CHAPTER 7

CONDENSERS

7.1 TYPES OF CONDENSERS USED IN INDUSTRIAL REFRIGERATION

The three main types of condensers used in general refrigeration systems are:

- air-cooled
- water-cooled
- evaporative

All of these serve the industrial refrigeration field as well. In comparison to the air-conditioning industry, however, a lower percentage of air-cooled condensers and a higher percentage of evaporative condensers are operating in industrial refrigeration plants. In industrial refrigeration practice, it is common to connect the evaporative condensers in parallel—a concept not normally used in air conditioning.

The three types of condensers are shown schematically in Fig. 7.1a, 7.1b, and 7.1c. The air-cooled condenser in Fig. 7.1a condenses refrigerant vapor by rejecting heat to ambient air blown over the finned condenser coil with the aid of a fan, usually a propeller type.

Most all water-cooled condensers (Fig. 7.1b) condense refrigerant in the shell and on the outside of tubes through which water passes. The condenser cooling water picks up heat in passing through the condenser and this warm water is cooled by circulating through a cooling tower (Section 7.6). While the shell-and-tube

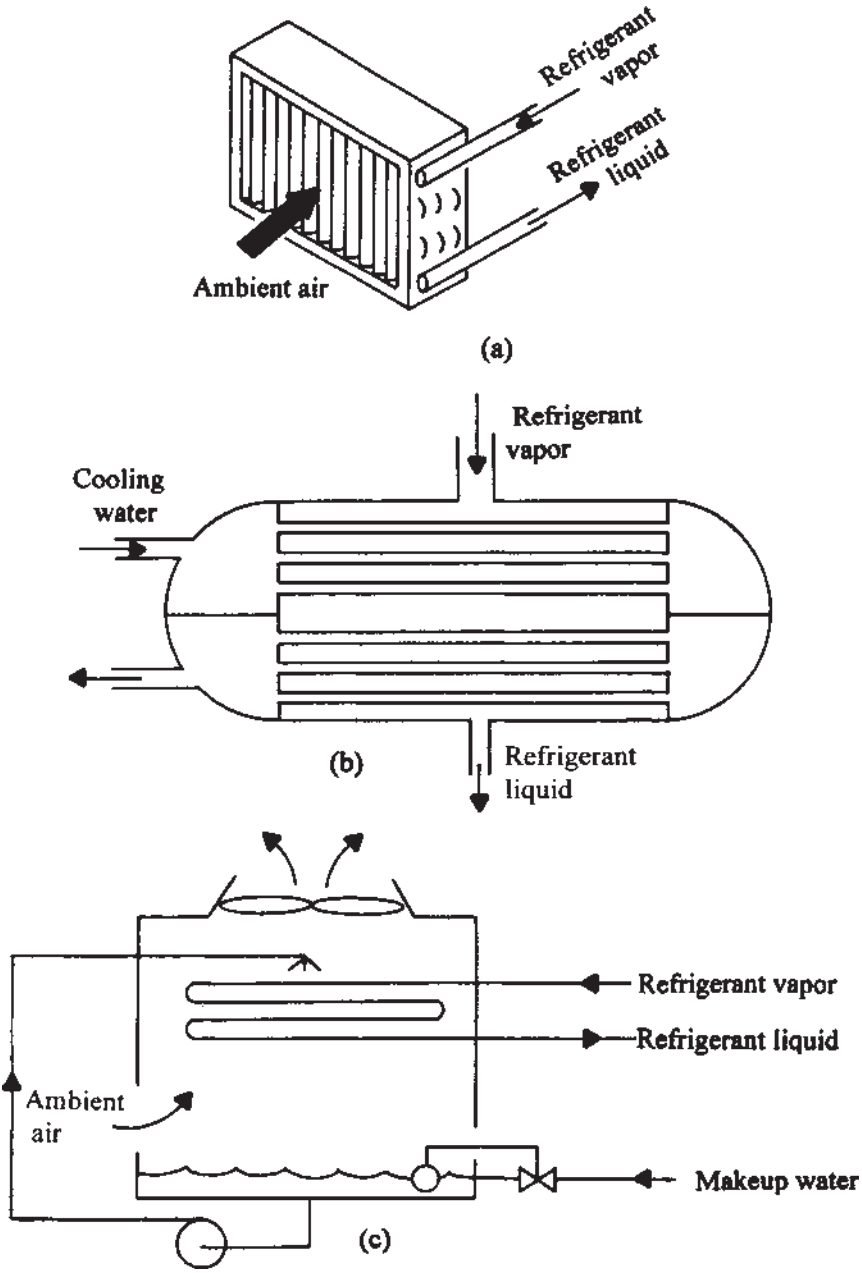

FIGURE 7.1
Types of condensers: (a) air-cooled, (b) water-cooled and (c) evaporative.

construction predominates for water-cooled condensers, plate-type condensers, sister of the plate-type evaporator explained in Sec. 6.31, are now appearing. The evaporative condenser of Fig. 7.1c might be considered a cooling tower, with the condenser tubes washed by the water spray. Ultimately, the heat rejected from the refrigeration plant is discharged to ambient air, except where the condenser is cooled by water from a well, lake, or stream.

This chapter first explores the condensing process outside and inside tubes. Next, the overall performance of water-cooled condensers and the translation of performance to noncatalog ratings is examined. An explanation of the performance of cooling towers, the constant companions of water-cooled condensers, is given. Because of their prevalence in industrial refrigeration plants, the emphasis of this chapter is on the performance, selection, application, and operation of evaporative condensers.

7.2 THE CONDENSING PROCESS

Nearly a century ago, heat-transfer pioneer, Willhelm Nusselt, proposed a model to predict the magnitude of a condensing coefficient for a special geometric situation[1]. Nusselt envisioned the condensation of vapor on a cold vertical plate, Fig. 7.2, as a process where vapor condenses on the plate and the condensate drains downward, with the condensate film becoming progressively thicker as it descends. The local condensing coefficient is taken to be the conductance through the condensate film—the conductivity of the liquid divided by the film thickness at that point. Nusselt developed the expression for the mean condensing coefficient as

$$h_c = 0.943 \left(\frac{g \rho^2 h_{fg} k^3}{\mu \Delta t L} \right)^{1/4} \tag{7.1}$$

where h_c = mean condensing coefficient, W/m²-°C (Btu/hr·ft2·°F)
g = acceleration due to gravity=9.81 m/s² (4.17×10^8 ft/hr²)
ρ = density of condensate, kg/m³ (lb/ft³)
h_{fg} = latent heat of vaporization of the refrigerant, kJ/kg (Btu/lb)
k = conductivity of condensate, W/m·°C (Btu/hr·ft·°F)
μ = viscosity of condensate, Pa s (lb/ft·hr)
Δt = temperature difference, vapor to the plate, °C (°F)
L = vertical length of plate, m (ft)

The immediate question is where, if at all, does condensation occur on a vertical plate in industrial practice? Actually, a very old condenser design oriented the tubes vertically and water flowed by gravity down the inside of the tubes to ease their cleaning. The refrigerant in the shell condensed on the outside of the vertical tubes.

A slight modification of Eq. 7.1 applies to the widely used horizontal shell-and-tube condenser, Fig. 7.1b. The product of the number of tubes in a vertical

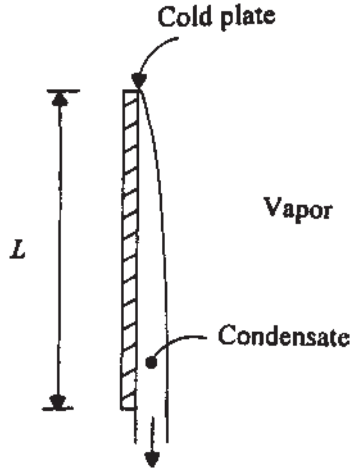

FIGURE 7.2
Condensation of a vapor on a cold vertical surface.

row multiplied by the diameter of the tubes replaces the vertical length of the plane L. White[2] found by experimental tests that the coefficient is 0.63 and Goto[3] measured 0.65, so the equation for N tubes of diameter D in a vertical row is:

$$h_c = 0.64\left(\frac{g\rho^2 h_{fg} k^3}{\mu \Delta t N D}\right)^{1/4} \tag{7.2}$$

Before leaving the condensing equations, an interesting comparison of the condensing coefficients of various refrigerants can be made. As Table 7.1 shows, the condensing coefficients of ammonia condensing on the outside of tubes far surpasses the coefficients of the other refrigerants shown. Experimental tests also show ammonia to have a higher condensing coefficient—five times that of the halocarbons in one study[4].

7.3 CONDENSATION INSIDE TUBES

In air-cooled and evaporative condensers, the refrigerant condenses inside tubes. The mechanism of condensation is complex and the flow regimes continue to change as the refrigerant passes through the tube[5]. Even though the state of the refrigerant is superheated vapor on entering the tube, condensation begins immediately and a spray regime develops. Later on the flow converts to annular then stratified with the liquid flowing along the bottom of the tube. Near the end of the condenser tube the flow regime is characterized as slug or plug.

TABLE 7.1
Condensing coefficients on the outside of tubes for several refrigerants. The condensing temperature is 30°C (86°F) and there are six 25-mm (1-in) tubes in a vertical row.

Refrigerant	Condensing coefficient	
	W/m²·°C	Btu/hr·ft²·°F
R-22	1142	201
R-134a	1046	184
Ammonia	5096	897

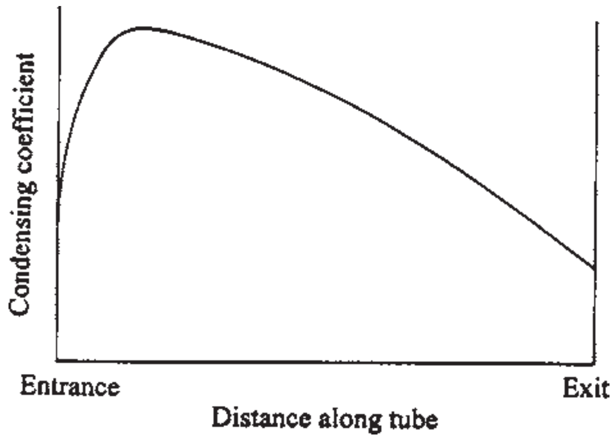

FIGURE 7.3
Variation in the condensing heat-transfer coefficient inside a tube.

Figure 7.3 shows relative values[6] of the condensing coefficient throughout a tube. At the entrance to the tube with its superheated vapor content the coefficient is low, which is typical of convection heat transfer with a gas. The coefficient increases once surface condensation begins and is usually at its highest value during annular flow. As more and more condensed liquid flows with the vapor, the surface available for condensation decreases. Near the end of the condenser tube the coefficient drops quite low, because the process has approached that of convection heat transfer to a liquid.

The low heat-transfer coefficient near the end of the condenser tube when all or most of the vapor has condensed is pertinent to the plant operator. The reason is that backing liquid into an air-cooled or evaporative condenser shifts some heat-transfer area into the liquid subcooling mode which exhibits low heat-transfer coefficients.

7.4 HEAT-REJECTION RATIO

The *heat-rejection ratio* (HRR) is defined as the ratio of the rate of heat rejected at the condenser to that absorbed at the evaporator.

$$\mathrm{HRR} = \frac{\text{rate of heat rejection at condenser}}{\text{rate of heat absorbed at evaporator}}$$

The designer and operator of the refrigeration system will usually characterize plant size by the refrigeration capacity. This capacity can be translated to a condenser capacity through the condenser-to-evaporator heat rejection ratio (HRR). The HRR is a function of the evaporating and condensing temperatures, but is also influenced by the compressor type and any supplementary cooling arrangements. The standard procedure for computing the HRR from catalog data of the compressor is to propose that the heat rejected at the condenser is composed of two contributions—the refrigerating capacity and the thermal equivalent of the power supplied to the compnessor. The standard equation for computing the HRR is, therefore,

$$\mathrm{HRR} = \frac{\text{refrigerating capacity} + \text{compressor power}}{\text{refrigerating capacity}} \tag{7.3}$$

where all the energy flow rates are expressed in the same units.

Figure 7.4 shows HRRs as functions of the evaporating and condensing temperatures. Changes of either of these temperatures affect both the refrigerating capacity and the power requirement of the compressor. The ideal HRR can be derived from knowledge of the Carnot cycle (Section 2.17), in which the ratio of area under the condensing line to that under the refrigeration line represents the HRR,

$$\mathrm{HRR} = \frac{T_{cond}}{T_{refrig}} \tag{7.4}$$

where the temperatures T are in absolute, thus °C+273.1 (°F+459.7). Equation 7.4 assumes a 100% efficiency of the cycle and the compressor, and an improved expression that can be used when compressor catalog data are not readily available is

$$\mathrm{HRR} = \left(\frac{T_{cond}}{T_{refrig}}\right)^{1.7} \tag{7.5}$$

Example 7.1. Estimate the HRR when the condensing temperature is 35°C (95°F) and the evaporating temperature is -10°C (14°F).

Solution. On the absolute scale the evaporating temperature is 263.1 K (473.6 R) and the condensing temperature is 308.1 K (554.6 R). The estimated HRR is

$$\mathrm{HRR} = \left(\frac{308.1}{263.1}\right)^{1.7} = 1.31$$

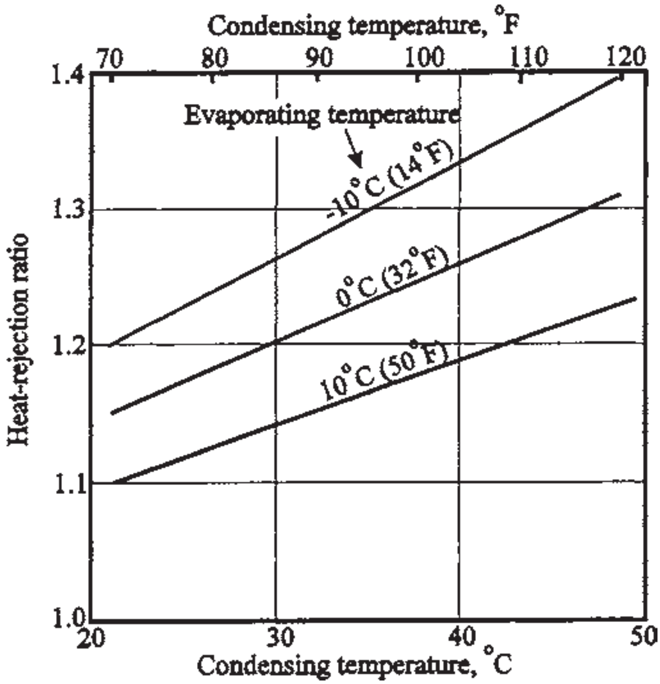

FIGURE 7.4
Typical values of the ratio of the heat rejected at the condenser to the refrigerating capacity, HRR, for ammonia and halocarbon refrigerants.

