Related Commercial Resources

CHAPTER 38

CONDENSERS

THE CONDENSER in a refrigeration system is a heat exchanger that rejects all the heat from the system. This heat consists of heat absorbed by the evaporator plus the heat from the energy input to the compressor. The compressor discharges hot, high-pressure refrigerant gas into the condenser, which rejects heat from the gas to some cooler medium. Thus, the cool refrigerant condenses back to the liquid state and drains from the condenser to continue in the refrigeration cycle.

Condensers may be classified by their cooling medium as (1) water-cooled, (2) air-cooled, (3) evaporative (air- and water-cooled), and (4) refrigerant-cooled (cascade systems). The first three types are discussed in this chapter; see Chapter 39 in the 2006 *ASHRAE Handbook—Refrigeration* for a discussion of cascade-cooled condensers.

WATER-COOLED CONDENSERS

HEAT REMOVAL

The heat rejection rate in a condenser for each unit of heat removed by the evaporator may be estimated from the graph in Figure 1. The theoretical values shown are based on Refrigerant 22 with 10°F suction superheat, 10°F liquid subcooling, and 80% compressor efficiency. Actually, the heat removed is slightly higher or lower than these values, depending on compressor efficiency. Usually, the heat rejection requirement can be accurately determined by adding the known evaporator load and the heat equivalent of the actual power required for compression (obtained from the compressor manufacturer's catalog). (Note that heat from the compressor is reduced by any independent heat rejection processes such as oil cooling, motor cooling, etc.)

The volumetric flow rate of condensing water required may be calculated as follows:

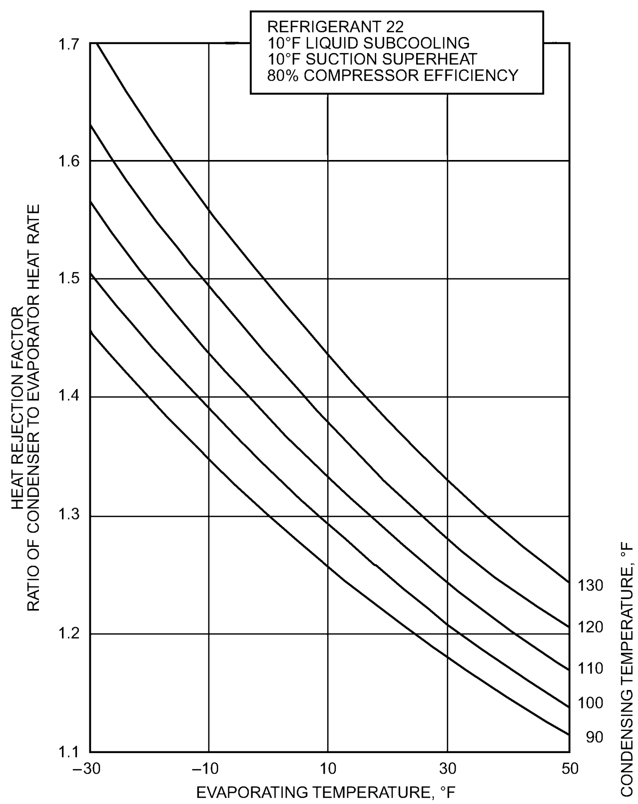


Fig. 1 Heat Removed in Condenser

$$Q = \frac{q_o}{\rho c_p (t_2 - t_1)} \qquad (1)$$

where

Q = volumetric flow rate of water, ft^3/h (multiply ft^3/h by 0.125 to obtain gpm)

The preparation of this chapter is assigned to TC 8.4, Air-to-Refrigerant Heat Transfer Equipment; TC 8.5, Liquid-to-Refrigerant Heat Exchangers; and TC 8.6, Cooling Towers and Evaporative Condensers.

q_o = heat rejection rate, Btu/h
ρ = density of water, lb/ft^3
t_1 = temperature of water entering condenser, °F
t_2 = temperature of water leaving condenser, °F
c_p = specific heat of water at constant pressure, Btu/lb·°F

Example 1. Estimate volumetric flow rate of condensing water required for the condenser of an R-22 water-cooled unit operating at a condensing temperature of 105°F, an evaporating temperature of 40°F, 10°F liquid subcooling, and 10°F suction superheat. Water enters the condenser at 86°F and leaves at 95°F. The refrigeration load is 100 tons.

Solution: From Figure 1, the heat rejection factor for these conditions is about 1.19.

q_o = 100 × 1.19 = 119 tons
ρ = 62.1 lb/ft^3 at 90.5°F
c_p = 1.0 Btu/(lb·°F)

From Equation (1):

$$Q = \frac{1496 \times 119}{62.1 \times 1.0(95-86)} = 319 \text{ gpm}$$

Note: The value 1496 is a unit conversion factor.

HEAT TRANSFER

A water-cooled condenser transfers heat by sensible cooling in the gas desuperheating and condensate subcooling stages and by transfer of latent heat in the condensing stage. Condensing is by far the dominant process in normal refrigeration applications, accounting for 83% of the heat rejection in Example 1. Because the tube wall temperature is normally lower than the condensing temperature at all locations in the condenser, condensation takes place throughout the condenser.

The effect of changes in the entering gas superheat is typically insignificant because of an inverse proportional relationship between temperature difference and heat transfer coefficient. As a result, an average overall heat transfer coefficient and the mean temperature difference (calculated from the condensing temperature corresponding to the saturated condensing pressure and the entering and leaving water temperatures) give reasonably accurate predictions of performance.

Subcooling affects the average overall heat transfer coefficient when tubes are submerged in liquid. The heat rejection rate is then determined as

$$q = UA\,\Delta t_m \tag{2}$$

where

q = total heat transfer rate, Btu/h
U = overall heat transfer coefficient, Btu/h·ft^2·°F
A = heat transfer surface area associated with U, ft^2
Δt_m = mean temperature difference, °F

Chapter 3 of the 2005 *ASHRAE Handbook—Fundamentals* describes how to calculate Δt_m.

Overall Heat Transfer Coefficient

The overall heat transfer coefficient U_o in a **water-cooled condenser with water inside the tubes** may be computed from calculated or test-derived heat transfer coefficients of the water and refrigerant sides, from physical measurements of the condenser tubes, and from a fouling factor on the water side, using the following equation:

$$U_o = \frac{1}{\left(\frac{A_o}{A_i}\frac{1}{h_w}\right) + \left(\frac{A_o}{A_i}r_{fw}\right) + \left(\frac{A_o}{A_m}\frac{t}{k}\right) + \left(\frac{1}{h_r\phi_s}\right)} \tag{3}$$

where

U_o = overall heat transfer coefficient, based on external surface and mean temperature difference between external and internal fluids, Btu/h·ft^2·°F
A_o/A_i = ratio of external to internal surface area
h_w = internal or water-side film coefficient, Btu/h·ft^2·°F
r_{fw} = fouling resistance on water side, ft^2·h·°F/Btu
t = thickness of tube wall, ft
k = thermal conductivity of tube material, Btu/h·ft·°F
A_o/A_m = ratio of external to mean heat transfer surface areas of metal wall
h_r = external or refrigerant-side coefficient, Btu/h·ft^2·°F
ϕ_s = surface fin efficiency (100% for bare tubes)

For **tube-in-tube condensers or other condensers where refrigerant flows inside the tubes**, the equation for U_o, in terms of water-side surface, becomes

$$U_o = \frac{1}{\left(\frac{A_o}{A_i}\frac{1}{h_r}\right) + r_{fw} + \left(\frac{t}{k}\right) + \left(\frac{1}{h_w}\right)} \tag{4}$$

where

h_r = internal or refrigerant-side coefficient, Btu/h·ft^2·°F
h_w = external or water-side coefficient, Btu/h·ft^2·°F

For **brazed or plate and frame condensers** $A_0 = A_i$; therefore the equation for U_o is

$$U_o = \frac{1}{(1/h_r) + r_{fw} + (t/k) + (1/h_w)} \tag{5}$$

where t is plate thickness.

