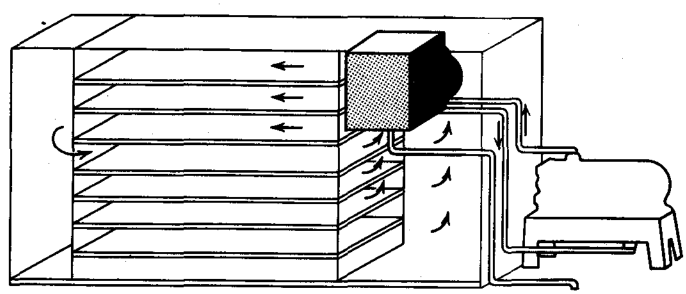


Fig. 9-2. Walk-in installation. Suspended blast freezer provides high-velocity air for fast freezing, saving valuable floor space in small areas. (Courtesy Carrier Corporation.)

Fig. 9-3. Suspended blast freezer applied to reach-in cabinet distributes blast air through shelves. (Courtesy Carrier Corporation.)



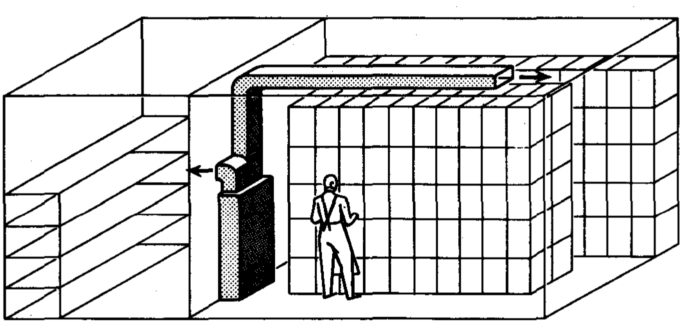


Fig. 9-4. Freezing in one room and storage in another is accomplished by single, floor-mounted blast freezers. (Courtesy Carrier Corporation.)

most of the enzymes, greatly increases the storage life of frozen vegetables. The time required for blanching varies with the type and variety of the vegetable and ranges from 1 to $1\frac{1}{2}$ min for green beans to 11 min for large ears of corn. Although much of the microbial population is destroyed along with the enzymes during the blanching process, many bacteria survive. To prevent spoilage by these viable bacteria, vegetables should be chilled to 50° F immediately after blanching and before they are packaged for the freezer.

Like vegetables, fruit must also be cleaned and washed to remove foreign materials and to reduce microbial contamination. Although fruit is perhaps even more subject to enzymic deterioration than are vegetables, it is never blanched to destroy the natural enzymes since to do so would destroy the natural fresh quality which is so desirable.

The enzymes causing the most concern with regard to frozen fruit are the ones which catalyze oxidation and result in rapid browning of the flesh. To control oxidation, fruit to be frozen is covered with a light sugar syrup. In some cases, ascorbic acid, citric acid, or sulfur dioxide are also used for this purpose.

As a general rule, meat products do not require any special processing prior to freezing. However, because of consumer demand, specially prepared meats and meat products are being frozen in increasing amounts. This is true also of poultry and sea foods.

Because of the relative instability of their fatty tissue, pork and fish are usually frozen as soon after chilling as possible. On the other hand, beef is frequently "aged" in a chilling cooler for several days before freezing. During this time the beef is tenderized to some extent by enzymic activity. However, the aging of beef decreases its storage life, particularly if the aging period exceeds 6 or 7 days.

With poultry, experiments indicate that poultry frozen within 12 to 24 hr after killing is more tender than that frozen immediately after killing. However, delaying freezing beyond 24 hr tends to reduce storage life without appreciable increasing tenderness.

9-26. Freezing Methods. Food products may be either sharp (slow) frozen or quick frozen. Sharp freezing is accomplished by placing the product in a low temperature room and allowing it to freeze slowly, usually in still air. The temperature maintained in sharp freezers ranges from 0° F to −40° F. Since air circulation is usually by natural convection, heat transfer from the product ranges from 3 hr to 3 days, depending upon the bulk of the product and upon the conditions in the sharp freezer. Typical items which are sharp frozen are beef and pork half-carcasses, boxed poultry, panned and whole fish, fruit in barrels and other large containers, and eggs (whites, yolks, or whole) in 10 and 30 lb cans.

Quick freezing is accomplished in any one or in any combination of three ways: (1) immersion, (2) indirect contact, and (3) air blast.

9-27. Air Blast Freezing. Air blast freezing utilizes the combined effects of low temperature and high air velocity to produce a high rate of heat transfer from the product. Although the method employed varies considerably with the application, blast freezing is accomplished by circulating high-velocity, low-temperature air around the product. Regardless of the method used, it is important that the arrangement of the freezer is such that air can circulate freely around all parts of the product.

Packaged blast freezers are available in both suspended and floor-mounted models. Typical applications are shown in Figs. 9-2 through 9-4. Blast freezing is frequently carried out in insulated tunnels, particularly where large quantities of product are to be frozen (Figs. 9-5 and 9-6). In some instances, the product is carried through the freezing tunnel and frozen on slow-moving, mesh conveyor belts. The unfrozen product is placed on the conveyor at one end of the tunnel and is frozen by the time it reaches the other end. Another method is to load the product on tiered dollies. The dollies are pushed into the tunnel and the product is frozen; whereupon they are pushed out of the freezing tunnel into a storage room (Fig. 9-5).

Although blast freezing is used to freeze nearly all types of products, it is particularly suitable for freezing products of nonuniform or irregular sizes and shapes, such as dressed poultry.

9-28. Indirect Contact Freezing. Indirect freezing is usually accomplished in plate freezers and involves placing the product on metal plates through which a refrigerant is circulated

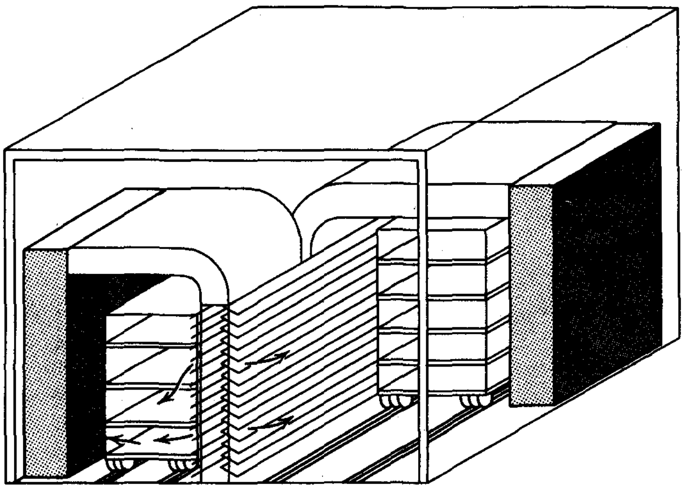


Fig. 9-5. Packaged blast freezers applied to freezing tunnel. High velocity, —15° F air is blasted through trucks. (Courtesy Carrier Corporation.)

(Fig. 9-7). Since the product is in direct thermal contact with the refrigerated plate, heat transfer from the product occurs primarily by conduction so that the efficiency of the freezer depends, for the most part, on the amount of contact surface. This type of freezer is particularly useful when products are frozen in small quantities.

One type of plate freezer widely used by the larger commercial freezers to handle small, flat, rectangular, consumer-size packages is the multiplate freezer. The multiplate freezer consists of a series of horizontal, parallel, refrigerated plates which are actuated by hydraulic pressure so that they can be opened to receive the product between them and then closed on the product with any desired pressure. When the plates are closed, the packages are held tightly between the plates. Since both the top and the bottom of the packages are in good thermal contact with the refrigerated plates, the rate of heat transfer is high and the product is quickly frozen.

9-29. Immersion Freezing. Immersion freezing is accomplished by immersing the product in a low temperature brine solution, usually either sodium chloride or sugar. Since the refrigerated liquid is a good conductor and is in good thermal contact with all the product, heat transfer is rapid and the product is completely frozen in a very short time.

Another advantage of immersion freezing is that the product is frozen in individual units rather than fused together in a mass.

The principal disadvantage of immersion freezing is that juices tend to be extracted from the product by osmosis. This results in contamination and weakening of the freezing solution. Too, where a sodium chloride brine is used, salt penetration into the product may sometimes be excessive. On the other hand, when fruit is frozen in a sugar solution, sugar penetration into the fruit is entirely beneficial.

The products most frequently frozen by immersion are fish and shrimp. Immersion is particularly suitable for freezing fish and shrimp at sea, since the immersion freezer is relatively compact and space aboard ship is at a premium. In addition, immersion freezing produces a "glaze" (thin coating of ice) on the surface of the product which helps to prevent dehydration of unpackaged products during the storage period.

9-30. Quick Freezing vs. Sharp Freezing. Quick frozen products are nearly always superior to those which are sharp (slow) frozen. D. K. Tressler, in 1932, summarized the views of R. Plank, H. F. Taylor, C. Birdseye, and G. A. Fitzgerald, and stated the following as the main advantages of quick freezing over slow freezing:

1. The ice crystals formed are much smaller, and therefore cause much less damage to cells.

2. The freezing period being much shorter, less time is allowed for the diffusion of salts and the separation of water in the form of ice.

3. The product is quickly cooled below the temperature at which bacterial, mold, and yeast

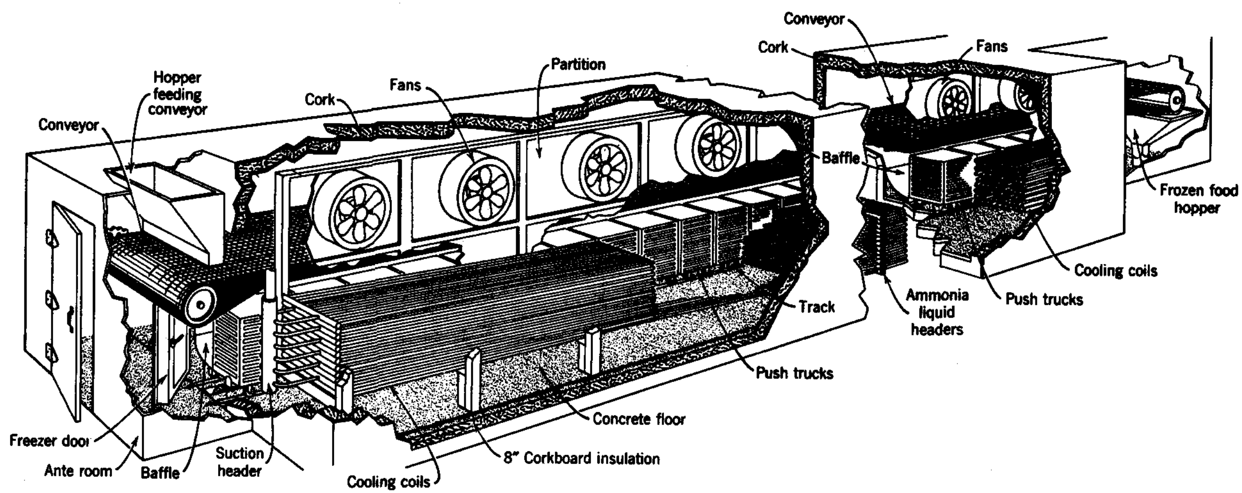


Fig. 9-6. Tunnel freezer for quick-freezing foods. (Courtesy Frick Company.)

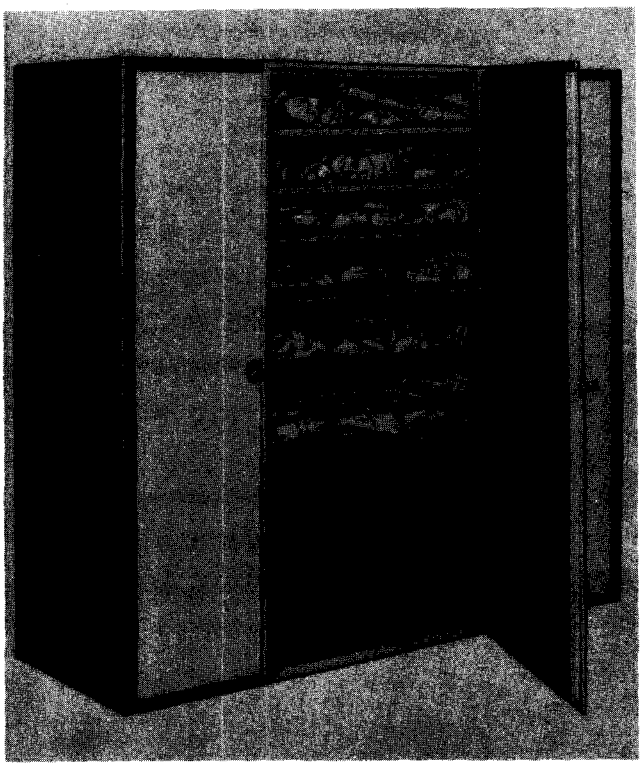


Fig. 9-7. Plate freezer for indirect contact freezing. (Courtesy Dole Refrigerating Company.)

growth occurs, thus preventing decomposition during freezing.*

The principal difference between quick freezing and sharp freezing is in the size, number, and location of the ice crystals formed in the product as cellular fluids are solidified. When a product is slow frozen, large ice crystals are formed which result in serious damage to the tissue of some products through cellular breakdown. Quick freezing, on the other hand, produces smaller ice crystals which are formed almost entirely within the cell so that cellular breakdown is greatly reduced. Upon thawing, products which have experienced considerable cellular damage are prone to lose excessive amounts of fluids through "drip" or "bleed," with a resulting loss of quality.

Ice-crystal formation begins in most products at a temperature of approximately 30° F and, although some extremely concentrated fluids still remain unfrozen even at temperatures below −50° F, most of the fluids are solidified by the time the product temperature is lowered to 25° F. The temperature range between 30° F and 25° F is often referred to as the zone of maximum ice-crystal formation, and rapid heat removal through this zone is desirable from the standpoint of product quality. This is particularly true for fruits and vegetables because both undergo serious tissue damage when slow frozen.

Since animal tissue is much tougher and much more elastic than plant tissue, the freezing rate is not as critical in the freezing of meats and meat products as it is in fruits and vegetables. Recent experiment indicates that poultry and fish suffer little, if any, cellular damage when slow frozen. This does not mean, however, that quick frozen meats are not superior to those which are slow frozen, but only that, for the standpoint of cellular damage, quick freezing is not as important in the freezing of meats as it is in fruits and vegetables. For example, poultry that is slow frozen takes on a darkened appearance which makes it much less attractive to the consumer. This alone is enough to justify the quick freezing of poultry. Too, in all cases, quick freezing reduces the processing time and, consequently, the amount

* *Air Conditioning Refrigerating Data Book*, Applications Volume, 5th Edition, American Society of Refrigerating Engineers, 1954–55, p. 1–02.

of bacterial deterioration. This is especially worthwhile in the processing of fish because of their tendency to rapid spoilage.

9-31. Packaging Materials. Dehydration, one of the principal factors limiting the storage life of frozen foods, is greatly reduced by proper packaging. Unpackaged products are subject to serious moisture losses not only during the freezing process but also during the storage period. While in storage, unpackaged frozen products lose moisture to the air continuously by sublimation. This eventually results in a condition known as "freezer-burn," giving the product a white, leathery appearance. Freezer-burn is usually accompanied by oxidation, flavor changes, and loss of vitamin content.

With few exceptions, all products are packaged before being placed in frozen storage. Although most products are packaged before freezing, some, such as loose frozen peas and lima beans, are packaged after the freezing process.

To provide adequate protection against dehydration and oxidation, the packaging material should be practically 100% gas and vapor proof and should fit tightly around the product to exclude as much air as possible. Too, air spaces in packages have an insulating effect which reduce the freezing rate and increase freezing costs.

The fact that frozen products are in competition to products preserved by other methods introduces several factors which must be taken into account when selecting packaging materials. When the product is to be sold directly to the consumer, the package must be attractive and convenient to use in order to stimulate sales. From a cost standpoint, the package should be relatively inexpensive and of such a nature that it permits efficient handling so as to reduce processing costs.

Some packaging materials in general use are aluminum foil, tin cans, impregnated paper-board cartons, paper-board cartons over-wrapped with vapor-proof wrappers, wax paper, cellophane, polyethylene, and other sheet plastics.

Frozen fish are often given an ice glaze (a thin coating of ice) which provides an excellent protective covering. However, since the ice glaze is very brittle, glazed fish must be handled carefully to avoid breaking the glaze. Too, since the ice glaze gradually sublimes to the air, the fish must be reglazed approximately once a month by dipping into fresh water or by spraying.

9-32. Frozen Storage. The exact temperature required for frozen storage is not critical, provided that it is sufficiently low and that it does not fluxuate. Although 0° F is usually adequate for short-term (retail) storage, −5° F is the best temperature for all-around long-term (wholesale) storage. When products having unstable fats (oxidizable, free, fatty acids) are stored in any quantity, the storage temperature should be held at −10° F or below in order to realize the maximum storage life.

When products are stored above −20° F, which is normally the case, the temperature of the storage room should be maintained constant with a variation of not more than 1° F in either direction. Variations in storage temperature cause alternate thawing and refreezing of some of the juices in the product. This tends to increase the size of the ice crystals in the product and eventually results in the same type of cellular damage as occurs with slow freezing.

Since many packaging materials do not offer complete protection against dehydration, the relative humidity should be kept at a high level (85% to 90%) in frozen storage rooms, particularly for long-term storage.

Proper stacking of the product is also essential. Stacking should always be such that it permits adequate air circulation around the product. It is particularly important to leave a good size air space between the stored product and the walls of the storage room. In addition to permitting air circulation around the product, this eliminates the possibility of the product absorbing heat directly from the warm walls.

9-33. Commercial Refrigerators. The term "commercial refrigerator" is usually applied to the smaller, ready-built, refrigerated fixtures of the type used by retail stores and markets, hotels, restaurants, and institutions for the processing, storing, displaying, and dispensing of perishable commodities. The term is sometimes applied also to the larger, custom-built refrigerated fixtures and rooms used for these purposes.

Although there are a number of special purpose refrigerated fixtures which defy classification, in general, commercial fixtures can

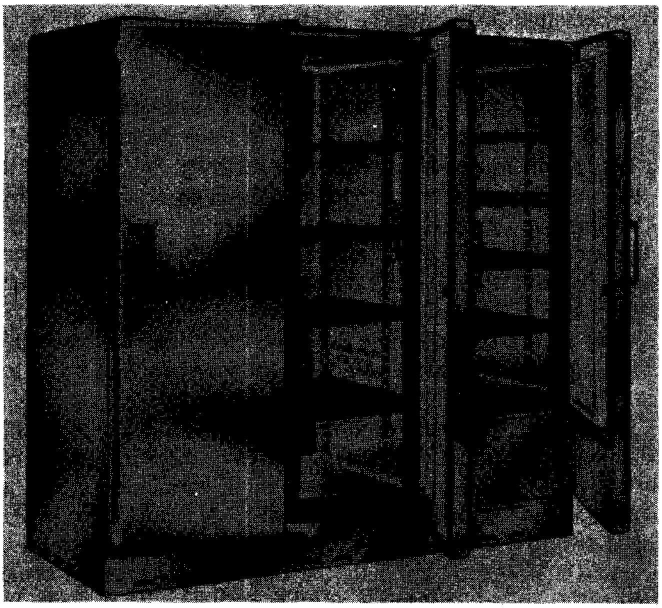


Fig. 9-8. Typical reach-in refrigerator. (Courtesy Tyler Refrigeration Corporation.)

be grouped into three principal categories: (1) reach-in refrigerators, (2) walk-in coolers, and (3) display cases.

9-34. Reach-In Refrigerators. The reach-in refrigerator is probably the most versatile and the most widely used of all commercial fixtures. Typical users are grocery stores, meat markets, bakeries, drug stores, lunch counters, restaurants, florists, hotels, and institutions of all kinds. Whereas some reach-in refrigerators serve only a storage function, others are used for both storage and display (Fig. 9-8). Those serving only the storage function usually have solid doors, whereas those used for display have glazed doors.

9-35. Walk-In Coolers. Walk-in coolers are primarily storage fixtures and are available in a wide variety of sizes to fit every need. Nearly all retail stores, markets, hotels, restaurants, institutions, etc., of any size employ one or more walk-in coolers for the storage of perishables of all types. Some walk-in coolers are equipped with glazed reach-in doors. This feature is especially convenient for the storing, displaying, and dispensing of such items as dairy products, eggs, and beverages. Walk-in coolers with reach-in doors are widely used in grocery stores, particularly drive-in groceries, for handling such items.

9-36. Display Cases. The principal function of any kind of display fixture is to display the product or commodity as attractively as possible in order to stimulate sales. Therefore, in the design of refrigerated display fixtures, first consideration is given to the displaying of the product. In many cases, this is not necessarily compatible with providing the optimum storage conditions for the product being displayed. Hence, the storage life of a product in a display fixture is frequently very limited, ranging from a few hours in some instances to a week or more in others, depending upon the type of product and upon the type of fixture.

Display fixtures are of two general types: (1) the self-service case, from which the customer serves himself directly, and (2) the service case, from which the customer is usually served by an attendant. The former is very popular in supermarkets and other large, retail, self-service establishments, whereas the service case finds use in the smaller groceries, markets, bakeries, etc. Typical service cases are shown in Figs. 9-9 and 9-10.

Self-service cases are of two types, open and closed, with the open type gaining rapidly in popularity. With the advent of the supermarket, the trend has been increasingly toward the open type self-service case, and the older, closed type self-service cases are becoming obsolete. Several of the more popular types of open self-service cases are shown in Figs. 9-11 and 9-12. These are used to display meat vegetables, fruit, frozen foods, ice cream, dairy products, delicatessen items, etc. The design of the case varies somewhat with the particular type of product being displayed. Too, designs are available for both wall and island installation. Although some provide additional storage space, others do not.

9-37. Special Purpose Fixtures. Although all the refrigerated fixtures discussed in the preceding sections are available in a variety of designs in order to satisfy the specific requirements of individual products and applications, a number of special purpose fixtures is manufactured which may or may not fall into one of the three general categories already mentioned. Some of the more common special purpose fixtures are

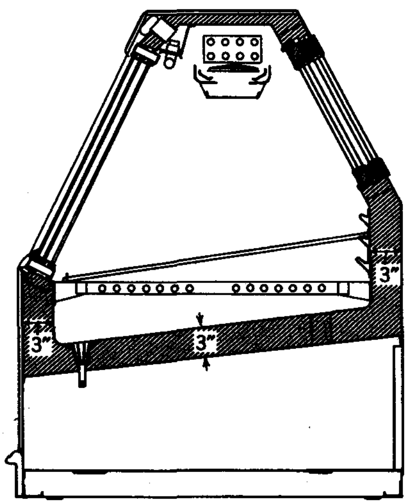


Fig. 9-9. Conventional single-duty service case for displaying meats. (Courtesy Tyler Refrigeration Corporation.)

beverage coolers, milk coolers (dairy farm), milk and beverage dispensers, soda fountains, ice cream makers, water coolers, ice makers, back-bar refrigerators, florist boxes, dough retarders, candy cases, and mortuary refrigerators.

9-38. Frozen Food Locker Plants. Normally, the function of a frozen food locker plant is to process and freeze foods for individual families and other groups, either for take-home storage or for storage at the locker plant. When storage is at the plant, the customer rents a storage space (locker) and calls at the plant for one or more packages as needed.

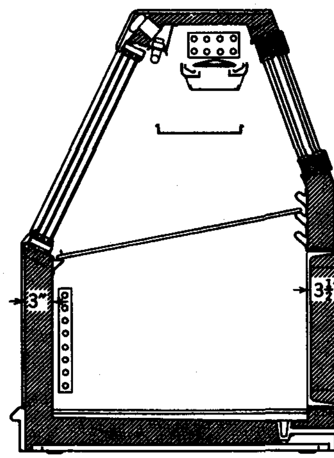


Fig. 9-10. Double-duty service case for displaying meats. (Courtesy Tyler Refrigeration Corporation.)

As a general rule, a locker plant furnishes all or most of the following facilities and/or services:

1. A chilling room for chilling freshly killed meats.
2. A cold storage room for holding products under refrigeration while awaiting preparation and processing prior to freezing.
3. A processing room where the products are processed and packaged for the freezer.

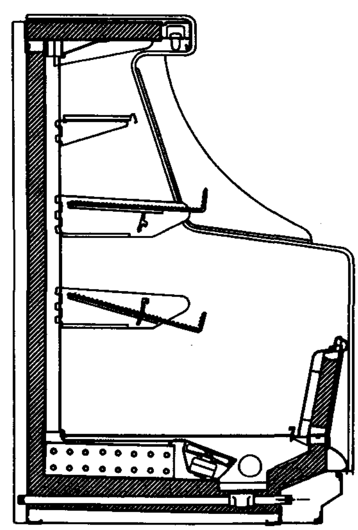


Fig. 9-11. High multishelf produce sales case. (Courtesy Tyler Refrigeration Corporation.)

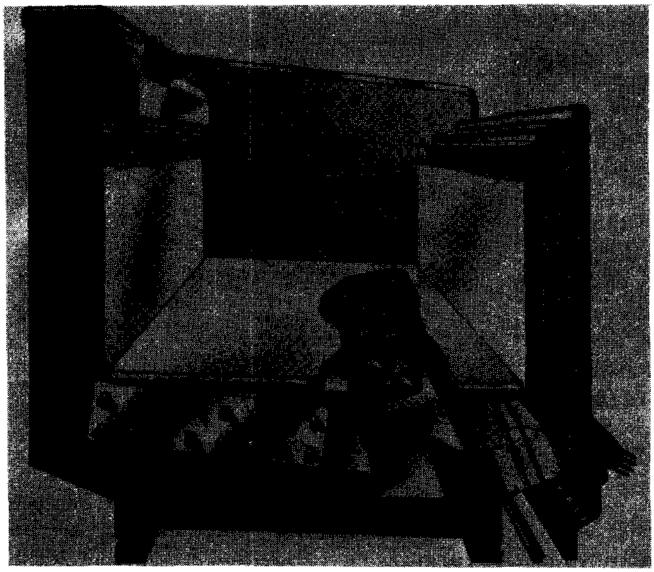


Fig. 9-12. Open-type display case for frozen foods and ice cream. (Courtesy Tyler Refrigeration Corporation.)

4. A freezing room or cabinet in which the food is frozen prior to being placed in storage.

5. A low temperature room containing the storage lockers.

6. A low temperature bulk storage room.

7. An aging room where certain meats are kept under refrigeration and allowed to age (tenderize) for periods usually ranging from 7 to 10 days.

8. A curing and smoking room for handling bacon, ham, sausage, and other cured meats.

Services such as slaughtering, lard rendering, sausage making, etc., are also provided by some plants.

The layout of a typical locker plant is shown in Fig. 9-13. The recommended design conditions for the various spaces in the locker plant are given in Fig. 9-14. The average size of the

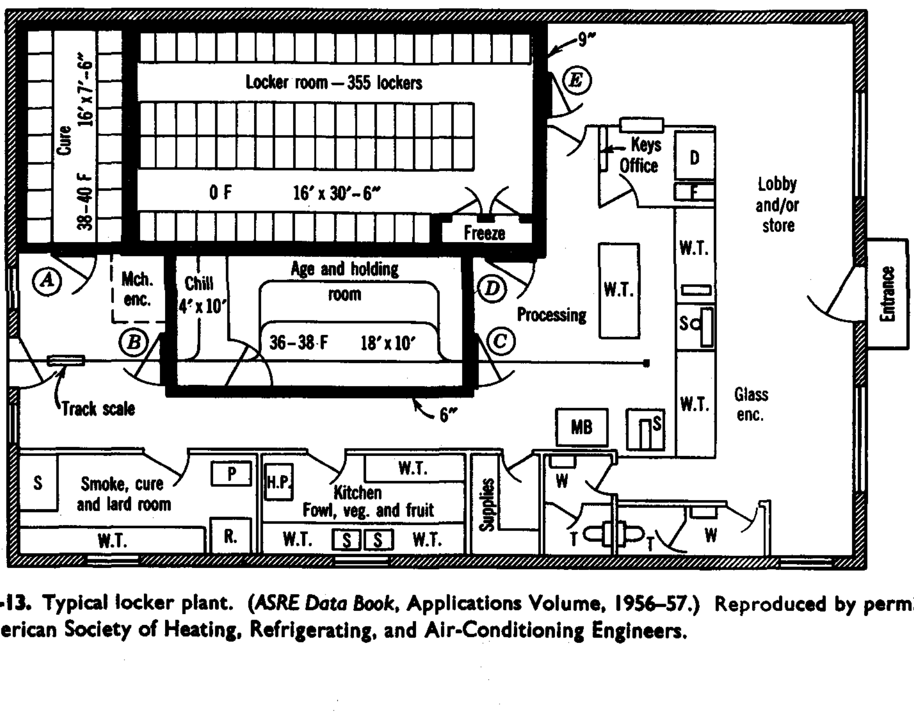


Fig. 9-13. Typical locker plant. (*ASRE Data Book*, Applications Volume, 1956–57.) Reproduced by permission of American Society of Heating, Refrigerating, and Air-Conditioning Engineers.

Locker Plant Design Conditions

Type of space	Room temperature	Refrigerant temperature	Insulation thickness, inches.
Work room, process room, and kitchen	Atmospheric	None	None
Chill room	34 to 36 F Design for 35 F	20 to 25 F below room temperature, for gravity circulation; 10 to 15 F below room temperature for forced air circulation	3 to 8
Aging room	34 to 36 F Design for 35 F	Same as chill room	3 to 8
Curing room	38 to 40 F Design for 40 F	Same as chill room	3 to 8
Freezing room (gravity air circulation)	−10 to −20 F	−20 to −30 F	6 to 12
Freezer cabinet (in locker room)	Not important	−15 to −20 F	1 or 2
Blast freezer	Depends on type of system used	−10 to −15 F	6 to 12
Locker room or bulk storage	0 F	−15 to −20 F	6 to 12

Fig. 9–14. (*ASRE Data Book*, Applications Volume, 1956–57. Reproduced by permission of American Society of Heating, Refrigerating and Air Conditioning Engineers.)

individual locker is 6 cu ft and the average product storage capacity is approximately 35 to 40 lb per cubic foot. Minimum product turnover is approximately 2 lb per locker per day. Standard practice is to base chilling room and freezer capacities on the handling of 2 to 4 lb of product per locker per day.

9-39. Summary. Recognizing that a thorough knowledge of the application itself is a prerequisite to good system design and proper equipment selection, we have devoted the material in this chapter to a brief survey of a few of the applications of mechanical refrigeration, with special emphasis being given to the area of commercial refrigeration.

Obviously, the applications of mechanical refrigeration are too many and too varied to permit detailed consideration of each and every type. Fortunately, this is neither necessary nor desirable since methods of system designing and equipment selection are practically the same for all types of applications. Commerical refrigeration was selected for emphasis because this area embraces a wide range of applications and because the problems encountered in this area are representative of those in the other areas. Hence, even though the discussion in this chapter and in those which follow deals chiefly with commercial refrigeration, the principals of system design and the methods of equipment selection developed therein may be applied to all types of mechanical refrigeration applications.

Although no attempt is made in this book to discuss air conditioning as such except in a very general way, it should be pointed out that most commercial refrigeration applications, particularly those concerned with product storage, involve air conditioning in that they ordinarily include close control of the temperature, humidity, motion, and cleanliness of the air in the refrigerated space.

10

Cooling Load Calculations

10-1. The Cooling Load. The cooling load on refrigerating equipment seldom results from any one single source of heat. Rather, it is the summation of the heat which usually evolves from several different sources. Some of the more common sources of heat which supply the load on refrigerating equipment are:

1. Heat that leaks into the refrigerated space from the outside by conduction through the insulated walls.
2. Heat that enters the space by direct radiation through glass or other transparent materials.
3. Heat that is brought into the space by warm outside air entering the space through open doors or through cracks around windows and doors.
4. Heat given off by a warm product as its temperature is lowered to the desired level.
5. Heat given off by people occupying the refrigerated space.
6. Heat given off by any heat-producing equipment located inside the space, such as electric motors, lights, electronic equipment, steam tables, coffee urns, hair driers, etc.

The importance of any one of these heat sources with relation to the total cooling load on the equipment varies with the individual application. Not all them will be factors in every application, nor will the cooling load in any one application ordinarily include heat from all these sources. However, in any given application, it is essential that consideration be given to all heat sources present and that all the heat evolving from them be taken into account in the over-all calculation.

10-2. Equipment Running Time. Although refrigerating equipment capacities are normally given in Btu per hour, in refrigeration applications the total cooling load is usually calculated for a 24-hr period, that is, in Btu per 24 hr. Then, to determine the required Btu per hour capacity of the equipment, the total load for the 24-hr period is divided by the desired running time for the equipment, viz:

$$\begin{array}{l}\text{Required Btu/hr}\\ \text{equipment}\\ \text{capacity}\end{array} = \frac{\text{Total cooling load, Btu/24 hr}}{\text{Desired running time}} \qquad (10\text{-}1)$$

Because of the necessity for defrosting the evaporator at frequent intervals, it is not practical to design the refrigerating system in such a way that the equipment must operate continuously in order to handle the load. In most cases, the air passing over the cooling coil is chilled to a temperature below its dew point and moisture is condensed out of the air onto the surface of the cooling coil. When the temperature of the coil surface is above the freezing temperature of water, the moisture condensed out of the air drains off the coil into the condensate pan and leaves the space through the condensate drain. However, when the temperature of the cooling coil is below the freezing temperature of water, the moisture condensed out of the air freezes into ice and adheres to the surface of the coil, thereby causing "frost" to accumulate on the coil surface. Since frost accumulation on the coil surface tends to insulate the coil and reduce the coil's capacity, the frost must be melted off periodically by raising the surface temperature of the coil above the freezing point of water and maintaining it at this level until the frost has melted off the coil and left the space through the condensate drain.

No matter how the defrosting is accomplished, the defrosting requires a certain amount of time, during which the refrigerating effect of the system must be stopped.

One method of defrosting the coil is to stop the compressor and allow the evaporator to warm up to the space temperature and remain at this temperature for a sufficient length of time to allow the frost accumulation to melt off the coil. This method of defrosting is called "off-cycle" defrosting. Since the heat required to melt the frost in off-cycle defrosting must come from the air in the refrigerated space, defrosting occurs rather slowly and a considerable length of time is required to complete the process. Experience has shown that when off-cycle defrosting is used, the maximum allowable running time for the equipment is 16 hr out of each 24-hr period, the other 8 hr being allowed for the defrosting. This means, of course, that the refrigerating equipment must have sufficient capacity to accomplish the equivalent of 24 hr of cooling in 16 hr of actual running time. Hence, when off-cycle defrosting is used, the equipment running time used in Equation 10-1 is approximately 16 hr.

When the refrigerated space is to be maintained at a temperature below 34° F, off-cycle defrosting is not practical. The variation in space temperature which would be required in order to allow the cooling coil to attain a temperature sufficiently high to melt off the frost during every off cycle would be detrimental to the stored product. Therefore where the space temperature is maintained below 34° F, some method of automatic defrosting is ordinarily used. In such cases the surface of the coil is heated artificially, either with electric heating elements, with water, or with hot gas from the discharge of the compressor (see Chapter 20).

Defrosting by any of these means is accomplished much more quickly than when off-cycle defrosting is used. Hence, the off-cycle time required is less for automatic defrosting and the maximum allowable running time for the equipment is greater than for the aforementioned off-cycle defrosting. For systems using automatic defrosting the maximum allowable running time is from 18 to 20 hr out of each 24-hr period, depending upon how often defrosting is necessary for the application in question. As a general rule, the 18 hr running time is used.

It is of interest to note that since the temperature of the cooling coil in comfort air conditioning applications is normally around 40° F, no frost accumulates on the coil surface and, therefore, no off-cycle time is required for defrosting. For this reason, air conditioning systems are usually designed for continuous run and cooling loads for air conditioning applications are determined directly in Btu per hour.

10-3. Cooling Load Calculations. To simplify cooling load calculations, the total cooling load is divided into a number of individual loads according to the sources of heat supplying the load. The summation of these individual loads is the total cooling load on the equipment.

In commercial refrigeration, the total cooling load is divided into four separate loads, viz: (1) the wall gain load, (2) the air change load, (3) the product load, and (4) the miscellaneous or supplementary load.

10-4. The Wall Gain Load. The wall gain load, sometimes called the wall leakage load, is a measure of the heat which leaks through the walls of the refrigerated space from the outside to the inside. Since there is no perfect insulation, there is always a certain amount of heat passing from the outside to the inside whenever the inside temperature is below that of the outside. The wall gain load is common to all refrigeration applications and is ordinarily a considerable part of the total cooling load. Some exceptions to this are liquid chilling applications, where the outside area of the chiller is small and the walls of the chiller are well insulated. In such cases, the leakage of heat through the walls of the chiller is so small in relation to the total cooling load that its effect is negligible and it is usually neglected. On the other hand, commercial storage coolers and residential air conditioning applications are both examples of applications wherein the wall gain load usually accounts for the greater portion of the total load.

10-5. The Air Change Load. When the door of a refrigerated space is opened, warm outside air enters the space to replace the more dense cold air which is lost from the refrigerated space through the open door. The heat which must be removed from this warm outside air to reduce its temperature to the space temperature becomes a part of the total cooling load on the equipment. This part of the total load is called the air change load.

The relationship of the air change load to the total cooling load varies with the application. Whereas in some applications the air change load is not a factor at all, in others it represents a considerable portion of the total load. For example, with liquid chillers, there are no doors or other openings through which air can pass and therefore the air change load is nonexistent. On the other hand, the reverse is true for air conditioning applications, where, in addition to the air changes brought about by door openings, there is also considerable leakage of air into the conditioned space through cracks around windows and doors and in other parts of the structure. Too, in many air conditioning applications outside air is purposely introduced into the conditioned space to meet ventilating requirements. When large numbers of people are in the conditioned space, the quantity of fresh air which must be brought in from the outside is quite large and the cooling load resulting for the cooling of this air to the temperature of the conditioned space is often a large part of the total cooling load in such applications.

In air conditioning applications, the air change load is called either the ventilating load or the infiltration load. The term ventilating load is used when the air changes in the conditioned space are the result of deliberate introduction of outside air into the space for ventilating purposes. The term infiltration load is used when the air changes are the result of the natural infiltration of air into the space through cracks around windows and doors. Every air conditioning application will involve either an infiltration load or a ventilating load, but never both in the same application.

Since the doors on commercial refrigerators are equipped with well-fitted gaskets, the cracks around the doors are tightly sealed and there is little, if any, leakage of air around the doors of a commercial fixture in good condition. Hence, in commercial refrigeration, the air changes are usually limited to those which are brought about by actual opening and closing of the door or doors.

10-6. The Product Load. The product load is made up of the heat which must be removed from the refrigerated product in order to reduce the temperature of the product to the desired level. The term product as used here is taken to mean any material whose temperature is reduced by the refrigerating equipment and includes not only perishable commodities, such as foodstuff, but also such items as welding electrodes, masses of concrete, plastic, rubber, and liquids of all kinds.

The importance of the product load in relation to the total cooling load, like all others, varies with the application. Although it is nonexistent in some applications, in others it represents practically the entire cooling load. Where the refrigerated cooler is designed for product storage, the product is usually chilled to the storage temperature before being placed in the cooler and no product load need be considered since the product is already at the storage temperature. However, in any instance where the product enters a storage cooler at a temperature above the storage temperature, the quantity of heat which must be removed from the product in order to reduce its temperature to the storage temperature must be considered as a part of the total load on the cooling equipment.

In some few instances, the product enters the storage fixture at a temperature below the normal storage temperature for the product. A case in point is ice cream which is frequently chilled to a temperature of 0° F or −10° F during the hardening process, but is usually stored at about 10° F, which is the ideal dipping temperature. When such a product enters storage at a temperature below the space temperature, it will absorb heat from the storage space as it warms up to the storage temperature and thereby produce a certain amount of refrigerating effect of its own. In other words, it provides what might be termed a negative product load which could theoretically be subtracted from the total cooling load. This is never done, however, since the refrigerating effect produced is small and is not continuous in nature.

The cooling load on the refrigerating equipment resulting from product cooling may be either intermittent or continuous, depending on the application. The product load is a part of the total cooling load only while the temperature of the product is being reduced to the storage temperature. Once the product is cooled to the storage temperature, it is no longer a source of heat and the product load ceases to be a part of the load on the equipment. An exception to this is in the storage of fruit and vegetables which give off respiration heat for the entire time they are in storage even though there is no further decrease in their temperature (see Section 10-17).

There are, of course, a number of refrigeration applications where product cooling is more or less continuous, in which case the product load is a continuous load on the equipment. This is true, for instance, in chilling coolers where the primary function is to chill the warm product to the desired storage temperature. When the product has been cooled to the storage temperature, it is usually moved out of the chilling room into a storage room and the chilling room is then reloaded with warm product. In such cases, the product load is continuous and is usually a large part of the total load on the equipment.

Liquid chilling is another application wherein the product provides a continuous load on the refrigerating equipment. The flow of the liquid being chilled through the chiller is continuous with warm liquid entering the chiller and cold liquid leaving. In this instance, the product load is practically the only load on the equipment since there is no air change load and the wall gain load is negligible, as is the miscellaneous load.

In air conditioning applications there is no product load as such, although there is often a "pull-down load," which, in a sense, may be thought of as a product load.

10-7. The Miscellaneous Load. The miscellaneous load, sometimes referred to as the supplementary load, takes into account all miscellaneous sources of heat. Chief among these are people working in or otherwise occupying the refrigerated space along with lights or other electrical equipment operating inside the space.

In most commercial refrigeration applications the miscellaneous load is relatively small, usually consisting only of the heat given off by lights and fan motors used inside the space.

In air conditioning applications, there is no miscellaneous load as such. This is not to say that human occupancy and equipment are not a part of the cooling load in air conditioning applications. On the contrary, people and equipment are often such large factors in the air conditioning load that they are considered as separate loads and are calculated as such. For example, in those air conditioning applications where large numbers of people occupy the conditioned space, such as churches, theaters, restaurants, etc., the cooling load resulting from human occupancy is frequently the largest single factor in the total load. Too, many air conditioning systems are installed for the sole purpose of cooling electrical, electronic, and other types of heat-producing equipment. In such cases, the equipment usually supplies the greater portion of the cooling load.

10-8. Factors Determining the Wall Gain Load. The quantity of heat transmitted through the walls of a refrigerated space per unit of time is the function of three factors whose relationship is expressed in the following equation:

$$Q = A \times U \times D \qquad (10\text{-}2)$$

where Q = the quantity of heat transferred in Btu/hr
A = the outside surface area of the wall (square feet)
U = the over-all coefficient of heat transmission (Btu/hr/sq ft/° F)
D = the temperature differential across the wall (° F)

The coefficient of transmission or "U" factor is a measure of the rate at which heat will pass through a 1 sq ft area of wall surface from the air on one side to the air on the other side for each 1° F of temperature difference across the wall. The value of the U factor is given in Btu per hour and depends on the thickness of the wall and on the materials used in the wall construction. Since it is desirable to prevent as much heat as possible from entering the space and becoming a load on the cooling equipment, the materials used in the construction of cold storage walls should be good thermal insulators so that the value of U is kept as low as is practical.

According to Equation 10-2, once the U factor is established for a wall, the rate of heat flow through the wall varies directly with the surface area of the wall and with the temperature differential across the wall. Since the value of U is given in Btu/hr/sq ft/° F, the total quantity of heat passing through any given wall in 1 hr can be determined by multiplying the U factor by the wall area in square feet and by the temperature difference across the wall in degrees Fahrenheit, that is, by application of Equation 10-2.

Example 10-1. Determine the total quantity of heat in Btu per hour which will pass through a wall 10 ft by 20 ft, if the U factor for the wall is 0.16 Btu/hr/sq ft/° F and the temperature on one side of the wall is 40° F while the temperature on the other side is 95° F.

Solution

Total wall area	= 10 ft × 20 ft
	= 200 sq ft
Temperature differential across wall, ° F	= 95° − 40°
	= 55° F
Applying Equation 10-2, the heat gain through the wall	= 200 × 0.16 × 55
	= 1760 Btu/hr

Since the value of U in Equation 10-2 is in Btu per hour, the result obtained from Equation 10-2 is in Btu per hour. To determine the wall gain load in Btu per 24 hr as required in refrigeration load calculations, the result of Equation 10-2 is multiplied by 24 hr. Hence, for calculation cooling loads in refrigeration applications, Equation 10-2 is written to include this multiplication, viz:

$$Q = A \times U \times D \times 24 \qquad (10\text{-}3)$$

10-9. Determination of the *U* Factor. Overall coefficients of transmission or U factors have been determined for various types of wall construction and these values are available in tabular form. Tables 10-1 through 10-3 list U values for various types of cold storage walls.

Example 10-2. From Table 10-1, determine the U factor for a wall constructed of 6-in. clay tile with 4 in. of corkboard insulation.

Solution. Turn to Table 10-1 and select the appropriate type of wall construction (third from top). In the next column select the desired thickness of clay tile (6 in.) and move to the right to the column listing values for 4 in. of insulation. Read the U factor of the wall, 0.064 Btu/hr/sq ft/° F.

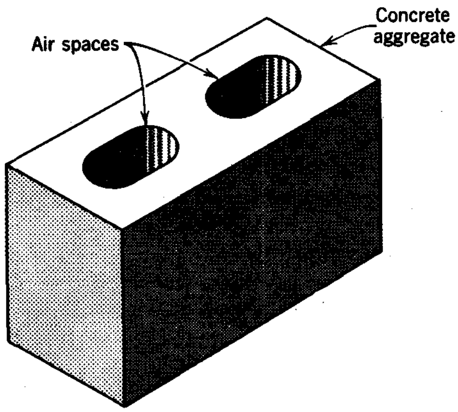


Fig. 10-1. Concrete aggregate building block.

Should it be necessary, the U factor for any type of wall construction can be readily calculated provided that either the conductivity or the conductance of each of the materials used in the wall construction is known. The conductivity or conductance of most of the materials used in wall construction can be found in tables. Too, this information is usually available from the manufacturer or producer of the material. Table 10-4 lists the thermal conductivity or the conductance of materials frequently used in the construction of cold storage walls.

The thermal conductivity or k factor of a material is the rate in Btu per hour at which heat passes through a 1 sq ft cross section of the material 1 in. thick for each 1° F of temperature difference across the material.

Whereas the thermal conductivity or k factor is available only for homogeneous materials and the value given is always for a 1 in. thickness of the material, the thermal conductance or C factor is available for both homogeneous and nonhomogeneous materials and the value given is for the specified thickness of the material.

For any homogeneous material, the thermal conductance can be determined for any given thickness of the material by dividing the k factor by the thickness in inches. Hence, for a homogeneous material,

$$C = \frac{k}{x} \qquad (10\text{-}4)$$

where x = the thickness of material in inches.

Example 10-3. Determine the thermal conductance for a 5 in. thickness of corkboard.

Solution

From Table 10-4, k factor of corkboard	= 0.30 Btu/hr/sq ft/in/° F
Applying Equation 10-4, C	$= \dfrac{0.30}{5}$
	= 0.06 Btu/hr/sq ft/° F

Since the rate of heat transmission through nonhomogeneous materials, such as the concrete building block in Fig. 10-1, will vary in the several parts of the material, the C factor from nonhomogeneous materials must be determined by experiment.

The resistance that a wall or a material offers to the flow of heat is inversely proportional to the ability of the wall or material to transmit heat. Hence, the over-all thermal resistance of a wall can be expressed as the reciprocal of the over-all coefficient of transmission, whereas the thermal resistance of an individual material can be expressed as the reciprocal of its conductivity or conductance, viz:

Over-all thermal resistance $= \dfrac{1}{U}$

Thermal resistance of an individual material $= \dfrac{1}{k}$ or $\dfrac{1}{C}$ or $\dfrac{x}{k}$

The terms $1/k$ and $1/C$ express the resistance to heat flow through a single material from surface to surface only and do not take into account the thermal resistance of the thin film of air which adheres to all exposed surfaces. In determining the over-all thermal resistance to the flow of heat through a wall from the air on one side to the air on the other side, the resistance of the air on both sides of the wall should be considered. Air film coefficients or surface conductances for average wind velocities are given in Table 10-5A.

When a wall is constructed of several layers of different materials the total thermal resistance of the wall is the sum of the resistances of the individual materials in the wall construction, including the air films, viz:

$$\frac{1}{U} = \frac{1}{f_i} + \frac{x}{k_1} + \frac{x}{k_2} + \frac{x}{k_n} + \frac{1}{f_o} \qquad (10\text{-}5)$$

Therefore

$$U = \frac{1}{\dfrac{1}{f_i} + \dfrac{x}{k_1} + \dfrac{x}{k_2} + \cdots \dfrac{x}{k_n} + \dfrac{1}{f_o}}$$

where $\dfrac{1}{f_i}$ = surface conductance of inside wall, floor, or ceiling

$\dfrac{1}{f_o}$ = surface conductance of outside wall, floor, or roof

NOTE. When nonhomogeneous materials are used, $1/C$ is substituted for x/k.

Example 10-4. Calculate the value of U for a wall constructed of 8 in. cinder aggregate building blocks, insulated with 4 in. of corkboard, and finished on the inside with 0.5 in. of cement plaster.

Solution

From Table 10-4,

8 in. cinder aggregate block $C = 0.60$

Corkboard $k = 0.30$

Cement plaster $k = 8.00$

From Table 10-5A,

inside surface conductance $f_i = 1.65$

outside surface conductance $f_o = 4.00$

Applying Equation 10-5, the over-all thermal resistance, $1/U$

$$= \frac{1}{4} + \frac{1}{0.6} + \frac{4}{0.3} + \frac{0.5}{8} + \frac{1}{1.65}$$
$$= 0.25 + 1.667 + 13.333 + 0.0625 + 0.607$$
$$= 15.92$$

Therefore, $U = 1/15.92$
$= 0.0622$ Btu/hr/sq ft/°F

For the most part, it is the insulating material used in the wall construction that determines the value of U for cold storage walls. The surface conductances and the conductances of the other materials in the wall have very little effect on the value of U because the thermal resistance of the insulating material is so large with relation to that of the air films and other materials. Therefore, for small coolers, it is sufficiently accurate to use the conductance of the insulating material alone as the wall U factor.

10-10. Temperature Differential across Cold Storage Walls. The temperature differential across cold storage walls is usually taken as the difference between the inside and outside design temperatures.

The inside design temperature is that which is to be maintained inside the refrigerated space and usually depends upon the type of product to be stored and the length of time the product is to be kept in storage. The recommended storage temperatures for various products are given in Tables 10-10 through 10-13.

The outside design temperature depends on the location of the cooler. For cold storage walls located inside a building, the outside design temperature for the cooler wall is taken as the inside temperature of the building. When cold storage walls are exposed to the outdoors, the outdoor design temperature for the region (Table 10-6) is used as the outside design temperature. The outdoor design temperatures given in Table 10-6 are average outdoor temperatures and include an allowance for normal variations in the outdoor design dry bulb temperature during a 24-hr period. These temperatures should not be used for calculating air conditioning loads.

10-11. Temperature Differential across Ceilings and Floors. When a cooler is located inside of a building and there is adequate clearance between the top of the cooler and the ceiling of the building to allow free circulation of air over the top of the cooler, the ceiling of the cooler is treated the same as an inside wall. Likewise, when the top of the cooler is exposed to the outdoors, the ceiling is treated as an outdoor wall. The same holds true for floors except when the cooler floor is laid directly on a slab on the ground. As a general rule, the ground temperature under a slab varies only slightly the year round and is always considerably less than the outdoor design dry bulb temperature for the region in summer. Ground temperatures used in determining the temperature differential across the floor of cold storage rooms are given in Table 10-6A and are based on the regional outdoor design dry bulb temperature for winter.

10-12. Effect of Solar Radiation. Whenever the walls of a refrigerator are so situated that they receive an excessive amount of heat by radiation, either from the sun or from some other hot body, the outside surface temperature of the wall will usually be considerably above the temperature of the ambient air. A familiar example of this phenomenon is the excessive surface temperature of an automobile parked in the sun. The temperature of the metal surface is much higher than that of the surrounding air. The amount by which the surface temperature exceeds the surrounding air temperature depends upon the amount of radiant energy striking the surface and upon the reflectivity of the surface. Recall (Section 2-21) that radiant energy waves are either reflected by or absorbed by any opaque material that they strike. Light-colored, smooth surfaces will tend to reflect more and absorb less radiant energy than dark, rough-textured surfaces. Hence, the surface temperature of smooth, light-colored walls will be somewhat lower than that of dark, rough-textured walls under the same conditions of solar radiation.

Since any increase in the outside surface temperature will increase the temperature differential across the wall, the temperature differential across sunlit walls must be corrected to compensate for the sun effect. Correction factors for sunlit walls are given in Table 10-7. These values are added to the normal temperature differential. For walls facing at angles to the directions listed in Table 10-7, average values can be used.

10-13. Calculating the Wall Gain Load. In determining the wall gain load, the heat gain through all the walls, including the floor and ceiling, must be taken into account. When the several walls or parts of walls are of different construction and have different *U* factors, the heat leakage through the different parts is computed separately. Walls having identical *U* factors may be considered together, provided that the temperature differential across the walls is the same. Too, where the difference in the value of *U* is slight and/or the wall area involved is small, the difference in the *U* factor can be ignored and the walls or parts of walls can be grouped together for computation.

Example 10-5. A walk-in cooler, 16 ft × 20 ft × 10 ft high is located in the southwest corner of a store building in Dallas, Texas (Fig. 10-2). The south and west walls of the cooler are adjacent to and a part of the south and west walls of the store building. The store has a 14 ft ceiling so that there is a 4 ft clearance between the top of the cooler and the ceiling of the store. The store is air conditioned and the temperature inside the store is maintained at approximately 80° F. The inside design temperature for the cooler is 35° F. Determine the wall gain load for the cooler if the walls of the cooler are of the following construction:

South and west (outside walls)	6 in. clay tile 6 in. corkboard 0.5 cement plaster finish on inside

North and east (inside walls)	1 in. board on both sides of 2 × 4 studs 3$\frac{5}{8}$ in. granulated cork
Ceiling	Same as north and east walls
Floor	4 in. corkboard laid on 5 in. slab and finished with 3 in. of concrete

Solution

Wall surface area

North wall	10 × 16 = 160 sq ft
West wall	10 × 20 = 200 sq ft
South wall	10 × 16 = 160 sq ft
East wall	10 × 20 = 200 sq ft
Ceiling	16 × 20 = 320 sq ft
Floor	16 × 20 = 320 sq ft

Wall U factors (Tables 10-1, 10-2, and 10-3)

North and east walls	0.079 Btu/hr/sq ft/° F
South and west walls	0.045
Ceiling	0.079
Floor	0.066
From Table 10-6, outside summer design dry bulb for Dallas	92° F
From Table 10-6A, design ground temperature for Dallas	70° F

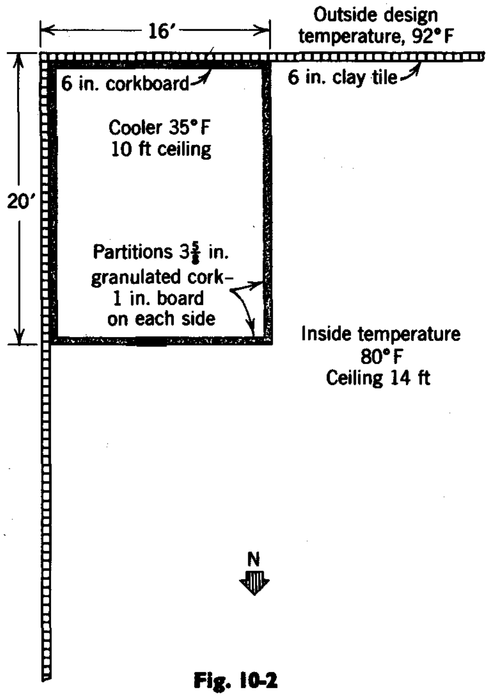


Fig. 10-2

A short method calculation may be used to determine the wall gain load for small coolers and for large coolers where the wall U factor

	Outside Design Temp.	Inside Design Temp.	Normal Wall T.D.	Correction Factor from Table 10-7	Design Wall T.D.
North wall	80° F	35° F	45° F	0	45° F
South wall	92° F	35° F	57° F	4° F	61° F
West wall	92° F	35° F	57° F	6° F	63° F
East wall	80° F	35° F	45° F	0	45° F
Ceiling	80° F	35° F	45° F	0	45° F
Floor	70° F	35° F	35° F	0	35° F

Applying Equation 10-2,

North wall	160 × 0.079 × 45 =	569 Btu/hr
West wall	200 × 0.045 × 63 =	567
South wall	160 × 0.045 × 61 =	439
East wall	200 × 0.079 × 45 =	711
Ceiling	320 × 0.079 × 45 =	1,137
Floor	320 × 0.066 × 35 =	739
Total wall gain load		4,162 Btu/hr

= 4,162 × 24 = 99,890 Btu/24 hr

and temperature difference are approximately the same for all the walls. Table 10-18 lists wall gain factors (Btu/24 hr sq ft) based on the thickness of the wall insulation and on the temperature differential across the wall. To compute the wall gain load in Btu/24 hr by the short method, multiply the total outside wall area (including floor and ceiling) by the appropriate wall gain factor from Table 10-18, viz:

Wall gain load = Outside surface area × wall gain factor

To select the appropriate wall gain factor from Table 10-18, find the thickness of the wall insulation in the extreme left-hand column of the table, move right to the column headed by the design wall temperature difference, and read the wall gain factor in Btu/24 hr/sq ft. For example, assume that the walls of a cooler are insulated with the equivalent of 4 in. of corkboard and that the temperature difference across the walls is 55° F. From Table 10-18, read the wall gain factor of 99 Btu/24 hr/sq ft (see Example 10-18).

10-14. Calculation the Air Change Load. The space heat gain resulting from air changes in the refrigerated space is difficult to determine with any real accuracy except in those few cases where a known quantity of air is introduced into the space for ventilating purposes. When the weight of outside air entering the space in a 24-hr period is known, the space heat gain resulting from air changes depends upon the difference in the enthalpy of the air at the inside and outside conditions and can be calculated by applying the following equation:

$$\text{Air change load} = W(h_o - h_i) \quad (10\text{-}6)$$

where W = weight of air entering space in 24 hr (lb/24 hr)
h_o = enthalpy of outside air (Btu/lb)
h_i = enthalpy of inside air (Btu/lb)

However, since air quantities are usually given in cubic feet rather than in pounds, to facilitate calculations the heat gain per cubic foot of outside air entering the space is listed in Tables 10-8A and 10-8B for various inside and outside air conditions. To determine the air change load in Btu per 24 hr, multiply the air quantity in cubic feet per 24 hr by the appropriate factor from Table 10-8A or 10-8B.

Where the ventilating air (air change) quantity is given in cubic feet per minute (cfm), convert cfm to cubic feet per 24 hr by multiplying by 60 min and by 24 hr.

Example 10-6. Three hundred cfm of air are introduced into a refrigerated space for ventilation. If the inside of the cooler is maintained at 35° F and the outside dry bulb temperature and humidity are 85° F and 50%, respectively, determine the air change load in Btu/24 hr.

Solution

Cubic feet of air per per 24 hr = cfm × 60 × 24
= 300 × 60 × 24
= 432,000 cu ft/24 hr

From Table 10-8,A heat gain per cubic feet = 1.86 Btu/cu ft

Ventilating (air change) load = cu ft/24 hr × Btu/cu ft
= 432,000 × 1.86
= 803,520 Btu/24 hr

Except in those few cases where air is purposely introduced into the refrigerated space for ventilation, the air changes occurring in the space are brought about solely by infiltration through door openings, The quantity of outside air entering a space through door openings in a 24-hr period depends upon the number, size, and location of the door or doors, and upon the frequency and duration of the door openings. Since the combined effect of all these factors varies with the individual installation and is difficult to predict with reasonable accuracy, it is general practice to estimate the air change quantity on the basis of experience with similar applications. Experience has shown that, as a general rule, the frequency and duration of door openings and, hence, the air change quantity, depend on the inside volume of the cooler and the type of usage. Tables 10-9A and 10-9B list the approximate number of air changes per 24 hr for various cooler sizes. The values given are for average usage (see table footnotes). The ASRE *Data Book* defines average and heavy usage as follows:

Average usage includes installations not subject to extreme temperatures and where the quantity of food handled in the refrigerator is not abnormal. Refrigerators in delicatessens and clubs may generally be classified under this type of usage.

Heavy usage includes installations such as those in busy markets, restaurant and hotel kitchens where the room temperatures are likely to be high, where rush periods place heavy loads on the refrigerator, and where large quantities of warm foods are often placed in it.*

* The *Refrigerating Data Book*, Basic Volume, The American Society of Refrigerating Engineers, 1949, New York, p. 327.

Example 10-7. A walk-in cooler 8 ft × 15 ft × 10 ft high is constructed of 4 in. of corkboard with 1 in. of wood on each side. The outside temperature is 95° F and the humidity is 50%. The cooler is maintained at 35° F and the usage is average. Determine the air change load in Btu/24 hr.

Solution. Since the walls of the cooler are approximately 6 in. thick (4 in. of corkboard and 2 in. of wood), the inside dimensions of the cooler are 1 ft less than the outside dimensions; therefore,

Inside volume	= 7 ft × 14 ft × 9 ft = 882 cu ft
From Table 10-9A, by interpolation, number of air changes per 24 hr for cooler volume of approximately 900 cu ft	= 19
Total quantity of air change per 24 hr	= Inside volume × air changes = 882 × 19 = 16,758 cu ft/24 hr
From Table 10-8A, heat gain per cubic feet	= 2.49 Btu/cu ft
Air change load	= cu ft/24 hr × Btu/cu ft = 16,758 × 2.49 = 41,727 Btu/24 hr

10-15. Calculation the Product Load. When a product enters a storage space at a temperature above the temperature of the space, the product will give off heat to the space until it cools to the space temperature. When the temperature of the storage space is maintained above the freezing temperature of the product, the amount of heat given off by the product in cooling to the space temperature depends upon the temperature of the space and upon the weight, specific heat, and entering temperature of the product. In such cases, the space heat gain from the product is computed by the following equation, (see Section 2-24):

$$Q = W \times C \times (T_2 - T_1) \qquad (10\text{-}7)$$

where Q = the quantity of heat in Btu
W = weight of the product (pounds)
C = the specific heat above freezing (Btu/lb/° F)
T_1 = the entering temperature (° F)
T_2 = the space temperature (° F)

Example 10-8. One thousand pounds of fresh, lean beef enter a cooler at 55° F and are chilled to the cooler temperature of 35° F in 24 hr. Calculate the product load in Btu/24 hr.

Solution

From Table 10-12, the specific heat of lean beef above freezing	= 0.75 Btu/lb/° F
Applying Equation 10-7, the product load, Btu/24 hr	= 1000 × 0.75 × (55 − 35) = 1000 × 0.75 × 20 = 15,000 Btu/24 hr

Notice that no time element is inherent in Equation 10-7 and that the result obtained is merely the quantity of heat the product will give off in cooling to the space temperature. However, since in Example 10-8 the product is to be cooled over a 24-hr period, the resulting heat quantity represents the product load for a 24-hr period. When the desired cooling time is less than 24 hr, the equivalent product load for a 24-hr period is computed by dividing the heat quantity by the desired cooling time for the product to obtain the hourly cooling rate and then multiplying the result by 24 hr to determine the equivalent product load for a 24-hr period. When adjusted to include these two factors, Equation 10-7 is written:

$$Q = \frac{W \times C \times (T_2 - T_1) \times 24 \text{ hr}}{\text{desired cooling time (hr)}} \qquad (10\text{-}8)$$

Example 10-9. Assume that it is desired to chill the beef in Example 10-8 in 6 hr rather than in 24 hr. Determine the product load in Btu/24 hr.

Solution. Applying Equation 10-8, product load, Btu/24 hr	$= \dfrac{1000 \times 0.75 \times (55 - 35) \times 24}{6}$ = 60,000 Btu/24 hr

Compare this result with that obtained in Example 10-8.

When a product is chilled and stored below its freezing temperature, the product load is calculated in three parts:

1. The heat given off by the product in cooling from the entering temperature to its freezing temperature.

2. The heat given off by the product in solidifying or freezing.
3. The heat given off by the product in cooling from its freezing temperature to the final storage temperature.

For parts 1 and 3, Equation 10-7 is used to determine the heat quantity. For part 1, T_1 in Equation 10-7 is the entering temperature of the product, whereas T_2 is the freezing temperature of the product (Tables 10-10 through 10-13). For part 3, T_1 in Equation 10-7 is the freezing temperature of the product and T_2 is the final storage temperature. The heat quantity for part 2 is determined by the following equation:

$$Q = W \times h_{if} \qquad (10\text{-}9)$$

where W = the weight of the product (pounds)
h_{if} = the latent heat of the product (Btu/lb)

When the chilling and freezing are accomplished over a 24-hr period, the summation of the three parts represents the product load for 24 hr. When the desired chilling and freezing time for the product are less than 24 hr, the summation of the three parts is divided by the desired processing time and then multiplied by 24 hr to determine the equivalent 24-hr product load.

Example 10-10. 500 pounds of poultry enter a chiller at 40° F and are frozen and chilled to a final temperature of −5° F for storage in 12 hr. Compute the product load in Btu/24 hr.

Solution

From Table 10-12,

Specific heat above freezing	= 0.79 Btu/lb/° F
Specific heat below freezing	= 0.37 Btu/lb/° F
Latent heat	= 106 Btu/lb
Freezing temperature	= 27° F
To cool poultry from entering temperature to freezing temperature, applying Equation 10-7	$= 500 \times 0.79 \times (40 - 27)$ = 5135 Btu
To freeze, applying Equation 10-9	$= 500 \times 106$ = 53,000 Btu
To cool from freezing temperature to final storage temperature, applying Equation 10-7	$= 500 \times 0.37 \times [27 - (-5)]$ = 5920 Btu
Total heat given up by product (summation of 1, 2, and 3)	= 64,000 Btu
Equivalent product load for 24-hr period Btu/24 hr	$= \dfrac{64{,}000 \times 24 \text{ hr}}{12 \text{ hr}}$ = 128,000 Btu/24 hr

10-16. Chilling Rate Factor. During the early part of the chilling period, the Btu per hour load on the equipment is considerably greater than the average hourly product load as calculated in the previous examples. Because of the high temperature difference which exists between the product and the space air at the start of the chilling period, the chilling rate is higher and the product load tends to concentrate in the early part of the chilling period (Section 9-23). Therefore, where the equipment selection is based on the assumption that the product load is evenly distributed over the entire chilling period, the equipment selected will usually have insufficient capacity to carry the load during the initial stages of chilling when the product load is at a peak.

To compensate for the uneven distribution of the chilling load, a chilling rate factor is introduced into the chilling load calculation. The effect of the chilling rate factor is to increase the product load calculation by an amount sufficient to make the average hourly cooling rate approximately equal to the hourly load at the peak condition. This results in the selection of larger equipment, having sufficient capacity to carry the load during the initial stages of chilling.

Chilling rate factors for various products are listed in Tables 10-10 through 10-13. The factors given in the tables are based on actual tests and on calculations and will vary with the ratio of the loading time to total chilling time. As an example, test results show that in typical beef and hog chilling operations the chilling rate is 50% greater during the first half of the chilling period than the average chilling rate for the entire period. The calculation without the chilling rate factor will, of course, show the average chilling rate for the entire period. To obtain this rate during the initial chilling period, it must be multiplied by 1.5. For convenience, the chilling rate factors are given in the tables in reciprocal form and are used in the denominator of the equation. Thus the chilling rate factor for beef as shown in the table is 0.67 (1/1.5).

Where a chilling rate factor is used, Equation 10-7 is written

$$Q = \frac{W \times C \times (T_2 - T_1)}{\text{Chilling rate factor}} \qquad (10\text{-}10)$$

As a general rule chilling rate factors are not used for the final stages of chilling from the freezing temperature to the final storage temperature of the product. Too, chilling rate factors are usually applied to chilling rooms only and are not normally used in calculation of the product load for storage rooms. Since the product load for storage rooms usually represents only a small percentage of the total load, the uneven distribution of the product load over the cooling period will not ordinarily cause overloading of the equipment and, therefore, no allowance need be made for this condition.

10-17. Respiration Heat. Fruits and vegetables are still alive after harvesting and continue to undergo changes while in storage. The more important of these changes are produced by respiration, a process during which oxygen from the air combines with the carbohydrates in the plant tissue and results in the release of carbon dioxide and heat. The heat released is called respiration heat and must be considered as a part of the product load where considerable quantities of fruit and/or vegetables are held in storage. The amount of heat evolving from the respiration process depends upon the type and temperature of the product. Respiration heat for various fruits and vegetables is listed in Table 10-14.

Since respiration heat is given in Btu per pound per hr, the product load accruing from respiration heat is computed by multiplying the total weight of the product by the respiration heat as given in Table 10-14, viz:

$$Q\ (\text{Btu/24 hr}) = \text{Weight of product (lb)} \times \text{respiration heat (Btu/lb/hr)} \times 24\ \text{hr} \qquad (10\text{-}11)$$

10-18. Containers and Packing Materials. When a product is chilled in containers, such as milk in bottles or cartons, eggs in crates, fruit and vegetables in baskets and lugs, etc., the heat given off by the containers and packing materials in cooling from the entering temperature to the space temperature must be considered as a part of the product load. Equation 10-7 is used to compute this heat quantity.

10-19. Calculating the Miscellaneous Load. the miscellaneous load consists primarily of the heat given off by lights and electric motors operating in the space and by people working in the space. The heat given off by lights is 3.42 Btu per watt per hour. The heat given off by electric motors and by people working in the space is listed in Tables 10-15 and 10-16, respectively. The following calculations are made to determine the heat gain from miscellaneous:

Lights: wattage × 3.42 Btu/watt/hr × 24 hr

Electric motors: factor (Table 10-15) × horsepower × 24 hr

People: factor (Table 10-16) × number of people × 24 hr

10-20. Use of Safety Factor. The total cooling load for a 24-hr period is the summation of the heat gains as calculated in the foregoing sections. It is common practice to add 5% to 10% to this value as a safety factor. The percentage used depends upon the reliability of the information used in calculating the cooling load. As a general rule 10% is used.

After the safety factor has been added, the 24-hr load is divided by the desired operating time for the equipment to determine the average load in Btu per hour (see Section 10-2). The average hourly load is used as a basis for equipment selection.

10-21. Short Method Load Calculations. Whenever possible the cooling load should be determined by using the procedures set forth in the preceding sections of this chapter. However, when small coolers (under 1600 cu ft) are used for general-purpose storage, the product load is frequently unknown and/or varies somewhat from day to day so that it is not possible to compute the product load with any real accuracy. In such cases, a short method of load calculation can be employed which involves the use of load factors which have been determined by experience. When the short method of calculation is employed, the entire cooling load is divided into two parts: (1) the wall gain load and (2) the usage or service load.

The wall gain load is calculated as outlined in Section 10-13. The usage load is computed by the following equation:

Usage load = interior volume × usage factor (10-12)

Notice that the usage factors listed in Table 10-17 vary with the interior volume of the cooler and with the difference in temperature between the inside and outside of the cooler. Too, an allowance is made for normal and heavy usage. Normal and heavy usage have already been defined in Section 10-14. No safety factor is used when the cooling load is calculated by the short method. The total cooling load is divided by the desired operating time for the equipment to find the average hourly load used to select the equipment.

Example 10-11. A walk-in cooler 18 ft × 10 ft × 10 ft high has 4 in. of corkboard insulation and standard wall construction consisting of two layers of paper and 1 in. of wood on each side (total wall thickness is 6 in.). The temperature outside the cooler is 85° F. 3500 lb of mixed vegetables are cooled 10° F to the storage temperature each day. Compute the cooling load in Btu/hr based on a 16-hr per day operating time for the equipment. The inside temperature is 40° F.

Solution

Outside surface area = 2 × 18 ft × 10 ft
= 360 sq ft
= 4 × 10 ft × 10 ft
= 400 sq ft
760 sq ft

Inside volume (since total wall thickness is 6 in., the inside dimensions are 1 ft less than the outside dimensions) = 17 ft × 9 ft × 9 ft
= 1377 cu ft

Wall gain factor (Table 10-18) (45° F TD and 4 in. insulation) = 81 Btu/hr/sq ft

Air changes (Table 10-9A) (by interpolation) = 16.8 per 24 hr

Heat gain per cubic foot of air (Table 10-8A) (50% RH) = 1.69 Btu/cu ft

Average specific heat of vegetables (Table 10-11) = 0.9 Btu/lb/° F

Average respiration heat of vegetables (Table 10-14) = 0.09 Btu/lb/hr

Wall gain load:
Area × wall gain factor
= 760 sq ft × 81 Btu/sq ft/24 hr
= 61,560 Btu/24 hr

Air change load:
Inside volume
× air changes × Btu/cu ft
= 1377 cu ft × 16.8
× 1.69 Btu/cuft = 23,100 Btu/24 hr

Product load:
Temperature reduction
$= M \times C \times (T_2 - T_1)$
= 3500 lb × 0.9 Btu/lb/° F
× 10° F = 31,500 Btu/24 hr

Respiration heat
$= M$ × reaction heat × 24 hr
= 3500 × 0.09 Btu/lb/hr
× 24 hr
= 7,560 Btu/24 hr

Summation: 123,720 Btu/24 hr

Safety factor (10%) = 12,370 Btu

Total cooling load = 136,100 Btu/24 hr

Required cooling capacity (Btu/hr) $= \dfrac{\text{Total cooling load}}{\text{Desired running time}}$

$= \dfrac{136{,}100 \text{ Btu/24 hr}}{16 \text{ hr}}$

= 8,500 Btu/hr

Example 10-12. The dimensions of a banana storage room located in New Orleans, Louisiana are 22 ft × 32 ft × 9 ft. The walls are 1 in. boards on both sides of 2 × 4 studs and insulated with 3⅝ in. granulated cork. The floor and roof are of the same construction as the walls. The floor is over a ventilated crawl space and the roof is exposed to the sun. The temperature outside the storage room is approximately the same as the outdoor design temperature for the region. 30,000 lb of bananas are ripened at 70° F and then cooled to 56° F in 12 hr for holding storage. Compute the Btu/hr cooling load. (NOTE: Because of the high storage temperature the evaporator will not collect frost and the equipment is designed for continuous run.)

Solution

Outside surface area	
Ceiling	= 704 sq ft
Walls and floor	= 1676 sq ft
Inside volume (21 ft × 31 ft × 8 ft)	= 5208 cu ft
Outside design temperature for New Orleans (Table 10-6)	= 89° F
Wall U factor (Table 10-2)	= 0.079 Btu/hr/sq ft/° F
Sun factor for roof (Table 10-7) (tar roofing)	= 20° F
Air changes (Table 10-9A) (by interpolation)	= 7/24 hr
Heat gain per cubic foot of air change (Table 10-8A) (interpolated)	= 1.4 Btu/cu ft
Specific heat of bananas (Table 10-10)	= 0.9 Btu/lb/° F
Reaction heat of bananas (Table 10-14)	= 0.5 Btu/lb/hr

Wall gain load:

Area × U × TD × 24 hr

Ceiling

704 × 0.079 × 53 × 24 = 70,740 Btu/24 hr

Walls and floor

1676 × 0.79 × 33 × 24 = 104,860 Btu/24 hr

Air change load:

Inside volume × air changes × Btu/cu ft = 5208 × 7 × 1.4 Btu/cu ft = 51,000 Btu/24 hr

Product load:

Temperature reduction

$$= \frac{M \times C \times (T_2 - T_1) \times 24\,\text{hr}}{\text{Chilling time (hr)}}$$

$$= \frac{30{,}000 \times 0.9 \times 14 \times 24}{12}$$

= 756,000 Btu/24 hr

Respiration heat

= M × reaction heat × 24 hr

= 30,000 × 0.5 × 24 = 360,000 Btu/24 hr

Summation:	1,342,600 Btu/24 hr
Safety factor (10%)	= 134,260 Btu
Total cooling load	1,476,860 Btu/24 hr

Required cooling capacity (Btu/hr)

$$= \frac{\text{Total cooling load}}{\text{Desired running time}}$$

$$= \frac{1{,}476{,}860\ \text{Btu/24 hr}}{24\ \text{hr}}$$

= 61,530 Btu/hr

Example 10-13. A chilling room 35 ft × 50 ft × 15 ft is used to chill 50,000 lb of fresh beef per day from an initial temperature of 100° F to a final temperature of 35° F in 18 hr. Four people work in the chiller during the loading period. The lighting load is 1500 watts. The floor, located over an unconditioned space, is a 5 in. concrete slab insulated with 4 in. of corkboard and finished with 3 in. of concrete. The ceiling, situated beneath an unconditioned space, is a 4 in. concrete slab with wood sleepers and insulated with 4 in. of corkboard. All of the walls are inside partitions adjacent to unconditioned spaces (90° F) except the east wall which is adjacent to a 35° F storage cooler. Wall construction is 4 in. cinder block insulated with 4 in. of corkboard and finished on one side with plaster. Compute the cooling load in Btu per hour based on a 16-hr operating time for the equipment.

Solution

Outside surface area	
Ceiling (35 ft × 50 ft)	= 1,750 sq ft
Floor (35 ft × 50 ft)	= 1,750 sq ft
Walls (except east) (120 ft × 15 ft)	= 1,800 sq ft
Inside volume (34 ft × 49 ft × 13.5 ft)	= 22,491 cu ft
Air changes (Table 10-9A) (by interpolation)	= 3.2 per 24 hr
Heat gain per cubic foot (Table 10-8A) (50% RH)	= 2.17 Btu/cu ft
Specific heat of beef (Table 10-12)	= 0.75 Btu/lb/° F
Chilling rate factor (Table 10-12)	= 0.67
Occupancy heat gain (Table 10-16)	= 900 Btu/hr/person
Ceiling U factor (Table 10-3)	= 0.069 Btu/hr/sq ft/° F

Floor U factor
(Table 10-3) $= 0.066$ Btu/hr/sq ft/°F
Wall U factor
(Table 10-1) $= 0.065$ Btu/hr/sq ft/°F

Wall gain load:
$A \times U \times TD \times 24$ hr
$= 5300 \times 0.067 \times 55 \times 24$ hr
$= 468{,}700$ Btu/24 hr

Air change load:
Inside volume
$\times$ air changes $\times$ Btu/cu ft
$= 22{,}491 \times 3.2 \times 2.17$
$= 156{,}000$ Btu/24 hr

Product load:

$$\frac{M \times C \times (T_2 - T_1) \times 24}{\text{Chilling time (hr)} \times \text{chilling rate factor}}$$

$$= \frac{50{,}000 \times 0.75 \times 65 \times 24}{18 \times 0.67}$$

$= 4{,}850{,}700$ Btu/24 hr

Miscellaneous:
Occupancy
$=$ No. of people $\times$ factor $\times$ 24 hr
$= 4 \times 900 \times 24$
$= 86{,}400$ Btu/24 hr

Lighting load
$=$ watts $\times 3.4 \times 24$ hr
$1500 \times 3.4 \times 24 =$ 122,400 Btu/24 hr

Summation: 5,684,200 Btu/24 hr
Safety factor (10%)
$=$ 568,420 Btu
6,252,620 Btu/24 hr

Required cooling capacity (Btu/hr)

$$= \frac{\text{Total cooling load}}{\text{Desired operating time}}$$

$$= \frac{6{,}252{,}620 \text{ Btu/24 hr}}{18 \text{ hr}}$$

$= 390{,}800$ Btu/hr

NOTE: (1) Since there is no temperature differential across the east wall, there is no gain or loss of heat through the wall and the wall is ignored in the cooling load calculations However, this wall should be insulated with at least the minimum amount of insulation to prevent excessive heat gains through this wall in the event that the adjacent refrigerated space should become inoperative. (2) Since the TD across all the walls, including floor and ceiling, is the same and since the difference in the wall U factors is slight, the walls may be lumped together for calculation. (3) Although the workmen are in the space for only 4 hr each day for the purpose of load calculation, they are assumed to be in the cooler continuously. This is because their occupancy occurs simultaneously with the chilling peak. If the occupancy occurred at any time other than at the peak, the occupancy load could be ignored.

Example 10-14. Three thousand lug boxes of apples are stored at 35° F in a storage cooler 50 ft $\times$ 40 ft $\times$ 10 ft. The apples enter the cooler at a temperature of 90° F and at the rate of 200 lugs per day each day for the 15 day harvesting period. The walls including floor and ceiling are constructed of 1 in. boards on both sides of 2×4 studs and are insulated with $3\frac{5}{8}$ in. of rock wool. All of the walls are shaded and the ambient temperature is 85° F. The average weight of apples per lug box is 59 lb. The lug boxes have an average weight of 4.5 lb and a specific heat value of 0.60 Btu/lb/° F. Determine the average hourly cooling load based on 16 hr operating time for the equipment.

Solution

Outside surface
area $= 5800$ sq ft
Inside volume
(49 ft $\times$ 39 ft $\times$ 9 ft) $= 17{,}200$ cu ft
Wall U factor
(Table 10-2) $= 0.072$ Btu/hr/sq ft/° F
Air changes
(Table 10-9A) (by
interpolation) $= 3.7$ per 24 hr
Heat gain per
cubic foot (Table
10-8A) $= 1.86$ Btu/cu ft
Specific heat of
apples (Table 10-10) $= 0.89$ Btu/lb/° F
Respiration heat
of apples (Table
10-14) (by interpo-
lation) $= 0.025$ Btu/lb/hr

Wall gain load:
$A \times U \times TD \times 24$ hr
$= 5800 \times 0.072 \times 50 \times 24$
$= 501{,}100$ Btu/24 hr

Air change load:
Inside volume $\times$ air changes
$\times$ Btu/cu ft
$= 17{,}000 \times 3.7 \times 1.86$
$= 117{,}000$ Btu/24 hr

Product load:
Temperature reduction

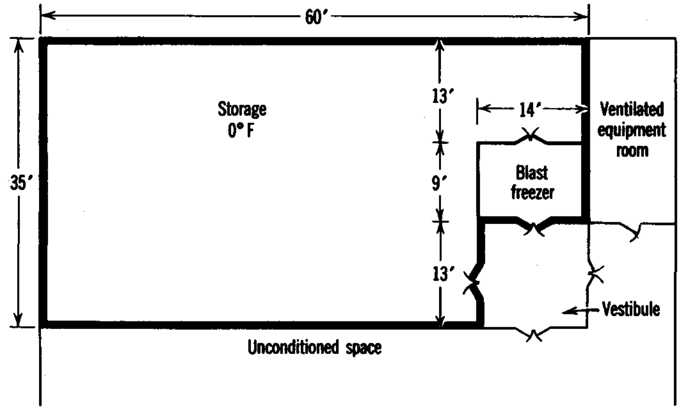
$$= \frac{M \times C \times (T_2 - T_1)}{\text{Chilling rate factor}}$$

Apples

$$\frac{(200 \times 59 \text{ lb}) \times 0.89 \times 55}{0.67}$$

$= 862{,}100$ Btu/24 hr

Fig. 10-3



Lug boxes

$$\frac{(200 \times 4.5 \text{ lb}) \times 0.6 \times 55}{0.67} = 44{,}300 \text{ Btu/24 hr}$$

Respiration

$= M \times$ reaction heat $\times$ 24 hr

$= (3000 \times 59 \text{ lb}) \times 0.025 \times 24$

	=	106,200 Btu/24 hr
Summation		1,630,700 Btu/24 hr
Safety factor (10%)	=	163,100 Btu
Total cooling load	=	1,793,800 Btu/24 hr

$$\text{Average hourly load} = \frac{\text{Total cooling load}}{\text{Running time}} = \frac{1{,}793{,}800 \text{ Btu/24 hr}}{16 \text{ hr}} = 112{,}100 \text{ Btu/24 hr}$$

Note: Load calculation and equipment selection is based on maximum loading which occurs on the fifteenth day.

Example 10-15. Twenty-two thousand pounds of dressed poultry are blast frozen on hand trucks each day (24 hr) in a freezing tunnel 14 ft × 9 ft × 10 ft high (see Fig. 10-3). The poultry is precooled to 45° F before entering the freezer where it is frozen and its temperature lowered to 0° F for storage. The lighting load is 200 watts. The hand trucks carrying the poultry total 1400 lb per day and have a specific heat of 0.25 Btu/lb/° F. The partitions adjacent to the equipment room and vestibule are constructed of 6 in. clay tile insulated with 8 in. of corkboard. Partitions adjacent to storage cooler are 4 in. clay tile with 2 in. corkboard insulation. The roof is a 6 in. concrete slab insulated with 8 in. of corkboard and covered with tar, felt, and gravel. The floor is a 6 in. concrete slab insulated with 8 in. of corkboard and finished with 4 in. of concrete. The floor is over a ventilated crawl space. Roof is exposed to the sun. The equipment room is well ventilated so that the temperature inside is approximately the outdoor design temperature for the region. The storage room is maintained at 0° F, whereas the temperature in the freezer is −10° F. The location is Houston, Texas. Determine the average hourly refrigeration load based on 20 hr per day operating time for the equipment.

Solution

Outside surface area	
Roof	
(9 ft + 14 ft)	= 126 sq ft
Floor	
(9 ft × 14 ft)	= 126 sq ft
N and E partitions	
(23 ft × 10 ft)	= 230 sq ft
S and W partitions	
(23 ft × 10 ft)	= 230 sq ft
Inside volume	
(8 ft × 9 ft × 13 ft)	= 936 cu ft
Summer outdoor design temperature	= 92° F
U factors	
Roof (Table 10-3)	= 0.036 Btu/hr/sq ft/° F
Floor (Table 10-3)	= 0.035 Btu/hr/sq ft/° F
N and E partitions (Table 10-2)	= 0.035 Btu/hr/sq ft/° F
S and W partitions (Table 10-2)	= 0.12 Btu/hr/sq ft/° F
Roof sun factor (Table 10-7)	= 20° F
Air changes (Table 10-9B)	= 13.5 per 24 hr
Heat gain per cubic foot (Table 10-8B)	= 3.56 Btu/cu ft

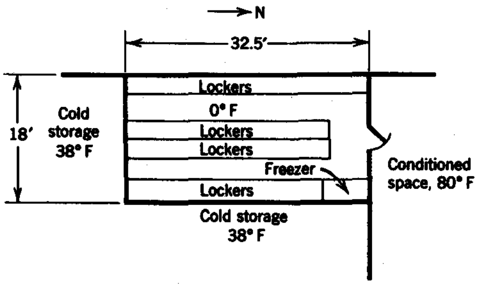


Fig. 10-4. Frozen food locker plant.

Specific heat of poultry
(Table 10-12)
Above freezing
= 0.79 Btu/lb/° F
Below freezing
= 0.37 Btu/lb/° F
Latent heat of poultry
(Table 10-12)
= 106 Btu/lb
Freezing temperature
27° F

Wall gain load:
$A \times U \times TD \times 24$ hrs
Floor
126 × 0.035 × 102 × 24
= 10,800 Btu/24 hr
Roof
126 × 0.036 × (102 + 20) × 24
= 13,300 Btu/24 hr
South and west partition
230 × 0.035 × 102 × 24
= 19,700 Btu/24 hr
North and east partition
230 × 0.12 × 10 × 24
= 6,600 Btu/24 hr

Air change load:
Inside volume × air changes
× Btu/cu ft
= 936 × 13.5 × 3.56
= 45,000 Btu/24 hr

Product Load:
Temperature reduction
$= M \times C \times (T_2 - T_1)$
Poultry
$$= \frac{22{,}000 \times 0.79 \times (45 - 27)}{0.67 \text{ (chilling factor)}}$$
= 302,700 Btu/24 hr
22,000 × 0.37 × (27 − 0)
= 219,700 Btu/24 hr
Trucks
$$= \frac{1{,}400 \times 0.25 \times (92 - 0)}{0.67}$$
= 48,000 Btu/24 hr

Freezing = M × latent heat
Poultry = 22,000 × 106
= 2,332,000 Btu/24 hr

Miscellaneous:
Lighting: 200 watts × 3.4
Btu/watt/hr × 24 hr
= 16,300 Btu/24 hr

Summation: 3,014,100 Btu/24 hr

Safety factor (10%) = 301,400 Btu

Total cooling load = 3,315,500 Btu/24 hr

Average hourly load
$$= \frac{3{,}315{,}500 \text{ Btu/24 hr}}{20 \text{ hr (running time)}}$$
= 165,775 Btu/hr

Example 10-16. A frozen food locker plant 18 ft × 32.5 ft × 10 ft, containing 353 individual lockers and an 8 ft freezing cabinet, is located in Tulsa, Oklahoma (see Fig. 10-4). The north and west wall are constructed of 8 in. clay tile with 6 in. of corkboard insulation. South and east walls are 4 in. clay tile with 4 in. of corkboard insulation. The roof is exposed to the sun and is constructed of 4 in. of concrete insulated with 8 in. of corkboard and covered with tar, felt, and gravel. The floor is a 5 in. concrete slab poured directly on the ground, insulated with 8 in. of corkboard, and finished with 3 in. of concrete. The product load on cabinet freezer is 700 lb of assorted meats per day. (Standard practice is to allow for 2 lb of product per locker per day.) In this instance, the product is precooled to 38° F before being placed in the freezer. The lighting load is 500 watts and the average occupancy is three people. Determine the average hourly refrigerating rate based on a 20-hr equipment operating time.

Solution
Outside surface area

Roof	(18 ft × 32.5 ft) = 585 sq ft
Floor	(18 ft × 32.5 ft) = 585 sq ft
South and east partition	(50.5 ft × 10 ft) = 505 sq ft
North partition	(18 ft × 10 ft) = 180 sq ft
West wall	(32.5 ft × 10 ft) = 325 sq ft

Inside volume
(16 ft × 30.5 ft × 8 ft)
= 3904 cu ft

Design outdoor temperature
(Table 10-6) = 92° F

Design ground temperature
(Table 10-6A) = 65° F

Roof sun factor
(Table 10-7) = 20° F

West wall sun factor
(Table 10-7) = 6° F

U factors
Roof (Table 10-3)
= 0.036 Btu/hr/sq ft/° F
Floor (Table 10-3)
= 0.046 Btu/hr/sq ft/° F
North and west walls
(Table 10-1) = 0.034 Btu/hr/sq ft/° F
South and east walls
(Table 10-1) = 0.066 Btu/hr/sq ft/° F

Air changes
(see Note 2 of Table 10-9B)
= 12.6(6.3 × 2)
per 24 hr

Heat gain per cubic foot
(Table 10-8B) = 3.0 Btu/cu ft

Specific heat (average for meat)
Above freezing = 0.8 Btu/lb/° F
Below freezing = 0.4 Btu/lb/° F
Latent heat
(average) = 100 Btu/lb
Freezing temperature (average) = 28° F
Occupancy factor
= 1300 Btu/hr/person

Wall Gain Load:
$A \times U \times TD \times 24$
Roof 585 × 0.036 × (92 + 20) × 24
= 56,600 Btu/24 hr
Floor 585 × 0.046 × 65 × 24
= 42,000 Btu/24 hr
South and east partition 505 × 0.066 × 38 × 24
= 30,400 Btu/24 hr
North partition 180 × 0.034 × 80 × 24
= 11,750 Btu/24 hr
West wall 325 × 0.034 × (92 + 6) × 24
= 26,000 Btu/24 hr

Air change load:
Inside volume × air changes × Btu/cu ft
= 3904 × 12.6 × 3.01
= 147,570 Btu/24 hr

Product load:
Temperature reduction
$= M \times C \times (T_2 - T_1)$
700 × 0.8 × (38 − 28)
= 5600 Btu/24 hr
700 × 0.4 × (28 − 0)
7840 Btu/24 hr
Freezing
= *M* × latent heat
= 700 × 100 70,000 Btu/24 hr

Miscellaneous:
Lights
= 500 watts × 3.4 Btu/hr
× 24 hr = 40,800 Btu/24 hr
Occupancy
= 3 × 1300 × 24
= 93,600 Btu/24 hr

Summation: = 531,560 Btu/24 hr
Safety factor
(10%) = 53,150 Btu
Total cooling
load = 584,710 Btu/24 hr
Average hourly
load $= \dfrac{584{,}710 \text{ Btu/24 hr}}{20 \text{ hr}}$
= 29,240 Btu/hr
Load on freezer $= \dfrac{91{,}780 \text{ Btu/24 hr}}{20 \text{ hr}}$
(product load only, including 10% safety factor)
= 4,590 Btu/hr
Load on locker room only = 29,240 Btu/hr
(total load less − 4,590 Btu/hr
freezer load) = 24,650 Btu/hr

Example 10-17. Five hundred gallons of partially frozen ice cream at 25° F are entering a hardening room 10 ft × 15 ft × 9 ft each day. Hardening is completed and the temperature of the ice cream is lowered to −20° F in 10 hr. The walls, including floor and ceiling, are insulated with 8 in. of corkboard and the overall thickness of the walls is 12 in. The ambient temperature is 90° F and the lighting load is 300 watts. Assume the average weight of ice cream is 5 lb per gallon, the average specific heat below freezing is 0.5 Btu/lb/° F, and the average latent heat per pound is 100 Btu.* Determine the average hourly load based on 18 hr operation.

* These values are variable and depend upon the composition of the mix, the percent of overrun, and the temperature of the ice cream leaving the freezer.

Solution

Outside surface area = 750 cu ft

Inside volume (8 ft × 13 ft × 7 ft) = 728 cu ft

Wall gain factor (Table 10-18) = 99 Btu/sq ft/24 hr

Air changes (Table 10-9B) (interpolated) = 16.7 per 24 hr

Heat gain per cubic foot (Table 10-8B) (50% RH) = 3.88 Btu/cu ft

Wall gain load:
area × wall gain factor
= 750 × 99 = 74,250 Btu/24 hr

Air change load:
Inside volume × air changes × Btu/cu ft
= 728 × 16.7 × 3.88 = 47,170 Btu/24 hr

Product load:

Temperature reduction

$$= \frac{M \times C(T_2 - T_1) \times 24 \text{ hr}}{\text{Chilling time}}$$

$$= \frac{(500 \times 5) \times 0.5 \times (25 - -20) \times 24}{10}$$

= 135,000 Btu/24 hr

Freezing

$$= \frac{M \times \text{latent heat} \times 24}{\text{Freezing time}}$$

$$= \frac{(500 \times 5) \times 100 \times 24}{10}$$

= 600,000 Btu/24 hr

Miscellaneous load:
Lighting:
300 watts × 3.4 × 24 = 24,480 Btu/24 hr

Summation: 880,900 Btu/24 hr
Safety factor (10%) = 88,090 Btu

968,990 Btu/24 hr

Average hourly load (968,990/18 hr) = 53,800 Btu/hr

Example 10-18. A cooler 10 ft × 12 ft × 9 ft is used for general purpose storage in a grocery store. The cooler is maintained at 35° F and the service load is normal. The walls are insulated with the equivalent of 4 in. of corkboard and the ambient temperature is 80° F. Determine the cooling load in Btu per hour based on a 16 hr operating time.

Solution

Outside surface area = 636 sq ft

Inside volume (9 ft × 11 ft × 8 ft) = 792 cu ft

Wall gain factor (Table 10-18) = 81 Btu/sq ft/24 hr

Usage factor (Table 10-17) = 50 Btu/cu ft/24 hr

Wall gain load:
area × wall gain factor
= 636 × 81 = 51,500 Btu/24 hr

Usage load:
Inside volume × usage factor
= 792 × 50 = 39,600 Btu/24 hr

Total cooling load = 91,100 Btu/24 hr

Average hourly load

$$\left(\frac{91{,}100 \text{ Btu/24 hr}}{16 \text{ hr}}\right) = 5{,}700 \text{ Btu/hr}$$

PROBLEMS

1. A cooler wall 10 ft by 18 ft is insulated with the equivalent of 3 in. of corkboard. Compute the heat gain through the wall in Btu/24 hr if the inside temperature is 37° F and the outside temperature is 78° F. *Ans.* 165,312 Btu/24 hr

2. The north wall of a locker plant is 12 ft by 60 ft and is constructed of 8 in. hollow clay tile insulated with 8 in. of corkboard. The locker plant is located in Houston, Texas and the inside temperature is maintained at 0° F. Determine the heat gain through the wall in Btu/24 hr. *Ans.* 54,000 Btu/24 hr

3. A cold storage warehouse in Orlando, Florida has a 30 ft by 50 ft flat roof constructed of 4 in. of concrete covered with tar and gravel and insulated with the equivalent of 4 in. of corkboard. If the roof is unshaded and the inside of the warehouse is maintained at 35° F, compute the heat gain through the roof in Btu/24 hr. *Ans.* 181,330 Btu/24 hr

4. A small walk-in cooler has an interior volume of 400 cu ft and receives heavy usage. If the inside of the cooler is maintained at 35° F and the outside design conditions are 90° F and 60% relative humidity, determine the air change load in Btu/24 hr. *Ans.* 50,300 Btu/24 hr

5. A frozen storage room has an interior volume of 2000 cu ft and is maintained at a temperature of −10° F. The usage is light and the outside design conditions (anteroom) are 50° F and 70% relative humidity. Compute the air change load in Btu/24 hr. *Ans.* 16,100 Btu/24 hr

6. Five thousand pounds of fresh, lean beef enter a chilling cooler at 100° F and are chilled to 38° F in 24 hr. Compute the chilling load in Btu/24 hr. *Ans.* 347,000 Btu/24 hr

7. Five hundred pounds of prepared, packaged beef enter a freezer at a temperature of 36° F. The beef is to be frozen and its temperature reduced to 0° F in 5 hr. Compute the product load. *Ans.* 267,900 Btu/hr

8. Fifty-five hundred crates of apples are in storage at 37° F. An additional 500 crates enter the storage cooler at a temperature of 85° F and are chilled to the storage temperature in 24 hr. The average weight of apples per crate is 60 lb. The crate weighs 10 lb and has a specific value of 0.6 Btu/lb/° F. Determine the total product load in Btu/24 hr. *Ans.* 1,679,800 Btu/24 hr

11

Evaporators

11-1. Types of Evaporators. As stated previously, any heat transfer surface in which a refrigerant is vaporized for the purpose of removing heat from the refrigerated space or material is called an evaporator. Because of the many different requirements of the various applications, evaporators are manufactured in a wide variety of types, shapes, sizes, and designs, and they may be classified in a number of different ways, such as type of construction, operating condition, method of air (or liquid) circulation, type of refrigerant control, and application.

11-2. Flooded and Dry-Expansion Evaporators. Evaporators fall into two general categories, flooded and dry expansion, according to their operating condition. The flooded type is always completely filled with liquid refrigerant, the liquid level being maintained with a float valve or some other liquid level control (Fig. 11-1). The vapor accumulating from the boiling action of the refrigerant is drawn off the top by the action of the compressor. The principal advantage of the flooded evaporator is that the inside surface of the evaporator is always completely wetted with liquid, a condition that produces a very high rate of heat transfer. The principal disadvantage of the flooded evaporator is that it is usually bulky and requires a relatively large refrigerant charge.

Liquid refrigerant is fed into the dry-expansion evaporator by an expansion device which meters the liquid into the evaporator at a rate such that all the liquid is vaporized by the time it reaches the end of the evaporator coil (Fig. 11-2). For either type, the rate at which the liquid is fed into the evaporator depends upon the rate of vaporization and increases or decreases as the heat load on the evaporator increases or decreases. However, whereas the flooded type is always completely filled with liquid, the amount of liquid present in the dry-expansion evaporator will vary with the load on the evaporator. When the load on the evaporator is light, the amount of liquid in the evaporator is small. As the load on the evaporator increases, the amount of liquid in the evaporator increases to accommodate the greater load. Thus, for the dry-expansion evaporator, the amount of liquid-wetted surface and, therefore, the evaporator efficiency, is greatest when the load is greatest.

11-3. Types of Construction. The three principal types of evaporator construction are: (1) bare-tube, (2) plate-surface, and (3) finned.

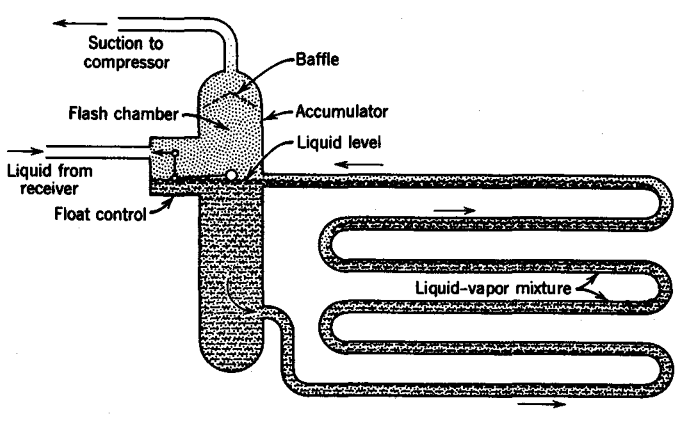


Fig. 11-1. Flooded evaporator. Notice accumulator and float control. Circulation of the refrigerant through the coil is by gravity. The vapor accumulated from the boiling action in the coil escapes to the top of the accumulator and is drawn off by the suction of the compressor.

Bare-tube and plate-surface evaporators are sometimes classified together as prime-surface evaporators in that the entire surface of both these types is more or less in contact with the vaporizing refrigerant inside. With the finned evaporator, the refrigerant-carrying tubes are the only prime surface. The fins themselves are not filled with refrigerant and are, therefore, only secondary heat transfer surfaces whose function is to pick up heat from the surrounding air and conduct it to the refrigerant-carrying tubes.

Although prime-surface evaporators of both the bare-tube and plate-surface types give satisfactory service on a wide variety of applications operating in any temperature range, they are most frequently applied to applications where the space temperature is maintained below 34° F and frost accumulation on the evaporator surface cannot be readily prevented. Frost accumulation on prime-surface evaporators does not affect the evaporator capacity to the extent that it does on finned coils. Furthermore, most prime surface evaporators, particularly the plate-surface type, are easily cleaned and can be readily defrosted manually by either brushing or scraping off the frost accumulation. This can be accomplished without interrupting the refrigerating process and jeopardizing the quality of the refrigerated product.

11-4. Bare-Tube Evaporators. Bare-tube evaporators are usually constructed of either steel pipe or copper tubing. Steel pipe is used for large evaporators and for evaporators to be employed with ammonia, whereas copper tubing is utilized in the manufacture of smaller evaporators intended for use with refrigerants other than ammonia. Bare-tube coils are available in a number of sizes, shapes, and designs, and are usually custom made to the individual application. Common shapes for bare-tube coils are flat zigzag and oval trombone, as shown in Fig. 11-3. Spiral bare-tube coils are often employed for liquid chilling.

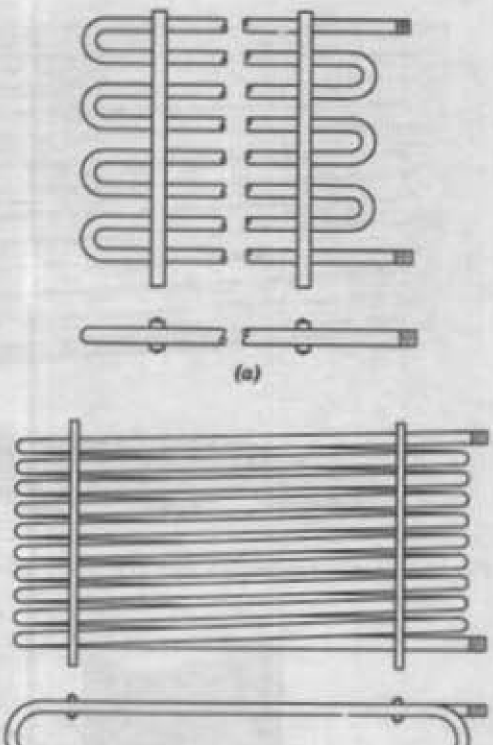


Fig. 11-3. Common designs for bare-tube coils. (a) Flat zigzag coil. (b) Oval trombone coil.

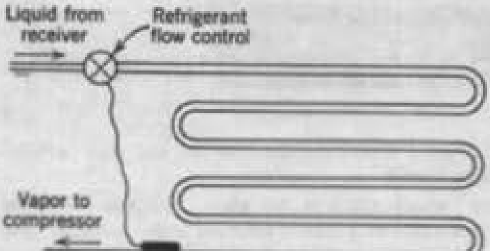


Fig. 11-2. Dry-expansion evaporator. Liquid refrigerant vaporizes progressively as it flows through coil and leaves coil as a vapor. Feeler bulb controls rate of flow through the orifice of the flow control.

11-5. Plate-Surface Evaporators. Plate-surface evaporators are of several types. Some are constructed of two flat sheets of metal so embossed and welded together as to provide a path for refrigerant flow between the two sheets (Fig. 11-4). This particular type of plate-surface evaporator is widely used in household refrigerators and home freezers because it is easily cleaned, economical to manufacture, and can be readily formed into any one of the various shapes required (Fig. 11-5).

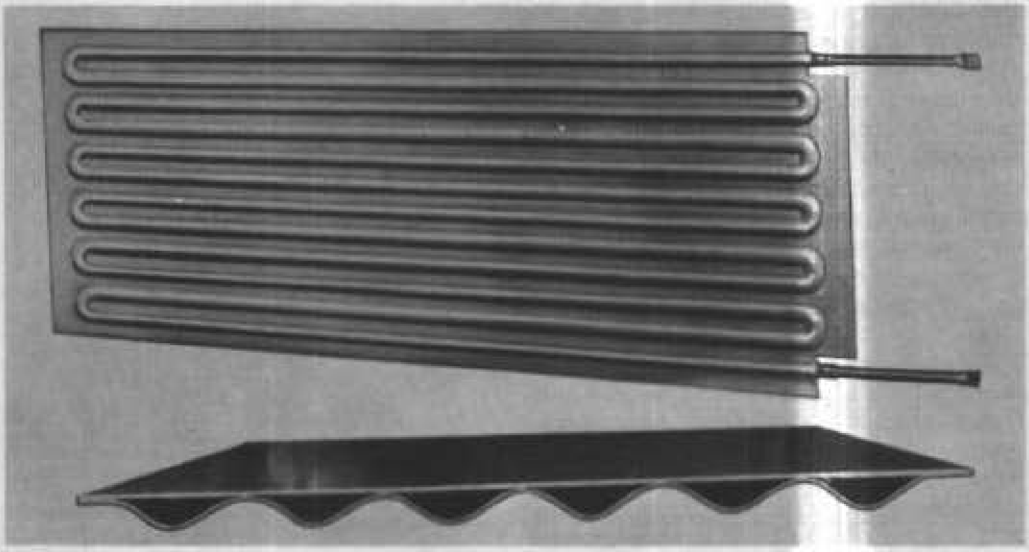


Fig. 11-4. Standard serpentine plate evaporator. (Courtesy Kold-Hold Division—Tranter Manufacturing, Inc.)

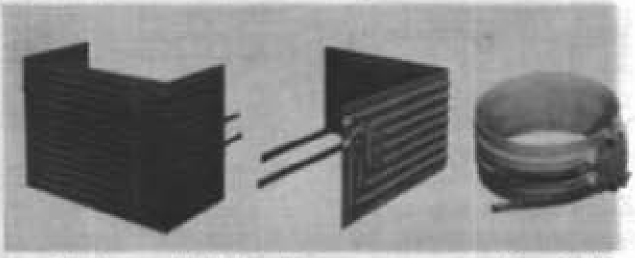


Fig. 11-5. Some typical shapes available in plate-type evaporators. (Courtesy Dean Products, Inc.)

(A) Outside jacket of plate. Heavy, electrically welded steel. Smooth surface.
(B) Continuous steel tubing through which refrigerant passes.
(C) Inlet from compressor.
(D) Outlet to compressor. Copper connections for all refrigerants except ammonia where steel connections are used.
(E) Fitting where vacuum is drawn and then permanently sealed.
(F) Vacuum space in dry plate. Space in hold-over plate contains eutectic solution under vacuum. No maintenance required due to sturdy, simple construction. No moving parts; nothing to wear or get out of order; no service necessary.

Fig. 11-6. Plate-type evaporator. (Courtesy Dole Refrigerating Company.)

Another type of plate-surface evaporator consists of formed tubing installed between two metal plates which are welded together at the edges (Fig. 11-6). In order to provide good thermal contact between the welded plated and the tubing carrying the refrigerant, the space between the plates is either filled with a eutectic solution or evacuated so that the pressure of the atmosphere exerted on the outside surface of the plates holds the plates firmly against the tubing inside. Those containing the eutectic solution are especially useful where a holdover capacity is required. Many are used on refrigerated trucks. In such applications, the plates are mounted either vertically or horizontally from the ceiling or walls of the truck (Fig. 11-7) and are usually connected to a central plant refrigeration system while the trucks are parked at the terminal during the night. The refrigerating capacity thus stored in the eutectic solution is sufficient to refrigerate the product during the next day's operations. The temperature of the plates is controlled by the melting point of the eutectic solution.

Plate-type evaporators may be used singly or in banks. Figure 11-8 illustrates how the plates can be grouped together for ceiling mounting in holding rooms, locker plants, freezers, etc. The plates may be manifolded for parallel flow of the refrigerant (Fig. 11-9) or they may be connected for series flow.

Plate-surface evaporators provide excellent shelves in freezer rooms and similar applications (Fig. 11-10). They are also widely used as partitions in freezers, frozen food display cases, ice cream cabinets, soda fountains, etc. Plate evaporators are especially useful for liquid cooling installations where unusual peak load conditions are encountered periodically. By building up an ice bank on the surface of the plates during periods of light loads, a holdover refrigerating capacity is established which will help the refrigerating equipment carry the load through the heavy or peak condition (Fig. 11-11).

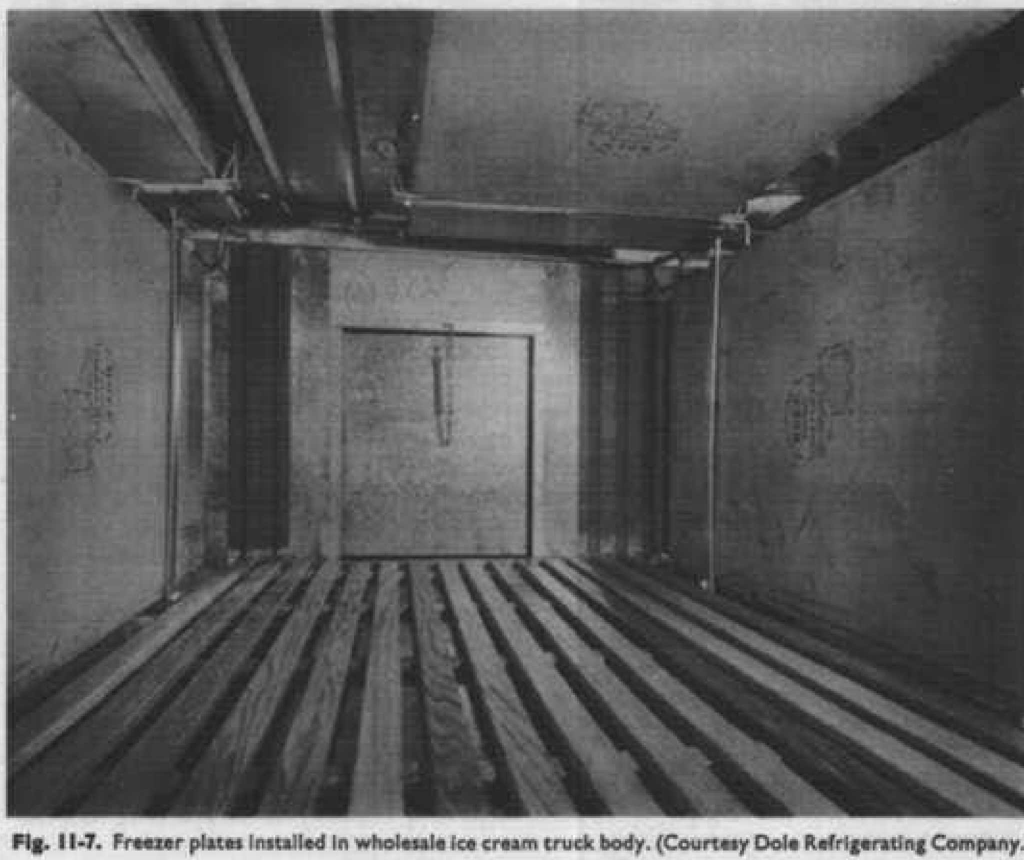


Fig. 11-7. Freezer plates installed in wholesale ice cream truck body. (Courtesy Dole Refrigerating Company.)

Fig. 11-8. Plate banks employed in low temperature storage rooms. (Courtesy Dole Refrigerating Company.)

Since this allows the use of smaller capacity equipment than would ordinarily be required by the peak load, a savings is affected in initial cost and, usually, also in operating expenses.

11-6. Finned Evaporators. Finned coils are bare-tube coils upon which metal plates or fins have been installed (Fig. 13-15). The fins, serving as secondary heat-absorbing surfaces, have the effect of increasing the over-all surface area of the evaporator, thereby improving its efficiency. With bare-tube evaporators, much of the air that circulates over the coil passes through the open spaces between the tubes and does not come in contact with the coil surface. When fins are added to a coil, the fins extend out into the open spaces between the tubes and act as heat collectors. They remove heat from that portion of the air which would not ordinarily come in contact with the prime surface and conduct it back to the tubing.

It is evident that to be effective the fins must be connected to the tubing in such a manner that good thermal contact between the fins and the tubing is assured. In some instances, the fins are soldered directly to the tubing. In others, the fins are slipped over the tubing and the tubing is expanded by pressure or some such means so that the fins bite into the tube surface and establish good thermal contact. A variation of the latter method is to flare the fin hole slightly to allow the fin to slip over the tube. After the fin is installed, the flare is straightened and the fin is securely locked to the tube.

Fin size and spacing depend in part on the particular type of application for which the coil is designed. The size of the tube determines the size of the fin. Small tubes require small fins. As the size of the tube increases, the size of the fin may be effectively increased. Fin spacing varies from one to fourteen fins per inch, depending primarily on the operating temperature of the coil.

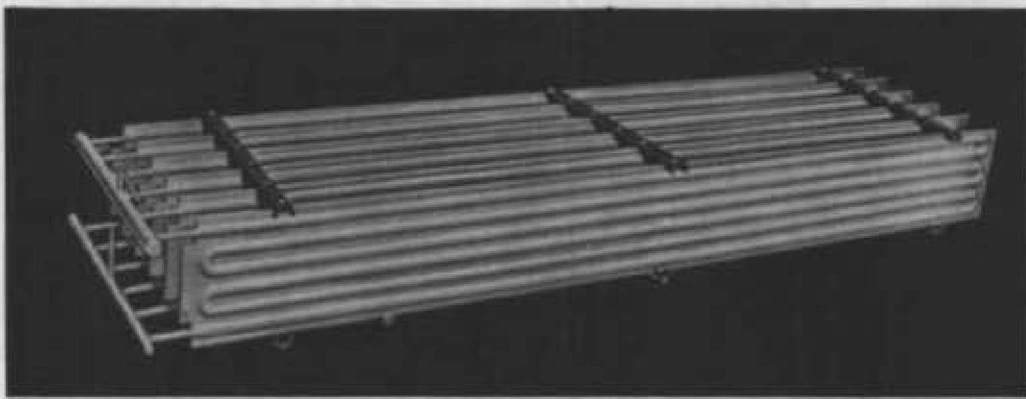


Fig. 11-9. Plate bank with plates manifolded for parallel refrigerant flow. Plates may also be connected for series flow. (Courtesy Kold-Hold Division—Tranter Manufacturing, Inc.)

Fig. 11-10. Plate evaporators employed as freezer shelves. Note that plates are arranged for series refrigerant flow. (Courtesy Kold-Hold Division—Tranter Manufacturing, Inc.)

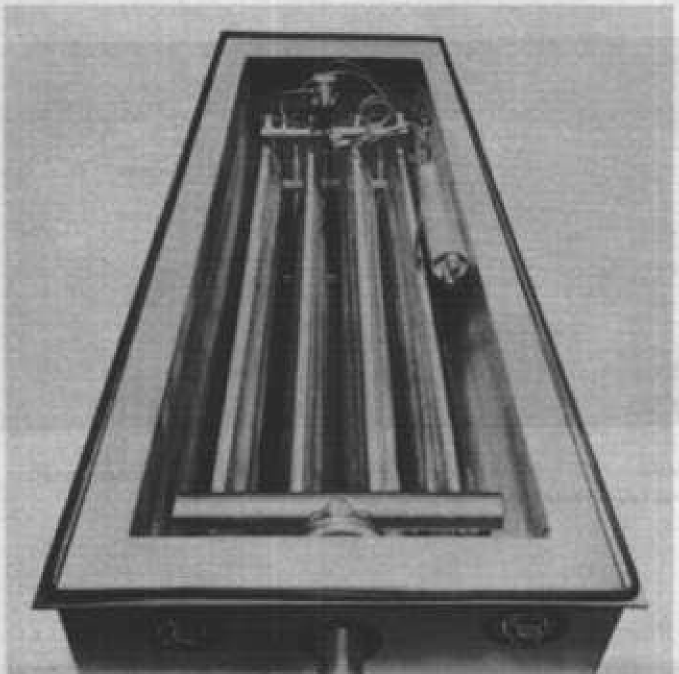


Fig. 11-11. Ice-Cel. Refrigeration holdover capacity is established by building up a bank of ice on plate evaporators. (Courtesy Dole Refrigerating Company.)

Frost accumulation on air-cooling coils operating at low temperatures is unavoidable, and since any frost accumulation on finned coils tends to restrict the air passages between the fins and to retard air circulation through the coil, evaporators designed for low temperature applications must have wide fin spacing (two or three fins per inch) in order to minimize the danger of blocking air circulation. On the other hand, coils designed for air conditioning and other installations where the coil operates at temperatures high enough so that no frost accumulates on the coil surface may have as many as fourteen fins per inch.

When air circulation over finned coils is by gravity, it is important that the coil offer as little resistance to air flow as is possible; therefore, in general, fin spacing should be wider for natural convection coils than for coils employing fans.

It has been determined that a definite relationship exists between the inside and outside surfaces of an evaporator. Since external finning affects only the outside surface, the addition of fins beyond a certain limit will not materially increase the capacity of the evaporator. In fact, in some instances, excessive finning may actually reduce the evaporator capacity by restricting the air circulation over the coil unnecessarily.

Since their capacity is affected more by frost accumulation than any other type of evaporator, finned coils are best suited to air-cooling application where the temperature is maintained above 34° F. When finned coils are used for low temperature operation, some means of defrosting the coil at regular intervals must be provided. This may be accomplished automatically by several means which are discussed in another chapter.

Because of the fins, finned coils have more surface area per unit of length and width than prime-surface evaporators and can therefore be built more compactly. Generally, a finned coil will occupy less space than either a bare-tube or plate-surface evaporator of the same capacity. This provides for a considerable savings in space and makes finned coils ideally suited for use with fans as forced convection units.

11-7. Evaporator Capacity. The capacity of any evaporator or cooling coil is the rate at which heat will pass through the evaporator walls from the refrigerated space or product to the vaporizing liquid inside and is usually expressed in Btu per hour. An evaporator selected for any specific application must always have sufficient capacity to allow the vaporizing refrigerant to absorb heat at the rate necessary to produce the required cooling when operating at the design conditions.

Heat reaches the evaporator by all three methods of heat transfer. In air-cooling applications most of the heat is carried to the evaporator by convection currents set up in the refrigerated space either by action of a fan or by gravity circulation resulting from the difference in temperature between the evaporator and the space. Too, some heat is radiated directly to the evaporator from the product and from the wall of the space. Where the product is in thermal contact with the outer surface of the evaporator, heat is transferred from the product to the evaporator by direct conduction. This is always true for liquid cooling applications where the liquid being cooled is always in contact with the evaporator surface. However, circulation of the cooled fluid either by gravity or by action of a pump is still necessary for good heat transfer.

Regardless of how the heat reaches the outside surface of the evaporator, it must pass through the walls of the evaporator to the refrigerant inside by conduction. Therefore, the capacity of the evaporator, that is, the rate at which heat passes through the walls, is determined by the same factors that govern the rate of heat flow by conduction through any heat transfer surface and is expressed by the formula

$$Q = A \times U \times D \tag{11-1}$$

where Q = the quantity of heat transferred in Btu/hr

A = the outside surface area of the evaporator (both prime and finned)

U = the over-all conductance factor in Btu/hr/sq ft of outside surface/° F D

D = the logarithmic mean temperature difference in degrees Fahrenheit between the temperature outside the evaporator and the temperature of the refrigerant inside the evaporator

11-8. U or Over-All Conductance Factor. The resistance to heat flow offered by the evaporator walls is the sum of three factors whose relationship is expressed by the following:

$$\frac{1}{U} = \frac{R}{f_i} + \frac{L}{K} + \frac{1}{f_0} \tag{11-2}$$

where U = the over-all conductance factor in Btu/hr/sq ft/° F D

f_i = the conductance factor of the inside surface film in Btu/hr/sq ft of inside surface/° F D

L/K = resistance to heat flow offered by metal of tubes and fins

f_0 = the conductance factor of the outside surface film in Btu/hr/sq ft of outside surface/° F D

R = ratio of outside surface to inside surface

Since a high rate of heat transfer through the evaporator walls is desirable, the U or conductance factor should be as high as possible. Metals, because of their high conductance factor, are always used in evaporator construction. However, a metal which will not react with the refrigerant must be selected. Iron, steel, brass, copper, and aluminum are the metals most commonly used. Iron and steel are not affected by any of the common refrigerants, but are apt to rust if any moisture is present in the system. Brass and copper can be used with any refrigerant except ammonia, which dissolves copper. Aluminum may be used with any refrigerant except methyl chloride. Magnesium and magnesium alloys cannot be used with the fluorinated hydrocarbons or with methyl chloride.

Of the three factors involved in Equation 11-2, the metal of the evaporator walls is the least significant. The amount of resistance to heat flow offered by the metal is so small, especially where copper and aluminum are concerned, that it is usually of no consequence. Thus, the U factor of the evaporator is determined primarily by the coefficients of conductance of the inside and outside surface films.

In general, because of the effect they have on the inside and outside film coefficients, the value of U for an evaporator depends on the type of coil construction and the material used, the amount of interior wetted surface, the velocity of the refrigerant inside the coil, the amount of oil present in the evaporator, the material being cooled, the condition of the external surface, the fluid (either gaseous or liquid) velocity over the coil, and the ratio of inside to outside surface.

Heat transfer by conduction is greater through liquids than through gases and the rate at which the refrigerant absorbs heat from the evaporator walls increases as the amount of interior wetted surface increases. In this respect, flooded evaporators, since they are always completely filled with liquid, are more efficient than the dry-expansion type. This principle also applies to the external evaporator surface. When the outside surface of the evaporator is in direct contact with some liquid or solid medium, the heat transfer by conduction to the outside surface of the evaporator is greater than when air is the medium in contact with the evaporator surface.

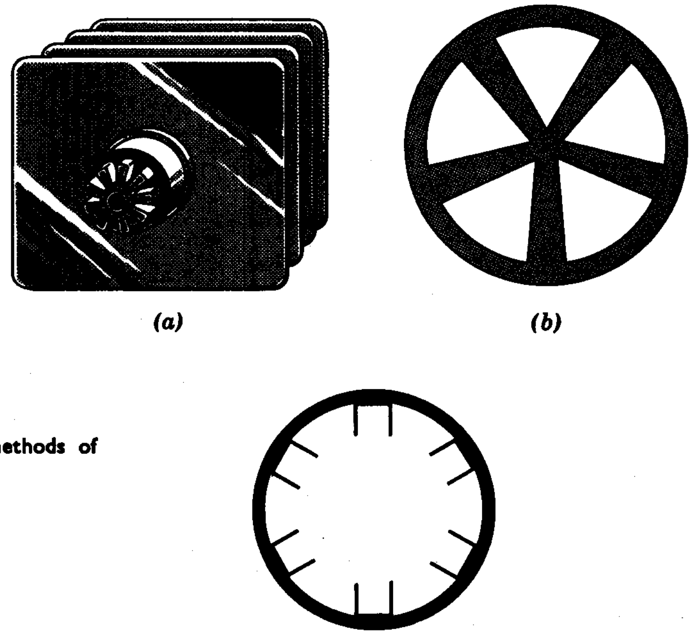
Any fouling of either the external or internal surfaces of the evaporator tends to act as thermal insulation and decreases the conductance factor of the evaporator walls and reduces the rate of heat transfer. Fouling of the external surface of air-cooling evaporators is usually caused by an accumulation of dust and lint from the air which adheres to the wet coil surfaces or by frost accumulation on the coil surface. In liquid-cooling applications, fouling of the external tube surface usually results from scale formation and corrosion. Fouling of the internal surface of the evaporator tubes is usually caused by excessive amounts of oil in the evaporator and/or low refrigerant velocities. At low velocities, vapor bubbles, formed by the boiling action of the refrigerant, tend to cling to the tube walls, thereby decreasing the amount of interior wetted surface. Increasing the refrigerant velocity produces a scrubbing action on the walls of the tube which carries away the oil and bubbles and improves the rate of heat flow. Thus, for a given tube size, the inside film coefficient increases as the refrigerant velocity in creases. The refrigerant velocity is limited, however, by the maximum allowable pressure drop through the coil and, if increased beyond a certain point, will result in a decrease rather than an increase in coil capacity. This depends to some extent on the method of coil circuiting and is discussed later. It can be shown also that the conductance of the outside surface film is improved by increasing the fluid velocity over the outside surface of the coil. But, here again, in many cases the maximum velocity is limited, this time by consideration other than the capacity of the evaporator itself.

Any increase in the turbulence of flow either inside or outside the evaporator will materially increase the rate of heat transfer through the evaporator walls. In general, internal turbulence increases with the difference in temperature across the walls of the tube, closer spacing of the tubes, and the roughness of the internal tube surface. In some instances, heat transfer is improved by internal finning.

Outside flow turbulence is influenced by fluid velocity over the coil, tube spacing, and the shape of the fins.

11-9. The Advantage of Fins. The advantage of finning depends on the relative values of the coefficients of conductance of the inside and outside surface films and upon R, the ratio of the outside surface to the inside surface. In any instance where the rate of heat flow from the inside surface of the evaporator to the liquid refrigerant is such that it exceeds the rate at which heat passes to the outside surface from the cooled medium, the over-all capacity of the evaporator is limited by the capacity of the outside surface. In such cases, the over-all value of U can be increased by using fins to increase the outside surface area to a point such that the amount of heat absorbed by the outside surface is approximately equal to that which can pass from the inside surface to the liquid refrigerant.

Because heat transfer is greater to liquids than to vapors, this situation often exists in air-cooling applications where the rate of heat flow from the inside surface to the liquid refrigerant is much higher than that from the air to the outside surface. For this reason, the use of finned evaporators for air-cooling applications is becoming more and more prevalent. On the other hand, in liquid-cooling applications, since liquid is in contact with both sides of the evaporator and the rate of flow is approximately equal for both surfaces, bare-tube evaporators perform at high efficiency and finning is usually unnecessary. In some applications, where fluid velocity over the outside of the evaporator is exceptionally high, the flow of heat to the outer surface may be greater than the flow from the inner surface to the refrigerant. When this occurs, the use of inner fins will improve evaporator capacity in that the amount of interior wetted surface is increased. Several methods of inner finning are shown in Fig. 11-12.



(a) (b) (c)

Fig. 11-12. Some methods of inner finning.

11-10. Logarithmic Mean Temperature Difference. As illustrated in Fig. 11-13, the temperature of air (or any other fluid) decreases progressively as it passes through the cooling coil. The drop in temperature takes place along a curved line (*A*) approximately as indicated. Assuming that the temperature of the refrigerant remains constant, it is evident that the difference in temperature between the refrigerant and the air will be greater at the point where the air enters the coil than at the point where it leaves, and that the average or mean difference in temperature will fall along the curve (*a*) at a point somewhere between the two extremes. Although the value obtained deviates slightly from the actual logarithmic mean, an approximate mean temperature difference may be calculated by the following equation:

$$D = \frac{(t_e - t_r) + (t_1 - t_r)}{2} \qquad (11\text{-}3)$$

where D = the arithmetic mean temperature
t_e = the temperature of the air entering the coil
t_1 = the temperature of the air leaving the coil
t_r = the temperature of the refrigerant in the tubes

For the values given in Fig. 11-13, the arithmetic mean temperature difference is

$$D = \frac{(40 - 20) + (30 - 20)}{2} = 15° \text{ F}$$

It must be remembered that the MTD as calculated by Equation 11-3 is slightly in error because of the curvature of the curved line *A* and would be the actual MTD only if the drop in air temperature occurred along a straight line, as indicated by the dotted line *B* in Fig. 11-13.

The actual logarithmic mean temperature, which is the midpoint of the curved line *A*, is given by equation

$$D = \frac{(t_e - t_r) - (t_1 - t_r)}{\ln \dfrac{(t_e - t_r)}{(t_1 - t_r)}} \qquad (11\text{-}4)$$

For the values given in Fig. 11-13, the logarithmic mean temperature difference will be

$$D = \frac{(40 - 20) - (30 - 20)}{\ln \dfrac{40 - 20}{30 - 20}} = \frac{10}{\ln 2} = 14.43° \text{ F}$$

The preceding calculations were made on the assumption that the refrigerant temperature remains constant. When this is not the case,

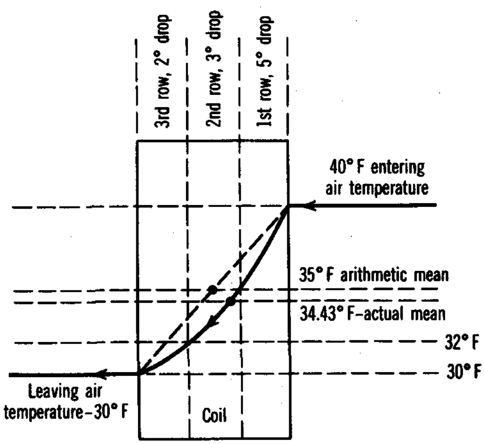


Fig. 11-13. Mean temperature of air passing through evaporator.

t_r will have two values. This condition is discussed in another chapter.

The log mean temperature difference hereafter called mean effective temperature difference (METD), may also be determined from Table 11-1.

11-11. The Effect of Air Quantity on Evaporator Capacity. Although not a part of the basic heat transfer equation, there are other factors external to the coil itself which greatly affect coil performance. Principal among these are the circulation, velocity, and distribution of air in the refrigerated space and over the coil. These factors are closely related and in many cases are dependent one on the other.

Except in liquid cooling and in applications where the product is in direct contact with the evaporator, most of the heat from the product is carried to the evaporator by air circulation: If air circulation is inadequate, heat is not carried from the product to the evaporator at a rate sufficient to allow the evaporator to perform at peak efficiency. It is important also that the circulation of air is evenly distributed in all parts of the refrigerated space and over the coil. Poor distribution of the circulating air results in uneven temperatures and "dead spots" in the refrigerated space, whereas the uneven distribution of air over the coil surface causes some parts of the surface to function less efficiently than others and lowers evaporator capacity.

The velocity of the air passing over the coil has a considerable influence on both the value of U and the METD and is important in determining evaporator capacity. When air velocity is low, the air passing over the coil stays in contact with the coil surface longer and is cooled through a greater range. Thus, the METD and the rate of heat transfer is low. As air velocity increases, a greater quantity of air is brought in contact with the coil per unit of time, the METD increases, and the rate of heat transfer improves. In addition, high air velocities tend to break up the thin film of stagnant air which is adjacent to all surfaces. Since this film of air acts as a heat barrier and insulates the surface, its disturbance increases the conductance of the outside surface film and the over-all value of U improves.

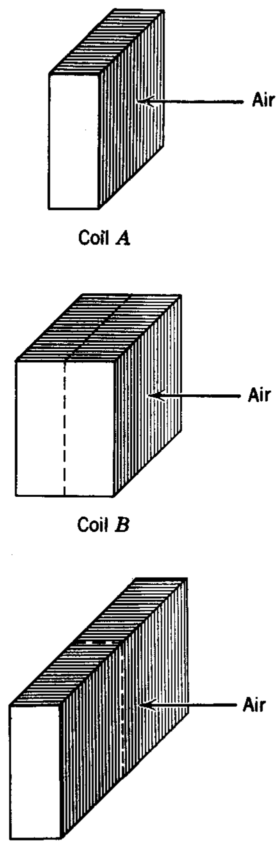


Fig. 11-14. Coils *B* and *C* both have twice the surface area of coil *A*. Coil *C* has twice the face area of coil *A* or coil *B*.

11-12. Surface Area. Equation 11-1 indicates that the capacity of an evaporator varies directly with the outside surface area. This is true only if the U factor of the evaporator and the METD remains the same. In many cases, the value of U and the METD are affected when the surface area of the evaporator is changed. In such cases, the capacity of the evaporator does not increase or decrease in direct proportion to the change in surface area. To illustrate, in Fig. 11-14, coils B and C each have twice the surface area of coil A, yet the increase in capacity over the capacity of coil A will be greater for coil C than for coil B. Provided the air velocity is the same (the total quantity of air circulated over coil C must be twice that circulated over coil A), the METD across C will be exactly the same as that across A and the capacity of C will therefore be twice the capacity of coil A.

Figure 11-15 shows how the METD is affected when the surface area of the coil is increased by increasing the number of rows (depth). Note that the drop in air temperature is much greater across the first row and diminishes as the air passes across each succeeding row. This is accounted for by the fact that the temperature difference between the air and the refrigerant is much greater across the first row, becomes less and less as the temperature of the air is reduced in passing across each row, and is least across the last row. It is evident then that the rate of heat transfer is greater for the first row and that the first row performs the most efficiently. For this reason, if the surface area of coil A in Fig. 11-14 is doubled by increasing the number of rows as in coil B, the METD will be decreased and the increase in capacity will not be nearly as great as when the surface area is increased as in coil C. For the same total surface area, a long, wide, flat coil will, in general, perform more efficiently than a short, narrow coil having more rows depth. However, in many instances, the physical space available is limited and compact coils arrangements must be used. In applications where it it is permissible, the loss of capacity resulting from increasing the number of rows can be compensated for to some extent by increasing the air velocity over the coil. Too, in some applications, the use of deep coils is desirable for the purpose of dehumidification.

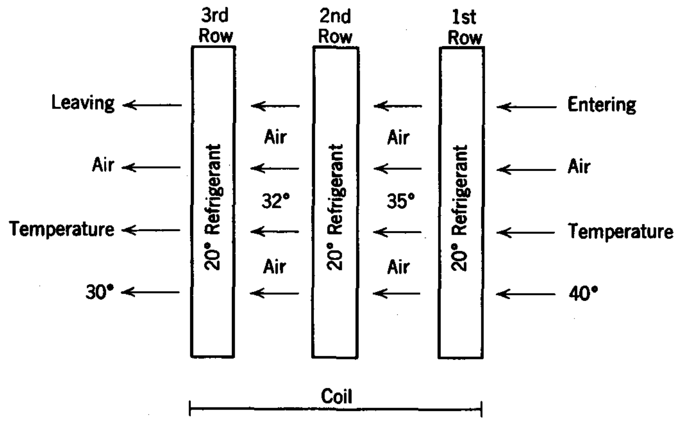


Fig. 11-15. Air temperature drop across typical three-row cooling coil.

11-13. Evaporator Circuiting. It was demonstrated in Chapter 8 that excessive pressure drop in the evaporator results in the suction vapor arriving at the suction inlet of the compressor at a lower pressure than is actually necessary, thereby causing a loss of compressor capacity and efficiency.

To avoid unnecessary losses in compressor capacity and efficiency, it is desirable to design the evaporator so that the refrigerant experiences a minimum drop in pressure. On the other hand, a certain amount of pressure drop is required to flow the refrigerant through the evaporator, and since velocity is a function of pressure drop, the drop in pressure must be sufficient to assure refrigerant velocities high enough to sweep the tube surfaces free of vapor bubbles and oil and to carry the oil back to the compressor. Hence, good design requires that the method of evaporator circuiting be such

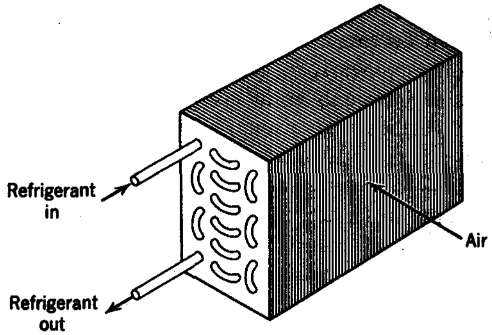


Fig. 11-16. Evaporator with one series refrigerant circuit.

that the drop in pressure through the evaporator is the minimum necessary to produce refrigerant velocities sufficient to provide a high rate of heat transfer and good oil return.

In general, the drop in pressure through any one evaporator circuit will depend upon the size of the tube, the length of the circuit, and the circuit load. By circuit load is meant the time-rate of heat flow through the tube walls of the circuit. The circuit load determines the quantity of refrigerant which must pass through the circuit per unit of time. The greater the circuit load, the greater must be the quantity of refrigerant flowing through the circuit and the greater will be the drop in pressure. Hence, for any given tube size, the greater the load on the circuit, the shorter the circuit must be in order to avoid excessive pressure drop.

Evaporators having only a single series refrigerant circuit, such as the one illustrated in Fig. 11-16, will perform satisfactorily within certain load limits. When the load limit is exceeded, the refrigerant velocity will be increased beyond the desired range and the pressure drop will be excessive.

Notice that the refrigerant enters at the top of the evaporator as a liquid and leaves at the bottom as a vapor. Since the volume of the refrigerant increases as the refrigerant vaporizes, the refrigerant velocity and the pressure drop per foot increase progressively as the refrigerant travels through the circuit, and are greatest at the end of the coil where the refrigerant is 100% vapor.

The excessive pressure drop occurring in the latter part of a single series circuit evaporator can be eliminated to a great extent by splitting the single circuit into two circuits in the lower portion of the evaporator (Fig. 11-17). When this is done, the refrigerant travels a single series path until the refrigerant velocity builds to the allowable maximum, at which time the circuit is split into two parallel paths for the balance of the travel through the evaporator. This has the effect of reducing the refrigerant velocity in the two paths to one-half the value it would have without the split, and the pressure drop per foot is reduced to one-eighth of the value it would have in the lower part of the evaporator with a single single series circuit.* This, of course, will permit greater loading of the coil without exceeding the allowable pressure drop. At the same time, the velocity in all parts of the coil is maintained within the desirable limits so that the rate of heat transfer is not unduly affected.

Another method of reducing the pressure drop through the evaporator is to install refrigerant headers at the top and bottom of the evaporator so that the refrigerant is fed simultaneously through a multiple of parallel circuits (Fig. 11-18). However, this arrangement is not too satisfactory and is not widely used. While the pressure drop through the evaporator is low, this method of circuiting ordinarily results in

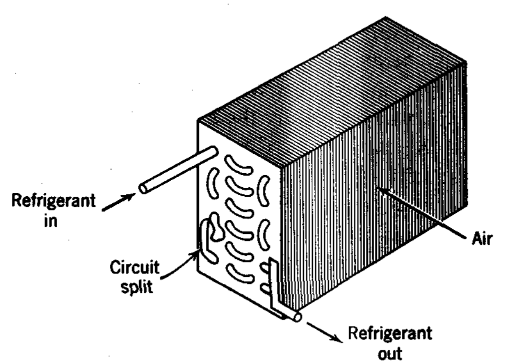


Fig. 11-17. Evaporator with split refrigerant circuit.

* Pressure drop increases as the square of the velocity. Reducing the velocity to one-half reduces the pressure drop to one-quarter of its original value. Then, since the length of each parallel branch is only one-half the length of a single circuit, the drop in pressure in the lower portion of the split coil is only one-eighth of the single circuit value.

reducing the refrigerant velocity below the desired minimum so that the inside film coefficient and the rate of heat transfer are also low. Another disadvantage of this type of circuiting is that the loading of the circuits is uneven. Since the temperature difference between the air passing over the coil and the refrigerant in the tubes is much greater across the first circuit (first row) than across the last circuit (last row), the loading of the first circuit is much greater than the loading of the last circuit. Hence, the refrigerant velocity and the drop in pressure through the several circuits are uneven and a large portion of the coil operates inefficiently. This criticism can be applied to some extent also for the circuit arrangements in Figs. 11-16 and 11-17. In all three arrangements, the lower portion of the evaporator will not perform as effectively as the upper portion because wetting of the internal tube surface will not be as great in the lower portion. This is because the refrigerant in the lower portion contains a high percentage of vapor, whereas in the upper portion the refrigerant is nearly all liquid.

It is for this reason also that the outside surface temperature of the coil is always lowest near the refrigerant inlet and highest near the outlet, in spite of the fact that the saturation temperature of the refrigerant is lowest at the outlet due to the drop in pressure through the coil.

The circuit arrangement shown in Fig. 11-19 is very effective and is widely used, particularly when circuit loading is heavy, as in the case of

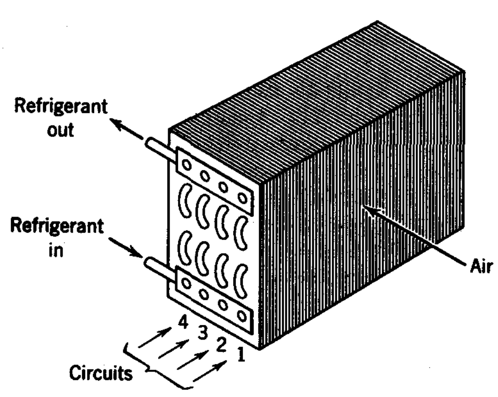


Fig. 11-18. Four-circuit evaporator with refrigerant headers on both inlet and outlet. Crossflow of air and refrigerant results in uneven circuit loading.

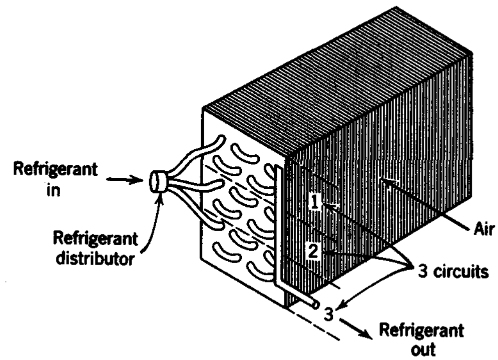


Fig. 11-19. Evaporator with refrigerant distributor and suction header. Notice counterflow arrangement for refrigerant and air.

an air conditioning coil where the temperature differential between the refrigerant and the air is large and where external finning is heavy. Notice that the air passes in counterflow to the refrigerant so that the warmest air is in contact with the warmest part of the coil surface. This provides the greatest mean temperature differential and the highest rate of heat transfer. Notice also that loading of the circuits is even. The number and length of the circuits that such a coil should have are determined by the size of the tube and the load on the circuits.

For the multipass, headered evaporator, the arrangement shown in Fig. 11-20 is much more desirable than that shown in Fig. 11-18. Counterflowing of the air and the refrigerant increases the METD and permits more even loading of the circuits.

11-14. Use of Manufacturer's Rating Tables. The mathematical evaluation of all the factors which influence evaporator capacity is usually impractical and in many cases impossible. For the most part, evaporator capacities must be determined by actual testing of the evaporator. The results of such tests are contained in the rating tables published by the various evaporator manufacturers.

The method of rating evaporators varies somewhat with the type of evaporator and with the particular manufacturer involved. However, the various rating methods are very similar and most manufacturers include, along with the evaporator rating tables, instructions as to how to use the ratings. In most cases, where the

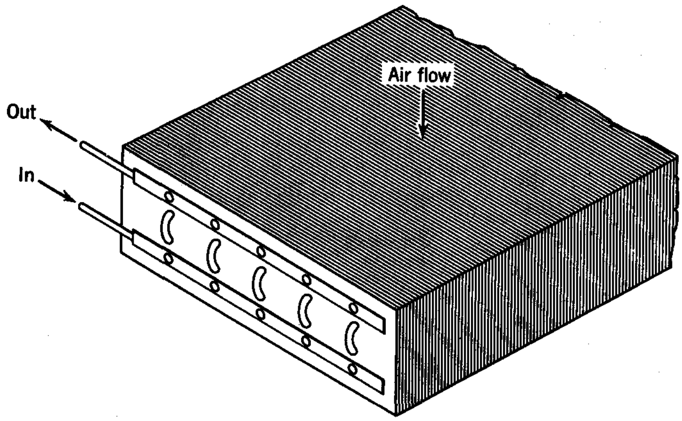


Fig. 11-20. Counterflow of refrigerant and air results in more even circuit loading and a higher mean temperature differential. Compare this arrangement with the cross-flow arrangement in Fig. 11-18.

evaporators are rated in accordance with ASRE Standards, the capacity data are reliable and are for operating conditions as normally encountered.

The selection of evaporators from manufacturer's rating tables is relatively simple once the conditions at which the evaporator is to operate are known. Typical evaporator rating tables, along with methods of evaporator selection, are discussed at the end of the chapter.

11-15. Evaporator TD. One of the most important factors to consider in selecting the proper evaporator for any given application is the evaporator TD. Evaporator TD is defined as the difference in temperature between the temperature of the air entering the evaporator and the saturation temperature of the refrigerant corresponding to the pressure at the evaporator outlet.*

Although more exact methods of rating evaporators are necessary in order to select evaporators for air conditioning applications and some product storage applications where space temperature and humidity are especially critical, ratings for most evaporators designed for product cooling applications are based on evaporator TD.

The relationship between evaporator capacity and evaporator TD is shown by the curve in Fig. 11-21. Notice that the capacity of the evaporator (Btu/hr) varies directly with the evaporator TD. That is, if an evaporator has a certain capacity at a 1° F TD, it will have exactly ten times that capacity if the TD is increased to 10° F, provided that all other conditions are the same.†

It is evident that a coil with a relatively small surface area operating at a relatively large TD can have the same capacity as another coil having a larger surface area but operating at a smaller TD. Thus, insofar as Btu per hour capacity alone is concerned, a small coil will have that same refrigerating effect as a larger one, provided that the TD at which the small coil operates is greater in proportion. However, it will be shown in the following sections that the temperature difference between the evaporator and the refrigerated space has considerable influence on the condition of the stored product and upon the operating efficiency of the entire system, and is usually, therefore, the determining factor in coil selection. Before an evaporator can be selected, it is necessary to first determine the TD at which it is expected to function. Once the desired temperature difference is known, an

* ASRE Standard 25–56, *Methods of Rating Air Coolers For Refrigeration.*

† Care should be taken not to confuse METD with evaporator TD. According to Equation 11-1, the Btu per hour capacity of any given evaporator (whose U factor and surface area are fixed at the time of manufacture) varies directly with the METD. However, assuming that the refrigerant temperature and all else remains constant, the METD between the air passing over the evaporator and the refrigerant in the evaporator will vary directly with the temperature of the air entering the evaporator. That is, if the temperature of the air entering the evaporator increases, the METD increases. Hence, the METD varies in proportion to the evaporator TD and, therefore, the capacity of the evaporator also varies in proportion to the evaporator TD.

evaporator having sufficient surface area to provide the required cooling capacity at the design TD can be selected.

11-16. The Effect of Coil TD on Space Humidity. The preservation of food and other products in optimum condition by refrigeration depends not only upon the temperature of the refrigerated space but also upon space humidity. When the humidity in the space is too low, excessive dehydration occurs in such products as cut meats, vegetables, dairy products, flowers, fruits, etc. On the other hand, when the humidity in the refrigerated space is too high, the growth of mold, fungus, and bacteria is encouraged and bad sliming conditions occur, particularly on meats and especially in the wintertime. Space humidity is of little importance, of course, when the refrigerated product is in bottles, cans, or other vapor-proof containers.

The most important factor governing the humidity in the refrigerated space is the evaporator TD.* The smaller the difference in temperature between the evaporator and the space, the higher is the relative humidity in the space. Likewise, the greater the evaporator TD, the lower is the relative humidity in the space.

When the product to be refrigerated is one that will be affected by the space humidity, an evaporator TD that will provide the optimum humidity conditions for the product should be selected. In such cases, the evaporator TD is the most important factor determining the evaporator selection. The design evaporator TD required for various space humidities is given in Table 11-2 for both natural-convection and forced-convection evaporators.

In applications where the space humidity is of no importance, the factors governing evaporator selection are: (1) system efficiency and economy of operation, (2) the physical space available for evaporator installation, and (3) initial cost.

11-17. The Effect of Air Circulation on Product Condition. As stated previously, circulation of air in the refrigerated space is essential to carry the heat from the product to the evaporator. When air circulation is inadequate, the capacity of the evaporator is decreased, the product is not cooled at a sufficient rate, the growth of mold and bacteria is encouraged, and sliming occurs on some products. On the other hand, too much air circulation can be as detrimental as too little. When the circulation of air is too great, the rate of moisture evaporation from the product surface increases and excessive dehydration of the product results. Excessive dehydration can be very costly in that it causes deterioration in product appearance and quality and shortens the life of the product. Furthermore, the loss of weight resulting from shrinkage and trimming is a considerable factor in dealer profits and in the price of perishable foods.

The desired rate of air circulation varies with the different applications and depends upon the space humidity, the type of product, and the length of the storage period.

With respect to product condition, air circulation and space humidity are closely associated. Poor air circulation has the same effect on the product as high humidity, whereas too much air circulation produces the same effect as low humidity. In many instances, it is difficult to determine whether product deterioration in a particular application is caused by faulty air circulation or poor humidity conditions. For the most part, product condition depends upon the combined effects of humidity and air circulation, rather than upon the effect of either one alone, and either of these two factors can be varied somewhat, provided that the other is varied in an off-setting direction. For example, higher than normal air velocities can be used without damage to the product when the space humidity is also maintained at a higher level.

The type of product and the amount of

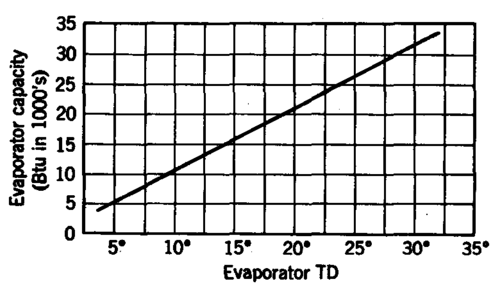


Fig. 11-21. Variation in evaporator capacity with evaporator TD.

* Some of the other factors which influence the space humidity are: air motion, system running time, type of system control, amount of exposed product surface, infiltration, outside air conditions, etc.

exposed surface should be given consideration when determining the desired rate of air circulation. Some products, such as flowers and vegetables, are more easily damaged by excessive air circulation than others and must be given special consideration. Cut meats, since they have more exposed surface, are more susceptible to loss of weight and deterioration than are beef quarters or sides, and air velocities should be lower. On the other hand, where the product is in vapor-proof containers, it will not be affected by high velocities and the rate of air circulation should be maintained at a high level to obtain the maximum cooling effect.

Recommended air velocities for product storage are given in Tables 10-10 through 10-13.

11-18. Natural Convection Evaporators. Natural convection evaporators are frequently used in applications where low air velocities and minimum dehydration of the product are desired. Typical installations are household refrigerators, display cases, walk-in coolers, reach-in refrigerators, and large storage rooms.

The circulation of air over the cooling coil by natural convection is a function of the temperature differential between the evaporator and the space. The greater the difference in temperature, the higher the rate of air circulation.

The circulation of air by natural convection is greatly influenced by the shape, size, and location of the evaporator, the use of baffles, and the placement of the stored product in the refrigerated space. Generally, shallow coils (one or two rows deep) extending the entire length of the cooler and covering the greater portion of the ceiling area are best. As the depth of the coil is increased, the coil offers greater resistance to the free circulation of air and the METD is thereby decreased with a resulting decrease in the coil capacity. Since cold air is denser than warm air and tends to fall to the floor, evaporators should be located as high above the floor as possible, but care should be taken to leave sufficient room between the evaporator and the ceiling to permit the free circulation of air over the top of the coil.

For coolers less than 8 ft wide, single, ceiling-mounted evaporators are frequently used. When the width of the cooler exceeds 8 ft, two or more evaporators should be used. In coolers where there is not sufficient head room to permit the use of overhead coils, side-wall evaporators may be used. If properly installed, these will function with approximately the same efficiency as overhead coils. Typical overhead and side-wall installations for large storage rooms are shown in Figs. 11-22 and 11-23, respectively.

In small coolers, baffles are used with natural convection coils to assure good air circulation. The baffles are installed in such a manner that they aid and direct the free flow of air over the coil and throughout the refrigerated space. The cold and warm air flues should each have an area approximately equal to one-sixth or one-seventh of the floor area of the cooler. Assuming that the flues extend the full length of the fixture, the width of the flues will then be proportional to the width of the cooler. Since

Fig. 11-22. Overhead installation of natural convection evaporator. Evaporator has cast aluminum fins. (Courtesy Detroit Ice Machine Company.)

Fig. 11-23. Sidewall installation of natural convection evaporator. (Courtesy Detroit Ice Machine Company)

warm air has a greater specific volume than cold air, some manufacturers recommend that the warm air flue be a little larger than the cold air flue. In Fig. 11-24, the width of the cold air flue is equal to $W/7$, whereas the width of the warm air flue is equal to $W/6$. The distance (A) from the coil to the ceiling should be approximately equal to the width of the warm air flue and never less than 3 in. Vertical side baffles should extend approximately 1 in. above and 3 to 4 in. below the coil. The horizontal baffles or coil decks should slope 1 to 2 in. per foot to give direction to the cold air flow and to drain off the condensate. Also, the coil decks must be insulated so that moisture does not condense on the undersurface of the deck and drip off on the product. The dimension (B) is 4 to 7 in., whereas (C) is usually 2 to 4 in.

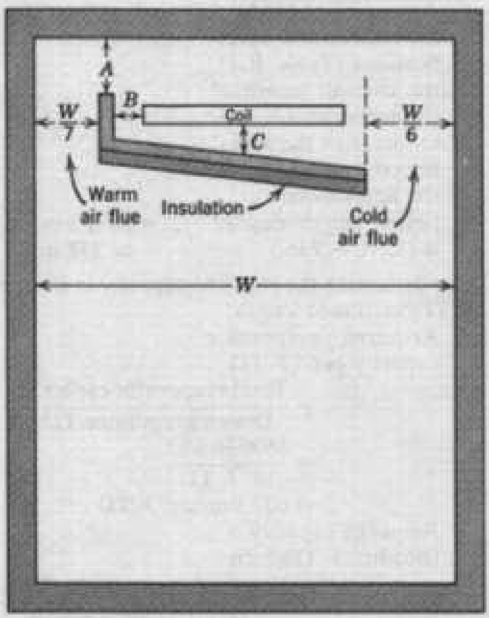


Fig. 11-24. Typical baffle arrangement for natural convection coil.

11-19. Coil-and-Baffle Assemblies. The availability of factory-built coil-and-baffle assemblies has practically eliminated the need for the custom building of baffles on the job. A typical ready built coil-and-baffle assembly is shown in Fig. 11-25. Since these assemblies are available in a wide variety of sizes and combinations (see Table R-1), they can be readily applied to almost any natural convection application.

11-20. Rating and Selection of Natural Convection Evaporators. Basic capacity ratings for natural convection evaporators, both prime surface and finned, are normally given in Btu/hr/°F TD. However, in some instances, where it will simplify evaporator selection, capacity ratings are given for TD's other than 1°F.

For the coil-and-baffle assemblies mentioned in the preceding section, the ratings given are per inch of finned length. For bare pipe evaporators, the ratings given are per square foot of external pipe surface, although in some instances bare pipe evaporators are rated per lineal foot of pipe.

Ratings for plate evaporators are given per square foot of plate surface. Both sides of the plate are considered when computing the area

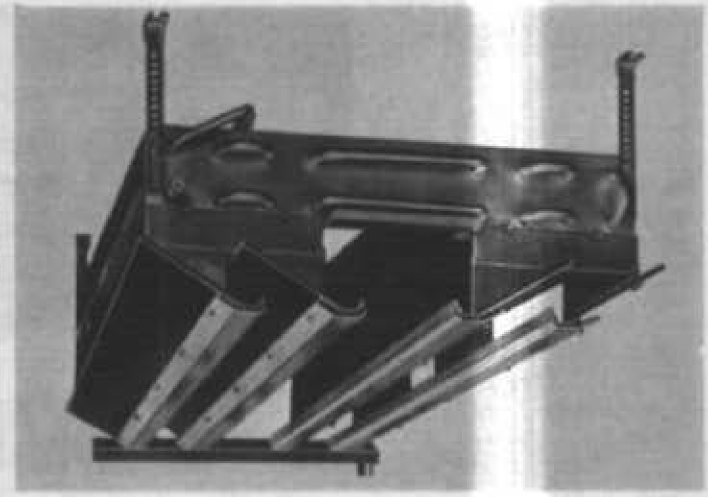


Fig. 11-25. Natural convection coil-and-baffle assembly. (Courtesy Dunham-Bush, Inc.)

of the plate. Frequently, ratings for plate evaporators apply to an entire plate or to a specific group or combination of plates.

Typical rating data for various types of natural convection evaporators are given in Tables R-1 through R-7. The use of these rating data in the selection of the various types of evaporators is best illustrated through the use of a series of examples.

Example 11-1. Select a natural convection coil-and-baffle assembly (Fig. 11-25 and Table R-1) for the vegetable storage cooler in Example 10-11.

Solution. Since the capacity ratings for this type of evaporator are given in Btu/hr/°F TD/in. of finned length, the required evaporator capacity must be reduced to this value before a selection can be made from the rating table. Also, recall that a natural convection evaporator should extend almost the full length of the cooler in order to assure adequate air circulation around the product.

From Example 10-11, inside dimension of cooler	= 17 ft × 9 ft
Required evaporator capacity (average hourly cooling load)	= 8500 Btu/hr
From Table 10-11, desired space humidity for mixed vegetable storage	= Approx. 87%
From Table 11-2, design evaporator TD required for 87% RH	= 14° F

To determine approximate finned length required:

Over-all length of cooler (inside)	= 17 ft
Allowing 1 ft on each end of evaporator for working space, the approximate over-all length of the evaporator is (see Fig. 11-26) (17 ft − 2 ft)	= 15 ft
According to the manufacturer's specifications (Table R-1), the over-all length of the evaporator is 7 in. longer than the actual finned length. Hence, the approximate finned length desired is (15 ft − 7 in.)	= 14 ft 3 in. or 171 in.

To determine the required capacity in Btu/hr/°F TD/in. finned length:

Required evaporator capacity per °F TD

$$= \frac{\text{Total evaporator capacity}}{\text{Design evaporator TD}}$$

$$= \frac{8500 \text{ Btu/hr}}{14° \text{ F TD}}$$

$$= 607 \text{ Btu/hr/° F TD}$$

Required capacity (Btu/hr/° F TD/inch

$$= \frac{\text{Required capacity per °F TD}}{\text{Desired finned length}}$$

$$= \frac{607 \text{ Btu/hr/° F TD}}{171 \text{ in. finned length}}$$

$$= 3.55 \text{ Btu/hr/° F/inch}$$

Because of the width of the cooler, a two-section evaporator will give the best results. Reference to Table R-1 indicates that Model #PK-16 with two fins per inch (½-in. fin spacing) has a capacity of 3.65 Btu/hr/° F/in.

Using this model evaporator, the required finned length is

$$\frac{607 \text{ Btu/hr/° F TD}}{3.65 \text{ Btu/hr/° F/in.}} = 166 \text{ in.}$$

The over-all length of the evaporator is 173 in. (166 in. + 7 in.) and, since the over-all length of the cooler inside is 204 in., the clearance between the evaporator and the cooler wall at each end is 15.5 in.

The width of the evaporator should always be checked against the width of the cooler to be sure that the evaporator can be installed in the space in accordance with the manufacturer's recommendations. For evaporators of this type, the manufacturer recommends that the side of the evaporator be not less than 6 in. nor more than 12 in. from the cooler wall (installation dimension *A* of Table R-1) and that the distance between the two sections of the evaporator (dimension *C*) be not less than 6 in. nor more than 8 ft.

The maximum allowable evaporator width can be determined by subtracting the minimum of dimensions *A* and *C* from the inside width of the cooler, viz:

Maximum width of evaporator = Inside width of cooler $- (A + C + C)$. In this particular case, the maximum allowable width of the evaporator is

$$108 \text{ in.} - (6 \text{ in.} + 6 \text{ in.} + 6 \text{ in.}) = 90 \text{ in.}$$

Since the combined width of the two sections of evaporator, Model #PK-16, is only 36 in.

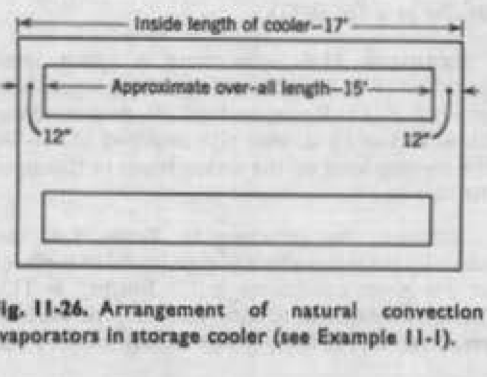


Fig. 11-26. Arrangement of natural convection evaporators in storage cooler (see Example 11-1).

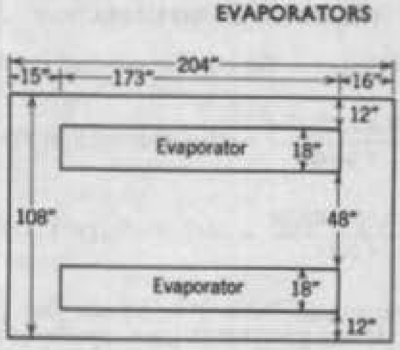


Fig. 11-27. Arrangement of natural convection evaporators in storage cooler (see Example 11-1).

(18 in. × 2), the evaporator is suitable for the cooler. A logical arrangement of the two evaporator sections in the cooler is shown in Fig. 11-27.

To order the evaporator, specify the model number, fin spacing, and finned length, viz: Model #PK-16.3-166 in.

NOTE. To avoid excessively long evaporators, which are inconvenient to ship and install, a multiple of evaporators should be used in large coolers. Typical arrangements for large coolers are shown in Fig. 11-28. These arrangements are also suitable for plate banks.

Example 11-2. Using Table R-3, select plate evaporators (banks) for ceiling installation in the locker room of Example 10-16. To assure good air circulation in the locker room, select evaporators for a 15° F TD.

Solution. Analysis of room dimensions (30.5 ft × 16 ft) indicates that four to six evaporators (two or three banks installed end-to-end over each aisle will be needed to provide good ceiling coverage and adequate air distribution. Reference to Table R-3 will show that plate banks are available in stock lengths of 108 in. (9 ft) and 144 in. (12 ft). Three banks 108 in. long (a total of 27 ft) or two banks 144 in. long (a total of 24 ft) could be installed end-to-end over each aisle and allow adequate working clearance at the ends.

Since the banks are already rated at the design TD of 15° F, the ratings can be used directly and the required capacity per bank can be determined by dividing the total hourly cooling load by the desired number of plate banks, viz:

$$\text{Required capacity per bank (Btu/hr)} = \frac{\text{Total cooling load (Btu/hr)}}{\text{Number of banks desired}}$$

In this instance, the capacity required per bank is

$$\frac{24{,}650 \text{ Btu/hr}}{6 \text{ banks}} = 4108 \text{ Btu/hr/15° F TD}$$

or

$$\frac{24{,}650 \text{ Btu/hr}}{4 \text{ banks}} = 6162 \text{ Btu/hr/15° F TD}$$

Referring to Table R-3, we see that plate bank, Model #5-12108-B, has a capacity of 4320 Btu/hr at a 15° F TD when operating below 32° F (frosted). This will permit good coverage of the ceiling and at the same time allow sufficient working space at the ends of the banks (see Fig. 11-29).

In ordering the evaporators, specify refrigerant and the type of connections desired (series or manifold), viz:

Model #6-12144-B, series connected for Refrigerant-12. Notice (Table R-3) that the manufacturer specifies that two refrigerant flow controls should be used with each bank for Refrigerant-12, whereas only one is needed when ammonia is the refrigerant.

Example 11-3. From Table R-4, select a plate-stand assembly for the freezing cabinet in Example 10-16. The inside dimensions of the cabinet are 28 in. × 90 in. and the freezing load is 4590 Btu/hr. Base plate selection on 10° F TD.

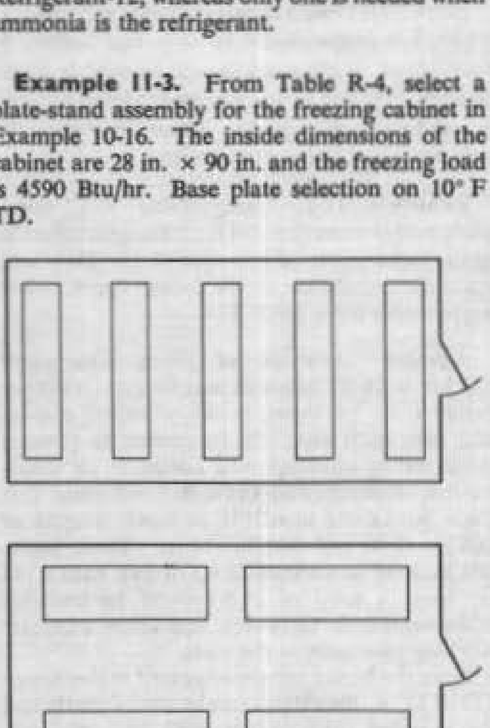


Fig. 11-28. Typical arrangements for natural convection evaporators in large coolers.

Solution. Inspection of Table R-4 will show that the ratings of the plate-stand assemblies are based on a 15° F TD. Since the design TD in this instance is only 10° F, it is necessary to determine the capacity the plate must have at a 15° F TD in order to have the desired design TD of 10° F. This is accomplished by dividing the average hourly load by the design TD of 10° F and then multiplying by the rating TD of 15° F, viz:

$$\frac{4590 \text{ Btu/hr} \times 15° \text{ F}}{10° \text{ F}} = 6885 \text{ Btu/hr}$$

Thus, it is determined that an evaporator having a capacity of 6885 Btu/hr at a 15° F TD will have the desired capacity of 4590 Btu/hr at the design TD of 10° F. This value can then be used to select the plate stand directly from the rating table.

By referring to Table R-4, plate stand, Model #7-2284-S, which is approximately 26 in. wide and 88 in. long (including piping connections), will fit the freezer cabinet and has a capacity of 7140 Btu/hr at a 15° F TD (4760 Btu/hr at 10° F TD). This provides a small safety factor and is therefore satisfactory for the application.

Alternate Solution. A rule of thumb used in selecting plate freezers for locker plant applications is to allow 0.5 sq ft of plate surface for each locker.

Allowing 0.5 sq ft of plate surface per locker per day, the plate surface required in this instance is

$$353 \text{ lockers} \times 0.5 = 176.5 \text{ sq ft}$$

By referring to Table R-2, the plate which best fits the cabinet, Model #2284 (22 in. × 84 in.), has a surface area (both sides of plate) of 27.24 sq ft per plate. Therefore, seven plates of this size are required. Seven plates have a total capacity of 7140 Btu/hr at a 15° F TD or 4760 Btu/hr at a 10° F TD.

Example 11-4. Assuming a space temperature of 0° F and a refrigerant temperature of −17° F (17° F evaporator TD), determine the lineal feet of 1¼ in. iron pipe required to handle the cooling load on the locker room in Example 10-16.

Solution. By referring to Table R-6, the capacity per square foot of pipe (outside surface) at the given conditions is 1.5 Btu/hr/° F TD. To determine the square feet of pipe surface required, divide the average hourly load by the

Fig. 11-29. Installation of plate banks in locker plant (see Example 11-2.)

capacity per square foot per degree TD and by the design TD, viz:

$$\frac{\text{Capacity required (Btu/hr)}}{\text{Pipe capacity (Btu/hr/sq ft/° F)} \times \text{TD}} = \text{Pipe surface (sq ft)}$$

In this instance, the square feet of pipe surface required is

$$\frac{24{,}650 \text{ Btu/hr}}{1.5 \times 17} = 966 \text{ sq ft}$$

By referring to Table R-7, 2.3 lineal ft of $1\frac{1}{4}$ in. pipe equals 1 sq ft of external pipe surface. Hence, the lineal feet of $1\frac{1}{4}$ in. pipe required is

$$966 \times 2.3 = 2220 \text{ ft}$$

11-21. Forced Convection Evaporators. Forced convection evaporators, commonly called "unit coolers" or "blower coils" in commercial refrigeration, are essentially finned coils encased in a metal housing and equipped with one or more fans to provide air circulation. Some typical unit coolers are shown in Fig. 11-30.

The total cooling capacity of any evaporator is directly related to the air quantity (cfm) circulated over the evaporator. The air quantity required for a given evaporator capacity is basically a function of two factors: (1) the sensible heat ratio and (2) the drop in the temperature of the air passing over the evaporator, viz:

$$\text{Cfm} = \frac{\text{Total capacity (Btu/hr)} \times \text{sensible heat ratio}}{\text{Temperature drop of the air} \times 1.08} \tag{11-5}$$

The sensible heat ratio is the ratio of the sensible cooling capacity of the evaporator to the total cooling capacity. When air is cooled below its dew point temperature, both the temperature and the moisture content of the air are reduced (Chapter 5). The temperature reduction is the result of sensible cooling, whereas the moisture removed is the result of latent cooling. Hence, for an evaporator having a total cooling capacity of one ton (12,000 Btu/hr) and a sensible heat ratio of 0.85, the sensible cooling capacity of the evaporator is 10,200 Btu/hr (85% of 12,000), whereas the latent cooling capacity is 1800 Btu/hr (12,000 − 10,200). Naturally, the sensible heat ratio of any evaporator will depend upon the conditions of the application, the design of the evaporator, and the air quantity. An average sensible heat ratio for unit coolers is approximately 0.85.

As a general rule, the air temperature drop through a well-designed unit cooler is approximately one-half the difference between the space temperature and the refrigerant temperature. For example, for an evaporator TD of 10° F, the air temperature drop through the unit cooler should be approximately 5° F.

The constant 1.08 in Equation 11-5 is a conversion factor involving air density (0.75), air specific heat (0.24 Btu/lb/° F), and minutes per hour (60).*

Example 11-5. Determine the approximate quantity of air (cfm) circulated over a unit cooler having a capacity of 20,000 Btu/hr if the sensible heat ratio is 0.85 and the design evaporator TD is 15° F.

Solution. Applying Equation 11-5, the air quantity

$$= \frac{20{,}000 \text{ Btu/hr} \times 0.85}{7.5 \times 1.08} = 2100 \text{ cfm}$$

As a general rule, air velocities across the face of unit coolers are maintained between 300 and

* See Section 14-4.

Fig. 11-30. Typical unit cooler designs. Notice that cooler designs are such that the air is not discharged directly on the stored product. (Courtesy Dunham-Bush, Inc.)