The value from Fig. 7.4 is 1.30.

Equation 7.3 is correct except for heat losses to the ambient or supplementary transfers of heat to other devices. The curves in Fig. 7.4 apply to open-type compressors, and the HRR will be higher for hermetic compressors servicing small halocarbon systems, because some of the motor heat enters the refrigerant stream. Also the HRR will be lowered if a reciprocating compressor uses water-cooled heads where the heat is rejected to a separate cooler or in a screw compressor where oil is cooled by a separate water or antifreeze circuit.

7.5 PERFORMANCE OF AIR AND WATER-COOLED CONDENSERS

Manufacturers of condensers provide performance data directed toward selecting equipment. By applying some fundamentals of heat transfer, a user can frequently translate catalog data to nondesign conditions. The strategy in extending catalog data to nondesign conditions is usually to compute the *UA* value (the product of the overall heat transfer coefficient and the heat-transfer area) and for situations where the *UA* remains essentially constant, apply this *UA* value to the new set of operating conditions. The temperature profiles are somewhat complex because of desuperheating and subcooling, as shown in Fig. 7.5a, but to approximate, assume the condensing temperature prevails throughout the condenser, as shown in Fig. 7.5b.

In the desuperheating section, the actual temperature difference between the refrigerant and cooling water is higher than the ideal, but this error is at least partially compensated for by the fact that the actual heat-transfer coefficient for the convection process is less than during condensation. Real condensers are rarely circuited strictly for counterflow or parallel flow. When one fluid is at a constant temperature, however, the flow pattern is immaterial, and an equation comparable to the one for evaporators, Eq. 6.11, applies:

$$q = UA\left[\frac{t_o - t_i}{\ln\left(\frac{t_c - t_i}{t_c - t_o}\right)}\right] \tag{7.6}$$

where q = rate of heat transfer, kW (Btu/hr)
UA = product of overall heat-transfer coefficient and area to which it applies, kW/°C (Btu/hr per °F)
t_c = temperature of condensing refrigerant, °C (°F)
t_i = temperature of entering cooling water, °C (°F)
t_o = temperature of leaving cooling water, °C (°F)

Example 7.2. The catalog for a Vilter 0.2 m×2.13 m (8 in×7 ft) R-22 condenser specifies a condensing capacity that accommodates a refrigeration load of 204 kW (58.1 tons) at the evaporator when the evaporating temperature is 4.4°C (40°F), the condensing temperature is 40.6°C (105°F), and a 9.8 L/s (156 gpm) flow rate of cooling water enters at 29.4°C (85°F).

What condensing temperature would prevail if the cooling water flow rate and its entering temperature remain constant, but the refrigeration capacity is half of the catalog value?

Solution. The rate of heat transfer q at the condenser with the original refrigeration load was:

$$q = (204 \text{ kW})(\text{heat rejection ratio}),$$

and at an evaporating temperature of 4.4°C (40°F) and a condensing temperature of 40.6°C (105°F), Fig. 7.4 shows a heat rejection ratio of 1.24, so q equals 253 kW (863,000 Btu/hr). The mass flow rate of cooling water is 9.8 kg/s (1300 lb/min), so the outlet water temperature t_o is:

$$t_o = 29.4 + \frac{253 \text{ kW}}{(9.8 \text{ kg/s})(4.19 \text{ kW/kg}\cdot{}^\circ\text{C})} = 29.4 + 6.2 = 35.6^\circ\text{C } (96.1^\circ\text{F})$$

The log-mean temperature difference is:

$$\text{LMTD} = \left[\frac{35.6 - 29.4}{\ln\left(\frac{40.6-29.4}{40.6-35.6}\right)}\right] = 7.69^\circ\text{C } (13.8^\circ\text{F})$$

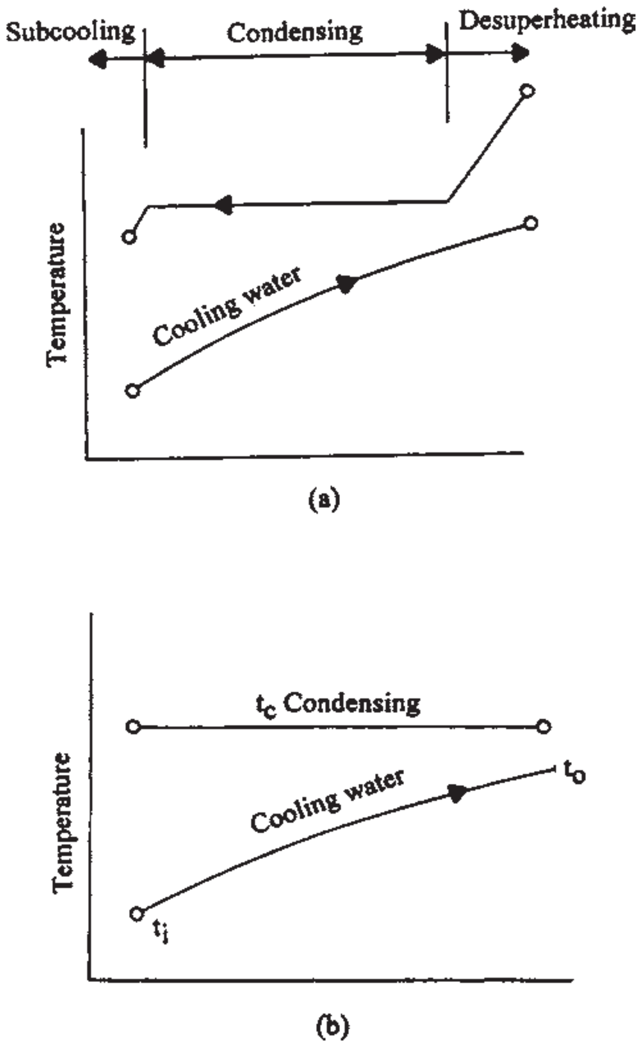

FIGURE 7.5
(a) Actual, and (b) idealized temperature profiles in a water-cooled condenser.

and the *UA* value is:

$$UA = \frac{253 \text{ kW}}{7.69°\text{C}} = 32.9 \text{ kW/°C } (62{,}400 \text{ Btu/hr per °F})$$

This *UA* value should remain essentially unchanged as the condensing capacity varies, so long as the water flow rate remains constant. At the half-load condition, the LMTD will be half the original value, 7.69÷2 or 3.85°C (6.92°F), and the condenser cooling water experiences half its original rise in temperature, so its new value is 29.4+3.1=32.5°C (90.5°F). The equation for the LMTD can be solved for the new condensing temperature,

$$\text{LMTD} = 3.85 = \left[\frac{32.5 - 29.4}{\ln\left(\frac{t_c - 29.4}{t_c - 32.5}\right)}\right]$$

$$\frac{t_c - 29.4}{t_c - 32.5} = e^{3.1/3.85} = 2.237$$

so the new t_c=35.0°C (95°F), which is a reduction from the original t_c of 40.6° (105°F).

If greater precision is desired, perform an iteration which uses a revised HRR based on the new condensing temperature.

Example 7.1 illustrates a situation where the U-value remains constant from one condition to another. If the rate of water flow changes, the heat-transfer coefficient on the water side will also change so that the U-value no longer remains constant. It is advisable to consult the manufacturer in such a case.

The tubes of water-cooled condensers are subject to fouling caused by impurities in the water. Some measurements[7] made with suspended solids in cooling tower water indicated that the fouling factor (which is additional heat transfer resistance) can easily be on the order of 0.00004 $m^2\cdot°C/W$ (0.0002 $hr\cdot ft^2\cdot°F/Btu$) or higher. In the condenser of Example 7.1 the water-side heat transfer area is 3.78 m^2 (40.7 ft^2), so the U-value of the condenser based on the water-side area is:

$$\frac{32{,}900 \text{ W/°C}}{3.78 \text{ m}^2} = 8701 \text{ W/m}^2 \cdot °\text{C}$$

or 1533 Btu/($hr\cdot ft^2\cdot°F$). The resistance is the reciprocal of this U-value or 0.000114 $m^2\cdot°C/W$ (0.000645 $hr\cdot ft^2\cdot°F/Btu$). The resistance when the fouling factor is included is 0.000154 m2·°C/W corresponding to a U-value of 6500 $W/m^2\cdot°C$ (1143.6 $Btu/hr\cdot ft^2\cdot°F$). Tube foulingin this case reduces the condensing capacity 25% compared with the clean condition. The user has some protection because condensers leave the factory with a higher U-value than indicated by catalog data. They are derated using a fouling factor specified in the catalog. The user, however, should be aware that if the cooling tower water is fouling the condenser, frequent tube cleaning can improve system performance.

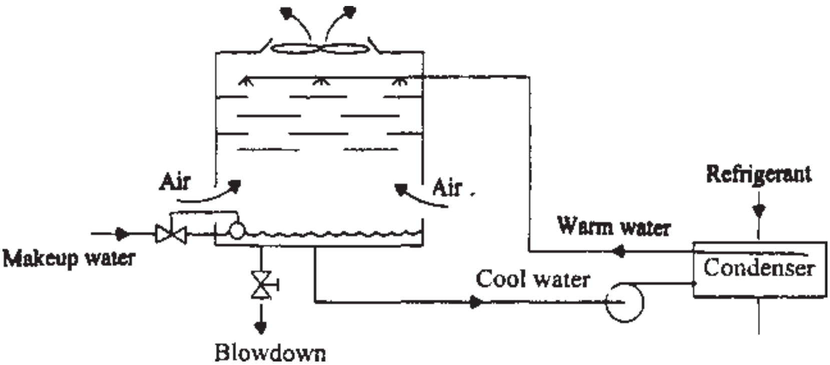

FIGURE 7.6
A cooling tower.

7.6 COOLING TOWERS

A cooling tower cools water by spraying it through a stream of ambient air. A schematic diagram of a cooling tower and the manner in which it serves the refrigeration condenser are shown in Fig. 7.6. The air- and water-flow patterns suggested in Fig. 7.6 are counterflow of air and water, a frequently used configuration. Another popular geometry is crossflow, in which the air is blown horizontally through the falling stream of water. Because some water evaporates into the air, a supply of makeup water must be provided. Also, because the makeup water contains some dissolved minerals, the concentration of these minerals in the sump water would progressively increase were a blowdown not provided.

The explanation of the heat- and mass-transfer process in a cooling tower starts with the recollection of the straight-line law first introduced in Sec. 6.13. The straight-line law states that when air is in contact with water, the change in air conditions is a straight line on the psychrometric chart directed toward the saturation line at the water temperature. This information is used to examine what happens to the enthalpy (heat content) of the air. If the enthalpy of air increases in the process, the enthalpy and temperature of the water must decrease. Consider first the special case shown in Fig. 7.7, where the wet-bulb temperature of the air equals the water temperature.

The path of the air moves toward the saturation line at the water temperature, which is along the wet-bulb temperature line. The wet-bulb temperature lines and the enthalpy lines are essentially parallel, so there is no change in the enthalpy of air, and the temperature of water does not change either. This is the process that takes place in evaporative coolers that reduce the air temperature in homes in arid regions.

If the temperature of the water is higher than the wet-bulb temperature of the air, as in Fig. 7.8, the enthalpy of the air increases from point 1 to point 2, so an energy balance requires that this heat must come from the water by cooling it from point 1′ to point 2′.

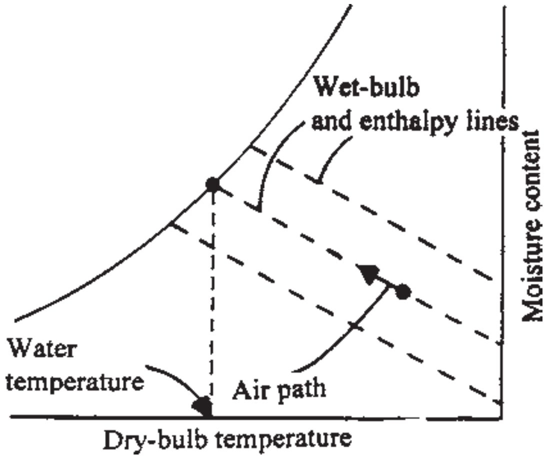

FIGURE 7.7
When the water temperature is the same as the wet-bulb temperature of the air, there is no change in water temperature.

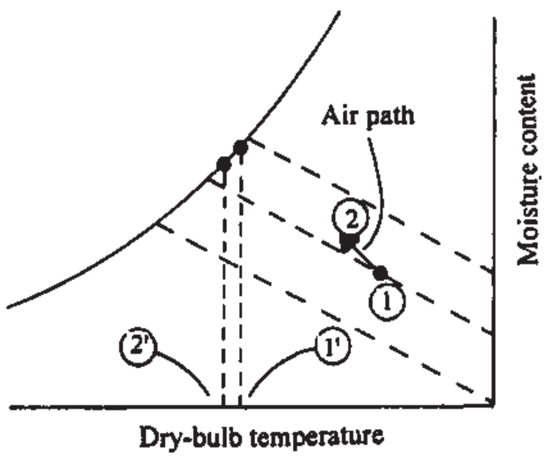

FIGURE 7.8
The enthalpy of air rises and the temperature of water drops when the water temperature is higher than the wet-bulb temperature of the air.

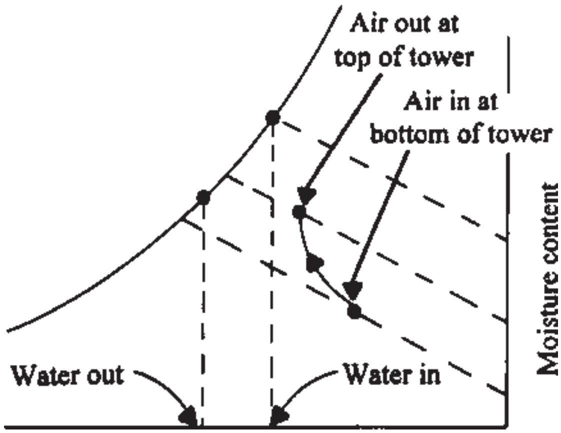

FIGURE 7.9
Conditions of air and water in a counterflow cooling tower.

When these elementary processes are expanded to a complete counterflow cooling tower, they show the pattern of air and water conditions as in Fig. 7.9. The air progressively increases in enthalpy, and while its dry-bulb temperature is shown decreasing in Fig. 7.9 as it rises through the tower, there could be situations where the temperature increases in passing through the tower.