Water-Side Film Coefficient

Values of the water-side film coefficient h_w may be calculated from equations in Chapter 3 of the 2005 *ASHRAE Handbook—Fundamentals*. For turbulent flow, at Reynolds numbers exceeding 10,000 in horizontal tubes and using average water temperatures, the general equation (McAdams 1954) is

$$\frac{h_w D}{k} = 0.023\left(\frac{DG}{\mu}\right)^{0.8}\left(\frac{c_p\mu}{k}\right)^{0.4} \tag{6}$$

where

D = inside tube diameter, ft
k = thermal conductivity of water, Btu/h·ft·°F
G = mass velocity of water, lb/h·ft^2
μ = viscosity of water, lb/ft·h
c_p = specific heat of water at constant pressure, Btu/lb·°F

The constant 0.023 in Equation (6) reflects plain inner diameter (ID) tubes. Bergles (1995) and Pate et al. (1991) discuss numerous water-side enhancement methods that increase the value of this constant.

Because of its strong influence on the value of h_w, a high water velocity should generally be maintained without initiating erosion or excessive pressure drop. Typical maximum velocities from 6 to 10 fps are common with clean water. Experiments by Sturley (1975) at velocities up to approximately 26 fps showed no damage to copper tubes after long operation. Water quality is the key factor affecting erosion potential (Ayub and Jones 1987). A minimum velocity of 3 fps is good practice when water quality is such that noticeable fouling or corrosion could result. With clean water, the velocity may be lower if it must be conserved or has a low temperature. In some cases, the minimum flow may be determined by a lower Reynolds number limit.

For brazed or plate and frame condensers, the equation is similar to Equation (6). However, the diameter D is replaced by H, which is the characteristic spacing between plates.

Refrigerant-Side Film Coefficient

Factors influencing the value of the refrigerant-side film coefficient h_r are

- Type of refrigerant being condensed
- Geometry of condensing surface [plain tube outer diameter (OD); finned-tube fin spacing, height, and cross-sectional profile; and plate geometry]
- Condensing temperature
- Condensing rate in terms of mass velocity or rate of heat transfer
- Arrangement of tubes in bundle and location of inlet and outlet connections
- Vapor distribution and rate of flow
- Condensate drainage
- Liquid subcooling

Values of refrigerant-side coefficients may be estimated from correlations in Chapter 4 of the 2005 *ASHRAE Handbook—Fundamentals*. Information on the effects of refrigerant type, condensing temperature, and loading (temperature drop across the condensate film) on the condensing film coefficient is in the section on Condensing in the same chapter. Actual values of h_r for a given physical condenser design can be determined from test data using a Wilson plot (Briggs and Young 1969; McAdams 1954).

The type of condensing surface has a considerable effect on the condensing coefficient. Most halocarbon refrigerant condensers use finned tubes where the fins are integral with the tube. Water velocities normally used are large enough for the resulting high water-side film coefficient to justify using an extended external surface to balance the heat transfer resistances of the two surfaces. Pearson and Withers (1969) compared refrigerant condensing performance of integral finned tubes with different fin spacing. Some other refrigerant-side enhancements are described by Pate et al. (1991) and Webb (1984a). The effect of fin shape on the condensing coefficient is addressed by Kedzierski and Webb (1990). Ghaderi et al. (1995) reviewed in-tube condensation heat transfer correlations for smooth and augmented tubes.

In the case of brazed-plate or plate-and-frame condensers inlet nozzle size, chevron angle, pitch, and depth of the nozzles are important design parameters. For a trouble-free operation, refrigerant should flow counter to the water flow. Little specific design information is available; however, film thickness is certainly a factor in plate condenser design because of the falling film nature along the vertical surface. Kedzierski (1997) showed that placing a brazed condenser in a horizontal position improved U_o by 17 to 30% because of the shorter film distance.

Huber et al. (1994a) determined condensing coefficients for R-134a, R-12, and R-11 condensing on conventional finned tubes with a fin spacing of 26 fins per inch (fpi) and a commercially available tube specifically developed for condensing halocarbon refrigerants (Huber et al. 1994b). This tube has a sawtooth-shaped outer enhancement. The data indicated that the condensing coefficients for the sawtoothed tube were approximately three times higher than for the conventional finned tube exchanger and two times higher for R-123.

Further, Huber et al. (1994c) found that for tubes with 26 fpi the R-134a condensing coefficients are 20% larger than those for R-12 at a given heat flux. However, on the sawtoothed tube, the R-134a condensing coefficients are nearly two times larger than those for R-12 at the same heat flux. The R-123 condensing coefficients were 10 to 30% larger than the R-11 coefficients at a given heat flux, with the largest differences occurring at the lowest heat fluxes tested. The differences in magnitude between the R-123 and R-11 condensing coefficients were the same for both the 26 fpi tube and the sawtoothed tube.

Physical aspects of a given condenser design (e.g., tube spacing and orientation, shell-side baffle arrangement, orientation of multiple water-pass arrangements, refrigerant connection locations, and number of tubes high in the bundle) affect the refrigerant-side coefficient by influencing vapor distribution and flow through the tube bundle and condensate drainage from the bundle. Butterworth (1977) reviewed correlations accounting for these variables in predicting the heat transfer coefficient for shell-side condensation. These effects are also surveyed by Webb (1984b). Kistler et al. (1976) developed analytical procedures for design within these parameters.

As refrigerant condenses on the tubes, it falls on the tubes in lower rows. Because of the added resistance of this liquid film, the effective film coefficient for lower rows should be lower than that for upper rows. Therefore, the average overall refrigerant film coefficient should decrease as the number of tube rows increases. Webb and Murawski (1990) present row effect data for five tube geometries. However, the additional compensating effects of added film turbulence and direct contact condensation on the subcooled liquid film make actual row effect uncertain.

Huber et al. (1994c, 1994d) determined that the row effect on finned tubes is nearly negligible when condensing low-surface-tension refrigerants such as R-134a. However, the finned-tube film coefficient for higher-surface-tension refrigerants such as R-123 can drop by as much as 20% in lower bundle rows. The row effect for the sawtoothed condensing tube is quite large for both R-134a and R-123, as the film coefficient drops by nearly 80% from top to bottom in a 30-row bundle.

Honda et al. (1994, 1995) demonstrated that row effects caused by condensate drainage and inundation are less for staggered tube bundles than for in-line tubes. In addition, performance improvements as high as 85% were reported for optimized two-dimensional fin profiles compared to conventional fin profiles. The optimized fin profiles differed from the sawtoothed profile tested by Huber et al. (1994b) primarily in that external fins were not notched. This observation coupled, with the differences in row effects between sawtoothed and conventional fin profiles reported by Huber et al. (1994c, 1994d), suggested that there is opportunity for further development and commercialization of two-dimensional fin profiles for large shell-and-tube heat exchanger applications.

Liquid refrigerant may be subcooled by raising the condensate level to submerge a desired number of tubes. The refrigerant film coefficient associated with the submerged tubes is less than the condensing coefficient. If the refrigerant film coefficient in Equation (3) is an average based on all tubes in the condenser, its value decreases as a greater portion of the tubes is submerged.

Tube-Wall Resistance

Most refrigeration condensers, except those using ammonia, have relatively thin-walled copper tubes. Where these are used, the temperature drop or gradient across the tube wall is not significant. If the tube metal has a high thermal resistance, as does 70/30 cupronickel, a considerable temperature drop occurs or, conversely, an increase in the mean temperature difference Δt_m or the surface area is required to transfer the same amount of heat as copper. Although the tube-wall resistance t/k in Equations (3) and (4) is an approximation, as long as the wall thickness is not more than 14% of the tube diameter, the error will be less than 1%. To improve the accuracy of the wall resistance calculation for heavy tube walls or low-conductivity material, see Chapter 3 of the 2005 *ASHRAE Handbook—Fundamentals*.

Surface Efficiency

For a finned tube, a temperature gradient exists from the root of a fin to its tip because of the fin material's thermal resistance. The surface efficiency ϕ_s can be calculated from the fin efficiency, which accounts for this effect. For tubes with low-conductivity material, high fins, or high values of fin pitch, the fin efficiency becomes increasingly significant. Wolverine Tube (1984) and Young and Ward (1957) describe methods to evaluate these effects.