500 ft per minute (fpm). Although higher velocities will result in higher transfer coefficients, they are not usually practical since they also increase fan horsepower requirements. When the fan horsepower is increased beyond a certain point, the additional heat given off by the fan motor resulting from the increase in horsepower will exceed the increase in unit cooler capacity resulting from higher air velocities. Hence, the net effect in such cases is to decrease, rather than increase, the overall capacity of the unit cooler. Too, where the air velocity exceeds 500 fpm there is a tendency for moisture to be blown from the face of the coil into the space and onto the product.

The air velocity (fpm) over the evaporator is a function of the air quantity (cfm) and the face area of the evaporator (sq ft), viz:

$$\text{Velocity (fpm)} = \frac{\text{Air quantity (cfm)}}{\text{Face area (sq ft)}} \quad (11\text{-}6)$$

Example 11-6. Determine the face area of the evaporator in Example 11-5 if the face velocity is to be maintained at 350 fpm.

Solution. By rearranging and applying Equation 11-6, the face area

$$= \frac{2100 \text{ cfm}}{350 \text{ fpm}} = 6 \text{ sq ft}$$

11-22. Rating and Selection of Unit Coolers. Basic ratings for unit coolers are given in Btu/hr/°F TD. For convenience, sometimes ratings are given for 10° F and 15° F TDs. As in the case of natural convection evaporators, the design TD for unit coolers depends primarily on the space humidity requirements. In general, for any given space humidity, the design TD for unit coolers is about 4° F to 6° F less than those used for natural convection evaporators (see Table 11-2).

Since the air quantity is usually fixed by the fan selection at the time of manufacture, realization of the rated capacity will depend primarily upon whether or not the coil is kept reasonably free of frost by adequate defrosting. When the space is maintained below 34° F, some means of automatic defrosting must be used (see Chapter 20).

Example 11-7. From Table R-8, select a forced convection evaporator (unit cooler) suitable for installation in a beer storage cooler having a calculated heat load of 16,600 Btu/hr. Since space humidity is not a factor, use 10° F TD for high system efficiency.

Solution. From Table R-8, select unit cooler Model #UC-180 having a capacity of 18,000 Btu/hr at a 10° F TD. Since the unit cooler fan motor operates inside the refrigerated space, the motor heat becomes a part of the space cooling load and must be added to the load calculations. From Table R-8, the heat given off by the fan motor is 24,000 Btu/24 hr. Since the fan operates continuously to provide air circulation in the refrigerated space, while the average hourly cooling load is based on a 16-hr running time, the average Btu per hour load resulting from the fan motor heat is

$$\frac{24{,}000 \text{ Btu/24 hr}}{16 \text{ hr}} = 1500 \text{ Btu/hr}$$

Hence, when the fan motor heat is considered, the average hourly cooling load for the beer cooler becomes 18,100 Btu/hr (16,600 + 1500). Since the unit cooler selected has a capacity of 18,000 Btu/hr, it will be adequate for the application.

Suggested locations for unit coolers in walk-in refrigerators are shown in Fig. 11-31.

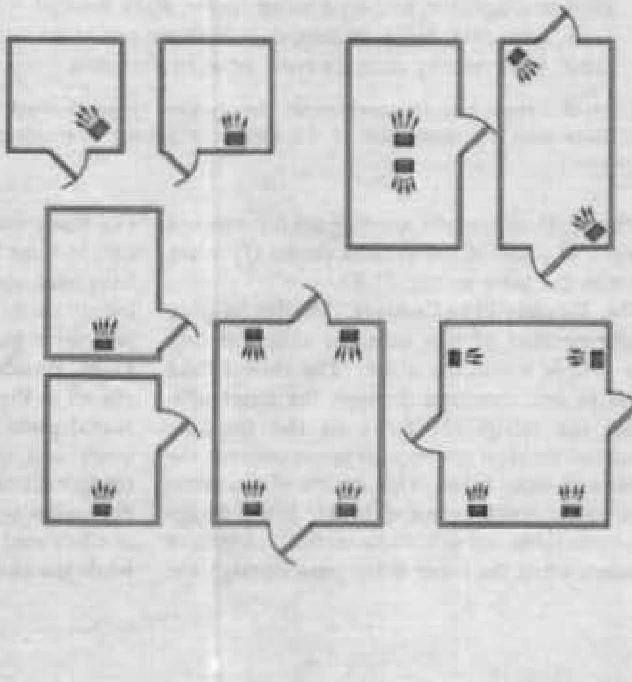


Fig. 11-31. Suggestions for location of unit coolers in walk-in refrigerators. (From the *ASRE Data Book*, Design Volume, 1957–58 edition, reproduced by permission of the American Society of Heating, Refrigerating and Air-Conditioning Engineers.)

11-23. Liquid Chilling Evaporators. As with air-cooling evaporators, liquid chilling evaporators vary in type and design according to the type of duty for which they are intended. Five general types of liquid chillers are in common use: (1) the double-pipe cooler, (2) the Baudelot cooler, (3) the tank-type cooler, (4) the shell-and-coil cooler, and (5) the shell-and-tube cooler. In all cases, the factors which influence the performance of liquid chillers are the same as those which govern the performance of air-cooling evaporators and all other heat transfer surfaces. Heat transfer coefficients for average designs of some of the various chiller types are listed in the table in Fig. 11-32.

Current conservative design values of heat transfer coefficient $\bar{U}$ based on outside surface for bare tube coolers, unless mentioned otherwise, are as follows:

	Min	Max
Flooded shell-and-tube cooler (water to ammonia or R-12)	50	150
Flooded shell-and-finned tube high velocity R-12 water cooler	30	150
Flooded shell-and-tube cooler (brine to ammonia)	45	100
Flooded shell-and-tube cooler (brine to R-12)	30	90
Dry-expansion shell-and-tube cooler, R-12 in tubes, water in shell	50	115
Baudelot cooler, flooded (ammonia or R-12 to water)	100	200
Baudelot cooler, dry-expansion (ammonia to water)	60	150
Baudelot cooler, dry-expansion (R-12 to water)	60	120
Double-pipe cooler (water to ammonia)	50	150
Double-pipe cooler (brine to ammonia)	50	125
Shell-and-coil cooler (water to ammonia)	10	25
Shell-and-coil cooler (water to R-12)	10	25
Spray-type shell-and-tube water coolers (ammonia or R-12)	150	250
Tank-and-agitator, coil-type water cooler, ammonia, flooded	80	125
Tank-and-agitator, coil-type water cooler, R-12 flooded	60	100
Tank, ammonia, brine cooling, coils between can in ice tank	15	40
Tank, high velocity raceway type, brine to ammonia	80	110

Fig. 11-32. Heat transfer coefficients for various types of liquid chillers. (Reprinted from 1955–56 *ASRE Data Book*, by permission of the American Society of Heating, Refrigerating, and Air-Conditioning Engineers.)

11-24. Double-Pipe Coolers. The double-pipe cooler consists of two tubes so arranged that one tube is inside the other. The chilled fluid flows in one direction through the inner tube while the refrigerant flows in the opposite direction through the annular space between the inner and outer tubes. One design of a double-pipe cooler is shown in Fig. 11-33. In this design the outer tubes are welded to vertical refrigerant headers while the inner tubes pass through the headers and are connected together by removable return bends (inset). The advantages claimed for this unit are rigid construction, the elimination of refrigerant joints, and easy accessibility of the inner tubes for cleaning.

Double-pipe coolers may be operated either dry-expansion or flooded. In either case, counterflowing of the fluids in the tubes produces a relatively high heat transfer coefficient. However, this type of cooler has the disadvantage of requiring more space, particularly head room, than some of the other cooler designs. For this reason, the double-pipe cooler is used only in some few special applications. A number have been used in the wine-making and brewing industries to chill wine and wort, and in the petroleum industry for the chilling of oils.

11-25. Baudelot Coolers. The Baudelot cooler shown in Fig. 11-34 consists of a series of horizontal pipes which are located one under the other and are connected together to form a refrigerant circuit or circuits. For either dry-expansion or flooded operation, the refrigerant is circulated through the inside of the tubes while the chilled liquid flows in a thin film over

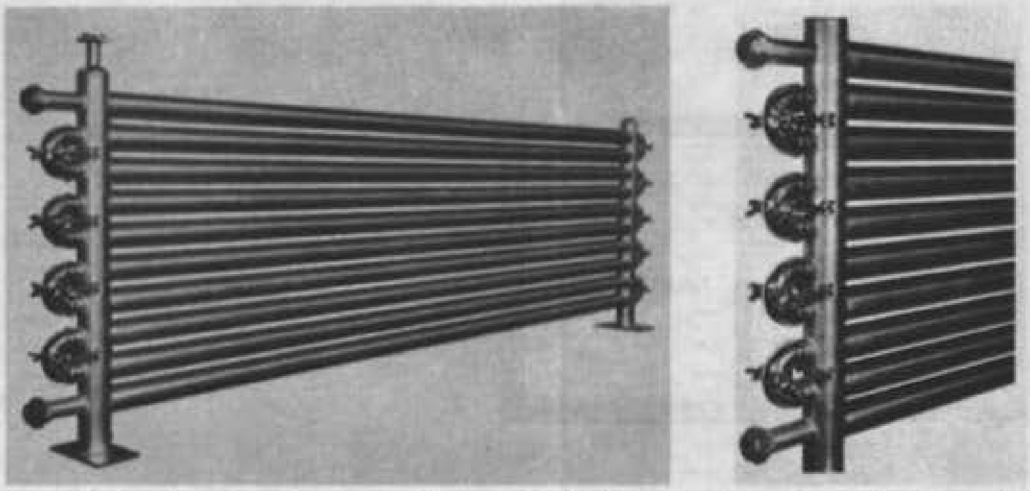


Fig. 11-33. Double pipe cooler. Removable return bends (right) are designed to make tube readily accessible for cleaning. (Courtesy Vilter Manufacturing Company.)

the outside. The liquid flows down over the tubes by gravity from a distributor located at the top of the cooler and is collected in a trough at the bottom. The fact that the chilled liquid is at atmospheric pressure and is open to the air makes the Baudelot cooler ideal for any liquid chilling application where aeration is a factor. The Baudelot chiller has been widely used for the cooling of milk, wine, and wort, and for the chilling of water for carbonation in bottling plants. With this particular type of chiller it is possible to chill liquid to a temperature very close to the freezing point without the danger of damaging the equipment if occasional freezing of the liquid occurs.

Another advantage of the Baudelot cooler, and one which is shared by the double-pipe cooler, is that the refrigerant circuit is readily

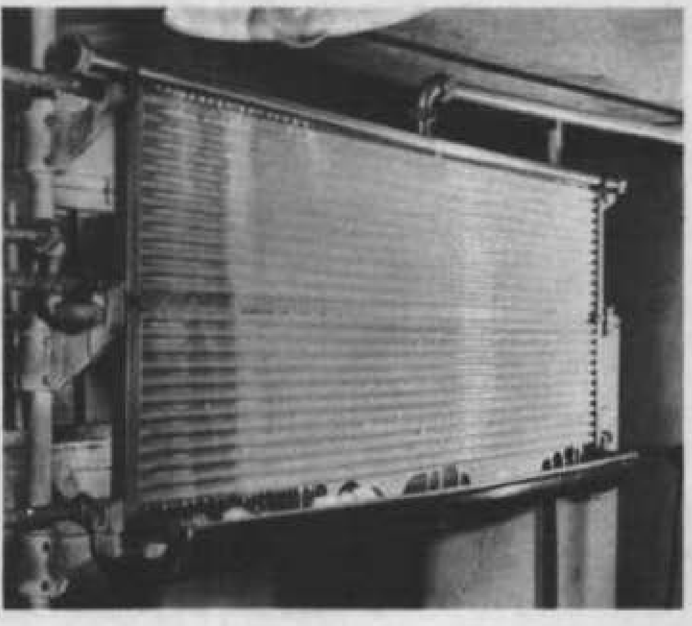


Fig. 11-34. Baudelot cooler employed in milk-cooling application. (Courtesy Dole Refrigerating Company.)

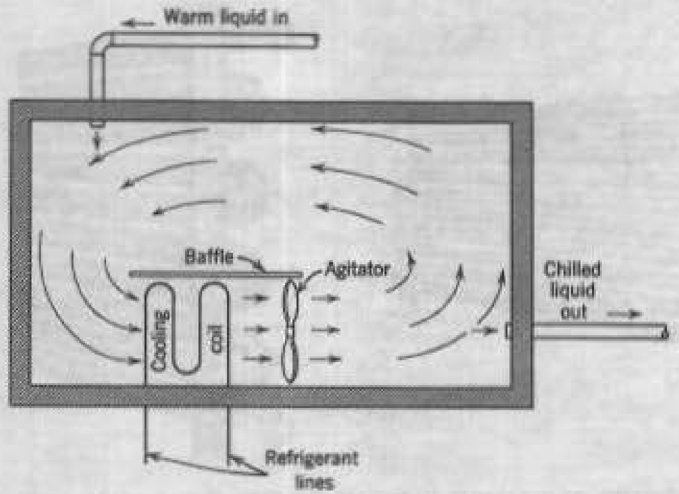


Fig. 11-35. Typical construction of tank-type liquid cooler.

split into several parts, a circumstance which permits precooling of the chilled liquid with cold water before the liquid enters the direct-expansion portion of the cooler (see Fig. 17-34).

11-26. Tank-Type Coolers. The tank-type liquid chiller consists essentially of a bare-tube refrigerant coil installed in the center or at one side of a large steel tank which contains the chilled liquid. Although completely submerged in the chilled liquid, the refrigerant coil is separated from the main body of the liquid by a baffle arrangement. As shown in Fig. 11-35, a motor driven agitator is utilized to circulate the chilled liquid over the cooling coil at relatively high velocity, usually between 100 and 150 ft per minute, the liquid being drawn in at one end of the coil compartment and discharged at the other end.

The spiral-shaped, bare-tube coils mentioned in Section 11-4 and the raceway-type coil illustrated in Fig. 11-36 are two coil designs frequently employed in tank-type chillers. With

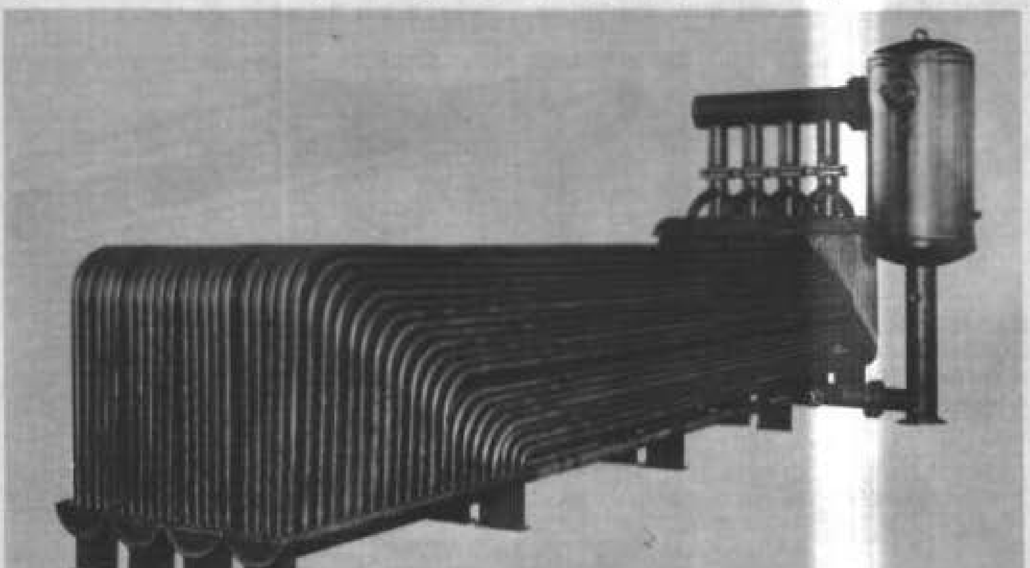


Fig. 11-36. Flooded raceway coil. (Courtesy Vilter Manufacturing Company.)

either design the coils are operated flooded. The Ice-Cel shown in Fig. 11-11 is another variation of the tank-type chiller.

Tank-type chillers can be applied to any liquid-chilling application where sanitation is not a primary factor, and are widely used for the chilling of water, brine, and other liquids to be used as secondary refrigerants. Because of their inherent holdover capacity, they are particularly suitable for applications subject to frequent and severe fluctuations in loading. In such cases, a comparatively large chilled-liquid storage tank is provided in order to minimize the rise in the temperature of the chilled liquid during periods of peak demand. The advantage gained by precooling is often considerable in cases where the liquid to be chilled enters the cooler at relatively high temperatures.

11-27. Shell-and-Coil Coolers. The shell-and-coil chiller is usually made up of one or more spiral-shaped, bare-tube coils enclosed in a welded steel shell (Fig. 11-37). As a general rule, the chiller is operated dry-expansion with the refrigerant in the coils and the chilled liquid in the shell. In a few cases, the chiller is operated flooded, in which case the refrigerant is in the shell and the chilled liquid passes through the tubes. The former arrangement has the advantage of providing a holdover capacity, thereby making this type of chiller ideal for small applications having high but infrequent peak loads. It is used primarily for the chilling of water for drinking and for other purposes where sanitation is a prime factor, as in bakeries and photographic laboratories.

When operated flooded with the refrigerant in the shell, this type of chiller becomes what is commonly referred to as an "instantaneous" liquid chiller. One of the disadvantages of this arrangement is that there is no holdover capacity. Since the liquid is not recirculated, it must be chilled instantaneously as it passes

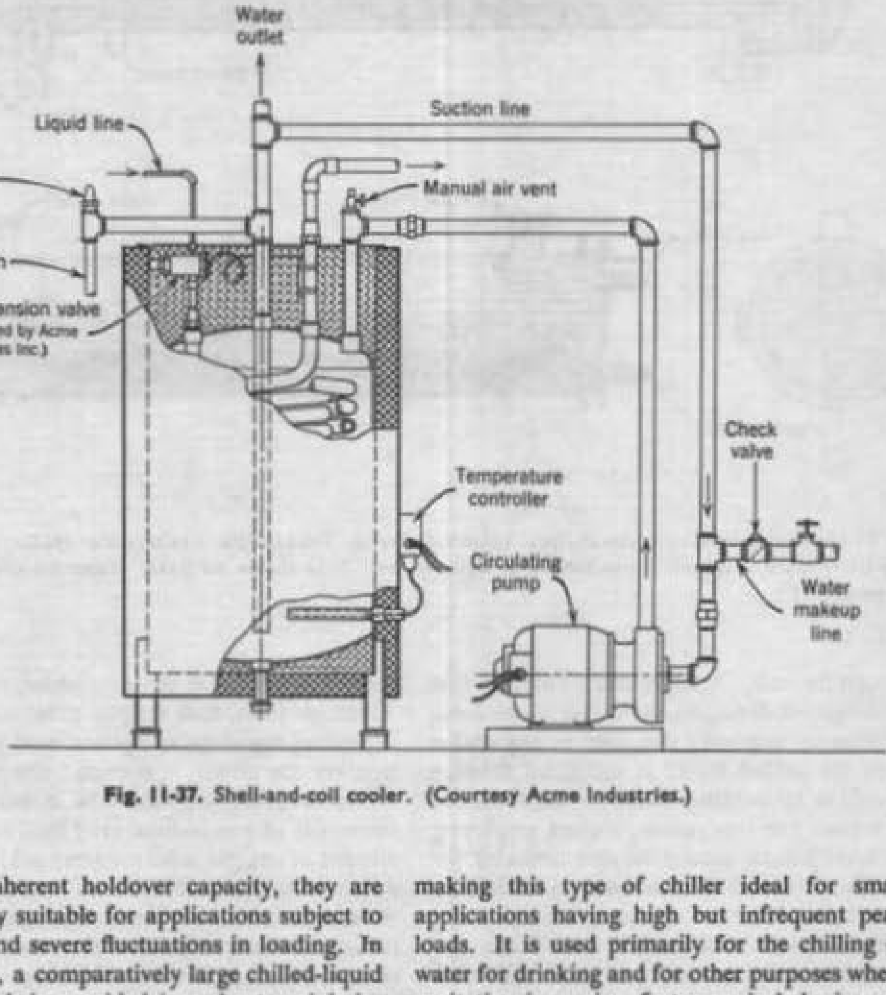


Fig. 11-37. Shell-and-coil cooler. (Courtesy Acme Industries.)

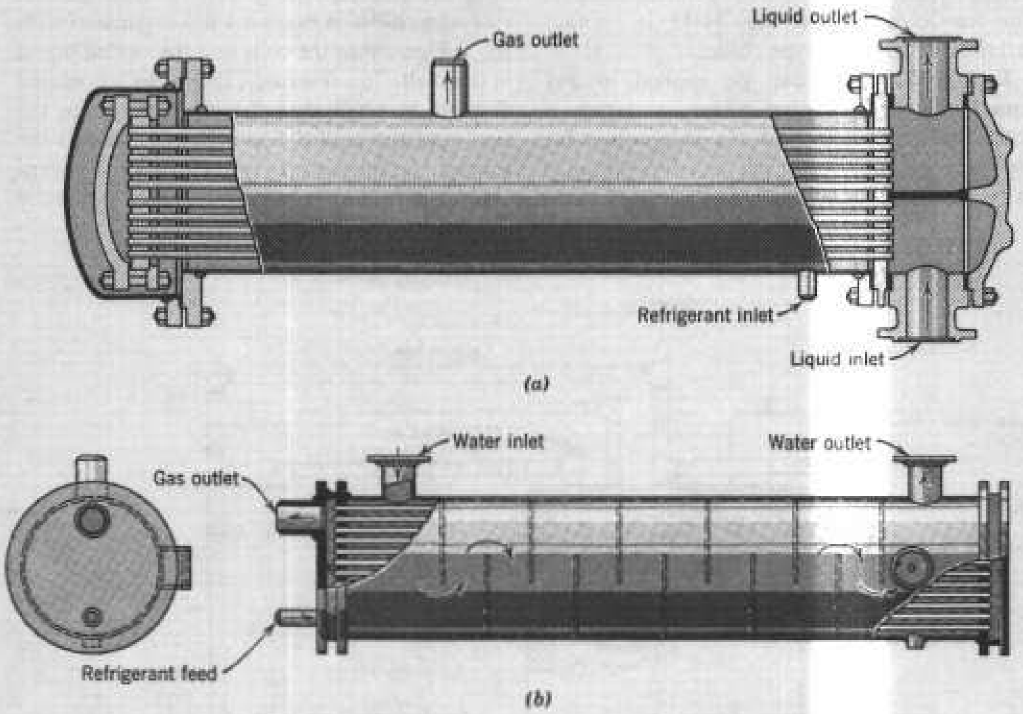


Fig. 11-38. Typical shell-and-tube chillers. (*a*) Hooded type. Tube bundle is removable. (*b*) Dry-expansion type (refrigerant in tubes). Note baffling of water circuit. Tube sheets are fixed. (Courtesy Worthington Corporation.)

through the coils. Another disadvantage is that the danger of damaging the chiller in the event of freeze-up is greatly increased in any chiller where the chilled liquid is circulated through the coils or tubes rather than over the outside of the tubes. For this reason, chillers employing this arrangement cannot be recommended for any application where it is required to chill the liquid below 38° F.

Instantaneous shell-and-coil chillers are used principally for the chilling of beer and other beverages in "draw-bars," in which case the beverage is usually precooled to some extent before entering the chiller.

11-28. Shell-and-Tube Chillers. Shell-and-tube chillers have a relatively high efficiency, require a minimum of floor space and head room, are easily maintained, and are readily adaptable to almost any type of liquid-chilling application. For these reasons, the shell-and-tube chiller is by far the most widely used type. Although individual designs differ somewhat, depending upon the refrigerant used and upon whether the chiller is operated dry-expansion or flooded, the shell-and-tube chiller consists essentially of a cylindrical steel shell in which a number of straight tubes are arranged in parallel and held in place at the ends by tube sheets. When the chiller is operated dry-expansion, the refrigerant is expanded into the tubes while the chilled liquid is circulated through the shell (Fig. 11-38*b*). When the chiller is operated flooded, the chilled liquid is circulated through the tubes and the refrigerant is contained in the shell, the level of the liquid refrigerant in the shell being maintained with some type of float control (Fig. 11-38*a*). In either case, the chilled liquid is circulated through the chiller and connecting piping by means of a liquid circulating pump, usually of the centrifugal type.

Shell diameters for shell-and-tube chillers range from approximately 6 to 60 in., and the number of tubes in the shell varies from fewer than 50 to several thousand. Tube diameters range from $\frac{5}{8}$ in. through 2 in., and tube lengths vary from 5 to 20 ft. Chillers designed for use with ammonia are equipped with steel tubes, whereas those intended for use with other refrigerants are usually equipped with copper tubes in order to obtain a higher heat transfer coefficient. Because of the relatively low film conductance of halocarbon refrigerants, chillers designed for use with these refrigerants are often equipped with tubes which are finned on the refrigerant side. In the case of dry-expansion chillers, the tubes are finned internally with longitudinal fins of the types shown in Fig. 11-12. For flooded operation, the tubes are finned externally using a very short fin which protrudes from the tube wall only approximately $\frac{1}{16}$th of an inch.

As a general rule, dry-expansion chillers are employed in small and medium tonnage installations requiring capacities ranging from 2 to approximately 250 tons, but are available in larger capacities. Flooded chillers, available in capacities ranging from approximately 10 through several thousand tons, are more frequently applied in the larger tonnage installations.

11-29. Dry-Expansion Chillers. The principal advantages of the dry-expansion chiller over the flooded type are the smaller refrigerant charge required and the assurance of positive oil return to the compressor. Too, as previously stated, the possibility of damage to the chiller in the event of freeze-up is always considerably less when the chilled liquid is circulated over the tubes rather than through them. The more important construction details of several designs of dry-expansion chillers are shown in Figs. 11-39 and 11-40.

In order to maintain the liquid velocity within the limits which will produce the most effective heat transfer-pressure drop ratio, the velocity of the chilled liquid circulated over the tubes is controlled by varying the length and spacing of the segmental baffles. When the flow rate and/or liquid viscosity is high, short, widely spaced baffles are used to reduce the velocity and minimize the pressure drop through the chiller. When the flow rate and/or liquid viscosity is low, longer, more closely spaced baffles are used in order to increase fluid velocity and improve the heat transfer coefficient (Fig. 11-41*a*).

The number and the length of the refrigerant circuits required to maintain the refrigerant velocity through the chiller tubes within reasonable limits depend on the total chiller load and on the relationship of the chilled liquid flow rate to the METD. Since these factors vary with the individual application, it follows that the optimum refrigerant circuit design also

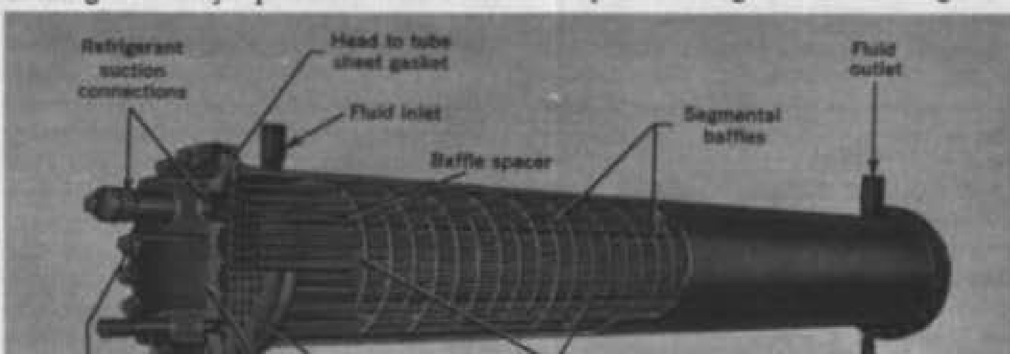


Fig. 11-39. Cutaway section illustrating construction details of dry-expansion chiller with fixed tube sheets. (Courtesy Acme Industries.)

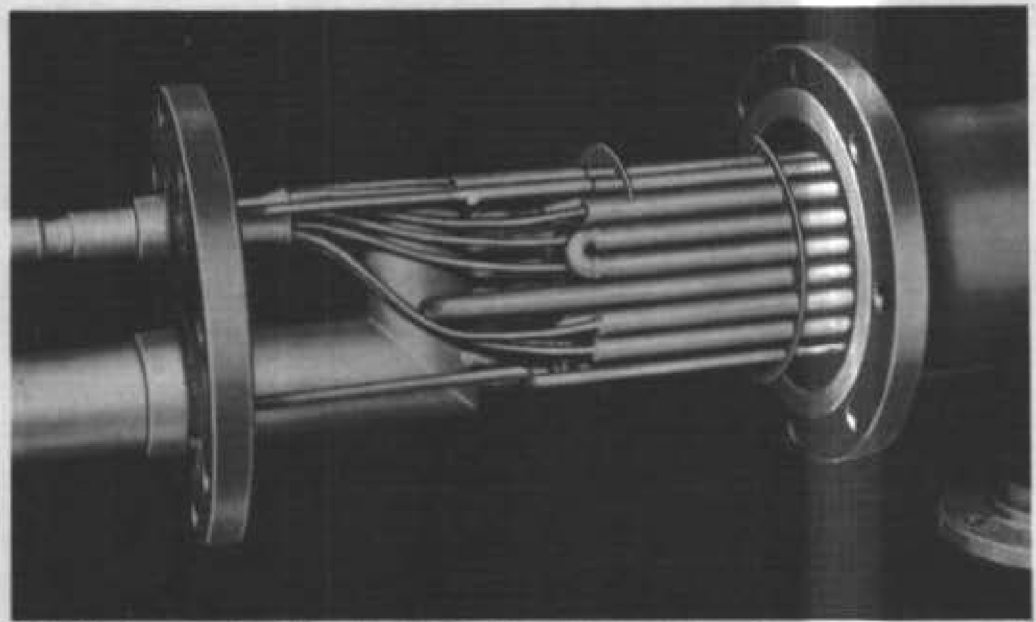


Fig. 11-40. Dry-expansion chiller with tube bundle partially removed to show tube arrangement and refrigerant distributors. Tube bundle can be removed as a unit. (Courtesy Kennard Division, American Air Filter Company, Inc.)

varies with the individual application. For this reason, chillers are made available with either single or multiple refrigerant circuits of varying lengths. For the design shown in Fig. 11-39, the number and length of the refrigerant circuits depend on the tube length and on the arrangement of the baffling in the end-plates or refrigerant heads which are bolted to the tube sheets at the ends of the chiller. The refrigerant circuit arrangement for any one model chiller can be changed by changing the refrigerant heads (Fig. 11-41*b*).

11-30. Flooded Chillers. Standard flooded chiller designs include both single and multipass tube arrangements. For single pass flow, the tubes are so arranged that the chilled liquid passes through all the tubes simultaneously and in only one direction.

Multipass circulation of the chilled liquid through the chiller is accomplished through the use of baffled end-plates or heads which are bolted to the ends of the chiller (Fig. 11-42). The arrangement of the end-plate baffling determines the number of passes the chilled liquid makes from one to the other before leaving the chiller. Although two, four, and six-pass arrangements are the most common, more passes are used in many instances.

As in the case of the dry-expansion chiller, some flooded chillers are designed with removable tube bundles, whereas others have fixed tube sheets. In the fixed tube sheet design, the tube sheets are welded to the shell so that the tube bundle is not removable. However, by unbolting the end-plates the tubes become readily accessible for cleaning and individual tubes can be removed and replaced if necessary. The chillers shown in Figs. 11-38*b* and 11-39 employ fixed tube sheets, whereas those in Figs. 11-38*a* and 11-40 have tube bundles.

In some flooded chiller designs, the shell is only partially filled with tubes in order to provide a large vapor-disengaging area and relatively low velocity in the space above the tubes (Fig. 11-43). This design eliminates the possibility of liquid carry-over into the suction line and therefore is particularly well suited to sudden heavy increases in loading.

In those chiller designs where the shell is completely filled with tubes, a surge drum or accumulator should be used to separate any entrained liquid from the vapor before the vapor enters the suction line. Some flooded chillers are equipped with built-in liquid-suction heat exchangers (Fig. 11-42). Although the primary function of the heat exchanger is to insure that only dry vapor enters the suction line, it has the additional effect of increasing the efficiency of the chiller in that it subcools the liquid approaching the chiller and thereby reduces the amount of flash gas that enters the cooler.

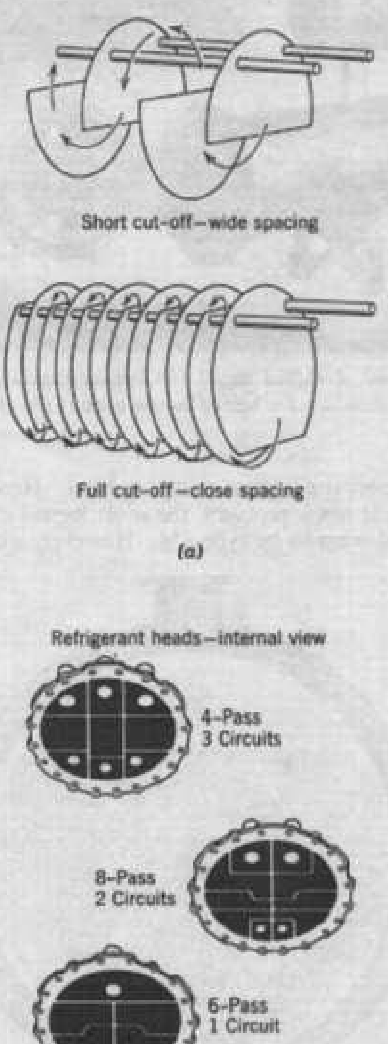


Fig. 11-41. (a) Baffle spacing in dry-expansion chillers. (b) Typical refrigerant heads for dry-expansion chiller. (Courtesy of Acme Industries.)

The vertical shell-and-tube chiller shown in Fig. 11-44 has the advantage of requiring a minimum amount of floor space. The chiller is operated flooded. The chilled liquid enters the chiller at the top and flows by gravity down the inside of the tubes. A circulating pump draws chilled liquid from the storage tank at the bottom and delivers it through the connecting piping. The return liquid is piped to the distributor box at the top, from where it again flows through the tubes. A specially designed distributor installed at the top of each tube (inset) imparts a swirling motion to the chilled liquid, which causes the liquid to flow in a comparatively thin film down the inside tube surfaces.

11-31. Spray-Type Chillers. The spray-type chiller is similar in construction to the conventional flooded chiller except that the liquid refrigerant is sprayed over the outside of the water tubes from nozzles located in a spray header above the tube bundle (Fig. 20-19). The unevaporated liquid drains from the tube into a sump at the bottom of the chiller from where it is recirculated to the spray nozzles by a low head liquid pump. A high recirculation rate assures continuous wetting of the tube surfaces and results in a high rate of heat transfer.

The principal advantages of this type of chiller is its high efficiency and relatively small refrigerant charge. Disadvantages are the high installation cost and the need for a liquid recirculating pump.

11-32. Chiller Selection Procedure. Although selection methods differ somewhat depending upon the type of chiller and the particular manufacturer, all are based on the simple fundamentals of heat transfer and fluid flow which have already been described. Almost without exception, manufacturers include sample selection procedure along with the design and capacity data in their equipment catalogs. The following selection procedure follows very closely that given in the catalog of one manufacturer for the selection of dry-expansion chillers.*

Example 11-8. It is desired to cool 50 gpm of water from 54° F to 46° F with a refrigerant temperature as measured at the cooler outlet of 40° F using Refrigerant-12.

* Acme Industries, Inc.

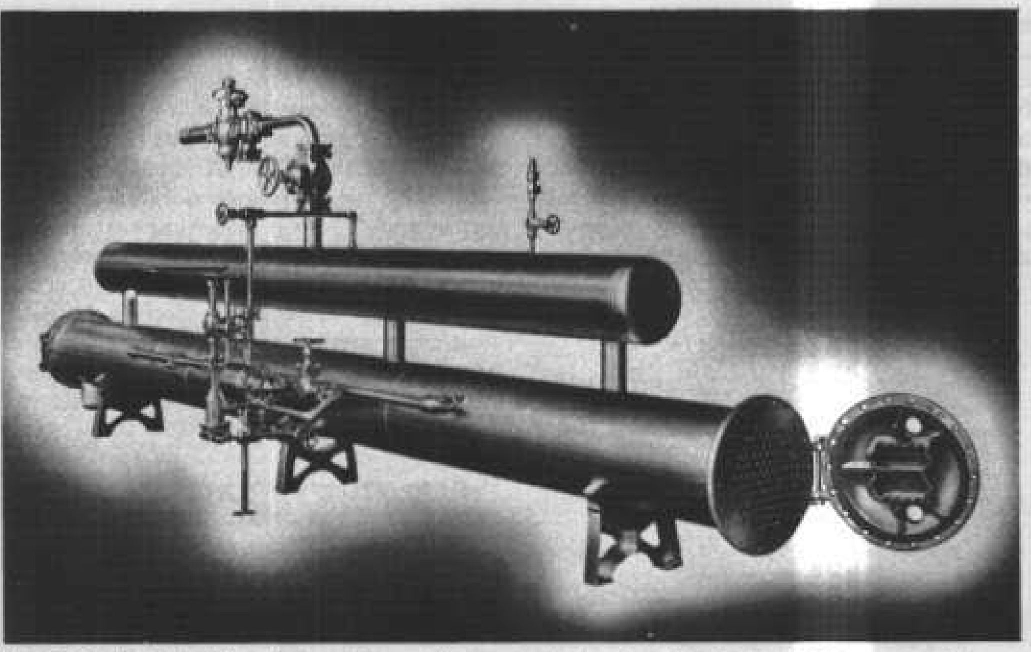


Fig. 11-42. Flooded chiller designed for multipass circulation of chilled liquid. Multipass circulation is accomplished by means of the baffled end-plates or water heads which are bolted to the ends of the chiller. (Courtesy Vilter Manufacturing Company.)

Solution

Step 1. Determine the total chiller load in tons.

$$\frac{\text{Gpm} \times 500 \times \text{cooling range}}{12{,}000 \text{ Btu/hr/ton}}$$

$$\frac{50 \times 500 \times (54 - 46)}{12{,}000} = 16.7 \text{ tons}$$

Step 2. Determine the mean effective temperature difference (METD).

Water in minus refrigerant temperature $54 - 40 = 14°$ F LTD

Water out minus refrigerant temperature $46 - 40 = 6°$ F STD

From Table 11-1, METD $= 9.47°$ F

Step 3. Select trial chiller (shell diameter and baffles spacing) from Fig. 1 of Table R-9. Enter Fig. 1 at 50 gpm on the lower vertical scale and move horizontally across the chart to the diagonal line representing the type unit desired. The number indicates shell diameter and the letter indicates baffle spacing. Possible choices are 10M, 12L, 8M, 12K, 10K, and 8L. As a general rule, small diameter chillers are more economical, whereas large diameter chillers are more compact. Type M baffling produces the lowest pumping head, whereas type L baffling produces the highest pumping head. Hence, if space is not a problem, the most logical choice would seem to be type 8M. However, a check

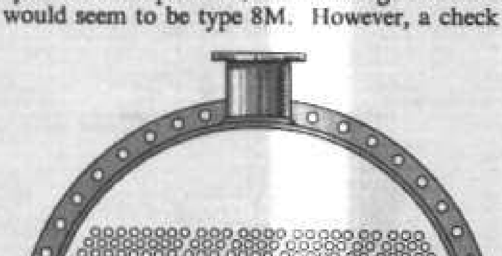


Fig. 11-43. Flooded chiller with shell only partially filled with tubes in order to provide a large vapor disengaging area above the tubes. (Courtesy Worthington Corporation.)

will show that neither 8M nor 8L is available with sufficient surface area in this instance. Therefore, select type 10M (8 to 30 tons). From the point of intersection move vertically upward to a diagonal line in the upper portion of Fig. 1 which represents a METD of 9.47° F as found in Step 2. From this intersection move horizontally to the scale at the left margin and read the loading of 1110 Btu/hr/sq ft (loading is the U value times the METD).

Step 4. Determine the surface area required.

$$\text{Surface area} = \frac{\text{Capacity (Btu/hr)}}{\text{Loading}}$$

$$\frac{200{,}000}{1110} = 180.2 \text{ sq ft}$$

Step 5. Select chiller length from Table R-9 to meet surface area requirements. A 10 in. × 14 ft DXH chiller has a surface area of 184 sq ft (Model No. DXH-1014).

Step 6. Determine the water pressure drop through the chiller. From the bottom of Fig. 1, Table R-9, the pressure drop per foot of length with type M baffling is 0.425 ft of water column.

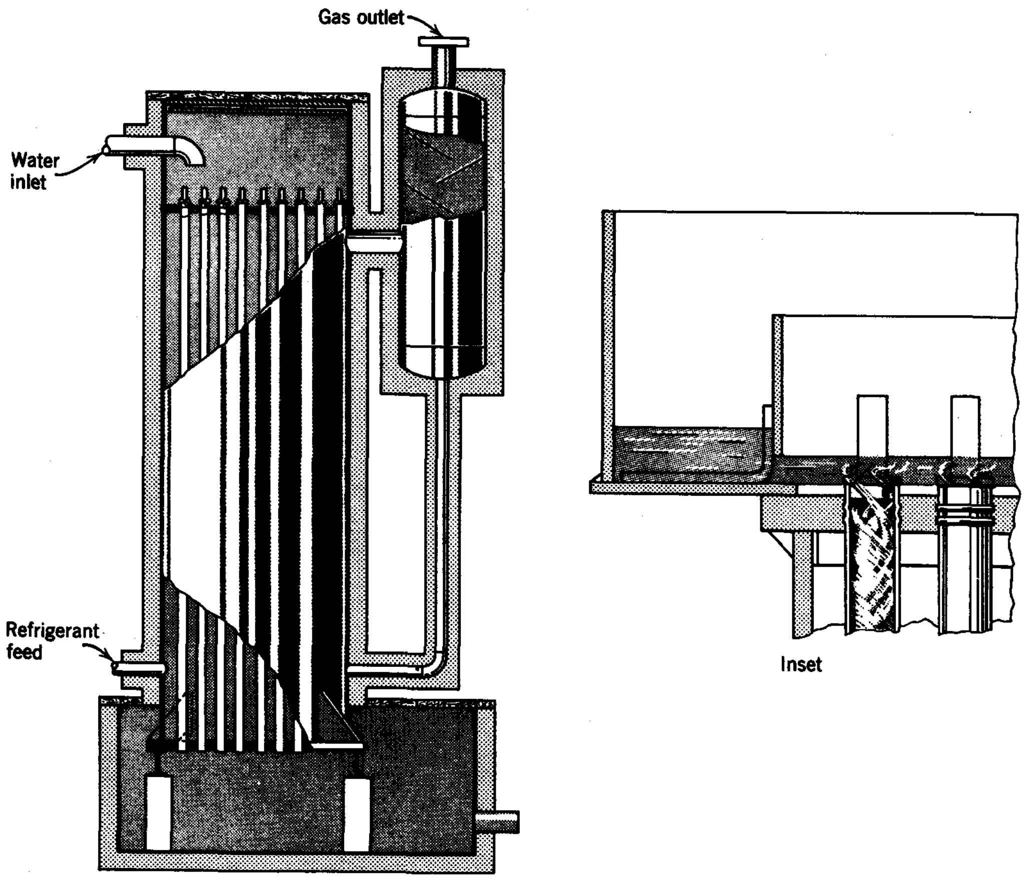


Fig. 11-44. Vertical shell-and-tube "Spira-Flo" chiller designed for flooded operation. The water flowing down through the tubes is given a swirling action by specially designed nozzles (inset). (Courtesy Worthington Corporation.)

Pressure drop = length (feet) × pressure drop/foot

14 feet × 0.425 = 5.95 ft H_2O

11-33. Direct and Indirect Systems. Any heat transfer surface into which a volatile liquid (refrigerant) is expanded and evaporated in order to produce a cooling effect is called a "direct-expansion" evaporator and the liquid so evaporated is called a "direct-expansion" refrigerant. A direct-expansion or "direct"

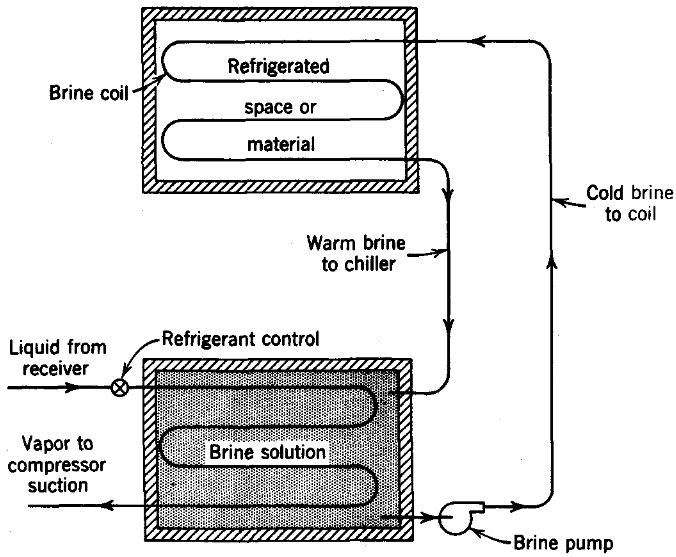


Fig. 11-45a. Indirect system.

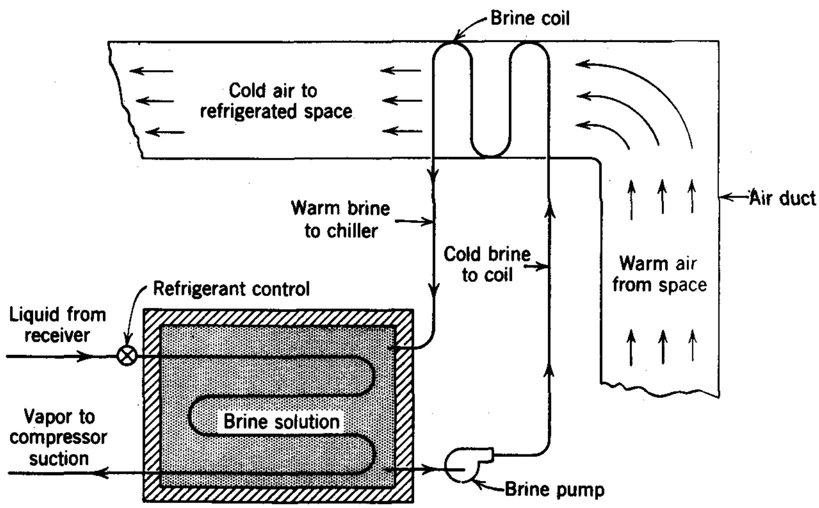


Fig. 11-45b. Indirect system—brine coil in communicating duct.

refrigerating system is one wherein the system evaporator, employing a direct-expansion refrigerant, is in direct contact with the space or material being refrigerated, or is located in air ducts communicating with such spaces. Up to this point, only direct refrigerating systems have been considered.

Very often it is either inconvenient or uneconomical to circulate a direct-expansion refrigerant to the area or areas where the cooling is required. In such cases, an indirect refrigerating system is employed. Water or brine (or some other suitable liquid) is chilled by a direct-expansion refrigerant in a liquid chiller and then pumped through appropriate piping to the space or product being refrigerated. The chilled liquid, called a secondary refrigerant, may be circulated directly around the refrigerated product or vessel or it may be passed through an air-cooling coil or some other type of heat transfer surface (Fig. 11-45). In either case, the secondary refrigerant, warmed by the absorption of heat from the refrigerated space or product, is returned to the chiller to be chilled and recirculated.

Indirect refrigerating systems are usually employed to an advantage in any installation where the space or product to be cooled is located a considerable distance from the condensing equipment. The reason for this is that long direct-expansion refrigerant lines are seldom practical. In the first place, they are expensive to install and they necessitate a large refrigerant charge. Too, long refrigerant lines, particularly long risers, create oil return problems and cause excessive refrigerant pressure losses which tend to reauce the capacity and efficiency of the system. Furthermore, leaks are more serious and are much more likely to occur in refrigerant piping than in water or brine piping.

Indirect refrigeration is required also in many industrial process cooling applications where it is often impractical to maintain a vapor tight seal around the product or vessel being cooled. Too, indirect systems are used to an advantage in any application where the leakage of refrigerant and/or oil from the lines may cause contamination or other damage to a stored product. The latter applies particularly to meat packing plants and large cold storage applications when ammonia is used as a refrigerant.

11-34. Secondary Refrigerants. Some commonly used secondary refrigerants are water, calcium chloride and sodium chloride brines, ethylene and propylene glycols, Methanol (methyl alcohol), and glycerin.

Almost without exception, water is used as the secondary refrigerant in large air conditioning systems and also in industrial process cooling installations where the temperatures maintained are above the freezing point of water. Water, because of its fluidity, high specific heat value, and high film coefficient, is an excellent secondary refrigerant. It also has the advantage of being inexpensive and relatively noncorrosive. In air conditioning applications, the chilled water is circulated through an air cooling coil or through a water spray unit. In either case, the air is both cooled and dehumidified. In the water spray unit, the water i, sprayed from nozzles and collected in a pan or basin at the bottom of the spray unit, from where it is returned to the chiller. Since the air passing through the water spray is chilled below its dew point temperature, a certain amount of water vapor is condensed from the air and is carried to the basin with the spray water. With either the cooling coil or the spray unit, the amount of cooling and dehumidification can be controlled by varying the amount and temperature of the chilled water.

Water is also used frequently as a secondary refrigerant in small beverage coolers and in farm coolers designed for cooling milk cans. In such cases, the water, because of its high conductivity, permits more rapid chilling of the product than would be possible with air. Too, the water supplies a holdover capacity which tends to level out load fluctuations resulting from intermittent loading of the cooler.

11-35. Brines. Obviously, water cannot be employed as a secondary refrigerant in any application where the temperature to be maintained is below the freezing point of water. In such cases, a brine solution is often used.

Brine is the name given to the solution which results when various salts are dissolved in water. If a salt is dissolved in water, the freezing temperature of the resulting brine will be below the freezing temperature of pure water. Up to a certain point, the more salt dissolved in the solution, the lower will be the freezing temperature of the brine. However, if the salt concentration is increased beyond a certain point, the freezing temperature of the brine will be raised rather than lowered. Hence, a solution of any salt in water has a certain concentration at which the freezing point of the solution is lowest. A solution at the critical concentration is called a eutectic solution. At any concentration above or below this critical concentration, the freezing temperature of the solution will be higher, that is, above the eutectic temperature.* When the salt content of the brine is less than that which is required for a eutectic solution, the excess water will begin to precipitate from the solution in the form of ice crystals at some temperature above the eutectic temperature. The exact temperature at which the ice crystals will begin to

* At any temperature other than the eutectic temperature, the term "freezing temperature" is used to mean the temperature at which ice or salt crystals begin to precipitate from the solution.

form depends upon the degree of the salt concentration and upon the relative solubility of the salt in water, the latter factor decreasing as the temperature of the solution decreases. The continued precipitation of ice crystals from the solution as the temperature is reduced causes a progressive increase in the concentration of the remaining brine until, at the eutectic temperature, a slush consisting of ice and eutectic brine will exist. The further removal of heat from this mixture will result in solidification of the eutectic brine. Solidification of the eutectic brine will take place at a constant temperature.

On the other hand, when the salt content of the brine is in excess of the amount required for a eutectic solution, the excess salt will begin to precipitate from the solution in the form of salt crystals at some temperature above the eutectic temperature. Continued precipitation of salt from the mixture as the temperature is reduced will result in a mixture of salt and eutectic brine when the eutectic temperature is reached. The further removal of heat from the mixture will result in solidification of the eutectic brine at constant temperature.

Two types of brine are commonly used in refrigeration practice: (1) calcium chloride and

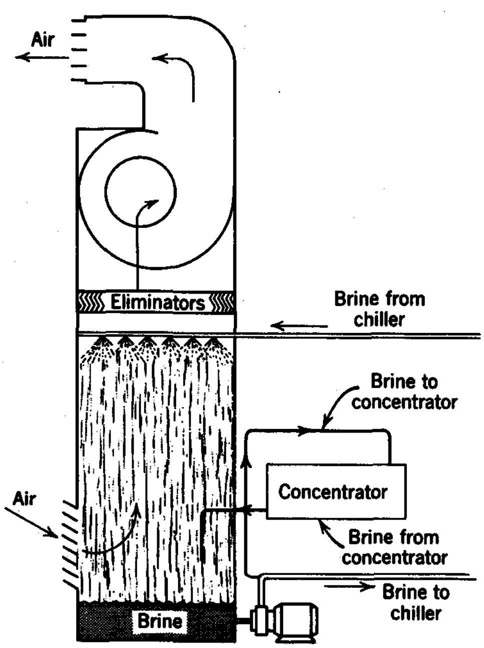


Fig. 11-46. Brine spray cooler.

(2) sodium chloride. The two brines are prepared from calcium chloride ($CaCl_2$) and sodium chloride (NaCl) salts, respectively, the latter salt being the common table variety.

Calcium chloride brine is used primarily in industrial process cooling, in product freezing and storage, and in other brine applications where temperatures below 0° F are required. The lowest freezing temperature which can be obtained with calcium chloride brine (the eutectic temperature) is approximately −67° F. The salt concentration in the eutectic solution is approximately 30% by weight. The freezing temperature of various concentrations of calcium chloride brine are given in Table 11-3, along with some of the other important properties of the brine.

The principal disadvantage of calcium chloride brine is its dehydrating effect and its tendency to impart a bitter taste to food products with which it comes in contact. For this reason, when calcium chloride brine is used in food freezing applications, the system must be designed so as to prevent the brine from coming into contact with the refrigerated product.

Sodium chloride brine is employed mainly in those applications where the possibility of product contamination prevents the use of calcium chloride brine. Sodium chloride brine is employed extensively in installations where the chilling and freezing of meat, fish, and other products are accomplished by means of a brine spray or fog.

The lowest temperature obtainable with sodium chloride brine is approximately −6° F. For this freezing temperature the salt concentration in the solution is approximately 23%. The thermal properties of sodium chloride brine at various concentrations is given in Table 11-4.

It is of interest to notice that the thermal properties of both calcium chloride and sodium chloride brines are somewhat less satisfactory than those of water. As the salt content of the brines is increased, the fluidity, specific heat value, and thermal conductance of the brines all decrease. Hence, the stronger the brine solution, the greater the quantity of brine that must be circulated in order to produce a given refrigerating effect.

Since the specific gravity of the brine increases as the salt concentration increases, the degree of salt concentration and the thermal properties of

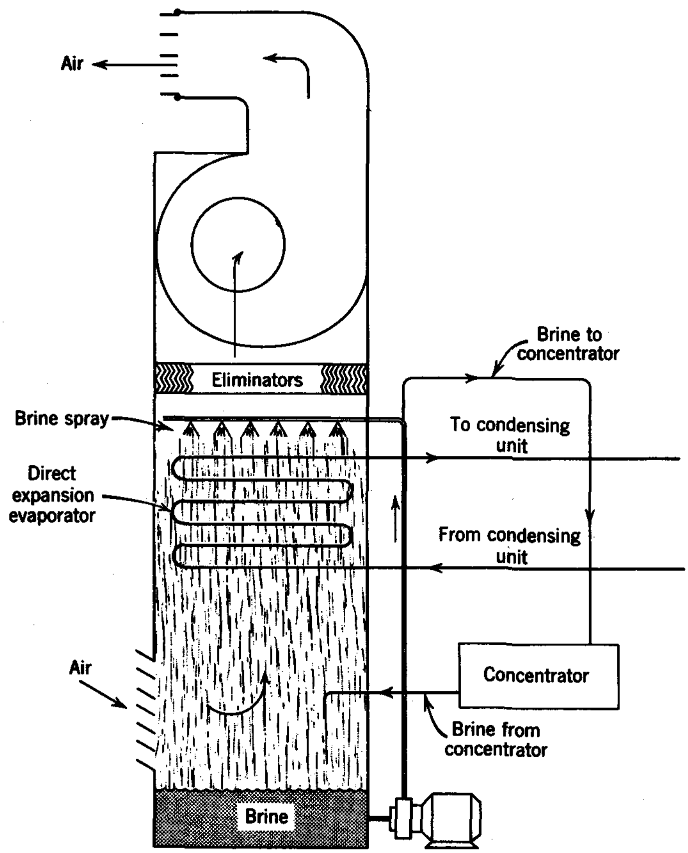


Fig. 11-47. Brine spray cooler.

the brine can be determined by measuring the specific gravity of the brine with a hydrometer.

11-36. Antifreeze Solutions. Certain water soluble compounds, generally described as antifreeze agents, are often used to depress the freezing point of water. The more widely known antifreeze agents are ethylene glycol, propylene glycol, Methanol (methyl alcohol), and glycerin. All these compounds are soluble in water in all proportions. The freezing temperature of water in solution with various percentages of each of these compounds is given in Table 11-5.

Propylene glycol is probably the most extensively used antifreeze agent in refrigeration service. In common with ethylene glycol, propylene glycol has a number of desirable properties. Unlike brine, glycol solutions are noncorrosive. They are also nonelectrolytic and therefore may be employed in systems containing dissimilar metals. Being extremely stable compounds, glycols will not evaporate under normal operating conditions. Because of the many advantages of glycol solutions, they are being used to replace brines in a number of installations, particularly in the brewing and dairy industries. The change-over from brine to glycol can be accomplished with practically no change in the plant facilities.

11-37. Brine Spray Units. Like chilled water, the chilled brine (or antifreeze solution) may be circulated directly around the refrigerated product or container, or it may be used to cool the air in a refrigerated space. When used to cool air, the chilled brine is circulated through a serpentine coil or through a brine spray unit. Two types of brine spray units which have been used extensively are shown in Figs. 11-46 and 11-47. In the former unit chilled brine from a brine chiller located outside the refrigerated space is sprayed down from spray nozzles and collected in the basin of the unit, from where it is returned to the brine chiller. In the latter type the brine is chilled by a direct-expansion coil located within the brine spray unit itself.

PROBLEMS

1. A walk-in cooler 8 ft by 9 ft by 9 ft high has walls 6 in. thick and is maintained at a temperature of 35° F. The load on the cooler is 7500 Btu/hr.

(*a*) Select a natural convection cooling coil (Plasti-Cooler) which will produce a relative humidity of approximately 85% in the cooler.

(*b*) Select a Unit Cooler which will produce approximately the same conditions in the cooler.

2. A freezing cabinet 6 ft high, 25 in. deep, and 80 in. wide has a freezing load of 3600 Btu/hr. Based on an evaporator TD of 10° F, determine the size and number of individual freezer plates to be used as shelves in the freezing cabinet.

3. The load on a tank-type brine cooler is 4500 Btu/hr. The brine is to be maintained at a temperature of 35° F with a refrigerant temperature of 19° F. Assuming little or no agitation of the brine, determine the lineal feet of $\frac{3}{4}$ in. pipe required for the evaporator.

4. It is desired to cool 100 gpm of water from 56° F to 46° F with a refrigerant temperature 38° F at the cooler outlet. Select an appropriate chiller and determine the water pressure drop through the chiller in psi.

12

Performance of Reciprocating Compressors

12-1. Refrigeration Compressors. Vapor compressors used in refrigeration are of three principal types: (1) reciprocating, (2) rotary, and (3) centrifugal. Of the three, the reciprocating compressor is by far the one most frequently used.

Rotary compressors are limited to use in very small fractional horsepower applications, such as home refrigerators and freezers and small commercial applications. Even in this limited area, rotary compressors represent only a small fraction of the total number used. Some rotary compressors are used also as booster compressors.* Their use for this purpose appears to be increasing.

Centrifugal compressors are used only on very large applications, usually at least 50 tons or above. In this area, they are widely accepted and are rapidly increasing in number because the number of large applications is growing steadily.

Only the performance of reciprocating compressors will be discussed in this chapter. Reciprocating compressor design, along with the design and performance of rotary and centrifugal compressors, is discussed elsewhere in the text at a more appropriate time and place. However, much that is said in this chapter about the performance of reciprocating compressors will apply also to the performance of rotary and centrifugal compressors.

* Booster compressors are discussed in Chapter 20.

12-2. The Compression Cycle. Before attempting to analyze the performance of the compressor, it is necessary to become familiar with the series of processes which make up the compression cycle of a reciprocating compressor.

A compressor, with the piston shown at four points in its travel in the cylinder, is illustrated in Fig. 12-1. As the piston moves downward on the suction stroke, low-pressure vapor from the suction line is drawn into the cylinder through the suction valves. On the upstroke of the piston, the low-pressure vapor is first compressed and then discharged as a high-pressure vapor through the discharge valves into the head of the compressor.

To prevent the piston from striking the valve plate, all reciprocating compressors are designed with a small amount of clearance between the top of the piston and the valve plate when the piston is at the top of its stroke. The volume of this clearance space is called the clearance volume and is the volume of the cylinder when the piston is at top dead center.

Not all the high-pressure vapor will pass out through the discharge valves at the end of the compression stroke. A certain amount will remain in the cylinder in the clearance space between the piston and the valve plate. The vapor which remains in the clearance space at the end of each discharge stroke is called the clearance vapor.

Reference to Figs. 12-2 and 12-3 will help to clarify the operation of the compressor. Figure 12-2 is a time-pressure diagram in which cylinder pressure is plotted against crank position. Figure 12-3 is a theoretical pressure-volume diagram of a typical compression cycle. The lettered points on the TP and PV diagrams correspond to the piston positions as shown in Fig. 12-1.

At point A, the piston is at the top of its stroke, which is known as top dead center. When the piston is at this position, both the suction and discharge valves are closed. The high pressure of the vapor trapped in the clearance space acts upward on the suction valves and holds them closed against the pressure of the suction vapor in the suction line. Because the pressure of the vapor in the head of the compressor is approximately the same as that of the vapor in the clearance volume, the discharge valves are held closed either by their own weight or by light spring loading.

As the piston moves downward on the suction stroke, the high-pressure vapor trapped in the clearance space is allowed to expand. The

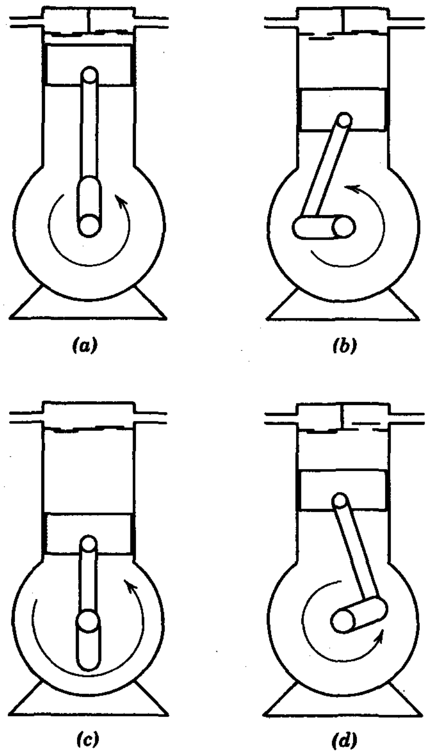


Fig. 12-1. *(a)* **Piston at top dead center.** *(b)* **Suction valves open.** *(c)* **Piston at bottom dead center.** *(d)* **Discharge valves open.**

expansion takes place along line *A–B* so that the pressure in the cylinder decreases as the volume of the clearance vapor increases. When the piston reaches point *B*, the pressure of the re-expanded clearance vapor in the cylinder becomes slightly less than the pressure of the vapor in the suction line; whereupon the suction valves are forced open by the higher pressure in the suction line and vapor from the suction line flows into the cylinder. The flow of suction vapor into the cylinder begins when the suction valves open at point *B* and continues until the

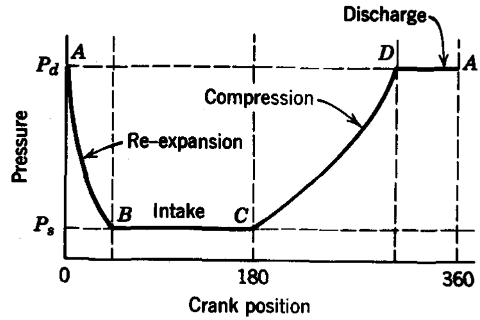


Fig. 12-2. Theoretical time-pressure diagram of compression cycle in which cylinder pressure is plotted against crank position.

piston reaches the bottom of its stroke at point *C*. During the time that the piston is moving from *B* to *C*, the cylinder is filled with suction vapor and the pressure in the cylinder remains constant at the suction pressure. At point *C*, the suction valves close, usually by spring action, and the compression stroke begins.

The pressure of the vapor in the cylinder increases along line *C–D* as the piston moves upward on the compression stroke. By the time the piston reaches point *D*, the pressure of the vapor in the cylinder has been increased until it is higher than the pressure of the vapor in the head of the compressor and the discharge valves are forced open; whereupon the high-pressure vapor passes from the cylinder into the hot gas line through the discharge valves. The flow of

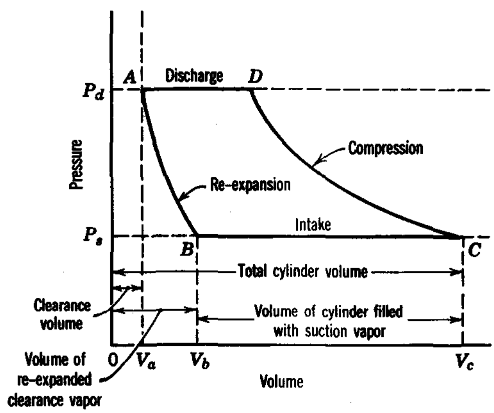


Fig. 12-3. Pressure-volume diagram of typical compression cycle.

the vapor through the discharge valves continues as the piston moves from D to A while the pressure in the cylinder remains constant at the discharge pressure. When the piston returns to point A, the compression cycle is completed and the crankshaft of the compressor has rotated one complete revolution.

12-3. Piston Displacement. The piston displacement of a reciprocating compressor is the total cylinder volume swept through by the piston in any certain time interval and is usually expressed in cubic feet per minute. For any single-acting, reciprocating compressor, the piston displacement is computed as follows:

$$V_p = \frac{\pi D^2 \times L \times N \times n}{4 \times 1728} \tag{12-1}$$

where V_p = the piston displacement in cubic feet per minute
D = the diameter of the cylinder (bore) in inches
L = the length of stroke in inches
N = revolutions of the crankshaft per minute (rpm)
n = the number of cylinders

The volume of the cylinder which is swept through by the piston each stroke (each revolution of the crankshaft) is the difference between the volume of the cylinder when the piston is at the bottom of its stroke and the volume of the cylinder when the piston is at the top of its stroke. This part of the cylinder volume is found by multiplying the cross-sectional area of the bore by the length of stroke. Thus:

Cross-sectional area of the bore in square inches $= \dfrac{\pi D^2}{4}$

Volume of cylinder swept through by the piston each stroke in cubic inches $= \dfrac{\pi D^2}{4} \times L$

Once the cylinder volume is known, the total cylinder volume swept through by the piston of a single cylinder compressor each minute in cubic inches can be determined by multiplying the cylinder volume by the rpm (N). When the compressor has more than one cylinder, the cylinder volume must also be multiplied by the number of cylinders (n). In either case, dividing the result by 1728 will give the piston displacement in cubic feet per minute.

Example 12-1. Calculate the piston displacement of a two cylinder compressor rotating at 1450 rpm, if the diameter of the cylinder is 2.5 in. and the length of stroke is 2 in.

Solution. Substituting in Equation 12-1,

$$\frac{3.1416 \times (2.5)^2 \times 2 \times 1455 \times 2}{4 \times 1728} = 16.52 \text{ cu ft/min}$$

12-4. Theoretical Refrigerating Capacity. The refrigerating capacity of any compressor depends upon the operating conditions of the system and, like system capacity, is determined by the weight of refrigerant circulated per unit of time and by the refrigerating effect of each pound circulated.*

The weight of refrigerant circulated per minute by the compressor is equal to the weight of the suction vapor that the compressor compresses per minute. If it is assumed that the compressor is 100% efficient and that the cylinder of the compressor fills completely with suction vapor at each downstroke of the piston, the volume of suction vapor drawn into the compressor cylinder and compressed per minute will be exactly equal to the piston displacement of the compressor. The weight of this volume of vapor, which is the weight of refrigerant circulated per minute, can be calculated by multiplying the piston displacement of the compressor by the density of the suction vapor at the compressor inlet.

Once the weight of refrigerant compressed per minute by the compressor has been determined, the theoretical refrigerating capacity of the compressor in tons can be found by multiplying the weight of refrigerant compressed per minute by the refrigerating effect per pound and then dividing by 200.

Example 12-2. The compressor in Example 12-1 is operating on a R-12 system at a suction temperature of 20° F. If the suction vapor reaching the compressor inlet is saturated and if the temperature of the liquid at the refrigerant control is 100° F, determine

(*a*) the total weight of refrigerant circulated per minute

(*b*) the theoretical refrigerating capacity of the compressor in tons.

* Since it is the compressor which circulates the refrigerant through the system, compressor capacity and system capacity are one and the same.

Solution

(*a*) From Example 12-1, piston displacement	= 16.52 cu ft/min
From Table 16-3, density of R-12 saturated vapor at 20° F	= 0.8921 lb/cu ft
Weight of refrigerant circulated per minute	= 16.52 × 0.8921 = 14.74 lb/min
(*b*) From Table 16-3, enthalpy of R-12 saturated vapor at 20° F	= 80.49 Btu/lb
Enthalpy of R-12 saturated liquid at 100° F	= 31.16 Btu/lb
Refrigerating effect	= 49.33 Btu/lb
Theoretical refrigerating capacity of compressor	= 14.74 × 49.33 = 727.12 Btu/min
Theoretical refrigerating capacity in tons	$= \dfrac{727.12}{200}$ = 3.63 tons

Since specific volume is the reciprocal of density, an alternate method of determining the weight of refrigerant circulated per minute by the compressor is to divide the piston displacement of the compressor by the specific volume of the suction vapor at the compressor inlet.

When the volume of vapor to be circulated per minute per ton for any given operating conditions is known, the capacity of the compressor in tons for the operating conditions in question may be found by dividing the piston displacement of the compressor by the volume of vapor to be compressed per minute per ton.

Example 12-3. For the conditions of Example 12-2, find (*a*) the weight of refrigerant circulated per minute per ton; (*b*) the volume of vapor to be compressed per minute per ton; and (*c*) the theoretical refrigerating capacity of the compressor in tons.

Solution

(*a*) From Example 12-2, refrigerating effect	= 49.33 Btu/lb
Weight of refrigerant circulated per minute per ton	$= \dfrac{200}{49.33}$ = 4.05 lb/min
(*b*) From Table 16-3, specific volume of R-12 saturated vapor at 20° F	= 1.121 cu ft/lb
Volume of vapor to be compressed per minute per ton	= 4.05 × 1.121 = 4.55 cu ft/min
(*c*) Piston displacement of compressor	= 16.52 cu ft/min
Theoretical refrigerating capacity of compressor in tons	$= \dfrac{16.52}{4.55}$ = 3.63 tons

12-5. Actual Refrigerating Capacity. The actual refrigerating capacity of a compressor is always less than its theoretical capacity as calculated in the previous examples. In the preceding examples it has been assumed: (1) that at each downstroke of the piston the cylinder of the compressor fills completely with suction vapor from the suction line and (2) that the density of the vapor filling the cylinder is the same as that in the suction line.

If these assumptions were correct, the actual refrigerating capacity would be exactly equal to the theoretical capacity. Unfortunately, this is not the case. Because of the compressibility of the refrigerant vapor and the mechanical clearance between the piston and the valve plate of the compressor, the volume of suction vapor filling the cylinder during the suction stroke is always less than the cylinder volume swept through by the piston. Too, it will be shown later that the density of the vapor filling the cylinder is less than the density of the vapor in the suction line. For these reasons, the actual volume of suction vapor at suction line conditions which is drawn into the cylinder of the compressor is always less than the piston displacement of the compressor and, therefore, the actual refrigerating capacity of the compressor is always less than its theoretical capacity.

12-6. Total Volumetric Efficiency. The actual volume of suction vapor compressed per minute is the actual displacement of the compressor. The ratio of the actual displacement of the compressor to its piston displacement is known as the total or real volumetric efficiency of the compressor. Thus:

$$E_v = \frac{V_a}{V_p} \times 100 \qquad (12\text{-}2)$$

where E_v = the total volumetric efficiency
V_a = actual volume of suction vapor compressed per minute
V_p = the piston displacement of the compressor

or

$$E_v = \frac{\text{Actual weight of suction vapor compressed} \times 100}{\text{Theoretical weight of suction vapor compressed}}$$

When the volumetric efficiency of the compressor is known, the actual displacement and refrigerating capacity can be found as follows:

$$V_a = V_p \times \frac{E_v}{100} \qquad (12\text{-}3)$$

and

$$\begin{matrix}\text{Actual} \\ \text{refrigerating} \\ \text{capacity}\end{matrix} = \begin{matrix}\text{Theoretical} \\ \text{refrigerating} \\ \text{capacity}\end{matrix} \times \frac{E_v}{100} \qquad (12\text{-}4)$$

Example 12-4. If the volumetric efficiency of the compressor in Example 12-3 is 76%, determine: (*a*) the actual volumetric displacement (*b*) the actual refrigerating capacity.

Solution

(*a*) From Example 12-1, piston displacement	= 16.52 cu ft/min
Actual volumetric displacement	= 16.52 × 0.76
	= 12.66 cu ft/min
(*b*) From Example 12-3, theoretical refrigerating capacity	= 3.63 tons
Actual refrigerating capacity	= 3.63 × 0.76
	= 2.76 tons

The actual refrigerating capacity of the compressor may also be determined as in Examples 12-2 and 12-3, if actual displacement is substituted for piston displacement.

12-7. Factors Influencing Total Volumetric Efficiency. The factors which tend to limit the volume of suction vapor compressed per working stroke, thereby determining the volumetric efficiency of the compressor, are the following:

1. Compressor clearance
2. Wiredrawing
3. Cylinder heating
4. Valve and piston leakage

12-8. The Effect of Clearance on Volumetric Efficiency. Because of compressor clearance and the compressibility of the refrigerant vapor, the volume of suction vapor flowing into the cylinder is less than the volume swept through by the piston. As previously shown, at the end of each compression stroke a certain amount of vapor remains in the cylinder in the clearance space after the discharge valves close. The vapor left in the clearance space has been compressed to the discharge pressure and, at the beginning of the suction stroke, this vapor must be re-expanded to the suction pressure before the suction valves can open and allow vapor from the suction line to flow into the cylinder. The piston will have completed a part of its suction stroke and the cylinder will already be partially filled with the re-expanded clearance vapor before the suction valves can open and admit suction vapor to the cylinder. Hence, suction vapor from the suction line will fill only that part of the cylinder volume which is not already filled with the re-expanded clearance vapor.

In Fig. 12-3, V_c is the total volume of the cylinder when the piston is at the bottom of its stroke. V_a, which represents the clearance volume, is the volume occupied by the clearance vapor at the end of the compression stroke. The difference between V_c and V_a then is the volume of the cylinder swept through by the piston each stroke. On the down stroke of the piston, the clearance vapor expands from V_a to V_b before the suction valves open. Therefore, the part of the cylinder volume which is filled with suction vapor during the balance of the suction stroke is the difference between V_b and V_c.

12-9. Theoretical Volumetric Efficiency. The volumetric efficiency of a compressor due to the clearance factor alone is known as the theoretical volumetric efficiency. It can be shown mathematically that the theoretical volumetric efficiency varies with the amount of clearance and with the suction and discharge pressures. The reason for this is easily explained.

12-10. Effect of Increasing the Clearance. If the clearance volume of the compressor is increased in respect to the piston displacement, the percentage of high-pressure vapor remaining in the cylinder at the end of the compression stroke will be increased. When re-expansion takes place during the suction stroke, a greater percentage of the total cylinder volume will be filled with the re-expanded clearance vapor and the volume of suction vapor taken in per stroke will be less than when the clearance volume is smaller. To obtain maximum volumetric efficiency, the clearance volume of a vapor compressor should be kept as small as possible.

It should be noted that this does not hold true for a reciprocating liquid pump. Since a liquid is not compressible, the liquid left in the clearance space at the end of the discharge stroke has the same specific volume as the liquid at the suction inlet. Therefore, there is no re-expansion of the liquid in the clearance during the suction stroke and the volume of liquid taken in each stroke is always equal to the volume swept by the piston, regardless of clearance.

12-11. Variation with Suction and Discharge Pressures. Increasing the discharge pressure or lowering the suction pressure will have the same effect on volumetric efficiency as increasing the clearance. If the discharge pressure is increased, the vapor in the clearance will be compressed to a higher pressure and a greater amount of re-expansion will be required to expand it to the suction pressure. Likewise, if the suction pressure is lowered, the clearance vapor must experience a greater re-expansion in expanding to the lower pressure before the suction valves will open.

On the other hand, for a constant discharge pressure, the amount of re-expansion that the clearance vapor experiences before the suction valves open diminishes as the suction pressure rises. It is evident, then, that the volumetric efficiency of the compressor increases as the suction pressure increases and decreases as the discharge pressure increases.

12-12. Compression Ratio. The ratio of the absolute suction pressure to the absolute discharge pressure is called the compression ratio. Thus,

$$R = \frac{\text{Absolute discharge pressure}}{\text{Absolute suction pressure}} \qquad (12\text{-}3)$$

where R = the compression ratio.

Example 12-5. Calculate the compression ratio of a R-12 compressor when the suction temperature is 20° F and the condensing temperature is 100° F.

Solution. From Table 16-3, absolute pressure of R-12 saturated vapor at 20° F $= 35.75$ psi

Absolute pressure of R-12 saturated vapor at 100° F $= 131.6$ psi

Compression ratio $= \dfrac{131.6}{35.75}$

$= 3.69$

Examination of Equation 12-3 indicates that the compression ratio is increased by either increasing the discharge pressure or lowering the suction pressure, or both.

In the preceding section it was shown that increasing the discharge pressure or lowering the suction pressure decreases the volumetric efficiency. It follows, then, that when the suction and discharge pressures are varied in such a direction that the compression ratio is increased, the volumetric efficiency of the compressor decreases. Likewise, decreasing the compression ratio will increase the volumetric efficiency. For a compressor of any given clearance, the volumetric efficiency varies inversely with the compression ratio.

12-13. The Effects of Wiredrawing. Wiredrawing is defined as a "restriction of area for a flowing fluid, causing a loss in pressure by (internal and external) friction without the loss of heat or performance of work; throttling."*

In order to have a flow of vapor from the suction line through the suction valves into the compressor cylinder, there must be a pressure differential across the valves sufficient to overcome the spring tension of the valves and valve weight and inertia. This means that the suction vapor experiences a mild, throttling expansion or drop in pressure as it flows through the suction valves and passages of the compressor. Therefore, the pressure of the suction vapor filling the cylinder of the compressor is always less than the pressure of the vapor in the suction line. As a result of the expanded condition of the vapor filling the cylinder, the volume of suction vapor taken in from the suction line each stroke is less than if the vapor filling the cylinder was at the suction line pressure.

A similar pressure differential is required across the discharge valves in order to cause the discharge vapor to flow through the valves into the condenser. To provide the necessary pressure differential across the discharge valves, the vapor in the cylinder must be compressed to a pressure somewhat higher than the actual condensing pressure. The vapor left in the clearance space at the end of the discharge stroke will be at this higher pressure. To re-expand from this higher pressure during the

* *Asre Data Book*, 1957–58 (page 39–27).

suction stroke, the clearance vapor must suffer a greater amount of re-expansion than if it had been compressed only to the condensing pressure. As a result of the greater expansion of the clearance vapor, a larger portion of the cylinder volume is filled with the re-expanded clearance vapor during the down stroke of the piston and the amount of suction vapor drawn in from the suction line is reduced.

Unlike the other factors which determine volumetric efficiency, wiredrawing is not directly affected by the compression ratio. In general, wiredrawing is a function of the velocity of the refrigerant vapor flowing through the valves and passages of the compressor. As the velocity of the vapor through the valves is increased, the effect of wiredrawing increases.

The refrigerant velocity through the valves of a compressor depends upon the design of the valves, the refrigerant used, and the speed of the compressor.

Wiredrawing is greatest for those refrigerants having the greatest specific volumes and the lowest latent heat values because the volume of vapor circulated per ton of refrigerating capacity is greater. This accounts for the large wiredrawing effect associated with R-12.

Increasing the speed of the compressor increases the piston displacement. Hence, the velocity of the vapor through the valves and the effects of wiredrawing are increased as the rpm are increased.

12-14. The Effects of Cylinder Heating. Another factor which tends to reduce the volumetric efficiency of the compressor is the heating of the suction vapor in the compressor cylinder. The suction vapor entering the compressor cylinder is heated by heat conducted from the hot cylinder walls and by friction which results from the turbulence of the vapor in the cylinder and from the fact that the refrigerant vapor is not a perfect gas. The heating causes the vapor to expand after entering the cylinder so that a smaller weight of vapor will fill the cylinder and thereby still further reduce the volume of vapor taken in from the suction line.

Cylinder heating increases as the compression ratio increases. At high compression ratios, the work of compression is greater and the discharge temperature is higher. This causes a rise in the temperature of the cylinder walls and other compressor parts so that the transfer of heat to the suction vapor occurs at a higher rate.

12-15. The Effect of Piston and Valve Leakage. Any back leakage of gas through either the suction or discharge valves or around the piston will decrease the volume of vapor pumped by the compressor. Because of precision manufacturing processes, there is very little leakage of gas around the pistons of a compressor in good condition. However, since it is not possible to design valves that will close instantaneously, there is always a certain amount of back leakage of gas through the suction and discharge valves.

As the pressure in the cylinder is lowered at the beginning of the suction stroke, a small amount of high-pressure vapor in the head of the compressor will leak back into the cylinder before the discharge valves can close tightly. Similarly, at the start of the compression stroke, some of the vapor in the cylinder will flow back through the suction valves into the suction line before the suction valves can close.

To assure prompt closing of the valves, both the suction and discharge valves are usually constructed of lightweight materials and are slightly spring loaded. However, since the spring tension increases wiredrawing, the amount of spring loading is critical.

For any given compressor, the amount of backleakage through the valves is a function of the compression ratio and the speed of the compressor. The higher the compression ratio, the greater is the amount of valve leakage. The effect of compressor speed on valve leakage is discussed later.

12-16. Determining the Total Volumetric Efficiency. The combined effects of all of the foregoing factors on the volumetric efficiency of the compressor varies with the design of the compressor and with the refrigerant used. Furthermore, for any one compressor the volumetric efficiency is not a constant amount; it changes with the operating conditions of the system. Therefore, the total volumetric efficiency of a compressor is difficult to predict mathematically and can be determined with accuracy only by actual testing of the compressor in a laboratory.

However, the results of such tests indicate that the volumetric efficiency of any one

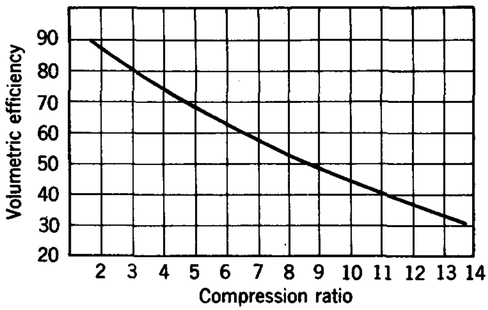


Fig. 12-4. Effect of compression ratio on volumetric efficiency of R-12 compressor.

compressor is primarily a function of the compression ratio and, for any given compression ratio, remains practically constant, regardless of the operating range. It has been determined also that compressors having the same design characteristics will have approximately the same volumetric efficiencies, regardless of the size of the compressor.

The relationship between the compression ratio and the volumetric efficiency of a typical R-12 compressor is illustrated by the curve in Fig. 12-4. In addition, in order to facilitate future calculations, the average volumetric efficiencies of a group of typical R-12 compressors at various compression ratios are given in Table 12-1. The values given are for compressors ranging in size from 5 to 25 hp. Smaller compressors will have slightly lower efficiencies, whereas larger compressors will have slightly higher efficiencies.

12-17. Variation in Compressor Capacity with Suction Temperature. Compressor performance and cycle efficiency will vary considerably with the operating conditions of the system. The most important factor governing the capacity of the compressor is the vaporizing temperature of the liquid in the evaporator, that is, the suction temperature. The large variations in compressor capacity which accompany changes in the operating suction temperature are primarily a result of a difference in the density of the suction vapor entering the suction inlet of the compressor. The higher the vaporizing temperature of the liquid in the evaporator, the higher is the vaporizing pressure and the greater is the density of the suction vapor. Because of the difference in the density of the suction vapor, each cubic foot of vapor compressed by the compressor will represent a greater weight of refrigerant when the suction temperature is high than when the suction temperature is low. This means that for any given position displacement, the weight of refrigerant circulated by the compressor per unit of time increases as the suction temperature increases.

The effect of suction temperature on compressor capacity is best illustrated by an actual example.

Example 12-6. Assuming 100% efficiency, if the liquid reaches the refrigerant control at 100° F in each case, determine the weight of refrigerant circulated per minute and the theoretical refrigerating capacity of the compressor in Example 12-1 when operating at each of the following suction temperatures: (*a*) 10° F and (*b*) 40° F.

Solution

(*a*) From Table 16-3, density of R-12 saturated vapor at 10° F	0.7402 lb/cu ft
From Example 12-1, piston displacement	= 16.52 cu ft/min
Weight of refrigerant circulated per minute at 10° F suction	= 16.52 × 0.7402 = 12.23 lb/min
From Table 16-3, enthalpy of R-12 saturated vapor at 10° F	= 79.36 Btu/lb
Enthalpy of R-12 liquid at 100° F	= 31.16 Btu/lb
Refrigerating effect	= 48.20 Btu/lb
Theoretical refrigerating capacity of compressor at 10° F suction, Btu/min	= 12.23 × 48.20 = 589.49 Btu/min
Theoretical refrigerating capacity in tons	$= \frac{589.49}{200}$ = 2.95 tons
(*b*) From Table 16-3, density of R-12 saturated vapor at 40° F	= 1.263 lb/cu ft
From Example 12-1, piston displacement	= 16.52 cu ft/min

Weight of refrigerant circulated per minute at 40° F suction	= 16.52 × 1.263 = 20.86 lb/min
From Table 16-3, enthalpy of R-12 saturated vapor at 40° F	= 82.71 Btu/lb
Enthalpy of R-12 liquid at 100° F	= 31.16 Btu/lb
Refrigerating effect	= 51.55 Btu/lb
Theoretical refrigerating capacity of compressor at 40° F suction, Btu/min	= 20.86 × 51.55 = 1075.33 Btu/min
Theoretical refrigerating capacity in tons	$= \dfrac{1075.33}{200}$ = 5.38 tons

In analyzing the results of Example 12-6 the following observations are of interest:

1. Although the piston displacement of the compressor is the same in each case, the weight of refrigerant circulated per minute by the compressor increases from 12.23 lb/min to 20.86 lb/min when the operating suction temperature is raised from 10° F to 40° F. The increase in the weight of refrigerant circulated results entirely from the greater density of the suction vapor entering the suction inlet of the compressor. In this instance, the percentage increase in the weight of refrigerant circulated is

$$= \frac{20.86 - 12.23}{12.23} \times 100 = 70.5\%$$

2. The theoretical refrigerating capacity of the compressor at the 10° F suction temperature is 2.95 tons, whereas at the 40° F suction temperature, the capacity increases to 5.38 tons. This represents an increase in refrigerating capacity of

$$= \frac{5.38 - 2.95}{2.95} \times 100 = 82.3\%$$

Although the increased density of the suction vapor at the higher suction temperature accounts for the greater part of the increase in compressor capacity, it is not the only reason for it. As indicated, the increase in the weight of refrigerant circulated is only 70.5%, whereas the total increase in compressor capacity is 82.3%. The additional 11.8% gain in capacity is brought about by an increase in the refrigerating effect of each pound of refrigerant circulated. Although the actual gain in refrigerating effect per pound is only 6.95%, when this increase is applied to the entire weight of refrigerant circulated at the higher suction temperature, the net gain in capacity over the original capacity which can be attributed to the greater refrigerating effect is 11.8% (1.705 × 0.0695 = 1.823 and 1.283 − 1.705 = 0.118 or 11.8%).

The actual variation in compressor capacity with changes in suction temperature is more pronounced than that indicated by theoretical computations. That is, the change in the actual compressor capacity with variations in suction temperature is always greater than the change in the theoretical capacity. The reason for this is that the compression ratio changes as the suction temperature changes. When the vaporizing temperature increases while the condensing temperature remains constant, the compression ratio is decreased and the volumetric efficiency of the compressor is improved. Hence, at the higher suction temperature, in addition to pumping a greater weight of refrigerant per unit of volume, the volume of vapor pumped by the compressor is also larger because of the improved efficiency.

Example 12-7. Assuming that the saturated discharge temperature is 100° F, determine the actual refrigerating capacity of the compressor in Example 12-6 when operating at each of the suction temperatures in question.

Solution

(*a*) From Table 16-3, absolute pressure corresponding to 100° F saturation temperature	131.6 psi
Absolute pressure corresponding to 10° F saturation temperature	= 29.35 psi
Compression ratio	$= \dfrac{131.6}{29.35}$ = 4.47
From Table 12-1, volumetric efficiency	= 76.3%
From Example 12-6, theoretical refrigerating capacity at 10° F suction	= 2.95 tons
Actual refrigerating capacity at 10° F suction	= 2.95 × 0.763 = 2.22 tons

(*b*) From Table 16-3, absolute pressure corresponding to 100° F saturation temperature = 131.6 psi

Absolute pressure corresponding to 40° F saturation temperature = 51.68 psi

Compression ratio $= \dfrac{131.6}{51.68}$
$= 2.55$

From Table 12-1, volumetric efficiency = 85.7%

From Example 12-6, theoretical refrigerating capacity at 40° F suction = 5.38 tons

Actual refrigerating capacity at 40° F suction = 5.38 × 0.857
= 4.61 tons

Whereas the theoretical increase in compressor capacity is only 82.3%, the actual increase in refrigerating capacity is

$$= \frac{4.61 - 2.22}{2.22} \times 100$$
$$= 107.7\%$$

12-18. Effect of Condensing Temperature on Compressor Capacity. In general, the refrigerating capacity of the compressor decreases as the condensing temperature increases and increases as the condensing temperature decreases. The effect that the condensing temperature has on compressor efficiency and capacity can be evaluated by comparing the results of the following example with those of Examples 12-6 and 12-7.

Example 12-8. Determine the theoretical and actual refrigerating capacities of the compressor in Example 12-1 for each of the two vaporizing temperatures given in Examples 12-6 and 12-7, if the condensing temperature in each case is 120° F rather than 100° F.

Solution

(*a*) For the 10° F vaporizing temperature.

From Example 12-1, piston displacement of compressor = 16.52 cu ft/min

From Table 16-3, density of R-12 saturated vapor at 10° F = 0.7402 lb/cu ft

Theoretical weight of refrigerant circulated per minute by com-compressor = 16.52 × 0.7402
= 12.23 lb

Refrigerating effect per pound at 10° F vaporizing and 120° F condensing = 43.20 Btu/lb

Theoretical refrigerating capacity of compressor = 12.23 × 43.20
= 527.34 Btu/min

Theoretical refrigerating capacity in tons $= \dfrac{527.34}{200}$
= 2.64 tons

From Table 16-3, absolute suction pressure = 29.35 psi

Absolute discharge pressure = 171.8 psi

Compression ratio $= \dfrac{171.8}{29.35}$
= 5.85

From Table 12-1, volumetric efficiency = 66.5%

Actual refrigerating capacity in tons = 2.645 × 0.665
= 1.76

(*b*) For the 40° F suction temperature.

From Example 12-1, piston displacement of compressor = 16.52 cu ft/min

From Table 16-3, density of R-12 saturated vapor at 40° F = 1.263 lb/cu ft

Theoretical weight of refrigerant circulated per minute by compressor = 16.52 × 1.263
= 20.86 lb

Refrigerating effect per pound at 40° F evaporating and condensing 120° F = 46.55 Btu/lb

Theoretical refrigerating capacity of compressor = 20.86 × 46.55
= 971 Btu/min

Theoretical refrigerating capacity in tons $= \dfrac{971}{200}$
= 4.85

From Table 16-3, absolute suction pressure = 51.68 psi

Absolute discharge pressure = 171.8 psi

Compression ratio	$= \dfrac{171.8}{51.68}$
	$= 3.32$
From Table 12-1, volumetric efficiency	78.5%
Actual refrigerating capacity in tons	$= 4.85 \times 0.785$
	$= 3.81$

Examining first the 10° F cycle, notice that raising the condensing temperature from 100° F to 120° F reduces the theoretical refrigerating capacity of the compressor from 2.95 tons to 2.64 tons and the actual capacity from 2.22 tons to 1.76 tons.

Since a 100% efficient compressor is assumed to displace a theoretical volume of vapor equal to its piston displacement and since the density of the suction vapor entering the compressor for any one vaporizing temperature is always the same regardless of the condensing temperature, the theoretical weight of refrigerant displaced by the compressor is the same at all condensing temperatures, and therefore the theoretical refrigerating capacity of the compressor for any condensing temperature depends only upon the refrigerating effect per pound of refrigerant circulated. Hence, the difference in the theoretical refrigerating capacity of the compressor at the two condensing temperatures results entirely from the difference in the refrigerating effect per pound.

The reduction in actual compressor capacity may be attributed to several factors: (1) a reduction in the refrigerating effect per pound and (2) a reduction in the volumetric efficiency of the compressor.

Increasing the condensing temperature while the suction temperature remains constant increases the compression ratio and reduces the volumetric efficiency of the compressor so that the actual volume of vapor displaced by the compressor per unit of time decreases. Therefore, even though the density of the vapor entering the compressor remains the same at all condensing temperatures, the actual weight of refrigerant circulated by the compressor per unit of time decreases because of the reduction in the quantity of vapor handled.

Increasing the condensing temperature increases the isentropic discharge temperature. In this instance, it is interesting to note (Fig. 7-7) that the increase in the isentropic discharge temperature is somewhat greater than that in the condensing temperature. Whereas the increase in condensing temperature is only 20° F (120° − 100°), the increase in the discharge temperature is 23.5° F (137.5° − 114°). This is accounted for by the greater work of compression at the higher compression ratio. Had the condensing temperature been increased in such a way that the compression ratio does not change (by increasing the suction temperature in proportion), the increase in the discharge temperature would have been approximately the same as that in the condensing temperature.

High discharge temperatures are undesirable and are to be avoided whenever possible. The higher the discharge temperature, the higher is the average temperature of the cylinder walls and the greater is the superheating of the suction vapor in the compressor cylinder. In addition to its adverse effect of compressor efficiency, high discharge temperatures tend to increase the rate of acid formation in the system, cause carbonization of the oil in the head of the compressor, and produce other effects detrimental to the equipment.

The loss of compressor efficiency and capacity resulting from an increase in the condensing temperature of the cycle is more serious when the suction temperature of the cycle is low than when the suction temperature is high. The desirability of operating a refrigerating system at the lowest practical condensing temperature has already been pointed out. This is of particular importance when the suction temperature of the cycle is low and the compressor is already operating at a relatively low efficiency.

When the cycle is operating at a 40° F vaporizing temperature, increasing the condensing temperature from 100° F to 120° F reduces the theoretical capacity of the compressor from 5.38 tons to 4.85 tons and the actual compressor capacity from 4.61 tons to 3.81 tons. The loss in theoretical capacity is

$$\frac{5.38 - 4.85}{5.38} \times 100 = 10\%$$

The loss in actual compressor capacity amounts to

$$\frac{4.61 - 3.81}{4.61} \times 100 = 17.4\%$$

For the 10° F cycle, the loss in theoretical compressor capacity is

$$\frac{2.95 - 2.64}{2.95} \times 100 = 10.5\%$$

and the loss in actual compressor capacity is

$$\frac{2.55 - 1.76}{2.55} \times 100 = 31\%$$

Note that the loss in theoretical capacity brought about by increasing the condensing temperature is approximately the same for both suction temperatures, whereas the loss in actual compressor capacity is much greater at the lower suction temperature. To a great extent, it is the loss in volumetric efficiency that causes the marked decrease in the actual capacity of the compressor at the higher condensing temperature. The change in volumetric efficiency for a given change in condensing temperature becomes greater as the suction temperature of the cycle decreases. This accounts for the greater effect that a change in condensing temperature has on compressor capacity when the suction temperature is low.

12-19. Compressor Horsepower. The theoretical horsepower required to drive the compressor may be found by multiplying the actual refrigerating capacity of the compressor in tons by the theoretical horsepower required per ton for the operating conditions in question.

Example 12-9. Find the theoretical horsepower required to drive the compressor in Example 12-4.

Solution. From Example 12-4, actual refrigerating capacity in tons	= 2.76 tons
From Fig. 7-9, theoretical horsepower required per ton	= 0.965 hp
The theoretical horsepower required to drive the compressor	= 2.76 × 0.965 = 2.66 hp

Notice that it is the actual refrigerating capacity of the compressor, rather than the theoretical refrigerating capacity, which must be used in determining the theoretical horsepower requirements of the compressor.

The theoretical horsepower as calculated in the preceding example is only an indication of the power which would be required by a 100% efficient compressor operating on an ideal compression cycle and does not represent the actual total horsepower which must be delivered to the shaft of the compressor. In actual practice, there are certain losses in power which accrue because of the mechanical friction in the compressor and because of the deviation of an actual compression cycle from the ideal compression cycle. Naturally, additional power must be supplied to the compressor to offset these losses. Therefore, the actual power required by a compressor will always be greater than the theoretical computations indicate.

12-20. Variation in Compressor Horsepower with Suction Temperature. Although the horsepower per ton of refrigerating capacity diminishes as the suction temperature rises, the horsepower required by the compressor may either increase or decrease, depending upon whether the work done by the compressor increases or decreases.

The total amount of work done by the compressor per unit of time in compressing the vapor and, hence, the power required to drive the compressor, is the function of only two factors: (1) the work of compression per pound of vapor compressed and (2) the weight of vapor compressed per unit of time.

The amount of work which is done in compressing the vapor from the suction pressure to the discharge pressure varies with the compression ratio. The greater the compression ratio, the greater is the work of compression. Therefore, when the suction temperature is raised while the condensing temperature remains the same, the compression ratio becomes smaller and the work of compression per pound is reduced. However, at the same time, because of the greater density of the suction vapor, the weight of vapor compressed by the compressor per unit of time increases. Since the saving in work done resulting from the reduction in the work per pound is seldom sufficient to outweigh the increase in the work of the compressor because of the increase in the weight of vapor compressed, raising the suction temperature will usually increase the power requirements of the compressor.

Example 12-10. Compute the theoretical horsepower required by the compressor in Example 12-7 at each of the suction temperatures listed.

Solution

(*a*) From Example 12-7, actual refrigerating capacity in tons at 10° F suction temperature	= 2.22 tons
From Fig. 7-9, theoretical horsepower per ton at 10° suction and 100° F condensing	= 1.13 hp
Theoretical horsepower of compressor at 10° F suction	= 2.22 × 1.13 = 2.51 hp
(*b*) From Example 12-7, actual refrigerating capacity in tons at 40° F suction temperature	= 4.61 tons
From Fig. 7-9, theoretical horsepower per ton at 40° F suction and 100° F condensing	= 0.683 hp
Theoretical horsepower of compressor at 40° F suction	= 4.61 × 0.683 = 3.15 hp

Although the horsepower per ton decreases 39.5% as the suction temperature is raised from 10° F to 40° F, because of the increase in the refrigerating capacity of the compressor, the horsepower required by the compressor increases from 2.51 hp to 3.15 hp. This represents an increase in the power required of

$$\frac{3.15 - 2.51}{2.51} \times 100 = 21\%$$

The increase in compressor horsepower with the suction temperature is relatively small in comparison to the increase in compressor capacity. In this instance, for a 30° F rise in suction temperature, the capacity of the compressor increased 107%, whereas compressor horsepower increased only 21%. The average increase in compressor capacity per degree of rise in suction temperature is 107%/30° F or 3.21%, whereas the increase in horsepower amounts to only 0.7% per degree of rise.

The relationship between compressor capacity and the horsepower of the compressor at various suction temperatures is shown by the curves in Fig. 12-5. The curves are for a typical R-12 compressor operating at a constant condensing temperature of 100° F.

As shown by the curve in Fig. 12-5 the horsepower required by a R-12 compressor increases as the suction temperature increases up to a certain point at which the horsepower required by the compressor is at a maximum. On reaching this point, if the suction temperature is further increased, the horsepower required by the compressor diminishes. This is not true, however, for compressors using ammonia as a refrigerant. For compressors using ammonia, the horsepower does not reach a maximum value, but continues to increase indefinitely as the suction temperature increases.

The suction temperature at which the horsepower required by a R-12 compressors reaches a maximum depends upon the condensing temperature and increases as the condensing temperature increases.

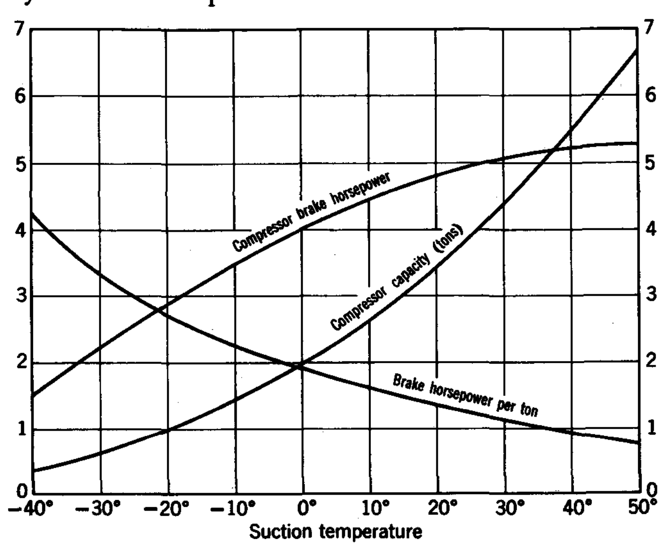


Fig. 12-5. Curves illustrate the effects of suction temperature on the capacity and horsepower of reciprocating compressors.

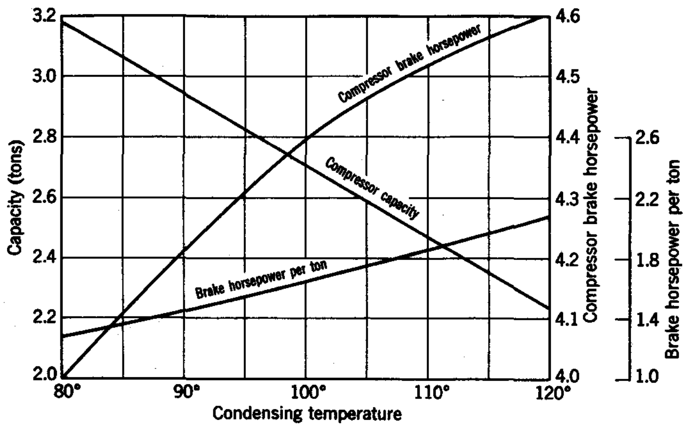


Fig. 12-6. Curves illustrate the effects of condensing temperature on capacity and horsepower of reciprocating compressors.

12-21. The Effect of Condensing Temperature on Compressor Horsepower. The curves in Fig. 12-6 illustrate the relationship between the horsepower required per ton of refrigerating capacity, the actual refrigerating capacity of the compressor, and the horsepower required by the compressor at various condensing temperatures when the suction temperature is kept constant. Note that, although the theoretical horsepower required per ton increases as the condensing temperature increases, the theoretical horsepower required by any one compressor will not increase in the same proportion. This is true because the decrease in the refrigerating capacity of the compressor which is coincident with an increase in the condensing temperature will offset to some extent the increase in the horsepower per ton.

For instance, according to Fig. 7-9, for a cycle operating at a 10° F vaporizing temperature, the theoretical horsepower required per ton increases from 1.13 to 1.52 when the condensing temperature of the cycle is increased from 100° F to 120° F. At the same time, Example 12-10 illustrates that the actual refrigerating capacity of one particular compressor drops from 2.22 tons to 1.76 tons when the condensing temperature is raised from 100° F to 120° F. The theoretical horsepower required by the compressor at the 100° F condensing temperature is

$$1.13 \times 2.22 = 2.51 \text{ hp}$$

For the 120° F condensing temperature, the theoretical horsepower required by the compressor is

$$1.52 \times 1.76 = 2.68 \text{ hp}$$

12-22. Brake Horsepower. The total horsepower which must be supplied to the shaft of the compressor is called the brake horsepower and may be computed from the theoretical horsepower by application of a factor called over-all compressor efficiency. The over-all efficiency is an expression of the relationship of the theoretical horsepower to the brake horsepower in percent. Written as an equation, the relationship is

$$E_o = \frac{Thp}{Bhp} \times 100 \tag{12-6}$$

and

$$Bhp = \frac{Thp}{E_o/100} \tag{12-7}$$

where E_o = the over-all efficiency in percent
Thp = the theoretical horsepower
Bhp = the brake horsepower

Example 12-11. Determine the brake horsepower required by the compressor in Example 12-14, if the over-all efficiency of the compressor is 80%.

Solution. From Example 12-4,
$Thp = 3.12$ hp
Applying Equation 12-7, the
$$Bhp = \frac{Thp}{E_o} = \frac{3.12}{0.80} = 3.9 \text{ hp}$$

The over-all efficiency is sometimes broken down into two components: (1) the compression efficiency and (2) the mechanical efficiency. In such cases, the relationship is

$$E_o = E_c \times E_m \tag{12-8}$$

where E_c = the compression efficiency in percent

E_m = the mechanical efficiency in percent

so that
$$Bhp = \frac{Thp}{E_c \times E_m} \qquad (12\text{-}9)$$

The compression efficiency of a compressor is a measure of the losses resulting from the deviation of the actual compression cycle from the ideal compression cycle, whereas the mechanical efficiency of the compressor is a measure of the losses resulting from the mechanical friction in the compressor. The principal factors which bring about the deviation of an actual compression cycle from the ideal compression cycle are: (1) wiredrawing, (2) the exchange of heat between the vapor and the cylinder walls, and (3) fluid friction due to the turbulence of the vapor in the cylinder and to the fact that the refrigerant vapor is not an ideal gas. Notice that the factors which determine the compression efficiency of the compressor are the same as those which influence the volumetric efficiency. As a matter of fact, for any one compressor, the volumetric and compression efficiencies are roughly the same and they vary with the compression ratio in about the same proportions. For this reason, the brake horsepower required per ton of refrigerating capacity can be approximated with reasonable accuracy by dividing the theoretical horsepower per ton by the volumetric efficiency of the compressor and then adding about 10% to offset the power loss due to the mechanical friction in the compressor. Written as an equation,

$$Bhp = \frac{M(h_d - h_c) \times 1.1}{42.42 \times E_v} \qquad (12\text{-}10)$$

Since the relationship between the various factors which influence the compression efficiency are difficult to evaluate mathematically, the compression efficiency of a compressor can be determined accurately only by actual testing of the compressor.

12-23. Indicated Horsepower. A device frequently used to determine the compression efficiency is the indicator diagram. An indicator diagram is a pressure-volume diagram of the actual compression cycle of the compressor which is produced during the actual testing of the compressor.

A theoretical indicator diagram for an ideal compression cycle is shown in Fig. 12-7. It has been illustrated previously that the area under a process diagram on a pressure-volume chart is a measure of the work of the process. In Fig. 12-7, notice that the area *dDCd* represents the work done by the piston in compressing the vapor during the isentropic process *CD*, and that the area *aADba* represents the work done by the piston in discharging the vapor from the cylinder during the constant pressure process *DA*, whereas the area *aABa* represents the work done back on the piston by the vapor during the isentropic re-expansion (of the clearance vapor) process *AB*. Since the work of process *AB* is work given back to the piston by the fluid, the net work input to the compression cycle is the sum of the work of processes *CD* and *DA*, less the work of process *AB*. Therefore, the net work of the compression cycle is represented by the area *BADCB*, the total area enclosed by the cycle diagram.

The work of the compression cycle as determined from the indicator diagram is called the indicated work and the horsepower computed from the indicated work is called the indicated horsepower.

Since the indicator diagram illustrated in Fig. 12-7 is a theoretical indicator diagram of an ideal compression cycle, the indicated work is the work of an ideal compression cycle and the indicated horsepower computed from the indicated work would, of course, be exactly

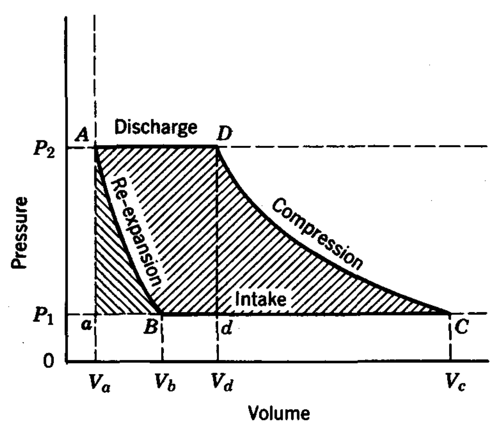


Fig. 12-7. Theoretical indicator diagram for an ideal compression cycle.

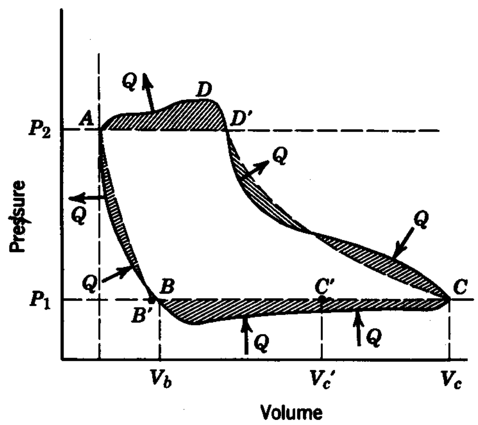


Fig. 12-8. Theoretical indicator diagram for an actual compression cycle.

equal to the theoretical horsepower. However, in actual practice, since the indicator diagram reproduces the true paths of the various processes which make up the actual compression cycle, the indicated work of the diagram is an accurate measure of the actual work of the compression cycle and, therefore, the indicated horsepower computed on the basis of the indicated work is the actual horsepower required to do the work of the actual compression cycle.

Care should be taken not to confuse indicated horsepower with brake horsepower. Although the indicated horsepower includes the power required to offset the losses resulting from the deviation of an actual compression cycle from the ideal cycle, it does not include the power required to overcome the losses resulting from the mechanical friction in the compressor. In other words, the indicated horsepower takes into account the compression efficiency but not the mechanical efficiency. Hence, brake horsepower differs from indicated horsepower in that the brake horsepower includes the power required to overcome the mechanical friction in the compressor, whereas the indicated horsepower does not. The horsepower necessary to overcome the mechanical friction in the compressor is sometimes referred to as the friction horsepower (*Fhp*), so that

$$Bhp = Ihp + Fhp \qquad (12\text{-}11)$$

The relationship of the indicated horsepower to the theoretical horsepower is

$$Ihp = \frac{Thp}{E_c} \qquad (12\text{-}12)$$

An indicator diagram of an actual compression cycle is shown in Fig. 12-8. The area *ABCD*, enclosed by the cycle diagram, is, of course, a measure of the work of the cycle. An ideal cycle, *AB'CD'*, is drawn in for comparison. Pressures P_1 and P_2 represent the pressure of the vapor entering and leaving the compressor. The areas above line P_2 and below line P_1 represent the increased work of the cycle due to wiredrawing. Notice that at the end of the suction and discharge strokes (points *C* and *A*), the piston velocity diminishes to zero and the pressure of the vapor tends to return to P_1 and P_2, respectively. The other deviations from the ideal cycle represent the losses resulting from the heating of the vapor in the compressor cylinder. Line *BC* indicates the approximate volume of the suction vapor at the end of the suction stroke, whereas line *BC'* represents the approximate volume of this same weight of vapor in the suction line. The deviation of the actual compression process from the isentropic can be seen by comparing the actual compression path *CD* to the isentropic path *CD'*.

The direction of the periodic heat transfer between the vapor and the cylinder walls at various times and points in the cycle is indicated by the arrows. The arrows pointing in denote heat transfer from the cylinder walls to the vapor, whereas arrows pointing out indicate heat transfer from the vapor to the cylinder walls.

The temperature of the cylinder walls of the compressor will fluctuate around some mean value which is between the suction and discharge temperatures of the vapor. During the latter part of the re-expansion process, during the period in which the vapor is being admitted to the cylinder, and during the initial part of the compression stroke, the cylinder wall temperature is greater than the vapor temperature and heat passes from the cylinder walls to the vapor. During the latter part of the compression stroke, during the discharge period, and during the early part of the suction stroke, the temperature of the vapor exceeds the cylinder wall temperature and heat passes from the vapor to the cylinder walls.

12-24. Isothermal vs. Isentropic Compression. Reference to Fig. 12-9 will show that if the compression process in the compressor was isothermal rather than isentropic the net work of the compression cycle would be reduced even though the work of the compression process itself is greater for isothermal compression than for isentropic compression. The reduction in the work of the cycle which would be realized through isothermal compression is indicated by the crosshatched area in Fig. 12-9.

Isothermal compression is not practical for a refrigeration compressor since it would result in the discharge of saturated liquid from the compressor. Furthermore, if a cooling medium were available at a temperature low enough to cool the compressor sufficiently to produce isothermal compression, the cooling medium could be used directly as the refrigerant and there would be no need for the refrigeration cycle.

12-25. Water-Jacketing the Compressor Cylinder. Any heat which is given up by the compressor cylinder to some external cooling medium represents, in effect, heat given up by the vapor during the compression process. Cooling of the vapor during compression causes the path of the compression process to shift from the isentropic path toward an isothermal path. Of course, the greater the amount of cooling, the greater will be the shift toward the isothermal.

If the temperature of the air surrounding the compressor were exactly the same as the temperature of the compressor cylinder, there would be no transfer of heat from the cylinder to the air and any heat given up by the vapor to the cylinder would be eventually reabsorbed by the vapor and the compression process would be approximately adiabatic. However, since there is nearly always some transfer of heat from the compressor to the surrounding air, compression is usually polytropic rather than isentropic. For an air-cooled compressor, the transfer of heat to the air will be slight and, therefore, the value of the polytropic compression exponent, n, will very nearly approach the isentropic compression exponent, k. Hence, the assumption of isentropic compression for the ideal cycle is ordinarily not too much in error for an air-cooled compressor.

Water-jacketing of the compressor cylinder results in lowering the temperature of the cylinder walls, and cooling of the vapor during compression will be greater for the compressor having a water jacket. Too, cylinder heating is reduced and the vapor is discharged from the compressor at a lower temperature. All of this has the effect of reducing the work of the compression cycle. However, the gain is usually not sufficient to warrant the use of a water jacket on most compressors, particularly compressors designed for R-12. For the most part, water-jacketing of the compressor is limited to compressors designed for use with refrigerants which have unusually high discharge temperatures, such as ammonia. Even then, the purpose of the jacketing is not so much for increasing compressor efficiency as it is to reduce the rate of oil carbonization and the formation of acids, both of which increase rapidly as the discharge temperature increases.

12-26. Wet Compression. Wet compression occurs when small particles of unvaporized liquid are entrained in the suction vapor entering the compressor. However, theoretical computations indicate that wet compression will bring about desirable gains in compression efficiency and reduce the work of compression. This would be true if the small particles of liquid vaporized during the actual compression of the vapor. However, in actual practice, this is not the case. Since heat transfer is a function of time and since compression of the vapor in a modern high-speed compressor takes place very

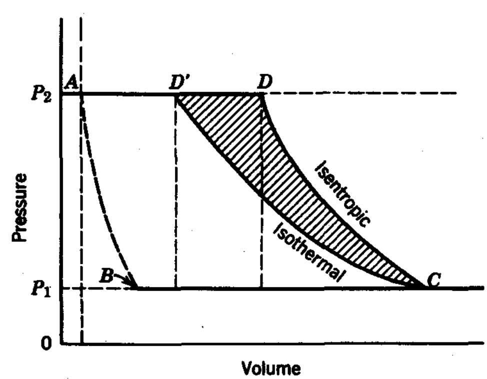


Fig. 12-9. Isentropic vs. isothermal compression.

rapidly, there is not sufficient time for the liquid to completely vaporize during the compression stroke. Hence, some of the liquid particles remain in the vapor in the clearance volume and vaporize during the early part of the suction stroke. This action reduces the volumetric efficiency of the compressor without benefit of the return of work to the piston by the expansion of the vaporizing particles.

A result similar to this is encountered when excessive cooling of the cylinder reduces the temperature of the vapor in the clearance below the saturation temperature corresponding to the discharge. Some of the clearance vapor will condense and the particles of liquid formed will vaporize during the early part of the suction stroke.

12-27. The Effect of Compressor Clearance on Horsepower. Theoretically, the clearance of the compressor has no effect on the horsepower, since the work done by the piston in compressing the clearance vapor is returned to the piston as the clearance vapor re-expands at the start of the suction stroke. However, since the refrigerant vapor is not an ideal gas, there is some loss of power in overcoming the internal friction of the fluid so that the power returned to the piston during the re-expansion of the clearance vapor will always be less than the power required to compress it. Hence, the clearance does have some, although probably slight, effect on the power requirements.

12-28. Compressor Speed. Since the speed of rotation is one of the factors determining piston displacement (Equation 12-1), the capacity of the compressor changes considerably when the speed of the compressor is changed. If the speed of the compressor is increased, the piston displacement is increased and the compressor displaces a greater volume of vapor per unit of time. Theoretically, based on the assumption that the volumetric efficiency of the compressor remains constant, the capacity of the compressor varies in direct proportion to the speed change. That is, if the speed of the compressor is doubled, the piston displacement and capacity of the compressor are also doubled. Likewise, if the speed of the compressor is reduced, the piston displacement and capacity of the compressor are reduced in the same proportion. However, the volumetric efficiency of the compressor does not remain constant during speed changes, and therefore the change in compressor capacity will not be proportional to the speed change.

The variation in the volumetric efficiency with changes in the speed of rotation is brought about principally by changes in the effects of wiredrawing, cylinder heating, and the back leakage of gas through the suction and discharge valves.

The amount of back leakage through the valves in percent per cubic foot of vapor displaced is at a maximum at low compressor speeds and decreases as the speed of the compressor is increased. Cylinder heating, too, is greatest at low compressor speeds. On the other hand, the effect of wiredrawing is at a minimum at low speeds and increases as the speed increases because of the increase in the velocity of the vapor passing through the valves. Hence, as the speed of rotation increases, the volumetric efficiency of the compressor due to the cylinder heating and valve leakage factors increases, while, at the same time, the volumetric efficiency due to the wiredrawing factor decreases. It follows, then, that there is one critical speed of rotation at which the combined effect of these factors are at a minimum and the volumetric efficiency is at a maximum. From an efficiency standpoint, this is the speed at which the compressor should be operated. At speeds higher than this critical speed, the volumetric efficiency of the compressor diminish because the loss of efficiency due to the wiredrawing effect will be greater than the gain resulting from the decrease in the effect of cylinder heating and valve leakage. Likewise, at speeds below the critical speed, the volumetric efficiency will be lower because the losses accruing from the increase in cylinder heating and valve leakage will be greater than the gain resulting from the decrease in the wiredrawing losses.

The critical speed will vary with the design of the compressor and with the refrigerant used, and can best be determined by actual test of the compressor.

It is general practice in the design of modern high-speed compressors to use large valve ports in order to reduce the wiredrawing effect to a practical minimum. These large openings in the valve-plate tend to increase the clearance volume and decrease the volumetric efficiency due to the clearance factor, but the advantages accruing from the reduction in the wiredrawing effect more than offsets the loss of efficiency due to the greater clearance. This is particularly true where the power requirements are concerned, since the loss of power due to wiredrawing is much greater than the loss of power due to the clearance factor.

12-29. Mechanical Efficiency. The mechanical friction in the compressor varies with the speed of rotation, but for any one speed, the mechanical friction, and therefore the friction horsepower, will remain practically the same at all operating conditions. Since the friction horsepower remains the same, it follows that the mechanical efficiency of the compressor depends entirely upon the loading of the compressor. As the total brake horsepower of the compressor increases due to loading of the compressor, the friction horsepower, being constant, will become a smaller and smaller percentage of the total horsepower and the mechanical efficiency will increase. It is evident that the mechanical efficiency of the compressor will be greatest when the compressor is fully loaded. The mechanical efficiency of the compressor will vary with the design of the compressor and with compressor speed. An average compressor of good design operating fully loaded at a standard speed should have a mechanical efficiency somewhat above 90%.

12-30. The Effect of Suction Superheat on Compressor Performance. It has been shown that superheating of the suction vapor causes the vapor to reach the compressor in an expanded condition. Therefore, when the vapor reaches the compressor in a superheated condition, the weight of refrigerant circulated by the compressor per minute is less than when the vapor reaches the compressor saturated. Whether or not the reduction in the weight of refrigerant circulated by the compressor reduces the refrigerating capacity of the compressor depends upon whether or not the superheating produces useful cooling. When the superheating produces useful cooling, the gain in refrigerating capacity resulting from the increase in the refrigerating effect per pound is usually sufficient to offset the loss in refrigerating capacity resulting from the reduction in the weight of refrigerant circulated. On the other hand, when the superheating produces no useful cooling, there is no offsetting gain in capacity and the refrigerating capacity of the compressor is reduced in inverse proportion to the increase in the specific volume of the suction vapor at the compressor inlet.

Regardless of whether or not the superheating produces useful cooling, the horsepower required to drive any one compressor is practically the same for a superheated cycle as for a saturated cycle. It was shown in Section 8-4 that, when superheating of the vapor produces useful cooling, both the horsepower required per ton and the refrigerating capacity of the compressor are the same for the superheated cycle as for the saturated cycle. It follows, then, that the horsepower required by any one compressor will be the same for both cycles. On the other hand, when superheating of the vapor produces no useful cooling, the horsepower per ton is greater than for the saturated cycle. However, at the same time, the refrigerating capacity of the compressor is less for the superheated cycle and the increase in the horsepower required per ton is more or less offset by the reduction in compressor capacity, so that the horsepower required by the compressor is still approximately the same as for the saturated cycle. Notice that, although the horsepower required by any one compressor is not appreciably changed by the superheating of the suction vapor, when the superheating does not produce useful cooling, the refrigerating capacity and efficiency of the compressor are materially reduced. This is particularly true when the compressor is operating at a low suction temperature. It should be noted also that superheating of the suction vapor reduces the amount of cylinder heating and the efficiency of the compressor is increased to some extent.

12-31. The Effect of Subcooling on Compressor Performance. When subcooling of the liquid refrigerant is accomplished in such a way that the heat given up by the liquid leaves the system, the specific volume of the suction vapor at the compressor inlet is unaffected by the subcooling and the weight of refrigerant circulated per minute by the compressor is the same as when no subcooling takes place. Since the refrigerating effect per pound is increased by the subcooling, the capacity of the compressor is increased by an amount equal to the amount of subcooling. Notice that the increase in the refrigerating capacity of the compressor resulting from the subcooling is accomplished without increasing the power requirements of the compressor. Therefore, subcooling improves compressor efficiency, provided the heat given up during the subcooling leaves the system.

When the heat given up during the subcooling does not leave the system, as when a heat exchanger is used, the gain in capacity due to the subcooling is approximately equal to the loss in capacity due to the superheating, and the refrigerating capacity of the compressor is very little affected. However, there is some small gain in compressor efficiency, since the superheating of the vapor in the heat exchanger reduces the effect of cylinder heating.

12-32. Compressor Rating and Selection. As previously stated, mathematical evaluation of all the factors which influence compressor performance is not practical. Hence, compressor capacity and horsepower requirements are determined accurately only by actual testing of the compressor. Table R-10A is a typical compressor rating table supplied by the compressor manufacturer for use in compressor selection. The ratings have been determined by actual testing of the compressor under operating conditions set forth in the compressor testing and rating standards of the American Society of Heating, Refrigerating, and Air Conditioning Engineers (see Table R-10B).

It has been shown in the foregoing sections that both the refrigerating capacity and the horsepower requirements of a compressor vary with the condition of the refrigerant vapor entering and leaving the compressor. Notice in Table R-10A that compressor refrigerating capacities (Btu/hr) and horsepower requirements are listed for various saturated suction and discharge temperatures. The saturated suction temperature is the saturation temperature corresponding to the pressure of the vapor at the suction inlet of the compressor, and the saturated discharge temperature is the saturation temperature corresponding to the pressure of the vapor at the discharge of the compressor.

Although compressor ratings are based on the saturated suction and discharge temperatures, ASHRAE test standards require a certain amount of suction superheat and specify that the actual temperature of the suction vapor entering the compressor be those listed in Table R-10C. Since the compressor ratings given in Table R-10A are in accordance with ASHRAE standards, it follows that in order to realize the listed ratings, the suction vapor must enter the suction inlet of the compressor at the conditions shown in Table R-10C. For example, for a compressor operating at a saturated suction of −40° F, the suction vapor should enter the compressor at a temperature of 35° F (superheated 75° F from −40° F to 35° F), if the listed rating is to be obtained. Likewise, for a compressor operating at a saturated suction of 40° F, the actual temperature of the suction vapor entering the compressor should be 65° F (superheated 25° F from 40° F to 65° F). Where the actual temperature of the suction vapor is less than that indicated in Table R-10C, the tabulated rating is corrected by using an appropriate multiplier to obtain the actual compressor capacity. The multipliers given in Table R-10D correct the ratings to a basis of no superheat for the saturated condition listed. Where the actual suction vapor temperature is intermediate between saturation and the temperature shown in Table R-10C, the multiplier is corrected accordingly.

The superheating is assumed to occur in the evaporator, in the suction line inside the refrigerated space, or in a liquid-suction heat exchanger so that the superheat produces useful cooling (Section 8-4). Superheating which occurs outside the refrigerated space should be disregarded with respect to the tabulated ratings.

The superheating requirement of the ASHRAE standards at first appears to complicate unnecessarily the compressor rating and selection procedure. However, this is not the case. The superheating requirement is very realistic in that the amount of superheating specified in the rating standards very nearly approaches that amount which would normally be expected in a well-designed application. Hence, the effect of the superheating requirement is to cause the compressor to be rated under conditions similar to those under which the compressor will be operating in the field. For this reason, except in unusual cases, no appreciable error will occur if the compressor ratings given in Table R-10A are used without correction of any kind. Furthermore, compressor capacity requirements are not usually critical within certain limits. There are several reasons for this. First of all, the methods of determining the required compressor capacity (cooling load calculations) are not in themselves exact. Too, it is seldom possible to select a compressor which has exactly the required capacity at the design conditions. Another reason that compressor capacity is not critical within reasonable limits is that the operating conditions of the system do not remain constant at all times, but vary from time to time with the loading of the system, the temperature of the condensing medium, etc. General practice is to select a compressor having a capacity equal to or somewhat in excess of the required capacity at the design operating conditions.

It was shown in Chapter 8 that subcooling of the liquid increases the refrigerating effect per pound and thereby increases compressor capacity. With regard to subcooling, the ratings given in Table R-10A are based on saturated liquid approaching the refrigerant control, that is, no subcooling. Where the liquid is subcooled by external means (Section 8-7), the capacity of the compressor may be increased approximately 2% for each 5° F of subcooling. Here again, for reasons outlined in the preceding paragraph, the effect of subcooling is usually neglected in selecting the compressor.

To select a compressor for a given application, the following data are needed:

1. The required refrigerating capacity (Btu/hr)
2. The design saturated suction temperature
3. The design saturated discharge temperature

Naturally, the required refrigerating capacity is the average hourly load as determined by the cooling load calculations. However, if an evaporator selection is made prior to the compressor selection, the compressor should be selected to match the evaporator capacity rather than the calculated load. The reasons for this are discussed in Chapter 13.

The design saturated suction temperature depends upon the design conditions of the application. Specifically, it depends upon the evaporator temperature (the saturation temperature of the refrigerant at the evaporator outlet) and upon the pressure loss in the suction line. For instance, assume an evaporator temperature of 28° F and a suction line pressure loss of approximately 3 psi. From Table 16-3, the saturation pressure of Refrigerant-12 corresponding to a temperature of 28° F is 41.59 psia. Allowing for the 3 psi pressure loss in the suction line, the pressure of the vapor at the suction inlet of the compressor is 38.59 psia (41.59 − 3). From Table 16-3, the saturation temperature corresponding to a pressure of 38.59 psia, and, therefore, the saturated suction temperature, is approximately 24° F.

The design saturated discharge temperature depends primarily on the size of the condenser selected and upon the quantity and temperature of the available condensing medium. Methods of condenser selection are discussed in Chapter 14.

12-33. Condensing Unit Rating and Selection. Since condensing unit capacity depends upon the capacity of the compressor, methods of rating and selecting condensing units are practically the same as those for rating and selecting compressors. The only difference is that, whereas compressor capacities are based on the saturated suction and discharge temperatures, condensing unit capacities are based on the saturated suction temperature and on the quantity and temperature of the condensing medium. Since the size of the condenser is fixed at the time of manufacture for any given condenser loading, the only variables determining the saturation temperature at the discharge of the compressor (and therefore the capacity of the compressor at any given suction temperature) is the quantity and temperature of the condensing medium. For air-cooled condensing units, when the quantity of the air passing over the condenser is fixed by the fan selection at the time of manufacture, the only variable determining the capacity of the condensing unit, other than the suction temperature, is the ambient air temperature (temperature of the air entering the condenser). Hence, ratings for air-cooled condensing units are based on the saturated suction temperature and the ambient air temperature.

Ratings for water-cooled condensing units are based on the saturated suction temperature and on the entering and leaving water temperatures.* Typical capacity ratings for air-cooled

* For any given condenser loading and entering water temperature, the leaving water temperature depends only on the quantity of water (gallons per minute) flowing through the condenser.

and water-cooled condensing units are given in Tables R-11 and R-12, respectively.

Example 12-12. A certain refrigeration application has a calculated cooling load of 33,000 Btu/hr. If the design saturated suction and discharge temperatures are 20° F and 100° F, respectively, select a compressor from Table R-10A which will meet the requirements of the application.

Solution. Locate the desired saturated discharge temperature in the first column of the table (100° F). Next, in the second column, locate the desired saturated suction temperature and read to the right until a compressor having a capacity equal to or somewhat in excess of the desired capacity is found. Select compressor, Model #5F20, which has a capacity of 34,000 Btu/hr at 1450 rpm.

Example 12-13. Assume that the design saturated suction in Example 12-12 is 0° F and make a new compressor selection.

Solution. Using the procedure outlined in the solution to Example 12-12, select compressor Model #5F30 which has a capacity of 36,000 Btu/hr at 1750 rpm.

Example 12-14. Determine the capacity of compressor Model #5F20 when operating at a 23° F saturated suction temperature, if the saturated discharge temperature is 100° F.

Solution. From Table R-10A,

Compressor capacity at 30° F suction = 44,200 Btu/hr

Compressor capacity at 20° F suction = 34,600 Btu/hr

Capacity change per 10° F change in saturated suction temperature = 44,200 − 34,600
= 9600 Btu/hr

Average capacity change per ° F change in suction temperature $= \dfrac{9600}{10}$
= 960 Btu/hr

Total capacity change for 3° F change in suction temperature = 960 × 3
= 2780 Btu/hr

Approximate capacity of compressor at a 23° F saturated suction = 34,000 + 2780
= 37,380 Btu/hr

Example 12-15. A certain refrigeration application has a calculated cooling load of 8750 Btu/hr and the ambient temperature is 90° F. If the design saturated suction temperature is 20° F, select an air-cooled condensing unit which will satisfy the requirements of the application.

Solution. From Table R-11, select a 1 hp condensing unit having a capacity of 9340 Btu/hr at the prescribed conditions.

PROBLEMS

1. A four-cylinder reciprocating compressor having a 2 in. bore and a 2.5 in. stroke is rotating at 100 rpm. Compute the piston displacement in cubic feet per minute.
Ans. 18.18 cfm

2. Assume the compressor in Problem 12-1 is operating on the cycle described in Problem 7-1 and compute the theoretical refrigerating capacity of the compressor in Btu/hr.
Ans. 2.47 tons

3. Using the conditions of Problem 12-2, determine:
(*a*) The volumetric efficiency of the compressor.
(*b*) The actual refrigerating capacity of the compressor in tons. *Ans.* 1.52 tons
(*c*) The brake horsepower required per ton (allow 10% for mechanical friction and assume the compression efficiency is the same as the volumetric efficiency).
Ans. 2.7 hp/ton
(*d*) The total brake horsepower required to drive the compressor. *Ans.* 4.1 hp

4. From the compressor rating tables, select a compressor which will satisfy the following conditions:
(*a*) Required capacity 24,900 Btu/hr
(*b*) Design saturated suction temperature 10° F
(*c*) Design saturated discharge temperature 105° F

13

System Equilibrium and Cycling Controls

13-1. System Balance. In the designing of a refrigerating system, one of the most important considerations is that of establishing the proper relationship or "balance" between the vaporizing and condensing sections of the system. It is important to recognize that whenever an evaporator and a condensing unit are connected together in a common system, a condition of equilibrium or "balance" is automatically established between the two such that the rate of vaporization is always equal to the rate of condensation. That is, the rate at which the vapor is removed from the evaporator and condensed by the condensing unit is always equal to the rate at which the vapor is produced in the evaporator by the boiling action of the liquid refrigerant. Since all the components in a refrigerating system are connected together in series, the refrigerant flow rate through all components is the same. It follows, therefore, that the capacity of all the components must be of necessity the same.

Obviously, then, where the system components are selected to have equal capacities at the system design conditions, the point of system equilibrium or balance will occur at the system design conditions. On the other hand, when the components selected do not have equal capacities at the system design conditions, system equilibrium will be established at operating conditions other than the system design conditions and the system will not perform satisfactorily.

In any event, it is important to understand that, regardless of the equipment selected, the system will always establish equilibrium at some conditions such that all the system components will have equal capacity. Hence, whether or not system equilibrium is established at the system design conditions depends entirely upon whether or not the equipment is selected to have approximately equal capacities at the system design conditions. This concept is best illustrated through the use of a series of examples.

Example 13-1. A walk-in cooler, having a calculated cooling load of 11,000 Btu/hr, is to be maintained at 35° F. The desired evaporator TD is 12° F and the ambient temperature is 90° F. Allowing 3° F (equivalent to approximately 2 lb) for the pressure drop in the suction line (see Section 12-32, select an air-cooled condensing unit and a unit cooler from manufacturer's catalog data.

Solution. Since the design space temperature is 35° F and the design evaporator TD is 12° F, the design evaporator temperature is 23° F (35° F − 12° F). When we allow for a 3° F loss in the suction line resulting from pressure drop, the saturation temperature at the compressor suction is 20° F (23° F − 3° F).

From Table R-11, select 1½ hp condensing unit, which has a capacity of 12,630 Btu/hr at a 20° F saturated suction and 90° F ambient air temperature. Although the condensing unit capacity is somewhat in excess of the calculated load of 11,000 Btu/hr, it is sufficiently close to make the condensing unit acceptable. However, to assure proper system balance, the unit cooler selection must now be based on the condensing unit capacity of 12,630 Btu/hr rather than on the calculated load of 11,000 Btu/hr.* Hence, a unit cooler having a capacity of approximately 12,630 Btu/hr at a 12° F TD is required.

From Table R-8, unit cooler Model #105 has a capacity of 10,500 Btu/hr at a 10° F TD. Using the procedure outlined in Section 11-23, it can be determined that this unit cooler will have a capacity of 12,600 Btu/hr when operating

* Either the evaporator or the condensing unit may be selected first. However, once either one has been selected, the other must be selected for approximately the same capacity.

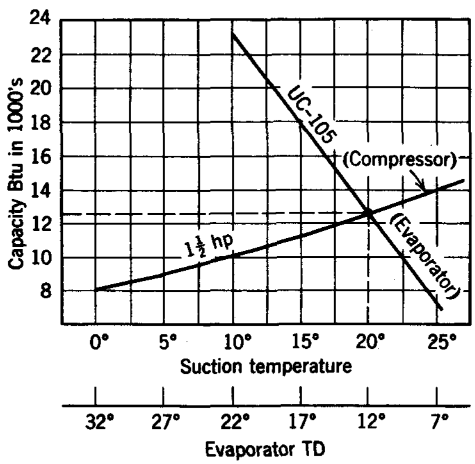


Fig. 13-1. Graphic analysis of system balance.

at a 12° F TD. Since this is very close to the condensing unit capacity, the unit cooler is ideally suited for the application.

13-2. Graphical Analysis of System Equilibrium. For any particular evaporator and condensing unit connected together in a common system, the relationship established between the two, that is, the point of system balance, can be evaluated graphically by plotting evaporator capacity against condensing unit capacity on a common graph. Using data taken from the manufacturers' rating tables, condensing unit capacity is plotted against suction temperature, whereas evaporator capacity is plotted against evaporator TD. A graphical analysis of the system described in Example 13-1 is shown in Fig. 13-1.

In order to understand the graphical analysis of system equilibrium in Fig. 13-1, it is important to recognize that, for any given space temperature, there is a fixed relationship between the evaporator TD and the compressor suction temperature. That is, for any given space temperature, once the evaporator TD is selected, there is only one possible suction temperature which will satisfy the design conditions of the system.

Notice that, in Example 13-1, for the design space temperature of 35° F and assuming a 3° F loss in the suction line, the only possible suction temperature that can coexist with the design evaporator TD of 12° F is 20° F. In this instance, the evaporator TD will be 12° F when, and only when, the suction temperature is 20° F. Any suction temperature other than 20° F will result in an evaporator TD either greater or smaller than 12° F. For example, assume a suction temperature of 25° F. Adding 3° F to allow for the suction line loss, the evaporator temperature is found to be 28° F (25° F + 3° F). Then subtracting the evaporator temperature from the space temperature, it is determined that the evaporator TD will be 7° F (35° F − 28° F) when the suction temperature is 25° F. Using the same procedure, it can be shown that if the suction temperature is reduced to 15° F, the evaporator TD will increase to 17° F, and when the suction temperature is 10° F, the evaporator TD will be 22° F, and so on.

Apparently, then, raising or lowering the suction temperature always brings about a corresponding adjustment in the evaporator TD. Provided that the space temperature is kept constant, raising the suction temperature reduces the evaporator TD, whereas lowering the suction temperature increases the evaporator TD.

With regard to Fig. 13-1, the following procedure is used in making a graphical analysis of the system equilibrium conditions:

1. On graph paper, lay out suitable scales for capacity (Btu/hr), suction temperature (° F), and evaporator TD (° F). The horizontal lines are used to represent capacity, whereas the vertical lines are given dual values, representing both suction temperature and evaporator TD. The latter is meaningful, however, only when the suction temperature and evaporator TD scales are so correlated that the two conditions which identify any one vertical line are conditions which, at the design space temperature, actually represent conditions that will occur simultaneously in the system. The procedure for correlating the suction temperature and evaporator TD scales was discussed in the preceding paragraphs.

2. Using manufacturer's catalog data, plot the capacity curve for the condensing unit. Since condensing unit capacity is not exactly proportional to suction temperature, the condensing unit capacity curve will ordinarily have a slight curvature. Hence, for accuracy, a capacity point is plotted for each of the suction

temperatures listed in the table and these points are connected with the "best-fitting" curve.

3. From the evaporator manufacturer's catalog data, plot the evaporator capacity curve. Since evaporator capacity is assumed to be proportional to the evaporator TD, the evaporator capacity curve is a straight line, the position and direction of which is adequately established by plotting the evaporator capacity at any two selected TDs. The evaporator capacity at any other TD will fall somewhere along a straight line drawn through these two points. In Fig. 13-1, evaporator TDs of 7° F and 12° F are used in plotting the two capacity points required to establish the evaporator capacity curve.

Notice in Fig. 13-1, that as the suction temperature increases, the evaporator TD decreases. This means, in effect, that as the suction temperature increases the capacity of the evaporator decreases while the capacity of the condensing unit (compressor) increases. Likewise, as the suction temperature decreases, the capacity of the evaporator increases while the capacity of the condensing unit decreases. The intersection of the two capacity curves indicates the point of system equilibrium. In this instance, because the evaporator and condensing unit have been selected to have equal capacities at the system design conditions, the point of system equilibrium occurs at the system design conditions (12° F TD and 20° F suction temperature). Although the total system capacity is somewhat greater than the calculated load, the difference is not sufficiently great to be of any particular consequence, and means only that the system will operate fewer hours out of each 24 than was originally anticipated.* The relationship between system capacity and the calculated load is discussed more fully later in the chapter.

It has already been pointed out that where the evaporator and condensing unit selected do not have equal capacities at the system design conditions, the point of system equilibrium will occur at conditions other than the design conditions. For instance, assume that unit cooler Model #UC-120, rather than Model #UC-105, is selected in the foregoing example. At the design evaporator TD of 12° F, this unit cooler has capacity of 14,000 Btu/hr, whereas the condensing unit selected has a capacity of only 12,630 Btu/hr at the design suction temperature of 20° F. Consequently, at the design conditions, evaporator capacity will be greater than condensing unit capacity, that is, vapor will be produced in the evaporator at a greater rate than it is removed from the evaporator and condensed by the condensing unit. Therefore, the system will not be in equilibrium at these conditions. Rather, the excess vapor will accumulate in the evaporator and cause an increase in the evaporator temperature and pressure. Since raising the evaporator temperature increases the suction temperature and, at the same time, reduces the evaporator TD, the condensing unit capacity will increase and the evaporator capacity will decrease. System equilibrium will be established when the evaporator temperature rises to some point where the suction temperature and evaporator TD are such that the condensing unit capacity and the evaporator are equal. In this instance, system equilibrium, as determined graphically (point *A* in Fig. 13-2), is established at a suction temperature of approximately 23° F. The evaporator TD is approximately 9° F, which is 3° F less than the design evaporator TD of 12° F and which will result in a space humidity somewhat higher than the design condition. The total system capacity is approximately 13,500 Btu/hr, which is about 23% greater than the calculated hourly load of 11,000 Btu/hr. This means that the system running time will be considerably shorter than originally calculated. For instance, if the original load calculation is based on a 16-hr running time, the system will operate only about 13 hr out of each 24.

The question immediately arises as to whether or not this system will perform satisfactorily. Although this would depend somewhat on the particular application, the answer is that it probably would not in the majority of cases. There are several reasons for this. First, the evaporator TD of 9° F is considerably less than the design TD of 12° F and would probably

* For simplicity, the heat given off by the evaporator fan motor has been neglected. If this heat is added to the cooling load, the total system capacity would be almost exactly equal to the calculated load. It is not often that equipment can be found which so nearly meets the requirements of an application as in this instance.

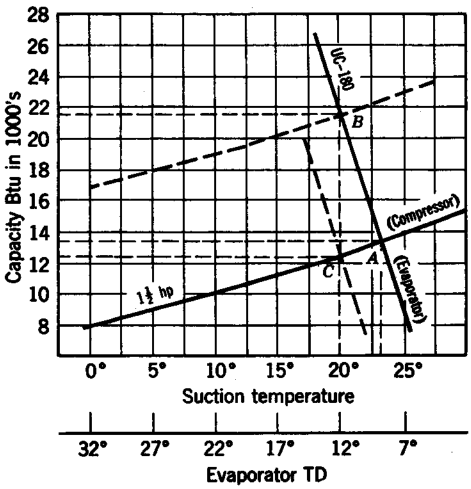


Fig. 13-2.

result in a space humidity too high for the application. Ordinarily, the humidity in the refrigerated space must be maintained within certain fixed limits. Assuming that the design TD is selected to produce the median condition within these limits, a one degree deviation from the design TD in either direction is usually the maximum which can be allowed if the space humidity is to be maintained within the limits specified for the application.

Another consideration is the fact that the system capacity is some 23% greater than the calculated load so that the system operating time will be relatively short. Although the system capacity exceeds the calculated load by a larger margin than good practice prescribes, this in itself would not ordinarily cause any serious problem in a majority of applications. However, since the shorter running time will also tend to aggravate the already existing problem of high humidity, especially in the wintertime, when the two conditions are taken together, it seems unlikely that the system would produce satisfactory results in any application where the space humidity is an important factor.

In the event that the equipment in question represents the best available selection, the question arises as to what can be done to bring the system into balance at conditions more in keeping with the design conditions. In this instance, since the problem is one of excessive evaporator capacity with relation to the condensing unit capacity at the design conditions, logical corrective measures prescribe either an increase in the condensing unit capacity or a reduction in the evaporator capacity in order to re-establish the point of system equilibrium at conditions nearer to the design conditions.

Which of these two measures will produce the most satisfactory results depends upon the relationship between the over-all system capacity and the calculated load. Whereas increasing either the condensing unit capacity or the evaporator capacity will always bring about an increase in the over-all system capacity, reducing either the condensing unit capacity or the evaporator capacity will always bring about a reduction in the over-all system capacity. Referring to Fig. 13-2 for the system under consideration, if the condensing unit capacity is increased to the evaporator capacity at the design conditions, system equilibrium will shift from point *A* to point *B*. On the other hand, if the evaporator capacity is reduced to the condensing unit capacity at the design conditions, the point of system equilibrium will shift from *A* to *C*. Notice that, although the system is balanced at the design conditions at either points *B* or *C*, the over-all system capacity at point *B* is considerably above the calculated load, whereas at point *C* the over-all system capacity very nearly approaches the calculated load. Hence, in this instance, it is evident that increasing the condensing unit capacity as a means of bringing the system into balance at the design conditions cannot be recommended, since it would also increase the over-all system capacity and therefore tend to aggravate the already existing problem of excessive system capacity with relation to the calculated load. On the other hand, in addition to bringing the system into balance at the design conditions, reducing the evaporator capacity will also have the beneficial effect of reducing the over-all system capacity and thereby bringing it more into line with the calculated load.

13-3. Decreasing or Increasing Evaporator Capacity. Reducing the evaporator capacity can be accomplished in several ways. One is to "starve" the evaporator, that is, to reduce the amount of liquid refrigerant in the evaporator by adjusting the refrigerant flow control so that the evaporator is only partially flooded with liquid. This effectively reduces the size of the evaporator, since that part of the evaporator which is not filled with liquid becomes, in effect, a part of the suction line.

Another method of reducing the capacity of the evaporator is to reduce the air velocity over the evaporator by slowing the evaporator fan or blower. However, this method has its limitations in that the air velocity must be maintained at a level sufficient to assure adequate air circulation in the refrigerated space. Too, reducing the air quantity causes a change in the sensible heat ratio of the evaporator. Depending upon the particular application, this may or may not be desirable.

As a general rule, there is little, if anything, that can be done to increase the capacity of an undersized evaporator. Occasionally, the evaporator capacity can be increased by increasing the air quantity. However, since increasing the air quantity also increases air velocity and fan horsepower requirements, this method has its limitations for reasons already discussed in Section 11-22.

In some cases, the evaporator surface area can be increased somewhat by using a length of either bare tubing or finned tubing as a "drier loop" or as additional evaporator surface. However, this too has its limitations because of the pressure drop accruing in the tubing.

13-4. Decreasing or Increasing Condensing Unit Capacity. Decreasing the condensing unit capacity can be accomplished in several ways, all of which involve decreasing the compressor displacement. Probably the simplest and most common method of reducing the condensing unit capacity is to reduce the speed of the compressor by reducing the size of the pulley on the compressor driver. The speed reduction required is approximately proportional to the desired capacity reduction.

The relationship between the speed of the compressor and the speed of the compressor driver is expressed in the following equation:

$$Rpm_1 \times D_1 = Rpm_2 \times D_2 \qquad (13\text{-}1)$$

where Rpm_1 = the speed of the compressor (rpm)

D_1 = the diameter of the compressor flywheel (inches)

Rpm_2 = the speed of the compressor driver (rpm)

D_2 = the diameter of the driver pulley (inches)

NOTE. Where the compressor driver is a four-pole, alternating-current motor operating on 60 cycle power, the approximate driver speed is 1750 rpm. For a two-pole, alternating-current motor, the approximate speed is 3500 rpm.

Example 13-2. A refrigeration compressor having a 10 in. flywheel is driven by a four-pole, alternating-current motor. If the diameter of the motor pulley is 4 in., determine the speed of the compressor.

Solution. Rearranging and applying Equation 13-1, Rpm_1

$$= \frac{Rpm_2 \times D_2}{D_1} = \frac{1750 \times 4}{10} = 700$$

Example 13-3. Determine the diameter of the motor pulley required to reduce the speed of the compressor in Example 13-2 from 700 to 600 rpm.

Solution. Rearranging and applying Equation 13-1, D_2

$$= \frac{Rpm_1 \times D_1}{Rpm_2} = \frac{600 \times 10}{1750} = 3.5 \text{ in.}$$

Another method of reducing condensing unit capacity is to reduce the volumetric efficiency of the compressor by increasing the clearance volume. This increase is accomplished by installing a thicker gasket between the cylinder housing and the valve-plate.

In some cases, small increases in condensing unit capacity can be obtained by merely increasing the speed of the compressor. However, when the capacity increase needed is substantial, it is usually more practical and more economical to use a larger size condensing unit and reduce the capacity as necessary. The reasons for this are several.

First, since the increase in compressor capacity will be accompanied by an increase in the horsepower requirements, any substantial increase in the compressor capacity will tend to overload the compressor driver and necessitate the use of a larger size. Too, some thought must be given to the condenser capacity. Here again, any increase in compressor capacity will tend to place a heavier load on the condenser. If the size of the condenser is not increased in proportion to the increase in the condenser load,

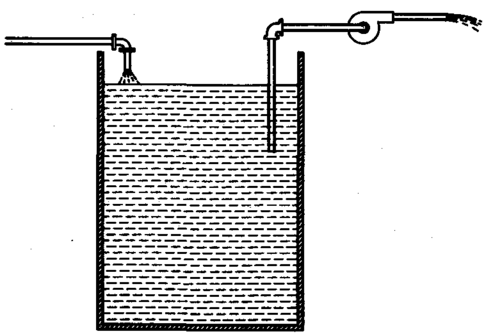


Fig. 13-3

excessive compressor discharge temperature and pressure will result. Not only will this materially reduce the life of the equipment and increase maintenance and operating costs, but it will also tend to nullify to some extent the gain in capacity originally accruing from the increase in compressor speed.

It is apparent from the foregoing that, in most cases, increasing the capacity of either the evaporator or the condensing unit is something which is not easily accomplished. Therefore, it is usually more practical and more economical to select oversized equipment rather than undersized equipment. When the evaporator or the condensing unit is oversized and capacity reduction is required to bring the system components into balance at the desired conditions, the capacity reduction can readily be made with little, if any, loss in system efficiency.

13-5. System Capacity vs. Calculated Load. The relationship between system capacity and system load is one which warrants careful consideration and which can be best explained by comparing the refrigerating system to a water pumping system. For example, assume that it is desired to maintain a constant water level in the tank shown in Fig. 13-3. If the water flows into the tank at a fixed and constant rate which is readily computable, the water in the tank can be maintained at a fixed level simply by installing a pumping system which has a capacity exactly equal to the flow rate of the water into the tank. Since the flow rate of the water entering the tank is constant and since the pumping rate is equal to the water flow rate, the pump will operate continuously and no other water level control of any kind will be needed.

On the other hand, if the flow rate of the water entering the tank varies from time to time, it is evident that if the level of the water is to be maintained within fixed limits, the pumping system must be selected to have a capacity equal to or somewhat in excess of the highest sustained flow rate of the water entering the tank. It is evident also that some means of cycling the pump "off" and "on" must be provided. Otherwise, during periods when the flow rate of the water entering the tank is less than maximum, the pumping rate will be excessive and the level of the water in the tank will be reduced below the desired level. One convenient and practical means of cycling the pump is to install a float control in the tank (Fig. 13-4). The float control is arranged to close the electrical contacts and start the pump when the water in the tank rises to a predetermined maximum level. When the water level in the tank falls to a predetermined lower limit, the float control acts to close the electrical contacts and stop the pump. In this way, intermittent operation of the pump will maintain a relatively constant water level in the tank.

The latter principle is readily applied to the refrigerating system. Since the cooling load on a refrigerating system varies from time to time, the system is usually designed to have a capacity equal to or somewhat in excess of the average maximum cooling load. This is done so that the temperature of the space or product can be maintained at the desired low level even under peak load conditions. As in the case of the water

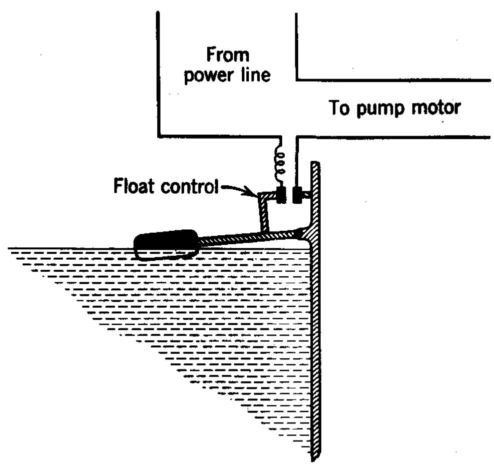


Fig. 13-4

pumping system, since the system refrigerating capacity will always exceed the actual cooling load, some means of cycling the system "off" and "on" is needed in order to maintain the temperature of the space or product at a constant level within reasonable limits and to prevent the temperature of the space or product from being reduced below the desirable minimum.

For any refrigerating system, the relative length of the "off" and "on" cycles will vary with the loading of the system. During periods of peak loading, the "running" or "on" cycles will be long and the "off" cycles will be short, whereas during periods of minimum loading the "on" cycles will be short and the "off" cycles will be long.*

13-6. Cycling Controls. The controls used to cycle a refrigerating system "on" and "off" are of two principal types: (1) temperature actuated (thermostatic) and (2) pressure actuated. Each of these types is discussed in the following sections.

13-7. Temperature Actuated Controls. Temperature actuated controls are called ther-

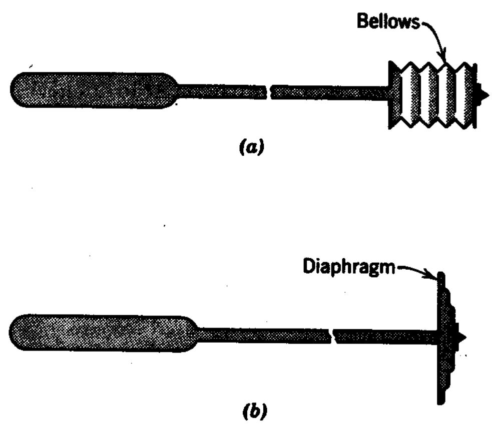


Fig. 13-5. Bulb-type temperature sensing element.

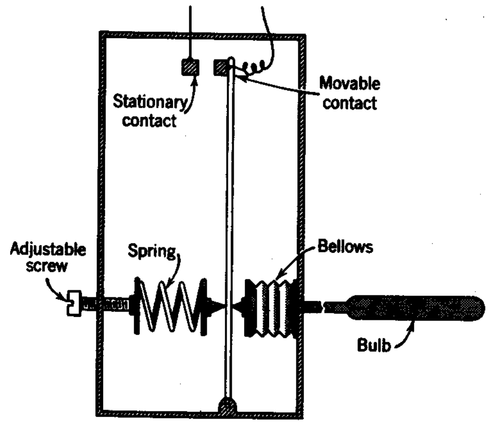


Fig. 13-6. Schematic diagram of simplified pressure control.

mostats. Whereas float controls are sensitive to and are actuated by changes in liquid level, thermostats are sensitive to and are actuated by changes in temperature. Thermostats are used to control the temperature level of a refrigerated space or product by cycling the compressor (starting and stopping the compressor driving motor) in the same way that float controls are used to control liquid level by cycling the pump (starting and stopping the pump motor).

13-8. Temperature Sensing Elements. Two types of elements are commonly used in thermostats to sense and relay temperature changes to the electrical contacts or other actuating mechanisms. One is a fluid-filled tube or bulb which is connected to a bellows or diaphragm and filled with a gas, a liquid, or a saturated mixture of the two (Figs. 13-5*a* and 13-5*b*).† Increasing the temperature of the bulb or tube increases the pressure of the confined fluid which acts through the bellows or diaphragm and a system of levers to close electrical contacts or to actuate other compensating mechanisms (Fig. 13-6). Decreasing the temperature of the tube or bulb will have the opposite effect.

* Unlike the water pumping system, refrigerating systems are designed to have sufficient capacity to permit "off" cycles even during periods of peak loading. This is necessary in order to allow time for defrosting of the evaporator. However, allowances are made for defrosting time in the load calculations (the 24-hr load is divided by the desired running time to obtain the average hourly load) and need not be further considered when selecting the equipment.

† The thermostat described here is called a remote-bulb thermostat. Although there are a number of different types of thermostats, this is the type most frequently used in commercial refrigeration applications. Thermostats are used for many purposes other than controlling a compressor driving motor, as, for example, opening and closing valves, starting and stopping damper motors, etc.

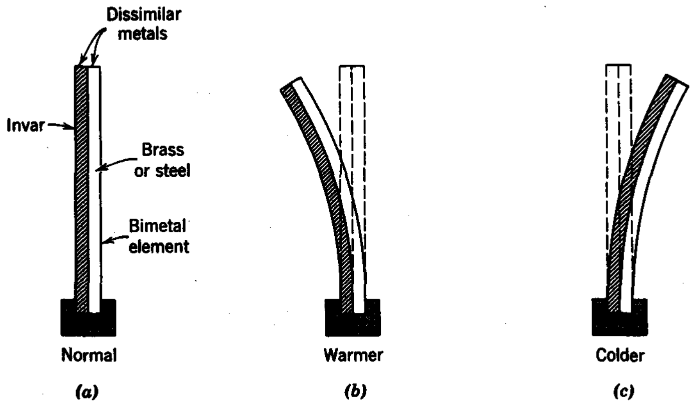


Fig. 13-7. Bimetal-type temperature sensing element.

Another and entirely different temperature sensing element is the compound bar, commonly called a bimetal element. The compound bar is made up of two dissimilar metals (usually Invar and brass or Invar and steel) bonded into a flat strip (Fig. 13-7*a*). Invar is an alloy which has a very low coefficient of expansion, whereas brass and steel have relatively high coefficients of expansions. Since the change in the length of the Invar per degree of temperature change will always be less than that of the brass or steel, increasing the temperature of the bimetal element causes the bimetal to warp in the direction of the Invar (the inactive metal) as shown in Fig. 13-7*b*, whereas decreasing the temperature of the bimetal element causes the bimetal to warp in the direction of the brass or steel (the active metal) as shown in Fig. 13-7*c*. This change in the configuration of the bimetal element with changes in temperature can be utilized directly or indirectly to open and close electrical contacts or to actuate other compensating mechanisms.

13-9. Differential Adjustment. Like float controls, thermostats have definite "cut-in" and "cut-out" points. That is, the thermostat is adjusted to start the compressor when the temperature of the space or product rises to some predetermined maximum (the cut-in temperature) and to stop the compressor when the temperature of the space or product is reduced to some predetermined minimum (the cut-out temperature).

The difference between the cut-in and cut-out temperatures is called the differential. In general, the size of the differential depends upon the particular application and upon the location of the temperature sensing element. Where the temperature sensing element of the thermostat is located in or on the product and controls the product temperature directly, the differential is usually small (2° F or 3° F). On the other hand, where the sensing element is located in the space and controls the space temperature, the differential is ordinarily about 6° F or 7° F. In many instances, the sensing element of the thermostat is clamped to the evaporator so that the space or product temperature is controlled indirectly by controlling the evaporator temperature, in which case the differential used must be larger (15° F to 20° F or more) in order to avoid short-cycling of the equipment.

When the thermostat controls the space or product temperature directly, the average space or product temperature is approximately the median of the cut-in and cut-out temperatures. Therefore, to maintain an average space temperature of 35° F, the thermostat can be adjusted for a cut-in temperature of approximately 38° F and a cut-out temperature of approximately 32° F.

On the other hand, when the space temperature is controlled indirectly by controlling the evaporator temperature, an allowance must be made in the cut-out setting to compensate for the evaporator TD. For example, for an average space temperature of 35° F and assuming an evaporator TD of 12° F, to compensate for the evaporator TD, the cut-out temperature would be set at 20° F (32° F − 12°) rather than at 32° F. Notice that the cut-in temperature is set at 38° F in either case. This is because the space temperature and the evaporator temperature are the same at the time that the system cycles on. After the compressor cycles off the evaporator continues to absorb heat from the space and warms up to the space temperature during the off cycle (Fig. 13-8). Therefore, when the space temperature rises to the cut-in temperature of 38° F, the evaporator will also be at the cut-in temperature of 38° F. As soon as the compressor is started, the evaporator temperature is quickly reduced below the space temperature by an amount approximately equal to the design evaporator TD. Therefore, in this instance, when the space temperature is reduced to 32° F (the desired minimum), the evaporator temperature (which the thermostat is controlling) will be approximately 20° F (12° F less than the space temperature).

Regardless of whether the thermostat controls the space temperature directly or indirectly, proper adjustment of the cut-in and cut-out temperatures is essential to good operation. If the cut-in and cut-out temperatures are set too close together (differential too small) the system will have a tendency to short-cycle (start and stop too frequently). This will materially reduce the life of the equipment and may result in other unsatisfactory conditions. On the other hand, if the cut-in and cut-out temperatures are set too far apart (differential too large), the on and off cycles will be too long and unnecessarily large fluctuations in the average space temperature will result. Naturally, this too is undesirable.

Although approximate cut-in and cut-out temperature settings for various types of applications have been determined by field experience, in many cases it is necessary to use trial-and-error methods to determine the optimum settings for a specific installation.

13-10. Range Adjustment. In addition to the differential, cycling controls have another adjustment, called the "range," which is also associated with the cut-in and cut-out temperatures. Although, like the differential, the range can be defined as the difference between the cut-in and cut-out temperatures, the two are not the same. For example, assume that a thermostat is adjusted for a cut-in temperature of 30° F and a cut-out temperature of 20° F. Whereas the differential is said to be 10° F (30° − 20°), the range is said to be between 30° F and 20° F.

Although it is possible to change the range without changing the differential, it is not possible to change the differential without changing the range. For instance, suppose that the thermostat previously mentioned is readjusted so that the cut-in temperature is raised to 35° F and the cut-out temperature is raised to 25° F. Although the differential is still 10° F (35° − 25°), the operating range of the control is 5° F higher than it was originally, that is, the operating range is now between 35° F and 25° F, whereas previously it was between 30° F and 20° F. In this instance, the range of the control is changed, but the differential remains the same. Under the new control setting the average space temperature will be maintained approximately 5° F higher than under the old setting.

Suppose now that the differential is increased 5° F by raising the cut-in temperature from 30° F

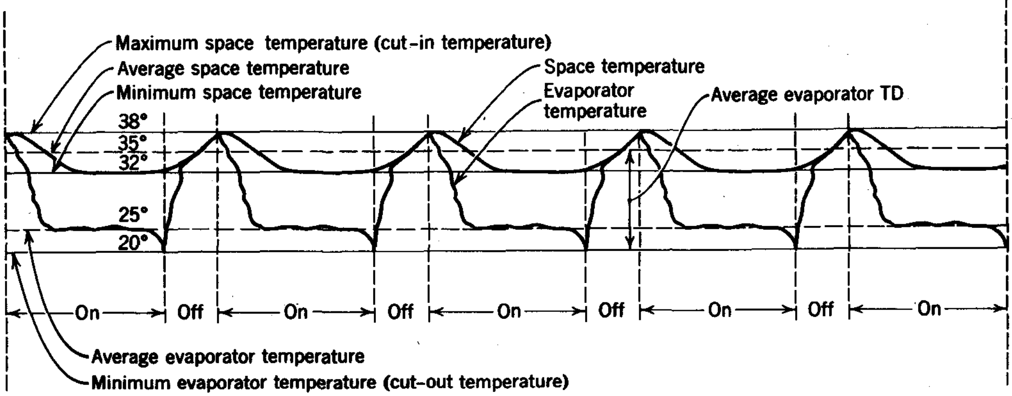


Fig. 13-8. Notice that when the unit cycles "on," the evaporator temperature is the same as the space temperature, whereas when the unit cycles "off," the evaporator temperature is lower than the space temperature by an amount equal to the design evaporator TD.

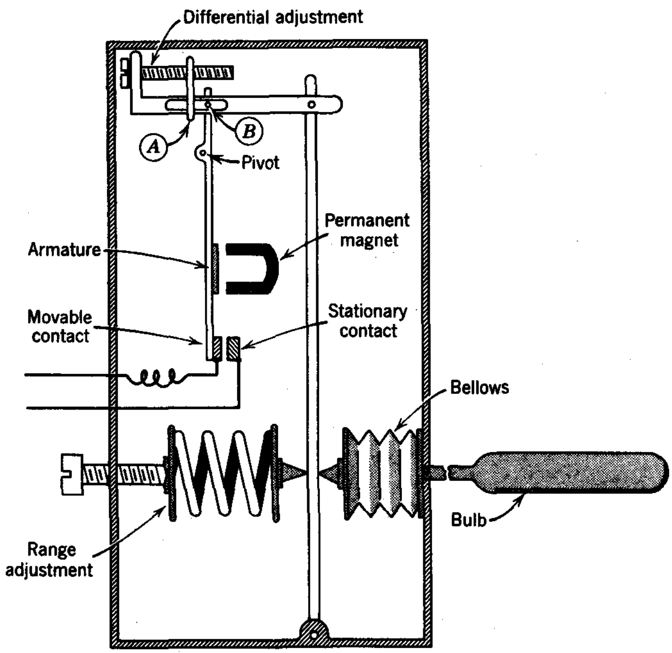


Fig. 13-9. Schematic diagram of thermostatic motor control illustrating range and differential adjustments.

to 35° F while the cut-out temperature is left at the original setting of 20° F. Notice that both the differential and the range are changed. The differential, originally 10° F, is now 15° F and the range, originally between 30° F and 20° F, is now between 35° F and 20° F. With this control setting, the running cycle will be somewhat longer because the differential is larger. Too, the average space temperature will be 2 or 3 degrees higher because the cut-in temperature is higher. If the differential had been increased by lowering the cut-out temperature 5° F rather than by raising the cut-in temperature 5° F, the operating range of the control would have shifted to the opposite direction and the average space temperature would have been 2 or 3 degrees lower than the original space temperature.

Typical range and differential adjustments are shown in Fig. 13-9. Turning the range-adjusting screw clockwise increases the spring tension which the bellows pressure must overcome in order to close the contacts and, therefore, raises both the cut-in and cut-out temperatures. Turning the range-adjusting screw counterclockwise decreases the spring tension and lowers both the cut-in and cut-out temperatures.

Turning the differential-adjusting screw clockwise causes the limit bar *A* to move toward the screw head, thereby increasing the travel of the pin *B* in the slot. This has the effect of increasing the differential by lowering the cut-out temperature. Turning the differential adjusting screw counterclockwise raises the cut-out temperature and reduces the differential. By manipulating both range and differential adjustments, the thermostat can be adjusted for any desired cut-in and cut-out temperatures.

The arrangement shown in Fig. 13-9 represents only one of a number of methods which can be employed to adjust the cut-in and cut-out temperatures. The particular method used in any one control depends on the type of control and on the manufacturer. For example, for the control shown in Fig. 13-9, changing the range adjustment changes both the cut-in and cut-out temperatures simultaneously, whereas for another type of control changing the range adjustment changes only the cut-in temperature. For still another type of control, changing the range adjustment changes only the cut-out temperature. However, whatever the method of adjustment, the principles involved are similar and the exact method of adjustment is readily determined by examining the control. In many cases, instructions for adjusting the control are given on the control itself.

If electrical contacts are permitted to open or close slowly, arcing will occur between the contacts, and burning or welding together of the contacts will result. Therefore, cycling controls which employ electrical contacts must all be equipped with some means of causing the contacts to open and close rapidly in order to avoid arcing. In Fig. 13-9, the armature and permanent magnet serve this purpose. As the pressure in the bellows increases and the movable contact moves toward the stationary contact, the strength of the magnetic field between the armature and the horseshoe magnet increases rapidly. When the movable contact approaches to within a certain, predetermined, minimum distance of the stationary contact, the strength of the magnetic field becomes great enough to overcome the opposing spring tension so that the armature is pulled into the magnet and the contacts are closed rapidly with a snap action.

As the pressure in the bellows decreases, spring tension acts to open the contacts. However, since the force of the spring is opposed somewhat by the force of magnetic attraction, the contacts will not separate until a considerable force is developed in the spring. This causes the contacts to snap open quickly so that arcing is again avoided.

Toggle mechanisms are also frequently used as a means of causing the contacts to open and close with a snap action. Too, some controls employ a mercury switch as a means of overcoming the arcing problem. A typical mercury switch is illustrated in Fig. 13-10. As the glass tube is tilted to the right, the pool of mercury enclosed in the tube make contact between the two electrodes. As the bulb is tilted back to the left, contact is broken. The surface tension of the mercury provides the snap action necessary to prevent arcing.

13-11. Space Control vs. Evaporator Control. When the sensing element of the thermostat is located in the space or in the product, the thermostat controls the space temperature or product temperature directly. Likewise, when the sensing element is clamped to the evaporator, the thermostat controls the evaporator temperature directly. In such cases, control of the space or product temperature is accomplished indirectly through evaporator temperature control. Which of these two methods of control is the most suitable for any given application depends upon the requirements of the application itself.

For applications where close control of the space or product temperature is desired, a thermostat which controls the space or product temperature directly will ordinarily give the best results. On the other hand, for applications where off-cycle defrosting is required and where minor fluctuations in the space or product temperature are not objectionable, indirect control of the space temperature by evaporator temperature control is probably the better method.

In order to assure complete defrosting of the evaporator, the evaporator must be allowed to warm up to a temperature of approximately 37° or 38° F during each off cycle. When the thermostat controls the evaporator temperature, the cut-in temperature of the thermostat can be set for 38° F. Since the evaporator must warm up to this temperature before the compressor can be cycled on, complete defrosting of the evaporator during each off cycle is almost certain. On the other hand, when the thermostat controls the space temperature, there is no assurance that the evaporator will always warm up sufficiently during the off cycle to permit adequate defrosting.

If we assume that the thermostat is properly adjusted, if the load on the system is relatively constant and the capacity of the system is sufficient to handle the load, no defrosting problems are likely to arise with either method of temperature control. However, if the system is subject to considerable changes in load, defrosting problems are sometimes experienced in applications where the thermostat controls the space temperature. When the system is

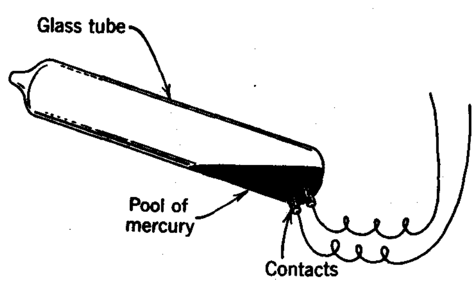


Fig. 13-10. Mercury contacts.

operating under peak load conditions, the temperature of the space tends to remain above the cut-out temperature of the thermostat for extended periods so that running cycles are long and frost accumulation on the evaporator is heavy. Too, under heavy load conditions, the space temperature warms up to the cut-in temperature of the thermostat very quickly during the off cycle so that the off cycles are usually short. Frequently, the off cycles are too short to allow adequate defrosting of the evaporator. In such cases, when the compressor cycles on again, the partially melted frost is caught on the evaporator and frozen into ice. Eventually the evaporator will be completely frozen over with ice, air flow over the evaporator will be severely restricted, and the system will become inoperative.