The key concept implicit in Figs. 7.7 through 7.9 is that the leaving water temperature can approach the wet-bulb temperature of entering air. For this reason, catalog data for cooling towers show the ambient condition that affects cooling tower performance as the wet-bulb temperature, and dry-bulb temperatures may not even be indicated. When a constant heat load is imposed on the condenser and its cooling water, the leaving water temperature rides up as the ambient wet-bulb temperature increases in a trend as shown in Fig. 7.10. Because the heat load and the water-flow rate are constant, a fixed drop in water temperature (5°C or 9°F in this case) prevails over the entire range shown in the graph.

7.7 EVAPORATIVE CONDENSERS

The schematic diagram of the evaporative condenser shown in Fig. 7.1c illustrates that the evaporative condenser combines the functions of an air-cooled condenser and cooling tower. Refrigerant condenses within the tubes, and these tubes are sprayed with water through which an air stream passes. The evaporation of some water into the air is the dominant process of rejecting heat to the atmosphere.

To provide a comparison of the three forms of condensing equipment, namely the air-cooled condenser, the water-cooled condenser/cooling tower combination, and the evaporative condenser—some of the characteristics of each are enumerated:

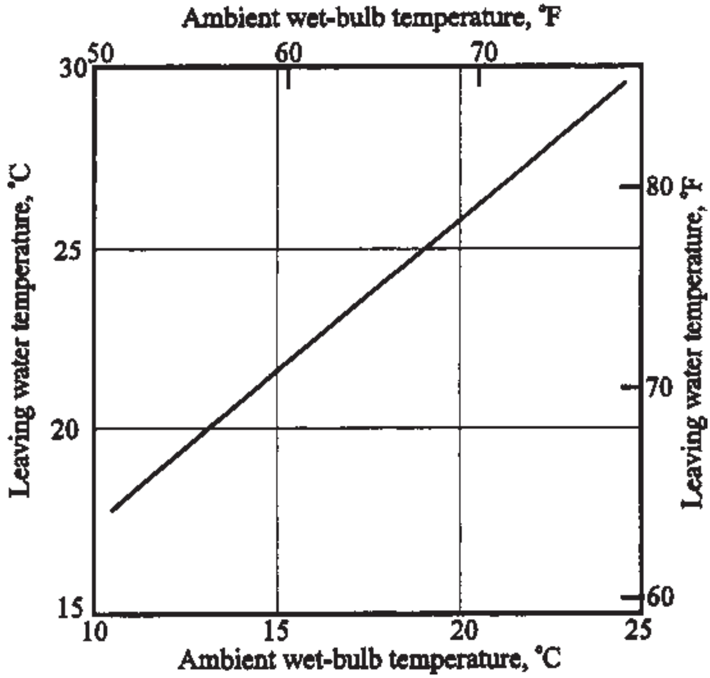

FIGURE 7.10
Leaving water temperature from a cooling tower as the ambient wet-bulb temperature changes. The heat load and water-flow rate are constant.

Air-cooled condenser. Usually lowest first cost of the three, and least maintenace cost as well, because no water circulates or evaporates.

Water-cooled condenser with cooling tower. Lower condensing temperature than with an air-cooled condenser, because the wet-bulb rather than the dry-bulb temperature of the air is the sink toward which the condensing temperature drives. When the distance between the compressor and the point of heat rejection is long, water can be piped to the cooling tower, rather than sending refrigerant, as must be done with the evaporative or air-cooled condenser.

Evaporative condenser. Compact and provides lower condensing temperatures than the air-cooled condenser and also lower than the water-cooled condenser/cooling tower combination. Figure 7.11 shows an evaporative condenser with a bit more detail than was presented in Fig. 7.1c.

The evaporative condenser is widely used in industrial refrigeration practice because it provides relatively low condensing temperatures. These temperatures conserve power and result in moderate compressor discharge temperatures

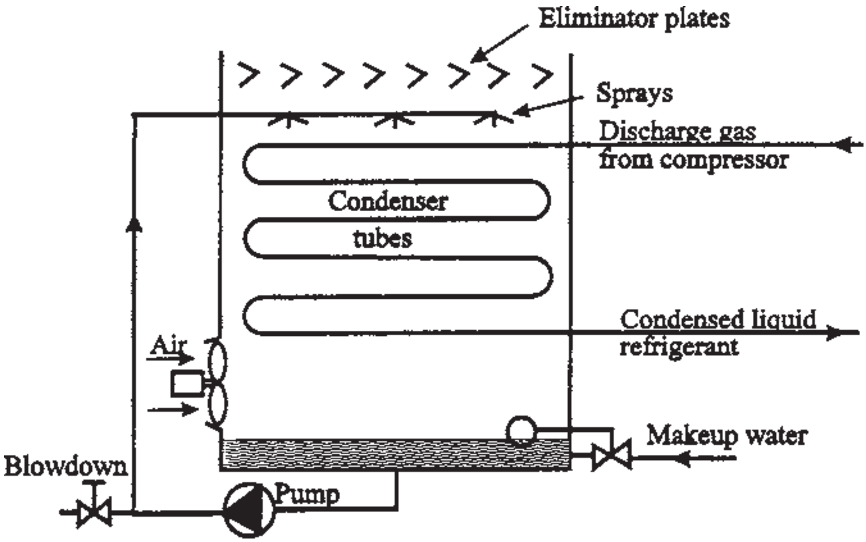

FIGURE 7.11
An evaporative condenser.

which may be important, especially for ammonia applications. The drawback of its maintenance requirements may not be prohibitive. Most industrial refrigeration plants employ service personnel, in contrast to smaller plants where the refrigeration facility is expected to operate without regular maintenance.

Most evaporative condensers are of the blow-through type using an axial-flow fan in preference to a centrifugal type. Eliminator plates are provided to avoid blowing water droplets out of the condenser. All condensers are equipped with blowdown to limit the buildup of minerals in the spray water, and this blow-down is usually installed in the pump discharge line, so that when the condenser is not in operation, the blowdown ceases.

7.8 NOMINAL SIZES AND RATES FOR EVAPORATIVE CONDENSERS

The design of an efficient evaporative condenser requires optimizing a number of factors, including tube size, tube length, tube spacing, refrigerant circuiting, air-flow rate, casing size, and spray-water flow rate. The condenser manufacturer/designer must draw on knowledge of refrigerant heat transfer, wetted-surface heat transfer (such as described in References 8 and 9), and a thorough understanding of fabrication economics and of the end-user operation. The condenser user is spared the responsibility of making most of these decisions and is best served with an understanding of how three variables affect the performance of an evaporative condenser: the wet-bulb temperature, air-flow rate, and spray-water flow rate. The next several sections will focus on those factors, but first a few nominal magnitudes applicable to many commercial evaporative condensers will be presented:

Heat-transfer area:

0.25 m^2 per kW of heat rejection (0.8 ft^2 per 1000 Btu/hr)

Spray water circulating rate:

0.018 L/s per kW of heat rejection (5 gph per 1000 Btu/hr)

Air volume flow rate:

0.03 m^3/s per kW of heat rejection (18 cfm per 1000 Btu/hr)

Air pressure drop through the condenser:

250–375 Pa (1 to 1–1/2 inches of water)

Rate of water evaporated:[10]

1.5 L/hr per kW of heat rejection (0.12 gph per 1000 Btu/hr)

Total rate of water consumption:[10]

with good quality makeup water the bleed rate may be as low as 50% of the evaporation rate, so the total rate evaporated and blown down may be about 2.2 L/hr per kW of heat rejection (0.18 gph per 1000 Btu/hr).

Many years ago the typical flow rate of spray water was quite low, perhaps 0.68 L/s per m^2 (1 gpm/ft^2), but this rate has gradually climbed so that it may run as high as 4.1 L/s per m^2 (6 gpm/ft^2) to achieve favorable capacity. The practical limit is reached when the spray water flow rate is so high that it restricts the air flow rate.

7.9 COMPARISON OF EVAPORATIVE CONDENSER WITH THE WATER-COOLED CONDENSER AND COOLING TOWER COMBINATION

Practically all of the remainder of this chapter will concentrate on evaporative condensers which are the predominant type used in industrial refrigeration. One of the reasons for the preference of evaporative condensers over the water-cooled condenser with cooling tower is the ability to achieve lower condensing temperatures. Figure 7.12 shows a heat rejection of 548 kW (1,870,000 Btu/hr) by means of an evaporative condenser (Fig. 7.12a) and the rejection of the same magnitude with a water-cooled condenser. The capacity of both the evaporative condenser and the cooling tower are controlled by the ambient wet-bulb temperature, which in this case is 25.6°C (78°F).

The comparison shows the ability to achieve a condensing temperature of 35°C (95°F) with the evaporative condenser, while with the water-cooled condenser the condensing temperature is 40.6°C (105°F), thus 5.6°C (10°F) higher. The superior performance of the evaporative condenser is explained by the avoidance of the intermediate fluid (the cooling-tower water) in the heat-transfer processes. The temperature of water leaving the cooling tower is 28.9°C (84°F) and can only approach the ambient wet-bulb temperature, and the condensing temperature can only approach the temperature of water returning to the tower which is 35.8°C (96.5°F).

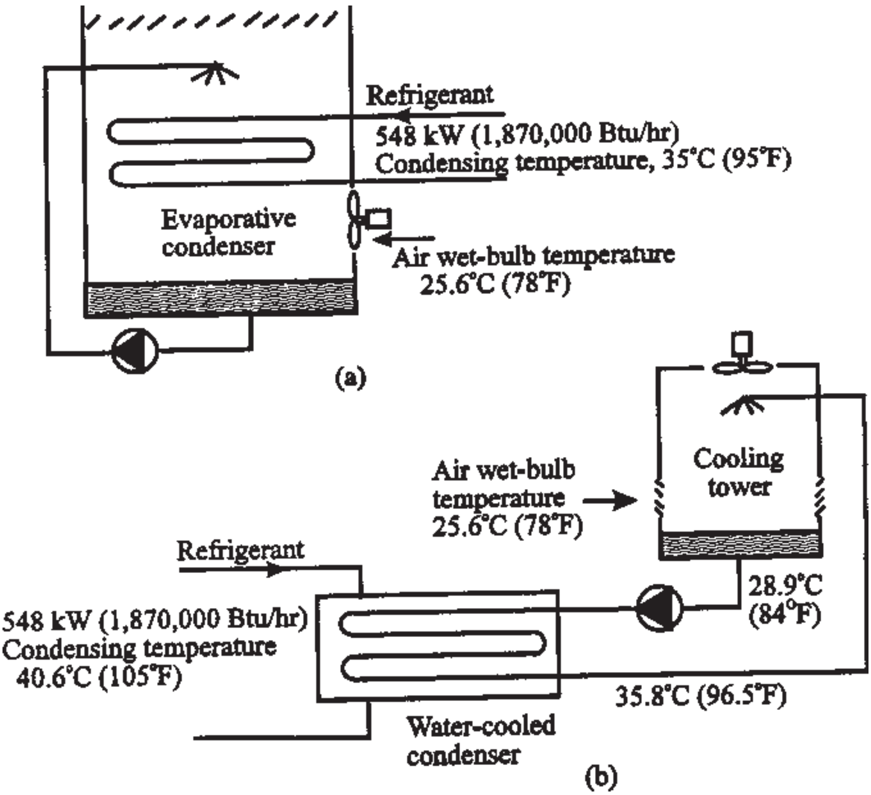

FIGURE 7.12
Achieving a lower condensing temperature with an evaporative condenser in comparison to the combination of a water-cooled condenser and cooling tower.

The comparison could be accused of being biased in that it would be possible to lower the condensing temperature in Fig. 7.12b by enlarging either or both the cooling tower or the water-cooled condenser. This observation is correct, but the sizes of all the components in the comparison of Fig. 7.12 are based on sizes typically chosen for this application. Nevertheless, the comparison is not complete until the comparative first costs of the condensing subsystem are evaluated. Some designers contend that the evaporative condenser in Fig. 7.12 might even cost less than the sum of the water-cooled condenser and cooling tower.

Another factor in favor of the evaporative condenser is the lower water-pumping costs. The spray-water flow rate in the evaporative condenser is typically about one-third that of the flow rate circulated between the water-cooled condenser and the cooling tower. Furthermore, the length of water line between the water-cooled condenser and cooling tower will likely be much longer. This saving in water-pumping power must be balanced against the higher pressure drop in the refrigerant lines, particularly the vapor line between the compressor and condenser. Normally the compressor and water-cooled condenser will be close-coupled. The need for water treatment exists in both concepts, so this feature is not a factor.

These advantages of the evaporative condenser influence the industrial refrigeration industry to predominately favor this type of condensing system. The air-conditioning industry, on the other hand, usually chooses water-cooled condensers, so it is reasonable to ask whether the air-conditioning industry is unaware of the secret. Such is not the case, because many air-conditioning systems experience long distances between the compressor and the ultimate heat rejector. The compressor and condenser may be in the basement and the cooling tower on the roof of a multistory building. In many industrial refrigeration plants the evaporative condenser is on the roof of the machine room that houses the compressors, and the distance separating them may be only 6 to 12 m (20 to 40 ft). Also, when a centrifugal compressor serves a water-chilling system, the refrigerant chosen has a high specific volume, making condensing in the tubes less practical.

7.10 INFLUENCE OF WET-BULB TEMPERATURE ON EVAPORATIVE CONDENSER CAPACITY

It was the conclusion in Section 7.6 that the leaving water temperature from a cooling tower is controlled by the wet-bulb temperature of ambient air. Because the same process of heat and mass transfer occurs in both devices, the wet-bulb temperature also has a dominant influence on the capacity of evaporative condensers. Figure 7.13 shows relative capacities of an ammonia evaporative condensers to changes in wet-bulb temperature and condensing temperatures. The capacities are relative to a condenser operating with a condensing temperature of 35°C (95° F) and a wet-bulb temperature of 25°C (77°F). The trends are as expected, namely the capacity increases with a given wet-bulb temperature as the condensing temperature increases. Furthermore, at a given condensing temperature the capacity increases with a reduction in wet-bulb temperature.