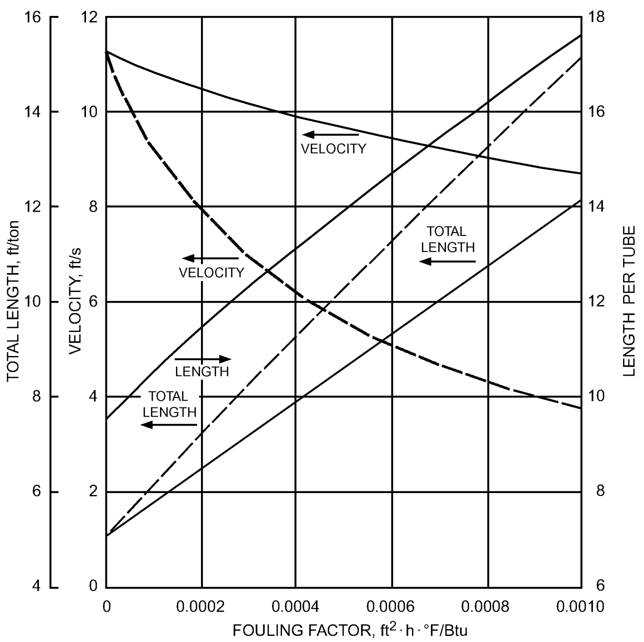
Fouling Factor

Manufacturers' ratings are based on clean equipment with an allowance for possible water-side fouling. The fouling factor r_{fw} is a thermal resistance referenced to the water-side area of the heat transfer surface. Thus, the temperature penalty imposed on the condenser equals the water-side heat flux multiplied by the fouling factor. Increased fouling increases overall heat transfer resistance because of the parameter $(A_o/A_i)r_{fw}$ in Equation (3). Fouling increases the Δt_m required to obtain the same capacity (with a corresponding increase in condenser pressure and system power) or lowers capacity.

Allowance for a given fouling factor has a greater influence on equipment selection than simply increasing the overall resistance (Starner 1976). Increasing the surface area lowers the water velocity. Consequently, the increase in heat transfer surface required for the same performance derives from both fouling resistance and the additional resistance that results from lower water velocity.

For a given tube surface, load, and water temperature range, the tube length can be optimized to give a desired condensing temperature and water-side pressure drop. The solid curves in Figure 2 show the effect on water velocity, tube length, and overall surface required due to increased fouling. A fouling factor of 0.00072 $ft^2 \cdot h \cdot °F/Btu$ doubles the required surface area compared to that with no fouling allowance.

A worse case occurs when an oversized condenser must be selected to meet increased fouling requirements but without the flexibility of increasing tube length. As shown by the dashed lines in Figure 2, water velocity decreases more rapidly as the total surface increases to meet the required performance. Here, the required surface area doubles with a fouling factor of only 0.00049 $ft^2 \cdot h \cdot °F/Btu$. If the application can afford more pumping power, water flow may be increased to obtain a higher velocity, which increases the water film heat transfer coefficient. This factor, plus the lower leaving water temperature, reduces the condensing temperature.



—— OPTIMIZED LENGTH (Δp = 10 psi)

— — — FIXED LENGTH (L = 9.2 ft)

t_{wi} = 85°F t_{wo} = 95°F t_{csat} = 103°F

TWO-PASS, EXTERNAL FIN, PLAIN ID TUBE (0.75 in. OD, 26 fpi)

h_{r_o} = 620 Btu/h · ft^2 · °F

Fig. 2 Effect of Fouling on Condenser

Fouling is a major unresolved problem in heat exchanger design (Taborek et al. 1972). The major uncertainty is which fouling factor to choose for a given application or water condition to obtain expected performance from the condenser: too low a fouling factor wastes compressor power, whereas too high a factor wastes heat exchanger material.

Fouling may result from sediment, biological growth, or corrosive products. Scale results from deposition of chemicals from the cooling water on the warmer surface of the condenser tube. Chapter 48 of the 2007 *ASHRAE Handbook—HVAC Applications* discusses water chemistry and water treatment factors that are important in controlling corrosion and scale in condenser cooling water.

Tables of fouling factors are available; however, in many cases, the values are greater than necessary (TEMA 1999). Extensive research generally found that fouling resistance reaches an asymptotic value with time (Suitor et al. 1976). Much fouling research is based on surface temperatures that are considerably higher than those found in air-conditioning and refrigeration condensers. Coates and Knudsen (1980) and Lee and Knudsen (1979) found that, in the absence of suspended solids or biological fouling, long-term fouling of condenser tubes does not exceed 0.0002 $ft^2 \cdot h \cdot °F/Btu$, and short-term fouling does not exceed 0.0001 $ft^2 \cdot h \cdot °F/Btu$ (ASHRAE 1982). These studies have resulted in a standard industry fouling value of 0.00025 $ft^2 \cdot h \cdot °F/Btu$ for condenser ratings (ARI *Guideline* E-1997). Periodic cleaning of condenser tubes (mechanically or chemically) usually maintains satisfactory performance, except in severe environments. ARI *Standard* 450 for water-cooled condensers should be consulted when reviewing manufacturers' ratings. This standard describes methods to correct ratings for different values of fouling.

WATER PRESSURE DROP

Water (or other fluid) pressure drop is important for designing or selecting condensers. Where a cooling tower cools condensing water, water pressure drop through the condenser is generally limited to about 10 psi. If condenser water comes from another source, pressure drop through the condenser should be lower than the available pressure to allow for pressure fluctuations and additional flow resistance caused by fouling.

Pressure drop through horizontal condensers includes loss through the tubes, tube entrance and exit losses, and losses through the heads or return bends (or both). The effect of tube coiling must be considered in shell-and-coil condensers. Expected pressure drop through tubes can be calculated from a modified Darcy-Weisbach equation:

$$\Delta p = N_p\left(K_H + f\frac{L}{D}\right)\frac{\rho V^2}{2g} \tag{7}$$

where

Δp = pressure drop, lb_f/ft^2
N_p = number of tube passes
K_H = entrance and exit flow resistance and flow reversal coefficient, number of velocity heads ($V^2/2g$)
f = friction factor
L = length of tube, ft
D = inside tube diameter, ft
ρ = fluid density, lb/ft^3
V = fluid velocity, fps
g = gravitational constant = 32.17 $lb_m \cdot ft/(lb_f \cdot s^2)$

For tubes with smooth inside diameters, the friction factor may be determined from a Moody chart or various relations, depending on the flow regime and wall roughness (see Chapter 2 of the 2005 *ASHRAE Handbook—Fundamentals*). For tubes with internal enhancement, the friction factor should be obtained from the tube manufacturer.

The value of K_H depends on tube entry and exit conditions and the flow path between passes. A minimum recommended value is 1.5. This factor is more critical with short tubes.

Predicting pressure drop for shell-and-coil condensers is more difficult than for shell-and-tube condensers because of the curvature of the coil and flattening or kinking of the tubes as they are bent. Seban and McLaughlin (1963) discuss the effect of curvature or bending of pipe and tubes on the pressure drop.

For brazed and plate-and-frame condensers, the total pressure drop is the sum of the pressure drop in the plate region and the pressure drops associated with the entry/exit ports. The general form of equation is similar to Equation (7) except $N_p = 1$.

LIQUID SUBCOOLING

The amount of condensate subcooling provided by the condensing surface in a shell-and-tube condenser is small, generally less than 2°F. When a specific amount of subcooling is required, it may be obtained by submerging tubes in the condensate. Tubes in the lower portion of the bundle are used for this purpose. If the condenser is multipass, then the subcooling tubes should be included in the first pass to gain exposure to the coolest water.

When means are provided to submerge the subcooler tubes to a desired level in the condensate, heat is transferred principally by natural convection. Subcooling performance can be improved by enclosing tubes in a separate compartment in the condenser to obtain the benefits of forced convection over the enclosed tubes.

Segmental baffles may be provided to produce flow across the tube bundle. Kern and Kraus (1972) describe how heat transfer performance can be estimated analytically by use of longitudinal or cross-flow correlations, but it is more easily determined by test because of the large number of variables. The refrigerant pressure drop along the flow path should not exceed the pressure difference permitted by the saturation pressure of the subcooled liquid.