13-12. Pressure Actuated Cycling Controls. Pressure actuated cycling controls are of two types: (1) low-pressure actuated and (2) high-pressure actuated. Low-pressure controls are connected to the low-pressure side of the system (usually at the compressor suction) and are actuated by the low-side pressure. High-pressure controls, on the other hand, are connected to the high-pressure side of the system (usually at the compressor discharge) and are actuated by the high-side pressure.

The design of both the low-pressure and the high-pressure controls is similar to that of the remote-bulb thermostat. The principal difference between remote-bulb thermostat and the pressure controls is the source of the pressure which actuates the bellows or diaphragm. Whereas the pressure actuating the bellows of the thermostat is the pressure of the fluid confined in the bulb, the pressures actuating the bellows of the low-and-high-pressure controls are the suction and discharge pressures of the compressor, respectively. Like the thermostat, both controls have cut-in and cut-out points which are usually adjustable in the field.

13-13. High-Pressure Controls. High-pressure controls are used only as safety controls. Connected to the discharge of the compressor, the purpose of the high-pressure control is to cycle the compressor off in the event that the pressure on the high-pressure side of the system becomes excessive. This is done in order to prevent possible damaging of the equipment. When the pressure on the high-pressure side of the system rises above a certain, predetermined pressure, the high-pressure control acts to break the circuit and stop the compressor. When the pressure on the high-pressure side of the system returns to normal, the high-pressure control acts to close the circuit and start the compressor. However, some high-pressure controls are equipped with "lock-out" devices which require that the control be reset manually before the compressor can be started again. Although high-pressure controls are desirable on all systems, because of the possibility of a water supply failure, they are essential on systems utilizing water-cooled condensers.

Since the condensing pressures of the various refrigerants are different, the cut-out and cut-in settings of the high-pressure control depend on the refrigerant used.

13-14. Low-Pressure Controls. Low-pressure controls are used both as safety controls and as temperature controls. When used as a safety control, the low-pressure control acts to break the circuit and stop the compressor when the low-side pressure becomes excessively low and to close the circuit and start the compressor when the low-side pressure returns to normal. Like high-pressure controls, some low-pressure controls are equipped with a lock-out device which must be manually reset before the compressor can be started.

Low-pressure controls are frequently used as temperature controls in commercial refrigeration applications. Since the pressure at the suction inlet of the compressor is governed by the saturation temperature of the refrigerant in the evaporator, changes in evaporator temperature are reflected by changes in the suction pressure. Therefore, a cycling control actuated by changes in the suction pressure can be utilized to control space temperature indirectly by controlling the evaporator temperature in the same way that the remote-bulb thermostat is used for this purpose. In such cases, the cut-in and cut-out pressures of the low-pressure control are the saturation pressures corresponding to the cut-in and cut-out temperatures of a remote-bulb thermostat employed in the same application. For example, assume that for a certain application the cut-in and cut-out temperature settings for a remote-bulb thermostat are 38° F and 20° F, respectively. If a low-pressure control is used in place of the thermostat, the cut-in pressure setting for the low-pressure control will be 50 psia (the saturation pressure of R-12 corresponding to a temperature of 38° F) and the cut-out pressure setting will be 36 psia (the saturation pressure of R-12 corresponding to a temperature of 20° F).*

As the evaporator warms up during the off cycle, the pressure in the evaporator increases accordingly. When the pressure in the evaporator rises to the cut-in pressure setting of the low-pressure control, the low-pressure control acts to close the circuit and start the compressor. Very soon after the compressor starts, the temperature and pressure of the evaporator are reduced to approximately the design evaporator temperature and pressure and they remain at this condition throughout most of the running cycle (see Fig. 13-8). Near the end of the running cycle the evaporator temperature and pressure are gradually reduced below the design conditions. When the evaporator pressure is reduced to the cut-out pressure setting of the low-pressure control, the control acts to break the circuit and stop the compressor.

Since the refrigerant vapor undergoes a drop in pressure while flowing through the suction line, the pressure of the vapor at the suction inlet of the compressor is usually 2 or 3 lb less than the evaporator pressure. This is particularly true when the compressor is located some distance from the evaporator. Since the low-pressure control is actuated by the pressure at the suction inlet of the compressor, the pressure drop accruing in the suction line must be taken into account when the pressure control settings are made. To compensate for the pressure loss in the suction line, the cut-out pressure setting is lowered by an amount equal to the pressure loss in the line. For example, assuming a 3-lb pressure loss in the suction line, when the pressure in the evaporator is 36 psia, the pressure at the suction inlet of the compressor will be 33 psia. Hence, if it is desired to cycle the compressor off when the pressure in the evaporator is reduced to 36 psia, the cut-out pressure of the low-pressure control is set for 33 psia. In this instance, failure to make an allowance for the pressure loss in the suction line would cause the control to cycle the compressor when the pressure in the evaporator was reduced to only 39 psia rather than the desired 36 psia. The system would have a tendency to short cycle because the differential is too small and unsatisfactory operation would result.

Pressure loss in the suction line in no way affects the cut-in setting of the control. Since pressure drop is a function of velocity or flow, there is no pressure drop in the suction line when the system is idle. As soon as the compressor cycles off, the pressure at the suction of compressor rises to the evaporator pressure so that at the time the compressor cycles on the pressure at the compressor inlet is the same as the evaporator pressure. Hence, the cut-in pressure setting of the control is made without regard for the pressure drop in the suction line.

Since the low-pressure control controls evaporator temperature rather than space temperature, it is an ideal temperature control for applications requiring off-cycle defrosting. This is particularly true for "remote" installations where the compressor is located some distance from the evaporator. In such installations, low-pressure temperature control has a distinct advantage over thermostatic temperature control in that it ordinarily results in a considerable saving in electrical wiring. Because of the remote bulb, the thermostat must always be located near the evaporator or space whose temperature is being controlled. This requires that a pair of electrical conductors be installed between the fixture and the condensing unit. On the other hand, the low-pressure control is located at the compressor near the power source so that the amount of control wiring needed is much less.

13-15. Dual-Pressure Controls. A dual-pressure control is a combination of the low-and-high-pressure controls in a single control. Ordinarily, only one set of electrical contact points are used in the control, although a separate bellows assembly is employed for each of the two pressures. A typical dual pressure control is shown in Fig. 13-11. This type of pressure control is frequently supplied as standard equipment on condensing units.

13-16. The Pump-Down Cycle. A commonly used method of cycling the condensing unit, known as a "pump-down cycle," employs both a

* When refrigerants other than R-12 are used in the system, the pressure settings will be the saturration pressures of those refrigerants corresponding to the desired cut-in and cut-out temperatures.

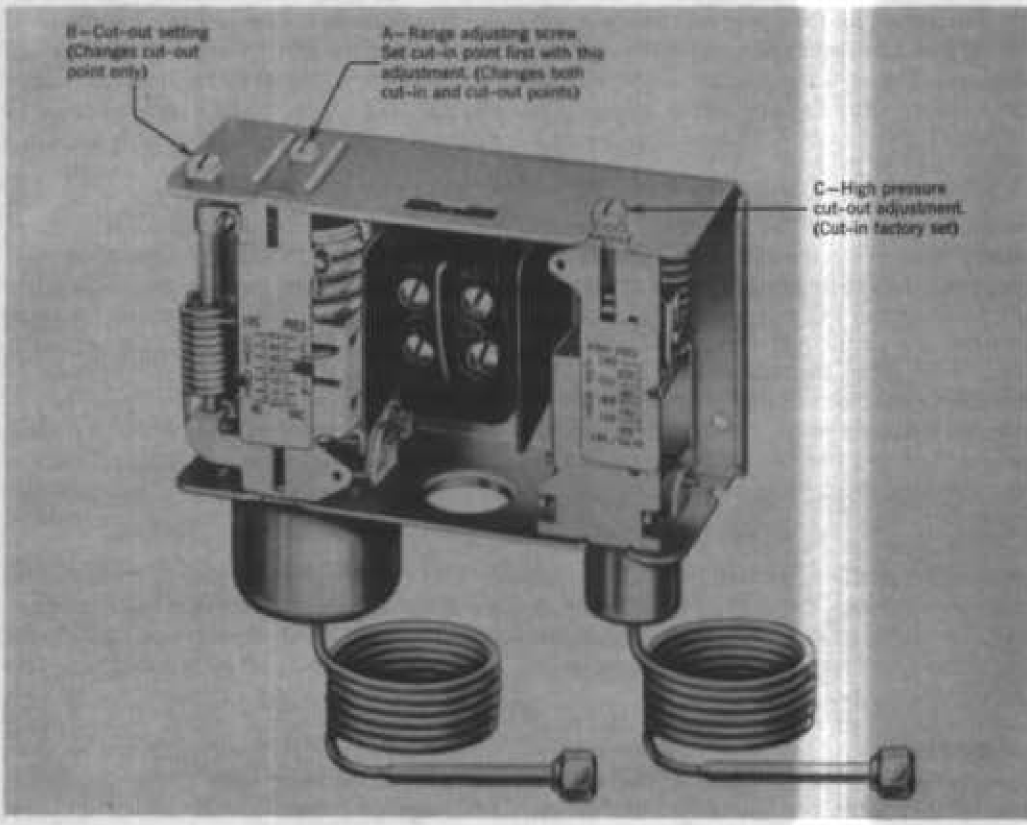


Fig. 13-11. Dual pressure control. (Courtesy of Penn Controls, Inc.)

thermostat and a low-pressure control. In a pump-down cycle, the space or evaporator temperature is controlled directly by the thermostat. However, instead of starting and stopping the compressor driver, the thermostat acts to open and close a solenoid valve installed in the liquid line, usually near the refrigerant flow control (Fig. 13-12). As the space or evaporator temperature is reduced to the cut-out temperature of the thermostat, the thermostat breaks the solenoid circuit, thereby de-energizing the solenoid and interrupting the flow of liquid refrigerant to the evaporator. Continued operation of the compressor causes evacuation of the refrigerant from that portion of the system beyond the point where the refrigerant flow is interrupted by the solenoid. When the pressure in the evacuated portion of the system is reduced to the cut-out pressure of the low pressure control, the low-pressure control breaks the compressor driver circuit and stops the compressor. When the temperature of the space or evaporator rises to the cut-in temperature of the thermostat, the thermostat closes the solenoid circuit and energizes the solenoid, thereby opening the liquid line and permitting liquid refrigerant to enter the evaporator. Since the evaporator is warm, the liquid entering the evaporator vaporizes rapidly so that the evaporator pressure rises immediately to the cut-in pressure of the low-pressure control, whereupon the low-pressure control closes the compressor driver circuit and starts the compressor.

The advantages of the pump-down cycle are many. One of the most important ones being that the amount of refrigerant absorbed by the oil in the crankcase of the compressor during the off cycle is substantially reduced. The problem of crankcase oil dilution by refrigerant absorption during the off cycle is fully discussed in Chapter 18.

13-17. Variations in System Capacity. It is worthwhile to notice that both the operating conditions and the capacity of a refrigerating system change as the load on the system changes. When the load on the system is heavy and the space temperature is high, the evaporator TD will be somewhat larger than the design evaporator TD and the capacity of the evaporator will be greater than the design evaporator capacity. Because of the higher evaporator capacity, the suction temperature will also be higher so that equilibrium is maintained between the vaporizing and condensing sections of the system. Hence, under heavy load conditions, the system operating conditions are somewhat higher than the average design conditions and the system capacity is somewhat greater than the average design capacity. Obviously, the horsepower requirements of the compressor are greatest at this peak load condition and the compressor driver must be selected to have sufficient horsepower to meet these requirements.

Conversely, when the load on the system is light, the space temperature will be lower than the average design space temperature, the evaporator TD will be less than the design TD, and the suction temperature will be lower than the design suction temperature. Therefore, the system operating conditions will be somewhat lower than the average design operating conditions and the system capacity will be somewhat less than the average design capacity.

During each running cycle the system passes through a complete series of operating conditions and capacities, the operating conditions and capacity being highest at the beginning of the running cycle when the space temperature is highest, and lowest at the end of the running cycle when the space temperature is lowest. However, during most of the running cycle, a well-designed system will operate very nearly at the design conditions.

13-18. Capacity Control. The importance of balancing the system capacity with the system load cannot be overemphasized. Any time the system capacity deviates considerably from the system load, unsatisfactory operating conditions will result. It has already been pointed out that good practice requires that the system be designed to have a capacity equal to or slightly in excess of the average maximum sustained load. This is done so that the system will have sufficient capacity to maintain the temperature and humidity at the desired level during periods of peak loading. Obviously, as the cooling load decreases, there is a tendency for the system to become oversized in relation to the load.

In applications where the changes in the average system load are not great, capacity control is adequately accomplished by cycling the system on and off as described in the preceding sections. In such cases, assuming that the cycling controls are properly adjusted, the relative length of the on and off cycles will vary with the load on the system. During periods when the load is heavy, on cycles will be long and off cycles will be short, whereas during periods when the load is light, on cycles will be short and off cycles will be long. Naturally, the degree of variation in the length of the on and off cycles will depend on the degree of load fluctuation. However, since the system must always be designed to have sufficient capacity to handle the maximum load, when the changes in the system load are substantial, it is evident that the system will be considerably oversized when the load is at a minimum. A system which is oversized for the load will usually prove to be as unsatisfactory as one that

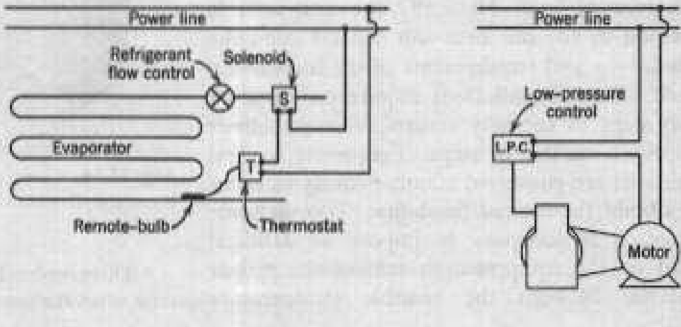


Fig. 13-12. Pump-down cycle.

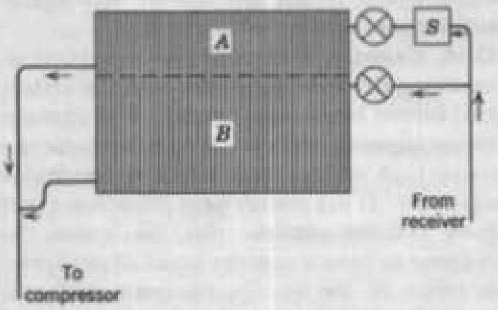


Fig. 13-13. Evaporator split into two segments for capacity control. Closing the solenoid valve in the liquid line of segment *A* renders this portion of the evaporator inoperative. The capacity reduction obtained is proportional to the surface area cycled out.

is undersized for the load. When the system is undersized for the load, the running time will be excessive, the space temperature will be high for extended periods, and the off cycles will be too short to permit adequate defrosting of the evaporator. On the other hand, where the system is oversized for the load, the off cycles will be too long and the equipment running time will be insufficient to remove the required amount of moisture from the space. This will result in unsatisfactory (higher than normal) humidity conditions in the refrigerated space.

For this reason, when changes in the system load are substantial, it is usually necessary to provide some means of automatically (or manually) varying the capacity of the system other than by cycling the system on and off. This is true also of large installations where the size of the equipment renders cycling the system on and off impractical.

There are many methods of bringing the refrigerating capacity into balance with the refrigerating load. Naturally, the most suitable method in any one case will depend upon the conditions and requirements of the installation itself. Some installations require only one or two steps of capacity control, whereas others require a number of steps. Frequently, several methods are employed simultaneously in order to obtain the desired flexibility. Too, in some cases, it is necessary to impose an artificial load on the equipment to achieve the proper balance between the sensible (temperature reduction) and latent (moisture reduction) loads.* In applications where the latent load is too large a percentage of the total load, satisfaction of the latent load will result in overcooling of the space unless sensible heat is artificially introduced into the space. In such cases, the sensible heat is usually added to the space in the form of reheat. The air is first passed across a cooling coil and cooled to the temperature necessary to reduce the moisture content to the desired level and the air is then reheated to the required dry bulb temperature. The reheating is accomplished with steam or hot water coils, with electric strip-heaters, or with hot gas from the compressor discharge.

In some installations, the refrigerating capacity of the system is adequately controlled by controlling the capacity of the compressor only. Since the flow rate of the refrigerant must be the same in all components, any change in the capacity of any one component will automatically result in a similar adjustment in the capacity of all the other components. Therefore, increasing or decreasing the capacity of the compressor will, in effect, increase or decrease the capacity of the entire system. However, it is important to notice that with this method of capacity control the operating conditions of the system will change as the capacity of the system changes.

Where it is desired to vary the capacity of the system without allowing the operating conditions of the system to change, it is necessary to

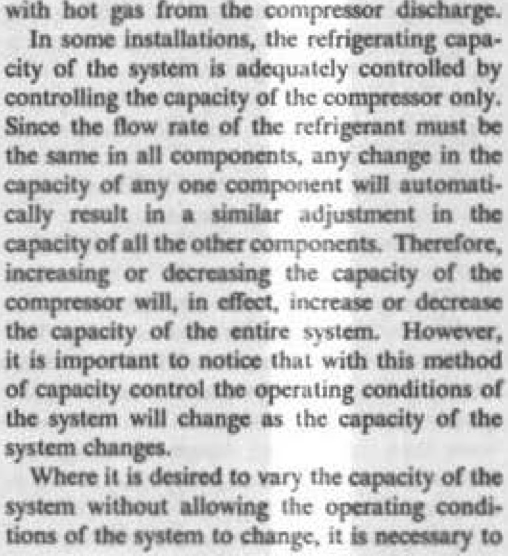


Fig. 13-14. Coil circuited for face control. (Courtesy Kennard Division, American Air Filter Company, Inc.)

* This problem is usually more acute in the wintertime when the transmission (wall gain) load is light.

control both the evaporator capacity and the compressor capacity directly.*

Some of the more common methods of controlling evaporator and compressor capacities are considered in the following sections.

13-19. Evaporator Capacity Control. Probably the most effective method of providing evaporator capacity control is to divide the evaporator into several separate sections or circuits which are individually controlled so that one or more sections or circuits can be cycled out as the load decreases (Fig. 13-13). Using this method, any percentage of the evaporator capacity can be cycled out in any desired number of steps. The number and size of the individual evaporator sections depend on the number of steps of capacity desired and the percent change in capacity required per step, respectively. The arrangement of the evaporator sections or circuiting depends on the relationship of the sensible load to the total load at the various load conditions. Basically, two circuit arrangements are possible. Evaporator circuiting can be arranged to provide either "face" control of "depth" control, or both (Fig. 13-14 and 13-15). When "face" control is used, the "sensible heat ratio" of the evaporator is not

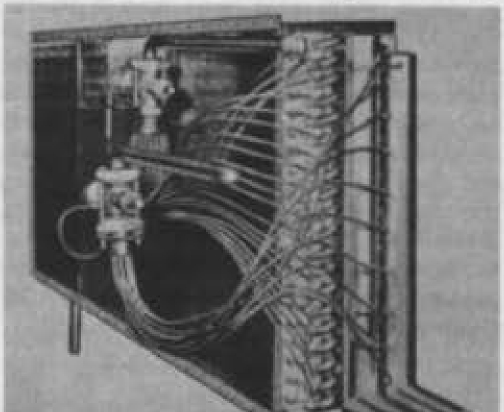


Fig. 13-15. Coil circuited for depth control. (Courtesy Kennard Division, American Air Filter Company, Inc.).

* Any reduction in system load and/or system capacity will also have some effect on the capacity of the condenser and on the size of the refrigerant lines. These topics are discussed in Chapters 14 and 17, respectively.

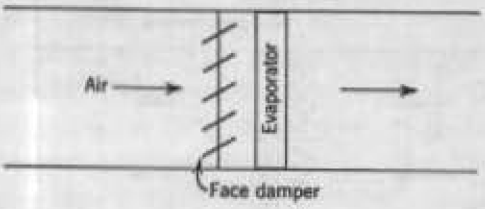


Fig. 13-16. Evaporator equipped with a face damper to vary the quantity of air passing over the evaporator. As the damper moves toward the closed position, the resistance against which the blower must work is increased so that the total quantity of air circulated decreases.

affected.† On the other hand, "depth" control always changes the sensible heat ratio. As a general rule, the more depth the evaporator has the greater is its latent cooling (moisture removal) capacity. Hence, as one or more rows of the evaporator are cycled out, the sensible heat ratio increases.

Another common method of varying the evaporator capacity is to vary the amount of air circulated over the evaporator through the use of "face" or "face-and-by-pass" dampers. (Figs. 13-16 and 13-17). Multispeed blowers can also be used for this purpose. Too, in some instances, multispeed blowers and dampers are used together in order to provide the desired balance.

In nearly all cases, application of any of the foregoing methods of evaporator capacity control will necessitate simultaneous control of compressor capacity.

13-20. Compressor Capacity Control. There are a number of different methods of controlling the capacity of reciprocating compressors. One method, already mentioned, is to vary the speed of the compressor by varying the speed of the compressor driver. When an engine or turbine is employed to drive the compressor, the compressor capacity can be modulated over a wide range by governor control of the compressor driver.

When an electric motor drives the compressor, only two speeds are usually available so that the compressor operates either at full capacity or at 50% capacity. When more than two speeds are desired, it is necessary to use two separate windings in the motor, in which case four speeds will be possible.

† Ratio of the sensible cooling capacity to the total cooling capacity.

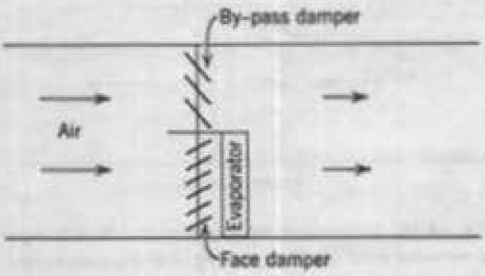


Fig. 13-17. Evaporator equipped with face and by-pass dampers for capacity control. Dampers are interconnected so that by-pass damper opens wider as face damper is closed off. With this arrangement the quantity of air passing over the evaporator can be regulated by allowing more or less air to by-pass the evaporator. However, regardless of the position of the dampers, the total quantity of air circulated remains practically the same.

Capacity control of multicylinder compressors is frequently obtained by "unloading" one or more cylinders so that they become ineffective. One method of accomplishing this is to by-pass the discharge from one or more cylinders back into the suction line as shown in Fig. 13-18. When the suction pressure drops to a certain predetermined value a solenoid valve in the by-pass line, actuated by a pressure control, opens and allows the discharge from one or more cylinders to flow through the by-pass line back into the suction line where it mixes with the incoming suction vapor. As long as the suction pressure remains below the cut-in setting of the control, the discharge from the unloaded cylinders continues to by-pass to the suction line. When the suction pressure rises to the cut-out setting of the pressure-control, the solenoid valve is de-energized and the by-pass line is closed so that the compressor is returned to full capacity operation.

Another method of unloading compressor cylinders is to depress the suction valves of the cylinder or cylinders to be unloaded so that they remain open during the compression (up) stroke. With the suction valves held open, the suction vapor drawn into the cylinder during the suction stroke is returned to the suction line during the compression stroke. A typical unloader of this type is shown in Fig. 13-19.

The operation of the unloader mechanism is as follows: when the suction pressure falls to the cut-in pressure of the pressure control, the control energizes the solenoid valve and admits high-pressure gas from the condenser to the unloader piston which acts to depress the suction valves and hold them open. When the suction pressure rises to the cut-out pressure of the pressure control, the solenoid valve is de-energized and the unloader piston is returned to the normal position.

In addition to providing capacity control, cylinder unloaders of all types are used to unload the compressor cylinders during compressor start-up so that the compressor starts in an unloaded condition, thereby reducing the inrush current demand.

When any of the capacity control methods described thus far are used, the horsepower requirements of the compressor decrease as the capacity decreases, although not in the same proportions.

Another method of controlling compressor capacity is to throttle the compressor suction. However, since it reduces compressor capacity without reducing compressor horsepower, this method is seldom used.

Still another means of controlling compressor capacity which is employed with good results is to operate two or more compressors in parallel (Fig. 13-20). Individual low-pressure controls are used to cycle the compressors. The cut-in and cut-out pressures of the individual controls are so adjusted that the compressors cycle off in sequence as the suction pressure decreases and cycle on in sequence when the suction pressure rises. Very often these compressors are equipped with cylinder unloaders to provide additional steps of control. Multiple compressor systems are discussed in detail in Chapter 20.

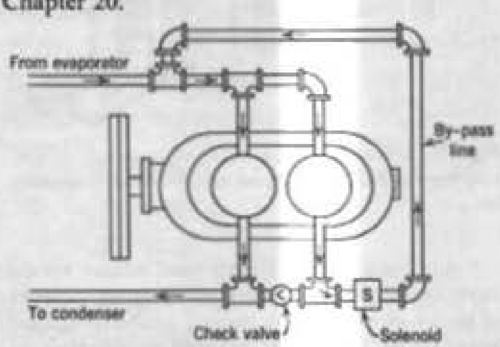


Fig. 13-18. Schematic diagram of cylinder by-pass.

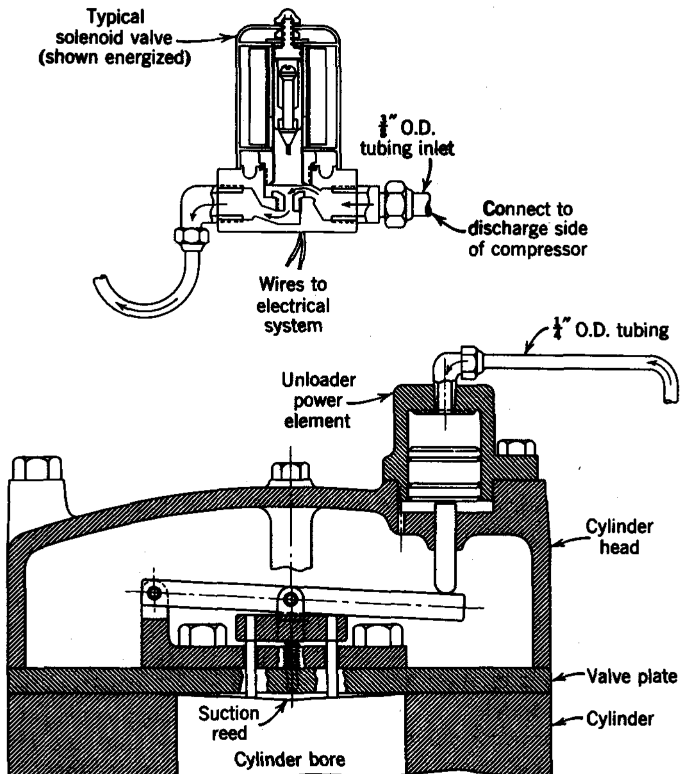


Fig. 13-19. Condenser pressure actuated cylinder unloader mechanism. (Courtesy Dunham-Bush, Inc.)

13-21. Multiple-System Capacity Control. Another method of controlling capacity is to employ two or more separate systems. The evaporators for the separate systems may be in the same housing and air stream or they may be in separate housings and air streams. In either case, separate compressors and condensers are used, although in some instances the condensers may be in a common housing.

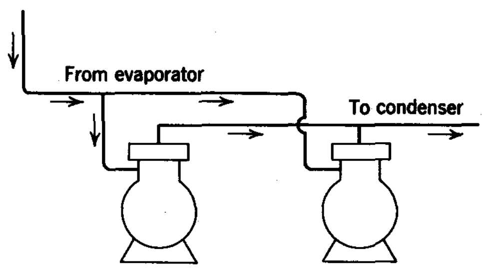


Fig. 13-20. Two compressors installed in parallel as a means of controlling compressor capacity. As the load diminishes, one compressor is cycled out to reduce the compressor capacity. Often, one compressor is equipped with cylinder unloaders to provide additional steps of control.

This method of capacity control is well suited to installations where only two steps of capacity control are required, as in chilling or combination chilling and storage applications. The use of two or more separate systems has the added advantage of providing a certain amount of insurance against losses accruing from equipment failure. Should one system become inoperative, the other can usually hold the load until repairs can be made.

PROBLEMS

1. Assuming a 2° F loss in saturated suction temperature due to refrigerant pressure drop in the suction line.
   (1) Select an air-cooled condensing unit to operate in conjunction with the natural convection evaporator in Problem 12-1.
   (2) Plot the evaporator and condensing unit capacities on a common graph and determine:
   (*a*) The saturated suction temperature at the point of system balance
   (*b*) The capacity of the system in Btu/hr at the point of system balance.

14

Condensers and Cooling Towers

14-1. Condensers. Like the evaporator, the condenser is a heat transfer surface. Heat from the hot refrigerant vapor passes through the walls of the condenser to the condensing medium. As the result of losing heat to the condensing medium, the refrigerant vapor is first cooled to saturation and then condensed into the liquid state.

Although brine or direct expansion refrigerants are sometimes used as condensing mediums in low temperature applications, in the great majority of cases the condensing medium employed is either air or water, or a combination of both.

Condensers are of three general types: (1) air-cooled, (2) water-cooled, and (3) evaporative. Air-cooled condensers employ air as the condensing medium, whereas water-cooled condensers utilize water to condense the refrigerant. In both the air-cooled and water-cooled condensers, the heat given off by the condensing refrigerant increases the temperature of the air or water used as the condensing medium.

Evaporative condensers employ both air and water. Although there is some increase in the temperature of the air passing through the condenser, the cooling of the refrigerant in the condenser results initially from the evaporation of the water from the surface of the condenser. The function of the air is to increase the rate of evaporation by carrying away the water vapor which results from the evaporating process.

14-2. The Condenser Load. Since the heat given up by the refrigerant vapor to the condensing medium includes both the heat absorbed in the evaporator and the heat of compression, the heat load on the condenser always exceeds that on the evaporator by an amount equal to the heat of compression. Since the work (heat) of compression per unit of refrigerating capacity depends upon the compression ratio, the heat load on the condenser per unit of evaporator load varies with the operating conditions of the system.

The quantity of heat liberated at the condenser per minute per ton of evaporator capacity at various suction and condensing temperatures can be estimated from Charts 14-1, 14-2, and 14-3. Chart 14-1 applies to R-12 systems, whereas Charts 14-2 and 14-3 apply to R-22 and R-717 (ammonia) systems, respectively. The values given are based on a simple saturated cycle.

Example 14-1. An R-12 system, operating at a 15° F suction temperature, has a condensing temperature of 100° F. Determine the load on the condenser in Btu per minute per ton.

Solution. In Chart 14-1, locate the 15° F suction temperature line at the base of the graph. Follow the line until it intersects the 100° F condensing temperature curve. The value on the left-hand index corresponding to this point is approximately 245 Btu per minute per ton.

It is evident that for any given set of operating conditions there is a fixed relationship between the condenser load and the evaporator load. For instance, for the R-12 system described in Example 14-1, the relationship between the condenser load and the evaporator load is such that 245 Btu are liberated at the condenser for each 200 Btu taken in at the evaporator.

Once the relationship between the condenser load and the evaporator load has been established for any given set of operating conditions, the total condenser load corresponding to any given total evaporator load can be easily computed. The following equation may be used:

$$Q_c = Q_e \times \frac{q_c}{q_e} \qquad (14\text{-}1)$$

where Q_c = the condenser load in Btu/hr
Q_e = the evaporator load in Btu/hr
q_c = the condenser load in Btu/min/ton (from Fig. 14-15)
q_e = the evaporator load in Btu/min/ton (always 200 Btu)

NOTE: Q_e may also be in Btu/min or in tons, in which case Q_c will be in Btu/min or in tons, respectively.

Example 14-2. For the system described in Example 14-1, determine the load on the condenser if the load on the evaporator is 35,000 Btu/hr.

Solution. Applying
Equation 14-1, the load $= 35{,}000 \times 245/200$
on the condenser, Q_c $= 42{,}875$ Btu/hr

It is important to notice that any increase or decrease in the load on the evaporator (system) will result in a proportional increase or decrease in the load on the condenser.

14-3. Condenser Capacity. Since heat transfer through the condenser walls is by conduction, condenser capacity is a function of the fundamental heat transfer equation:

$$Q = A \times U \times D \qquad (14\text{-}2)$$

where Q = the condenser capacity (Btu/hr)
A = the surface area of the condenser (sq ft)
U = the transfer coefficient of the condenser walls (Btu/hr/sq ft/° F)
D = the log mean temperature difference between the condensing refrigerant and the condensing medium

Examination of the factors in Equation 14-2 will show that for any fixed value of U the capacity of the condenser depends on the surface area of the condenser and on the temperature difference between the condensing refrigerant and the condensing medium. It is evident also that for any one condenser of specific size and design, wherein the surface area and the U factor are both fixed at the time of manufacture, the capacity of the condenser depends only on the temperature differential between the refrigerant and the condensing medium. Therefore, for any one specific condenser, the capacity of the condenser is increased or decreased only by increasing or decreasing the temperature differential. Furthermore, if it is assumed that the average temperature of the condensing medium is constant, it follows that an increase or a decrease in the capacity of the condenser is brought about only by an increase or a decrease, respectively, in the condensing temperature.

Since the condenser capacity must always be equal to the condenser load, it is evident from the foregoing that, for any given condensing load, the larger the surface area of the condenser, the smaller will be the required temperature differential and the lower will be the condensing temperature. Too, since the load on the condenser is always proportional to the load on the evaporator (system), any increase or decrease in the load on the evaporator will be reflected by an increase or a decrease, respectively, in the condensing temperature.

14-4. Quantity and Temperature Rise of Condensing Medium. In both the air-cooled and water-cooled condensers, all the heat given off by the condensing refrigerant increases the temperature of the condensing medium. Therefore, in accordance with Equation 2-8, the temperature rise experienced by the condensing medium in passing through the condenser is directly proportional to the condenser load and inversely proportional to the quantity and specific heat of the condensing medium, viz:

$$(T_2 - T_1) = \frac{Q_s}{M \times C} \qquad (14\text{-}3)$$

where T_1 = the temperature of the air or water entering the condenser (T_e)
T_2 = the temperature of the air or water leaving the condenser (T_1)
$(T_2 - T_1)$ = the temperature rise (ΔT) experienced in the condenser
Q_s = the load on the condenser in Btu per hour
M = the weight of air or water circulated through the condenser in pounds per hour
C = the specific heat at constant pressure of the air or water

Assuming that C has a constant value, for any given condenser load (Q_s), Equation 14-2 contains only two variables, M and ΔT, the value of each being inversely proportional to the value of the other, viz:

$$M = \frac{Q_s}{C \times \Delta T} \qquad (14\text{-}4)$$

$$\Delta T = \frac{Q_s}{C \times M} \qquad (14\text{-}5)$$

Therefore, for any given condenser load, if the temperature rise of the condensing medium is known, the quantity of condensing medium circulated through the condenser in pounds per hour can be determined by applying Equation 14-4. Likewise, if the quantity circulated is known, the temperature rise through the condenser can be computed by applying Equation 14-5.

Average specific heat values for air and water are 0.24 Btu/lb and 1 Btu/lb, respectively. By substituting the appropriate value for C, Equations 14-4 and 14-5 can be written to apply specifically to either water or air, viz:

for water $$M = \frac{Q_s}{\Delta T} \qquad (14\text{-}6)$$

$$\Delta T = \frac{Q_s}{M} \qquad (14\text{-}7)$$

and for air $$M = \frac{Q_s}{0.24 \times \Delta T} \qquad (14\text{-}8)$$

$$\Delta T = \frac{Q_s}{0.24 \times M} \qquad (14\text{-}9)$$

Since general practice is to express air and water quantities in cubic feet per minute (cfm) and in gallons per minute (gpm), respectively, it is usually desirable to compute condensing medium quantities in these units rather than in pounds per hour.

To convert pounds of water per hour into gallons per minute, divide by 60 min to reduce pounds per hour to pounds per minute, and then divide by 8.33 lb per gallon to convert pounds per minute to gallons per minute, viz:

$$\text{gpm} = \frac{M(\text{lb/hr})}{60 \text{ min} \times 8.33 \text{ lb/gal}}$$

If these conversion factors are incorporated into Equation 14-6, the water quantity can be computed directly in gpm. The following equation results:

$$\text{gpm} = \frac{Q_s}{60 \times 8.33 \times \Delta T}$$

or, combining constants ($60 \times 8.33 = 500$),

$$\text{gpm} = \frac{Q_s}{500 \times \Delta T} \qquad (14\text{-}10)$$

To reduce pounds of air per hour to cubic feet per minute, divide pounds per hour by 60 min to determine pounds per minute and then multiply by the specific volume of the air to convert from pounds per minute to cubic feet per minute, viz:

$$\text{cfm} = \frac{M(\text{lb/hr}) \times v(\text{cu ft/lb})}{60 \text{ min}}$$

Assuming the specific volume of the air to be the specific volume of standard air (13.34 cu ft/lb), incorporation of these conversion factors into Equation 14-7 results in the following:

$$\text{cfm} = \frac{Q_s \times 13.34 \text{ cu ft/lb}}{0.24 \times 60 \times \Delta T}$$

or, combining constants ($13.34/0.24 \times 60 = 1/1.08$),

$$\text{cfm} = \frac{Q_s}{1.08 \times \Delta T} \qquad (14\text{-}11)$$

Example 14-3. If the load on a water-cooled condenser is 150,000 Btu/hr and the temperature rise of the water in the condenser is 10° F. What is the quantity of water circulated in gpm?

Solution. Applying Equation 14-10, the water quantity in gpm
$$= \frac{150{,}000}{500 \times 10} = 30 \text{ gpm}$$

Example 14-4. The load on a water-cooled condenser is 90,000 Btu/hr. If the quantity of water circulated through the condenser is 15 gallons per minute, determine the temperature rise of the water in the condenser.

Solution. Rearranging and applying Equation 14-10, ΔT
$$= \frac{90{,}000}{500 \times 15} = 12° \text{ F}$$

Example 14-5. Thirty-six gallons of water per minute are circulated through a water-cooled condenser. If the temperature rise of the water in the condenser is 12° F, compute the load on the condenser in Btu/hr.

Solution. Rearranging and applying Equation 14-10, the load on Q_s the condenser,
$$= 500 \times \Delta T \times \text{gpm} = 500 \times 12 \times 36 = 216{,}000 \text{ Btu/hr}$$

Example 14-6. The load on an air-cooled condenser is 121,500 Btu/hr. If the desired temperature rise of the air in the condenser is 25° F, determine the air quantity in cfm which must be circulated over the condenser.

Solution. Applying Equation 14-11, the air quantity in cfm
$$= \frac{121{,}500}{1.08 \times 25} = 4500 \text{ cfm}$$

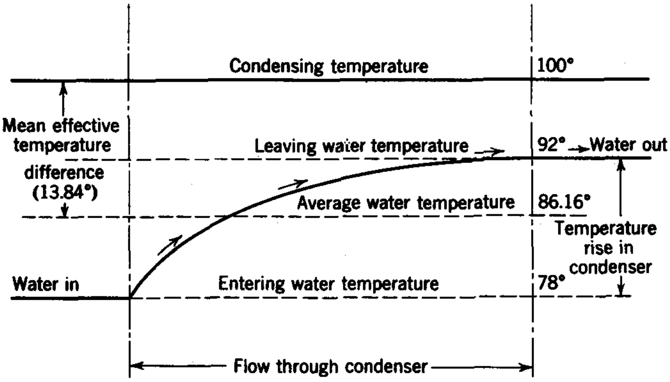


Fig. 14-1. Water temperature rise through condenser.

Example 14-7. Three thousand cfm of air are circulated over an air-cooled condenser. If the load on the condenser is 64,800 Btu/hr, compute the temperature rise of the air passing over the condenser.

Solution. Rearranging and applying Equation 14-11, $\Delta T = \dfrac{64{,}800}{1.08 \times 3000} = 20° \text{ F}$

For any given condenser and condenser loading, the condensing temperature of the refrigerant in the condenser will depend only upon the average temperature of the condensing medium flowing through the condenser. The lower the average temperature of the condensing medium the lower is the condensing temperature. For example, assume that the size and loading of a condenser are such that the required mean temperature differential between the refrigerant and the condensing medium is 15° F. If the average temperature of the condensing medium is 90° F, the condensing temperature will be 105° F (90 + 15), whereas if the average temperature of the condensing medium is 85° F, the condensing temperature will be 100° F (85 + 15).

The average temperature of the condensing medium flowing through the condenser depends upon both the initial temperature of the condensing medium entering the condenser and the temperature rise experienced in the condenser. Since the temperature rise of the condensing medium decreases as the flow rate increases, the greater the quantity of condensing medium circulated the lower is the average temperature of the condensing medium. Therefore, for any given condenser loading, the greater the flow rate of the condensing medium the lower will be the condensing temperature.

14-5. Condenser Application. As a general rule, for any given condenser load, the size of the condenser and the quantity of condensing medium circulated will depend upon the entering temperature of the condensing medium and upon the desired condensing temperature. A careful analysis of the data in Sections 14-3 and 14-4 will show that the condensing temperature of the refrigerant in the condenser is a function of three variables: (1) the entering temperature of the condensing medium, (2) the temperature rise in the condenser, and (3) the temperature difference between the refrigerant and the condensing medium. This relationship is illustrated in Fig. 14-1.

Recalling that the temperature rise in the condenser varies inversely with the flow rate of the condensing medium and that the temperature differential between the refrigerant and the condensing medium varies inversely with the size (surface area) of the condenser,* it is evident that:

1. For any given condensing surface and flow rate, the condensing temperature will increase or decrease as the entering temperature of the condensing medium increases or decreases.

2. For any given entering temperature, the larger the condensing surface and the higher the flow rate, the lower will be the condensing temperature.

3. For any given entering temperature, the

* Assuming the transfer coefficient to be constant.

amount of condensing surface required for a given condensing temperature decreases as the flow rate of the condensing medium increases.

With regard to the latter statement, this means in effect that the same condensing temperature can be maintained with either a small condensing surface and a high flow rate or a large condensing surface and a low flow rate. However, it should be recognized that the flow rate of the condensing medium is fixed within certain limits by the size and design of the condenser. If the flow rate through the condenser is too low, flow will be streamlined rather than turbulent and a low transfer coefficient will result. On the other hand, if the flow rate is too high, the pressure drop through the condenser becomes excessive, with the result that the power required to circulate the condensing medium also becomes excessive.

Since the design entering temperature of the condensing medium is usually fixed by conditions beyond the control of the system designer, it follows that the size and design of the condenser and the flow rate of the condensing medium are determined almost entirely by the design condensing temperature.

Although low condensing temperatures are desirable in that they result in high compressor efficiency and low horsepower requirements for the compressor, this does not necessarily mean that the use of a large condensing surface and a high flow rate in order to provide a low condensing temperature will always result in the most practical and economical installation. Other factors which must be taken into account and which tend to limit the size of the condenser and/or the quantity of condensing medium circulated are initial cost, available space, and the power requirements of the fan, blower, or pump circulating the condensing medium. Too, where water is used as the condensing medium and the water leaving the condenser is wasted to the sewer (see Section 14-9), the availability and cost of the water must also be considered.

The limitations imposed on condenser size by the factors of initial cost and available space are self-evident. As for the power requirements of the fan, blower, or pump circulating the condensing medium, it has already been stated that the power required to circulate the condensing medium increases as the flow rate increases. If the flow rate is increased beyond a certain point, the increase in the power required to circulate the condensing medium will more than offset the reduction in the power requirements of the compressor accruing from the increased flow rate. Therefore, the quantity of condensing medium which can be economically circulated is limited by the power requirements of the fan, blower, or pump.

Obviously, the optimum flow rate for the condensing medium is the one which will result in the lowest over-all operating costs for the system. This will vary somewhat with the conditions of the individual installation, being influenced by the type of application, the size and type of condenser used, fouling rates, and the design conditions for the region, along with such practical considerations as the cost and availability of water, utility costs, local codes and restrictions, etc. For example, since good system efficiency prescribes lower condensing temperatures for low temperature applications than for high temperature applications, it follows that for the same condenser load the optimum condensing medium flow rate will usually be higher for a low temperature application than for a high temperature application. Too, where the entering temperature of the condensing medium is relatively high, larger condensing surfaces and higher flow rates are required to provide reasonable condensing temperatures than where the entering temperature of the condensing medium is lower.

14-6. Air-Cooled Condensers. The circulation of air over an air-cooled condenser may be either by natural convection or by action of a fan or blower. Where air circulation is by natural convection, the air quantity circulated over the condenser is low and a relatively large condensing surface is required. Because of their limited capacity natural convection condensers are used only on small applications, principally domestic refrigerators and freezers.

Natural convection condensers employed on domestic refrigerators are usually either plate surface or finned tubing. When finned tubing is used, the fins are widely spaced so that little or no resistance is offered to the free circulation of air. Too, wide fin spacing reduces the possibility of the condenser being fouled with dirt and lint.

The plate-type condenser is mounted on the back of the refrigerator in such a way that an air flue is formed to increase air circulation. Finned tube condensers are mounted either on the back of the refrigerator or at an angle underneath the refrigerator. Regardless of condenser type or location, it is essential that the refrigerator be so located that air is permitted to circulate freely through the condenser at all times. Too, warm locations, such as one adjacent to an oven, should be avoided whenever possible.

A number of domestic freezer manufacturers utililize the outer shell of the freezer (outside wall surface) as a condensing surface. This is accomplished by bonding bare tubing to the inside surface of the outer shell so that the entire outer shell becomes a plate-type heat transfer surface. The use of these "wrap-around" condensers permits a considerable reduction in the size of the freezer (6 to 8 in. on both length and width), not only because it eliminates the space ordinarily required for the condenser but also because it allows the use of 3 to 4 in. of insulation in the walls where normally 6 to 8 in. is required in order to prevent moisture from condensing on the outside surface of the freezer. The slightly higher operating costs which accrue as a result of reducing the amount of wall insulation is more than offset by the savings in space that this practice makes possible.

Air-cooled condensers employing fans or blowers to provide "forced-air" circulation can be divided into two groups according to the location of the condenser: (1) chassis-mounted and (2) remote.

A chassis-mounted air-cooled condenser is one that is mounted on a common chassis with the compressor and compressor driver so that it is an integral part of the air-cooled "condensing unit" (Fig. 6-12). Although chassis-mounting of the air-cooled condenser makes possible a very compact, completely self-contained condensing unit which is ideally suited for use on small commercial fixtures, this arrangement has certain inherent disadvantages which make chassis-mounting impractical in larger applications.

The principal disadvantage of chassis-mounted air-cooled condensers is that the physical size of the condenser is limited to the dimensions of the chassis. Because of the limitation in physical size, chassis-mounted condensers of the type shown in Fig. 6-12 are rarely found in capacities larger than 2 tons.*

Another disadvantage of the chassis-mounted air-cooled condenser is their susceptibility to fouling. Since most condensing units are mounted on the floor, the condenser air tends to sweep across the floor so that dirt, lint, and other foreign materials are picked up from the floor and carried to the surface of the condenser, thereby "clogging" the condenser and restricting the air flow.

Too, on "open-type" air-cooled condensing units the condenser fan is usually mounted on the shaft of the compressor driver (Fig. 6-12). Naturally this limits both the size and the location of the fan so that the quantity of air circulated over the condenser is always less than that which would produce maximum efficiency at full load conditions. Notice also that, because of the fan location, the distribution of the air over the condenser surface is very uneven, being much greater on the end of the condenser directly in front of the fan.

Remote air-cooled condensers are used in sizes from 1 ton up to 100 tons or more and may be mounted either indoors or outdoors. When located indoors, provisions must be made for an adequate supply of outside air to the condenser (Fig. 14-2). If the condenser is installed in a warm location, such as in an attic or boiler room, ducts should be used to carry the air into the condenser and back to the outside. Because of the large quantity of air required, only the smaller sizes are mounted indoors.

When located outdoors, the air-cooled condenser may be mounted on the ground, on the roof, or on the side of a wall, with roof locations being the most popular. Typical wall and roof installations are shown in Figures 14-3 and 14-4, respectively. In all cases, the condenser should be so oriented that the prevailing winds for the area in the summertime will aid rather than retard the action of the fan. In the event that such orientation is not possible, wind deflectors should be installed on the discharge side of the condenser (Fig. 14-5).

* This is the approximate condenser capacity required on a 3 hp, commercial, air-cooled condensing. Approximately 25% of the motor horsepower is required to drive the fan. Naturally, this reduces the horsepower available to the compressor.

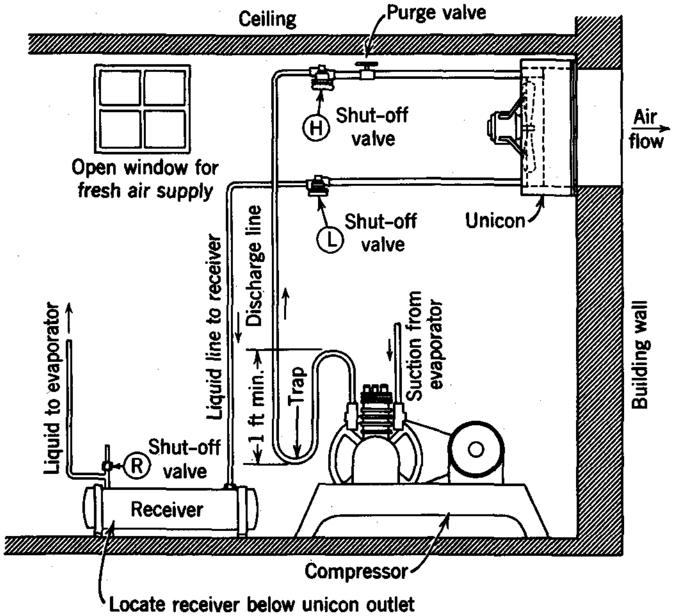


Fig. 10-2. Indoor installation of air-cooled condenser. (Courtesy Kramer Trenton Company.)

One significant outgrowth of the remote air-cooled condenser has been the development of a new type of air-cooled condensing unit which is designed specifically for remote installation. The air-cooled condensing unit illustrated in Fig. 14-6 is typical of these newer designs. This type is rapidly gaining in popularity and is now available in almost any desired capacity.

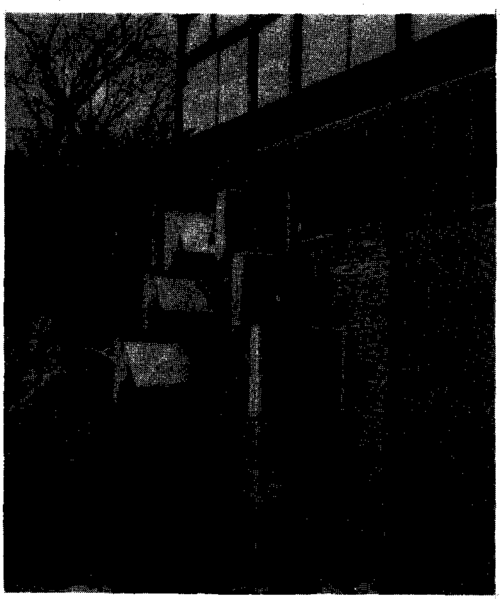


Fig. 14-3. Remote air-cooled condensers installed on outside wall. (Courtesy Kramer Trenton Company.)

14-7. Air Quantity and Velocity. For an air-cooled condenser there is a definite relationship between the size (face area) of the condenser and the quantity of air circulated in that the velocity of the air through the condenser is critical within certain limits. Good design prescribes the minimum air velocity that will produce turbulent flow and a high transfer coefficient. Increasing the air velocity beyond this point causes an excessive pressure drop through the condenser and results in an unnecessary increase in the power requirements of the fan or blower circulating the air.

The velocity of the air passing through an air-cooled condenser is a function of the free face area of the condenser and the quantity of air circulated. The relationship is given in the following equation:

$$\text{Air velocity (fpm)} = \frac{\text{Air quantity (cfm)}}{\text{Free face area (sq ft)}}$$

The free face area of the condenser is the area of the free air spaces between the tubes and fins. The actual free area per unit of face area varies with the design of the condenser, being dependent upon the size, number, and arrangement of the tubes and fins.

Fig. 14-4. Remote air-cooled condensers mounted on roof. (Courtesy Dunham-Bush, Inc.)

Normally, air velocities over air-cooled condensers are between 500 and 1000 fpm. However, because of the many variables involved, the optimum air velocity for a given condenser design is best determined by experiment. For this reason, most air-cooled condensers come from the factory already equipped with fans or blowers so that the air quantity and velocity over the condenser are fixed by the manufacturer. In all cases, to realize peak performance from an air-cooled condenser, the manufacturer's recommendations as to the air quantities should be carefully followed.

14-8. Rating and Selection of Air-Cooled Condensers. Capacity ratings for air-cooled condensers are usually given in Btu/hr for various operating conditions. It has already been shown that since the surface area and the value of U are fixed at the time of manufacture, the capacity of any one condenser depends only on the mean temperature difference between the air and the condensing refrigerant. Since

Fig. 14-5. Remote air-cooled condensers equipped with wind deflectors. (Courtesy Kramer Trenton Company.)

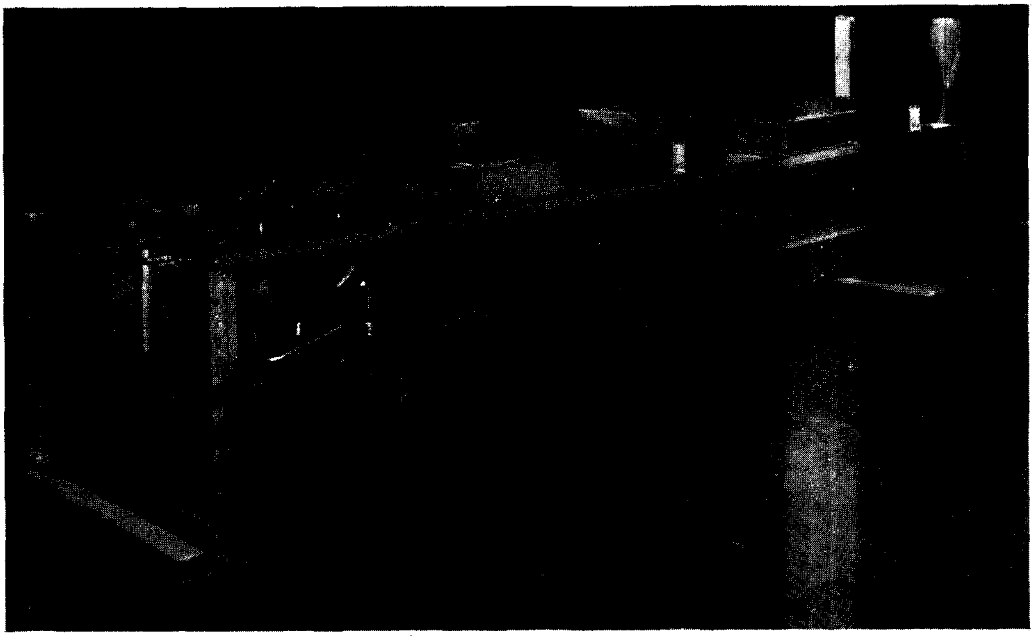


Fig. 14-6. Air-cooled condensing unit designed for remote installation. Notice generous size of condenser. (Courtesy Kramer Trenton Company.)

most air-cooled condensers come equipped with fans or blowers, the quantity of air circulated over the condenser is also fixed so that the average temperature of the air passing over the condenser depends only on the dry bulb temperature of the entering air and the load on the condenser. Obviously, then, in such cases, the capacity of the condenser is directly proportional to the temperature difference between the dry bulb temperature of the entering air and the condensing temperature. This temperature differential is often referred to as the "temperature split" in order to distinguish it from the mean effective temperature differential.*

Table R-13 is a typical manufacturer's rating table for air-cooled condensers. The basic ratings given in Table R-13A are based on 90° F ambient air temperature, 120° F condensing temperature, and 40° F evaporating temperature. For other design conditions multiply the basic rating from Table R-13A by the correction factors for variation in evaporating temperatures (Table R-13B) and for variation in entering air and condensing temperatures (Table R-13C).

* The temperature split is always proportional to the METD.

In order to select a condenser from the rating tables, the following design data must be known:

(1) The design suction and condensing temperatures
(2) The compressor capacity in Btu/hr
(3) The design outdoor dry bulb temperature (use values in Table 10-6A. Round off to next highest multiple of 5)

Example 14-8. From Table R-13, select an air-cooled condenser for a compressor having a capacity of 75,000 Btu/hr if the design evaporating and condensing temperatures are 20° F and 110° F, respectively, and the outdoor design dry bulb for the region is 90° F.

Solution. From Table R-13B, the correction factor for 20° F suction temperature $= 0.95$

From Table R-12C, the correction factor for condensing temperature of 110° F and entering air temperature of 90° F $= 0.665$

Required capacity of condenser at basic rating conditions

$$= \frac{75{,}000}{0.95 \times 0.665}$$
$$= 118{,}700 \text{ Btu/hr}$$

From Table R-13A, select condenser Model #BD1000 which has a capacity of 120,000 Btu/hr at the basic rating conditions.

Experience has shown that as a general rule selecting an air-cooled condenser on the basis of a condensing temperature of 110° F will result in the most economical condenser size. Hence, the actual size of the condenser selected will depend upon the outdoor design dry bulb temperature for the region in question. The higher the dry bulb temperature, the larger the condenser required. For example, for a condensing temperature of 110° F, if the dry bulb temperature is 85° F, the condenser can be selected for a 25° F temperature split, whereas if the dry bulb temperature is 90° F, the condenser must be selected for a 20° F temperature split, which will require a larger size.

14-9. Water-Cooled Condenser Systems. Systems employing water-cooled condensers can be divided into two general categories: (1) waste-water systems and (2) recirculated water systems. In waste-water systems the water supply for the condenser is usually taken from the city main and wasted to the sewer after passing through the condenser (Fig. 14-7). In recirculated water systems the water leaving the condenser is piped to a water cooling tower where its temperature is reduced to the entering temperature, after which the water is recirculated through the condenser (Fig. 14-8).

Naturally, where the condenser water is wasted to the sewer, the availability and cost of the water are important factors in determining the quantity of water circulated per unit of condenser load. As a general rule, an economical balance between water and power costs prescribes a water flow rate of approximately 1.5 gal per minute per ton of capacity.

The high cost of water, along with limited sewer facilities and recurring water shortages in many regions, has tended to limit waste-water systems to very small sizes. Too, many cities have placed severe restrictions on waste-water systems, particularly where the water supply is taken from the city main and wasted to the sewer.

When the condenser water is recirculated the power required to circulate the water through the water system must be taken into account in determining the water flow rate. Experience has shown that, in general, a water flow rate of between 2.5 and 3 gal per minute per ton usually provides the most economical balance between compressor horsepower and pump horsepower.

In some instances, the water supply for a waste-water system is taken from a well or from some nearby body of water, such as a river, lake, pond, etc., in which case both the cost of the water and the pumping horsepower must be considered in determining the optimum water flow rate.

To a large extent, the quantity of water circulated through the condenser determines the design of the water circuit in the condenser. Since heat transfer is a function of time, it follows that where low water quantities necessitate a high temperature rise in the condenser, the water must remain in contact with the condensing refrigerant for a longer period than when the water flow rate is high and the temperature rise required is smaller. Hence, where the water flow rate is low, the number of water circuits through the condenser are few and the circuits are long so that the water will remain in the condenser for enough time to permit the required amount of heat to be absorbed. On the other hand, when the flow rate is high and the temperature rise low, more circuits are used and the circuits are shorter in order to reduce the pressure drop to a minimum. This is illustrated in Figs. 14-9*a* and 14-9*b*. In Fig. 14-9*a*, the two water circuits through the condenser are connected in series for a low flow rate and a high temperature rise. The water enters through opening *A* and leaves through opening *C*. Opening *B* is capped. In Fig. 14-9*b*, the two water circuits are connected in parallel for a high flow rate and a low temperature rise. The water enters through opening *B* and leaves through openings *A* and *C*.

In designing the condenser water circuit particular attention must be given to the water

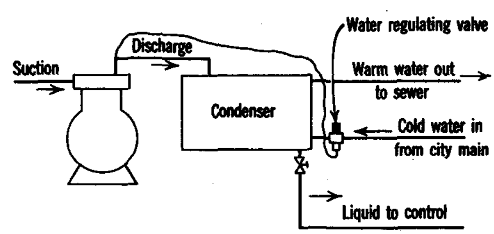


Fig. 14-7. Waste water system.

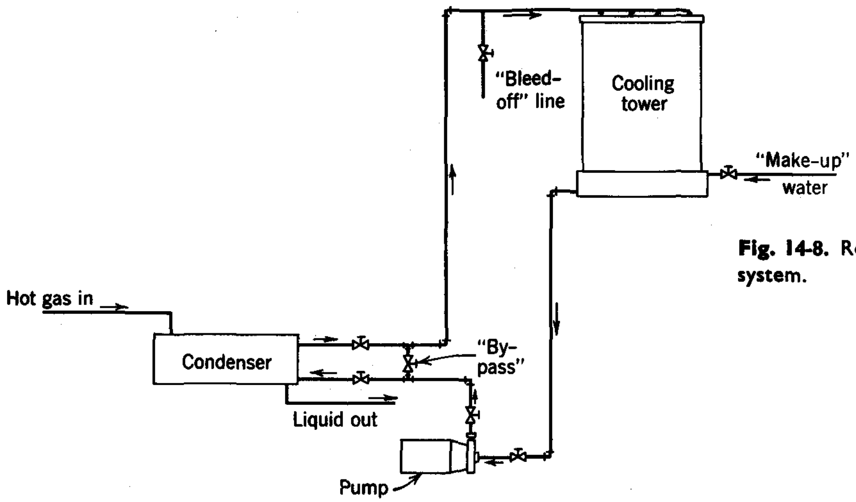


Fig. 14-8. Recirculating water system.

velocity and pressure drop through the condenser. In all cases the minimum permissible velocity is that which will produce turbulent flow and a high transfer coefficient. Since pressure drop is a function of velocity, the pressure drop through the condenser increases as the water velocity increases. For this reason, the maximum permissible velocity in any one case is usually determined by the allowable pressure drop.* For waste-water systems, where the water is forced through the condenser by city main pressure, the pressure drop through the condenser is not critical as long as it is within the limits of turbulent flow and the available city main pressure. In such cases, high velocities are recommended in order to take advantage of the higher transfer coefficient. On the other hand, when the water is circulated by action of a pump, a high pressure drop through the condenser will increase the pumping head and the power required to circulate the water. Therefore, for recirculating water systems, the optimum water velocity is one which will provide the most economical balance between a high transfer coefficient and a low pumping head.

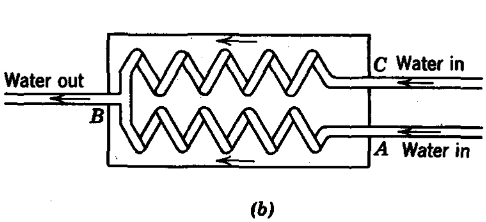


Fig. 14-9. (*a*) Water circuit connected for series flow. (*b*) Water circuit connected for parallel flow.

* Excessive velocity will usually cause erosion of the water tubes, particularly at points where the water changes direction. The maximum velocity recommended by Air Conditioning and Refrigeration Institute (ARI) is 8 fps.

In Figs. 14-9*a* and 14-9*b*, it is of interest to notice that for the same flow rate the velocity and pressure drop through the circuit arrangement in Fig. 14-9*a* are approximately four times as great as that through the circuit arrangement in Fig. 14-9*b*. Too, because of the higher velocity, the transfer coefficient is somewhat higher for the condensing surface in Fig. 14-9*a* and less condensing surface is required for the same heat transfer capacity.

14-10. Fouling Rates. Another factor which must be considered in selecting a water-cooled condenser is fouling of the tube surface on the water side. The fouling is caused primarily by mineral solids which precipitate out of the water and adhere to the tube surface. The scale thus formed on the tube not only reduces the water side transfer coefficient, but it also tends to restrict the water tube and reduce the quantity of water circulated, both of which will cause serious increases in the condensing pressure.

In general, the rate of tube fouling is influenced by: (1) the quality of the water used with regard to the amount of impurities contained therein, (2) the condensing temperature, and (3) the frequency of tube cleaning with relation to the total operating time.

Most manufacturers of water-cooled condensers give condenser ratings for clean tubes and for four stages of tube fouling in accordance with the scale factors given in Table 14-1 for various types of water. These scale factors are an index of the reduction in the tube transfer coefficient resulting from the scale deposit. In selecting a water-cooled condenser, a minimum scale factor of 0.0005 should always be used. Under no circumstances should a condenser be selected on the basis of clean tubes. However, when the condensing temperature is low (leaving water temperature less than 100° F) and the condenser tubes are to be cleaned frequently, the fouling factor from Table 14-1 may be reduced to the next lowest value. The use of scale factors will be illustrated later in the chapter.

14-11. Water-Cooled Condensers. Water-cooled condensers are of three basic types: (1) double-tube, (2) shell-and-coil, and (3) shell-and-tube.

As its name implies, the double-tube condenser consists of two tubes so arranged that one is inside of the other (Fig. 14-10). Water is piped through the inner tube while the refrigerant flows in the opposite direction in the space between the inner and outer tubes. With this arrangement, some air-cooling of the refrigerant is provided in addition to the water-cooling. Counterflowing of the fluids in any type of heat exchanger is always desirable since it results in the greatest mean temperature difference between the fluids and, therefore, the highest rate of heat transfer.

Several types of double-tube condensers are shown in Figs. 14-11 and 14-12. The type shown in Fig. 14-11 can be cleaned mechanically by removing the end-plates (inset). The type shown in Fig. 14-12 is cleaned by circulating approved chemicals through the water tubes (see Section 14-23).

Equipped with water-regulating valves (Section 14-20), double-tube condensers make excellent "booster" condensers for use with chassis-mounted air-cooled condensers during periods of peak loading. Since the water valve can be adjusted to open and allow water to flow through the condenser only when the condensing pressure rises to some predetermined level, the amount of water used is relatively small in comparison to the savings in power afforded by the increased compressor efficiency.

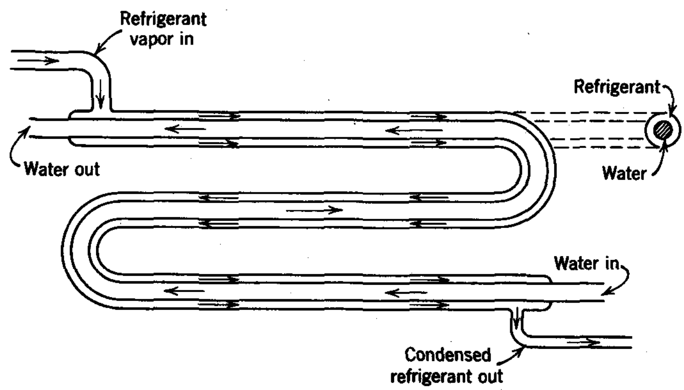


Fig. 14-10. Double-tube water-cooled condenser.

The shell-and-coil condenser is made up of one or more bare-tube or finned-tube coils enclosed in a welded steel shell (Fig. 14-9). The condensing water circulates through the coils while the refrigerant is contained in the shell surrounding the coils. Hot refrigerant vapor enters at the top of the shell and condenses as it comes in contact with the water coils. The condensed liquid drains off the coils into the bottom of the shell, which often serves also as the receiver tank. Care should be taken not to

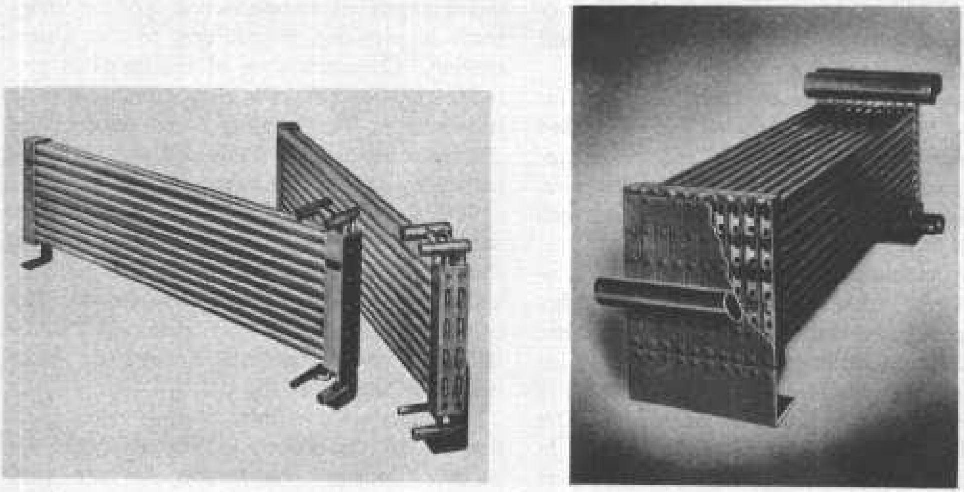


Fig. 14-11. Double-pipe condensers with mechanically cleanable tubes. (Courtesy Halstead and Mitchell.)

overcharge the system with refrigerant since an excessive accumulation of liquid in the condenser will tend to cover too much of the condensing surface and cause an increase in the discharge temperature and pressure.

Most shell-and-coil condensers are equipped with a split water circuit. The two parts of the circuit are connected in series for waste-water systems (Fig. 14-9*b*) and in parallel for recirculating systems (Fig. 14-9*a*). As a general rule, shell-and-coil condensers are used only for small installations up to approximately 10 tons capacity.

Shell-and-coil condensers are cleaned by circulating an approved chemical through the water coils.

The shell-and-tube condenser consists of a cylindrical steel shell in which a number of straight tubes are arranged in parallel and held in place at the ends by tube sheets. Construction is almost identical to that of the flooded-type shell-and-tube liquid chiller. The condensing water is circulated through the tubes, which may be either steel or copper, bare or extended surface. The refrigerant is contained in the steel shell between the tube sheets. Water circulates in the annular spaces between the tube sheets and the end-plates, the end-plates being baffled to act as manifolds to guide the water flow through the tubes. The arrangement of the end-plate baffling determines the number of passes the water makes through the condenser from one end to the other before leaving the condenser. The number of passes may be as few as two or as many as twenty.

For any given total number of tubes, the number of tubes per pass varies inversely with the number of passes. For example, assuming that a condenser has a total of forty tubes, if there are two passes, the number of tubes per pass is twenty, whereas if there are four passes, the number of tubes per pass is ten.

It is important to notice that for the same total number of tubes and the same water quantity, the velocity of the water and the pressure drop through the condenser will be four times as great for a four-pass condenser as for a two-pass condenser. Because of the higher velocity the transfer coefficient will be higher for the four-pass condenser and a smaller condensing surface will be required for a given heat transfer capacity. However, on the other hand, because of the high pressure drop, the power required to circulate the water will be greater. Hence, for a waste-water system, the four-pass condenser is probably the best selection, whereas for a recirculating system, the two-pass condenser is probably the better of the two.*

Shell-and-tube condensers are available in capacities ranging from 2 tons up to several

* This example is intended only to illustrate the principles of design and should not be construed to mean that four-pass condensers are undesirable for recirculating systems.

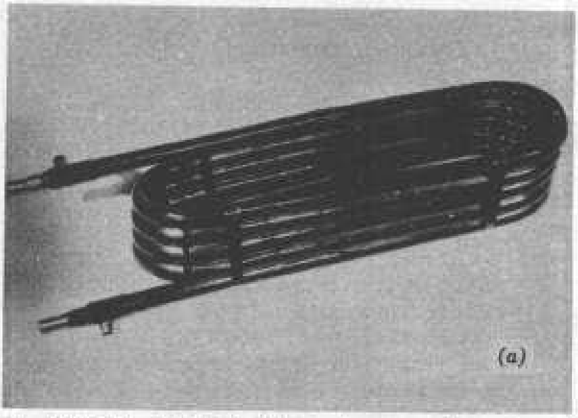
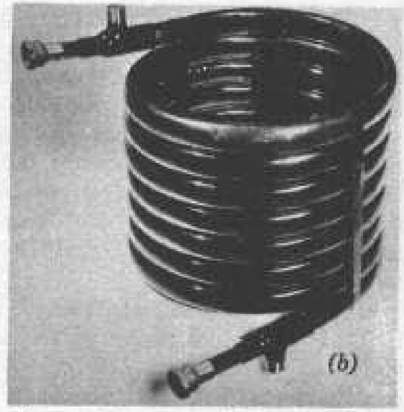



Fig. 14-12. Typical double-pipe condenser configurations. (*a*) Trombone configuration. (*b*) Helix configuration. (Courtesy Edwards Engineering Corporation.)

hundred tons or more. Shell diameters range from approximately 4 in. up to 60 in., whereas tube length varies from approximately 3 ft to 20 ft. The number and the diameter of the tubes depend on the diameter of the shell. Tube diameters of $\frac{5}{8}$ in. through 2 in. are common, whereas the number of tubes in the condenser varies from as few as six or eight to as many as a thousand or more. The end-plates of the condenser are removable to permit mechanical cleaning of the water tubes.

Single-pass, vertical shell-and-tube condensers are sometimes employed on large ammonia installations. The construction of the vertical shell-and-tube condenser is similar to that of the vertical shell-and-tube chiller illustrated in Fig. 11-44. The vertical condenser is equipped with a water box at the top to distribute the water to the tubes and a drain at the bottom to carry the water away. Each tube is equipped at the top with a distributor fitting which imparts a rotating motion to the water to assure adequate wetting of the tube. The hot refrigerant vapor usually enters at the side of the shell near the middle of the condenser and the liquid leaves the condenser at the side of the shell near the bottom. The height of vertical shell-and-tube condensers ranges from 12 ft to 18 ft. The tubes are mechanically cleanable.

14-12. Rating and Selection of Water-Cooled Condensers.* The ratings shown in Table R-14 are based on condensing temperatures of 102° and 105° F, 20° and 10° water rise and 0.0005 scale factor which is the minimum recommended in ARI standards.

Where other conditions exist, the following procedure should be followed in selecting the proper condenser.

Condensers must not be selected for less than 0.5 gpm per tube below which streamline instead of turbulent water flow occurs. ARI standards indicate that the water velocity should not exceed 8 fps which is 5.75 gpm per tube for Acme STF and SRF condensers.

It is necessary to have the following information to select a proper condenser:

1. Total tons (low side).
2. Evaporator temperature.
3. Condensing temperature.
4. Water temperature "in."
5. Water temperature "out," or gpm available.
6. Type of water or required scale factor.

Then proceed as follows:

1. Determine the corrected tons to be used in selecting the proper condenser by reference to Fig. 2, Table R-14. The factor obtained for the desired evaporator temperature and condensing temperature is multiplied by the actual tons to obtain corrected tons.

2. Determine the water temperature rise and gpm per ton. Knowing either factor, the other may be obtained by reference to Fig. 3, Table R-14. Use corrected tons to determine the total gpm required.

* The material in this section is reprinted directly from the manufacturer's catalog, the only alteration being the table designations. Courtesy of Acme Industries, Inc.

3. Determine the temperature differences between the condensing temperature and the "water in" and "water out" temperatures and find the METD by referring to Table 11-1.

4. Make preliminary selection of condenser shell diameter by reference to Table R-14, basing the selection on the corrected tons found in step 1. Find the number of tubes per pass and then by referring to step 2, find the gpm per tube.

5. Select the desired scale factor by reference to Table 14-1 which suggests scale factors for various types of water. The most commonly used factor is 0.0005 and it should be borne in mind when selecting a factor that a determination is being made of the frequency of cleaning which will be required.

6. Referring to Fig. 1, Table R-14, determine the rate of heat transfer "U" for the gpm per tube in step 4 and the scale factor in step 5.

7. Calculate the surface required by use of the following formula.

$$\text{Square feet of surface} = \frac{\text{Corrected tons} \times 14{,}400}{U \times \text{METD}}$$

8. Select a condenser having at least the required surface from Table R-14. Be sure to use the shell diameter determined in the preliminary selection of step 4.

9. Make final checks on selection.

a. Using the gpm per tube from step 4 and the nominal tube length shown in Table R-14 for the model selected in step 8, refer to Fig. 4 of Table R-14 to obtain water pressure drop through condenser.

b. Obtain nominal operating charge from the last column of Table R-14. This is the maximum weight of liquid refrigerant which can be allowed in the shell during the operating period covering some of the lower tubes. Larger shell diameters or separate receivers may be used where greater storage capacity is needed during operation.

c. Determine the pump down capacity from Table R-14. If less than the total weight of refrigerant to be used in the system and provision for complete pump-down are required, an additional receiver should be used.

Example 14-9. Select an R-12 condenser to meet the following conditions:

Refrigeration load	30 tons
Condensing temperature	100° F
Suction temperature	30° F
Water available 2 gpm/ton river water reasonably clean at	78°F
Maximum tube length	12 ft
Maximum water pressure drop	7.5 psi

Solution

1. From Fig. 2, the correction factor for 30° F suction temperature and 100° F condensing temperature is 1.013.

Corrected tons $30 \times 1.013 = 30.4$ tons

2. From Fig. 3, for 2 gpm/ton the water temperature rise is found to be 14.4°.

Total gpm $30.4 \times 2 = 60.8$

Water "out" temperature 78 plus 14.4 = 92.4° F

3. GTD $100 - 78 = 22°$

LTD $100 - 92.4 = 7.6°$

From Table 11-1, METD = 13.55° F

4. Refer to Table R-14. Use of four passes will usually give an economical selection for 75° F water in and 95° F water out which approximates the required water conditions. Note that a $10\frac{3}{4}$ shell will probably be needed. This shell has sixty tubes.

$$\text{gpm per tube} \quad \frac{\text{Total gpm} \times \text{number of passes}}{\text{Number of tubes in condenser}} = \frac{60.8 \times 4}{60} = 4.05 \text{ gpm per tube}$$

5. Referring to Table 14-1, for clean river water and over 3 fpm velocity, the suggested scale factor is 0.001.

6. From Fig. 1, the U factor for 4.05 gpm per tube and 0.001 scale factor is 121.5 Btu per hour per square foot of extended surface per °F METD.

7. Square feet required

$$\frac{\text{Corrected tons} \times 14{,}400}{U \text{ factor} \times \text{METD}} = \frac{30.4 \times 14{,}400}{121.5 \times 13.55} = 266 \text{ sq ft}$$

8. Referring to Table R-14, a Model STF-1010 has 289 sq ft external tube surface and should be selected. When installed the water connection should be made for four-pass operation.

9. (*a*). For water pressure drop, refer to Fig. 4 and note that the pressure drop for 4.05 gpm per tube in an STF-1010 condenser connected for four passes is 7.1 psi. (*b*). Table R-14 shows a nominal operating charge of 38 lb of R-12, which will normally be sufficient for a 30-ton

installation. However, if more operating storage is needed, a separate receiver may be chosen, or alternately a different condenser selection may be made if more economical. (*c*). Table R-14 also shows pump-down capacity which is 252 lb of R-12. Usually this will be sufficient, but if greater pump-down capacity is required, a separate receiver tank must be used.

14-13. Simplified Ratings. Simplified ratings, based on the horsepower of the compressor driver, are available for most air-cooled and water-cooled condensers, particularly in smaller sizes. Since the power required by the compressor varies with both the evaporator load and the compression ratio, it provides a reasonable index of the condenser load at all operating conditions. Table R-15, which applies to double-tube condensers of the type shown in Fig. 14-12, is a typical simplified condenser rating table.

14-14. Cooling Towers. Cooling towers are essentially water conservation or recovery devices. Warm water from the condenser is pumped over the top of the cooling tower from where it falls or is sprayed down to the tower basin. The temperature of the water is reduced as it gives up heat to the air circulating through the tower.

Although there is some sensible heat transfer from the water to the air, the cooling effect in a cooling tower results almost entirely from the evaporation of a portion of the water as the water falls through the tower. The heat to vaporize the portion of water that evaporates is drawn from the remaining mass of the water so that the temperature of the mass is reduced. The vapor resulting from the evaporating process is carried away by the air circulating through the tower. Since both the temperature and the moisture content of the air are increased as the air passes through the tower, it is evident that the effectiveness of the cooling tower depends to a large degree on the wet bulb temperature of the entering air. The lower the wet bulb temperature of the entering air, the more effective is the cooling tower.

The efficiency of a cooling tower is influenced by all the factors governing the rate at which the water will evaporate into the air (see Section 4-8). Some of the factors which determine cooling tower efficiency are: (1) the mean difference in vapor pressure between the air and the water in the tower, (2) the amount of exposed water surface and the length (time) of exposure, (3) the velocity of the air passing through the tower, and (4) the direction of the air flow with relation to the exposed water surface (parallel, transverse, or counter).

For any given water temperature entering the tower, the vapor pressure difference is essentially a function of the wet bulb temperature of the entering air. In general, the lower the entering wet bulb temperature, the greater the vapor pressure differential and the greater the tower capacity.

The exposed water surface includes: (1) the surface of the water in the tower basin, (2) all wetted surfaces in the tower, and (3) the combined surface of the water droplets falling through the tower.

Theoretically, the lowest temperature to which the water can be cooled in a cooling tower is the wet bulb temperature of the entering air, in which case the water vapor in the leaving air will be saturated. In actual practice, it is not possible to cool the water to the wet bulb temperature of the air. In most cases, the temperature of the water leaving the tower will be 7° to 10° F above the wet bulb temperature of the entering air. Too, the air leaving the tower will always be somewhat less than saturated.

The temperature difference between the temperature of the water leaving the tower and the wet bulb temperature of the entering air is called the tower "approach." As a general rule, all other conditions being equal, the greater the quantity of water circulated over the tower the closer the leaving water temperature approaches the wet bulb temperature of the air. However, the quantity of water which can be economically circulated over the tower is somewhat limited by the power requirements of the pump.

The temperature reduction experienced by the water in passing through the tower (the difference between the entering and leaving water temperatures) is called the "range" of the tower. Naturally, to maintain equilibrium in the condenser water system, the tower "range" must always be equal to the temperature rise of the water in the condenser.*

The load on a cooling tower can be approximated by measuring the water flow rate over the

* Except where a condenser by-pass is used. See Section 14-17.

tower and the entering and leaving water temperatures. The following equation is applied:

$$\text{Tower load(Btu/min)} = \text{flow rate(gpm)} \times 8.33 \times (\text{entering water temperature} - \text{leaving water temperature}) \quad (14\text{-}12)$$

Example 14-10. Determine the approximate load on a cooling tower if the entering and leaving water temperatures are 96° F and 88° F, respectively, and the flow rate of the water over the tower is 30 gpm.

Solution. Applying 14-12, the tower load (Btu/min) $= 30 \times 8.33 \times (96 - 88)$ $= 2000$ Btu/min

Since the load on the tower is equal to the load on the condenser, the approximate refrigerating capacity of the system can be computed by dividing the tower load by the condenser load in Btu/min/ton corresponding to the operating conditions of the system.

Example 14-11. Compute the refrigerating capacity of an R-13 system operating on the cooling tower of Example 14-10, if the evaporating and condensing temperatures are 20° F and 110° F, respectively.

Solution. From Fig. 14-1, the load on the condenser

= 247 Btu/min/ton

The approximately refrigerating capacity of the system

$$= \frac{\text{Tower load (Btu/min)}}{\text{Condenser load (Btu/min/ton)}} = \frac{2000}{247} = 8.1 \text{ tons}$$

Since the heat absorbed per pound of water evaporated is approximately 1000 Btu, assuming a condenser load of 250 Btu/min/ton, the quantity of water evaporated per ton of refrigeration (evaporator) is approximately 0.25 lb per minute or 2 gal per hour.

In addition to the water lost by evaporation, water is lost from the cooling tower by "drift" and by "bleed-off." A small amount of water in the form of small droplets is entrained and carried away by the air passing through the tower. Water lost in this manner is called the drift loss. The amount of drift loss from a tower depends on the design of the tower and the wind velocity.

"Bleed-off" is the continuous or intermittent wasting of a certain percentage of the circulated water in order to avoid a build-up in the concentration of dissolved mineral solids and other impurities in the condenser water. Without bleed-off the concentration of dissolved mineral solids in the condenser will build up quite rapidly as a result of the evaporation taking place in the cooling tower. Since the scaling rate is proportional to the quality of the water, as the concentration of mineral solids in the water increases the scaling rate also increases.

The amount of bleed-off required to maintain the concentration of dissolved mineral solids at a reasonable level depends upon the cooling range, the water flow rate, and the initial water conditions. Suggested bleed-off rates for various cooling ranges are given in Table 14-2. To determine the quantity of water loss by bleed-off, multiply the water flow rate over the tower by the factor obtained from Table 14-2.

Example 14-12. Determine the quantity of water lost by bleed-off if the water flow rate over the tower is 30 gpm and the range is 10° F.

Solution. From Table 14-1, the percent bleed-off required $= 0.33\%$

The quantity of water lost by bleed-off $= 30 \text{ gpm} \times 0.0033$ $= 0.099$ gpm

The bleed-off line should be located in the hot water return line near the top of the tower so that water is wasted only when the pump is running (Fig. 14-8).

Make-up water, to replace that lost by evaporation, drift, and bleed-off, is piped to the tower basin through a float valve which tends to maintain a constant water level in the basin.

14-15. Cooling Tower Design. According to the method of air circulation, cooling towers are classified as either natural draft or mechanical draft. When air circulation through the tower is by natural convection, the tower is called a natural draft or atmospheric tower. When air circulation through the tower is by action of a fan or blower, the tower is called a mechanical draft tower. Mechanical draft towers may be further classified as either induced draft or forced draft, depending on whether the fan or

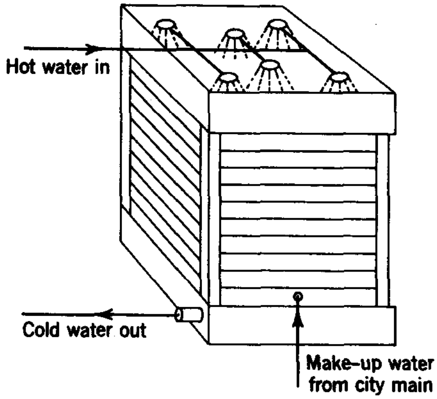


Fig. 14-13. Natural draft-cooling tower.

blower draws the air through the tower or forces (blows) it through. A schematic diagram of a spray-type natural draft tower is shown in Fig. 14-13. Schematic diagrams of induced draft and forced draft towers are shown in Figs. 14-14 and 14-15, respectively.

In the spray-type atmospheric tower, the warm water from the condenser is pumped to the top of the tower where it is sprayed down through the tower through a series of spray nozzles. Since the amount of exposed water surface depends primarily on the spray pattern, a good spray pattern is essential to high efficiency. The type of spray pattern obtained depends on the design of the nozzles. For most nozzle designs, a water pressure drop of 7 to 10 lb per square inch will produce a suitable spray pattern.

Some natural draft towers contain decking or filling (usually of redwood) to increase the amount of wetted surface in the tower and to break up the water into droplets and slow its fall to the bottom of the tower. Atmospheric towers containing decking are called "splash-deck." Often, in splash-deck towers, no spray nozzles are used and the water is broken up into droplets by the "splash-impact" method.

The quantity and velocity of the air passing through a natural draft cooling tower depend on the wind velocity. Hence, the capacity of a natural draft tower varies with the wind velocity, as does the amount of "drift" experienced. Too, natural draft towers must always be located out-of-doors in places where the wind can blow freely through the tower. In commercial applications, roof installations are common.

Since air circulation through mechanical draft towers is by action of a fan or blower, small mechanical draft towers can be installed indoors as well as out-of-doors, provided that an adequate amount of outside air is ducted into and out of the indoor location. Too, since larger air quantities and higher velocities can be used, the capacity of a mechanical draft tower per unit of physical size is considerably greater than that of the natural draft tower. In addition, most mechanical draft towers contain some sort of decking or fill to improve further the efficiency. Spray eliminators must be used in mechanical draft towers to prevent excessive drift losses.

14-16. Cooling Tower Rating and Selection. Table R-16 contains rating data for the spray-type, natural draft cooling tower illustrated in Fig. 14-13 and is a typical cooling tower rating table. Notice that the tower ratings are given in tons, based on a heat transfer capacity of 250

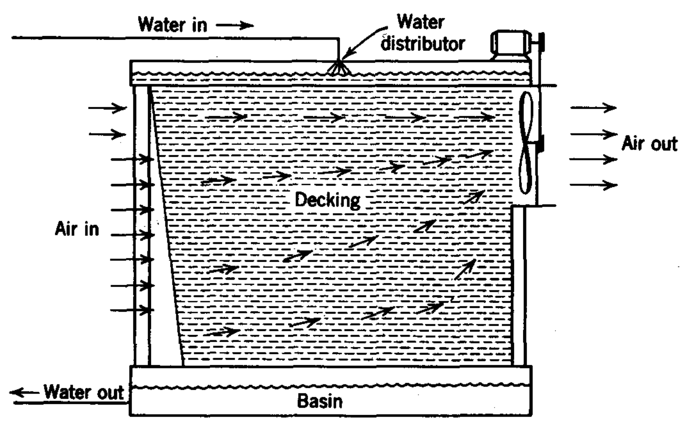


Fig. 14-14. Small induced draft-cooling tower.

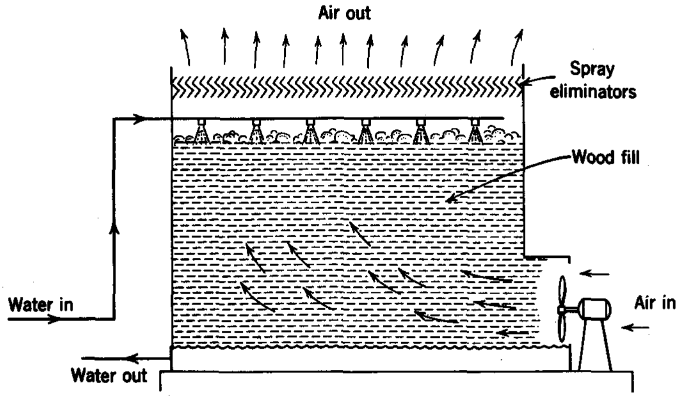


Fig. 14-15. Forced draft-cooling tower.

Btu/min/ton. Nominal tower ratings are based on a 3 mi per hour wind velocity, and 80° F design wet bulb temperature, and a water flow rate over the tower of 4 gpm per ton. Tower performance at conditions other than those listed in the table can be determined by using the rating correction chart that accompanies the table.

To select the proper tower from the rating table, the following data must be known:

1. Desired tower capacity in tons (compressor capacity)
2. Design wet bulb temperature
3. Desired leaving water temperature (condenser entering water temperature or tower approach)

or

1. Desired flow rate over the tower (gpm)
2. Design wet bulb temperature
3. Desired entering and leaving water temperatures (tower cooling range and tower approach)

Example 14-13. From Table R-16, select a cooling tower to meet the following conditions:

1. Required tower capacity	= 20 tons
2. Design wet bulb temperature	= 78° F
3. Desired leaving water temperature	= 86° F

Solution. From Table R-16, select tower, Model #CSA-66, which has a capacity of 20.7 tons at the desired conditions when the flow rate over the tower is 3 gpm per ton. Hence, for 20-tons capacity, a total of 60 gpm (20 × 3) must be circulated over the tower. As shown in the table, the entering water temperature will be approximately 96° F.

Example 14-14. It is desired to cool 90 gpm from 96° F to 86° F when the design wet bulb is 78° F. Select the proper tower from Table R-16.

Solution

Tower range	= 96 − 86 = 10°
Tower approach	= 86 − 78 = 8°
From rating correction chart, range-approach factor	= 1.1
From wet bulb correction chart, wet bulb factor	= 1.04
Nominal gpm	= 90 × 1.1 × 1.04
	= 103 gpm

From Table R-16, for 103 gpm nominal, select tower, Model #SA-68.

Example 14-15. It is required to cool water for 30 tons at 5 gpm/ton to a 5° F approach of an 80° F wet bulb. Select the proper tower from Table R-16.

Solution

Total gpm required for 30 tons at 5 gpm/ton	= 30 × 5 = 150 gpm
From rating cortion chart, rating correction factor for 5 gpm/ton and 5° approach	= 1.15
From wet bulb correction chart, wet bulb correction factor	= 1.0
Nominal gpm	= 150 × 1.15 × 1.0
	= 172.6 gpm

From Table R-16, for 172.6 gpm nominal, select tower Model #SA-612.

14-17. Condenser By-Pass. For any given tower range and approach, the entering and leaving water temperatures will depend only on the wet bulb temperature of the air. Hence, in regions (particularly coastal areas) where the outdoor wet bulb temperature is relatively high, a closer approach to the wet bulb temperature is required in order to maintain a reasonable condensing temperature with an economical condenser size than in areas where the wet-bulb temperature is lower. It has already been shown that, in general, the greater the quantity of water circulated over the tower per unit of capacity the closer the leaving water temperature will approach the wet bulb temperature. Therefore, in regions having a high wet bulb temperature, it is usually desirable to circulate a greater quantity of water over the tower than can be economically circulated through the condenser because of the excessive pumping head encountered. This can be accomplished by installing a condenser by-pass line as shown in Fig. 14-8. Through the use of a condenser by-pass, a certain, predetermined portion of the water circulated over the tower is permitted to by-pass the condenser, thereby reducing the over-all pumping head.

The advantage of the condenser by-pass is that it makes possible the maintenance of reasonable condensing temperatures with moderate condenser and tower sizes without greatly increasing the pumping head. The quantity of water flowing through the by-pass is regulated by the hand valve in the by-pass line. Once the hand valve has been adjusted for the proper flow rate through the by-pass, the handle should be removed from the valve so that the valve adjustment cannot be changed indiscriminately. An excessive amount of water flowing through the by-pass will not only tend to starve the condenser and raise the condensing pressure, but it may also cause the pump motor to become overloaded, thereby rendering the entire system inoperative. The desired flow rate through the by-pass is determined by subtracting the flow rate through the condenser from the flow rate over the tower. This will be illustrated presently.

Since the cooling tower capacity must of necessity be equal to the condenser capacity at the design conditions, it follows that:

$$\text{Tower gpm} \times \text{tower range} \times 500 = \text{condenser gpm} \times \text{condenser rise} \times 500$$

Eliminating the constant,

$$\text{Tower gpm} \times \text{tower range} = \text{condenser gpm} \times \text{condenser rise} \qquad (14\text{-}13)$$

Example 14-16. A compressor on a refrigerating system has a capacity of 25 tons. The design wet bulb temperature is 80° F. The desired condenser water entering temperature is 87° F and the desired temperature rise through the condenser is 10° F. Select a cooling tower from Table R-16 and determine:

1. The total gpm circulated over the tower
2. The temperature of the water entering the tower
3. The tower cooling range
4. The temperature of the water leaving the condenser
5. The gpm circulated through the condenser
6. The gpm circulated through the by-pass

Solution. From Table R-16, tower, Model #SA-58 has a capacity of 25 tons at an 80° F wet bulb temperature and a 7° approach. This capacity is based on a water flow rate of 4 gpm/ton and on a cooling range of 7.5° (94.5 − 87).

Total gpm over the tower for 25 tons

$$= 25 \text{ tons} \times 4 \text{ gpm/ton}$$
$$= 100 \text{ gpm}$$

From Table R-16, the tower entering water temperature

$$= 94.5° \text{ F}$$

Tower range

$$= 94.5 - 87 = 7.5°$$

Water temperature leaving condenser

$$= 87 + 10 = 97° \text{ F}$$

Rearranging and applying Equation 14-13, condenser gpm

$$= \frac{\text{Tower gpm} \times \text{tower range}}{\text{Condenser rise}}$$
$$= \frac{100 \times 7.5}{10}$$
$$= 75 \text{ gpm}$$

Gpm circulated through by-pass

$$= \text{Tower gpm} - \text{condenser gpm}$$
$$= 100 - 75$$
$$= 25 \text{ gpm}$$

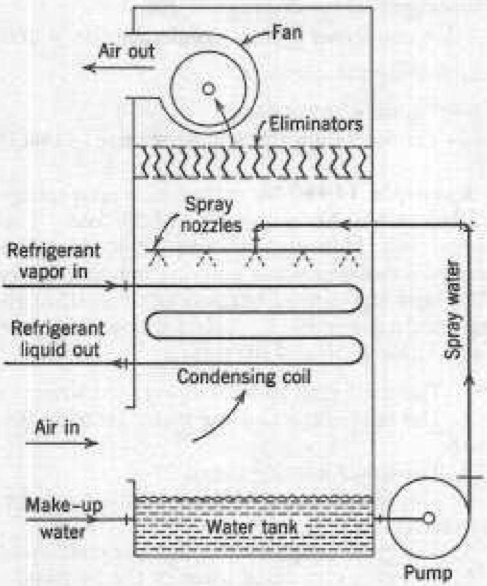


Fig. 14-16. Schematic diagram of evaporative condenser.

14-18. Evaporative Condensers. An evaporative condenser is essentially a water conservation device and is, in effect, a condenser and a cooling tower combined into a single unit. A diagram of a typical evaporative condenser is shown in Fig. 14-16.

As previously stated, both air and water are employed in the evaporative condenser. The water, pumped from the sump up to the spray header, sprays down over the refrigerant coils and returns to the sump. The air is drawn in from the outside at the bottom of the condenser by action of the blower and is discharged back to the outside at the top of the condenser. In some cases, both pump and blower are driven by the same motor. In others, separate motors are used. The eliminators installed in the air stream above the spray header are to prevent entrained water from being carried over into the blower. An alternate arrangement, with the blower located on the entering air side of the condenser, is shown in Fig. 14-17.

Although the actual thermodynamic processes taking place in the evaporative condenser are somewhat complex, the fundamental process is that of evaporative cooling. Water is evaporated from the spray and from the wetted surface of the condenser into the air, the source of the vaporizing heat being the condensing refrigerant in the condenser coil.

The cooling produced is approximately 1000 Btu per pound of water evaporated. All the heat given up by the refrigerant in the condenser eventually leaves the condenser as either sensible heat or latent heat (moisture) in the discharge air. Since both the temperature and the moisture content of the air are increased as the air passes through the condenser, the effectiveness of the condenser depends, in part, on the wet bulb temperature of the entering air. The lower the wet bulb temperature of the entering air the more effective is the evaporative condenser.

To facilitate cleaning and scale removal, the condensing coil is usually made up of bare rather than finned tubing. The amount of coil surface used per ton of capacity varies with the manufacturer and depends to a large extent on the amount of air and water circulated.

Generally, the capacity of the evaporative condenser increases as the quantity of air circulated through the condenser increases. As a practical matter, the maximum quantity of air

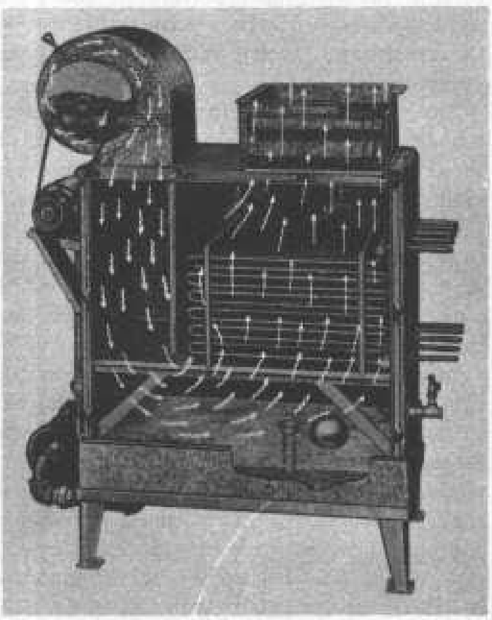


Fig. 14-17. Cutaway view of "Dri-Fan" evaporative condenser. Funnel-shaped overflow drain provides automatic bleed-off. (Courtesy Refrigeration Engineering, Inc. A proprietary design of Refrigeration Engineering, Inc.)

which can be circulated through the condenser is limited by the horsepower requirements of the fan and by the maximum air velocity that can be permitted through the eliminators without the carry over of water particles.

The quantity of water circulated over the condenser should be sufficient to keep the tube surface thoroughly wetted in order to obtain maximum efficiency from the tube surface and to minimize the rate of scale formation. However, a water flow rate in excess of the amount required for adequate wetting of the tubes will only increase the power requirements of the pump without materially increasing the condenser capacity.

Assuming a condenser load of 15,000 Btu per hour per ton, the water lost by evaporation is approximately 15 lb (2 gal) per hour per ton (15,000/1000). In addition to the water lost by evaporation, a certain amount of water is lost by drift and by bleed-off. The amount of water lost by drift and by bleed-off is approximately 1.5 to 2.5 gal per hour per ton, depending upon the design of the condenser and the quality of water used. Hence, total water consumption for an evaporative condenser is between 3 and 4 gal per hour per ton.

Some evaporative condensers are available equipped with desuperheating coils, which are usually installed in the leaving air stream. The hot gas leaving the compressor passes first through the desuperheating coils where its temperature is reduced before it enters the condensing coils. The desuperheating coils tend to increase the over-all capacity of the condenser and reduce the scaling rate by lowering the temperature of the wetted tubes. Too, often the receiver tank is located in the sump of the evaporative condenser in order to increase the amount of liquid subcooling.

14-19. Rating and Selection of Evaporative Condensers. Table R-17 is a typical evaporative condenser rating table. Notice that the ratings are based on the temperature difference between the condensing temperature and the design wet bulb temperature. The following sample selection is reprinted directly from the manufacturer's catalog data:*

Example 14-17. Select an evaporative condenser for the following conditions:

6-ton evaporator load (Refrigerant-12)
20° evaporator temperature
78° entering wet bulb temperature
105° F condensing temperature

Solution. Since the rating table is in terms of evaporator load at 40° F, it is necessary to correct for other evaporator temperatures by using a correction factor from R-17B as follows:
Tons × evaporator correction factor
= Rating table tons
Therefore, 6 × 1.05 = 6.3 tons.

Referring to Table R-17A, the E-135F has a capacity of only 5.6 tons at 78° F entering wet bulb and 105° F condensing temperature. It does, however, have the required capacity of 6.3 tons at between 105° F and 110° F condensing temperature.

The compressor ratings should then be checked to see if the compressor originally selected has the required capacity at between 105° F and 110° F condensing temperature. If not, it will be necessary to select the next larger size evaporative condenser or compressor to do the job.

The next larger size evaporative condenser, the E-270F, has a capacity of 11.2 tons at the given conditions; however, the required capacity of 6.3 tons will be obtained at a condensing temperature between 90 and 95° F. The compressor selection should then be made for these conditions.

14-20. Water Regulating Valves. The water flow rate through a water-cooled condenser on a waste water system is automatically controlled by a water regulating valve (Fig. 14-18). The valve is installed on the water line at the inlet of the condenser and is actuated by the compressor discharge (Fig. 14-7). When the compressor is in operation, the valve acts to modulate the flow of water through the condenser in response to changes in the condensing pressure. An increase in the condensing pressure tends to collapse the bellows further and open the valve wider against the tension of the range spring, thereby increasing the water flow rate through the condenser. Likewise, as the condensing pressure decreases, the valve moves toward the closed position so that the flow rate through the condenser is reduced accordingly. Although the regulating valve tends to maintain the condensing pressure constant within reasonable limits, the condensing pressure will usually be considerably higher during periods of peak loading than during those of light loading.

* McQuay Products.

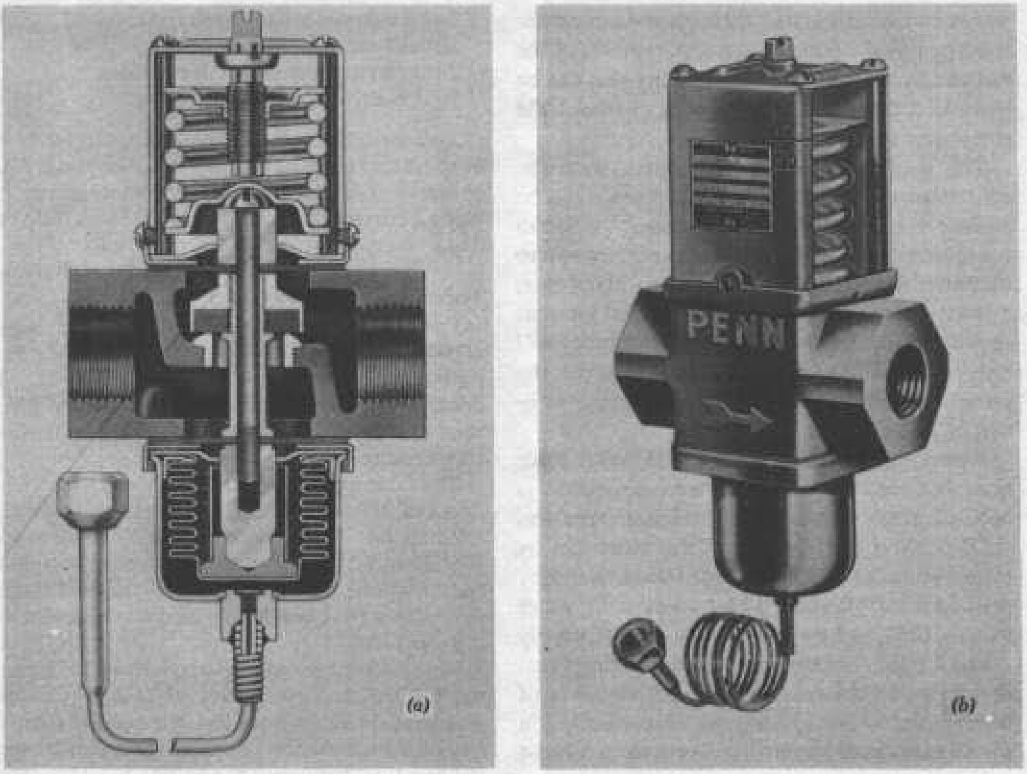


Fig. 14-18. Typical threaded-type water regulating valve. Larger sizes are available with flange connections. (*a*) Cross-sectional view showing principal parts. (*b*) Exterior view. (Courtesy Penn Controls, Inc.)

When the compressor cycles off, the water valve remains open and water continues to flow through the condenser until the pressure in the condenser is reduced to a certain predetermined minimum, at which time the valve closes off completely and shuts off the water flow. When the compressor cycles on again, the water valve remains closed until the pressure in the condenser builds up to the valve opening pressure, at which time the valve opens and permits water to flow through the condenser. The opening pressure of the valve is approximately 7 psi above the shut-off pressure.

The water valve is set for the desired shut-off pressure by adjusting the tension of the range spring. The minimum operating pressure for the valve, that is, the shut-off pressure, must be set high enough so that the valve will not remain open and permit water to flow through the condenser when the compressor is on the off cycle. Since the saturation temperature of the refrigerant in the condenser can never be lower than the ambient temperature at the condenser, the shut-off point of the water valve should be set at a saturation pressure corresponding to the maximum ambient temperature in the summertime at the condenser location. Too, the shut-off pressure of the valve must be high enough so that the minimum condensing temperature in the wintertime is sufficiently high to provide a pressure differential across the refrigerant control large enough to assure its proper operation.

The capacity of water regulating valves varies with the size of the valve and the pressure drop across the valve orifice. The available pressure drop across the valve orifice is determined by subtracting the pressure drop through the condenser and water piping from the total pressure drop available at the water main.

Water regulating valves are usually selected from flow charts (Table R-18). In order to select the proper valve from the flow chart, the following data must be known: (1) the desired water quantity in gpm; (2) the maximum ambient temperature in the summertime; (3) the desired condensing temperature; and (4) the available water pressure drop across the valve.

The following selection procedure and sample selection are reprinted directly from the literature of the manufacturer:*

1. Draw horizontal line across upper half of Flow Chart (Table R-18) through the required flow rate.

2. Determine refrigerant condensing pressure rise above valve opening point.

   a. Valve closing point (to assure closure under all conditions) must be the refrigerant condensing pressure equivalent to the highest ambient air temperature expected at time of maximum load. Read this in psig from "Saturated Vapor Table" for refrigerant selected.

   b. Read from the same table the operating condensing pressure corresponding to selected condensing temperature.

   c. Valve opening point will be about 7 psi above closing point.

   d. Subtract opening pressure from operating pressure. This gives the condensing pressure rise.

3. Draw horizontal line across lower half of Flow Chart through this value.

4. Determine the water pressure drop through the valve—this is the pressure actually available to force the water through the valve.

   a. Determine the minimum water pressure available from city mains or other source.

   b. From condensing unit manufacturer's tables read pressure drop through condenser corresponding to required flow.

   c. Add to this estimated or calculated drop through piping, etc., between water valve and condenser, and from condenser to drain (or sump of cooling tower).

   d. Subtract total condenser and piping drop from available water pressure. This is the available pressure drop through the valve.

Example 14-18. The required flow for an R-12 system is found to be 27 gpm. Condensing pressure is 125 psig and the maximum ambient temperature estimated at 86° F. City water pressure is 40 psig and manufacturer's table gives drop through condenser and accompanying piping and valves as 15 psi. Drop through installed piping approximately 4 psi. Select proper size of water regulating valve from Table R-18.

* By permission of Penn Controls, Inc., Goshen, Indiana.

Solution

1. Draw a line through 27 gpm—see dotted line, upper half of Flow Chart (Table R-18).
2. Closing point of valve is pressure of R-12 corresponding to 86° F ambient = 93 psig.
3. Opening point of valve is 93 + 7 = 100 psig.
4. Condensing pressure rise = 125 − 100 = 25 psi.
5. Draw line through 25 psi—see dotted line, lower half of Flow Chart.
6. Available water pressure drop through valve = 40 − 19 = 21 psi.
7. Interpolate just over the 20 psi curve—circle on lower half of Flow Chart.
8. Draw vertical line upward from this point to flow line—circle on Flow Chart marks this intersection.
9. This intersection falls between curves for 1 in. and 1¼ in. valves. The 1¼ in. valve is required.

14-21. Condenser Controls. For reasons of economy, the condensing medium is circulated through the condenser only when the compressor is operating. Hence, common practice is to cycle the condenser fan and/or pump on and off with the compressor. This is usually accomplished by electrically interlocking the fan and/or pump circuit with the compressor driver circuit. Method of interlocking electrical circuits are discussed in Chapter 21.

Whereas high pressure controls are always desirable as safety devices on any type of system, they are absolutely essential on all equipment employing water as the condensing medium in order to protect the equipment against damage from high condensing pressures and temperatures in the event that the water supply becomes restricted or is shut-off completely. The high pressure control has already been discussed in Section 13-13.

If a refrigerating system is to function properly and efficiently, the condensing temperature must be maintained within certain limits. As previously described, high condensing temperatures cause losses in compressor capacity and efficiency, excessive power consumption, and, in some cases, overloading of the compressor driver and/or serious damage to the compressor itself.

An abnormally low condensing temperature, on the other hand, will cause an insufficient pressure differential across the refrigerant control (condensing pressure to vaporizing pressure), which reduces the capacity of the control and results in starving of the evaporator and general unbalancing of the system.

As a general rule, low condensing temperatures result from either one or both of two principal causes: (1) low ambient temperatures and (2) light refrigerating loads. Naturally, the problem of low condensing temperatures is more acute in the wintertime when the ambient temperature and the refrigerating load are both apt to be low.

To maintain the condensing temperature at a sufficiently high level, it is necessary to make some provision for reducing or controlling the capacity of the condenser during periods when the ambient temperature is low and/or the refrigerating load is light. Although the methods employed to control the capacity of the condenser vary somewhat with the type of condenser used, all involve reducing either the quantity of condensing medium circulated or the amount of effective condensing surface. Condenser capacity control devices are usually actuated by pressure or temperature controls which respond to condensing pressure or temperature.

With regard to air-cooled condensers, the condensing temperature is maintained within the desired limits by varying the air quantity through the condenser or by causing a portion of the condenser to become filled with liquid so as to reduce the amount of effective condensing surface.

The air quantity through the condenser is varied by cycling the fan or blower or by the use

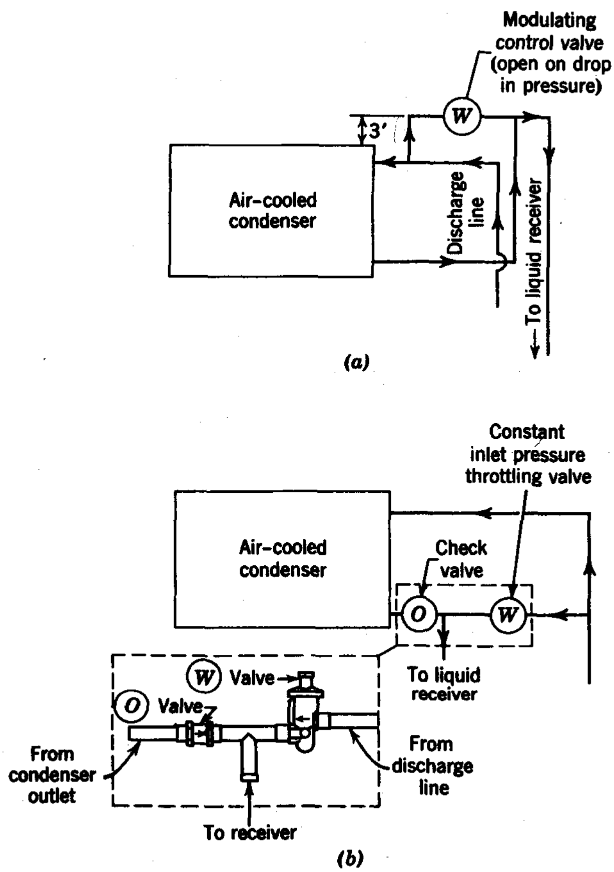


Fig. 14-19. Winterstat control of air-cooled condensers. (*a*) Loop Winterstat may be used wherever 3 feet of head room is available above the top of the condenser. This type is the simplest and lowest in cost. (*b*) No-loop Winterstat is employed where head room is not available above condenser. Valves *O* and *W* are supplied as an integral unit and must be mounted at the level of the liquid outlet of the condenser. (The Winterstat is a proprietary design of the Kramer Trenton Company and is manufactured under the following patent numbers: 2,564,310; 2,761,287; and 2,869,330.)

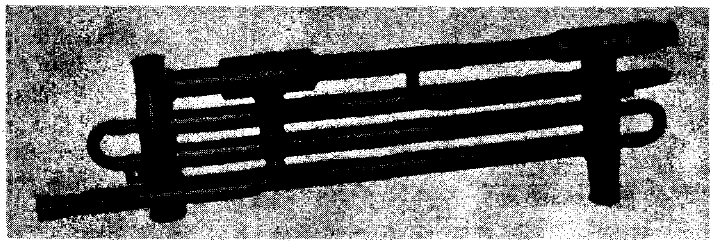


Fig. 14-20. Pressure stabilizer. (A proprietary design of Dunham-Bush, Inc.) (Courtesy Dunham-Bush, Inc.)

of dampers placed in the air stream. Because it tends to cause large fluctuations in the condensing temperature, cycling of the fan cannot be recommended as a means of controlling the capacity of air-cooled condensers. Modulating dampers installed in the air stream provide satisfactory control of the air quantity in many cases, although some difficulty is experienced with dampers when the condenser is exposed to high wind velocities.

A more satisfactory method of controlling the capacity of air-cooled condensers is to vary the amount of effective condensing surface by causing the liquid refrigerant to back up into the lower portion of the condenser whenever the condensing pressure drops below the desired minimum. To accomplish this, one design of capacity control employs a modulating valve installed in a by-pass line between the inlet and outlet of the condenser (Fig. 14-19). As the receiver pressure falls, the modulating valve opens and allows high-pressure vapor from the compressor discharge to flow through the by-pass line, thereby restricting the flow of liquid refrigerant from the condenser and causing the liquid to back up into the lower portion of the unit. The amount of discharge vapor by-passed, and therefore the amount of liquid refrigerant retained in the lower portion of the condenser, is automatically controlled by the modulating valve and depends upon the receiver tank pressure.

Another device used to restrict the amount of effective condensing surface is called a "pressure stabilizer" (Fig. 14-20). The following description of the operation of the pressure stabilizer is reprinted directly from the manufacturer's engineering data.*

The pressure stabilizer is a heat transfer surface which transfers the heat from the hot gas discharge of the compressor to the subcooled liquid leaving the condenser. This heat exchange is controlled by the regulating valve installed between the condenser and the receiver. This valve is set at the desired operating pressure, and throttles from the open position to the closed position as the condensing pressure drops. The throttling action backs up the liquid in the condenser, thus reducing the amount of effective condensing surface. The subcooled liquid coming from the condenser is forced through the heat exchanger portion of the pressure stabilizer and receives enough heat from the hot gas to satisfactorily establish the balanced pressure temperature relationship in the receiver. This assures satisfactory condensing pressure and a solid column of liquid at the refrigerant control.

The pressure stabilizer is designed with a predetermined pressure drop to insure against liquid refrigerant reheating during warm weather operations. During high ambient air temperatures, where the condensing temperature is above the setting of the regulating valve, the liquid flows through the valve, which is fully open, and thereby by-passes the heat exchanger section (Fig. 14-21*a*). In Fig. 14-21*b*, as the ambient temperature drops to 50° F the condensing temperature drops below the setting of the regulating valve. The valve then modulates toward the closed position, and this action limits the flow of liquid through the regulating valve. Consequently, the liquid backs up in the condenser until the condensing surface is reduced approximately 60%. The liquid which is forced to pass through the heat exchanger section is then heated up to the saturation temperature.

When the ambient temperature drops to 0° F (Fig. 14-21*c*), the regulating valve throttles to hold 120 psi in the condenser. The liquid logs in the condenser so that approximately 10% of the surface is utilized to condense the hot gas.

With regard to evaporative condensers, capacity control is best obtained through regulation of the air quantity through the condenser, which can be accomplished either by cycling the blower

* Courtesy Dunham-Bush, Inc.

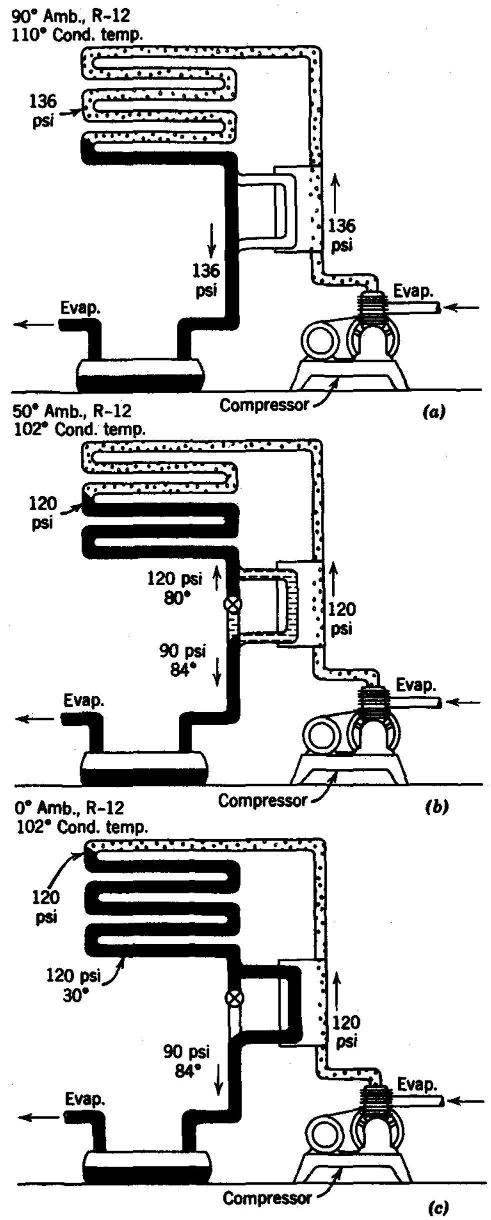


Fig. 14-21. Air-cooled condenser control employing pressure stabilizer. (Courtesy Dunham-Bush, Inc.)

or by installing dampers in the air stream. Of the two methods, the latter is usually the most satisfactory, especially where modulating dampers are used and the air quantity can be varied through a wide range.

Cycling of the pump as a means of controlling the capacity of an evaporative condenser cannot be recommended. Each time the pump cycles off a thin film of scale is formed on the condenser tubes. Consequently, frequent cycling of the condenser pump greatly increases the scaling rate, which reduces the efficiency of the condenser and greatly increases maintenance costs.

With reference to water-cooled condensers, recall that for a given load and condensing surface, the condensing temperature varies with the quantity and temperature of the water entering the condenser. Where waste water is used, the modulating action of the water-regulating valve controls the water flow rate through the condenser and maintains the condensing temperature above the desired minimum so that low condensing temperatures are not usually a problem with waste water systems. On the other hand, since the flow rate of the water through the condenser on a recirculating water system is maintained constant, the condensing temperature decreases as the temperature of the water leaving the tower decreases. Therefore, when the ambient air temperature is low, the condensing temperature will also be low unless some means is provided for restricting the flow rate through the condenser or for increasing the temperature of the water leaving the tower.

One method of controlling the condensing temperature in a recirculating water system is to install a water-regulating valve in the water line at the inlet to the condenser. The modulating action of the water valve will restrict the water flow rate through the condenser in response to a drop in the condensing pressure. When a water-regulating valve is used in a recirculating water system, the pressure drop through the valve must be taken into account in computing the total pumping head.*

Where mechanical draft cooling towers are used, the condensing temperature can be maintained at the desired level through regulation of the tower leaving water temperature. As in the case of the evaporative condenser, this can be accomplished by cycling the tower fan or by installing dampers in the air stream.

* Except in those cases where they have a specific function, water-regulating valves should never be used in recirculating water systems, since they tend to restrict the water flow and increase the pumping head unnecessarily.

14-22. Winter Operation. When the compressor and/or condenser are so located that they are exposed to low ambient temperatures, the pressure in these parts may fall considerably below that in the evaporator during the compressor off-cycle. In such cases, the liquid refrigerant, which otherwise would remain in the evaporator, very often tends to migrate to the area of lower pressure in the compressor and condenser. With no liquid refrigerant in the evaporator, an increase in evaporator temperature is not reflected by a corresponding increase in the evaporator pressure, and, where the system is controlled by a low pressure motor control, the rise in evaporator pressure may not be sufficient to actuate the control and cycle the system on in response to an increase in the evaporator temperature.

Corrective measures are several. One is to install a thermostatic motor control in series with the low pressure control. The thermostat is adjusted to cycle the system on and off, whereas the low pressure control serves only as a safety device. Another, and usually more practical, solution is to isolate the condenser during the off-cycle. One method of isolating the condenser during the off-cycle is illustrated in Fig. 14-22. The check valve (*C*) in the condenser liquid line prevents the refrigerant from boiling off in the receiver and backflowing to the condenser during the off-cycle. The (*R*) valve, which closes on drop of pressure at the valve inlet, closes when the compressor stops, preventing the flow of refrigerant from the evaporator, through the compressor valves and discharge line, into the condenser. With the condenser isolated, the evaporator pressure can build up and start the compressor regardless of the ambient temperature at the condenser.

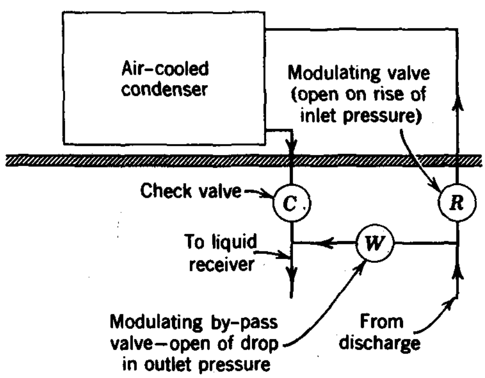


Fig. 14-22. Sure-start Winterstat provides normal head and receiver pressures when the compressor starts by allowing the compressor to impose its full discharge pressure on the liquid through the open (W) valve. When the receiver pressure is up to normal, the (*R*) valve opens and allows the discharge gas to flow to the condenser. (Courtesy Kramer Trenton Company.)

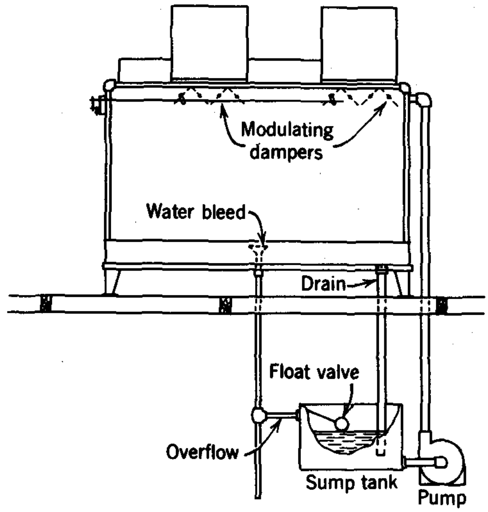


Fig. 14-23. Evaporative condenser equipped with modulating dampers for capacity control. Protected auxiliary pump is designed to prevent freezing during winter operation. (Courtesy Refrigeration Engineering Inc.)

Another and rather obvious problem concerning the operation of evaporative condensers and cooling towers in the wintertime is the danger of freezing when the equipment is exposed to freezing temperatures. In general, the measures employed to prevent freezing are similar to those used to prevent low condensing temperatures, that is, controlling the air quantity through the tower by the use of dampers or by cycling the fan. In addition, an auxiliary sump must be installed in a warm location and the piping arranged so that the water drains by gravity into the auxiliary sump and does not remain in the tower or condenser sump (Figs. 14-23 and 14-24).

14-23. Condenser and Tower Maintenance. As a general rule, air-cooled condensers require little maintenance other than regular lubrication of the fan and motor bearings. However, the

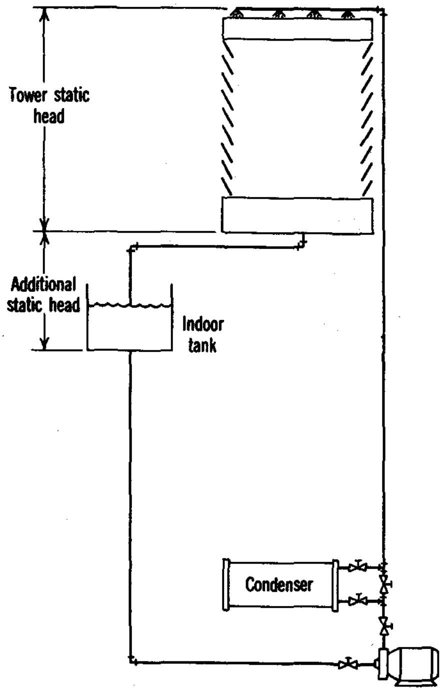


Fig. 14-24. Protected indoor tank.

fan blades and condensing surface should be inspected occasionally for the accumulation of dust and other foreign materials. These parts should be kept clean in order to obtain high efficiency from the condenser.

Any type of condenser employing water is subject to scaling of the condenser tubes, corrosion, and the growth of algae and bacterial slime on all wetted surfaces. The latter is controlled by frequent cleaning of the infected parts and by the use of various algaecides which are available commercially.

As previously stated, the scaling rate depends primarily upon the condensing temperature and the quality of water used. The scaling rate will be relatively low where the condenser leaving water temperature is below 100° F. Too, the importance of providing for the recommended amount of bleed-off cannot be overemphasized with regard to keeping the scaling rate at a minimum. In addition, a number of chemical companies have products which when added to the sump water considerably reduce the scaling rate.

Scale can be removed from the condenser tubes by applying an approved inhibited acid compound, many of which are available in either liquid or powder form. After the tower or condenser sump has been drained, cleaned, and filled with fresh water, the cleaning compound can be added directly to the sump water. The pump is then started and the cleaner is circulated through the system until the system is clean, at which time the sump is again drained, flushed, and filled with clean water before the system is placed in normal operation.

It should be pointed out that descaling compounds have an acid base and should not be allowed to contact grass, shrubs, or painted surfaces. Therefore, it is usually advisable to remove the cooling tower spray nozzles, if any, in order to minimize the danger of damaging shrubs or painted surfaces with drift from the tower.

When rapid descaling of the condenser tubes is required, an inhibited solution (18%) of muriatic acid may be used. However, muriatic acid should be used only on the condenser tubes. The system pump should not be used to circulate the acid. A small pump having an acid resistant impeller (brass or nylon) may be used for this purpose (see Fig. 14-25). After the condenser is clean, it should be flushed with clean water or with an acid neutralizer as recommended by the manufacturer.

Corrosion is usually greatest in areas near salt water or in industrial areas where relative

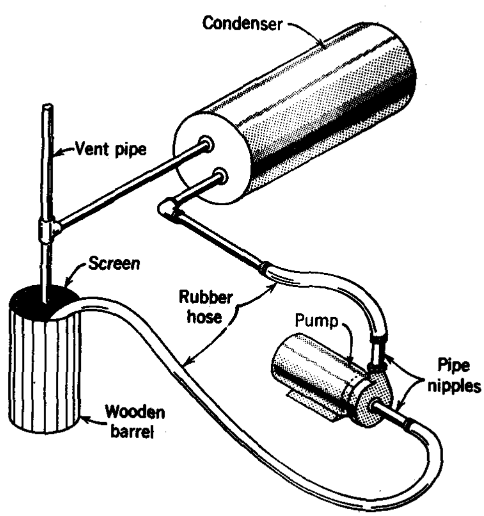


Fig. 14-25. Apparatus for descaling condenser.

large concentrations of sulfur and other industrial fumes are found in the atmosphere. Corrosion damage is minimized by regular cleaning and painting of the affected parts and by application of protective coatings of various types.

PROBLEMS

1. An R-12 system is operating at an evaporator temperature of 0° F and a condensing temperature of 100° F. From Chart 14-1, determine the heat load on the condenser in Btu per minute per ton of refrigeration. *Ans.* 257 Btu/min/ton

2. An R-22 system operating with a 40° F evaporator and a 110° F condenser has an evaporator load of 10 tons. Determine the heat load on the condenser in Btu/hr.
Ans. 141,000 Btu/hr

3. The heat rejected to a water-cooled condenser is 120,000 Btu/hr. How many square feet of effective tube surface must this condenser have if the *U* factor of the condenser is 100 Btu/hr/sq ft/° F and the METD is 5° F at the desired gpm? *Ans.* 240 sq ft

4. The heat load on the evaporator of an air conditioning system is 60,000 Btu/hr. If the coefficient of performance of the system is 4 : 1, what is the heat load on the condenser in Btu/hr?
Ans. 75,000 Btu/hr

5. An R-12 waste water system operating at a 40° F suction temperature and a 105° F condensing temperature has an evaporator load of 5 tons. If the condenser is selected for a 12° F water temperature rise, how many gpm must be circulated through the condenser?
Ans. 11.5 gpm

6. Seventy-two gallons of water per minute are circulated through a water-cooled condenser. If the temperature rise of the water in the condenser is 14° F, what is the heat load on the condenser? *Ans.* 504,000 Btu/hr

7. An R-12 air conditioning system operating with an evaporator temperature of 40° F and a condensing temperature of 120° F has an evaporator load of 60,000 Btu/hr. 4500 cfm of air are circulated over the condenser. If the temperature of the air entering the condenser is 90° F, compute: (*a*) the leaving air temperature and (*b*) the METD.
Ans. (*a*) 104.6° F (*b*) 21.89° F

8. If the air-cooled condenser in Problem 7 has a free face area of 5.5 sq ft, what is the velocity of the air through the condenser?
Ans. 818 fpm

9. From Table R-12, select an air-cooled condenser for a compressor having a capacity of 42,000 Btu/hr if the design suction and discharge temperatures are 40° F and 130° F, respectively, and the outdoor design dry bulb temperature for the region is 95° F.

10. Select a shell-and-tube water-cooled condenser for an R-12 system to meet the following conditions:

Refrigeration load and evaporator	60 tons
Evaporator temperature	40° F
Condensing temperature	110° F
Water quantity	2.5 gpm/ton

Untreated cooling tower water enters condenser at 85° F.

11. Rework Problem 10 using a condensing temperature of 120° F.

12. A cooling tower and a water-cooled condenser (with by-pass) are operating with a condenser load of 240,000 Btu/hr. Forty-eight gpm are circulated through the condenser and 32 gpm are by-passed. The ambient wet bulb temperature is 78° F and the tower approach is 7° F. Determine:

(*a*) The temperature of the water entering the condenser. *Ans.* 85° F

(*b*) The temperature of the water leaving the condenser. *Ans.* 95° F

(*c*) The temperature of the water entering the cooling tower. *Ans.* 91° F

(*d*) The tower range. *Ans.* 6° F

13. A compressor on a Refrigerant-12 system has a capacity of 50 tons. The design wet bulb temperature is 78° F. The desired condenser water entering temperature is 85° F and the desired temperature rise through the condenser is 12° F. Select a cooling tower from Table R-15 and determine:

(*a*) The total gpm circulated over the tower

(*b*) The temperature of the water entering the tower

(*c*) The temperature of the water leaving the condenser

(*d*) The tower range

(*e*) The gpm circulated through the condenser

(*f*) The gpm by-passed

14. Select an evaporative condenser for the following conditions:

Refrigerant-12 system
Evaporator load—10 tons
Evaporator temperature—40° F
Wet bulb temperature of entering air—78° F
Condensing temperature—105° F

15

Fluid Flow, Centrifugal Liquid Pumps, Water and Brine Piping

15-1. Fluid Pressure. The total pressure exerted by any fluid is the sum of the static and velocity pressures of the fluid, viz:

$$p_t = p_s + p_v \qquad (15\text{-}1)$$

where p_t = the total pressure
p_s = the static pressure
p_v = the velocity pressure

All flowing fluids possess kinetic energy and therefore exert a force or pressure in the direction of flow. The pressure exerted by a fluid which is the direct result of fluid motion or velocity is called the velocity pressure of the fluid. Any pressure exerted by a fluid which is not the direct result of fluid motion or velocity, regardless of the force causing the pressure, is called the static pressure of the fluid. For fluids at rest (static), the velocity pressure is equal to zero and the total pressure is equal to the static pressure. Whereas velocity pressure acts only in the direction of flow, static pressure acts equally in all directions. This is easily demonstrated through the use of an example employing a gravitational column.

It was shown in Chapter 1 that the action of gravity on any body causes the body to exert a force which is commonly referred to as the weight of the body. For a solid material, because of the rigid molecular structure, the gravitational force or pressure is exerted in a downward direction only. However, because of the loose molecular structure of fluids, the gravitation force or pressure exerted at any point in a body of fluid acts equally in all directions—up, down, and sideways, and always at right angles to any containing surfaces. When no force other than the force of gravity is acting on the fluid, the pressure at any depth in a body of fluid is proportional to the weight of fluid above that depth. When an external force in addition to the force of gravity is applied to the liquid, the pressure at any depth in the fluid is proportional to the weight of the fluid above that depth, plus the pressure caused by the external force.

For example, assume that a flat-bottomed container 1 sq ft in cross section and 10 ft high is filled to the top with water at a temperature of 60° F (Fig. 15-1). Since water at 60° F has a density of 62.4 lb per cubic foot, if the pressure of the atmosphere on the surface of the water is neglected, the total force acting on the bottom of the tank due to the weight of the water alone is 624 lb (10 × 62.4). Since the base area of the tank is 1 sq ft, the pressure exerted on the bottom of the tank is 624 psf or 4.33 psi (624/144). Since this pressure acts equally in all directions, it is exerted on the sides of the tank at the base as well as on the bottom of the tank.

Assume now that level *A* in the water column is exactly 1 ft below the surface of the water. The volume and weight of water above this level are 1 cu ft and 62.4 lb, respectively. Since this weight of water is also evenly distributed over an area of 1 sq ft, the fluid pressure acting in all directions from any point at level *A* is 62.4 psf or 0.433 psi. Similarly, the volume and weight of water above level *B*, which is located 5 ft below the surface of the water, are 5 cu ft and 312 lb (5 × 62.4), respectively, and the fluid pressure at this level is 312 fsf or 2.165 psi.

If the force exerted on the top of the water by the pressure of the atmosphere is taken into account, the pressure of the water at any level in the tank will be increased by an amount equal to the pressure of the atmosphere. Assuming normal sea level pressure, the fluid pressures at levels *A* and *B* are 15.129 psi (0.433 + 14.696) and 16.861 (2.165 + 14.696), respectively, while the pressure at the base of the tank is 19.026 psi (4.33 + 14.696). However, it should be recognized that since the pressure of the atmosphere is exerted also on the outside of the tank the pressure tending to burst the tank is still only that resulting from the gravitational effect on the water alone.

For any noncompressible fluid (liquid), the pressure exerted by the fluid at any level in a fluid column is directly proportional to the depth of the fluid at that level.* Hence, the pressure of a liquid at any level in a column of liquid can be determined by multiplying the depth at that level times the density of the fluid, viz:

$$\text{Pressure (psf)} = \text{depth (ft)} \times \text{density (lb/cu ft)} \tag{15-2}$$

$$\text{Pressure (psi)} = \frac{\text{depth (ft)} \times \text{density (lb/cu ft)}}{144} \tag{15-3}$$

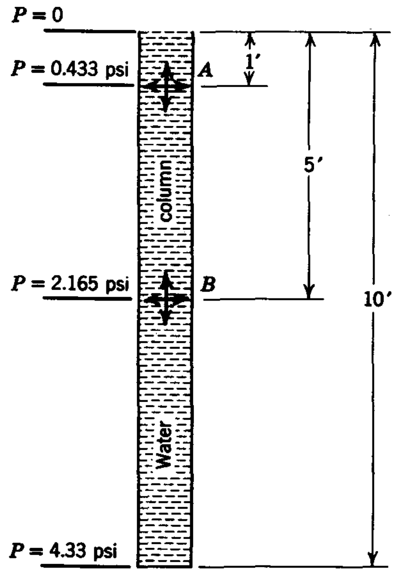


Fig. 15-1. Illustrating head-pressure relationship.

15-2. Head-Pressure Relationship. The vertical distance between any two levels in a column of liquid is called the "head" of the liquid at the lower level with respect to the upper level. For example, with respect to level *B* in Fig. 15-1, the head of the water at the base of the column is 5 ft. With respect to the top of the column, the head of the water at the base of the column is 10 ft. Similarly, with respect to the top, the water heads at levels *A* and *B* are 1 ft and 5 ft, respectively.

Since the depth of the liquid at any level in a liquid column is equal to the head of the liquid at that level with respect to the top of the column, the head can be substituted for depth in Equation 15-3 and the following relationship between head and pressure is established:

$$\text{Pressure (psi)} = \frac{\text{Head (ft)} \times \text{density (3/cu ft)}}{144} \tag{15-4}$$

$$\text{Head (ft)} = \frac{\text{Pressure (psi)} \times 144}{\text{Density (lb/cu ft)}} \tag{15-5}$$

It is evident from the foregoing that there is a definite and fixed relationship between the head and the pressure of any liquid, the head-pressure ratio for any given liquid being dependent upon the density of the liquid. For example, in the case of water, the head-pressure ratio is 2.31 ft to 1 psi. For mercury, the head-pressure ratio is 2.04 in. to 1 psi. This means that a pressure of 1 psi is equivalent to head of 2.31 ft of water column or 2.04 in. of mercury column. Conversely, a 1 ft column of water (1 ft water head) is equivalent to 0.433 psi, whereas a 1 ft column of mercury (1 ft mercury head) is equivalent to 24.48 psi.

With respect to the head-pressure relationship, the following general statements can be made:

1. For any liquid of given and uniform density, the pressure exerted by the liquid is directly proportional to the head of the liquid.
2. At any given head, the pressure exerted by any liquid is directly proportional to the density of the liquid. Liquids having different densities will exert different pressures at the same head.

15-3. Static and Velocity Heads. The total head of any fluid is the sum of the static and velocity heads of the fluid, viz:

$$h_t = h_s + h_v \tag{15-6}$$

where, h_t = the total head in feet
h_s = the static head in feet
h_v = the velocity head in feet

The static head of any liquid is expressed as the height in feet (or inches) of a gravitational column of that liquid which would be required

* This is not true of a compressible fluid because the density of a compressible fluid varies with the depth.

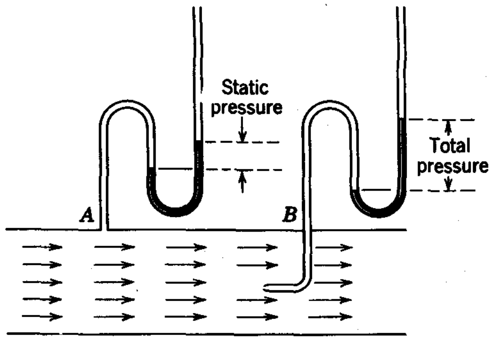


Fig. 15-2. Illustrating relationship between the static, velocity, and total pressures of a fluid flowing in a circuit.

to produce a base pressure equal to the static pressure of the liquid. That is, the head in feet of liquid column equivalent to the static pressure of the liquid is called the static head of the liquid. Likewise, the head in feet of liquid column equivalent to the velocity pressure of a liquid is called the velocity head of the liquid.

The fundamental relationship between velocity and velocity head is established by Galileo's law, which states in effect that all falling bodies, regardless of weight, accelerate at equal rates and that the final velocity of any falling body, neglecting friction, depends only upon the height from which the body falls. Hence, the height in feet from which a body must fall in order to attain a given velocity is the velocity head corresponding to that velocity. The velocity head corresponding to any given velocity can be determined by applying the following equation:

$$h_v = \frac{V^2}{2g} \qquad (15\text{-}7)$$

where, h_v = the velocity head in feet
V = the velocity in feet per second (fps)
g = the acceleration due to gravity (32.2 ft/sec/sec)

By combining and/or rearranging Equations 15-7 and 15-4, the following relationships are established:

To convert velocity head to velocity pressure,

$$p_v = \frac{h_v \times \rho}{144} \qquad (15\text{-}8)$$

To convert velocity to velocity pressure,

$$p_v = \frac{V^2 \times \rho}{2g \times 144} \qquad (15\text{-}9)$$

To convert velocity head to velocity,

$$V = \sqrt{2g \times h_v} \qquad (15\text{-}10)$$

To convert velocity pressure to velocity,

$$V = \sqrt{\frac{2g \times p_v \times 144}{\rho}} \qquad (15\text{-}11)$$

15-4. Head-Energy Relationship. Although the term "head" itself is entirely independent of weight or density, it should be recognized that the head of any fluid is numerically equal to the energy per pound of fluid. For this reason, head is often used to express energy per pound of fluid.

The basic relationship of head to energy or work is shown in the following equation:

$$\text{Energy or work (ft-lb)} = \text{mass (lb)} \times \text{head (ft)} \qquad (15\text{-}12)$$

Since velocity head (h_v) is equal to $V^2/2g$ (Equation 15-7), it follows that the total velocity (kinetic) energy (E_k) of any given mass (M) of fluid flowing at any given velocity (V) can be expressed as

$$E_k = M \times \frac{V^2}{2g}$$

The fact that the preceding equation is identical to Equation 1-7 indicates that the velocity head of a fluid is an expression of the kinetic energy per pound of fluid. Similarly, it can be shown also that the static head of a fluid is an expression of the potential energy per pound of fluid.

In any fluid column of uniform and constant density, the potential energy per pound of fluid is the same at all levels in the column. However, the potential energy at various levels is differently divided between the energy of position and the energy of pressure (head) depending upon the elevation. For example, in Fig. 15-1, 1 lb of water at the uppermost level in the tank has a potential energy of position with relation to the base of 10 ft-lb (1 lb × 10 ft) in accordance with Equation 1-8. Since the head at this level is zero, the potential energy of pressure (head) is also zero. On the other hand, 1 lb of water at the base of the tank has no potential energy of positions, but has pressure or head energy of 10 ft-lb (1 lb × 10 ft), according to Equation 15-12. Likewise, 1 lb of water at a level midway in the water column also has potential energy in the amount 10 ft-lb, the energy being evenly divided between the energy of position and the energy of pressure.

15-5. Static Head-Velocity Head Relationship in Flowing Fluids. The fact that the static pressure of a fluid is exerted equally in all directions, whereas the velocity pressure of the fluid is exerted only in the direction of flow, makes it relatively simple to measure the static and velocity pressures (or heads) of a fluid flowing in a conduit. This is illustrated in Fig. 15-2. Notice that tube A is so connected to the conduit that the opening of the tube is exactly perpendicular to the line of flow. Since only the static pressure of the fluid will act in this direction, the height of the fluid column in tube A is a measure of the static pressure or static head of the fluid in the conduit. On the other hand, tube B is so arranged in the conduit that the opening of the tube is directly in the line of flow. Since both the static pressure and the velocity pressure of the flowing fluid act on the opening of tube B, the height of the liquid column in tube B is a measure of the total pressure or total head of the fluid. Since the total pressure or head of a fluid is the sum of the static and velocity pressures or heads, it follows that the difference in the heights of the two fluid columns is a measure of the velocity pressure or velocity head of the fluid in the conduit.

If losses because of friction are neglected, the total pressure or head of a flowing fluid will be the same at all points along the conduit. However, the total head may be differently divided between static head and velocity head at the several points, depending upon the velocity of the fluid at these points.

For any given flow rate (quantity of flow), the velocity of the fluid flowing in a conduit varies inversely with the cross-sectional area of the conduit. This relationship is expressed by the basic equation

$$V = \frac{Q}{A} \tag{15-13}$$

where V = the velocity in feet per second
Q = the flow rate in cubic feet per second
A = the cross-sectional area of the conduit in square feet

NOTE. When Q is in cubic feet per minute, V will be in feet per minute.

In accordance with Equation 15-13, the fluid velocity (and velocity head) in section B of the conduit in Fig. 15-3 is greater than that in sections A and C, since the cross-sectional area of section B is less than that of sections A and C. Assuming that the total head of the fluid is the same at all points in the conduit, it follows then that the static head-velocity head ratio in section B is different from that in sections A and C. As the fluid flows through the reducer between sections A and B, static head is converted to velocity head (pressure is converted to velocity). Conversely, as the fluid flows through the increaser between sections B and C, velocity head is converted back into static head (velocity is converted to pressure).

In view of the head-energy relationship, it is evident that the conversion of static head to velocity head is in fact a conversion of potential energy (pressure) into kinetic energy (velocity). Likewise, the conversion of velocity head to static head represents a conversion of kinetic energy (velocity) to potential energy (pressure).

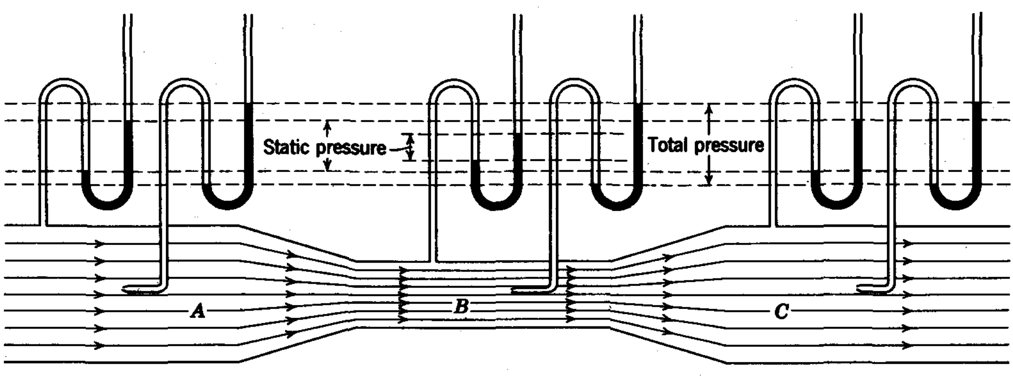


Fig. 15-3. Illustrating changes in static-velocity pressure ratio resulting from changes in conduit area.

15-6. Friction Head. It has already been established that a fluid flowing in a conduit will suffer losses in energy (converted into heat) as a result of the work of overcoming friction. These energy losses are frequently expressed in terms of pressure drop or head loss. The pressure drop in psi or the head loss in feet experienced by a fluid flowing between any two points in a conduit is known as the friction head or friction loss between these two points.

The amount of pressure drop or head loss suffered by a fluid due to friction in flowing through a conduit varies with a number of factors: (1) the viscosity and specific gravity of the fluid, (2) the velocity of the fluid, (3) the hydraulic radius (ratio of perimeter to diameter) of the conduit, (4) the roughness of the internal surface of the conduit, and (5) the length of the conduit.

Obviously, the mathematical evaluation of all these factors is too laborious for most practical purposes. As a general rule, the friction loss in piping is determined from charts and tables.

The pressure (friction) loss in psi per hundred feet of straight pipe is given in Charts 15-1 and 15-2 for various flow rates in various sizes of pipe. Chart 15-1 applies to smooth copper tube, whereas Chart 15-2 applies to fairly rough pipe. Since the pressure loss for a given pipe size and flow rate is proportional to the length of the pipe, the pressure loss through any given length of straight pipe is determined by the following equation:

$$\text{Total pressure loss (ft)} = \frac{\text{Total length of pipe (ft)}}{100} \times \text{pressure loss/100 ft (psi)} \qquad (15\text{-}14)$$

Pipe fittings, such as elbows, tees, valves, etc., offer a greater resistance to flow than does straight pipe and therefore must be taken into account in determining the total friction loss through the piping. For convenience, this is done by considering the fittings as having a resistance equal to a certain length of straight pipe called the "equivalent length." Table 15-1 lists the equivalent length of straight pipe for various types of fittings and valves. Notice that the equivalent length varies with the size of the pipe.

When the equivalent length of the fittings is added to the actual length of straight pipe, the result is called the "total equivalent length." This value is then applied in Equation 15-14 to determine the total friction loss through the piping.

Example 15-1. A water piping system consists of 128 ft of 2 in. straight pipe, 6 standard elbows, and 2 gate valves (full open). Using fairly rough pipe, if the flow rate through the system is 40 gpm, determine:

(*a*) The total equivalent length of straight pipe

(*b*) The total friction loss through the piping in psi and in feet of water column.

Solution. From Table 15-1, the equivalent lengths of 2 in. standard elbows and 2 in. gate valves (full open) are 5 ft and 1.2 ft, respectively. From Chart 15-2, for a flow rate of 40 gpm, the friction loss per hundred feet of 2 in. nominal pipe is 3 psi. From Table 1-1, a pressure of 1 psi is equivalent to 2.31 ft of water column.

(*a*) Total equivalent length

Straight pipe	= 128.0 ft
Six 2 in. elbows @ 5 ft	= 30.0
Six 2 in. gate valves @ 1.2 ft	= 2.4
	= 160.4 ft

(*b*) Applying Equation 15-14, the total friction loss through the piping

$$= \frac{160.4}{100} \times 3$$
$$= 4.8 \text{ psi}$$

Converting to ft H_2O

$$= 4.8 \times 2.31$$
$$= 11.09 \text{ ft } H_2O$$

Although the pressure loss determined from Charts 15-1 and 15-2 apply only to water, the charts can be used for other fluids by multiplying the water pressure loss obtained from these charts by the correction factors listed in Table 15-2.

15-7. Centrifugal Pumps. Liquid pumps used in the refrigerating industry to circulate chilled water or brine, and the condenser water are usually of the centrifugal type.

A centrifugal pump consists mainly of a rotating vane-type impeller that is enclosed in a stationary casing. The liquid being pumped is drawn in through the "eye" of the impeller and is thrown to the outer edge or periphery of the impeller by centrifugal force. Considerable velocity and pressure are imparted to the liquid in the process. The liquid leaving the periphery of the impeller is collected in the casing and directed through the discharge opening (Fig. 15-4).

Frequently, the impeller of the pump is mounted directly on the shaft of the pump-driving motor so that the pump and motor are an integral unit (Fig. 15-5). In other cases, the pump and motor are separate units and are connected together by a flexible coupling.

In general, the capacity of a centrifugal pump depends on the design and size of the pump and on the speed of the motor. For a pump of specific size, design, and speed, the volume of liquid handled varies with the pumping head

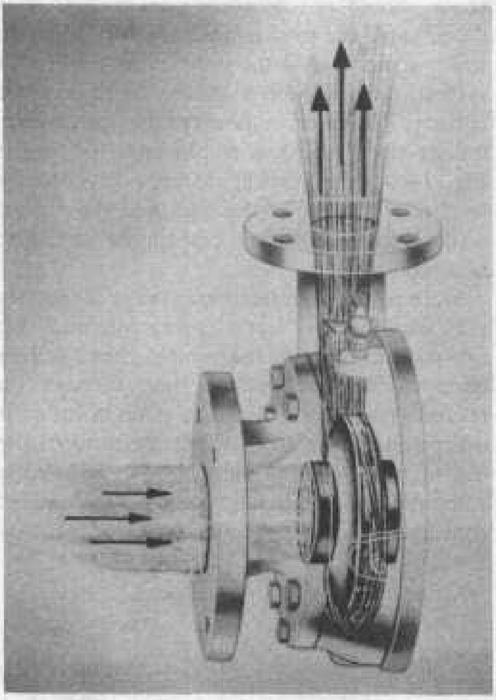


Fig. 15-4. Fluid flow through centrifugal pump. (Courtesy Ingersoll-Rand Company.)

against which the pump must work. A characteristic head-capacity curve for a typical centrifugal pump is shown in Fig. 15-6. Notice that the pumping head is maximum when the valve on the discharge of the pump is closed, at which time the pump delivery is zero. As the valve is opened, the pumping head decreases and the deliver rate increases.

Centrifugal pumps are rated in gpm of delivery at various pumping heads, that is, centrifugal pumps are rated to deliver a certain gpm against a certain pumping head. Although

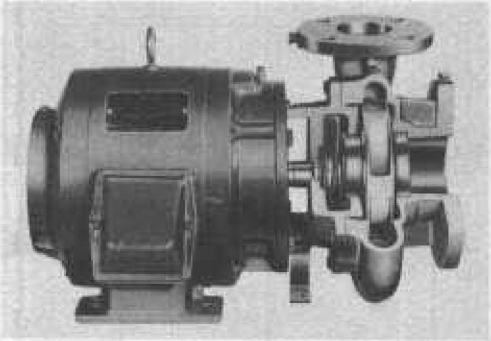


Fig. 15-5. Typical centrifugal pump and motor assembly. (Courtesy Bell & Gossett Company.)

pump ratings are available in table form, more frequently they are taken from head-capacity curves (see Chart R-19). In either case, before the proper pump can be selected from the manufacturer's ratings, it is necessary to know the required gpm and the total pumping head against which the pump must operate.

15-8. Total Pumping Head. The total pumping head is the sum of the static head and the friction head.

The static head is the vertical distance between the "free liquid level" and the highest point to which the liquid must be lifted by the pump. For the condenser-water circulating system in Fig. 15-7, the static head, measured in feet of water column, is the vertical distance in feet between the free water level in the tower basin and the tower spray header. Because of the water head in the tower basin, the water in the discharge pipe will stand to the level of the

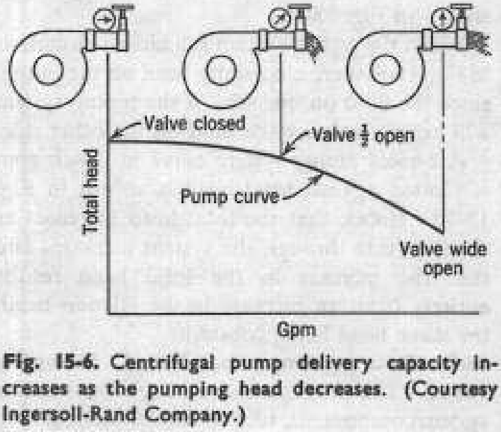


Fig. 15-6. Centrifugal pump delivery capacity increases as the pumping head decreases. (Courtesy Ingersoll-Rand Company.)

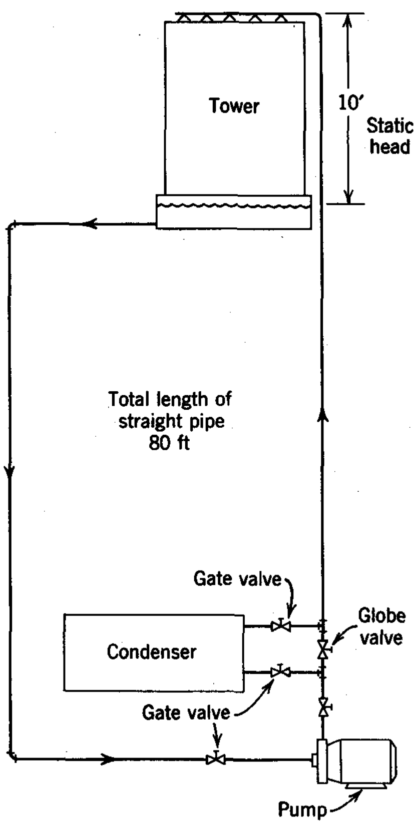


Fig. 15-7. Condenser-water circulating system.

water in the tower basin of its own accord. Therefore, the distance the water is actually lifted by the pump is only the distance from the water level in the tower basin up to the spray header. Contrast this with the pumping system shown in Fig. 15-8.

When the piping system is a closed circuit, as in Fig. 15-9, there is no static head on the pump, since the fluid on one side of the piping system will exactly balance the fluid on the other side.

A typical piping system curve in which gpm is plotted against total head is shown in Fig. 15-10. Notice that the total head increases as the flow rate through the system increases and that the increase in the total head results entirely from an increase in the friction head, the static head being constant.

15-9. Determining the Total Pumping Head. The pressure loss through the various system components, such as condensers, chillers, and cooling towers, are found in the manufacturers' rating tables.

When more than one condenser (or chiller, etc.) is used in the system, the condensers are piped in parallel and only the condenser with the largest pressure drop is considered in computing the pumping head.

The pressure loss through the cooling tower, as given by the tower manufacturer, is the total head and includes both the tower static and friction heads. Therefore, the static head of the tower should not be considered separately in determining the total pumping head. When the tower static head is the only static head in the system, the static head should be disregarded entirely. However, in the event that an auxiliary indoor storage tank is employed, as shown in Fig. 14-24, the vertical distance between the level of the water in the tank and the normal water level in the tower basin must be treated as a separate static head.

Since pump manufacturers always express the pumping head in "feet of water column," it is necessary to compute the pumping head in these units. When the pressure loss through the several system components is given in psi or in other units of pressure, it must be converted to feet of water column before it can be used in computing the pumping head. The required conversion factors are found in Table 1-1.

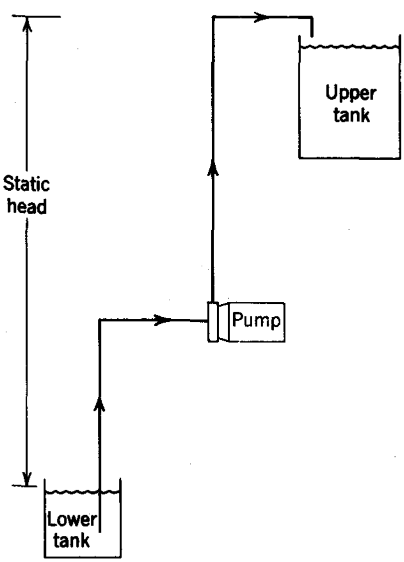


Fig. 15-8

Example 15-2. The recirculating water system shown in Fig. 15-7 is for a 10-ton refrigerating system. The flow rate over the tower is 40 gpm (4 gpm/ton). The flow rate through the condenser is 30 gpm (3 gpm/ton), with 10 gpm (1 gpm/ton) flowing through the condenser bypass. From the manufacturers' rating tables, the tower head based on 4 gpm/ton is 24 ft of water column, whereas the pressure drop through the condenser for 30 gpm is 11.2 psi or 25.9 ft of water column (11.2 × 2.31). If the size of the piping is 2 in. nominal, determine the total pumping head and select the proper pump from Chart R-19.

Solution. Total equivalent length of pipe:

Straight pipe	= 80.0 ft
3—2 in. standard elbows at 5 ft	= 15.0
2—2 in. tees (side outlet) at 12 ft	= 24.0
4—2 in. gate valves (open) at 1.2 ft	= 4.8
	= 123.8 ft

From Chart 15-2, the pressure loss per 100 ft of pipe (40 gpm and 2 in. pipe) = 3 psi

Applying Equation 15-14, the total pressure loss through the piping $= \dfrac{123.8}{100} \times 3$ = 3.71 psi

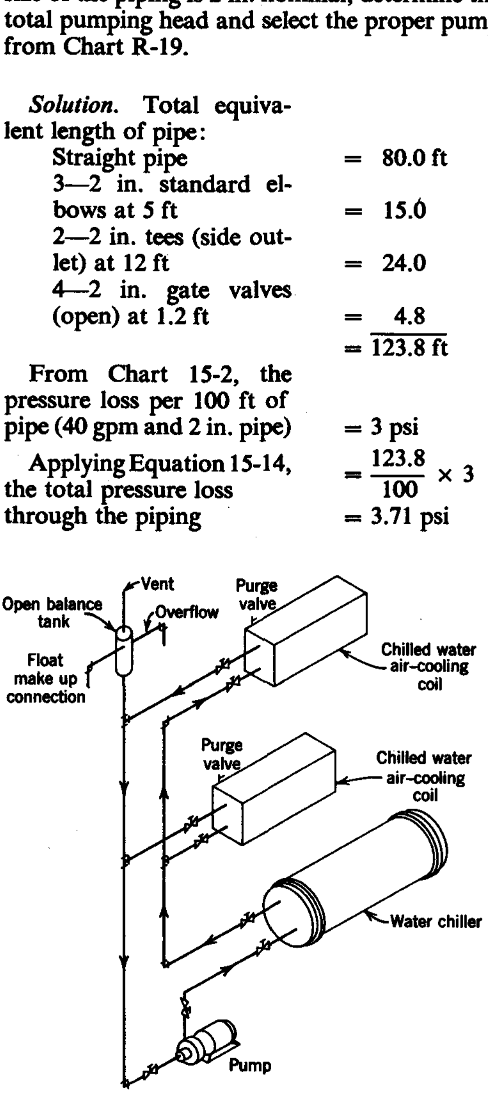


Fig. 15-9. Closed chilled water (or brine) circulating system. To compute pumping head use circuit having greatest friction loss. There is no static head.

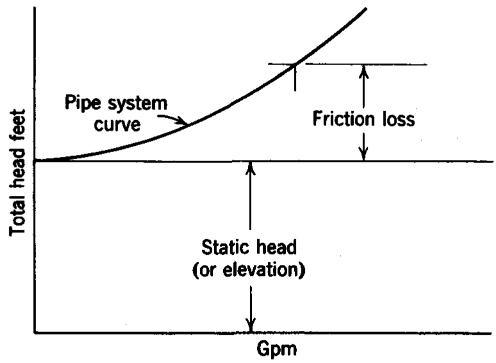


Fig. 15-10. Friction head of piping system increases as flow rate through system increases. (Courtesy Ingersoll-Rand Company.)

Converting to ft H_2O	= 3.71 × 2.31
	= 8.58 ft H_2O
Total pumping head	
Piping	= 8.58 ft
Condenser	= 25.90
Tower	= 24.00
	= 57.58 ft H_2O

From Chart R-19, select pump Model #1531-28, which has a delivery capacity of 40 gpm at a 57-ft head.

Example 15-3. At the required flow rate of 100 gpm, a certain water system has a pumping head of 60 ft of water column. Select the proper pump from Chart R-19.

Solution. Reference to Chart R-19 shows that pump Model #1531-30 is the smallest pump which can be used. However, since this pump will deliver 125 gpm at a 60-ft head, to obtain the desired flow rate of 100 gpm, the pumping head must be increased to 73 ft of water. This is accomplished by throttling the pump with a globe valve installed on the discharge side of the pump. (The pump should never be throttled on the suction side.)

15-10. Power Requirements. The power required to drive the pump depends upon the delivery rate in pounds per minute, the total pumping head, and the efficiency of the pump, viz:

$$\text{Bhp} = \frac{\text{Pounds per minute} \times \text{total head in feet}}{33{,}000 \times \text{pump efficiency}}$$

Since the flow rate is usually in gpm, a more practical equation is

$$\text{Bhp} = \frac{\text{Gpm} \times \text{total head} \times 8.33 \text{ lb/gal}}{33{,}000 \times \text{efficiency}}$$

Combining constants,

$$\text{Bhp} = \frac{\text{Gpm} \times \text{total head in feet}}{3960 \times \text{efficiency}} \qquad (15\text{-}15)$$

Equation 15-15 applies to water. When a liquid other than water is handled, the specific gravity of the liquid must be taken into account, viz:

$$\text{Bhp} = \frac{\text{Gpm} \times \text{total head} \times \text{specific gravity}}{3960 \times \text{efficiency}} \qquad (15\text{-}16)$$

From Equation 15-16, it is evident that the power required by the pump increases as the delivery rate, total head, or specific gravity increases, and decreases as the pump increases.

Typical pump horsepower and efficiency curves are shown in Fig. 15-11. Notice that pump horsepower is lowest at no delivery and increases progressively as the delivery rate increases. Hence, any decrease in the pumping head will cause an increase in both the delivery rate and the power requirements of the pump.

Pump efficiency, also lowest at no delivery, increases to a maximum as the flow rate is increased and then decreases as the flow rate is further increased. The pump efficiency curve in Fig. 15-11 indicates that the highest efficiency is obtained when the pump is selected to deliver the desired gpm when operating at some point near the midpoint of its head-capacity curve.

15-11. Water Piping Design. In general, the water piping should be designed for the minimum friction loss consistent with reasonable initial costs so that the pumping requirements are maintained at a practical minimum. Water

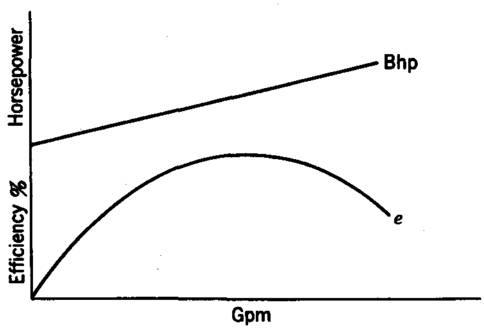


Fig. 15-11. Variations in pump horsepower and efficiency with delivery rate. (Courtesy Ingersoll-Rand Company.)

lines should be kept as short as possible and a minimum amount of fittings should be used.

Standard weight steel pipe or Type "L" copper tubing are usually employed for condenser water piping. Pipe sizes which will provide water velocities in the neighborhood of 5 to 8 fps at the required flow rate will usually prove to be the most economical. For example, assume that 150 gpm of water are to be circulated through 100 equivalent feet of piping. The following approximate values of velocity and friction loss are shown in Chart 15-2 for a flow rate of 150 gpm through various sizes of pipe:

Pipe Size (inches)	Velocity (fps)	Friction Loss per 100 ft (psi)	(ft H_2O)
2	15.5	31.5	72.8
2½	10.0	10.5	24.3
3	7.1	4.8	11.1
3½	5.2	2.0	4.6
4	3.9	1.1	2.5

Notice that whereas increasing the pipe size from 2 to 3 in. results in a considerable reduction in the friction loss (from 72.8 to 11.1 ft), a further increase in the pipe size from 3 to 4 in. reduces the friction loss by only an additional 9.4 ft of water column (11.1 to 2.54 ft), which will not ordinarily justify the increase in the cost of the pipe. Depending upon the characteristics of the available pump, either 3 in. or 3½ in. pipe should be used. For instance, assume two separate systems having pumping heads, exclusive of the friction loss in the piping, of 55 ft of water column and 65 ft of water column, respectively. Reference to Chart R-19 indicates that the only suitable pump for either of the systems is Model 31531-32, which has a delivery rate of 150 gpm at a 70-ft head. Therefore, for the system having the 55-ft head, the permissible friction loss in the piping is 15 ft (70 − 55), whereas for the system having the 65-ft head, the permissible friction loss in the piping is only 5 ft (70 − 65). For the latter system, 3½ in. pipe must be used, since the use of 3 in. pipe would result in a total pumping head in excess of the allowable 70 ft and necessitate the use of the next larger size pump. On the other hand, for the former system, 3 in. pipe is the most practical size. The use of 3½ in. pipe in this instance would result in a total pumping head of only 61 ft and would necessitate throttling of the pump discharge in order to raise the pumping head to 70 ft and obtain the desired flow rate of 150 gpm.

In designing the piping system, care should be taken to include all valves and fittings necessary for the proper operation and maintenance of the water circulating system. It is good practice to install a globe valve on the discharge side of the pump to regulate the water flow rate when the latter is critical. Too, where the piping is long and/or the quantity of water in the system is large, shut-off valves installed on the inlet and outlet of both the pump and the condenser will permit repairs to these pieces of equipment without the necessity of draining the tower. A drain connection should be installed at the lowest point in the piping and the piping should be pitched downward so as to assure complete drainage during winter shut down.

The pump must always be located at some point below the level of the water in the tower basin in order to assure positive and continuous priming of the pump. When quiet operation is required, the pump may be isolated from the piping with short lengths of rubber hose. Automobile radiator hose is suitable for this purpose.

PROBLEMS

1. A water piping system consists of 135 ft of 2.5 in. nominal Type L smooth copper tube, 5 standard elbows, and 2 globe valves (full open). If the flow rate through the pipe is 60 gpm, determine:

(*a*) The total equivalent length of straight pipe. *Ans.* 297.5 equivalent ft

(*b*) The total friction loss through the pipe is feet of water column. *Ans.* 9.28 ft H_2O

2. Rework Problem 1 using fairly rough pipe. *Ans.* 12.02 ft H_2O

3. A recirculating condenser water system consists of 100 ft of straight pipe, and 6 standard 90° elbows. At the desired flow rate the pressure drop through the condenser is 7.5 psi and the pressure drop over the tower is 10 ft of water column. If 60 gpm are circulated through the system, determine:

(*a*) The total equivalent length of pipe. *Ans.* 130 equivalent ft

(*b*) The total head against which the pump must operate. *Ans.* 42.92 ft H_2O

4. From the manufacturer's rating curves, select a pump to fit the conditions of Problem 3.

5. For a Refrigerant-12 system, select a water regulating valve to meet the following conditions:

(*a*) Desired condensing temperature range—90° to 105°.

(*b*) Maximum entering water temperature—85° F.

(*c*) Desired water quantity through condenser at maximum loading—9 gpm.

(*d*) Pressure available at city main during period of peak loading—50 psi.

(*e*) Pressure loss through condenser and water piping—12 psi.

16

Refrigerants

16-1. The Ideal Refrigerant. Generally speaking, a refrigerant is any body or substance which acts as a cooling agent by absorbing heat from another body or substance. With regard to the vapor-compression cycle, the refrigerant is the working fluid of the cycle which alternately vaporizes and condenses as it absorbs and gives off heat, respectively. To be suitable for use as a refrigerant in the vapor-compression cycle, a fluid should possess certain chemical, physical, and thermodynamic properties that make it both safe and economical to use.

It should be recognized at the onset that there is no "ideal" refrigerant and that, because of the wide differences in the conditions and requirements of the various applications, there is no one refrigerant that is universally suitable for all applications. Hence, a refrigerant approaches the "ideal" only to the extent that its properties meet the conditions and requirements of the application for which it is to be used.

Table 16-1 lists a number of fluids having properties which render them suitable for use as refrigerants.* However, it will be shown presently that only a few of the more desirable ones are actually employed as such. Some, used extensively as refrigerants in the past, have been discarded as more suitable fluids were developed. Others, still in the development stage, show promise for the future. Tables 16-2 through 16-6 list the thermodynamic properties of some of the refrigerants in common use at the present time. The use of these tables has already been described in an earlier chapter.