Even though Fig. 7.13 indicates that the temperature difference between the condensing refrigerant and the entering wet-bulb influences the capacity, it is not to be assumed that the heat-rejection capacity is proportional to this difference in temperature. For an air-cooled condenser the heat-transfer rate is proportional to the temperature difference between the condensing refrigerant and the dry-bulb temperature of the entering air. For a water-cooled condenser the capacity is also proportional to the temperature difference between the refrigerant and entering water. For an evaporative condenser, as Fig. 7.14 shows, the level of temperatures as well as the temperature difference controls the capacity. This trend indicates that if an evaporative condenser develops a certain

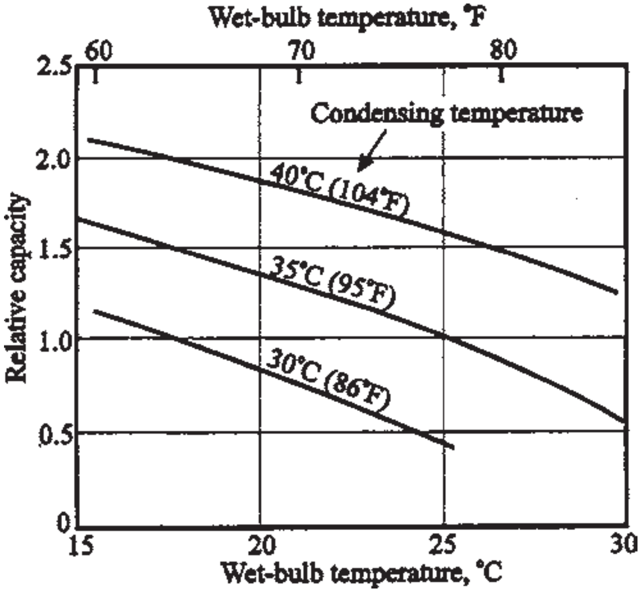

FIGURE 7.13
Relative heat-rejection capacity of an ammonia evaporative condenser as a function of the condensing and wet-bulb temperatures. The reference point is a 35°C (95°F) condensing temperature and a wet-bulb temperature of 25°C (77°F).

heat-rejection capacity with a temperature difference between condensing refrigerant and the ambient wet-bulb temperature, for example, 40°C to 25°C (104°F to 77°F), the capacity of the condenser will be less if the same temperature difference exists at a lower level, for example, 30°C to 15°C (86°F to 59°F).

The reason for this behavior lies in the evaporation process on which the evaporative condenser operates. The major heat-transfer mechanism in the evaporative condenser is due to the vaporization of water from the condenser tubes, and the rate of this vaporization is proportional to the difference of water-vapor pressure of the liquid water on the tube and the water vapor pressure in saturated air that surrounds the tube. Examination of a psychrometric chart (Fig. 6.18 or 6.20) shows that at the low level of temperature, the saturation curve flattens out so that a given difference in temperature translates to a lower difference in water-vapor pressure.

7.11 CATALOG SELECTION—TWO METHODS

Most catalogs of evaporative condensers show tables for selecting condensers by two different methods. One is the *condenser capacity* or *heat-rejection method* and the other is the *refrigeration capacity method*. The condenser capacity

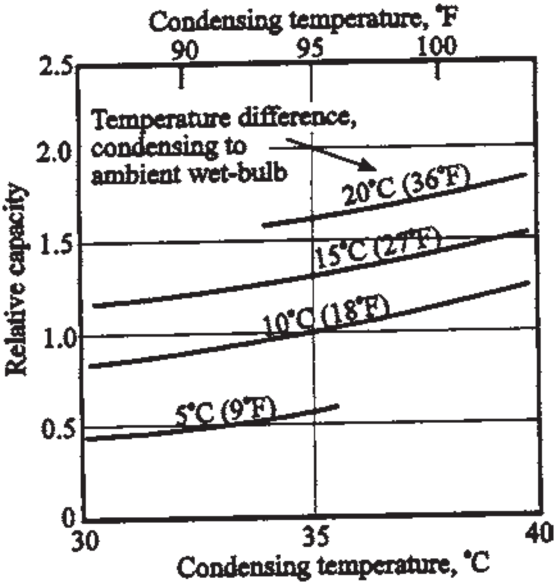

FIGURE 7.14
The level of temperature as well as the temperature difference between refrigerant and the air wet-bulb temperature influence the capacity.

method is most straightforward because it deals with the actual heat-transfer rate at the condenser. The refrigeration capacity method is simply a tool to facilitate quick selections, because the user of the catalog will normally think in terms of the refrigeration capacity which then becomes the entry point of the catalog.

The condenser capacity method will be explored first. Two tables are associated with this method—one table presents the capacity factors as functions of condensing and wet-bulb temperatures and the other table is nominal heat rejection capacity of various models of condensers. Most manufacturers show the capacity factors in tabular form. The presentation in Figure 7.15 is graphical. Table 7.2 shows a excerpt of a manufacturer's table[11] of nominal capacity of a series of condensers.

The capacity factors shown in catalogs and in Fig. 7.15 may at first seem to present trends opposite of expectations. For example, the capacity factor would be expected to increase as the wet-bulb temperature drops and the condensing temperature increases. But the main purpose of the catalog is to facilitate selection of condensers, not to analyze their performance. The consistency of the method can be demonstrated by an example.

> **Example 7.3.** Using Table 7.2 and Fig. 7.15, select a condenser to reject 586 kW (2,000,000 Btu/hr) while operating at a condensing temperature of 35°C (95°F) and a wet-bulb temperature of 25°C (77°F).

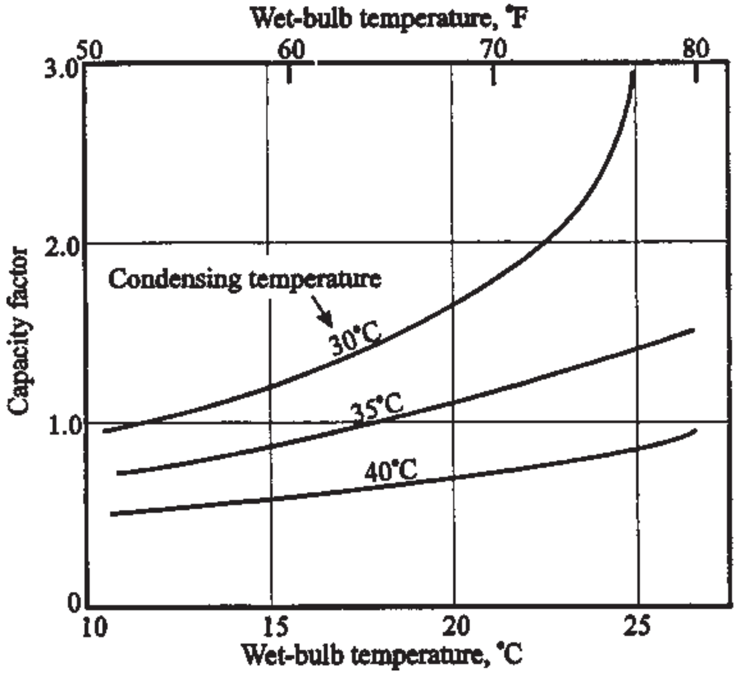

FIGURE 7.15
Capacity factors for selection of an evaporative condenser in conjunction with Table 7.2 using the condenser heat-rejection method.

TABLE 7.2
Nominal capacity of a line of evaporative condensers.

Model	Heat rejection rate		Model	Heat rejection rate	
	Thousands of Btu/hr	kW		Thousands of Btu/hr	kW
A	1544	452	F	2426	711
B	1764	517	G	2720	797
C	1911	561	H	3014	883
D	2058	603	I	3381	991
E	2205	646	J	3675	1077

Solution. Figure 7.15 shows a capacity factor of 1.4 at the prevailing condensing and wet-bulb temperatures and this factor is multiplied by the desired heat rejection rate:

$$586 \times 1.4 = 820 \text{ kW} \qquad \text{or} \qquad 2000 \times 1.4 = 2800 \text{ thousands of Btu/hr}$$

Referring to Table 7.2, Model G shows a capacity of 796 kW (2720 thousands of Btu/hr) which is low, but Model H with a capacity of 883 kW (3014 thousands of Btu/hr) would have adequate capacity.

The application of the capacity factors of Figure 7.15 in the example explains what might have appeared to be an incongruity. The capacity of a condenser is highest at high condensing temperatures and low wet-bulb temperatures, but Fig. 7.15 shows just the opposite trends. The purpose of the capacity factor is to aid in selecting the condenser and is multiplied by the specified heat-rejection rate. In so doing, when the capacity factor is low, a smaller condenser will be adequate, and the capacity factor is low when the condensing temperature is high and the wet-bulb temperature low.

If the refrigeration capacity method is chosen, a different capacity factor table and nominal rating table will be available from the manufacturer. Section 7.4 pointed out that the ratio of heat rejected at the condenser to the refrigeration capacity, the HRR, depends on the condensing temperature and the evaporating temperature. The capacity factor applicable to this method must therefore incorporate these two temperatures if the selection process is entered with the refrigerating capacity. A capacity factor table comparable to Fig. 7.15 offers the chance of incorporating the influence of the condensing temperature on the HRR, so a separate table correcting for the evaporating temperature will also be provided by the manufacturer.

Of these two selection techniques, the heat-rejection method is more powerful in accommodating system complexities, such as might occur in two-stage plants. The refrigeration capacity method is useful for quick estimates of the condenser size.

7.12 CAPACITY CONTROL

Capacity control of a condenser means reducing its capacity. This understanding of capacity control raises the question of why the condenser capacity should ever be reduced. When the condenser operates at full capacity, the condensing temperature will follow the wet-bulb temperature as it drops, and thus the compressor power will be reduced. In general the recommended strategy is to operate the condenser with full capacity, dropping the condensing temperature until limited by one or more of the following conditions:

- the condensing pressure is too low to adequately feed level-control valves and expansion valves
- the pressure of defrost gas is too low to achieve a satisfactory defrost,
- if the plant uses screw compressors with their oil cooled by direct injection of refrigerant, the pressure of the liquid must be high enough to force an adequate flow rate of liquid into the compressor
- savings in compressor power by further lowering of the condensing temperature are less than savings that would be possible in pump and fan motors of the condensers

If the pressure of liquid is too low, expansion valves and liquid-level controllers will not be able to pass enough refrigerant and capacity of evaporator coils will drop. In the case of a level controller to an intermediate-pressure sub-cooler/desuperheater, subcooling of the liquid will suffer and desuperheating of discharge gas from the low-stage compressor will degrade. It would be possible to use valves with larger ports, but this change could cause unstable feeding when the liquid pressure is high.

In order to defrost a coil the saturation temperature of the defrost gas must be well above 0°C (32°F). Tests[12] on an ammonia coil showed that successful defrosts could be achieved with a saturation temperature of the defrost gas of 15° C (59°F) resulting in saturation temperature inside the coil of approximately 10° C (50°F). Before a plant operator would set the minimum condensing temperature at 15°C (59°F) for unattended operation, defrosts of coils should be monitored for a period of time to be confident that no residual frost or ice remains on the coil following this defrost.

Manufacturers of compressors who apply direct injection for oil cooling recommend[13] minimum condensing temperatures of approximately 21°C (70°F) in R-22 and ammonia systems to provide adequate pressure for injection of the liquid.

The question of reducing air or water flow to save power required by the fan and pump motors will be addressed further in Section 7.15.

Still another situation which may influence the minimum condensing pressure occurs if the plant recovers heat from the high side of the system for uses elsewhere in the facility. A simple economic analysis may be able to establish whether it is cheaper to pay for the recovered heat from other sources and reap the saving on compressor power with a lower condensing pressure.

The two principal methods of reducing the condenser capacity are to reduce or cycle the flow of spray water or the air flow. Adjusting the flow rate of spray water will be be dismissed rather quickly in the next section, and the regulation of air flow evaluated in Section 7.14.

7.13 CAPACITY CONTROL— VARYING THE FLOW RATE OF SPRAY WATER

Reducing the flow rate of spray water by throttling the flow with a regulating valve or reducing the speed of the pump motor will lower the heat-transfer capacity of the condenser. Tests[8] suggest that the condenser capacity near its normal operating point varies as the flow rate to the 0.22 power:

$$\text{Condenser capacity} = (\text{Constant})(\text{spray-water flow rate})^{0.22} \qquad (7.7)$$

thus if the flow rate is reduced by 20% the capacity of the condenser would drop to 95% of its original value. At lower flow rates the drop in capacity is more precipitous until a complete interruption of the spray-water flow to dry operation drops the condenser capacity significantly.

Usually reducing the spray water flow rate is not recommended. If the rate is dropped much below the design value, areas of the tubes may become alternately dry and wet. The result is excessive scaling on that tube surface. Avoidance of scale is also one of the reasons for opposing cycling of the pump for capacity control. The second reason is that the frequent stopping and starting of the motor accelerates its wear.

7.14 CAPACITY CONTROL—VARYING THE AIR FLOW RATE

An equation[8] for the air-flow rate comparable to Equation 7.7 for the rate of spray-water flow is:

$$\text{Condenser capacity} = (\text{constant})(\text{air flow rate})^{0.48} \tag{7.8}$$

This equation matches closely some other data[14] that is shown in Fig. 7.16. If the air-flow rate is reduced by 50%, for example, the heat-rejection capacity of the condenser with a given combination of condensing and wet-bulb temperatures is 72% of the base capacity. One manufacturer[15], on the other hand, suggests that a reduction in air-flow rate will result in 58% of rated capacity. Some of the ways in which the air-flow rate can be regulated are:

- Variable-frequency drive of fan motor
- Two speed fan motors
- Pony motors
- Fan dampers
- Fan cycling on a single-fan unit
- Shutting down one fan in a multiple-fan condenser

Variable-frequency inverters driving the fan motor give the most precise regulation, but currently the first cost of the assembly makes this method the most expensive in overall first cost. Two-speed fan motors are available that operate with 1800/1200 rpm combination using a two-winding construction or 1800/900 rpm using a single winding. The 1800/1200 rpm combination requires an expensive motor but a low-cost starter, while the 1800/900 rpm combination offers a low-cost motor but an expensive starter. The pony motor arrangement mounts a different-speed motor on each end of the shaft and only one is powered while the other idles. Fan dampers are sometimes used, but in the hostile environment of the condenser the parts sometimes fail to move easily. Cycling the fan of a single-fan unit is a direct approach, but the condensing pressure oscillates and may cause control problems elsewhere in the system. One of the widely used methods of controlling the air flow is to cycle one or more fans in a multiple-fan unit. Such condensers must be equipped with baffles between the cells or much of the air delivered by one fan simply flows backward through an adjacent fan.