CIRCUITING

Varying water flow rate in a condenser can significantly affect the saturated condensing temperature, which affects performance. Figure 3 shows the change in condensing temperature for one, two, or three passes in a particular condenser. For example, at a loading of 22,000 Btu/h per tube, a two-pass condenser with a 10°F range would have a condensing temperature of 102.3°F. At the same load with a one-pass, 5°F range, this unit would have a condensing temperature of 99.3°F. The one-pass option does, however, require twice the water flow rate with an associated increase in pumping power. Three-pass design may be favorable when costs associated with water flow outweigh gains from a lower condensing temperature. Hence, different numbers of passes (if an option) and ranges should be considered against other parameters (water source, pumping power, cooling tower design, etc.) to optimize overall performance and cost.

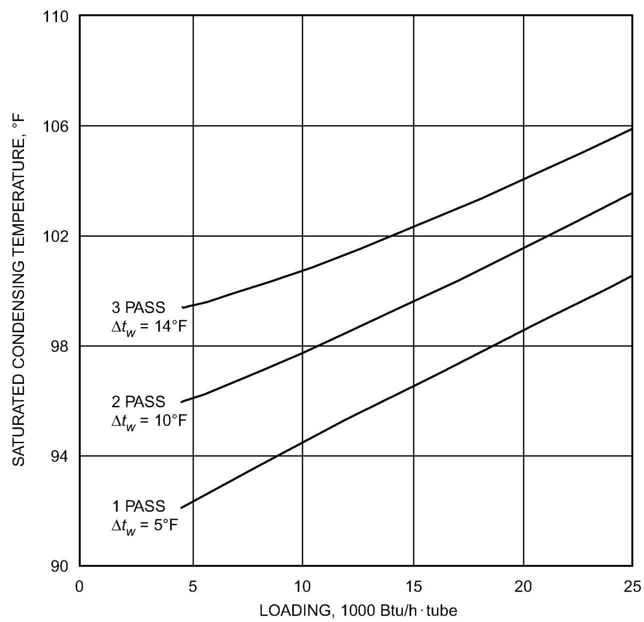


Fig. 3 Effect of Condenser Circuiting

CONDENSER TYPES

The most common types of water-cooled refrigerant condensers are (1) shell-and-tube, (2) shell-and-coil, (3) tube-in-tube, and (4) brazed-plate. The type selected depends on the cooling load, refrigerant used, quality and temperature of the available cooling water, amount of water that can be circulated, location and space allotment, required operating pressures (water and refrigerant sides), and cost and maintenance concerns.

Shell-and-Tube Condensers

Built in sizes from 1 to 10,000 tons, these condensers condense refrigerant outside the tubes and circulate cooling water through the tubes in a single or multipass circuit. Fixed-tube-sheet, straight-tube construction is usually used, although U-tubes that terminate in a single tube sheet are sometimes used. Typically, shell-and-tube condenser tubes run horizontally. Where floor installation area is limited, the tubes may be vertical, but this orientation has poor condensate draining, which reduces the refrigerant film coefficient. Vertical condensers with open water systems have been used with ammonia.

Gas inlet and liquid outlet nozzles should be located with care. The proximity of these nozzles may adversely affect condenser performance by requiring excessive amounts of liquid refrigerant to seal the outlet nozzle from inlet gas flow. This effect can be diminished by adding baffles at the inlet and/or outlet connection.

Halocarbon refrigerant condensers have been made with many materials, including all prime surface or finned, ferrous, or nonferrous tubes. Common tubes are nominal 0.75 and 1.0 in. OD copper tubes with integral fins on the outside. These tubes are often available with fin heights from 0.035 to 0.061 in. and fin spacings of 19, 26, and 40 fins/in. For ammonia condensers, prime surface steel tubes, 1.25 in. OD and 0.095 in. average wall thickness, are common.

Many tubes designed for enhanced heat transfer are available (Bergles 1995). On the inside of the tube, common enhancements include longitudinal or spiral grooves and ridges, internal fins, and other devices to promote turbulence and augment heat transfer. On the refrigerant side, condensate surface tension and drainage are important in design of the tube outer surface. Tubes are available with the outsides machined or formed specifically to enhance condensation and promote drainage. Heat transfer design equations should be obtained from the manufacturer.

The electrohydrodynamics (EHD) technique couples a high-voltage, low-current electric field with the flow field in a fluid with low electrical conductivity to achieve higher heat transfer coefficients. Ohadi et al. (1995) experimentally demonstrated enhancement factors in excess of tenfold for both boiling and condensation of refrigerants such as R-134a. For condensation, the technique responds equally well to augmentation of condensation over (or inside) both vertical and horizontal tubes. Most passive augmentation techniques perform poorly for condensation enhancement in vertical orientation. Additional details of the EHD technique are in Chapter 3 of the 2005 *ASHRAE Handbook—Fundamentals*.

Because water and refrigerant film resistances with enhanced tubes are reduced, the effect of fouling becomes relatively great. Where high levels of fouling occur, fouling may easily account for over 50% of the total resistance. In such cases, the advantages of enhancement may diminish. On the other hand, water-side augmentation, which creates turbulence, may reduce fouling. The actual value of fouling resistances depends on the particular type of enhancement and service conditions (Starner 1976; Watkinson et al. 1974).

Similarly, refrigerant-side enhancements may not show as much benefit in very large tube bundles as in smaller bundles. This is because of the row effect addressed in the section on Refrigerant-Side Film Coefficient.

The tubes are either brazed into thin copper, copper alloy, or steel tube sheets, or rolled into heavier nonferrous or steel tube sheets. Straight tubes with a maximum OD less than the tube hole diameter and rolled into tube sheets are removable. This construction facilitates field repair in the event of tube failure.

The required heat transfer area for a shell-and-tube condenser can be found by solving Equations (1), (2), and (3). The mean temperature difference is the logarithmic mean temperature difference, with entering and leaving refrigerant temperatures taken as the saturated condensing temperature. Depending on the parameters fixed, an iterative solution may be required.

Shell-and-U-tube condenser design principles are the same as those outlined for horizontal shell-and-tube units, with one exception: water pressure drop through the U-bend of the U-tube is generally less than that through the compartments in the water header where the direction of water flow is reversed. The pressure loss is a function of the inside tube diameter and the ratio of the inside tube diameter to bending centers. Pressure loss should be determined by test.

Shell-and-Coil Condensers

Shell-and-coil condensers circulate cooling water through one or more continuous or assembled coils contained within the shell. The refrigerant condenses outside the tubes. Capacities range from 0.5 to 15 tons. Because of the type of construction, the tubes are neither replaceable nor mechanically cleanable.

Again, Equations (1), (2), and (3) may be used for performance calculations, with the saturated condensing temperature used for the entering and leaving refrigerant temperatures in the logarithmic mean temperature difference. The values of h_w (the water-side film coefficient) and, especially, the pressure loss on the water side require close attention; laminar flow can exist at considerably higher Reynolds numbers in coils than in straight tubes. Because the film coefficient for turbulent flow is greater than that for laminar flow, h_w as calculated from Equation (6) will be too high if the flow is not turbulent. Once flow has become turbulent, the film coefficient will be greater than that for a straight tube (Eckert 1963). Pressure drop through helical coils can be much greater than through smooth, straight tubes for the same length of travel. The section on Water Pressure Drop outlines the variables that make accurate determination of the pressure loss difficult. Pressure loss and heat transfer rate should be determined by test because of the many variables inherent in this condenser.

Tube-in-Tube Condensers

These condensers consist of one or more assemblies of two tubes, one within the other, in which the refrigerant vapor is condensed in either the annular space or the inner tube. These units are built in sizes ranging from 0.3 to 50 tons. Both straight-tube and axial-tube (coaxial) condensers are available.