* Since some of the fluorocarbon refrigerants, first introduced to the industry under the trade name "Freon," are now produced under several different trade marks, the ASRE, in order to avoid the confusion inherent in the use of either proprietary or chemical names, has adopted a numbering system for the identification of the various refrigerants. Table 16-1 lists the ASRE number designation, along with the chemical name and formula for each of the compounds listed.

16-2. Safe Properties. Ordinarily, the safe properties of the refrigerant are the prime consideration in the selection of a refrigerant. It is for this reason that some fluids, which otherwise are highly desirable as refrigerants, find only limited use as such. The more prominent of these are ammonia and some of the straight hydrocarbons.

To be suitable for use as a refrigerant, a fluid should be chemically inert to the extent that it is nonflammable, nonexplosive, and nontoxic both in the pure state and when mixed in any proportion with air. Too, the fluid should not react unfavorably with the lubricating oil or with any material normally used in the construction of refrigerating equipment. Nor should it react unfavorably with moisture which despite stringent precautions is usually present at least to some degree in all refrigerating systems. Furthermore, it is desirable that the fluid be of such a nature that it will not contaminate in any way foodstuff or other stored products in the event that a leak develops in the system.

16-3. Toxicity. Since all fluids other than air are toxic in the sense that they will cause suffocation when in concentrations large enough to preclude sufficient oxygen to sustain life, toxicity is a relative term which becomes meaningful only when the degree of concentration and the time of exposure required to produce harmful effects are specified.

The toxicity of most commonly used refrigerants has been tested by National Fire Underwriters. As a result, the various refrigerants are separated into six groups according to their degree of toxicity, the groups being arranged in descending order (Column 1 of Table 16-7). Those falling into Group 1 are highly toxic and are capable of causing death or serious injury in relatively small concentrations and/or short exposure periods. On the other hand, those classified in Group 6 are only mildly toxic, being capable of causing harmful effects only in relatively large concentrations. Since injury from the latter group is caused more by oxygen deficiency than by any harmful effects of the fluids themselves, for all practical purposes the fluids in Group 6 are considered to be nontoxic. However, it should be pointed out that some refrigerants, although nontoxic when mixed with air in their normal state, are subject to decomposition when they come in contact with an open flame or an electrical heating element. The products of decomposition thus formed are highly toxic and capable of causing harmful effects in small concentrations and on short exposure. This is true of all the fluorocarbon refrigerants (see Column 3 of Table 16-7).

16-4. Flammability and Explosiveness. With regard to flammability and explosiveness, most of the refrigerants in common use are entirely nonflammable and nonexplosive (Column 2 of Table 16-7). Notable exceptions to this are ammonia and the straight hydrocarbons. Ammonia is slightly flammable and explosive when mixed in rather exact proportions with air. However with reasonable precautions, the hazard involved in using ammonia as a refrigerant is negligible.

Straight hydrocarbons, on the other hand, are highly flammable and explosive, and their use as refrigerants except in special applications and under the surveillance of experienced operating personnel is not usually permissible. Because of their excellent thermal properties, the straight hydrocarbons are frequently employed in ultra-low temperature applications. In such installations, the hazard incurred by their use is minimized by the fact that the equipment is constantly attended by operating personnel experienced in the use and handling of flammable and explosive materials.

The "American Standard Safety Code for Mechanical Refrigeration" sets forth in detail the conditions and circumstances under which the various refrigerants can be safely used. Most local codes and ordinances governing the use of refrigerating equipment are based on this code, which is sponsored jointly by the ASRE and ASA.

The degree of hazard incurred by the use of toxic refrigerants depends upon a number of factors, such as the quantity of refrigerant used with relation to the size of the space into which the refrigerant may leak, the type of occupancy, whether or not open flames are present, the odor of the refrigerant, and whether or not experienced personnel are on duty to attend the equipment. For example, a small quantity of even a highly toxic refrigerant presents little hazard when used in relatively large spaces in that it is not possible in the event of a leak for the concentration to reach a harmful level. Too, the danger inherent in the use of toxic refrigerants is somewhat tempered by the fact that toxic refrigerants (including decomposition products) have very noticeable odors which tend to serve as a warning of their presence. Hence, toxic refrigerants are usually a hazard only to infants and others who, by reason of infirmity or confinement, are unable to escape the fumes. At the present time, ammonia is the only toxic refrigerant that is used to any great extent, and its use is ordinarily limited to packing plants, ice plants, and large cold storage facilities where experienced personnel are usually on duty.

16-5. Economic and Other Considerations. Naturally, from the viewpoint of economical operation, it is desirable that the refrigerant have physical and thermal characteristics which will result in the minimum power requirements per unit of refrigerating capacity, that is, a high coefficient of performance. For the most part, the properties of the refrigerant which influence the coefficient of performance are: (1) the latent heat of vaporization, (2) the specific volume of the vapor, (3) the compression ratio, and (4) the specific heat of the refrigerant in both the liquid and vapor states.

Except in very small systems, a high latent heat value is desirable in that the weight of refrigerant circulated per unit of capacity is less. When a high latent heat value is accompanied by a low specific volume in the vapor state, the efficiency and capacity of the compressor are greatly increased. This tends not only to decrease the power consumption but also to reduce the compressor displacement required, which permits the use of smaller, more compact equipment. However, in small systems, if the latent heat value of the refrigerant is too high, the amount of refrigerant circulated will be insufficient for accurate control of the liquid.

A low specific heat for the liquid and a high specific heat for the vapor are desirable in that both tend to increase the refrigerating effect per pound, the former by increasing the subcooling effect and the latter by decreasing the

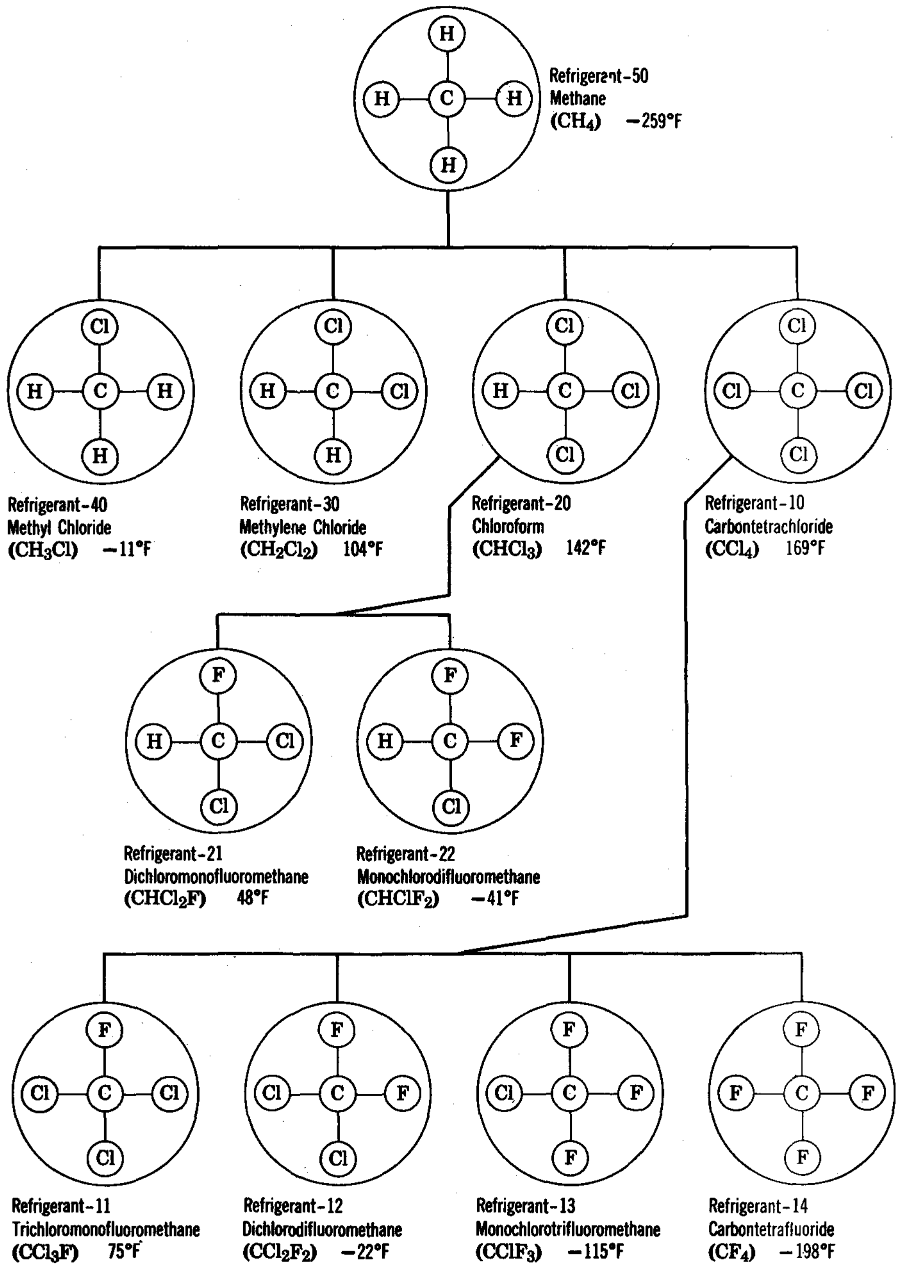


Fig. 16-1. Methane series refrigerants.

superheating effect. When both are found in a single fluid, the efficiency of a liquid-suction heat exchanger is much improved.

The effect of compression ratio on the work of compression and, consequently, on the coefficient of performance, has already been discussed in a previous chapter. Naturally, all other factors being equal, the refrigerant giving the lowest compression ratio is the most desirable. Low compression ratios result in low power consumption and high volumetric efficiency, the latter being more important in smaller systems since it permits the use of small compressors.

Too, it is desirable that the pressure-temperature relationship of the refrigerant is such that the pressure in the evaporator is always above atmospheric. In the event of a leak on the low pressure side of the system, if the pressure in the low side is below atmospheric, considerable amounts of air and moisture may be drawn into the system, whereas if the vaporizing pressure is above atmospheric, the possibility of drawing in air and moisture in the event of a leak is minimized.

Reasonably low condensing pressures under normal atmospheric conditions are also desirable in that they allow the use of lightweight materials in the construction of the condensing equipment, thereby reducing the size, weight, and cost of the equipment.

Naturally, the critical temperature and pressure of the refrigerant must be above the maximum temperature and pressure which will be encountered in the system. Likewise, the freezing point of the refrigerant must be safely below the minimum temperature to be obtained in the cycle. These factors are particularly important in selecting a refrigerant for a low temperature application.

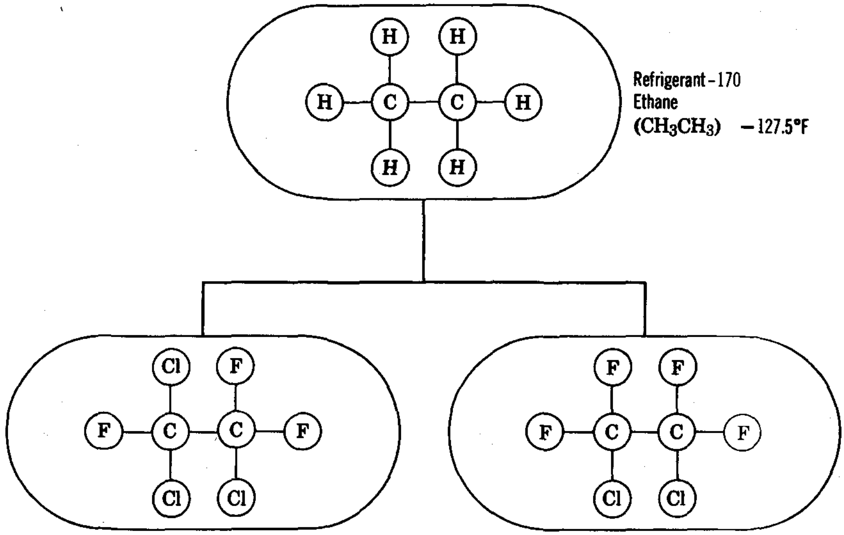
In Table 16-8, a comparison is given of the performance of the various refrigerants at standard ton conditions (5° F evaporator and 86° F condensing). Notice particularly that, with the exception of air, carbon dioxide, and ethane, the horsepower required per ton of refrigeration is very nearly the same for all the refrigerants listed. For this reason, efficiency and economy of operation are not usually deciding factors in the selection of the refrigerant. More important are those properties which tend to reduce the size, weight, and initial cost of the refrigerating equipment and which permit automatic operation and a minimum of maintenance.

16-6. Early Refrigerants. In earlier days, when mechanical refrigeration was limited to a few large applications, ammonia and carbon dioxide were practically the only refrigerants available. Later, with the development of small, automatic domestic and commercial units, refrigerants such as sulfur dioxide and methyl chloride came into use, along with methylene chloride, which was developed for use with centrifugal compressors. Methylene chloride and carbon dioxide, because of their safe properties, were extensively used in large air conditioning applications.

With the exception of ammonia, all these refrigerants have fallen into disuse and are found only in some of the older installations, having been discarded in favor of the more suitable fluorocarbon refrigerants as the latter were developed. The fluorocarbons are practically the only refrigerants in extensive use at the present time. Again, an exception to this is ammonia which, because of its excellent thermal properties, is still widely used in such installations as ice plants, skating rinks, etc. A few other refrigerants also find limited use in special applications.

16-7. Development of the Fluorocarbons. The search for a completely safe refrigerant with good thermal properties led to the development of the fluorocarbon refrigerants in the late 1920's. The fluorocarbons (fluoronated hydrocarbons) are one group of a family of compounds known as the halocarbons (halogenated hydrocarbons). The halocarbon family of compounds are synthesized by replacing one or more of the hydrogen atoms in methane (CH_4) or ethane (C_2H_6) molecules, both of which are pure hydrocarbons, with atoms of chlorine, fluorine, and/or bromine, the latter group comprising the halogen family. Halocarbons developed from the methane molecule are known as "methane series halocarbons." Likewise, those developed from the ethane molecule are referred to as "ethane series halocarbons."

The composition of the methane series halocarbons is shown in Fig. 16-1. Notice that the basic methane molecule consists of one atom of carbon (C) and four atoms of hydrogen (H). If the hydrogen atoms are replaced progressively



Refrigerant-170
Ethane
(CH_3CH_3) −127.5°F

Refrigerant-113
Trichlorotrifluoroethane
(CCl_2FCClF_2) 117.6°F

Refrigerant-114
Dichlortetrafluoroethane
$(CClF_2CClF_2)$ 38.4°F

Fig. 16-2. Ethane series refrigerants.

with chlorine (Cl) atoms, the resulting compounds are methyl chloride (CH_3Cl), methylene chloride (CH_2Cl_2), chloroform ($CHCl_3$), and carbontetrachloride (CCl_4), respectively, the last two being the base molecules for the more popular fluorocarbons of the methane series.

If the chlorine atoms in the carbontetrachloride molecule are now replaced progressively with fluorine atoms, the resulting compounds are trichloromonofluoromethane (CCl_3F), dichlorodifluoromethane (CCl_2F_2), monochlorotrifluoromethane ($CClF_3$), and carbontetrafluoride (CF_4), respectively. In the same order, the ASRE refrigerant standard number designations for these compounds are Refrigerants-11, 12, 13, and 14, the last figure in the numbers being an indication of the number of fluorine atoms in the molecule.

The molecular structure of Refrigerants-21 and 22, which are also fluorocarbons of the methane series, is shown in Fig. 16-1. Notice the presence of the hydrogen atom in each of these two compounds, an indication that they are derivatives of the chloroform molecule rather than the carbontetrachloride molecule.

Figure 16-2 shows the molecular structure of Refrigerants-113 and 114, the only two fluorocarbons of the ethane series in common use. The presence of the two carbon atoms identifies the basic molecule as ethane, rather than methane, which has only one carbon atom.

The individual characteristics of these and other refrigerants are discussed in the following sections.

16-8. The Effect of Moisture. It is a well-established fact that moisture will combine in varying degrees with most of the commonly used refrigerants, causing the formation of highly corrosive compounds (usually acids) which will react with the lubricating oil and with other materials in the system, including metals. This chemical action often results in pitting and other damage to valves, seals, bearing journals, cylinder walls, and other polished surfaces. It may also cause deterioration of the lubricating oil and the formation of metallic and other sludges which tend to clog valves and oil passages, score bearing surfaces, and otherwise reduce the life of the equipment. Moisture corrosion also contributes to compressor valve failure and, in hermetic motor-compressors, often causes breakdown of the motor winding insulation, which results in shorting or grounding of the motor.

Although a completely moisture-free refrigerating system is not possible, good refrigerating practice demands that the moisture content of the system be maintained below the level which will produce harmful effects in the system. The minimum moisture level which will produce harmful effects in a refrigerating system is not clearly defined and will vary considerably, depending upon the nature of the refrigerant, the quality of the lubricating oil, and the operating temperatures of the system, particularly the compressor discharge temperature.

Moisture in a refrigerating system may exist as "free water" or it may be in solution with the refrigerant. When moisture is present in the system in the form of free water, it will freeze into ice in the refrigerant control and/or in the evaporator, provided that the temperature of the evaporator is maintained below the freezing point of the water. Naturally, the formation of ice in the refrigerant control orifice will prevent the flow of liquid refrigerant through that part and render the system inoperative until such time that the ice melts and flow through the control is restored. In such cases, refrigeration is usually intermittent as the flow of liquid is started and stopped by alternate melting and freezing of the ice in the control orifice.

Since free water exists in the system only when the amount of moisture in the system exceeds the amount that the refrigerant can hold in solution, freeze-ups are nearly always an indication that the moisture content of the system is above the minimum level that will produce corrosion. On the other hand, the mere absence of freeze-ups cannot be taken to mean that the moisture content of the system is necessarily below the level which will cause corrosion, since corrosion can occur with some refrigerants at levels well below those which will result in free water. Too, it must be recognized that freeze-ups do not occur in air conditioning systems or in any other system where the evaporator temperature is above the freezing point of water. For this reason, high temperature systems are often more subject to moisture corrosion than are systems operating at lower evaporator temperatures, since relatively large quantities of moisture can go unnoticed in such systems for relatively long periods of time.

Since the ability of an individual refrigerant to hold moisture in solution decreases as the temperature decreases, it follows that the moisture content in low temperature systems must be maintained at a very low level in order to avoid freeze-ups. Hence, moisture corrosion in low temperature systems is usually at a minimum.

The various refrigerants differ greatly both as to the amount of moisture they will hold in solution and as to the effect that the moisture has upon them. For example, the straight hydrocarbons, such as propane, butane, ethane, etc., absorb little if any moisture. Therefore, any moisture contained in such systems will be in the form of free water and will make its presence known by freezing out in the refrigerant control. Since this moisture must be removed immediately in order to keep the system operative, moisture corrosion will not usually be a problem when these refrigerants are used.

Ammonia and sulfur dioxide, on the other hand, have an affinity for water and therefore are capable of absorbing moisture in such large quantities that free water is seldom found in systems employing these two refrigerants. However, the effects produced by the combination of the water and the refrigerant are entirely different for the two refrigerants.

In ammonia systems, the combination of water and ammonia produces aqua ammonia, a strong alkali, which attracts nonferrous metals, such as copper and brass, but has little if any effect on iron or steel or any other materials in the system. For this reason, ammonia systems can be operated successfully even when relatively large amounts of moisture are present in the system.

In the case of sulfur dioxide, the moisture and sulfur dioxide combine to form sulfurous acid (H_2SO_3), which is highly corrosive. In view of the high solubility of water in SO_2, the amount of acid formed can be quite large. Hence, corrosion in sulfur dioxide systems can be very heavy.

The halocarbon refrigerants hydrolyze only slightly and therefore form only small amounts of acids or other corrosive compounds. As a general rule, corrosion will not occur in systems employing halocarbon refrigerants when the moisture content is maintained below the level which will cause freeze-ups, provided that high quality lubricating oils are used and that discharge temperatures are reasonably low.

16-9. Refrigerant-Oil Relationship. With a few exceptions, the oil required for lubrication of the compressor is contained in the crankcase of the compressor where it is subject to contact with the refrigerant. Hence, as already stated, the refrigerant must be chemically and physically stable in the presence of oil, so that neither the refrigerant nor the oil is adversely affected by the relationship.

Although some refrigerants, particularly sulfur dioxide and the halocarbons, react with the lubricating oil to some extent, under normal operating conditions the reaction is usually slight and therefore of little consequence, provided that a high quality lubricating oil is used and that the system is relatively clean and dry. However, when contaminants, such as air and moisture, are present in the system in any appreciable amount, chemical reactions involving the contaminants, the refrigerant, and the lubricating oil often occur which can result in decomposition of the oil, the formation of corrosive acids and sludges, copper plating, and/or serious corrosion of polished metal surfaces. High discharge temperatures greatly accelerate these processes, particularly oil decomposition, and often result in the formation of carbonaceous deposits on discharge valves and pistons and in the compressor head and discharge line. This condition is aggravated by the use of poorly refined lubricating oils containing a high percentage of unsaturated hydrocarbons, the latter being very unstable chemically.

Because of the naturally high discharge temperature of Refrigerant-22 (see Table 16-8), breakdown of the lubricating oil, accompanied by motor burnouts, is a common problem with hermetic motor-compressor units employing this refrigerant, particularly when used in conjunction with air-cooled condensers and long suction lines.

Copper plating of various compressor parts is often found in systems employing halocarbon refrigerants. The parts usually affected are the highly polished metal surfaces which generate heat, such as seals, pistons, cylinder walls, bearing surfaces, and valves. The exact cause of copper plating has not been definitely determined, but considerable evidence does exist that moisture and poor quality lubricating oils are contributing factors.

Because copper is never used with ammonia, copper plating is not found in ammonia systems. However, neither is it found in sulfur dioxide systems, although copper has been employed extensively with this refrigerant.

In any event, regardless of the nature of and/or the cause of unfavorable reactions between the refrigerant and the lubricating oil, these disadvantages can be greatly minimized or eliminated by the use of high quality lubricating oils, having low "pour" and/or "floc" points (see Section 18-16), by maintaining the system relatively free of contaminants, such as air and moisture, and by designing the system so that discharge temperatures are reasonably low.

16-10. Oil Miscibility. With regard to the refrigerant-oil relationship, one important characteristic which differs for the various refrigerants is oil miscibility, that is, the ability of the refrigerant to be dissolved into the oil and vice versa.

With reference to oil miscibility, refrigerants may be divided into three groups: (1) those which are miscible with oil in all proportions under conditions found in the refrigerating system, (2) those which are miscible under conditions normally found in the condensing section, but separate from the oil under the conditions normally found in the evaporator section, and (3) those which are not miscible with oil at all (or only very slightly so) under conditions found in the system.

As to whether or not oil miscibility is a desirable property in a refrigerant there is some disagreement. In any event, the fact of oil miscibility, or the lack of it, has little if any significance insofar as the selection of the refrigerant is concerned. However, since it greatly influences the design of the compressor and other system components, including the refrigerant piping, the degree of oil miscibility is an important refrigerant characteristic and therefore should be considered in some detail.

With regard to the oil, one of the principal effects of an oil miscible refrigerant is to dilute the oil in the crankcase of the compressor, thereby lowering the viscosity (thinning) of the oil and reducing its lubricating qualities. To compensate for refrigerant dilution, the compressor lubricating oil used in conjunction with oil-miscible refrigerants should have a higher initial viscosity than that used for similar duty with nonmiscible refrigerants.

Viscosity may be defined as a measure of fluid friction or as a measure of the resistance that a fluid offers to flow. Hence, thin, low viscosity fluids will flow more readily than thicker, more viscous fluids. To provide adequate lubrication for the compressor, the viscosity of the lubricating oil must be maintained within certain limits. If the viscosity of the oil is too low, the oil will not have sufficient body to form a protective film between the various rubbing surfaces and keep them separated. On the other hand, if the viscosity of the oil is too high, the oil will not have sufficient fluidity to penetrate between the rubbing surfaces, particularly where tolerances are close. In either case, lubrication of the compressor will not be adequate.

Any oil circulating through the system with the refrigerant will have an adverse affect on the efficiency and capacity of the system, the principal reason being that the oil tends to adhere to and to form a film on the surface of the condenser and evaporator tubes, thereby lowering the heat transfer capacity of these two units. Since the oil becomes more viscous and tends to congeal as the temperature is reduced, the problem with oil is greatest in the evaporator and becomes more acute as the temperature of the evaporator is lowered.

Since the only reason for the presence of oil in the refrigerating system is to lubricate the compressor, it is evident that the oil will best serve its function when confined to the compressor and not allowed to circulate with the refrigerant through other parts of the system. However, since, with few exceptions, the system refrigerant unavoidably comes into contact with the oil in the compressor, a certain amount of oil in the form of small particles will be entrained in the refrigerant vapor and carried over through the discharge valves into the discharge line. If the oil is not removed from the vapor at this point, it will pass into the condenser and liquid receiver from where it will be carried to the evaporator by the liquid refrigerant. Obviously, in the interest of system efficiency and in order to maintain the oil in the crankcase at a constant level, some provision must be made for removing this oil from the system and returning it to the crankcase where it can perform its lubricating function.

The degree of difficulty experienced in bringing about the return of oil to the crankcase depends primarily on three factors: (1) the oil miscibility of the refrigerant, (2) the type of evaporator used, and (3) the evaporator temperature.

When an oil-miscible refrigerant is employed, the problem of oil return is greatly simplified by the fact that the oil remains in solution with the refrigerant. This permits the oil to be carried along through the system by the refrigerant and, subsequently, to be returned to the crankcase through the suction line, provided that the evaporator and the refrigerant piping are properly designed.

Unfortunately, when nonmiscible refrigerants are used, once the oil passes into the condenser, the return of the oil to the crankcase is not so easily accomplished. The reason for this is that, except for a small amount of mechanical mixing, the refrigerant and the oil will remain separate, so that only a small portion of the oil is actually carried along with the refrigerant. For example, in the case of ammonia, which is lighter than oil, a large percentage of the oil will separate from the liquid ammonia and settle out at various low points in the system. For this reason, oil drains should be provided at the bottom of all receivers, evaporators, accumulators, and other vessels containing liquid ammonia, and provisions should be made for draining the oil from these points, either continuously or periodically, and returning it to the crankcase. This may be accomplished manually or automatically.

When flooded-type evaporators are used, the refrigerant velocity will not usually be sufficient to permit the refrigerant vapor to entrain the oil and carry it over into the suction line and back to the crankcase. Hence, even with oil miscible refrigerants, where flooded-type evaporators are employed, it is often necessary to make special provisions for oil return. The methods used to insure the continuous return of the oil from the evaporator to the crankcase in such cases is described in Chapter 19.

Since the oil acts to lubricate the refrigerant flow control and other valves which may be in the system, the circulation of a small amount of oil with the refrigerant is not ordinarily objectionable. However, because of the adverse effect on system capacity, the amount of oil should be kept to a practical minimum. Too, since the oil in circulation comes initially from the compressor crankcase, an excessive amount in circulation may cause the oil level in the crankcase to fall below the minimum level required for adequate lubrication of the compressor parts.

In order to minimize the circulation of oil, an oil separator or trap is sometimes installed in the discharge line between the compressor and the condenser (see Section 19-12).

As a general rule, discharge line oil separators should be employed in any system where oil return is likely to be inadequate and/or where the amount of oil in circulation is apt to be excessive or to cause an undue loss in system capacity and efficiency. Specifically, discharge line oil separators are recommended for all systems employing nonmiscible refrigerants (or refrigerants which are not oil miscible at the evaporator conditions), not only because of the difficulty experienced in returning the oil from the evaporator to the crankcase but also because the presence of even small amounts of oil in the evaporators of such systems will usually cause considerable loss of evaporator efficiency and capacity.

The same thing is usually true for systems employing miscible refrigerants when the evaporator temperature is below 0° F. Oil separators are recommended also for all systems using flooded evaporators, since oil return from this type of evaporator is apt to be inadequate because of low refrigerant velocities.

Although oil separators are very effective in removing oil from the refrigerant vapor, they are not 100% efficient. Therefore, even though an oil separator is used, some means must still be provided for returning to the crankcase the small amount of oil which will always pass through the separator and find its way into other parts of the system. Too, since oil separators can often cause serious problems in the system if they are not properly installed, the use of oil separators should ordinarily be limited to those systems where the nature of the refrigerant or the particular design of the system requires their use. Oil separators are discussed in more detail in Chapter 19.

16-11. Leak Detection. Leaks in a refrigerating system may be either inward or outward, depending on whether the pressure in the system at the point of leakage is above or below atmospheric pressure. When the pressure in the system is above atmospheric at the point of leakage, the refrigerant will leak from the system to the outside. On the other hand, when the pressure in the system is below atmospheric, there is no leakage of refrigerant to the outside, but air and moisture will be drawn into the system. In either case, the system will usually become inoperative in a very short time. However, as a general rule, outward leaks are less serious than inward ones, usually requiring only that the leak be found and repaired and that the system be recharged with the proper amount of refrigerant. In the case of inward leaks, the air and moisture drawn into the system increase the discharge pressure and temperature and accelerates the rate of corrosion. The presence of moisture in the system may also cause freeze-up of the refrigerant control. Furthermore, after the leak has been located and repaired, the system must be completely evacuated and dehydrated before it can be placed in operation. A refrigerant drier should also be installed in the system.

The necessity of maintaining the system free of leaks demands some convenient means for checking a new system for leaks and for detecting leaks if and when they occur in systems already in operation. New systems should be checked for leaks under both vacuum and pressure.

One method of leak detection universally used with all refrigerants employs a relatively viscous soap solution which is relatively free of bubbles. The soap solution is first applied to the pipe joint or other suspected area and then examined with the help of a strong light. The formation of bubbles in the soap solution indicates the presence of a leak. For adequate testing with a soap solution, the pressure in the system should be 50 psig or higher.

The fact that sulfur and ammonia vapors produce a dense white smoke (ammonia sulfite) when they come into contact with one another provides a convenient means of checking for leaks in both sulfur dioxide and ammonia systems. To check for leaks in a sulfur dioxide system, a cloth swab saturated with stronger ammonia (approximately 28% available in any drug store) is held near, but not in contact with, all pipe joints and other suspected areas. A leak is indicated when the ammonia swab gives off a white smoke.

Ammonia systems are checked in the same way except that a sulfur candle is substituted for the ammonia swab. Dampened phenophthalein paper, which turns red on contact with ammonia vapor, may also be used to detect ammonia leaks.

A halide torch is often used to detect leaks in systems employing any of the halocarbon refrigerants. The halide torch consists of a copper element which is heated by a flame. Air to support combustion is drawn in through a rubber tube, one end of which is attached to the torch. The free end of the tube is passed around all suspected areas. The presence of a halocarbon vapor is indicated when the flame changes from its normal color to a bright green or purple. The halide torch should be used only in well-ventilated spaces.

For carbon dioxide and the straight hydrocarbons, the only method of leak detection is the soap solution previously mentioned.

16-12. Ammonia. Ammonia is the only refrigerant outside of the fluorocarbon group that is being used to any great extent at the present time. Although ammonia is toxic and also somewhat flammable and explosive under certain conditions, its excellent thermal properties make it an ideal refrigerant for ice plants, packing plants, skating rinks, large cold storage facilities, etc., where experienced operating personnel are usually on duty and where its toxic nature is of little consequence.

Ammonia has the highest refrigerating effect per pound of any refrigerant. This, together with a moderately low specific volume in the vapor state, makes possible a high refrigerating capacity with a relatively small piston displacement.

The boiling point of ammonia at standard atmospheric pressure is −28° F. The evaporator and condenser pressures at standard ton conditions of 5° F and 86° F are 19.6 psig and 154.5 psig, respectively, which are moderate, so that lightweight materials can be used in the construction of the refrigerating equipment. However, the adiabatic discharge temperature is relatively high, being 210° F at standard ton conditions, which makes water cooling of the compressor head and cylinders desirable. Too, high suction superheats should be avoided in ammonia systems.

Although pure anhydrous ammonia is noncorrosive to all metals normally used in refrigerating systems, in the presence of moisture, ammonia becomes corrosive to nonferrous metals, such as copper and brass. Obviously, these metals should never be used in ammonia systems.

Ammonia is not oil miscible and therefore will not dilute the oil in the compressor crankcase. However, provisions must be made for the removal of oil from the evaporator and an oil separator should be used in the discharge line of all ammonia systems.

Ammonia systems may be tested for leaks with sulfur candles, which give off a dense white smoke in the presence of ammonia vapor, or by applying a thick soap solution around the pipe joints, in which case a leak is indicated by the appearance of bubbles in the solution.

16-13. Sulfur Dioxide. Sulfur dioxide (SO_2) is produced from the combustion of sulfur. It is highly toxic, but nonflammable and nonexplosive. In the 1920s and 1930s, sulfur dioxide was widely used in domestic refrigerators and in small commercial fixtures. Today, it is found only in a few of the older commercial units, having been replaced first by methyl chloride and later by the more desirable fluorocarbon refrigerants.

The boiling point of sulfur dioxide at atmospheric pressure is approximately 14° F. Saturation pressures at standard ton conditions of 5° F and 86° F are 5.9 in. Hg and 51.8 psig, respectively.

Sulfur dioxide is not oil miscible. However, unlike ammonia and carbon dioxide, liquid sulfur dioxide is heavier than oil so that the oil floats on top of the refrigerant. Since this characteristic simplifies the problem of oil return, it accounts for the popularity enjoyed by sulfur dioxide in the past for small automatic equipment.

Like most common refrigerants, sulfur dioxide in the pure state is noncorrosive to metals normally used in the refrigerating system. However, it combines with moisture to form sulfurous acid (H_2SO_3) and sulfuric acid (H_2SO_4), both of which are highly corrosive.

16-14. Carbon Dioxide. Carbon dioxide (CO_2) is one of the first refrigerants used in mechanical refrigerating systems. It is odorless, nontoxic, nonflammable, nonexplosive, and noncorrosive. Because of its safe properties, it has been widely used in the past for marine service and for air conditioning in hospitals, theaters, hotels, and in other places where safety is the prime consideration. Although a few of these older installations are still in service, at the present time the use of carbon dioxide as a refrigerant is limited for the most part to extremely low temperature applications, particularly in the production of solid CO_2 (dry ice).

One of the chief disadvantages of carbon dioxide is its high operating pressures, which under standard ton conditions of 5° F and 86° F are 317.5 psig and 1031 psig, respectively. Naturally, this requires the use of extra heavy piping and equipment. However, because of the high vapor density of CO_2, the volume of vapor handled by the compressor is only 0.96 cu ft per minute per ton at 5° F, so that compressor sizes are small.

Another disadvantage of carbon dioxide is that the horsepower required per ton is approximately twice that of any of the commonly used refrigerants. For carbon dioxide, the theoretical horsepower required per ton at standard conditions is 1.84, whereas for ammonia, the horsepower required per ton is only 0.989, the latter value being typical for most refrigerants.

Since its boiling temperature at atmospheric pressure (−109.3° F) is below its freezing temperature (−69.9° F) at this pressure, carbon dioxide cannot exist in the liquid state at atmospheric pressure nor at any pressure below its triple point pressure of 75.1 psia. At any pressure under 75.1 psia, solid carbon dioxide sublimes directly into the vapor state and therefore below this pressure is found only in the solid and vapor states. Because of the low critical temperature of CO_2 (87.8° F), relatively low condensing temperatures are required for liquefaction. Carbon dioxide is nonmiscible in oil and therefore will not dilute the oil in the crankcase of the compressor. Like ammonia, it is lighter than oil. Hence, oil return problems are similar to those encountered in an ammonia system.

Leak detection is by soap solution only.

16-15. Methyl Chloride. Methyl chloride (CH_3Cl) is a halocarbon of the methane series. It has many of the properties desirable in a refrigerant, which accounts for its wide use in the past in both domestic and commercial applications. Its boiling point at atmospheric pressure is −10.65° F. Evaporator and condenser pressures at standard ton conditions are 6.5 psig and 80 psig, respectively.

Although methyl chloride is considered nontoxic, in large concentrations it has an anesthetic effect similar to that of chloroform, a compound to which it is closely related. Methyl chloride is moderately flammable and is explosive when mixed with air in concentrations between 8.1 and 17.2% by volume. The hazard resulting from these properties is the principal reason for the discarding of methyl chloride in favor of the safer fluorocarbon refrigerants.

Methyl chloride is corrosive to aluminum, zinc, and magnesium, and the compounds formed in combination with these materials are both flammable and explosive. Hence, these metals should not be used in methyl chloride systems. In the presence of moisture, methyl chloride forms a weak hydrochloric acid, which is corrosive to both ferrous and nonferrous metals. Too, since natural rubber and the synthetic, Neoprene, are dissolved by methyl chloride, neither is suitable gasket material for use in methyl chloride systems.

Oil return in methyl chloride systems is simplified by the fact that methyl chloride is oil miscible. However, in selecting the compressor lubricating oil, crankcase dilution must be taken into account.

Leaks in a methyl chloride system are found with the aid of a soap solution which is applied to the suspected joints. The presence of methyl chloride vapor may be detected with a halide leak detector. However, this method is not recommended because of the flammability of methyl chloride.

16-16. Methylene Chloride (Carrene I). Methylene chloride (CH_2Cl_2), another halocarbon of the methane series, has a boiling point of 103.5° F at atmospheric pressure, a characteristic which permits the refrigerant to be stored in sealed cans rather than in compressed gas cylinders. Under standard ton conditions, the evaporator and condenser pressures are both below atmospheric pressure, being 27.6 in. Hg and 9.5 in. Hg, respectively. Since the volume of the vapor handled per ton of refrigerating capacity is quite large (74.3 cu ft/min/ton at

5° F), centrifugal compressors, which are particularly suited to handling large volumes of low pressure vapor, are required.

Although it dissolves natural rubber, methylene chloride is noncorrosive even in the presence of moisture. It is also nontoxic and nonflammable. Because of its safe properties, it has been widely used in large air conditioning installations.

The fact that methylene chloride is oil miscible is of little consequence, since in centrifugal compressors the oil and refrigerant do not ordinarily come in contact with one another.

A halide torch or soap solution may be used to detect leaks. However, the pressure in the system must be built up above atmospheric in either case.

16-17. Refrigerant-11. Refrigerant-11 (CCl_3F) is a fluorocarbon of the methane series and has a boiling point at atmospheric pressure of 74.7° F. Operating pressures at standard ton conditions are 24 in. Hg and 3.6 psig, respectively, which is very similar to those of methylene chloride. Although the theoretical horsepower required at standard ton conditions (0.927) is approximately the same as that for methylene chloride, the compressor displacement required at these conditions (36.32 cu ft/min/ton) is only approximately one-half that required for methylene chloride.

Like other fluorocarbon refrigerants, Refrigerant-11 dissolves natural rubber. However, it is noncorrosive, nontoxic, and nonflammable. The low operating pressures and the relatively high compressor displacement required necessitate the use of a centrifugal compressor.

Refrigerant-11 is used mainly in the air conditioning of small office buildings, factories, department stores, theaters, etc. A halide torch may be used for leak detection.

16-18. Refrigerant-12. Although its supremacy is being seriously challenged in some areas by Refrigerant-22, Refrigerant-12 (CCl_2F_2) is by far the most widely used refrigerant at the present time. It is a completely safe refrigerant in that it is nontoxic, nonflammable, and nonexplosive. Furthermore, it is a highly stable compound which is difficult to break down even under extreme operating conditions. However, if brought into contact with an open flame or with an electrical heating element, Refrigerant-12 will decompose into products which are highly toxic (see Section 16-3).

Along with its safe properties, the fact that Refrigerant-12 condenses at moderate pressures under normal atmospheric conditions and has a boiling temperature of −21° F at atmospheric pressure makes it a suitable refrigerant for use in high, medium, and low temperature applications and with all three types of compressors. When employed in conjunction with multistage centrifugal type compressors, Refrigerant-12 has been used to cool brine to temperatures as low as −110° F.

The fact that Refrigerant-12 is oil miscible under all operating conditions not only simplifies the problem of oil return but also tends to increase the efficiency and capacity of the system in that the solvant action of the refrigerant maintains the evaporator and condenser tubes relatively free of oil films which otherwise would tend to reduce the heat transfer capacity of these two units.

Although the refrigerating effect per pound for Refrigerant-12 is relatively small as compared to that of some of the other popular refrigerants, this is not necessarily a serious disadvantage. In fact, in small systems, the greater weight of Refrigerant-12 which must be circulated is a decided advantage in that it permits closer control of the liquid. In larger systems, the disadvantage of the low latent heat value is offset somewhat by a high vapor density, so that the compressor displacement required per ton of refrigeration is not much greater than that required for Refrigerants-22, 500, and 717. The horsepower required per ton of capacity compares favorably with that required for other commonly used refrigerants.

A halide torch is used for leak detection.

16-19. Refrigerant-13. Refrigerant-13 ($CClF_3$) was developed for and is being used in ultra-low temperature applications, usually in the low stage of a two or three stage cascade system. It is also being used to replace Refrigerant-22 in some low temperature applications.

The boiling temperature of Refrigerant-13 is −144.5° F at atmospheric pressure. Evaporator temperatures down to −150° F are practical. The critical temperature is 83.9° F. Since condensing pressures and the compressor displacement required are both moderate, Refrigerant-13 is suitable for use with all three types of compressors.

Refrigerant-13 is a safe refrigerant. It is not miscible with oil. A halide torch may be used for leak detection.

16-20. Refrigerant-22. Refrigerant-22 ($CHClF_2$) has a boiling point at atmospheric pressure of −41.4° F. Developed primarily as a low temperature refrigerant, it is used extensively in domestic and farm freezers and in commercial and industrial low temperature systems down to evaporator temperatures as low as −125° F. It also finds wide use in packaged air conditioners, where, because of space limitations, the relatively small compressor displacement required is a decided advantage.

Both the operating pressures and the adiabatic discharge temperature are higher for Refrigerant-22 than for Refrigerant-12. Horsepower requirements are approximately the same.

Because of the high discharge temperatures experienced with Refrigerant-22, suction superheat should be kept to a minimum, particularly where hermetic motor-compressors are employed. In low temperature applications, where compression ratios are likely to be high, water cooling of the compressor head and cylinders is recommended in order to avoid overheating of the compressor. Air-cooled condensers used with Refrigerant-22 should be generously sized.

Although miscible with oil at temperatures found in the condensing section, Refrigerant-22 will often separate from the oil in the evaporator. The exact temperature at which separation occurs varies considerably with the type of oil and the amount of oil mixed with the refrigerant. However, no difficulty is usually experienced with oil return from the evaporator when a properly designed serpentine evaporator is used and when the suction piping is properly designed. When flooded evaporators are employed, oil separators should be used and special provisions should be made to insure the return of oil from the evaporator. Oil separators should always be used on low temperature applications.

The principal advantage of Refrigerant-22 over Refrigerant-12 is the smaller compressor displacement required, being approximately 60% of that required for Refrigerant-12. Hence, for a given compressor displacement, the refrigerating capacity is approximately 60% greater with Refrigerant-22 than with Refrigerant-12. Too, refrigerant pipe sizes are usually smaller for Refrigerant-22 than for Refrigerant-12. For evaporator temperatures between −20 and −40° F, still another advantage added to Refrigerant-22 is that the evaporator pressures for Refrigerant-22 at these temperatures are above atmospheric, whereas for Refrigerant-12 the evaporator pressures will be below atmospheric. However, all this should not be taken to mean that Refrigerant-22 is superior to Refrigerant-12 in all applications. As a matter of fact, except in those applications where space limitations necessitate the use of the smallest possible equipment and/or where the evaporator temperature is between −20° F and −40° F, Refrigerant-12, because of its lower discharge temperatures and greater miscibility with oil, is probably the more desirable of the two refrigerants.

The ability of Refrigerant-22 to absorb moisture is considerably greater than that of Refrigerant-12 and therefore less trouble is experienced with freeze-ups in Refrigerant-22 systems. Although some consider this to be an advantage, the advantage gained is questionable, since any amount of moisture in a refrigerating system is undesirable.

Being a fluorocarbon, Refrigerant-22 is a safe refrigerant. A halide torch may be used for leak detection.

16-21. Refrigerant-113. Refrigerant-113 (CCl_2FCClF_2) boils at 117.6° F under atmospheric pressure. Operating pressures at standard ton conditions are 27.9 in. Hg and 13.9 in. Hg, respectively. Although the compressor displacement per ton is somewhat high (100.76 cu ft/min/ton at standard ton conditions), the horsepower required per ton compares favorably with other common refrigerants. The low operating pressures and the large displacement required necessitate the use of a centrifugal type compressor.

Although used mainly in comfort air conditioning applications, it is also employed in industrial process water and brine chilling down to 0° F.

Refrigerant-113 is a safe refrigerant. A halide torch may be used for leak detection.

16-22. Refrigerant-114. Refrigerant-114 (CCl_2CClF_2) has a boiling point of 38.4° F under atmospheric pressure. Evaporating and condensing pressures at standard ton conditions are 16.1 in. Hg and 22 psig, respectively. The compressor displacement required is relatively low for a low pressure refrigerant (19.59 cu ft/min/ton at standard conditions) and the horsepower required compares favorably with that required by other common refrigerants.

Refrigerant-114 is used with centrifugal compressors in large commercial and industrial air conditioning installations and for industrial process water chilling down to −70° F. It is also used with vane-type rotary compressors in domestic refrigerators and in small drinking water coolers.

Like Refrigerant-22, Refrigerant-114 is oil miscible under conditions found in the condensing section, but separates from oil in the evaporator. However, because of the type of equipment used with Refrigerant-114 and the conditions under which it is used, oil return is not usually a problem.

Refrigerant-114 is a safe refrigerant. A halide torch may be used for leak detection.

16-23. Straight Hydrocarbons. The straight hydrocarbons are a group of fluids composed in various proportions of the two elements hydrogen and carbon. Those having significance as refrigerants are methane, ethane, butane, propane, ethylene, and isobutane. All are extremely flammable and explosive. Too, since all act as anesthetics in varying degrees, they are considered mildly toxic. Although none of these compounds will absorb moisture to any appreciable extent, all are extremely miscible with oil under all conditions.

Although a few of the straight hydrocarbons (butane, propane, and isobutane) have been used in small quantities for domestic refrigeration, their use is ordinarily limited to special applications where an experienced attendent is on duty. Ethane, methane, and ethylene are employed to some extent in ultra-low temperature applications, usually in the lower stage of two and three stage cascade systems. However, even in these applications, it is likely that they will be replaced in the future by Refrigerants-13 and 14, the latter being used only in pilot plants at the present time.

Leak detection is by soap solution only.

16-24. Refrigerant-500. Refrigerant-500, commonly known as Carrene 7*, is an azeotropic mixture† of Refrigerant-12 (73.8% by weight) and Refrigerant-152a (26.2%). It has a boiling point at atmospheric pressure of −28° F. Evaporator and condenser pressures at standard ton conditions are 16.4 psig and 113.4 psig, respectively. Although the horsepower requirements of Refrigerant-500 are approximately the same as those for Refrigerants-12 and 22, the compressor displacement required is greater than that required for Refrigerant-22, but somewhat less than that required for Refrigerant-12.

The principal advantage of Refrigerant-500 lies in the fact that its substitution for Refrigerant-12 results in an increase in compressor capacity of approximately 18%. This makes it possible to use the same direct connected compressor (as in a hermetic motor-compressor unit) on either 50 or 60 cycle power with little or no change in the refrigerating capacity or in the power requirements.

It will be shown in Chapter 21 that the speed of an alternating current motor varies in direct proportion to the cycle frequency. Therefore, an electric motor operating on 50 cycle power will have only five-sixths of the speed it has when operating on 60 cycle power. For this reason, the displacement of a direct connected compressor is reduced approximately 18% when a change is made from 60 to 50 cycle power. Since the increase in capacity per unit of displacement accruing from the substitution of Refrigerant-500 for Refrigerant-12 is almost exactly equal to the loss of displacement suffered when changing for 60 to 50 cycle power, the same motor-compressor assembly is made suitable for use with both frequencies by the simple expedient of changing refrigerants.

16-25. Refrigerant Drying Agents. Refrigerant drying agents, called desiccants are frequently employed in refrigerating systems to remove moisture from the refrigerant. Some of the most commonly used desiccants are silica gel (silicon dioxide), activated alumina (aluminum oxide), and Drierite (anhydrous calcium sulfate). Silica gel and activated alumina are adsorption-type desiccants and are available in granular form. Drierite is an absorption type desiccant and is available in granular form and in cast sticks.

* A proprietary refrigerant of the Carrier Corporation.

† An azeotropic mixture is a mixture of two or more liquids, which, when mixed in precise proportions, form a compound having a boiling temperature which is independent of the boiling temperatures of the individual liquids.

17

Refrigerant Flow Controls

17-1. Types and Function. There are six basic types of refrigerant flow controls: (1) the hand expansion valve, (2) the automatic expansion valve, (3) the thermostatic expansion valve, (4) the capillary tube, (5) the low pressure float, and (6) the high pressure float.

Regardless of type, the function of any refrigerant flow control is twofold: (1) to meter the liquid refrigerant from the liquid line into the evaporator at a rate commensurate with the rate at which vaporization of the liquid is occurring in the latter unit, and (2) to maintain a pressure differential between the high and low pressure sides of the system in order to permit the refrigerant to vaporize under the desired low pressure in the evaporator while at the same time condensing at a high pressure in the condenser.

17-2. Hand Expansion Valves. Hand expansion valves are hand-operated needle valves (Fig. 17-1). The rate of liquid flow through the valve depends on the pressure differential across the valve orifice and on the degree of valve opening, the latter being manually adjustable. Assuming that the pressure differential across the valve remains the same, the flow rate through a hand expansion valve will remain constant at all times without regard for either the evaporator pressure or the evaporator loading.

The principal disadvantage of the hand expansion valve is that it is unresponsive to changes in the system load and therefore must be manually readjusted each time the load on the system changes in order to prevent either starving or overfeeding of the evaporator, depending upon the direction of the load shift. Too, the valve must be opened and closed manually each time the compressor is cycled on and off.

Obviously the hand expansion valve is suitable for use only on large systems where an operator is on duty and where the load on the system is relatively constant. When automatic control is desired and/or when the system is subject to frequent load fluctuations, some other type of refrigerant flow control is required.

At the present time, the principal use of the hand expansion valve is as an auxiliary refrigerant control installed in a by-pass line (Fig. 17-29). It is also frequently used to control the flow rate through oil bleeder lines (Fig. 19-12).

17-3. Automatic Expansion Valves. A schematic diagram of an automatic expansion valve is shown in Fig. 17-2. The valve consists mainly of a needle and seat, a pressure bellows or diaphragm, and a spring, the tension of the latter being variable by means of an adjusting screw. A screen or strainer is usually installed at the liquid inlet of the valve in order to prevent the entrance of foreign materials which may cause stoppage of the valve. The construction of a typical automatic expansion valve is shown in Fig. 17-3.

The automatic expansion valve functions to maintain a constant pressure in the evaporator by flooding more or less of the evaporator surface in response to changes in the evaporator load. The constant pressure characteristic of the valve results from the interaction of two opposing forces: (1) the evaporator pressure and (2) the spring pressure. The evaporator pressure, exerted on one side of the bellows or diaphragm, acts to move the valve in a closing direction, whereas the spring pressure, acting on the opposite side of the bellows or diaphragm, acts to move the valve in an opening direction. When the compressor is running, the valve functions to maintain the evaporator pressure in equilibrium with the spring pressure.

As the name implies, the operation of the valve is automatic and, once the tension of the spring is adjusted for the desired evaporator pressure, the valve will operate automatically to regulate the flow of liquid refrigerant into the evaporator so that the desired evaporator pressure is maintained, regardless of evaporator loading. For example, assume that the tension of the spring is adjusted to maintain a constant pressure in the evaporator of 10 psig. Thereafter, any time the evaporator pressure tends to fall below 10 psig, the spring pressure will exceed the evaporator pressure causing the valve to move in the opening direction, thereby increasing the flow of liquid to the evaporator and flooding more of the evaporator surface. As more of the evaporator surface becomes effective, the rate of vaporization increases and

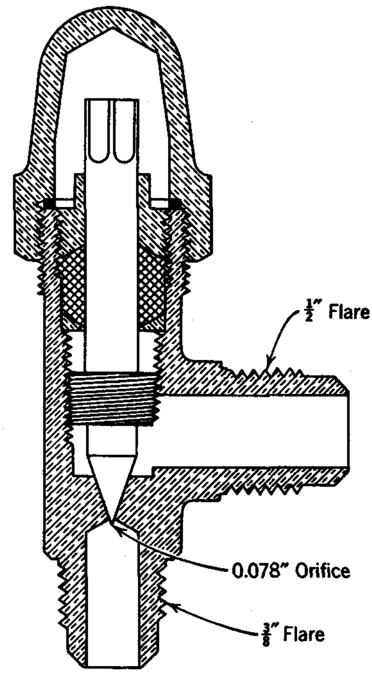


Fig. 17-1. Small capacity hand-expansion valve. (Courtesy Mueller Brass Company.)

the evaporator pressure rises until equilibrium is established with the spring pressure. Should the evaporator pressure tend to rise above the desired 10 psig, it will immediately override the pressure of the spring and cause the valve to move in the closing direction, thereby throttling the flow of liquid into the evaporator and reducing the amount of effective evaporator surface. Naturally, this decreases the rate of vaporization and lowers the evaporator pressure until equilibrium is again established with the spring pressure.

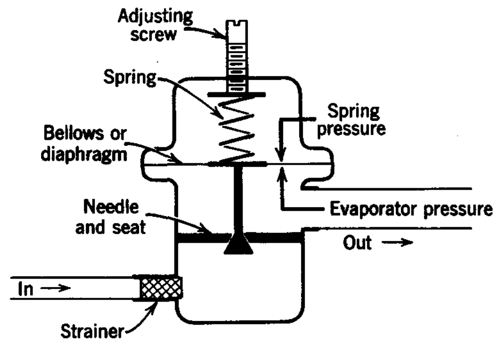


Fig. 17-2. Schematic diagram of automatic expansion valve.

It is important to notice that the operating characteristics of the automatic expansion valve are such that the valve will close off tightly when the compressor cycles off and remain closed until the compressor cycles on again. As previously described, vaporization continues in the evaporator for a short time after the compressor cycles off and, since the resulting vapor is not removed by the compressor, the pressure in the evaporator rises. Hence, during the off cycle, the evaporator pressure will always exceed the spring pressure and the valve will be tightly closed. When the compressor cycles on, the evaporator pressure will be immediately reduced below the spring pressure, at which time the valve will open and admit sufficient liquid to the evaporator to establish operating equilibrium between the evaporator and spring pressures.

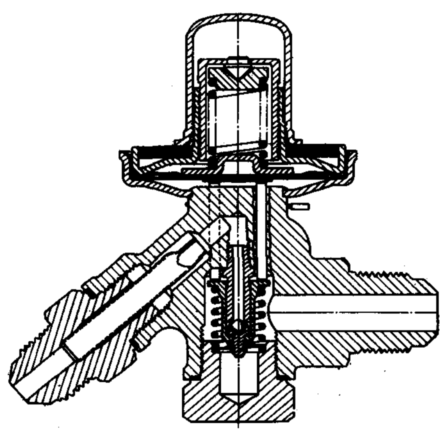


Fig. 17-3. Typical automatic expansion valve. (Courtesy Controls Company of America.)

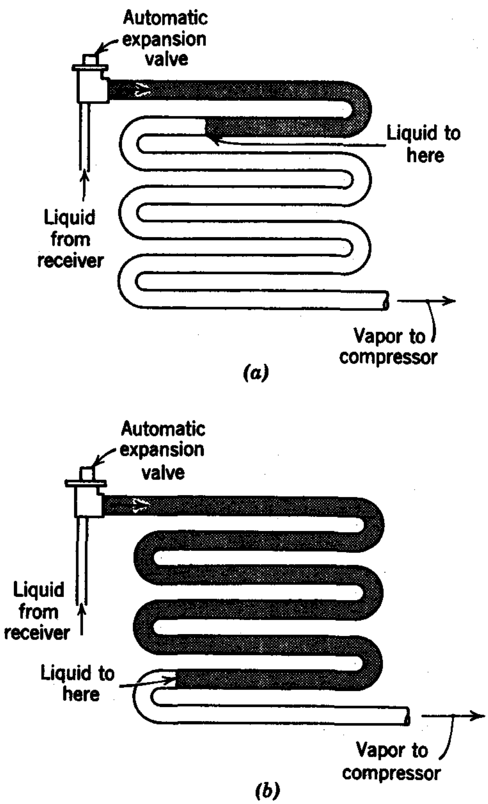


Fig. 17-4. Operating characteristics of the automatic expansion valve under varying load conditions. (*a*) Heavy load conditions. (*b*) Minimum load conditions.

The chief disadvantage of the automatic expansion valve is its relatively poor efficiency as compared to that of other refrigerant flow controls. In view of the evaporator-compressor relationship, it is evident that maintaining a constant pressure in the evaporator requires that the rate of vaporization in the evaporator be kept constant. To accomplish this necessitates severe throttling of the liquid in order to limit the amount of effective evaporator surface when the load on the evaporator is heavy and the heat transfer capacity per unit of evaporator surface is high (Fig. 17-4*a*). As the load on the evaporator decreases and the heat transfer capacity per unit of evaporator surface is reduced, more and more of the evaporator surface must be flooded with liquid if a constant rate of vaporization is to be maintained (Fig. 17-4*b*). As a matter of fact, if the load on the evaporator is permitted to fall below a certain level, the automatic expansion valve, in an attempt to keep the evaporator pressure up, will overfeed the evaporator to the extent that liquid will enter the suction line and be carried to the compressor where it may cause serious damage. However, in a properly designed system, overfeeding is not likely to occur, since the thermostat will usually cycle the compressor off before the space or product temperature is reduced to a level such that the load on the evaporator will fall below the critical point.

Obviously, since it permits only a small portion of the evaporator to be filled with liquid during periods when the load on the system is heavy, the constant pressure characteristic of the automatic expansion valve severely limits the capacity and efficiency of the refrigerating system at a time when high capacity and high efficiency are most desired. Too, because the evaporator pressure is maintained constant throughout the entire running cycle of the compressor, the valve must be adjusted for a pressure corresponding to the lowest evaporator temperature required during the entire running cycle (see Fig. 17-5). This results in a considerable loss in compressor capacity and efficiency, since advantage cannot be taken of the higher suction temperatures which would ordinarily exist with a full-flooded evaporator during the early part of the running cycle.

Another disadvantage of the automatic expansion valve, which can also be attributed to its constant pressure characteristic, is that it cannot be used in conjunction with a low pressure motor control, since proper operation of the latter part depends on a rather substantial change in the evaporator pressure during the running cycle, a condition which obviously

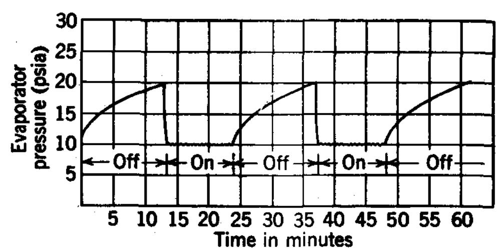


Fig. 17-5. Operating characteristics of the automatic expansion valve.

cannot be met when an automatic expansion valve is used as the refrigerant flow control.

In view of its poor efficiency under heavy load conditions, the automatic expansion valve is best applied only to small equipment having relatively constant loads, such as domestic refrigerators and freezers and small, retail ice-cream storage cabinets. However, even in these applications the automatic expansion valve is seldom used at the present time, having given way to other types of refrigerant flow controls which are more efficient and sometimes lower in cost.

Some automatic expansion valves are now being employed as "condenser by-pass valves." As such thay are installed in a by-pass line between the condenser and the suction line (Fig. 17-6) where they serve to regulate the flow of hot gas which is by-passed from the condenser directly into the suction line in order to prevent the evaporator pressure from dropping below a predetermined desired minimum. In such cases, the valve is set for the minimum desired evaporator pressure. As long as the pressure in the evaporator remains above the desired minimum, the valve will remain closed and no gas is by-passed from the condenser into the suction line. However, any time the evaporator tends to fall below the desired minimum, the by-pass valve opens and permits hot gas from the condenser to pass directly into the suction line in an amount just sufficient to maintain the minimum evaporator pressure. In this respect, the condenser by-pass serves the same function as the cylinder by-pass type of compressor capacity control.* However, unlike the cylinder by-pass, the condenser by-pass does not unload the compressor in any way. Hence, with the condenser by-pass, there is no reduction in the work of compression or in the power requirements of the compressor. For this reason, the condenser by-pass is not generally recommended as a means of controlling the capacity of the compressor.

Care should be taken to connect the by-pass line to the condenser at a point low enough on the condenser to insure that slightly "wet" vapor, rather than superheated vapor, is by-passed to the suction line. Superheated vapor directly from the discharge of the compressor, if by-passed to the suction line, will cause excessive discharge temperatures and result in overheating of the compressor and possible carbonization of the lubricating oil. On the other hand, wet vapor, that is, vapor containing small particles of liquid, will tend to reduce the operating temperature of the compressor through the cooling effect produced by vaporization of the

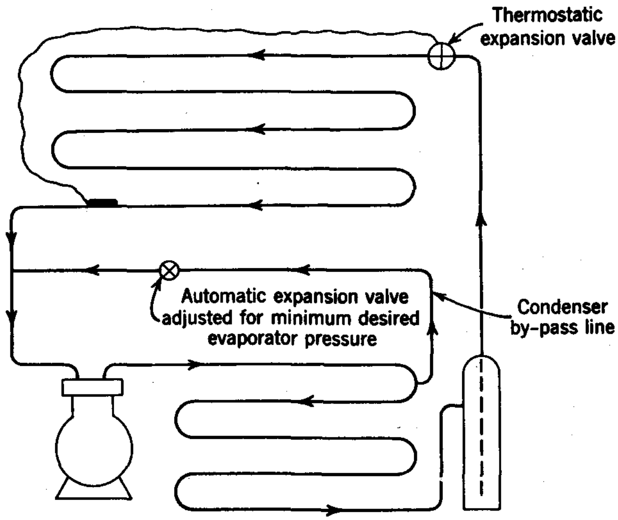


Fig. 17-6. Automatic expansion valve employed as condenser by-pass valve.

* The use of the condenser by-pass is common on some types of automobile air conditioning units. In such cases, the condenser by-pass serves to offset the high compressor capacity which accrues as a result of the increased piston displacement at high speeds.

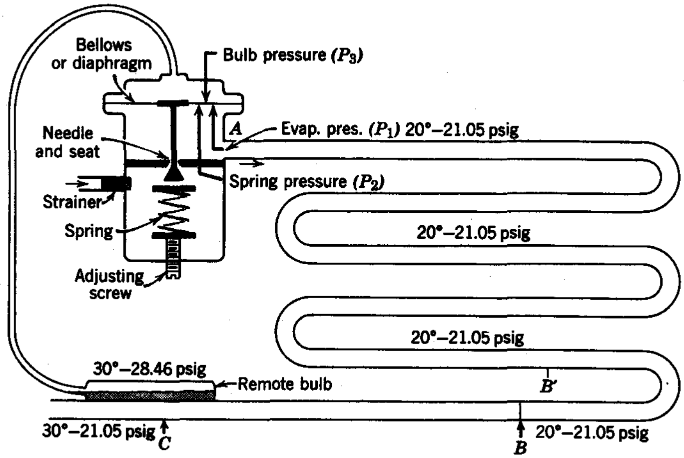


Fig. 17-7. Illustrating operating principle of conventional liquid-charged thermostatic expansion valve.

liquid particles in the compressor cylinder. Too, vaporization of the liquid particles in the compressor reduces the volumetric efficiency of the compressor, which, under the circumstances, is also beneficial since it will reduce the amount of vapor which must be by-passed from the condenser in order to maintain the minimum evaporator pressure.

17-4. Thermostatic Expansion Valves. Because of its high efficiency and its ready adaptability to any type of refrigeration application, the thermostatic expansion valve is the most widely used refrigerant control at the present time. Whereas the operation of the automatic expansion valve is based on maintaining a constant pressure in the evaporator, the operation of the thermostatic expansion valve is based on maintaining a constant degree of suction superheat at the evaporator outlet, a circumstance which permits the latter control to keep the evaporator completely filled with refrigerant under all conditions of system loading, without the danger of liquid slopover into the suction line. Because of its ability to provide full and effective use of all the evaporator surface under all load conditions, the thermostatic expansion valve is a particularly suitable refrigerant control for systems which are subject to wide and frequent variations in loading.

Figure 17-7 is a schematic diagram of a thermostatic expansion valve showing the principal parts of the valve, which are: (1) a needle and seat, (2) a pressure bellows or diaphragm, (3) a fluid-charged remote bulb which is open to one side of the bellows or diaphragm through a capillary tube, and (4) a spring, the tension of which is usually adjustable by an adjusting screw. As in the case of the automatic expansion valve and all other refrigerant controls, a screen or strainer is usually installed at the liquid inlet of the valve to prevent the entrance of foreign material which may cause stoppage of the valve.

The characteristic operation of the thermostatic expansion valve results from the interaction of three independent forces, viz: (1) the evaporator pressure, (2) the spring pressure, and (3) the pressure exerted by the saturated liquid-vapor mixture in the remote bulb.*

As shown in Fig. 17-7, the remote bulb of the expansion valve is clamped firmly to the suction line at the outlet of the evaporator, where it is responsive to changes in the temperature of the refrigerant vapor at this point. Although there is a slight temperature differential between the temperature of the refrigerant vapor in the suction line and the temperature of the saturated liquid-vapor mixture in the remote bulb, for all practical purposes the temperature of the two are the same and therefore it may be assumed that the pressure exerted by the fluid in the bulb is always the saturation pressure of the liquid-vapor mixture in the bulb corresponding to the temperature

* With some exceptions which are discussed later, the fluid in the remote bulb is the refrigerant used in the system. Hence, the remote bulb of a thermostatic expansion valve employed on a Refrigerant-12 system would ordinarily be charged with Refrigerant-12.

of the vapor in the suction line at the point of bulb contact.

Notice that the pressure of the fluid in the remote bulb acts on one side of the bellows or diaphragm through the capillary tube and tends to move the valve in the opening direction, whereas the evaporator pressure and the spring pressure act together on the other side of the bellows or diaphragm and tend to move the valve in a closing direction. The operating principles of the thermostatic expansion valve are best described through the use of an example.

With reference to Fig. 17-7, assume that Refrigerant-12 liquid is vaporizing in the evaporator at a temperature of 20° F so that the evaporator pressure (P_1) is 21.05 psig, the saturation pressure of Refrigerant-12 corresponding to a temperature of 20° F. Assume further that the tension of the spring is adjusted to exert a pressure (P_2) of 7.41 psi, so that the total pressure tending to move the valve in the closing direction is 28.46 psi, the sum of P_1 and P_2 (21.05 + 7.41). If the pressure drop in the evaporator is ignored, it can be assumed that the temperature and pressure of the refrigerant are the same throughout all parts of the evaporator where a liquid-vapor mixture of the refrigerant is present. However, at some point B near the evaporator outlet all the liquid will have vaporized from the mixture and the refrigerant at this point will be in the form of a saturated vapor at the vaporizing temperature and pressure. As the refrigerant vapor travels from point B through the remaining portion of the evaporator, it will continue to absorb heat from the surroundings, thereby becoming superheated so that its temperature is increased while its pressure remains constant. In this instance, assume that the refrigerant vapor is superheated 10° from 20 to 30° F during its travel from point B to the remote bulb location at point C. The saturated liquid-vapor mixture in the remote bulb, being at the same temperature as the superheated vapor in the line, will then have a pressure (P_3) of 28.46 psig, the saturation pressure of Refrigerant-12 at 30° F, which is exerted on the diaphragm through the capillary tube and which constitutes the total force tending to move the valve in the opening direction.

Under the conditions just described, the force tending to open the valve is exactly equal to the force tending to close the valve ($P_1 + P_2 = P_3$) and the valve will be in equilibrium. The valve will remain in equilibrium until such time that a change in the degree of suction superheat unbalances the forces and causes the valve to move in one direction or the other.

By careful analysis of the foregoing example it can be seen that for the conditions described the valve will be in equilibrium when and only when the degree of superheat of the suction vapor at the remote bulb location is 10° F, which is exactly the amount required to offset the pressure exerted by the spring. Any change in the degree of suction superheat will cause the valve to move in a compensating direction in order to restore the required amount of superheat and reestablish equilibrium. For instance, if the degree of suction superheat becomes less than 10° F, the pressure in the remote bulb will be less than the combined evaporator and spring pressures and the valve will move toward the closed position, thereby throttling the flow of liquid into the evaporator until the superheat is increased to the required 10° F. On the other hand, if the superheat becomes greater than 10° F, the pressure in the remote bulb will exceed the combined evaporator and spring pressures and the valve will move toward the open position, thereby increasing the flow of liquid into the evaporator until the superheat is reduced to the required 10° F.

In all cases, the amount of superheat required to bring a thermostatic expansion valve into equilibrium depends upon the pressure setting of the spring. It is for this reason that the spring adjustment is called the "superheat adjustment." Increasing the tension of the spring increases the amount of superheat required to offset the spring pressure and bring the valve into equilibrium. A high degree of superheat is usually undesirable in that it tends to reduce the amount of effective evaporator surface. On the other hand, decreasing the spring tension reduces the amount of superheat required to maintain the valve in a condition of equilibrium and therefore tends to increase the amount of effective surface. However, if the valve superheat is set too low, the valve will lose control of the refrigerant to the extent that it will alternately "starve" and "overfeed" the evaporator, a condition often called "hunting."

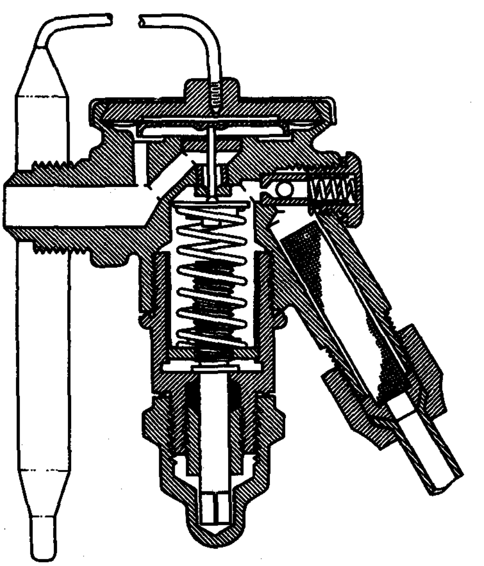


Fig. 17-8. *Conventional liquid-charged, internally equalized thermostatic expansion valve. (Courtesy General Controls.)*

As a general rule, thermostatic expansion valves are adjusted for a superheat of 7° to 10° by the manufacturer. Since this superheat setting is ordinarily satisfactory for most applications, it should not be changed except when absolutely necessary.

Once the valve is adjusted for a certain superheat, the valve will maintain approximately that superheat under all load conditions, regardless of the evaporator temperature and pressure, provided that the capacity and operating range of the valve are not exceeded. For instance, in the preceding example, assume that because of an increase in system loading the rate of vaporization in the evaporator increases to the extent that all the liquid is vaporized by the time the refrigerant leaves point B', rather than point B, in Fig. 17-7. The greater travel of the vapor before reaching point C will cause the superheat to exceed 10° F, in which case the increased bulb pressure resulting from the higher vapor temperature at point C will cause the valve to open wider and increase the flow of liquid to the evaporator, whereupon more of the evaporator surface will be filled with liquid so that the superheat is again reduced to the required 10° F. However, it is important to notice that when equilibrium is again established, the evaporator temperature and pressure will be higher than before because of the increased rate of vaporization. Furthermore, since the valve will maintain a constant superheat of approximately 10° F, the temperature of the vapor at point C will also be higher because of the increase in the evaporator temperature, as will the temperature and pressure of the fluid in the remote bulb.

Obviously, then, unlike the automatic expansion valve, the thermostatic expansion valve cannot be set to maintain a certain evaporator temperature and pressure, only a constant superheat. When a thermostatic expansion valve is used as a refrigerant control, the evaporator temperature and pressure will vary with the loading of the system, as described in Chapter 13. A typical internally equalized thermostatic expansion valve is shown in Fig. 17-8.

17-5. Externally Equalized Valves. Since the refrigerant undergoes a drop in pressure because of friction as it flows through the evaporator, the saturation temperature of the refrigerant is always lower at the evaporator outlet than at the evaporator inlet. When the refrigerant pressure drop through the evaporator is relatively small, the drop in saturation temperature is also small and therefore of little consequence. However, when the pressure drop experienced by the refrigerant in the evaporator is of appreciable size, the saturation temperature of the refrigerant at the evaporator outlet will be considerably lower than that at the evaporator inlet, a circumstance which adversely affects the operation of the expansion valve in that it necessitates a higher degree of suction superheat in order to bring the valve into equilibrium. Since more of the evaporator surface will be needed to satisfy the higher superheat requirement, the net effect of the evaporator pressure drop, unless compensated for through the use of an external equalizer, will be to reduce seriously the amount of evaporator surface which can be used for effective cooling.

For example, assume that a Refrigerant-12 evaporator is fed by a standard, internally equalized thermostatic expansion valve and that the saturation pressure and temperature of the refrigerant at the evaporator inlet (point A) is 21.05 psig and 20° F, respectively, the former being the evaporator pressure (P_1) exerted on the diaphragm of the valve. If the valve spring is adjusted for a pressure (P_2) of 7.41 psi, a bulb pressure (P_3) of 28.46 psig (21.05 + 7.41) will be required for valve equilibrium.

If it is assumed that the refrigerant pressure drop in the evaporator is negligible, as in the previous example, the saturation pressure and temperature of the refrigerant at the evaporator outlet will be approximately the same as those at the evaporator inlet, 21.05 psig and 20° F, and the amount of suction superheat required for operation of the valve will be only 10° (30° − 20°), as shown in Fig. 17-7. On the other hand, assume now that in flowing through the evaporator, the refrigerant experiences a drop in pressure of 10 psi, in which case the saturation pressure at the evaporator outlet will be approximately 11 psig (21 − 10), 10 psi less than the inlet pressure. Since the saturation temperature corresponding to 11 psig is approximately 4° F, it is evident that a suction superheat of approximately 26° F will be required to provide the 30° F suction vapor temperature which is necessary at the point of bulb contact in order to bring the valve into equilibrium.

In order to satisfy the greater superheat requirement, vaporization of the liquid must be completed prematurely in the evaporator (point B' in Fig. 17-7) so that a considerable portion of the evaporator surface becomes relatively ineffective. Naturally, the loss of effective evaporator surface will materially reduce the over-all capacity and efficiency of the system.

Although an external equalizer does not reduce the evaporator pressure drop in any way, it does compensate for it so that full and effective use of all the evaporator surface can be obtained. Notice in Fig. 17-9 that the externally equalized valve is so constructed that the evaporator pressure (P_1) which acts on the valve diaphragm is the evaporator outlet pressure rather than the evaporator inlet pressure. This is accomplished by completely isolating the valve diaphragm from the evaporator inlet pressure, while at the same time permitting the evaporator outlet pressure to be exerted on the diaphragm through a small diameter tube which is connected to the evaporator outlet or to the suction line 6 to 8 in. beyond the remote bulb location on the compressor side, as shown in Fig. 17-9. An expansion valve equipped with an external equalizer connection is shown in Fig. 17-10.

Since the evaporator pressure (P_1) exerted on the diaphragm of the externally equalized valve is the evaporator outlet pressure rather than the evaporator inlet pressure, the effect of the evaporator pressure drop is nullified to the extent that the degree of suction superheat required to operate the valve is approximately the same as when the evaporator pressure drop is negligible. Notice in Fig. 17-9 that the evaporator (outlet) pressure (P_1) exerted on the

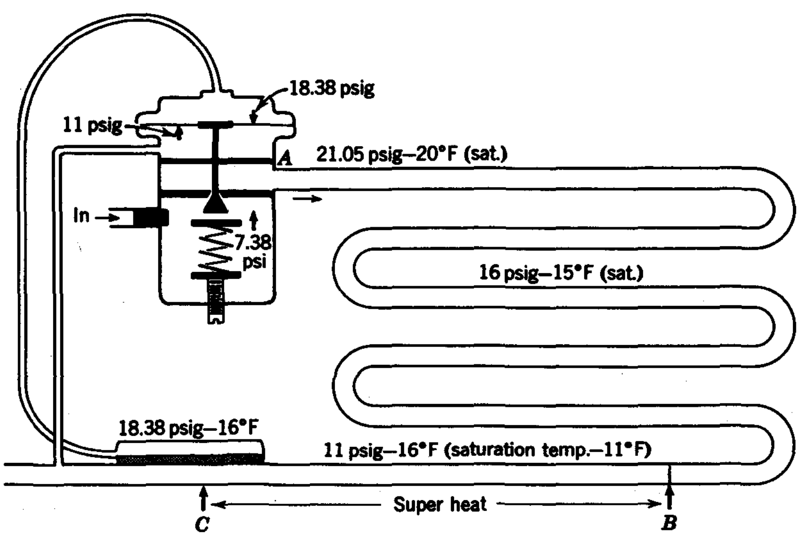


Fig. 17-9. Schematic diagram of externally equalized thermostatic expansion valve.

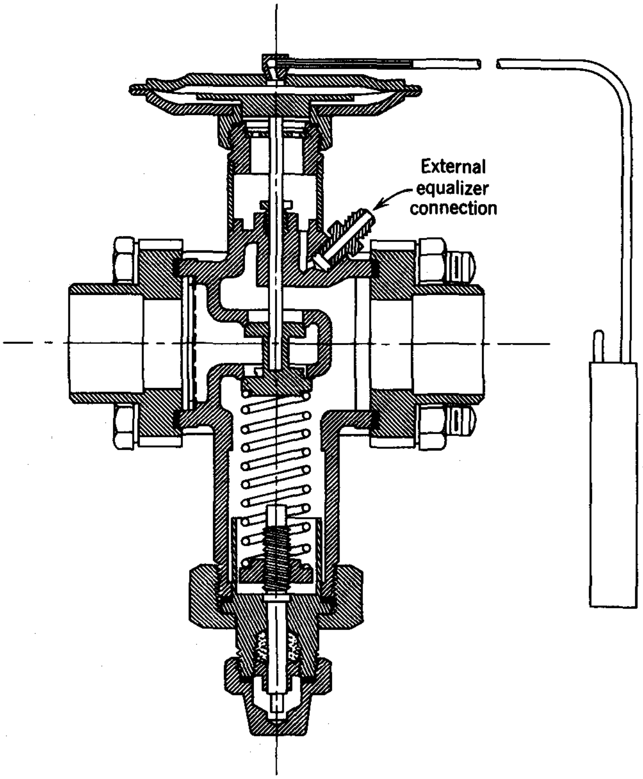


Fig. 17-10. Externally equalized thermostatic expansion valve. (Courtesy Sporlan Valve Company.)

diaphragm is 11 psig, which, when added to the spring pressure (P_2) of 7.41 psi, constitutes a total pressure of 18.41 psi, tending to move the valve in the closing direction. Hence, a bulb pressure of 18.41 psig, corresponding to a saturation temperature of approximately 16° F, is required for equilibrium. Since the saturation temperature corresponding to the suction vapor pressure of 11 psig is 4° F, a suction superheat of only 12° F (16° − 4°) is necessary to provide valve equilibrium.

17-6. Pressure Limiting Valves. The propensity of a conventional liquid charged thermostatic expansion valve for keeping the evaporator completely filled with refrigerant, without regard for the evaporator temperature and pressure, has some disadvantages as well as advantages. Although this characteristic is desirable in that it insures full and efficient use of all the evaporator surface under all conditions of loading, it is, at the same time, undesirable in that it also permits overloading of the compressor driver because of excessive evaporator pressures and temperatures during periods of heavy loading.

Another disadvantage of the conventional thermostatic expansion valve is its tendency to open wide and overfeed the evaporator when the compressor cycles on, which in many cases permits liquid to enter the suction line with possible damage to the compressor. Overfeeding at start-up is caused by the fact that the evaporator pressure drops rapidly when the compressor is started and the bulb pressure remains high until the temperature of the bulb is cooled to the normal operating temperature by the suction vapor. Naturally, because of the high bulb pressure, the valve will be unbalanced in the open direction during this period and overfeeding of the evaporator will occur until the bulb pressure is reduced.

Fortunately, these operating difficulties can be overcome through the use of thermostatic expansion valves which have built-in pressure limiting devices. The pressure limiting devices act to throttle the flow of liquid to the evaporator by taking control of the valve away from the remote bulb when the evaporator pressure rises to some predetermined maximum. Not only does this protect the compressor driver from overload during periods of heavy loading, it also tends to eliminate liquid flood-back to the compressor because of overfeeding at start-up.

The maximum operating pressure (MOP) of the expansion valve can be limited either by mechanical means or by the use of a gas charged remote bulb. The former is accomplished by placing a spring or a collapsible cartridge between the diaphragm and the valve stem or push-rods which actuate the valve pin. In the collapsible cartridge-type (Fig. 17-11), the cartridge, which is filled with a noncondensible gas, acts as a solid link between the diaphragm and the valve stem as long as the evaporator pressure is less than the pressure of the gas in the cartridge. Hence, control of the valve is vested in the remote bulb and the valve operates as a conventional thermostatic expansion valve as long as the evaporator pressure is less than the pressure of the gas in the cartridge. However, when the evaporator pressure exceeds the cartridge pressure, the cartridge collapses, thereby taking control of the valve away from the bulb and allowing the superheat spring to throttle the valve until the evaporator pressure is reduced below the cartridge pressure, at which time the cartridge will again act as a solid link, thereby returning control of the valve to the remote bulb.

The maximum evaporator pressure, called the maximum operating pressure (MOP) of the valve, depends on the pressure of the gas in the cartridge and can be changed simply by changing the valve cartridge. Cartridges are available for almost any desired maximum operating pressure.

The operation of the spring-type pressure limiting valve (Fig. 17-12) is similar to that of the collapsible cartridge type in that the spring acts as a solid link between the valve diaphragm and the valve stem or push-rods whenever the pressure in the evaporator is less than the spring tension. When the evaporator pressure rises to a point where it exceeds the tension for which the spring is adjusted, the spring collapses and the flow of refrigerant to the evaporator is throttled until the evaporator pressure is again reduced below the spring tension. Obviously, the maximum operating pressure of the valve depends on the degree of spring tension, which in some cases is adjustable in the field.

In addition to the overload protection afforded by pressure limiting valves, they also tend to eliminate the possibility of liquid flood-back to the compressor at start-up. The fact that the evaporator pressure must be reduced below the MOP of the valve before the valve can open delays the valve opening

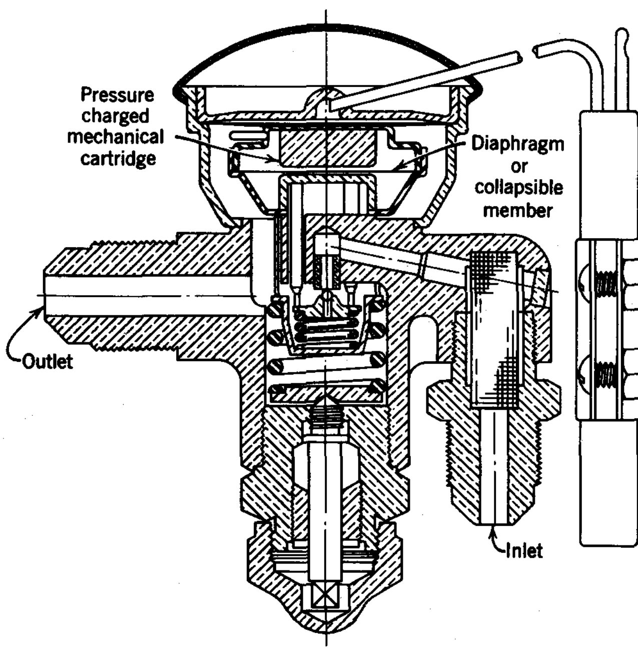


Fig. 17-11. Cartridge-type pressure limiting valve. (Courtesy Alco Valve Company.)

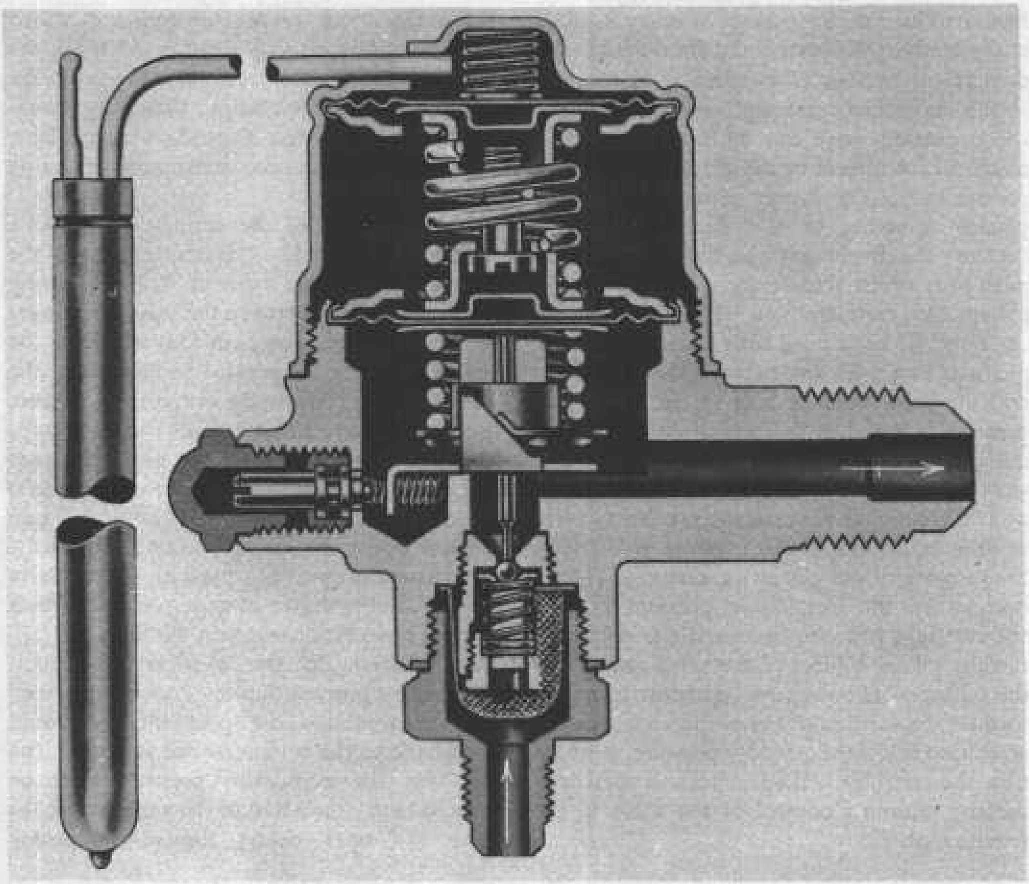


Fig. 17-12. Spring-type pressure limiting valve. (Courtesy Detroit Controls Division, American Radiator and Standard Sanitary Corporation.)

sufficiently to permit the suction vapor to cool the remote bulb and reduce the bulb pressure before the valve opens. Hence, the valve does not open wide and overfeed the evaporator when the compressor is started.

The "pull-down" characteristics of a pressure limiting expansion valve as compared to those of a conventional expansion valve is shown in Fig. 17-13.

17-7. Gas-Charged Expansion Valves. The gas-charged thermostatic expansion valve is essentially a pressure limiting valve, the pressure limiting characteristic of the valve being a function of its limited bulb charge.

The remote bulb of the gas-charged expansion valve, like that of the liquid charged valve, is always charged with the system refrigerant. However, whereas in the liquid charged valve, the remote bulb charge is sufficiently large to assure that a certain amount of liquid is always present in the remote bulb, in the gas charged valve the bulb charge is limited so that at some predetermined bulb temperature all the liquid will have vaporized and the bulb charge will be in the form of a saturated vapor. Once the bulb charge is in the form of a saturated vapor, further increases in the bulb temperature (additional superheat) will have very little effect on the bulb pressure. Hence, by limiting the amount of charge in the remote bulb, the maximum pressure which can be exerted by the remote bulb is also limited. As in the case of mechanical pressure limiting devices, limiting the pressure exerted by the remote bulb also limits the evaporator pressure, since valve equilibrium is established only when the bulb pressure (P_3) is equal to the sum of the evaporator pressure (P_1) and the spring pressure (P_2), the latter pressure being constant. Therefore, any time the evaporator pressure exceeds the maximum operating pressure of the valve, the sum of the evaporator and spring pressures will always exceed the bulb pressure and the valve will be closed.

For example, suppose the system illustrated in Fig. 17-7 is equipped with a gas charged thermostatic expansion valve having a MOP of 25 psig with a superheat setting of 10° F, in which case the bulb charge will be so limited that it will become 100% saturated vapor when the bulb temperature reaches the saturation temperature corresponding to 32.41 psig, the latter being the sum of the maximum evaporator pressure (25 psig) and the spring pressure equivalent to 10° F of superheat (7.41 psi). Once the bulb reaches this temperature, additional superheating of the suction vapor will have very little effect on the bulb pressure and therefore will not cause the valve to open wider.

In this instance, any time the evaporator pressure exceeds 25 psig, the sum of the evaporator and spring pressures will exceed the maximum bulb pressure of 32.41 psig and the valve will be closed. On the other hand, any time the evaporator pressure is below 25 psig, the sum of the evaporator and spring pressures will be less than the maximum bulb pressure and the bulb will be in control of the valve. The valve will then respond normally to any changes in the suction superheat.

Because of its pressure limiting characteristics, the gas charged thermostatic expansion valve provides the same compressor overload and flood-back protection as do mechanical pressure limiting valves.

Since the evaporator pressure is limited indirectly by limiting the bulb pressure (charge), any change in the superheat setting (spring pressure) will cause the maximum evaporator pressure (the MOP of the valve) to change. Since the bulb pressure (P_3) is always equal to the evaporator pressure (P_1) plus the spring pressure (P_2), increasing the superheat setting (P_2) will decrease the MOP (P_1). Likewise, decreasing the superheat setting will increase the maximum operating pressure of the valve.

In view of the limited bulb charge, some precautions must be observed when installing a gas charged expansion valve. The valve body must be in a warmer location than the remote bulb and the tube connecting the remote bulb to the power head must not be allowed to touch a surface colder than the remote bulb; otherwise the bulb charge will condense at the coldest point and the valve will become inoperative because of the lack of liquid in the remote bulb (Fig. 17-14). Too, care should be taken to so locate the remote bulb that the liquid does not drain from the bulb by gravity.

17-8. Importance of Pressure Limiting Valves. The importance of pressure limiting valves is readily understood when it is recognized that many refrigeration systems are subject to occasional "pull-down" loads which are substantially greater than the average system load under normal operation. Since evaporator pressures and temperatures are abnormally high during these pull-down periods, the

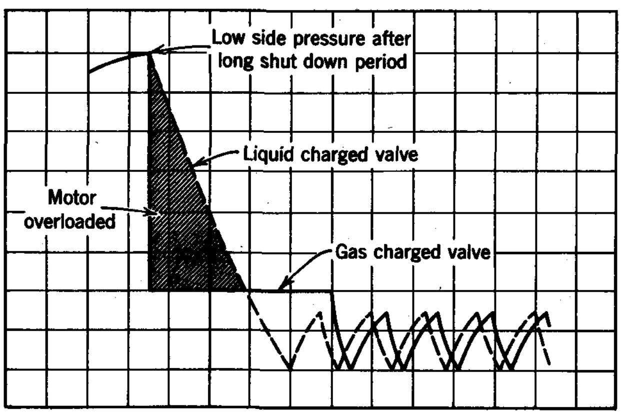


Fig. 17-13. Chart showing comparative performance of the gas charged and liquid charged valves during "pull-down." (Courtesy Detroit Controls Division, American Radiator and Standard Sanitary Corporation.)

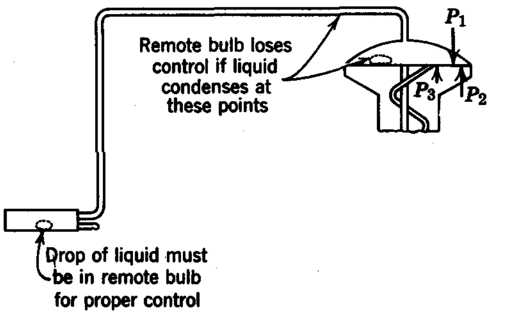


Fig. 17-14. Gas-charged thermo expansion valve. (Courtesy Alco Valve Company.)

capacity and horsepower requirements of the compressor are greatly increased, which often results in overloading of the compressor driver.

Obviously, two solutions to the problem are possible. One is to increase the size of the compressor driver so that it has sufficient power to carry the load during the overload period. The other is to limit the maximum evaporator pressure in order to avoid excessive compressor loading. Which is the better solution depends upon the requirements of the particular installation. In applications where rapid reduction of the space or product temperature is desirable, the former solution is the one recommended. However, since motoring the compressor for an occasional peak load condition unnecessarily increases both the initial and operating costs of the system, in applications where rapid reduction of the load is not required, it is usually more practical to limit the evaporator pressure to some maximum reasonably near the average evaporator pressure under normal operating conditions and then motor the compressor accordingly. This will ordinarily result in the use of a smaller size motor, thereby effecting a saving in both the initial and operating costs of the system.

For this reason, pressure limiting expansion valves of all types are widely used at the present time, particularly in air conditioning applications. As a general rule, a pressure limiting expansion valve is selected to have a MOP approximately 5 to 10 psi above the average evaporator pressure encountered at normal loading of the system. In ordering such a valve, the desired MOP must be stated.

17-9. Cross-Charged Expansion Valves. Although expansion valves having a bulb charged with the system refrigerant are suitable for most medium and high temperature applications, they are not ordinarily satisfactory for low temperature applications. The reason for this becomes evident upon examination of the pressure-temperature relationship of any common refrigerant.

A pressure-temperature curve for Refrigerant-12 is shown in Fig. 17-15. Notice that the change in pressure per degree of temperature change decreases considerably as the temperature of the refrigerant decreases. Therefore, the amount of superheat required to cause a given increase in remote bulb pressure is much greater at low temperatures than at high

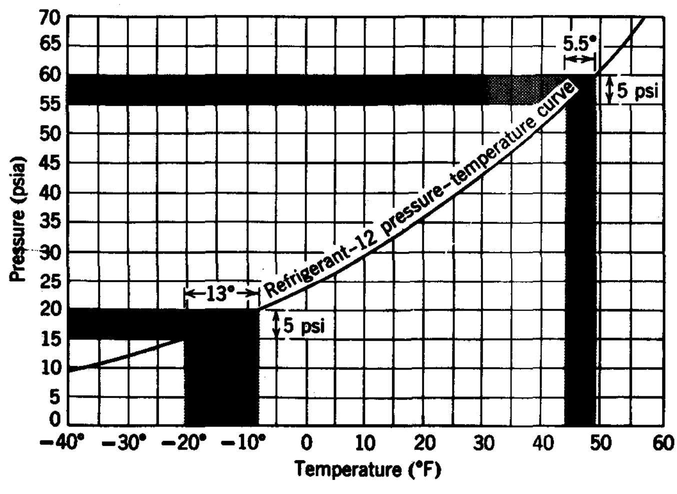


Fig. 17-15. The effect of temperature range on valve superheat.

temperatures. For example, notice in Fig. 17-15, that at a temperature of 45° F, a temperature change of only approximately 5.5° F will cause a pressure change of 5 psi, whereas at −20° F, a temperature change of approximately 13° F is required for a 5 psi change in pressure. Obviously, then, when the expansion valve bulb is charged with the system refrigerant, the amount of suction superheat necessary to actuate the valve becomes excessive at low temperatures, with the result that much of the evaporator surface becomes ineffective. Hence, for low temperature applications, good practice prescribes the use of an expansion valve having a bulb charged with some fluid other than the system refrigerant, usually one which has a boiling point somewhat below that of the system refrigerant, so that the pressure change in the bulb per degree of suction superheat is more substantial at the desired operating temperature of the valve. This will permit operation of the valve with a normal amount of suction superheat.

Expansion valves whose bulbs are charged with fluids other than the system refrigerant are called "cross-charged" valves because the pressure-temperature curve of the fluid crosses the pressure-temperature curve of the system refrigerant. Notice in Fig. 17-16 that the pressure-temperature curve of the bulb fluid (curve *B*) is somewhat flatter than that of the system refrigerant (curve *A*), so that as the evaporator pressure increases, a greater amount of suction superheat is required to bring the valve into equilibrium. Curve *C* in Fig. 17-16 indicates the bulb temperature necessary to bring the valve into equilibrium if the superheat spring is adjusted for a pressure of 5 psi. Because of the higher superheat requirement at high evaporator temperatures, the cross-charged valve has a pressure limiting effect which affords a certain amount of protection against motor overload and compressor flood-back at start-up.

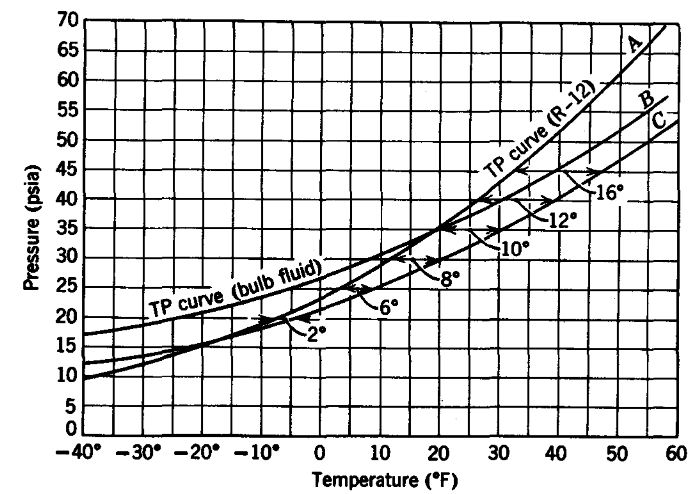


Fig. 17-16. Operating characteristics of cross-charged thermostatic expansion valve.

Cross-charged expansion valves are by no means limited to low temperature applications. They are also used extensively in commercial applications where pressure limiting is desired and where their varying superheat characteristic is not objectionable. In such applications, they are often preferable to the gas-charged valve because they have less tendency to "hunt" and because the remote bulb location is not so critical. However, cross-charged valves are not suitable for air conditioning installations since these systems require a constant superheat for optimum performance.

Since a cross-charged valve will perform satisfactorily only within a given temperature range, several different types of cross charges are required for the various temperature ranges. Naturally, in ordering a cross-charged valve, the desired operating temperature range must be stated so that a valve with the proper cross charge can be selected.

17-10. Multioutlet Valves and Refrigerant Distributors. When an evaporator has more than one refrigerant circuit, the refrigerant from the expansion valve is delivered to the

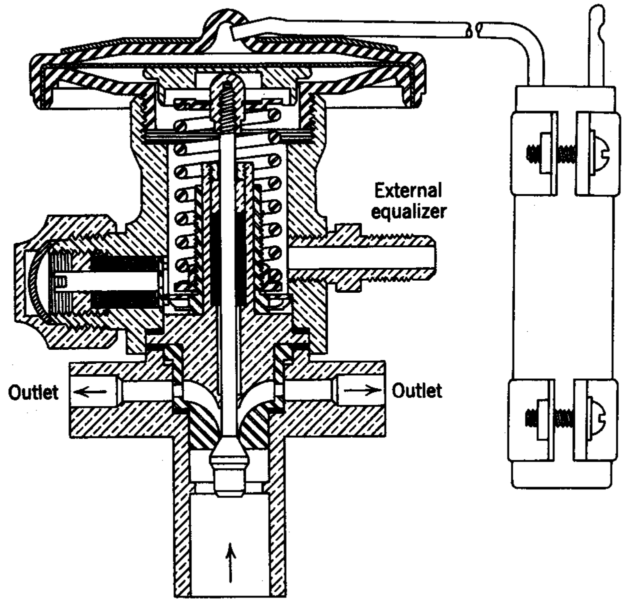
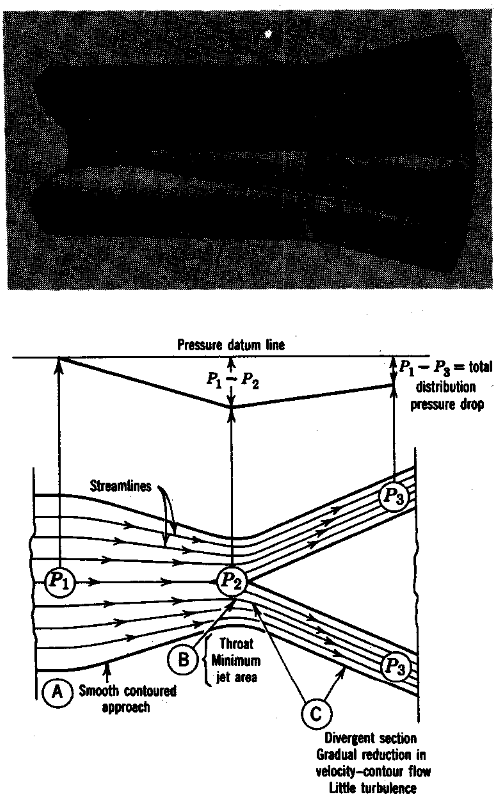


Fig. 17-17. Multioutlet thermostatic expansion valve. (Courtesy Alco Valve Company.)



Pressure recovery—low pressure drop

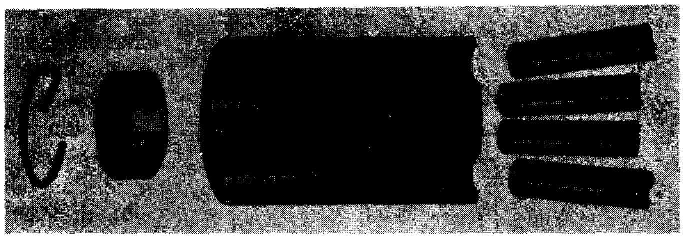
Fig. 17-18. Flow through a venturi-type distributor. (Courtesy Alco Valve Company.)

various evaporator circuits through a refrigerant distributor. In some instances the refrigerant distributor is an integral part of the valve itself. In others, it is a completely separate unit. In either case, it is important that the design of the distributor is such that the liquid-vapor mixture leaving the valve be evenly distributed to all of the evaporator circuits, if peak evaporator performance is to be expected.

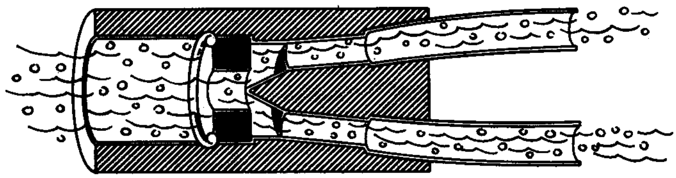
A multioutlet expansion valve incorporating a refrigerant distributor is shown in Fig. 17-17. Since expansion and distribution of the refrigerant occur simultaneously within the valve itself, this, along with the carefully proportioned passages through the radial distributor, assures even distribution of a homogeneous mixture of liquid and vapor to each of the several evaporator circuits.

Four different types of refrigerant distributors are in common use at the present time: (1) the venturi type, (2) the pressure drop type, (3) the centrifugal type, and (4) the manifold type. Any of these distributors can be used with any standard, single-outlet expansion valve.

A venturi-type distributor, along with its flow characteristics, is shown in Fig. 17-18. This type of distributor utilizes the venturi principle of a large percentage of pressure recovery, and depends on contour flow for equal distribution of the liquid-vapor mixture



(a)



(b)

Fig. 17-19. Pressure drop-type distributor. (Courtesy Sporlan Valve Company.)

to each of the evaporator circuits. This distributor provides a minimum of turbulence and a minimum over-all pressure loss, the pressure loss being confined only to wall frictional losses. It may be mounted in any position.

A pressure drop-type distributor and its flow pattern is shown in Fig. 17-19. The following description is condensed from the manufacturer's catalog data.*

The distributor consists of a body or housing, the outlet end of which is drilled to receive the tubes connecting the distributor to the evaporator. The inlet end is recessed to receive an interchangeable nozzle which is held in place by a snap ring. The refrigerant, after leaving the expansion valve, enters the distributor inlet and passes through the nozzle. The nozzle orifice is sized to produce a pressure drop which increases the velocity of the liquid, thereby homogeneously mixing the liquid and vapor and eliminating the effect of gravity. The nozzle orifice centers the flow of refrigerant so that it impinges squarely on the center of the conical button inside the distributor body. The outlet passage holes are accurately spaced around the base of the conical button so that the mixture coming off the button divides evenly as it enters these holes. The orifice size of the nozzle determines the capacity of the distributor, and the pressure drop prevents separation of flash gas from the liquid, causing a homogeneous mixture of liquid and vapor to pass through the distributor.

A centrifugal-type distributor is illustrated in Fig. 17-20. This type of distributor depends upon a high entrance velocity to create a

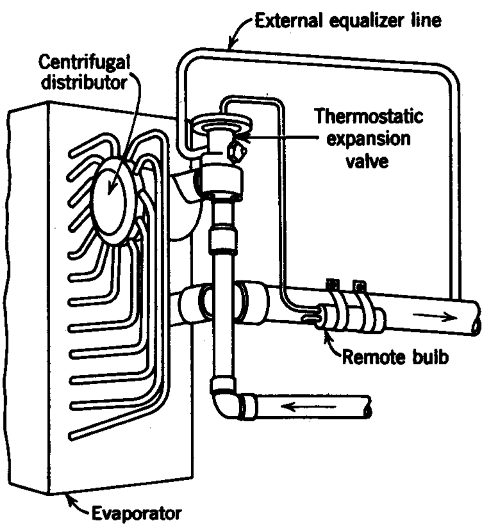


Fig. 17-20. Single outlet thermostatic expansion valve and centrifugal-type distributor. (Courtesy Alco Valve Company.)

* Sporlan Valve Company.

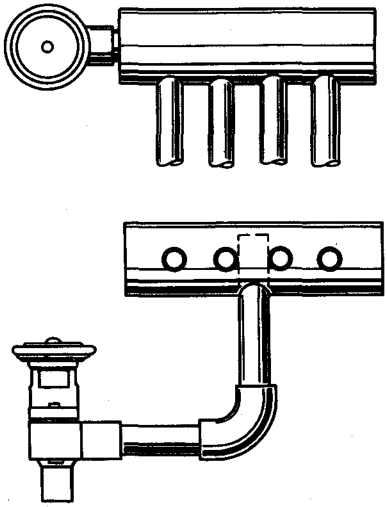


Fig. 17-21. Single outlet thermostatic expansion valve and manifold-type distributor. (Courtesy Alco Valve Company.)

swirling effect which maintains a homogeneous mixture of the liquid and flash gas and which distributes the mixture evenly to each of the evaporator tubes.

A manifold or Weir type distributor is illustrated in Fig. 17-21. This type of distributor depends upon level mounting and low entrance velocities to insure even distribution of the refrigerant to the evaporator circuits. A baffle is often installed in the header in order to minimize the tendency to overfeed the evaporator circuits directly in front of the heater inlet connection. Too, an elbow installed between the expansion valve and the header inlet will usually reduce the refrigerant velocity and help prevent unequal distribution of the refrigerant to the evaporator circuits.

Some typical distributor applications are shown in Fig. 17-22.

17-11. Expansion Valve Location. For best performance, the thermostatic expansion valve should be installed as close to the evaporator as possible. With the exception of a refrigerant distributor, where one is used, there should be no restrictions of any kind between the evaporator and the expension valve. When it is necessary to locate a hand valve on the outlet side of the valve, the hand valve should have a full sized port.

Since there is enough liquid in a liquid charged expansion valve to insure that control of the valve will remain with the bulb under all conditions, a liquid charged thermostatic expansion valve can be installed in any position (power head up, down, or sideways), either inside or outside of the refrigerated space, without particular concern for the relative temperatures of the valve body and remote bulb. On the other hand, gas charged valves must be installed so that the valve body is always warmer than the remote bulb, preferably with the power head up.

With the exception of the manifold-type distributor, when a refrigerant distributor is used, the valve should be installed as close to the distributor as possible.

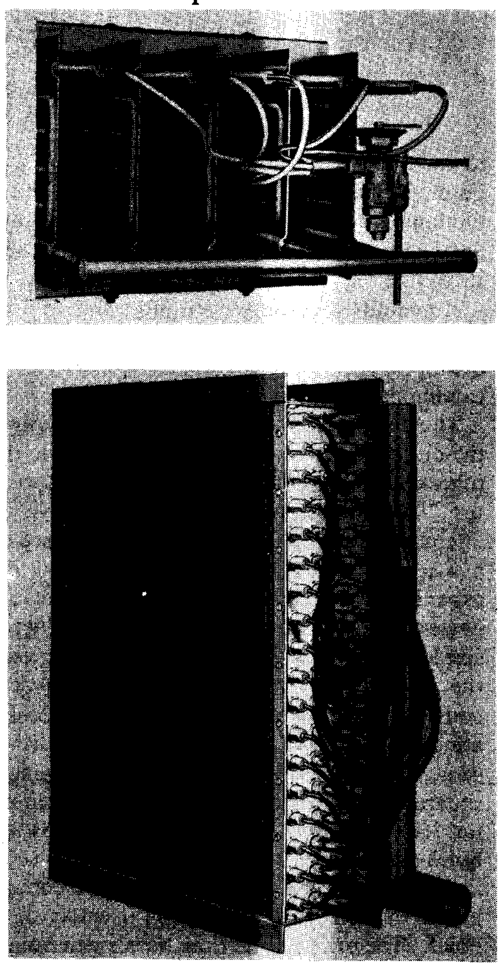


Fig. 17-22. Illustrating applications of refrigerant distributors. (Courtesy Sporlan Valve Company.)

17-12. Remote Bulb Location. To a large extent, the performance of the thermostatic expansion valve depends upon the proper location and installation of the remote bulb. When an external remote bulb is used (mounted on the outside, rather than the inside of the refrigerant piping) as is normally the case, the bulb should be clamped firmly (with metal clamps) to a horizontal section of the suction line near the evaporator outlet, preferably inside the refrigerated space.

Since the remote bulb must respond to the temperature of the refrigerant vapor in the suction line, it is essential that the entire length of the remote bulb be in good thermal contact with the suction line. When an iron pipe or steel suction line is used, the suction line should be cleaned thoroughly at the point of bulb location and painted with aluminum paint in order to minimize corrosion. On suction lines under $\frac{7}{8}$ in. OD, the remote bulb is usually installed on top of the line. For suction lines $\frac{7}{8}$ in. OD and above, a remote bulb located in a position of 4 or 8 o'clock (Fig. 17-23) will normally give satisfactory control of the valve. However, since this is not true in all cases, the optimum bulb location is often best determined by trial and error.

It is important also that the remote bulb be so located that it is not unduly influenced by temperatures other than the suction line temperature, particularly during the compressor off cycle. If the temperature of the bulb is permitted to rise substantially above that of the evaporator during the off cycle, the valve will open allowing the evaporator to become filled with liquid refrigerant, with the result that liquid will flood back to the compressor when the compressor starts. When the remote bulb is located inside the refrigerated space, the temperature difference between the fixture temperature and the evaporator temperature is not usually large enough to affect adversely expansion valve operation. However, when it is necessary to locate the bulb outside the refrigerated space, both the bulb and the suction line must be well insulated from the surroundings. The insulation must be nonhydroscapoic and must extend at least 1 ft or more beyond the bulb location on both sides of the bulb.

Care must be taken also to locate the thermal bulb at least $1\frac{1}{2}$ ft from the point where an uninsulated suction line leaves a refrigerated fixture. When the bulb is located on the suction line too close to the point where the line leaves the refrigerated space, heat conducted along the suction line from the outside may cause the bulb pressure to increase to the extent that the valve will open and permit liquid to fill the evaporator during the off cycle.

On air conditioning applications, when suitable pressure limiting valves are employed, the remote bulb may be located either outside or inside the air duct, but always out of the direct air stream. On brine tanks or water coolers, the bulb should always be located below the liquid level at the coldest point.

Whenever the bulb location is such that there is a possibility that the valve may open on the off cycle, a solenoid valve should be installed in the liquid line directly in front of the expansion valve so that positive shut-off of the liquid during the off cycle is assured. The system then operates on a pump-down cycle.

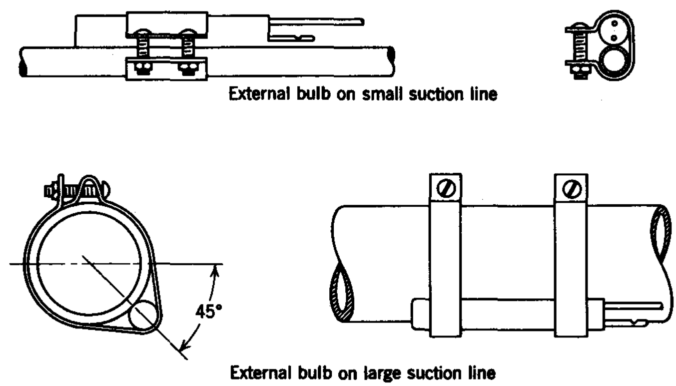
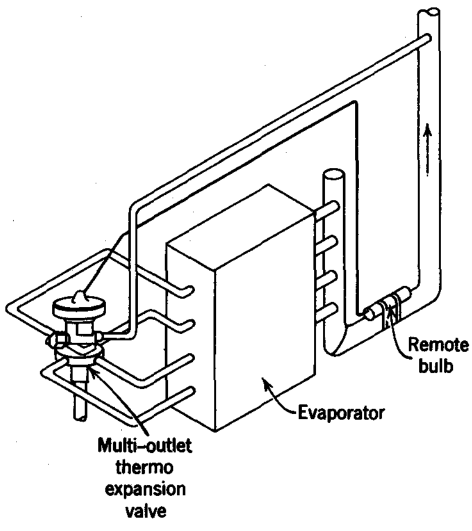
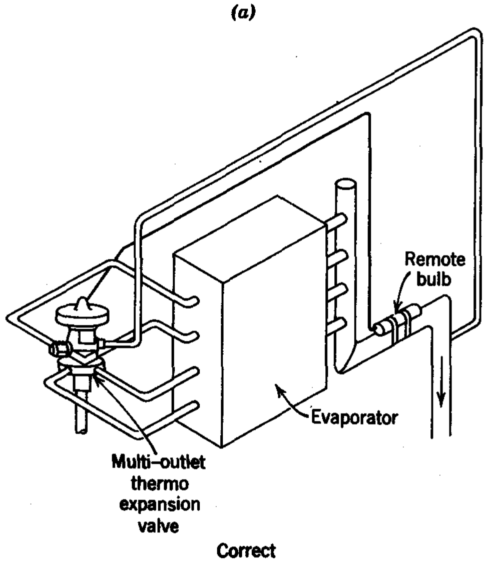


Fig. 17-23. Usual location of expansion valve remote bulb. (Courtesy Alco Valve Company.)



Incorrect
(a)



Correct
(b)

Fig. 17-24. (*a*) **Remote bulb location shown trapped. (*b*) Remote bulb location shown free draining. (Courtesy Alco Valve Company.)**

Under no circumstances should a remote bulb ever be located where the suction line is trapped. Any accumulation of liquid in the suction line at the point where the remote bulb is located will cause irregular operation (hunting) of the expansion valve. Except in a few special cases, the remote bulb must be located on the evaporator side of a liquid-suction heat exchanger.

Several of the more common incorrect remote bulb applications are shown in Figs. 17-24 through 17-26, along with recommended corrections for piping and remote bulb locations to avoid these conditions. In Fig. 17-24, liquid can trap in the suction line at the evaporator outlet, causing the loss of operating superheat and resulting in irregular operation of the valve due to alternate drying and filling of the trap. If valve operation becomes too irregular, liquid may be blown back to the compressor by the gas which forms in the evaporator behind the trap.

Figure 17-25 illustrates the proper remote bulb location to avoid trapped oil or liquid from affecting the operation of the expansion valve when the suction line must rise at the evaporator outlet. Liquid or oil accumulating in the trap during the off cycle will not affect the remote bulb and can evaporate without "slugging" to the compressor when the compressor is started. This piping arrangement is often used deliberately on large installations to avoid the possibility of liquid slugging to the compressor.

Figure 17-26 illustrates the incorrect application of the remote bulb on the suction header of an evaporator. With poor air circulation through the evaporator, liquid refrigerant can pass through some of the evaporator circuits without being evaporated and without affecting the remote bulb, a condition which can cause flood back to the compressor. The correct remote bulb location is shown by the dotted lines. However, correcting the remote bulb location in this instance will do nothing to

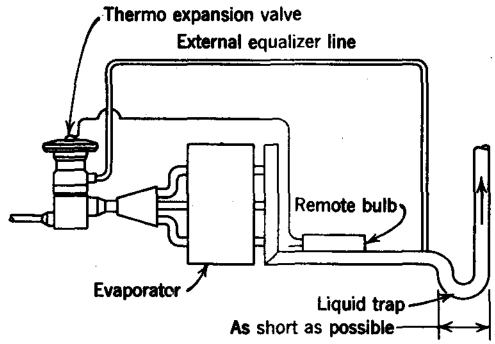


Fig. 17-25. Recommended remote bulb location and schematic piping for rising suction line. (Courtesy Alco Valve Company.)

improve the poor air distribution, but will only prevent flood back to the compressor. The air distribution must be approached as a separate problem.

Since a trapped or partially trapped suction line at the remote bulb location will cause poor expansion valve performance, care should always be taken to arrange the suction piping from the evaporator so that oil and liquid will be drained away from the remote bulb location by gravity.

Location of the remote bulb on a vertical section of suction line should be avoided whenever possible. However, in the event that no other location is possible, the bulb should be installed well above the liquid trap on a suction riser.

In some instances, it is necessary or desirable to employ a remote bulb well, so that the remote bulb is in effect installed inside of the suction line (see Fig. 17-27). As a general rule, a remote bulb well should be used when low superheats are required or when the remote bulb is likely to be influenced by heat conducted down the suction line from a warm space. It is desirable also in installations where the suction line is very short or where the size of the suction line exceeds 2⅛ in. OD.

17-13. External Equalizer Location. In general, an external equalizer should be used in any case where the pressure drop through the evaporator is sufficient to cause a drop in the saturation temperature of the refrigerant in excess of 2° F at evaporator temperatures above 0° F or in excess of approximately 1° F at evaporator temperatures below 0° F. Naturally,

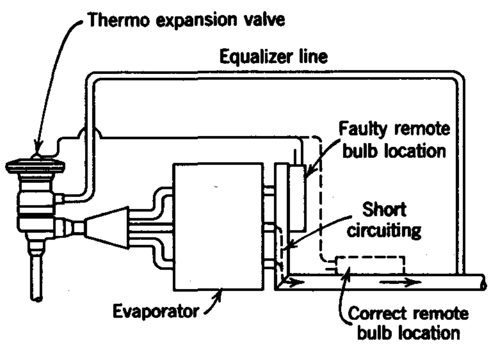


Fig. 17-26. Correct remote bulb location on "short circuiting" evaporator to prevent "flood back." (Courtesy Alco Valve Company.)

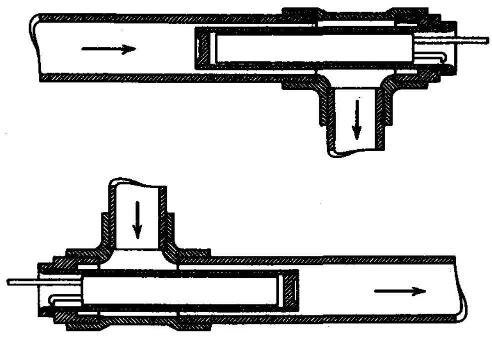


Fig. 17-27. Remote bulb well and suggested piping. (Courtesy Alco Valve Company.)

because of the pressure-temperature relationship of the refrigerant, the maximum permissible pressure drop will vary with the individual refrigerant and with the operating temperature range. For example, with regard to Refrigerant-12, the permissible pressure drop is approximately 2.5 psi when the evaporator temperature is about 40° F, whereas for evaporator temperatures below 0° F, the permissible evaporator pressure drop is only approximately 0.5 psi.

External equalizers are required also whenever multioutlet expansion valves or refrigerant distributors are employed, regardless of whether or not the pressure drop through the evaporator is excessive. In such installations, the external equalizer is required in order to compensate for the pressure drop suffered by the refrigerant in passing through the distributor.

It should be emphasized at this point that the refrigerant pressure loss in a refrigerant distributor in no way affects the capacity or efficiency of the system, provided that the expansion valve has been properly selected (see Section 17-14). In any system, with or without a distributor, the refrigerant must undergo a drop in pressure between the high and low pressure side of the system. On systems without distributors, all of this pressure drop is taken across the valve. When a distributor is employed, a portion of the pressure drop occurs in the distributor and the remainder across the valve. The total pressure drop and refrigerating effect are the same in either case.

This is not true, however, when the pressure loss is in the evaporator itself. As described in Section 8-9, any refrigerant pressure drop in the evaporator tends to reduce the capacity and efficiency of the system. Hence, evaporators having excessive pressure drops should be avoided whenever possible.

As a general rule, the external equalizer connection is made on the suction line 6 to 8 in. beyond the expansion valve bulb on the compressor side. However, in applications where the external equalizer is used to offset pressure drop through a refrigerant distributor and the pressure drop through the evaporator is not excessive, the external equalizer may be connected either to one of the feeder tubes or to one of the evaporator return bends at approximately the midpoint of the evaporator. When the external equalizer is connected to a horizontal line, it should be installed on top of the line in order to avoid the drainage of oil or liquid into the equalizer tube.

17-14. Expansion Valve Rating and Selection. Before the proper size valve can be selected, a decision must be made as to the exact type of valve desired with respect to bulb charge, pressure limiting, the possible need for an external equalizer, and the size of the valve inlet and outlet connections. Obviously, the nature and conditions of the application will determine the type of bulb charge and also whether or not a pressure limiting valve is needed. An externally equalized valve should be used whenever the pressure drop through the evaporator is substantial and/or when a refrigerant distributor is employed. The size of the inlet and outlet connections of the valve should be equal to those of the liquid line and evaporator, respectively. A slight reduction in size at the evaporator inlet is permissible.

Once a decision has been made on all of the foregoing, the proper size valve can be selected from the manufacturer's catalog ratings. Table R-20 is a typical thermostatic expansion valve rating table. The expansion valves are rated in tons of refrigerating capacity (or Btu/hr) at various operating conditions. Normally, valve ratings are based on a condensing temperature of 100° F with zero degrees of subcooling, but with solid liquid approaching the valve.

In order to select the proper size valve from the rating table, the following data should be known: (1) the evaporator temperature, (2) the system capacity in tons, and (3) the available pressure difference across the valve. In general, the first two factors determine the required liquid flow rate through the valve, whereas the latter determines the size orifice required to deliver the desired flow rate, the flow rate through the orifice being proportional to the pressure differential across the valve.

The pressure difference across the valve can never be taken as the difference between the suction and discharge pressures as measured at the compressor. When these two pressures are used as a basis for determining the pressure difference across the expansion valve, an allowance must always be made for the pressure losses which accrue between the expansion valve and the compressor on both the low and high pressure sides of the system. This includes the refrigerant distributor when one is used.

When the available pressure difference across the expansion valve has been determined, a valve should be selected from the manufacturer's rating table which has a capacity equal to or slightly in excess of the system capacity at the system design operating conditions.

Example 17-1. From Table R-20, select the proper size expansion valve for a Refrigerant-12 system, if the desired capacity is 8 tons at a 20° F evaporator temperature and the available pressure drop across the expansion valve is approximately 65 psi.

Solution. From Table R-20, select expansion valve Model #TJL1100F, which has a capacity of 8.1 tons at a 20° F evaporator temperature when the pressure drop across the valve is 60 psi. The letters in the model number indicate valve type and refrigerant.

Thermostatic expansion valves will not operate satisfactorily at less than 50% of their rated capacity. Therefore, when the load on the system is likely to fall below 50% of the design load, the evaporator should be split into two or more separate circuits, with each circuit being fed by an individual expansion valve. With this arrangement it is possible to cycle out portions of the evaporator as the load on the system fall off so that the load on any one expansion valve never drops below 50% of the design capacity of the valve.

17-15. Capillary Tubes. The capillary tube is the simplest of all refrigerant flow controls, consisting merely of a fixed length of small diameter tubing installed between the condenser and the evaporator in place of the conventional liquid line (Fig. 17-28). Because of the high frictional resistance resulting from its length and small bore, the capillary tube acts to restrict or to meter the flow of liquid from the condenser to the evaporator and also to maintain the required operating pressure differential between these two units.

For any given tube length and bore, the resistance of the tube is fixed or constant so that the liquid flow rate through the tube at any one time is always proportional to the pressure differential across the tube, said pressure differential being the difference between the vaporizing and condensing pressures of the system. Likewise, the greater the frictional resistance of the tube (the longer the tube and/or the smaller the bore), the greater is the pressure differential required for a given flow rate.

Since the capillary tube and the compressor are connected in series in the system, it is evident that the flow capacity of the tube must of necessity be equal to the pumping capacity of the compressor when the latter is in operation. Consequently, if the system is to perform efficiently and balance out at the design operating conditions, the length and bore of the tube must be such that the flow capacity of the tube at the design vaporizing and condensing pressures is exactly equal to the pumping capacity of the compressor at these same conditions.

In the event that the resistance of the tube is such that the flow capacity of the tube is either greater than or less than the pumping capacity of the compressor at the design conditions, a balance will be established between these two components at some operating conditions other than the system design conditions. For example, if the resistance of the tube is too great (tube too long and/or bore too small), the capacity of the tube to pass liquid refrigerant from the condenser to the evaporator will be less than the pumping capacity of the compressor at the design conditions, in which case the evaporator will become starved while the excess liquid will back-up in the lower portion of the condenser at the entrance to the capillary tube. Naturally, starving of the evaporator will result in lowering the suction pressure, whereas the build-up of liquid in the condenser will result in a reduction of the effective condensing surface and, consequently, an increase in the condensing temperature. Hence, the net effect of too much restriction in the capillary tube is to lower the suction pressure and raise the condensing pressure. Since both these conditions tend to increase the flow capacity of the tube and, at the same time, decrease the pumping capacity of the compressor, it is evident that the system will eventually establish equilibrium at some operating conditions where the capacity of the tube and the capacity of the compressor are exactly the same. In this instance, the point of balance will be at a lower suction pressure and a higher condensing pressure than the system design pressures. Too, since the capacity of the compressor is reduced at these conditions, the overall system capacity will be less than the design capacity.

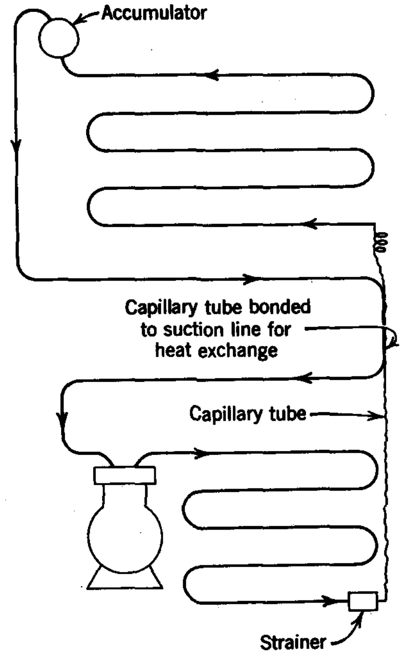


Fig. 17-28. Capillary tube system.

On the other hand, when the tube does not have enough resistance (tube too short and/or bore too large), the flow capacity of the tube will be greater than the pumping capacity of the compressor at the design conditions, in which case overfeeding of the evaporator will result with the danger of possible liquid flood-back to the compressor. Also, there will be no liquid seal in the condenser at the entrance to the tube and, therefore, uncondensed gas will be allowed to enter the tube along with the liquid. Obviously, the introduction of latent heat into the evaporator in the form of uncondensed gas will have the effect of reducing the system capacity. Furthermore, because of the excessive flow rate through the tube, the compressor will not be able to reduce the evaporator pressure to the desired low level.

From the foregoing it is evident that the design of the capillary tube must be such that the flow capacity of the tube is identical to the pumping capacity of the compressor at the system design conditions. It is evident also that a system employing a capillary tube will operate at maximum efficiency only at one set of operating conditions. At all other operating conditions, the efficiency of the system will be somewhat less than maximum. However, it should be pointed out that the capillary tube is self-compensating to some extent and, if properly designed and applied, will give satisfactory service over a reasonable range of operating conditions. Normally, as the load on the system increases or decreases, the flow capacity of the capillary tube increases or decreases, respectively, because of the change in condensing pressure which ordinarily accompanies these changes in system loading.

The capillary tube differs from other types of refrigerant flow controls in that it does not close off and stop the flow of liquid to the evaporator during the off cycle. When the compressor cycles off, the high and low side pressures equalize through the open capillary tube and any residual liquid in the condenser passes to the low pressure evaporator where it remains until the compressor cycles on again. For this reason, the refrigerant charge in a capillary tube system is critical and no receiver tank is employed between the condenser and the capillary tube. In all cases, the refrigerant charge should be the minimum which will satisfy the requirements of the evaporator and at the same time maintain a liquid seal in the condenser at the entrance to the capillary tube during the latter part of the operating cycle. Any refrigerant in excess of this amount will only back up in the condenser, thereby increasing the condensing pressure which, in turn, reduces the system efficiency and tends to unbalance the system by increasing the flow capacity of the tube. If the overcharge is sufficiently large, overloading of the compressor driver may also result. However, of more importance is the fact that all of the excess liquid in the condenser will pass to the evaporator during the off cycle. Being at the condensing temperature, a substantial amount of such liquid will cause the evaporator to warm up rapidly, thereby causing defrosting of the evaporator and/or short cycling of the compressor. Moreover, where a considerable amount of liquid enters the evaporator during the off cycle, flood-back to the compressor is likely to occur when the compressor cycles on.

Other than its simple construction and low cost, the capillary tube has the additional advantage of permitting certain simplifications in the refrigerating system which further reduces manufacturing costs. Because the high and low pressure equalize through the capillary tube during the off cycle, the compressor starts in an "unloaded" condition. This allows the use of a low starting torque motor to drive the compressor; otherwise a more expensive type of motor would be required. Furthermore, the small and critical refrigerant charge required by the capillary tube system results not only in reducing the cost of the refrigerant but also in eliminating the need for a receiver tank. Naturally, all these things represent a substantial savings in the manufacturing costs. Thus, capillary tubes are employed almost universally on all types of domestic refrigeration units, such as refrigerators, freezers, and room coolers. Many are used also on small commercial packaged units, particularly packaged air conditioners.

Capillary tubes should be employed only on those systems which are especially designed for their use. They are best applied to close-coupled, packaged systems having relatively constant loads and employing hermetic motor-compressors. Specifically, a capillary tube should not be used in conjunction with an open type compressor. Because of the critical refrigerant charge, an open type compressor may lose sufficient refrigerant by seepage around the shaft seal to make the system inoperative in only a very short time.

The use of capillary tubes on remote systems (compressor located some distance from the evaporator) should also be avoided as a general rule. Such systems are very difficult to charge accurately. Furthermore, because of the long liquid and suction lines, a large charge of refrigerant is required, all of which concentrates in the evaporator during the off cycle. Serious flood-back to the compressor is likely to occur at start-up unless an adequate designed accumulator is installed in the suction line.

Condensers designed for use with capillary tubes should be so constructed that liquid drains freely from the condenser into the capillary tube in order to prevent the trapping of liquid in the condenser during the off cycle. Any liquid trapped in the condenser during the off cycle will evaporate and pass through the tube to the evaporator in the vapor state rather than in the liquid state. The subsequent condensation of this vapor in the evaporator will unnecessarily add latent heat to the evaporator, thereby reducing the capacity of the system.

Too, the diameter of the condenser tubes should be kept as small as is practical so that a minimum amount of liquid backed-up in the condenser at the tube inlet will cause a maximum increase in the condensing pressure and therefore a maximum increase in the flow capacity of the tube.

Evaporators intended for use with capillary tubes should provide for liquid accumulation at the evaporator outlet in order to prevent liquid flood-back to the compressor at start-up (Fig. 17-28). The function of the accumulator is to absorb the initial surge of liquid from the evaporator as the compressor starts. The liquid then vaporizes in the accumulator and returns to the compressor as a vapor. To expedite the return of oil to the compressor crankcase, liquid from the evaporator usually enters at the bottom of the accumulator, whereas the suction to the compressor is taken from the top.

In most cases, best performance is obtained when the capillary tube is connected directly between the condenser and the evaporator without an intervening liquid line. When the condenser and evaporator are too far apart to make direct connection practical, some other type of refrigerant control should ordinarily be used.

Bonding (soldering) the capillary tube to the suction line for some distance in order to provide a heat transfer relationship between the two is usually desirable in that it tends to minimize the formation of flash gas in the tube. Flash gas, formed in the tube because of the gradual expansion of the liquid as its pressure is reduced, seriously reduces the flow capacity of the tube. When the tube is not bonded to the suction line, the tube must be shortened sufficiently to offset the throttling action of the vapor in the tube.

17-16. Flooded Evaporator Control. Refrigerant flow controls employed with flooded evaporators are usually of the float type. The float control consists of a buoyant member (hollow metal ball, cylinder or pan) which is responsive to refrigerant liquid level and which acts to open and close a valve assembly to admit more or less refrigerant into the evaporator in accordance with changes in the liquid level in the float chamber. The float chamber may be located on either the low pressure side or high pressure side of the system. When the float is located on the low pressure side of the system, the float control is called a low pressure float control. When the float is located on the high pressure side of the system, the float control is known as a high pressure float control.

The principal advantage of the flooded evaporator lies in the higher evaporator capacity and efficiency which is obtained therefrom. With flooded operation, the refrigerant in all parts of the evaporator is predominately in the liquid state and a high refrigerant side tube coefficient is produced, as compared to that obtained with the dry-expansion type evaporator wherein the refrigerant in the evaporator is predominately in the vapor state, especially in the latter part of the evaporator. For this reason, float controls (flooded evaporators) are used extensively in large liquid chilling installations where advantage can be taken of the high refrigerant side conductance coefficient. On the other hand, because of their bulk and the relatively large refrigerant charge required, float controls are seldom employed on small applications, having been discarded in this area in favor of the smaller, more versatile thermostatic expansion valve or the simpler, more economical capillary tube.

17-17. Low Pressure Float Control. The low pressure float control (low side float) acts to maintain a constant level of liquid in the evaporator by regulating the flow of liquid refrigerant into that unit in accordance with the rate at which the supply of liquid is being depleted by vaporization. It is responsive only

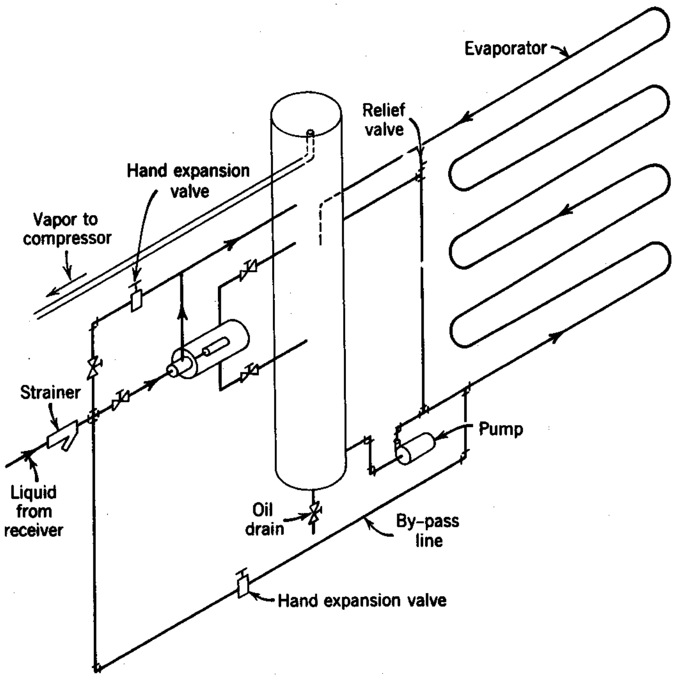


Fig. 17-29. Low side float valve controlling liquid level in accumulator. Note liquid pump used to recirculate refrigerant through evaporator.

to the level of liquid in the evaporator and will maintain the evaporator filled with liquid refrigerant to the desired level under all conditions of loading without regard for the evaporator temperature and pressure.

Operation of the low pressure float valve may be either continuous or intermittent. With continuous operation, the low pressure float valve has a throttling action in that it modulates toward the open or closed position to feed more or less liquid into the evaporator in direct response to minor changes in the evaporator liquid level. For intermittent operation, the valve is so designed that it responds only to minimum and maximum liquid levels, at which points the valve is either fully open or fully closed as the result of a toggle arrangement built into the valve mechanism.

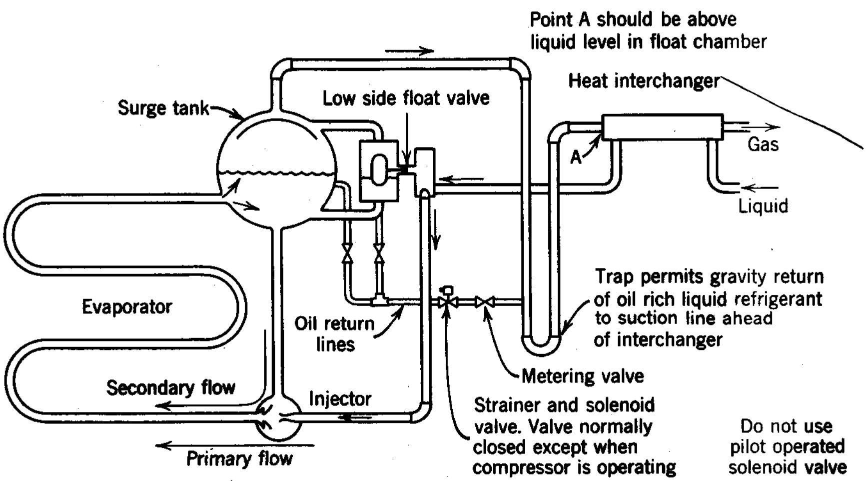


Fig. 17-30. Flooded evaporator (recirculating injector circulation). (Courtesy General Electric.)

The low pressure float may be installed directly in the evaporator or accumulator in which it is controlling the liquid level (Fig. 11-1), or it may be installed external to these units in a separate float chamber (see Fig. 17-29).

On large capacity systems, a by-pass line equipped with a hand expansion valve is usually installed around the float valve in order to provide refrigeration in the event of float valve failure. Too, hand stop valves are usually installed on both sides of the float valve so that the latter can be isolated for servicing without the necessity for evacuating the large refrigerant charge from the evaporator (Fig. 17-29). Notice also in Fig. 17-29 the liquid pump employed to provide forced circulation of the refrigerant through the evaporator tubes. It is of interest to compare this method of recirculation to the injection method of recirculation shown in Fig. 17-30, and with the gravity recirculation method shown in Fig. 11-1. The hand expansion valve in the by-pass line around the liquid pump in Fig. 17-29 is to provide refrigeration in the event of pump failure.

Low pressure float valves may be used in multiple or in parallel with thermostatic expansion valves. In many instances, a single low pressure float valve can be used to control the liquid flow into several different evaporators.

17-18. High Pressure Float Valves. Like the low pressure float valve, the high pressure float valve is a liquid level actuated refrigerant flow control which regulates the flow of liquid to the evaporator in accordance with the rate at which the liquid is being vaporized. However, whereas the low pressure float valve controls the evaporator liquid level directly, the high pressure float valve is located on the high pressure side of the system and controls the amount of liquid in the evaporator indirectly by maintaining a constant liquid level in the high pressure float chamber (Fig. 17-31).

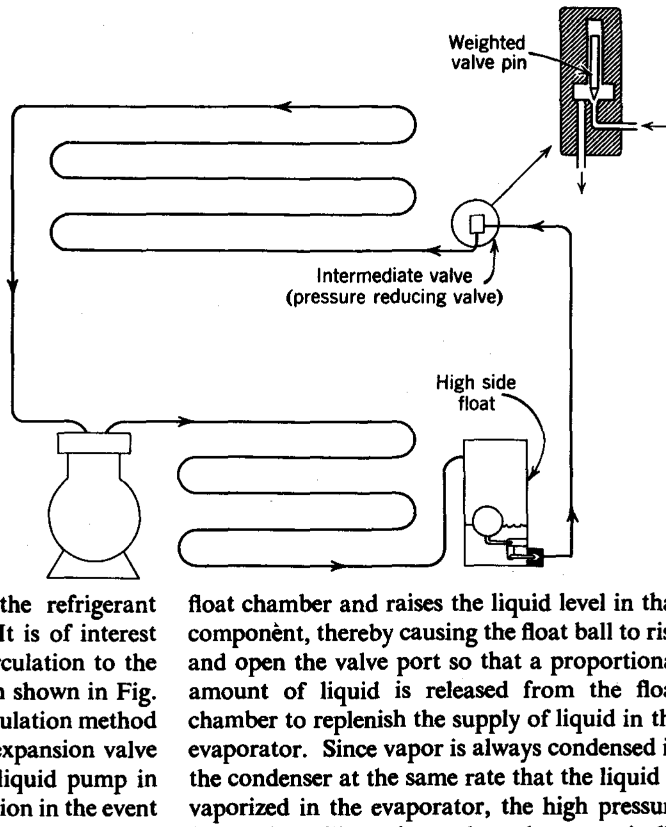


Fig. 17-31. High pressure float valve.

The operating principle of the high pressure float valve is relatively simple. The refrigerant vapor from the evaporator condenses into the liquid state in the condenser and passes into the float chamber and raises the liquid level in that component, thereby causing the float ball to rise and open the valve port so that a proportional amount of liquid is released from the float chamber to replenish the supply of liquid in the evaporator. Since vapor is always condensed in the condenser at the same rate that the liquid is vaporized in the evaporator, the high pressure float valve will continuously and automatically feed the liquid back to the evaporator at a rate commensurate with the rate of vaporization, regardless of the system load. When the compressor stops, the liquid level in the float chamber drops, causing the float valve to close and remain closed until the compressor is started again.

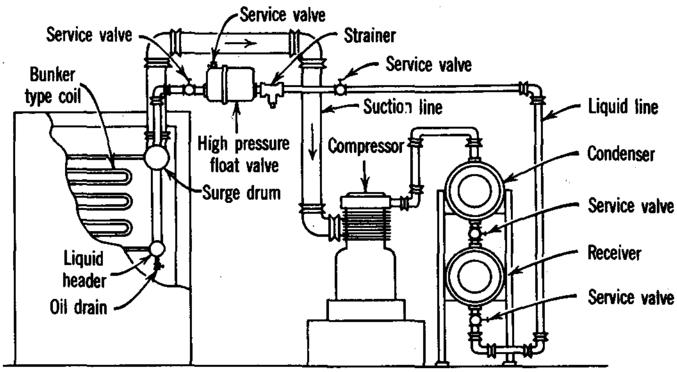


Fig. 17-32. Typical high pressure float valve application. (Courtesy Alco Valve Company.)

Since the high pressure float valve permits only a small and fixed amount of refrigerant to remain in the high pressure side of the system, it follows that the bulk of the refrigerant charge is always in the evaporator, and that the refrigerant charge is critical. An overcharge of refrigerant will cause the float valve to overfeed the evaporator with the result that liquid refrigerant will flood back to the compressor. Moreover, if the system is seriously overcharged, the float valve will not throttle the liquid flow sufficiently to allow the compressor to reduce the evaporator pressure to the desired low level. On the other hand, if the system is undercharged, operation of the float will be erratic and the evaporator will be starved.

The high pressure float valve may be used with a dry-expansion type evaporator as shown in Fig. 17-31, or with a flooded-type evaporator as shown in Fig. 17-32. With the latter, liquid refrigerant is expanded into the surge drum (low pressure receiver) from where it flows into the evaporator through the drop leg pipe attached to the bottom of the surge drum. The suction vapor is drawn off at the top of the surge drum, as is the flash gas resulting from the expansion of the liquid as it passes through the float valve. To prevent liquid flood-back during changes in loading, the surge drum should have a volume equal to at least 25% of the evaporator volume.

The construction of a typical high pressure float valve is illustrated in Fig. 17-33. Notice that the float valve opens on a rising liquid level and that the construction of the valve pin and float arm pivot are such that the weight of the float ball will move the valve pin in a closing direction as the liquid refrigerant recedes in the float chamber. Notice also that the float ball is so positioned that the valve seat is always submerged in the liquid refrigerant in order to eliminate the possibility of wire-drawing by high velocity gas passing through the valve pin and seat. Too, the high pressure float assembly

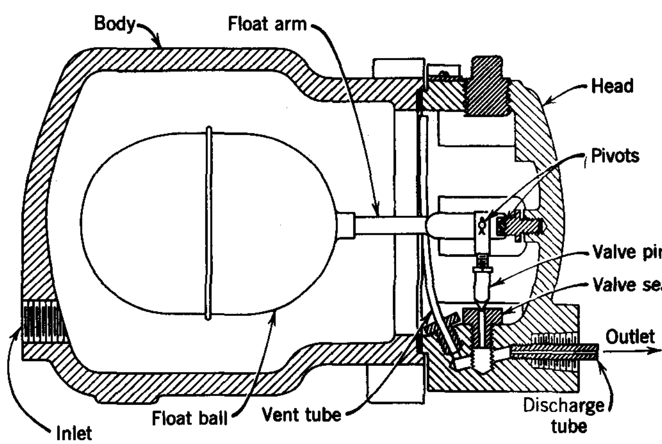


Fig. 17-33. High pressure float valve. (Courtesy Alco Valve Company.)

contains a vent tube to prevent the float chamber from becoming gas bound by noncondensible gases which may otherwise collect in the chamber and build up a pressure, thereby preventing liquid refrigerant from entering the chamber. The use of the vent tube makes possible the installation of the high pressure float valve at a point above or below the condenser without the danger of gas binding.

Unlike the low pressure float control, the high pressure float control, being independent of the evaporator liquid level, may be installed either above or below that unit. However, the float valve should be located as close to the evaporator as possible and always in a horizontal line in order to insure free action of the float ball and valve assembly. When the float is located some distance from the evaporator, it is usually necessary to provide some means of maintaining a high liquid pressure in the line between the float valve and the evaporator in order to prevent premature expansion of the liquid before it reaches the evaporator. In small systems, this is accomplished by installing an "intermediate" valve in the liquid line at the entrance to the evaporator (see inset of Fig. 17-31). In larger systems, a pilot valve is ordinarily employed for this purpose (see Section 17-21).

Because of their operating characteristics, high pressure float controls cannot be employed in multiple or in parallel with other types of refrigerant flow controls.

17-19. Float Switch. As shown in Fig. 17-34, a float switch can be employed to control the level of liquid in the evaporator. The float switch consists of two principal parts: (1) a float chamber equipped with a ball float which rises and falls with the liquid level in the evaporator and float chamber and (2) a mercury switch which is actuated by the ball float to open and close a liquid line solenoid valve when the level of liquid in the evaporator falls and rises, respectively. A hand expansion valve is installed in the liquid line between the solenoid valve and the evaporator to throttle the liquid refrigerant and prevent surging in the evaporator from the sudden inrush of liquid when the liquid line solenoid opens and to eliminate short cycling of the solenoid valve.

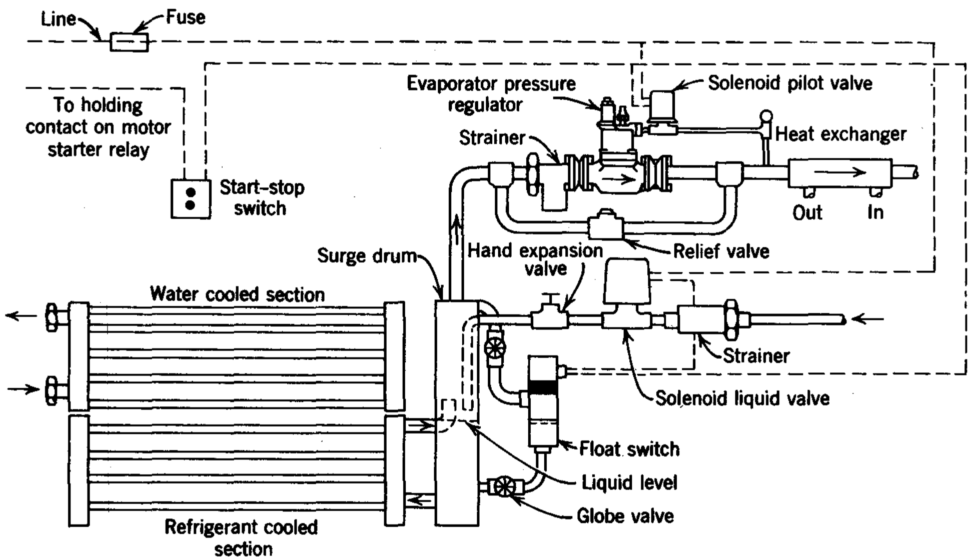


Fig. 17-34. Baudelot cooler with float switch, solenoid liquid valve, evaporator pressure regulator, and solenoid pilot valve. (Courtesy Alco Valve Company.)

Float switches have many applications for operating electrical devices associated with the refrigerating system. They may be arranged for

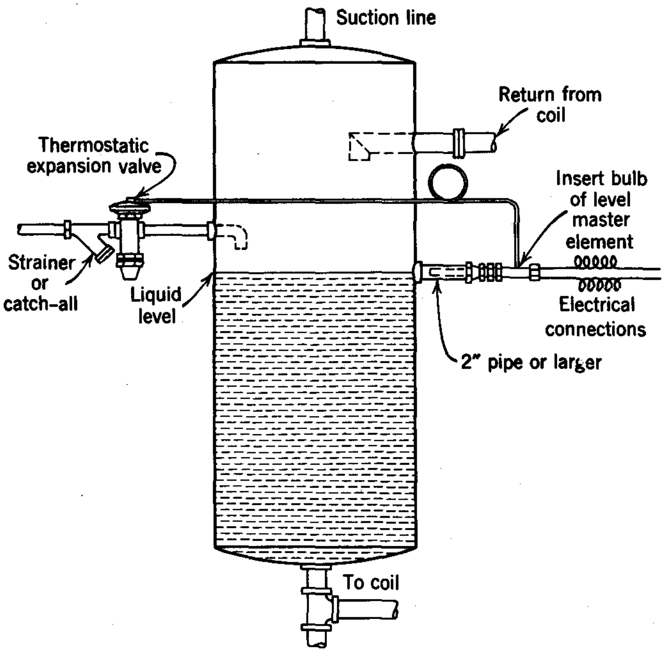


Fig. 17-35. Thermostatic expansion valve used as liquid levelcontrol. (Courtesy Sporlan Valve Company.)

reverse action (close on rise) by employing a reverse acting switch.

17-20. Liquid Level Control with Thermostatic Expansion Valve. A thermostatic expansion valve with a specially designed thermal element can also be used to control the liquid level in flooded-type evaporators. Figure 17-35 illustrates a typical installation on a vertical surge drum.

The specially designed thermal element is an insert bulb consisting of a low wattage electric heating element (approximately 15 watts) and a reservoir for the thermostatic charge. The heating element of the thermal bulb is a means of providing an artificial superheat to the thermostatic charge, which increases the bulb pressure and results in opening the port of the expansion valve allowing more refrigerant to be

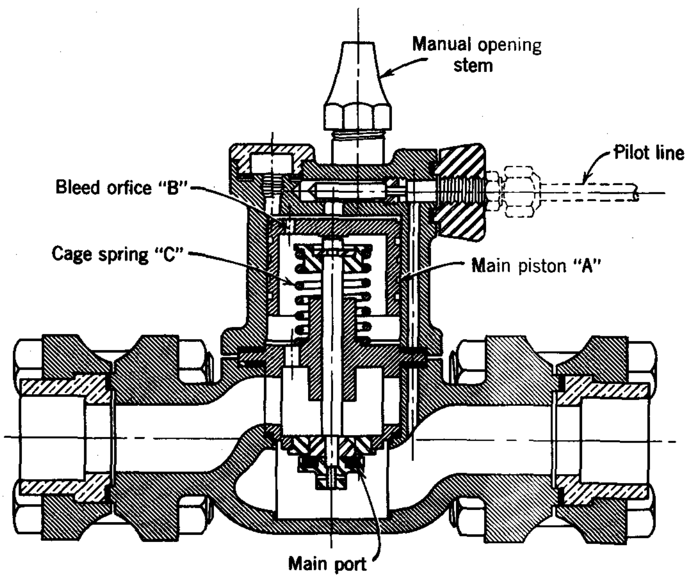


Fig. 17-36. Pilot-operated expansion valve—main regulator. (Courtesy Alco Valve Company.)

fed to the evaporator. As the liquid level in the evaporator rises and more liquid comes in contact with the bulb the effect of the heater element is overcome, thereby decreasing the superheat and allowing the thermostatic expansion valve to throttle to the point of equilibrium or eventual shut-off.

The thermostatic expansion valve is installed in the liquid line and may be arranged to feed liquid directly into the evaporator or accumulator (surge tank), into an accumulator drop leg, or into a coil header.

17-21. Pilot Control Valves. Pilot operated liquid control valves are employed on large tonnage installations. The pilot valve actuating the liquid control valve is usually a thermostatic expansion valve, a low pressure float valve, or a high pressure float valve. A liquid control valve designed for use with a thermostatic expansion valve pilot is illustrated in Fig. 17-36. The liquid control valve opens when pressure is supplied to the top of piston "A" from the pilot line. The small bleeder port "B" in the top of the piston vents this pressure to the outlet (evaporator) side of the liquid control valve. When the pressure supply to the top of the piston is cut off, the cage spring "C" closes the liquid control valve.

Figure 17-37 illustrates a pilot operated expansion valve installed on a direct expansion shell and tube chiller. The externally equalized thermostatic pilot valve supplies pressure to the top of the piston in response to changes in the temperature and pressure of the suction vapor. When the superheat in the suction vapor increases, indicating the need of greater refrigerant flow, the pilot thermo valve moves in an opening direction and supplies a greater pressure to the top of the piston thereby moving the piston in an opening direction and providing a greater flow of refrigerant. Conversely, when the suction vapor superheat decreases, indicating the need of a reduction in refrigerant flow, the pilot valve moves in a closing direction. This provides less pressure on top of the piston, permitting the piston to move in a closing direction and provide a smaller flow. In operation, the pilot valve and the main piston assume intermediate or throttling positions depending on the load.

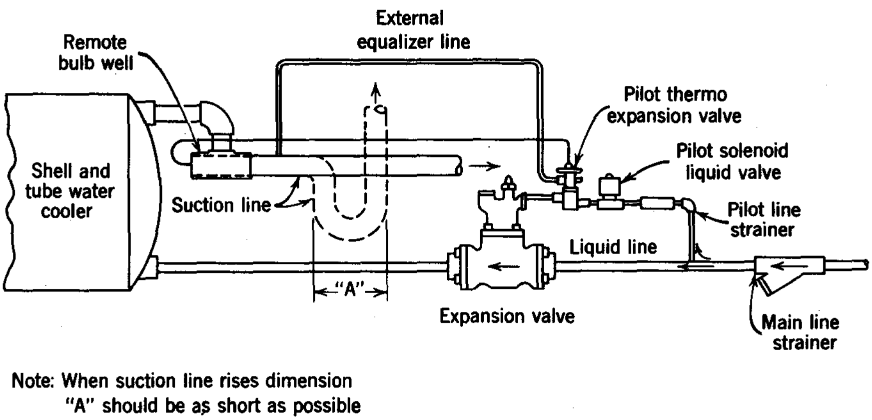


Fig. 17-37. Pilot-operated thermo expansion valve on shell-and-tube-water cooler with refrigerant in the tubes. (Courtesy Alco Valve Company.)

A liquid control valve designed for use with a high pressure float valve as a pilot is illustrated in Fig. 17-38. A system employing this type of refrigerant control is illustrated in Fig. 17-39. Operation of the high pressure float pilot is similar to that of the thermo expansion valve pilot. On a rise in the level in the pilot receiver, the float opens and admits high pressure liquid through the pilot line to the liquid control valve piston. This pressure acts against a spring and opens the valve stem to admit liquid to the evaporator. As the level in the pilot receiver descends, the pilot valve closes and the high pressure in the pilot line is bled off to the low

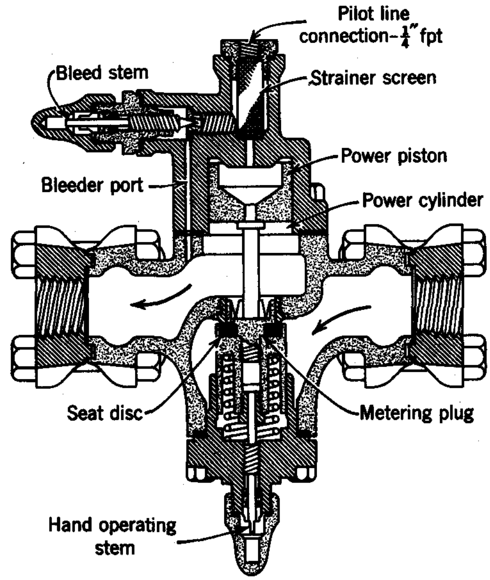


Fig. 17-38. Liquid control valve—high pressure. (Courtesy York Corporation.)

side through an adjustable internal bleeder port in the liquid control valve. A pressure gage should be installed in the pilot line to facilitate adjusting the control valve during the initial start-up.

A liquid control valve designed for use with a low pressure float pilot is shown in Fig. 17-40. A typical application of a low pressure float pilot is shown in Fig. 17-41. As the liquid level in the cooler drops, the low pressure float pilot opens and allows the pressure in the pilot line to be relieved to the cooler so that the high pressure liquid acting on the bottom of the liquid control valve piston can lift the piston and admit more refrigerant to the cooler. As the level in the cooler builds up, the pilot float closes and high pressure liquid is bled into the area above the control piston through an internal bleeder in the bottom of the piston. The resulting high pressure on top of the piston causes the piston to drop down and close the main valve port. The latter bleeder is not adjustable, but modulation can be obtained by adjusting the pressure on the spring above the control piston. A gage is installed in the pilot line to aid in the adjustment of the valve at initial start-up.

17-22. Solenoid Valves. Solenoid valves are widely used in refrigerant, water, and brine lines in place of manual stop valves in order to provide automatic operation. A few of their many functions in the refrigerating system are described at appropriate places in this book.

A solenoid valve is simply an electrically operated valve which consists essentially of a coil of insulated copper wire and an iron core or armature (sometimes called a plunger) which is drawn into the center of the coil magnetic field when the coil is energized. By attaching a valve stem and pin to the coil armature, a valve port can be opened and closed as the coil is energized and de-energized, respectively.

Although there are a number of mechanical variations, solenoid valves are of two principal types: (1) direct acting and (2) pilot operated. Small solenoid valves are usually direct acting (Fig. 17-42), whereas the larger valves are pilot operated (Fig. 17-43). In the direct acting valve, the valve stem attached to the coil armature controls the main valve port directly. In the pilot operated type, the coil armature controls only the pilot port rather than the main valve port. When the coil is energized, the armature is drawn into the coil magnetic field and the pilot port *A* is opened. This releases the pressure on top of the floating main piston *B* through the open pilot port, thereby causing a pressure unbalance across the piston. The higher pressure under the piston forces the piston to move upward, opening the main valve port *C*. When the coil is de-energized, the armature drops out of the coil magnetic field and closes the pilot port. The pressure immediately builds up on top of the main piston, causing the piston to drop and close off the main valve port.

Except where the solenoid valve is specially designed for horizontal installation, the solenoid valve must always be mounted in a vertical position with the coil on top.

In selecting a solenoid valve, the size of the valve is determined by the desired flow rate through the valve and never by the size of the line in which the valve is to be installed. Consideration must also be given to the

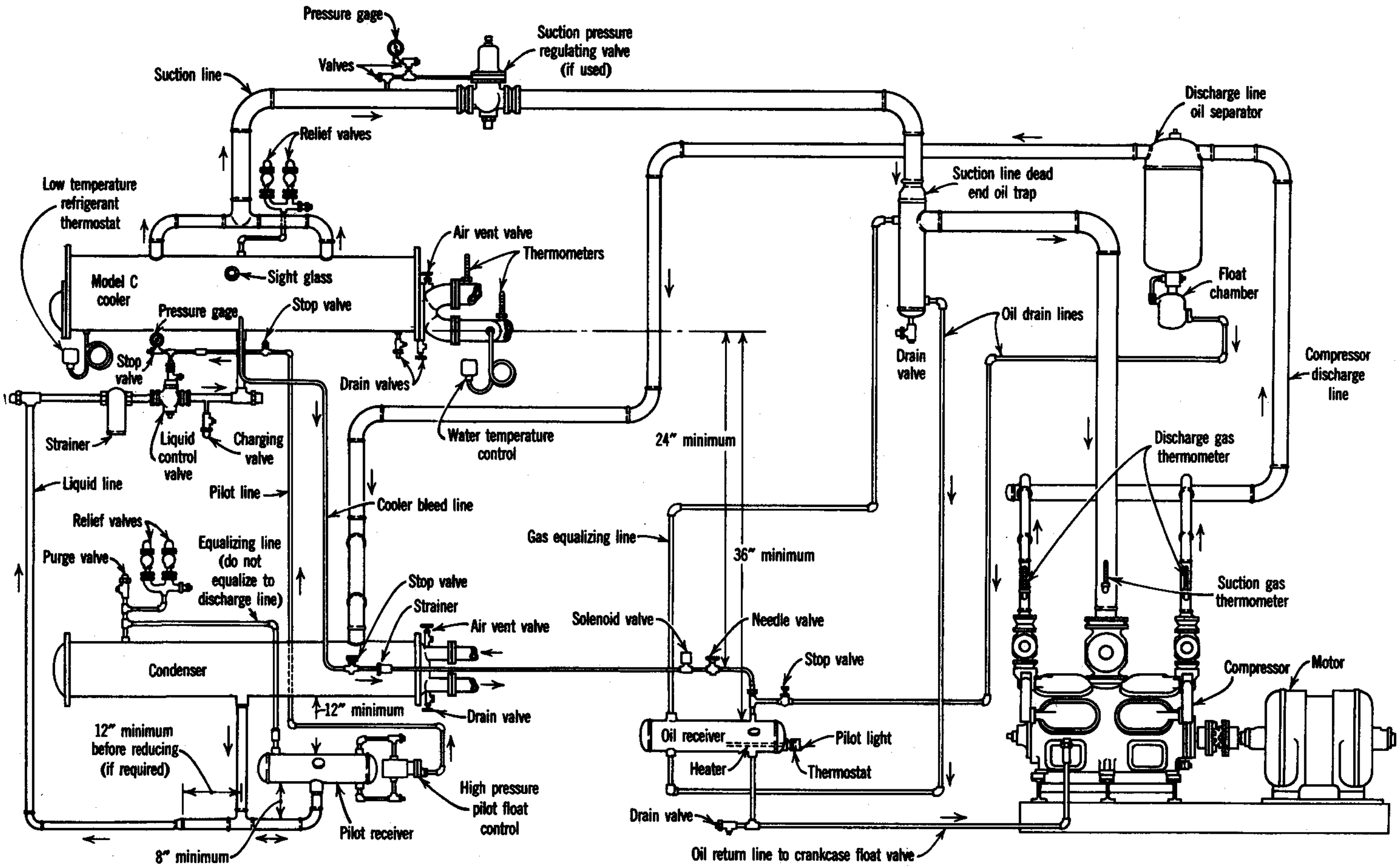


Fig. 17-39. Application of high pressure float pilot control valve. (Courtesy York Corporation.)

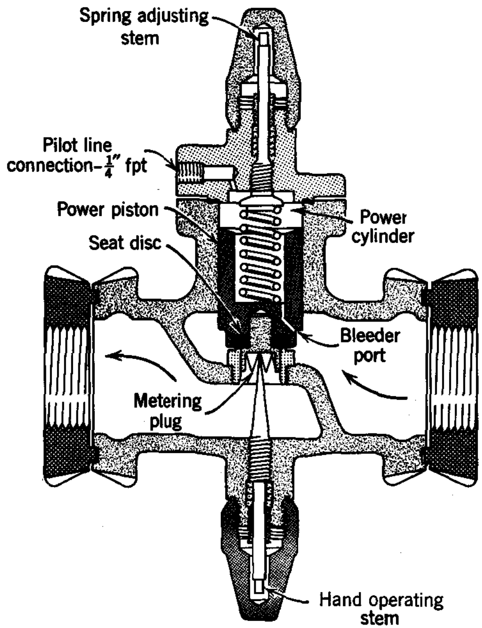


Fig. 17-40. Liquid control valve—low pressure. (Courtesy York Corporation.)

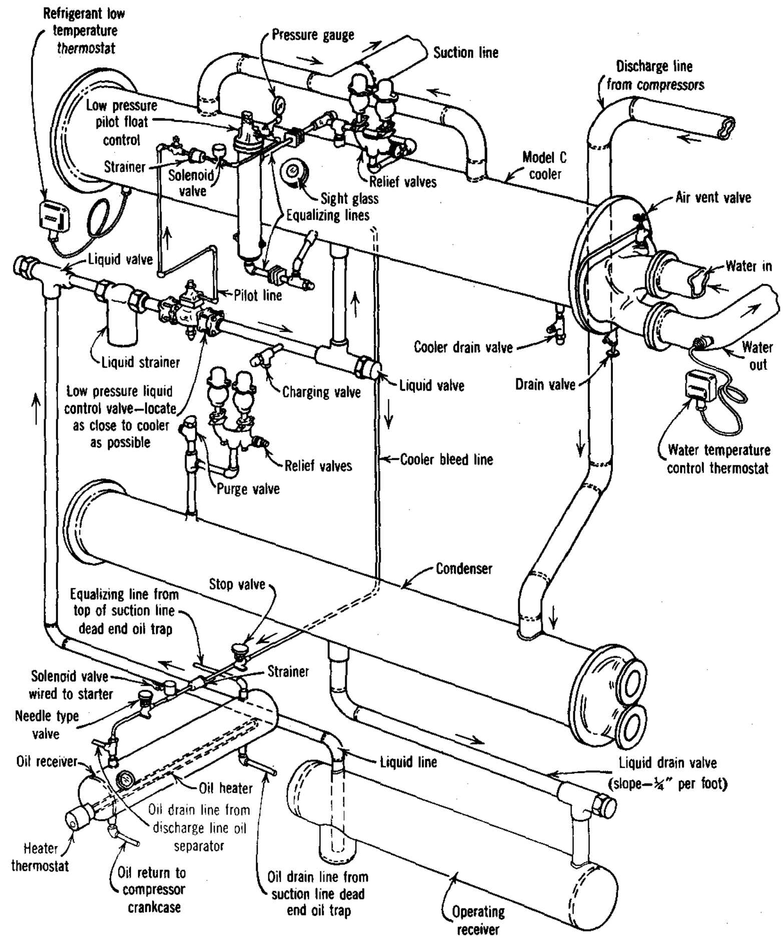


Fig. 17-41. Typical system—low pressure float control. (Courtesy York Corporation.)

maximum allowable pressure difference across the valve and to the pressure drop through the valve.

17-23. Suction Line Controls. Suction line controls are of two general types: (1) evaporator pressure regulators and (2) suction pressure regulators.

The function of the evaporator pressure regulator is to prevent the evaporator pressure, and therefore the evaporator temperature, from dropping below a certain predetermined minimum, regardless of how low the pressure in the suction line may drop because of the action of the compressor. It is important to recognize that the evaporator pressure regulator does not maintain a constant pressure in the evaporator but merely limits the minimum evaporator pressure. Evaporator pressure regulators are available with either throttling action (modulating) or snap-action (fully open or fully closed). The differential between the closing and opening points of the snap-action control not only gives close control of the product temperature but also provides for automatic defrosting of air-cooling evaporators when the temperatures in the space are sufficiently high to permit off-cycle defrosting.

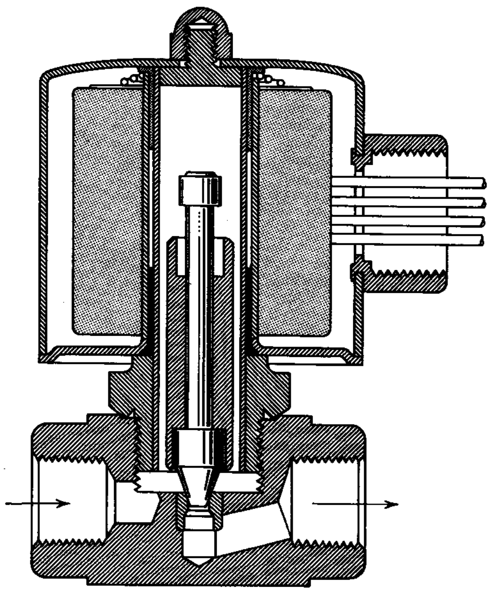


Fig. 17-42. Small, direct-acting solenoid valve. (Courtesy Sporlan Valve Company.)

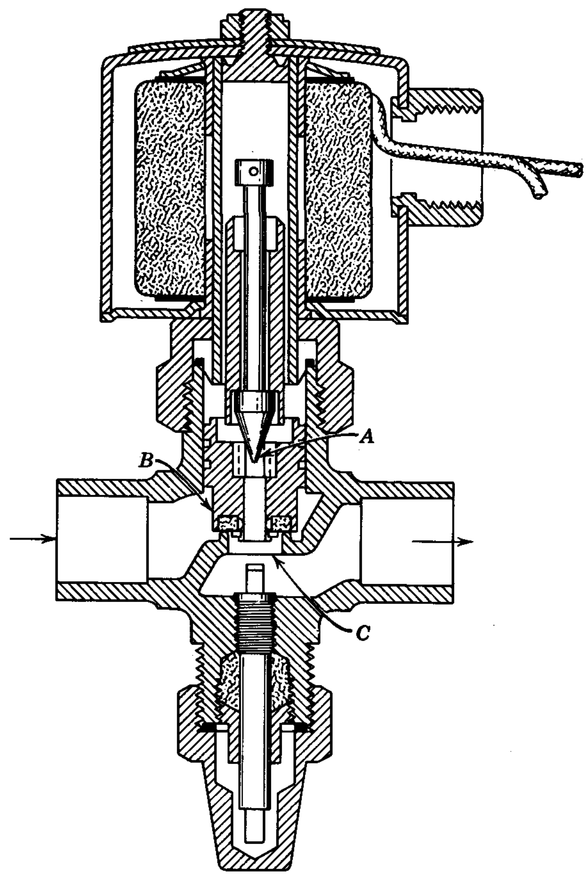


Fig. 17-43. Pilot-operated solenoid valve of the floating piston type. (Courtesy Sporlan Valve Company.)