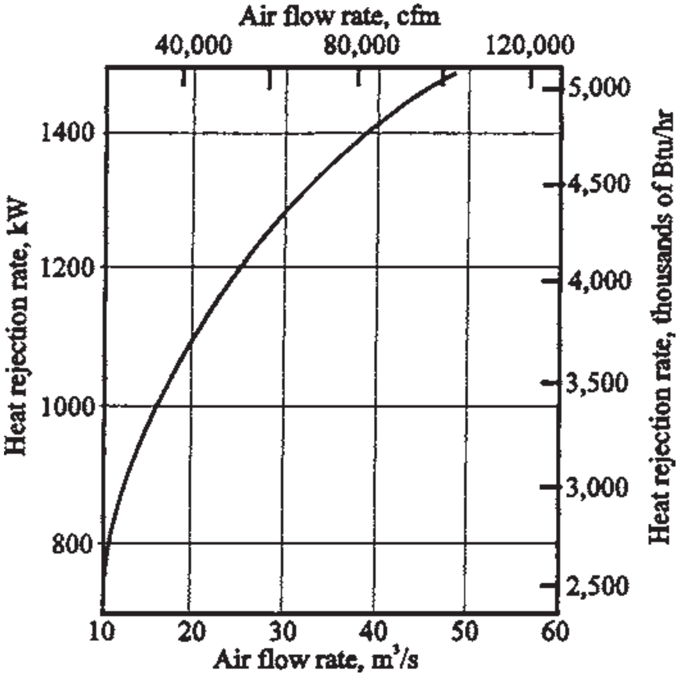

FIGURE 7.16
Effect of air-flow rate on the heat-rejection capacity of an evaporative condenser with given condensing and wet-bulb temperatures.

7.15 JUDICIOUS REDUCTIONS IN AIR-FLOW RATE

Most plants operate at the combination of maximum refrigerating capacity and design ambient temperature only a small percentage of the total time. In most cases, the refrigeration capacity is less and/or the ambient temperature is lower than design, offering the opportunity to reduce the condensing pressure. Up to a certain point, the plant operator should exploit the reduced condensing temperature to conserve compressor power. The catalog data for reciprocating and screw compressors in Chapters 4 and 5, respectively, indicate a power saving of the order of 3% per °C reduction in condensing temperature (1.7% per °F) in the range of 35°C (95°F) and 0°C (32°F) evaporating temperatures.

In operating regimes of perhaps 50% or more refrigerating capacity and wet-bulb temperatures above 15°C (59°F), the evaporative condenser would be operated with full fan and spray-water pump capacities. For lower refrigerating capacities and wet-bulb temperatures, the designer of the system may have specified controls that limit how low the condensing temperature (pressure) can drop. The setting of that minimum pressure is at the operator's discretion.

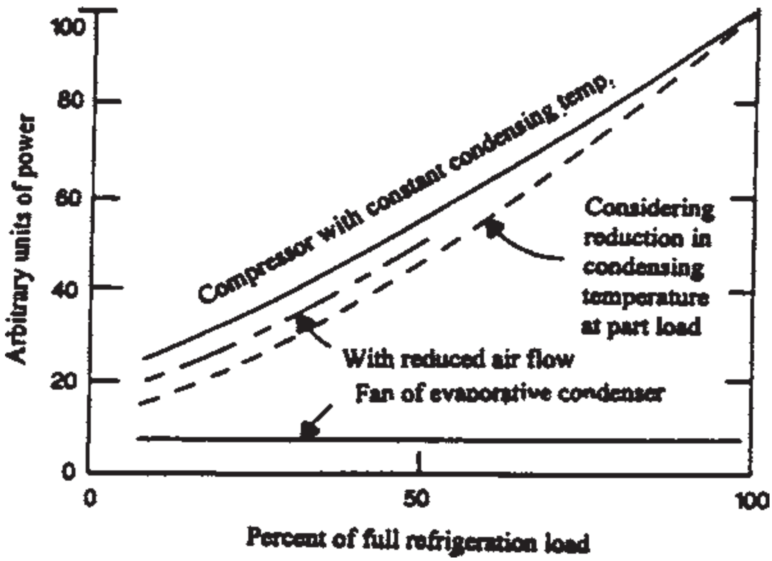

FIGURE 7.17
Relative power requirements of the compressor and the fan of an evaporative condenser. The evaporating temperature is in the range of 5°C (41 °F) and the wet-bulb temperature is constant.

The conclusion has been reached that regulating the rate of air flow is the preferable method of reducing condenser capacity. An additional method in a multiple-condenser installation is to shut down completely one condenser and bring it back on line when necessary. With the exception of variable-speed drives, all of the methods of capacity control are a form of cycling wherein the condensing pressure drops low enough to reduce the capacity. When the condensing pressure passes through the control dead band, it rises to a point where full fan operation resumes. No good purpose is served by rapidly cycling the control, because it causes excessive wear on the affected motors and possibly erratic feeding of the evaporators serviced by expansion valves and pressure fluctuations downstream of level-control valves.

The next consideration is how to operate the fans to minimize combined compressor and fan power. The power required by the fans is small, relative to that required by the compressor, at least at full load. Figure 7.17 shows relative power requirements for the compressor and condenser fans as a function of the refrigeration load.

At full refrigeration capacity, the fan motors draw about 5%–8% of the power drawn by the compressor motor. As the refrigeration load decreases, the compressor power drops, but the fan power remains constant if the air-flow rate is not reduced.

Three curves are shown for the compressor power—the top one applies if the condensing temperature remains constant through the entire load range. The condensing temperature progressively drops as the load decreases because, as Fig. 7.13 shows, the condensing temperature falls as the load on the condenser drops. The intermediate compressor power curve in Fig. 7.17 represents the compressor power at reduced air flow rate, and suggests a trade-off between savings in fan power and compressor power.

The optimum conditions to shift from full fan operation to partial operation may be different for each plant. However, to conserve energy, many plant operators are overly influenced by the visibility of all the condenser fans operating and are not as conscious of the extra compressor power required.

7.16 SUBCOOLING THE REFRIGERANT IN THE CONDENSER

A reasonable desire is for the refrigerant liquid that flows out to the system to be subcooled. Conveying saturated liquid in pipes is always fraught with the drawback of flashing some liquid into vapor, which can restrict the mass flow when the refrigerant reaches an expansion valve or a level-control valve. Liquid can be subcooled in the tubes of the evaporative condenser by operating with liquid backed up into the tubes. This condition is not desirable, however, because heat-transfer area that should be available for condensation is reduced. The result will be a higher-than-necessary condensing temperature. The latter part of this chapter is devoted to proper drain piping, which has the objective of not backing liquid up into the condenser.

Even if subcooled liquid is produced in the condenser, this subcooled liquid flows to the receiver, as in Fig. 7.18, and here the liquid warms to a saturated state. Because both vapor and liquid are present in the receiver, the liquid is saturated. The heat-transfer process that takes place in the receiver is that of condensing some vapor to warm the subcooled liquid. In order to develop subcooled liquid a separate coil must be provided in the condenser that draws saturated liquid from the receiver and subcools it in the coil, as shown in Fig. 7.18.

7.17 POSITIONING THE CONDENSER

Not always is there complete freedom in where to place condensers, because the condensers should be close to the compressors that they serve, the walls of the machine room or other buildings may tend to obstruct air flow, and the condensers must be placed where their weight can be structurally supported. Within these limitations there are two major objectives to be kept in mind when siting the condensers. One is that the rate of air flow should not be restricted, or the reductions in capacity as discussed in Sec. 7.12 on capacity control will occur. The second precaution is to place the condensers so that there is a minimum of recirculation of discharge air from the same or other condensers entering a condenser. Recirculation results in a wet-bulb temperature of air entering the condenser that is higher than the ambient wet-bulb temperature.

Figure 7.19 shows several arrangements of condensers with Figs. 7.19a and

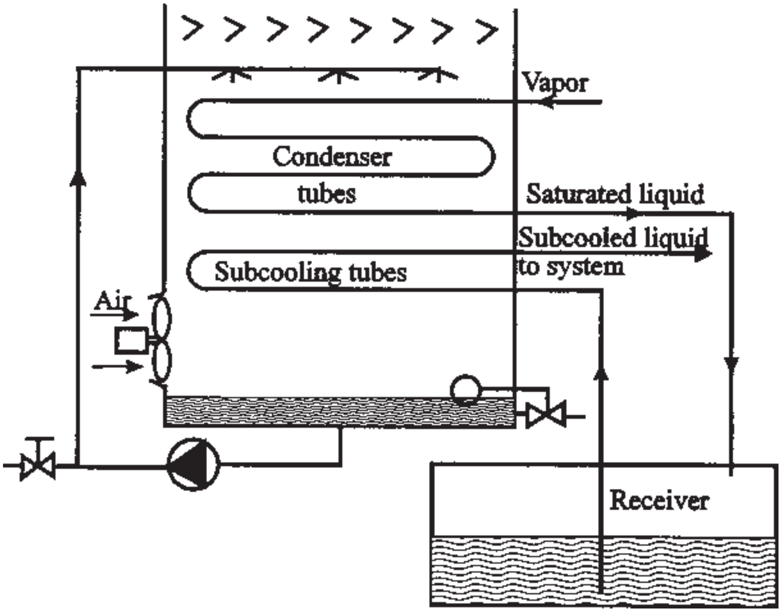

FIGURE 7.18
Liquid subcooler as auxiliary coil in the condenser case.

7.19b illustrating several problems while the placement in Fig. 7.19c offering a favorable placement. When the inlet of the condenser is close to a wall, as in Fig. 7.19a, the total air flow may be restricted. Furthermore some of the discharge air may be induced into the high-velocity air stream flowing down between the wall the condenser. In the arrangement of Fig.7.19b, the condensers appear too close to one another so that the condenser on the right draws some discharge air into its inlet. Manufacturers of evaporative condensers often recommended minimum spacing distances between condensers. The positioning shown in Fig. 7.19c provides ready access of ambient air to both condensers.

7.18 WINTER OPERATION OF EVAPORATIVE CONDENSERS.

Many plants operate in geographic regions where ambient temperatures fall to near or below freezing. Two measures to prevent the spray water from freezing in such instances are:

(1) to locate the sump in a warm area, as in Fig. 7.20

(2) to drain the water from the condenser and operate the condenser dry.

The indoor sump in Fig. 7.20 must be able to accommodate all the water normally in suspension in the condenser during operation. The cost of pumping the spray water will be slightly higher than experienced when the sump is integral

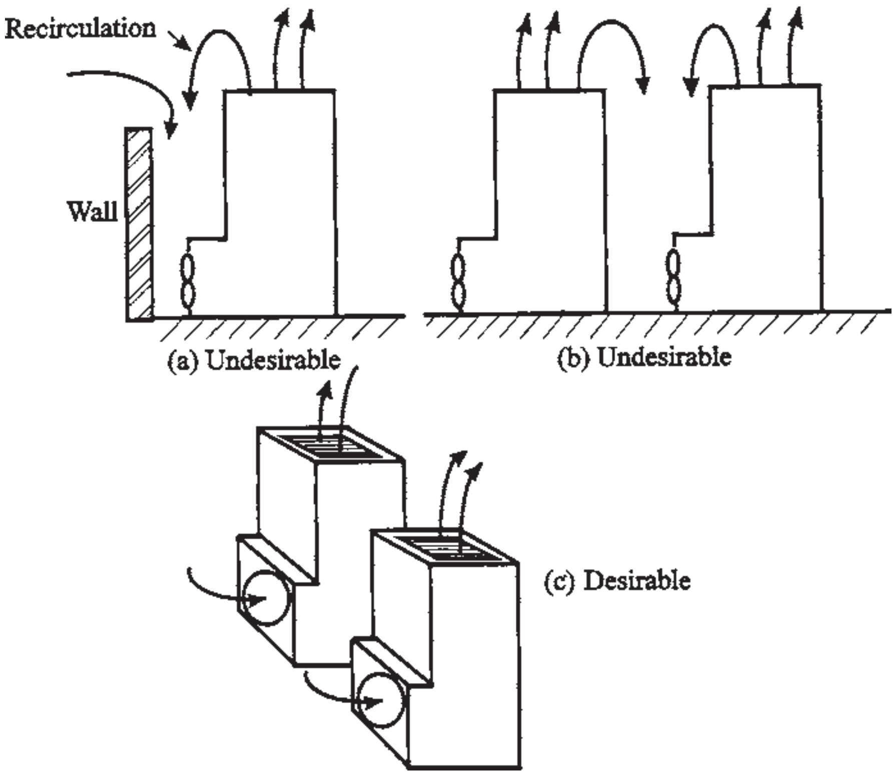

FIGURE 7.19
The condensers should be positioned so that the flow of entering air is not obstructed and as little as possible discharge air recirculates to the entrance.

to the condenser because of the additional head attributable to the difference in elevation between the condenser and the sump.

The indoor sump will prevent freezing of the main body of water in extremely cold weather, but the water droplets that drift out of the condenser may freeze close to the condenser causing icing conditions. An alternative to the indoor sump is to drain the condenser and operate dry. The heat-transfer capacity[16] of a condenser operating dry is strikingly less than when the condenser operates with the water sprays. Figure 7.21 shows relative capacities of a condenser operating dry compared to wet operation. It is significant that the dry condenser does not duplicate the wet capacity until the ambient dry-bulb temperature is about -30°C (-22° F). In many industrial refrigeration plants the load drops off as the ambient temperature drops, so the condenser can be shifted to dry operation when the ambient temperature drops below freezing. Such is not the case, however, for a plant with a predominantly process load which is only slightly affected by the ambient conditions. The sharp reduction in heat-transfer

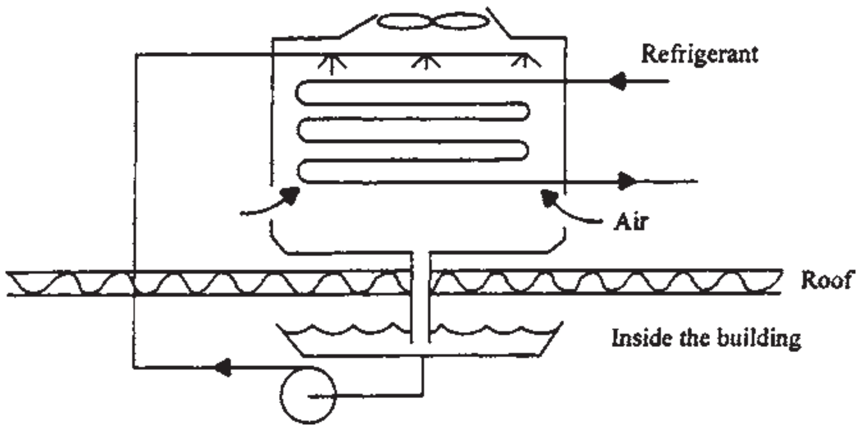

FIGURE 7.20
Locating the sump indoors to prevent freezing of the spray water during winter.