Equations (1) and (2) can be used to size a tube-in-tube condenser. Because the refrigerant may undergo a significant pressure loss through its flow path, the refrigerant temperatures used to calculate the mean temperature difference should be selected carefully. Refrigerant temperatures should be consistent with the model used for the refrigerant film coefficient. The logarithmic mean temperature difference for either counterflow or parallel flow should be used, depending on the piping connections. Equation (3) can be used to find the overall heat transfer coefficient when water flows in the tubes, and Equation (4) may be used when water flows in the annulus.

Tube-in-tube condenser design differs from those outlined previously, depending on whether the water flows through the inner tube or through the annulus. Condensing coefficients are more difficult to predict when condensation occurs within a tube or annulus, because the process differs considerably from condensation on the outside of a horizontal tube. Where water flows through the annulus, disagreement exists regarding the appropriate method that should be used to calculate the waterside film coefficient and the water pressure drop. The problem is further complicated if the tubes are spiraled.

The water side is mechanically cleanable only when the water flows in straight tubes and cleanout access is provided. Tubes are not replaceable.

Brazed-Plate and Plate-and-Frame Condensers

Brazed-plate condensers are constructed of plates brazed together to form an assembly of separate channels. Capacities range from 0.5 to 100 tons. The plate-and-frame condenser is a standard design in which plate pairs are laser-welded to form a single cassette. Refrigerant is confined to the space between the welded plates and is exposed to gaskets only at the ports. Such condensers have a higher range of capacity.

The plates, typically stainless steel, are usually configured with a wave pattern, which results in high turbulence and low susceptibility to fouling. The design has some ability to withstand freezing and, because of the compact design, requires a low refrigerant charge. The construction of brazed units does not allow mechanical cleaning, and internal leaks usually cannot be repaired. Thus, it can be beneficial to use filtration or separators on open cooling towers to keep the water clean, or to use closed-circuit cooling towers. Plate-and-frame units can be cleaned on the water side.

Performance calculations are similar to those for other condensers; however, very few correlations are available for the heat transfer coefficients.

NONCONDENSABLE GASES

When first assembled, most refrigeration systems contain gases, usually air and water vapor. These gases are detrimental to condenser performance, so it is important to evacuate the entire refrigeration system before operation.

For low-pressure refrigerants, where the operating pressure of the evaporator is less than ambient pressure, even slight leaks can be a continuing source of noncondensables. In such cases, a purge system, which automatically expels noncondensable gases, is recommended. Figure 4 shows some examples of refrigerant loss associated with using purging devices at various operating conditions. As a general rule, purging devices should emit less than one part refrigerant per part of air as rated in accordance with ARI *Standard* 580 (see ASHRAE *Guideline* 3).

When present, noncondensable gases collect on the high-pressure side of the system and raise the condensing pressure above that corresponding to the temperature at which the refrigerant is

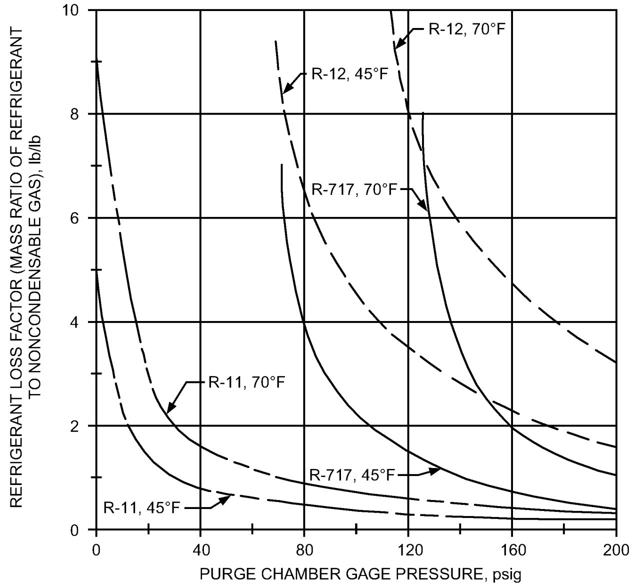


Fig. 4 Loss of Refrigerant During Purging at Various Gas Temperatures and Pressures

actually condensing. This increases power consumption and reduces capacity. Also, if oxygen is present at a point of high discharge temperature, the oil may oxidize.

The excess pressure is caused by the partial pressure of the noncondensable gas. These gases form a resistance film over some of the condensing surface, thus lowering the heat transfer coefficient. Webb et al. (1980) showed how a small percentage of noncondensables can cause major decreases in the refrigerant film coefficient in shell-and-tube condensers. (See Chapter 4 of the 2005 *ASHRAE Handbook—Fundamentals*.) The noncondensable situation of a given condenser is difficult to characterize because such gases tend to accumulate in the coldest and least agitated part of the condenser or in the receiver. Thus, a fairly high percentage of noncondensables can be tolerated if the gases are confined to areas far from the heat transfer surface. One way to account for noncondensables is to treat them as a refrigerant or gas-side fouling resistance in Equation (3). Some predictions are presented by Wanniarachchi and Webb (1982).

As an example of the effect on system performance, experiments performed on a 250 ton R-11 chiller condenser revealed that 2% noncondensables by volume caused a 15% reduction in the condensing coefficient. Also, 3% and 8% noncondensables by volume caused power increases of 2.6% and 5%, respectively.

Huber et al. (1994b) determined that noncondensable gases have a more severe effect on the condensing coefficient of tubes with sawtoothed enhancements than on conventional finned tubes. For a noncondensable gas concentration of 0.5%, the coefficient for the tube with a fin spacing of 26 fpi dropped by 15%, whereas the coefficient for the sawtoothed tube dropped by nearly 40%. At a noncondensable concentration of 5%, degradation was similar for both tubes.

The presence of noncondensable gases can be tested by shutting down the refrigeration system while allowing the condenser water to flow long enough for the refrigerant to reach the same temperature as the water. If the condenser pressure is higher than the pressure corresponding to the refrigerant temperature, noncondensable gases are present. This test may not be sensitive enough to detect the presence of small amounts of noncondensables, which can, nevertheless, decrease shell-side condensing coefficients.

CODES AND STANDARDS

Pressure vessels must be constructed and tested under the rules of appropriate codes and standards. The introduction of the current ASME *Boiler and Pressure Vessel Code*, Section VIII, gives guidance on rules and exemptions.

The more common applicable codes and standards include the following:

- ARI *Standard* 450 covers industry criteria for standard equipment, standard safety provisions, marking, fouling factors, and recommended rating points for water-cooled condensers.
- ASHRAE *Standard* 22 covers recommended testing methods.
- ARI *Standard* 580 covers methods of testing, evaluating, and rating the efficiency of noncondensate gas purge equipment.
- ASHRAE *Standard* 15 specifies design criteria, use of materials, and testing. It refers to the ASME *Boiler and Pressure Vessel Code*, Section VIII, for refrigerant-containing sides of pressure vessels, where applicable. Factory test pressures are specified, and minimum design working pressures are given by this code. This code requires pressure-limiting and pressure-relief devices on refrigerant-containing systems, as applicable, and defines the setting and capacity requirements for these devices.
- ASME *Boiler and Pressure Vessel Code,* Section VIII covers the safety aspects of design and construction. Most states require condensers to meet these requirements if they fall within the scope of the ASME code. Some of the exceptions from meeting the requirements listed in the ASME code are as follows:
  - Condenser shell ID is 6 in. or less.
  - Working pressure is 15 psig or less.
  - The fluid (water) portion of the condenser need not be built to the requirements of the ASME code if the fluid is water or an aqueous glycol solution, the design pressure does not exceed 300 psig, and the design temperature does not exceed 210°F.

  Condensers meeting the requirements of the ASME code will have an ASME stamp. The ASME stamp is a U or UM inside a four-leaf clover. The U can be used for all condensers; the UM can be used (considering local codes) for those with net refrigerant-side volume less than 1.5 ft^3 if less than 600 psig, or less than 5 ft^3 if less than 250 psig.
- UL *Standard* 207 covers specific design criteria, use of materials, testing, and initial approval by Underwriters Laboratories. A condenser with the ASME U stamp does not require UL approval.

Design Pressure

Refrigerant-side pressure should be chosen per ASHRAE *Standard* 15. Standby temperature and temperatures encountered during shipping of units with a refrigerant charge should also be considered.