The throttling-type evaporator pressure regulator (Fig. 17-44) is never fully closed while the compressor is operating. As the load on the evaporator decreases and the evaporator

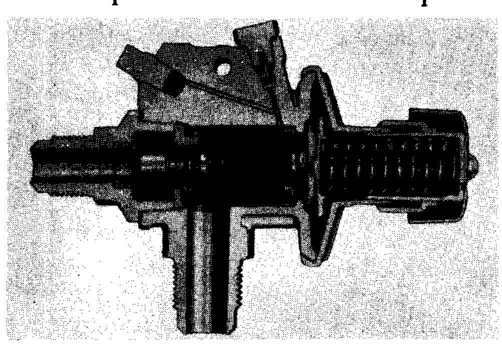


Fig. 17-44. Throttling-type evaporator pressure regulator. (Courtesy Controls Company of America.)

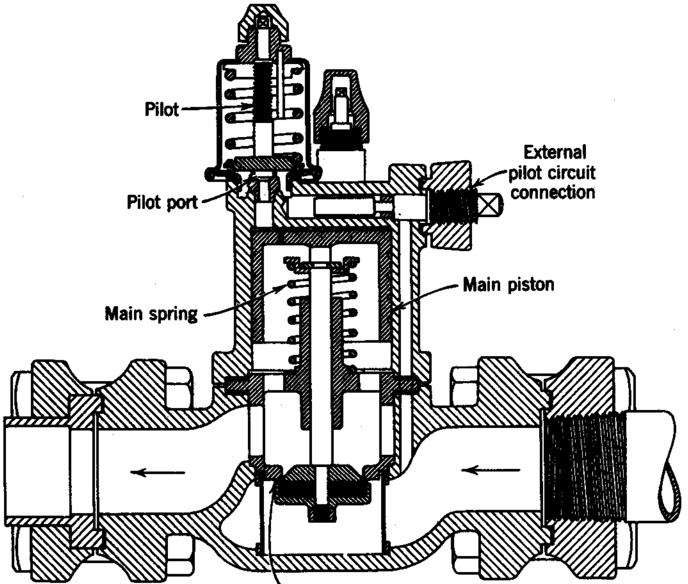


Fig. 17-45. Pilot-operated evaporator pressure regulator. (Courtesy Alco Valve Company.)

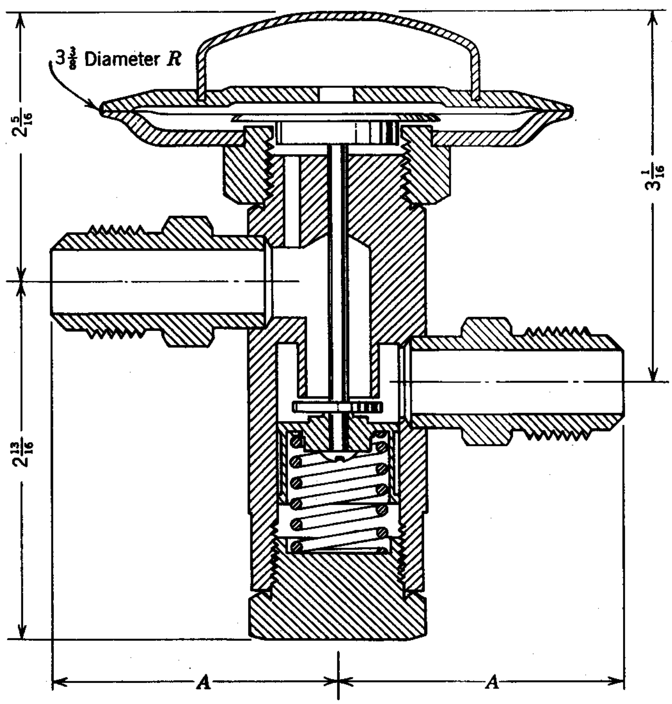


Fig. 17-46. Crankcase pressure regulator. (Courtesy Sporlan Valve Company.)

pressure tends to fall below the preset minimum regulator pressure, the regulator modulates toward the closed position to throttle the suction vapor to the compressor, thereby maintaining the evaporator pressure above the desired minimum. As the load on the evaporator increases and the evaporator pressure rises above the regulator setting, the regulator modulates toward the open position so that at full load the regulator is in the full open position.

Evaporator pressure regulators can be used in any installation when the evaporator pressure or temperature must be maintained above a certain minimum. They are widely used with water and brine chillers in order to prevent freeze-ups during periods of minimum loading. They are also used frequently in air-cooling applications where proper humidity control prescribes a minimum evaporator temperature. In multiple evaporator systems where the evaporators are all operated at approximately the same temperature, a single evaporator pressure regulator can be installed in the suction main to control the pressure in all the evaporators. On the other hand, where a multiple of evaporators connected to a single compressor are operated at different temperatures, a separate evaporator pressure regulator must be installed in the suction line of each of the higher temperature evaporators (see Section 20-16). This arrangement prevents the pressures in the warmer evaporators from dropping below the desired minimum while the compressor continues to operate to satisfy the coldest evaporator.

In large sizes, evaporator pressure regulators are pilot operated. The valve shown in Fig. 17-45 is designed for either internal or external pilot control. With internal pilot control, operation of the regulator is similar to that of the pilot solenoid described in the previous section. With external pilot control, operation of the regulator is similar to that of the pilot operated liquid control valve described in Section 17-21.

It is of interest to notice that the solenoid pilot shown in Fig. 17-34 has nothing to do with the pressure regulating function of the evaporator pressure regulator. The use of the solenoid pilot permits the evaporator pressure to serve also as a suction stop valve.

The function of the suction pressure regulator (Fig. 17-46), sometimes called a "crankcase pressure regulator" or a "suction pressure hold-back valve," is to limit the suction pressure at the compressor inlet to a predetermined maximum, regardless of how high the pressure in the evaporator rises because of an increase in the evaporator load. The purpose of the suction pressure regulator is to protect the compressor driver from overload during periods when the evaporator pressure is above the normal operating pressure for which the compressor driver was selected. Suction pressure regulators are recommended for use on any installation where motor protection is desired because the system is subject to:

1. High starting loads.
2. Surges in suction pressure.
3. High suction pressure caused by hot gas defrosting or reverse cycle (heat pump) operation.
4. Prolonged operation at excessive suction pressures.

Like evaporator pressure regulators, suction pressure regulators in large sizes are pilot operated.

18

Compressor Construction and Lubrication

18-1. Types of Compressors. Three types of compressors are commonly used for refrigeration duty: (1) reciprocating, (2) rotary, and (3) centrifugal. The reciprocating and rotary types are positive displacement compressors, compression of the vapor being accomplished mechanically by means of a compressing member. In the reciprocating compressor, the compressing member is a reciprocating piston, whereas in the rotary compressor, the compressing member takes the form of a blade, vane, or roller. The centrifugal compressor, on the other hand, has no compressing member, compression of the vapor being accomplished primarily by action of the centrifugal force which is developed as the vapor is rotated by a high speed impeller.

All three compressor types have certain advantages in their own field of use. For the most part, the type of compressor employed in any individual application depends on the size and nature of the installation and on the refrigerant used.

18-2. Reciprocating Compressors. The reciprocating compressor is by far the most widely used type, being employed in all fields of refrigeration. It is especially adaptable for use with refrigerants requiring relatively small displacement and condensing at relatively high pressures. Among the refrigerants used extensively with reciprocating compressors are Refrigerants 12, 22, 500 (Carrene 7), and 717 (ammonia).

As a general rule, because of limited valve areas, reciprocating compressors cannot be employed economically with low pressure refrigerants which require a large volumetric displacement per ton of capacity. Although best applied to systems having evaporator pressures above one atmosphere and relatively high condensing pressures, reciprocating compressors have also been used very successfully in both low temperature and ultra-low temperature installations.

Reciprocating compressors are available in sizes ranging from $\frac{1}{8}$ hp in small domestic units up through 100 hp or more in large industrial installations. The fact that reciprocating compressors can be manufactured economically in a wide range of sizes and designs, when considered along with its durability and its high efficiency under a wide variety of operating conditions, accounts for its widespread popularity in the refrigeration field.

Reciprocating compressors are of two basic types: (1) single-acting, vertical, enclosed compressors and (2) double-acting, horizontal compressors employing crossheads and piston rods. In single-acting compressors, compression of the vapor occurs only on one side of the piston and only once during each revolution of the crankshaft, whereas in double-acting compressors, compression of the vapor occurs alternately on both sides of the piston so that compression occurs twice during each revolution of the crankshaft.

Vertical compressors are usually of the enclosed type wherein the piston is driven directly by a connecting rod working off the crankshaft, both connecting rod and crankshaft being enclosed in a crankcase which is pressure tight to the outside, but open to contact with the system refrigerant (Fig. 18-1). Horizontal compressors, on the other hand, usually employ crankcases which are open (vented) to the outside, but isolated from the system refrigerant, in which case the piston is driven by a piston rod connected to a crosshead, which in turn, is actuated by a connecting rod working off the crankshaft (Fig. 18-2).

Because of its design, the horizontal compressor is obviously not practical in small sizes

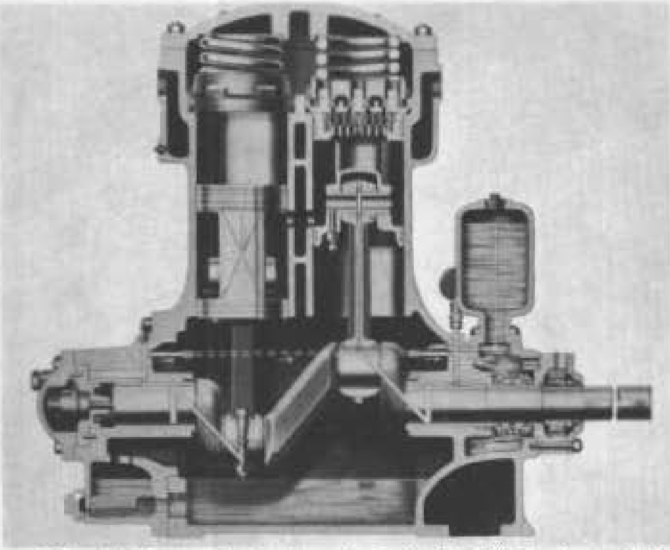


Fig. 18-1. Large capacity, reciprocating compressor showing details of lubrication system. (Courtesy Vilter Manufacturing Company.)

and therefore is limited to the larger industrial applications. As compared to the vertical type, the horizontal compressor requires more floor space, but less head room. Also, while it is more expensive than the vertical type, it is also more accessible for maintenance since the crankcase is not exposed to the system refrigerant. The principal disadvantage of the horizontal compressor is that the packing or seal around the piston rod is subject to both the suction and discharge pressures, whereas in the vertical type compressor, the packing or

Fig. 18-2. Donble-acting, horizontal compressor. Cylinder clearance can be adjusted manually to obtain capacity control. (Courtesy Worthington Corporation.)

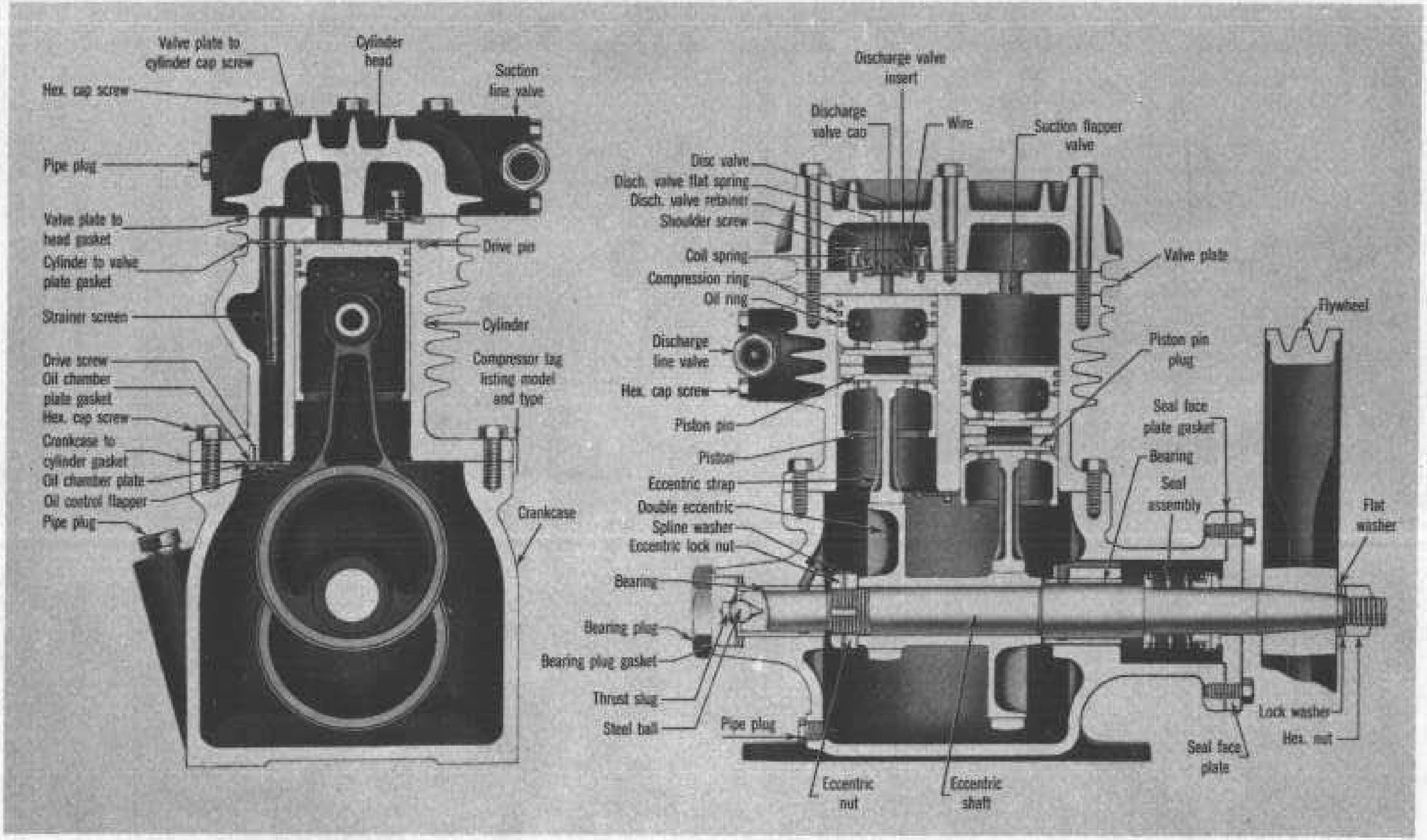


Fig. 18-3. Typical cross section of two cylinder, eccentric-type refrigeration compressor. (Courtesy Dunham-Bush, Inc.)

seal around the crankshaft is subject only to the suction pressure. This disadvantage is made more serious because it is usually more difficult to maintain a pressure tight seal around the reciprocating piston rod of the horizontal compressor than it is around the rotating shaft of the vertical type compressor.

Vertical, single-acting reciprocating compressors differ considerably in design according to the type of duty for which they are intended. Numerous combinations of the following design features are used in order to obtain the desired flexibility: (1) the number and arrangement of the cylinders, (2) type of pistons, (3) type and arrangement of valves, (4) crank and piston speeds, (5) bore and stroke, (6) type of crankshaft, (7) method of lubrication, etc.

18-3. Cylinders. The number of cylinders varies from as few as one to as many as sixteen. In multicylinder compressors, the cylinders may be arranged in line, radially, or at an angle to each other to form a *V* or *W* pattern. For two and three cylinder compressors the cylinders are usually arranged in line. Where four or more cylinders are employed, *V*, *W*, or radial arrangements are ordinarily used. In-line arrangements have the advantage of requiring only a single valve plate, whereas *V*, *W*, and radial arrangements provide better running balance and permit the cylinders to be staggered so that the over-all compressor length is less.

Compressor cylinders are usually constructed of close-grained cast iron which is easily machined and not subject to warping. For small compressors, the cylinders and crankcase housing are often cast in one piece, a practice which permits very close alignment of the working parts. For larger compressors, the cylinders and crankcase housing are usually cast separately and flanged and bolted together. As a general rule, the cylinders of the larger compressors are usually equipped with replaceable liners or sleeves.

Small compressors often have fins cast integral with the cylinders and cylinder head to increase cylinder cooling, whereas cylinder castings for larger compressors frequently contain water jackets for this purpose.

18-4. Pistons. Pistons employed in refrigeration compressors are of two common types: (1) automotive and (2) double-trunk. For the most part, the type of piston used depends on the method of suction gas intake and on the location of the suction valves. Automotive-type pistons are used when the suction gas enters the cylinder through suction valves located in the cylinder head (valve plate) as shown in Fig. 18-3. Double-trunk pistons are ordinarily used in medium and large compressors, in which case the suction gas enters through ports in the cylinder wall and in the side of the piston and passes into the cylinder through suction valves located in the top of the piston (Fig. 18-1). Notice that the bottom of the piston contains a bulkhead that seals off the hollow portion of the piston from the crankcase.

Because of small piston clearances (approximately 0.003 in. per inch of cylinder diameter), the oil film on the cylinder wall is usually sufficient to prevent gas blow-by around the pistons in small compressors. For this reason, rings are seldom used on pistons less than 2 in. in diameter. However, these pistons are provided with oil grooves to facilitate lubrication of the cylinder walls. Automotive-type pistons having diameters above 2 in. are usually equipped with two compression rings and one oil ring, the latter sometimes being located at the bottom of the piston. Double-trunk pistons are equipped with from one to three compression rings at the top and one or two oil rings at the bottom.

As a general rule, pistons are manufactured from close-grained cast iron, as are the rings. However, a number of aluminum pistons are in use. The use of cast iron permits closer tolerances. When aluminum pistons are used, they are usually equipped with at least one compression ring.

18-5. Suction and Discharge Valves. Since it influences to a greater or lesser degree all the factors which determine both the volumetric and compression efficiencies of the compressor, the design of the compressor suction and discharge valves is one of the most important considerations in compressor design. Furthermore, it will be shown later that valve design determines to a large extent the over-all design of the compressor.

The friction loss (wiredrawing effect) suffered by the vapor in flowing through the compressor valves and passages is primarily a function of vapor velocity and increases as the velocity of the vapor increases. Therefore, in order to minimize the wiredrawing losses, the valves should be designed to provide the largest possible restricted area (opening) and to open with the least possible effort. Too, whenever practical, the valves should be so located as to provide for straight-line flow (uniflow) of the vapor through the compressor valves and passages. In all cases, the valve openings must be sufficiently large to maintain vapor velocities within the maximum limits. The maximum permissible vapor velocity may be defined as that velocity beyond which the increase in the wiredrawing effect will produce a marked reduction in the volumetric efficiency of the compressor and/or a material increase in the power requirements of the compressor.

To minimize back leakage of the vapor through the valves, the valves should be designed to close quickly and tightly. In order to open easily and close quickly, the valves should be constructed of lightweight material and be designed for a low lift. They should be strong and durable and they should operate quietly and automatically. Furthermore, they should be so designed and placed that they do not increase the clearance volume of the compressor.

Although there are numerous modifications within any one type, valves employed in refrigeration compressors can be grouped into three basic types: (1) the poppet, (2) the ring plate, (3) the flexing or reed. All three types operate automatically, opening and closing in response to pressure differentials caused by changes in the cylinder pressure. To facilitate rapid closing of the valve, most discharge valves and some suction valves are spring loaded.

18-6. Poppet Valves. The poppet valve is similar to the automotive valve, except that the valve stem is much shorter. The valve is enclosed in a cage which serves both as a valve seat and valve stem guide and also as a retainer for the valve spring. A spring, dashpot, or bleeder arrangement is also included in the assembly to cushion and limit the valve travel. Except for minor differences, the design of the suction and discharge poppet valves is essentially the same, the principal difference being that the suction poppet valve is beveled on the stem side of the valve face, whereas the discharge poppet valve is beveled on the opposite side.

The poppet valve, one of the first types used in refrigeration compressors, is essentially a slow speed valve and as such is limited at the present time to a few types of slow speed compressors. In high speed machines, the poppet valve has been discarded in favor of either the ring plate valve or the flexing valve, both of which are more adaptable to high speed operation than is the poppet valve. The principal advantage of the poppet valve is that

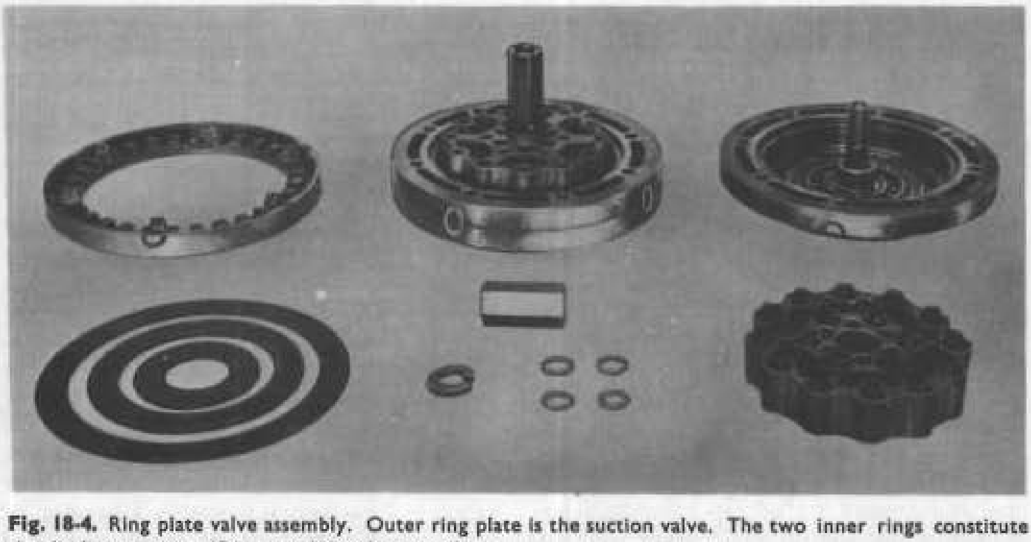


Fig. 18-4. Ring plate valve assembly. Outer ring plate is the suction valve. The two inner rings constitute the discharge valve. (Courtesy Frick Company.)

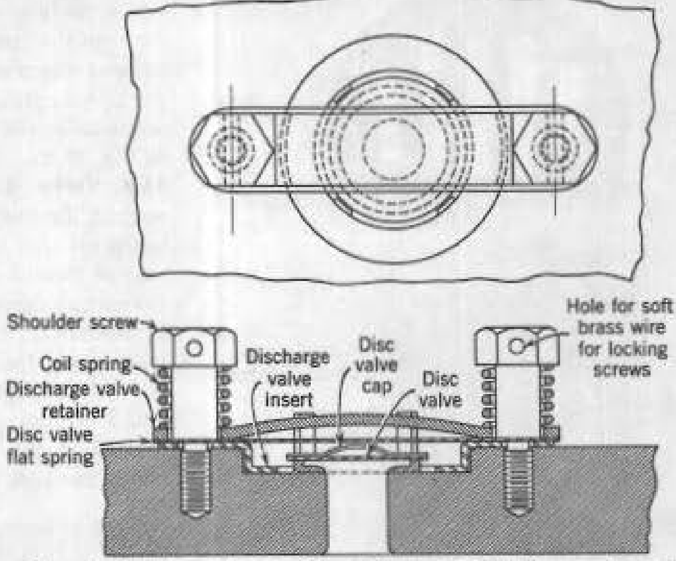


Fig 18-5. Discharge valve assembly (disc valve). (From the *ASRE Data Book*, Design Volume, 1957–58 Edition. Reproduced by permission of the American Society of Heating, Refrigerating and Air-Conditioning Engineers)

it can be mounted flush and therefore does not increase the clearance volume of the compressor.

18-7. Ring Plate Valves. The ring plate valve (Fig. 18-4) consists of a valve seat, one or more ring plates, one or more valve springs, and a retainer. The ring plates are held firmly against the valve seat by the valve springs, which also help to provide rapid closure of the valves. The function of the retainer is to hold the valve springs in place and to limit the valve lift.

The ring plate valve is suitable for use in both slow speed and high speed compressors and it may be used as either the suction or the discharge valve. In fact, when both suction and discharge valves are located in the head, they are usually contained in the same ring plate assembly. For example, in Fig. 18-4, the outer ring serves as the suction valve, whereas the two smaller rings serve as the discharge valve.

One modification of the ring plate valve is the disc valve (Fig. 18-5), which is simply a thin metal disc held in place on the valve seat by a retainer.

18-8. Flexing Valves. Flexing valves vary in individual design to a much greater extent than do either the poppet or ring plate-types. One popular type of flexing valve suitable for use in medium and large compressors is the Feather valve* (Fig. 18-6), which consists of a valve seat, a series of ribbon steel strips, and a valve guard or retainer. The flexible metal strips fit over slots in the valve seat and are held in place by the valve guard. The operation of the Feather valve is illustrated in Fig. 18-7. It is important to notice that in order to allow the valve reeds to flex under pressure, they are not tightly secured at either end. The principal advantage of the Feather valve is that the reeds are lightweight and easily opened and are so designed that they provide a large restricted area, all of which tend to reduce the wiredrawing effect to a minimum.

One disadvantage of all flexing valves, and one that is shared by the ring plate type, is that they cannot be mounted flush as can the poppet valve. Because of the presence of the valve port spaces, the clearance volume is necessarily increased in all compressors employing either ring plate or flexing valves of any design.

A flexing valve design widely used in smaller compressors is the flapper valve, of which there are enumerable variations. The flapper valve is a thin steel reed, which is usually fastened securely at one end while the opposite unfastened end rests on the valve seat over the valve port. The free end of the reed flexes or "flaps" to cover and uncover the valve port (Fig. 18-8).

A flapper valve design frequently employed in discharge valves, called a "beam" valve, is shown in Fig. 18-9. The valve reed is held in place over the valve port by a spring-loaded

* A proprietary design of the Worthington Pump and Machinery Company.

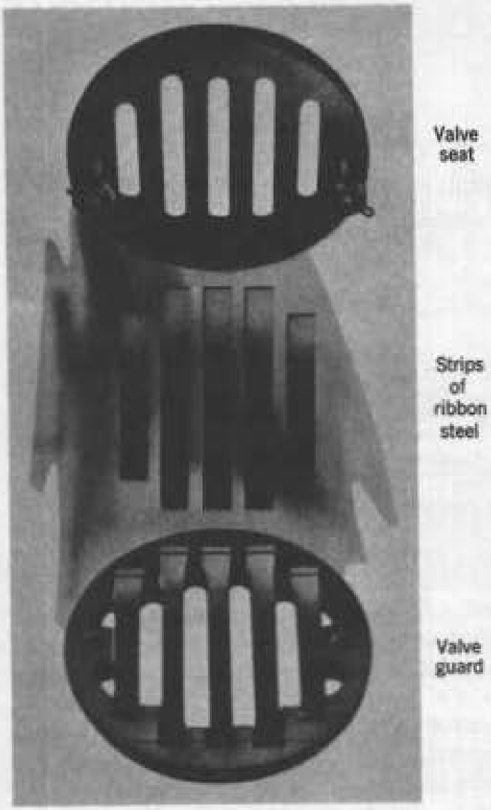


Fig. 18-6. One popular design of flexing valve. (Courtesy Worthington Corporation.)

beam which is arched in the center to permit the reed to flex upward at this point. The ends of the reed are slotted and are held down by only the tension of the coil springs in order to allow the ends of the reed to move as the reed flexes up and down at the center. The spring-loaded beam also acts as a safety device to protect the compressor against damage in the event that a slug of liquid refrigerant or oil enters the valve port. Since the valves are designed to handle vapor, there is not usually sufficient clearance to pass a slug of liquid of any kind. However, with the arrangement in Fig. 18-9, the whole valve assembly will lift to pass liquid slugs. Under ordinary discharge pressures, the tension of the springs is ample to hold the beam down firmly on the ends of the reed.

Another type of flexing valve in common use is the diaphragm valve. The diaphragm valve consists of a flexible metal disc which is held down on the valve seat by a screw or bolt through the center of the disc. The disc flexes up and down to uncover and cover the valve port. A diaphragm valve used as a suction valve mounted in the crown of a piston is illustrated in Fig. 18-10.

18-9. Valve Location. As previously described, the discharge valves are usually located in the cylinder head, whereas the suction valves may be located either in the head, in which case the suction vapor enters the cylinder through the cylinder head, or in the crown of the piston, in which case the suction vapor enters through the side of the cylinder. As a general rule, with larger compressors, the suction valves are located in the piston and the suction vapor enters through the cylinder wall. With small

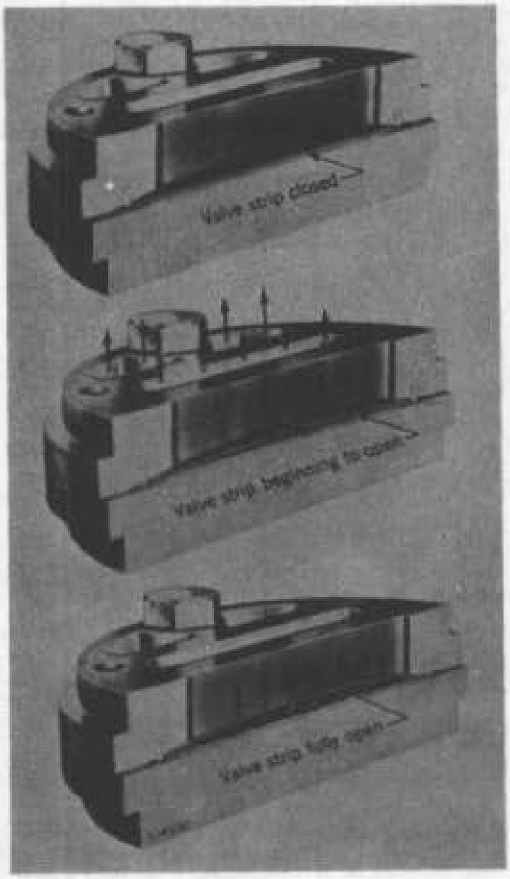


Fig. 18-7. Illustrating the operation of the Worthington Feather valve. (Courtesy Worthington Corporation.)

and medium compressors, the suction valves are usually located in the cylinder head. When both valves are placed in the cylinder head, the head must be partitioned to permit separation of the suction and discharge vapors.

Most large compressors are equipped with secondary safety heads which are located at the end of the cylinder and held in place by heavy coil springs (Fig. 18-1). Under normal discharge pressures, the safety head is held firmly in place by the springs. However, in the event that a slug of liquid or some other noncompressible material enters the cylinder, the safety head will rise under the increased pressure and permit the material to pass into the cylinder head, thereby preventing damage to the compressor. In smaller compressors, the discharge valve is usually designed to provide this protection.

In large compressors, the valves and seats are removable for replacement. In small compressors, the suction and discharge valves are usually incorporated into a valve plate assembly, which is removed and replaced as a unit (see Fig. 18-9).

18-10. Crank and Piston Speeds. In an effort to reduce the size and weight of the compressor, the trend in modern compressor design is toward higher rotational speeds. Since single-cylinder piston displacement is a function of bore, stroke, and rpm, it follows that as the rpm is increased, the bore and stroke can be decreased proportionally without loss of displacement, provided that the volumetric efficiency of the compressor remains the same.

Rotative speeds between 500 and 1750 rpm are quite common, whereas some compressors are being operated successfully at speeds up to 3500 rpm. The maximum rotational speed of the compressor is more or less limited by the maximum allowable piston velocity.

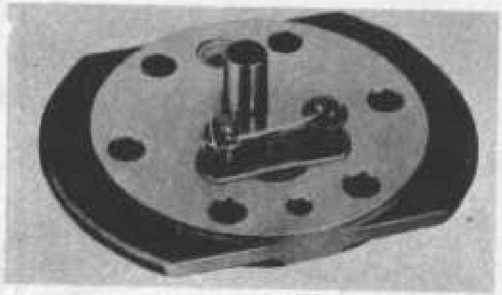


Fig. 18-9. Compressor valve plate assembly. (Courtesy Tecumseh Products Company.)

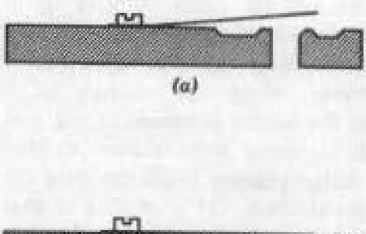


Fig. 18.-8. Flapper-type flexing valve. (*a*) Port open. (*b*) Port closed.

Theoretically, there is no limit to piston speeds. However, as a practical matter, piston speeds are limited to a maximum of approximately 800 fpm, the limiting factor being the available valve area.

Since considerable difficulty is experienced in finding sufficient space in the compressor for valve arrangements, valve areas tend to be somewhat limited. Hence, when piston velocities are increased beyond 800 fpm, the velocity of the vapor through the valves will usually become excessive, with the result that the volumetric efficiency of the compressor is decreased while the power required by the compressor is increased.

Piston velocity is a function of compressor rpm and the length of the piston stroke. The following relationship exists:

$$\text{Piston velocity (fpm)} = \text{rpm} \times \text{stroke (ft)} \times 2 \text{ strokes per revolution}$$

For example, a compressor having a 4 in. stroke and rotating at 1200 rpm will have a piston speed of

$$\frac{1200 \times 4\text{ in.} \times 2}{12} = 800\text{ fpm}$$

If the rotational speed of the compressor is increased to 3600 rpm, in order to maintain a piston velocity of 800 fpm, the length of stroke will have to be reduced to

$$\frac{12 \times 800\text{ fpm}}{3600\text{ rpm} \times 2} = 1.33\text{ in.}$$

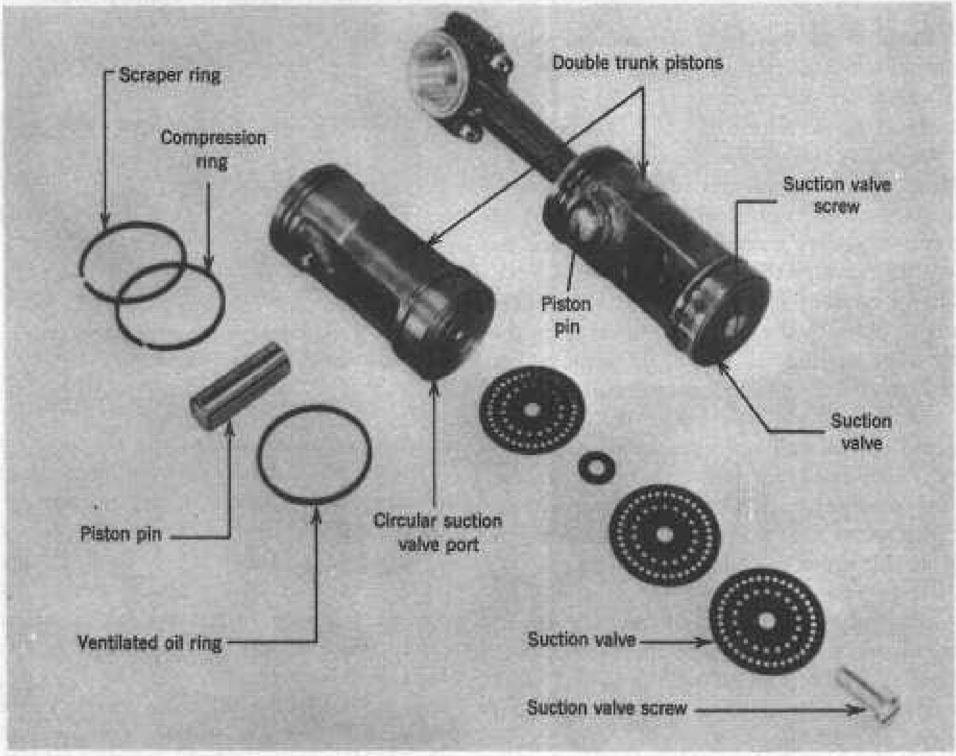


Fig. 18-10. Diaphragm-type suction valve. This same type of valve is often mounted in a valve plate and used as a discharge valve. (Courtesy York Corporation.)

Obviously, then, the maximum speed at which an individual compressor can be rotated without exceeding allowable piston velocities depends upon the length of stroke. The shorter the stroke, the higher is the maximum permissible rpm. This accounts for the fact that the volumetric efficiency of a compressor will usually remain constant or increase slightly as the speed of the compressor is increased up to a certain point beyond which, if the speed is further increased, the efficiency of the compressor will decrease while the power required per ton of refrigerating capacity will be greater (see Section 12-28).

18-11. Bore and Stroke. The relationship of the bore to the stroke differs somewhat with the individual compressor. Although the bore dimension may be either less than or greater than that of the stroke, the general trend in high speed compressors is toward a large bore and a short stroke. When the suction and discharge valves are both located in the head, a large bore is usually required in order to provide sufficient valve area. Too, since the piston stroke and the compressor rpm are both limited somewhat by the maximum allowable piston velocity, it follows that the only practical means of increasing single-cylinder piston displacement is to increase the size of the bore. The increase in piston displacement accruing from an increase in the size of the bore need not increase the vapor velocity through the compressor valves since the increase in the bore also increases the available valve area.

However, since the amount of blow-by around the piston increases as the size of the bore is increased with relation to the stroke, good design practice limits the bore dimension to approximately 125% of that of the stroke. If the bore is increased beyond this point, the blow-by around the piston becomes excessive.

Since compressors working on low temperature or "booster" applications (see Section

20-11) must handle relatively large volumes of vapor per ton of capacity, they are usually designed with a large bore and a short stroke in order to obtain the maximum piston displacement per cylinder.

Cylinder bores range from approximately 1 in. in small domestic compressors up to approximately 18 in. in some of the large industrial types.

18-12. Cranks, Rods, and Bearings. Crankshafts employed in large compressors are of the crank-throw type and are usually constructed of forged steel or alloy cast iron. All bearing-journals are highly polished and are usually case-hardened, particularly where brass or aluminum bearings are used. As a general rule, the crankshaft has a standard taper on the flywheel end, the flywheel being fastened to the crankshaft with one or more woodruff keys and a locknut arrangement. Crankshaft bearings are usually of the sleeve type, although antifriction (roller or ball) bearings are sometimes used for the mains. Common bearing materials are bronze, aluminum, and babbit.

The eccentric-type shaft, which consists of a cast iron eccentric mounted on a straight steel shaft (see Fig. 18-3) is often used in smaller compressors. The eccentric is counterbalanced and is fastened to the shaft by a key-and-lock screw arrangement. Since the bearing of the connecting rod completely encircles the eccentric, the entire eccentric acts as a bearing surface.

Connecting rods are constructed of bronze, aluminum, forged steel, or cast iron. Wrist pins are usually case-hardened steel. Wrist-pins bearings are generally of the sleeve type made of bronze and pressed into the rod. Bronze, aluminum, and cast iron are often used without bearings, in which case the shaft is usually case-hardened.

18-13. Crankshaft Seals. In order to prevent the leakage of refrigerant and oil from the crankcase (or the leakage of air into the crankcase in the event that the pressure in the crankcase is below atmospheric), a seal or packing must be provided at the point where the crankshaft passes through the crankcase. One of the oldest methods of sealing the crankshaft, still employed on some large ammonia compressors, is through the use of a stuffing box (Fig. 18-2). The stuffing box is a cylindrical housing which is cast as an integral part of the crankcase where the shaft emerges, and which is bored to an inside diameter somewhat larger than the diameter of the crankshaft. A series of packing rings, placed over the shaft and inserted into the stuffing box, fills the space in the stuffing box between the shaft and the stuffing box housing. The packing is held in place by a threaded gland nut which, when tightened, causes the packing rings to swell and press tightly against the shaft and housing, thereby affecting a vapor tight seal between the two. Because of the pressure of the rings against the rotating shaft, the rings will eventually wear and permit refrigerant to seep around the shaft, whereupon the packing gland nut must be tightened again to reestablish a tight seal. Although the stuffing box seal is satisfactory for large ammonia installations where an operator is on duty to tighten the packing gland as the occasion requires, they are not suitable for small compressors or for large compressors designed for automatic operation.

A crankshaft seal suitable for use on automatic equipment must be self-adjusting to compensate for wear and for varying crankcase pressures. It must not leak under pressure or vacuum when the shaft is rotating or idle. It must be self-lubricating, have a reasonably long life, and be easily replaceable in the field. Although there are a number of different seal designs which meet these qualifications and which are in use at the present time, one relatively simple design of crankshaft seal, which is rapidly gaining in popularity, is shown in Fig. 18-11. The seal consists essentially of a spring-loaded bronze or hard carbon seal nose which is sealed to the crankshaft with a synthetic rubber gasket. The spring holds the seal nose firmly against a highly polished steel seal face which is a part of the seal plate. An oil film between these two smooth surfaces form an effective vapor-tight seal. Notice that sealing occurs in three places: (1) at the rubber gasket between the seal nose and the crankshaft, (2) between the seal nose and seal face, and (3) at the gasket between the seal plate and the crankcase housing.

The compressor shown in Fig. 18-1 employs a double shaft seal. Notice that the seal

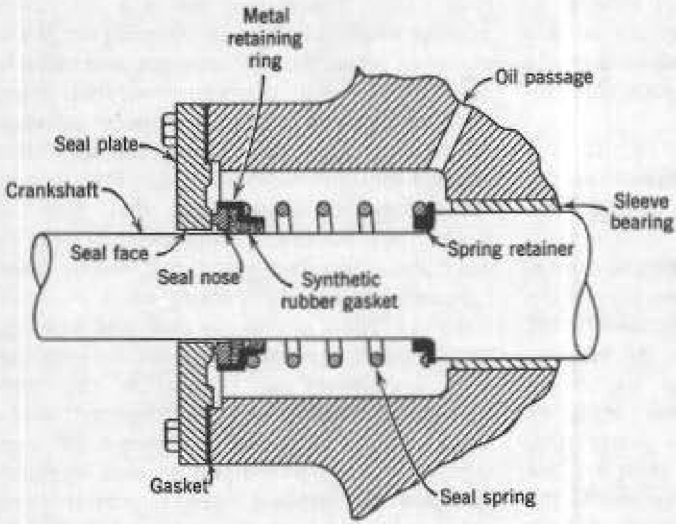


Fig. 18-11. Crankshaft seal.

remains completely submerged in oil during both the running and off cycles.

18-14. Compressor Lubricating Oils. The fact that the compressor lubricating oil usually comes into contact with, and often mixes with, the system refrigerant make it necessary that the oil used to lubricate refrigeration compressors be specially prepared for that purpose. Some of the more important properties of the oil which must be considered when selecting the compressor lubricating oil are: (1) chemical stability, (2) pour and/or floc point, (3) dielectric strength, and (4) viscosity. In evaluating these oil properties with relation to an individual compressor, all the following factors should be taken into account: (1) the type and design of the compressor, (2) the nature of the refrigerant to be used, (3) the evaporator temperature, and (4) the compressor discharge temperature.

18-15. Chemical Stability. The importance of chemical stability is emphasized by the fact that it is necessary for the compressor lubricating oil to perform its lubricating function continuously and effectively without undergoing change for long periods of time. Since changing the oil in a hermetic motor-compressor is not usually practical, the same oil frequently remains in these units throughout the life of the unit, which is often ten years or more. Because of the high discharge temperatures encountered in hermetic motor-compressor units, particularly where air-cooled condensers are used, the ability of the oil to remain stable and resist decomposition under high temperature is especially important when selecting a lubricating oil for these units.

For the most part, the chemical stability of an oil is closely related to the amount of unsaturated hydrocarbons present in the oil. The smaller the percentage of unsaturated hydrocarbons contained in the oil, the more stable is the oil. For refrigeration service, a high quality oil with a very low percentage of unsaturated hydrocarbons is desired. These oils are usually light in color, being just off from a water-white.

18-16. Pour, Cloud, and Floc Points. The pour point of an oil is the lowest temperature at which the oil will flow, or "pour," when tested under certain specified conditions. Of two oils having the same viscosity, one may have a higher pour point than the other because of a greater wax content. Pour point is an important consideration in selecting an oil for low temperature systems. Naturally, the pour point of the oil should be well above the lowest temperature to be obtained in the evaporator. If the pour point of the oil is too high, the oil tends to congeal on the surface of the evaporator tubes, causing a loss in evaporator efficiency. Since this oil is not returned to the compressor, inadequate lubrication of the compressor may also result.

Since all lubricating oils contain a certain amount of paraffin, wax will precipitate from any oil if the temperature of the oil is reduced to a sufficiently low level. Because the oil becomes cloudy at this point, the temperature at which the wax begins to precipitate from the oil is called the cloud point of the oil. If the cloud point of the oil is too high, wax will precipitate from the oil in the evaporator and in the refrigerant control. Although a small amount of wax in the evaporator does little harm, a small amount of wax in the refrigerant control will cause stoppage of that part, with the result that the system will become inoperative.

The floc point of the oil is the temperature at which wax will start to precipitate from a mixture of 90% Refrigerant-12 and 10% oil by volume. Since the use of an oil soluble refrigerant lowers the viscosity of the oil and affects both the pour and cloud points, where oil miscible refrigerants are employed, the floc point of the oil is a more important property than the pour or cloud points. The use of 10% oil in the oil-refrigerant mixture to determine the floc point seems quite realistic, since the tendency of an oil-refrigerant mixture to separate wax increases as the amount of oil in the mixture increases and since the amount of oil circulating with the refrigerant seldom exceeds 10% and is usually much less.

Because floc point of the oil is a measure of the relative tendency of the oil to separate wax when mixed with an oil soluble refrigerant, it is an important consideration when selecting an oil for use with an oil miscible refrigerant at evaporator temperatures below 0° F. However, floc point has no significance when a non-miscible refrigerant is used.

18-17. Dielectric Strength. The dielectric strength of an oil is a measure of the resistance that the oil offers to the flow of electric current. It is expressed in terms of the voltage required to cause an electric current to arc across a gap one-tenth of an inch wide between two poles immersed in the oil. Since any moisture, dissolved metals, or other impurities contained in the oil will lower its dielectric strength, a high dielectric strength is an indication that the oil is relatively free of contaminants. This is especially important in oils used with hermetic motor-compressor units, since an oil of low dielectric strength may contribute to grounding or shorting of the motor windings.

18-18. Viscosity. Viscosity has already been defined in Section 16-10 as the resistance that a fluid offers to flow. With regard to the lubricating oil, viscosity may also be defined as a measure of the "body" of the oil or of the ability of the oil to perform its lubricating function by forming a protective film or coating between the various moving parts of the compressor, thus keeping the parts separated and preventing wear. In order to provide adequate lubrication for the compressor, the viscosity of the oil must be maintained within reasonable limits. If the viscosity of the oil is too low, the oil will not have sufficient body to keep the moving parts separated and thin film lubrication will result, accompanied by excessive wearing of the rubbing surfaces. Too, since, in addition to its lubricating function, the oil frequently must serve as a sealing agent between the low and high pressures in the compressor, excessive blow-by of vapor around the pistons (in a reciprocating compressor) or vanes (in a rotary compressor) may occur when the viscosity of the oil is low. On the other hand, when the viscosity of the oil is too high, fluid friction will be excessive and the power consumption of the compressor will be increased. Furthermore, in extreme case, a high viscosity oil may not have sufficient fluidity to penetrate between the various rubbing surfaces, particularly when tolerances are close, with the result that the lubrication of the compressor parts will be inadequate.

The viscosity of a lubricating oil is usually measured in Saybolt Seconds Universal (SSU), which is an index of the time in seconds required for a given quantity of oil (60 mm) at a controlled temperature (usually 100° F) to flow by gravity from a reservoir into a flask through a capillary tube of specified internal diameter (0.1765 cm) and length (1.225 cm). An oil having a temperature of 100° F and requiring 300 sec to pass through the tube is said to have a viscosity of 300 SSU at 100° F.

The viscosity of the lubricating oil changes considerably with the temperature, increasing as the temperature decreases. The effect of temperature on the viscosity of a typical lubricating oil is shown graphically in Fig. 18-12 (see top line—0% refrigerant dilution). Notice that the oil has a viscosity at 100° F of approximately 175 SSU, but increases to approximately 1800 SSU when the temperature of the oil is reduced to 40° F.

Shown also in Fig. 18-12 is the effect of

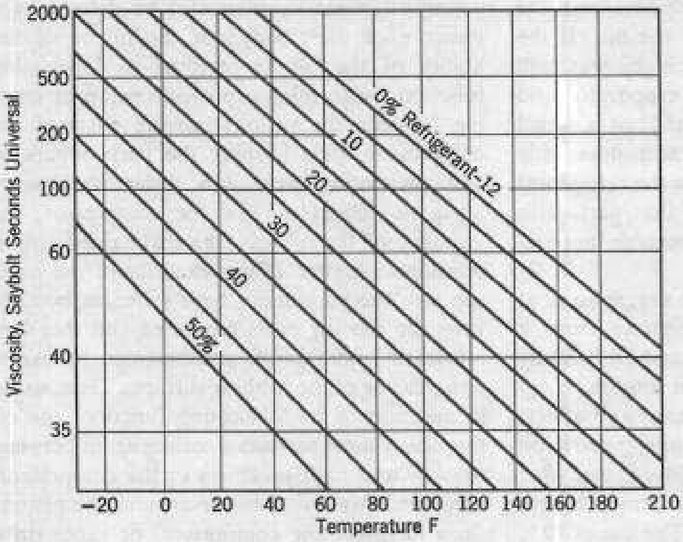


Fig. 18-12. Viscosity temperature curves of solution of Refrigerant-12 in oil. (From *ASRE Data Book*, Design Volume, 1957–58 Edition. Reproduced by permission of the American Society of Heating, Refrigerating and Air-Conditioning Engineers.)

refrigerant dilution on the viscosity of the lubricating oil. Notice, for example, that pure oil having a viscosity of 175 SSU at 100° F has a viscosity of about 60 SSU at this same temperature when diluted with 15% Refrigerant-12.

It is evident from the foregoing that both the operating temperature range and the effect of refrigerant dilution must be taken into account in selecting the proper viscosity oil. In all cases, the compressor manufacturer's recommendations should be followed when they are available. When such data are not available, the values given in Fig. 18-13 may be used as a guide.

18-19. Methods of Lubrication. Methods of lubricating the compressor vary somewhat depending upon the type and size of the compressor and upon the individual manufacturer. However, for the most part, lubrication methods can be grouped into two general types: (1) splash and (2) forced feed. Although forced feed lubrication can be found even in very small compressors, as a general rule, small, vertical, enclosed compressors up through approximately 15 hp are splash lubricated. Above this size, most compressors employ some type of forced feed lubrication. Often, a combination of the splash and forced feed methods is found in a single compressor.

In the splash method of lubrication, the compressor crankcase acts as an oil sump and is filled with oil to a level approximately even with the bottom of the main crank bearings. With each revolution of the crankshaft, the connecting rod and crankshaft (or eccentric) dip into the oil causing the oil to be splashed up on the cylinder walls, bearings, and other rubbing surfaces. Usually, small cavities or oil reservoirs are located at each end of the crankcase housing immediately over the main bearings. These cavities collect oil which feed by gravity down into the main bearings and shaft seal (Fig. 18-3). In some instances, connecting rods are rifle-drilled to carry oil to the wrist-pin bearings. Too, oil scoops or dippers are sometimes installed on the end of the connecting rods to increase splashing and/or to aid in forcing oil through rifle-drilled oil passages.

A modified type of splash lubrication, sometimes called flooded lubrication, employs slinger rings, discs, screws, or similar devices to raise the oil to a level above the crankshaft or main bearings, from where it is allowed to flood over the bearings and/or feed through oil channels to the various rubbing surfaces (Fig. 6-14). This method is particularly suitable for small, high speed compressors where the conventional splash system may result in excessive oil carryover because of violent splashing of the oil in the crankcase.

In the forced feed method of lubrication, the oil is forced under pressure through oil tubes and/or rifle-drilled passages in the crankshaft and connecting rods to the various rubbing surfaces. After performing its lubricating function, the oil drains by gravity back into a sump located in the crankcase or the compressor. The oil is circulated under pressure developed by a small oil sump located in the crankcase of the compressor, usually at the end of the crankshaft (Fig. 18-1). Since most oil pumps are automatically reversible, the direction of crank rotation is not usually critical with regard to compressor lubrication. However, this is not true of all compressors, particularly those employing oil dippers with splash lubrication. When rotation is critical, an arrow denoting the proper direction of rotation is usually embossed on the flywheel or crankcase housing.

Oil strainers are always placed at the suction inlet of the oil pump to prevent the entrance of foreign material into the pump or bearings. Although not required, oil filters are worthwhile in all forced feed lubrication systems to eliminate the possibility of plugged oil line resulting from the accumulation of sludges or other residue. An oil pressure failure safety switch (Section 21-20) should be employed in conjunction with all forced feed lubrication systems.

In some large compressors, cylinders are lubricated by mechanical forced feed lubricators which are located external to the compressor crankshaft. In such cases, the cylinder lubrication system is entirely separate from the internal pressure lubricating system (Fig. 18-14).

The bearings and cross-heads of horizontal, open crankcase compressors are usually splash lubricated (Fig. 18-2). Oil from the crankcase is carried to the cross-head by splash-fed troughs. Cylinders and piston rod packing glands are lubricated by mechanical forced feed lubricators similar to those used to lubricate the cylinders of large vertical compressors.

18-20. Liquid Refrigerant in the Compressor Crankcase. The presence of liquid refrigerant in the compressor crankcase is always undesirable for a number of reasons. In the first place, excessive dilution of the crankcase oil by liquid refrigerant can result in inadequate lubrication of the compressor parts. More important, however, is that fact that the liquid refrigerant will vaporize in the crankcase and cause foaming of the oil, with the result that the amount of oil carried over into the discharge line is materially increased. Under

A. Small Systems

Refrigerant	Type of Compressor	SU at 100 F Oil Viscosity
Ammonia	Reciprocating	150–300
Carbon dioxide	Reciprocating	280–300
Sulfur dioxide	Reciprocating	70–200
Sulfur dioxide	Rotary	280–300
Methyl chloride	Reciprocating	280–300
Refrigerant-30	Centrifugal	280–300
Refrigerant-30	Rotary	150–300
Refrigerant-11	Centrifugal	280–300
Refrigerant-12	Centrifugal and reciprocating	280–300
Refrigerant-21	Reciprocating	280–300
Refrigerant-113	Centrifugal	280–300
Refrigerant-114	Rotary	280–300

B. Industrial Refrigeration
(Ammonia and carbon dioxide compressors with splash, force-feed, or gravity circulating systems)

Type of Compressor	SU Viscosity Range
Where oil may enter refrigeration system or compressor cylinders	150–300 at 100 F
Where oil is prevented from entering system or cylinders:	
In force feed or gravity systems	500–600 at 100 F
In splash systems	150–160 at 100 F
Steam-driven compressor cylinders when condensate is reclaimed for ice making	140–165 at 210 F

C. Miscellaneous Equipment

Type of Requirement	SU Viscosity at 100 F Range
Bearings:	
Ring oiled, normal temperature	280–300
Ring oiled, low temperature	100–115
Chain oiled	280–300
Ball and roller bearings:	
Oil lubricated	280–300
Grease lubricated	—
Wick oilers	280–600

Fig. 18-13. Lubrication recommendations. (From *ASRE Data Book*, Design Volume, 1957–58 Edition, by permission of the American Society of Heating, Refrigerating and Air-Conditioning Engineers.)

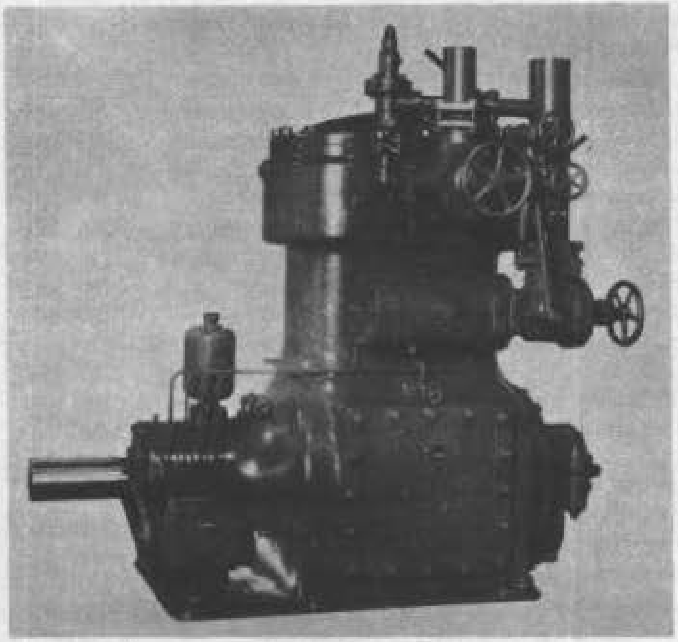


Fig. 18-14. Single-acting, vertical compressor with enclosed crankcase. (Courtesy Vilter Manufacturing Company.)

certain conditions, oil foaming may become so severe that all the oil is pumped out of the crankcase. Not only will this leave the compressor without lubrication but there is also the possibility that noncompressible slugs of liquid refrigerant and oil will enter the cylinder and cause serious damage to the compressor in the form of broken valves and pistons and bent or broken rods and shafts. Too, where considerable oil foaming occurs in compressors employing forced feed lubrication, lubrication will often be inadequate because the oil pump is unable to develop sufficient pressure to deliver the oil to the various rubbing surfaces.

Furthermore, vaporization of liquid refrigerant in the crankcase tends to reduce the capacity and efficiency of the compressor in that the resulting vapor is drawn into the cylinder and displaces vapor which would otherwise be taken from the suction line.

Liquid refrigerant may gain entrance into the crankcase in a number of ways:

1. Improper application or adjustment of the refrigerant flow control will often cause continuous or intermittent overfeeding of the evaporator, in which case liquid refrigerant will slop-over from the evaporator into the suction line and be carried to the compressor crankcase. As described in Chapter 17, this condition is more likely to occur during start-up than at any other time. In any event, it is easily prevented or corrected by proper application and adjustment of the refrigerant control and/or by properly designed suction piping.

2. Liquid refrigerant may drain by gravity into the compressor crankcase from the evaporator and/or suction piping during the off cycle. This condition is also caused by faulty system design, particularly with reference to the evaporator and suction piping. A leaking refrigerant control may also be a contributing factor. Here, again, the condition is readily corrected or prevented by proper design.

3. Any time the temperature at the compressor crankcase falls below that of the evaporator, liquid refrigerant will boil off in the evaporator and condense in the compressor crankcase. Naturally, this can occur only during the off cycle and only when the compressor is so located that the ambient temperature at the crankcase can fall below that of the evaporator. It is prevalent in the wintertime in installations where the compressor is located outside or in a basement or some other cold location. The only solution, of course, is to maintain the temperature of the crankcase above the saturation temperature of the refrigerant vapor. This can be accomplished by installing an electrical heating element in the

crankcase or by moving the compressor to a warmer location.

Because of the tendency of the lubrication oil to absorb oil miscible refrigerant vapors, a certain amount of liquid refrigerant will always be dissolved into the lubricating oil in any system employing an oil miscible refrigerant, assuming that the refrigerant vapor and the oil are permitted to come in contact with one another, as is usually the case. For the most part the percentage refrigerant that can be dissolved into the crankcase oil depends on three factors: (1) the degree of miscibility of the refrigerant, (2) the pressure of the refrigerant vapor, and (3) the temperature of the lubricating oil. For any one refrigerant, the percentage refrigerant that will be dissolved into the oil depends only on the pressure of the refrigerant vapor, the temperature of the oil, and the length of time that the two are in contact under steady conditions.

The solubility of Refrigerant-12 in oil under various conditions of temperature and pressure is shown graphically in Fig. 18-15. Notice that the percentage of Refrigerant-12 which can be dissolved in the oil increases considerably as the temperature of the oil decreases and as the pressure of the refrigerant vapor increases. For example, when the temperature of the oil is 100° F and the refrigerant vapor pressure is 20 psi, the maximum percentage of Refrigerant-12 which can be present in the oil-refrigerant mixture in the crankcase is only approximately 8% by weight. However, if the oil is cooled to 80° F while the refrigerant vapor pressure is increased to 60 psi, the percentage refrigerant in the oil-refrigerant mixture could be as high as 42%. In other words, under the latter conditions, the so-called lubricating oil in the crankcase could actually be 42% Refrigerant-12.

The practical significance of the foregoing can be illustrated by the use of an example. Suppose that during the compressor off cycle, the pressure on the low side of a Refrigerant-12 system rises to 38 psig, whereas the crankcase cools to a temperature of 80° F. Assuming that the off cycle is of sufficient length to permit equilibrium to be established, the percentage of liquid refrigerant in the oil-refrigerant mixture in the crankcase will be approximately 20% by weight, as determined from Fig. 18-15. Suppose now that the compressor cycles on and that the pressure in the evaporator and in the crankcase is immediately reduced to 25 psig. At this lower pressure, the maximum percentage of refrigerant that can be present in the oil-refrigerant mixture is only approximately 13%. Therefore, in a matter of only a few seconds approximately one-third of the refrigerant (7% by weight of the total mixture) must of necessity vaporize out of the mixture in order to establish the new percentage. Naturally, the vaporization of this much refrigerant out of the mixture in such a short time will cause severe foaming of the oil with the result that a considerable amount of oil is drawn into the compressor.

There are several ways to reduce oil foaming and the loss of oil from the crankcase during compressor start-up. One common method is to equip the compressor with an oil check valve, which is installed in the oil passage between the suction inlet of the compressor and the crankcase (Fig. 18-16). With oil miscible refrigerants, the oil pumped over into the system ordinarily returns to the compressor with the suction vapor. On entering the suction inlet, the oil is separated from the vapor by impingement before the vapor enters the cylinder. The separated oil

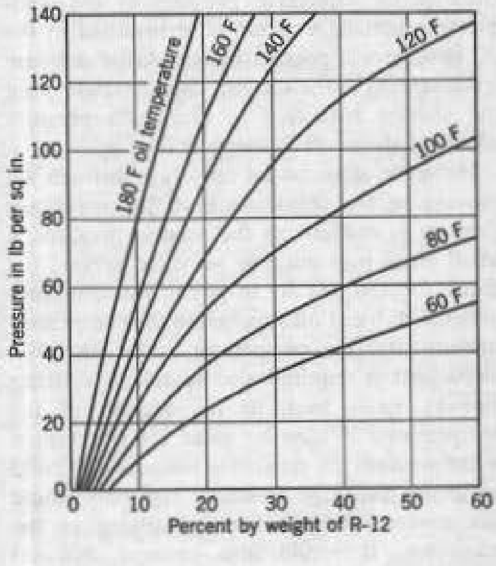


Fig. 18-15. Temperature-pressure relationship of Refrigerant-12 oil mixtures (pressure in psig). (From *ASRE Data Book*, Design Volume, 1957–58 Edition. Reproduced by permission of the American Society of Heating, Refrigerating and Air-Conditioning Engineers.)

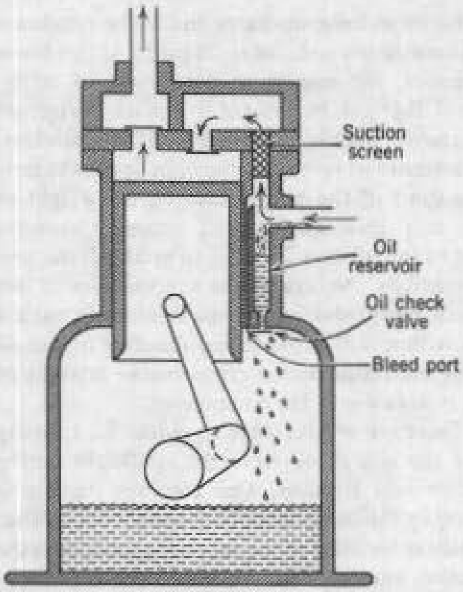


Fig. 18-16. Illustrating oil check value and bleed port.

drains from the inlet chamber to the crankcase through an oil passage provided for this purpose. Since this oil passage also serves to equalize the crankcase pressure to the compressor suction, a check valve installed in the oil passage will prevent the crankcase pressure from venting to the suction, thereby eliminating the sudden reduction in crankcase pressure which produces oil foaming at start-up.

However, since no oil can drain through the passage to the crankcase until the crankcase pressure is reduced to the suction pressure, a small bleed port must be provided around the check (or through it) to permit the crankcase pressure to bleed off slowly into the compressor suction after the compressor cycles on. The bleed port is required also to relieve cylinder blow-by gases back to the suction of the compressor. If blow-by gases are not vented to the suction, the crankcase pressure will build up to the discharge pressure. Not only would this prevent the oil from returning to the crankcase, it would also cause a material increase in the power requirements of the compressor.

During the normal running cycle, the crankcase pressure is approximately the same as the suction, and minor fluctuation in the suction pressure produced by the throttling action of the refrigerant control will supply the pressure differential necessary to cause the oil to flow through the check valve into the crankcase. However, the suction inlet chamber of the compressor must be large enough to serve as a reservoir for all the oil that returns to the compressor during the time that the crankcase pressure is too high to permit oil drainage into the crankcase.

Another method of reducing the amount of oil foaming at start-up, and one which is rapidly growing in popularity, is to install a small wattage heating element in the compressor crankcase. The crankcase heater is wired to come on when the compressor cycles off and serves to keep the oil in the crankcase warm during the offcycle so that the amount of refrigerant which can be dissolved into the oil is relatively small. However, care should be taken to wire the heater to the secondary of the main disconnect so that it cannot be turned off unless the main disconnect is pulled.

Still another method of reducing oil foaming at start-up is to operate the system on a pump-down cycle, in which case the evaporator is completely evacuated and the crankcase pressure reduced to a low level before the compressor cycles off. The resulting low pressure in the crankcase limits the amount of refrigerant absorbed by the oil. The pump-down cycle used alone or in conjunction with either the oil check valve or the crankcase heater is very effective in reducing oil foaming.

18-21. Rotary Compressors. Rotary compressors in common use are of two general designs. One employs a cylindrical steel roller which revolves on an eccentric shaft, the latter being mounted concentrically in a cylinder (Fig. 18-17). Because of the shaft eccentric, the cylindrical roller is eccentric with the cylinder and touches the cylinder wall at the point of minimum clearance. As the shaft turns, the roller rolls around the cylinder wall in the direction of shaft rotation, always maintaining contact with the cylinder wall. With relation to the camshaft, the inside surface of the cylinder roller moves counter to the direction of shaft rotation in the manner of a crankpin bearing. A spring-loaded blade, mounted in a slot in the cylinder wall, bears firmly against the roller at all times. The blade moves in and out of the cylinder slot to follow the roller as the latter rolls around the cylinder wall.

Cylinder heads or end-plates are used to close the cylinder at each end and to serve as supports for the camshaft. Both the roller and blade extend the full length of the cylinder with only working clearance being allowed between these parts and the end-plates. Suction and discharge ports are located in the cylinder wall near the blade slot, but on opposite sides. The flow of vapor through both the suction and discharge ports is continuous, except for the instant that the roller covers one or the other of the ports. The suction and discharge vapors are separated in the cylinder at the point of contact between the blade and roller on one side and between the roller and cylinder wall on the other side.

The point on the cylinder wall in contact with the roller changes continuously as the roller travels around the cylinder. At one point during each compression cycle the roller will cover the discharge ports, at which time only low pressure vapor will be in the cylinder. The manner in which the vapor is compressed by the roller is illustrated by the sequence of drawings in Fig. 18-17.

The whole cylinder assembly is enclosed in a housing and operates submerged in a bath of oil. Notice that the high pressure vapor is discharged into the space above the oil level in the housing from where it passes into the discharge line. All rubbing surfaces in the compressor including the end-plates are highly polished and closely fitted. Although no suction valves are needed, a check or flapper valve is installed in the discharge passage to eliminate back-feeding of the discharge vapor into the cylinder. When the compressor is operating, an oil film forms a seal between the high and low pressure areas. However, when the compressor stops, the oil seal is lost and the high and low pressures equalize in the compressor. A check valve must be placed in the suction line (or discharge line) to prevent the high pressure discharge gas from backing up through the compressor and suction line

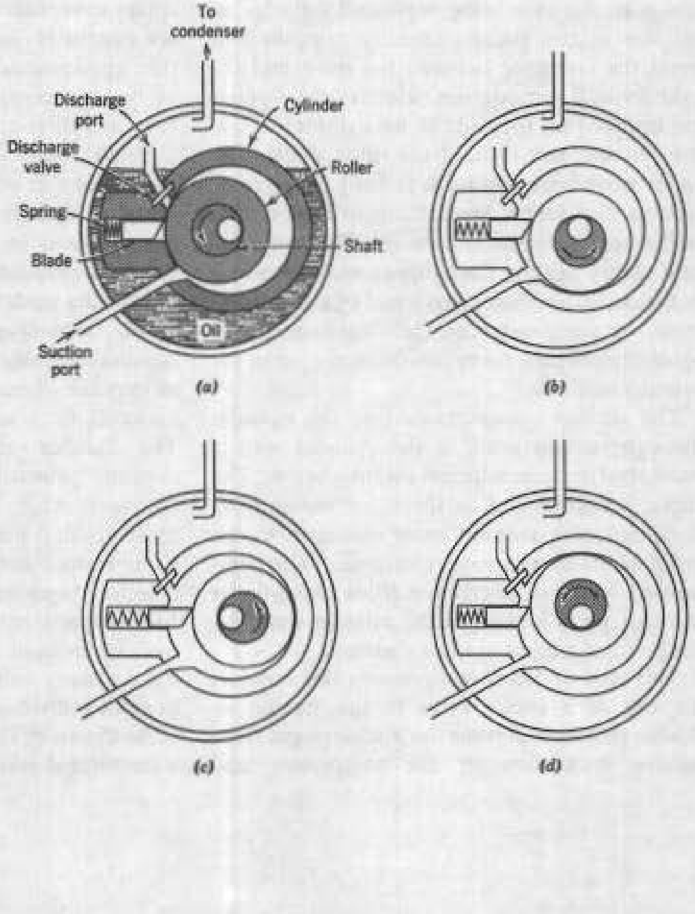


Fig. 18-17. Blade-type rotary compressor.

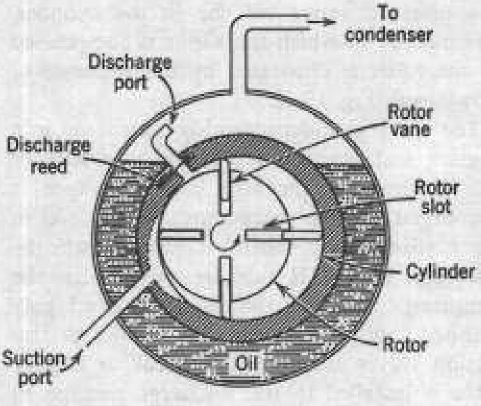


Fig. 18-18. Vane-type rotary compressor.

into the evaporator when the compressor cycles off.

Another design of rotary compressor employs a series of rotating vanes or blades which are installed equidistant around the periphery of a slotted rotor (Fig. 18-18). The rotor shaft is mounted eccentrically in a steel cylinder so that the rotor nearly touches the cylinder wall on one side, the two being separated only by an oil film at this point. Directly opposite this point the clearance between the rotor and the cylinder wall is maximum. Heads or end-plates are installed on the ends of the cylinder to seal the cylinder and to hold the rotor shaft. The vanes move back and forth radially in the rotor slots as they follow the contour of the cylinder wall when the rotor is turning. The vanes are held firmly against the cylinder wall by action of the centrifugal force developed by the rotating rotor. In some instances, the blades are spring-loaded to obtain a more positive seal against the cylinder wall.

The suction vapor drawn into the cylinder through suction ports in the cylinder wall is entrapped between adjacent rotating vanes. The vapor is compressed as the vanes rotate from the point of maximum rotor clearance to the point of minimum rotor clearance. The compressed vapor is discharged from the cylinder through ports located in the cylinder wall near the point of minimum rotor clearance.

This type of rotary compressors also requires the use of a check valve in the suction or discharge line to prevent the discharge gas from leaking back through the compressor and suction line to the evaporator when the compressor cycles off.

Although rotary compressors are positive displacement machines, because of their rotary motion and the smoother, more constant flow of the suction and discharge gases, they are much less subject to mechanical vibration and to the pronounced discharge pulsations associated with the reciprocating compressor. However, like reciprocating compressors, rotary compressors experience volumetric and compression losses resulting from blow-by around the compressing element, back leakage through valves, cylinder heating, clearance, and wire-drawing. As a general rule, the efficiency of rotary compressors is relatively high, being about 65 to 80%, depending on the individual design and the operating conditions.

Rotary compressors are particularly suitable for applications requiring a relatively large compressor displacement at moderate operating pressures. Under these conditions, the rotary compressors will usually have a distinct displacement advantage over reciprocating types of comparable size. For this reason, large rotary compressors of the rotating vane design are extensively used in industrial low temperature applications (Fig. 18-19), being employed as booster compressors in the low stages of two- and three-stage cascade systems.

Small rotary compressors have been used successfully in domestic units for a number of years. Although a few rotary compressors have been utilized in commercial installations, the difficulty encountered in the manufacture of the larger sizes tends to limit their use in this area.

18-22. Centrifugal Compressors. The centrifugal compressor consists essentially of a series of impeller wheels mounted on a steel shaft and enclosed in a cast iron casing (Fig. 18-20). The number of impeller wheels employed depends primarily on the magnitude of the thermodynamic head which the compressor must develop during the compression process. Compressors employing two, three, and four wheels (stages of compression) are common. More wheels may be used when the required increase in head is sufficiently large to demand it. As many as twelve wheels have been used in some individual cases.

As shown in Fig. 18-21, the impeller wheel of a centrifugal compressor consists of two discs, a hub disc and a cover disc, with a number of blades or vanes mounted radially between them. To resist corrosion and erosion, the impeller blades are usually constructed either of stainless steel or of high carbon steel with a lead coating. A typical two-stage rotor is shown in Fig. 18-22.

The operating principles of the centrifugal compressor are similar to those of the centrifugal fan or pump. Low-pressure, low-velocity vapor from the suction line is drawn in the inlet cavity or "eye" of the impeller wheel along the axis of the rotor shaft. On entering the impeller wheel, the vapor is forced radially outward between the impeller blades by action of the centrifugal force developed by the rotating wheel, and is discharged from the blade tips into the compressor housing at high velocity and at increased temperature and pressure. The high-pressure, high-velocity vapor discharged from the periphery of the wheel is collected in specially designed passages in the casing which reduce the velocity of the vapor and direct the vapor to the inlet of the next stage impeller or, in the case of the last stage impeller, to a discharge chamber, from where the vapor passes through the discharge line to the condenser. The refrigerant flow path through a two-stage centrifugal compressor is shown diagrammatically in Fig. 18-23.

The rotating impeller wheels are essentially the only moving parts of the centrifugal compressor and as such are the source of all the energy imparted to the vapor during the compression process. The action of the impeller is such that both the static and velocity heads of

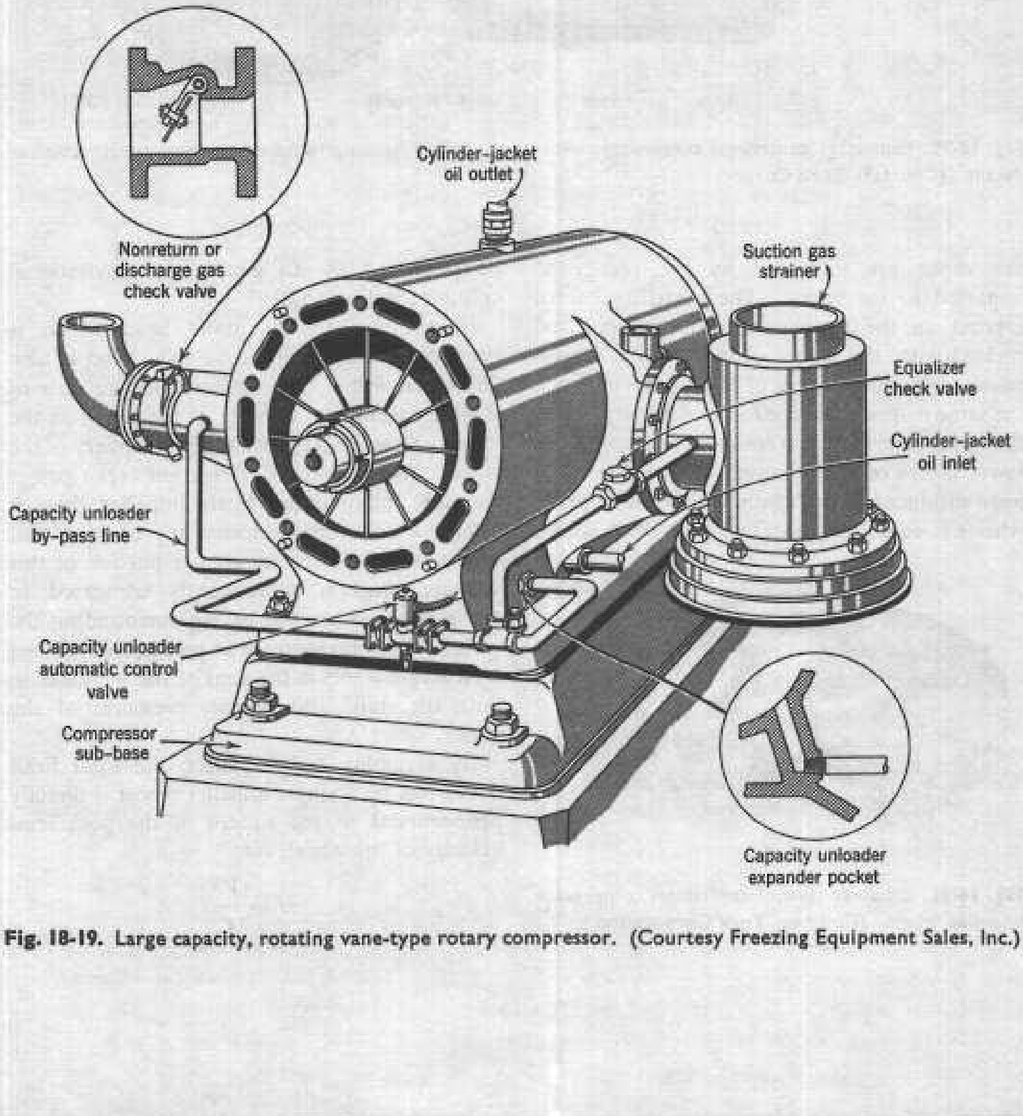


Fig. 18-19. Large capacity, rotating vane-type rotary compressor. (Courtesy Freezing Equipment Sales, Inc.)

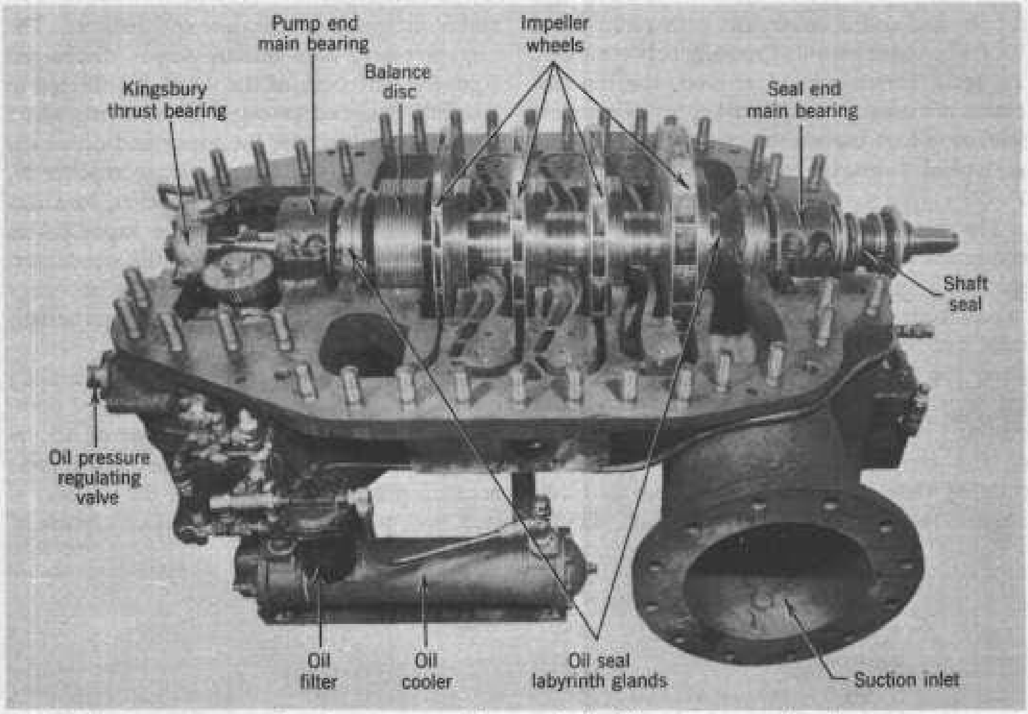


Fig. 18-20. Four-stage centrifugal compressor with upper half of housing removed to show construction of details. (Courtesy York Corporation.)

the vapor are increased by the energy so imparted to the vapor. The centrifugal force exerted on the vapor confined between, and rotated with, the blades of the impeller wheels causes self-compression of the vapor in much the same manner that the force of gravity causes the upper layers of a gas column to compress the lower layers of the column. Hence, the static head produced centrifugally within the impeller wheels is equal to the static head which would be produced by an equivalent gravitational column (Section 15-3).

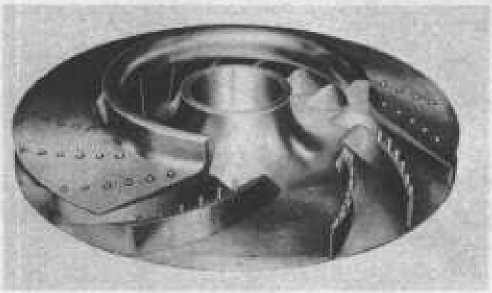


Fig. 18-21. Cutaway view of centrifugal compressor impeller wheel. (Courtesy York Corporation.)

In addition to the static head which is produced centrifugally, a velocity head is also developed within the impeller wheel because of the increase in the velocity of the vapor as the vapor passes from the eye to the periphery of the wheel. As the mass of refrigerant vapor passes through, and is rotated by the impeller wheel, it attains a rotational velocity approaching that of the wheel. Since the greater portion of this velocity head is subsequently converted to static head within the casing surrounding the wheels, the total increase in pressure developed by a single wheel is the sum of the increases in both the static and velocity pressures of the vapor.

By assuming radial blades, the total head developed by a single impeller wheel is directly proportional to the square of the peripheral velocity of the wheel, viz:

$$H = \frac{V^2}{g}$$

where H = the total head in feet

V = the periphial velocity of the wheel in fps

g = the gravitational constant

The total increase in pressure produced per wheel is

$$p = \frac{H \times \rho}{144} = \frac{V^2 \times \rho}{144 \times g}$$

where p = the pressure in psi

ρ = the mean density of the vapor in lb/cu ft

From the foregoing it is evident that for a refrigerant of given density, the total increase in pressure developed by a single wheel depends only on the tip velocity of the impeller blades, this tip velocity in turn being proportional to the rotational speed of the rotor shaft and to the diameter of the impeller wheel. However, since the maximum tip velocity is limited by the strength of materials and by the sonic speed of the refrigerant, it follows that the maximum increase in pressure which can be obtained with a single impeller wheel is also limited. For this reason, single-stage centrifugal compressors, such as the one shown in Fig. 18-24, can be used only in those few applications where, because of a small temperature head (difference between vaporizing and condensing temperatures), the increase in pressure (head) required is relatively small. As a general rule, two or more impeller wheels must be used in order to obtain the necessary pressure increase, in which case compression of the vapor occurs in stages as the vapor passes from one wheel to the next. Assuming equal vapor velocities at the inlet and outlet of the compressor, the total increase in pressure in the compressor is the sum of the pressure increases produced by the individual wheels. Notice that in any series of wheels the wheels are made progressively smaller in size in the direction of vapor flow in order to compensate for the reduced volume of the vapor resulting from prior compression in the preceding wheel or wheels (Fig. 18-20).

Since the head of a fluid is an expression of the energy per pound of fluid, it follows that the head developed by the compressor during the compression process is numerically equal to the work done in foot-pounds per pound of vapor compressed, and that the magnitude of the head which must be produced by the compressor depends on the refrigerant used and on the difference between the saturated suction and discharge temperatures. Therefore, it is evident that, for any given set of operating conditions, the head which must be produced by the compressor, that is, the diameter, speed, and number of wheels required, will be the same

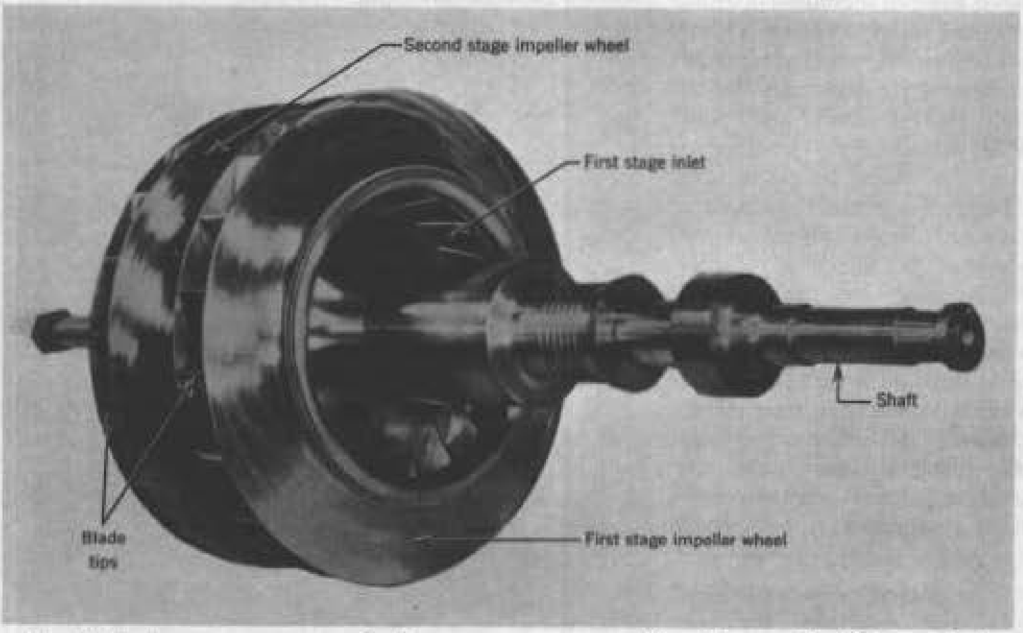


Fig. 18-22. Two-stage rotor centrifugal compressor rotor assembly. (Courtesy York Corporation.)

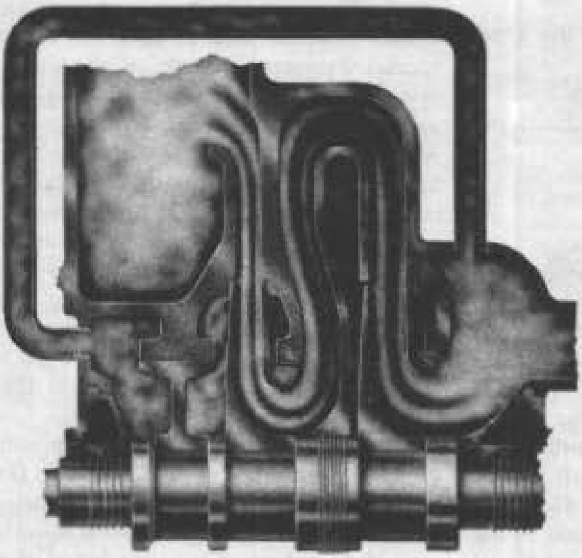


Fig. 18-23. Diagram of gas flow through centrifugal compressor. (Courtesy York Corporation.)

for small capacity compressors as for large capacity compressors. For this reason, centrifugal compressors are not practical in small sizes.

For good wheel performance, the diameter, width, and eye dimensions of the impeller wheel must be maintained within certain ratio limits. Since the width of the impeller must be reduced as the volume of vapor handled is reduced in order to insure stable operation at low gas volumes, the wheel width could become very narrow, resulting in high friction losses and poor wheel performance. Therefore, to keep the wheels in proportion, it becomes necessary to reduce the diameter of the wheels as the width of the wheels is reduced. At the same time, the speed of rotation and/or the number of wheels must be increased in order to maintain the required head. Since this tends to increase manufacturing and other costs, most manufacturers agree that the smallest practical size of centrifugal compressor is one which discharges to the condenser approximately 500 cu ft of vapor per minute. The exact tonnage this represents depends upon the refrigerant used and on the operating conditions. At the present time, centrifugal compressors are available in sizes ranging from approximately 35 tons to well over 2000 tons.

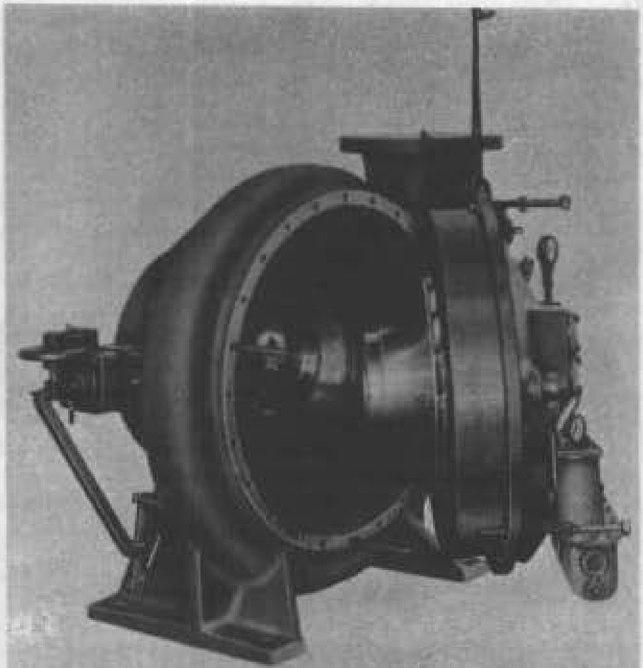


Fig. 18-24. Single-stage centrifugal compressor. (Courtesy York Corporation.)

Centrifugal compressors are essentially high speed machines. Rotative speeds of between 3000 and 8000 rpm are quite common, with much higher speeds being used in some individual cases. Because of their high rotative speeds, centrifugal compressors are capable of handling large volumes of vapor in relatively small sizes. Although especially suited for use with low pressure refrigerants requiring a large compressor displacement at moderate compression ratios, they have been applied successfully in all temperature ranges with both low and high pressure refrigerants.

Some of the more common refrigerants employed with centrifugal compressors are Refrigerants-11, -12, -113, and ammonia. The high displacement required per ton of refrigeration with Refrigerants-11 and 113 make these refrigerants ideal for use with centrifugal compressors in high temperature applications where the displacement required per ton of capacity is relatively low. Their use in such applications permits small refrigerating capacities without requiring small compressor frames and wheel sizes. On the other hand, when the required refrigerating capacity is large and/or the evaporator design temperature is low, refrigerants which require a relatively small displacement per ton capacity, such as Refrigerants-12 and ammonia, will ordinarily allow the use of smaller compressors to produce the same tonnage. In any event, because of the difference in the operating pressures, head requirements, and other characteristics of the several refrigerants, the compressor must be designed to fit the refrigerant as well as the application.

Centrifugal compressor efficiencies are relative high in all sizes and over a wide range of operating conditions, being about 70 to 80% as a general rule, although values well over 80% are obtained in many instances. Efficiency losses in a centrifugal compressor are due primarily to irreversible changes resulting from turbulence and fluid friction.

18-23. Centrifugal Compressor Construction and Lubrication. For maximum-pressor efficiency, the conversion of velocity pressure into static pressure in the casing must occur gradually and smoothly and without an appreciable loss in the total pressure head. To accomplish this, a series of diffuser vanes is often installed in the casing passages which convey the vapor from the discharge of one wheel to the inlet of the next. The diffuser vanes are curved in a direction opposite to that of vapor discharge from the impeller wheels and are so designed (area increasing in the direction of vapor flow) that velocity reduction and the accompanying increase in static pressure take place gradually and smoothly and with a minimum loss of energy. When diffuser vanes are not employed, gradual velocity reduction is obtained by discharging the vapor directly into scroll- or volute-shaped passages which guide the vapor from one wheel to the next. A compressor of volute design is shown in Fig. 18-25. In some instances, diffuser vanes and volutes are used together in a single compressor.

The back leakage of refrigerant between the several wheels or stages is limited to a practical minimum by the use of labyrinth-type seals which are arranged between the rotor and the stationary partitions. The labyrinth seal consists essentially of a series of thin steel strips which are fastened to the rotor and which match lands and grooves in the stationary partitions (Fig. 18-26). The labyrinth of passages provided by this type of seal causes a drop in the pressure of the refrigerant gas as it passes through each restricted area formed by the shaft sealing strips and housing. As the pressure drops, the velocity of the gas increases. However, on entering the next pocket, the gas encounters a large quantity of gas at rest and the increased velocity acquired during passage through the restriction is dissipated by the production of turbulence in the pockets. The leakage through the labyrinth seal is proportional to the clearance between the shaft sealing strips and the compressor housing and is also a function of the number of restrictions or pockets provided.

The fact that the vapor pressure on the discharge side of the impeller wheels is always greater than the pressure on the suction or inlet side of the wheels causes the rotor assembly to develop an axial thrust toward the suction inlet of the compressor. To offset this thrust, a balance disc is usually installed on the rotor shaft on the discharge side of the high stage impeller wheel (Fig. 18-20). This disc, equipped with a labyrinth seal, acts as a floating partition at the end of the discharge space. The pressure on the outboard side of the balance disc is

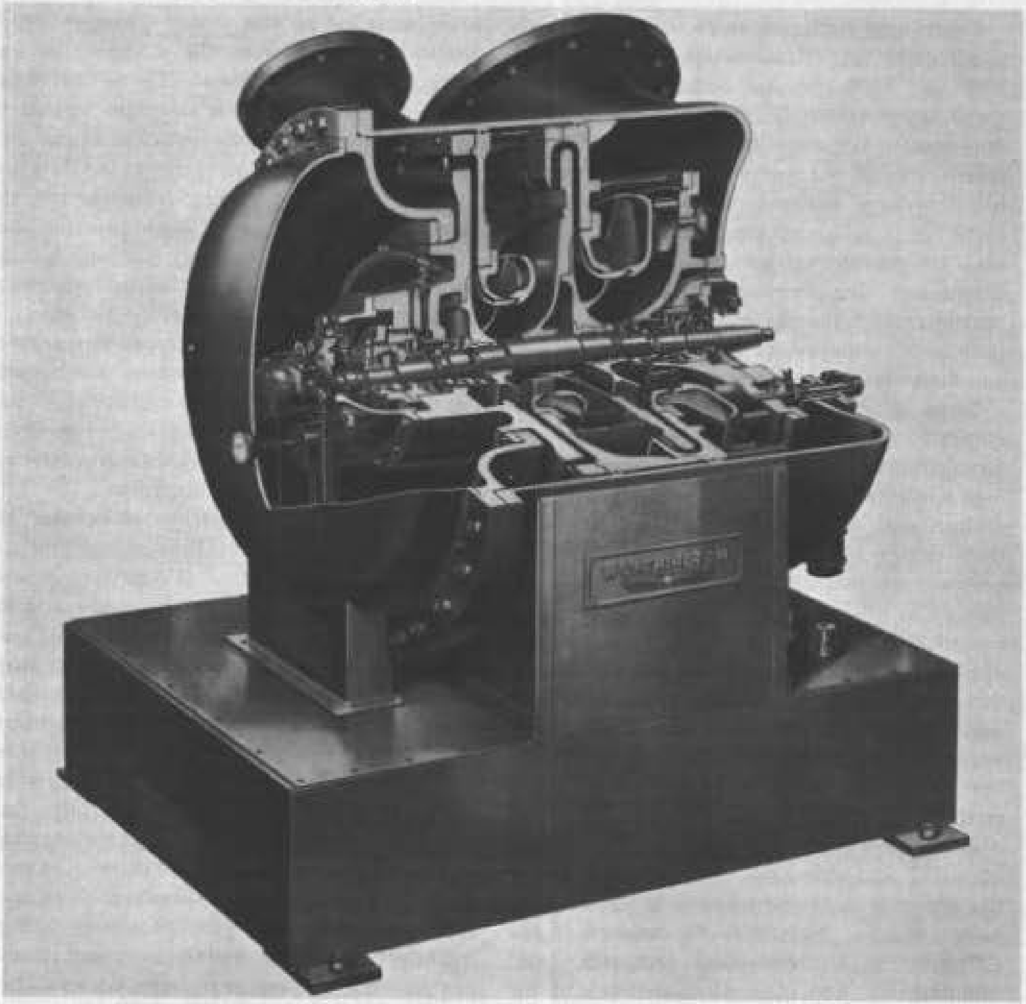


Fig. 18-25a. Volute-type compressor. (Courtesy Worthington Corporation.)

equalized to the suction inlet of the low stage impeller through an equalizer line (Fig. 18-23), whereas the inboard side of the disc is subject to the discharge pressure of the high stage wheel. When the balance disc is properly sized, the pressure differential across the disc will exactly balance the natural thrust of the rotor assembly (Fig. 18-27).

In one design of a three-stage compressor (Fig. 18-25*b*), the impellers are so positioned on the shaft that the axial thrust developed by the third-stage impeller opposes the thrust developed by the first- and second-stage impellers. Since the third-stage impeller is the highest pressure impeller, the thrust produced by it is sufficient to counter substantially the combined thrust of the other two impellers.

The rotor assembly is supported radially in the housing by two main bearings, one located at each end of the rotor shaft (Fig. 18-20). A Kingsbury-type thrust bearing mounted on the discharge end of the shaft positions the rotor assembly axially in the casing. Since the axial thrust of the rotor is usually neutralized by one means or another, the load on the thrust bearing is ordinarily very light. As in the case of the open-type reciprocating and rotary compressors, a shaft seal is employed between the compressor housing and the rotor shaft in order to prevent inward or outward leakage at the point where the shaft protrudes from the compressor housing.

Centrifugal compressors are pressure lubricating either by a submerged type oil pump

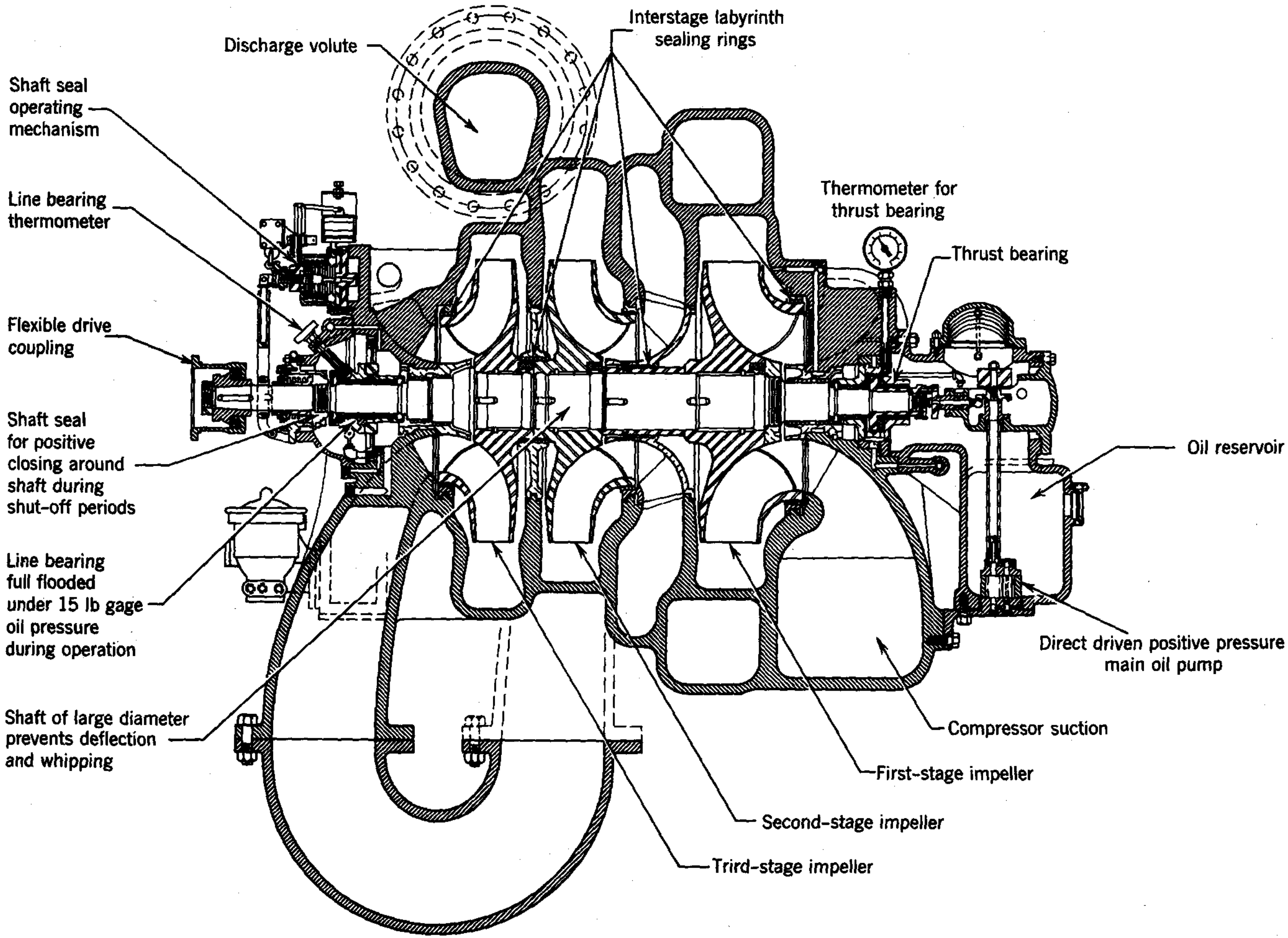


Fig. 18-25b. Sectional elevation of three-stage volute centrifugal compressor. (Courtesy Worthington Corporation.)

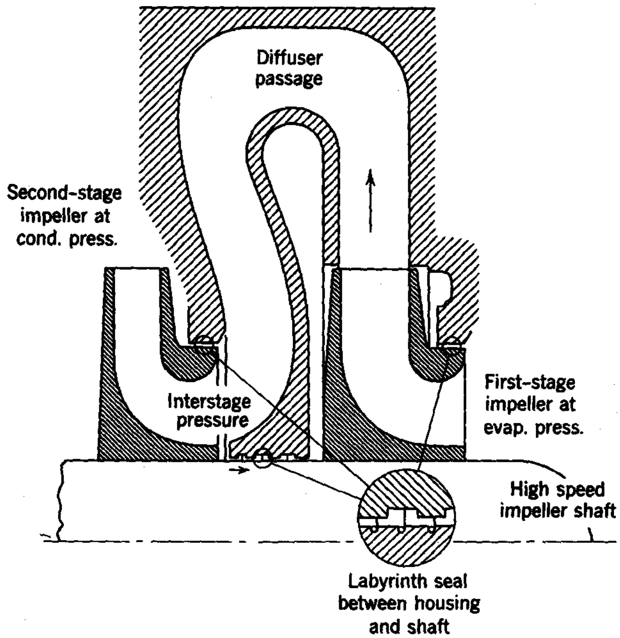


Fig. 18-26. Labyrinth seal between impellers. (Courtesy York Corporation.)

driven directly from the rotor shaft or by a separate, externally mounted, motor-driven oil pump with an external oil reservoir. The principal parts of the compressor requiring lubrication are the two main bearings, the Kingsbury thrust bearing, and the shaft seal. Since these parts are so located that they do not come into direct contact with the system refrigerant during normal operation, lubrication is simplified in that there is little or no contamination of the refrigerant by the compressor lubricating oil. The leakage of oil along the rotor shaft from the main bearings into the refrigerant spaces is minimized by the use of oil seal labyrinth glands which are installed on the shaft on the inboard side of each of the main bearings (Fig. 18-20). Oil coolers are employed to maintain oil temperature during normal operation. Oil heaters are usually installed in the oil reservoir to prevent excessive refrigerant dilution of the oil during periods of shut-down. Oil filters are standard equipment on all centrifugal compressors. Compressors employing a shaft-driven oil pump must also be equipped with an auxiliary oil pump to supply oil pressure during start-ups and at other times when the shaft driven pump cannot supply adequate lubrication for the compressor parts.

18-24. Performance of Centrifugal Compressors. In addition to its ability to maintain a relatively high efficiency over a wide range of load conditions, and its high volumetric displacement per unit of size, there are certain other desirable performance characteristics inherent in the design of a centrifugal compressor. Principal among these is its relatively flat head-capacity characteristic as compared to that of positive displacement compressors. This, along with an extreme sensitivity to changes in speed, greatly simplifies the problem of capacity control and tends to give the centrifugal compressor a decided advantage over the reciprocating type in any large tonnage installation where the evaporator temperature must be maintained relatively constant despite wide variations in evaporator loading.

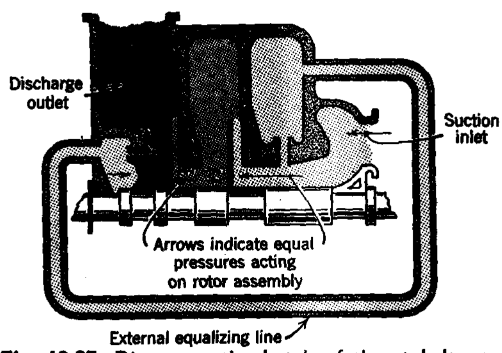


Fig. 18-27. Diagrammatic sketch of thrust balance. (Courtesy York Corporation.)

Like the centrifugal pump or blower, the delivery capacity (in cfm or in tons refrigeration) of an individual centrifugal compressor will decrease as the thermodynamic head produced by the compressor increases. Conversely, it is true also that as the delivery rate of the compressor is reduced the head produced by the compressor must increase. Therefore, since the maximum head which the compressor is capable of developing is limited by the peripheral speed of the impeller wheels, it follows that the minimum delivery capacity of the compressor is also limited. If the load on the evaporator becomes too small, the thermodynamic head necessary to handle the reduced volume of vapor will exceed the maximum head which the compressor can produce. When this point is reached, compressor operation becomes unstable and the compressor begins to "surge" or "hunt." However, with proper capacity control methods, the load on a centrifugal compressor can be reduced to as little as 10% of the design load without exceeding the pumping limit of the compressor.

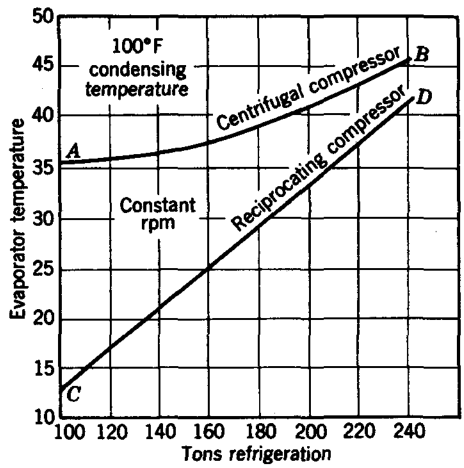


Fig. 18-28. Centrifugal versus reciprocating performance. (Courtesy York Corporation.)

As in the case of the reciprocating compressor, the capacity and the power requirements of the centrifugal vary with the vaporizing and condensing temperatures of the cycle and with the speed of the compressor. With reference to these variables, the performance of the centrifugal compressor is compared to that of the reciprocating type in Figs. 18-28 through 18-31. Some of the more important differences in the performance of the two compressors become apparent on careful examination of these data.

Notice in Fig. 18-28 that a reduction in refrigerating capacity from 240 to 100 tons is accomplished by the centrifugal compressor with a

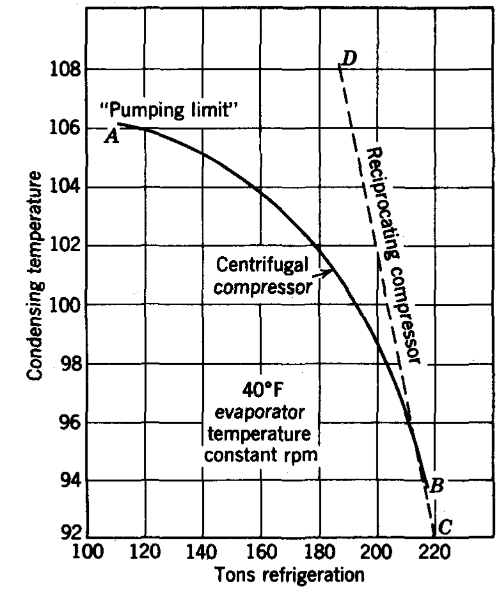


Fig. 18-29. Centrifugal versus reciprocating performance. (Courtesy York Corporation.)

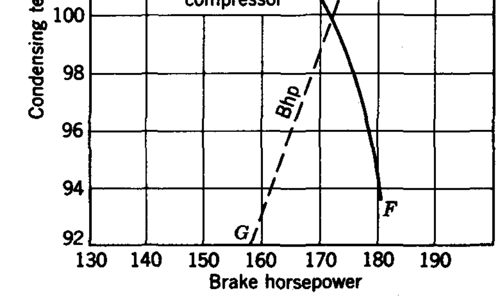


Fig. 18-30. Centrifugal versus reciprocating performance. (Courtesy York Corporation.)

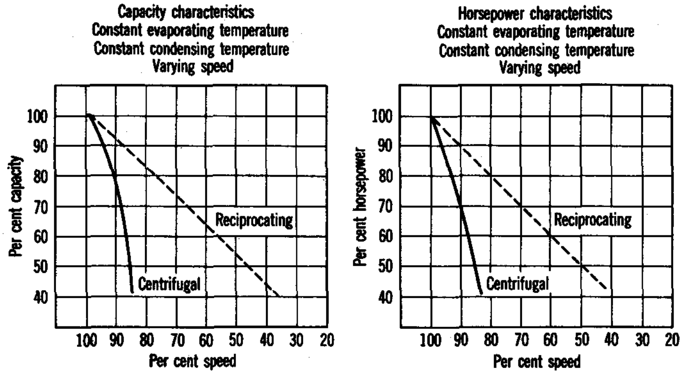


Fig. 18-31. (Courtesy Worthington Corporation.)

corresponding change in evaporator temperature of only 10° F as compared to a 29° F change required by the reciprocating compressor to effect the same tonnage reduction. This means in effect that a centrifugal compressor will maintain a more constant evaporator temperature over a much wider range of loading than will the reciprocating type. Naturally, this is an important advantage in any installation requiring the maintenance of a constant evaporator temperature under varying load conditions.

Too, the fact that a rather substantial change in capacity is brought about by only a small change in the suction temperature makes practical the use of suction throttling devices as a means of controlling the capacity of a centrifugal compressor, a practice which cannot be recommended for reciprocating compressors.

It is of interest to notice also that the operating range of the centrifugal compressor is definitely limited by the "surging" or "hunting" characteristic of the compressor. In Fig. 18-30, the centrifugal evaporator temperature cannot fall below 35° F regardless of the reduction of the evaporator loading. A further decrease in evaporator would cause the compressor to reach its "pumping limit" and a rise in the evaporator temperature would occur. By comparison, the positive displacement reciprocating compressor will continue to reduce the evaporator temperature and pressure as the evaporator load is reduced until a capacity balance is obtained between the evaporator loading and the compressor capacity.

Figure 18-31 shows a performance comparison between centrifugal compressors operating at constant speed and evaporator temperature but with varying condensing temperature. Curve *A–B* illustrates that the centrifugal compressor experiences a rapid reduction in capacity as the condensing temperature increases. This characteristic of a centrifugal compressor makes it possible to control compressor capacity by varying the quantity and temperature of the condenser water. The capacity of the compressor can be reduced by this means until point *A* is reached, beyond which a further increase in the condensing temperature will cause the required thermodynamic head to exceed the developed head of the compressor for the given speed and tons capacity, with the result that hunting will occur.

The reduction in the capacity of the reciprocating compressor with a rise in the condensing temperature is relatively small as compared to that experienced by the centrifugal compressor. Regardless of the increase in condensing temperature, the reciprocating compressor will continue to have a positive displacement and produce a refrigerating effect.

Figure 18-29 compares the power requirements of the centrifugal and reciprocating compressors under conditions of varying condensing temperature. Whereas the centrifugal shows a reduction in power requirements with an increase in the condensing temperature to correspond with the rapid fall off in capacity shown in Fig. 18-31, the reciprocating compressor shows a small increase in power requirements to correspond with the small change in refrigeration tonnage shown in Fig. 18-29 for that machine.

Figure 18-30 also illustrates the nonoverloading characteristic of the centrifugal compressor. Notice that an increase in condensing temperature causes a reduction in both the refrigerating capacity and the power requirements of the compressor, although the horsepower required per ton increases.

With regard to compressor speed, the centrifugal compressor is much more sensitive to speed changes than is the reciprocating type. Whereas the change in the capacity of the reciprocating compressor is approximately proportional to the speed change, according to the performance curves in Fig. 18-31, a speed change of only 12% will cause a 50% reduction in the capacity of the centrifugal compressor.

18-25. Capacity Control. Capacity control of centrifugal compressors is usually accomplished by one of the following three methods: (1) varying the speed of the compressor, (2) varying the suction pressure by means of a suction throttling damper, or (3) varying the condensing temperature through control of the condenser water. Often, some combination of these methods is used.

Because of its extreme sensitivity to changes in speed, the centrifugal compressor is ideally suited for capacity regulation by means of variable speed drives, such as steam turbines and wound-rotor induction motors. When constant speed drives, such as synchronous or squirrel cage motors, are employed, speed control can be obtained through the use of a hydraulic or magnetic clutch installed between the drive and the step-up gear.

18-26. Centrifugal Refrigeration Machines. Centrifugal compressors are available for refrigeration duty only as an integral part of a centrifugal refrigerating machine. Because of the relatively flat head-capacity characteristic of the centrifugal compressor, and the resulting limitation in the operating range, the component parts of a centrifugal refrigerating system must be very carefully balanced. Too, since very marked changes in capacity accrue with only minor changes in the suction or condensing temperatures, the centrifugal refrigerating system

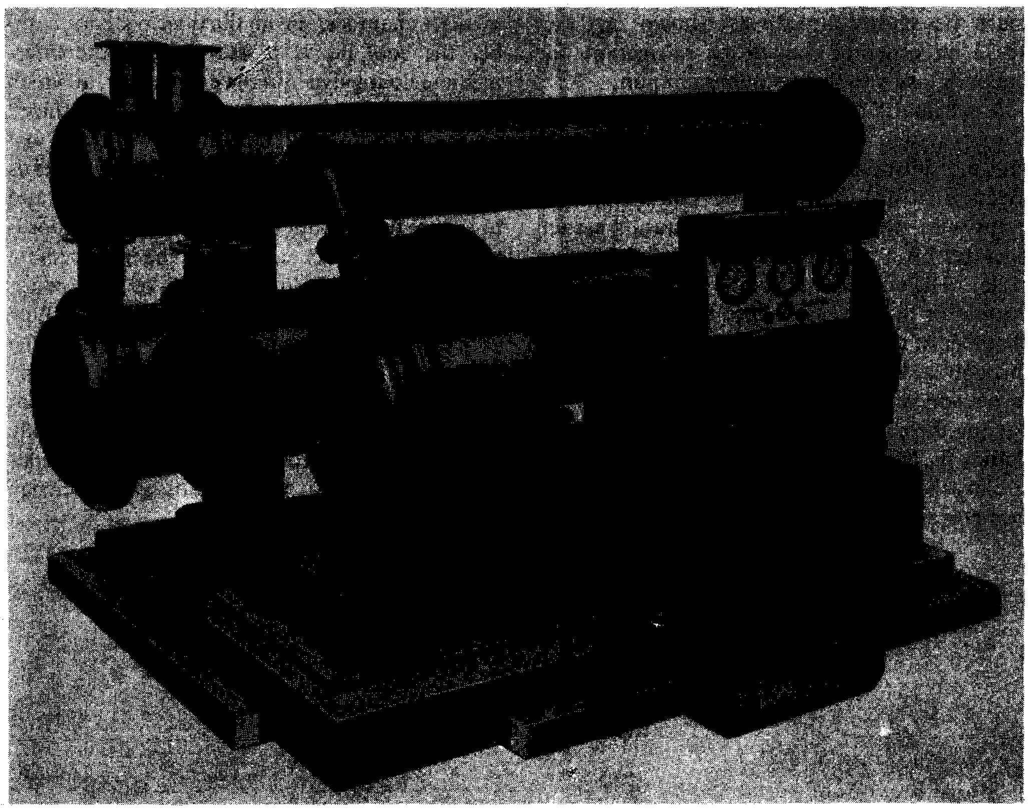


Fig. 18-32. Centrifugal refrigerating machine. (Courtesy Worthington Corporation)

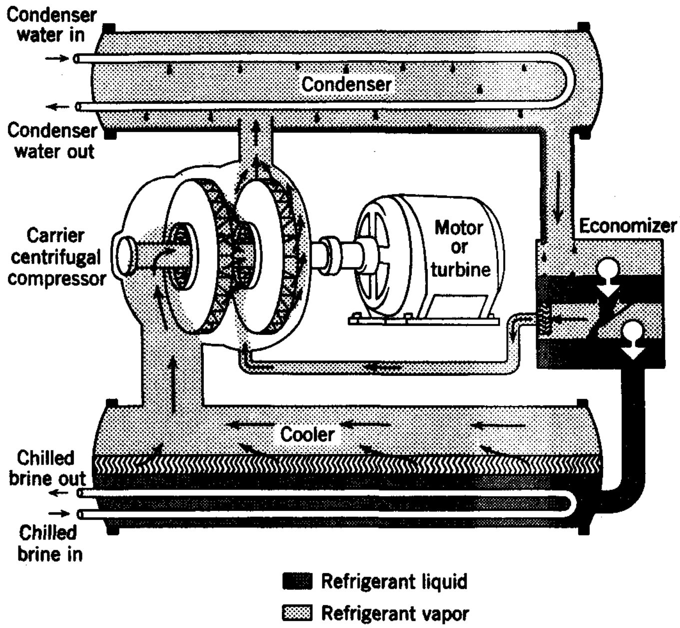


Fig. 18-33. Flow diagram for typical centrifugal refrigerating machine. (Courtesy Carrier Corporation.)

must be close-coupled, as shown in Fig. 18-32, in order to reduce the refrigerant line pressure losses to an absolute minimum.

A schematic diagram of a centrifugal refrigerating system is shown in Fig. 18-33. Except for the introduction of a flash intercooler (Section 20-12) between the condenser and evaporator, the centrifugal refrigerating system operates on a conventional vapor-compression cycle. High pressure liquid drains from the bottom of the condenser into the high pressure chamber of the intercooler, from where it passes through a high pressure float valve into the intermediate chamber of the intercooler. In passing through the float valve, a portion of the liquid flashes into the vapor state, thereby cooling the balance of the liquid to the temperature corresponding to the pressure in the intermediate chamber. From the intermediate chamber the cool liquid passes through the intermediate float valve into the evaporator, at which time the temperature of the liquid is reduced to the evaporator temperature by additional flashing. Hence, the effect of the intercooler is to increase the refrigerating effect per pound and to reduce the amount of flash gas in the evaporator. Since the flash vapor from the intermediate chamber is taken into the suction of the second-stage impeller, the pressure of this vapor will be above the evaporator pressure and therefore the power required to compress it to the condensing pressure will be less. Too, the cool vapor from the intercooler reduces the temperature of the discharge vapor from the first-stage impeller with the result that the capacity and efficiency of the system are increased.

19

Refrigerant Piping and Accessories

19-1. Piping Materials. In general, the type of piping material employed for refrigeration piping depends upon the size and nature of the installation, the refrigerant used, and the cost of materials and labor. Specific minimum requirements for refrigerant piping, with regard to type and weight of piping materials, methods of joining, etc., are set forth in the American Standard Safety Code for Mechanical Refrigeration (ASA Standard B9.1). Since the specifications in this standard represent good, safe, piping practice, they should be closely followed. Too, in all cases, local codes and ordinances must be taken into account.

The materials most frequently used for refrigerant piping are black steel, wrought iron, copper, and brass. All these are suitable for use with all the common refrigerants, except that copper and brass may not be used with ammonia, since, in the presence of moisture, ammonia attacks nonferrous metals.

Copper tubing has the advantage of being lighter in weight, more resistant to corrosion, and easier to install than either wrought iron or black steel. With all refrigerants except ammonia, refrigerant lines up to $4\frac{1}{8}$ in. OD may be either copper or steel. All lines above this size should be steel. However, general practice is to use all steel pipe in any installation where a considerable amount of piping exceeds 2 in. in size. Wrought iron pipe, although more expensive than black steel, is sometimes used in place of the latter because of its greater resistance to corrosion.

Steel pipe should be of either the seamless or lap-welded types, except that butt-welded pipe may be used in sizes up to 2 in. All steel pipe 1 in. or smaller should be Schedule-80 (extra heavy). Above this size, Schedule-40 (standard weight) pipe may be used, except that liquid lines up to $1\frac{1}{2}$ in. should be Schedule-80.

Copper tubing is available in either hard or soft temper. The hard drawn tubing comes in 20 ft straight lengths, whereas the soft temper is usually packaged in 25 and 50 ft coils. Only types K and L are suitable for refrigerant lines.

Soft temper copper tubing may be used for refrigerant lines up to $\frac{7}{8}$ in. OD, and is recommended for use where bending is required, where the tubing is hidden, and/or where flare connections are used. Hard temper tubing should be used for all sizes above $\frac{7}{8}$ in. OD and for smaller sizes when rigidity is desired.

19-2. Pipe Joints. Depending on the type and size of the piping, joints for refrigerant piping may be screwed, flanged, flared, welded, brazed, or soldered. When refrigerant pressures are below 250 psi, screwed joints may be used on pipe sizes up to 3 in. For higher pressures, screwed joints are limited to pipe sizes $1\frac{1}{4}$ in. and smaller. Above these sizes, flanged joints of the tongue and groove type should be used. Screwed-on flanges are limited to the pipe sizes listed above. For larger sizes, welded-neck flanges are required. A joint compound, suitable for refrigerant piping and applied to the male threads only, should be used with all screw connections.

Welding is probably the most commonly used method of joining iron and steel piping. Pipes 2 in. and over are usually butt-welded, whereas those $1\frac{1}{2}$ in. and smaller are generally socket-welded. Branch connections should be reinforced.

Flared compression fittings may be used for connecting soft temper copper tubing up to size $\frac{3}{4}$ in. OD. Above this size and for hard temper copper tubing, joints should be made with sweat fittings using a hard solder. Hard solders are silver brazing alloys with melting temperatures above 1000° F. Soft solder (95% tin and 5% antimony), having a melting point below 500° F, may be used for tubing $\frac{1}{2}$ in. OD and smaller. A suitable noncorrosive soldering flux should be used with both types of solder.

Flare fittings should be forged brass, and sweat fittings may be either wrought copper or forged brass. Cast sweat fittings are not suitable for refrigeration duty. As a general rule, better fitting joints will result when tubing and fittings are obtained from the same manufacturer.

19-3. Location. In general, refrigerant piping should be located so that it does not present a safety hazard, obstruct the normal operation and maintenance of the equipment, or restrict the use of adjoining spaces. When the requirements of refrigerant flow will permit, piping would be at least $7\frac{1}{2}$ ft above the floor, unless installed against the wall or ceiling. The piping code prohibits refrigerant piping in public hallways, lobbies, stairways, elevator shafts, etc., except that it may be placed across a hallway provided that there are no joints in the hallway and that nonferrous pipe 1 in. and smaller is encased in rigid metal conduit.

The arrangement of the piping should be such that it is easily installed and readily accessible for inspection and maintenance. In all cases, the piping should present a neat appearance. All lines should be run plumb and straight, and parallel to walls, except that horizontal suction lines, discharge lines, and condenser to receiver lines should be pitched in the direction of flow.

All piping should be supported by suitable ceiling hangers or wall brackets. The supports should be close enough together to prevent the pipe from sagging between the supports. As a general rule, supports should not be more than 8 to 10 ft apart. A support should be placed not more than 2 ft away from each change in direction, preferably on the side of the longest run. All valves in horizontal piping should be installed with the valve stems in a horizontal position whenever possible. All valves in copper tubing smaller than 1 in. OD should be supported independently of the piping. Risers may be supported either from the floor or from the ceiling.

When piping must pass through floors, walls, or ceilings, sleeves made of pipe or formed galvanized steel should be placed in the openings. The pipe sleeves should extend 1 in. beyond each side of the openings and curbs should be used around pipe sleeves installed in floors.

Provisions must be made also for the thermal expansion and contraction of the piping which usually amounts to approximately $\frac{3}{4}$ in. per hundred feet of piping. This is not ordinarily a serious problem, since refrigerant piping is usually three dimensional and therefore sufficiently flexible to absorb the small changes in length. However, care should be taken not to anchor rigidly both ends of a long straight length of pipe.

19-4. Vibration and Noise. In most cases, the vibration and noise in refrigerant piping originates not in the piping itself but in the connected equipment. However, regardless of the source, vibration, and the objectional noise associated with it, is greatly reduced by proper piping design. Often, relatively small vibrations transmitted to the piping from the connected equipment are amplified by improperly designed piping to the extent that serious damage to the piping and/or the connected equipment results.

For the most part, vibration in refrigerant piping is caused by the rigid connection of the piping to a reciprocating compressor, by gas pulsations resulting from the opening and closing of the valves in a reciprocating compressor, and by turbulence in the refrigerant gas due to high velocity. When centrifugal and rotary compressors are used, vibration and noise in the refrigerant piping is not usually a serious problem, being caused only by the latter of the above three factors. The reason lies in the rotary motion of the centrifugal and rotary compressors and in the smooth flow of the gas into and out of these units, as compared to the pulsating flow through the reciprocating-type compressor.

Since a small amount of vibration is inherent in the design of certain types of equipment, such as reciprocating compressors, it is not possible to eliminate vibration completely. However, if the piping immediately adjacent to such equipment is designed with sufficient flexibility, the vibration will be absorbed and dampened by the piping rather than transmitted and amplified by it. On small units piped with soft temper copper tubing, the desired flexibility is obtained by forming vibration loops in the suction and discharge lines near the point where these lines are connected to the compressor. If properly designed and placed, these loops will act as springs to absorb and dampen compressor vibration and prevent its transmission through the piping to other parts of the system. Where the compressor is piped with rigid piping, vibration eliminators (Fig. 19-1) installed in the suction and discharge lines near the compressor are usually effective in dampening compressor vibration. Vibration eliminators should be placed in a vertical line for best results.

On larger systems, adequate flexibility is ordinarily obtained by running the suction and discharge piping approximately 30 pipe diameters in each of two or three directions before anchoring the pipe. In all cases, isolation type hangers and brackets should be used when piping is supported by or anchored to building construction which may act as a sounding board to amplify and transmit vibrations and noise in the piping.

Although vibration and noise resulting from gas pulsations can occur in both the suction and discharge lines of reciprocating compressors, it is much more frequent and more intense in the discharge line. As a general rule, these gas pulsations do not cause sufficient vibration and noise to be of any consequence. Occasionally, however, the frequency of the pulsations and the design of the piping are such that resonance is established, with the result that the pulsations are amplified and sympathetic vibration (as with a tuning fork) is set up in the piping. In some instances, vibration can become so severe that the piping is torn loose from its supports. Fortunately, the condition can be remedied by changing the speed of the compressor, by installing a discharge muffler, and/or by changing the size of length of the discharge line. Since changing the speed of the compressor is not usually practical, the latter two methods are better solutions, particularly when used together.

When vibration and noise are caused by gas turbulence resulting from high velocity, the usual remedy is to reduce the gas velocity by increasing the size of the pipe. Sometimes this can be accomplished by installing a supplementary pipe.

19-5. General Design Considerations. Since many of the operational problems encountered in refrigeration applications can be traced directly to improper design and/or installation of the refrigerant piping and accessories, the importance of proper design and installation procedures cannot be overemphasized. In general, refrigerant piping should be so designed and installed as to:

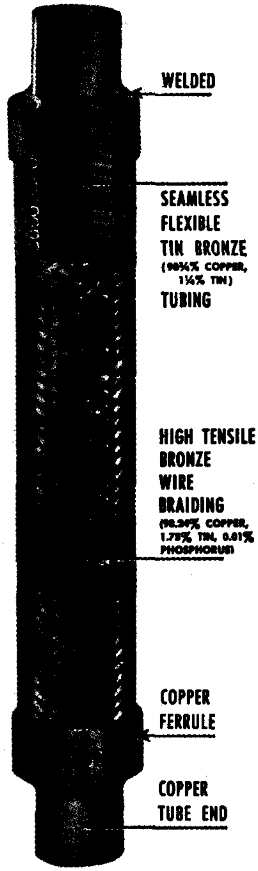


Fig. 19-1. Vibration eliminator. (Courtesy Anaconda Metal Hose Division, The American Brass Company.)

1. Assure an adequate supply of refrigerant to all evaporators
2. Assure positive and continuous return of oil to the compressor crankcase
3. Avoid excessive refrigerant pressure losses which unnecessarily reduce the capacity and efficiency of the system
4. Prevent liquid refrigerant from entering the compressor during either the running or off cycles, or during compressor start-up
5. Avoid the trapping of oil in the evaporator or suction line which may subsequently return to the compressor in the form of a large "slug" with possible damage to the compressor.

19-6. Suction Line Size. Because of its relative location in the system, the size of the suction piping is usually more critical than that of the other refrigerant lines. Undersizing of the suction piping will cause an excessive refrigerant pressure drop in the suction line and result in a considerable loss in system capacity and efficiency. On the other hand, oversizing of the suction piping will often result in refrigerant velocities which are too low to permit adequate oil return from the evaporator to the compressor crankcase. Therefore, the optimum size for the suction piping is one that will provide the minimum practical refrigerant pressure drop commensurate with maintaining sufficient vapor velocity to insure adequate oil return.

Most systems employing oil miscible refrigerants are so designed that oil return from the evaporator to the compressor is through the suction line, either by gravity flow or by entrainment in the suction vapor. When the evaporator is located above the compressor and the suction line can be installed without risers or traps, the oil will drain by gravity from the evaporator to the compressor crankcase, provided that all horizontal piping is pitched downward in the direction of the compressor. In such cases, the minimum vapor velocity in the suction line is of little importance and the suction piping can be sized to provide the minimum practical pressure drop without regard for the velocity of the vapor. This holds true also for any system employing a nonmiscible refrigerant and for any other system where special provisions are made for oil return.

On the other hand, when the location of the evaporator and/or other conditions are such that a riser is required in the suction line, the riser must be sized small enough so that the resulting vapor velocity in the riser under minimum load conditions will be sufficiently high to entrain the oil and carry it up the riser and back to the compressor. The minimum vapor velocity required for oil entrainment in suction risers for various suction temperatures and pipe sizes are given in Charts 19-1A and B for Refrigerants-12 and 22, respectively. These velocities should be increased by 25% to determine the minimum design velocity for a suction riser.

In accordance with Equation 15-13, for any given flow rate and pipe size, the refrigerant velocity in the pipe can be calculated by dividing flow rate in cfm by the internal area of the pipe in square feet. The refrigerant flow rate in pounds per minute per ton at various operating conditions can be determined from Charts 19-2A, B, and C for Refrigerant-12, 22, and ammonia, respectively. Internal areas for various pipe sizes are listed in Table 19-1.

Example 19-1. A Refrigerant-12 system, with a capacity of 40 tons, is operating at a 20° F evaporator temperature and a 110° F condensing temperature. Compute the refrigerant velocity in the suction line, if the line is $3\frac{1}{8}$ in. OD copper tube.

Solution. From Chart 19-2A, the flow rate in pounds per minute per ton	= 4.26
From Table 16-3, the specific of R-12 saturated vapor at 20° F	= 1.097 cu ft/lb
Refrigerant flow rate in cfm for 40 tons	= 4.26 × 1.097 × 40 = 187 cfm
From Table 19-1, the internal area of $3\frac{1}{8}$ in. OD copper tube	= 6.81 sq in.
Applying Equation 15-13, the refrigerant velocity in the suction pipe	$= \frac{187 \text{ cfm} \times 144}{6.81 \text{ sq in.}}$ = 3850 fpm

In the interest of high system efficiency, good design requires that the suction piping be sized so that the over-all refrigerant pressure drop in the line does not cause a drop in the saturated suction temperature of more than one or two degrees for Refrigerants-12 and 22, or more than one degree for ammonia. Since the pressure-temperature relationship of all refrigerants changes with the temperature range, the maximum permissible pressure drop in the suction piping varies with the evaporator temperature, decreasing as the evaporator temperature decreases. For instance, for Refrigerant-12 vapor at 40° F, the maximum permissible pressure drop in the suction piping (equivalent to a 2° F drop in saturation temperature) is 1.8 psi, whereas for Refrigerant-12 vapor at −40° F, the maximum permissible pressure drop in the suction line is only 0.4 psi.

Tonnage capacities of various sizes of iron pipe and type L copper tubing at various suction temperatures are listed in Tables 19-2, 19-3, and 19-4 for Refrigerants-12, 22, and ammonia, respectively. The values listed in the tables are based on a suction line pressure loss equivalent to 2° F per 100 ft of pipe for Refrigerants-12 and 22, and 1° F for ammonia. The condensing temperature is taken as 100° F for Refrigerant-12 and as 105° F for Refrigerant-22 and ammonia. In all cases, tonnage capacities at other condensing temperatures can be determined by applying the correction factors given at the bottom of each table. Equations are also given at the bottom of the tables for correcting tonnages for other pressure losses and equivalent lengths. The following example will serve to illustrate the use of the tables.

Example 19-2. A 40-ton, Refrigerant-12 system has an evaporator temperature of 20° F and a condensing temperature of 110° F. If a suction pipe 30 ft long containing six standard elbows is required, determine:
(*a*) the size of type L copper tubing required and
(*b*) the over-all pressure drop in the suction line in psi.

Solution. Adding 50% to the straight length of pipe as a fitting allowance establishes a trial equivalent length of 45 ft (30 ft × 1.5). From Table 19-2, $3\frac{1}{8}$ in. OD copper tubing has a capacity of 34 tons based on a condensing temperature of 100° F and a suction line pressure loss equivalent to 2° F per 100 ft of pipe. Since the pressure loss is proportional to the length of pipe and since the equivalent length of pipe is only 45 ft in this instance, this pipe size may be sufficient and a trial calculation should be made. From Table 15-1, $3\frac{1}{8}$ in. OD (3 in. nominal) standard elbows have an equivalent length of 3.8 ft.

Actual equivalent length of suction piping:

Straight pipe length	= 30 ft
6 ells at 3.8 ft	= 22.8 ft
Total equivalent length	= 52.8 ft

Correction factor from Table 19-3 to correct tonnage for 110° F condensing temperature is 0.9.	= 34 × 0.9
Corrected tonnage	= 30.6 tons

Suction line pressure loss in °F

$$= \frac{\text{Actual equiv. length}}{50} \times \left(\frac{\text{Actual tons}}{\text{Table tons}}\right)^{1.8}$$
$$= \frac{52.8}{50} \times \left(\frac{40}{30.6}\right)^{1.8}$$
$$= 1.55 \times (1.113)^{1.8}$$
$$= 1.88^\circ \text{ F}$$

From the chart at the bottom of Table 19-2, the pressure loss in psi corresponding to 1.88° F at a 20° F suction temperature is approximately 1.3 psi.

When the suction piping is sized on the basis of a one or two degree drop in the saturated suction temperature, as in the preceding example, the resulting vapor velocity will ordinarily be sufficiently high to insure the return of oil up a suction riser during periods of minimum loading. However, exceptions to this are likely to occur in any system where the evaporator temperature is low, where the suction line is excessively long, and/or where the minimum system loading is less than 50% of the design load. When any of the above conditions exist, the vapor velocity should be checked for the minimum load condition to be sure that it will be above the minimum required for successful oil entrainment in risers.

The widespread use of automatic capacity control on modern compressors, in order to vary the capacity of the compressor to conform to changes in the system load, tends to complicate the design of all of the refrigerant piping. Through the use of automatic capacity control, single compressors are capable of unloading down to as little as 25% of the maximum design capacity. When two or more such compressors are connected in parallel, the system can be designed to unload down to as little as 10% of the combined maximum design capacity of the compressors. Obviously, when the system capacity is varied over such a wide range, any suction piping sized small enough to insure vapor velocities sufficiently high to carry oil up a riser during periods of minimum loading will cause a prohibitively high refrigerant pressure drop during periods of maximum loading. On the other hand, sizing the pipe for a low pressure drop at maximum loading will result in riser velocities too low to return oil. Fortunately, the vapor velocity in horizontal piping

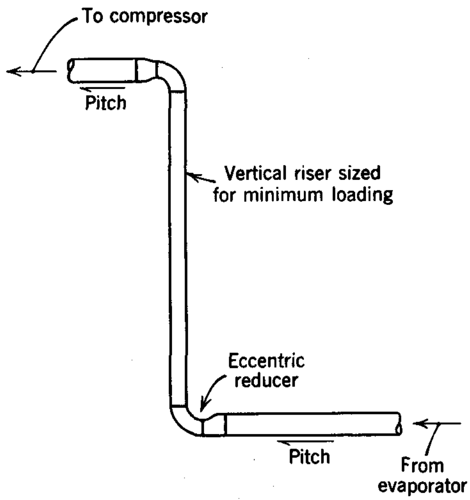


Fig. 19-2. Illustrating method of reducing the size of a vertical suction riser. (Courtesy York Corporation.)

is not critical and the problem is mainly one of riser design. In most cases, when the minimum system load is not less than 25% of the design load, the problem can be solved by reducing the size of the riser only, with the balance of the suction piping being sized for a low pressure drop at maximum loading.

Example 19-3. Assume that the minimum load for the system described in Example 19-2 is 25% of the design load, or 10 tons. Compute the velocity of the vapor in the suction piping under the minimum load condition and check in Table 19-1 to determine if it is sufficiently high to carry oil of the 10 ft riser.

Solution. From Example 19-1, the flow rate in pounds per minute per ton is 4.26 and the specific volume of the vapor is 1.097 cu ft/lb, so that the flow rate through the suction piping in cfm for 10 tons is 46.8 (4.26 × 1.097 × 10). From Table 19-1, the internal area of $3\frac{1}{8}$ in. OD copper tubing is 6.81 sq in. Therefore, the velocity of the vapor in the suction piping is

$$\frac{46.8 \times 144}{6.81} = 993 \text{ fpm}$$

From Chart 19-1A, the minimum velocity for oil entrainment with 20° F vapor in a $3\frac{1}{8}$ in. OD suction riser is approximately 1430 fpm. Increasing the table value by 25%, the minimum design velocity is found to be 1775 fpm (1430 × 1.25). It is evident that the vapor velocity at minimum loading will be too low for oil return up the riser.

Therefore, the size of the riser must be reduced. Try the next smaller pipe size, which is $2\frac{5}{8}$ in. OD. From Chart 19-1A, the minimum velocity for oil entrainment with 20° F vapor in a $2\frac{5}{8}$ in. OD suction riser is 1300 fpm. By increasing the table value by 25%, the minimum design velocity is 1625 fpm. From Table 19-1, the internal area of $2\frac{5}{8}$ in. OD copper tubing is 4.76 sq in. At the minimum load of 10 tons, the vapor velocity in the riser will be

$$\frac{46.8 \times 144}{4.76} = 1415 \text{ fpm}$$

Since this is still too low for adequate oil entrainment, try the next smaller pipe size, which is $2\frac{1}{8}$ in. OD. Using the same procedure, it is found that the minimum design velocity for oil entrainment with 20° F vapor in a $2\frac{1}{8}$ in. OD suction riser is 1500 fpm, and that the vapor velocity at the minimum load of 10 tons is 2180 fpm. Therefore, use $2\frac{1}{8}$ in. OD copper tube for the riser and $3\frac{1}{8}$ in. OD copper tube for the balance of the suction piping.

The refrigerant pressure loss in the riser in degrees is (see Note 3 at the bottom of Table 19-2)

$$\frac{10}{50} \times \left(\frac{40}{(12.1)}\right)^{1.8} = 0.248° \text{ F}$$

When this is added to the pressure loss in the balance of the suction piping, the over-all pressure loss will still be well within the acceptable limits.

Figure 19-2 illustrates the proper method of reducing the line size at a vertical riser. An eccentric reducer with its flat side down should be used at the bottom connection before entry to the elbow. This is done to prevent forming an area of low gas velocity which could trap a layer of oil extending the length of the horizontal line. At the top of the riser, the line size is increased beyond the elbow with a standard reducer, so that any oil reaching this point cannot drain back into the riser.

19-7. Double-Pipe Risers. As a general rule, when the suction riser is reasonably short and the minimum system load does not fall below 25% of the maximum design load, undersizing of the riser to provide adequate vapor velocity during periods of minimum loading will not cause a significant increase in the over-all suction line pressure drop, particularly if the horizontal portion of the piping is liberally sized. On the other hand, when the suction riser is quite long and/or when the minimum system loading is less than 25% of the design loading, undersizing of the riser to conform to the requirements of minimum loading will ordinarily result in an excessive pressure loss in the suction piping during periods of maximum loading, especially in low temperature installations. In such cases, the double-pipe riser, shown in Fig. 19-3, should be employed.

The small diameter riser is sized for the minimum load condition, whereas the combined capacity of the two pipes is designed for the maximum load condition. The larger riser is trapped slightly below the horizontal line at the bottom. During periods of minimum loading, oil will settle in the trap and block the flow of vapor through the larger riser, thereby increasing the flow rate and velocity in the smaller riser to a level high enough to insure oil return up the riser. As the system load increases, the velocity increases in the small riser until the pressure drop across the riser is sufficient to clean the oil out of the trap and permit flow through both pipes.

Notice that the trap at the bottom of the large riser is made up of two 45° elbows and one 90° elbow. This is done to keep the volume of the trapped oil as small as possible. Notice also that inverted loops are used to connect both risers to the top of the upper horizontal line, so that oil reaching the upper line cannot drain back into the risers.

19-8. General Design of Suction Piping. The suction piping should always be so arranged as to eliminate the possibility of liquid refrigerant (or large slugs of oil) entering the compressor during either the running or off cycles, or during compressor start-up. Generally, unless the system is operated on a pump-down cycle, it is good practice to install a liquid-suction heat exchanger in the suction line of all systems employing dry-expansion evaporators. The reason for this is that thermostatic expansion valves frequently do not close off tightly during the compressor off cycle, thereby permitting off cycle leakage of liquid refrigerant into the evaporator from the liquid line. When the compressor starts, the excess liquid often slops over into the suction line and is carried to the compressor unless a liquid-suction heat exchanger is employed to trap the liquid and vaporize it before it reaches the compressor. The liquid-suction heat exchanger also serves to trap and vaporize any liquid which may carry over into the suction line because of overfeeding of the expansion valve during start-up or during sudden changes in the evaporator loading.

Ordinarily, the liquid-suction heat exchanger can be safely omitted if the system is operated on a pump-down cycle, in which case the liquid refrigerant will be pumped out of the evaporator before the compressor cycles off, and the liquid line solenoid installed ahead of the refrigerant control will prevent liquid refrigerant from entering the evaporator from the liquid line, even though the expansion valve itself may not close off tightly.

When the evaporator is located above the compressor, and the system is not operated on a pump-down cycle, the suction line should be trapped immediately beyond the expansion valve bulb, as shown in Fig. 19-4, so that liquid refrigerant cannot drain by gravity from the evaporator to the compressor during the off

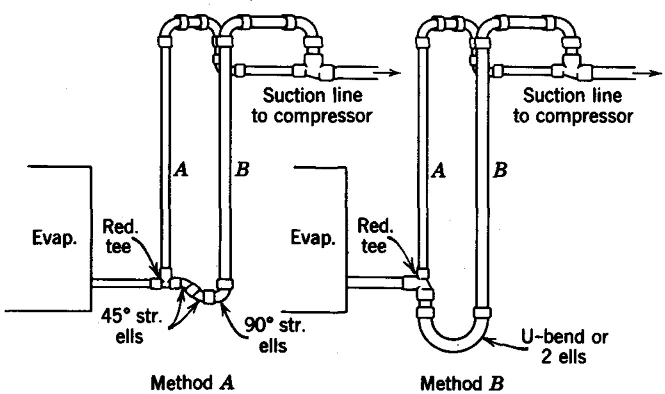


Fig. 19-3. Double suction riser construction. (Courtesy Carrier Corporation.)

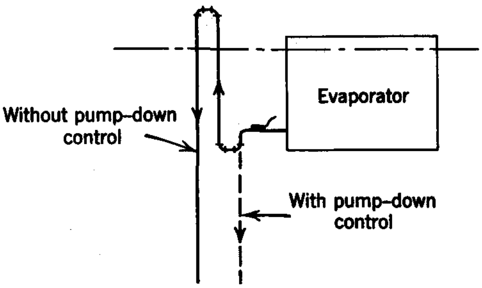


Fig. 19-4. Evaporator located above compressor.

cycle. If the system is operated on a pump-down cycle, the trap may be omitted and the piping arranged for free draining (dotted lines in Fig. 19-4), since all the liquid is pumped from the evaporator before the compressor cycles off.

When the evaporator is located below the compressor and the suction riser is installed immediately adjacent to the evaporator, the riser should be trapped as shown in Fig. 19-5 to prevent liquid refrigerant from trapping at the thermal bulb location. In the event that trapping of the line is not practical, the trap may be omitted and the thermal bulb moved to a position on the vertical riser approximately 12 to 18 in. above the horizontal header, as shown by the dotted line in Fig. 19-5.

When a multiple of evaporators is to be connected to a common suction main, each evaporator (or separately fed evaporator segment) should be connected to the main with an individual riser, as shown in Fig. 19-6. Since thermostatic expansion valves do not perform properly when the load on the evaporator falls below 50% of the design capacity of the valve, the flow rate through the individual risers should never drop below 50% of the design flow rate. Therefore, the use of individual risers for each evaporator (or separately fed segment) should eliminate the problem of oil return at minimum loading. Figures 19-7 through 19-9 illustrate some of the various methods of connecting multiple evaporators to a common suction main when it is not practical to use individual risers.

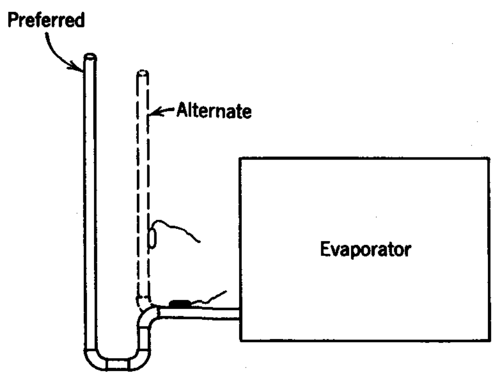


Fig. 19-5. Evaporator below compressor.

The suction piping at the compressor should be brought in above the level of the compressor

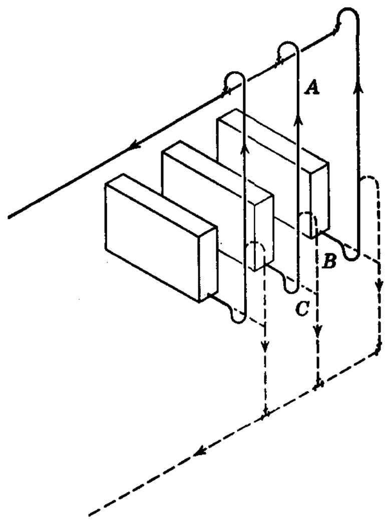


Fig. 19-6. Multiple evaporators, individual suction lines.

suction inlet. The piping should be designed without liquid traps and so arranged that the oil will drain by gravity from the suction line into the compressor. When multiple compressors are connected to a common suction header, the piping should be designed so that oil return to the several compressors is as nearly equal as possible. The lines to the individual compressors should always be connected to the side of the header. Some typical piping arrangements for multiple compressors are shown in Figs. 19-10 and 19-11.

19-9. Discharge Piping. Sizing of the discharge piping is similar to that of the suction piping. Since any refrigerant pressure drop in the discharge piping tends to increase the compressor discharge pressure and reduce the capacity and efficiency of the system, the discharge piping should be sized to provide the minimum practical refrigerant pressure drop. Tonnage capacities for various sizes of discharge pipes are given in Tables 19-2, 19-3, and 19-4. The values listed in the tables are based on an overall refrigerant pressure drop per 100 ft of equivalent length corresponding to a 2° F drop in the saturation temperature of Refrigerants-12 and

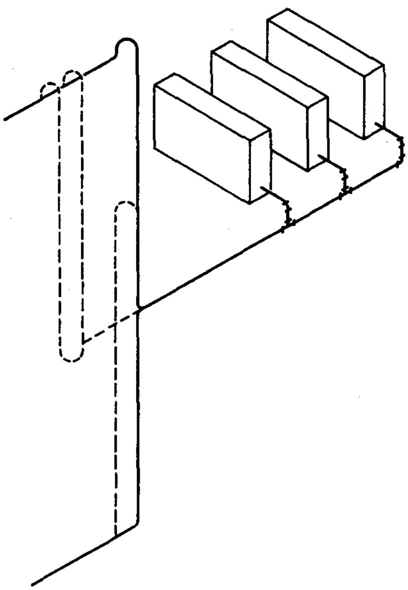


Fig. 19-7. Multiple evaporators, common suction line.

22, and for a 1° F loss in saturation temperature for ammonia. The procedure for sizing the discharge piping is the same as that used for the suction piping.

All horizontal discharge piping should be pitched downward in the direction of the refrigerant flow so that any oil pumped over from the compressor into the discharge line will drain toward the condenser and not back into the compressor head. Although the minimum vapor velocity in horizontal discharge piping is not ordinarily critical, special attention must be given to the vapor velocity in discharge line risers. As in the case of a suction riser, the discharge line riser must be designed so that the

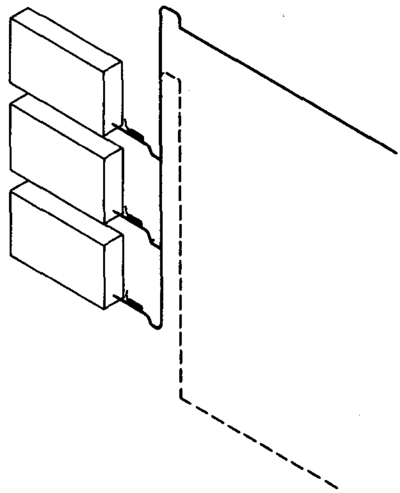


Fig. 19-8. Evaporators at different levels connected to a common suction riser.

vapor velocity in the riser under minimum load conditions is sufficiently high to entrain the oil and carry it up the riser. Minimum vapor velocities for oil entrainment in discharge risers are given in Charts 19-1C and D for Refrigerants-12 and 22, respectively. The table values should be increased by 25% to determine the minimum design velocity. When the system capacity varies over a wide range, a double-pipe riser may be necessary, unless a discharge line oil separator is used. When an oil separator is installed

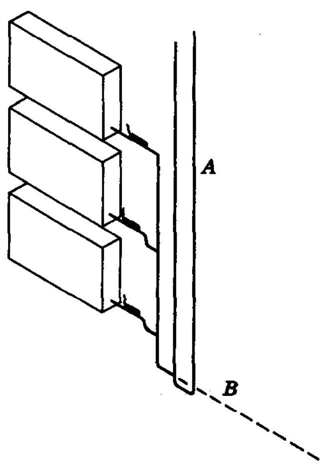


Fig. 19-9. Evaporators at different levels connected to a double suction riser.

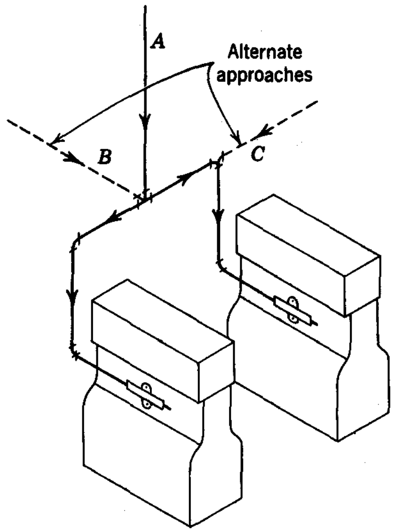


Fig. 19-10. Suction piping for compressors connected in parallel.

in the discharge line, the vapor velocity in the discharge riser is not critical and the riser should be sized for a low pressure drop, since any oil which is not carried up the riser during periods of minimum loading will drain back into the separator (Fig. 19-12).

When the compressor is not operating, the oil adhering to the inside surface of a discharge riser tends to drain by gravity to the bottom of the riser. If the riser is more than 8 to 10 ft long, the amount of oil draining from the riser may be quite large. Therefore, the discharge line from the compressor should be looped to the floor to form a trap so that oil cannot drain from the discharge piping into the head of the compressor. Since this trap will also collect any liquid refrigerant which may condense in the discharge riser during the compressor off cycle, it is especially important when the discharge piping is in a cooler location than the liquid receiver and/or condenser. Additional traps, one for 25 ft of vertical rise, should be installed in the discharge riser when the vertical rise exceeds 25 ft, as shown in Fig. 19-13. The horizontal width of the traps should be held to a minimum and can be constructed of two standard 90° elbows. The depth of the traps should be approximately 18 in.

The traps may be omitted if an oil separator is used, since any oil or liquid refrigerant draining from the vertical riser during the off cycle will drain into the oil separator. However, certain precautions must be taken to eliminate the possibility of liquid refrigerant passing from the oil separator to the compressor crankcase during the off cycle (see Section 19-12).

A purge valve, to permit purging of noncondensible gases from the system, should be installed at the highest point in the discharge piping or condenser.

When two or more compressors are connected together for parallel operation, the discharge piping at the compressors must be arranged so that the oil pumped over from an active compressor does not drain into an idle one. Under no circumstances should the piping be arranged so that the compressors discharge directly into one another. As a general rule, it is good practice to carry the discharge from each compressor nearly to the floor before connecting to a common discharge main (Fig. 19-14). With this piping arrangement, a discharge line trap is not required at the riser, since the lower horizontal header serves this purpose.

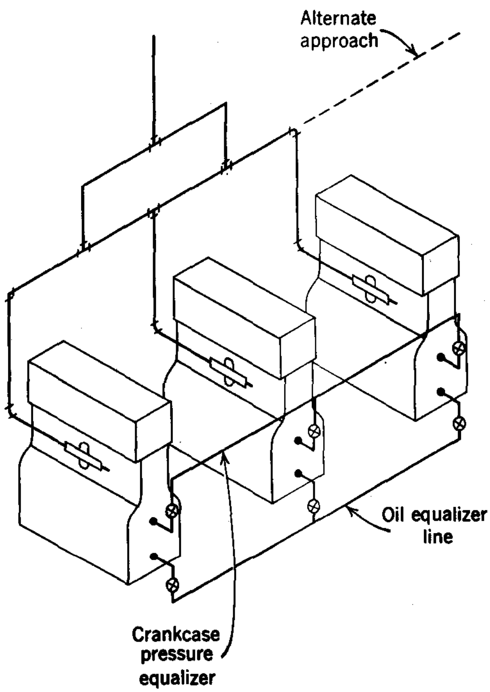


Fig. 19-11. Suction piping for compressors connected in parallel.

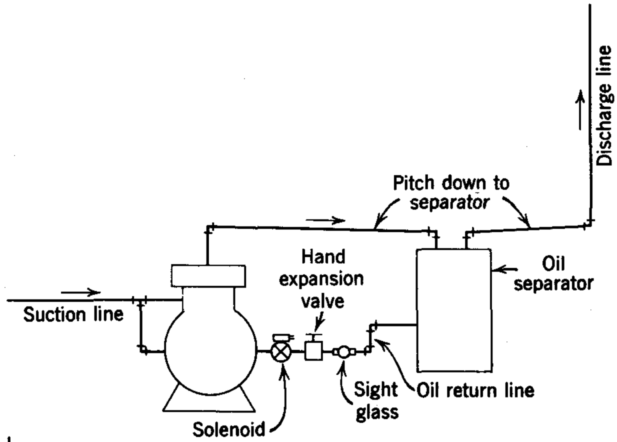


Fig. 19-12. Arrange for preventing liquid return to compressor crankcase.

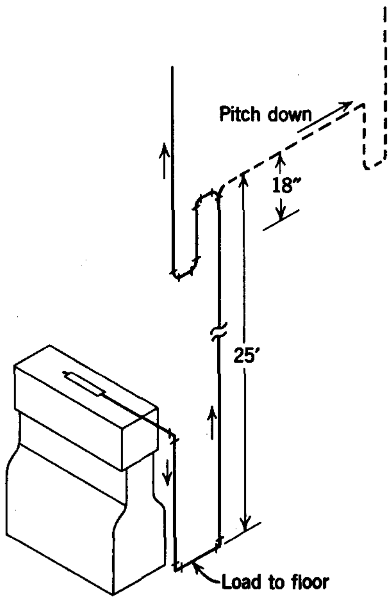


Fig. 19-13. Piping of discharge riser.

In the event that the discharge header must be located above the compressors, the discharge from the individual compressors should be connected to the top of the header, as shown in Fig. 19-15, so that oil cannot drain from the header into the head of an idle compressor.

In order to reduce the noise and vibration created by the compressor discharge pulsations, discharge mufflers are recommended for all multiple compressor installations and for a single compressor installation where the noise of the discharge pulsations may become objectionable. Discharge mufflers must be installed for free draining, either in a horizontal line or in a down-comer, as shown in Fig. 19-14, but never in a riser.

19-10. Liquid Lines. The function of the liquid line is to deliver a solid stream of subcooled liquid refrigerant from the receiver tank to the refrigerant flow control at a sufficiently high pressure to permit the latter unit to operate efficiently. Since the refrigerant is in the liquid state, any oil entering the liquid line is readily carried along by the refrigerant to the evaporator, so that there is no problem with oil return in liquid lines. For this reason, the design of the liquid piping is somewhat less critical than that of the other refrigerant lines, the problem encountered being mainly one of preventing the liquid from flashing before it reaches the refrigerant control.

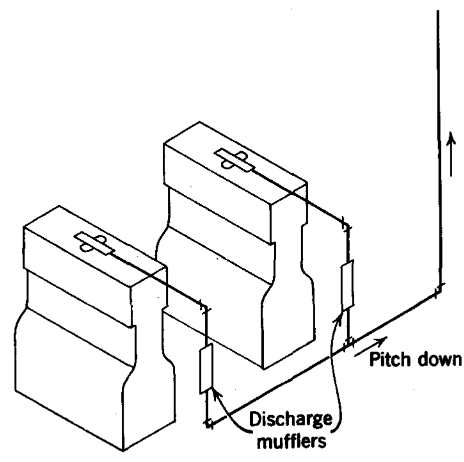


Fig. 19-14. Discharge piping of multiple compressors connected in parallel.

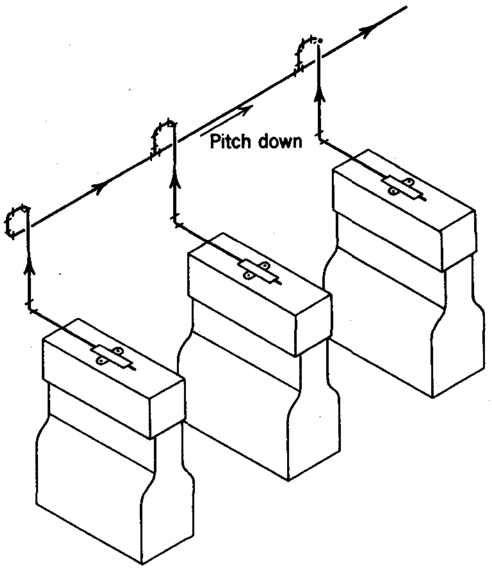


Fig. 19-15. Discharge piping for multiple compressors connected in parallel.

Flash gas in the liquid line reduces the capacity of the refrigerant control, causes erosion of the valve pin and seat, and often results in erratic control of the liquid refrigerant to the evaporator. To avoid flashing of the liquid in the liquid line, the pressure of the liquid in the line must be maintained above the saturation pressure corresponding to the temperature of the liquid.

Since the liquid leaving the condenser is usually subcooled 5 to 10° F, flashing of the liquid will not ordinarily occur if the over-all pressure drop does not exceed 5 to 10 psi. However, if the pressure drop is much in excess of 10 psi, it is very likely that some form of liquid subcooling will be required if flashing of the liquid is to be prevented. In most cases, a liquid-suction heat exchanger and/or a water-cooled subcooler will supply the necessary subcooling. In extreme cases, a direct-expansion subcooler may be required. (Fig. 19-16.)

The amount of subcooling required in any individual installation can be determined by computing the liquid line pressure drop. Pressure drop in the liquid line results not only from friction losses but also from the loss of head due to vertical lift. Tonnage capacities of various sizes of liquid pipes are listed in Tables 19-2, 19-3, and 19-4. The pressure drop per foot of vertical lift is found in Table 19-6.

Example 19-4. A Refrigerant-12 system has a capacity of 35 tons. The equivalent length of liquid line including fittings and accessories is 60 ft. If the line contains a 20 ft riser, determine:

(*a*) the size of the liquid line required
(*b*) the over-all pressure drop in the line
(*c*) the amount of subcooling (°F) required to prevent flashing of the liquid.

Solution

(*a*) From Table 19-2, 1⅜ in. OD copper tubing has a capacity of 35.1 tons based on a 1.8 psi pressure drop per 100 equivalent feet of pipe.

(*b*) For 60 ft equivalent length, the friction loss in the pipe = 1.8 psi × 0.6
= 1.00 psi

Pressure loss due to 20 ft lift = 0.55 psi × 20
= 11.00 psi

Over-all pressure drop = 1 + 11.0
= 12.0 psi

(*c*) Assuming the condensing temperature to be 100° F, the pressure at the condenser is 131.6 psia. The pressure at the refrigerant control is 119.6 psia, which corresponds to a saturation temperature of approximately 93° F. The amount of subcooling required is approximately 7° F (100° − 93°).

19-11. Condenser to Receiver Piping. Since the amount of refrigerant in the evaporator and condenser varies with the loading of the system, a liquid receiver tank is required on all systems employing hand expansion valves, automatic

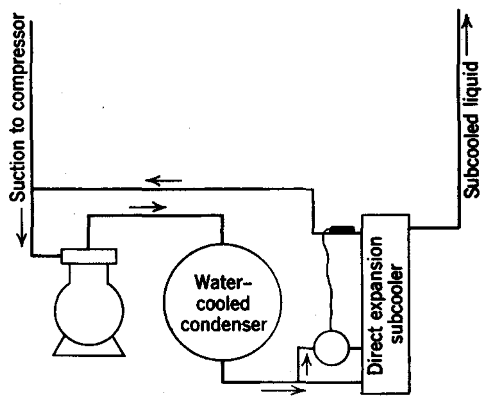


Fig. 19-16.

expansion valves, thermostatic expansion valves, or low pressure float valves. Some exceptions to this are installations using water-cooled condensers, wherein the water-cooled condenser also serves as the liquid receiver. In addition to accommodating fluctuations in the refrigerant charge, the receiver tends to keep the condenser drained of liquid, thereby preventing the liquid level from building up in the condenser and reducing the amount of effective condenser surface. The liquid receiver serves also as a pump-down storage tank for the liquid refrigerant.

In general, the condenser to receiver piping must be so designed and sized as to allow free draining of the liquid from the condenser at all times. If the pressure in the receiver is permitted to rise above that in the condenser, vapor binding of the receiver will occur and the liquid refrigerant will not drain freely from the condenser. Vapor binding of the receiver is likely to occur in any installation where the receiver is so located that it can become warmer than the condenser. The problem of vapor binding is more acute in the wintertime and during periods of reduced loading.

Although the exact preventative measures which can be taken to eliminate vapor binding of the receiver depends somewhat on the type of condenser, in every instance it involves proper equalization of the receiver pressure to the condenser.

Basically, there are two types of liquid receivers: the through-flow and the surge (Fig. 19-17). The through-flow type may be either bottom inlet or top inlet. With the through-flow type receiver, all the liquid from the condenser drains into the receiver before passing into the liquid line. The surge type differs from the through-flow in that only a part of the liquid from the condenser, that part not required in the evaporator, enters the receiver. With the surge-type receiver, the refrigerant liquid enters and leaves the receiver through the same opening.

When a top-inlet, through-flow type receiver is used, equalization of the receiver pressure to the condenser can be accomplished directly through the condenser to receiver piping, provided that the piping is sized so that the refrigerant velocity does not exceed 100 fpm and that the line is not trapped at any point (Fig. 19-18). All horizontal piping should be pitched toward the receiver at least ¼ in. per foot. When a stop valve is placed in the line, it should be located a minimum distance of 8 in. below the liquid outlet of the condenser and should be installed so that the valve stem is in a horizontal position. In the event that a trap in the line is unavoidable, a separate equalizing line must be installed from the top of the receiver to the condenser, as shown in Fig. 19-19. When an equalizing line is used, the condenser to receiver piping may be sized for a refrigerant velocity of 150 fpm.

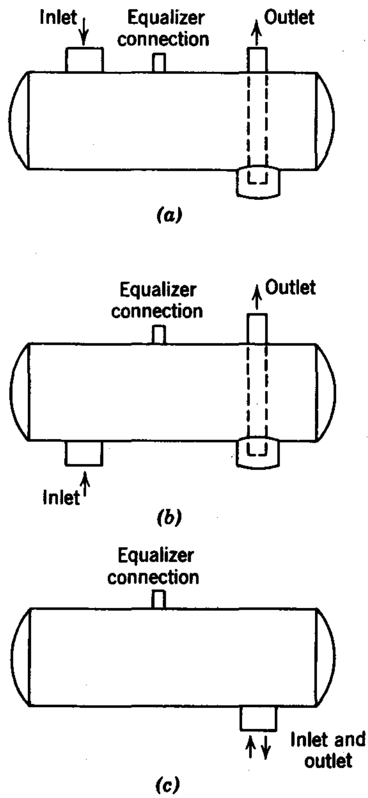


Fig. 19-17. (*a*) Top inlet through-flow receiver. (*b*) Bottom inlet through-flow receiver. (*c*) Surge-type receiver.

All bottom inlet receivers of both the through-flow and surge types must be equipped with equalizing lines. The minimum vertical distance (h_2 of Fig. 19-19) between the outlet of the condenser and the maximum liquid level in the receiver to prevent the back-up of liquid in the

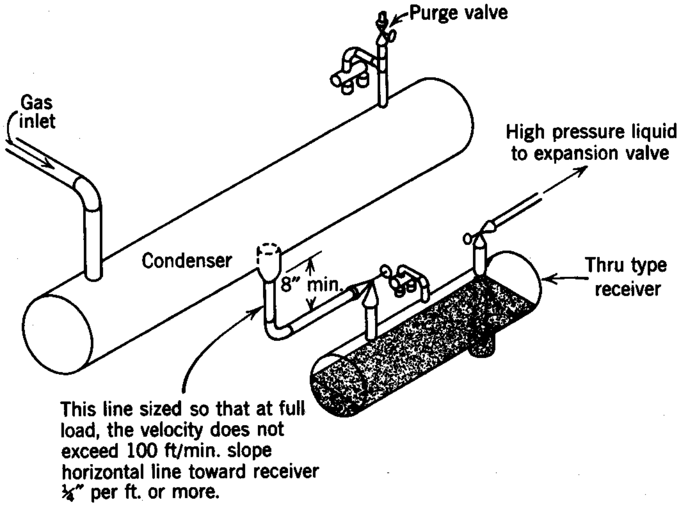


Fig. 19-18. Top inlet through type receiver hookup. (Courtesy York Corporation.)

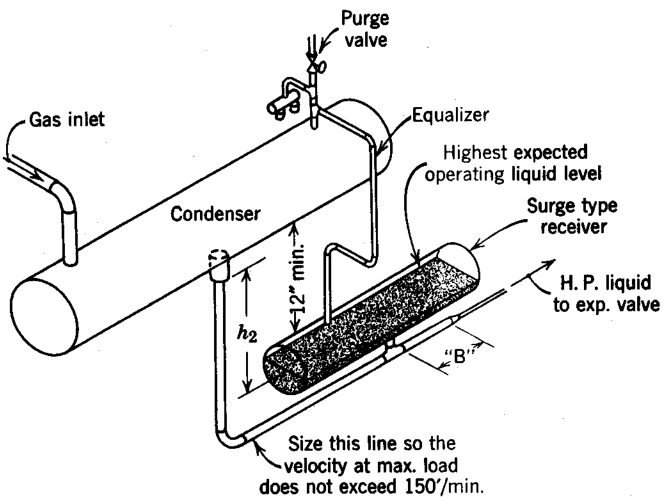


Fig. 19-19. Surge-type receiver hookup. (Courtesy York Corporation.)

Maximum Velocity of Drain line, lbs/min	Type Valve between Condenser and Receiver	h_2 Required inches
150	None	14
150	Angle	16
150	Globe	28
100	None, Angle, or Globe	14

Size drain line to receiver for maximum velocity of 150 ft/min. If a valve is located in this line, the trapping height limitation may require a larger size line to minimize the pressure drop.

condenser is listed at the bottom of Fig. 19-19. This value increases as the pressure loss in the condenser to receiver line increases, but should never be less than 12 in.

Figure 19-20 illustrates a satisfactory piping arrangement for multiple condensers connected in parallel to a top inlet receiver. A piping arrangement for multiple condensers with a bottom inlet receiver is shown in Fig. 19-21. The vertical distance h_2 is determined from the bottom of Fig. 19-19.

19-12. Oil Separators. As a general rule, discharge line oil separators should be employed in any system when oil return is likely to be inadequate or difficult to accomplish and/or when the amount of oil in circulation is apt to be excessive or to cause an undue loss in the efficiency of the various heat transfer surfaces. Specifically, discharge line oil separators are recommended for: (1) all systems employing nonmiscible refrigerants, (2) low temperature systems, (3) all systems employing nonoil-returning evaporators, such as flooded liquid chillers, when oil bleeder lines or other special provisions must be made for oil return, and (4) any system where capacity control and/or long suction or discharge risers cause serious piping design problems.

Discharge line oil separators are of two basic types: (1) impingement and (2) chiller. The impingement-type separator (Fig. 19-22) consists of a series of screens or baffles through which the oil laden refrigerant vapor must pass.

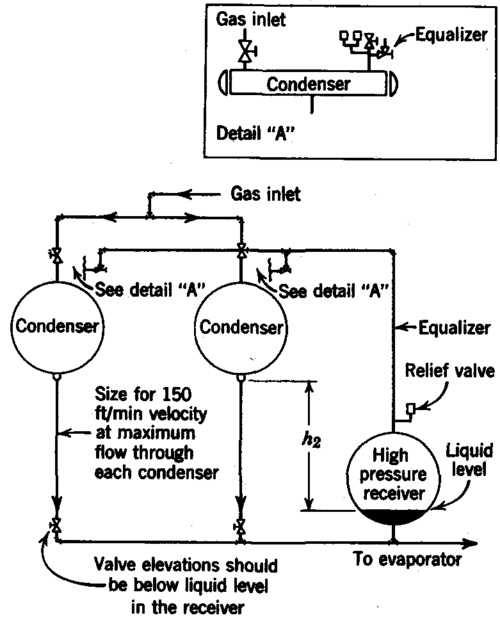


Fig. 19-21. Parallel shell-and-tube condensers with bottom inlet receiver. (Courtesy York Corporation.)