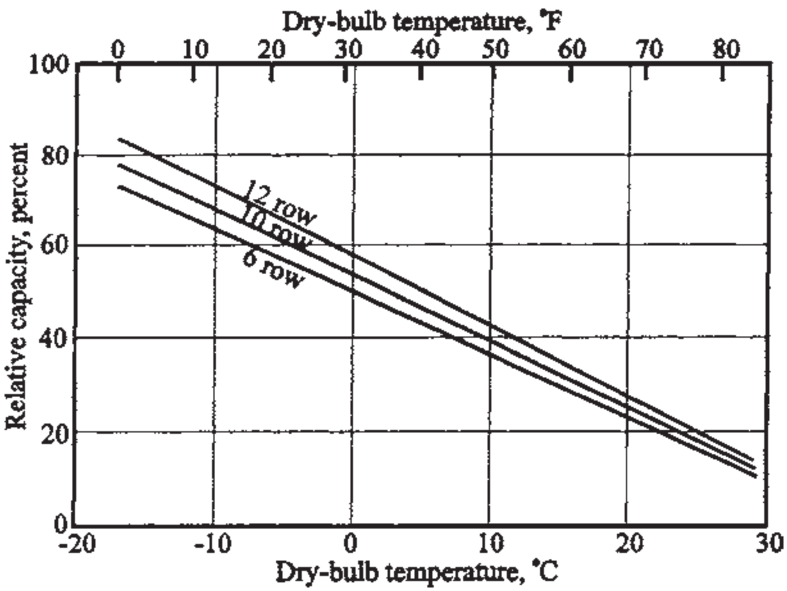

FIGURE 7.21
Heat-transfer capacity of a condenser operating dry compared to rated capacity operating wet with a condensing temperature of 35°C (95°F).

TABLE 7.3
Evaporating temperatures corresponding to standard atmospheric pressure of 101 kPa (14.7 psia).

Refrigerant	Evaporating temperature below which air could be drawn into the system through leaks
Ammonia	−33.5°C (−28.3°F)
R-22	−40.8°C (−41.5°F)
R-404a	−46°C (−50.8°F)
R-507	−46.7°C (−52°F)

capacity under dry operation emphasizes that the evaporation process is the dominant one for transfer of heat to air in the evaporative condenser.

One incentive for a plant to shift to dry operation is that the costs of water treatment are usually a function of operating time, and this cost can be eliminated by dry operation.

7.19 PURGING THE CONDENSER OF AIR

Air and other noncondensible gases may enter a system through leaks in seals, gaskets, or uncapped valves. Air may also be present because of imperfect evacuation before the initial charging of the system or due to impurities in the refrigerant or oil. Another way that air gains access to the system is when an evaporator coil or a compressor is opened. Air could be drawn into the system through leaks in the low-pressure portion of the system when operating with refrigerant pressures below atmospheric, which will occur at evaporating temperatures shown in Table 7.3. Air drawn into the system on the low-pressure side is eventually pumped to the condenser where the liquid seal prevents it from traveling further.

The presence of noncondensables in condensers penalizes the system performance through the artificial elevation of the condensing pressure. As pictured in Fig. 7.22, the noncondensables add their partial pressure to that of the refrigerant vapor and thus increase the pressure against which the compressor must work. A further penalty is the reduction in the heat-transfer coefficient by requiring the refrigerant to diffuse through the noncondensables on its way to the tube surface where it condenses.

A test of the need for purging is to compare the actual pressure to the saturation pressure at the temperature of liquid at a location where liquid and vapor are in equilibrium, such as in the receiver of Fig. 7.22. If the actual pressure p is significantly higher than the saturation pressure corresponding to t, purging is warranted. Purging may be performed on rare occasions in small systems, but is often done frequently by automatic purgers on large systems. These automatic purgers proceed from one purge point to another to extract gas.

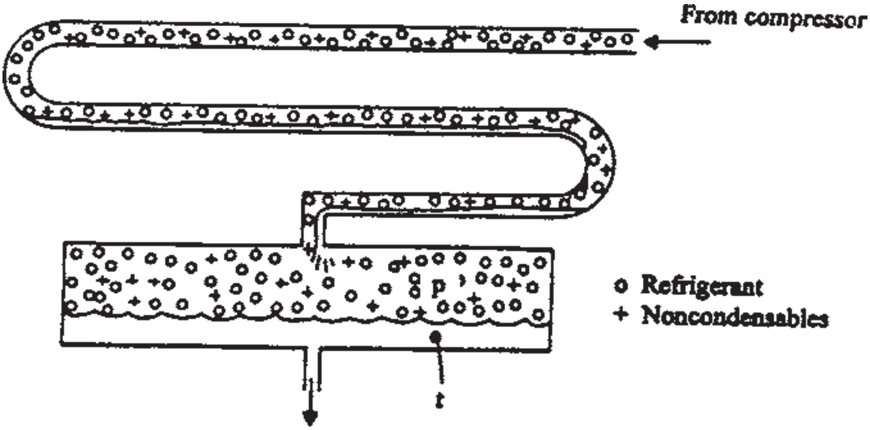

FIGURE 7.22
Noncondensables in a condenser.

There are preferable locations for purging, and basically these are (1) on the high-pressure side of the system, (2) where only vapor exists, and (3) where the vapor velocity is low. Air at a given pressure and temperature is more dense than ammonia, and not as dense as the halocarbon refrigerants, but no appreciable settling of one of the constituents can be aniticipated. The air diffuses quite uniformly throughout slow-moving refrigerant.

The three principal concepts available for purging are

- direct venting of the air-refrigerant mixture
- compression of the mixture, condensing as much as possible of the refrigerant, and venting the vapor mixture that is now rich in noncondensables
- condensation of refrigerant using a small evaporator, followed by venting of the air-refrigerant mixture

Figure 7.23a shows the first method, a primitive, manual technique. Vapor is released from a high-pressure vessel, such as the receiver, and this vapor is mostly refrigerant but also contains a small amount of the noncondensables that are the target. In the case of ammonia, the discharge bubbles through a container of water to absorb the ammonia. As venting proceeds, more refrigerant liquid vaporizes, so the concentration of noncondensables decreases, but never drops to zero. This method wastes considerable refrigerant to expel a small amount of noncondensables.

The second method of purging, as shown in Figure 7.23b, consists of drawing a sample from the vessel with a small compressor that elevates the pressure and condenses some refrigerant on a water-cooled coil. The vapor vented from

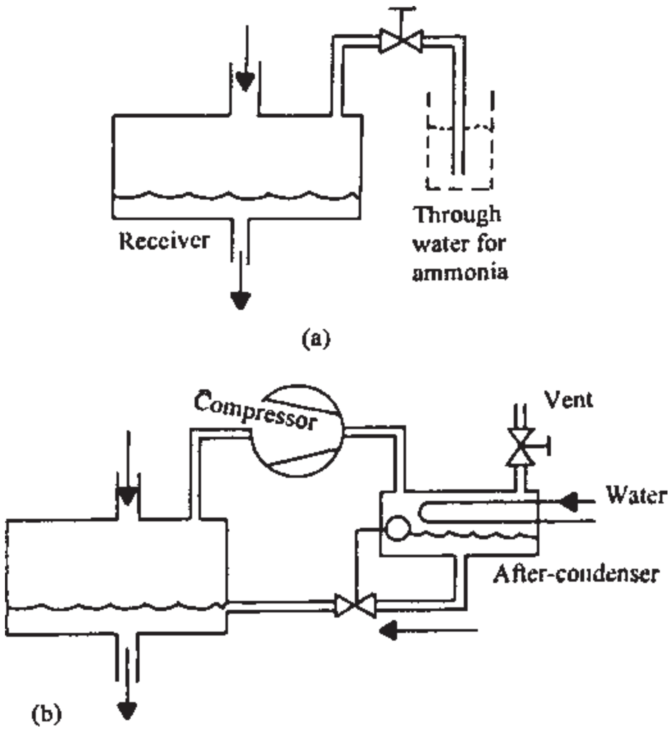

FIGURE 7.23
Two methods of purging noncondensables, (a) direct venting, and (b) compression of the refrigerant followed by condensation.

this after-condenser is higher in noncondensable content than at the original sampling position. This purging concept is widely applied in centrifugal-compressor water-chiller systems using such low-pressure refrigerants as R-123, but sees limited application in industrial refrigeration.

The third concept in purging (Figure 7.24), which is widely used in industrial refrigeration, avoids the need of a separate compressor, and instead uses a low temperature developed in a small evaporator. The air- refrigerant mixture from the condenser or receiver bubbles through cold liquid and condenses most of the refrigerant. This concept is embodied in automatic purgers which move from one purge point to another allowing enough time at each for a satisfactory purge. Commercial models of refrigerated purgers employ refinements in handling the vented stream leaving the after- condenser. The proper procedure is to purge one point at a time, because if one solenoid control valve serves two or more purge points, the pressure at these positions will be equalized during purging. In the later sections of this chapter, the need to properly regulate pressure differentials will be emphasized.

A manual technique is sometimes used for a massive purge that would require a long time for automatic purgers to handle. The method applies to a

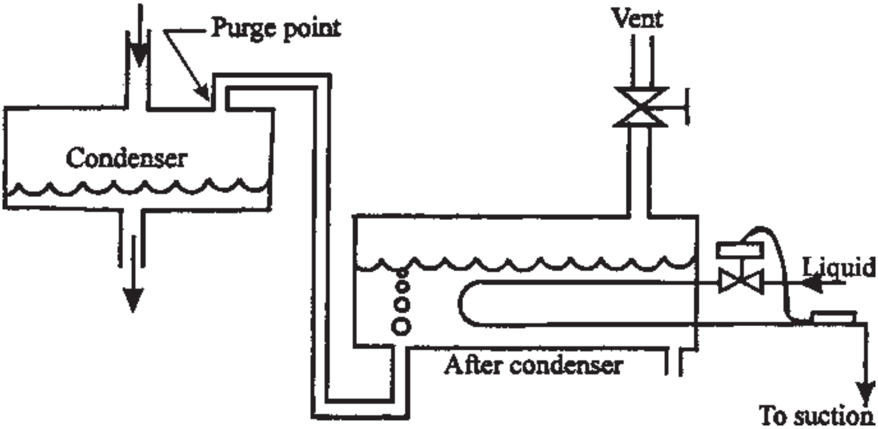

FIGURE 7.24
Purger for industrial refrigeration systems condenses refrigerant on a low-temperature evaporator.

multiple-coil condenser, such as shown in Fig. 7.25, that is equipped with individual valves from the discharge gas header and individual vents to atmosphere. If one coil is to be purged, the valve in the gas line is closed, but its fan and spray water continue to operate. Other parallel coils continue to operate normally, so the temperature in the coil being purged drops and ammonia vapor in the coil condenses. As the vapor condenses, the vapor volume decreases and liquid is drawn from the condensate line. After a period of time most of the ammonia vapor has condensed, leaving the small volume of mostly noncondensables which can be vented to the atmosphere or to a vessel of water.

7.20 INTENTIONAL SUBCOOLING IN A CONDENSER

An objective to be pursued in the next several sections is to effectively drain the condenser of liquid so that the maximum amount of surface area will be available for condensation. Before addressing that goal, however, a technique sometimes used in commercial refrigeration (e.g., supermarkets) that operate year-round will be shown. The condenser pressure must be prevented from dropping so low that the thermal expansion valves (typically used in that type of system) are unable to pass a sufficient flow rate of refrigerant. The condensing pressure must be artificially prevented from dropping during cold weather operation, and a procedure, such as the one shown in Fig. 7.26, backs liquid into the condenser to reduce its heat-transfer capacity and maintain the desired level of condensing pressure.

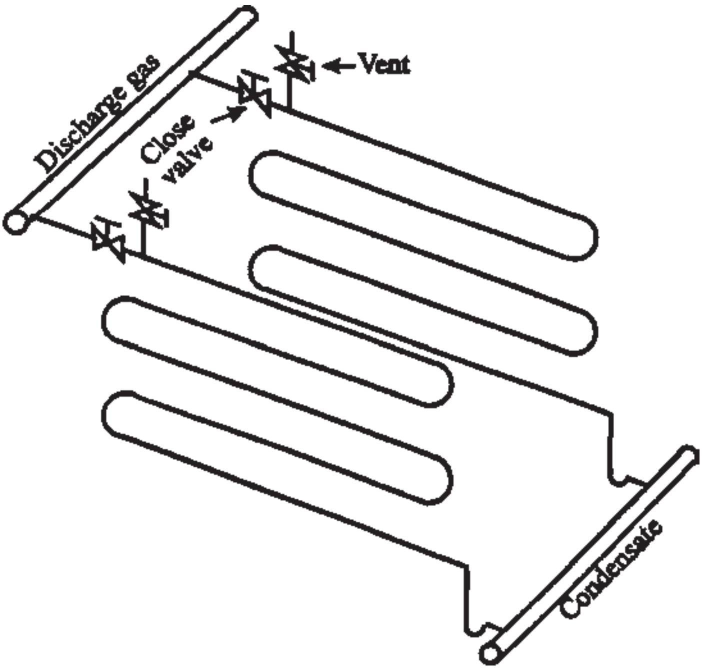

FIGURE 7.25
Manual purge of condenser tubes.

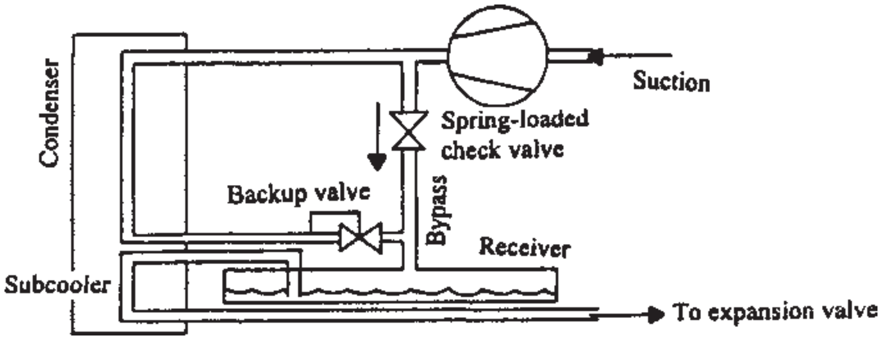

FIGURE 7.26
Condenser piping arrangement used on some commercial refrigeration systems to prevent drop of condensing pressure below a preset point.