Required fluid- (water-) side pressure varies, depending largely on the following conditions: static head, pump head, transients due to pump start-up, and valve closing. A common water-side design pressure is 150 psig, although with taller building construction, requirements for 300 psig are not uncommon.

OPERATION AND MAINTENANCE

When a water-cooled condenser is selected, anticipated operating conditions, including water and refrigerant temperatures, have usually been determined. Standard practice allows for a fouling factor in the selection procedure. A new condenser, therefore, operates at a condensing temperature lower than the design point because it has not yet fouled. Once a condenser starts to foul or scale, economic considerations determine how frequently it should be cleaned. As scale builds up, the condensing temperature and subsequent power increase while the unit capacity decreases. This effect can be seen in Figure 5 for a condenser with a design fouling factor of 0.00025 $ft^2 \cdot h \cdot °F/Btu$.

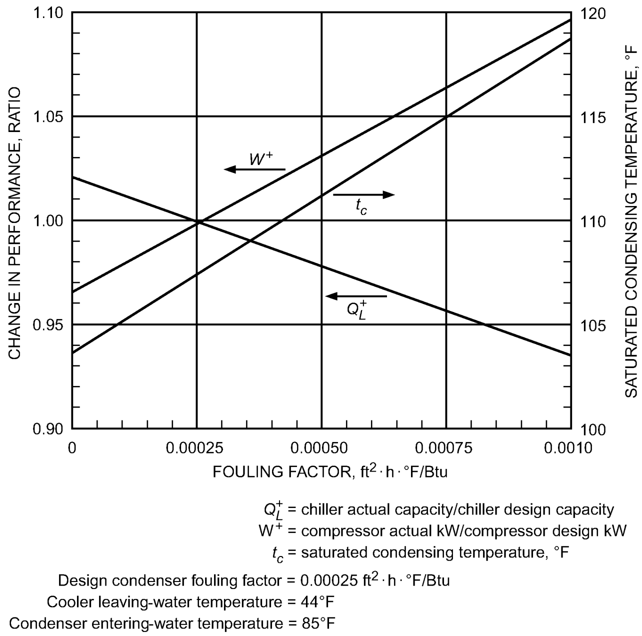


Fig. 5 Effect of Fouling on Chiller Performance

At some point, the increased cost of power can be offset by the labor cost of cleaning.

Local water conditions, as well as the effectiveness of chemical water treatment, if used, make the use of any specific maintenance schedule difficult. Cleaning can be done either mechanically with a brush or chemically with an acid solution. In applications where water-side fouling may be severe, online cleaning can be accomplished by brushes installed in cages in the water heads. By using valves, flow is reversed at set intervals, propelling the brushes through the tubes (Kragh 1975). The most effective method depends on the type of scale formed. Expert advice in selecting the particular method of tube cleaning is advisable.

Occasionally, one or more tubes may develop leaks because of corrosive impurities in the water or through improper cleaning. These leaks must be found and repaired as soon as possible; this is normally done by replacing the leaky tubes, a procedure best done through the original condenser manufacturer. In large condensers, where the contribution of a single tube is relatively insignificant, a simpler approach may be to seal the ends of the leaking tube.

If the condenser is located where water can freeze during the winter, special precautions should be taken when it is idle. Opening all vents and drains may be sufficient, but water heads should be removed and tubes blown free of water.

If refrigerant vapor is to be released from the condenser and there is water in the tubes, the pumps should be on and the water flowing. Otherwise, freezing can occur.

The condenser manufacturer's installation recommendations on orientation, piping connections, space requirements for tube cleaning or removal, and other important factors should be followed.

AIR-COOLED CONDENSERS

An air-cooled condenser uses ambient air to remove the heat of condensation from the refrigerant in a compression-type refrigeration system. Individual air-cooled condensers range in size from a few hundred Btu/h to about 100 tons, though individual units may be coupled together for larger systems. The smallest air-cooled condensers are used in residential refrigerators and may have no fans, relying only on gravity circulation of the ambient air. Larger condensers almost always use one or more motor-driven fans for air circulation. Midsized condensers are frequently provided as part of an integral package (called a condensing unit) with the compressor. All but the smallest condensers use finned coils, which may be formed or otherwise bent around a compressor/fan combination to form outdoor residential condensing units up to about 5 tons. Larger air-cooled condensers almost always use coils with a rectangular profile, either flat or with a single corner bent shape, and a minimum number of rows of finned tubing. Fans may be positioned to either draw or blow the air though the condenser coil. Fan motors are generally connected to operate when the compressor operates.

Air-cooled condensers may be located either adjacent to or remote from the compressor and may be designed for indoor or outdoor operation. For indoor use, larger condensers use centrifugal fans for ducted discharge to the outdoors. Outdoor coil orientation is generally horizontal, with propeller fans above the coil providing draw-through air circulation. For nonstandard applications, air discharge may be either vertical (top) or horizontal (side). Interconnecting piping connects the condenser coil to the compressor and to the expansion device or a liquid receiver.

TYPES

Air-cooled condensers may vary in air- and refrigerant-side design, but there are three main coil construction types: plate-and-fin, integral-fin, and microchannel.

Plate-and-Fin

Coils are commonly constructed of copper, aluminum, or steel tubes, ranging from 0.25 to 0.75 in. in diameter. For fluorocarbon and hydrocarbon refrigerants, copper is most common. Aluminum or steel tube condensers are generally used for ammonia. Fins attached to the tubes can be aluminum or copper.

Copper tubes generally have a round profile, with fins perpendicularly bound to their exterior. This arrangement normally requires no further air-side protection but may be prone to corrode when subject to some industrial or furnace flue gases, or salt-atmosphere-induced oxidation. Some inherent corrosion protection is provided by the slight galvanic effect of aluminum fins. Aluminum round-tube coil manufacturing is similar to that of round copper tube coils, except gas shield arc welding or special brazing alloys are used for joints. Corrosion protection is necessary for aluminum-to-copper joints. Steel tube condensers with steel fins are painted for indoor use or hot-dip galvanized for outdoor applications. Brazed-steel condensers are very common on small refrigeration units.

Tube diameter selection compromises between factors such as compactness, manufacturing tooling cost, header arrangement, air resistance, and refrigerant flow resistance. A smaller diameter gives more flexibility in coil circuit design and a lower refrigerant charge, but at increased cost.

Other core tubing choices include copper tube that is internally enhanced by fins or grooves known as microfin tubing. The helical microfinning is very small, on the order of 0.008 in. height in a tiny spiral with an optimum helical angle of 15 to 20° throughout the internal surface of the tube. Over 60 such grooved fins could fit in a 3/8 in. tube's cross section. This pattern is common on seamless (drawn) tubes; on welded (strip) tubes, internal grooving can be augmented with a double cross-hatch fin pattern. Although a microfin tube has a significant refrigerant-side heat transfer advantage, tube-side effect does not have dominant heat transfer conductance in the coil's overall heat transfer function. The majority of heat transfer resistance is on the air side, primarily because of an extended fin surface. Usually, an air-cooled condenser's fin-to-tube b ratio is around 25 to 1. Therefore, a 50% increase in tube-side heat transfer by microfinning may result in only a 5 to 15% overall heat transfer coefficient gain in the coil design, depending on refrigerant type, its transport velocities, and the fin surface enhancement used. Occasionally, a bare-tube condenser (i.e., without a secondary surface) is used where airborne dirt loading is expected to be excessive. For these applications, enhanced bare-tube coils could be half the size of regular bare-tube coils.

When conditions are normal, and even in a saline atmosphere, a copper tube aluminum-finned coil is generally used. The most common is the aluminum plate fin, from 0.012 to 0.006 in. thick. Most plate fins have extruded tube collars. Straight tubes are passed through these nested fin collars and expanded to fit the collars by either mechanical or, less frequently, hydraulic expansion. This results in a rigid coil assembly that has maximum tube-to-fin thermal conductance. Spiral- and spline-fin coils, tightly wound onto the individual tubes that make up the coil core, are also used. Common fin spacing for each type ranges from 8 to 20 fpi.