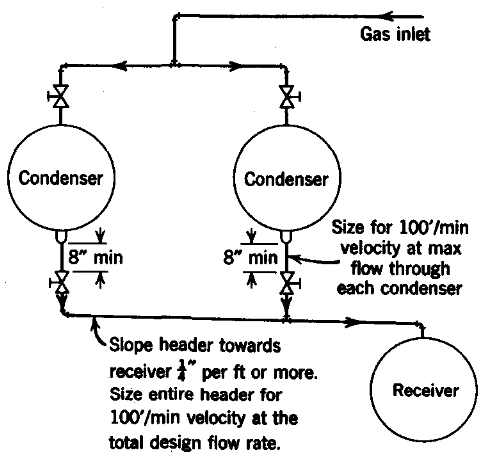


Fig. 19-20. Parallel shell-and-tube condensers with top inlet receiver. (Courtesy York Corporation.)

On entering the separator, the velocity of the refrigerant vapor is considerably reduced because of the larger area of the separator with relation to that of the discharge line, whereupon the oil particles, having a greater momentum than that of the refrigerant vapor, are caused to impinge on the surface of the screens or baffling. The oil then drains by gravity from the screens or baffles into the bottom of the separator, from where it is returned through a float valve to the compressor crankcase or, preferably, to the suction inlet of the compressor (Fig. 19-12).

The water-cooled, chiller-type separator, sometimes called an oil chiller, is similar in construction to the water-cooled condenser. Water is circulated through the tubes while the discharge vapor passes through the shell. The oil is separated from the vapor by precipitation on the cold water tubes, from where it drains into a drop-leg sump. The oil may be drained manually from the sump or returned automatically to the compressor through a float valve. (Fig. 19-23.) The water flow rate through the separator must be carefully controlled so that the refrigerant vapor is not cooled below its condensing temperature, in which case liquid

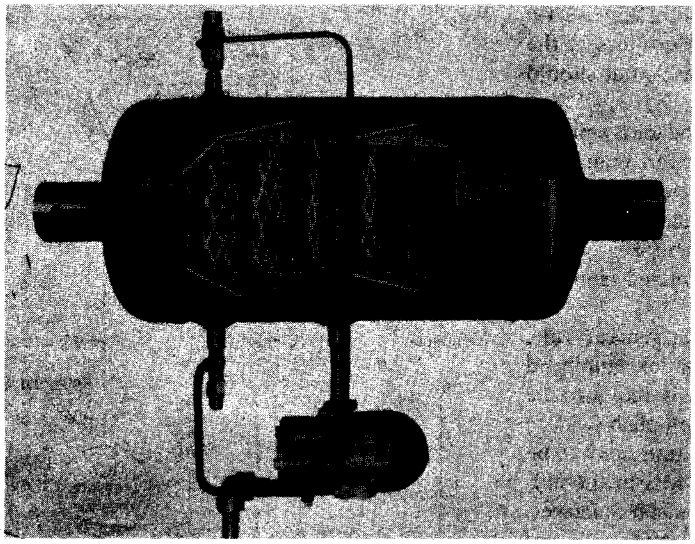


Fig. 19-22. Impingement-type oil separator with float drainer. (Courtesy York Corporation.)

refrigerant could be condensed in the separator and passed to the compressor crankcase through the float valve.

In some low temperature applications, a direct-expansion, chiller-type separator is installed in the liquid line. The operation and installation of the direct-expansion oil separator are similar to those of the direct-expansion liquid subcooler described in Section 19-10. The liquid refrigerant passing through the separator is chilled to a temperature below the pour point of the oil, thereby causing the oil to congeal on the chiller tubes. The oil is drained from the separator periodically by taking the separator out of service and allowing it to warm up to a temperature above the pour point of the oil.

Although properly applied oil separators are usually very effective in removing oil from the refrigerant vapor, they are not 100% efficient. Therefore, even when an oil separator is used, some means should be provided for removing the small amount of oil which will always pass through the separator and find its way into other parts of the system. Some of the various methods of accomplishing this are described in the following sections.

The principal hazard associated with the use of a discharge line oil separator is the possibility of liquid refrigerant passing from the oil separator to the compressor crankcase when the compressor is idle. The liquid refrigerant may drain into the separator from the discharge piping or it may condense in the separator itself during the off cycle.

While the compressor is operating, the temperature of the oil separator is relatively high and the possibility of liquid refrigerant condensing in the oil separator is rather remote, particularly if the separator is located reasonably close to the compressor. However, after the compressor cycles off, the separator tends to cool to the condensing temperature, at which time some of the high pressure refrigerant vapor is likely to condense in the separator. This raises the liquid level in the separator and causes the float to open and pass a mixture of oil and

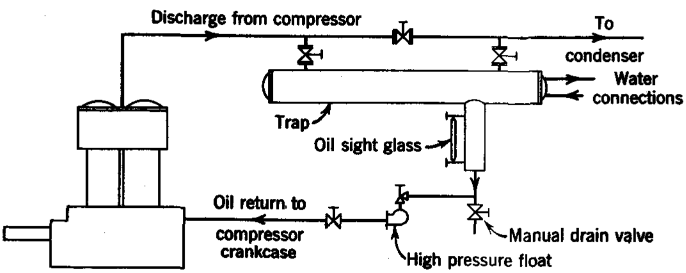


Fig. 19-23. Application of chiller-type oil separator. (Courtesy York Corporation.)

liquid refrigerant to the compressor crankcase. Condensation of the refrigerant in the separator is most apt to occur when the oil separator is installed in a cooler location than the condenser, in which case the liquid will boil off in the condenser and condense in the separator.

To eliminate the possibility of liquid refrigerant draining from the oil separator into the compressor crankcase during the off cycle, the oil drain line from the separator should be connected to the suction inlet of the compressor, rather than to the crankcase, and the line should be equipped with a solenoid valve, a sight glass, a hand expansion valve, and a manual shut-off valve (see Fig. 19-12). With the help of the sight glass, the hand throttling valve can be adjusted so that the liquid (oil and refrigerant) from the separator is bled slowly into the suction inlet of the compressor. The solenoid valve is interlocked with the compressor motor starter so that it is energized (open) only when the compressor is operating. This arrangement prevents the liquid refrigerant and oil in the separator from draining to the compressor during the off cycle, but permits slow draining into the suction inlet of the compressor when the compressor is operating.

To minimize the condensation of refrigerant vapor in the oil separator during the off cycle, oil separators should be installed near the compressor in as warm a location as possible. The separators should also be well insulated in order to retard the loss of heat from the separator after the compressor cycles off.

In some instances, the oil from the separator is drained into an oil receiver where it is stored until needed in the compressor crankcase (Fig. 17-39). The oil receiver contains a heating element which boils off the liquid refrigerant to the suction line. Oil from the oil receiver is admitted to the compressor crankcase as needed through a float valve located in the compressor crankcase.

Oil receivers cannot be used with compressors equipped with crankcase oil check valves. Since the oil receiver is at the suction pressure, the higher crankcase pressure could force all the oil out of the crankcase into the oil receiver. When oil receivers are employed, oil check valves must be removed and crankcase heaters installed.

19-13. Ammonia Piping. Since ammonia is a nonmiscible refrigerant, the oil pumped over into the discharge line from the compressor is not readily carried along through the system by the refrigerant. Therefore, an oil separator should be installed in the discharge line of all ammonia systems to reduce to a minimum the amount of oil that passes into the system. Provisions must be made also to remove from the system and return to the crankcase the small amount of oil that gets by the separator.

Oil, being heavier than liquid ammonia, tends to separate from the ammonia and settle out at various low points in the system. For this reason, oil sumps are provided at the bottom of all receivers, evaporators, accumulators, and other vessels in the system containing liquid ammonia, and provisions are made for draining the oil from these points either continuously or periodically. Since the amount of oil involved is small, when an operator is on duty the draining is usually done manually and the oil discarded. Of course, this requires that the crankcase oil be replenished periodically.

Since the lubricating oil is not returned to the compressor through the refrigerant piping, the minimum vapor velocity in ammonia piping is of no consequence and the piping is sized for a low pressure drop without regard for the minimum vapor velocity.

19-14. Nonoil Returning Halocarbon Evaporators. The design of some halocarbon evaporators is such that the oil reaching the evaporator cannot be entrained by the refrigerant vapor and carried over into the suction line and back to the compressor. The more common of these evaporators are flooded liquid chillers and certain types of air-cooling evaporators that are operated semiflooded by bottom-feeding with a thermostatic expansion valve. In both cases, the problem with oil return results from the lack of sufficient refrigerant velocity and turbulence in the evaporator to permit entraining the oil and carrying it over into the suction line.

With the semiflooded evaporator, oil return from the evaporator is usually accompanied by adjusting the expansion valve for a low superheat and slightly overfeeding the evaporator so that a small amount of the oil-rich liquid refrigerant in the evaporator is continuously carried over into the suction line. This arrangement will ordinarily keep the oil concentration in the evaporator within reasonable limits. As shown in Fig. 19-24, a liquid-suction heat exchanger is installed in the suction line to evaporate the liquid refrigerant from the oil-refrigerant mixture before the latter reaches the compressor.

Flooded liquid chillers are usually equipped with oil bleeder lines which permit a measured amount of the oil-rich liquid in the chiller to be bled off into the suction line or, in some instances, into an oil receiver. A throttling valve is installed in the bleeder line so that the flow rate through the line can be adjusted. The bleeder line also contains a solenoid valve which is wired to open only when the compressor is operating, so that flow through the bleed line does not occur when the compressor is idle.

Since Refrigerant-12 is completely oil miscible at all temperatures above the pour point of the oil, the oil bleeder connection on Refrigerant-12 chillers may be located at any point below the liquid level in the chiller. Refrigerant-22, on the other hand, is partially oil miscible at evaporator temperatures. Therefore, when refrigerant turbulence in the evaporator is relatively low, there is a tendency for the oil-refrigerant mixture in the evaporator to separate into two layers, with the upper layer containing the greater concentration of oil. For this reason, the oil bleeder connection of Refrigerant-22 chillers should be located just above the midpoint of the liquid level in the chiller.

As shown in Fig. 17-30, when the bleed off is directly to the suction line, a liquid-suction heat exchanger is required in order to evaporate the liquid refrigerant from the bleed mixture before it reaches the compressor. Figures 17-39 and 17-41 illustrate typical piping arrangements when bleed off is to an oil receiver.

19-15. Crankcase Piping for Parallel Compressors. When two or more compressors are operating in parallel off a common suction line, it is very unlikely that the oil returning through the suction line will be evenly distributed to the several compressors. Too, it is very unlikely that the amount of oil pumped over by any two compressors will be exactly the same even when the compressors are alike in both design and size. For these reasons, when compressors are piped for parallel operation, it is necessary to interconnect the crankcases of the several compressors both above and below the crankcase oil level, as shown in Fig. 19-11.

This requires that the compressors be so placed on their respective foundations that the oil tappings of the individual compressors are all at exactly the same level. The crankcase oil equalizing line may be installed either level with or, preferably, below the level of the crankcase oil tappings. Under no circumstances should the oil equalizing line be allowed to rise above the level of the crankcase tappings.

Since any small difference in the crankcase pressure of the several compressors will cause a difference in the crankcase oil levels, it is

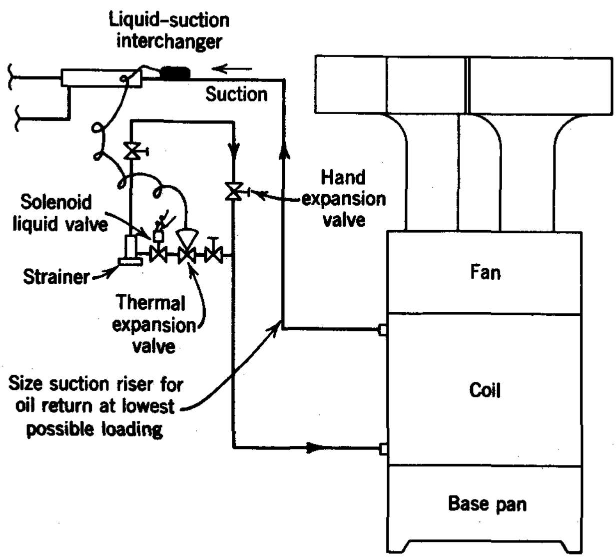


Fig. 19-24. Forced circulation air cooler with direct expansion feed of flooded-type coil. (Courtesy Carrier Corporation.)

refrigerating capacity and the power requirements of the compressor, although the horsepower required per ton increases.

With regard to compressor speed, the centrifugal compressor is much more sensitive to speed changes than is the reciprocating type. Whereas the change in the capacity of the reciprocating compressor is approximately proportional to the speed change, according to the performance curves in Fig. 18-31, a speed change of only 12% will cause a 50% reduction in the capacity of the centrifugal compressor.

18-25. Capacity Control. Capacity control of centrifugal compressors is usually accomplished by one of the following three methods: (1) varying the speed of the compressor, (2) varying the suction pressure by means of a suction throttling damper, or (3) varying the condensing temperature through control of the condenser water. Often, some combination of these methods is used.

Because of its extreme sensitivity to changes in speed, the centrifugal compressor is ideally suited for capacity regulation by means of variable speed drives, such as steam turbines and wound-rotor induction motors. When constant speed drives, such as synchronous or squirrel cage motors, are employed, speed control can be obtained through the use of a hydraulic or magnetic clutch installed between the drive and the step-up gear.

18-26. Centrifugal Refrigeration Machines. Centrifugal compressors are available for refrigeration duty only as an integral part of a centrifugal refrigerating machine. Because of the relatively flat head-capacity characteristic of the centrifugal compressor, and the resulting limitation in the operating range, the component parts of a centrifugal refrigerating system must be very carefully balanced. Too, since very marked changes in capacity accrue with only minor changes in the suction or condensing temperatures, the centrifugal refrigerating system

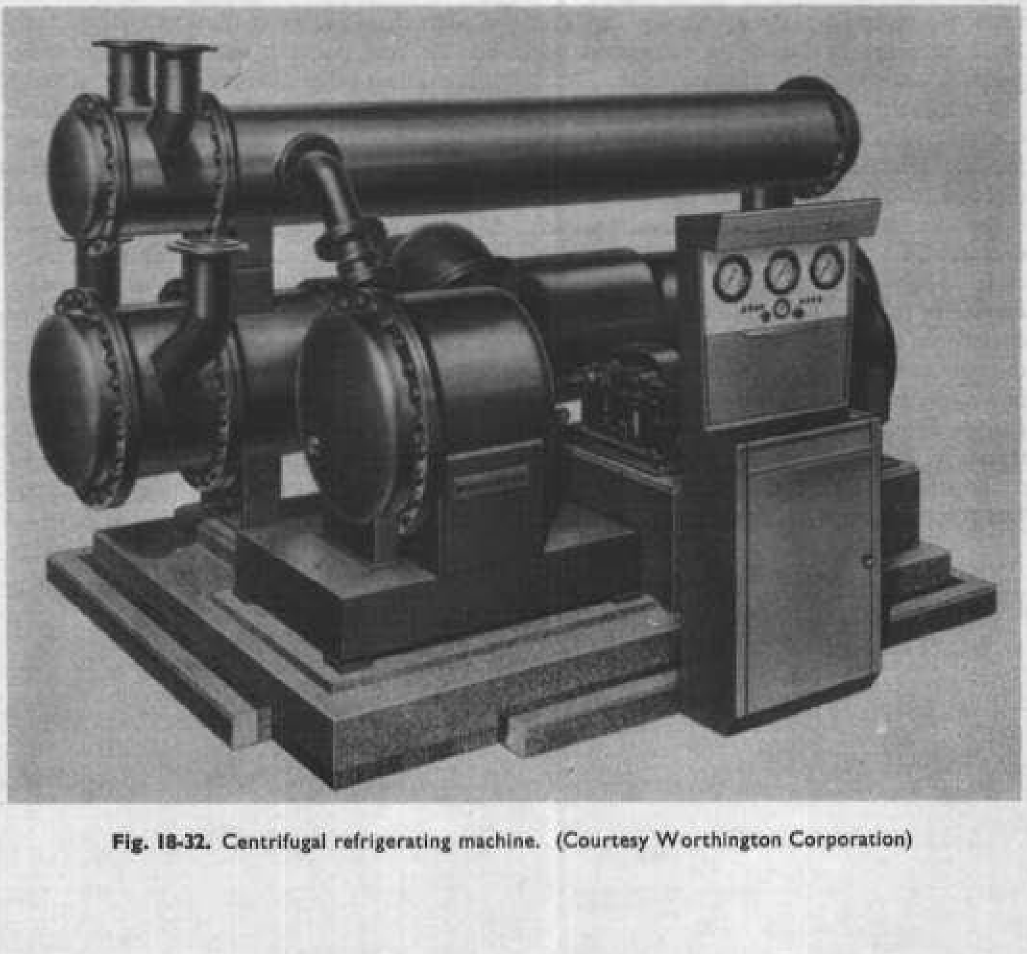


Fig. 18-32. Centrifugal refrigerating machine. (Courtesy Worthington Corporation)

the piping to other parts of the system. Where the compressor is piped with rigid piping, vibration eliminators (Fig. 19-1) installed in the suction and discharge lines near the compressor are usually effective in dampening compressor vibration. Vibration eliminators should be placed in a vertical line for best results.

On larger systems, adequate flexibility is ordinarily obtained by running the suction and discharge piping approximately 30 pipe diameters in each of two or three directions before anchoring the pipe. In all cases, isolation type hangers and brackets should be used when piping is supported by or anchored to building construction which may act as a sounding board to amplify and transmit vibrations and noise in the piping.

Although vibration and noise resulting from gas pulsations can occur in both the suction and discharge lines of reciprocating compressors, it is much more frequent and more intense in the discharge line. As a general rule, these gas pulsations do not cause sufficient vibration and noise to be of any consequence. Occasionally, however, the frequency of the pulsations and the design of the piping are such that resonance is established, with the result that the pulsations are amplified and sympathetic vibration (as with a tuning fork) is set up in the piping. In some instances, vibration can become so severe that the piping is torn loose from its supports. Fortunately, the condition can be remedied by changing the speed of the compressor, by installing a discharge muffler, and/or by changing the size of length of the discharge line. Since changing the speed of the compressor is not usually practical, the latter two methods are better solutions, particularly when used together.

When vibration and noise are caused by gas turbulence resulting from high velocity, the usual remedy is to reduce the gas velocity by increasing the size of the pipe. Sometimes this can be accomplished by installing a supplementary pipe.

19-5. General Design Considerations. Since many of the operational problems encountered in refrigeration applications can be traced directly to improper design and/or installation of the refrigerant piping and accessories, the importance of proper design and installation procedures cannot be overemphasized. In

Fig. 19-1. Vibration eliminator. (Courtesy Anaconda Metal Hose Division, The American Brass Company.)

general, refrigerant piping should be so designed and installed as to:

1. Assure an adequate supply of refrigerant to all evaporators
2. Assure positive and continuous return of oil to the compressor crankcase
3. Avoid excessive refrigerant pressure losses which unnecessarily reduce the capacity and efficiency of the system
4. Prevent liquid refrigerant from entering the compressor during either the running or off cycles, or during compressor start-up
5. Avoid the trapping of oil in the evaporator or suction line which may subsequently return to the compressor in the form of a large "slug" with possible damage to the compressor.

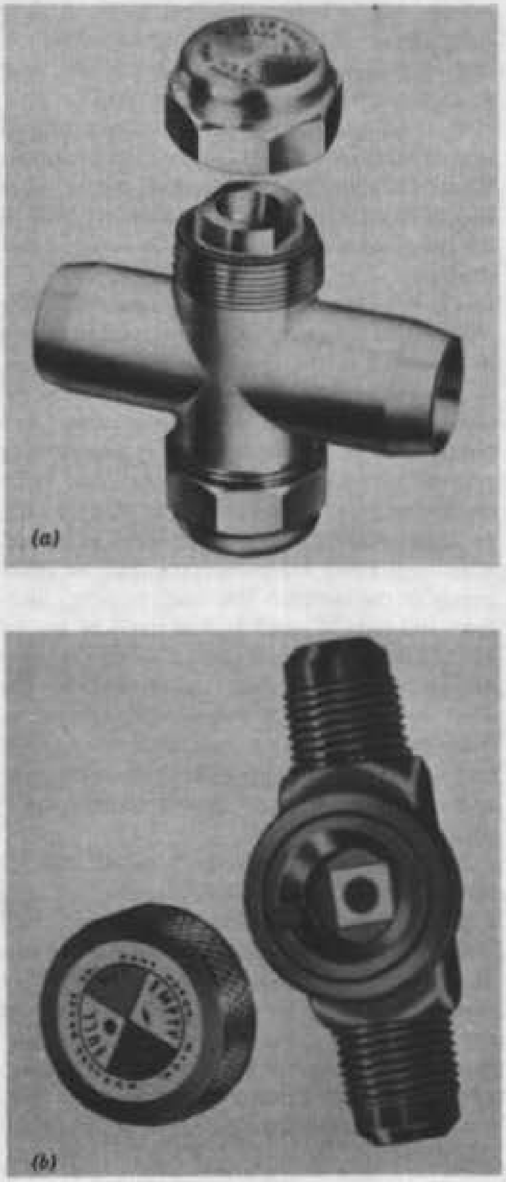


Fig. 19.25. Typical liquid indicators or sight glasses. Notice moisture indicator incorporated in single port sight glass. The color of the moisture indicator denotes the relative moisture content of the system. (*a*) Double port sight glass. (*b*) Single port sight glass. (Courtesy Mueller Brass Company.)

necessary also to equalize the crankcase pressures. This is done by interconnecting the crankcases above the crankcase oil level with a crankcase pressure equalizing line. This line may be installed level with, or above, the crankcase tappings of the compressors. The crankcase pressure equalizing line must not be allowed to drop below the level of the crankcase tappings and it must not contain any liquid traps of any kind.

Both the oil equalizing line and the crankcase pressure equalizing line should be the same size as the crankcase tappings. Manual shut-off valves should be installed in both lines between the compressors so that individual compressors can be valved off for maintenance or repairs without the necessity of shutting down the entire system.

19-16. Liquid Indicators (Sight Glasses). A liquid indicator or sight glass installed in the liquid line of a refrigerating system provides a means of determining visually whether or not the system has a sufficient charge of refrigerant. If the system is short of refrigerant, the vapor bubbles appearing in the liquid stream will be easily visible in the sight glass. The sight glass should be installed as close to the liquid receiver as possible, but far enough downstream from any valves so that the resulting disturbance does not appear in the sight glass. When liquid lines are long, an additional sight glass is frequently installed in front of the refrigerant control (or liquid line solenoid, when one is used) to determine if a solid stream of liquid is reaching the refrigerant control. Bubbles appearing in the sight glass at this point indicate that the liquid is flashing in the liquid line as a result of excessive pressure drop, in which case the bubbles can be eliminated only by reducing the liquid line pressure drop or by further subcooling of the liquid refrigerant. Typical sight glasses are shown in Fig. 19-25.

19-17. Refrigerant Dehydrators. Refrigerant driers (Fig. 19-26) are recommended for all refrigerating systems employing a halocarbon refrigerant. In small systems the drier is usually installed directly in the liquid line. In larger systems the by-pass arrangement shown in Fig. 19-27 is employed. With the latter method of installation, the drier cartridge can be removed and reinstalled without interrupting the operation of the system. Too, the drier can be used intermittently as needed. When the drier is not being used valves *A* and *B* are open and valve *C* is closed. When the drier is in service valves *B* and *C* are open and valve *A*

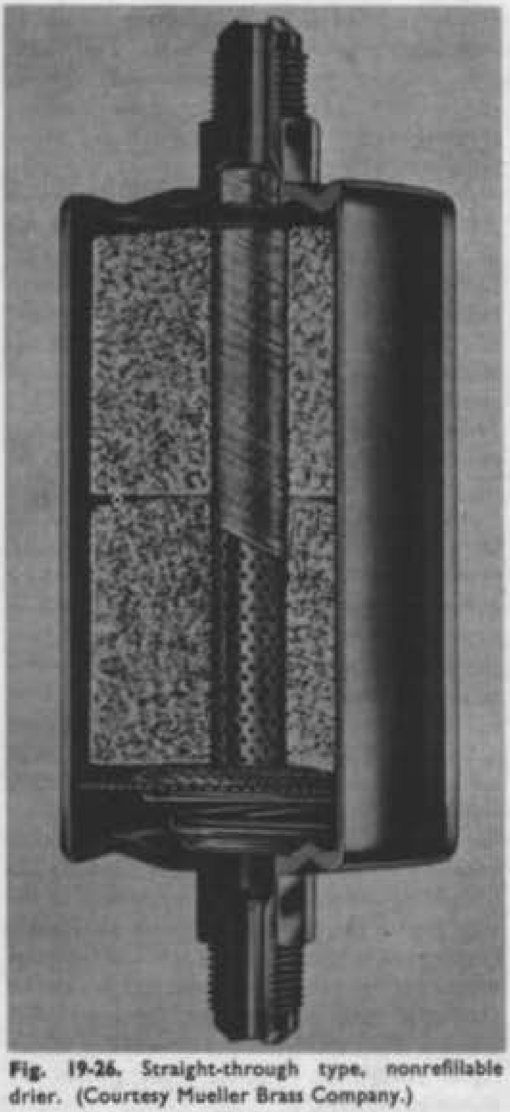


Fig. 19-26. Straight-through type, nonrefillable drier. (Courtesy Mueller Brass Company.)

is closed. Under no circumstances should valves *B* and *C* ever be closed at the same time, except when the drier cartridge is being changed. With valves *B* and *C* both closed, cold liquid could be trapped in the drier, which, upon warming, could create tremendous hydraulic pressures and burst the drier casing.

19-18. Strainers. Strainers should be installed immediately in front of all automatic valves in all refrigerant lines. When two or more automatic valves are installed close together, a single strainer, placed immediately upstream of the valves, may be used. In all cases, the strainer should be amply sized so that the accumulation of foreign material in the strainer will not cause an excessive refrigerant pressure drop.

Most refrigerant compressors come equipped with a strainer in the suction inlet chamber. When installing the refrigerant piping, care should be taken to arrange the suction piping at the compressor so as to permit servicing of this strainer.

19-19. Pressure Relief Valves. Pressure relief valves are safety valves designed to relieve the pressure in the system to the atmosphere, or to the out-of-doors through a vent line, in the event that the pressure in the system rises to an unsafe level for any reason. Most refrigerating systems have at least one pressure relief valve (or fusible plug) mounted on the receiver tank or water-cooled condenser. In many instances, additional relief valves are required at other points in the system. The exact number, location, and type of relief devices required are set forth in the American Standard Safety Code for Mechanical Refrigeration, and depends for the most part on the type and size of the system. Since local codes vary somewhat in this respect, they should always be considered when designing an installation. A typical pressure relief valve is illustrated in Fig. 19-28.

A fusible plug is sometimes substituted for the pressure relief valve. A fusible plug is simply a pipe plug which has been drilled and filled with a metal alloy designed to melt at some predetermined fixed temperature (Fig. 19-29). The design melting temperature of the fusible plug depends on the pressure-temperature relationship of the refrigerant employed in the system.

19-20. Receiver Tank Valves. Receiver tank valves are usually of the packed type, equipped

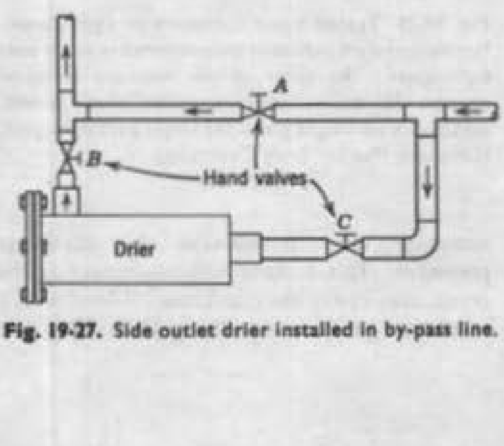


Fig. 19-27. Side outlet drier installed in by-pass line.

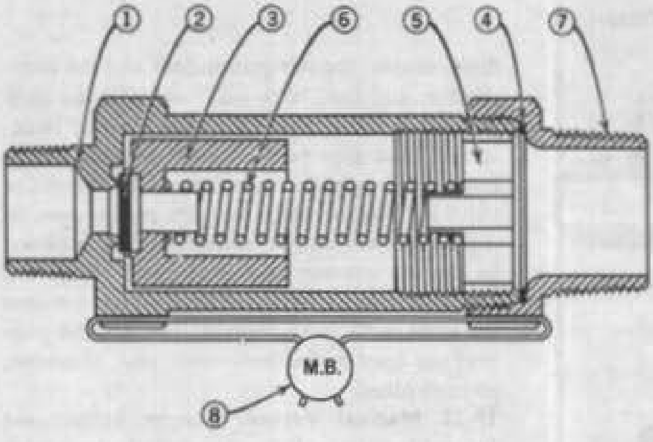


Fig. 19-28. Typical pressure relief valve. (Courtesy Mueller Brass Company.)

1. Valve body
2. Seat disc
3. Disc holder
4. Gasket
5. Spring retainer
6. Spring
7. Outlet connection
8. Lead seal and locking wire (prevents alteration of factory setting)

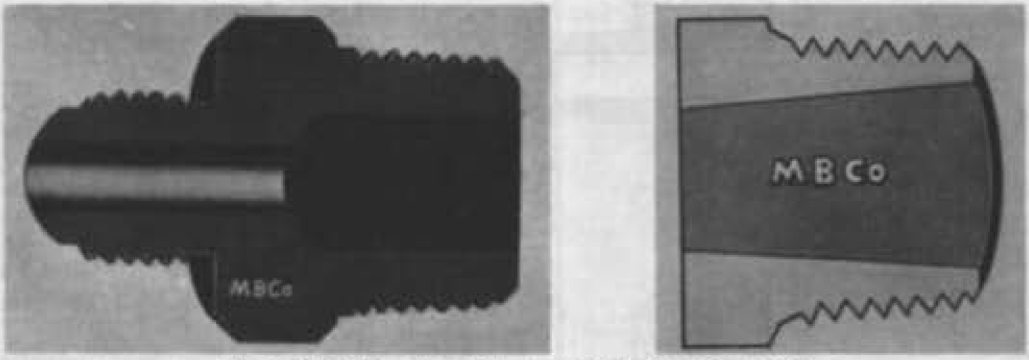


Fig. 19-29. Fusible plugs. (Courtesy Mueller Brass Company.)

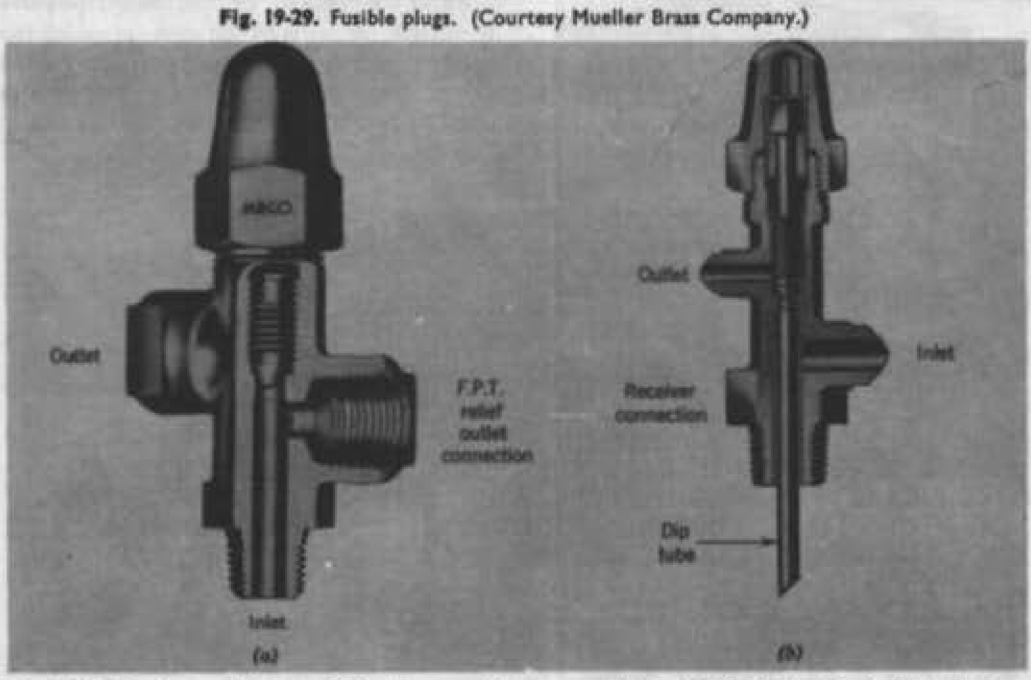


Fig. 19-30. Receiver tank valves. (a) Angle type with pressure relief outlet (nonbackseating). (b) Angle type with dip tube (nonbackseating). (Courtesy Mueller Brass Company.)

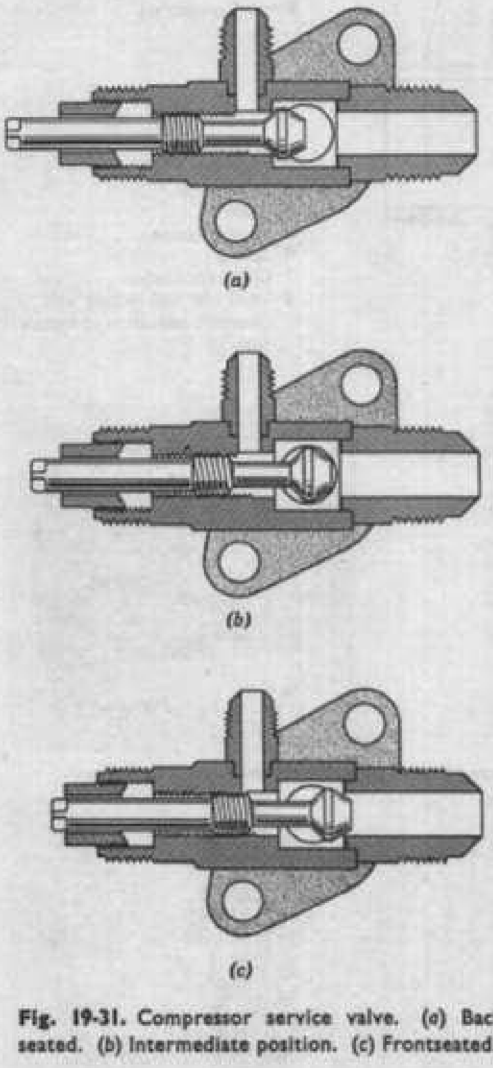


Fig. 19-31. Compressor service valve. (a) Backseated. (b) Intermediate position. (c) Frontseated.

with a cap seal and designed for direct installation on the receiver tank. (Fig. 19-30.) When designed for installation on top of the receiver tank, the valves must be provided with dip tubes so that the liquid refrigerant can be drawn from the bottom of the receiver. Some valves also have tappings to accommodate a relief valve or fusible plug.

19-21. Compressor Service Valves. Compressor service valves are usually designed to bolt directly to the compressor housing. As shown in Fig. 19-31, they have both "front" and "back" seats. The "front seat" controls the flow between the refrigerant lines and the compressor, and the "back seat" controls the gage port of the valve. When the valve stem is "backseated," the gage port is closed and the refrigerant line is open to the compressor. When the valve is "front-seated," the gage port is open to the compressor and the refrigerant line is closed to the compressor and gage port. With the valve stem in an intermediate position between the seats, both the refrigerant line and the gage port are open to the compressor and, of course, to each other.

19-22. Manual Valves. Manual valves used for refrigeration duty may be of the globe, angle, or gate type. Since the piping code prohibits the use of gate valves in refrigerant lines, except in large installations where an operating attendant is on duty, they are used primarily in water and brine lines. Gate valves have a very low pressure drop, but do not permit throttling and therefore can be employed only where full-flow or no-flow conditions are desired. Both globe and angle valves are suitable for throttling. Since the angle valve offers the least resistance to flow, its use is recommended whenever practical.

Either the "packed" or "packless" type valve is suitable for refrigeration duty, provided the valve has been designed for that purpose. Packed valves should be of the back-seating type in order to permit packing under pressure and to

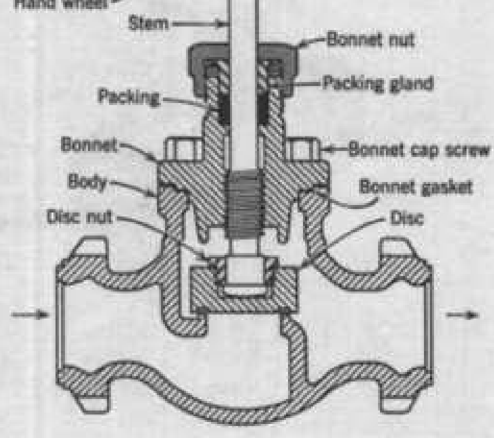


Fig. 19-32. Packed-type manual valve. (Courtesy Vilter Manufacturing Company.)

reduce the possibility of leakage through the packing in the full-open position (Fig. 19-32.) Many packed valves are equipped with cap seals which completely cover and seal the valve stem, thereby eliminating the possibility of leakage when the valve is not in use.

PROBLEMS

1. A Refrigerant-12 system with a design capacity of 65 tons is operating with a suction temperature of 40° F and a condensing temperature of 110° F. The suction line is 40 ft long and contains 2 elbows and a 10 ft riser. The discharge line is 60 ft long and has 5 elbows and a 15 ft riser. The condenser to receiver liquid line is 50 ft long with 3 elbows. The liquid line is 50 ft long with 4 elbows and a 30 ft riser. Using Type L copper tube for the refrigerant piping, determine:

(*a*) The suction pipe size. *Ans.* $3\frac{1}{8}$ in. OD

(*b*) The over-all pressure drop (°F) in the suction pipe at design capacity. *Ans.* 2° F

(*c*) The refrigerant velocity in the suction pipe at design capacity. *Ans.* 4200 fpm

(*d*) The refrigerant velocity in the suction pipe at 50% of design capacity. *Ans.* 2100 fpm

(*e*) The discharge pipe size. *Ans.* $2\frac{5}{8}$ in. OD

(*f*) The refrigerant velocity in the discharge pipe at 50% of design capacity. *Ans.* 1100 fpm

(*g*) Liquid line size. *Ans.* $1\frac{5}{8}$ in. OD

(*h*) The over-all pressure drop in the liquid line at design capacity. *Ans.* 4 psi

(*i*) The condenser to receiver pipe size. *Ans.* $2\frac{5}{8}$ in. OD

2. Determine the size suction and discharge risers required to insure adequate oil return when the system in Problem 1 is operating at 25% of design capacity.
Ans. (*a*) $2\frac{5}{8}$ in. OD (*b*) $2\frac{1}{2}$ in. OD

3. Rework Problem 1 using Refrigerant-22.
Ans. (*a*) $2\frac{5}{8}$ in. OD (*b*) 1.23° F (*c*) 4000 fpm (*d*) 2000 fpm (*e*) $2\frac{1}{8}$ in. OD (*f*) 2000 fpm (*g*) $1\frac{3}{8}$ in. OD (*h*) 7 psi (*i*) $2\frac{5}{8}$ in, OD

4. Rework Problem 2 using Refrigerant 22.
Ans. (*a*) $2\frac{1}{8}$ in. OD (*b*) $1\frac{1}{2}$ in. OD

20

Defrost Methods, Low-Temperature Systems, and Multiple Temperature Installations

20-1. Defrosting Intervals. The necessity for periodically defrosting air-cooling evaporators which operate at temperatures low enough to cause frost to collect on the evaporator surface has already been established. How often the evaporator should be defrosted depends on the type of evaporator, the nature of the installation, and the method of defrosting. Large, bare-tube evaporators, such as those employed in breweries, cold storage plants, etc., are usually defrosted only once or twice a month. On the other hand, finned blower coils are frequently defrosted as often as once or twice each hour. In some low temperature installations defrosting of the evaporator is continuous by brine spray or by some antifreeze solution.

In general, the length of the defrost period is determined by the degree of frost accumulation on the evaporator and by the rate at which heat can be applied to melt off the frost. For the most part, the degree of frost accumulation will depend on the type of installation, the season of the year, and the frequency of defrosting. As a general rule, the more frequently the evaporator is defrosted the smaller is the frost accumulation and the shorter is the defrost period required.

20-2. Methods of Defrosting. Defrosting of the evaporator is accomplished in a number of different ways, all of which can be classified as either "natural defrosting" or "supplementary-heat defrosting" according to the source of the heat used to melt off the frost. Natural defrosting, sometimes called "shut-down" or "off-cycle" defrosting, utilizes the heat of the air in the refrigerated space to melt the frost from the evaporator, whereas supplementary-heat defrosting is accomplished with heat supplied from sources other than the space air. Some common sources of supplementary heat are water, brine, electric heating elements, and hot gas from the discharge of the compressor.

All methods of natural defrosting require that the system (or evaporator) be shut down for a period of time long enough to permit the evaporator temperature to rise to a level well above the melting point of the frost. The exact temperature rise required and the exact length of time the evaporator must remain shut down in order to complete the defrosting vary with the individual installation and with the frequency of defrosting. However, in every case, since the heat to melt the frost comes from the space air, the temperature in the space must be allowed to rise to whatever level is necessary to melt off the evaporator frost, which is usually about 37° to 40° F. For this reason, natural defrosting is not ordinarily practical in any installation when the design space temperature is below 34° F.

The simplest method of defrosting is to shut the system down manually until the evaporator warms up enough to melt off the frost, after which the system is started up again manually. When several evaporators connected to the same condensing unit are located in different spaces or fixtures, the evaporators can be taken out of service and defrosted one at a time by manually closing a shut-off valve located in the liquid line of the evaporator being defrosted. When defrosting is completed, the evaporator is put back into service by opening the shut-off valve.

If automatic defrosting is desired, a clock timer can be used to shut the system down for a fixed period of time at regular intervals. Both the number and the length of the defrost periods can be adjusted to suit the individual installation. As a general rule, natural convection evaporators are defrosted only once a day, in which case the defrost cycle is usually started around midnight and lasts for several hours. On the other hand, unit coolers should be defrosted at least once every 3 to 6 hr. Since it is usually undesirable to keep the system out of service for any longer than is necessary, the length of the defrost period should be carefully adjusted so that the system is placed back in service as soon as possible after defrosting.

In one variation of the time defrost, the defrost cycle is initiated by the defrost timer and terminated by a temperature or pressure control that is actuated by the evaporator temperature or pressure. With this method, the defrost period is automatically adjusted to the required length, since the evaporator temperature (or pressure) will rise to the cut-in setting of the control as soon as defrosting is completed.

The most common method of natural defrosting is the "off-cycle" defrost. As described in an earlier chapter, off-cycle defrosting is accomplished by adjusting the cycling control so that the evaporator temperature rises to 37° F or 38° F during every off cycle. If the system has been properly designed, the evaporator will be maintained relatively free of frost at all times, since it will be completely defrosted during each off cycle.

20-3. Water Defrosting. For evaporator temperatures down to approximately minus 40° F, defrosting can be accomplished by spraying water over the surface of the evaporator coils. For evaporator temperatures below minus 40° F, brine or some antifreeze solution should be substitued for the water. A typical water defrost system is illustrated in Fig. 20-1.

Although water defrosting can be made automatic, it is often designed for manual operation. Ordinarily, the following procedure is used to carry out the defrosting:

1. A stop valve in the liquid line is closed and the refrigerant is evacuated from the evaporator,

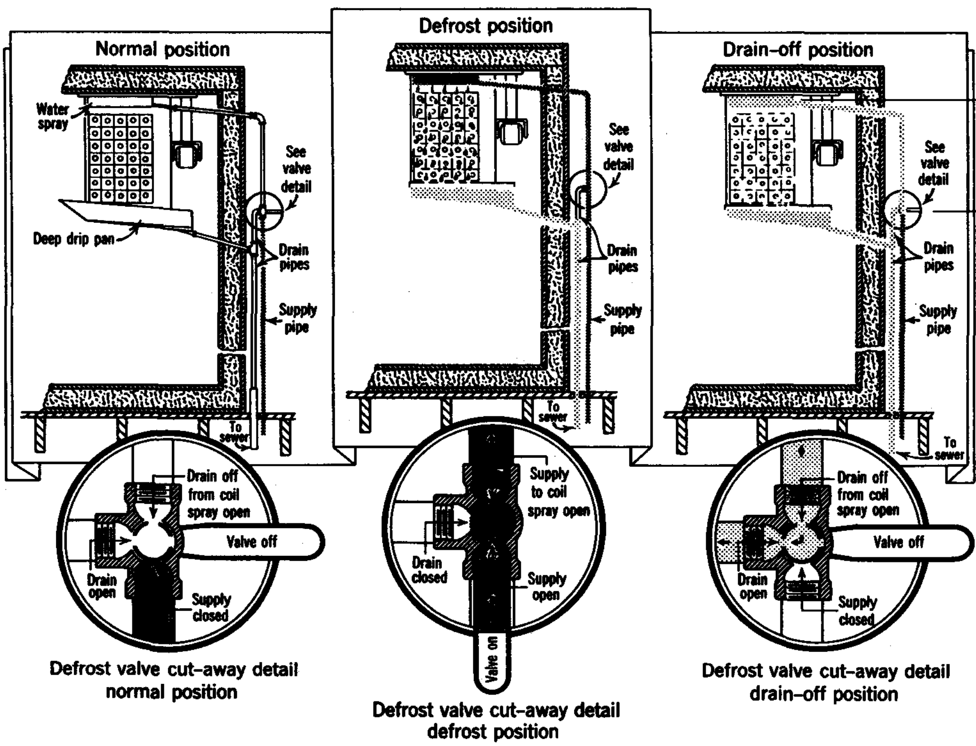


Fig. 20-1. Typical water defrost system. (Courtesy Dunham-Bush, Inc.)

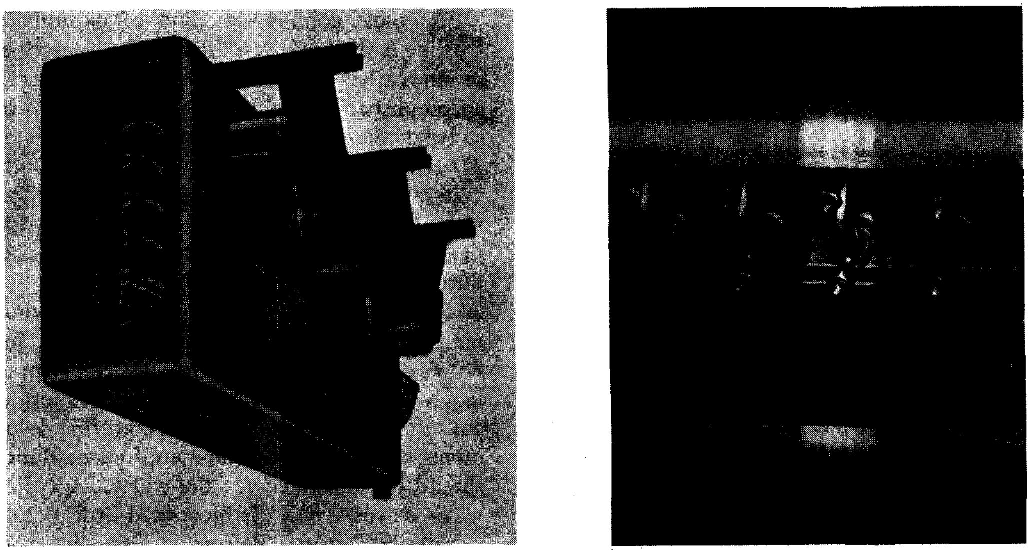


Fig. 20-2. Evaporator equipped for electric defrosting. Heater elements are installed through the center of the tubes. Inset shows details of mechanical sealing of the heater elements. (Courtesy Dunham-Bush, Inc.)

after which the compressor is stopped and the evaporator fans are turned off so that the water spray is not blown out into the refrigerated space. If the evaporator is equippped with louvers, these are closed to isolate further the evaporator and prevent fogging of the refrigerated space.

2. The water sprays are turned on until the evaporator is defrosted, which requires approximately 4 to 5 minutes. After the sprays are turned off, several minutes are allowed for draining of the water from the evaporator coils and drain pan before the evaporator fans are started and the system put back in operation.

To eliminate the possibility of water freezing in the drain line, the evaporator should be located close to an outside wall and the drain line should be amply sized and so arranged that the water is drained from the space as rapidly as possible. A trap is installed in the drain line outside the refrigerated space to prevent warm air being drawn into the space through the drain line during normal operation. In some instances, a float valve is employed in the drain pan to shut off the water spray and prevent overflowing into the space in the event that the drain line becomes plugged with ice.

When brine or an antifreeze solution replaces the water spray, the defrosting solution is returned to a reservoir and recirculated, rather than wasted. Unless the reservoir is large enough so that the addition of heat is not required, some means of reheating the solution in the reservoir may be necessary. Since the water from the melting frost will weaken the solution, the defrost system is equipped with a "concentrator" to boil off the excess water and return the solution to its initial concentration.

One manufacturer circulates a heated glycol solution through the inner tube of a double-tube evaporator coil. The principal advantage gained is that the glycol is not diluted by the melting frost.

20-4. Electric Defrosting. Electric resistance heaters are frequently employed for the defrosting of finned blower coils. An evaporator equipped with defrost heaters is shown in Fig. 20-2. Ordinarily, the drain pan and drain line are also heated electrically to prevent refreezing of the melted frost in these parts.

The electric defrost cycle can be started and stopped manually or a defrost timer may be used to make defrosting completely automatic. In either case, the defrosting procedure is the same. The defrost cycle is initiated by closing a solenoid valve in the liquid line causing the evaporator to be evacuated, after which the compressor cycles off on low pressure control.

At the same time, the heating elements in the evaporator are energized and the evaporator fans turned off so that the heat is not blown out into the refrigerated space. After the evaporator is defrosted, the heaters are de-energized and the system put back in operation by opening the liquid line solenoid and starting the evaporator fans.

20-5. Hot Gas Defrosting. Hot gas defrosting has many variations, all of which in some way utilize the hot gas discharged from the compressor as a source of heat to defrost the evaporator. One of the simplest methods of hot gas defrosting is illustrated in Fig. 20-3. A by-pass equipped with a solenoid valve is installed between the compressor discharge and the evaporator. When the solenoid valve is opened, the hot gas from the compressor discharge by-passes the condenser and enters the evaporator at a point just beyond the refrigerant control. Defrosting is accomplished as the hot gas gives up its heat to the cold evaporator and condenses into the liquid state. Some of the condensed refrigerant stays in the evaporator while the remainder returns to the compressor where it is evaporated by the compressor heat and recirculated to the evaporator.

This method of hot gas defrosting has several disadvantages. Since no liquid is vaporized in the evaporator during the defrost cycle, the amount of hot gas available from the compressor will be limited. As defrosting progresses, more liquid remains in the evaporator and less refrigerant is returned to the compressor for recirculation, with the result that the system tends to run out of heat before the evaporator is completely defrosted.

Another, and more serious, disadvantage of this method is the possibility that a large slug of liquid refrigerant will return to the compressor and cause damage to that unit. This is most likely to occur either at the beginning of the defrost cycle or immediately after defrosting is completed.

Fortunately, both these weaknesses can be overcome by providing some means of re-evaporating the liquid which condenses in the evaporator before it is returned to the compressor. The particular means used to re-evaporate the liquid is the principal factor distinguishing one method of hot gas defrosting from another.

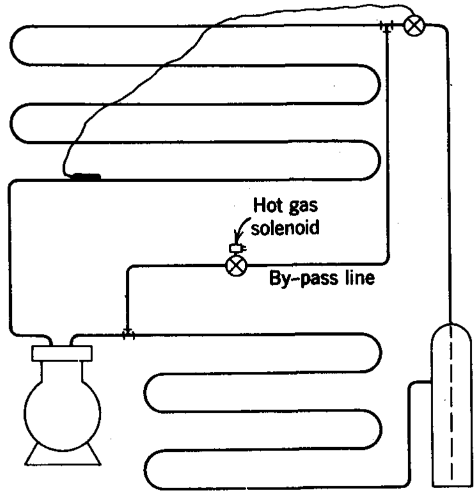


Fig. 20-3. Simple hot gas defrost system.

20-6. Re-evaporator Coils. One common method of hot gas defrosting employs a re-evaporator coil in the suction line to re-evaporate the liquid, as shown in Fig. 20-4. During the normal running cycle the solenoid valve in the suction line is open and the suction vapor from the evaporator by-passes the re-evaporator coil in order to avoid an excessive suction line pressure loss. At regular intervals (usually 3 to 6 hr) the defrost timer starts the defrost cycle by opening the solenoid in the hot gas line and closing the solenoid in the suction by-pass line. At the same time, the evaporator fans are stopped and the re-evaporator fan is started. The liquid condensed in the evaporator is re-evaporated in the re-evaporator coil and returned as a vapor to the compressor, where it is compressed and recirculated to the evaporator. When defrosting is completed, the defrost cycle may be terminated by the defrost timer or by an evaporator-temperature actuated temperature control. In either case, the system is placed back in operation by closing the hot gas solenoid, opening the suction solenoid, stopping the re-evaporator fan and starting the evaporator fans.

20-7. Defrosting Multiple Evaporator Systems. When two or more evaporators are connected to a common condensing unit, the evaporators may be defrosted individually, in which case the operating evaporator can serve as

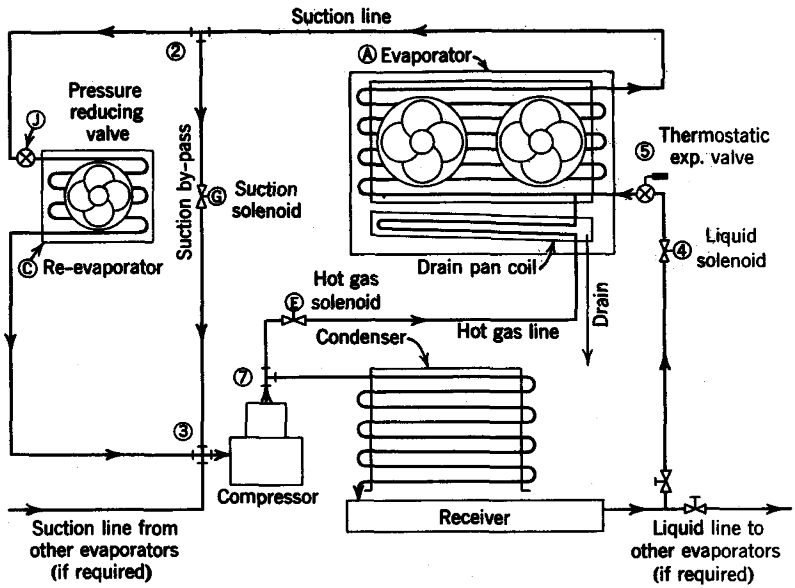


Fig. 20-4. Hot gas defrost system employing re-evaporator coil. (Courtesy Kramer-Trenton Company.)

a re-evaporator for the refrigerant condensed in the evaporator being defrosted. A flow diagram of this arrangement is illustrated in Fig. 20-5.

20-8. Reverse Cycle Defrosting. By employing the reverse cycle (heat pump) principle, the condenser can be utilized as a re-evaporator coil to re-evaporate the refrigerant that condenses in the evaporator during the defrost cycle. An automatic expansion valve is used to meter the liquid refrigerant into the condenser for re-evaporation. Flow diagrams for normal operation and defrosting are shown in Figs. 20-6(*a*) and 20-6(*b*), respectively. Modern practice replaces valves *A*, *B*, *C*, and *D* of Fig. 20-6 with a single four-way valve as illustrated in Fig. 20-7.

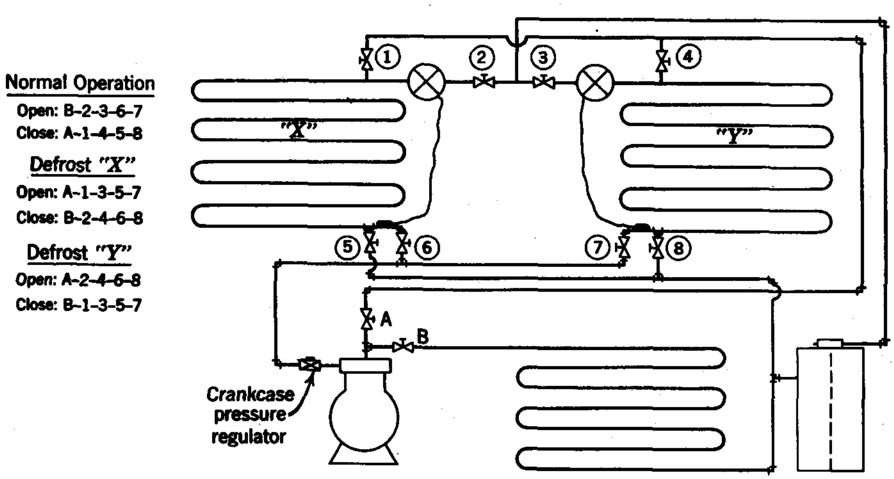


Fig. 20-5. Hot gas defrost—multiple evaporator system.

20-9. Heat Bank Defrosting. The Thermobank* method of hot gas defrosting employs a water bank to store a portion of the heat ordinarily discarded at the condenser when the evaporator is being refrigerated. During the defrost cycle, the heat stored in the water bank is used to re-evaporate the refrigerant condensed in the defrosting evaporator.

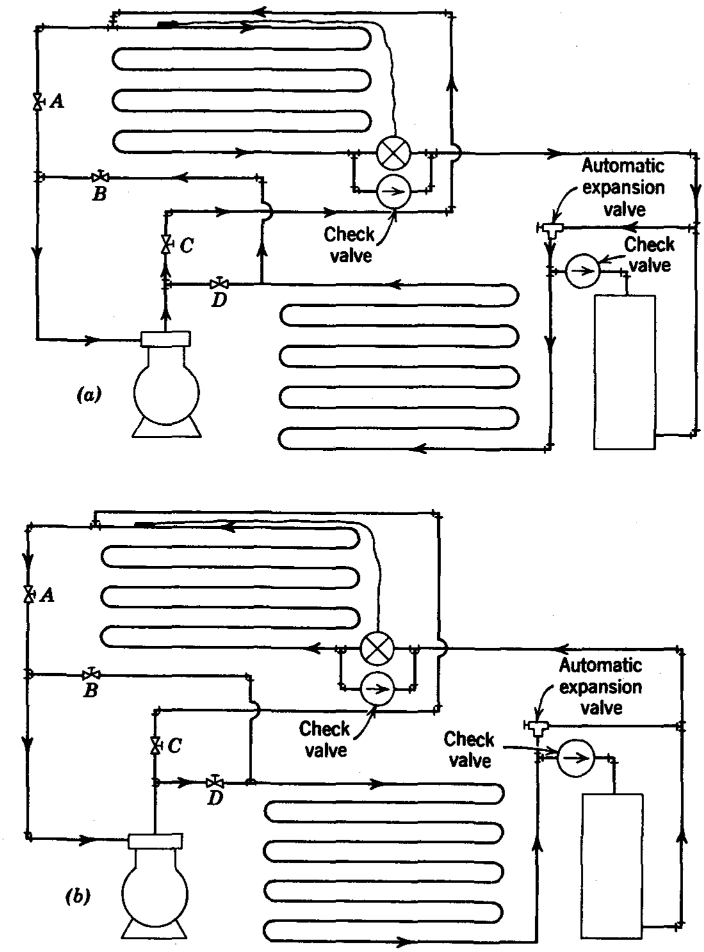


Fig. 20-6. (*a*) Reverse cycle hot gas defrost system (defrost cycle). (*b*) Reverse cycle hot gas defrost system (normal operation).

During normal operation (Fig. 20-8*a*), the discharge gas from the compressor passes through the heating coil in the water bank first and then goes to the condenser, so that a portion of the heat ordinarily discarded at the condenser is stored in the bank water. Notice that the suction vapor by-passes the holdback valve and bank during the refrigerating cycle in order to avoid unnecessary suction line pressure loss and superheating of the suction vapor by the bank water. Also, to control the maximum water bank temperature, a by-pass is built into the water bank heating coil. The by-pass is so sized that a greater portion of the discharge gas by-passes the heater coil and flows directly to the condenser as the temperature of the bank water increases.

When the frost reaches a predetermined thickness, the defrost cycle (Fig. 20-8*b*) is initiated by an electric timer which opens the hot gas solenoid valve, closes the suction solenoid valve, and stops the evaporator fans. Hot gas is discharged into the evaporator where it condenses and defrosts the coil. The condensed refrigerant flows to the holdback valve which acts as a constant pressure expansion

* A proprietary design of the Kramer-Trenton Company.

valve and feeds liquid to the re-evaporator coil immersed in the bank water. In this process, the bank water actually freezes on the outside of the re-evaporator coil. The heat stored in the bank is transferred to the refrigerant which evaporates completely in the re-evaporator coil. liquid refrigerant in the coil and suction line is re-evaporated. The timer then returns the system to normal operation. When normal operation is resumed, the bank water is promptly restored to its original temperature by the hot gas passing through the heating coil.

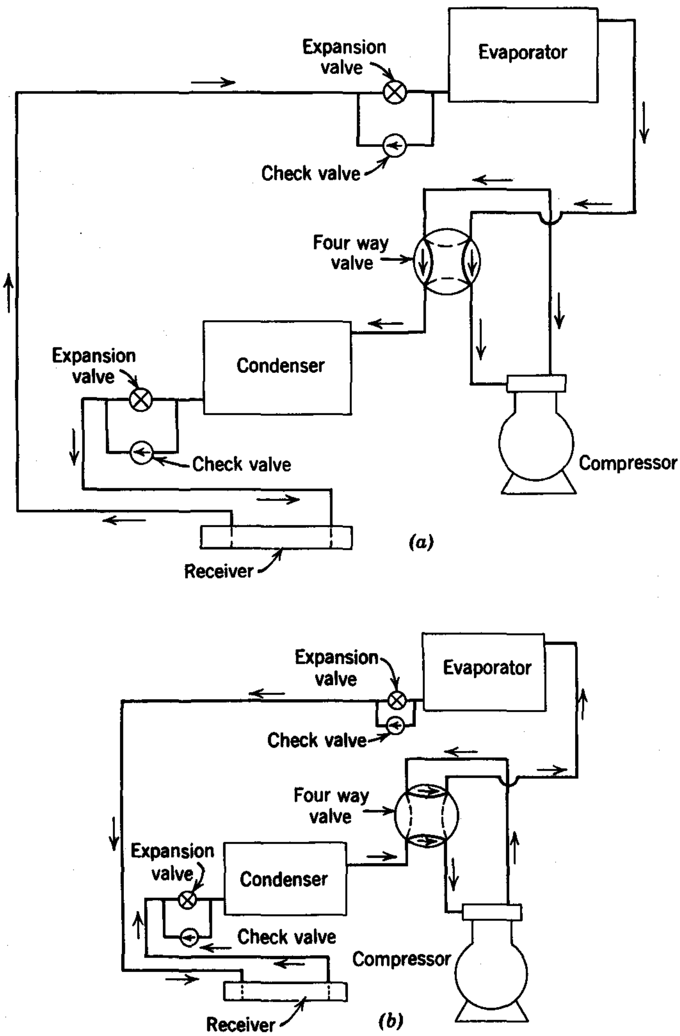


Fig. 20-7. (*a*) Reverse cycle hot gas defrost—normal operation. (*b*) Reverse cycle hot gas defrost—defrost cycle.

Thus both sensible and latent heat are abstracted from the bank water, making available vast heat quantities for fast defrost and the refrigerant returns to the suction inlet of the compressor completely evaporated.

Defrost is completed in approximately 6 to 8 min. This is followed by a post-defrost period lasting a few minutes after the closing of the hot gas solenoid valve. During post-defrost any

20-10. Vapot Defrosting. A schematic diagram of the Vapomatic defrosting system is shown in Fig. 20-9, along with an enlarged view of the Vapot,* which is the heart of this hot gas defrost system. The Vapot, which is actually a specially designed suction line accumulator, traps the liquid refrigerant condensed in the

* Proprietary designs of Refrigeration Engineering, Inc.

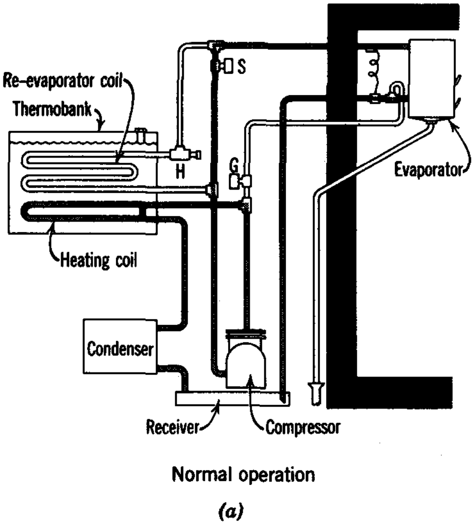


Fig. 20-8. Thermobank hot gas defrosting. (Courtesy Kramer-Trenton Company.)

evaporator and, by means of a carefully sized bleed tube, continuously feeds a measured amount of the liquid back to the compressor with the suction vapor. The small amount of liquid feeding back to the compressor is vaporized by the heat of compression and returned to the evaporator. In this way, the Vapot provides a continuous source of latent heat for defrosting the evaporator and at the same time eliminates the possibility of large slugs of liquid returning to the compressor. The heat exchanger in the Vapot has no significance in the defrost cycle.

The defrost cycle is initiated by a defrost timer which opens the hot gas solenoid valve and stops the evaporator fans. An evaporator temperature control terminates the defrost and restores the system to normal operation.

20-11. Multistage (Booster) Compression. It has already been established that the capacity and efficiency of any refrigerating system diminish rapidly as the difference between the suction and condensing temperatures is increased by a reduction in the evaporator temperature. The losses experienced are due partially to the rarification of the suction vapors at the lower

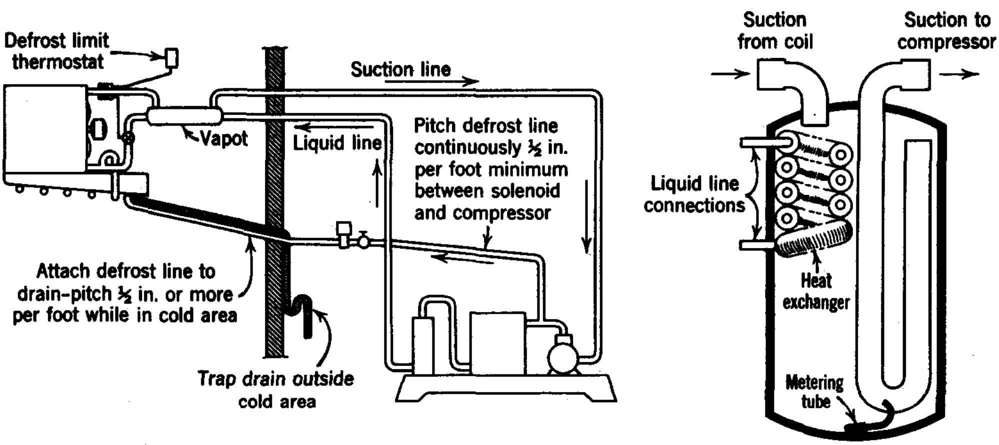


Fig. 20-9. (Left) Typical application of Vapot. (Right) Enlarged view of Vapot. (Courtesy Recold Corporation.)

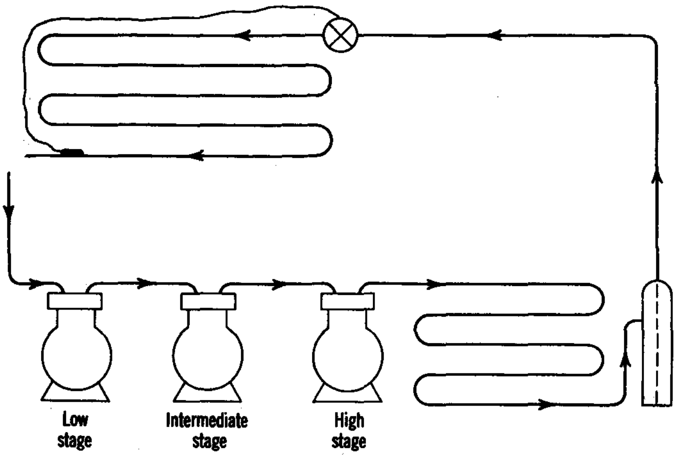


Fig. 20-10. Three-stage, direct staged compression system.

evaporator temperatures and partially to the increase in the compression ratio. Since any increase in the ratio of compression is accompanied by a rise in the discharge temperature, discharge temperatures also tend to become excessive as the evaporator temperature is reduced.

Whereas conventional single-stage systems will usually give satisfactory results with evaporator temperatures down to $-40°$ F, provided that condensing temperatures are reasonably low, for evaporator temperatures below minus 40° F, some form of multistage compression must be employed in order to avoid excessive discharge temperatures and to maintain reasonable operating efficiencies. In larger installations, multistage operation should be considered for any evaporator temperature below 0° F.

All methods of accomplishing multistage compression can be grouped into two basic types: (1) direct staging and (2) cascade staging. The direct staging method employs two or more compressors connected in series to compress a single refrigerant in successive stages. A flow diagram of a simple, three-stage, direct staged multicompression system is shown in Fig. 20-10. Notice that the pressure of the refrigerant vapor is raised from the evaporator pressure to the condenser pressure in three increments, the discharge vapor from the lower stage compressors being piped to the suction of the next higher stage compressor.

Cascade staging involves the use of two or more separate refrigerant circuits which employ refrigerants having progressively lower boiling points (Fig. 20-11). The compressed refrigerant vapor from the lower stage is condensed in a heat exchanger, usually called a cascade condenser, which is also the evaporator of the next higher stage refrigerant.

Both methods of multistaging have relative advantages and disadvantages. The particular method which will produce the best results in a given installation depends for the most part on the size of the installation and on the degree of low temperature which must be attained. In some instances, a combination of the cascade and direct staging methods can be used to an advantage. In these cases, the compound compression (direct staging) is usually applied to the lower stage of the cascade.

20-12. Intercoolers. With direct staging, cooling of the refrigerant vapor between the several stages of compression (desuperheating) is necessary in order to avoid overheating of the higher stage compressors. Since the refrigerant gas is superheated during the compression process, if the gas is not cooled before it enters the next stage compressor, excessive discharge temperatures will result with subsequent overheating of the higher stage machines.

Because of the large temperature differential between the condenser and the evaporator, cooling of the liquid refrigerant is also desirable in order to avoid heavy losses in refrigerating effect because of excessive flashing of the liquid in the refrigerant control, and the accompanying increase in the volume of vapor which must be handled by the low stage compressor.

Three common methods of gas desuperheating and liquid cooling for direct staged systems are illustrated in Fig. 20-12. The intercooler shown in Fig. 20-12*a* is an "open" or "flash" type intercooler. The liquid from the condenser is expanded into the intercooler where its temperature is reduced by flashing to the saturation temperature corresponding to the intercooler pressure. Since suction from the intercooler is taken into the high stage compressor, the temperature of the liquid leaving the intercooler to go to the low temperature evaporator is the saturation temperature corresponding to the intermediate pressure (pressure between stages). Since the refrigeration expended in cooling the liquid to the intermediate temperature is accomplished much more economically at the level of the high stage suction than at that of the low stage suction, cooling of the liquid in the intercooler has the effect of reducing the horsepower per ton as well as the displacement required for the low stage compressor.

The discharge gas from the low stage compressor is desuperheated by causing it to bubble up through the liquid in the intercooler, after which the discharge gas passes to the suction of the high stage compressor along with the flash gas from the intercooler.

The principal advantages of the flash type intercooler are in its simplicity and low cost, and the fact that the temperature of the liquid refrigerant is reduced to the saturation temperature corresponding to the intermediate pressure. The chief disadvantage is that the pressure of the liquid going to the evaporator is reduced to the intermediate pressure in the intercooler. This reduces the pressure drop available at the expansion valve and necessitates oversizing of the valve, which often results in sluggish operation. Too, since the low-temperature, low-pressure liquid leaving the intercooler is saturated, there is a tendency for the liquid to flash in the liquid line between the intercooler and the evaporator. For this reason, the liquid line should be designed for the minimum possible pressure drop.

A shell-and-coil intercooler, sometimes called a "closed type" intercooler, is illustrated in Fig. 20-12*b*. This type of intercooler differs from the flash type in that only a portion of the liquid from the condenser is expanded into the intercooler, whereas the balance, that portion going to the evaporator, passes through the coil submerged in the intercooler liquid. Therefore, with the shell-and-coil type intercooler, the pressure of the liquid is not reduced to the intermediate pressure, that is, the liquid cooling occurs in the form of subcooling rather than as a reduction in the saturation temperature. The advantages gained by this method, of course, are the higher liquid pressures made available at the expansion valve and the elimination of flash gas in the liquid line. With good intercooler design the liquid can be cooled to within 10 to 20 degrees of the saturation temperature corresponding to the intermediate pressure.

Vapor velocities in both types of flooded intercoolers should be limited to a maximum of

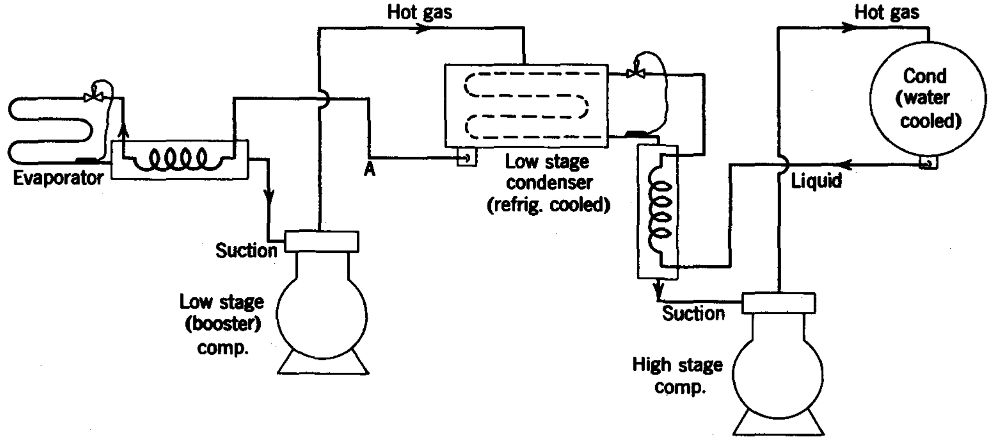


Fig. 20-11. Cascade system (two-stage). (Courtesy Carrier Corporation.)

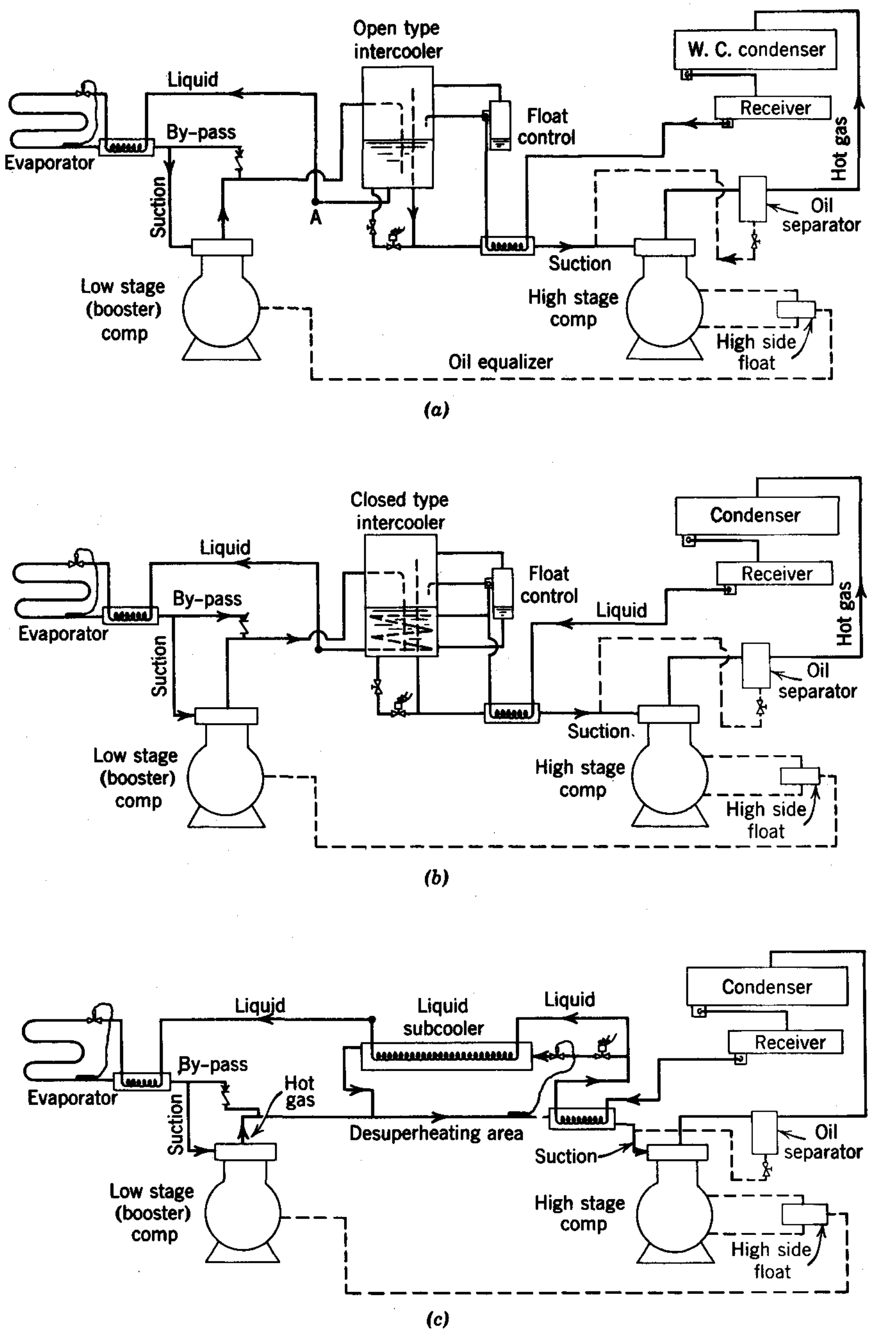


Fig. 20-12. Various types of gas and liquid intercoolers. (*a*) Direct staged system—open, flash-type intercooler. (*b*) Direct staged system—closed, shell-and-coil type intercooler. (*c*) Direct staged system—dry-expansion intercooler. (Courtesy Carrier Corporation.)

200 fpm and ample separation area should be allowed above the liquid level in the intercoolers in order to prevent liquid carryover into the high stage compressor.

Another arrangement, employing a dry-expansion intercooler, is shown in Fig. 20-12*c*. This type of intercooler is not suitable for ammonia systems, but is widely used with Refrigerants-12 and 22. The liquid from the condenser is subcooled as it passes through the coil in the intercooler. Desuperheating is accomplished by overfeeding of the intercooler so that a small amount of liquid is carried over into the desuperheating area where it is vaporized by the hot gas from the discharge of the low stage compressor. The gas is cooled by vaporizing the liquid and by mixing with the cold vapor from the intercooler.

Since ammonia has a very high latent heat value, liquid cooling is not as important in ammonia systems as in systems employing fluorocarbon refrigerants. For this reason, liquid cooling is sometimes neglected in ammonia systems, in which case the discharge vapor from the low stage compressor is usually desuperheated by injecting a small amount of liquid ammonia directly into the line connecting the low and high stage compressors (Fig. 20-13). The vaporization of the liquid ammonia in this line provides the necessary gas cooling.

In some ammonia systems, the discharge gas is cooled in a water-cooled intercooler that is similar in design to the shell-and-tube or shell-and-coil water-cooled condensers. The effectiveness of this type of intercooler depends on the temperature of the available water and increases as the available water temperature decreases. As a general rule, water-cooled intercoolers will not produce enough gas cooling to lower the power requirements, but will usually provide sufficient cooling to keep the discharge temperature within the maximum limit and thereby prevent overheating of the high stage compressor.

20-13. Direct Staging vs. Cascade Staging. Direct staging requires the use of refrigerants that have boiling points low enough to provide the low temperatures desired in the evaporator and which, at the same time, are condensable under reasonable pressures with air or water at normal temperatures. This requirement tends to limit the degree of low temperature that can be attained by the direct staging method. The practical low limit is approximately −125° F with either Refrigerants-12 or 22 and −90° F with ammonia. Below these temperatures cascade staging is ordinarily required, with some high-pressure, low boiling point refrigerant, such as ethane, ethylene, R-13, R-13Bl, or R-14, being used in the lower stage. Because of their extremely high pressures at normal condensing temperatures and/or their relatively low critical temperatures, these high pressure refrigerants must be condensed at rather low temperatures and therefore are cascaded with R-12, R-22, or propane. A three-stage cascade system employing ethylene, methane, and propane in the low, intermediate, and high stages respectively is illustrated in Fig. 20-14.

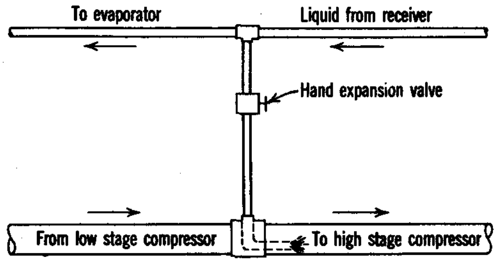


Fig. 20-13. Liquid injection gas intercooling.

The chief disadvantage of cascade staging is the overlap of refrigerant temperatures in the cascade condenser, which tends to reduce the thermal efficiency of the system somewhat below that of the direct staged system. On the other hand, cascade staging makes possible the use of high density, high pressure refrigerants in the lower stages, which will usually result in a considerable reduction in the displacement required for the low stage compressor. The use of high pressure refrigerants also simplifies the design of the low stage evaporator in that higher refrigerant pressure losses through the evaporator can be permitted without incurring excessive losses in system capacity and efficiency. Too, since the refrigerants in the several stages do not intermingle, and each stage is a separate system within itself, the problem of oil return to the compressors is somewhat less critical than in the direct staged system.

A single stage Refrigerant-12 system and a two-stage, direct staged Refrigerant-12 system operating between the same temperature limits

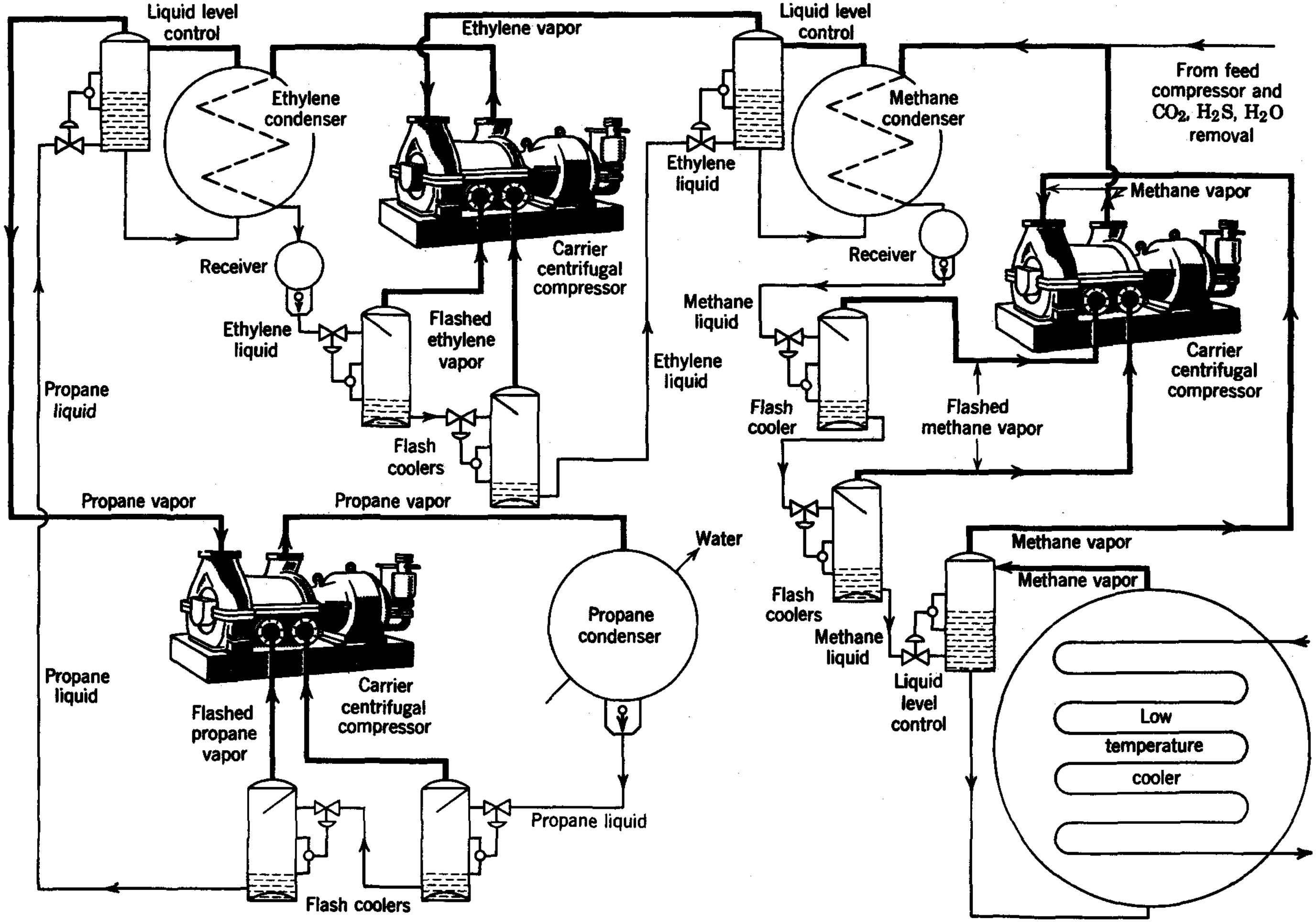


Fig. 20-14. Three-stage cascade system employing ethylene, methane, and propane in the low, intermediate, and high stages, respectively. (Courtesy Carrier Corporation)

are compared on pressure-enthalpy coordinates in Fig. 20-15. If the volumetric and compression efficiencies of the compressors were considered, the difference between the single and multistage compression systems would be approximately twice as great as that indicated by the values because of the difference in the compression ratios.

20-14. Oil Return in Multistage Systems. Since the several stages of the cascade system are actually separate and independent systems, oil return is accomplished in the individual stages in the same manner as in any other single stage system operating under the same conditions. This is not true of the direct staged system. Whenever two or more compressors are interconnected, either in parallel or in series, *there is no* assurance that oil return to the individual compressors will be in equal amounts. Therefore, some means of insuring equal distribution of the oil among the several compressors must be provided. When the compressors are connected in parallel, the oil can be maintained at the same level in all the compressors by interconnecting the crankcases as shown in Fig. 19-11. However, this simple method of oil equalization requires that the crankcase pressures in the several compressors be exactly the same (Section 19-15) and therefore is not practical when the compressors are connected in series, since the higher pressures existing in the crankcase of the high stage compressors would force the oil through the equalizing lines into the lower pressures existing in the crankcases of the low stage compressors.

One common method of equalizing the oil levels in the crankcases of compressors connected in series is shown in Fig. 20-12. An oil separator installed in the discharge line of the high stage compressor separates the oil from the discharge gas and returns it to the suction inlet of that machine. High side float valves maintain the desired oil level in the high stage compressors by continuously draining the excess oil returning to these compressors to the next lower stage compressor through the oil transfer lines. For manual operation, hand stop valves (normally closed) can be substituted for the float drainers, in which case the hand valves are opened periodically to adjust the oil levels by bleeding oil from the higher stage compressors to the lower stage compressors.

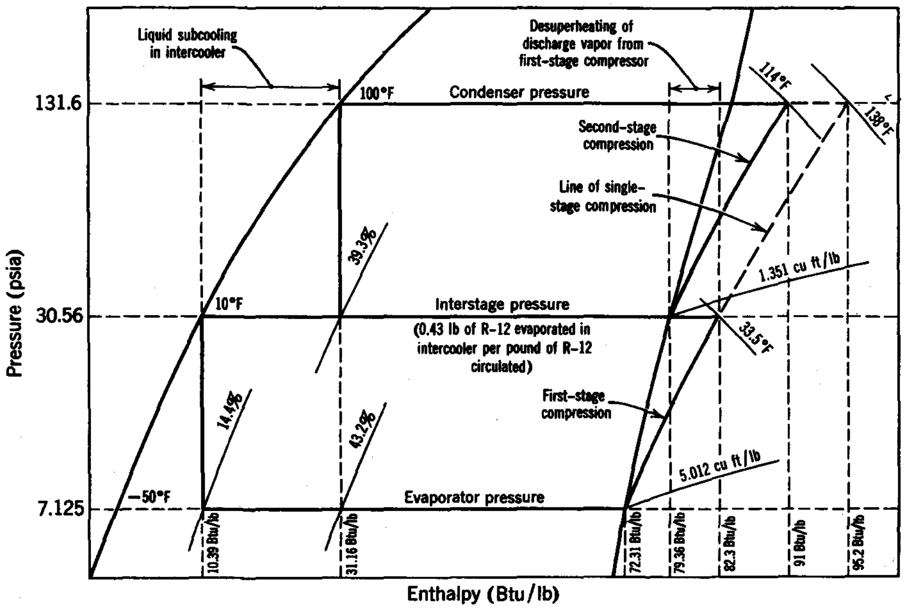


Fig. 20-15. *Ph* **diagram of two-stage direct-staged R-12 system with flash intercooler. First-stage compressor displacement—16.19 cfm/ton. Second-stage compressor displacement—6.24 cfm/ton including vapor from intercooler. Compression ratio for each stage is approximately 4.3 to 1.**

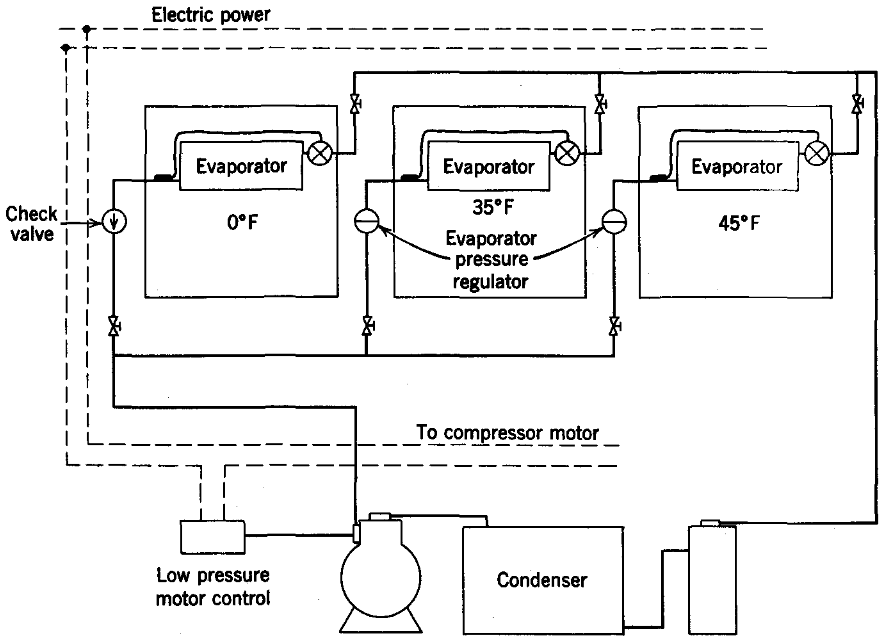


Fig. 20-16. Three-evaporator multiple temperature installation. Manual stop valves in suction and liquid lines permit isolation of the individual evaporators for maintenance.

20-15. Multiple Temperature System. A multiple temperature system is one wherein two or more evaporators operating at different temperatures and located in different spaces or fixtures (or sometimes in the same space or fixture) are connected to the same compressor or condensing unit. The chief advantages gained by this type of operation are a savings in space and a reduction in the initial cost of the equipment. However, since higher operating costs will usually more than offset the initial cost advantage, a multiple temperature system is economically justifiable only in small capacity installations where operating costs in any case are relatively small. One obvious disadvantage of the multiple temperature system is that in the event of compressor breakdown all spaces served by the compressor will be without refrigeration, thereby causing the possible loss of product which otherwise would not occur.

A typical three-evaporator multiple temperature system is illustrated in Fig. 20-16. An evaporator pressure regulator valve is installed in the suction line of each of the warmer evaporators in order to maintain the pressure, and therefore the saturation temperature of the refrigerant, in these units at the desired high level. A check valve is installed in the suction line of the lowest temperature evaporator to prevent the higher pressure from the warmer evaporators from backing up into the cold evaporator when the former are calling for refrigeration. Since the check valve will remain closed as long as the pressure in the suction main is above the pressure in the low temperature evaporator, it is evident that the low temperature evaporator will receive little, if any, refrigeration until the refrigeration demands of the high temperature evaporators are satisfied.

For this reason, if a multiple temperature system is to perform satisfactorily, the load on the lowest temperature evaporator must account for at least 60%, and preferably more, of the total system load. When the high temperature evaporator(s) constitute more than 40% of the total load, the refrigeration demands of the high temperature evaporator(s) will cause the compressor to operate a greater portion of the time at suction pressures too high to permit adequate refrigeration of the low temperature evaporator, with the result that temperature control in that unit will be erratic.

The evaporator pressure regulators installed in the suction line of the warmer evaporators may be either the throttling or snap-action types, the latter being employed whenever "off-cycle" defrosting of the high temperature evaporator is desired (Section 17-23).

In multiple temperature systems, a low pressure motor control is ordinarily used to cycle the compressor on and off, the cut-in and cut-out pressures of the control being adjusted to suit the conditions required in the low temperature evaporator.

When the compressor is operating, the pressure at the suction inlet of the compressor will depend on the rate at which vapor is being generated in all the evaporators. If the combined load on the several evaporators is high, the suction pressure will also be high. When the refrigeration demands of one of the higher temperature evaporators is satisfied, the evaporator pressure regulator will close (or throttle) so that little or no vapor enters the suction main from that unit, thereby causing a reduction in the suction pressure. As previously mentioned, whether or not the low temperature evaporator is being refrigerated at any given time depends on whether the suction pressure is below or above the pressure in that evaporator. However, the low temperature evaporator will always be open to the compressor (refrigerated) at any time that the demands of the high temperature evaporators are satisfied and the regulator valves are closed (or throttled). When the demands of the low temperature evaporator are also satisfied, the suction pressure will drop below the cut-out setting of the low pressure control and the compressor will cycle off.

With the compressor on the off cycle, any one of the evaporators is capable of cycling the compressor on again. For the low temperature evaporator, a gradual rise in pressure in that unit to the cut-in pressure of the low pressure control will start the compressor, whether or not either of the high temperature evaporators also requires refrigeration.

Since the pressure maintained in the high temperature evaporators, even at the lower limit, is always above the cut-in setting of the low pressure control, if either of the evaporator pressure regulators opens one of the high temperature evaporators to the suction main, the suction pressure will immediately rise above the cut-in setting of the low pressure control and the compressor will cycle on.

The fact that the pressure in the high temperature evaporators is always above the cut-in setting of the low pressure control poses somewhat of a problem when throttling-type evaporator pressure regulators are employed on the high temperature evaporators. Since this type of regulator is open any time the evaporator pressure is above the pressure setting of the regulator, it will often cause the compressor to cycle on when the evaporator it is controlling does not actually require refrigeration. As soon as the compressor starts, the pressure in the evaporator is immediately reduced below the regulator setting and the regulator closes, causing the compressor to cycle off again on the low pressure control.

To prevent short cycling of the compressor on the low pressure control when throttling type evaporator pressure regulators are employed, it is usually necessary to install a solenoid stop valve in the suctions line of the high temperature evaporators in order to obtain positive shut-down of these evaporators during the compressor off-cycle. With pilot operated evaporator pressure regulators, positive close-off of the regulator during the compressor off cycle can be obtained by controlling the regulator with a temperature or solenoid pilot.

In small systems, a surge tank installed in the suction main will usually prevent short cycling of the compressor. Because of the relatively large volume of the surge tank, it is capable of absorbing reasonable pressure increases occurring in the high temperature evaporator and thereby preventing the suction pressure from rising prematurely to the cut-in pressure of the low pressure control. The size of the surge tank required depends on the ratio of the high temperature load to the low temperature load and on the temperature differential between the high and low temperature, the size of the tank required increasing as each of these factors increases. Obviously, this method of preventing short cycling is best applied to systems where the high temperature load is only a small portion of the total load and/or where the difference in temperature between the high and low temperature evaporators is relatively small.

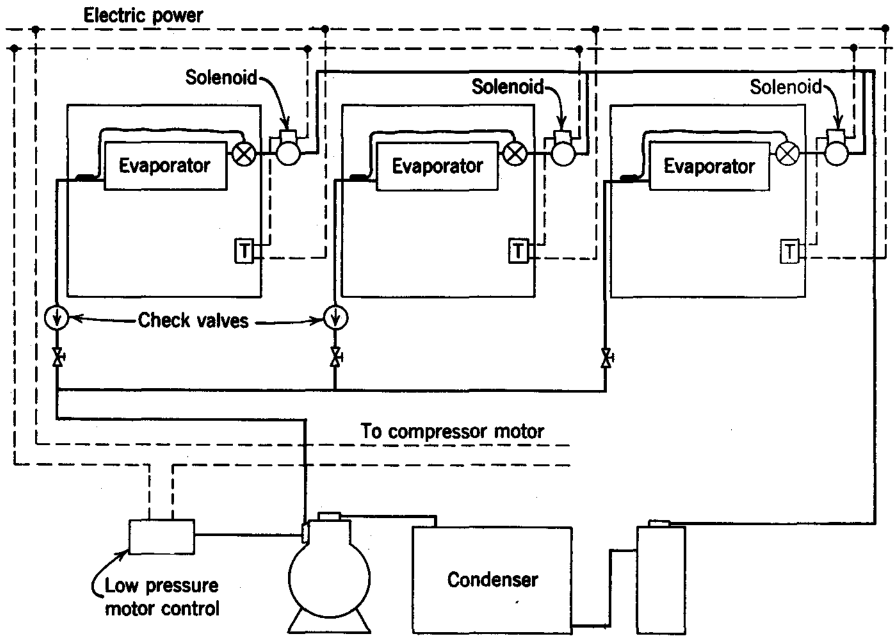


Fig. 20-17. Multiple unit installation employing thermostat-solenoid control. Manual stop valves in suction lines permit isolation of individual evaporators for maintenance.

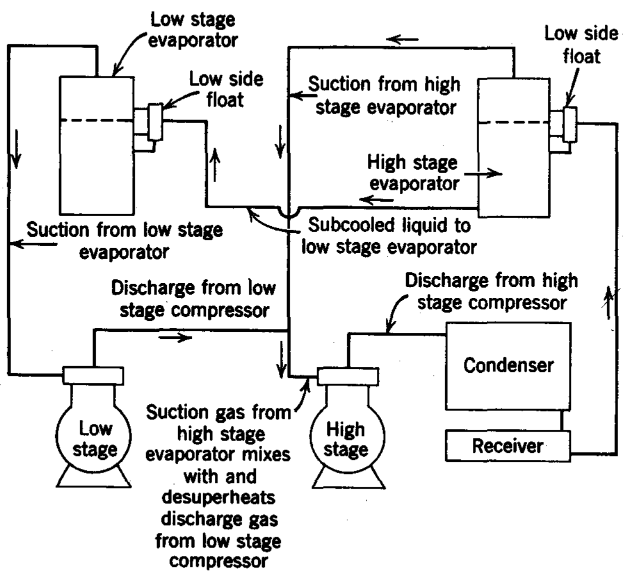


Fig. 20-18. A multiple temperature system employing two-staged direct staged compression.

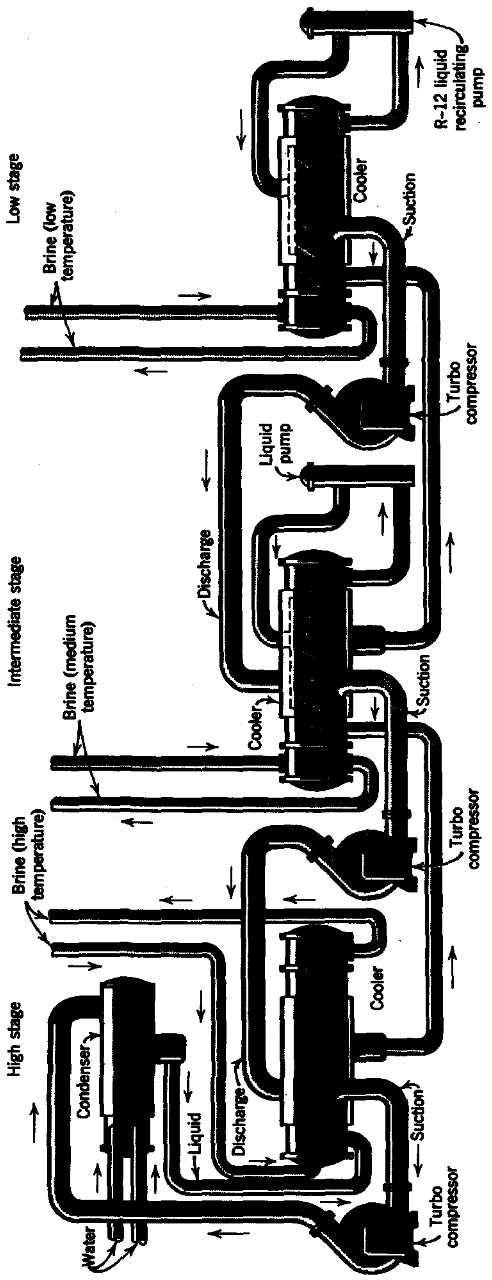


Fig. 20-19. Three-stage, multiple temperature system employing Refrigerant-12 with centrifugal compressors. (Courtesy York Corporation.)

20-16. Solenoid Controlled Multiple Temperature Systems. Thermostatically controlled solenoid stop valves, installed in either the liquid or the suction lines of the high temperature evaporators, are frequently used to obtain multiple temperature operation. A typical installation employing solenoid valves in the liquid line is shown in Fig. 20-17. The operation of this type of system is similar to that of systems employing snap-action regulators in the suction line, except that there is no control of the evaporator pressure and temperature. A space-temperature actuated thermostat controls the solenoid valves. When the space temperature rises, the thermostat contacts close, energizing the solenoid coil and opening the liquid line to the evaporator. The pressure in the evaporator rises as the liquid enters the evaporator, causing the low pressure control to cycle the compressor on, if the latter is not already running. If the compressor is already running, the entrance of liquid into the evaporator will cause a rise in the operating suction pressure.

When the temperature in the space is reduced to the desired low level, the thermostat contacts open, de-energizing the solenoid coil and closing the liquid line, whereupon the evaporator pumps down to the operating suction pressure, or to the cut-out pressure in the event that none of the other evaporators is calling for refrigeration. Since all the evaporators pump-down as they are cycled out, the system receiver tank must be large enough to hold the entire system refrigerant charge. This is not true, however, if the solenoids are installed in the suction line rather than in the liquid line.

Placing the solenoid valves in the suction line has the disadvantage of requiring larger, more expensive valves. Too, in the event of a leaky refrigerant control, there is always the danger that liquid will accumulate in the evaporator while the suction solenoid is closed and flood back to the compressor when the solenoid is opened.

When solenoids are employed to obtain multiple temperature operation, there is no direct control of the evaporator pressure and temperature, since the pressure in the evaporator at any given time will depend upon the number of evaporators open to the compressor. Obviously, with this type of operation it is very difficult to maintain a balanced relationship between the space and evaporator temperatures, and therefore humidity control in the refrigerated space becomes very indefinite. For this reason, when humidity control is important, it is usually necessary to install a throttling-type evaporator pressure regulator in the suction lines of the high temperature evaporators in order to maintain the pressure and temperature in these evaporators at the necessary high level. However, evaporator pressure regulators should be used only with suction line solenoids. Short cycling of the compressor may result if they are employed in conjunction with liquid line solenoids.

With either suction line or liquid line solenoids, check valves should be installed in the suction lines of the lower temperature evaporators to avoid excessive pressures and temperatures in these units when the warmer evaporators are open to the suction line.

20-17. Multiple Temperature Operation in Staged Systems. A multiple temperature system employing direct staged compression is illustrated in Fig. 20-18. Notice that the high temperature evaporator serves also as the gas and liquid cooler for the lower stage. The high stage compressor must be selected to handle the high temperature load in addition to the load passed along by the low stage compressor. A three-stage, multiple temperature, cascade-staged system employing Refrigerant-12 in all three stages is shown in Fig. 20-19.

21

Electric Motors and Control Circuits

21-1. Electric Motors. Single-phase and three-phase alternating current motors of various types are employed in the refrigerating industry as drives for compressors, pumps, and fans. A few two-phase and direct current motors are also used on occasion.

Single-phase motors range in size from approximately $\frac{1}{20}$ hp up through 10 hp, whereas three-phase motors are available in sizes ranging from approximately $\frac{1}{6}$ hp on up, although the latter are seldom employed in sizes below 1 hp. When three-phase power is available, the three-phase motor is usually preferred to the single-phase type in integral horsepower sizes because of its greater simplicity and lower cost.

Practically all power in the United States is generated as 60-cycle, three-phase alternating current and is supplied at the point of use as single-phase and/or three-phase alternating current. Voltages available at the point of use depend somewhat on the type of transformer connection. In areas where power consumption is predominantly single-phase low voltage, transformers are "*Y*" connected so that the low voltage load can be distributed evenly among the three transformers. As shown in Fig. 21-1*a*, voltages supplied from a "*Y*" connected transformer bank are 120 V and 208 V single-phase and 208 V three-phase. When the power load is predominantly three-phase, the transformers are delta (Δ) connected as shown in Fig. 21-1*b*, in which case the supply voltages are 115 V and 230 V single-phase and 230 V three-phase. With the open delta arrangement shown in Fig. 21-1*c*, three-phase power for isolated users can be supplied with only two transformers.

Power is frequently delivered to commercial establishments at 460 V and is available to large industrial users at much higher voltages by special arrangement with the power company. Naturally, all motors must be selected to conform to the characteristics of the available power supply. Power companies guarantee to maintain voltages within plus or minus 10% of the design voltage. Most motors will operate satisfactorily within these voltage limits. Many motors are designed so that they can be operated on either low or high voltage by reconnecting external motor leads.

In addition to the type of power supply available, some of the other more important factors that must be taken into account in order to select the proper type motor are:

1. The conditions prevailing at the point of installation with respect to the ambient temperature and to the presence of dust, moisture, or explosive materials.
2. Starting torque requirements (loaded or unloaded starting).
3. Starting current limitations.
4. Single or multispeed operation.
5. Continuous or intermittent operation.
6. Efficiency and power factor (not important for small motors).

All motors generate a certain amount of heat due to power losses in the windings. If this heat is not dissipated to the surroundings, motor temperature will become excessive and breakdown of the winding insulation will result. Open type (ventilated) motors are designed to operate at temperatures approximately 40° C (72° F) above the ambient temperature under full load, whereas totally enclosed motors are designed for a 55° C temperature rise at full load. Both types are guaranteed by the manufacturer to operate continuously under full load conditions without overheating when the ambient temperature does not exceed 40° C. When higher ambient temperatures are encountered, it is sometimes necessary to employ a

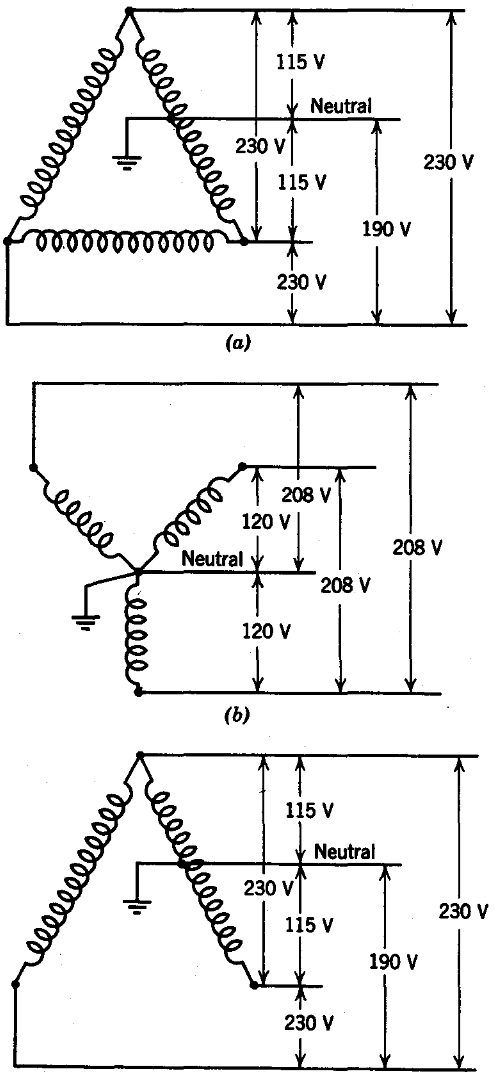


Fig. 21-1. (*a*) Delta connected transformers. (*b*) "Y" connected transformers. (*c*) Transformers connected open delta.

motor designed to operate at a proportionally lower temperature rise.

Most motors can be operated with small overloads for reasonable periods of time without damage. However, since the heat generated in the motor increases as the load on the motor increases, continuous operation of the motor under overload conditions will cause excessive winding temperatures and materially shorten the life of the insulation.

Motors may be classified according to the type of enclosure as: (1) open, (2) totally enclosed, (3) splash-proof, and (4) explosion-proof. Open-type motors are designed so that air is circulated directly over the windings to carry away the motor heat. This type of motor can be used in any application where the air is relatively free of dust and moisture, the motor is not subject to wetting, and the hazards of fire or explosion do not exist. Splash-proof motors are designed for installation out-of-doors or in any other location where the motor may be subject to wetting. Totally enclosed motors are designed for use where dust and moisture conditions are severe. These motors are unventilated and must dissipate their heat to the surrounding air through the motor housing. Explosion-proof motors are designed for installation in hazardous locations, as when explosive dusts or gases are contained in the air.

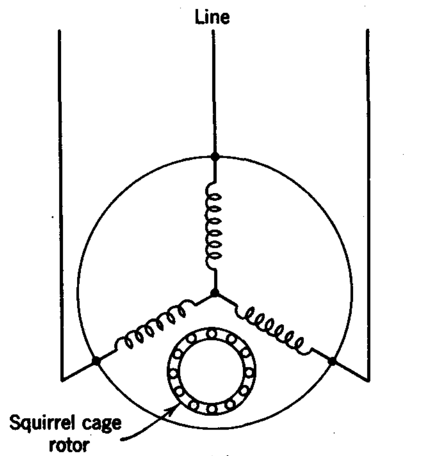
For the most part, motor starting torque requirements depend on the load characteristics of the driven machine. Low starting torque motors may be used with any machine that starts unloaded. On the other hand, high starting torque motors must be used when the driven machine starts under load. Since motor starting (locked rotor) currents often exceed five to six times the full load current of the motor, where large motors are employed, a low starting current characteristic is desirable in order to reduce the starting load on the wiring, transformers, and generating equipment.

Alternating current motors may be classified according to their principle of operation as either induction motors or synchronous motors. An induction motor is one wherein the magnetic field of the rotor is induced by currents flowing in the stator windings. A synchronous motor is one wherein the rotor magnetic field is produced by energizing the rotor directly from an external source.

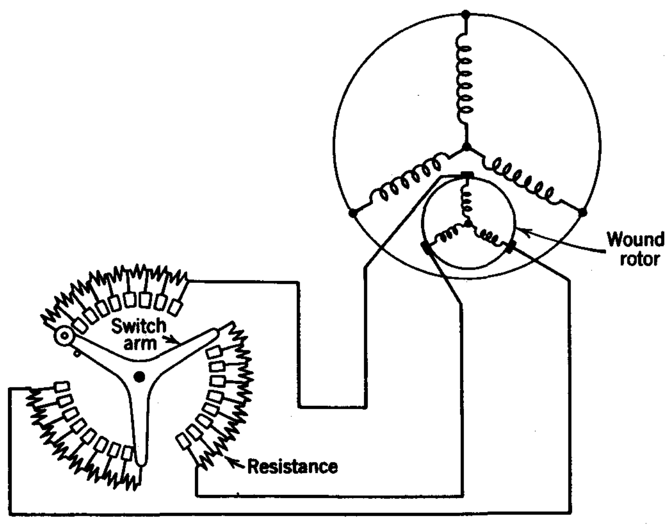
21-2. Three-Phase Induction Motors. Three-phase induction motors are of two general types: (1) squirrel cage and (2) wound rotor (slip ring). The two types are similar in construction except for rotor design. As shown in Fig. 21-2, each has three separate stator windings, one for each phase, which are evenly and alternately distributed around the stator core to establish the desired number of poles. A four-pole, three-phase motor will have twelve poles, four poles for each of the three phases.

When the stator is energized, three separate currents, each 120 electrical degrees out-of-phase with the other two, flow in the stator windings and produce a rotating magnetic field in the stator. At the same time, the currents induced in the rotor windings establish a magnetic field in the rotor. The magnetic poles of the rotor field are attracted by, and tend to follow, the poles of the rotating stator field, causing the rotor to rotate as shown in Fig. 21-3.

The rotor of an induction motor always rotates at a speed somewhat less than that of the rotating stator field. If the speed of the rotor were the same as that of the field, the conductors of the rotor winding would be standing still with respect to the rotating stator field rather than cutting across it, in which case no voltage would be induced in the rotor and the rotor would have no magnetic polarity. Therefore, it is necessary that the rotor turn at a speed slightly less than that of the stator field so that the conductors of the rotor winding continuously cut the flux of the stator field as the latter

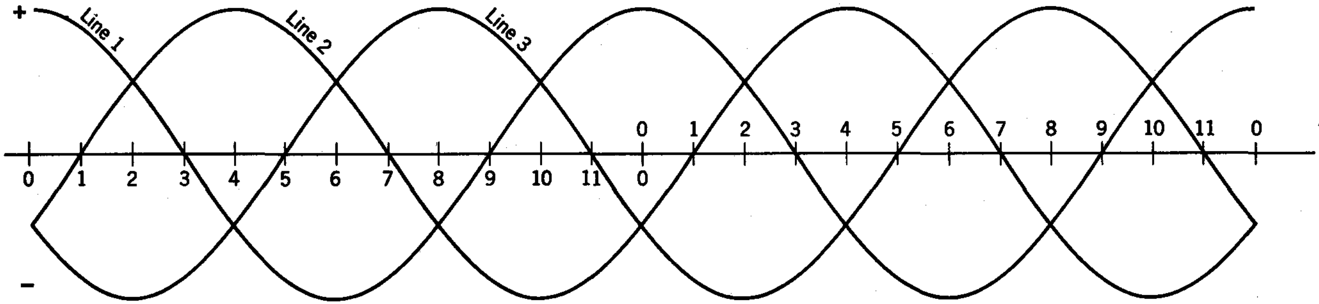


(a)

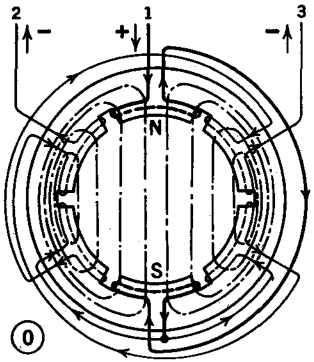


(b)

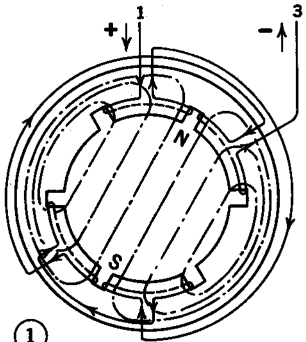
Fig. 21-2. (a) Squirrel cage polyphase motor. (b) Wound rotor (slip ring) polyphase motor.



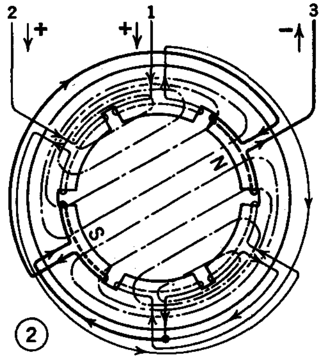
This diagram of three-phase voltages covers two complete cycles. The numbers on it refer to the numbers on the diagrams below. Each diagram shows the condition in the armature at the instant indicated by the corresponding number on this curve. The action of the magnetic field is smooth and regular; the rise and fall of currents in the conductors are also smooth and regular.



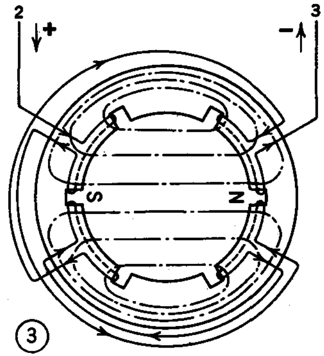
The current entering the motor on line 1 divides equally and leaves the motor on line 2 and line 3.



Now the current in line 2 is zero and that flowing in at line 1 leaves at line 3. The magnetic field revolves clockwise.



The current in line 1 is small and joining that from line 2 flows out in line 3 which carries a maximum negative current.



This and the following diagrams show how the magnetic field continues to rotate throughout the remainder of the cycle.

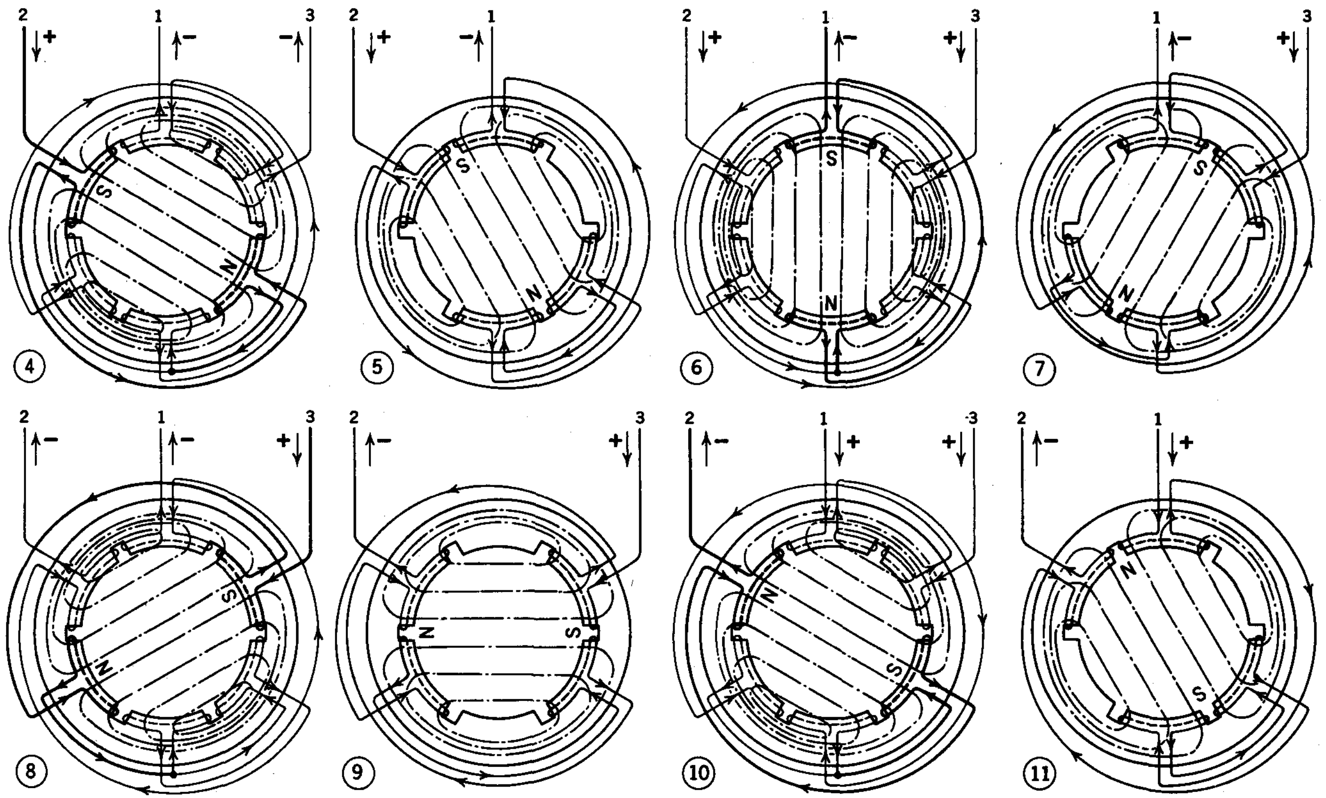


Fig. 21-3. This series of twelve diagrams shows the electric and magnetic conditions in a two-pole three-phase motor at the end of twelve equal parts of one cycle.

slips by. The difference between the speed of the rotor and that of the stator field is called the "magnetic slip" or "rotor slip." The greater the load on the motor, the greater is the amount of rotor slip. However, since the amount of slip changes only slightly as the load increases or decreases, and is very small even at full load, three-phase induction motors are usually considered to be constant speed motors.

Neglecting rotor slip, the speed of any alternating current motor is a function of the frequency and the number of stator poles. The synchronous speed of an alternating current motor can be determined by the following equation:

$$\text{Motor speed (rpm)} = \frac{\text{Frequency} \times 120}{\text{Number of poles}} \tag{21-1}$$

For a four-pole alternating current motor operating on 60-cycle power, the synchronous speed is

$$\frac{60 \times 120}{4} = 1800 \text{ rpm}$$

Rotor slip is usually expressed as a percentage of the synchronous speed. For instance, for a motor having a synchronous speed of 1800 rpm and operating at 1750 rpm, the percentage slip is

$$\frac{1800 - 1750}{1800} = 2.78\%$$

Rotor slip is also a measure of the power losses in the motor. In this instance, 2.78% of the total power input to the motor is converted to heat in the motor. Hence, there is a definite relationship between the amount of slip and the efficiency of the motor. The higher the slip, the lower is the efficiency of the motor.

The starting torque of a motor is the turning effort or torque that the motor develops at the instant of starting when full voltage is applied to the motor terminals. Starting torque is usually expressed as a percentage of full load torque and depends to some extent on the resistance of the rotor winding. An increase in rotor resistance increases the starting torque, but also increases the amount of rotor slip and decreases motor efficiency.

21-3. Three-Phase Squirrel Cage Motors. The rotor winding of a squirrel cage motor consists of bar-type copper conductors embedded in a laminated iron core and connected together (short-circuited) at the ends with heavy end-rings, giving the winding the appearance of a squirrel cage, from which the motor derives its name.

The squirrel cage induction motor is by far the most common type of three-phase motor and is available in a number of designs which provide a variety of starting torque-starting current characteristics. Two designs frequently employed with refrigerating equipment are designs B and C. Design B motors develop a locked rotor or starting torque between 125% and 275% of full load torque with relatively low starting currents. The normal torque-low starting current characteristic of this design makes it ideal for use as a drive for blowers, fans, and pumps, and for compressors which are started unloaded. Design C motors have a high-starting torque-low starting current characteristic which makes them suitable as drives for compressors which must start under load. Design C motors develop a starting torque between 225% to 275% of full load torque, but are slightly less efficient than the design B motor.

Multispeed operation of squirrel cage induction motors can be obtained by proper design of the stator windings. Since motor speed decreases as the number of poles increase, it follows that by doubling the number of stator poles, the speed of the motor can be reduced by one-half. For a single-winding motor, the number of poles can be changed in a two to one ratio by bringing extra leads outside of the motor. When more than two speeds are desired, the motor is wound with two or more separate windings for each phase. Two speeds are available with each separate winding.

21-4. Wound Rotor (Slip Ring) Motors. Wound rotor motors are employed in applications where the excessive starting current of a large squirrel cage motor would be objectionable and/or where a number of operating speeds are desired in the range between one-half and maximum speed.

A slip ring or wound rotor motor induction motor differs from the squirrel cage type only in the rotor winding. The rotor winding consists of insulated coils, grouped to form definite pole areas so that the rotor has the same number of poles as the stator. The terminal connections of the rotor windings are brought out to slip rings. The leads from the brushes on the slip rings are connected to external resistors, as shown in Fig. 21-2*b*. The operating principle of the slip ring motor is the same as that of the squirrel cage motor, except that by inserting external resistance in the rotor circuit when starting high-starting torque can be developed with low values of starting current. As the motor accelerates, the resistance is gradually cut out of the rotor circuit until at full speed the rotor windings are short-circuited. With the rotor winding short-circuited, the motor operates with low slip and high efficiency.

The speed of the wound rotor motor can be varied from maximum down to approximately 50% of maximum by inserting resistance in the rotor circuit. The starting resistors can be used for this purpose provided they are designed for a large enough current capacity to prevent excessive heating in continuous service. At reduced speeds, the wound rotor tends to lose its constant speed characteristic and the speed of the motor will vary somewhat with the load.

21-5. Synchronous Motors. The synchronous motor is so named because the field (rotor) poles are synchronized with the rotating poles of the armature (stator) windings. Therefore, the speed of the synchronous motor depends only on the frequency of the power supply and the number of poles, and is independent of motor load. The armature winding of the synchronous motor is similar to those of the squirrel cage and wound rotor motors. The field (rotor) winding consists of a series of coils which make up the field poles. The field coils are connected through slip rings to a direct current power source and are so connected together that alternate north and south poles are formed when the field winding is energized with direct current. The direct current is usually supplied by a small direct current generator, called an exciter, which is mounted on the motor shaft. The rotor is also equipped with a squirrel cage winding, called the "damper" winding, which is used to start the motor. The damper winding can be designed for a variety of starting torque-starting current characteristics.

When polyphase power is applied to the armature winding, the rotating magnetic field acts on the squirrel cage damper winding and causes the rotor to rotate. Since the motor starts as a squirrel cage motor, the speed will be slightly less than synchronous speed. After the motor comes up to speed, direct current is applied to the field windings. This produces alternate north and south poles on the rotor which lock the rotor into synchronization with the rotating armature field.

By adjusting the flow of direct current to the field coils, the synchronous motor can be operated at unity power factor. Therefore, synchronous motors can be applied to an advantage in any large installation where constant speed and high efficiency are desired. However, the big advantage of the synchronous motor is that it can be used to correct the low power factor that results from heavily inductive loads. By increasing the flow of direct current through the field winding (overexciting the field), the synchronous motor is operated with a leading power factor which can be adjusted to offset exactly the lagging power factor produced by inductive loads. Power factor correction will in no way affect the load-carrying capacity of the synchronous motor.

21-6. Single-Phase Motors. Single-phase motors commonly used in the refrigerating industry are of the following types: (1) split-phase, (2) capacitor start, (3) capacitor start and run, (4) permanent capacitor, and (5) shaded pole. All these motors are induction motors and all employ a squirrel cage rotor. The principal factor which distinguishes one type from another is the particular method used to produce a starting torque.

When a single-phase stator winding is energized, current flow is simultaneous in all the stator poles and no rotating stator field is produced. Furthermore, the current induced in the squirrel cage rotor winding is such that the magnetic field set up in the rotor is exactly in line with the magnetic field of the stator. The condition occurring can be compared to the "dead center" condition of a single piston engine. Therefore, there is no tendency for the rotor to rotate. However, if the rotor is started to rotate by some means, the current induced in the rotor winding will lag slightly behind the current in the stator winding. This causes the rotor field to lag the stator field and produces a torque that keeps the rotor turning. Hence, once the rotor of a single-phase motor is

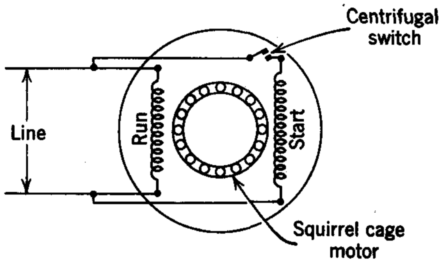


Fig. 21-4. Split-phase motor.

started, a rotating field is produced and the motor operates in a manner similar to that of the three-phase squirrel cage motor.

21-7. Split-Phase Motors. In order to produce a starting torque in the single-phase motor and make the motor self-starting, a second stator winding, called the "starting" or "auxiliary" winding, is employed in addition to the phase winding, the latter winding being referred to as the "main" or "running" winding. The relative position of the two windings in the stator of a four-pole, single-phase motor is shown in Fig. 21-4. Notice that the starting and running windings are connected in parallel directly across the single-phase line. In the split-phase type motor, the starting winding is wound with small wire so that the winding has a high resistance and low inductance, whereas the running winding is wound with large wire to have a low resistance and a high inductance. Both windings are energized at the instant of starting. However, because of the higher inductance of the running winding with relation to that of the starting winding, the current flow in the running winding lags the current flow in the starting winding by approximately 30 electrical degrees. Since the currents flowing in the two windings are 30 degrees out-of-phase with each other, the single phase is "split" to give the effect of two phases and a rotating field is set up in the stator which produces a starting torque and causes the rotor to rotate. When the rotor has accelerated to approximately 70% of maximum speed, which is a matter of a second or two, a centrifugal mechanism mounted on the rotor shaft opens a switch in the starting winding. With the starting winding disconnected, the motor continues to operate on the running winding alone. Since the starting winding is wound with relative small wire, it will heat very quickly and, if allowed to remain in the circuit for an appreciable length of time, will be destroyed by overheating.

Since the maximum phase split that can be achieved with the split-phase motor is approximately 30 electrical degrees, the split-phase motor has a relatively low starting torque and can be used only with machines which start unloaded. These motors are generally available in sizes ranging from $\frac{1}{20}$ to $\frac{1}{3}$ hp for both 115 V and 230 V operation. They are used primarily as drives for small fans, blowers, and pumps.

21-8. Capacitor Start Motors. The capacitor start motor is identical to the split-phase motor in both construction and operation, except that a capacitor is installed in series with the starting winding, as shown in Fig. 21-5. Too, the starting winding of the capacitor start motor is usually wound with larger wire than that used for the starting winding of the split-phase motor. The use of a capacitor in series with the starting winding causes the current in this winding to lead the voltage, whereas the current in the running winding lags the voltage by virtue of the high inductance of that winding. With this arrangement, the phase displacement between the two windings can be made to approach 90 electrical degrees so that true two-phase starting is achieved. For this reason, the starting torque of the capacitor start motor is very high, a circumstance which makes it an ideal drive for small compressors that must be started under full load.

As in the case of the split-phase motor, the starting winding of the capacitor start motor is taken out of the circuit when the rotor

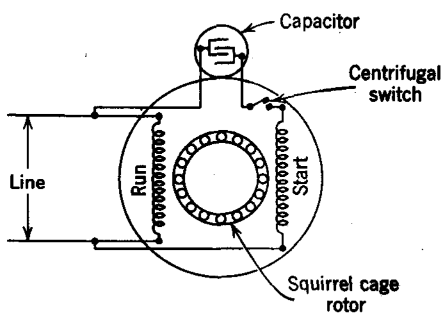


Fig. 21-5. Capacitor start motor.

approaches approximately 70% of maximum speed, and thereafter the motor operates on the running winding alone. Capacitor start motors are generally available in sizes ranging from $\frac{1}{6}$ through $\frac{3}{4}$hp for both 115V and 230V operation.

21-9. Capacitor Start and Run Motors. Construction of the capacitor start and run motor is identical to that of the capacitor start motor with the exception that a second capacitor, called a "running" capacitor, is installed in series with the starting winding but in parallel with the starting capacitor and starting switch, as shown in Fig. 21-6. The operation of the capacitor start-and-run motor differs from that of the capacitor start-and-split-phase motors is that the starting or auxiliary winding remains in the circuit at all times. At the instant of starting, the starting-and-running capacitors

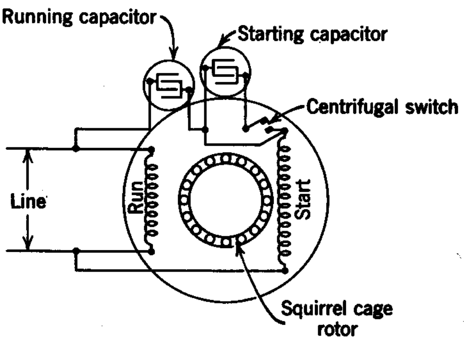


Fig. 21-6. Capacitor start and run motor.

are both in the circuit in series with the auxiliary winding so that the capacity of both capacitors is utilized during the starting period. As the rotor approaches 70% of rated speed, the centrifugal mechanism opens the starting switch and removes the starting capacitor from the circuit, and the motor continues to operate with both main and auxiliary windings in the circuit. The function of the running capacitor in series with the auxiliary winding is to correct power factor. As a result, the capacitor start-and run-motor not only has a high starting torque but also an excellent running efficiency. These motors are generally available in sizes ranging from approximately $\frac{1}{8}$ through 10 hp, and are widely used as drives for refrigeration compressors in single-phase applications.

21-10. Permanent Capacitor Motors. Construction of the permanent capacitor motor is

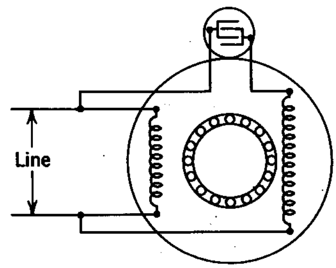


Fig. 21-7. Permanent capacitor motor.

similar to that of the capacitor start-and-run motor, except that no starting capacitor or starting switch is used. The capacitor shown in series with the auxiliary winding in Fig. 21-7 remains in the circuit continuously. The capacitor is sized for power factor correction but is used also as a starting capacitor. However, since the capacitor is too small to provide a large degree of phase displacement, the starting torque of the permanent capacitor is very low. These motors are available only in small fractional horsepower sizes. They are used mainly as drives for small fans which are mounted directly on the motor shaft. The chief advantage of this type of motor is that it lends itself readily to speed control down to 50% of rated speed. Also, it does not require a starting switch.

21-11. Shaded Pole Motors. Construction of the shaded pole motor differs somewhat from that of the other single-phase motors in that the main stator winding is arranged to form salient poles, as shown in Fig. 21-8. The auxiliary winding consists of a shading coil, which surrounds a portion of one side of each stator pole. The shading coil usually consists of a single turn of heavy copper wire which is

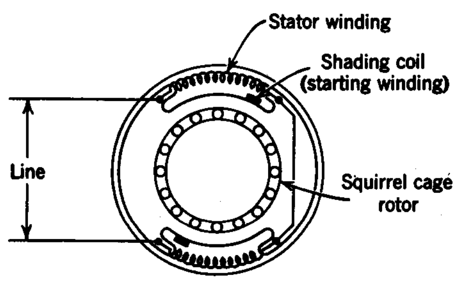


Fig. 21-8. Shaded pole motor.

short-circuited and carries only induced current. In operation, the flux produced by the induced current in the shading coil distorts the magnetic field of the stator poles and thereby produces a small starting torque. Shaded pole motors are widely used as drives for small fans which are mounted directly on the motor shaft. They are available in sizes ranging from $\frac{1}{125}$ through approximately $\frac{1}{20}$ hp. In addition to its ready adaptability to speed control, the main advantages of the shaded pole motor are its simple construction and low cost.

21-12. Hermetic Motors. Motors frequently employed in hermetic motor-compressor units are three-phase squirrel cage motors and split-phase, capacitor start, and capacitor start-and-run single-phase motors. Whereas the split-phase and capacitor start motors are limited to small fractional horsepower units, the capacitor start-and-run motor is used in sizes from $\frac{1}{2}$ through 10 hp. Three-phase squirrel cage motors are employed from 3 hp up.

Although air, water, oil, and liquid refrigerant are sometimes used as cooling mediums to carry away the heat of hermetic motors, the large majority of hermetic motors are suction vapor cooled. For this reason, hermetic motor-compressor units should never be operated for any appreciable length of time without a continuous flow of suction vapor through the unit.

In the single-phase hermetic motor, a specially designed starting relay replaces the shaft-mounted centrifugal mechanism as a means of disconnecting the starting winding (or starting capacitor) from the circuit after the motor starts. Three types of starting relays have been used, namely: (1) the hot wire or timing relay, (2) the current coil relay, and (3) the voltage coil or potential relay.

21-13. Hot Wire Relays. The hot wire relay depends on the heating effect of the high-starting current to cause the thermal expansion of a special alloy wire, which in turn acts to open the starting contacts and remove the starting winding from the circuit. As shown in Fig. 21-9, the hot wire relay contains two sets of contacts, "S" and "M," which are in series with the starting and running windings, respectively. Both sets of contacts are closed at the instant of starting so that both windings are connected to the line (Fig. 21-9*a*). The high-starting currents heats the wire and causes it to expand sufficiently to pull contacts "S" open and remove the starting winding from the circuit. After the starting winding is out of the circuit, the normal running current through the running winding will generate enough heat to maintain the "S" contacts in the open position, but not enough to cause additional expansion of the wire and open contacts "M" (Fig. 21-9*b*). However, if for any reason the motor draws a sustained overcurrent, the wire will expand further and pull open contacts "M," removing the running winding from the circuit (Fig. 21-9*c*). Contacts "M" are actually overload contacts which act as overcurrent protection for the motor. The mechanical arrangement of the two sets of contacts is such that contacts "M" cannot open without also opening contacts "S."

Since the action of the hot wire relay depends on the amount of current flow through the alloy wire, these relays must be sized to fit the current characteristics of the motor. They are best applied to the split-phase type motor.

21-14. Current Coil Relays. The current coil relay is used primarily with capacitor start motors. It is a magnetic type relay and is actuated by the change in the current flow in the running winding during the starting and running periods. The coil of the relay, which is made up of a relatively few turns of large wire, is connected in series with the running winding. The relay contacts, which are normally open, are connected in series with the starting winding, as shown in Fig. 21-10.

When the motor is energized, the high locked rotor current passing through the running winding and through the relay coil produces a relatively strong magnet around the coil and causes the relay armature to "pull-in" and close the starting contacts energizing the starting winding. With the starting winding energized, the rotor begins to rotate and a counter emf is induced in the stator windings which opposes the line voltage and reduces the current through the windings and relay coil. As the current flow through the relay coil diminishes, the coil field becomes too weak to hold the armature, whereupon the armature falls out of the coil field by gravity (or by spring action) and opens the starting contacts. The motor then runs on the running winding alone.

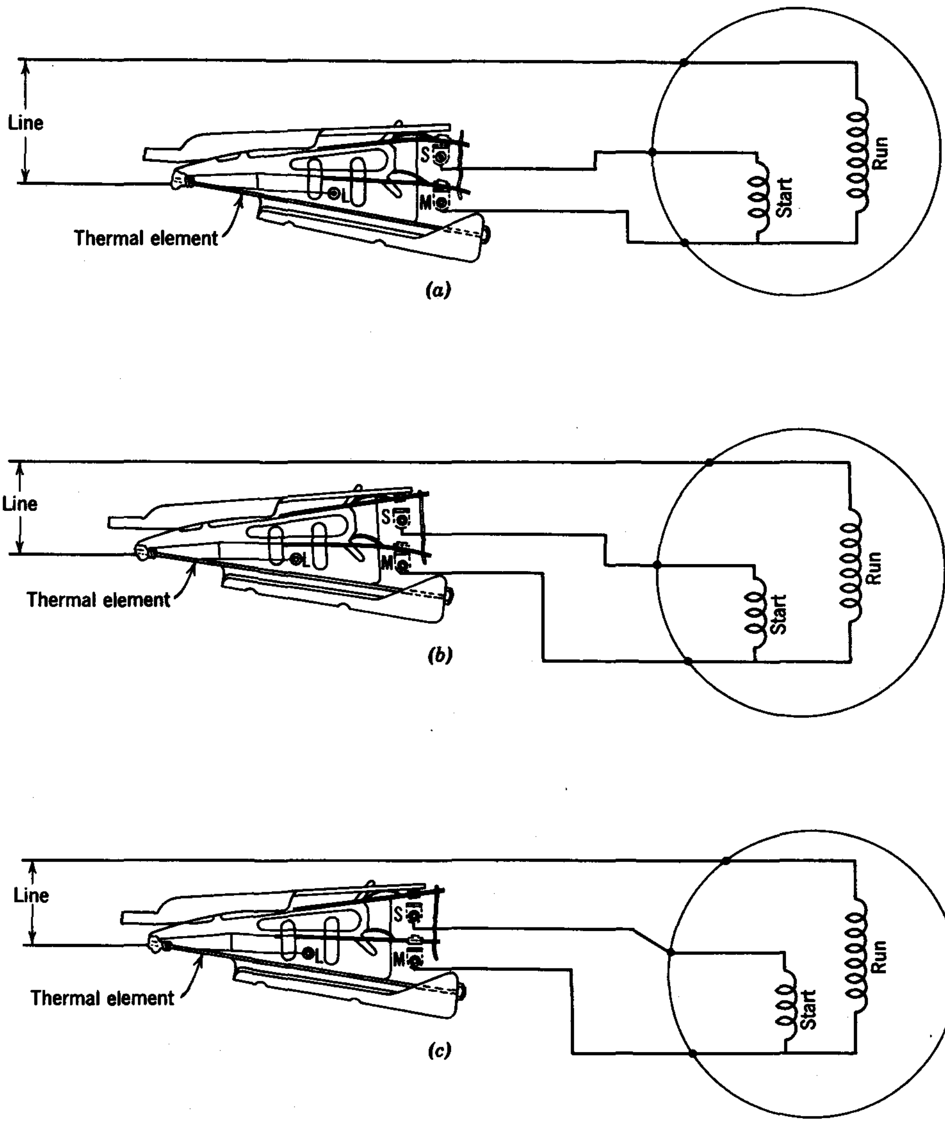


Fig. 21-9. Hot wire starting relay. *(a)* Starting position. *(b)* Run position. *(c)* Overload position.

21-15. Potential Relays. Potential or voltage coil relays are employed with capacitor start and capacitor start-and-run motors. The potential relay differs from the current coil type in that the coil is wound with many turns of small wire and is connected in parallel with (across) the starting winding, rather than in series with the running winding, as shown in Fig. 21-11. The relay contacts are connected in series with the starting capacitor and are closed when the motor is not running. When the motor is energized, both the starting and running windings are in the circuit. As the motor starts and comes up to speed, the voltage in the starting winding increases to a value considerably above that of the line voltage (approximately 150%), as a result of the action of the capacitor(s) in series with this winding.*

* The vector sum of the voltages across the starting winding and capacitor(s) is equal to the line voltage.

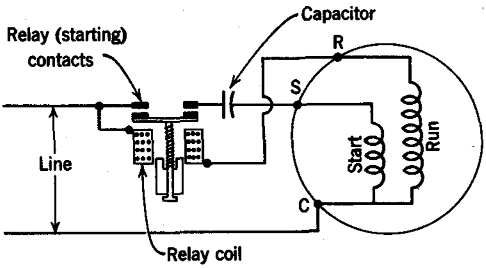


Fig. 21-10. Current-coil type starting relay.

The high voltage generated in the starting winding produces a relatively high current flow through the relay coil and causes the coil armature to pull in and open the starting contacts. With the capacitor start motor, opening the relay contacts disconnects both the starting winding and starting capacitor from the circuit. With the capacitor start-and-run motor, only the starting capacitor is disengaged. With either type or motor, the starting winding voltage decreases somewhat when the starting contacts open, but remains high enough to hold the coil armature in the field and keep the starting contacts open until the motor is stopped.

21-16. Thermal Overload Protection for Hermetic Motors. All hermetic motor-compressors should be equipped with some type of thermal device which will protect the motor against overheating regardless of the cause. Thermal overload devices of this type are usually designed to be fastened directly to, and in good thermal contact with, the motor-compressor housing, so that they are sensitive not only to motor overcurrent but also to overheating resulting from high discharge temperatures and other such causes.

Although some types of motor starting relays contain built-in overcurrent protection, these are usually sensitive only to motor overcurrent and do not provide protection against overheating from other causes.

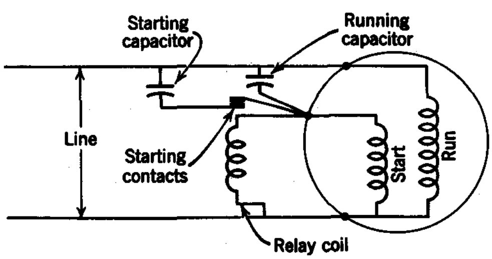


Fig. 21-11. Potential type starting relay.

21-17. Motor Starting Devices. For fractional horsepower motors the motor starting equipment sometimes consists only of a direct acting (line voltage) manual switch, thermostat, or low pressure control installed in the motor circuit between the motor and the power source (Fig. 21-12*a*). The control acts to open and close the motor circuit to stop and start the motor, respectively. Safety controls, such as high pressure cut-outs, overcurrent protective devices, etc., are connected in series with the operating or "cycling" control, as shown in Fig. 21-12*b*. The contacts of the safety controls are normally closed and do not open to break the circuit unless called on to perform their protective function.

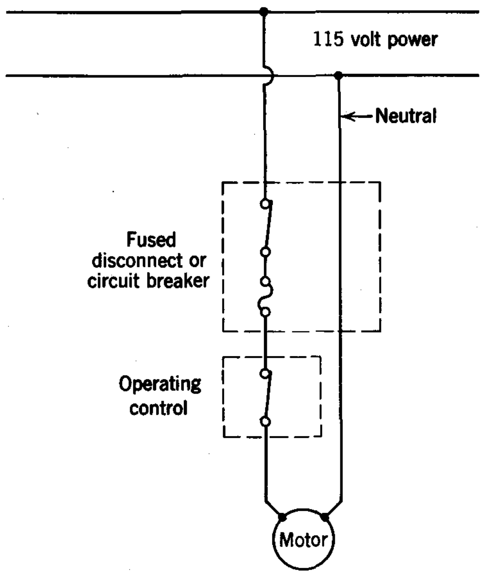


Fig. 21-12*a*. Direct acting, line voltage controls with 115 V single-phase motor.

With low voltage, single-phase power, the line voltage controls are installed in the "hot" line, never in the neutral (Fig. 21-12*a*). With high voltage, single-phase power, the controls may be installed in either one or both of the power lines (Fig. 21-12*b*). In the case of three-phase power, at least two of the three power lines must be opened to disconnect the motor from the power source. This requires the use of double-pole controls, as illustrated in Fig. 21-12*c*. However, in all cases, regardless of the type of power supplied, all "hot" lines must be protected individually with a properly sized fuse or circuit breaker.

Since the contacts of direct acting controls must be heavy enough to carry the full load current of the motor(s) they are controlling, these controls tend to become unwieldy when the full load current of the motor exceeds 15 or 20 amperes. Therefore, general practice is to control larger motors indirectly through a magnetic contactor.

A magnetic contactor or motor starter is essentially an electrical relay which in its simplest form consists of a coil of insulated wire, called a holding coil, and an armature to which the electrical contacts are attached (Fig. 21-13). The operation of the magnetic contactor is similar to that of the solenoid valve described in Chapter 17. When the holding coil is energized, the armature is pulled into the coil magnetic field, thereby closing the electrical contacts and connecting the motor to the power source. When the holding coil is de-energized, the armature drops out of the coil field, causing the contacts to open and disconnect the motor from the power source. When a magnetic contactor is employed, the motor is controlled indirectly by controlling the contactor holding coil. Therefore, the operating control is installed in series with the holding coil in the holding coil circuit rather than directly in the motor circuit.

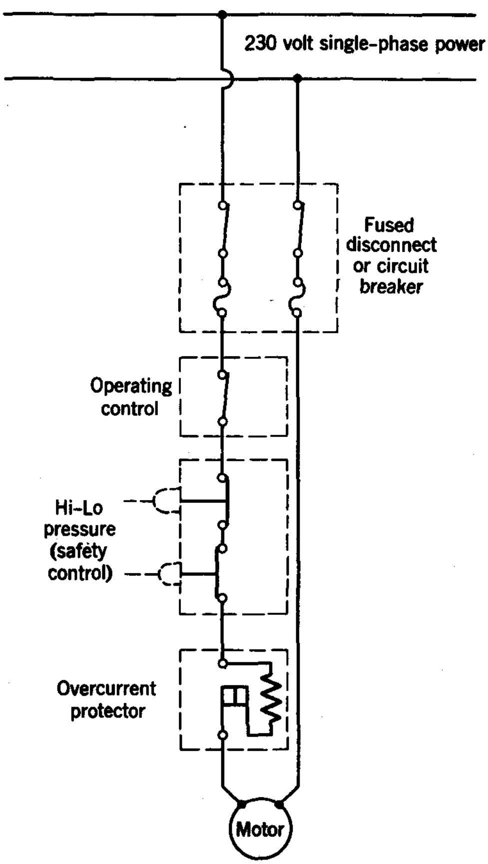


Fig. 21-12*b*. *Direct acting, line voltage controls used* with 230 V single-phase power.

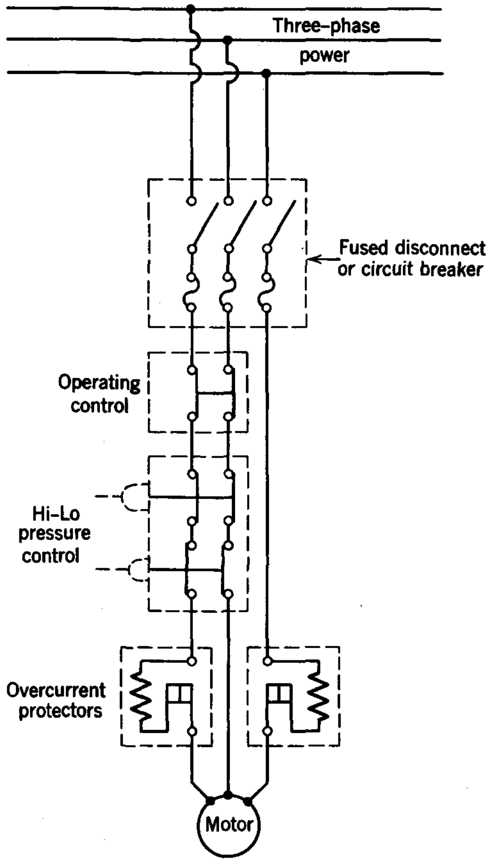


Fig. 21-12c. Direct acting, line voltage controls used with 230 V three-phase power.

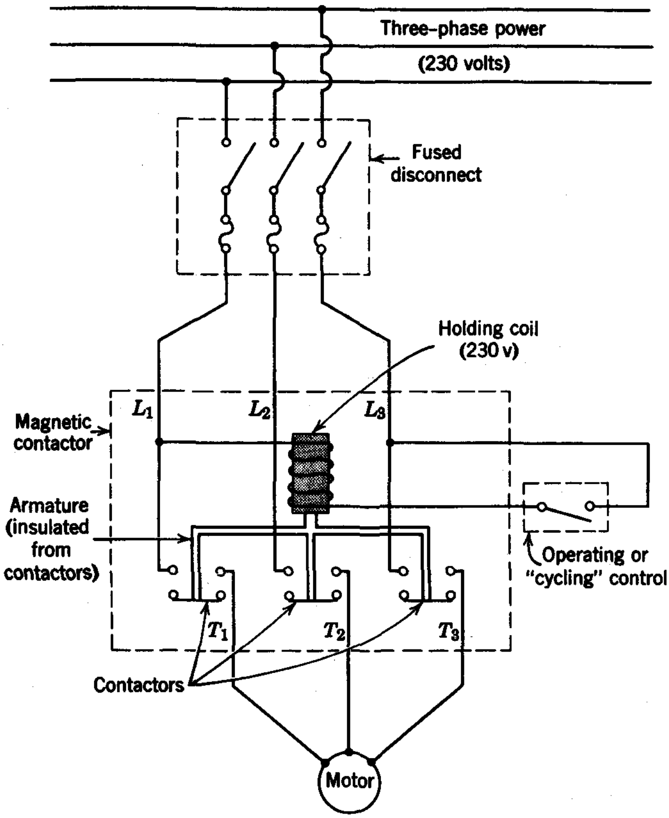


Fig. 21-13. Motor controlled indirectly through magnetic contactor.

The advantages gained by employing magnetic contactors to connect motors to the power source are several. First, since the current required to energize the holding coil is small, the contacts of the operating and safety controls can be of relatively light construction, which results in a reduction in both the size and the cost of the controls. Second, since the holding coil circuit is electrically independent of the motor circuit, the holding coil circuit voltage may be different from that of the motor circuit. This permits the use of low voltage (usually 24 V) control circuits, which are safer and generally less expensive to buy and install. A magnetic contactor employing a low voltage control circuit is shown in Fig. 21-14.

Holding coils for magnetic contactors are manufactured for all standard voltages and frequencies and are readily interchangeable in the field. The holding coil voltages most commonly used are 24, 115, 230, and 460.

21-18. Reduced Voltage Starters. The magnetic contactor described in the preceding section is called an "across-the-line" starter, and is so named because it connects the motor directly across the line at full voltage immediately when the holding coil is energized. This type of motor starter is suitable for motors up to 20 or 25 hp and is more widely used than any other type. However, in order to prevent excessive current surges in the power lines during the starting period, general practice is to start squirrel cage motors above 25 hp under reduced voltage. Reduced voltage starting is accomplished through the use of reduced voltage starters which introduce resistors or auto-transformers into the motor circuit during the starting period. A resistance type reduced voltage starter is illustrated in Fig. 21-15. When the operating control closes, the #1 holding coil is energized and the main contacts (#1) close, thereby connecting the motor to the power source through the resistors. This allows the motor to start under reduced voltage and, at the same time, energizes the timing relay. After a predetermined time interval, the

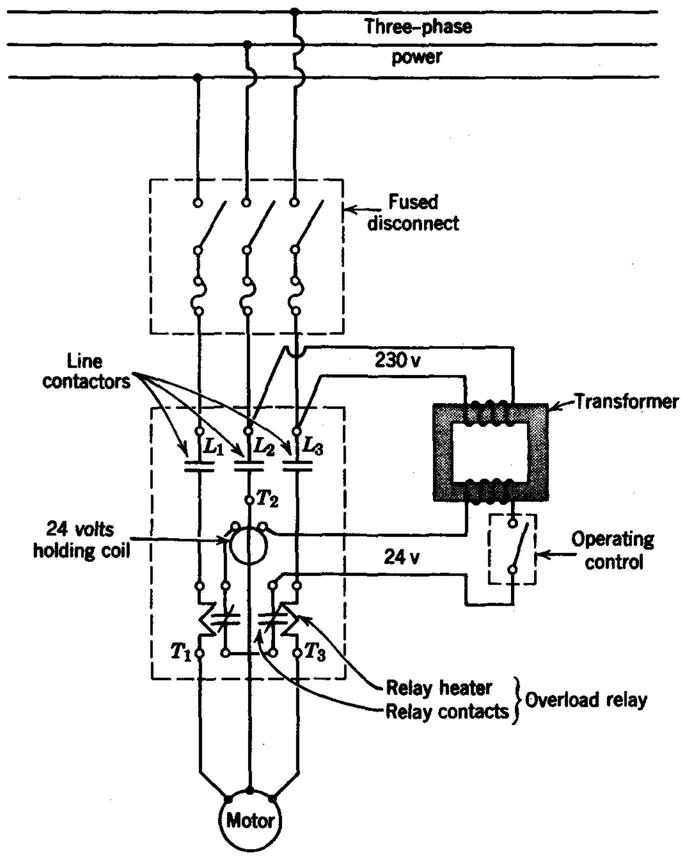


Fig. 21-14. Magnetic contactor with low voltage control circuit.

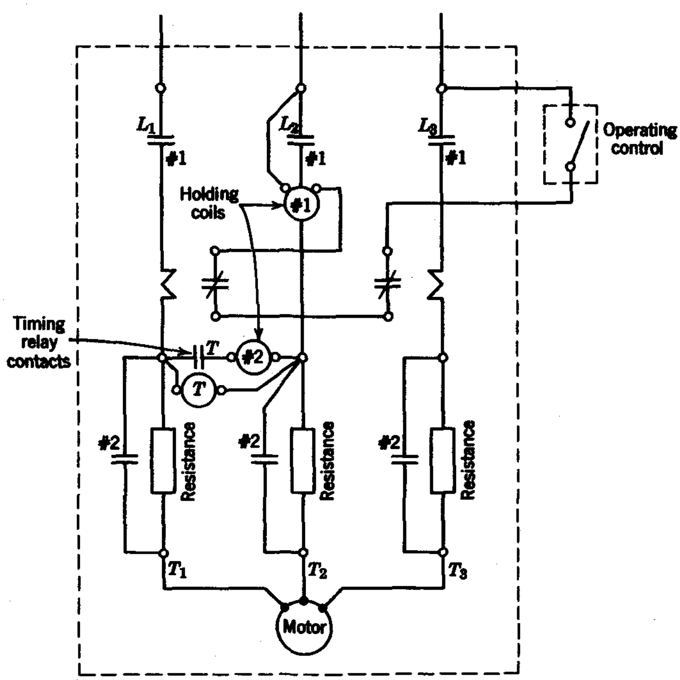


Fig. 21-15. Resistance type reduced voltage starter.

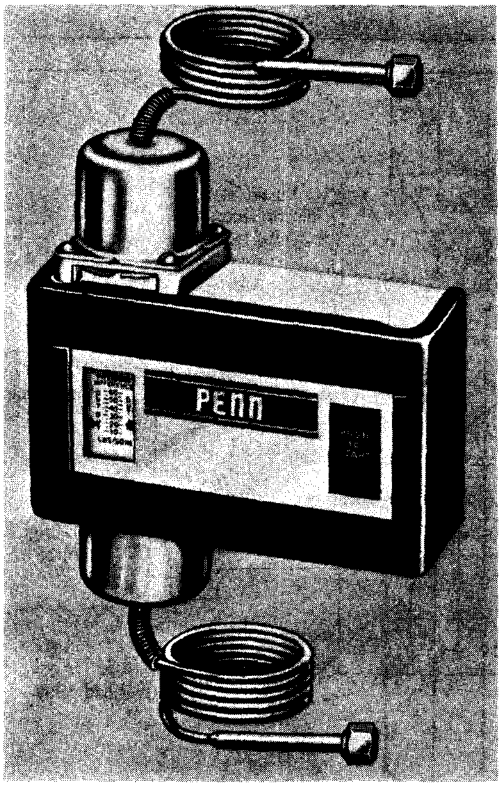


Fig. 21-16. External view of oil pressure failure control. (Courtesy Penn Controls, Inc., Goshen, Indiana.)

timing relay closes and energizes the #2 holding coil, which in turn closes the #2 contactors and shunts the resistors out of the motor circuit.

21-19. Motor Overcurrent Protection. It is important to recognize that line fuses and circuit breakers are designed to protect the circuit only and do not provide overcurrent protection for the motor. Therefore, unless the motor is equipped with a built-in thermal overload, separate overcurrent protection must be provided in the circuit of each motor. To satisfy the need for overcurrent protection, many magnetic motor starters come equipped with overload relays.

The "overload relay" consists essentially of two parts: (1) a heater element installed in the motor circuit and (2) a set of contacts installed in the holding coil circuit. In the event that the motor is subjected to a sustained overcurrent, the temperature of the heater element increases above normal and the excess heat given off by the heater causes warping of a bimetal element (or melting of a special alloy metal) which opens the overload contacts in the holding coil circuit. This de-energizes the holding coil which in turn disengages the motor from the power source. A time delay action built into the overload relay prevents tripping of the overload during the motor starting period and during momentary overloads.

21-20. Oil Pressure Failure Control. Another safety control frequently encountered in the control circuits of refrigeration equipment is the oil pressure failure control. The function of this control is to cycle the compressor off when the useful oil pressure developed by the oil pump falls below a predetermined minimum, or in the event that the oil pressure fails to build up to the minimum safe level within a predetermined time interval after the compressor is started. An external view of the oil pressure failure control is shown in Fig. 21-16.

In studying the operating characteristics of the oil pressure failure control it is important to recognize that the total oil pressure, as measured by an oil pressure gage, is the sum of the crankcase (suction) pressure and the pressure developed by the oil pump, and therefore is not the true or useful oil pressure. To determine the useful oil pressure, the suction pressure must be subtracted from the total oil pressure, the difference between the two being the useful oil pressure developed by the oil pump.

To be effective, the oil pressure failure switch must be actuated by the useful oil pressure rather than by the total oil pressure. This is accomplished by using two pressure bellows opposed to each other, as shown in Fig. 21-17. One bellows is connected to the crankcase and reflects crankcase pressure, whereas the other bellows is connected to the discharge of the oil pump and reflects total oil pressure. The pressure differential between the two bellows pressures is equal to the useful oil pressure and is utilized to actuate the pressure differential switch of the oil pressure failure control.

A time delay relay incorporated into the oil pressure failure control allows the compressor to operate 90 to 120 sec with the oil pressure below the safe level. This permits the compressor to start with zero oil pressure and also prevents unnecessary shut-down of the compressor in the event that the oil pressure momentarily falls below the minimum safe limit. However, if the oil pressure does not rise to the safe level within the alloted time, the oil pressure failure control will shut-down the compressor. Before the compressor can be restarted, the oil pressure failure control must be reset manually.

Referring to Fig. 21-17, notice that the timing relay consists of a timing switch and a heater element. The timing switch is connected in series with the holding coil of the magnetic starter, and the heater is connected in parallel with the holding coil. The pressure differential switch is connected in series with, and controls the operation of, the relay heater. The resistor in series with the relay heater limits the current flow through the heater and makes the oil pressure failure control adaptable to both 115 V and 230 V control circuits.

Since the oil pump operates only when the compressor is operating, the total oil pressure will be exactly equal to the crankcase pressure during the compressor off cycle, and both the timing relay heater and the holding coil are energized. If, after the compressor starts, the useful oil pressure builds up to the cut-in pressure of the oil pressure safety control, the differential pressure switch will open and remove the relay heater from the circuit. This action will allow the compressor to continue normal operation. On the other hand, if the useful oil pressure does not build up to the cut-in pressure of the control within the alloted time, the differential pressure switch will not open and the heater is left in the circuit. Continued operation of the relay heater will cause the bimetal of the timing switch to warp and open the timing contacts. This breaks the holding coil circuit and stops the compressor.

If the useful oil pressure falls below the cut-out point of the oil pressure failure control while the compressor is operating, the differential pressure switch closes and energizes the relay heater. If the oil pressure does not build up to the cut-in pressure again within the alloted time interval, continued operation of the heater will open the timing switch and stop the compressor.

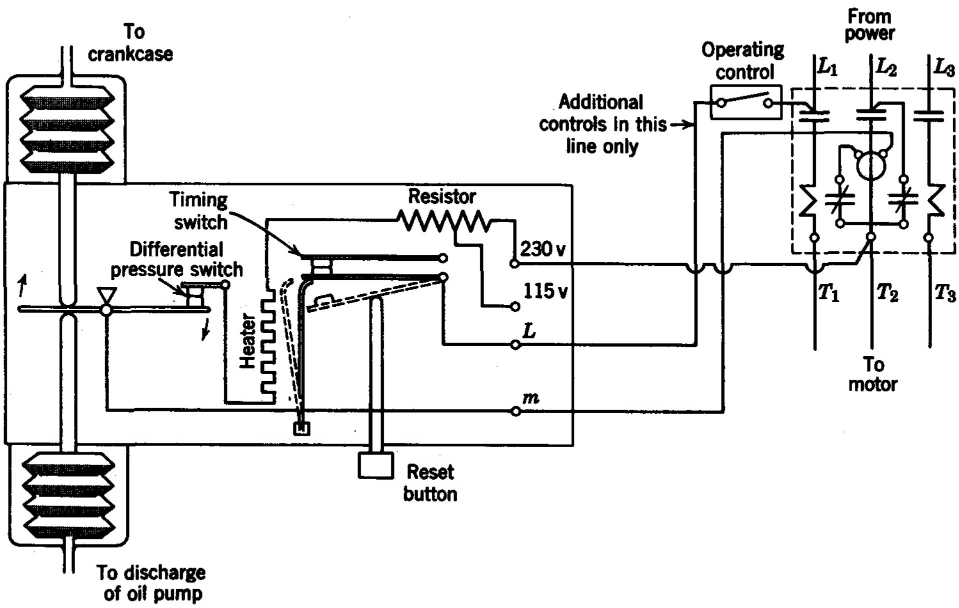


Fig. 21-17. Oil pressure failure control.

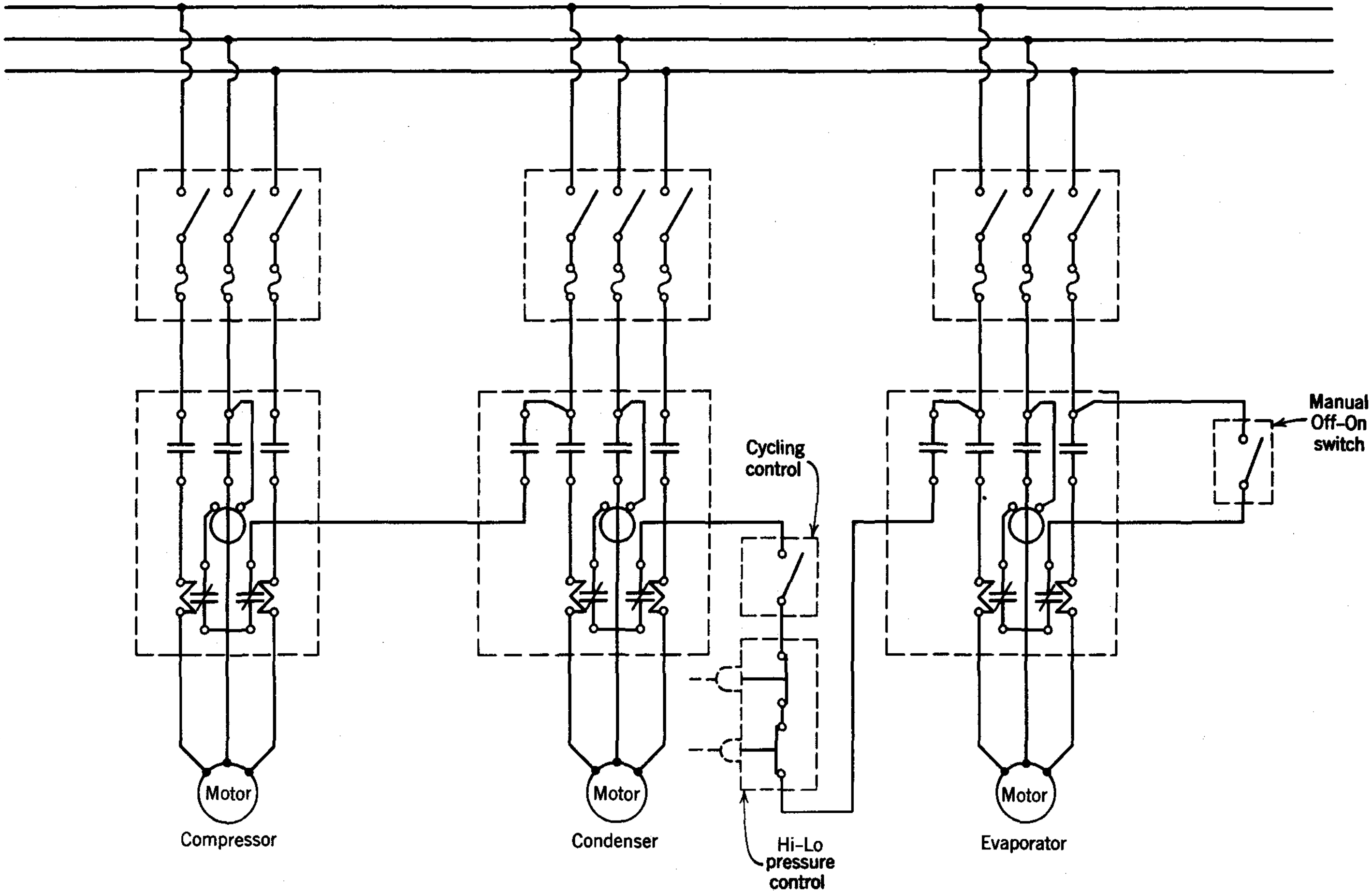


Fig. 21-18. Interlocking control circuits.

As indicated in the foregoing, the oil pressure failure control has both a cut-in pressure and a cut-out pressure. These should be set in accordance with the compressor manufacturer's instructions whenever such data are available. In the absence of these data, general practice is to set the cut-in point of the control for a pressure approximately 5 psi below the useful oil pressure when the compressor is in operation. The cut-out point is usually set for a pressure approximately 5 psi below the cut-in pressure. For example, assume that the crankcase pressure is 37 psig and the total oil pressure is 72 psig, so that the useful oil pressure is 35 psi (72 − 37). The cut-in pressure should be set at approximately 30 psi (35 − 5) and the cut-out pressure at approximately 25 psi (30 − 5).

21-21. Interlocking Controls. As a general rule, a refrigerating system employs at least three motors: the compressor motor, the evaporator blower motor, and the condenser fan (or pump) motor. Good design practice requires that the controls of these motors be so interlocked that the compressor cannot operate unless the evaporator blower and the condenser fan or pump are operating. One of the more common methods of achieving the desired interlocking is illustrated in Fig. 21-18. In this instance, the evaporator is permitted to operate continuously and is controlled with a manual off-on switch. With this particular control arrangement, the fan control is the lead control and as such may be used to start and stop the entire system.

The holding coil of the condenser starter is wired through an auxiliary contact in the evaporator blower starter. Since the auxiliary contact will be closed only when the holding coil of the blower starter is energized, the condenser fan or pump cannot be started without first starting the evaporator blower. Likewise, the holding coil of the compressor starter is connected through an auxiliary contact in the condenser starter so that the compressor starts unless the condenser and evaporator starters are energized. Notice also that the cycling control

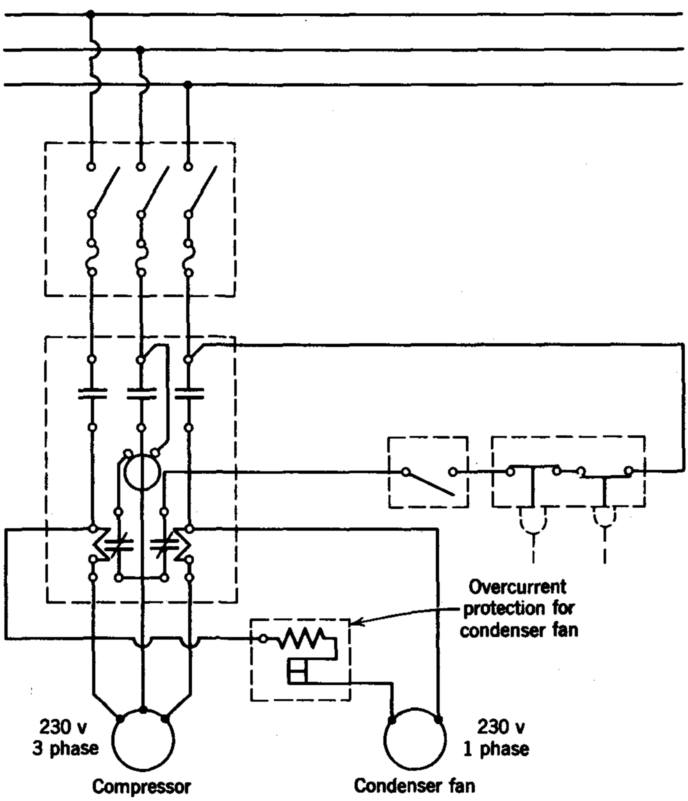


Fig. 21-19. Two motors operating through one magnetic contactor.