The operating efficiencies offered by lower condensing pressures are thus not available, although the subcooling coil does reduce the enthalpy of liquid passing to the expansion valve and thereby increases the refrigerating effect.

7.21 PIPING OF HIGH-PRESSURE RECEIVERS

The condenser normally drains its liquid refrigerant into the high-pressure receiver, which is a vessel providing storage space for the condensate. The considerations that surround the condenser/receiver subsystem are:

- piping arrangements of the receiver
- connecting the vapor space in the receiver to vapor locations in the condenser
- the role of refrigerant pressure drop in the condenser

The two typical piping configurations of the receiver are shown in Figure 7.27. Figure 7.27a shows the top inlet or mixing-type receiver where all the condensate passes through the receiver. Usually the outlet of this receiver is as shown, with the tip of the outlet line not quite touching the bottom of the receiver. This construction avoids passing solid contaminants on to the system.

In the bottom inlet receiver of Figure 7.27b, most of the liquid passes directly to the evaporators. The only liquid flow in or out of the receiver is associated with the rise or fall in liquid level. The liquid level may shift because of transient differences in the condensation rate and the rate of liquid flow to the evaporators. Some designers prefer the bottom inlet receiver over the top inlet for two reasons: (1) the ability to use subcooled liquid when it is available from the condenser, and (2) inherent trapping of the liquid line which, as explained later, is a necessity for good drainage in multiple condenser installations. In the top inlet receiver of Figure 7.27a, the liquid and vapor are in equilibrium, so saturated liquid passes on to the low side of the system. Even if subcooled liquid enters the receiver, it will quickly assume the temperature of the stored liquid—a temperature heavily influenced by the machine room temperature. Because the machine room temperature will be higher in most cases than that of the liquid coming from the condenser, the available subcooling is lost in the top inlet receiver.

One of the considerations mentioned above is to provide a connection from the vapor in the receiver to vapor space in the condenser. The liquid level in the receiver of an industrial refrigeration system is almost constantly rising or falling because liquid flow rate from the condenser is not precisely the same as the rate demanded of the low side of the system. When an excess flow enters the receiver, there is a tendency to compress the vapor, which builds up the pressure in the receiver and temporarily restricts the flow of condensate from the condenser. If the flow rate demanded by the system exceeds temporarily the rate provided by the condenser, the pressure in the receiver drops and some of the liquid vaporizes. The requirement of a connection between the vapor in the receiver with vapor in the condenser will be a thread running through

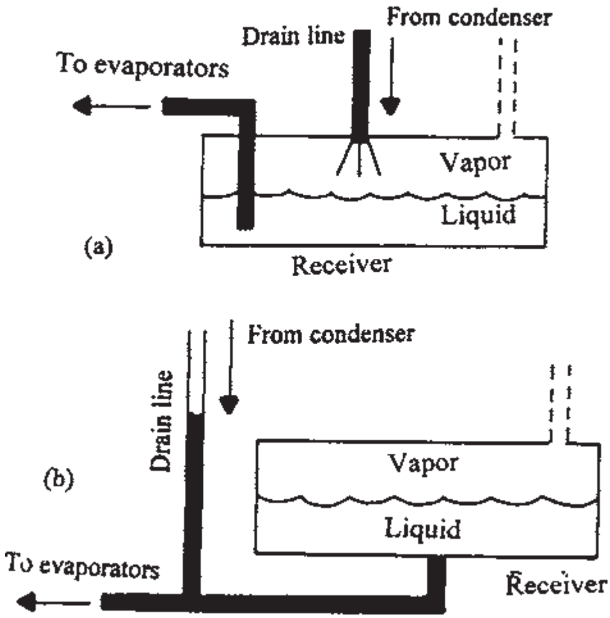

FIGURE 7.27
Piping arrangements for the receiver: (a) top inlet, mixing type, and (b) bottom inlet or surge receiver.

recommended procedures for drain piping that will be presented in the next several sections.

7.22 DRAINING CONDENSATE—SINGLE CONDENSER

The arrangement to insure proper draining of a single condenser depends on the type of receiver. If the receiver is top inlet, good drainage of the condensate results if the vapor in the receiver can flow freely counterflow to the condensate, as appears in Fig. 7.28. The provisions[15] for achieving open channel flow in the drain line are:

- avoid a horizontal drain line, if possible, sloping the line with a pitch of at least 1 in 50
- choose a large-size pipe based on liquid velocities no higher than about 0.5 m/s (100 fpm)
- do not place a valve in the sloping drain line with its stem upward. Instead use angle valves which introduce much less pressure drop than straight-through globe valves. If a straight-through valve is used, place it in the vertical portion of the drain line

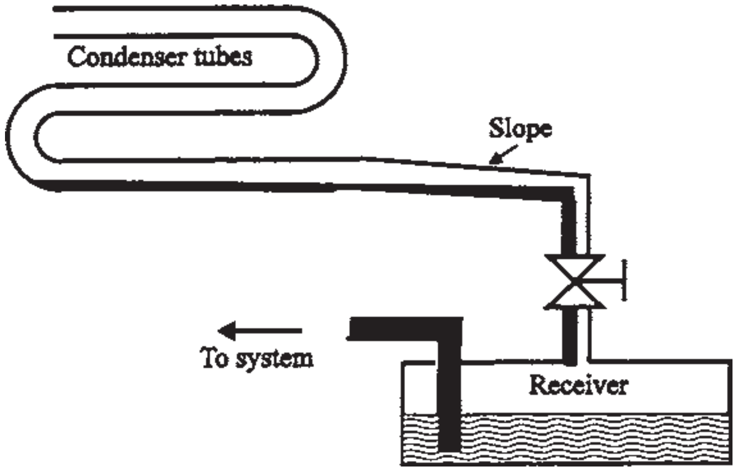

FIGURE 7.28
Proper draining of a single condenser to a top inlet receiver.

If the receiver is bottom inlet, as in Fig. 7.29, the liquid is inherently trapped and the vapor in the receiver has no access to the vapor space in the condenser. For this reason an equalizer line should be connected from the top of the receiver to the outlet of the condenser.

7.23 PRESSURE PROFILES IN A CONDENSER

Figure 7.30 shows changes in pressure that develop in a condenser when the equalizer line connects the top of the receiver to the inlet to the condenser. As the condensing refrigerant flows down through the condenser, a pressure drop occurs, $-\Delta p$. The equalizer line forces the pressure in the receiver to be the same as that of the condenser inlet, so some means must be found to recover the drop in pressure that occurs in the condenser. The $+\Delta p$ to cancel the $-\Delta p$ derives from the static head of a column of liquid refrigerant. In Figure 7.30 the liquid column is adequate to provide this gain in pressure. If the difference in elevation from the bottom of the condenser to the liquid level in the receiver is inadequate, the system achieves the necessary column of liquid by backing liquid into the condenser. Liquid that is forced back into the condenser is the root of many low-capacity problems with condensers. Incidentally, the arrangement shown in Figure 7.30 is a satisfactory alternate to the configuration of Figure 7.29, but Figure 7.29 is preferred because it does not require as much length of liquid column and the equalizer line is shorter.

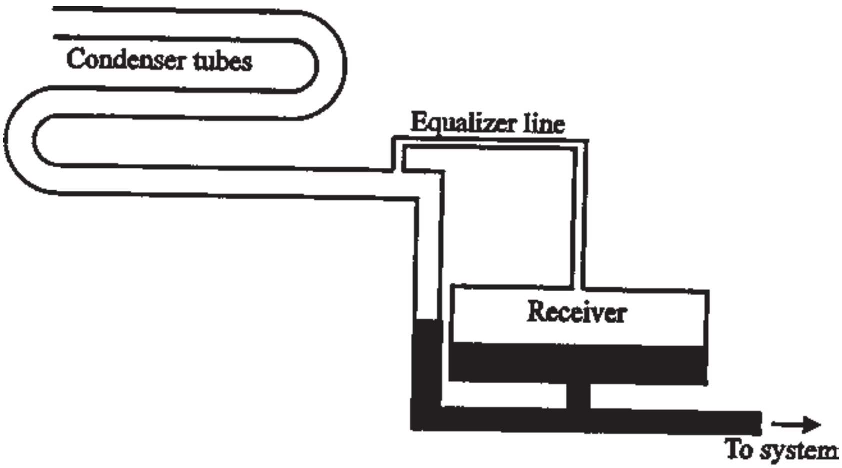

FIGURE 7.29
Proper draining of a single condenser to a bottom inlet receiver.

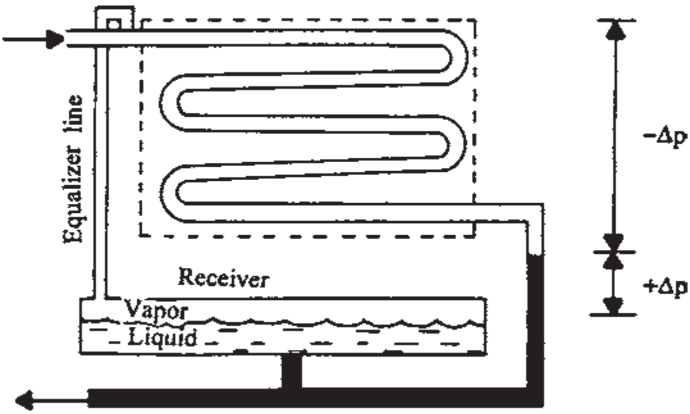

FIGURE 7.30
Pressure profiles when the receiver is equalized to the top of the condenser.

7.24 DRAINING CONDENSATE—PARALLEL CONDENSERS

A characteristic of industrial refrigeration plants is the use of parallel condensers and compressors. This fact is especially pertinent to ammonia systems, and not always true for halocarbon systems where the oil transfer between parallel units sometimes becomes a problem. Plants of even moderate size are usually designed for parallel condensers to offer flexibility in meeting a wide range of load variation. When condensers are piped in parallel, the following rules should be observed:

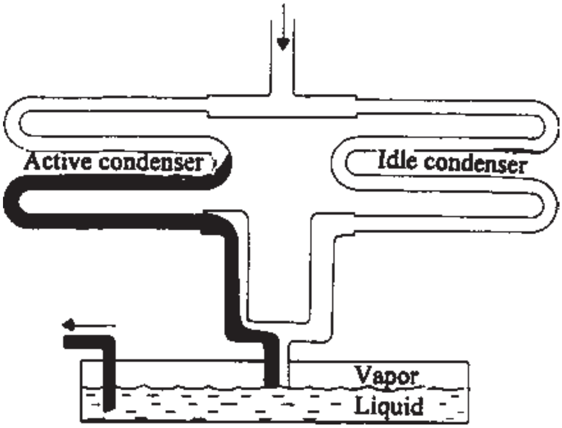

FIGURE 7.31
Backup of liquid into one of two parallel condensers.

- trap the liquid drain lines
- provide a generous length of vertical drain line
- install an equalizer line between the receiver and the entrance of the condensers

The purpose of trapping the liquid drain lines is to aid in drainage of liquid from all condensers. The potential drainage problem with multiple parallel condensers is illustrated in Fig. 7.31 where the condenser on the right has, at least at this moment, a low flow rate of refrigerant. Some reasons for the low flow rate include: a different design of condenser than the other or the fans are completely or partially shut down.

The pressure drop through both condensers must be the same because there are two common points in the piping—at the inlet and the outlet of the condensers. The only way the active condenser on the left can operate with the low pressure drop of the condenser on the right is if liquid backs up into the condenser tubes. Assume that the drain lines are large, so the liquid head in the tubes on the left condenser supplements the available pressure difference.

To avoid the problem of liquid backup into one of the condensers, liquid lines from both condensers should be trapped, using arrangements such as those shown in Fig. 7.32. Once again, the pressure drop through the two condensers from their common inlet to their common outlet is identical, but because the liquid lines are trapped, the drain line is full, and the difference in liquid level in the vertical drain line compensates for the difference in pressure drop. Thus, the liquid head in the left condenser builds up in the drain line and not in the condenser tubes where it would adversely affect the condenser performance. The bottom inlet receiver in Figure 7.33 inherently provides liquid traps for each condenser.

Implied in the correct piping of Figures 7.32 and 7.33 is an adequate length of the vertical leg. An estimate of the needed length of liquid column can be derived from the knowledge that the maximum pressure drop in an operating ammonia condenser is usually about 3.4 kPa (1/2 psi), which is the maximum pressure difference to be compensated for between an idle and an operating condenser. A liquid column of 0.6 m (2 ft) would compensate for this pressure difference, but most designers attempt to place the condenser high enough to permit a 1.2-m (4-ft) column of ammonia. Liquid R-22 has twice the density of liquid ammonia which would suggest that only half the length of liquid column would be needed for R-22, but the pressure drop in an R-22 condenser is about four times that of an ammonia condenser. The reason for the higher pressure drop with R-22 is because of its low latent heat, requiring perhaps six times the flow of R-22 compared to ammonia for a given heat-rejection capacity of the condenser. The recommended length of liquid column in an R-22 installation is 2.4 m (8 ft).

An equalizer line is required since the vapor from the top of the receiver cannot vent back to the condensers through the drain line because of the liquid traps as it could in the single condenser in Figure 7.28. Also the pressure at the outlet of the condenser tubes may be different, so the equalizer line must be connected to the top of the condensers where the pressure is the same for both condensers.

7.25 DRAINING CONDENSATE—THERMOSYPHON OIL COOLING

When the oil for screw compressors is cooled by refrigerant circulating by means of a thermosyphon, some of the same principles of condenser draining apply, but there are some additional considerations. A vessel, called the thermosyphon receiver, is needed to separate the liquid from the liquid/vapor mixture that rises from the oil cooler. The vapor thus separated passes to the discharge gas header, as shown in Figure 7.33. The return of liquid/vapor from the oil cooler should flow to the thermosyphon receiver and not to the discharge header of the condenser. Were the liquid from the oil cooler to enter the condenser, it would overload it with liquid and reduce its heat-transfer capacity.