Stock fin coatings, as well as entire (completed) coil dip-process coatings, are readily available from specialty chemical processors. These coatings are for use in specific corrosive atmospheres, especially in a dry-surface coil application, such as aluminum-finned air-cooled condenser coils in seacoast environments

Integral-Fin

Integral-fin condensers can be made of either copper of aluminum. These condensers are made by extruding or forming fins directly from the tube material. Copper tubing can be formed using cold compression to form large fins on the outside of the tubing. Where the internal heat transfer coefficient is important, these formed tubes can be stacked to form condensers. Using aluminum, the common method of forming condensers is to rake the surface of the tubing to form protrusions. Because aluminum is a very soft material, these tubes with the “spiny” fins can be coiled to form complete condensers.

Microchannel

Small all-aluminum condensers, used in automotive and aviation applications where lightness and compactness are paramount, are made with flattened tubes having hydraulic radii from 0.01 to 0.12 in. These oval tubes are formed into serpentines with zigzag aluminum fins nested horizontally between tubing runs. New, improved versions of these heat exchangers feature individual oval tubes connected to manifold header assemblies. The entire assembly is furnace-brazed and a diffusion layer of zinc applied for corrosion protection. This type of flat tube and horizontal fin arrangement has evolved into **microchannel coils**, which are used in residential condensing units and commercial air-conditioning systems. Although an advancement, they have some limitations that are uncommon in conventional coils.

Functioning solely as a condenser, microchannel coils have a higher heat transfer efficiency than a conventional plate or spine-fin coil, both per unit volume and per unit surface area. Manufacturers' laboratory tests have shown microchannel coils, compared to a conventional 12 fpi, flat-fin, 3/8 in. copper tube core, to be 90% higher in heat transfer coefficient for similar face areas. Compared to an 18-spine fin spiral design, 3/8 in. aluminum tube, this brazed aluminum design appears to be 44% higher in heat transfer coefficient. In addition to lower air-side pressure drop, microchannel refrigerant-side pressure drop tends to be lower, depending on circuitry. However, equal refrigerant distribution in the inlet header is less favorable than with conventional coils. Another use restriction is in the heat pump reverse cycle: horizontal-tube and horizontal-fin construction gives microchannel coils a less than desirable high defrost (condensate) water retention ability. Versatility is thus restricted to a specific unit's size and design, and not one-fits-many as for conventional coil designs. The equipment designer should consider how to avoid such deficits, including field repair not being an option, when applying microchannel coils at the system level.

FANS AND AIR REQUIREMENTS

Condenser coils can be cooled by natural convection or wind or by propeller, centrifugal, and vaneaxial fans. Because efficiency increases sharply by increasing air speed across the coil, forced convection with fans predominates.

Unless unusual operating or application conditions exist, fan/motor selections are made by balancing operating and first costs with size and sound requirements. Common air quantities are 600 to 1200 cfm/ton [14,400 Btu/h at 30°F temperature difference (TD)] at 400 to 800 fpm. Fan power requirements generally range from 0.1 to 0.2 hp/ton.

The type of fan depends primarily on static pressure and unit shape requirements. Propeller fans are well suited to units with a low static-pressure drop and free air discharge. Fan blade speeds are selected in the range of 515 to 1750 rpm (1140 rpm is common). Direct-drive propeller fans up to 36 in. in diameter are used in single and multiple assemblies in virtually all sizes of condenser units. Because of the propeller fan's low starting torque requirements, the most common is a permanent split-capacitor (PSC) fractional horsepower motor, having internal inherent protection. These motors are most commonly used up to about 3/4 hp, with shaded-pole motors used in the smallest sizes. Larger motors are generally polyphase induction or synchronous for the lowest-speed fans. In outdoor upward-blow applications, special attention must be paid to motor thrust bearing selection and to weather protection. When the fan blade is positioned above a vertically installed motor, a shaft-mounted slinger disk is sometimes supplied to further protect the motor and its bearings. In industrial applications, totally enclosed weather-resistant motors are sometimes used.

For free air discharge, direct-drive propeller fans are more efficient than centrifugal fans. A belt drive arrangement on propeller and centrifugal fan(s) offers more flexibility but usually requires greater maintenance, and is used most frequently on centrifugal and propeller fans over 36 in. in diameter. Centrifugal fans are used in noise-sensitive applications and/or where significant external air resistance is expected, as with extended ductwork. Vaneaxial fans are often more efficient than centrifugal fans and are frequently used in the largest sizes, although special provision for the weight of the fans, motors, and belt drive must be made. A partially obstructed fan inlet or outlet can drastically increase the noise level. Support brackets close to the fan inlet or partially obstructing the fan outlet can noticeably effect fan noise and efficiency.

HEAT TRANSFER AND PRESSURE DROP

In condensers, heat is transferred by (1) desuperheating, (2) condensing, and (3) subcooling. Figure 6 shows the changes of state of R-134a passing through an air-cooled condenser coil and the corresponding temperature change of the cooling air as it passes through the coil. Desuperheating and subcooling zones vary from 5 to 10% of the total heat transfer, depending on the entering gas and leaving liquid temperatures, but Figure 6 is typical. Good design usually has full condensing occurring in approximately 85% of the condenser area, based on an azeotropic refrigerant. Generally this happens at a fairly constant temperature, though some drop in saturated condensing temperature through the condenser coil can occur when there is significant pressure drop in the condensing coil.

Chapter 3 of the 2005 *ASHRAE Handbook—Fundamentals* lists equations for heat transfer coefficients in the single-phase flow sections. For condensation from saturated vapors, one of the best calculation methods is the Shah correlation (Shah 1979, 1981). Also, Ghaderi et al. (1995) reviewed available correlations for condensation heat transfer in smooth and enhanced tubes.

The overall heat transfer coefficient of an air-cooled condenser can be expressed by Equation (5c) in Chapter 22. It is suggested that direct use of experimental testing for extracted coinciding thermal flow resistance data is preferable to the completely theoretical approach.

For optimum vapor desuperheating and liquid subcooling (i.e., inlet and outlet regions of the coil) a cross-counterflow arrangement of refrigerant and air produces the best performance. In the condensing portion (i.e., halfway through the coil core), the latent heat dissipates and the change of state occurs, with a concurrent refrigerant velocity drop. Because of the change of state, the velocity and volume of the refrigerant drop substantially as it becomes a liquid. In general, most air-cooled condenser designs provide an adequate heat transfer surface area and complementary pressure drop to cause the refrigerant to subcool an average of 2 to 6°F before it leaves the condenser coil. The subcooling section may be integrated into the main coil by routing the liquid outlet tubes through coil rows on the air inlet side of the coil. Some manufacturers use "tripod" circuitry, where two or more parallel tubes of he condensing portion are joined to form a single continuing circuit. Overall capacity increases about 0.5% per 1°F subcooling at the same suction and discharge pressure. To ensure proper operation, liquid quality needs to be 100% at the entrance to the evaporator's inlet flow control device.

Proper condenser design also involves calculating pressure drops for the gas and liquid flow areas in the coil. Chapter 4 of the 2005 *ASHRAE Handbook—Fundamentals* describes an estimation of two-phase pressure drop. The overall pressure drop in a typical condenser ranges from 1 to 4°F equivalent saturated temperature change.

CONDENSERS REMOTE FROM COMPRESSOR

Remote air-cooled condensers are used for refrigeration and air conditioning from 0.5 to over 500 tons. Larger systems use multiple condensers. Larger air-cooled condensers with multiple individual circuits are used when the installation has an array of independent refrigeration systems (e.g., supermarkets). The most common coil-fan arrangements are horizontal coils with upflow air (top discharge), and vertical coil with horizontal airflow (front discharge). The choice of design depends primarily on the intended application and the surroundings.

Refrigerant piping and electrical wiring are the interconnections between the compressor and remote condenser unit. Receivers are most often installed close to the compressor and not the condenser. When selecting installation points for any high-side equipment, major concerns include direct sunlight (high solar load), summer and winter conditions, prevailing wind, elevation differences, and length of piping runs. Recirculated condenser air is a common cause of excessive air temperatures entering the condenser and usually results from a poor choice in locating the condenser, and/or badly accommodating its airflow pattern.