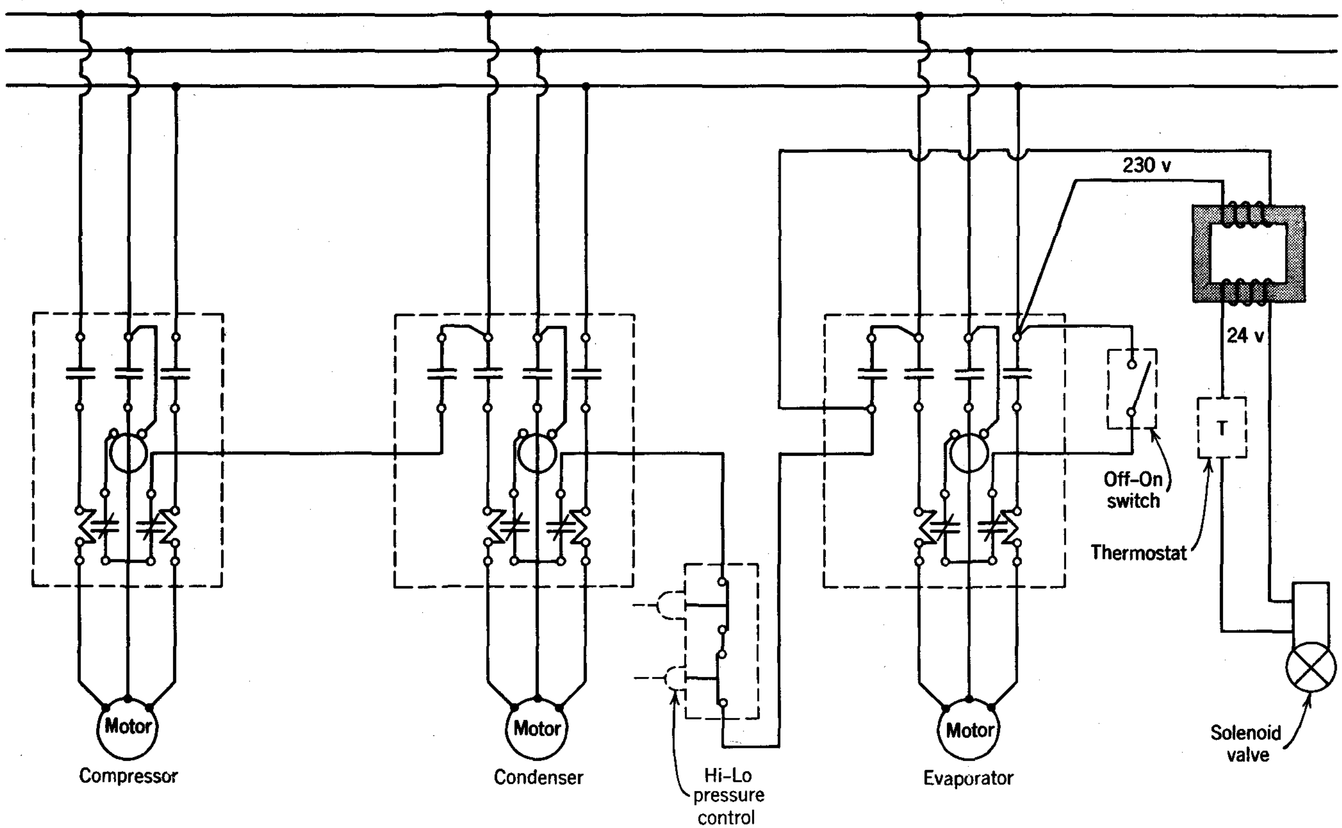


Fig. 21-20. Pump-down system. A time delay relay should be installed in solenoid circuit to prevent solenoid energizing at the same time that the evaporator blower is started.

(thermostat) controls the condenser starter rather than the compressor starter. This arrangement permits the condenser fan or pump to cycle off and on with the compressor.

Another common method of accomplishing the same result is to operate the compressor and condenser motors through the same magnetic contactor, as shown in Fig. 21-19. This method is usually confined to small, packaged equipment and requires that separate overcurrent protection be provided for each motor.

A wiring diagram for a simple pump-down system with interlocking control is illustrated in Fig. 21-20. Notice particularly the method of interlocking low voltage and high voltage control circuits.

Tables and Charts

TABLE 1-1. Pressure Conversion Factors

Multiply	By	To Obtain
Atmosphere	29.92	Inches of mercury
Atmosphere	33.93	Feet of water
Atmosphere	14.70	Pounds per square inch
Atmosphere	1.058	Tons per square foot
Feet of water	0.881	Inches of mercury (at 32° F)
Feet of water	62.37	Pounds per square foot
Feet of water	0.4335	Pounds per square inch
Feet of water	0.02950	Atmospheres
Inches of mercury (at 62° F)	13.57	Inches of water (at 62° F)
Inches of mercury (at 62° F)	1.131	Feet of water (at 62° F)
Inches of mercury (at 62° F)	70.73	Pounds per square foot
Inches of mercury (at 62° F)	0.4912	Pounds per square inch
Inches of water (at 62° F)	0.07355	Inches of mercury
Inches of water (at 62° F)	0.03613	Pounds per square inch
Inches of water (at 62° F)	5.202	Pounds per square foot
Inches of water (at 62° F)	0.002458	Atmospheres

TABLE 3-1. Properties of Gases

Gas	C_p	C_v	k	R
Air	0.2375	0.169	1.406	53.3
Ammonia	0.508	0.399	1.273	90.5
Carbon dioxide	0.207	0.162	1.28	35.1
Carbon monoxide	0.243	0.173	1.403	55.1
Hydrogen	3.41	2.42	1.41	765.9
Nitrogen	0.244	0.173	1.41	55.1
Oxygen	0.218	0.156	1.40	48.3
Sulfur dioxide	0.154	0.123	1.26	24.1

TABLE 4-1. Properties of Saturated Steam

	Absolute Pressure		Specific Volume			Enthalpy			Entropy		
Temp., °F, t (1)	Psi, p (2)	In. Hg, p (3)	Sat. liquid, v_f (4)	Evap., v_{fg} (5)	Sat. vapor, v_g (6)	Sat. liquid, h_f (7)	Evap., h_{fg} (8)	Sat. vapor, h_g (9)	Sat. liquid, S_f (10)	Evap., S_{fg} (11)	Sat. vapor, S_g (12)
32	0.08854	0.1803	0.01602	3306	3306	0.00	1075.8	1075.8	0.0000	2.1877	2.1877
33	0.09223	0.1878	0.01602	3180	3180	1.01	1075.2	1076.2	0.0020	2.1821	2.1841
34	0.09603	0.1955	0.01602	3061	3061	2.02	1074.7	1076.7	0.0041	2.1764	2.1805
35	0.09995	0.2035	0.01602	2947	2947	3.02	1074.1	1077.1	0.0061	2.1709	2.1770
36	0.10401	0.2118	0.01602	2837	2837	4.03	1073.6	1077.6	0.0081	2.1654	2.1735
37	0.10821	0.2203	0.01602	2732	2732	5.04	1073.0	1078.0	0.0102	2.1598	2.1700
38	0.11256	0.2292	0.01602	2632	2632	6.04	1072.4	1078.4	0.0122	2.1544	2.1666
39	0.11705	0.2383	0.01602	2536	2536	7.04	1071.9	1078.9	0.0142	2.1489	2.1631
40	0.12170	0.2478	0.01602	2444	2444	8.05	1071.3	1079.3	0.0162	2.1435	2.1597
41	0.12652	0.2576	0.01602	2356	2356	9.05	1070.7	1079.7	0.0182	2.1381	2.1563
42	0.13150	0.2677	0.01602	2271	2271	10.05	1070.1	1080.2	0.0202	2.1327	2.1529
43	0.13665	0.2782	0.01602	2190	2190	11.06	1069.5	1080.6	0.0222	2.1274	2.1496
44	0.14199	0.2891	0.01602	2112	2112	12.06	1068.9	1081.0	0.0242	2.1220	2.1462
45	0.14752	0.3004	0.01602	2036.4	2036.4	13.06	1068.4	1081.5	0.0262	2.1167	2.1429
46	0.15323	0.3120	0.01602	1964.3	1964.3	14.06	1067.8	1081.9	0.0282	2.1113	2.1395
47	0.15914	0.3240	0.01603	1895.1	1895.1	15.07	1067.3	1082.4	0.0302	2.1060	2.1362
48	0.16525	0.3364	0.01603	1828.6	1828.6	16.07	1066.7	1082.8	0.0321	2.1008	2.1329
49	0.17157	0.3493	0.01603	1764.7	1764.7	17.07	1066.1	1083.2	0.0341	2.0956	2.1297
50	0.17811	0.3626	0.01603	1703.2	1703.2	18.07	1065.6	1083.7	0.0361	2.0903	2.1264
51	0.18486	0.3764	0.01603	1644.2	1644.2	19.07	1065.0	1084.1	0.0380	2.0852	2.1232
52	0.19182	0.3906	0.01603	1587.6	1587.6	20.07	1064.4	1084.5	0.0400	2.0799	2.1199
53	0.19900	0.4052	0.01603	1533.3	1533.3	21.07	1063.9	1085.0	0.0420	2.0747	2.1167
54	0.20642	0.4203	0.01603	1481.0	1481.0	22.07	1063.3	1085.4	0.0439	2.0697	2.1136
55	0.2141	0.4359	0.01603	1430.7	1430.7	23.07	1062.7	1085.8	0.0459	2.0645	2.1104
56	0.2220	0.4520	0.01603	1382.4	1382.4	24.06	1062.2	1086.3	0.0478	2.0594	2.1072
57	0.2302	0.4686	0.01603	1335.9	1335.9	25.06	1061.6	1086.7	0.0497	2.0544	2.1041
58	0.2386	0.4858	0.01604	1291.1	1291.1	26.06	1061.0	1087.1	0.0517	2.0493	2.1010
59	0.2473	0.5035	0.01604	1248.1	1248.1	27.06	1060.5	1087.6	0.0536	2.0443	2.0979

TABLE 5-1. Properties of Saturated Water Vapor with Air at Low Temperatures

Temp, °F	Pressure of saturated vapor × 10^5		Weight of saturated vapor				Volume, cu ft barometer, 29.92 in. Hg		Enthalpy per pound		
	In. Hg	Psi	Per cubic feet: Pounds × 10^5	Per cubic feet: Grains	Per pound of dry air: Pounds × 10^5	Per pound of dry air: Grains	Of 1 lb of dry air	Of 1 lb of dry air + vapor to saturate it	Dry air 0°F datum	Vapor 32°F datum	Dry air with vapor to saturate it
(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)	(10)	(11)	(12)
−25	946.4	464.87	1.8016	0.12611	19.68	1.3776	10.95	10.95	−6.011	1048.0	−5.805
−24	1,003.	492.67	1.9049	0.13334	20.86	1.4602	10.97	10.97	−5.770	1048.4	−5.551
−23	1,064.	522.64	2.0162	0.14113	22.13	1.5491	11.00	11.00	−5.529	1048.9	−5.297
−22	1,126.	553.09	2.1287	0.14901	23.42	1.6394	11.02	11.02	−5.288	1049.3	−5.042
−21	1,192.	585.51	2.2484	0.15739	24.79	1.7353	11.05	11.05	−5.047	1049.8	−4.787
−20	1,262.0	619.89	2.3750	0.16625	26.25	1.8375	11.07	11.07	−4.807	1050.2	−4.531
−19	1,337.	656.73	2.5105	0.17574	27.81	1.9467	11.10	11.10	−4.566	1050.7	−4.274
−18	1,416.	695.54	2.6527	0.18569	29.45	2.0615	11.13	11.13	−4.325	1051.1	−4.015
−17	1,496.	734.84	2.7963	0.19574	31.12	2.1784	11.15	11.1	−4.085	1051.6	−3.758
−16	1,584.	778.06	2.9542	0.20679	32.95	2.3065	11.18	11.18	−3.844	1052.0	−3.497
−15	1,675.0	822.76	3.1168	0.21818	34.84	2.4388	11.20	11.21	−3.604	1052.5	−3.237
−14	1,772.	870.41	3.2899	0.23029	36.86	2.5802	11.23	11.24	−3.363	1052.9	−2.975
−13	1,874.	920.51	3.4714	0.24300	38.98	2.7286	11.25	11.26	−3.123	1053.4	−2.712
−12	1,980.	972.58	3.6596	0.25617	41.19	2.8833	11.28	11.29	−2.883	1053.8	−2.449
−11	2,093.	1,028.1	3.8599	0.27019	43.54	3.0478	11.30	11.31	−2.642	1054.3	−2.183
−10	2,210.0	1,085.6	4.0666	0.28466	45.98	3.2186	11.33	11.34	−2.402	1054.7	−1.917
−9	2,335.	1,147.0	4.2871	0.30009	48.58	3.4006	11.35	11.36	−2.162	1055.2	−1.649
−8	2,463.	1,209.8	4.5120	0.31584	51.25	3.5875	11.38	11.39	−1.921	1055.6	−1.380
−7	2,502.	1,229.0	4.5734	0.32014	52.06	3.6442	11.40	11.41	−1.681	1056.1	−1.131
−6	2,745.	1,348.3	5.0066	0.35046	57.12	3.9984	11.43	11.44	−1.441	1056.5	−0.8375
−5	2,898.0	1,423.5	5.2738	0.36917	60.30	4.2210	11.45	11.46	−1.201	1057.0	−0.5636
−4	3,055.	1,500.6	5.5473	0.38831	63.57	4.4499	11.48	11.49	−0.9604	1057.4	−0.2882
−3	3,222.	1,582.6	5.8379	0.40865	67.05	4.6935	11.50	11.51	−0.7203	1057.9	−0.01098
−2	3,397.	1,668.6	6.1414	0.42990	70.69	4.9483	11.53	11.54	−0.4802	1058.3	+0.2679
−1	3,580.	1,758.5	6.4583	0.45208	74.50	5.2150	11.55	11.57	−0.2401	1058.8	+0.5487
0	3,773.0	1,853.3	6.7914	0.47500	78.52	5.5000	11.58	11.59	0	1059.2	+0.8317

From "Heating, Ventilating and Air Conditioning Guide," Chap. 1, 1939; compiled by W. M. Sawdon, vapor pressures converted from International Critical Tables. Reproduced by permission of American Society of Heating, Refrigerating, and Air-Conditioning Engineers.

TABLE 5-2. Properties of Saturated Water Vapor with Air, 0 to 164°F

Temp, °F	Pressure of saturated vapor		Weight of saturated vapor				Volume, cu ft barometer, 29.92 in. Hg		Enthalpy per pound		
			Per cubic feet		Per pound of dry air						
	In. Hg	Psi	Pounds	Grains	Pounds	Grains	Of 1 lb of dry air	Of 1 lb of dry air + vapor to saturate it	Dry air 0°F datum	Vapor 32°F datum	Dry air with vapor to saturate it
(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)	(10)	(11)	(12)
0	0.03773	0.01853	0.000067914	0.475	0.0007852	5.50	11.58	11.59	0.0000	1059.2	0.8317
1	0.03975	0.01963	0.000071395	0.500	0.0008275	5.79	11.60	11.62	0.2401	1059.7	1.117
2	0.04186	0.02056	0.000075021	0.525	0.0008714	6.10	11.63	11.64	0.4801	1060.1	1.404
3	0.04409	0.02166	0.000078851	0.552	0.0009179	6.43	11.65	11.67	0.7201	1060.6	1.694
4	0.04645	0.02282	0.000082890	0.580	0.0009671	6.77	11.68	11.70	0.9601	1061.0	1.986
5	0.04886	0.02400	0.000087005	0.609	0.001017	7.12	11.70	11.72	1.200	1061.5	2.280
6	0.05144	0.02527	0.000091399	0.640	0.001071	7.50	11.73	11.75	1.440	1061.9	2.577
7	0.05412	0.02658	0.000095955	0.672	0.001127	7.89	11.75	11.77	1.680	1062.4	2.877
8	0.05692	0.02796	0.00010070	0.705	0.001186	8.30	11.78	11.80	1.920	1062.8	3.180
9	0.05988	0.02941	0.00010572	0.740	0.001247	8.73	11.80	11.83	2.160	1063.3	3.486
10	0.06295	0.03092	0.00011090	0.776	0.001311	9.18	11.83	11.85	2.400	1063.7	3.795
11	0.06618	0.03251	0.00011634	0.814	0.001379	9.65	11.86	11.88	2.640	1064.2	4.108
12	0.06958	0.03418	0.00012206	0.854	0.001450	10.15	11.88	11.91	2.880	1064.6	4.424
13	0.07309	0.03590	0.00012794	0.896	0.001523	10.66	11.91	11.93	3.120	1065.1	4.742
14	0.07677	0.03771	0.00013410	0.939	0.001600	11.20	11.93	11.96	3.359	1065.5	5.064
15	0.08067	0.03963	0.00014062	0.984	0.001682	11.77	11.96	11.99	3.599	1066.0	5.392
16	0.08469	0.04160	0.00014732	1.031	0.001766	12.36	11.98	12.01	3.839	1066.4	5.722
17	0.08895	0.04369	0.00015440	1.081	0.001855	12.99	12.00	12.04	4.079	1066.9	6.058
18	0.09337	0.04586	0.00016174	1.132	0.001947	13.63	12.03	12.07	4.319	1067.3	6.397
19	0.09797	0.04812	0.00016935	1.185	0.002043	14.30	12.06	12.09	4.559	1067.8	6.741

TABLE 5-2 *(Continued)*

Temp, °F	Pressure of saturated vapor		Weight of saturated vapor				Volume, cu ft barometer, 29.92 in. Hg		Enthalpy per pound		
			Per cubic feet		Per pound of dry air						
	In. Hg	Psi	Pounds	Grains	Pounds	Grains	Of 1 lb of dry air	Of 1 lb of dry air + vapor to saturate it	Dry air 0°F datum	Vapor 32°F datum	Dry air with vapor to saturate it
(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)	(10)	(11)	(12)
20	0.1028	0.05050	0.00017747	1.242	0.002144	15.01	12.08	12.12	4.798	1068.2	7.088
21	0.1078	0.05295	0.00018564	1.299	0.002250	15.75	12.11	12.15	5.038	1068.7	7.443
22	0.1132	0.05560	0.00019439	1.361	0.002361	16.53	12.13	12.18	5.278	1069.1	7.802
23	0.1186	0.05826	0.00020335	1.423	0.002476	17.33	12.16	12.20	5.518	1069.6	8.166
24	0.1244	0.06111	0.00021276	1.489	0.002596	18.17	12.18	12.23	5.758	1070.0	8.536
25	0.1304	0.06405	0.00022255	1.558	0.002722	19.05	12.21	12.26	5.998	1070.5	8.912
26	0.1366	0.06710	0.00023278	1.629	0.002853	19.97	12.23	12.29	6.237	1070.9	9.292
27	0.1432	0.07034	0.00024342	1.704	0.002991	20.94	12.26	12.32	6.477	1071.4	9.682
28	0.1500	0.07368	0.00025445	1.781	0.003133	21.93	12.28	12.34	6.717	1071.8	10.075
29	0.1571	0.07717	0.00026597	1.862	0.003283	22.99	12.31	12.37	6.957	1072.3	10.477
30	0.1645	0.08080	0.00027797	1.946	0.003439	24.07	12.33	12.40	7.197	1072.7	10.886
31	0.1722	0.08458	0.00029043	2.033	0.003601	25.21	12.36	12.43	7.437	1073.2	11.302
32	0.1803	0.08856	0.00030343	2.124	0.003771	26.40	12.38	12.46	7.677	1073.6	11.726
33	0.1879	0.09230	0.00031471	2.203	0.003931	27.52	12.41	12.49	7.917	1074.1	12.139
34	0.1957	0.09610	0.00032690	2.288	0.004094	28.66	12.43	12.51	8.157	1074.5	12.556
35	0.20360	0.1000	0.0003394	2.376	0.004262	29.83	12.46	12.54	8.397	1075.0	12.979
36	0.21195	0.1041	0.0003527	2.469	0.004438	31.07	12.48	12.57	8.636	1075.4	13.409
37	0.22050	0.1083	0.0003662	2.563	0.004618	32.33	12.51	12.60	8.876	1075.9	13.845
38	0.22925	0.1126	0.0003799	2.660	0.004803	33.62	12.53	12.63	9.116	1076.3	14.285
39	0.23842	0.1171	0.0003943	2.760	0.004996	34.97	12.56	12.66	9.356	1076.8	14.736
40	0.24778	0.1217	0.0004090	2.863	0.005194	36.36	12.59	12.69	9.596	1077.2	15.191
41	0.25755	0.1265	0.0004243	2.970	0.005401	37.80	12.61	12.72	9.836	1077.7	15.657
42	0.26773	0.1315	0.0004401	3.081	0.005616	39.31	12.64	12.75	10.08	1078.1	16.13
43	0.27832	0.1367	0.0004566	3.196	0.005840	40.88	12.66	12.78	10.32	1078.6	16.62
44	0.28911	0.1420	0.0004735	3.315	0.006069	42.48	12.69	12.81	10.56	1079.0	17.11

TABLE 5-2 (*Continued*)

45	0.30031	0.1475	0.0004909	3.436	0.006306	44.14	12.71	12.84	10.80	1079.5	17.61
46	0.31191	0.1532	0.0005088	3.562	0.006553	45.87	12.74	12.87	11.04	1079.9	18.12
47	0.32393	0.1591	0.0005274	3.692	0.006808	47.66	12.76	12.90	11.28	1080.4	18.64
48	0.33635	0.1652	0.0005465	3.826	0.007072	49.50	12.79	12.93	11.52	1080.8	19.16
49	0.34917	0.1715	0.0005663	3.964	0.007345	51.42	12.81	12.96	11.76	1081.3	19.70
50	0.36241	0.1780	0.0005866	4.106	0.007626	53.38	12.84	12.99	12.00	1081.7	20.25
51	0.37625	0.1848	0.0006078	4.255	0.007921	55.45	12.86	13.02	12.23	1082.2	20.80
52	0.39051	0.1918	0.0006296	4.407	0.008226	57.58	12.89	13.06	12.47	1082.6	21.38
53	0.40496	0.1989	0.0006516	4.561	0.008534	59.74	12.91	13.09	12.71	1083.1	21.95
54	0.42003	0.2063	0.0006746	4.722	0.008856	61.99	12.94	13.12	12.95	1083.5	22.55
55	0.43570	0.2140	0.0006984	4.889	0.009192	64.34	12.96	13.15	13.19	1084.0	23.15
56	0.45179	0.2219	0.0007228	5.060	0.009536	66.75	12.99	13.19	13.43	1084.4	23.77
57	0.46828	0.2300	0.0007477	5.234	0.009890	69.23	13.01	13.22	13.67	1084.9	24.40
58	0.48538	0.2384	0.0007735	5.415	0.01026	71.82	13.04	13.25	13.91	1085.3	25.05
59	0.50310	0.2471	0.0008003	5.602	0.01064	74.48	13.06	13.29	14.15	1085.8	25.70
60	0.52142	0.2561	0.0008278	5.795	0.01103	77.21	13.09	13.32	14.39	1086.2	26.37
61	0.54035	0.2654	0.0008562	5.993	0.01144	80.08	13.11	13.35	14.63	1086.7	27.06
62	0.55970	0.2749	0.0008852	6.196	0.01186	83.02	12.14	13.39	14.87	1087.1	27.76
63	0.57985	0.2848	0.0009153	6.407	0.01229	86.03	13.16	13.42	15.11	1087.6	28.48
64	0.60042	0.2949	0.0009460	6.622	0.01274	89.18	13.19	13.46	15.35	1088.0	29.21
65	0.62179	0.3054	0.0009778	6.845	0.01320	92.40	13.21	13.49	15.59	1088.5	29.96
66	0.64378	0.3162	0.0010105	7.074	0.01368	95.76	13.24	13.53	15.83	1088.9	30.73
67	0.66638	0.3273	0.0010440	7.308	0.01417	99.19	13.26	13.57	16.07	1089.4	31.51
68	0.68980	0.3388	0.0010816	7.571	0.01468	102.8	13.29	13.60	16.31	1089.8	32.31
69	0.71382	0.3506	0.0011140	7.798	0.01520	106.4	13.31	13.64	16.55	1090.3	33.12
70	0.73866	0.3628	0.0011507	8.055	0.01574	110.2	13.34	13.68	16.79	1090.7	33.96
71	0.76431	0.3754	0.0011884	8.319	0.01631	114.2	13.37	13.71	17.03	1091.2	34.83
72	0.79058	0.3883	0.0012269	8.588	0.01688	118.2	13.40	13.75	17.27	1091.6	35.70
73	0.81766	0.4016	0.0012667	8.867	0.01748	122.4	13.42	13.79	17.51	1092.1	36.60
74	0.84555	0.4153	0.0013075	9.153	0.01809	126.6	13.44	13.83	17.75	1092.5	37.51
75	0.87448	0.4295	0.0013497	9.448	0.01873	131.1	13.47	13.87	17.99	1093.0	38.46
76	0.90398	0.4440	0.0013927	9.749	0.01938	135.7	13.49	13.91	18.23	1093.4	39.42
77	0.93452	0.4590	0.0014371	10.06	0.02005	140.4	13.52	13.95	18.47	1093.9	40.40
78	0.96588	0.4744	0.0014825	10.38	0.02075	145.3	13.54	13.99	18.71	1094.3	41.42
79	0.99825	0.4903	0.0015295	10.71	0.02147	150.3	13.57	14.03	18.95	1094.8	42.46

TABLE 5.2 (*Continued*)

Temp, °F	Pressure of saturated vapor		Weight of saturated vapor				Volume, cu ft barometer, 29.92 in. Hg		Enthalpy per pound		
	In. Hg	Psi	Per cubic feet		Per pound of dry air		Of 1 lb of dry air	Of 1 lb of dry air + vapor to saturate it	Dry air 0°F datum	Vapor 32°F datum	Dry air with vapor to saturate it
			Pounds	Grains	Pounds	Grains					
(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)	(10)	(11)	(12)
80	1.0316	0.5067	0.0015777	11.04	0.02221	155.5	13.59	14.08	19.19	1095.2	43.51
81	1.0661	0.5236	0.0016273	11.39	0.02298	160.9	13.62	14.12	19.43	1095.7	44.61
82	1.1013	0.5409	0.0016781	11.75	0.02377	166.4	13.64	14.16	19.67	1096.1	45.72
83	1.1377	0.5588	0.0017304	12.11	0.02459	172.1	13.67	14.21	19.91	1096.6	46.88
84	1.1752	0.5772	0.0017841	12.49	0.02543	178.0	13.69	14.26	20.15	1097.0	48.05
85	1.2135	0.5960	0.0018389	12.87	0.02629	184.0	13.72	14.30	20.39	1097.5	49.24
86	1.2527	0.6153	0.0018950	13.27	0.02718	190.3	13.74	14.34	20.63	1097.9	50.47
87	1.2933	0.6352	0.0019531	13.67	0.02810	196.7	13.77	14.39	20.87	1098.4	51.74
88	1.3346	0.6555	0.0020116	14.08	0.02904	203.3	13.79	14.44	21.11	1098.8	53.02
89	1.3774	0.6765	0.0020725	14.51	0.03002	210.1	13.82	14.48	21.35	1099.3	54.35
90	1.4211	0.6980	0.0021344	14.94	0.03102	217.1	13.84	14.53	21.59	1099.7	55.70
91	1.4661	0.7201	0.0021982	15.39	0.03205	224.4	13.87	14.58	21.83	1100.2	57.09
92	1.5125	0.7429	0.0022634	15.84	0.03312	231.8	13.89	14.63	22.07	1100.6	58.52
93	1.5600	0.7662	0.0023304	16.31	0.03421	239.5	13.92	14.69	22.32	1101.1	59.99
94	1.6088	0.7902	0.0023992	16.79	0.03535	247.5	13.94	14.73	22.56	1101.5	61.50
95	1.6591	0.8149	0.0024697	17.28	0.03652	255.6	13.97	14.79	22.80	1102.0	63.05
96	1.7108	0.8403	0.0025425	17.80	0.03772	264.0	13.99	14.84	23.04	1102.4	64.62
97	1.7638	0.8663	0.0026164	18.31	0.03896	272.7	14.02	14.90	23.28	1102.9	66.25
98	1.8181	0.8930	0.0026925	18.85	0.04024	281.7	14.04	14.95	23.52	1103.3	67.92
99	1.8741	0.9205	0.0027700	19.39	0.04156	290.9	14.07	15.01	23.76	1103.8	69.63
100	1.9316	0.9487	0.0028506	19.95	0.04293	300.5	14.10	15.07	24.00	1104.2	71.40
101	1.9904	0.9776	0.0029316	20.52	0.04433	310.3	14.12	15.12	24.24	1104.7	73.21
102	2.0507	1.0072	0.0030156	21.11	0.04577	320.4	14.15	15.18	24.48	1105.1	75.06
103	2.1128	1.0377	0.0031017	21.71	0.04726	330.8	14.17	15.25	24.72	1105.6	76.97
104	2.1763	1.0689	0.0031887	22.32	0.04879	341.5	14.20	15.31	24.96	1106.0	78.92

TABLE 5.2 (*Continued*)

105	2.2414	1.1009	0.0032786	22.95	0.05037	352.6	14.22	15.37	25.20	1106.5	80.93
106	2.3084	1.1338	0.0033715	23.60	0.05200	364.0	14.25	15.44	25.44	1106.9	83.00
107	2.3770	1.1675	0.0034650	24.26	0.05368	375.8	14.27	15.50	25.68	1107.4	85.13
108	2.4473	1.2020	0.0035612	24.93	0.05541	387.9	14.30	15.57	25.92	1107.8	87.30
109	2.5196	1.2375	0.0036603	25.62	0.05719	400.3	14.32	15.64	26.16	1108.3	89.54
110	2.5939	1.274	0.0037622	26.34	0.05904	413.3	14.35	15.71	26.40	1108.7	91.86
111	2.6692	1.311	0.0038669	27.07	0.06092	426.4	14.37	15.78	26.64	1109.2	94.21
112	2.7486	1.350	0.0039729	27.81	0.06292	440.4	14.39	15.85	26.88	1109.6	96.70
113	2.8280	1.389	0.0040816	28.57	0.06493	454.5	14.42	15.93	27.12	1110.1	99.20
114	2.9094	1.429	0.0041911	29.34	0.06700	469.0	14.45	16.00	27.36	1110.5	101.76
115	2.9929	1.470	0.0043047	30.13	0.06913	483.9	14.47	16.08	27.60	1111.0	104.40
116	3.0784	1.512	0.0044208	30.95	0.07134	499.4	14.50	16.16	27.84	1111.4	107.13
117	3.1660	1.555	0.0045372	31.76	0.07361	515.3	14.52	16.24	28.08	1111.9	109.92
118	3.2576	1.600	0.0046620	32.63	0.07600	532.0	14.55	16.32	28.32	1112.3	112.85
119	3.3492	1.645	0.0047846	33.49	0.07840	548.8	14.57	16.41	28.56	1112.8	115.80
120	3.4449	1.692	0.0049115	34.38	0.08093	566.5	14.60	16.50	28.80	1113.2	118.89
121	3.5406	1.739	0.005040	35.28	0.08348	584.4	14.62	16.58	29.04	1113.7	122.01
122	3.6404	1.788	0.005173	36.21	0.08616	603.1	14.65	16.68	29.28	1114.1	125.27
123	3.7422	1.838	0.005311	37.18	0.08892	622.4	14.67	16.77	29.52	1114.6	128.63
124	3.8460	1.889	0.005450	38.15	0.09175	642.3	14.70	16.87	29.76	1115.0	132.06
125	3.9519	1.941	0.005590	39.13	0.09466	662.6	14.72	16.96	30.00	1115.5	135.59
126	4.0618	1.995	0.005734	40.14	0.09770	683.9	14.75	17.06	30.24	1115.9	139.26
127	4.1718	2.049	0.005882	41.17	0.1008	705.6	14.77	17.17	30.48	1116.4	143.01
128	4.2858	2.105	0.006031	42.22	0.1040	728.0	14.80	17.27	30.72	1116.8	146.87
129	4.4039	2.163	0.006188	43.32	0.1074	751.8	14.83	17.38	30.96	1117.3	150.96
130	4.5220	2.221	0.006344	44.41	0.1107	774.9	14.85	17.49	31.20	1117.7	154.93
131	4.6441	2.281	0.006504	45.53	0.1143	800.1	14.88	17.61	31.45	1118.2	159.26
132	4.7703	2.343	0.006671	46.70	0.1180	826.0	14.90	17.73	31.69	1118.6	163.68
133	4.8986	2.406	0.006839	47.87	0.1218	852.6	14.93	17.85	31.93	1119.1	168.24
134	5.0289	2.470	0.007010	49.07	0.1257	879.9	14.95	17.97	32.17	1119.5	172.89
135	5.1633	2.536	0.007185	50.30	0.1297	907.9	14.98	18.10	32.41	1120.0	177.67
136	5.2997	2.603	0.007364	51.55	0.1339	937.3	15.00	18.23	32.65	1120.4	182.67
137	5.4402	2.672	2.007547	52.83	0.1382	967.4	15.03	18.36	32.89	1120.9	187.80
138	5.5827	2.742	0.007732	54.12	0.1427	998.9	15.05	18.50	33.13	1121.3	193.14
139	5.7293	2.814	0.007923	55.46	0.1473	1,031.1	15.08	18.65	33.37	1121.8	198.61

TABLE 5.2 *(Continued)*

Temp, °F	Pressure of saturated vapor		Weight of saturated vapor				Volume, cu ft barometer, 29.92 in. Hg		Enthalpy per pound		
	In. Hg	Psi	Per cubic feet		Per pound of dry air		Of 1 lb of dry air	Of 1 lb of dry air + vapor to saturate it	Dry air 0°F datum	Vapor 32°F datum	Dry air with vapor to saturate it
			Pounds	Grains	Pounds	Grains					
(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)	(10)	(11)	(12)
140	5.8779	2.887	0.008116	56.81	0.1521	1,064.7	15.10	18.79	33.61	1122.2	204.30
141	6.0306	2.962	0.008313	58.19	0.1570	1,099.0	15.13	18.94	33.85	1122.7	210.11
142	6.1874	3.039	0.008516	59.61	0.1622	1,135.4	15.15	19.10	34.09	1123.1	216.26
143	6.3482	3.118	0.008724	61.07	0.1675	1,172.5	15.18	19.26	34.33	1123.6	222.53
144	6.5111	3.198	0.008933	62.53	0.1730	1,211.0	15.20	19.43	34.57	1124.0	229.02
145	6.6781	3.280	0.009148	64.04	0.1787	1,250.9	15.23	19.60	34.81	1124.5	235.76
146	6.8471	3.363	0.009366	65.56	0.1846	1,292.2	15.25	19.78	35.05	1124.9	242.71
147	7.0222	3.449	0.009590	67.13	0.1908	1,335.6	15.28	19.96	35.29	1125.4	250.02
148	7.1993	3.536	0.009817	68.72	0.1971	1,379.7	15.30	20.15	35.53	1125.8	257.43
149	7.3805	3.625	0.010040	70.28	0.2037	1,425.9	15.33	20.35	35.77	1126.3	265.20
150	7.5658	3.716	0.010284	71.99	0.2105	1,473.5	15.35	20.55	36.02	1126.7	273.19
151	7.7551	3.809	0.010526	73.68	0.2176	1,523.2	15.38	20.76	36.26	1127.2	281.54
152	7.9485	3.904	0.010772	75.40	0.2250	1,575.0	15.40	20.97	36.50	1127.6	290.21
153	8.1460	4.001	0.011022	77.15	0.2327	1,628.9	15.43	21.20	36.74	1128.1	299.25
154	8.3476	4.100	0.011279	78.95	0.2407	1,684.9	15.45	21.43	36.98	1128.5	308.61
155	8.5532	4.201	0.011539	80.77	0.2490	1,743.0	15.48	21.67	37.22	1129.0	318.34
156	8.7650	4.305	0.011807	82.65	0.2577	1,803.9	15.50	21.93	37.46	1129.4	328.51
157	8.9788	4.410	0.012077	84.54	0.2667	1,866.9	15.53	22.19	37.70	1129.9	339.04
158	9.1986	4.518	0.012354	86.48	0.2761	1,932.7	15.56	22.46	37.94	1130.3	350.02
159	9.4206	4.627	0.012634	88.44	0.2858	2,000.6	15.58	22.74	38.18	1130.8	361.36
160	9.6486	4.739	0.012919	90.43	0.2961	2,072.7	15.61	23.03	38.43	1131.2	373.38
161	9.8807	4.853	0.013211	92.48	0.3067	2,146.9	15.63	23.33	38.67	1131.7	385.76
162	10.119	4.970	0.013509	94.56	0.3179	2,225.3	15.66	23.65	38.91	1132.1	398.80
163	10.361	5.089	0.013812	96.68	0.3295	2,306.5	15.68	23.98	39.15	1132.5	412.34
164	10.608	5.210	0.014120	98.84	0.3416	2,391.2	15.71	24.33	39.39	1133.0	426.42

From "Heating, Ventilating and Air Conditioning Guide," Chap. 1, 1939; compiled by W. M. Sawdon, vapor pressures converted from International Critical Tables. Reproduced by permission of American Society of Heating, Refrigerating, and Air-Conditioning Engineers.

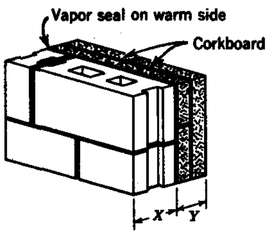
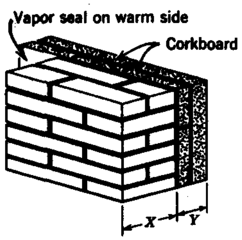
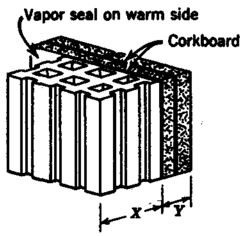
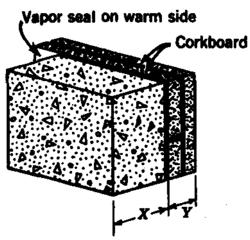
TABLE 7-1. Temperature and Enthalpy of Discharge Vapor after Isentropic Compression

Saturated Suction Temperature	Condensing Temperature 80°		90°		100°	
	t	*h*	*t*	*h*	*t*	*h*
−40°	111.0°	91.6	121.0°	92.3	132.0°	93.9
−30°	105.0°	90.5	116.0°	92.0	127.5°	93.2
−20°	102.0°	90.2	112.5°	91.4	124.0°	92.6
−10°	97.5°	89.5	108.5°	90.7	119.9°	91.9
0°	95.0°	89.2	106.0°	90.3	117.0°	91.5
10°	92.0°	88.7	103.5°	89.9	114.0°	90.9
20°	90.0°	88.4	102.0°	89.6	112.0°	90.6
30°	88.0°	88.1	99.0°	89.1	110.8°	90.4
40°	86.0°	87.7	97.0°	88.8	109.5°	90.2
50°	84.0°	87.4	95.5°	88.6	107.0°	89.8

Saturated Suction Temperature	Condensing Temperature 110°		120°		130°	
	t	*h*	*t*	*h*	*t*	*h*
−40°	143.0°	95.1	155.0°	96.3	166.5°	97.3
−30°	138.0°	94.3	150.5°	95.5	161.5°	96.6
−20°	135.5°	93.7	147.0°	94.8	157.5°	95.8
−10°	131.6°	93.1	143.0°	94.2	154.0°	95.2
0°	128.5°	92.6	141.0°	93.7	152.0°	94.8
10°	126.5°	92.1	137.5°	93.2	148.5°	94.3
20°	124.0°	91.7	136.0°	92.8	147.2°	93.9
30°	122.0°	91.4	133.5°	92.5	146.0°	93.6
40°	120.0°	91.1	132.5°	92.2	143.5°	93.2
50°	118.0°	90.8	131.0°	92.0	142.0°	92.9

TABLE 10-1. Heat Transmission Coefficients (*U*) for Cold Storage Rooms

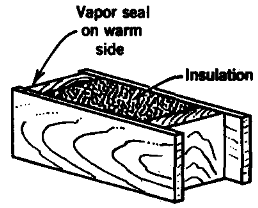
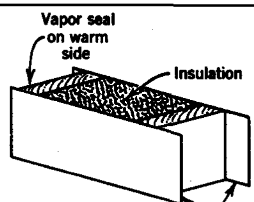
Btu per hour per square foot per degree F difference between air on the two sides. Wind velocity 15 mph.

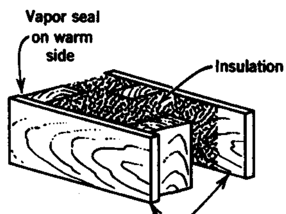
	Wall Thickness *X* Inches	Thickness of Insulation, *Y* Inches						
		2	3	4	5	6	7	8
	Concrete block 8	0.12	0.085	0.066	0.054	0.046	0.040	0.035
	Concrete block 12	0.12	0.083	0.065	0.053	0.045	0.039	0.035
	Cinder block 8	0.11	0.081	0.064	0.052	0.045	0.039	0.034
	Cinder block 12	0.11	0.079	0.063	0.052	0.044	0.039	0.034
	Common brick 8	0.11	0.081	0.064	0.053	0.045	0.039	0.034
	Common brick 12	0.10	0.076	0.061	0.050	0.043	0.038	0.034
	Clay tile 4	0.12	0.085	0.066	0.054	0.046	0.040	0.035
	Clay tile 6	0.11	0.081	0.064	0.053	0.045	0.039	0.035
	Clay tile 8	0.11	0.081	0.064	0.052	0.045	0.039	0.034
	Concrete 6	0.13	0.089	0.069	0.056	0.047	0.041	0.036
	Concrete 8	0.12	0.087	0.068	0.055	0.047	0.040	0.036
	Concrete 10	0.12	0.086	0.067	0.055	0.046	0.040	0.035
	Concrete 12	0.12	0.085	0.066	0.054	0.046	0.040	0.035

From *Carrier Design Data.* Reproduced by permission of Carrier Corporation.

TABLE 10-2. Heat Transmission Coefficients (*U*) for Cold Storage Rooms

Btu per hour per square foot per degree F difference between the air on the two sides. Outside wind velocity 15 mph.

Type of Construction	Insulating Material	Thickness of Insulation (Inches)						
		$3\frac{5}{8}$	$5\frac{5}{8}$	2	3	4	5	6
	Granulated cork	0.079	0.055					
	Rock or palco wool	0.072	0.050					
	Sawdust	0.097	0.069					
1" board on both sides of studs[a]	Corkboard	-----	-----	0.11	0.084	0.067	0.055	0.047
 Sheet steel on both sides of studs[a]	Glass or rock wool fill	0.084	0.055	-----	0.100	0.077	0.062	0.052

Type of Construction	Insulating Material	Thickness of Insulation (Inches)		
		8	10	12
	Granulated cork	0.040	0.033	0.027
	Palco or rock wool	0.036	0.029	0.025
1" board both sides—2" x 4" studs—16" ℄[b]	Sawdust	0.051	0.042	0.035

NOTES:

[a] Coefficients corrected for 2 x 4 or 2 x 6 studs, on 16 in. centers.

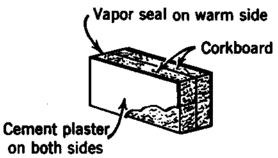
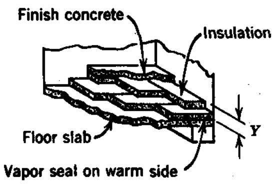
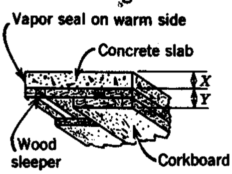
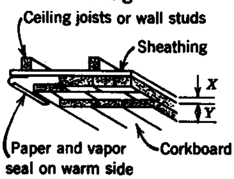
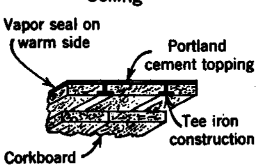
[b] Coefficients corrected for 2 x 4 studs

* Actual thickness $\frac{25}{32}$ in.

From *Carrier Design Data.* Reproduced by permission of Carrier Corporation.

TABLE 10-3. Heat Transmission Coefficients (*U*) for Cold Storage Rooms

Btu per hour per square foot per degree F difference between air on the two sides. Wind velocity 15 mph.

<table>
<tr><th rowspan="2"></th><th rowspan="2">Wall, Floor or Ceiling Thickness X (Inches)</th><th colspan="7">Thickness of Insulation, Y Inches</th></tr>
<tr><th>2</th><th>3</th><th>4</th><th>5</th><th>6</th><th>7</th><th>8</th></tr>
<tr><td>Self-supporting partition*

</td><td>Cork partition</td><td>0.13</td><td>0.089</td><td>0.069</td><td>0.056</td><td>0.047</td><td>0.041</td><td>0.036</td></tr>
<tr><td rowspan="8">Floor *

</td><td></td><td colspan="7">Corkboard[a]</td></tr>
<tr><td>Slab 2
Finish 2</td><td>0.12</td><td>0.087</td><td>0.067</td><td>0.055</td><td>0.046</td><td>0.040</td><td>0.035</td></tr>
<tr><td>Slab 5
Finish 3</td><td>0.12</td><td>0.084</td><td>0.066</td><td>0.054</td><td>0.046</td><td>0.040</td><td>0.035</td></tr>
<tr><td>Slab 6
Finish 4</td><td>0.11</td><td>0.083</td><td>0.065</td><td>0.054</td><td>0.045</td><td>0.039</td><td>0.035</td></tr>
<tr><td></td><td colspan="7">Foamglas[a]</td></tr>
<tr><td>Slab 2
Finish 2</td><td>0.15</td><td>0.11</td><td>0.087</td><td>0.071</td><td>0.060</td><td>0.053</td><td>0.046</td></tr>
<tr><td>Slab 5
Finish 3</td><td>0.15</td><td>0.11</td><td>0.084</td><td>0.070</td><td>0.059</td><td>0.052</td><td>0.046</td></tr>
<tr><td>Slab 6
Finish 4</td><td>0.14</td><td>0.10</td><td>0.083</td><td>0.069</td><td>0.059</td><td>0.051</td><td>0.045</td></tr>
<tr><td rowspan="2">Ceiling*

</td><td>Concrete 4</td><td>0.12</td><td>0.089</td><td>0.069</td><td>0.056</td><td>0.048</td><td>0.042</td><td>0.036</td></tr>
<tr><td>Concrete 8</td><td>0.12</td><td>0.086</td><td>0.067</td><td>0.055</td><td>0.047</td><td>0.041</td><td>0.036</td></tr>
<tr><td>Ceiling*

</td><td>Wood 25/32 (actual)</td><td>0.11</td><td>0.082</td><td>0.064</td><td>0.053</td><td>0.045</td><td>0.039</td><td>0.035</td></tr>
<tr><td>Ceiling*

</td><td></td><td>0.13</td><td>0.092</td><td>0.072</td><td>0.059</td><td>0.050</td><td>0.043</td><td>0.038</td></tr>
</table>

[a] These values may also be used for floors on ground.
* Surface conductance for still air, 1.65, used on both sides

From *Carrier Design Data*. Reproduced by permission of Carrier Corporation.

TABLE 10-4. Thermal Conductivity of Materials Used in Cold Storage Rooms

Material	Thermal Conductivity (k) (Btu per hour per sq ft per °F per inch thickness)	Thermal Conductance (C) (Btu per hour per sq ft per °F per test thickness)	Authority for 2nd and 3rd Columns	Practical* Thermal Conductivity (k) (Btu per hour per sq ft per °F per inch thickness)
Brick, common	5.0	—	1	—
Cement plaster	8.0	—	1	—
Concrete	12.0	—	1	—
Cinder aggregate block 8″	—	0.60	1	—
Cinder aggregate block 12″	—	0.53	1	—
Gravel aggregate block 8″	—	1.0	1	—
Gravel aggregate block 12″	—	0.80	1	—
Corkboard	0.28	—	2	0.30
Cork, granulated coarse	0.31	—	2	0.34
Foamglas	0.40	—	4	0.40
Glass wool, density 1.5 lb per cu ft	0.27	—	1	0.30
Mineral wool board	0.33	—	3	0.36
Redwood bark, palco wool	0.26	—	1	0.29
Rock wool, density 10.0 lb per cu ft	0.27	—	1	0.30
Sawdust, various woods	0.41	—	2	0.45
Tile, hollow clay 4″	—	1.0	1	—
Tile, hollow clay 6″	—	0.64	1	—
Tile, hollow clay 8″	—	0.60	1	—
Wood, yellow pine, or fir	0.80	—	1	—

Authorities: (1) *ASHVE Guide* 1945. (2) *ASRE Data Book*, Vol. 1—1943. (3) *ASRE Data Book*, Vol. 2—1942. (4) Pittsburgh Corning Corporation.

* These conductivities were used for insulating materials in calculation of heat transmission coefficients. Most of these values have been increased 10% above laboratory test values to allow for the effect of moisture gain in the insulating material and for imperfect workmanship. This also assumes adequate vapor sealing. When no vapor sealing is applied or where the workmanship is poor the value of the insulation is largely destroyed. It is extremely difficult to get a good vapor seal with loose fill type insulation.

Foamglas. If a combination of corkboard and Foamglas is used, 1 in. of Foamglas is equivalent to $\frac{3}{4}$ in. of corkboard.

Mineral Wool Board. For estimating purposes use heat transmission coefficients for corkboard increased by 15%.

From *Carrier Design Data*. Reproduced by permission of Carrier Corporation.

TABLE 10-5. *U* Factors for Glass

Number of Panes	Btu/hr/sq ft/° F
1	1.13
2	0.46
3	0.29
4	0.21

From *ASRE Data Book*, Design Volume, 1949 Edition, by permission of the American Society of Heating, Refrigerating, and Air-Conditioning Engineers.

TABLE 10-5A. Surface Conductance (f) for Building Structures

		Surface Conductance (Btu per hour per square foot per ° F)	
Surface	Exposure	Winter	Summer
Ceilings	Inside	1.65	1.20
Roofs	Outside	6.00*	4.00†
Walls	Inside	1.65	1.65
Walls	Outside	6.00*	4.00†

* Average wind velocity 15 mph.
† Average wind velocity 8 mph.
From *Carrier Design Data.* Reproduced by permission of Carrier Corporation.

TABLE 10-6. Refrigeration Design Ambient Temperature Guide*

Location	Average Ambient Temp.	Maximum Ambient Temp.
Alabama		
Birmingham	88	99
Mobile	88	97
Arizona		
Flagstaff	75	90
Phoenix	100	113
Tucson	84	98
Arkansas		
Fort Smith	91	103
Little Rock	90	100
California		
Bakersfield	96	114
Fresno	94	111
Los Angeles	83	94
Oakland	75	89
Sacramento	90	108
San Diego	75	80
San Francisco	75	83
Colorado		
Colorado Springs	83	94
Denver	83	98
Grand Junction	88	102
Pueblo	83	100
Connecticut		
Hartford	83	94
New Haven	83	95
New London	83	93
Norwalk	83	96
Delaware		
Dover	87	96
Milford	87	98
Wilmington	87	94
District of Columbia		
Washington	89	98
Florida		
Jacksonville	88	96
Miami	88	90
Orlando	88	97
Tallahassee	88	100
Tampa	88	95
Georgia		
Atlanta	87	95
Savannah	89	99
Idaho		
Boise	89	105
Pocatello	83	100
Illinois		
Cairo	89	101
Chicago	87	98
Peoria	88	100
Quincy	90	103
Rockford	87	101
Springfield	90	102

TABLE 10-6 (*Continued*)

Location	Average Ambient Temp.	Maximum Ambient Temp.
Indiana		
Evansville	90	100
Fort Wayne	87	100
Indianapolis	89	99
South Bend	87	101
Terre Haute	90	100
Iowa		
Burlington	90	101
Davenport	90	100
Des Moines	90	102
Dubuque	90	99
Keokuk	90	101
Mason City	86	97
Sioux City	90	102
Kansas		
Concordia	93	108
Dodge City	92	106
Hutchinson	92	108
Salina	95	111
Topeka	92	105
Wichita	91	104
Kentucky		
Lexington	86	98
Louisville	88	99
Louisiana		
Baton Rouge	88	98
New Orleans	89	98
Shreveport	92	102
Maine		
Eastport	70	81
Portland	81	93
Maryland		
Baltimore	89	99
Cumberland	87	102
Massachusetts		
Boston	84	94
Fall River	81	90
Lawrence	81	94
Worcester	81	92
Michigan		
Alpena	82	95
Detroit	86	99
Grand Rapids	86	98
Jackson	86	99
Lansing	86	96
Marquette	81	96
Saginaw	88	101

Location	Average Ambient Temp.	Maximum Ambient Temp.
Minnesota		
Duluth	79	92
Minneapolis	90	102
St. Cloud	88	101
Mississippi		
Jackson	90	99
Vicksburg	90	96
Missouri		
Hannibal	90	102
Kansas City	92	103
St. Joseph	92	103
St. Louis	92	103
Springfield	88	98
Montana		
Billings	85	104
Butte	75	96
Havre	82	99
Helena	82	102
Nebraska		
Lincoln	94	106
North Platte	89	103
Omaha	92	104
Nevada		
Reno	84	101
Tonopah	84	96
New Hampshire		
Concord	81	92
New Jersey		
Atlantic City	83	92
Paterson	85	95
Trenton	85	96
New Mexico		
Albuquerque	83	99
Santa Fe	81	90
New York		
Albany	83	96
Binghamton	83	94
Buffalo	80	89
Elmira	83	97
New York	85	93
Poughkeepsie	83	95
Rochester	83	95
Syracuse	83	96
Watertown	83	93

TABLE 10-6 (*Continued*)

Location	Average Ambient Temp.	Maximum Ambient Temp.	Location	Average Ambient Temp.	Maximum Ambient Temp.
North Carolina			Tennessee		
Asheville	81	93	Chattanooga	87	98
Charlotte	86	98	Knoxville	87	98
Raleigh	86	98	Memphis	89	99
Wilmington	86	95	Nashville	87	98
Winston-Salem	86	97	Texas		
North Dakota			Dallas	92	102
Bismarck	87	103	El Paso	92	102
Devils Lake	84	100	Fort Worth	92	104
Ohio			Houston	92	99
Akron	86	98	San Antonio	92	102
Canton	86	97	Utah		
Cincinnati	88	100	Modena	80	97
Cleveland	83	95	Salt Lake City	88	101
Columbus	88	98	Vermont		
Dayton	88	99	Burlington	80	91
Toledo	87	99	Virginia		
Youngstown	86	97	Lynchburg	87	99
Oklahoma			Norfolk	87	95
Oklahoma City	92	104	Richmond	87	98
Tulsa	92	105	Washington		
Oregon			Olympia	75	90
Portland	81	95	Seattle	75	86
Pennsylvania			Spokane	75	102
Altoona	82	96	Walla Walla	87	105
Erie	83	92	West Virginia		
Harrisburg	85	97	Charleston	87	102
Philadelphia	87	97	Clarksburg	84	97
Pittsburgh	85	96	Huntington	87	100
Scranton	82	95	Parkersburg	86	98
Rhode Island			Wheeling	86	101
Providence	83	94	Wisconsin		
South Carolina			Green Bay	85	97
Charleston	88	98	La Crosse	87	99
Columbia	88	99	Madison	87	96
South Dakota			Milwaukee	87	99
Huron	93	107	Wyoming		
Pierre	94	110	Cheyenne	79	94
Rapid City	87	103	Lander	80	98
Sioux Falls	88	102	Sheridan	86	102

* Do not use these temperatures for air conditioning design.

From *ASRE Data Book*, Design Volume, 1949 Edition, by permission of the American Society of Heating, Refrigerating, and Air-Conditioning Engineers.

TABLE 10-6A. Design Ground Temperatures

Location	Ground Temperature
Alabama	
Birmingham	70° F
Mobile	75
Arizona	
Flagstaff	60
Phoenix	80
Tucson	80
Arkansas	
Fort Smith	70
Little Rock	70
California	
Bakersfield	75
Fresno	80
Los Angeles	75
Oakland	65
Sacramento	80
San Diego	65
San Francisco	65
Colorado	
Colorado Springs	60
Denver	60
Grand Junction	60
Pueblo	55
Connecticut	
Hartford	65
New Haven	65
New London	65
Norwalk	65
Delaware	
Dover	65
Milford	65
Wilmington	65
District of Columbia	
Washington	65
Florida	
Jacksonville	80
Miami	80
Orlando	80
Tallahassee	80
Tampa	80
Georgia	
Atlanta	70
Savannah	75
Idaho	
Boise	60° F
Pocatello	60
Illinois	60
Cairo	
Chicago	60
Peoria	60
Quincy	60
Rockford	60
Springfield	60
Indiana	
Evansville	65
Fort Wayne	60
Indianapolis	60
South Bend	60
Terre Haute	65
Iowa	
Burlington	60
Davenport	60
Des Moines	60
Bubuque	60
Keokuk	60
Mason City	60
Sioux City	60
Kansas	
Concordia	60
Dodge City	60
Hutchinson	60
Salina	60
Topeka	60
Wichita	60
Kentucky	
Lexington	65
Louisville	65
Louisiana	
Baton Rouge	75
New Orleans	75
Shreveport	70
Maine	
Eastport	60
Portland	60
Maryland	
Baltimore	65
Cumberland	65

TABLE 10-6A *(Continued)*

Location	Ground Temperature
Massachusetts	
Boston	65° F
Fall River	60
Lawrence	60
Worcester	60
Michigan	
Alpena	60
Detroit	60
Grand Rapids	60
Jackson	60
Lansing	60
Marquette	60
Saginaw	60
Minnesota	
Duluth	50
Minneapolis	55
St. Cloud	55
Mississippi	
Jackson	75
Vicksburg	75
Missouri	
Hannibal	60
Kansas City	60
St. Joseph	60
St. Louis	60
Springfield	60
Montana	
Billings	55
Butte	55
Havre	50
Helena	55
Nebraska	
Lincoln	60
North Platte	55
Omaha	60
Nevada	
Reno	65
Tonopah	70
New Hampshire	
Concord	55
New Jersey	
Atlantic City	70
Paterson	70
Trenton	70
New Mexico	
Albuquerque	70° F
Santa Fe	65
New York	
Albany	60
Binghamton	60
Buffalo	65
Elmira	60
New York	65
Poughkeepsie	60
Rochester	60
Syracuse	60
Watertown	60
North Carolina	
Asheville	70
Charlotte	70
Raleigh	70
Wilmington	75
Winston-Salem	75
North Dakota	
Bismarck	50
Devils Lake	50
Ohio	
Akron	65
Canton	65
Cincinnati	65
Cleveland	65
Columbus	60
Dayton	65
Toledo	60
Youngstown	60
Oklahoma	
Oklahoma City	65
Tulsa	65
Oregon	
Portland	70
Pennsylvania	
Altoona	65
Erie	65
Harrisburg	70
Philadelphia	70
Pittsburgh	65
Scranton	65
Rhode Island	
Providence	65

TABLE 10-6A *(Continued)*

Location	Ground Temperature	Location	Ground Temperature
South Carolina		Virginia	
Charleston	75° F	Lynchburg	75° F
Columbia	75	Norfork	75
		Richmond	70
South Dakota		Washington	
Huron	55	Olympia	60
Pierre	55	Seattle	75
Rapid City	55	Spokane	60
Sioux Falls	55	Walla Walla	60
Tennessee		West Virginia	
Chattanooga	70	Charleston	65
Knoxville	70	Clarksburg	65
Memphis	70	Huntington	65
Nashville	70	Parkersburg	65
Texas		Wheeling	65
Dallas	70	Wisconsin	
El Paso	70	Green Bay	55
Fort Worth	70	La Crosse	55
Houston	75	Madison	55
San Antonio	75	Milwaukee	55
Utah		Wyoming	
Modena	60	Cheyenne	55
Salt Lake City	60	Lander	55
Vermont		Sheridan	55
Burlington	60		

TABLE 10-7. Allowance for Solar Radiation

(Degrees Fahrenheit to be added to the normal temperature difference for heat leakage calculations to compensate for sun effect—not to be used for air-conditioning design)

Type of Surface	East Wall	South Wall	West Wall	Flat Roof
Dark-colored surfaces such as: Slate roofing, Tar roofing, Black paints	8	5	8	20
Medium-colored surfaces, such as: Unpainted wood, Brick, Red tile, Dark cement, Red, gray, or green paint	6	4	6	15
Light-colored surfaces, such as: White stone, Light-colored cement, White paint	4	2	4	9

From *ASRE Data Book*, Design Volume, 1957–1958 Edition, by permission of the American Society of Heating, Refrigerating, and Air-Conditioning Engineers.

TABLE 10-8A. Btu per Cubic Foot of Air Removed in Cooling to Storage Conditions above 30°

Storage Room Temp., °F	Inlet Air Temperature, °F: 85			90			95		100	
	Inter. Air Relative Humidity, %: 50	60	70	50	60	70	50	60	50	60
65	0.65	0.85	1.12	0.93	1.17	1.44	1.24	1.54	1.58	1.95
60	0.85	1.03	1.26	1.13	1.37	1.64	1.44	1.74	1.78	2.15
55	1.12	1.34	1.57	1.41	1.66	1.93	1.72	2.01	2.06	2.44
50	1.32	1.54	1.78	1.62	1.87	2.15	1.93	2.22	2.28	2.65
45	1.50	1.73	1.97	1.80	2.06	2.34	2.12	2.42	2.47	2.85
40	1.69	1.92	2.16	2.00	2.26	2.54	2.31	2.62	2.67	3.06
35	1.86	2.09	2.34	2.17	2.43	2.72	2.49	2.79	2.85	3.24
30	2.00	2.24	2.49	2.26	2.53	2.82	2.64	2.94	2.95	3.35

Reprinted from *Refrigeration Engineering Data Book* by courtesy of American Society of Refrigerating Engineers.

TABLE 10-8B. Btu per Cubic Foot Removed in Cooling to Storage Conditions below 30°

Storage Room Temp., °F	Inlet Air Temperature, °F: 40		50		80		90		100	
	Inter. Air Relative Humidity, %: 70	80	70	80	50	60	50	60	50	60
30	0.24	0.29	0.58	0.66	1.69	1.87	2.26	2.53	2.95	3.35
25	0.41	0.45	0.75	0.83	1.86	2.05	2.44	2.71	3.14	3.54
20	0.56	0.61	0.91	0.99	2.04	2.22	2.62	2.90	3.33	3.73
15	0.71	0.75	1.06	1.14	2.20	2.39	2.80	3.07	3.51	3.92
10	0.85	0.89	1.19	1.27	2.38	2.52	2.93	3.20	3.64	4.04
5	0.98	1.03	1.34	1.42	2.51	2.71	3.12	3.40	3.84	4.27
0	1.12	1.17	1.48	1.56	2.68	2.86	3.28	3.56	4.01	4.43
−5	1.23	1.28	1.59	1.67	2.79	2.98	3.41	3.69	4.15	4.57
−10	1.35	1.41	1.73	1.81	2.93	3.13	3.56	3.85	4.31	4.74
−15	1.50	1.53	1.85	1.93	3.05	3.25	3.67	3.96	4.42	4.86
−20	1.63	1.68	2.01	2.09	3.24	3.44	3.88	4.18	4.66	5.10
−25	1.77	1.80	2.12	2.21	3.38	3.56	4.00	4.30	4.78	5.21
−30	1.90	1.95	2.29	2.38	3.55	3.76	4.21	4.51	5.00	5.44

Reprinted from *Refrigeration Engineering Data Book* by courtesy of American Society of Refrigerating Engineers.

TABLE 10-9A. Average Air Changes per 24 Hours for Storage Rooms above 32° F due to Door Opening and Infiltration

(Does not apply to rooms using ventilating ducts or grilles)

Volume cu ft	Air Changes per 24 hr	Volume cu ft	Air Changes per 24 hr	Volume cu ft	Air Changes per 24 hr	Volume cu ft	Air Changes per 24 hr
250	38.0	1,000	17.5	6,000	6.5	30,000	2.7
300	34.5	1,500	14.0	8,000	5.5	40,000	2.3
400	29.5	2,000	12.0	10,000	4.9	50,000	2.0
500	26.0	3,000	9.5	15,000	3.9	75,000	1.6
600	23.0	4,000	8.2	20,000	3.5	100,000	1.4
800	20.0	5,000	7.2	25,000	3.0		

NOTE: For storage room with anterooms, reduce air changes to 50% of values in table.
For heavy duty usage, add 50% to values given in table.

From *ASRE Data Book*, Design Volume, 1949 Edition, by permission of the American Society of Heating, Refrigerating, and Air-Conditioning Engineers.

TABLE 10-9B. Average Air Changes per 24 Hours for Storage Rooms below 32° F due to Door Opening and Infiltration

(Does not apply to rooms using ventilating ducts or grilles)

Volume cu ft	Air Changes per 24 hr	Volume cu ft	Air Changes per 24 hr	Volume cu ft	Air Changes per 24 hr	Volume cu ft	Air Changes per 24 hr
250	29.0	1,000	13.5	5,000	5.6	25,000	2.3
300	26.2	1,500	11.0	6,000	5.0	30,000	2.1
400	22.5	2,000	9.3	8,000	4.3	40,000	1.8
500	20.0	2,500	8.1	10,000	3.8	50,000	1.6
600	18.0	3,000	7.4	15,000	3.0	75,000	1.3
800	15.3	4,000	6.3	20,000	2.6	100,000	1.1

NOTE: (1) For storage rooms with anterooms, reduce air changes to 50% of values in table.
For heavy duty usage, add 50% to values given in table.
(2) For locker plant rooms, double the above table values.

From *ASRE Data Book*, Design Volume, 1949 Edition, by permission of the American Society of Heating, Refrigerating, and Air-Conditioning Engineers.

TABLE 10-10. Design Data for Fruit Storage

FRUITS	TYPE OF STORAGE	DESIGN ROOM CONDITIONS: Temperature: Recommended Deg F	Temperature: Permissible Range Deg F	Relative Humidity: Recommended %	Relative Humidity: Permissible Range %	Grains per lb Air at Recommended Condition	Maximum Storage Period	CHILLING DATA: Product Temp. Deg F: Start	Product Temp. Deg F: Finish	Time Hr	Rate Factor	Est Prod Latent Heat Btu/lb 24 Hr (Ex. see Note f)	SPECIFIC HEAT Btu/lb/Deg F: Before Freezing	After Freezing	Latent Heat of Fusion Btu/lb	Water Content %	Freezing Point Deg F	Maximum Air Motion in Room Ft./Min
Apples	Short	36k	35-40	87	85-88	26.0						4.0	0.89	0.43	122	84	28.9	90
	Long	30k	30-32	87b	85-88	20.8	48 Mo$_g$					0.2						60
	Chill Start	40		85		31.0		80	32	24	0.67	24.0f						150
	Chill Finish	30		85		20.4						0.3						60d
Apricots	Short	35	35-40	85	80-85	25.2						4.0	0.92	0.80	122	86	28.1	90
	Long	32	31-32	85a	80-85	22.3	7-14 Days					0.3						60
	Chill Start	40		85		31.0		80	33	20	0.67	20.0f						150
	Chill Finish	32		85		22.3						0.3						60d
Avocados	Short	40k	40-53	85b	85-90	31.0						4.5	0.91	0.49	136	94	27.2	90
	Long	38k	37-53	85b	85-90	28.8	10 Days					0.3						90
	Chill Start	40		85		31.0		80	39	22	0.67	22.0f						250
	Chill Finish	53		85		23.2						0.3						90d
Bananas i (See Document 2D-84)	Ripening	70	62-70	95	90-95	104.7		Heating 56°-70°				2.0	0.90		108	75	26-30	90
	Chill Start	70		95		104.7		68	56	12	0.1	11.0f						150
	Chill Finish	56		90		60	10 Days					1.0						90d
	Holding Green	56	56-60	92	90-95	61.3						1.0						90
	Holding Ripe	56	56-60	87	85-90	58.8						1.0						90
Berries (General)	Short	36	35-40	86	80-85	25.2						6.3	0.90	0.49	128	84	28-30	90
	Long	32	31-32	85b	80-85	22.3	3-10 Da.$_g$					0.3						60
	Chill Start	40		86		31.0		80	34	20	0.67	20.0f						150
	Chill Finish	32		86		22.3						0.3						60d
Cranberries	Short	36	36-40	85	85-90	26.4						5.0	0.91	0.47	122	88	27.3	90
	Long	36	36-40	85b	85-90	26.4	1-3 Mo					0.2						90
	Chill Start	40		86		31.0		70	38	20	0.67	18.0f						150
	Chill Finish	36		86		26.2						0.2						90d
Dates (Cured)	Short	36k	36-40	70c	65-75	20.8						0.10	0.36		26	18	−4	150
	Long	28k	28-32	70c	65-75	15.4	3-6 mo$_g$					0.05						150
Dried Fruits	Short	35	35-40	70c	70-75	20.8						0.10	0.47	0.32	43	30		150
	Long	32 L	32-36	70c	70-75	18.6	9-12 Mo					0.07						150
Figs and Dates (Fresh)	Short	40	40-50	75	65-75	27.5						5.0	0.71	0.44	116	90	28.3	90
	Long	34	34-36	70	65-75	20.0	15 Days					0.4						90
Grapes (American Eastern)	Short	36	35-40	85	80-85	25.2						5.0	0.90	0.61	112	77	28.0	90
	Long	31	31-32	85b	80-85	21.3	3-8 Wk$_g$					0.4						90
	Chill Start	40		85		31.0		70	34	20	0.80	14.0f						250
	Chill Finish	32		86		22.3						0.4						90d
Grapes (Vinifera California)	Short	36	35-40	86	85-90	25.2						5.0	0.85	0.59	112	79	24.3	90
	Long	30	30-31	85b	85-90	20.4	3-6 Mo					0.4						90
	Chill Start	40		86		31.0		70	34	20	0.80	14.0f						250
	Chill Finish	32		86		22.3						0.4						90d
Grapefruit	Short	40	40-45	90	85-90	32.0						2.0	0.91	0.49	120	88	28.4	90
	Long	32 j	32-34	85b	85-90	22.3	6-8 Wk					0.3						90
	Chill Start	40		85		31.0		75	34	22	0.70	19.0f						250
	Chill Finish	32		85		22.3						0.3						90d
Lemons	Short	55	55-60	85b	85-90	54.5						3.0	0.91	0.49	126	88	28.1	90
	Long	55	55-60	85b	85-90	54.5	1-4 Mo					0.3						90
	Chill Start	60		85		65.5		75	57	20	1.0	10.0f						250
	Chill Finish	55		85		54.5						0.3						90d
Limes	Short	45	45-50	85b	85-90	37.5						4.0	0.91	0.49	126	88	29.3	60
	Long	45	45-48	90b	85-90	39.6	6-8 Wk					0.2						60
	Chill Start	50		85		45.2		75	47	20	0.90	14.0f						180
	Chill Finish	45		85		37.5						0.2						60d
Oranges	Short	40	40-45	86	85-90	31.0						4.0	0.91	0.44	125	81	28.0	90
	Long	32	32-34	85b	85-90	22.3	8-10 Wk$_g$					0.3						90
	Chill Start	40		86		31.0		75	32	22	0.70	19.0f						250
	Chill Finish	32		86		22.3						0.3						90d

TABLE 10-10 (*Continued*)

FRUITS	TYPE OF STORAGE	Design Room Conditions: Temperature: Recommended Deg F	Design Room Conditions: Temperature: Permissible Range Deg F	Design Room Conditions: Relative Humidity: Recommended %	Design Room Conditions: Relative Humidity: Permissible Range %	Design Room Conditions: Grains per lb Air at Recommended Condition	Maximum Storage Period	Chilling Data: Product Temp. Deg F: Start	Chilling Data: Product Temp. Deg F: Finish	Chilling Data: Time Hr	Chilling Data: Rate Factor	Est Prod Latent Heat Btu/lb 24 Hr (Ex. see Note f)	Specific Heat Btu/lb/Deg F: Before Freezing	Specific Heat Btu/lb/Deg F: After Freezing	Latent Heat of Fusion Btu/lb	Water Content %	Freezing Point Deg F	Maximum Air Motion in Room Ft./Min
Peaches	Short	35	35-40	85a	80-85	25.2						5.1	0.91	0.41	128	90	29.2	60
	Long	32	31-33	85b	80-85	22.3	2-4 Wk$_g$					0.3						60
	Chill Start	40		85		31.0		85	34	24	0.62	23.0f						150
	Chill Finish	32		85		22.3						0.3						60d
Pears	Short	35	35-40	90a	85-90	26.8					0.80	6.0	0.91	0.49	122	84	27-28	60
	Long	31k	29-31	90b	85-90	22.7	1-7 Mo$_g$					0.3						60
	Chill Start	40		85		31.0		70	34	24		17.0f						150
	Chill Finish	32		85		22.3						0.3						60d
Pineapples	Short	40	40-45	85	85-90	31.0						3.0	0.90	0.50	128	88		150
	Long Ripe	40	40-45	85b	85-90	31.0	2-4 Wk					0.1					29.9	150
	Green	50	50-60	90b	85-90	48.0	3-4 Wk					0.1					29.1	150
	Chill Start	45		85		37.5		85	40	3	0.67	24.0f						250
	Chill Finish	38		85		28.8						0.1						150d
Plums and Prunes (Fresh)	Short	35	35-40	85	80-85	25.2						4.0	0.88	0.48	116	80	28.0	90
	Long	32	31-32	85b	80-85	22.3	3-8 Wk$_g$					0.3						90
	Chill Start	40		80		29.1		80	34	20	0.67	20.0f						280
	Chill Finish	32		80		21.1						0.3						90d
Quinces	Short	35	35-40	85	80-85	25.2						4.0	0.90	0.49	122	85	28.0	60
	Long	32	31-32	85b	80-85	22.3	2-3 Mo					0.3						60
	Chill Start	40		85		31.0		80	32	24	0.67	24.0f						180
	Chill Finish	32		85		22.3						0.3						60d

From *Carrier Design Data.* Reproduced by permission of Carrier Corporation.

TABLE 10-11. Design Data for Vegetable Storage

VEGETABLES	TYPE OF STORAGE	DESIGN ROOM CONDITIONS: Temperature		Relative Humidity		Grains per lb Air at Recommended Condition	Maximum Storage Period	CHILLING DATA: Product Temp. Deg F		Time Hr	Rate Factor	Est Prod Latent Heat Btu/lb 24 Hr (Ex. see Note f)	SPECIFIC HEAT Btu/lb/Deg F		Latent Heat of Fusion Btu/lb	Water Content %	Freezing Point Deg F	Maximum Air Motion in Room Ft./Min
		Recommended Deg F	Permissible Range Deg F	Recommended %	Permissible Range %			Start	Finish				Before Freezing	After Freezing				
Asparagus	Short	40	40-45	90	85-90	32.8						6.0	0.91	0.49	136	94.0	29.8	90
	Long	32	32-36	90a	85-90	23.7	30 Days					0.5						60
	Chill Start	40		85		31.0		60	34	24	0.90	13.0f						150
	Chill Finish	33		85		23.2						0.5						60d
Beans,	Short	40	40-45	90	85-90	32.8						3.0	0.87	0.47	119	83.0	29.7	90
Green	Long	33	32-40	90a	85-90	24.6	30 Days					0.7						60
	Chill Start	40		85		31.0		80	35	20	0.67	15.0f						150
	Chill Finish	33		85		23.2						0.7						60d
Beans	Short	40	40-45	90	85-90	32.8	15 Days Shelled					3.0	0.78	0.36	99	68.5	28.4	90
(Lima)	Long	33	32-40	90a	85-90	24.6	30 Days Unshelled					0.6						60
Beets,	Short	40	40-45	90	85-90	32.8						2.0	0.90	0.48	129	90.0	26.9	90
Tops Off	Long	32	32-36	95	95-98	25.0	1-3 Mo					0.3						60
Beets,	Short	40	40-45	90	85-90	32.8						3.0	0.90	0.48	129	90.0	31.0	90
Tops On	Long	32	32-36	90a	85-90	23.7	10-14 Days					0.4						60
	Chill Start	40		90		32.8		70	34	24	0.80	17.0f						150
	Chill Finish	32		90		23.7						0.4						60d
Broccoli	Short	40	40-45	90	90-95	32.8						4.0	0.90	0.48	136	93.0	29.2	90
	Long	32	32-35	98	90-95	23.7	7-10 Days					0.5						60
	Chill Start	40		90		32.8						14.0f						150
	Chill Finish	33		90		24.6		80	34	24	0.80	0.5						90d
Brussel	Short	40	40-45	95	90-95	34.5						5.0	0.91	0.49	136	94.5	31.0	90
Sprouts	Long	33	32-35	95b	90-95	26.0	3-4 Wk					0.5						60
	Chill Start	40		90		32.8		80	34	24	0.80	14.0f						150
	Chill Finish	33		90		24.6						0.5						90d
Cabbage	Short	36	36-40	95	90-95	28.2						7.0	0.93	0.47	132	91.5	31.2	90
	Long	32	32-36	95b	90-95	25.0	3-4 Mo					0.5						60
	Chill Start	40		90		32.8		70	34	24	0.80	17.0f						150
	Chill Finish	32		90		23.7						0.5						60d
Carrots,	Short	40	40-45	90	85-90	32.8						2.0	0.93	0.45	126	88.0	30.4	90
Tops Off	Long	32	32-36	95	95-98	26.0	4-5 Mo					0.3						60
Carrots,	Short	40	40-45	90	85-90	32.8						4.0	0.86	0.45	126	88.0	31.0	60
Tops On	Long	32	32-36	90b	85-90	23.7	10-14 Days					0.5						60
	Chill Start	40		90		32.8		70	34	24	0.80	17.0f						150
	Chill Finish	32		90		23.7						0.5						60d
Cauliflower	Short	36	35-40	90	85-90	26.8						4.0	0.90	0.46	133	92.5	30.1	90
	Long	32	32-36	90a	85-90	23.7	2-3 Wk					0.3						60
	Chill Start	40		90		32.8		70	34	24	0.80	17.0f						150
	Chill Finish	32		90		23.7						0.3						60d
Celery p	Short	36	35-40	90	90-95	26.8						4.0	0.91	0.46	136	94.5	29.7	90
	Long (Wetted)	32	31-32	90a	90-95	23.7	2-4 Mo					1.0						60
Corn	Short	36	35-40	90	85-90	26.8						7.0	0.86	0.38	108	75.5	28.9	90
(Green)	Long	32	31-32	90a	85-90	23.7	4-8 Days					0.5						60
	Chill Start	40		85		31.0		70	34	24	0.80	17.0f						150
	Chill Finish	32		85		22.3						0.5						60d
Cucumbers	Short	50	50-60	85	80-85	45.2						3.0	0.93	0.48	137	95.5	30.5	90
	Long	45	45-50	85	80-85	37.5	10-14 Days					0.2						90
	Chill Start	60		80		61.7		70	52	24	1.0	13.0f						250
	Chill Finish	50		80		42.6						0.2						150d
Endive p	Short	36	35-40	90	90-95	26.8						4.0	0.90	0.46	136	89.0	30.9	90
	Long (Iced)	36	32-36	90a	90-95	26.8	2-3 Wk					1.0						90

TABLE 10-11 (*Continued*)

VEGETABLES	TYPE OF STORAGE	DESIGN ROOM CONDITIONS: Temperature Recommended Deg F	Temperature Permissible Range Deg F	Relative Humidity Recommended %	Relative Humidity Permissible Range %	Grains per lb Air at Recommended Condition	Maximum Storage Period	CHILLING DATA: Product Temp. Deg F Start	Product Temp. Deg F Finish	Time Hr	Rate Factor	Est Prod Latent Heat Btu/lb 24 Hr (Ex. see Note f)	SPECIFIC HEAT Btu/lb/Deg F Before Freezing	SPECIFIC HEAT After Freezing	Latent Heat of Fusion Btu/lb	Water Content %	Freezing Point Deg F	Maximum Air Motion in Room Ft./Min
Lettuce p	Short	35	35-40	90	90-95	26.8						7.0	0.90	0.46	136	89.0	31.2	90
	Long (Iced)	36	32-36	90a	90-95	26.8	2-3 Wk					1.0						60
Melons	Short	45	45-50	85	75-85	37.5						3.0	0.91	0.46	115	85.0	29.0	90
Watermelons Honeydews	Long	36	36-40	85a	75-85	26.2	2-4 Wk					0.2						150
Cantaloupes	Long	32	32-35	85	75-78	22.3	7-10 Days					0.2	0.91	0.47	128	89.0	29.0	90
	Chill Start	40		85		31.0		80	34	24	0.90	14.0f						250
	Chill Finish	32		85		22.3						.2						150d
Onions	Short	50	50-60	75	70-75	40.0						2.0	0.91	0.51	130	89.0	30.1	150
	Long	32	32-36	75	70-75	19.8	6-8 Mo					0.2						150
	Chill Start	40		75		27.5		70	34	24	0.80	10.0f						250
	Chill Finish	32		75		19.8						0.2						150d
Parsnips	Short	36	35-40	95	90-95	28.2						4.0	0.84	0.44	119	83.0	28.9	60
	Long	32	32-36	95b	90-95	25.0	2-4 Mo					0.5						60
	Chill Start	40		90		32.8		70	34	24	0.80	17.0f						150
	Chill Finish	32		90		23.7						0.5						90d
Peas (Green)	Short	36	36-40	90	85-90	26.8						3.0	0.82	0.45	107	80.0	28.9	90
	Long	32	32-34	90b	85-90	23.7	1-2 Wk					0.5						90
	Chill Start	40		85		31.0		80	34	20	0.67	14.0f						180
	Chill Finish	33		85		23.2						0.5						90d
Potatoes (Eating)		50r	50-70	85	85-90	46.2						3.0	0.84	0.47	113	78.5	28.9	150
Potatoes (Seed Stock)		36m	36-50	85a	85-90	26.4						0.5					28.9	150
Sauerkraut (In Kegs)	Short	45	45-50	80	75-80	36.3						3.0	0.92	0.52	128	89.0	26.0	150
	Long	30	30-32	80c	75-80	19.2	5 Mo					0.2						90
Spinach	Short	35	35-40	95a	90-95	28.2						7.0	0.92	0.51	129	90.0	30.3	90
	Long	32	32-36	95a	90-95	25.0	10-14 Days					0.5						60
Sweet Potato n	Short	55	55-60	85	80-85	54.5						3.0	0.84	0.42	102	78.0	28.5	150
	Long	55	55-60	85a	80-85	54.5	4-6 Mo					0.4						180
Tomatoes (Green)	Short	55	55-60	85	85-90	54.5						3.0	0.92	0.46	132	95.0	30.6	90
	Long	55	55-60	85	85-90	54.5	3-5 Wk					0.4						60
	Ripening	65	65-70	85	85-90	78.2						2.0						90
	Chill Start	70		85		93.3		80	52	34	1.0	14.0f						150
	Chill Finish	50		85		45.2						0.4						90d
(Ripe)	Long	45	40-50	85a	85-90	37.5	7-10 Days					3.0						90
Turnips	Short	36	35-40	95	95-98	28.2						4.0	0.90	0.45	128	89.5	30.5	90
	Long	32	32-36	95a	95-98	25.0	4-6 Mo					0.5						60
	Chill Start	40		95		34.5		70	34	24	0.80	17.0f						150
	Chill Finish	32		95		26.0						0.5						60d
Vegetables (Wetted, Mixed)	Short	40	40-45	86b	86-90	31.0						5.0	0.90	0.45	130	90.0	30.8	90
	Long	36	36-40	87b	86-90	26.0	2-4 Mo					1.2						90
	Chill Start	50		90		48.0		80	38	18	0.70	23.0f						180
	Chill Finish	36		90		26.8						1.2						90d

From *Carrier Design Data*. Reproduced by permission of Carrier Corporation.

TABLE 10-12. Design Data for Meat Storage

MEATS	TYPE OF STORAGE	Design Room Conditions: Temperature: Recommended Deg F	Design Room Conditions: Temperature: Permissible Range Deg F	Design Room Conditions: Relative Humidity: Recommended %	Design Room Conditions: Relative Humidity: Permissible Range %	Design Room Conditions: Grains per lb Air at Recommended Condition	Maximum Storage Period	Chilling Data: Product Temp. Deg F: Start	Chilling Data: Product Temp. Deg F: Finish	Chilling Data: Time Hr	Chilling Data: Rate Factor	Est Prod Latent Heat Btu/lb 24 Hr (Ex. see Note f)	Specific Heat Btu/lb/Deg F: Before Freezing	Specific Heat Btu/lb/Deg F: After Freezing	Latent Heat of Fusion Btu/lb	Water Content %	Freezing Point Deg F	Maximum Air Motion in Room Ft./Min
Bacon	Short	55	50-60	65	55-65	41.7	15 Days					2.5	0.50	0.30	9	20		150
	Hardening	28†	28-30	75	70-80	16.4						1.2						90
	Slicing Room	50	50-55	40	35-40	21.3												60
Beef-Combined Chill and Holding	Chill Start	38		85b		28.8		100	44	24	0.56	18.0f	0.75	[illegible]	[illegible]	72	31.3	250
	Chill Finish	33		85b		23.2						5.0						90d
Beef- Dried	Long	55	55-60	65	65-70	41.7	6 Mo					0.1	.22-.34	.19-.26	7-22	5-15		150
Beef- Fresh	Short	35	35-40	87b	85-90	26.0						5.0	0.75	0.40	98	72	31.3	60
	Long	30	30-32	87b	85-90	20.8	3 Wk					1.7						60
	Chill Start	45		87		38.3		100	44	18	0.67	22.0f						250
	Chill Finish	30		87		20.8						1.7						150d
Brined Meat	Short	40	40-45	85	80-85	31.0						1.0	0.75					150
	Long	31	31-32	85	80-85	21.3	6 Mo					0.8						150
Cut Meat	Short	34	34-38	87a	85-90	24.8	5 Days					5.6	0.72	0.40	95	65	29	60
Fish Frozen	Long	0	(−5)–0	85c	80-85	4.65	6 Mo					0.1	0.76	0.41	101	70	28	250
Iced	Short	34	34-38	85c	80-85	24.3						5.7						90
	Long	30	30-32	85a	80-85	20.4	15 Days					0.4						90
Hams, & Loins Fresh	Short	34	34-38	85	85-87	24.3						3.4	0.68	0.38	86.5	52	31.3	60
	Long	28	28-30	85b	85-87	18.5	3 Wk					1.8						60
	Short	55	50-60	65	55-65	41.7						1.3	0.60	0.32		57		150
Smoked	Chill Start g	60		70		53.9		105	57	8	1.00	5.0f						150
	Chill Finish g	55		70		44.8						.3						90d
Hog 18 Hrs Chilling	Chill Start	45		85		37.5		105	35	18	0.67	24.0f	0.68	0.38	86.5	60	27	250
	Chill Finish	30		85		20.4						1.9						150d
14 Hrs	Chill Start	38		90		30.1		106	35	14	0.67	23.0f						250
	Chill Finish	28		90		19.7						1.9						150d
Lamb	Short	34	34-38	90	85-90	25.8						3.4	0.67	0.30	83.5	58	29	60
	Long	28	28-30	90b	85.90	19.7	2 Wk					1.3						60
	Chill Start	45		90		39.6		100	40	5	0.75	19.0f						250
	Chill Finish	30		90		21.6						1.3						90d
Offal (Livers, Hearts, etc.)	Chill Start	40		85		31.0		90	35	18	0.70	21.0f	0.75	0.42	103	72		150
	Chill Finish	32		85		22.3						1.3						90d
Oysters Shell	Short	35	35-40	90c	85-90	26.8						4.2	0.83	0.44	116	80.4	27	90
	Long	32	32-38	90c	85-90	23.7	15 Days					0.5						90
Tub	Short	35	35-40	70	70-75	20.8						2.3	0.90	0.46	125	87	27	150
	Long	32	32-38	70	70-75	18.6	10 Days					0.2						150
Pork (Fresh)	Short	34	34-38	85	85-90	24.3	15 Days					3.4	0.68	0.38	86.5	60	28	90
Poultry Fresh	Long	28	28-30	87b	85-90	19.0	10 Days					0.4	0.79	0.37	106	74	27	60
Frozen	Long	0	(−5)–0	85	85-90	4.65	10 Mo					0.2						150
Wet Picked	Chill Start g	45		85		37.5		85	40	5	1.00	17.0f						150
	Chill Finish	32		85		22.3						0.4						90d

TABLE 10-12 (*Continued*)

MEATS	TYPE OF STORAGE	DESIGN ROOM CONDITIONS: Temperature: Recommended Deg F	Temperature: Permissible Range Deg F	Relative Humidity: Recommended %	Relative Humidity: Permissible Range %	Grains per lb Air at Recommended Condition	Maximum Storage Period	CHILLING DATA: Product Temp. Deg F: Start	Product Temp. Deg F: Finish	Time Hr	Rate Factor	Est Prod Latent Heat Btu/lb 24 Hr (Ex. see Note f)	SPECIFIC HEAT Btu/lb/Deg F: Before Freezing	SPECIFIC HEAT Btu/lb/Deg F: After Freezing	Latent Heat of Fusion Btu/lb	Water Content %	Freezing Point Deg F	Maximum Air Motion in Room Ft./Min
Sausage Casings (Salted)	Short	40	40-45	80c	75-80	29.1						0.2	0.60					150
	Long	31	31-32	80c	75-80	20.1	4 Mo					0.0						150
Franks and Smoked	Short	35	35-40	85a	80-90	25.2	48 Hr					4.3	0.86	0.56	86	60	29	60
	Chill Start	42		80		31.6		70	35	2	1.00	9.0f						150
	Chill Finish	32		80		21.1												60d
Fresh	Short	35	35-40	85a	85-90	25.2	7 Days					4.3	0.89	0.56	93	65	26	60
	Chill Start	42		85		33.6		70	35	2	1.00	9.0f						150
	Chill Finish	32		85		22.3												60d
Mfg. Room		55	55-60	40	35-40	25.5						0.0						60
Smoked Summer	Short	40	35 40	85	80-90	31.0	6 Mo					3.2	0.86	0.56	86	60	25	60
	Drying	50	48-56	70	65-80	37.2						5.0						60
	Long	32	32 34	70	70-75	18.6	6-8 Mo					2.0						60
Wrapping Room		45	45-50	85	80-85	37.5						0.0						60
Veal	Short	34	34-38	87b	85-90	24.8						3.6	0.71	0.39	91	63	29	60
	Long	28	28-30	87b	85-90	19.0	15 Days					1.3						60
	Chill Start	45		90		39.6		100	40	6	0.75	21.0f						90
	Chill Finish	30		90		21.6						1.3						60d

From *Carrier Design Data.* Reproduced by permission of Carrier Corporation.

TABLE 10-13. Design Data for Miscellaneous Storage

MISCELLANEOUS	TYPE OF STORAGE	DESIGN ROOM CONDITIONS: Temperature: Recommended Deg F	Temperature: Permissible Range Deg F	Relative Humidity: Recommended %	Relative Humidity: Permissible Range %	Grains, per lb Air at Recommended Condition	CHILLING DATA: Maximum Storage Period	Product Temp. Deg F: Start	Product Temp. Deg F: Finish	Time Hr	Rate Factor	Est Prod Latent Heat Btu/lb 24 Hr (Ex. see Note f)	SPECIFIC HEAT Btu/lb/Deg F: Before Freezing	SPECIFIC HEAT Btu/lb/Deg F: After Freezing	Latent Heat of Fusion Btu/lb	Water Content %	Freezing Point Deg F	Maximum Air Motion in Room Ft./Min
Beer (Wholesaler)																		
Wooden Keg	Short	35	35-40	85	80-85	25.2	6 Mo					7.0u	1.0			72.0	28	150
Metal Keg	Short	35	35-40	70c	65-70	20.8	6 Mo					0.4u						150
Butter or	Short	40	35-45	80c	75-80	29.1	10 Days					2.0	0.64	0.34	15	15.0	30.0w	150
Honey But'r	Long	0	(−5)−0	85	80-85	4.65	6 Mo					0.3						250
Candy	Long	65	60-70	55	50-55	50.3	6 Mo						0.93					60
Caviar	Short	40	40-45	85	80-85	31.0						2.0					20	150
(In Tubs)	Long	34	34-36	85b	80-85	24.3	15 Day					0.3						150
Cheese	Short	40	40-45	80b	75-80	29.1						2.3	0.64	0.36	79	55.0	17	90
American	Long	32	30-34	80b	75-80	21.1	15 Mo					0.5						90
Camembert	Short	40	40-45	85	80-85	31.0						2.5	0.70	0.40	86	60.0	18	90
	Long	40	34-34	85b	80-85	31.0	90 Days					0.2						90
Limburger	Short	40	40-45	85	80-85	31.0						2.5	0.70	0.40	86	60.0	19	90
	Long	31	30-34	85b	80-85	21.3	60 Days					0.3						90
Roquefort	Short	45	45-50	80	75-80	35.3						2.0	0.65	0.32	79	55.0	3	90
	Long	40	30-34	80b	75-80	29.1	60 Days					0.2						90
Swiss	Short	40	40-45	80	75-80	29.1						2.3	0.64	0.36	79	55.0	15	90
	Long	38	30-34	80b	75-80	27.8	60 Days					0.2						90
Chocolate (For Coating)	Long	60	60-70	55	50-55	42.1	6 Mo					0.1	0.56	0.30	40	0.5	85-95	60
Cream	Short	35	35-40	80c	75-80	23.8						2.0u	0.85	0.40	90	55.0	28	150d
(40%)	Long	5	(−5)−0	80	80-85	5.60	4 Mo					0.1u						150
Eggs Crated	Short	40	40-45	85b	80-85	31.0						3.6	0.85	0.45	100	74.2	31.6	90
(See Doc.	Long	30	30-31	85b	85-87	20.4	12 Mo					0.2						60
2D-85)	Chill Start	40		85b		31.0		45	30	10	0.85	7.0f						90
	Chill Finish	30		85b		20.4						0.2						60d
Eggs, Frozen	Long	5	(−5)−0	60		4.26	18 Mo					0.06u		0.45	100			250
10 lb cans	Chill Start	0		85c		4.65		40	5	24	0.67	9.0f						250
Doc. 2D-85	Chill Finish	0		85c		4.65						0.06						250
Fur, Woolens	Fumigated	35	35-40	65	60-65	19.3	6 Mo					0.1	0.40					150
(See Doc. 2D-83)	Ref only	15	15-18	70	65-70	8.2	6 Mo					0.1						150
Flour	Long	78	78-82	60	60-65	86.0	6 Mo						0.38	0.28		13.5		60
Flowers, Cut General		40k	33-40	85	85-90	31.0	3-14 Days					120 Btu	0.92				27-31	60
Orchids Gardenias		45	45-50	85	85-90	37.5	1 Wk					Per Sq Ft Floor					28-31	60
Hides, Curing		55	50-55	85	80-85	54.5						0.2	0.40					150
Storage	Long	36	32-40	75	70-75	23.1	5 Yr					0.1	0.40					150
Ice Cream	Hardening																	
5 Gal Cans	Start	0		85c		4.65		22	−10	8	0.75	1.1f,u	0.77		37	60.0	28.5-0	250
(See Doc.	Finish	−20		85c		1.55						0.1u						250
2D-86)	Start	0		85c		4.65		26	−10	8	0.75	1.3f,u			62			250
	Finish	−20		85c		1.55						0.1u						250
Lard	Short	45	45-50	80c	75-80	35.3						2.0	0.60		90		70	150
	Long	32	32-34	80c	75-80	21.1	6 Mo					0.3						150
Maple	Short	45	45-50	70c	65-70	29.9						0.7	0.24	0.21	7	5		250
Sugar	Long	31	31-32	70c	65-70	17.7	5 Mo					0.1						250

TABLE 10-13 (*Continued*)

MISCELLANEOUS	TYPE OF STORAGE	Design Room Conditions: Temperature Recommended Deg F	Design Room Conditions: Temperature Permissible Range Deg F	Design Room Conditions: Relative Humidity Recommended %	Design Room Conditions: Relative Humidity Permissible Range %	Design Room Conditions: Grains per lb Air at Recommended Condition	Maximum Storage Period	Chilling Data: Product Temp. Deg F Start	Chilling Data: Product Temp. Deg F Finish	Chilling Data: Time Hr	Chilling Data: Rate Factor	Est Prod Latent Heat Btu/lb 24 Hr (Ex. see Note f)	Specific Heat Btu/lb/Deg F: Before Freezing	Specific Heat Btu/lb/Deg F: After Freezing	Latent Heat of Fusion Btu/lb	Water Content %	Freezing Point Deg F	Maximum Air Motion in Room Ft./Min
Maple Syrup	Short	45	45-50	70c	65-70	29.9						0.7	0.49	0.31	52	36.0		250
	Long	31	31-32	70c	65-70	17.7	5 Mo					0.1						250
Milk Bottled	Short	35	35-40	70c	65-75	20.8	5 Days					2.0	0.90	0.49	124	87.5	31	250
and Wet	Chill Start	40		80c		29.1		45	35	10	0.85	8.0f						250
Dec. 20-59	Chill Finish	34		80c		23.0						0.1						250
Nuts, in Shells	Short	40	40-45	70c	65-75	25.3						0.50	0.25	0.22	3-10	2-8		150
	Long	32	32-40	70c	65-75	18.6	8-12 Mo					0.08						150
Nuts, Shelled	Short	40	40-45	70	65-75	25.3						0.50	0.30	0.24	4-14	3-10		150
	Long	32	32-40	70	65-75	18.6	6-10 Mo					0.06						150
Oleo	Short	45	45-50	80c	75-80	36.3						2.0	0.48					150
	Long	34	34-36	80c	75-80	23.0	90 Days					0.3						150
Vaccine Serum	Long	43	40-45	70	65-70	28.5	4 Mo					0.0						150
Shrubs	Long	28	24-29	70	60-80	15.4	6-8 Mo						0.60	0.35		50.0		

From *Carrier Design Data*. Reproduced by permission of Carrier Corporation.

TABLE 10-14. Reaction Heat from Fruits and Vegetables

FRUITS

Commodity	Temperature Deg F	Btu per hr per lb
Apples	32	.018
	40	.030
	60	.120
Apricots	32	.023
	40	.036
	60	.170
Bananas		
Holding	54	.069
Ripening	68	.190
Chilling	70-56	.500§
Berries	36	.115
	60	.345
Cherries	32	.032
	60	.250
Cranberries	32	.014
	40	.019
	50	.036
Dates, Fresh	32	.014
	40	.019
	50	.036
Grapefruit	32	.0096
	40	.022
	60	.058
Grapes	32	.0075
	40	.014
	60	.050
Lemons	32	.012
	40	.017
	60	.062
Limes	32	.012
	40	.017
	60	.062
Oranges	32	.017
	40	.029
	60	.104
Peaches	32	.023
	40	.036
	60	.170
Pears	32	.016
	60	.230
Plums	32	.032
	60	.250
Quinces	32	.018
	40	.030
	60	.120
Strawberries	32	.068
	40	.120
	60	.360

VEGETABLES

Commodity	Temperature Deg F	Btu per hr per lb
Asparagus	32	.035
	40	.170
Beans, Lima	32	.170
	60	.820
Beans, String	32	.099
	40	.140
	60	.470
Beets	32	.055
	40	.085
	60	.150
Brussel Sprouts	32	.059
	40	.095
	60	.280
Cabbage	32	.059
	40	.095
	60	.280
Cauliflower	32	.059
	40	.095
	60	.280
Carrots	32	.045
	40	.073
	60	.170
Celery	32	.059
	40	.095
	60	.280
Corn, Sweet	32	.035
	40	.170
Cucumber	32	.028
	40	.041
	60	.175
Endive	40	.200
Lettuce	32	.240
	40	.330
	60	.960
Melons (Except Watermelons)	32	.028
	40	.041
	60	.175
Mushrooms	32	.130
	50	.460
Onions	32	.018
	50	.039
	70	.075
Parsnips	32	.045
	40	.073
	60	.170
Peas	32	.170
	60	.820
Peppers	32	.057
	60	.180
Potatoes	32	.014
	40	.030
	70	.060
Spinach	40	.200
Sweet Potatoes	40	.070
Tomatoes		
(Green)	60	.130
(Ripe)	40	.027
Turnips	32	.040
	40	.050

From *Carrier Design Data.* Reproduced by permission of Carrier Corporation.

TABLE 10-15. Heat Equivalent of Electric Motors

Btu/hp-hr

Motor hp	Connected Load in Refr. Space[1]	Motor Losses Outside Refr. Space[2]	Connected Load Outside Refr. Space[3]
$\frac{1}{8}$ to $\frac{1}{2}$	4250	2545	1700
$\frac{1}{2}$ to 3	3700	2545	1150
3 to 20	2950	2545	400

[1] For use when both useful output and motor losses are dissipated within refrigerated space; motors driving fans for forced circulation unit coolers.

[2] For use when motor losses are dissipated outside refrigerated space and useful work of motor is expended within refrigerated space; pump on a circulating brine or chilled water system, fan motor outside refrigerated space driving fan circulating air within refrigerated space.

[3] For use when motor heat losses are dissipated within refrigerated space and useful work expended outside of refrigerated space; motor in refrigerated space driving pump or fan located outside of space.

From *ASRE Data Book*, Design Volume, 1949 Edition, by permission of the American Society of Heating, Refrigerating, and Air-Conditioning Engineers.

TABLE 10-16. Heat Equivalent of Occupancy

Cooler Temperature, F	Heat Equivalent/Person Btu/hr
50	720
40	840
30	950
20	1050
10	1200
0	1300
−10	1400

From *ASRE Data Book*, Design Volume, 1949 Edition, by permission of the American Society of Heating, Refrigerating, and Air-Conditioning Engineers.

TABLE 10-17. Usage Heat Gain, Btu per 24 Hour for One Cubic Foot Interior Capacity

Volume Cubic Feet	Service	Temperature Reduction in ° F (Outside temperature minus storage temperature)								
		40°	45°	50°	55°	60°	65°	70°	75°	80°
15	Normal	108	122	135	149	162	176	189	203	216
	Heavy	134	151	168	184	201	218	235	251	268
50	Normal	97	109	121	133	145	157	169	182	194
	Heavy	124	140	155	171	186	202	217	233	248
100	Normal	85	96	107	117	128	138	149	160	170
	Heavy	114	128	143	157	171	185	200	214	228
200	Normal	74	83	93	102	111	120	130	139	148
	Heavy	104	117	130	143	156	169	182	195	208
300	Normal	68	77	85	94	102	111	119	128	136
	Heavy	98	110	123	135	147	159	172	184	196
400	Normal	65	73	81	89	97	105	113	122	130
	Heavy	95	107	119	130	142	154	166	178	190
600	Normal	61	68	76	84	91	99	106	114	122
	Heavy	91	103	114	125	137	148	160	171	182
800	Normal	59	67	74	81	89	96	104	111	118
	Heavy	89	100	112	123	134	145	156	167	178
1000	Normal	57	64	72	79	86	93	100	107	114
	Heavy	86	97	108	119	130	140	151	162	173
1200	Normal	55	62	69	76	83	90	97	104	110
	Heavy	84	95	105	116	126	137	147	158	168
1600	Normal	51	58	64	70	77	83	90	96	102
	Heavy	79	89	99	108	118	128	138	148	158

From *ASRE Data Book*, Design Volume, 1949 Edition, by permission of the American Society of Heating, Refrigerating, and Air-Conditioning Engineers.

TABLE 10-18. Wall Heat Gain

(Btu per sq ft per 24 hr)

Insulation Cork or Equivalent in.	Temp. Difference (Ambient Temp. Minus Refrigerator Temp)., F																	
	1	40	45	50	55	60	65	70	75	80	85	90	95	100	105	110	115	120
3	2.4	96	108	120	132	144	156	168	180	192	204	216	228	240	252	264	267	288
4	1.8	72	81	90	99	108	117	126	135	144	153	162	171	180	189	198	207	216
5	1.44	58	65	72	79	87	94	101	108	115	122	130	137	144	151	159	166	173
6	1.2	48	54	60	66	72	78	84	90	96	102	108	114	120	126	132	138	144
7	1.03	41	46	52	57	62	67	72	77	82	88	93	98	103	108	113	118	124
8	0.90	36	41	45	50	54	59	63	68	72	77	81	86	90	95	99	104	108
9	0.80	32	36	40	44	48	52	56	60	64	68	72	76	80	84	88	92	96
10	0.72	29	3[illegible]	36	40	43	47	50	54	58	61	65	68	72	76	79	83	86
11	0.66	26	30	33	36	40	43	46	50	53	56	60	63	66	69	73	76	79
12	0.60	24	27	30	33	36	39	42	45	48	51	54	57	60	63	66	69	72
13	0.55	22	25	28	30	33	36	39	41	44	47	50	52	55	58	61	63	66
14	0.51	20	23	26	28	31	33	36	38	41	43	46	49	51	54	56	59	61
Single glass	27.0	1080	1220	1350	1490	1620	1760	1890	2030	2160	2290	2440	2560	2700	2840	2970	3100	3240
Double glass	11.0	440	500	550	610	660	715	770	825	880	936	990	1050	1100	1160	1210	1270	1320
Triple glass	7.0	280	320	350	390	420	454	490	525	560	595	630	665	700	740	770	810	840

NOTE: Where wood studs are used multiply the above values by 1.1.
From *ASRE Data Book*, Design Volume, 1955–56 Edition, by permission of the American Society of Heating, Refrigerating, and Air-Conditioning Engineers.

TABLE 11-1. Mean Effective Temperature Differences

	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20
1	1.00	1.44	1.82	2.16	2.48	2.79	3.08	3.37	3.64	3.91	4.17	4.43	4.68	4.93	5.17	5.41	5.65	5.88	6.11	6.34
2	1.44	2.00	2.47	2.89	3.28	3.64	3.99	4.33	4.65	4.97	5.28	5.58	5.88	6.17	6.45	6.73	7.01	7.28	7.55	7.82
3	1.82	2.47	3.00	3.51	3.95	4.33	4.73	5.11	5.40	5.82	6.17	6.49	6.82	7.15	7.46	7.77	8.08	8.37	8.67	8.97
4	2.16	2.89	3.51	4.00	4.48	4.93	5.36	5.77	6.17	6.55	6.92	7.28	7.64	8.00	8.32	8.66	8.98	9.31	9.63	9.94
5	2.48	3.28	3.95	4.48	5.00	5.49	5.94	6.38	6.81	7.21	7.61	8.00	8.37	8.74	9.10	9.46	9.81	10.15	10.49	10.82
6	2.79	3.64	4.33	4.93	5.49	6.00	6.37	7.01	7.40	7.85	8.27	8.70	9.08	9.47	9.98	10.22	10.61	10.96	11.30	11.67
7	3.08	3.99	4.73	5.36	5.94	6.37	7.00	7.63	7.86	8.39	8.87	9.32	9.67	10.10	10.52	10.86	11.26	11.65	12.04	12.37
8	3.37	4.33	5.10	5.77	6.38	7.01	7.63	8.00	8.49	8.96	9.42	9.86	10.30	10.72	11.13	11.54	11.94	12.33	12.72	13.10
9	3.64	4.65	5.40	6.17	6.81	7.40	7.86	8.49	9.00	9.58	10.06	10.52	10.97	11.24	11.70	12.14	12.57	12.99	13.39	13.92
10	3.91	4.97	5.82	6.55	7.21	7.85	8.39	8.96	9.58	10.00	10.49	10.97	11.43	11.89	12.33	12.77	13.19	13.61	14.02	14.43
11	4.17	5.28	6.17	6.92	7.61	8.27	8.87	9.42	10.06	10.49	11.00	11.49	11.96	12.42	12.94	13.33	13.79	14.22	14.65	15.06
12	4.43	5.58	6.49	7.28	8.00	8.70	9.32	9.86	10.52	10.97	11.49	12.00	12.50	12.99	13.45	13.90	14.45	14.80	15.23	15.66
13	4.68	5.88	6.82	7.64	8.37	9.08	9.67	10.30	10.97	11.43	11.96	12.50	13.00	13.48	13.91	14.44	14.90	15.35	15.80	16.26
14	4.93	6.17	7.15	8.00	8.74	9.47	10.10	10.72	11.24	11.89	12.42	12.99	13.48	14.00	14.58	14.93	15.46	15.90	16.38	16.81
15	5.17	6.45	7.46	8.32	9.10	9.98	10.52	11.13	11.70	12.33	12.94	13.45	13.91	14.58	15.00	15.87	16.00	16.46	16.90	17.39
16	5.41	6.73	7.77	8.66	9.46	10.22	10.86	11.54	12.14	12.77	13.33	13.90	14.44	14.93	15.87	16.00	16.29	16.98	17.31	17.93
17	5.65	7.01	8.08	8.98	9.81	10.61	11.26	11.94	12.57	13.19	13.79	14.45	14.90	15.46	16.00	16.29	17.00	17.51	18.07	18.51
18	5.88	7.28	8.37	9.31	10.15	10.96	11.65	12.33	12.99	13.61	14.22	14.80	15.35	15.90	16.46	16.98	17.51	18.00	18.35	18.99
19	6.11	7.55	8.67	9.63	10.49	11.30	12.04	12.72	13.39	14.02	14.65	15.23	15.80	16.38	16.90	17.31	18.07	18.35	19.00	19.23
20	6.34	7.82	8.95	9.94	10.82	11.67	12.37	13.10	13.92	14.43	15.06	15.66	16.26	16.81	17.39	17.93	18.51	18.99	19.23	20.00
21	6.57	8.08	9.25	10.25	11.15	12.00	12.74	13.47	14.19	14.83	15.47	16.08	16.69	17.26	17.83	18.35	18.96	19.43	20.24	20.49
22	6.79	8.34	9.54	10.56	11.47	12.35	13.11	13.84	14.57	15.22	15.87	16.50	17.11	17.71	18.28	18.84	19.40	19.96	20.45	20.99
23	7.02	8.60	9.82	10.86	11.79	12.68	13.44	14.20	14.89	15.61	16.27	16.92	17.53	18.12	18.72	19.27	19.90	20.38	20.90	21.46
24	7.24	8.85	10.01	11.16	12.11	13.02	13.79	14.56	15.27	15.99	16.64	17.31	17.95	18.55	19.15	19.73	20.33	20.86	21.48	21.94
25	7.46	9.11	10.38	11.46	12.43	13.34	14.14	14.92	15.65	16.37	17.05	17.74	18.35	18.95	19.58	20.14	20.76	21.30	21.86	22.41
26	7.67	9.36	10.65	11.75	12.74	13.67	14.46	15.26	16.02	16.75	17.43	18.11	18.76	19.38	20.01	20.60	21.20	21.77	22.34	22.87
27	7.89	9.61	10.92	12.05	13.05	13.99	14.81	15.62	16.38	17.11	17.82	18.50	19.20	19.79	20.42	21.01	21.63	22.19	22.76	23.33
28	8.10	9.85	11.19	12.33	13.35	14.31	15.15	15.96	16.75	17.48	18.20	18.89	19.55	20.20	20.83	21.44	22.04	22.62	23.20	23.77
29	8.32	10.01	11.46	12.62	13.65	14.63	15.49	16.31	17.10	17.85	18.57	19.27	19.94	20.60	21.24	21.85	22.49	23.07	23.66	24.22
30	8.53	10.34	11.73	12.90	13.95	14.94	15.79	16.64	17.46	18.20	18.94	19.64	20.33	20.99	21.64	22.27	22.90	23.48	24.08	24.66
31	8.74	10.58	11.98	13.19	14.25	15.25	16.12	16.98	17.81	18.56	19.31	20.02	20.71	21.27	22.09	22.67	23.31	23.92	24.50	25.10
32	8.94	10.82	12.26	13.47	14.55	15.57	16.45	17.31	18.11	18.91	19.66	20.39	21.09	21.77	22.45	23.08	23.72	24.33	24.94	25.53
33	9.15	11.06	12.51	13.74	14.84	15.87	16.75	17.64	18.46	19.26	20.03	20.76	21.47	22.18	22.83	23.47	24.13	24.75	25.35	25.96
34	9.36	11.29	12.76	14.02	15.13	16.17	17.08	17.97	18.80	19.61	20.37	21.12	21.85	22.53	23.22	23.88	24.53	25.15	25.79	26.30
35	9.56	11.53	13.03	14.29	15.47	16.48	17.40	18.29	19.14	19.96	20.72	21.48	22.22	22.92	23.60	24.27	24.94	25.58	26.19	26.95
36	9.77	11.76	13.28	14.56	15.70	16.77	17.71	18.62	19.48	20.30	21.08	21.85	22.58	23.30	23.99	24.66	25.33	25.97	26.62	27.28
37	9.97	12.00	13.53	14.83	15.99	17.07	18.01	18.94	19.81	20.64	21.43	22.20	22.95	23.66	24.37	25.04	25.72	26.36	27.01	27.64
38	10.17	12.23	13.78	15.10	16.27	17.36	18.32	19.25	20.14	20.97	21.78	22.55	23.30	24.05	24.73	25.43	26.11	26.77	27.41	28.06
39	10.37	12.45	14.04	15.37	16.55	17.67	18.63	19.57	20.47	21.31	22.13	22.91	23.67	24.41	25.12	25.81	26.50	27.16	27.80	28.48
40	10.57	12.68	14.29	15.63	16.83	17.95	18.92	19.88	20.80	21.64	22.46	23.26	24.02	24.77	25.49	26.19	26.89	27.56	28.21	28.23

Courtesy Acme Industries, Inc.

TABLE 11-2. Evaporator Design TD

Relative Humidity, %	Design TD, ° F Natural Convection	Forced Convection
95–91	12–14	8–10
90–86	14–16	10–12
85–81	16–18	12–14
80–76	18–20	14–16
75–70	20–22	16–18

For temperatures 10° F and below, an evaporator TD of 10° F is generally used for forced convection evaporators.

TABLE 11-3. Properties of Pure Calcium Chloride Brine

Pure $CaCl_2$ % by wt	Specific gravity 60 F / 60 F	Baumé density 60 F	Specific heat 60 F Btu per lb F	Crystallization starts F	Weight per gallon $CaCl_2$ lb/gal	Water lb/gal	Brine lb/gal	Weight per cubic foot $CaCl_2$ lb/cu ft	Water lb/cu ft	Brine lb/cu ft
0	1.000	0.0	1.000	32.0	0.000	8.34	8.34	0.00	62.40	62.40
5	1.044	6.1	0.924	27.7	0.436	8.281	8.717	3.26	61.89	65.15
6	1.050	7.0	0.914	26.8	.526	8.234	8.760	3.93	61.59	65.52
7	1.060	8.2	0.898	25.9	.620	8.231	8.851	4.63	61.51	66.14
8	1.069	9.3	0.884	24.6	.714	8.212	8.926	5.34	61.36	66.70
9	1.078	10.4	0.869	23.5	.810	8.191	9.001	6.05	61.22	67.27
10	1.087	11.6	0.855	22.3	0.908	8.168	9.076	6.78	61.05	67.83
11	1.096	12.6	0.842	20.8	1.006	8.137	9.143	7.52	60.81	68.33
12	1.105	13.8	0.828	19.3	1.107	8.120	9.227	8.27	60.68	68.95
13	1.114	14.8	0.816	17.6	1.209	8.093	9.302	9.04	60.47	69.51
14	1.124	15.9	0.804	15.5	1.313	8.064	9.377	9.81	60.27	70.08
15	1.133	16.9	0.793	13.5	1.418	8.034	9.452	10.60	60.04	70.64
16	1.143	18.0	0.779	11.2	1.526	8.010	9.536	11.40	59.86	71.26
17	1.152	19.1	0.767	8.6	1.635	7.984	9.619	12.22	59.67	71.89
18	1.162	20.2	0.756	5.9	1.747	7.956	9.703	13.05	59.46	72.51
19	1.172	21.3	0.746	2.8	1.859	7.927	9.786	13.90	59.23	73.13
20	1.182	22.1	0.737	− 0.4	1.970	7.883	9.853	14.73	58.90	73.63
21	1.192	23.0	0.729	− 3.9	2.085	7.843	9.928	15.58	58.61	74.19
22	1.202	24.4	0.716	− 7.8	2.208	7.829	10.037	16.50	58.50	75.00
23	1.212	25.5	0.707	−11.9	2.328	7.792	10.120	17.40	58.23	75.63
24	1.223	26.4	0.697	−16.2	2.451	7.761	10.212	18.32	58.00	76.32
25	1.233	27.4	0.689	−21.0	2.574	7.721	10.295	19.24	57.70	76.94
26	1.244	28.3	0.682	−25.8	2.699	7.680	10.379	20.17	57.39	77.56
27	1.254	29.3	0.673	−31.2	2.827	7.644	10.471	21.13	57.12	78.25
28	1.265	30.4	0.665	−37.8	2.958	7.605	10.563	22.10	56.84	78.94
29	1.276	31.4	0.658	−49.4	3.090	7.565	10.655	23.09	56.53	79.62
29.87	1.290	32.6	0.655	−67.0	3.16	7.59	10.75	23.65	56.80	80.45
30	1.295	33.0	0.653	−50.8	3.22	7.58	10.80	24.06	56.70	80.76
32	1.317	34.9	0.640	−19.5	3.49	7.49	10.98	26.10	56.04	82.14
34	1.340	36.8	0.630	+ 4.3	3.77	7.40	11.17	28.22	55.35	83.57

From *ASRE Data Book*, Design Volume, 1957–58 Edition, by permission of The American Society of Heating, Refrigerating, and Air-Conditioning Engineers.

TABLE 11-4. Properties of Pure Sodium Chloride Brine

Pure NaCl % by wt	Specific gravity 59 F / 39 F	Baumé density 60 F	Specific heat 59 F Btu/lb deg F	Crystallization starts F	Weight per gallon: NaCl lb/gal	Weight per gallon: Water lb/gal	Weight per gallon: Brine lb/gal	Weight per cubic foot: NaCl lb/cu ft	Weight per cubic foot: Water lb/cu ft	Weight per cubic foot: Brine lb/cu ft
0	1.000	0.0	1.000	32.0	0.000	8.34	8.34	0.000	62.40	62.4
5	1.035	5.1	0.938	27.0	0.432	8.22	8.65	3.230	61.37	64.6
6	1.043	6.1	0.927	25.5	0.523	8.19	8.71	3.906	61.19	65.1
7	1.050	7.0	0.917	24.0	0.613	8.15	8.76	4.585	60.91	65.5
8	1.057	8.0	0.907	23.2	0.706	8.11	8.82	5.280	60.72	66.0
9	1.065	9.0	0.897	21.8	0.800	8.09	8.89	5.985	60.51	66.5
10	1.072	10.1	0.888	20.4	0.895	8.05	8.95	6.690	60.21	66.9
11	1.080	10.8	0.879	18.5	0.992	8.03	9.02	7.414	59.99	67.4
12	1.087	11.8	0.870	17.2	1.090	7.99	9.08	8.136	59.66	67.8
13	1.095	12.7	0.862	15.5	1.188	7.95	9.14	8.879	59.42	68.3
14	1.103	13.6	0.854	13.9	1.291	7.93	9.22	9.632	59.17	68.8
15	1.111	14.5	0.847	12.0	1.392	7.89	9.28	10.395	58.90	69.3
16	1.118	15.4	0.840	10.2	1.493	7.84	9.33	11.168	58.63	69.8
17	1.126	16.3	0.833	8.2	1.598	7.80	9.40	11.951	58.36	70.3
18	1.134	17.2	0.826	6.1	1.705	7.76	9.47	12.744	58.06	70.8
19	1.142	18.1	0.819	4.0	1.813	7.73	9.54	13.547	57.75	71.3
20	1.150	19.0	0.813	+ 1.8	1.920	7.68	9.60	14.360	57.44	71.8
21	1.158	19.9	0.807	− 0.8	2.031	7.64	9.67	15.183	57.12	72.3
22	1.166	20.8	0.802	− 3.0	2.143	7.60	9.74	16.016	56.78	72.8
23	1.175	21.7	0.796	− 6.0	2.256	7.55	9.81	16.854	56.45	73.3
24	1.183	22.5	0.791	+ 3.8	2.371	7.51	9.88	17.712	56.09	73.8
25	1.191	23.4	0.786	+16.1	2.488	7.46	9.95	18.575	55.72	74.3
25.2	1.200			+32.0						

From *ASRE Data Book*, Design Volume, 1957–58 Edition, by permission of The American Society of Heating, Refrigerating, and Air-Conditioning Engineers.

TABLE 11-5. Freezing Points of Aqueous Solutions

Alcohol % by Wt	Alcohol Deg F	Glycerine % by Wt	Glycerine Deg F	Ethylene Glycol % by Vol	Ethylene Glycol Deg F	Propylene Glycol % by Vol	Propylene Glycol Deg F
5	28.0	10	29.1	15	22.4	5	29.0
10	23.6	20	23.4	20	16.2	10	26.0
15	19.7	30	14.9	25	10.0	15	22.5
20	13.2	40	4.3	30	3.5	20	19.0
25	5.5	50	−9.4	35	−4.0	25	14.5
30	−2.5	60	−30.5	40	−12.5	30	9.0
35	−13.2	70	−38.0	45	−22.0	35	2.5
40	−21.0	80	−5.5	50	−32.5	40	−5.5
45	−27.5	90	+29.1			45	−15.0
50	−34.0	100	+62.6			50	−25.5
55	−40.5					55	−39.5
						59	−57.0

Above 60% fails to crystallize at −99.4 F

From *ASRE Data Book*, 1957–58 Edition, by permission of the American Society of Heating, Refrigerating, and Air-Conditioning Engineers.

TABLE 12-1. Approximate Volumetric Efficiency of Refrigerant-12 Compressors at Various Compression Ratios

Compression Ratio	Volumetric Efficiency	Compression Ratio	Volumetric Efficiency
2	87.3	6.2	62.2
2.2	86.0	6.4	61.2
2.4	84.9	6.6	60.2
2.6	83.5	6.8	59.2
2.8	82	7	58.2
3	80.8	7.2	57.2
3.2	79.5	7.4	56.3
3.4	78.3	7.6	55.3
3.6	77.2	7.8	54.4
3.8	76.0	8	53.5
4	74.9	8.2	52.6
4.2	73.7	8.4	51.7
4.4	72.5	8.6	50.8
4.6	71.3	8.8	49.9
4.8	70.1	9	49
5	69.0	9.2	48.1
5.2	67.9	9.4	47.2
5.4	66.8	9.6	46.4
5.6	65.7	9.8	45.7
5.8	64.5	10	44.9
6	63.3		

TABLE 14-1. Scale Factors—Water

Temperature of Water	125° F or Less	
Types of Water	Water Velocity Ft/Sec	
	3 Ft and Less*	Over 3 Ft†
Sea water	0.0005	0.0005
Brackish water	0.002	0.001
Cooling tower and artificial spray pond: Treated make-up	0.001	0.001
Untreated	0.003	0.003
City or well water (Such as Great Lakes)	0.001	0.001
Great Lakes	0.001	0.001
River water: Minimum	0.002	0.001
Mississippi	0.003	0.002
Delaware, Schuylkill	0.003	0.002
East River and New York Bay	0.003	0.002
Chicago Sanitary Canal	0.008	0.006
Muddy or silty	0.003	0.002
Hard (Over 15 grains) gal	0.003	0.003
Engine jacket	0.001	0.001
Distilled	0.0005	0.0005

* 2.16 gpm per tube is equivalent to a water velocity of 3 ft per second.

† This table is presented by permission of the Tubular Exchanger Manufacturers Association, Inc., New York.

TABLE 14-2. Bleed-Off Rates

Cooling Range Deg. F	Percent Bleed-off
6	0.15
7½	0.22
10	0.33
15	0.54
20	0.75

Courtesy The Marley Company.

TABLE 15-1. Equivalent Length in Feet to be Added to Run Owing to Valves and Fittings

Type of Fitting	Nominal Pipe Sizes—Inches ½	¾	1	1¼	1½	2	2½	3	3½	4	4½	5	6	7	8	9	10	11	12
Gate valve—open	0.3	0.5	0.6	0.8	0.9	1.2	1.4	1.7	2.0	2.5	2.7	3.0	3.5	4.0	4.5	5.0	6.0	6.5	7.0
Globe valve—open	16	21	26	35	43	54	65	80	95	110	120	140	160	180	210	250	280	305	330
Angle valve—open	8	11	14	18	20	25	31	40	45	51	60	70	80	91	110	125	140	152	165
Standard 45 deg elbow	0.8	1.0	1.3	1.6	2.0	2.5	3.0	3.8	4.5	5.0	5.8	6	8	8.5	10	11	13	14	15
Standard 90 deg elbow	1.5	2.0	2.5	3.5	4.5	5.0	6.5	8.0	10	11	13	14	16	18	20	23	26	28	30
Medium sweep 90 deg elbow	1.4	1.8	2.3	3.0	3.5	4.5	5.2	6.8	8	9	10	11	14	15	17	19	21	23	25
Long sweep 90 deg elbow	1.0	1.5	2.0	2.5	3.0	3.5	4	5	6	7	8	9	10	12	14	16	18	19	20
Square elbow 90 deg	3.0	4.5	5.5	7.5	9	12	14	17	20	22	24	26	33	38	44	50	53	55	57
Close return bend	3.5	5	6	8	10	13	15	18	20	24	26	30	35	42	49	54	61	66	72
Stand tee—full-size branch*	3.0	4.5	5.5	7.5	9	12	14	17	20	22	24	26	33	38	44	50	53	55	57
Stand tee—through run	1.0	1.5	2.0	2.5	3.0	3.5	4	5	6	7	8	9	10	12	14	16	18	19	20
Sudden enlargement from d to D†																			
d/D = ¼	1.5	2.0	2.5	3.5	4.5	5.0	6.5	8.0	10	11	13	14	16	18	20	23	26	28	30
d/D = ½	1.0	1.3	1.6	2.2	2.6	3.3	3.8	4.9	5.6	6.4	7.0	8.1	10	11	13	15	16	17	18
d/D = ¾	0.3	0.5	0.6	0.8	0.9	1.2	1.4	1.7	2.0	2.5	2.7	3.0	3.5	4.0	4.5	5.0	6.0	6.5	7.0
Sudden contraction from D to d†																			
d/D = ¼	0.8	1.0	1.3	1.6	2.0	2.5	3.0	3.8	4.5	5.0	5.8	6	8	8.5	10	11	13	14	15
d/D = ½	0.6	0.8	1.0	1.3	1.5	1.8	2.3	2.8	3.4	3.6	4.3	4.8	5.6	6.4	7.5	8.5	9.5	11	12
d/D = ¾	0.3	0.5	0.6	0.8	0.9	1.3	1.4	1.7	2.0	2.5	2.7	3.0	3.5	4.0	4.5	5.0	6.0	6.5	7.0
Ordinary pipe entrance with upstream end of pipe flush with inside of tank	0.9	1.3	1.5	2.0	2.4	3.0	3.6	4.5	5.1	6.0	6.6	7.5	9.0	11	12	14	15	17	18
Entrance with pipe projecting into tank beyond inside face (borda entrance)	1.5	2.0	2.5	3.5	4.0	5.0	6.0	7.8	9.0	10	12	13	15	17	19	21	24	27	30

* Pressure drop through side outlet, or from side outlet through run.
† Equivalent feet of the smaller diameter pipe, "d."
Courtesy York Corporation.

TABLE 15-2. Pressure Drop Correction Factors*

Liquid	Freeze at ° F	Specific Gravity at (° F)		Friction Correction Factor Temperature—° F								
		−20	+20	−20	−10	0	10	20	30	40	50	60
Calcium brine												
Sp gr = 1.10	20.3	—	1.11	—	—	—	—	1.21	1.19	1.15	1.12	1.11
Sp gr = 1.20	−5.8	—	1.21	—	—	1.49	1.44	1.38	1.33	1.28	1.26	1.24
Sp gr = 1.25	−26.0	1.27	1.26	1.85	1.75	1.66	1.57	1.50	1.44	1.40	1.37	1.34
Sodium brine												
Sp gr = 1.10	14.9	—	1.11	—	—	—	1.27	1.21	1.19	1.15	1.12	1.11
Sp gr = 1.18	−6.0	—	1.19	—	1.58	1.50	1.44	1.39	1.33	1.28	1.25	1.22
Ammonia (liquid)	−107.8	0.68	0.65	0.65	0.65	0.65	0.65	0.65	0.65	0.65	0.65	0.65
Alcohol (ethyl) (100%)	−114.6	0.83	0.81	0.97	0.95	0.93	0.92	0.91	0.91	0.90	0.90	0.90
Alcohol (ethyl) (40%)	−22	0.93	0.91	1.45	1.39	1.33	1.29	1.23	1.19	1.15	1.12	1.10
Alcohol (methyl) (100%)	−97	0.84	0.82	0.85	0.85	0.85	0.84	0.84	0.84	0.83	0.83	0.83
Alcohol (methyl) (30%)	−5.8	—	0.96	1.32	1.26	1.22	1.19	1.16	1.12	1.09	1.07	1.05
Ethylene glycol (60%)	−59.0	1.10	1.09	1.87	1.83	1.78	1.72	1.62	1.57	1.46	1.40	1.36
Ethylene glycol (50%)	−38.0	1.09	1.08	1.83	1.74	1.64	1.54	1.48	1.42	1.37	1.31	1.26
Ethylene glycol (30%)	2.0	—	1.06	—	—	—	1.34	1.27	1.22	1.17	1.13	1.11
Refrigerant-11 (liquid)	−168	1.60	1.55	1.42	1.42	1.42	1.42	1.42	1.42	1.42	1.42	1.42
Refrigerant-12 (liquid)	−252	1.49	1.42	1.32	1.32	1.32	1.32	1.32	1.32	1.32	1.32	1.32
Methyl chloride	−144	1.02	0.97	1.13	1.13	1.13	1.12	1.12	1.12	1.11	1.11	1.11
Methylene chloride	−142.1	1.40	1.33	1.65	1.63	1.62	1.60	1.59	1.58	1.56	1.54	1.52

* To obtain pressure drop from flow of above liquids through pipes, multiply pressure drop for water flow (of equal quantity through same pipe) by factors from above table.

Courtesy York Corporation.

TABLE 16-1. ASRE Refrigerant Numbering System

ASRE Standard Refrigerant Designation	Chemical Name	Chemical Formula	Molecular Weight	Boiling Point, F	Status[1]
Halocarbon Compounds					
10	Carbontetrachloride	CCl_4	153.8	170.2	
11	Trichloromonofluoromethane	CCl_3F	137.4	74.8	C
12	Dichlorodifluoromethane	CCl_2F_2	120.9	−21.6	C
13	Monochlorotrifluoromethane	$CClF_3$	104.5	−114.6	C
13B1	Monobromotrifluoromethane	$CBrF_3$	148.9	−72.0	S
14	Carbontetrafluoride	CF_4	88.0	−198.4	S
20	Chloroform	$CHCl_3$	119.4	142	
21	Dichloromonofluoromethane	$CHCl_2F$	102.9	48.1	D
22	Monochlorodifluoromethane	$CHClF_2$	86.5	−41.4	C
23	Trifluoromethane	CHF_3	70.0	−119.9	D
30	Methylene chloride	CH_2Cl_2	84.9	105.2	C
31	Monochloromonofluoromethane	CH_2ClF	68.5	48.0	
32	Methylene fluoride	CH_2F_2	52.0	−61.4	
40	Methyl chloride	CH_3Cl	50.5	−10.8	C
41	Methyl fluoride	CH_3F	34.0	−109	
(50	Methane	CH_4	16.0	−259	C)[2]
110	Hexachloroethane	CCl_3CCl_3	236.8	365	
111	Pentachloromonofluoroethane	CCl_3CCl_2F	220.3	279	
112	Tetrachlorodifluoroethane	CCl_2FCCl_2F	203.8	199.0	
112a	Tetrachlorodifluoroethane	CCl_3CClF_2	203.8	195.8	
113	Trichlorotrifluoroethane	CCl_2FCClF_2	187.4	117.6	C
113a	Trichlorotrifluoroethane	CCl_3CF_3	187.4	114.2	
114	Dichlorotetrafluoroethane	$CClF_2CClF_2$	170.9	38.4	C
114a	Dichlorotetrafluoroethane	CCl_2FCF_3	170.9	38.5	C
114B2	Dibromotetrafluoroethane	$CBrF_2CBrF_2$	259.9	117.5	D
115	Monochloropentafluoroethane	$CClF_2CF_3$	154.5	−37.7	D
116	Hexafluoroethane	CF_3CF_3	138.0	−108.8	
120	Pentachloroethane	$CHCl_2CCl_3$	202.3	324	
123	Dichlorotrifluoroethane	$CHCl_2CF_3$	153	83.7	
124	Monochlorotetrafluoroethane	$CHClFCF_3$	136.5	10.4	
124a	Monochlorotetrafluoroethane	CHF_2CClF_2	136.5	14	D
125	Pentafluoroethane	CHF_2CF_3	120	−55	
133a	Monochlorotrifluoroethane	CH_2ClCF_3	118.5	43.0	D
140a	Trichloroethane	CH_3CCl_3	133.4	165	
142b	Monochlorodifluoroethane	CH_3CClF_2	100.5	12.2	S
143a	Trifluoroethane	CH_3CF_3	84	−53.5	
150a	Dichloroethane	CH_3CHCl_2	98.9	140	
152a	Difluoroethane	CH_3CHF_2	66	−12.4	C
160	Ethyl chloride	CH_3CH_2Cl	64.5	54.0	
(170	Ethane	CH_3CH_3	30	−127.5	C)[2]
218	Octafluoropropane	$CF_3CF_2CF_3$	188	−36.4	
(290	Propane	$CH_3CH_2CH_3$	44	−44.2	C)[2]
Cyclic Organic Compounds					
C316	Dichlorohexafluorocyclobutane	$C_4Cl_2F_6$	233	140	
C317	Monochloroheptafluorocyclobutane	C_4ClF_7	216.5	77	
C318	Octafluorocyclobutane	C_4F_8	200	21.1	D
Azeotropes					
500	Refrigerants-12/152a 73.8/26.2 wt %*	CCl_2F_2/CH_3CHF_2	99.29	−28.0	C
501	Refrigerants-22/12 75/25 wt %	$CHClF_2/CCl_2F_2$	93.1	−42	
502	Refrigerants-11/115 48.8/51.2 wt %	$CHClF_2/CClF_2CF_3$	112	−50.1	

TABLE 16-1 *(Continued)*

ASRE Standard Refrigerant Designation	Chemical Name	Chemical Formula	Molecular Weight	Boiling Point, F	Status[1]
Miscellaneous Organic Compounds					
Hydrocarbons					
50	Methane	CH_4	16.0	−259	C
170	Ethane	CH_3CH_3	30	−127.5	C
290	Propane	$CH_3CH_2CH_3$	44	−44.2	C
600	Butane	$CH_3CH_2CH_2CH_3$	58.1	31.3	
601	Isobutane	$CH(CH_3)_3$	58.1	14	
(1150	Ethylene	$CH_2{=}CH_2$	28.0	−155.0	C)[3]
(1270	Propylene	$CH_3CH{=}CH_2$	42.1	−53.7	C)[3]
Oxygen Compounds					
610	Ethyl ether	$C_2H_5OC_2H_5$	74.1	94.3	
611	Methyl formate	$HCOOCH_3$	60.0	89.2	
Sulfur Compounds					
620					
Nitrogen Compounds					
630	Methyl amine	CH_3NH_2	31.1	20.3	
631	Ethyl amine	$C_2H_5NH_2$	45.1	61.8	
Inorganic Compounds					
717	Ammonia	NH_3	17	−28.0	C
718	Water	H_2O	18	212	
729	Air		29	−318	
744	Carbon dioxide	CO_2	44	−109 (subl.)	C
744A	Nitrous oxide	N_2O	44	−127	
764	Sulfur dioxide	SO_2	64	14.0	C
Unsaturated Organic Compounds					
1112a	Dichlorodifluoroethylene	$CCl_2{=}CF_2$	133	67	
1113	Monochlorotrifluoroethylene	$CClF{=}CF_2$	116.5	−18.2	
1114	Tetrafluoroethylene	$CF_2{=}CF_2$	100	−105	
1120	Trichloroethylene	$CHCl{=}CCl_2$	131.4	187	
1130	Dichloroethylene	$CHCl{=}CHCl$	96.9	118	
1132a	Vinylidene fluoride	$CH_2{=}CF_2$	64	−119	
1140	Vinyl chloride	$CH_2{=}CHCl$	62.5	7.0	
1141	Vinyl fluoride	$CH_2{=}CHF$	46	−98	
1150	Ethylene	$CH_2{=}CH_2$	28.0	−155.0	C
1270	Propylene	$CH_3CH{=}CH_2$	42.1	−53.7	C

* Carrier Corp. Document 2-D-127, p. 1.

1. Denotes that as of October 1956, the status of these refrigerants as regards commercial evolution is as follows: C, S, or D. C—Commercial S—Semi-commercial D—Development.

2. The compounds methane, ethane, and propane appear in the halocarbon section in their proper numerical positions, but in parentheses since these products are not halocarbons.

3. The compounds ethylene and propylene appear in the hydrocarbon section as parenthetical items in order to indicate that these compounds are hydrocarbons. Ethylene and propylene are properly identified under Unsaturated Organic Compounds.

From the *ASRE Data Book*, Design Volume, 1957-58 Edition, by permission of the American Society of Heating, Refrigerating, and Air-Conditioning Engineers.

TABLE 16-2. Refrigerant-11 (Trichloromonofluoromethane) Properties of Liquid and Saturated Vapor

Temp F t	Pressure psia	Pressure psig	Liquid, sp vol cu ft/lb v_f	Vapor, sp vol cu ft/lb v_g	Enthalpy, datum −40 F Btu per lb Liquid h_f	Enthalpy Vapor h_g	Entropy, datum −40 F Btu per lb F Liquid s_f	Entropy Vapor s_g
−90	0.100	29.72*	0.00964	288.6	−9.89	81.7	−0.0250	0.2226
−80	.157	29.60*	.00961	189.0	−7.89	82.9	− .0197	.2192
−70	.240	29.43*	.00967	127.58	−5.91	84.0	− .0146	.2160
−60	.356	29.19*	.00974	87.5	−3.94	85.2	− .0096	.2133
−50	.518	28.86*	.00981	61.1	−1.97	86.3	− .0047	.2108
−40	0.739	28.42*	0.00988	44.21	0.00	87.48	0.0000	0.2085
−36	0.847	28.20*	.00991	38.93	0.79	87.96	.0019	.2076
−32	0.968	27.95*	.00993	34.37	1.58	88.44	.0037	.2068
−28	1.103	27.67*	.00996	30.44	2.36	88.91	.0055	.2060
−24	1.253	27.37*	.00999	27.03	3.15	89.39	.0073	.2053
−20	1.420	27.03*	0.01002	24.06	3.94	89.87	0.0091	0.2046
−16	1.605	26.65*	.01005	21.47	4.73	90.35	.0109	.2040
−12	1.810	26.24*	.01008	19.20	5.52	90.83	.0127	.2033
− 8	2.035	25.78*	.01011	17.21	6.31	91.31	.0145	.2027
− 4	2.283	25.27*	.01015	15.47	7.10	91.79	.0162	.2021
0	2.555	24.72*	0.01018	13.94	7.89	92.27	0.0179	0.2015
4	2.852	24.11*	.01021	12.58	8.68	92.75	.0197	.2010
5†	2.931	23.95*	.01022	12.27	8.88	92.88	.0201	.2009
8	3.179	23.45*	.01024	11.38	9.48	93.24	.0213	.2005
12	3.534	22.73*	.01027	10.31	10.28	93.72	.0231	.2000
16	3.923	21.94*	.01031	9.359	11.07	94.21	.0248	.1996
20	4.342	21.08*	0.01034	8.519	11.87	94.69	0.0264	0.1991
24	4.801	20.15*	.01037	7.760	12.68	95.18	.0281	.1987
28	5.294	19.14*	.01041	7.087	13.48	95.66	.0297	.1983
32	5.830	18.05*	.01044	6.481	14.28	96.15	.0314	.1979
36	6.411	16.87*	.01048	5.934	15.08	96.63	.0330	.1976
40	7.032	15.61*	0.01051	5.447	15.89	97.11	0.0346	0.1972
44	7.702	14.24*	.01055	5.006	16.70	97.60	.0362	.1969
48	8.422	12.78*	.01058	4.607	17.52	98.08	.0378	.1966
52	9.199	11.20*	.01062	4.245	18.33	98.56	.0394	.1963
56	10.02	9.53*	.01066	3.921	19.15	99.05	.0410	.1960
60	10.90	7.73*	0.01069	3.636	19.96	99.53	0.0426	0.1958
64	11.85	5.80*	.01073	3.356	20.78	100.01	.0442	.1955
68	12.87	3.72*	.01077	3.107	21.61	100.49	.0457	.1953
72	13.95	1.53*	.01081	2.883	22.43	100.97	.0473	.1950
76	15.09	0.39	.01085	2.679	23.26	101.45	.0489	.1948
80	16.31	1.61	0.01088	2.492	24.09	101.93	0.0504	0.1947
84	17.60	2.90	.01092	2.322	24.93	102.41	.0519	.1945
86†	18.28	3.58	.01094	2.242	25.34	102.65	.0527	.1944
88	18.97	4.27	.01096	2.165	25.76	102.89	.0535	.1943
92	20.43	5.73	.01101	2.020	26.60	103.36	.0550	.1941
96	21.97	7.27	.01105	1.887	27.43	103.83	.0565	.1940
100	23.60	8.90	0.01109	1.765	28.27	104.30	0.0580	0.1938
104	25.33	10.63	.01113	1.652	29.12	104.77	.0595	.1937
108	27.15	12.45	.01117	1.548	29.97	105.24	.0610	.1936
112	29.05	14.35	.01122	1.452	30.82	105.71	.0625	.1935
116	31.07	16.37	.01126	1.363	31.67	106.17	.0639	.1934
120	33.20	18.50	0.01130	1.281	32.53	106.63	0.0654	0.1933
124	35.42	20.72	.01135	1.206	33.38	107.09	.0669	.1932
128	37.74	23.04	.01139	1.135	34.24	107.55	.0683	.1931
132	40.23	25.53	.01144	1.068	35.10	108.00	.0698	.1930
136	42.80	28.10	.01149	1.007	35.97	108.46	.0712	.1929
140	45.50	30.80	0.01154	0.9505	36.84	108.91	0.0727	0.1929
144	48.35	33.65	.01159	.8970	37.71	109.35	.0741	.1928
148	51.31	36.61	.01163	.8476	38.59	109.80	.0755	.1927
152	54.41	39.71	.01168	.8014	39.46	110.24	.0770	.1927
156	57.65	42.95	.01173	.7581	40.35	110.69	.0784	.1927
160	61.04	46.34	0.01179	0.7176	41.23	111.12	0.0798	0.1926

* Inches of mercury below one standard atmosphere
† Standard cycle temperatures.

Courtesy E. I. du Pont de Nemours & Co., Inc.

TABLE 16-3. Dichlorodifluoromethane (Refrigerant-12) Properties of Saturated Vapor

Temp.	Pressure		Volume		Density		Heat content from — 40°			Entropy from — 40°		Temp.
°F	Abs. lb./in.²	Gage lb./in.²	Liquid ft.³/lb.	Vapor ft.³/lb.	Liquid lb./ft.³	Vapor lb./ft.³	Liquid Btu./lb.	Latent Btu./lb.	Vapor Btu./lb.	Liquid Btu./lb. °F.	Vapor Btu./lb.°F	°F
t	p	p_d	v_f	v_g	$1/v_f$	$1/v_g$	h_f	h	h_g	s_f	s_g	t
—155	0.1163	29.68*	0.00954	232.29	104.86	0.004305	—24.61	84.61	60.00	—0.0686	0.2092	—155
—150	0.1527	29.61*	0.00957	179.79	104.46	0.005562	—23.50	84.07	60.57	—0.0650	0.2065	—150
—145	.1985	29.52*	.00961	140.52	104.05	.007117	—22.39	83.53	61.14	— .0615	.2040	—145
—140	.2554	29.40*	.00965	110.92	103.64	.009016	—21.29	83.01	61.72	— .0580	.2017	—140
—135	.3256	29.26*	.00969	88.34	103.22	.01132	—20.19	82.49	62.30	— .0546	.1995	—135
—130	.4116	29.08*	.00973	70.94	102.80	.01410	—19.10	81.98	62.88	— .0512	.1975	—130
—125	0.5160	28.87*	0.00977	57.42	102.38	0.01742	—18.02	81.48	63.46	—0.0480	0.1955	—125
—120	.6417	28.61*	.00981	46.84	101.95	.02125	—16.94	80.98	64.04	— .0448	.1937	—120
—115	.7921	28.31*	.00985	38.49	101.52	.02598	—15.85	80.48	64.63	— .0416	.1919	—115
—110	.9709	27.94*	.00989	31.84	101.08	.03141	—14.78	80.00	65.22	— .0385	.1903	—110
—105	1.182	27.51*	.00994	26.51	100.64	.03773	—13.71	79.52	65.81	— .0355	.1888	—105
—100	1.430	27.01*	0.00998	22.20	100.20	0.04504	—12.64	79.04	66.40	—0.0325	0.1873	—100
— 95	1.719	26.42*	.01003	18.71	99.75	.05344	—11.58	78.57	66.99	— .0295	.1860	— 95
— 90	2.054	25.74*	.01007	15.86	99.30	.06305	—10.51	78.10	67.59	— .0266	.1847	— 90
— 85	2.441	24.95*	.01012	13.51	98.85	.07400	— 9.46	77.64	68.18	— .0238	.1835	— 85
— 80	2.885	24.05*	.01016	11.57	98.39	.08640	— 8.40	77.17	68.77	— .0210	.1823	— 80
— 75	3.393	23.01*	0.01021	9.958	97.92	0.1004	— 7.35	76.71	69.36	—0.0182	0.1813	— 75
— 70	3.971	21.84*	.01026	8.608	97.46	.1162	— 6.30	76.25	69.95	— .0155	.1802	— 70
— 65	4.626	20.50*	.01031	7.474	96.99	.1338	— 5.25	75.79	70.54	— .0128	.1793	— 65
— 60	5.365	19.00*	.01036	6.516	96.51	.1535	— 4.20	75.33	71.13	— .0102	.1783	— 60
— 55	6.195	17.31*	.01041	5.704	96.04	.1753	— 3.15	74.87	71.72	— .0076	.1774	— 55
— 50	7.125	15.42*	0.01047	5.012	95.55	0.1995	— 2.11	74.42	72.31	—0.0050	0.1767	— 50
— 45	8.163	13.31*	.01052	4.420	95.07	.2263	— 1.06	73.97	72.91	— .0025	.1759	— 45

TABLE 16-3 *(Continued)*

Temp.	Pressure		Volume		Density		Heat content from — 40°			Entropy from — 40°		Temp.
°F t	Abs. lb./in.² p	Gage lb./in.² pd	Liquid ft.³/lb. v_f	Vapor ft.³/lb. v_g	Liquid lb./ft.³ $1/v_f$	Vapor lb./ft.³ $1/v_g$	Liquid Btu./lb. h_f	Latent Btu./lb. h	Vapor Btu./lb. h_g	Liquid Btu./lb. °F. s_f	Vapor Btu./lb.°F s_g	°F t
— 40	9.32	10.92*	0.0106	3.911	94.58	0.2557	0	73.50	73.50	0	0.17517	— 40
— 38	9.82	9.91*	.0106	3.727	94.39	.2683	0.40	73.34	73.74	0.00094	.17490	— 38
— 36	10.34	8.87*	.0106	3.553	94.20	.2815	0.81	73.17	73.98	.00188	.17463	— 36
— 34	10.87	7.80*	.0106	3.389	93.99	.2951	1.21	73.01	74.22	.00282	.17438	— 34
— 32	11.43	6.66*	.0107	3.234	93.79	.3092	1.62	72.84	74.46	.00376	.17412	— 32
— 30	12.02	5.45*	0.0107	3.088	93.59	0.3238	2.03	72.67	74.70	0.00471	0.17387	— 30
— 28	12.62	4.23*	.0107	2.950	93.39	.3390	2.44	72.50	74.94	.00565	.17364	— 28
— 26	13.26	2.93*	.0107	2.820	93.18	.3546	2.85	72.33	75.18	.00659	.17340	26
— 24	13.90	1.63*	.0108	2.698	92.98	.3706	3.25	72.16	75.41	.00753	.17317	— 24
— 22	14.58	0.24*	.0108	2.583	92.78	.3871	3.66	71.98	75.64	.00846	.17296	— 22
— 20	15.28	0.58	0.0108	2.474	92.58	0.4042	4.07	71.80	75.87	0.00940	0.17275	— 20
— 18	16.01	1.31	.0108	2.370	92.38	.4219	4.48	71.63	76.11	.01033	.17253	— 18
— 16	16.77	2.07	.0108	2.271	92.18	.4403	4.89	71.45	76.34	.01126	.17232	— 16
— 14	17.55	2.85	.0109	2.177	91.97	.4593	5.30	71.27	76.57	.01218	.17212	— 14
— 12	18.37	3.67	.0109	2.088	91.77	.4789	5.72	71.09	76.81	.01310	.17194	— 12
— 10	19.20	4.50	0.0109	2.003	91.57	0.4993	6.14	70.91	77.05	0.01403	0.17175	— 10
— 8	20.08	5.38	.0109	1.922	91.35	.5203	6.57	70.72	77.29	.01496	.17158	— 8
— 6	20.98	6.28	.0110	1.845	91.14	.5420	6.99	70.53	77.52	.01589	.17140	— 6
— 4	21.91	7.21	.0110	1.772	90.93	.5644	7.41	70.34	77.75	.01682	.17123	— 4
— 2	22.87	8.17	.0110	1.703	90.72	.5872	7.83	70.15	77.98	.01775	.17107	— 2
0	23.87	9.17	0.0110	1.637	90.52	0.6109	8.25	69.96	78.21	0.01869	0.17091	0
2	24.89	10.19	.0110	1.574	90.31	.6352	8.67	69.77	78.44	.01961	.17075	2
4	25.96	11.26	.0111	1.514	90.11	.6606	9.10	69.57	78.67	.02052	.17060	4
5†	26.51	11.81	.0111	1.485	90.00	.6735	9.32	69.47	78.79	.02097	.17052	5†
6	27.05	12.35	.0111	1.457	89.88	.6864	9.53	69.37	78.90	.02143	.17045	6
8	28.18	13.48	.0111	1.403	89.68	.7129	9.96	69.17	79.13	.02235	.17030	8

* Inches of mercury below one atmosphere. † Standard ton temperatures.

TABLE 16-3 (*Continued*)

Temp.	Pressure		Volume		Density		Heat content from − 40°			Entropy from − 40°		Temp.
°F	Abs. lb./in.²	Gage lb./in.²	Liquid ft.³/lb.	Vapor ft.³/lb.	Liquid lb./ft.³	Vapor lb./ft.³	Liquid Btu./lb.	Latent Btu./lb.	Vapor Btu./lb.	Liquid Btu./lb.°F	Vapor Btu./lb.°F	°F
t	p	p_d	v_f	v_g	$1/v_f$	$1/v_g$	h_f	h	h_g	s_f	s_g	t
10	29.35	14.65	0.0112	1.351	89.45	0.7402	10.39	68.97	79.36	0 02328	0.17015	10
12	30.56	15.86	.0112	1.301	89.24	.7687	10.82	68.77	79.59	.02419	.17001	12
14	31.80	17.10	.0112	1.253	89.03	.7981	11.26	68.56	79.82	.02510	.16987	14
16	33.08	18.38	.0112	1.207	88.81	.8288	11.70	68.35	80.05	.02601	.16974	16
18	34.40	19.70	.0113	1.163	88.58	.8598	12.12	68.15	80.27	.02692	.16961	18
20	35.75	21.05	0.0113	1.121	88.37	0.8921	12.55	67 94	80.49	0.02783	0.16949	20
22	37.15	22.45	.0113	1.081	88.13	.9251	13.00	67.72	80.72	.02873	.16938	22
24	38.58	23.88	.0113	1.043	87.91	.9588	13.44	67.51	80.95	.02963	.16926	24
26	40.07	25.37	.0114	1.007	87.68	.9930	13.88	67.29	81.17	.03053	.16913	26
28	41.59	26.89	.0114	0.973	87.47	1.028	14.32	67.07	81.39	.03143	.16900	28
30	43.16	28.46	0.0115	0.939	87.24	1.065	14.76	66.85	81.61	0.03233	0.16887	30
32	44.77	30.07	.0115	.908	87.02	1.102	15.21	66.62	81.83	.03323	.16876	32
34	46.42	31.72	.0115	.877	86.78	1.140	15.65	66.40	82.05	.03413	.16865	34
36	48.13	33.43	.0116	.848	86.55	1.180	16.10	66.17	82.27	.03502	.16854	36
38	49.88	35.18	.0116	.819	86.33	1.221	16.55	65.94	82.49	.03591	.16843	38
40	51.68	36.98	0.0116	0.792	86.10	1.263	17.00	65.71	82.71	0.03680	0.16833	40
42	53.51	38.81	.0116	.767	85.88	1.304	17.46	65.47	82.93	.03770	.16823	42
44	55.40	40.70	.0117	.742	85.66	1.349	17.91	65.24	83.15	.03859	.16813	44
46	57.35	42.65	.0117	.718	85.43	1.393	18.36	65.00	83.36	.03948	.16803	46
48	59.35	44.65	.0117	.695	85.19	1.438	18.82	64.74	83.57	.04037	.16794	48

TABLE 16-3 (*Continued*)

50	61.39	46.69	0.0118	0.673	84.94	1.485	19.27	64.51	83.78	0.04126	0.16785	50
52	63.49	48.79	.0118	.652	84.71	1.534	19.72	64.27	83.99	.04215	.16776	52
54	65.63	50.93	.0118	.632	84.50	1.583	20.18	64.02	84.20	.04304	.16767	54
56	67.84	53.14	.0119	.612	84.28	1.633	20.64	63.77	84.41	.04392	.16758	56
58	70.10	55.40	.0119	.593	84.04	1.686	21.11	63.51	84.62	.04480	.16749	58
60	72.41	57.71	0.0119	0.575	83.78	1.740	21.57	63.25	84.82	0.04568	0.16741	60
62	74.77	60.07	.0120	.557	83.57	1.795	22.03	62.99	85.02	.04657	.16733	62
64	77.20	62.50	.0120	.540	83.34	1.851	22.49	62.73	85.22	.04745	.16725	64
66	79.67	64.97	.0120	.524	83.10	1.909	22.95	62.47	85.42	.04833	.16717	66
68	82.24	67.54	.0121	.508	82.86	1.968	23.42	62.20	85.62	.04921	.16709	68
70	84.82	70.12	0.0121	0.493	82.60	2.028	23.90	61.92	85.82	0.05009	0.16701	70
72	87.50	72.80	.0121	.479	82.37	2.090	24.37	61.65	86.02	.05097	.16693	72
74	90.20	75.50	.0122	.464	82.12	2.153	24.84	61.38	86.22	.05185	.16685	74
76	93.00	78.30	.0122	.451	81.87	2.218	25.32	61.10	86.42	.05272	.16677	76
78	95.85	81.15	.0123	.438	81.62	2.284	25.80	60.81	86.61	.05359	.16669	78
80	98.76	84.06	0.0123	0.425	81.39	2.353	26.28	60.52	86.80	0.05446	0.16662	80
82	101.7	87.00	.0123	.413	81.12	2.423	26.76	60.23	86.99	.05534	.16655	82
84	104.8	90.1	.0124	.401	80.87	2.495	27.24	59.94	87.18	.05621	.16648	84
86†	**107.9**	**93.2**	**.0124**	**.389**	**80.63**	**2.569**	**27.72**	**59.65**	**87.37**	**.05708**	**.16640**	**86†**
88	111.1	96.4	.0124	.378	80.37	2.645	28.21	59.35	87.56	.05795	.16632	88
90	114.3	99.6	0.0125	0.368	80.11	2.721	28.70	59.04	87.74	0.05882	0.16624	90
92	117.7	103.0	.0125	.357	79.86	2.799	29.19	58.73	87.92	.05969	.16616	92
94	121.0	106.3	.0126	.347	79.60	2.880	29.68	58.42	88.10	.06056	.16608	94
96	124.5	109.8	.0126	.338	79.32	2.963	30.18	58.10	88.28	.06143	.16600	96
98	128.0	113.3	.0126	.328	79.06	3.048	30.67	57.78	88.45	.06230	.16592	98
100	131.6	116.9	0.0127	0.319	78.80	3.135	31.16	57.46	88.62	0.06316	0.16584	100
102	135.3	120.6	.0127	.310	78.54	3.224	31.65	57.14	88.79	.06403	.16576	102
104	139.0	124.3	.0128	.302	78.27	3.316	32.15	56.80	88.95	.06490	.16568	104
106	142.8	128.1	.0128	.293	78.00	3.411	32.65	56.46	89.11	.06577	.16560	106
108	146.8	132.1	.0129	.285	77.73	3.509	33.15	56.12	89.27	.06663	.16551	108

* Inches of mercury below one atmosphere.

† Standard ton temperatures.

TABLE 16-3 *(Continued)*

Temp.	Pressure		Volume		Density		Heat content from − 40°			Entropy from − 40°		Temp.
°F	Abs. lb./in.²	Gage lb./in.²	Liquid ft.³/lb.	Vapor ft.³/lb.	Liquid lb./ft.³	Vapor lb./ft.³	Liquid Btu./lb.	Latent Btu./lb.	Vapor Btu./lb.	Liquid Btu./lb.°F	Vapor Btu./lb.°F	°F
t	p	p_d	v_f	v_g	$1/v_f$	$1/v_g$	h_f	h	h_g	s_f	s_g	t
110	150.7	136.0	0.0129	0.277	77.46	3.610	33.65	55.78	89.43	0.06749	0.16542	110
112	154.8	140.1	.0130	.269	77.18	3.714	34.15	55.43	89.58	.06836	.16533	112
114	158.9	144.2	.0130	.262	76.89	3.823	34.65	55.08	89.73	.06922	.16524	114
116	163.1	148.4	.0131	.254	76.60	3.934	35.15	54.72	89.87	.07008	.16515	116
118	167.4	152.7	.0131	.247	76.32	4.049	35.65	54.36	90.01	.07094	.16505	118
120	171.8	157.1	0.0132	0.240	76.02	4.167	36.16	53.99	90.15	0.07180	0.16495	120
122	176.2	161.5	.0132	.233	75.72	4.288	36.66	53.62	90.28	.07266	.16484	122
124	180.8	166.1	.0133	.277	75.40	4.413	37.16	53.24	90.40	.07352	.16473	124
126	185.4	170.7	.0133	.220	75.10	4.541	37.67	52.85	90.52	.07437	.16462	126
128	190.1	175.4	.0134	.214	74.78	4.673	38.18	52.46	90.64	.07522	.16450	128
130	194.9	180.2	0.0134	0.208	74.46	4.808	38.69	52.07	90.76	0.07607	0.16438	130
132	199.8	185.1	.0135	.202	74.13	4.948	39.19	51.67	90.86	.07691	.16425	132
134	204.8	190.1	.0135	.196	73.81	5.094	39.70	51.26	90.96	.07775	.16411	134
136	209.9	195.2	.0136	.191	73.46	5.247	40.21	50.85	91.06	.07858	.16396	136
138	215.0	200.3	.0137	.185	73.10	5.405	40.72	50.43	91.15	.07941	.16380	138
140	220.2	205.5	0.0138	0.180	72.73	5.571	41.24	50.00	91.24	0.08024	0.16363	140

† Standard ton temperatures.

TABLE 16-4. Refrigerant-13 (monochlorotrifluoromethane) Properties of Liquid and Saturated Vapor

Temp F	Pressure psia	Specific volume cu ft per lb		Enthalpy datum −40 F Btu per lb			Entropy datum −40 F Btu per lb R		
		Liquid	Vapor	Liquid	Latent	Vapor	Liquid	Vaporization	Vapor
−200	0.4329	0.009466	61.33	−34.551	73.096	38.545	−0.10081	0.28147	0.18066
−190	0.7490	0.009574	36.74	−32.429	72.029	39.600	−0.09313	0.26708	0.17395
−180	1.238	0.009685	22.99	−30.298	70.790	40.672	−0.08575	0.25375	0.16800
−170	1.967	0.009801	14.942	−28.208	69.904	41.696	−0.07858	0.24131	0.16273
−160	3.104	0.009920	9.750	−26.083	68.808	42.725	−0.07213	0.22960	0.15747
−150	4.464	0.01004	6.976	−24.010	67.783	43.773	−0.06491	0.21887	0.15396
−140	6.455	0.01017	4.950	−21.902	66.696	44.794	−0.05844	0.20863	0.15019
−130	9.080	0.01031	3.605	−19.792	65.596	45.804	−0.05209	0.19896	0.14687
−120	12.48	0.01045	2.681	−17.671	64.473	46.802	−0.04590	0.18980	0.14390
−110	16.81	0.01060	2.031	−15.527	63.316	47.789	−0.03977	0.18106	0.14129
−100	22.23	0.01075	1.5642	−13.387	62.138	48.751	−0.03286	0.17275	0.13889
− 90	28.89	0.01091	1.2232	−11.241	60.941	49.700	−0.02806	0.16484	0.13678
− 80	36.98	0.01109	0.9689	− 9.052	59.672	50.620	−0.02230	0.15716	0.13486
− 70	46.68	0.01127	0.7766	− 6.843	58.362	51.519	−0.01665	0.14977	0.13312
− 60	58.19	0.01146	0.6289	− 4.604	56.993	52.389	−0.01106	0.14259	0.13153
− 50	71.71	0.01167	0.5139	− 2.320	55.546	53.226	−0.00548	0.13558	0.13009
− 40	87.43	0.01189	0.4234	0.000	54.023	54.023	0.00000	0.12872	0.12872
− 30	105.6	0.01213	0.3512	2.363	52.416	54.779	0.00545	0.12199	0.12744
− 20	126.4	0.01239	0.2930	4.809	50.668	55.477	0.01096	0.11524	0.12620
− 10	150.1	0.01268	0.2454	7.484	48.630	56.113	0.01686	0.10814	0.12500
0	176.8	0.01299	0.2066	10.052	46.638	56.690	0.02234	0.10146	0.12380
10	206.8	0.01335	0.17443	12.696	44.479	57.175	0.02789	0.09470	0.12259
20	240.4	0.01375	0.14732	15.443	42.100	57.543	0.03351	0.08777	0.12128
30	277.9	0.01422	0.12437	18.247	39.472	57.719	0.03921	0.08061	0.11982
40	319.6	0.01477	0.10455	21.370	36.450	57.820	0.04516	0.07295	0.11811
45	342.2	0.01509	0.09565	22.979	34.769	57.748	0.04826	0.06889	0.11715
50	365.9	0.01546	0.08734	24.651	32.958	57.609	0.05143	0.06466	0.11609
55	390.8	0.01588	0.07945	26.418	30.946	56.364	0.05473	0.06013	0.11486
60	417.0	0.01637	0.07189	28.310	28.677	56.987	0.05824	0.05518	0.11342
65	444.5	0.01696	0.06468	30.322	26.137	56.459	0.06193	0.04981	0.11174
70	473.4	0.01771	0.05767	32.515	23.193	55.708	0.06591	0.40379	0.10970
75	503.7	0.01874	0.05207	35.110	19.382	54.492	0.07059	0.03625	0.10684
80	535.5	0.02047	0.04131	38.527	13.565	52.092	0.07672	0.02513	0.10185
83.93	561.3	0.02772	0.02772	45.271	—	45.271	0.08898	—	0.08898

Courtesy E. I. du Pont de Nemours & Co., Inc.

TABLE 16-5. Refrigerant-22 (Monochlorodifluoromethane) Properties of Liquid and Saturated Vapor

Temp F	Pressure		Liquid, density	Vapor, sp vol	Enthalpy, datum −40 F Btu per lb		Entropy, datum −40 F Btu per lb F	
t	psia	psig	lb/cu ft $1/v_f$	cu ft/lb v_g	Liquid h_f	Vapor h_g	Liquid s_f	Vapor s_g
−155	0.19901	29.51*	97.67	188.1	−29.07	86.78	−0.0808	0.2996
−150	0.2605	29.39*	97.33	146.1	−27.79	87.36	−0.0767	0.2952
−145	0.3375	29.23*	96.99	114.5	−26.52	87.94	− .0727	.2912
−140	0.4332	29.04*	96.63	90.61	−25.25	88.53	−0.0687	0.2874
−135	0.5511	28.80*	96.27	72.33	−23.99	89.11	− .0647	.2837
−130	0.6949	28.51*	95.91	58.21	−22.73	89.70	−0.0609	0.2803
−125	0.8692	28.15*	95.53	47.23	−21.47	90.29	− .0571	.2770
−120	1.079	27.72*	95.15	38.60	−20.22	90.88	−0.0534	0.2738
−115	1.329	27.21*	94.76	31.77	−18.98	91.47	− .0497	.2708
−110	1.626	26.61*	94.37	26.33	−17.73	92.07	−0.0461	0.2680
−105	1.976	25.90*	93.97	21.96	−16.48	92.67	− .0425	.2653
−100	2.386	25.06*	93.56	18.43	−15.23	93.27	−0.0390	0.2627
− 95	2.865	24.09*	93.14	15.54	−13.98	93.87	− .0356	.2602
− 90	3.417	22.96*	92.72	13.20	−12.73	94.47	−0.0322	0.2579
− 85	4.055	21.67*	92.29	11.26	−11.47	95.08	− .0288	.2556
− 80	4.787	20.18*	91.85	9.650	−10.22	95.68	−0.0255	0.2535
− 78	5.100	19.55*	91.67	9.086	− 9.72	95.92	− .0242	.2526
− 76	5.430	18.87*	91.49	8.561	− 9.21	96.16	− .0229	.2518
− 74	5.79	18.14*	91.31	8.072	− 8.70	96.40	− .0216	.2510
− 72	6.17	17.37*	91.13	7.616	− 8.20	96.64	− .0203	.2502
− 70	6.57	16.55*	90.95	7.192	− 7.69	96.88	−0.0253	0.2494
− 68	6.99	15.70*	90.77	6.795	− 7.19	97.12	− .0177	.2487
− 66	7.40	14.86*	90.58	6.426	− 6.68	97.36	− .0164	.2479
− 64	7.86	13.93*	90.39	6.079	− 6.17	97.60	− .0151	.2472
− 62	8.35	12.93*	90.21	5.755	− 5.67	97.84	− .0138	.2465
− 60	8.86	11.89*	90.03	5.452	− 5.16	98.08	−0.0126	0.2458
− 58	9.39	10.81*	89.84	5.166	− 4.65	98.32	− .0113	.2451
− 56	9.94	9.69*	89.65	4.900	− 4.13	98.56	− .0100	.2444
− 54	10.51	8.53*	89.46	4.650	− 3.61	98.80	− .0087	.2438
− 52	11.11	7.31*	89.27	4.415	− 3.09	99.04	− .0075	.2431
− 50	11.74	6.03*	89.08	4.192	− 2.58	99.28	−0.0062	0.2425
− 48	12.40	4.68*	88.88	3.986	− 2.06	99.52	− .0050	.2418
− 46	13.09	3.28*	88.68	3.793	− 1.54	99.76	− .0037	.2412
− 44	13.80	1.83*	88.49	3.611	− 1.02	100.00	− .0025	.2406
− 42	14.54	0.326*	88.30	3.440	− 0.51	100.23	− .0012	.2400
− 40	15.31	0.610	88.10	3.279	0.00	100.46	0.0000	0.2394
− 38	16.12	1.42	87.90	3.126	0.53	100.70	.0013	.2389
− 36	16.97	2.27	87.70	2.981	1.05	100.93	.0025	.2383
− 34	17.85	3.15	87.50	2.844	1.58	101.17	.0037	.2377
− 32	18.77	4.07	87.29	2.713	2.10	101.40	.0050	.2372
− 30	19.72	5.02	87.09	2.590	2.62	101.63	0.0062	0.2367
− 28	20.71	6.01	86.89	2.474	3.15	101.86	.0074	.2361
− 26	21.73	7.03	86.69	2.365	3.69	102.10	.0086	.2356
− 24	22.79	8.09	86.48	2.262	4.22	102.33	.0099	.2351
− 22	23.88	9.18	86.27	2.165	4.75	102.56	.0111	.2346
− 20	25.01	10.31	86.06	2.074	5.28	102.79	0.0123	0.2341
− 18	26.18	11.48	85.85	1.987	5.82	103.02	.0135	.2336
− 16	27.39	12.69	85.64	1.905	6.40	103.25	.0147	.2331
− 14	28.64	13.94	85.43	1.827	6.90	103.48	.0159	.2326
− 12	29.94	15.24	85.21	1.752	7.43	103.70	.0170	.2321
− 10	31.29	16.59	84.99	1.681	7.96	103.92	0.0182	0.2316
− 8	32.69	17.99	84.78	1.613	8.49	104.14	.0194	.2312
− 6	34.14	19.44	84.56	1.549	9.02	104.36	.0205	.2307
− 4	35.64	20.94	84.34	1.488	9.55	104.58	.0217	.2302
− 2	37.19	22.49	84.12	1.429	10.09	104.80	.0228	.2298
0	38.79	24.09	83.90	1.373	10.63	105.02	0.0240	0.2293
2	40.43	25.73	83.68	1.320	11.17	105.24	0.0251	0.2289
4	42.14	27.44	83.45	1.270	11.70	105.45	.0262	.2285
5	43.02	28.33	83.34	1.246	11.97	105.56	.0268	.2283
6	43.91	29.21	83.23	1.221	12.23	105.66	.0274	.2280
8	45.74	31.04	83.01	1.175	12.76	105.87	.0285	.2276
10	47.63	32.93	82.78	1.130	13.29	106.08	0.0296	0.2272
12	49.58	34.88	82.55	1.088	13.82	106.29	.0307	.2268
14	51.59	36.89	82.32	1.048	14.36	106.50	.0319	.2264
16	53.66	38.96	82.09	1.009	14.90	106.71	.0330	.2260
18	55.79	41.09	81.86	0.9721	15.44	106.92	.0341	.2257

* Inches mercury below one atmosphere.