A significant fact is that there is appreciable flow of vapor through the vapor line from the thermosyphon receiver, so there is a pressure drop in this line. Thus the pressure in the thermosyphon receiver is higher than that of the condenser inlet and certainly higher than the condenser outlet. To overcome this pressure difference a liquid column must develop somewhere. Figure 7.33 shows bottom inlet to the thermosyphon receiver with individual liquid columns for each condenser. Entrance of the condensate to the top of the thermosyphon receiver would work satisfactorily if the drains were equipped with the P-traps of Figure 7.32a. Even if the installation consists of only one condenser, there is

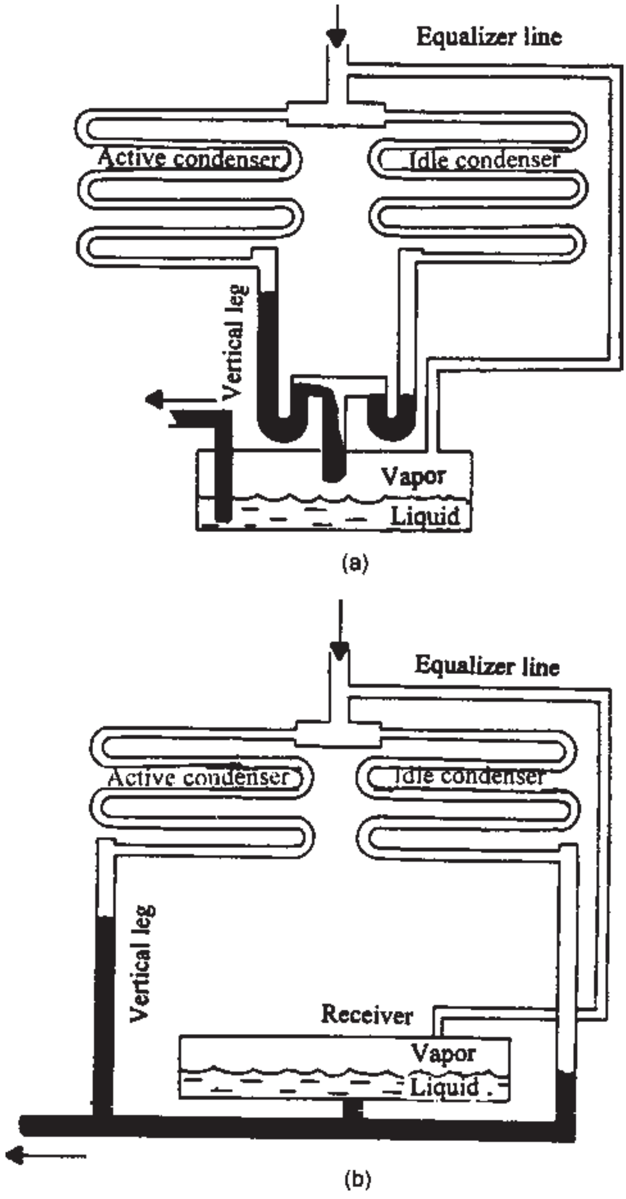

FIGURE 7.32
Trapping the liquid lines and using the liquid head in the vertical drain line to compensate for differences in pressure drop in the condensers in (a) a top inlet receiver and (b) a bottom inlet receiver.

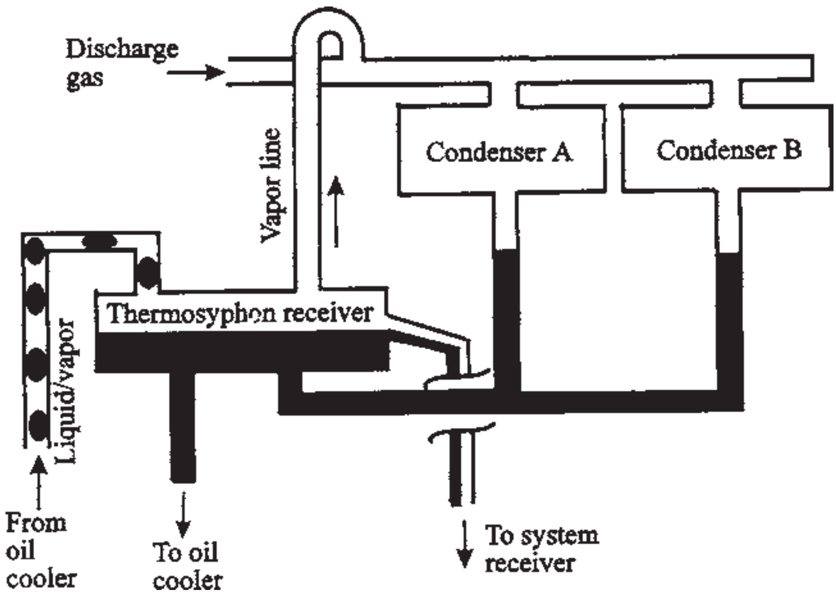

FIGURE 7.33
Trapping the liquid drain lines from multiple condensers that serve a thermosyphon receiver.

still the need to build up pressure from the outlet of the condenser to the thermosyphon receiver. A convenient means of providing a liquid seal to form a liquid trip is to immerse the drain line from the condenser below the liquid level maintained by the overflow to the system receiver.

7.26 SIZE OF EQUALIZER LINES AND THERMOSYPHON VAPOR LINES

Under normal circumstances equalizer lines between the receiver and the inlet or outlet of the condenser can be small, because it is not expected that they need carry high flow rates. They only need to convey a volume flow rate of vapor equal to the rate of change of liquid volume. There is one situation where the equalizer line should be generously sized and that is during a shutdown of a plant located in a cold ambient temperature but with the receiver located in a warm machine room. There will be a constant vaporization of the liquid in the receiver causing vapor to flow through the equalizer line to the discharge gas header where it will condense in the condenser. The pressure drop resulting from the vapor flow in the equalizer line must be compensated by a liquid column that will develop and back liquid into the condenser. For high pressure drops of, for example, 25 kPa (3.7 psi), an ammonia liquid column of 4.6 m (15 ft) may

TABLE 7.4
Flow and refrigeration capacity of ammonia thermosyphon vapor lines and equalizer lines

Pipe size, inch	Thermosyphon vapor line		Equalizer lines	
	kg/s	lb/min	kW	tons
3/4			176	50
1			352	100
1-1/4			528	150
1-1/2	0.0529	7	792	225
2	0.0907	12	1056	300
2-1/2	0.166	22	1760	500
3	0.295	39	3520	1000
4	0.529	70	7040	2000
5	0.907	120		
6	1.66	220		
8	5.29	700		

be needed. This elevation could be enough to push all of the liquid of the system into the condenser resulting in difficulty in restarting the plant.

As was pointed out in the previous section, the vapor line between the thermosyphon receiver and the discharge gas header (Figure 7.33) is more than a low-flow equalizer line, but carries a heavy flow of vapor and will be of larger size than an equalizer line. Table 5.3 in the chapter on screw compressors listed recommended pipe sizes for this line[17], which are reproduced here in Table 7.4 along with recommended equalizer line sizes[18].

7.27 SUPPLEMENTARY COOLING REQUIREMENTS PROVIDED BY EVAPORATIVE CONDENSERS.

In addition to the major task of condensing refrigerant, evaporative condensers are sometimes called on to cool some other fluid as well. The prime example of this supplementary cooling requirement, and a frequent one, is to cool the oil injected in screw compressors. To accomplish this assignment, pump water from the sump to the oil-cooling heat exchanger and return the warmed water to the sump. This additional cooling load reflects itself in the performance of the condenser, as shown in Figure 7.34, where the solid lines show the performance without, and the dashed lines with external cooling by the sump water.

When no supplementary cooling is demanded, the water temperature in the sump at A is the same as that sprayed over the tubes at the top of the condenser, point B. This equality of temperatures is inherent because, as Figure 7.1c shows, there is a direct connection from the sump to the sprays. When the sump water provides supplementary cooling, the spray water temperature at B' will be higher than temperature A. The consequence of the higher spray-water temperature

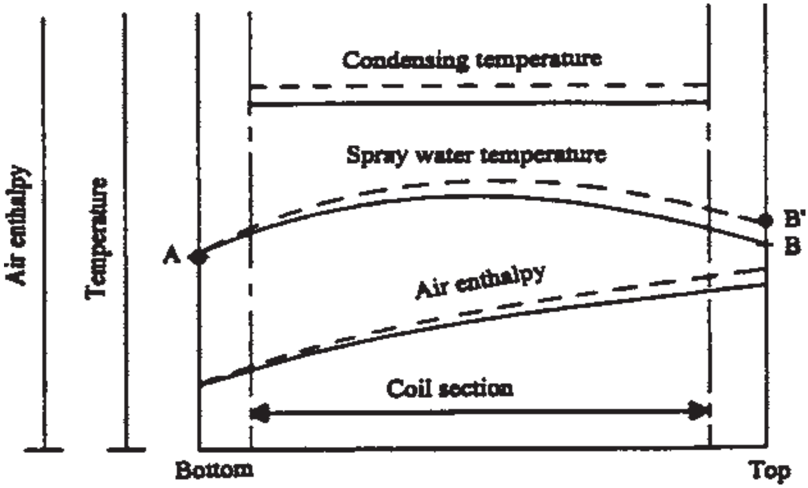

FIGURE 7.34
The solid lines show the temperatures of refrigerant and spray water and the enthalpy of air when the condenser provides no supplementary cooling, and the dashed lines show the performance when the sump water performs external cooling.

is a greater rise in air enthalpy through the condenser, and a higher condensing temperature of the refrigerant.

If an additional load is imposed on the condenser by adding heat to the sump water, the condenser will accommodate the addition—afact which is shown in Fig. 7.34 by the increased rise in air enthalpy through the condenser. The increased capacity does not come without a cost, however, and that cost is the increased condensing temperature of the refrigerant.

Manufacturers of evaporative condensers usually recommend a separate closed circuit evaporative cooler for the supplementary cooling duty to permit separate control of the refrigerant temperature and the fluid temperature, and to allow for dry operation during cold weather.

7.28 WATER TREATMENT IN EVAPORATIVE CONDENSERS

Evaporation of spray water is the dominant process of heat rejection in evapoartive condensers. The amount of heat transfer attributible to the difference in dry-bulb temperatures between the condenser tubes and the air is minor. Because the makeup water contains some minerals and other chemicals, and because the water vapor leaves the condenser with none of these impurities, the concentration of foreign materials in the spray water tends to increase. To keep the concentration of impurities under control, blowdown of some of the spray water should be provided whenever the condenser operates. Manufacturers of evaporative condensers typically recommend that the blowdown rate be approximately equal to the evaporation rate. The rate of makeup water is thus twice the blowdown rate.

Almost all sources of makeup water must be treated in some way to avoid one or more of the problems that beset the surfaces of the condenser that are in contact with water. These difficulties include scaling, fouling, and corrosion. Scaling is the deposit of a hard layer of minerals, usually calcium carbonate ($CaCO_3$), on the tube surfaces. A layer of scale 0.8 mm (1/32 in) thick on the tubes can be expectecd to reduce the capacity of the condenser by 30%. This layer of scale is usually due to high mineral concentration in the makeup water and if calcium carbonate precipitates it can form scale on the tubes. Generally a $CaCO_3$ concentration of less than about 170 ppm will be satisfactory. It should not be implied, however, that softened water with a mineral concentration of perhaps 30 ppm is desirable. Softened water may result in excessive corrosion.

Fouling usually refers to the accumulation of nonscale solids, such as dirt, silt, sand, algae, fungi, and bacteria.

Corrosion is a distressing problem in galvanized steel because it often takes the form of deteriorating the zinc coating, which exposes the steel to oxidation. Corrosion is an electrochemical process where an electrical potential develops between two different metals. When current flows as a result of this difference in voltage in the presence of an electrolyte, such as water with dissolved solids, one of the metals dissolves. A particularly sensitive situation is where the zinc coating on steel has deteriorated, because it is between these points that current will flow. Controlling the pH value of the spray water is particularly helpful in retarding corrosion. The pH value is an indicator of the alkalinity or the acidity of a solution, with a pH value of 7 being defined as neutral. Maintenance of a pH value of between 6 and 8 is usually recommended[19]. Corrosion inhibitors approved by the condenser manufacturer are also treatments to prevent corrosion.

A problem called *white rust* has appeared in the past decade and may be associated with the prohibition on the use of chromates for corrosion protection. White rust is the accumulation of a white, waxy, nonprotective zinc corrosion product on galvanized surfaces[20]. Typically it will appear suddenly and progress rapidly over the wetted, galvanized steel components of the condenser. If not corrected, white rust may lead to premature failure of the galvanized coating. One of the most effective methods of preventing white rust is to assure the *passivation* of the condenser during initial operation. Passivation is the natural formation of a protective, light crystalline film on the zinc surface. A recent trend in water treatment has been toward greater alkalinity (high pH), which some experts suspect promotes white rust. Indications are that soft water (less than 30 ppm total hardness) combined with high pH valves exacerbate the problem. Passivation is facilitated by initial use of untreated water and also by the use of phosphates. A water treatment expert should be consulted for the unique approach applicable to local conditions.

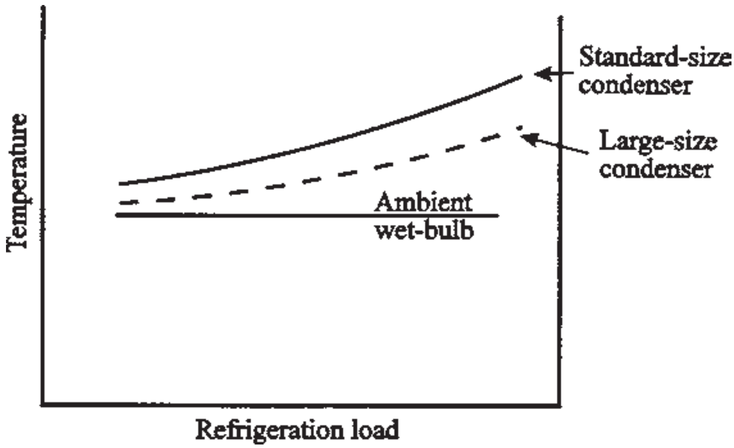

FIGURE 7.35
Condensing temperatures as affected by the refrigeration load and the condenser size.

7.29 THE CONDENSER AS A PART OF THE SYSTEM

The condenser operates according to its own rules, but it is also a part of the refrigeration system and therefore interacts with the other components. Except in the low-condensing temperature regions where certain operating limits may apply, it is always desirable to operate at as low a condensing temperature as possible. But, as Fig. 7.35 shows, an increase m the refrigeration load increases the condensing temperature in order to transfer a higher heat flow rate.

It is unfortunate that the condensing temperature increases and the compressor power per unit refrigerating capacity increases at just the time when the refrigeration rate is high. Figure 7.35 also shows that the condensing temperatures can be lowered at all ranges of refrigeration load if a large condenser is chosen. Certainly there would be additional first cost associated with the larger condenser, and perhaps additional fan power, but the compressor power drops throughout its life. A further advantage of the large condenser with its lower condensing temperature is that the peak refrigerating capacity of the compressor can be increased slightly.

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