TABLE 16-5 (Continued)

Temp F t	Pressure psia	Pressure psig	Liquid, density lb/cu ft $1/v_f$	Vapor, sp vol cu ft/lb v_g	Enthalpy, datum −40 F Btu per lb Liquid h_f	Enthalpy Vapor h_g	Entropy, datum −40 F Btu per lb F Liquid s_f	Entropy Vapor s_g
20	57.98	43.28	81.63	0.9369	15.98	107.13	0.0352	0.2253
22	60.23	45.53	81.39	.9032	16.52	107.33	.0364	.2249
24	62.55	47.85	81.16	.8707	17.06	107.53	.0375	.2246
26	64.94	50.24	80.92	.8398	17.61	107.73	.0379	.2242
28	67.40	52.70	80.69	.8100	18.17	107.93	.0398	.2239
30	69.93	55.23	80.45	0.7816	18.74	108.13	0.0409	0.2235
32	72.53	57.83	80.21	.7543	19.32	108.33	.0421	.2232
34	75.21	60.51	79.97	.7283	19.90	108.52	.0433	.2228
36	77.97	63.27	79.73	.7032	20.49	108.71	.0445	.2225
38	80.81	66.11	79.49	.6791	21.09	108.90	.0457	.2222
40	83.72	69.02	79.25	0.6559	21.70	109.09	0.0469	0.2218
42	86.69	71.99	79.00	.6339	22.29	109.27	.0481	.2215
44	89.74	75.04	78.76	.6126	22.90	109.45	.0493	.2211
46	92.88	78.18	78.51	.5922	23.50	109.63	.0505	.2208
48	96.10	81.40	78.26	.5726	24.11	109.80	.0516	.2205
50	99.40	84.70	78.02	0.5537	24.73	109.98	0.0528	0.2201
52	102.8	88.10	77.77	.5355	25.34	110.14	.0540	.2198
54	106.2	91.5	77.51	.5184	25.95	110.30	.0552	.2194
56	109.8	95.1	77.26	.5014	26.58	110.47	.0564	.2191
58	113.5	98.8	77.01	.4849	27.22	110.63	.0576	.2188
60	117.2	102.5	76.75	0.4695	27.83	110.78	0.0588	0.2185
62	121.0	106.3	76.50	.4546	28.46	110.93	.0600	.2181
64	124.9	110.2	76.24	.4403	29.09	111.08	.0612	.2178
66	128.9	114.2	75.98	.4264	29.72	111.22	.0624	.2175
68	133.0	118.3	75.72	.4129	30.35	111.35	.0636	.2172
70	137.2	122.5	75.46	0.4000	30.99	111.49	0.0648	0.2168
72	141.5	126.8	75.20	.3875	31.65	111.63	.0661	.2165
74	145.9	131.2	74.94	.3754	32.29	111.75	.0673	.2162
76	150.4	135.7	74.68	.3638	32.94	111.88	.0684	.2158
78	155.0	140.3	74.41	.3526	33.61	112.01	.0696	.2155
80	159.7	145.0	74.15	0.3417	34.27	112.13	0.0708	0.2151
82	164.5	149.8	73.89	.3313	34.92	112.24	.0720	.2148
84	169.4	154.7	73.63	.3212	35.60	112.36	.0732	.2144
86	174.5	159.8	73.36	.3113	36.28	112.47	.0744	.2140
88	179.6	164.9	73.09	.3019	36.94	112.57	.0756	.2137
90	184.8	170.1	72.81	0.2928	37.61	112.67	0.0768	0.2133
92	190.1	175.4	72.53	.2841	38.28	112.76	.0780	.2130
94	195.6	180.9	72.24	.2755	38.97	112.85	.0792	.2126
96	201.2	186.5	71.95	.2672	39.65	112.93	.0803	.2122
98	206.8	192.1	71.65	.2594	40.32	113.00	.0815	.2119
100	212.6	197.9	71.35	0.2517	40.98	113.06	0.0827	0.2115
102	218.5	203.8	71.05	.2443	41.65	113.12	.0839	.2111
104	224.6	209.9	70.74	.2370	42.32	113.16	.0851	.2107
106	230.7	216.0	70.42	.2301	42.98	113.20	.0862	.2104
108	237.0	222.3	70.11	.2233	43.66	113.24	.0874	.2100
110	243.4	228.7	69.78	0.2167	44.35	113.29	0.0886	0.2096
112	249.9	235.2	69.45	.2104	45.04	113.34	.0898	.2093
114	256.6	241.9	69.12	.2043	45.74	113.38	.0909	.2089
116	263.4	248.7	68.78	.1983	46.44	113.42	.0921	.2085
118	270.3	255.6	68.44	.1926	47.14	113.46	.0933	.2081
120	277.3	262.6	68.10	0.1871	47.85	113.52	0.0945	0.2078
122	384.4	269.7	67.75	.1825	48.6	113.57		
124	391.6	276.9	67.40	.1772	49.4	113.61		
126	398.8	284.1	67.05	.1724	50.2	113.65		
128	306.1	291.4	66.70	.1675	50.8	113.69		
130	313.5	298.8	66.35	0.1629	51.5	113.71		
132	321.0	306.3	66.00	.1585	52.3	113.74		
134	328.7	314.0	65.65	.1538	53.1	113.77		
136	336.6	321.9	65.25	.1492	53.8	113.79		
138	344.6	329.9	64.85	.1449	54.6	113.80		
140	352.7	338.0	64.45	0.1408	55.3	113.81		
142	361.0	346.3	64.05	.1368	56.1	113.80		
144	369.7	355.0	63.65	.1330	56.9	113.79		
146	379.0	364.3	63.25	.1292	57.7	113.78		
148	388.8	374.1	62.85	.1253	58.4	113.76		
150	399.0	384.3	62.45	0.1216	59.2	113.74		
152	407.0	392.3	62.02	.1179	60.0	113.71		
154	416.0	401.3	61.58	.1141	60.8	113.67		
156	426.0	411.3	61.13	.1105	61.6	113.62		
158	436.5	421.8	60.67	.1070	62.5	113.56		
160	448.0	433.3	60.20	0.1035	63.5	113.50		

Courtesy E. I. du Pont de Nemours & Co., Inc.

TABLE 16-6. Refrigerant-717 (*Ammonia*) Properties of Liquid and Saturated Vapor

Temp F	Pressure		Liquid, density	Vapor, sp vol	Enthalpy, datum −40 F Btu per lb		Entropy, datum −40 F Btu per lb F	
t	psia	psig	lb/cu ft $1/v_f$	cu ft/lb v_g	Liquid h_f	Vapor h_g	Liquid s_f	Vapor s_g
−105	1.00	*27.9	45.71	223.14	−68.5	570.3	−0.1774	1.6243
−104	1.04	27.8	45.67	214.23	−67.5	570.7	− .1744	1.6205
−103	1.08	27.7	45.63	205.90	−66.4	571.2	− .1714	1.6167
−102	1.14	27.7	45.59	197.70	−65.4	571.6	− .1685	1.6129
−101	1.19	27.5	45.55	190.08	−64.3	572.1	− .1655	1.6092
−100	1.24	*27.4	45.51	182.90	−63.3	572.5	−0.1626	1.6055
− 99	1.29	27.3	45.47	175.42	−62.2	572.9	− .1597	1.6018
− 98	1.35	27.2	45.43	168.48	−61.2	573.4	− .1568	1.5982
− 97	1.41	27.0	45.40	161.98	−60.1	573.8	− .1539	1.5945
− 96	1.47	26.9	45.36	155.92	−59.1	574.3	− .1510	1.5910
− 95	1.52	*26.8	45.32	150.30	−58.0	574.7	−0.1481	1.5874
− 94	1.59	26.7	45.28	144.68	−57.0	575.1	− .1452	1.5838
− 93	1.66	26.5	45.24	139.27	−55.9	575.6	− .1423	1.5803
− 92	1.73	26.4	45.20	134.06	−54.9	576.0	− .1395	1.5768
− 91	1.79	26.2	45.16	129.06	−53.8	576.5	− .1366	1.5734
− 90	1.86	*26.1	45.12	124.28	−52.8	576.9	−0.1338	1.5699
− 89	1.94	26.0	45.08	119.75	−51.7	577.3	− .1309	1.5665
− 88	2.02	25.8	45.04	115.37	−50.7	577.8	− .1281	1.5631
− 87	2.11	25.6	45.00	111.31	−49.6	578.2	− .1253	1.5597
− 86	2.18	25.5	44.96	107.39	−48.6	578.6	− .1225	1.5504
− 85	2.27	*25.3	44.92	103.63	−47.5	579.1	−0.1197	1.5531
− 84	2.36	25.1	44.88	99.87	−46.5	579.5	− .1169	1.5498
− 83	2.46	24.9	44.84	96.28	−45.4	579.9	− .1141	1.5465
− 82	2.55	24.7	44.80	92.86	−44.4	580.4	− .1113	1.5432
− 81	2.65	24.5	44.76	89.65	−43.3	580.8	− .1085	1.5400
− 80	2.74	*24.3	44.73	86.54	−42.2	581.2	−0.1057	1.5368
− 79	2.85	24.1	44.68	83.50	−41.2	581.6	− .1030	1.5336
− 78	2.96	23.9	44.64	80.61	−40.1	582.1	− .1002	1.5304
− 77	3.07	23.6	44.60	77.90	−39.1	582.5	− .0975	1.5273
− 76	3.19	23.4	44.56	75.30	−38.0	582.9	− .0947	1.5242
− 75	3.30	*23.2	44.52	72.80	−37.0	583.3	−0.0920	1.5211
− 74	3.43	22.9	44.48	70.35	−35.9	583.8	− .0892	1.5180
− 73	3.56	22.7	44.44	68.01	−34.9	584.2	− .0865	1.5149
− 72	3.69	22.4	44.40	65.78	−33.8	584.6	− .0838	1.5119
− 71	3.82	22.2	44.36	63.70	−32.8	585.0	− .0811	1.5089
− 70	3.94	*21.9	44.32	61.65	−31.7	585.5	−0.0784	1.5059
− 69	4.09	21.6	44.28	59.60	−30.7	585.9	− .0757	1.5029
− 68	4.24	21.3	44.24	57.64	−29.6	586.3	− .0730	1.4999
− 67	4.39	21.0	44.19	55.78	−28.6	586.7	− .0703	1.4970
− 66	4.54	20.7	44.15	54.01	−27.5	587.1	− .0676	1.4940
− 65	4.69	*20.4	44.11	52.34	−26.5	587.5	−0.0650	1.4911
− 64	4.86	20.1	44.07	50.79	−25.4	588.0	− .0623	1.4883
− 63	5.03	19.6	44.03	49.26	−24.4	588.4	− .0596	1.4854
− 62	5.20	19.3	43.99	47.74	−23.3	588.8	− .0570	1.4826
− 61	5.38	18.9	43.95	46.23	−22.2	589.2	− .0543	1.4797
− 60	5.55	*18.6	43.91	44.73	−21.2	589.6	−0.0517	1.4769
− 59	5.74	18.2	43.87	43.37	−20.1	590.0	− .0490	1.4741
− 58	5.93	17.8	43.83	42.05	−19.1	590.4	− .0464	1.4713
− 57	6.13	17.4	43.78	40.79	−18.0	590.8	− .0438	1.4686
− 56	6.33	17.0	43.74	39.56	−17.0	591.2	− .0412	1.4658
− 55	6.54	*16.6	43.70	38.38	−15.9	591.6	−0.0386	1.4631
− 54	6.75	16.2	43.66	37.24	−14.8	592.1	− .0360	1.4604
− 53	6.97	15.7	43.62	36.15	−13.8	592.4	− .0334	1.4577
− 52	7.20	15.3	43.58	35.09	−12.7	592.9	.0307	1.4551
− 51	7.43	14.8	43.54	34.06	−11.7	593.2	.0281	1.4524

* Inches of mercury below one standard atmosphere (29.92 in.)

U. S. Dept. of Commerce, Bureau of Standards, Thermodynamic Properties of Ammonia, Circular No. 142 (1923) and Circular No. 472 (1948).

TABLE 16-6 (*Continued*)

Temp F	Pressure		Liquid, density	Vapor, sp vol	Enthalpy, datum −40 F Btu per lb		Entropy, datum −40 F Btu per lb F	
t	psia	psig	lb/cu ft $1/v_f$	cu ft/lb v_g	Liquid h_f	Vapor h_g	Liquid s_f	Vapor s_g
−50	7.67	*14.3	43.49	33.08	−10.6	593.7	−0.0256	1.4497
−49	7.91	13.8	43.45	32.12	−9.6	594.0	.0230	1.4471
−48	8.16	13.3	43.41	31.20	−8.5	594.4	.0204	1.4445
−47	8.42	12.8	43.37	30.31	−7.4	594.9	.0179	1.4419
−46	8.68	12.2	43.33	29.45	−6.4	595.2	.0178	1.4393
−45	8.95	*11.7	43.28	28.62	−5.3	595.6	.0127	1.4368
−44	9.23	11.1	43.24	27.82	−4.3	596.0		1.4342
−43	9.51	10.6	43.20	27.04	−3.2			1.4317
−42	9.81	10.0	43.16	26.29	−2.1			1.4292
−41	10.10	9.3	43.12	25.56	−1.1			1.4267
−40	10.41	*8.7	43.08	24.86	0.0	597.6	0.0000	1.4242
−39	10.72	8.1	43.04	24.18	1.1	598.0	.0025	1.4217
−38	11.04	7.4	42.99	23.53	2.1	598.3	.0051	1.4193
−37	11.37	6.8	42.95	22.89	3.2	598.7	.0076	1.4169
−36	11.71	6.1	42.90	22.27	4.3	599.1	.0101	1.4144
−35	12.05	*5.4	42.86	21.68	5.3	599.5	0.0126	1.4120
−34	12.41	4.7	42.82	21.10	6.4	599.9	.0151	1.4096
−33	12.77	3.9	42.78	20.54	7.4	600.2	.0176	1.4072
−32	13.14	3.2	42.73	20.00	8.5	600.6	.0201	1.4048
−31	13.52	2.4	42.69	19.48	9.6	601.0	.0226	1.4025
−30	13.90	*1.6	42.65	18.97	10.7	601.4	0.0250	1.4001
−29	14.30	0.8	42.61	18.48	11.7	601.7	.0275	1.3978
−28	14.71	0.0	42.57	18.00	12.8	602.1	.0300	1.3955
−27	15.12	0.4	42.54	17.54	13.9	602.5	.0325	1.3932
−26	15.55	0.8	42.48	17.09	14.9	602.8	.0350	1.3909
−25	15.98	1.3	42.44	16.66	16.0	603.2	0.0374	1.3886
−24	16.42	1.7	42.40	16.24	17.1	603.6	.0399	1.3863
−23	16.88	2.2	42.35	15.83	18.1	603.9	.0423	1.3840
−22	17.34	2.6	42.31	15.43	19.2	604.3	.0488	1.3818
−21	17.81	3.1	42.26	15.05	20.3	604.6	.0472	1.3796
−20	18.30	3.6	42.22	14.68	21.4	605.0	0.0497	1.3774
−19	18.79	4.1	42.18	14.32	22.4	605.3	.0521	1.3752
−18	19.30	4.6	42.13	13.97	23.5	605.7	.0545	1.3729
−17	19.81	5.1	42.09	13.62	24.6	606.1	.0570	1.3708
−16	20.34	5.6	42.04	13.29	25.6	606.4	.0594	1.3686
−15	20.88	6.2	42.00	12.97	26.7	606.7	0.0618	1.3664
−14	21.43	6.7	41.96	12.66	27.8	607.1	.0642	1.3643
−13	21.99	7.8	41.91	12.36	28.9	607.5	.0666	1.3621
−12	22.56	7.9	41.87	12.06	30.0	607.8	.0690	1.3600
−11	23.15	8.5	41.82	11.78	31.0	608.1	.0714	1.3579
−10	23.74	9.0	41.78	11.50	32.1	608.5	0.0738	1.3558
−9	24.35	9.7	41.74	11.23	33.2	608.8	.0762	1.3537
−8	24.97	10.3	41.69	10.97	34.3	609.2	.0768	1.3516
−7	25.61	10.9	41.65	10.71	35.4	609.5	.0809	1.3495
−6	26.26	11.6	41.60	10.47	36.4	609.8	.0833	1.3474
−5	26.92	12.2	41.56	10.23	37.5	610.1	0.0857	1.3454
−4	27.59	12.9	41.52	9.991	38.6	610.5	.0880	1.3433
−3	28.28	13.6	41.47	9.763	39.7	610.8	.0904	1.3413
−2	28.98	14.3	41.43	9.541	40.7	611.1	.0928	1.3393
−1	29.69	15.0	41.38	9.326	41.8	611.4	−.0951	1.3372
0	30.42	15.7	41.34	9.116	42.9	611.8	0.0975	1.3352
1	31.16	16.5	41.29	8.912	44.0	612.1	0.0998	1.3332
2	31.92	17.2	41.25	8.714	45.1	612.4	.1022	1.3312
3	32.69	18.0	41.20	8.521	46.2	612.7	.1045	1.3292
4	33.47	18.8	41.16	8.333	47.2	613.0	.1069	1.3273
5	34.27	19.6	41.11	8.150	48.3	613.3	.1092	1.3253
6	35.09	20.4	41.07	7.971	49.4	613.6	0.1115	1.3234
7	35.92	21.2	41.01	7.798	50.5	613.9	.1138	1.3214
8	36.77	22.1	40.98	8.629	51.6	614.3	.1162	1.3195
9	37.63	22.9	40.93	7.464	52.7	614.6	.1185	1.3176
10	38.51	23.8	40.89	7.304	53.8	614.9	.1208	1.3157
11	39.40	24.7	40.84	7.148	54.9	615.2	0.1231	1.3137
12	40.31	25.6	40.80	6.996	56.0	615.5	.1254	1.3118
13	41.24	26.5	40.75	6.847	57.1	615.8	.1277	1.3099
14	42.18	27.5	40.71	6.703	58.2	616.1	.1300	1.3081
15	43.14	28.4	40.66	6.562	59.2	616.3	.1323	1.3062

* Inches of mercury below one standard atmosphere (29.92 in.).

TABLE 16-6 (*Continued*)

Temp F t	Pressure		Liquid, density	Vapor, sp vol	Enthalpy, datum −40 F Btu per lb		Entropy, datum −40 F Btu per lb F	
	psia	psig	lb/cu ft $1/v_f$	cu ft/lb v_g	Liquid h_f	Vapor h_g	Liquid s_f	Vapor s_g
16	44.12	29.4	40.61	6.425	60.3	616.6	0.1346	1.3043
17	45.12	30.4	40.57	6.291	61.4	616.9	.1369	1.3025
18	46.13	31.4	40.52	6.161	62.5	617.2	.1392	1.3006
19	47.16	32.5	40.48	6.034	63.6	617.5	.1415	1.2988
20	48.21	33.5	40.43	5.910	64.7	617.8	.1437	1.2969
21	49.28	34.6	40.38	5.789	65.8	618.0	0.1460	1.2951
22	50.36	35.7	40.34	5.671	66.9	618.3	.1483	1.2933
23	51.47	36.8	40.29	5.556	68.0	618.6	.1505	1.2951
24	52.59	37.9	40.25	5.443	69.1	618.9	.1528	1.2897
25	53.78	39.0	40.20	5.334	70.2	619.1	.1551	1.2879
26	54.90	40.2	40.15	5.227	71.3	619.4	0.1573	1.2861
27	56.08	41.4	40.10	5.123	72.4	619.7	.1596	1.2843
28	57.28	42.6	40.06	5.021	73.5	619.9	.1618	1.2825
29	58.50	43.8	40.01	4.922	74.6	620.2	.1641	1.2808
30	59.74	45.0	39.96	4.825	75.7	620.5	.1663	1.2790
31	61.00	46.3	39.91	4.730	76.8	620.7	0.1686	1.2773
32	62.29	47.6	39.86	4.637	77.9	621.0	.1708	1.2755
33	63.59	48.9	39.82	4.547	79.0	621.2	.1730	1.2738
34	64.91	50.2	39.77	4.459	80.1	621.5	.1753	1.2731
35	66.26	52.6	39.72	4.373	81.2	621.7	.1775	1.2704
36	67.63	52.9	39.67	4.289	82.3	622.0	0.1797	1.2686
37	69.02	54.3	39.63	4.207	83.4	622.2	.1819	1.2669
38	70.43	55.7	39.58	4.126	84.6	622.5	.1841	1.2652
39	71.87	57.2	39.54	4.048	85.7	622.7	.1863	1.2635
40	73.32	58.6	39.49	3.971	86.8	623.0	.1885	1.2618
41	74.80	60.1	39.44	3.897	87.9	623.2	0.1908	1.2602
42	76.31	61.6	39.39	3.823	89.0	623.4	.1930	1.2585
43	77.83	63.1	39.34	3.752	90.1	623.7	.1952	1.2568
44	79.38	64.7	39.29	3.682	91.2	623.9	.1974	1.2552
45	80.96	66.3	39.24	3.614	92.3	624.1	.1996	1.2535
46	82.55	67.9	39.19	3.547	93.5	624.4	0.2018	1.2519
47	84.18	69.5	39.14	3.481	94.6	624.6	.2040	1.2502
48	85.82	71.1	39.10	3.418	95.7	624.8	.2062	1.2486
49	87.49	72.8	39.05	3.355	96.8	625.0	.2083	1.2469
50	89.19	74.5	39.00	3.294	97.9	625.2	.2105	1.2453
51	90.91	76.2	38.95	3.234	99.1	625.5	0.2127	1.2437
52	92.66	78.0	38.90	3.176	100.2	625.7	.2149	1.2421
53	94.43	79.7	38.85	3.119	101.3	625.9	.2171	1.2405
54	96.23	81.5	38.80	3.063	102.4	626.1	.2192	1.2389
55	98.06	83.4	38.75	3.008	103.5	626.3	.3214	1.2373
56	99.91	85.2	38.70	2.954	104.7	626.5	0.2236	1.2357
57	101.8	87.1	38.65	2.902	105.8	626.7	.2257	1.2341
58	103.7	89.0	38.60	2.851	106.9	626.9	.2279	1.2325
59	105.6	90.9	38.55	2.800	108.1	627.1	.2301	1.2310
60	107.6	92.9	38.50	2.751	109.2	627.3	.2322	1.2294
61	109.6	94.9	38.45	2.703	110.3	627.5	0.2344	1.2278
62	111.6	96.9	38.40	2.656	111.5	627.7	.2365	1.2262
63	113.6	98.9	38.35	2.610	112.6	627.9	.2387	1.2247
64	115.7	101.0	38.30	2.565	113.7	628.0	.2408	1.2231
65	117.8	103.1	38.25	2.520	114.8	628.2	.2430	1.2216
66	120.0	105.3	38.20	2.477	116.0	628.4	0.2451	1.2201
67	122.1	107.4	38.15	2.435	117.1	628.6	.2473	1.2186
68	124.3	109.6	38.10	2.393	118.3	628.8	.2494	1.2170
69	126.5	111.8	38.05	2.352	119.4	628.9	.2515	1.2155
70	128.8	114.1	38.00	2.312	120.5	629.1	.2537	1.2140
71	131.1	116.4	37.95	2.273	121.7	629.3	0.2558	1.2125
72	133.4	118.7	37.90	2.235	122.8	629.4	.2579	1.2110
73	135.7	121.0	37.84	2.197	124.0	629.6	.2601	1.2095
74	138.1	123.4	37.79	2.161	125.1	629.8	.2622	1.2080
75	140.5	125.8	37.74	2.125	126.2	629.9	.2643	1.2065
76	143.0	128.3	37.69	2.089	127.4	630.1	0.2664	1.2050
77	145.4	130.7	37.64	2.055	128.5	630.2	.2685	1.2035
78	147.9	133.2	37.58	2.021	129.7	630.4	.2706	1.2020
79	150.5	135.8	37.53	1.988	130.8	630.5	.2728	1.2006
80	153.0	138.3	37.48	1.955	132.0	630.7	.2749	1.1991

TABLE 16-6 (*Continued*)

Temp F t	Pressure		Liquid, density	Vapor, sp vol	Enthalpy, datum −40 F Btu per lb		Entropy, datum −40 F Btu per lb F	
	psia	psig	lb/cu ft $1/v_f$	cu ft/lb v_g	Liquid h_f	Vapor h_g	Liquid s_f	Vapor s_g
81	155.6	140.9	37.43	1.923	133.1	630.8	0.2769	1.1976
82	158.3	143.6	37.37	1.892	134.3	631.0	.2791	1.1962
83	161.0	146.3	37.32	1.861	135.4	631.1	.2812	1.1947
84	163.6	149.0	37.26	1.831	136.6	631.3	.2833	1.1933
85	166.4	151.7	37.21	1.801	137.8	631.4	.2854	1.1918
86	169.2	154.5	37.16	1.772	138.9	631.5	0.2875	1.1904
87	172.0	157.3	37.11	1.744	140.1	631.7	.2895	1.1889
88	174.8	160.1	37.05	1.716	141.2	631.8	.2917	1.1875
89	177.7	163.0	37.00	1.688	142.4	631.9	.2937	1.1860
90	180.6	165.9	36.95	1.661	143.5	632.0	.2958	1.1846
91	183.6	168.9	36.89	1.635	144.7	632.1	0.2979	1.1832
92	186.6	171.9	36.84	1.609	145.8	632.3	.3000	1.1818
93	189.6	174.9	36.78	1.584	147.0	632.3	.3021	1.1804
94	192.7	178.0	36.73	1.559	148.2	632.5	.3041	1.1789
95	195.8	181.1	36.67	1.534	149.4	632.6	.3062	1.1775
96	198.9	184.2	36.62	1.510	150.5	632.6	0.3083	1.1761
97	202.1	187.4	36.56	1.487	151.7	632.8	.3104	1.1747
98	205.3	190.6	36.51	1.464	152.9	632.9	.3125	1.1733
99	208.6	193.9	36.45	1.441	154.0	632.9	.3145	1.1719
100	211.9	197.2	36.40	1.419	155.2	633.0	.3166	1.1705
101	215.2	200.5	36.34	1.397	156.4	633.1	0.3187	1.1691
102	218.6	203.9	36.29	1.375	157.6	633.2	.3207	1.1677
103	222.0	207.3	36.23	1.354	158.7	633.3	.3228	1.1663
104	224.4	210.7	36.18	1.334	159.9	633.4	.3248	1.1649
105	228.9	214.2	36.12	1.313	161.1	633.4	.3269	1.1635
106	232.5	217.8	36.06	1.293	162.3	633.5	0.3289	1.1621
107	236.0	221.3	36.01	1.274	163.5	633.6	.3310	1.1607
108	239.7	225.0	35.95	1.254	164.6	633.6	.3330	1.1593
109	243.3	228.6	35.90	1.235	165.8	633.7	.3351	1.1580
110	247.0	232.3	35.84	1.217	167.0	633.7	.3372	1.1566
111	250.8	236.1	35.78	1.198	168.2	633.8	0.3392	1.1552
112	354.5	239.8	35.72	1.180	169.4	633.8	.3413	1.1538
113	258.4	243.7	35.67	1.163	170.6	633.9	.3433	1.1524
114	262.2	247.5	35.61	1.145	171.8	633.9	.3453	1.1510
115	266.2	251.5	35.55	1.128	173.0	633.9	.3474	1.1497
116	270.1	255.4	35.49	1.112	174.2	634.0	0.3495	1.1483
117	274.1	259.4	35.43	1.095	175.4	634.0	.3515	1.1469
118	278.2	263.5	35.38	1.079	176.6	634.0	.3515	1.1455
119	282.3	267.6	35.32	1.063	177.8	634.0	.3556	1.1441
120	286.4	271.7	35.26	1.047	179.0	634.0	.3576	1.1427
121	290.6	275.9	35.20	1.032	180.2	634.0	0.3597	1.1414
122	294.8	280.1	35.14	1.017	181.4	634.0	.3618	1.1400
123	299.1	284.4	35.08	1.002	182.6	634.0	.3638	1.1386
124	303.4	288.7	35.02	0.987	183.9	634.0	.3659	1.1372
125	307.8	293.1	34.96	0.973	185.1	634.0	.3679	1.1358

TABLE 16-7. Relative Safety of Refrigerants

Refrigerant	ASAB9 Safety Code Group Classification	Nat'l Fire Underwriters Group Number	Toxicity, Lethal or Serious Injury[3]: Refrigerant in Air: Duration of Exposure (hr)	Refrigerant in Air: % by Vol	Refrigerant in Air: lb/1000 cu ft	Products of Decomposition by Flame: Duration of Exposure (min)	Products of Decomposition by Flame: % by Vol[4]	Flammable or Explosive Limits of Concentration in Air: % by Vol
Methane	3[1]	+5						4.9–15.0
R-14	1[1]	6[1]						Nonflam.
Ethylene	3[1]	+5						3.0–25.0
Nitrous oxide			8	0.0025				Nonflam.
R-13	1[1]	6[1]						Nonflam.
Ethane	3	5	2	37.4–51.7				3.3–10.6
Carbon dioxide	1	5	½ to 1	29–30	33.2–34.3			Nonflam.
Kulene-131	1[1]	6[1]						Nonflam.
Propane	3	5	2	37.5–51.7	42.4–58.5			2.3–7.3
R-22	1	5A				16	1.0	Nonflam.[2]
Ammonia	2	2	½	0.5–0.6	0.221–0.256			16.0–25.0
Carrene-7	1	5A	2	19.4–20.3	50.2–52.2	25	1.1	Nonflam.
R-12	1	6	2	28.5–30.4	89.6–95.7	20	1.0	Nonflam.
Methyl chloride	2	4	2	2–2.5	2.62–3.28	30	2.4	8.1–17.2
Isobutane	3	+5						1.8–8.4
Sulfur dioxide	2	1	$\frac{1}{12}$	0.7	1.165			Nonflam.
Butane	3	5	2	37.5–51.7				1.6–6.5
R-114	1	6	2	20.1–21.5	90.5–96.8	15	1.0	Nonflam.
R-21	1		½	10.2	27.1			Nonflam.
Ethyl chloride	2	4	1	4.0	6.72	18	2.0	3.7–12.0
R-11	1	5	2	10	35.7	5	1.0	Nonflam.
Methyl formate	2	3	1	2–2.5	3.12–3.9			4.5–20.0
Methylene chloride	1	4A	½	5.1–5.3	11.25–11.7	20	1.0	Nonflam.
R-113	1	4	1	4.8–5.2	23.3–25.2	16	1.2	Nonflam.
Dichlorethylene	2	4	1	2–2.5	5.04–6.3	5	2.1	5.6–11.4

[1] Unofficial.
[2] Very slightly flammable, but for practical purposes considered nonflammable.
[3] To guinea pigs.
[4] Initial concentration.
From the *ASRE Data Book*, Design Volume, 1957–58 Edition, by permission of the American Society of Heating Refrigerating, and Air-Conditioning Engineers.

TABLE 16-8. Comparative Refrigerant Characteristics Performance Based on 5 F Evaporation and 86 F Condensation

Refrigerant	Chemical symbol	Molecular weight	Boiling temperature at 0 psig, F	Freezing temperature at 0 psig, F	Critical temperature, F	Critical pressure, psia	Evaporator pressure at 5 F, psig	Condensing pressure at 86 F, psig	Ratio of compression at 86 F, 5 F (psia/psia)	Net refrigerating effect of liquid 86 F, 5 F, Btu/lb	Refrigerant circulated per ton, lb/min	Liquid circulated per ton 86 F, 5 F, cu in./min	Specific volume of vapor 5 F, cu ft/lb	Compressor displacement per ton 86 F, 5 F, cfm	Horsepower per ton 86 F, 5 F, hp	Coefficient of performance 86 F, 5 F	Temperature of compressor discharge, F
Air		28.95	−318.0		−221.0	547.0	0	58.8	5.00	28.5	7.02		11.7	82.3	2.82	1.67	
Methane	CH_4	16.03	−258.9	−297.0	−115.8	673.0	‡										
Refrigerant 14	CF_4	88.00	−198.2	−312.0	− 49.9	542.4	‡										
Ethylene	C_2H_4	28.03	−155.0	−272.0	48.8	731.8	399.8	‡									
Ethane	C_2H_6	30.04	−127.5	−278.0	90.1	708.3	221.3	661.1	2.86	58.6	3.41	342.9	0.53	1.82	1.953	2.41	122
Nitrous Oxide	N_2O	46.01	−127.0	−128.0	96.5	1050.0	294.3	922.3	3.03	85.2	2.35	71.2	0.28	0.66	1.310	3.60	
Refrigerant 13	$CClF_3$	104.40	−114.5	−296.0	83.9	561.3	177.1	‡			0						
Carbon Dioxide	CO_2	44.00	−109.3	− 69.9	87.8	1071.1	317.5	1031.0	3.15	55.5	3.62	167.1	0.27	0.96	1.840	2.56	151
Refrigerant 13B1	CF_3Br	148.93	− 73.6	−226.0	153.5	587.0	63.2	247.1	3.36	29.3	6.86	123.8	0.38	2.63	1.030	4.25	124
Propylene	C_3H_6	42.08	− 53.7	−301.0	196.5	667.2	37.0	167.0	3.51	173.0	1.1	61.5	2.61	3.03	1.046	4.51	108
Propane	C_3H_8	44.06	− 44.2	−309.8	202.0	661.5	27.2	140.5	3.70	121.0	1.65	94.0	2.48	4.09	1.030	4.58	97
Refrigerant 22	$CHClF_2$	86.48	− 41.4	−256.0	204.8	716.0	28.3	159.8	4.06	69.3	2.89	68.0	1.25	3.60	1.011	4.66	131
Ammonia	NH_3	17.03	− 28.0	−107.9	271.4	1657.0	19.6	154.5	4.94	474.4	0.422	19.6	8.15	3.44	0.989	4.76	210
Refrigerant 500	†	99.29	− 28.0	−254.0	221.1	631.0	16.4	113.4	4.12	61.1	3.27	79.3	1.52	4.97	1.022	4.61	105
Refrigerant 12	CCl_2F_2	120.9	− 21.6	−252.0	232.7	582.0	11 8	93.2	4.07	51.1	3.92	83.9	1.49	5.81	1.002	4.70	100
Methyl Chloride	CH_3Cl	50.48	− 10.8	−144.0	289.4	968.7	6.5	80.0	4.48	150.2	1.33	40.9	4.47	5.95	0.962	4.90	172
Isobutane	C_4H_{10}	58.12	10.3	−229.0	272.7	537 0	3 3*	44.8	4.54	111.5	1.79	91.0	6.41	11.50	1.083	4.36	80
Sulfur Dioxide	SO_2	64.06	14.0	−103.9	314.8	1141.5	5.9*	51.8	5.63	141.4	1.41	26.6	6.42	9.09	0.968	4.87	191
Methylamine	CH_5N	31.05	20.3	−134.5	314.0	1082.0	9.9*	46.8	6.13	304.0	0.66	28.2	15.54	10.23	0.978	4.81	
Butane	C_4H_{10}	58.12	31.3	−211.0	306.0	550.1	13.2*	26.9	5.07	128.6	1.56	75.9	9.98	15.52	0.953	4.95	88
Refrigerant 114	$C_2Cl_2F_4$	170.93	38.4	−137.0	294.3	474.0	16.1*	22.0	5.42	43.1	4.64	89.2	4.22	19.59	1.015	4.64	86
Refrigerant 21	$CHCl_2F$	102.93	48.0	−211.0	353.3	750.0	19.2*	16.5	5.96	89.4	2.24	45.7	9.13	20.43	0.943	5.05	142
Ethyl Chloride	C_2H_5Cl	64.52	54.0	−217.7	369.0	764.0	20.5*	12.4	5.83	142.3	1.45	45.8	17.06	24.82	0.906	5.21	106
Ethylamine	C_2H_7N	45.06	61.8	−115.0	362.0	815.0	23.1*	10.0	7.40	225.5	0.89	349.0	32.32	38.67	0.855	5.52	
Refrigerant 11	CCl_3F	137.38	74.7	−168.0	388.4	635.0	24.0*	3.6	6.24	67.5	2.96	56.0	12.27	36.32	0.927	5.09	
Methyl Formate	$C_2H_4O_2$	60.04	89.2	−147.5	418.0	607.0	26.3*	1.6*	7.74	189.2	1.06	29.9	48.25	51.00			
Ethyl Ether	$C_4H_{10}O$	74.08	94.3	−177.3	522.1	380.8	26.9*	4.9*	8.20	126.3	1.58	62.9	35.00	55.40	0.822	5.74	
Methylene Chloride	CH_2Cl_2	84.93	105.2	−142.0	480.0	670.0	27.6*	9.5*	8.60	134.6	1.49	30.9	49.90	74.30	0.963	4.90	205
Refrigerant 113	$C_2Cl_3F_3$	187.39	117.6	− 31.0	417.4	495.0	27.9*	13.9*	8.02	53.7	3.73	66.5	27.04	100.76	0.960	4.92	86
Dichloroethylene	$C_2H_2Cl_2$	96.93	118.0	− 70.0	470.0	795.0	28.3*	15.8*	8.42	114.3	1.75	38.3	63.60	111.20	0.973	4.83	
Trichloroethylene	C_2HCl_3	131.37	187.0	−124.3	520.0	728.0	29.6*	26.2*	11.65	91.7	2.18	41.6	229.40	502.00	0.980	4.82	
Water (40 F and 86 F)	H_2O	18.02	212.0	32.0	706.1	3226.0	29.7*	28.6*	5.06	1025.3	0.195	5.5	12444.40	476.70	1.125	4.10	282

* Inches of Mercury Vacuum.
† Azeotropic Mixture of Genetron-100 (CH_3CHF_2) and Freon-12 (CCl_2F_2).
‡ Above Critical.

From *ASRE Data Book*, Design Volume, 1957–1958 Edition, by permission of the American Society of Heating, Refrigerating, and Air-Conditioning Engineers.

TABLE 19-1. Dimensions and Physical Data—Copper, Brass, or Seamless Steel Tubing

Size in Inches OD	Nominal Pipe Size Inches	External Dia. Inches	Type	Internal Dia. Inches	Thickness of Metal Inches	Transverse Area Square Inches		Lin. Ft. of Pipe Per Square Foot of		Length of Pipe in Ft. Containing 1-Cu. Foot	Length of Pipe in Ft. Containing 1-Gallon	Length of Pipe in Ft. Occupying 1-Cu. Ft. of Space	Weight Per Foot Pounds
						External	Internal	External Surface	Internal Surface				
1/4	1/8	.250	—	.190	.030**	.049	.028	15.25	20.00	5090.0	681.0	2940.0	.080
3/8	1/4	.375	K	.311	.032**	.110	.076	10.45	12.29	1895.0	253.0	1310.0	.134
1/2	3/8	.500	K	.402	.049	.196	.127	7.65	9.50	1135.0	151.0	735.0	.269
			L	.430	.035		.144		8.89	1001.0	133.5		.198
5/8	1/2	.625	K	.527	.049	.306	.218	6.10	7.25	660.5	88.0	470.0	.344
			L	.545	.040		.232		7.00	621.0	82.6		.284
3/4	5/8	.750	K	.652	.049	.539	.333	5.10	5.85	432.5	57.5	267.0	.418
			L	.660	.042		.341		5.79	422.0	56.1		.362
7/8	3/4	.875	K	.745	.065	.598	.435	4.36	5.12	331.0	44.0	240.5	.641
			L	.785	.045		.482		4.86	299.0	39.8		.454
1 1/8	1	1.125	K	.995	.065	.989	.775	3.39	3.84	186.0	24.7	145.9	.839
			L	1.025	.050		.825		3.72	174.7	23.2		.653
1 3/8	1 1/4	1.375	K	1.245	.065	1.481	1.215	2.78	3.06	118.9	15.8	97.3	1.04
			L	1.265	.055		1.255		3.02	115.0	15.3		.882
1 5/8	1 1/2	1.625	K	1.481	.072	2.070	1.725	2.35	2.57	83.5	11.1	69.6	1.36
			L	1.505	.060		1.771		2.54	81.4	10.8		1.14
2 1/8	2	2.125	K	1.959	.083	3.540	3.000	1.80	1.95	48.0	6.39	40.6	2.06
			L	1.985	.070		3.090		1.92	46.6	6.20		1.75
2 5/8	2 1/2	2.625	K	2.435	.095	5.400	4.620	1.45	1.57	31.2	4.15	27.6	2.92
			L	2.465	.080		4.760		1.55	30.2	4.01		2.48
3 1/8	3	3.125	K	2.907	.109	7.750	6.620	1.22	1.31	21.8	2.90	18.6	4.00
			L	2.945	.090		6.810		1.29	21.1	2.80		3.33
3 5/8	3 1/2	3.625	K	3.385	.120	10.350	8.96	1.05	1.13	16.1	2.14	13.9	5.12
			L	3.425	.100		9.21		1.11	15.6	2.07		4.29
4 1/8	4	4.125	K	3.857	.134	13.320	11.620	.93	.99	12.4	1.65	7.50	6.51
			L	3.905	.110		11.920		.98	12.1	1.61		5.38
5 1/8	5	5.125	K	4.805	.160	20.530	18.100	.75	.79	7.95	1.06	7.04	9.67
			L	4.875	.125		18.600		.78	7.75	1.03		7.61
6 1/8	6	6.125	K	5.741	.192	29.400	25.80	.62	.67	5.59	.74	4.90	13.87
			L	5.845	.140		26.61		.66	5.41	.72		10.20
8 1/8	8	8.125	K	7.583	.271	51.700	44.80	.47	.50	3.22	.43	2.78	25.90
			L	7.725	.200		46.60		.49	3.09	.41		19.29

Courtesy York Corporation.

TABLE 19-2. Refrigerant Line Capacities in Tons for Refrigerant-12 (Single or High Stage Applications)

Tons of Refrigeration Resulting in a Line Friction Drop per 100 Ft Equivalent Pipe Length Corresponding to 2 F (ΔT) Change in Saturation Temp

Line Size Type L Copper OD	Suction Lines* Suction Temp F −40 $\Delta P = 0.49$	−20 $\Delta P = 0.72$	0 $\Delta P = 1.01$	20 $\Delta P = 1.38$	40 $\Delta P = 1.82$	Discharge Lines* $\Delta P = 3.66$ Sat. −40	Suct. 0	Temp F 40	Liquid Lines Line Size Type L Copper OD	Condenser to Receiver Velocity = 100 fpm	Receiver* to System $\Delta T = 1$ F $\Delta P = 1.8$ psi
1/2				0.21	0.31	0.46	0.54	0.67	1/2	1.16	2.03
5/8		0.17	0.26	0.40	0.58	0.85	0.98	1.23	5/8	2.65	3.81
7/8	0.25	0.42	0.68	1.04	1.50	2.23	2.58	3.22	7/8	6.94	10.10
1 1/8	0.51	0.87	1.39	2.10	3.10	4.60	5.30	6.65	1 1/8	11.85	20.5
1 3/8	0.87	1.52	2.40	3.70	5.36	7.8	9.0	11.3	1 3/8	18.10	35.1
1 5/8	1.41	2.44	3.86	5.82	8.50	12.4	14.4	18.0	1 5/8	25.5	57.5
2 1/8	2.94	5.03	8.00	12.1	17.6	25.8	30.0	37.4	2 1/8	44.4	117.8
2 5/8	5.20	8.94	14.2	21.3	31.4	45.5	52.5	66.0	2 5/8	68.4	207.8
3 1/8	8.35	14.3	22.7	34.0	49.5	73.0	85.0	106.0	3 1/8	97.5	344.0
3 5/8	12.4	21.2	33.8	50.6	73.5	107.0	124.0	155.0	3 5/8	132.0	508.0
4 1/8	17.4	29.9	47.7	71.0	103.0	152.0	176.0	220.0	4 1/8	173.0	704.0
5 1/8	31.7	54.0	85.3	128.0	187.0	270.0	314.0	392.0			
6 1/8	50.8	86.0	137.0	206.0	299.0	428.0	494.0	620.0			

Steel IPS	SCH	−40	−20	0	20	40	Sat. −40	Suct. 0	Temp F 40	Steel IPS	SCH	Condenser to Receiver	Receiver* to System
1/2	40			0.30	0.45	0.64	0.92	1.07	1.34	1/2	80	3.43	3.23
3/4	40	0.24	0.41	0.64	0.96	1.39	1.96	2.26	2.83	3/4	80	6.25	7.27
1	40	0.46	0.78	1.22	1.82	2.68	3.75	4.35	5.42	1	80	10.4	14.3
1 1/4	40	0.97	1.60	2.52	3.78	5.41	7.8	9.0	11.3	1 1/4	80	18.6	30.1
1 1/2	40	1.50	2.41	3.76	5.62	8.12	11.4	13.2	16.5	1 1/2	80	25.5	47.3
2	40	2.81	4.69	7.40	10.9	15.7	21.6	25.1	31.4	2	40	48.0	111.9
2 1/2	40	4.44	7.42	11.6	17.3	24.7	34.7	40.2	50.2	2 1/2	40	68.3	173.0
3	40	8.04	13.2	20.6	30.6	43.8	61.0	70.8	88.4	3	40	104.0	311.8
4	40	16.03	27.0	42.8	62.9	90.2	125.0	146.0	182.0	4	40	179.0	634.0
5	40	30.0	49.1	78.7	114.0	165.0	228.0	264.0	330.0				
6	40	48.2	78.6	124.0	182.0	268.0	365.0	421.0	528.0				
8	48	98.4	161.0	254.0	376.0	541.0	745.0	865.0	1080.0				
10	40	180.0	297.0	458.0	678.0	972.0	1350.0	1570.0	1960.0				
12	ID	286.0	475.0	729.0	1080.0	1520.0	2130.0	2460.0	3090.0				

NOTES:

* (1) *Basis of Table: 100 F Condensing Temp, 2 F ΔT per 100 ft Equivalent Length (except liquid lines).*

(2) *For other ΔT's and Equivalent Lengths,*

$$\text{Line Capacity (Tons)} = \text{Table Tons} \times \left(50 \times \frac{\text{Actual } \Delta T \text{ loss desired, F}}{\text{Actual equiv. length, ft}}\right)^{0.55}$$

(3) *For other Tons and Equivalent Lengths–*

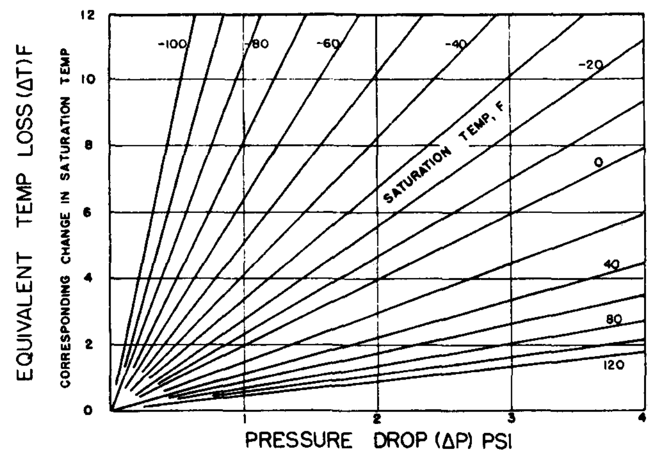
$$\Delta T \text{ for a given pipe size} = \frac{\text{Actual Equiv. Length, ft}}{50} \times \left(\frac{\text{Actual Tons}}{\text{Table Tons}}\right)^{1.8}$$

(4) *Values based on 100 F Condensing Temp. For capacities at other condensing temp. multiply table value by line capacity multiplier below:*

Line	*Condensing Temp, F*					
	80	*90*	*100*	*105*	*110*	*120*
Suction Lines	*1.11*	*1.06*	*1.00*	*0.97*	*0.94*	*0.88*
Discharge Lines	*0.88*	*0.94*	*1.00*	*1.04*	*1.07*	*1.16*

(5) *Tabulated data taken from chapter 9 of the ASRE 1957–58 Design Data Book. Initially developed from ARI preliminary data.*

(6) *For the Equivalent Change in saturation temp for various line pressure drops in psi refer to Fig.*



TEMPERATURE EQUIVALENT (ΔT) OF PRESSURE DROP, F (REFRIGERANT 12)

From *ASRE Data Book*, Design Volume, 1957–58 Edition, by permission of the American Society of Heating, Refrigerating, and Air-Conditioning Engineers.

TABLE 19-3. Refrigerant Line Capacities for Refrigerant-22 (Single or High Stage Applications)

Tons of Refrigeration Resulting in a Line Friction Drop per 100 Ft Equivalent Pipe Length Corresponding to 2 F (ΔT) Change in Saturation Temp

Line Size Type L Copper OD		*Suction Lines**					*Discharge Lines** $\Delta P = 6.1$			*Liquid Lines*			
		Suction Temp, F								*Line Size Type L Copper OD*		*Condenser to Receiver Velocity = 100 fpm*	*Receiver to System $\Delta T = 1$ F*
		−40 $\Delta P = 0.79$	*−20* $\Delta P = 1.15$	*0* $\Delta P = 1.6$	*20* $\Delta P = 2.22$	*40* $\Delta P = 2.93$	*Sat. −40*	*Suct. 0*	*Temp 40*				
1/2					0.40	0.59	1.0	1.1	1.2	1/2		2.24	3.5
5/8			0.32	0.49	0.75	1.10	2.1	2.3	2.5	5/8		3.57	6.4
7/8		0.35	0.87	1.31	2.00	1.89	4.9	5.4	5.2	7/8		7.41	17.0
1 1/8		1.08	1.74	2.65	4.03	5.82	9.8	10.7	11.8	1 1/8		12.7	34.4
1 3/8		1.88	3.01	4.61	7.03	9.98	17.0	18.6	20.5	1 3/8		19.2	60.0
1 5/8		2.90	4.78	7.23	10.26	15.95	26.4	29.0	31.9	1 5/8		27.2	95.0
2 1/8		6.21	9.97	15.2	23.2	33.2	55.0	60.3	66.3	2 1/8		47.3	200.0
2 5/8		10.8	17.5	26.5	40.3	58.1	96.0	105.0	115.6	2 5/8		73.2	354.0
3 1/8		17.3	27.1	43.3	64.5	23.1	155.0	170.0	187.3	3 1/8		104.1	572.0
3 5/8		25.9	41.9	63.9	96.8	139.5	233.0	255.0	281.0	3 5/8		141.1	860.0
4 1/8		36.9	59.2	89.7	136.0	196.0	327.0	358.0	394.0	4 1/8		183.0	1200.0
5 1/8		66.1	106.7	162.0	245.0	355.0	588.0	644.0	709.0				
Steel										*Steel*			
IPS	*SCH*									*IPS*	*SCH*		
1/2	40		0.38	0.56	0.84	1.20	2.0	2.2	2.4	1/2	80	4.66	5.5
3/4	40	0.49	0.78	1.18	1.77	2.52	4.1	4.5	5.0	3/4	80	6.17	12.2
1	40	0.94	1.49	2.24	3.24	4.72	7.8	8.5	9.4	1	80	13.2	24.4
1 1/4	40	1.95	3.01	4.67	6.83	9.73	16.1	17.6	19.4	1 1/4	80	22.9	51.5
1 1/2	40	2.89	4.63	7.04	10.4	14.7	24.1	26.5	29.1	1 1/2	80	37.1	78.0
2	40	5.60	8.90	13.0	19.9	28.2	46.6	51.0	56.2	2	40	51.5	185.0
2 1/2	40	8.90	14.2	21.5	31.9	45.9	74.7	82.0	90.0	2 1/2	40	73.3	297.0
3	40	15.9	25.2	38.0	56.5	80.1	132.0	144.0	159.0	3	40	113.0	510.0
3 1/2	40	23.1	36.1	55.1	81.0	116.0	189.0	207.0	228.0	3 1/2	40	151.5	704.0
4	40	32.1	50.8	76.7	112.7	159.5	260.0	285.0	314.0	4	40	195.0	1060.0
5	40	57.8	91.0	138.6	204.0	292.0	477.0	520.0	575.0				
6	40	94.1	148.6	224.0	329.0	472.0	775.0	850.0	937.0				
8	40	199.0	316.0	474.0	704.0	996.0	1650.0	1810.0	1992.0				
10	40	294.2	550.0	840.0	1226.0	1760.0	2880.0	3150.0	3470.0				
12	ID	555.0	877.0	1340.0	1935.0	2795.0	4640.0	5080.0	5590.0				

NOTES:

* (*1*) *Basis of Table: 105 F Condensing Temperature, 2 F ΔT per 100 ft Equivalent Length (except liquid lines).*

(*2*) *For Other ΔT's and Equivalent Lengths*

Line Capacity (Tons) = Table Tons $\times$

$$\left(50\times\frac{\text{Actual }\Delta T\text{ loss desired, F}}{\text{Actual Equiv. length, ft}}\right)^{0.55}$$

(*3*) *For other Tons and Equivalent Lengths*

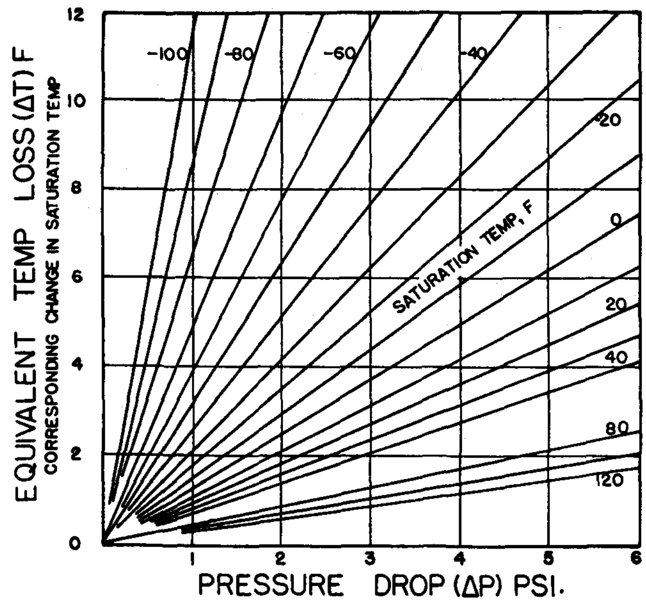
$$\Delta T\text{ for a given pipe size} = \frac{\text{Actual Equiv. Length, ft}}{50}\times\left(\frac{\text{Actual Tons}}{\text{Table Tons}}\right)^{1.8}$$

(*4*) *For the equivalent change in saturation temperature for various line pressure drops in psi, refer to Fig.*

(*5*) *Data developed from preliminary ARI information. Subject to correction.*

(*6*) *For other condensing temperatures, multiply table tons by the following factors:*

Condensing Temp F	*Suction Lines*	*Hot Gas Lines*
80	*1.13*	*0.77*
90	*1.08*	*0.86*
100	*1.03*	*0.95*
110	*0.97*	*1.04*
120	*0.91*	*1.13*



TEMP EQUIV. (ΔT) OF PRESSURE DROP, F
(REFRIGERANT 22)

From *ASRE Data Book*, Design Volume, 1957–58 Edition, by permission of the American Society of Heating, Refrigerating, and Air-Conditioning Engineers.

TABLE 19-4. Refrigerant Line Capacities for Refrigerant-717 (Ammonia) (Single or High Stage Applications)

Tons of Refrigeration Resulting in a Line Friction Drop per 100 Ft Equivalent Pipe Length Corresponding to 1 F (ΔT) Change in Saturation Temp

*Suction Lines**							*Discharge Lines*	*Liquid Lines*			
Line Size		*Suction Temperature F*					$\Delta P = 3.3$	*Line Size*		*Condenser to Receiver Velocity = 100 fpm*	*Receiver to System* $\Delta P = 3.3$
IPS	*SCH*	-40 $\Delta P = 0.32$	-20 $\Delta P = 0.52$	0 $\Delta P = 0.78$	20 $\Delta P = 1.08$	40 $\Delta P = 1.48$		*IPS*	*SCH*		
3/8	80										
1/2	80						3.63	1/2	80	13.5	29.7
3/4	80				2.58	3.75	7.98	3/4	80	24.9	66.7
1	80		2.11	3.46	5.14	7.50	15.9	1	80	41.5	130.0
1 1/4	40	3.24	5.57	8.90	13.4	19.4	41.2	1 1/4	40	86.2	281.0
1 1/2	40	4.83	8.75	13.70	20.2	29.4	57.5	1 1/2	40	117.2	439.0
2	40	9.34	16.4	26.2	39.4	57.3	118.9	2	40	193.5	1004.0
2 1/2	40	15.0	26.0	42.2	62.5	91.2	187.2	2 1/2	40	276.0	1599.0
3	40	26.9	46.0	73.9	111.0	162.0	338.2	3	40	425.0	2341.0
4	40	56.1	94.5	151.0	226.0	327.0	676.0	4	40	736.0	5750.0
5	40	102.0	172.0	272.0	408.0	592.0	1228.0	5	40		
6	40	160.0	280.0	445.0	662.0	958.0	1986.0	6	40		
8	40	338.0	570.0	908.0	1355.0	1960.0	4120.0	8	40		
10	40	605.0	1030.0	1640.0	2430.0	3555.0		10	40		
12	ID	975.0	1660.0	2640.0	3940.0	5680.0		12	ID		

NOTES:

(1) *Basis of Table: 100 F Condensing Temperature, 1 F ΔT per 100 ft equivalent length. Discharge and liquid lines based on 0 F suction.*

(2) *For other ΔT's and Equivalent Lengths,*

$$\text{Line Capacity (tons)} = \text{Table Tons} \times \left(\frac{50 \times \text{Actual } \Delta T \text{ Loss Desired, F}}{\text{Actual Equiv. Length, ft}}\right)^{0.55}$$

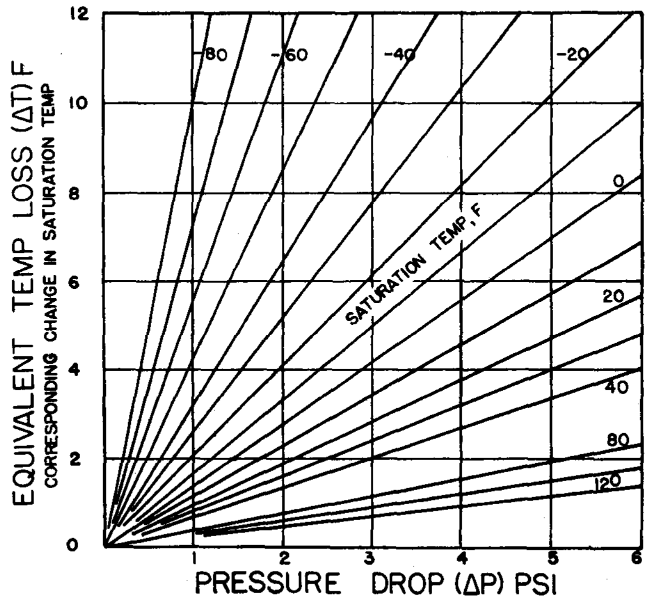
(3) *For other Tons and Equivalent Lengths,*

$$\Delta T \text{ for a given pipe size} = \frac{\text{Actual Equiv. Length, ft}}{50} \times \left(\frac{\text{Actual Tons}}{\text{Table Tons}}\right)^{1.8}$$

(4) *Values based on 100 F condensing temp. For capacities at other condensing temp, multiply table value by line capacity multiplier:*

Line	Condensing Temperature F 70	80	90	100
Suction Lines	*1.0*	*1.0*	*1.0*	*1.0*
Discharge Lines	*0.70*	*0.80*	*0.90*	*1.0*

(5) *For the Equivalent Change in saturation temp for various line pressure drops in psi, refer to Fig.*
* (6) *Taken from chapter 9 of the 1957–58 ASRE Design Data Book. Initially developed from ARI preliminary data.*



TEMPERATURE EQUIVALENT (ΔT) OF PRESSURE DROP (AMMONIA)

From *ASRE Data Book*, Design Volume, 1957–58 Edition, by permission of the American Society of Heating, Refrigeration, and Air-Conditioning Engineers.

TABLE 19-5. Refrigerant Line Capacities for Intermediate or Low Stage Duty (Tons) for Refrigerants 12, 22, and Ammonia

*Refrigerant and ΔT Equivalent of Friction Drop**	*Line Size Type L Copper OD*	*Suction Lines**							*Discharge Lines**	*Line Size Type L Copper OD*	*Liquid Lines*	
		Suction Temp F									*Condenser to Receiver V = 100 fpm*	*Receiver to System**
		−90	*−80*	*−70*	*−60*	*−50*	*−40*	*−30*				
	½									½		2.1
Refrigerant 12	⅝									⅝		3.9
	⅞				0.2	0.3	0.4	0.5	0.9	⅞		11.0
	1⅛	0.17	0.24	0.3	0.4	0.6	0.8	1.0	1.8	1⅛		21.5
	1⅜	0.30	0.42	0.6	0.8	1.1	1.4	1.7	3.2	1⅜		37.0
	1⅝	0.47	0.67	0.9	1.2	1.7	2.2	2.7	5.0	1⅝		60.0
	2⅛	1.00	1.40	1.9	2.5	3.5	4.6	5.7	10.5	2⅛		125.0
2 F ΔT Per 100 ft	2⅝	1.7	2.4	3.3	4.5	6.1	8.0	10.0	18.5	2⅝		220.0
Equiv. Length	3⅛	2.8	3.9	5.4	7.3	10.0	13.0	16.2	30.0	3⅛	See Table 19–2	350.0
	3⅝	4.1	5.9	8.2	10.8	15.0	19.5	24.3	44.0	3⅝		
	4⅛	6.0	8.5	11.7	15.6	21.5	28.0	35.0	65.0	4⅛		
	5⅛	10.6	15.1	20.8	27.8	38.5	50.0	62.5	113.0	5⅛		
	6⅛	18.1	25.8	35.4	47.2	65.4	85.0	106.0	180.0	6⅛		
	½									½		3.6
Refrigerant 22	⅝								0.6	⅝		7.0
	⅞	0.16	0.23	0.31	0.44	0.57	0.75	0.94	1.5	⅞		18.0
	1⅛	0.34	0.48	0.65	0.91	1.19	1.55	1.93	3.0	1⅛		36.0
	1⅜	0.59	0.81	1.12	1.59	2.07	2.7	3.4	5.2	1⅜		63.0
	1⅝	0.93	1.34	1.8	2.5	3.3	4.3	5.4	8.5	1⅝		100.0
	2⅛	1.9	2.8	3.7	5.2	6.8	8.9	11.1	17.5	2⅛		210.0
2 F ΔT Per 100 ft	2⅝	3.5	5.0	6.6	9.4	12.3	16.0	20.0	31.0	2⅝		375.0
Equiv. Length	3⅛	5.5	8.0	10.6	15.0	19.6	25.5	32.0	50.0	3⅛	See Table 19–3	
	3⅝	8.4	12.0	16.0	22.6	29.5	38.5	48.0	75.0	3⅝		
	4⅛	12.0	17.2	22.9	32.3	42.3	55.0	68.8	105.0	4⅛		
	5⅛	21.2	30.6	41.0	57.5	75.0	98.0	122.0	190.0	5⅛		
	6⅛	34.8	50.0	66.5	94.0	123.0	160.0	200.0	305.0	6⅛		
	Steel IPS SCH				*−60*	*−50*	*−40*	*−30*		*Steel IPS SCH*		
										⅜ 80		17.0
	½ 40				0.26	0.38	0.50	0.62	1.0	½ 80		34.0
	¾ 40				0.55	0.76	1.05	1.30	2.1	¾ 80		75.0
	1 40				1.05	1.53	2.00	2.50	4.1	1 80		150.0
Refrigerant 717	1¼ 40				2.15	3.15	4.10	5.10	8.5	1¼ 80		305.0
(Ammonia)	1½ 40				3.4	5.0	6.5	8.1	12.5	1½ 80		490.0
	2 40				6.3	9.2	12.0	15.0	25.0	2 40	See Table 19–4	
	2½ 40				10.3	15.0	19.5	24.3	40.0	2½ 40		
1 F ΔT Per 100 ft	3 40				18.4	26.8	35.0	43.7	71.0	3 40		
Equiv. Length	3½ 40				27.3	39.8	52.0	65.0	105.0	3½ 40		
	4 40				37.8	55.2	72.0	90.0	145.0	4 40		
	5 40				68.3	100.0	130.0	162.0	260.0	5 40		
	6 40				110.0	161.0	210.0	262.0	425.0	6 40		
	8 40				258.0	376.0	490.0	610.0		8 40		

NOTES:

* *(1) Values in this table are tons of refrigeration resulting in a line friction drop per 100 ft of equivalent pipe length corresponding to the (ΔT) change in saturation temp indicated in the left hand column under the refrigerant designation.*

(2) Values based on 0 F saturated discharge temp. For capacities at other saturated discharge temp, multiply table value by proper line capacity multiplier:

Sat. Discharge Temp, F	*Line Capacity Multipliers*				
	Refrigerant 12 Suction	*Refrigerant 12 Discharge*	*Refrigerant 22 Suction*	*Refrigerant 22 Discharge*	*Ammonia Discharge*
−30	*1.12*	*0.55*	*1.09*	*0.58*	
−20	*1.07*	*0.70*	*1.06*	*0.71*	
−10	*1.03*	*0.85*	*1.03*	*0.85*	*0.77*
0	*1.00*	*1.00*	*1.00*	*1.00*	*1.00*
10	*0.96*	*1.25*	*0.97*	*1.20*	*1.23*
20	*0.93*	*1.50*	*0.94*	*1.45*	*1.45*
30	*0.90*	*1.80*	*0.90*	*1.80*	*1.67*

(3) For other ΔT's and Equivalent Lengths,

$$\textit{Line Capacity (Tons)} = \textit{Table Tons} \times \left(\frac{100}{\textit{Actual Equiv. Length, ft}} \times \frac{\textit{Actual } \Delta T \textit{ Loss Desired, F}}{\textit{Table } \Delta T \textit{ Loss, F}}\right)^{0.55}$$

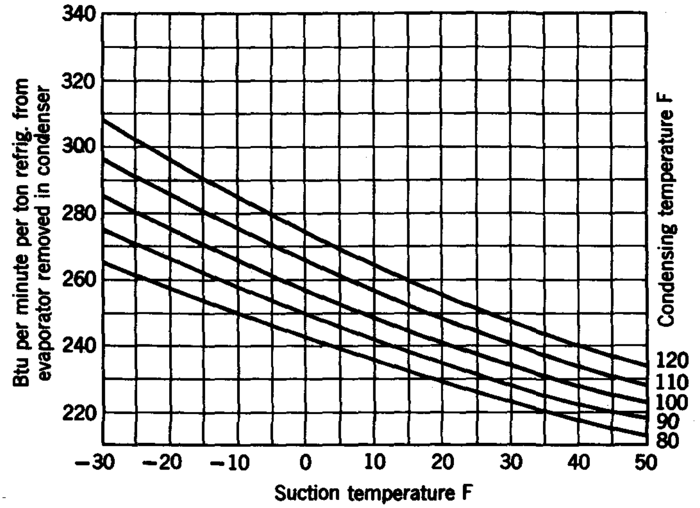
(4) For other Tons and Equivalent Lengths in a given pipe size,

$$\Delta T(F) = \textit{Table } \Delta T \times \frac{\textit{Actual Equiv. Length, ft}}{100} \times \left(\frac{\textit{Actual Tons}}{\textit{Table Tons}}\right)^{1.8}$$

(5) Values obtained from Carrier Corp. data.

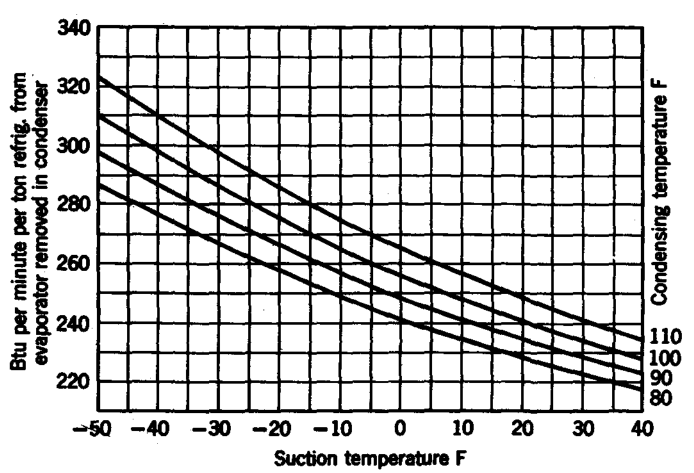
From *ASRE Data Book*, Design Volume, 1957–58 Edition, by permission of the American Society of Heating, Refrigerating, and Air-Conditioning Engineers.

CHART 14-1. Btu per Minute Removed in Refrigerant-12 Condenser per Ton of Refrigerating Effect Occurring in Evaporator



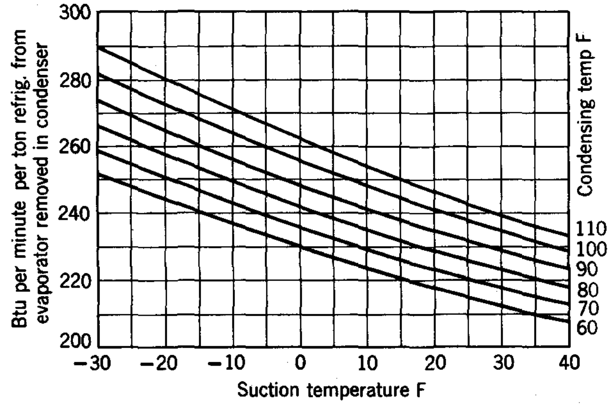
From *ASRE Data Book*, Design Volume 1957–58 Edition, by permission of the American Society of Heating, Refrigerating, and Air-Conditioning Engineers.

CHART 14-2. Btu per Minute Removed in Refrigerant-22 Condenser per Ton of Refrigerating Effect Occurring in Evaporator



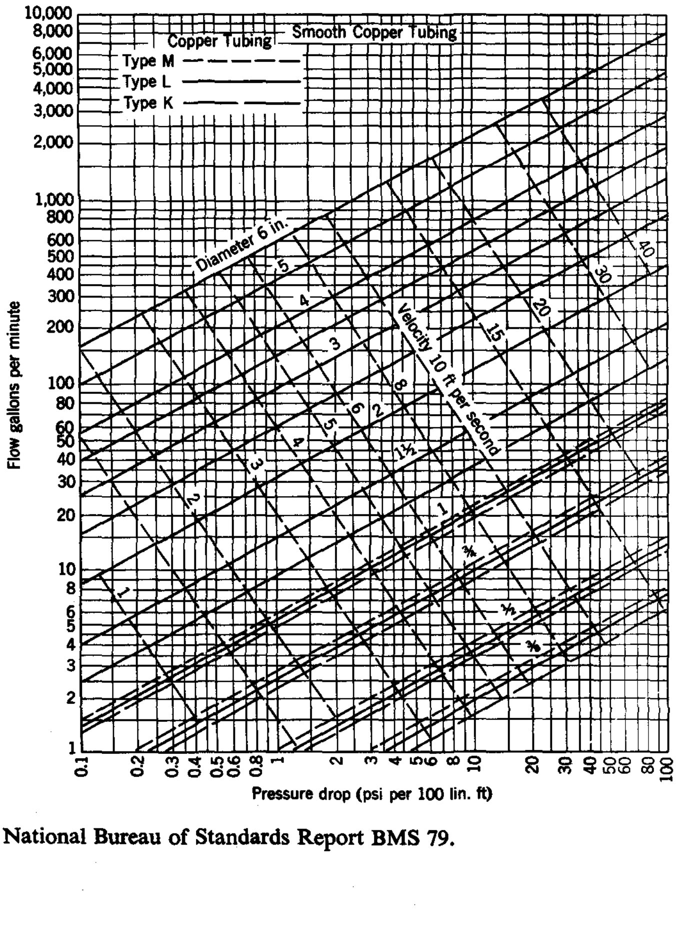
From *ASRE Data Book*, Design Volume 1957–58 Edition, by permission of The American Society of Heating, Refrigerating, and Air-Conditioning Engineers.

CHART 14-3. Btu per Minute Removed in Ammonia Condenser per Ton Refrigerating Effect Occurring in Evaporator



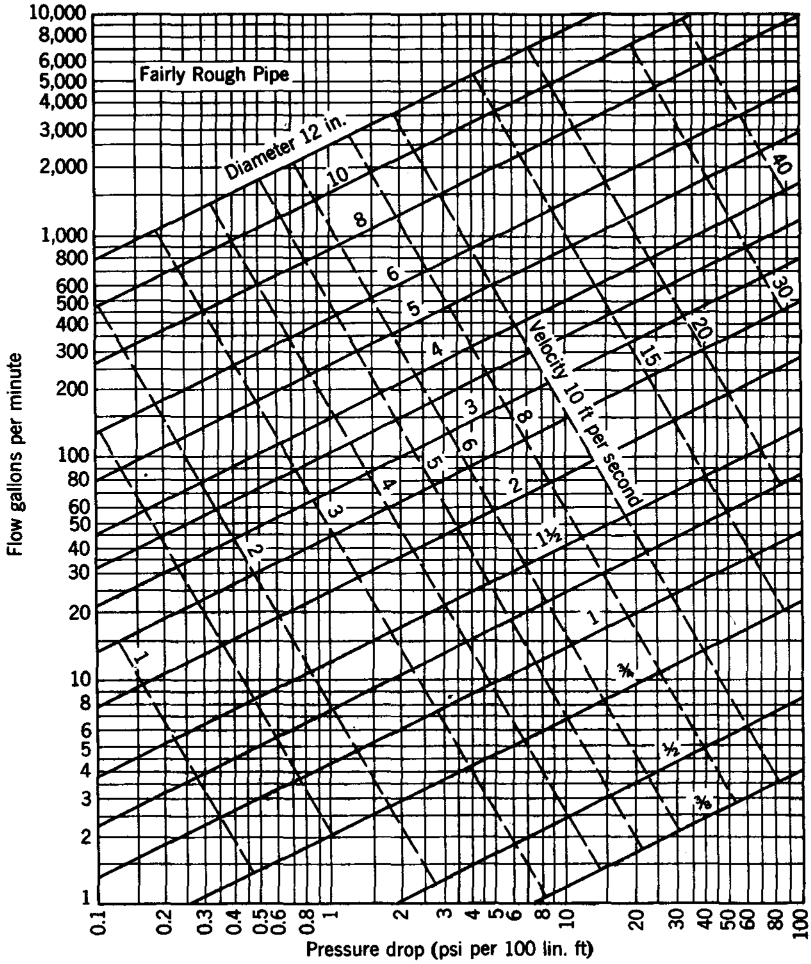
From *ASRE Data Book*, Design Volume 1957–58 Edition, by permission of The American Society of Heating, Refrigerating and Air-Conditioning Engineers.

CHART 15-1. Resistance to Flow of Water Through Smooth Copper Tubing



National Bureau of Standards Report BMS 79.

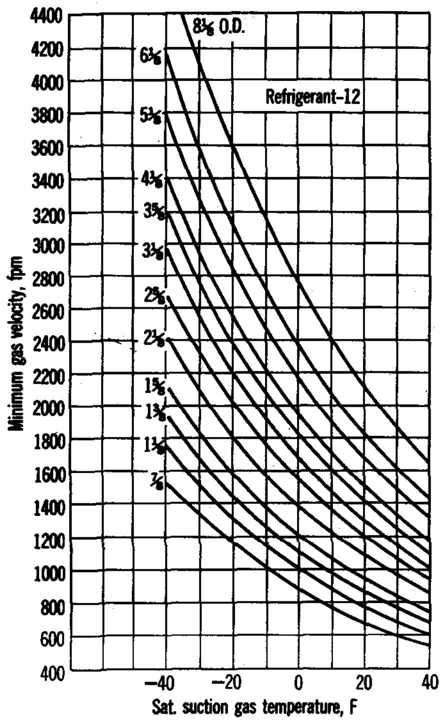
CHART 15-2. Resistance to Flow of Water Through Fairly Rough Pipe



National Bureau of Standards Report BMS 79.

CHART 19-1A. Minimum Gas Velocity for Oil Entrainment in Copper-Tube Suction Risers

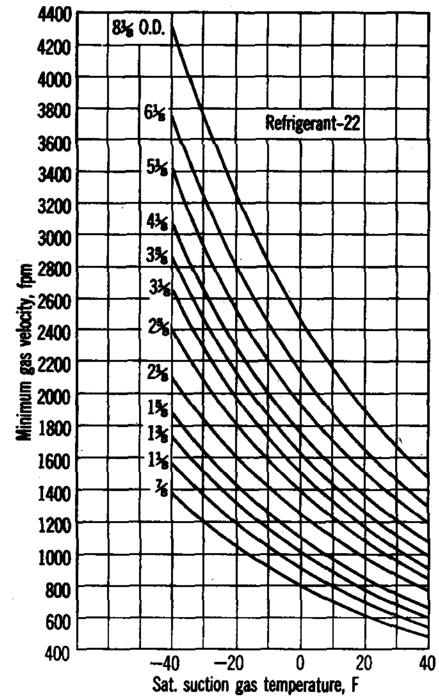
Design for At Least 25% Greater Velocity at Lowest Partial Loading



Courtesy Carrier Corporation.

CHART 19-1B. Minimum Gas Velocity for Oil Entrainment in Copper-Tube Suction Risers

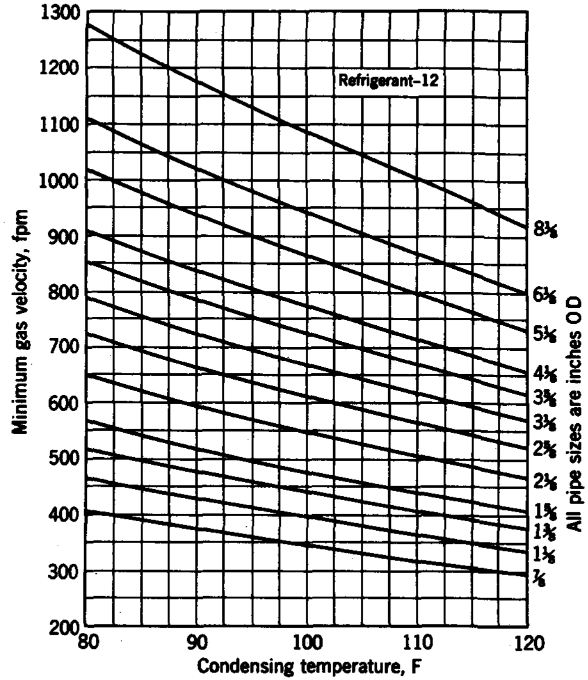
Design for At Least 25% Greater Velocity at Lowest Partial Loading



Courtesy Carrier Corporation.

CHART 19-1C Minimum Gas Velocity for Oil Entrainment Up Vertical Hot Gas Risers

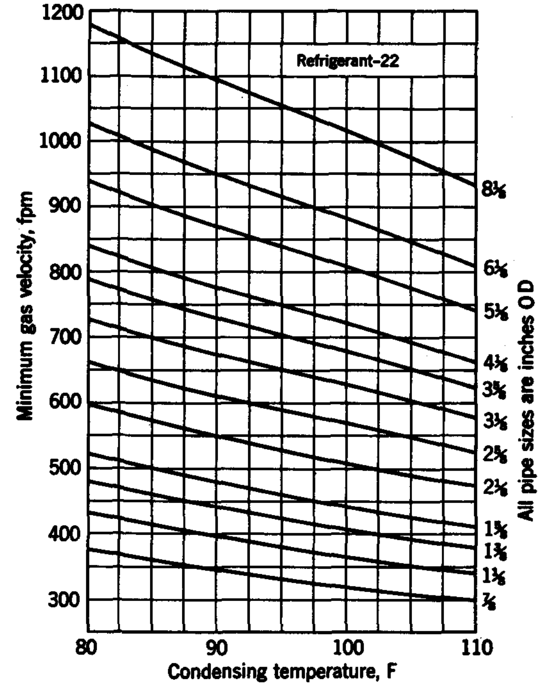
At Least 25% Greater Velocity at Lowest Partial Loading



Courtesy Carrier Corporation

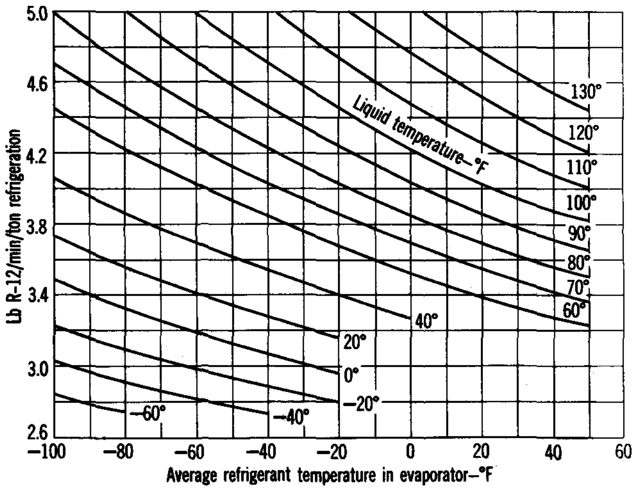
CHART 19-1D Minimum Velocity for Oil Entrainment Up Vertical Hot Gas Risers

At Least 25% Greater Velocity at Lowest Partial Loading



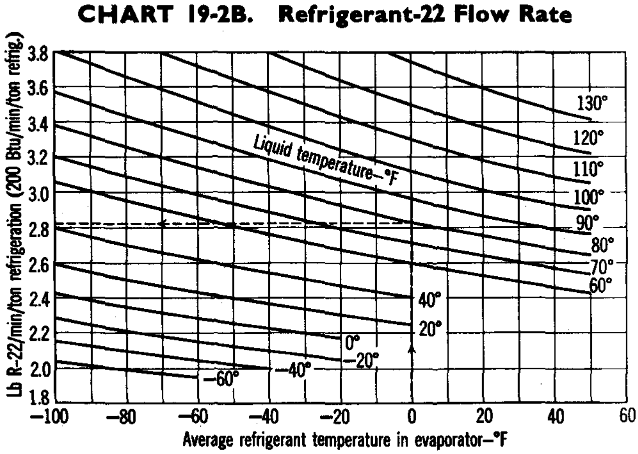
Courtesy Carrier Corporation

CHART 19-2A. Refrigerant-12 Flow Rate



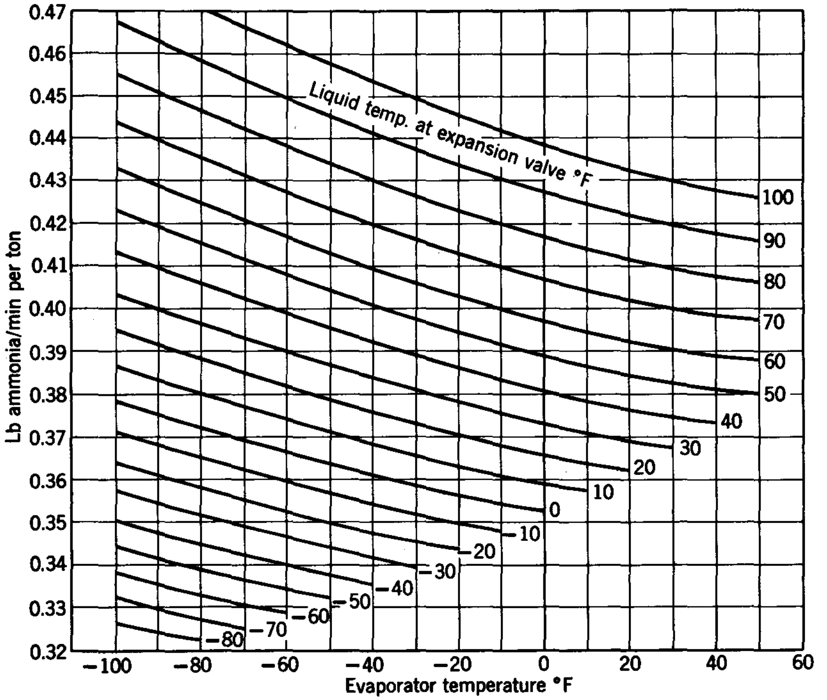
Courtesy York Corporation.

CHART 19-2B. Refrigerant-22 Flow Rate



Courtesy York Corporation.

CHART 19-2C. Ammonia Flow Rate



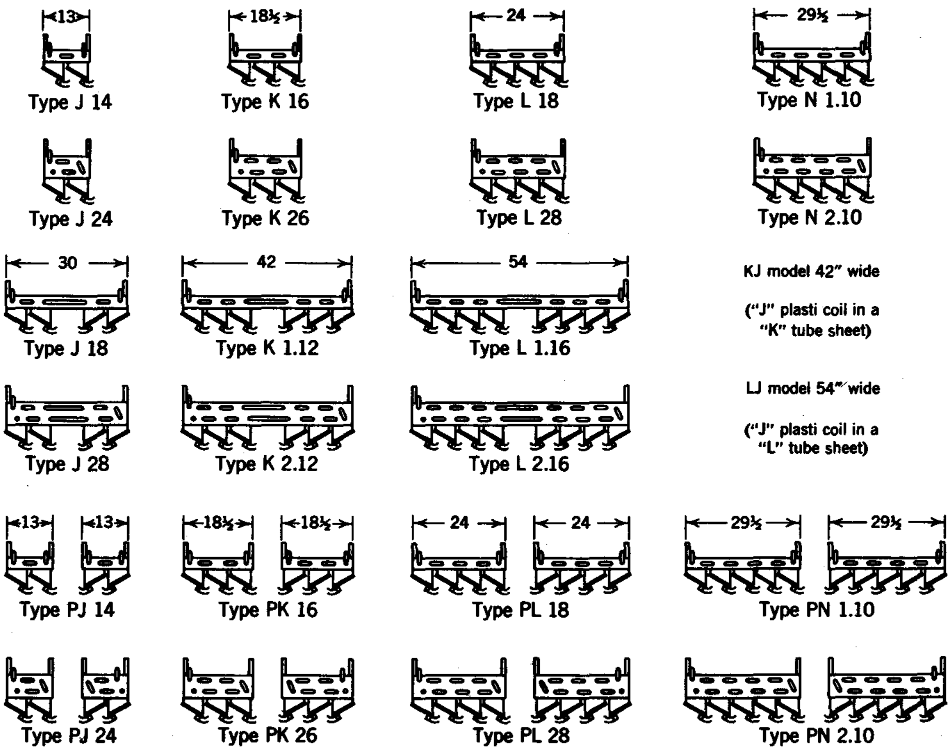
Courtesy Carrier Corporation.

The equipment rating tables reprinted in this book are intended only to illustrate methods of equipment rating and selection. For this reason, many are incomplete and therefore do not represent the full line of the manufacturer.

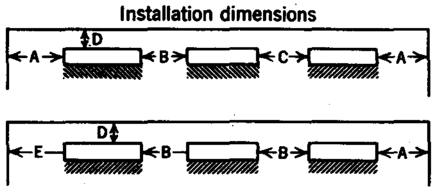
TABLE R-1. Natural Convection Cooling Coils Capacities Btu/hr/inch Finned Length

Model	(7″ less than overall) Width	No. Tubes	Fin Spac.	Surface Sq Ft per in.	Btu per Hr per in. 1° TD	15° TD
			Single Row High Coils			
J-14	13″	4	⅓	1.26	1.47	22.1
			½	0.86	1.21	18.2
K-16	18½″	6	⅓	1.89	2.21	33.1
			½	1.29	1.82	27.3
L-18	24″	8	⅓	2.52	2.94	44.1
			½	1.72	2.44	35.5
N-1.10	29½″	10	⅓	3.15	3.68	55.2
			½	2.15	3.04	45.9
J-18	30″	8	⅓	2.52	2.94	44.2
			½	1.72	2.44	36.6
K-1.12	42″	12	⅓	3.78	4.42	66.3
			½	2.58	3.65	54.8
L-1.16	54″	16	⅓	5.04	5.88	88.4
			½	3.44	4.87	73.2
PJ-14	13″*	8	⅓	2.52	2.94	44.2
			½	1.72	2.44	36.6
PK-16	18½″*	12	⅓	3.78	4.42	66.2
			½	2.58	3.65	54.8
PL-18	24″*	16	⅓	5.04	5.88	88.3
			½	3.44	4.87	73.0
PN-1.10	29½*	20	⅓	6.30	7.33	110.0
			½	4.30	6.10	91.1
			Two Row High Coils			
J-24	13″	8	⅓	2.52	2.65	39.8
			½	1.72	2.23	33.5
K-26	18½*	12	⅓	3.78	3.96	59.4
			½	2.58	3.34	50.1
L-28	24″	16	⅓	5.04	5.30	79.5
			½	3.44	4.45	66.8
N-2.10	29½″	20	⅓	6.30	6.62	99.3
			½	4.30	5.57	83.6
J-28	30″	16	⅓	5.04	5.30	79.5
			½	3.44	4.45	66.8
K-2.12	42″	24	⅓	7.56	7.92	118.8
			½	5.16	6.68	100.2
L-2.16	54″	32	⅓	10.08	10.60	159.0
			½	6.88	8.90	133.5
PJ-24	13″*	16	⅓	5.04	5.30	79.5
			½	3.44	4.45	66.8
PK-26	18½″*	24	⅓	7.56	7.92	118.8
			½	5.16	6.68	100.2
PL-28	24″*	32	⅓	10.08	10.60	159.0
			½	6.88	8.90	133.5
PN-2.10	29½″*	40	⅓	12.60	13.25	198.8
			½	8.60	11.12	166.8

* Width of each section.
Courtesy Dunham-Bush, Inc.



Installation dimensions



Dimensions	A	B	C	D	E
Maximum	12 in.	5 ft.	8 ft.	8 in.	4 ft.
Minimum	6 in.	10 in.	6 in.	4 in.	4 in.

Note:
If the width of the box requires additional plasti-units, care should be taken to see that they are installed in a similar manner as shown above.

TABLE R-2. Single Plate Evaporators

Catalog Number	Width, Inches	Length, Inches	Feet of Pass	Sq Ft Surface	Total "K" per Plate Below 32°	Total "K" per Plate Above 32°	Total Btu's per Hour at 15° TD Below 32°	Total Btu's per Hour at 15° TD Above 32°
1224	12	24	9.9	4.18	10	13	150	195
1236	12	36	15.8	6.26	16	18	240	270
1248	12	48	21.8	8.35	21	24	315	360
1260	12	60	27.8	10.45	26	30	390	450
1272	12	72	33.8	12.55	31	36	465	540
1284	12	84	39.8	14.65	37	42	555	630
12108	12	108	51.8	18.83	48	54	720	810
12144	12	144	69.8	25.2	63	72	945	1080
2230	22	30	25.7	9.64	24	28	360	420
2236	22	36	31.6	11.58	29	33	435	495
2248	22	48	43.6	15.4	38	44	570	660
2260	22	60	55.6	19.3	48	55	720	825
2272	22	72	67.6	23.2	58	66	870	990
2284	22	84	79.6	27.24	68	79	1020	1185
22108	22	108	103.6	34.9	87	100	1305	1500

Courtesy Kold-Hold Division, Tranter Manufacturing Co.

TABLE R-3. Plate Banks

Catalog Number	Size, Inches	No. Plates	Feet of Pass	No. of Expansion Valves—R-12	No. of Expansion Valves—Ammonia	Total Btu's per Hour at 15° TD Below 32°	Total Btu's per Hour at 15° TD Above 32°
4-1248-B	12 × 48	4	87	1	1	1260	1440
5-1248-B	12 × 48	5	109	1	1	1575	1880
6-1248-B	12 × 48	6	131	1	1	1890	2160
4-1260-B	12 × 60	4	111	1	1	1560	1800
5-1260-B	12 × 60	5	139	1	1	1950	2250
6-1260-B	12 × 60	6	166	1	1	2340	2700
4-1272-B	12 × 72	4	136	1	1	1860	2160
5-1272-B	12 × 72	5	170	1	1	2325	2700
6-1272-B	12 × 72	6	204	2	1	2790	3240
4-1284-B	12 × 84	4	160	1	1	2220	2520
5-1284-B	12 × 84	5	200	2	1	2775	3150
6-1284-B	12 × 84	6	240	2	1	3300	3780
4-12108-B	12 × 108	4	207	2	1	2880	3240
5-12108-B	12 × 108	5	258	2	1	3600	4050
6-12108-B	12 × 108	6	310	2	1	4320	4860
4-12144-B	12 × 144	4	279	2	1	3780	4320
5-12144-B	12 × 144	5	348	2	1	4725	5400
6-12144-B	12 × 144	6	418	2	1	5670	6480

Courtesy Kold-Hold Division, Tranter Manufacturing Co.

TABLE R-4. Plate Stands

Catalog Number	Width, Inches	Length, Inches	No. of Plates	Feet of Pass	No. of Expansion Valves—F-12	No. of Expansion Valves—Ammonia	Total BTU's per hour at 15° TD	
							Below 32°	Above 32°
4-2230-S	22	30	4	103	1	1	1440	1680
5-2230-S	22	30	5	128	1	1	1800	2100
6-2230-S	22	30	6	154	1	1	2160	2520
7-2230-S	22	30	7	180	2	1	2520	2940
8-2230-S	22	30	8	206	2	1	2880	3360
4-2248-S	22	48	4	175	1	1	2280	2640
5-2248-S	22	48	5	218	2	1	2850	3300
6-2248-S	22	48	6	262	2	1	3420	3960
7-2248-S	22	48	7	306	2	1	3990	4620
8-2248-S	22	48	8	350	2	1	4560	5280
4-2260-S	22	60	4	222	2	1	2880	3300
5-2260-S	22	60	5	278	2	1	3600	4125
6-2260-S	22	60	6	333	2	1	4320	4950
7-2260-S	22	60	7	390	3	1	5040	5775
8-2260-S	22	60	8	445	3	2	5760	6600
4-2272-S	22	72	4	270	2	1	3480	3960
5-2272-S	22	72	5	338	2	1	4350	4950
6-2272-S	22	72	6	405	3	1	5220	5940
7-2272-S	22	72	7	474	3	2	6090	6930
8-2272-S	22	72	8	540	3	2	6960	7920
4-2284-S	22	84	4	319	2	1	4080	4740
5-2284-S	22	84	5	398	3	1	5100	5925
6-2284-S	22	84	6	479	3	2	6120	7110
7-2284-S	22	84	7	558	3	2	7140	8295
8-2284-S	22	84	8	637	4	2	8160	9480
4-22108-S	22	108	4	415	2	1	5220	6000
5-22108-S	22	108	5	518	3	2	6525	7500
6-22108-S	22	108	6	622	3	2	7830	9000
7-22108-S	22	108	7	725	4	2	9135	10500
8-22108-S	22	108	8	830	4	2	10440	12000

Courtesy Kold-Hold Division, Tranter Manufacturing Co.

TABLE R-5. "K" Factors for Bare Pipe Coils in Liquid*

Desired Liquid Temp.	Refr. Temp.	"K"	Desired Liquid Temp.	Refr. Temp.	"K"
65° (*a*)	38°	15.7	35° (*c*)	19°	13.5
60 (*a*)	38	15.5	30 (*c*)	15	13.0
55 (*a*)	38	15.2	25 (*c*)	11	12.5
50 (*a*)	36	15.0	20 (*c*)	7	12.0
45 (*a*)	32	14.5	15 (*c*)	3	11.2
40 (*a*)	28	14.0	10 (*c*)	−1	10.5
35 (*a*)	24	13.5	5 (*c*)	−5	9.8
35 (*b*)	19	12.5	0 (*c*)	−9	9.0
35 (*b*)	15	10.8	−5 (*c*)	−12	8.2
35 (*b*)	11	10.0	−10 (*c*)	−16	7.5
35 (*b*)	7	9.0			

(*a*) Water cooling. (*b*) Water cooling, ice formation on coils. (*c*) Brine cooling.
* For dry expansion tubing or pipe submerged in water or brine without agitation. (Courtesy Vilter Manufacturing Company.)

TABLE R-6. "K" Factors for Bare Pipe Coils in Air*

Refrig. Temp. °F	Room Temperature Degrees Fahrenheit										
	−20°	−10°	0°	10°	20°	30°	36°	40°	44°	50°	60°
32										2.30	2.49
28									2.11	2.50	2.52
24								2.11	2.49	2.51	2.52
20							2.11	2.49	2.49	2.47	
14						1.79	2.50	2.52	2.48	2.52	
12						1.80	2.49	2.49	2.52		
9					1.40	1.79	2.50	2.49	2.49		
6					1.39	2.01	2.48	2.51			
3					1.40	1.99	2.48	2.53			
0				1.39	1.59	1.99	2.51				
−4				1.39	1.59	1.99	2.50				
−8			1.30	1.49	1.80	2.01					
−13			1.39	1.60	1.74	1.98					
−17			1.50	1.70	1.79						
−25		1.50	1.60	1.80							
−30	1.39	1.70	1.79	1.80							
−40	1.59	1.80	1.80								
−50	1.79	1.80									

* For iron pipe coils with gravity air circulation. (Courtesy Vilter Manufacturing Company.)

TABLE R-7. Lineal Foot of Pipe per Sq Ft of External Surface

Pipe Size	Lineal Feet
$\frac{1}{2}''$	4.55
$\frac{3}{4}''$	3.64
$1''$	2.90
$1\frac{1}{4}''$	2.30
$1\frac{1}{2}''$	2.01
$2''$	1.61

Courtesy Vilter Manufacturing Company.

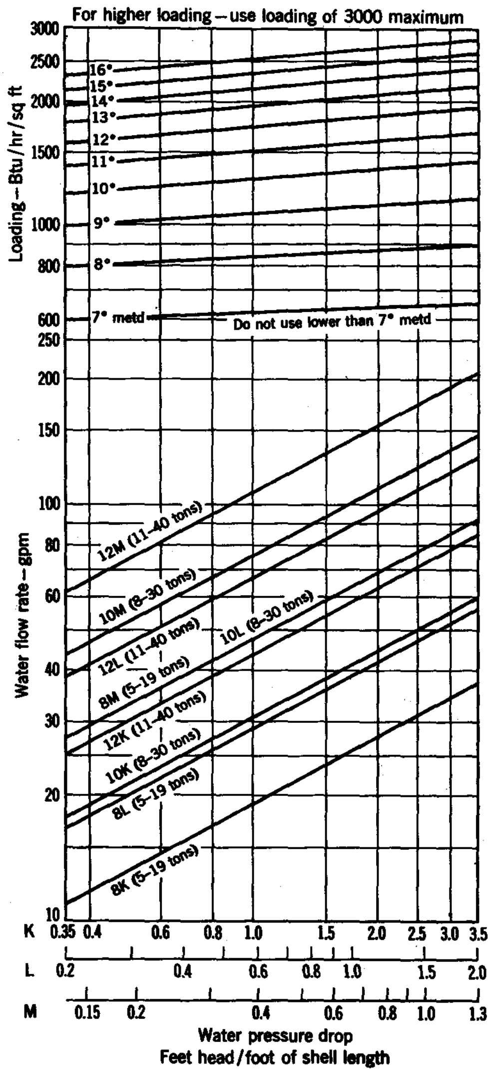
TABLE R-8. Unit Cooler Capacity Ratings and Specifications

	Btu/Hr Rating			Core		Motor and Fan					
Model	10° TD	15° TD	Surf. FT^2	Circuits	Tube	HP	Motor Heat Btu/24 Hr	Fan	Rpm	Cfm	Air Throw
UC25	2,500	3,750	67	1	$\frac{1}{2}''$	$\frac{1}{25}$	7,600	$10''$	1500	390	20
UC35	3,500	5,250	93	1	$\frac{1}{2}''$	$\frac{1}{25}$	8,000	$12''$	1500	510	20
UC45	4,500	6,750	156	1	$\frac{1}{2}''$	$\frac{1}{20}$	11,500	$14''$	1500	700	17
UC65	6,500	9,750	210	1	$\frac{5}{8}''$	$\frac{1}{12}$	12,600	$16''$	1140	1,000	23
UC85	8,500	12,750	266	1	$\frac{3}{4}''$	$\frac{1}{12}$	13,350	$16''$	1140	1,480	27
UC105	10,500	15,750	328	Split	$\frac{3}{4}''$	$\frac{1}{6}$	15,100	$18''$	1140	1,730	25
UC120	12,000	18,000	378	Split	$\frac{3}{4}''$	(2) $\frac{1}{20}$	18,000	(2) $14''$	1140	1,950	25
UC180	18,000	27,000	566	Split	$\frac{3}{4}''$	(2) $\frac{1}{12}$	25,200	(2) $16''$	1140	2,550	20
UC240	24,000	36,000	755	Two	$\frac{3}{4}''$	(2) $\frac{1}{6}$	34,000	(2) $18''$	1140	4,050	27
UC320	32,000	48,000	1030	3	$\frac{3}{4}''$	(2) $\frac{1}{4}$	75,000	(2) $22''$	1140	6,000	28

Courtesy Dunham-Bush Inc.

TABLE R-9 Water Chillers

Acme Model No.*	Effective Tube Area Sq Ft	Capacity Range Tons	Std. No. of Circuits	Number of Tubes
DXH-805	41			
806	50			
807	59			
808	67			
809	76	5.4 to 19	1	44
810	85			
811	94			
812	102			
813	111			
814	119			
DXH-1005	64			
1006	77			
1007	91			
1008	104			
1009	118			
1010	131	8 to 30	1	68
1011	145			
1012	158			
1013	171			
1014	184			
1015	197			
1016	211			
DXH-1206	105			
1207	123			
1208	141			
1209	159			
1210	177			
1211	195	11 to 40	2	92
1212	213			
1213	231			
1214	249			
1215	267			
1216	285			
DXH-1406	136			
1407	159			
1408	184			
1409	207			
1410	231			
1411	255	14 to 52	2	120
1412	278			
1413	302			
1414	325			
1415	349			
1416	372			
DXH-1606	187			
1607	219			
1608	251			
1609	283			
1610	315			
1611	348	20 to 71	2	164
1612	380			
1613	412			
1614	444			
1615	477			
1616	509			
DXH-2006	286			
2007	335			
2008	384			
2009	437			
2010	487			
2011	535			
2012	583	31 to 111	2	252
2013	633			
2014	683			
2015	733			
2016	782			
2017	830			
2018	880			
2020	976			



Courtesy Acme Industries.

TABLE R-10A. Compressor Rating Data

SAT. DISCH. TEMP. °F	SAT. SUCTION		5F20				5F30				5F40				5F60			
			1450 RPM		1750 RPM		1450 RPM		1750 RPM		1450 RPM		1750 RPM		1450 RPM		1750 RPM	
	Temp. °F	Pressure Psig*	Btu/Hr	BHP	Btu/Hr	BHP	Btu/Hr	BHP	Btu/Hr	BHP	Btu/Hr	BHP	Btu/Hr	BHP	Btu/Hr	BHP	Btu/Hr	BHP
80° 84.1 Psig	—40	10.92"	5,300	1.5	6,400	1.8	7,950	2.1	9,600	2.7	10,600	2.9	12,800	3.6	15,900	4.4	19,200	5.4
	—30	5.45"	8,450	1.8	10,200	2.3	12,680	2.7	15,300	3.4	16,900	3.6	20,400	4.5	25,400	5.5	30,600	6.8
	—20	.6	12,600	2.2	15,200	2.7	18,900	3.3	22,800	4.1	25,200	4.4	30,400	5.4	37,800	6.6	45,600	8.2
	—10	4.5	17,500	2.5	21,200	3.2	26,350	3.8	31,800	4.7	35,100	5.1	42,400	6.3	52,700	7.7	63,600	9.5
	0	9.2	24,000	2.9	29,000	3.6	36,000	4.4	43,500	5.4	48,100	5.8	58,000	7.2	72,100	8.8	87,000	10.9
	10	14.7	31,600	3.2	38,200	4.0	47,500	4.8	57,300	6.0	63,300	6.4	76,400	8.0	94,900	9.7	114,600	12.0
	20	21.1	40,600	3.4	49,000	4.2	60,900	5.1	73,500	6.4	81,200	6.8	98,000	8.5	121,800	10.3	147,000	12.7
	30	28.5	50,700	3.5	61,200	4.3	76,000	5.2	91,800	6.4	101,400	6.9	122,400	8.6	152,000	10.4	183,600	12.9
	40	37.0	63,000	3.4	76,000	4.2	94,400	5.1	114,000	6.4	126,000	6.8	152,000	8.5	189,000	10.3	228,000	12.7
	50	46.7	76,800	3.3	92,800	4.1	115,300	5.0	139,200	6.2	153,300	6.7	185,600	8.3	231,000	10.0	278,400	12.4
90° 99.6 Psig	—40	10.92"	4,470	1.4	5,400	1.7	6,710	2.0	8,100	2.5	8,940	2.7	10,800	3.4	13,420	4.1	16,200	5.1
	—30	5.45"	7,450	1.8	9,000	2.2	11,180	2.7	13,500	3.4	14,900	3.6	18,000	4.5	22,400	5.4	27,000	6.7
	—20	.6	11,400	2.3	13,800	2.8	17,150	3.4	20,700	4.2	22,900	4.5	27,600	5.6	34,300	6.8	41,400	8.4
	—10	4.5	16,100	2.7	19,400	3.3	24,100	4.0	29,100	5.0	32,200	5.3	38,800	6.6	48,200	8.1	58,200	10.0
	0	9.2	22,000	3.1	26,600	3.8	33,000	4.6	39,900	5.7	44,100	6.2	53,200	7.6	66,100	9.3	79,800	11.5
	10	14.7	29,100	3.4	35,200	4.2	43,700	5.1	52,800	6.4	58,300	6.8	70,400	8.5	87,400	10.2	105,600	12.7
	20	21.1	37,500	3.7	45,200	4.6	56,100	5.5	67,800	6.9	74,900	7.4	90,400	9.2	112,200	11.1	135,600	13.8
	30	28.5	47,400	3.8	57,200	4.8	71,000	5.8	85,800	7.1	94,800	7.7	114,400	9.5	142,000	11.5	171,600	14.3
	40	37.0	59,000	3.9	71,200	4.9	88,400	5.9	106,800	7.3	118,100	7.8	142,400	9.7	177,000	11.8	213,600	14.6
	50	46.7	72,000	3.8	87,000	4.8	108,100	5.8	130,500	7.1	144,200	7.7	174,000	9.5	216,400	11.5	261,000	14.3
100° 116.9 Psig	—40	10.92"	3,480	1.2	4,200	1.5	5,210	1.9	6,300	2.3	6,960	2.5	8,400	3.1	10,420	3.7	12,600	4.6
	—30	5.45"	6,300	1.7	7,600	2.2	9,440	2.6	11,400	3.2	12,600	3.5	15,200	4.3	18,900	5.3	22,800	6.5
	—20	.6	10,100	2.2	12,200	2.7	15,180	3.3	18,300	4.1	20,200	4.5	24,400	5.6	30,300	6.6	36,600	8.2
	—10	4.5	14,600	2.7	17,600	3.4	21,900	4.1	26,400	5.1	29,200	5.5	35,200	6.8	43,750	8.3	52,800	10.2
	0	9.2	19,900	3.2	24,000	4.0	29,800	4.8	36,000	6.0	39,800	6.4	48,000	8.0	59,600	9.7	72,000	12.0

	10	14.7	26,700	3.6	32,200	4.4	40,000	5.4	48,300	6.7	53,300	7.2	64,400	8.9	80,000	10.8	96,600	13.3
	20	21.1	34,600	3.9	41,800	4.8	52,000	5.8	62,700	7.2	69,300	7.7	83,600	9.6	104,000	11.6	125,400	14.4
	30	28.5	44,200	4.1	53,400	5.1	66,400	6.2	80,100	7.7	88,400	8.3	106,800	10.2	132,700	12.4	160,200	15.3
	40	37.0	55,200	4.3	66,600	5.3	82,700	6.4	99,900	8.0	110,300	8.6	133,200	10.6	165,400	12.8	199,800	15.9
	50	46.7	67,400	4.3	81,400	5.3	101,200	6.4	122,100	8.0	134,900	8.6	162,800	10.7	202,000	12.9	244,200	16.0
105° 126.2 Psig	—30	5.45"	5,720	1.6	6,900	2.1	8,570	2.5	10,350	3.0	11,440	3.3	13,800	4.1	17,200	5.0	20,700	6.1
	—20	.6	9,280	2.2	11,200	2.7	13,930	3.3	16,800	4.0	18,600	4.4	22,400	5.4	27,900	6.5	33,600	8.1
	—10	4.5	13,750	2.7	16,600	3.4	20,650	4.1	24,900	5.1	27,500	5.5	33,200	6.8	41,300	8.3	49,800	10.2
	0	9.2	19,000	3.2	22,900	4.0	28,500	4.8	34,400	6.0	38,000	6.4	45,800	8.0	56,900	9.7	68,700	12.0
	10	14.7	25,600	3.6	30,900	4.5	38,400	5.5	46,400	6.8	51,200	7.3	61,800	9.0	76,500	10.9	92,700	13.5
	20	21.1	33,300	4.0	40,200	4.9	50,000	5.9	60,300	7.4	66,700	7.9	80,400	9.8	100,000	11.9	120,600	14.7
	30	28.5	42,600	4.2	51,400	5.3	63,900	6.4	77,100	7.9	85,200	8.5	102,800	10.5	127,800	12.7	154,200	15.7
	40	37.0	53,200	4.5	64,200	5.5	79,800	6.6	96,300	8.3	106,300	8.9	128,400	11.0	159,600	13.3	192,600	16.5
	50	46.7	65,400	4.5	78,900	5.6	98,000	6.8	118,400	8.4	130,700	9.1	157,800	11.3	196,000	13.6	236,700	16.8
110° 136.0 Psig	—30	5.45"	5,140	1.5	6,200	1.9	7,700	2.3	9,300	2.8	10,280	3.1	12,400	3.8	15,400	4.6	18,600	5.7
	—20	.6	8,450	2.1	10,200	2.6	12,680	3.2	15,300	3.9	16,900	4.2	20,400	5.2	25,350	6.4	30,600	7.9
	—10	4.5	12,900	2.7	15,600	3.3	19,400	4.0	23,400	5.0	25,800	5.4	31,200	6.7	38,800	8.2	46,800	10.2
	0	9.2	18,100	3.2	21,800	4.0	27,100	4.8	32,700	6.0	36,200	6.4	43,600	8.0	54,100	9.7	65,400	12.0
	10	14.7	24,500	3.6	29,600	4.5	36,800	5.5	44,400	6.8	49,000	7.3	59,200	9.0	73,000	11.0	88,800	13.6
	20	21.1	32,000	4.0	38,600	5.0	48,000	6.0	57,900	7.5	64,000	8.0	77,200	10.0	95,900	12.1	115,800	15.0
	30	28.5	40,900	4.3	49,400	5.4	61,400	6.5	74,100	8.0	81,900	8.6	98,800	10.7	122,800	13.0	148,200	16.1
	40	37.0	51,200	4.6	61,800	5.7	76,800	6.8	92,700	8.5	102,300	9.2	123,600	11.4	153,700	13.7	185,400	17.0
	50	46.7	63,300	4.7	76,400	5.9	94,800	7.1	114,600	8.8	126,500	9.5	152,800	11.8	190,000	14.3	229,200	17.7
120° 157.1 Psig	—20	.6	6,950	2.0	8,400	2.4	10,430	3.0	12,600	3.7	13,900	3.9	16,800	4.9	20,900	5.9	25,200	7.3
	—10	4.5	11,100	2.6	13,400	3.2	16,650	3.9	20,100	4.8	22,200	5.2	26,800	6.4	33,500	7.7	40,400	9.6
	0	9.2	16,060	3.2	19,400	3.9	24,100	4.7	29,100	5.9	32,200	6.3	38,800	7.8	48,200	9.5	58,200	11.8
	10	14.7	22,200	3.7	26,800	4.6	33,300	5.5	40,200	6.9	44,400	7.4	53,600	9.2	66,600	11.1	80,400	13.8
	20	21.1	29,500	4.2	35,600	5.2	44,200	6.2	53,400	7.7	59,000	8.3	71,200	10.3	88,400	12.5	106,800	15.5
	30	28.5	37,600	4.6	45,400	5.7	56,500	6.8	68,200	8.5	75,300	9.1	90,800	11.3	112,800	13.7	136,200	17.0
	40	37.0	47,200	4.9	57,000	6.1	70,800	7.3	85,500	9.1	94,400	9.8	114,000	12.2	141,700	14.7	171,000	18.2
	50	46.7	59,000	5.1	71,200	6.4	88,400	7.7	106,800	9.5	118,000	10.2	142,400	12.7	177,000	15.4	213,600	19.1

Courtesy Carrier Corporation.

TABLE R-10B. Compressor Rating Data

	UNIT		5F20	5F30	5F40	5F60	5H40	5H60	5H80	5H40-60 Duplex	5H60-60 Duplex	5H80-80 Duplex
GENERAL DATA	Nominal Horsepower		5	7½	10	15	25	40	50	60	75	100
	No. of Cylinders		2	3	4	6	4	6	8	10	12	16
	Bore, Inches		2½	2½	2½	2½	3¼	3¼	3¼	3¼	3¼	3¼
	Stroke, Inches		2	2	2	2	2¾	2¾	2¾	2¾	2¾	2¾
DISPLACEMENT	CFM at 1750 RPM		19.90	29.80	39.80	59.60	92.40	138.4	184.7	230.8	276.8	369.4
CAPACITY BTU PER HR (At 1750 RPM with F-12)	ASRE Group I (—10° and 100°)[1]		17,600	26,400	35,200	52,800	84,000	126,000	168,000	210,000	252,000	336,000
	ASRE Group II (5° and 100°)[1]		28,000	42,000	56,000	84,000	132,200	198,500	264,000	331,000	397,000	528,800
	ASRE Group III (20° and 105°)[1]		40,200	60,300	80,400	120,600	188,750	283,000	377,500	471,500	566,000	755,000
	ASRE Group IV (40° and 110°)[1]		61,800	92,700	123,600	185,400	283,000	425,000	566,000	708,000	850,000	1,132,000
SPEED DATA	Maximum RPM	F-12 and C-7[6]	1750	1750	1750	1750	1750	1750	1750	1750	1750	1750
		F-22	1450	1450	1450	1450	1450	1450	1450	1450	1450	1450
	Minimum RPM	For Unloader Operation	600	700	800	900	800	900	1100	900	900	1100
		For Lubrication	400	400	400	400	400	400	400	400	400	400
OIL DATA	Minimum Oil Pressure for Unloader Operation		22	28	35	35	35	35	35	35	35	35
	Recommended Oil Pressure at 1750 RPM-PSI		45	45	45	45	45	45	45	45	45	45
	Normal Oil Level in Bulls Eye		Cent. Line	Cent. Line	⅜" above C.L.	⅜" above C.L.	Cent. Line	Cent. Line	Cent. Line	Cent. Line	Cent. Line	Cent. Line
	Oil Charge (Factory) Pints		5	5½	12	13	18	21	41	39	42	82
NET WEIGHTS LB	Compressor Only[3]		175	215	315	400	610	795	1115	......	......	......
	Compressor Unit with Motor and Drive[3]	Direct Drive	......	......	794	910	1350	1810	2330	3250	3620	4710
		Belt Drive	465	565	866	986	1480	1980	......	......	......	......
	Condensing Unit (with Standard Condenser Package)[4]	Direct Drive	......	......	1110	1280	1930	2610	3280	......	......	......
		Belt Drive	545	690	1180	1360	2050	2780	......	......	......	......
	Condensing Unit with Small Cond. Pkg.	Belt Drive	......	645	1090	1310	1860	2570	......	......	......	......
CONNECTION SIZES - INCHES	Compressor Suction Gas		1⅜ OD	1⅝ OD	2⅛ OD	2⅛ OD	2⅝ OD	3⅛ OD	3⅝ OD	......[5]	......[5]	......[5]
	Compressor Discharge Gas		1⅛ OD	1⅜ OD	1⅜ OD	1⅝ OD	2⅛ OD	2⅝ OD	3⅛ OD	......[5]	......[5]	......[5]

Courtesy Carrier Corporation.

TABLE R-10C. Specified Suction Temperature at Compressor Inlet (R-12 and R-500)

Saturated suction temperature ° F	−40	−30	−20	−10	0 and above
Actual suction temperature ° F	35	45	55	65	65

Courtesy Carrier Corporation.

TABLE R-10D. Correction Factors

Saturation temperature of suction gas (° F)	−40	−30	−20	−10	0	10	20	30	40	50
Factor	0.90	0.91	0.92	0.93	0.94	0.95	0.96	0.97	0.987	0.997

Courtesy Carrier Corporation.

TABLE R-11. Air Cooled Condensing Units

$\frac{1}{2}$ hp

Application and Speed	Sat. Suction Temp. (F)	Sat. Suction Press. (Psig)	80 F Btu/Hr	80 F Disch. Press. Psig	80 F Kw	90 F Btu/Hr	90 F Disch. Press. (Psig)	90 F Kw	100 F Btu/Hr	100 F Disch. Press. (Psig)	100 F Kw
			Ambient Air Temperature (F)								
High temp. 1725 rpm	45	41.7	6,760	149	0.73	6,280	169	0.76	5,810	187	0.80
	40	37.0	6,300	143	0.71	5,830	162	0.74	5,480	181	0.78
	35	32.6	5,810	137	0.68	5,370	154	0.71	4,950	175	0.75
	30	28.5	5,340	133	0.65	4,930	148	0.68	4,520	169	0.73
	25	24.6	4,900	126	0.62	4,490	142	0.64	4,100	163	0.70
Med. temp. 1725 rpm	25	24.6	5,780	141	0.73	5,360	157	0.75	4,960	179	0.76
	20	21.1	5,300	136	0.71	4,900	151	0.72	4,500	172	0.74
	15	17.7	4,800	130	0.68	4,420	145	0.69	4,040	165	0.71
	10	14.7	4,320	125	0.66	3,960	141	0.67	3,600	160	0.69
	5	11.8	3,900	119	0.63	3,560	136	0.64	3,220	154	0.66
	0	9.2	3,490	113	0.60	3,190	131	0.61	2,880	148	0.63
Low temp. 1725 rpm	0	9.2	4,020	118	0.60	3,670	134	0.61	3,320	153	0.63
	−5	6.7	3,560	115	0.57	3,250	129	0.58	2,940	149	0.60
	−10	4.5	3,140	112	0.54	2,840	125	0.55	2,600	145	0.57
	−15	2.5	2,740	108	0.51	2,450	120	0.52	2,180	141	0.54
	−20	0.6	2,360	107	0.48	2,090	119	0.49	1,830	138	0.51
	−25	2.28*	2,000	105	0.44	1,750	117	0.45	1,520	135	0.48

¾ hp

Application and Speed	Sat. Suction Temp. (F)	Sat. Suction Press. (Psig)	80 F Btu/Hr	80 F Disch. Press. (Psig)	80 F Kw	90 F Btu/Hr	90 F Disch. Press. (Psig)	90 F Kw	100 F Btu/Hr	100 F Disch. Press. (Psig)	100 F Kw
High temp. 1725 rpm	45	41.7	9,860	148	1.10	9,160	170	1.11	8,480	186	1.16
	40	37.0	9,130	143	1.02	8,470	162	1.05	7,780	181	1.10
	35	32.6	8,460	137	0.95	7,820	154	0.99	7,200	175	1.04
	30	28.5	7,850	132	0.90	7,220	148	0.95	7,610	169	0.99
	25	24.6	7,250	126	0.85	6,650	142	0.90	6,060	162	0.95
Med. temp. 1725 rpm	25	24.6	7,960	140	1.10	7,380	155	1.10	6,830	175	1.13
	20	21.1	7,350	134	0.98	6,780	149	1.04	6,250	169	1.07
	15	17.7	6,610	128	0.95	6,090	142	0.98	5,560	162	1.00
	10	14.7	6,000	123	0.90	5,500	139	0.93	5,000	157	0.96
	5	11.8	5,450	117	0.85	4,980	133	0.88	4,500	151	0.91
	0	9.2	4,900	111	0.80	4,490	127	0.83	4,050	145	0.86
Low temp. 1725 rpm	0	9.2	5,720	118	0.89	5,200	138	0.91	4,730	154	0.92
	−5	6.7	5,100	116	0.83	4,640	133	0.85	4,200	151	0.87
	−10	4.5	4,540	114	0.78	4,090	129	0.80	3,720	148	0.82
	−15	2.5	4,030	111	0.72	3,600	124	0.74	3,200	144	0.76
	−20	0.6	3,550	109	0.67	3,140	122	0.69	2,770	141	0.71
	−25	2.28*	3,090	106	0.62	2,710	120	0.64	2,360	138	0.66

* Inches of Mercury Vacuum.

NOTES:

1. Refrigeration effect is given in Btu per hour. To obtain tons of refrigeration effect, divide by 12,000.
2. Refrigeration effect values given are based upon an actual suction gas temperature of 80°. To obtain this gas temperature usually requires the use of a liquid-suction interchanger.
3. Selection of condensing unit should be made on the basis of the maximum air temperature surrounding the condensing unit. When condenser is not connected directly to outdoors, or where poor ventilation and heat dissipation exists, an ambient temperature higher than the maximum outdoor temperature should be the basis for selection.
4. Operation at suction temperatures lower than those shown is permissible. Operation at suction temperatures higher than those shown will result in overloading of the compressor motor. When low or medium temperature range unit selections are made, it is usually necessary to use some form of suction pressure control to prevent overloading of the compressor motor during pull-down or other abnormal conditions producing a high suction pressure.
5. Power input to motor is given in kilowatts, and includes the power required by the condenser fan. Kilowatt values given are for single phase 60 cycle a-c and d-c motors. To obtain approximate Bhp, divide Kw by 1.04.

TABLE R-11 *(Continued)*

1 hp

Application and Speed	Sat. Suction Temp. (F)	Sat. Suction Press. (Psig)	Ambient Air Temperature (F) 80 F			90 F			100 F		
			Btu/Hr	Disch. Press. (Psig)	Kw	Btu/Hr	Disch. Press. (Psig)	Kw	Btu/Hr	Disch. Press. (Psig)	Kw
High temp. 1725 rpm	45	41.7	13,430	151	1.50	12,500	172	1.55	11,580	190	1.61
	40	37.0	12,450	145	1.41	11,540	164	1.46	10,660	184	1.51
	35	32.6	11,680	139	1.32	10,790	156	1.36	9,920	177	1.40
	30	28.5	10,750	133	1.23	9,860	151	1.27	9,060	171	1.31
	25	24.6	9,900	127	1.13	9,080	143	1.17	8,300	164	1.22
Med. temp. 1725 rpm	25	24.6	10,930	142	1.52	10,140	158	1.57	9,400	179	1.62
	20	21.1	10,100	137	1.41	9,340	152	1.46	8,570	172	1.51
	15	17.7	9,250	131	1.30	8,500	145	1.35	7,753	164	1.40
	10	14.7	8,320	125	1.21	7,620	140	1.26	6,920	159	1.31
	5	11.8	7,410	119	1.11	6,790	135	1.16	6,120	154	1.21
	0	9.2	6,360	113	1.01	5,830	130	1.06	5,260	149	1.11
Low temp. 1725 rpm	0	9.2	8,000	120	1.21	7,300	138	1.24	6,620	157	1.27
	−5	6.7	7,220	118	1.14	6,580	134	1.17	5,960	153	1.20
	−10	4.5	6,400	116	1.07	5,790	130	1.10	5,200	149	1.13
	−15	2.5	5,730	113	0.99	5,120	125	1.02	4,550	145	1.05
	−20	0.6	5,000	110	0.92	4,410	123	0.95	3,880	142	0.98
	−25	2.28*	4,300	106	0.86	3,770	120	0.89	3,290	138	0.92

$1\frac{1}{2}$ hp

Application and Speed	Sat. Suction Temp. (F)	Sat. Suction Press. (Psig)	80 F Btu/Hr	80 F Disch. Press. (Psig)	80 F Kw	90 F Btu/Hr	90 F Disch. Press. (Psig)	90 F Kw	100 F Btu/Hr	100 F Disch. Press. (Psig)	100 F K
			Ambient Air Temperature (F)								
	45	41.7	20,820	148	1.98	19,370	166	2.07	17,920	188	2.18
	40	37.0	19,460	144	1.90	17,900	160	1.98	16,660	182	2.10
High temp. 1725 rpm	35	32.6	18,110	139	1.82	16,720	154	1.90	15,400	175	2.01
	30	28.5	16,670	133	1.75	15,360	148	1.83	14,080	169	1.94
	25	24.6	15,220	126	1.68	13,970	142	1.76	12,760	163	1.87
	25	24.6	15,220	126	1.68	13,970	142	1.76	12,760	163	1.87
	20	21.1	13,700	122	1.62	12,630	137	1.70	11,630	158	1.81
Med. temp. 1725 rpm	15	17.7	12,300	118	1.56	11,310	132	1.64	10,330	152	1.74
	10	14.7	11,050	115	1.51	10,140	128	1.59	9,210	147	1.68
	5	11.8	9,800	111	1.45	8,960	123	1.53	8,090	142	1.62
	0	9.2	8,700	107	1.40	7,870	139	1.48	7,010	137	1.56
	0	9.2	11,300	120	1.73	10,240	135	1.78	9,360	154	1.85
	−5	6.7	10,100	116	1.61	9,200	131	1.67	8,320	149	1.73
Low temp. 1725 rpm	−10	4.5	9,040	113	1.50	8,070	128	1.56	7,330	145	1.62
	−15	2.5	7,970	109	1.39	7,120	124	1.45	6,330	140	1.51
	−20	0.6	6,940	107	1.31	6,150	122	1.37	5,420	137	1.43
	−25	2.28*	5,900	104	1.22	5,170	119	1.28	4,510	134	1.34

* Inches of Mercury Vacuum.

NOTES:

1. Refrigeration effect is given in Btu per hour. To obtain tons of refrigeration effect, divide by 12,000.
2. Refrigeration effect values given are based upon an actual suction gas temperature of 80°. To obtain this gas temperature usually requires the use of a liquid-suction interchanger.
3. Selection of condensing unit should be made on the basis of the maximum air temperature surrounding the condensing unit. When condenser is not connected directly to outdoors, or where poor ventilation and heat dissipation exists, an ambient temperature higher than the *maximum* outdoor temperature should be the basis for selection.
4. Operation at suction temperatures lower than those shown is permissible. Operation at suction temperatures higher than those shown will result in overloading of the compressor motor. When low or medium temperature range unit selections are made, it is usually necessary to use some form of suction pressure control to prevent overloading of the compressor motor during pull-down or other abnormal conditions producing a high suction pressure.
5. Power input to motor is given in kilowatts, and includes the power required by the condenser fan. To obtain approximate Bhp, divide Kw by 0.99.

TABLE R-11 (*Continued*)

2 hp

Application and Speed	Sat. Suction Temp. (F)	Sat. Suction Press. (Psig)	80 F Btu/Hr	80 F Disch. Press. (Psig)	80 F Kw	90 F Btu/Hr	90 F Disch. Press. (Psig)	90 F Kw	100 F Btu/Hr	100 F Disch. Press. (Psig)	100 F Kw
High temp. 1725 rpm	45	41.7	27,260	153	2.69	25,370	173	2.83	23,500	194	2.96
	40	37.0	25,490	148	2.60	23,430	167	2.72	21,830	188	2.85
	35	32.6	23,720	142	2.50	21,880	160	2.61	20,160	181	2.74
	30	28.5	21,860	136	2.40	20,100	154	2.51	18,450	175	2.64
	25	24.6	20,000	130	2.30	18,320	147	2.40	16,750	168	2.53
Med. temp. 1725 rpm	25	24.6	20,000	130	2.30	18,320	147	2.40	16,750	168	2.53
	20	21.1	17,960	126	2.20	16,610	142	2.30	14,300	162	2.42
	15	17.7	16,160	122	2.10	14,870	137	2.20	13,580	156	2.31
	10	14.7	14,490	119	2.00	13,310	133	2.10	12,100	151	2.22
	5	11.8	12,820	115	1.90	11,750	128	2.00	10,600	145	2.12
	0	9.2	11,300	112	1.80	10,400	124	1.90	9,300	140	2.02
Low temp. 1725 rpm	0	9.2	15,870	122	2.43	14,650	138	2.51	13,360	158	2.60
	−5	6.7	14,220	118	2.28	12,960	134	2.36	11,720	153	2.44
	−10	4.5	12,690	115	2.14	11,410	130	2.22	10,290	148	2.30
	−15	2.5	11,160	111	1.99	9,970	126	2.07	8,850	143	2.15
	−10	0.6	9,750	109	1.87	8,640	124	1.95	7,610	140	2.03
	−25	2.28*	8,340	107	1.75	7,310	121	1.83	6,360	136	1.91

3 hp

Application and Speed	Sat. Suction Temp. (F)	Sat. Suction Press. (Psig)	80 F Btu/Hr	80 F Disch. Press. (Psig)	80 F Kw	90 F Btu/Hr	90 F Disch. Press. (Psig)	90 F Kw	100 F Btu/Hr	100 F Disch. Press. (Psig)	100 F Kw
			Ambient Air Temperature (F)								
High temp. 1750 rpm	45	41.7	43,050	152	4.02	40,000	173	4.20	37,050	191	4.40
	40	37.0	30,230	147	3.86	37,000	165	4.03	34,430	185	4.24
	35	32.6	37,400	141	3.70	34,560	158	3.86	31,800	179	4.07
	30	28.5	34,500	136	3.56	31,750	152	3.72	29,150	173	3.93
	25	24.6	31,600	130	3.42	28,950	146	3.58	26,450	166	3.78
Med. temp. 1750 rpm	25	24.6	31,600	130	3.42	28,950	146	3.58	26,450	166	3.78
	20	21.1	28,350	126	3.30	26,250	141	3.46	24,120	160	3.66
	15	17.7	25,550	122	3.17	23,500	136	3.34	21,460	154	3.53
	10	14.7	22,900	119	3.07	21,000	131	3.23	19,100	149	3.43
	5	11.8	20,200	115	2.96	18,510	127	3.13	16,700	143	3.32
	0	9.2	17,000	112	2.50	15,500	123	3.03	13,900	138	3.20
Low temp. 1750 rpm	0	9.2	20,080	120	3.08	18,480	138	3.20	16,800	159	3.40
	−5	6.7	17,980	116	2.86	16,380	133	2.98	14,810	152	3.10
	−10	4.5	16,020	113	2.66	14,410	129	2.78	13,000	146	2.90
	−15	2.5	14,050	110	2.46	12,550	125	2.58	11,170	141	2.70
	−20	0.6	12,240	109	2.29	10,850	123	2.42	9,560	138	2.54
	−25	2.28*	10,420	107	2.13	9,150	120	2.25	7,960	135	2.37

* Inches of Mercury Vacuum.

NOTES:

1. Refrigeration effect is given in Btu per Hour. To obtain tons of refrigeration effect, divide by 12,000.
2. Refrigeration effect values given are based upon an actual suction gas temperature of 80°. To obtain this gas temperature usually requires the use of a liquid-suction interchanger.
3. Selection of condensing unit should be made on the basis of the maximum air temperature surrounding the condensing unit. When condenser is not connected directly to outdoors, or where poor ventilation and heat dissipation exists, an ambient temperature higher than the maximum outdoor temperature should be the basis for selection.
4. Operation at suction temperatures lower than those shown is permissible. Operation at suction temperatures higher than those shown will result in overloading of the compressor motor. When low or medium temperature range unit selections are made, it is usually necessary to use some form of suction pressure control to prevent overloading of the compressor motor during pull-down or other abnormal conditions producing a high suction pressure.
5. Power input is given in kilowatts, and includes the power required by the condenser fan. Kilowatt values given are for three phase 60 cycle a-c motors. To obtain approximate Bhp, divide Kw by 0.93.

TABLE R-12. Water-Cooled Condensing Units

Usage	Sat. Suction Temp. (F)	Sat. Suction Press. (Psig)	1/3 hp Btu/Hr	1/3 hp Disch. Press. (Psig)	1/3 hp Kw	1/2 hp Btu/Hr	1/2 hp Disch. Press. (Psig)	1/2 hp Kw	3/4 hp Btu/Hr	3/4 hp Disch. Press. (Psig)	3/4 hp Kw	1 hp Btu/Hr	1 hp Disch. Press. (Psig)	1 hp Kw
	45	41.7	4,300	130	0.43	7,580	129	0.80	10,890	131	1.25	14,810	133	1.49
High temp.	40	37.0	3,890	127	0.42	6,860	126	0.77	9,860	128	1.20	13,410	130	1.45
75° Water In	35	32.6	3,550	125	0.41	6,250	124	0.75	8,980	126	1.16	12,230	127	1.40
95° Water Out	30	28.5	3,260	123	0.40	5,740	122	0.72	8,250	123	1.01	11,230	125	1.36
	25	24.6	3,000	120	0.39	5,280	120	0.70	7,580	121	0.97	10,320	123	1.32
	25	24.6	3,970	117	0.44	6,400	118	0.70	8,720	119	0.95	12,030	120	1.31
Med. Temp.	20	21.1	3,630	115	0.43	5,860	116	0.68	7,990	117	0.92	11,010	118	1.27
75° Water In	15	17.7	3,300	113	0.42	5,310	113	0.66	7,250	114	0.89	10,000	116	1.23
90° Water Out	10	14.7	2,970	112	0.41	4,770	112	0.64	6,510	112	0.86	8,980	114	1.19
	5	11.8	2,630	110	0.40	4,230	110	0.62	5,770	110	0.83	7,960	112	1.15
	0	9.2	2,290	108	0.39	3,680	109	0.60	5,030	108	0.80	6,940	110	1.11
	0	9.2	2,790	107	0.42	4,360	108	0.62	5,950	111	0.90	8,510	110	1.24
Low temp.	− 5	6.7	2,390	105	0.41	3,740	106	0.60	5,100	108	0.87	7,290	108	1.19
75° Water In	−10	4.5	2,030	103	0.40	3,180	104	0.59	4,330	105	0.80	6,200	106	1.14
85° Water Out	−15	2.5	1,720	101	0.39	2,700	102	0.57	3,670	102	0.76	5,260	103	1.09
	−20	0.6	1,440	99	0.38	2,250	100	0.56	3,060	100	0.71	4,380	101	1.04
	−25	2.28*	1,180	96	0.37	1,850	97	0.54	2,520	97	0.66	3,600	98	0.99

Usage	Sat. Suction Temp. (F)	Sat. Suction Press. (Psig)	1½ hp Btu/Hr	1½ hp Disch. Press. (Psig)	1½ hp Kw	2 hp Btu/Hr	2 hp Disch. Press. (Psig)	2 hp Kw	3 hp Btu/Hr	3 hp Disch. Press. (Psig)	3 hp Kw
	45	41.7	23,000	131	1.65	29,980	134	2.15	47,300	135	3.25
High temp.	40	37.0	20,800	129	1.60	27,150	131	2.08	42,850	132	3.10
75° Water In	35	32.6	18,970	126	1.55	24,740	128	2.00	39,080	129	2.95
95° Water Out	30	28.5	17,410	124	1.50	22,700	126	1.93	35,850	127	2.82
	2!	24.6	16,020	122	1.45	20,850	124	1.85	32,940	124	2.68
	25	24.6	16,020	122	1.45	20,850	124	1.85	32,940	124	2.68
Med. temp.	20	21.1	14,630	119	1.40	19,080	122	1.72	30,120	122	2.56
75° Water In	15	17.7	13,300	116	1.35	17,330	119	1.60	27,350	119	2.43
90° Water Out	10	14.7	11,930	115	1.30	15,670	117	1.52	24,550	117	2.30
	5	11.8	10,580	114	1.25	13,800	115	1.45	21,780	115	2.18
	0	9.2	9,220	113	1.20	12,010	113	1.38	18,980	113	2.06
	0	9.2	12,020	109	1.58	16,870	109	2.09	21,080	112	2.95
Low Temp.	−5	6.7	10,300	107	1.50	14,460	106	1.96	18,060	109	2.80
75° Water In	−10	4.5	8,740	105	1.43	12,460	104	1.83	15,620	106	2.65
85° Water Out	−15	2.5	7,550	102	1.35	10,660	101	1.70	13,320	103	2.50
	−20	0.6	6,490	100	1.29	9,090	99	1.57	11,410	101	2.35
	−25	2.28*	5,430	97	1.23	7,690	97	1.44	9,610	99	2.20

* Inches of Mercury Vacuum.

NOTES:

1. Refrigeration effect is given in Btu per Hr. To obtain tons of refrigeration, divide by 12,000.
2. Refrigeration effect values given are based upon an actual suction gas temperature of 65° F. To obtain this gas temperature usually requires the use of a liquid-suction interchanger.
3. Operation at suction temperatures lower than those shown is permissible. Operation at suction temperatures higher than shown will result in overloading of the compressor motor. When low or medium temperature range unit selections are made, it is usually necessary to use some form of suction pressure control to prevent overloading of the compressor motor during pull-down or other abnormal conditions producing high suction pressure.
4. For each 10° lower entering water temperature, increase above capacities 6%
   For each 10° higher entering water temperature, decrease above capacities 6%
5. Power input to motors is given in kilowatts. To obtain approximate Bhp divide Kw by the factor from the table below:

Motor hp	Factor	Motor hp	Factor
⅓	1.09	1½	0.96
½	1.07	2	0.94
¾	1.04	3	0.92
1	1.02		

6. Condenser water quantity (gpm) at full load $= \dfrac{\text{Btu/hr} \times F}{(\text{leaving water temp.} - \text{entering water temp.}) \times 500}$

Where F = 1.2 for high temperature usage
F = 1.3 for medium temperature usage
F = 1.4 for low temperature usage

TABLE R-13. Air-Cooled Condenser Ratings

TABLE A. Basic Rating Table 40° Suction 90° Air Entering

Model No.	Compr. Capacity-Btu/Hr 120° Condensing Temp.				Fan Motors				Fan Speed
	Tons	Btu/hr	Cfm	Dia. Inches	HP	Total Watts	Amps	Rpm	Rpm
RC75	0.88	10,600	1,125	14	$\frac{1}{25}$	78		1,500	1,500
RC100	1.11	13,340	1,450	16	$\frac{1}{15}$	175		1,050	1,050
RC150	1.61	19,300	2,450	18	$\frac{1}{6}$	230		1,140	1,140
RC200	2.36	28,300	2,850	20	$\frac{1}{6}$	230		1,140	1,140
RC300	3.18	38,200	3,400	20	$\frac{1}{6}$	330		1,140	1,140
BD300	3.31	39,800	3,450	24	$\frac{1}{3}$		1.3	1,725	625
RC500	5.14	61,700	5,500	20	2–$\frac{1}{6}$	460		1,140	1,140
BD500	5.13	61,600	5,500	30	$\frac{1}{2}$		1.7	1,725	500
RC750	7.92	95,000	6,900	24	2–$\frac{1}{4}$	860		1,140	1,140
BD750	7.57	90,900	7,100	36	$\frac{3}{4}$		2.8	1,725	455
RC1000	10.34	124,000	9,100	20	3–$\frac{1}{4}$	1,290		1,140	1,140
BD1000	10.00	120,000	8,800	36	1		3.2	1,725	485
BD1500	17.32	208,000	14,750	48	$1\frac{1}{2}$		4.8	1,725	325
BD2000	19.75	237,000	13,600	48	$1\frac{1}{2}$		4.8	1,725	325
BD3000	34.64	416,000	29,500	48	2–$1\frac{1}{2}$		9.6	1,725	325
BD4000	39.50	474,000	27,200	48	2–$1\frac{1}{2}$		9.6	1,725	325

Courtesy Kramer-Trenton Company.

TABLE B. Correction Factors for Suction Temperature Lower than 40°

Suction temp. °F	−30	−20	−10	0	+10	+20	+30	+40
Conversion factor	0.76	0.81	0.85	0.89	0.92	0.95	0.98	1.00

Courtesy Kramer-Trenton Company.

TABLE C. Correction Factors for Temp. Diff. (Condensing temp.—ent. air. temp.)

Entering Air D.B.	Condensing Temperature ° F						
	100	105	110	115	120	125	130
70	1.00	1.17	1.33	1.50	1.67	1.83	2.00
80	0.665	0.834	1.00	1.17	1.33	1.5	1.67
90	0.333	0.50	0.665	0.834	1.00	1.17	1.33
100	—	—	0.333	0.50	0.665	0.834	1.00

Courtesy Kramer-Trenton Company.

TABLE R-14. Refrigerant Condensers—Capacity and Engineering Data

Shell O.D. Inches	No. of Tubes	Model Number ▲	Total Effective Sq. Ft. of Surface	Nominal Rating—Tons*: 75°—95° Water 102° Cond. Temp. Capacity Tons*		Water P.D. PSI		Nominal Rating—Tons*: 85°—95° Water 105° Cond. Temp. Capacity Tons*		Water P.D. PSI		Pump Down Capacity Pounds Freon-12 **	Nominal Operating Charge lbs. F-12 ***
				4-Pass.	2-Pass.	4-Pass.	2-Pass.	4-Pass.	2-Pass.	4-Pass.	2-Pass.		
		STF 84	73.5	5.41	‡	.24	‡	9.93	‡	2.4	‡	61	8.2
		STF 85	93.4	9.38	‡	.72	‡	14.4	8.7	5.1	.35	77	10
		STF 86	113	13.6	‡	1.50	‡	19.3	12.7	9.2	.74	93	13
		STF 87	133	17.9	‡	2.65	‡	§	16.8	§	1.3	109	15
		STF 88	153	22.4	12.2	4.2	.20	§	21.1	§	2.1	125	17
		STF 89	173	26.9	16.0	6.3	.40	§	25.5	§	3.0	141	19
8⅝	40	STF 810	193	31.7	20.1	8.7	.64	§	30.2	§	4.3	158	21
		STF 811	213	36.6	24.3	12.2	.95	§	35.1	§	6.0	174	24
		STF 812	232	41.6	28.5	15.9	1.30	§	40.2	§	7.8	190	26
		STF 813	252	§	32.8	§	1.78	§	§	§	§	206	28
		STF 814	272	§	37.2	§	2.30	§	§	§	§	222	30
		STF 815	292	§	41.6	§	2.95	§	§	§	§	238	32
		STF 816	312	§	46.1	§	3.65	§	§	§	§	254	34
		STF 104	110	8.1	‡	.24	‡	14.9	‡	2.4	‡	98	15
		STF 105	140	14.1	‡	.72	‡	21.6	13.0	5.1	.35	124	18
		STF 106	170	20.4	‡	1.50	‡	29.0	19.1	9.2	.74	150	22
		STF 107	200	26.9	‡	2.65	‡	§	25.2	§	1.3	175	26
		STF 108	229	33.6	18.3	4.2	.20	§	31.7	§	2.1	201	30
		STF 109	259	40.4	24.0	6.3	.40	§	38.2	§	3.0	226	34
10¾	60	STF 1010	289	47.5	30.2	8.7	.64	§	45.3	§	4.3	252	38
		STF 1011	319	54.9	36.4	12.2	.95	§	52.6	§	6.0	278	42
		STF 1012	349	62.5	42.7	15.9	1.30	§	60.3	§	7.8	304	46
		STF 1013	378	§	49.2	§	1.78	§	§	§	§	330	50
		STF 1014	408	§	55.8	§	2.30	§	§	§	§	355	53
		STF 1015	438	§	62.5	§	2.95	§	§	§	§	381	57
		STF 1016	468	§	69.2	§	3.65	§	§	§	§	407	61
		STF 124	169	12.4	‡	.24	‡	22.8	‡	2.4	‡	136	20
		STF 125	215	21.6	‡	.72	‡	33.1	20.0	5.1	.35	172	26
		STF 126	260	31.3	‡	1.50	‡	44.4	29.2	9.2	.74	207	31
		STF 127	306	41.2	‡	2.65	‡	§	38.7	§	1.3	243	36
		STF 128	352	51.5	28.0	4.2	.20	§	48.6	§	2.1	279	42
		STF 129	397	61.9	36.8	6.3	.40	§	58.7	§	3.0	315	47
		STF 1210	443	73.0	46.3	8.7	.64	§	69.5	§	4.3	351	53
		STF 1211	489	84.3	55.9	12.2	.95	§	80.7	§	6.0	386	58
12¾	92	STF 1212	535	95.8	65.6	15.9	1.30	§	92.5	§	7.8	422	63
		STF 1213	580	§	75.5	§	1.78	§	§	§	§	458	69
		STF 1214	626	§	85.6	§	2.30	§	§	§	§	494	74
		STF 1215	672	§	95.7	§	2.95	§	§	§	§	530	80
		STF 1216	718	§	106	§	3.65	§	§	§	§	566	85
		STF 1217	763	§	117	§	4.5	§	§	§	§	602	90
		STF 1218	809	§	127	§	5.5	§	§	§	§	638	96
		STF 1219	854	§	138	§	6.6	§	§	§	§	674	101
		STF 1220	900	§	149	§	7.8	§	§	§	§	710	107

		STF 144	220	16.2	‡	.24	‡	29.8	‡	2.4	‡	159	26
		STF 145	280	28.1	‡	.72	‡	43.2	26.1	5.1	.35	202	33
		STF 146	339	40.8	‡	1.50	‡	57.9	38.1	9.2	.74	244	40
		STF 147	399	53.7	‡	2.65	‡	§	50.4	§	1.3	287	46
		STF 148	458	67.2	36.6	4.2	.20	§	63.3	§	2.1	329	53
		STF 149	518	80.7	48.0	6.3	.40	§	76.5	§	3.0	372	60
		STF 1410	578	95.1	60.3	8.7	.64	§	90.6	§	4.3	414	67
		STF 1411	637	110	72.9	12.2	.95	§	105	§	6.0	457	74
14	120	STF 1412	697	125	85.5	15.9	1.30	§	121	§	7.8	499	81
		STF 1413	757	§	98.4	§	1.78	§	§	§	§	542	88
		STF 1414	816	§	111.7	§	2.30	§	§	§	§	584	94
		STF 1415	876	§	125	§	2.95	§	§	§	§	626	101
		STF 1416	936	§	138	§	3.65	§	§	§	§	669	108
		STF 1417	995	§	152	§	4.5	§	§	§	§	712	115
		STF 1418	1055	§	166	§	5.5	§	§	§	§	754	122
		STF 1419	1115	§	180	§	6.6	§	§	§	§	797	129
		STF 1420	1174	§	195	§	7.8	§	§	§	§	839	136
		STF or SRF 164	301	22.2	‡	.24	‡	40.7	‡	2.4	‡	208	35
		STF or SRF 165	382	38.5	‡	.72	‡	59.0	35.7	5.1	.35	264	44
		STF or SRF 166	464	55.8	‡	1.50	‡	79.1	52.0	9.2	.74	319	53
		STF or SRF 167	545	73.4	‡	2.65	‡	§	68.9	§	1.3	374	62
		STF or SRF 168	627	91.8	50.0	4.2	.20	§	86.5	§	2.1	430	72
		STF or SRF 169	708	110	65.6	6.3	.40	§	105	§	3.0	485	81
		STF or SRF 1610	790	130	82.5	8.7	.64	§	124	§	4.3	541	90
		STF or SRF 1611	871	150	99.7	12.2	.95	§	144	§	6.0	596	99
16	164	STF or SRF 1612	953	171	117	15.9	1.30	§	165	§	7.8	652	108
		STF or SRF 1613	1034	§	135	§	1.78	§	§	§	§	707	118
		STF or SRF 1614	1116	§	153	§	2.30	§	§	§	§	763	127
		STF or SRF 1615	1197	§	171	§	2.95	§	§	§	§	818	136
		STF or SRF 1616	1278	§	189	§	3.65	§	§	§	§	874	145
		STF or SRF 1617	1360	§	208	§	4.5	§	§	§	§	929	155
		STF or SRF 1618	1441	§	227	§	5.5	§	§	§	§	983	164
		STF or SRF 1619	1523	§	246	§	6.6	§	§	§	§	1038	173
		STF or SRF 1620	1604	§	266	§	7.8	§	§	§	§	1092	182

NOTES:
* Nominal tons rating are based on a scale factor of 0.0005.
** Pump down capacity is based on 80% of the free volume in the shell with R-12 at 102° F.
*** Condenser capacities based on free drainage of condensed liquid from shell. Normal operating charge is based on allowances for R-12 vapor in shell, liquid film on tubes, and liquid in bottom of shell.
‡ Water rate below turbulent flow.
§ Water velocity exceeds maximum of 8' sec, prescribed in ACRMA standards. All Model STF condensers are furnished as standard with dual pass heads for 2 or 4 passes. All Model SRF condensers are available in either 4-pass or 2-pass construction. The number of passes required must be specified on all orders.

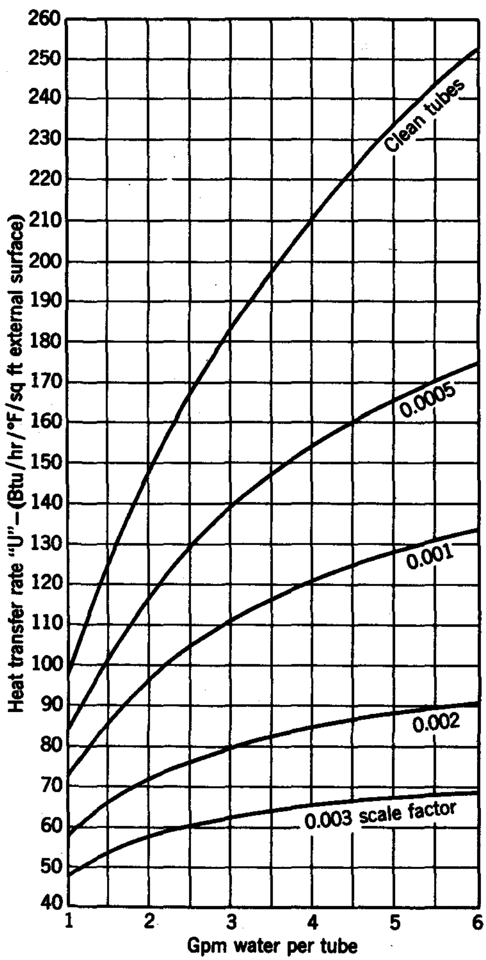

Heat transfer rate "U"—(Btu/hr/°F/sq ft external surface)
260
250
240
230
220
210
200
190
180
170
160
150
140
130
120
110
100
90
80
70
60
50
40
Clean tubes
0.0005
0.001
0.002
0.003 scale factor
1
2
3
4
5
6
Gpm water per tube

Fig. 1

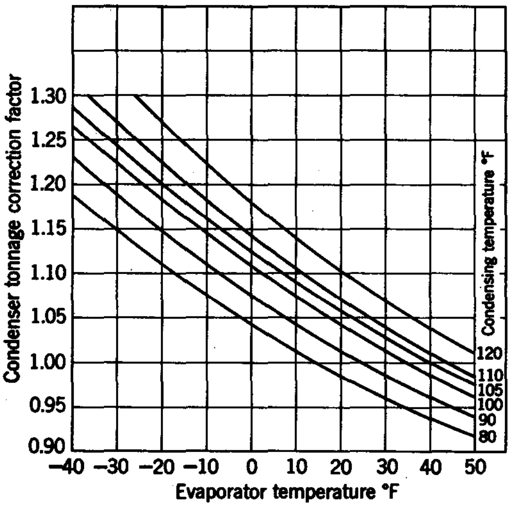


Fig. 2

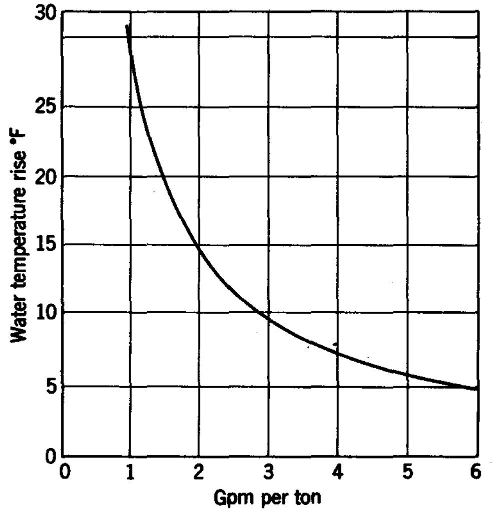


Fig. 3

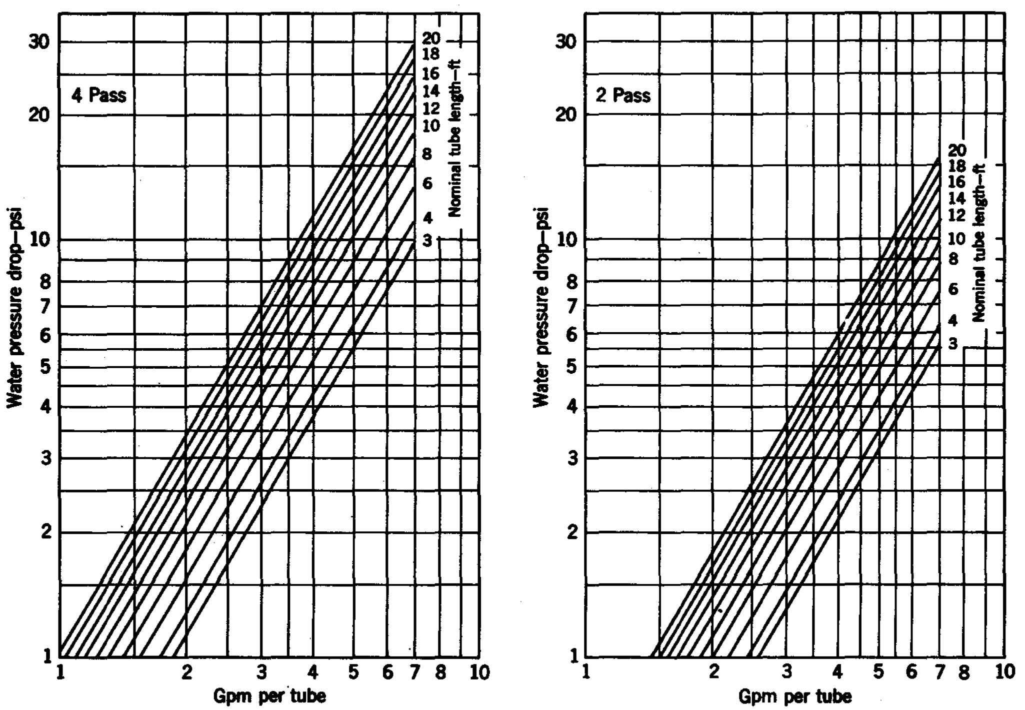


Fig. 4

TABLE R-15. Quick Selection Table—Water-Cooled Condensers

Catalog Number	Stock Number	*Nominal HP Rating	Size and Type of Connections: Water In and Out SAE Flare	Refrigerant In SAE Flare	Refrigerant Out SAE Flare	Dimension in Inches: Height	Length	Depth	No. of Sections	Shipping Weight (Appr.)	Cleaning Tool Catalog Number
EL-33	1-EL	$\frac{1}{3}$	$\frac{1}{2}$″	$\frac{1}{2}$	$\frac{3}{8}$	$8\frac{3}{4}$	18	1	1	13	836
EL-50	2-EL	$\frac{1}{2}$	$\frac{1}{2}$″	$\frac{1}{2}$	$\frac{3}{8}$	$8\frac{3}{4}$	21	1	1	18	836
EL-75	3-EL	$\frac{3}{4}$	$\frac{1}{2}$″	$\frac{1}{2}$	$\frac{3}{8}$	$10\frac{3}{4}$	21	1	1	20	836
EL-100	4-EL	1	$\frac{1}{2}$″	$\frac{5}{8}$	$\frac{3}{8}$	$10\frac{3}{4}$	27	1	1	25	836
EL-150	5-EL	$1\frac{1}{2}$	$\frac{1}{2}$″	$\frac{5}{8}$	$\frac{3}{8}$	$12\frac{3}{4}$	33	1	1	30	836
				O.D. Swt.							
EL-200	6-EL	2	$\frac{1}{2}$″	$\frac{7}{8}$	$\frac{1}{2}$	14	34	1	1	35	1036
EL-300	7-EL	3	$\frac{1}{2}$″	$\frac{7}{8}$	$\frac{1}{2}$	$16\frac{1}{4}$	34	1	1	39	1036

* For Booster application—Use one size smaller when used in combination with air-cooled condensers.
Courtesy Halstead Mitchell.

TABLE R-16. Ratings for Atmospheric Cooling Tower

Capacities based on 3 mph wind velocity

		Refrigeration						Tons			20° Range—14° Approach			
		5 Gpm	Per Ton	4 Gpm per Ton				3 Gpm per Ton						
	Tower	91	93	90½	93½	92½	94½	90	95	96	Gas and			Comp.
	Basin	85	87	83	86	85	87	80	85	86	Gasoline	Diesel	Steam	Air
	W.B.	78	80	75	78	78	80	70	75	78	Engine	Engine	Condensing	100 lb
Tower No.	Nom. Gpm						(Nom.)				HP	HP	lb/hr	Cfm
CSA 23	12	2.7	2.6	2.8	3.1	2.9	3.0	3.4	3.8	3.4	20	33	100	670
CSA 33	20	4.5	4.4	4.8	5.1	4.8	5.0	5.7	6.4	5.8	30	50	150	1,000
CSA 34	24	5.4	5.3	5.7	6.1	5.7	6.0	6.8	7.7	6.9	40	67	200	1,330
CSA 44	32	7.2	7.0	7.6	8.2	7.7	8.0	9.1	10.2	9.2	50	83	250	1,670
CSA 45	40	9.0	8.8	9.5	10.2	9.6	10.0	11.4	12.8	11.5	60	100	300	2,000
CSA 55	50	11.2	11.0	11.8	12.8	12.0	12.5	14.2	16.0	14.4	80	133	400	2,670
CSA 66	72	16.2	15.8	17.1	18.4	17.2	18.0	20.5	23.0	20.7	110	183	550	3,670
SA 33	22	5.0	4.8	5.2	5.6	5.3	5.5	6.3	7.0	6.3	40	67	200	1,330
SA 34	30	6.7	6.6	7.1	7.7	7.2	7.5	8.6	9.6	8.6	50	83	250	1,670
SA 44	40	9.0	8.8	9.5	10.2	9.6	10.0	11.4	12.8	11.5	60	100	300	2,000
SA 45	50	11.2	11.0	11.8	12.8	12.0	12.5	14.2	16.0	14.4	80	133	400	2,670
SA 46	60	13.5	13.2	14.2	15.3	14.4	15.0	17.1	19.2	17.1	100	167	500	3,330
SA 56	75	16.9	16.5	17.8	19.2	18.0	18.8	21.4	24.0	21.6	120	200	600	4,000
SA 58	100	22.5	22	23.7	25.5	24	25.0	28.5	32.0	29	160	267	800	5,330
SA 68	120	27	26	28	31	29	30.0	34	38	34	200	333	1,000	6,670
SA 610	150	34	33	36	38	36	37.5	43	48	43	250	410	1,250	8,200
SA 612	180	41	40	43	46	43	45.0	51	57	52	300	500	1,500	10,000
SA 615	225	50	49	53	57	54	56.3	64	72	64	350	580	1,750	11,600
SA 616	240	54	53	57	61	57	60.0	68	77	69	400	670	2,000	13,400
SA 618	270	61	60	65	70	65	67.5	77	87	78	450	750	2,250	15,000
SA 620	300	67	66	71	77	72	75.0	86	96	86	500	830	2,500	16,600
SA 624	360	81	79	85	92	86	90.0	103	115	104	600	1,000	3,000	20,000
SA 824	400	90	88	95	102	96	100.0	114	128	115	700	1,167	3,500	23,300
SA1224	450	102	100	107	115	108	112.5	129	144	130	800	1,330	4,000	26,700
SA1230	550	124	121	131	140	132	137.5	157	176	158	1,000	1,670	5,000	33,300
SA1236	650	146	143	154	166	156	162.5	186	208	187	1,200	2,000	6,000	40,000
SA1242	750	169	165	178	192	180	187.5	214	240	216	1,400	2,330	7,000	46,700
SA1248	850	192	187	202	218	204	212.5	243	272	245	1,600	2,670	8,000	53,300
SA1254	950	213	210	226	243	228	237.5	271	305	274	1,800	3,000	9,000	60,000
SA1260	1100	247	240	261	280	263	275.0	314	350	315	2,000	3,330	10,000	66,700
SA1266	1200	270	260	280	310	290	300.0	340	380	340	2,200	3,670	11,000	73,300

Courtesy Star Cooling Towers, Inc.

TABLE R-17. Evaporative Condenser Ratings

TABLE A. (In terms of evaporator load at +40° F evaporator)

Model No.	Cond. Temp. (° F)	Wet Bulb Temperature of Entering Air (°F) 60°	65°	70°	75°	78°	80°	85°
	90	2.6	2.2	1.9	1.5	1.2	1.1	0.6
	95	3.2	2.9	2.5	2.1	1.9	1.7	1.2
E-80F	100	3.8	3.5	3.2	2.8	2.5	2.3	1.9
	105	4.5	4.2	3.9	*3.5	3.3	3.1	2.6
	110	5.3	5.1	4.8	4.4	4.1	4.0	3.5
	115	6.3	6.0	5.7	5.3	5.1	4.9	4.5
	90	4.4	3.9	3.3	2.6	2.1	1.8	0.9
	95	5.5	5.0	4.3	3.7	3.3	2.9	2.1
E-135F	100	6.6	6.1	5.5	4.8	4.4	4.1	3.2
	105	7.8	7.3	6.7	*6.0	5.6	5.4	4.5
	110	9.4	8.9	8.3	7.6	7.2	6.9	6.1
	115	11.0	10.5	9.9	9.3	8.9	8.6	7.8
	90	8.8	7.8	6.6	5.2	4.2	3.6	1.8
	95	11.0	10.0	8.6	7.4	6.6	5.8	4.2
E-270F	100	13.2	12.2	11.0	9.6	8.8	8.2	6.4
	105	15.6	14.6	13.4	*12.0	11.2	10.8	9.0
	110	18.8	17.8	16.6	15.2	14.4	13.8	12.2
	115	22.0	21.0	19.8	18.6	17.8	17.2	15.6

* ASRE standard rating conditions.

TABLE B

Evaporator Temperature Correction Factors

Evaporator Temp. (° F)	Correction Factor	Evaporator Temp. (° F)	Correction Factor
50	0.97	0	1.11
40	1.0	−10	1.16
30	1.03	−20	1.20
20	1.05	−30	1.26
10	1.09		

Courtesy McQuay, Inc.

TABLE R-18. Water Valve Selection Table

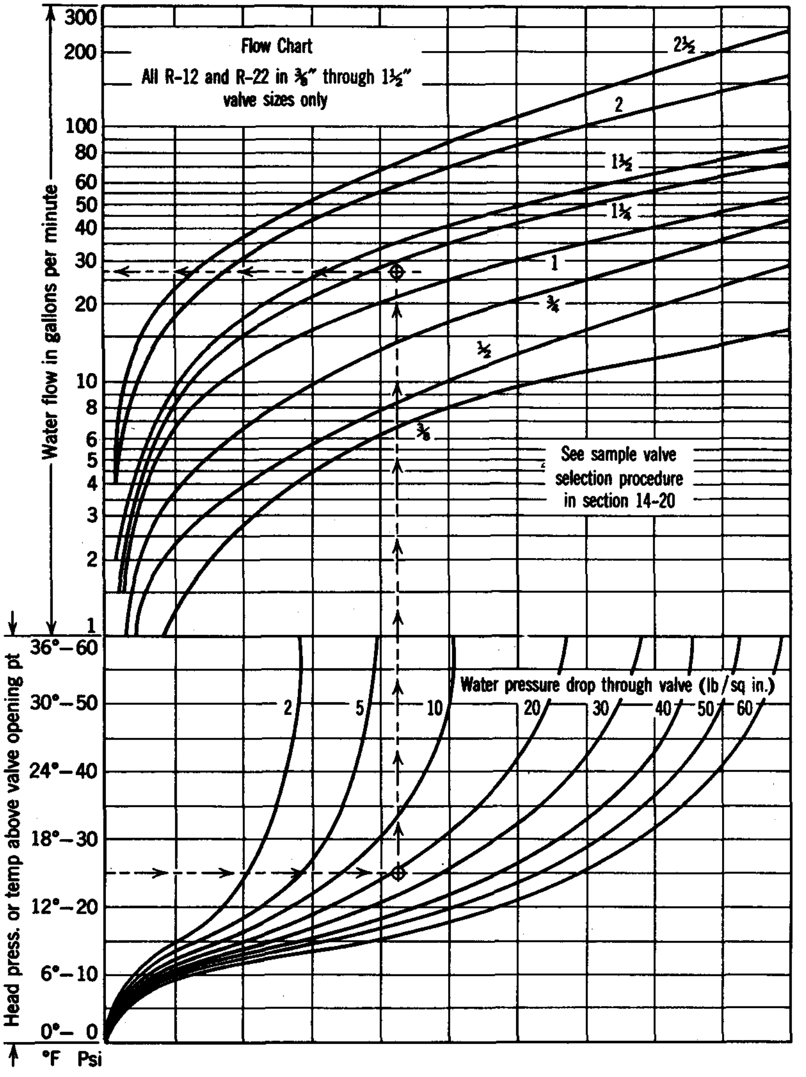
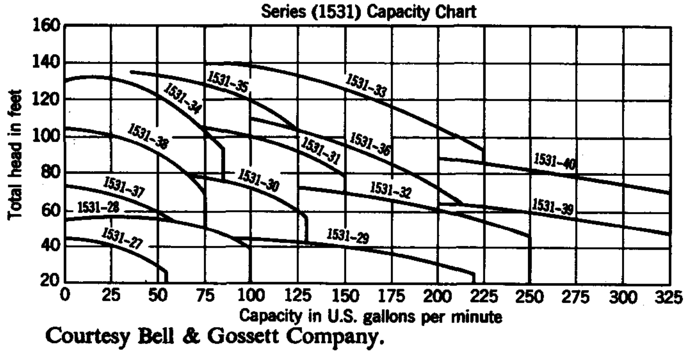


TABLE R-19. Centrifugal Pump Capacity Table



Courtesy Bell & Gossett Company.

TABLE R-20. Thermostatic Expansion Valve Selection Table

New Type No.	EVAPORATOR TEMPERATURE °F.																							
	40°				20°				0°				–10°				–20°				–30°			
	PRESSURE DIFFERENCE ACROSS VALVE, PSI																							
	40	60	80	100	60	80	100	120	75	100	125	150	75	100	125	150	75	100	125	150	75	100	125	150
	TONS OF REFRIGERATION																							
TK25F	.24	.30	.35	.39	.22	.25	.28	.31	.17	.20	.22	.25	.15	.17	.19	.21	.12	.14	.16	.18	.10	.12	.14	.16
TK50F	.49	.60	.70	.78	.44	.50	.56	.62	.34	.40	.44	.50	.30	.34	.38	.42	.24	.28	.32	.36	.20	.24	.28	.32
TK100F	1.1	1.3	1.5	1.7	1.0	1.1	1.2	1.3	.75	.86	.97	1.1	.63	.72	.81	.89	.52	.60	.67	.73	.42	.49	.55	.60
TK200F	1.9	2.2	2.5	2.9	1.7	1.9	2.1	2.2	1.3	1.5	1.7	1.9	1.1	1.2	1.4	1.5	.90	1.0	1.1	1.2	.71	.83	.93	1.1
TK300F	2.8	3.3	3.8	4.3	2.5	2.8	3.0	3.3	1.9	2.2	2.5	2.8	1.6	1.8	2.0	2.2	1.3	1.5	1.7	1.9	1.1	1.2	1.4	1.5
TL50F TCL50F	.49	0.6	.70	.78	.44	.50	.56	.62	.34	.40	.44	.50	.30	.34	.38	.42	.24	.28	.32	.36	.20	.24	.28	.32
TL100F TCL100F	1.1	1.3	1.5	1.7	1.0	1.1	1.2	1.3	.75	.86	.97	1.1	.63	.72	.81	.89	.52	.60	.67	.73	.42	.49	.55	.60
TL200F TCL200F	1.9	2.3	2.6	3.0	1.7	1.9	2.2	2.4	1.3	1.5	1.7	1.9	1.1	1.3	1.4	1.6	.91	1.1	1.2	1.3	.75	.86	.97	1.1
TL300F TCL300F	2.9	3.5	4.0	4.5	2.6	3.0	3.3	3.6	2.0	2.3	2.6	2.9	1.7	2.0	2.2	2.4	1.4	1.6	1.8	2.0	1.1	1.3	1.5	1.6
TL400F TCL400F	3.5	4.3	5.0	5.6	3.2	3.6	4.0	4.5	2.5	2.9	3.2	3.5	2.1	2.4	2.7	2.9	1.7	2.0	2.2	2.4	1.4	1.6	1.8	2.0
TL600F TCL600F	4.9	6.0	7.0	7.8	4.4	5.0	5.7	6.2	3.5	4.0	4.5	4.9	2.9	3.3	3.7	4.1	2.4	2.7	3.1	3.4	1.9	2.2	2.5	2.7
TAL650F TDL650F	5.3	6.5	7.5	8.4	4.7	5.5	6.1	6.7	3.8	4.4	4.9	5.4	3.1	3.6	4.1	4.4	2.6	3.0	3.3	3.6	2.1	2.4	2.7	3.0
TJL800F	6.9	8.5	9.8	11.0	6.2	7.2	8.0	8.8	4.9	5.7	6.4	6.9	4.1	4.8	5.3	5.8	3.4	3.9	4.4	4.8	2.8	3.2	3.6	3.9
TJL1100F	9.0	11.0	12.7	14.2	8.1	9.3	10.4	11.4	6.4	7.4	8.3	9.1	5.3	6.1	6.9	7.5	4.4	5.0	5.6	6.2	3.6	4.1	4.6	5.0
TEL1400F	11.4	14.0	16.1	18.1	10.3	11.8	13.2	14.6	8.1	9.3	10.5	11.4	6.8	7.8	8.8	9.6	5.6	6.4	7.2	7.9	4.5	5.2	5.9	6.4
TEL1600F	13.5	16.5	19.0	21.3	12.1	13.9	15.6	17.2	9.5	11.0	12.3	13.5	8.0	9.2	10.4	11.2	6.6	7.6	8.5	9.3	5.3	6.2	6.9	7.6
TEL2200F	17.9	22.0	25.4	28.4	16.1	18.6	20.8	22.7	12.8	14.8	16.6	18.1	10.6	12.3	13.8	15.0	8.7	10.0	11.3	12.3	7.1	8.2	9.2	10.1
TIL2700F	22.0	27.0	31.2	34.8	19.8	22.8	25.4	27.9	15.7	18.2	20.3	22.2	13.1	15.0	16.9	18.4	10.7	12.3	13.8	15.1	8.8	10.1	11.3	12.3
TIL3300F	26.9	33.0	38.0	42.6	24.3	27.9	31.2	34.3	19.0	22.0	24.7	26.9	16.0	18.5	20.8	22.6	13.1	15.1	17.0	18.5	10.7	12.3	13.9	15.1
THL4200F	34.3	42.0	48.5	54.2	30.8	35.5	39.6	43.4	24.4	28.2	31.6	34.6	20.3	23.4	26.2	28.7	16.6	19.1	21.4	23.5	13.6	15.7	17.6	19.2
THL5000F	40.8	50.0	57.6	64.6	36.8	42.3	47.3	52.0	28.9	33.3	37.4	40.8	24.3	28.0	31.5	34.3	19.9	22.9	25.7	28.1	16.2	18.7	21.0	22.9

*New valve size.

Based on 100° F. condensing temperature, 1° subcooled liquid, and a maximum superheat change of 4° F. (Temperature rise of remote bulb required to move valve pin from closed to rated open position.) For each 10° subcooling, the capacities are increased approximately 6%.

Courtesy Alco Valve Company.

Index

TABLES

CHARTS

EQUIPMENT RATING TABLES

This Wiley International Edition is part of a continuing program of paperbound text books especially designed for students and professional people overseas. It is an unabridged reprinting of the original hardbound edition, which is also available from your bookseller.

Wiley International Editions include titles in the fields of:

- Chemical Engineering
- Civil Engineering
- Electrical Engineering
- Mechanics
- Metallurgy
- Mechanical Engineering
- Business Management
- Statistics
- Psychology
- Agricultural Engineering and Agriculture
- Biological Science
- Chemistry
- Earth Sciences
- Mathematics
- Physics

JOHN WILEY & SONS, INC., U.S.A.
TOPPAN COMPANY, LTD., JAPAN

PRINCIPLES OF REFRIGERATION DOSSAT

1807
WILEY
TOPPAN

JW-073(P-11)