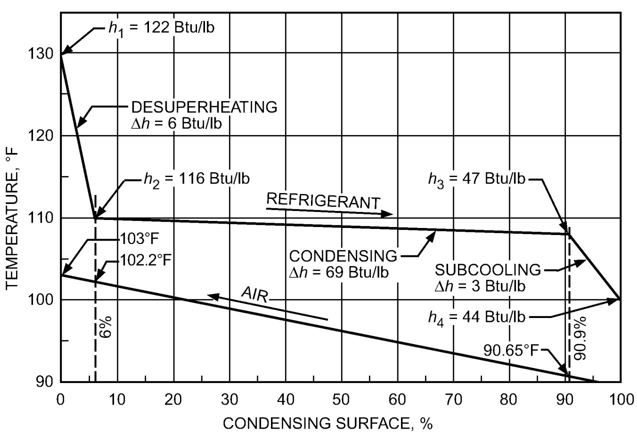


Fig. 6 Temperature and Enthalpy Changes in Air-Cooled Condenser with R-134a

CONDENSERS AS PART OF CONDENSING UNIT

When the condenser coil is included with the compressor as a packaged assembly, it is called a condensing unit. Condenser units are further categorized as indoor or outdoor units, depending on rain protection, electrical controls, and code approvals. Factory-assembled condensing units consist of one or more compressors, a condenser coil, electrical controls in a panel, a receiver, shutoff valves, switches, and sometimes other related units. Precharged interconnecting refrigerant lines to and from the evaporator may be offered as a kit with smaller units. Open, semihermetic or full-hermetic compressors are used for single or parallel (multiplexing) piping connections, in which case an accumulator, oil separator, and its crankcase oil return is likely to be included in the package. Indoor units lack weather enclosures, allowing easy access for service to all components; outdoor unit access is obtained by removing cabinet panels or opening hinged covers.

Noise is a major concern in both the design and installation of condensing units, mainly because of the compressor. The following can be done to reduce unwanted sound:

- Avoid a straight-line path from the compressor to the listener.
- Use acoustical material inside the cabinet and to envelop the compressor. Streamline air passages as much as possible. Use top-mounted, vertical air discharge.
- Cushion or suspend the compressor on a suitable base.
- Use a lighter-gage base rather than a heavier one to minimize vibration transmission to other panels.
- Where possible, ensure that natural frequencies of panels and refrigerant lines are different from basic compressor and fan frequencies.
- Avoid refrigerant lines with many bends, which are more likely to produce numerous pulsation forces and natural frequencies that cause audible sounds.
- Be aware that a condenser with a very low refrigerant pressure drop may have one or two passes resonant with the compressor discharge pulsations.
- Use wire basket fan assembly supports to help isolate noise and vibration.
- Fan selection is important; steeply pitched propeller blades and unstable centrifugal fans can be serious noise producers.
- Fan noise is more objectionable at night. Because air temperatures are generally lower at night, fan speed can usually be reduced then to reduce noise levels without negatively affecting performance.
- Design to avoid fan and compressor short-cycling.

See Chapter 47 of the 2007 *ASHRAE Handbook—HVAC Applications* for more information on sound and vibration control.

WATER-COOLED VERSUS AIR-COOLED CONDENSING

Where small units (less than 3 hp) are used and abundant low-cost water is available, both first and operating costs may be lower for water-cooled equipment. Air-cooled equipment generally requires up to a 20% larger compressor and/or longer run time, thus increasing costs. When comparing systems, operation and maintenance of the complete system as well as local conditions need to be considered. Chronic water shortages can affect selection. When cooling towers are used to produce condenser cooling water, the initial cost of water cooling may be higher. Also, the lower operating cost of water cooling may be offset by cooling tower pump, and maintenance costs. Water-cooled condensers are discussed in detail at the beginning of this chapter.

TESTING AND RATING

Condenser and condensing unit manufacturers specify, select, and test motor and fan combinations as complete assemblies for specific published capacities. Test measurements of capacity should be made in accord with ASHRAE *Standard* 20. Other common tests by manufacturers include motor winding and bearing temperature rise during highest ambient operating conditions, duty cycling at lowest ambient, and mounting positions and noise amplitude. To ensure user safety, most condenser coil designs are submitted for listing by Underwriters Laboratories.

Condensers are rated in terms of total heat rejection (THR), which is the total heat removed in desuperheating, condensing, and subcooling the refrigerant. This value is the product of the mass flow of the refrigerant and the difference in enthalpies of the refrigerant vapor entering and the liquid leaving the condenser coil.

A condenser may also be rated in terms of net refrigeration effect (NRE) or net heat rejection (NHR), which is the total heat rejection less the heat of compression added to the refrigerant in the compressor. This is the typical expression of a refrigeration system's capacity.

For open compressors, the THR is the sum of the actual power input to the compressor and the NRE. For hermetic compressors, the THR is obtained by adding the NRE to the total motor power input and subtracting any heat losses from the compressor and discharge line surfaces. Surface heat losses are generally 0 to 10% of the power consumed by the motor. All quantities must be expressed in consistent units. Table 1 recommends factors for converting condenser THR ratings to NRE for both open and hermetic reciprocating compressors.

Air-cooled condenser ratings are based on the temperature difference (TD) between the dry-bulb temperature of air entering the coil and the saturated condensing temperature (which corresponds to the refrigerant pressure at the condenser outlet). Typical TD values are 10 to 15°F for low-temperature systems at a –20 to –40°F evaporator temperature, 15 to 20°F for medium-temperature systems at a 20°F evaporator temperature, and 25 to 30°F for air-conditioning systems at a 45°F evaporator temperature. The THR capacity of the condenser is considered proportional to the TD. That is, the capacity at 30°F TD is considered to be double the capacity for the same condenser selected for 15°F TD.

The 1% design day condition is suggested for condenser selection.

The specific design dry-bulb temperature must be selected carefully, especially for refrigeration serving process cooling. An entering air temperature that is higher than expected quickly causes compressor discharge pressure and power to exceed design. This can cause an unexpected shutdown, usually at a time when it can least be tolerated. Also, congested or unusual locations may create entering air temperatures higher than general ambient conditions. Recirculated condenser air is usually the result of a poor choice in locating the installation.

The capacity of an air-cooled condenser equipped with an integral subcooling circuit varies depending on the refrigerant charge. The charge is greater when the subcooling circuit is full of liquid, which increases subcooling. When the subcooling circuit is used for condensing, the refrigerant charge is lower, condensing capacity is greater, and liquid subcooling is reduced. Laboratory testing should be in accordance with ASHRAE *Standard* 20, which requires liquid leaving the condenser coil (under test). Currently no industry certification program exists for air-cooled condensers.

CONTROL OF AIR-COOLED CONDENSERS

For a refrigeration system to function properly, the condensing pressure and temperature must be maintained within certain limits. An increase in condensing temperature causes a loss in capacity, requires extra power, and may overload the compressor motor. A condensing pressure that is too low hinders flow through conventional liquid feed devices. This hindrance starves the evaporator and causes a loss of capacity, unbalances the distribution of refrigerant in the evaporator, possibly causes strips of ice to form across the face of a freezer coil, and trips off the unit on low pressure.

Some systems use low-pressure-drop thermostatic expansion valves (balance port TXVs). These low-head (usually surge-receiving) systems require an additional means of control to ensure that liquid line subcooled refrigerant enters the expansion valve. Supplemental electronic controls, valves, or, at times, liquid pumps are included. This flow control equipment is often found on units that operate year-round to provide medium- and low-temperature food processing refrigeration, where a precise temperature must be held.

Table 1 Net Refrigeration Effect Factors for Reciprocating Compressors Used with Air-Cooled and Evaporative Condensers

Saturated Suction Temperature, °F	Condensing Temperature, °F										
	85	90	95	100	105	110	115	120	125	130	135
Open Compressors[a,b]											
–40	0.71	0.70	0.69	0.68	0.67	0.65	0.64	0.63	0.62	0.60	—
–20	0.77	0.76	0.74	0.73	0.72	0.71	0.70	0.69	0.67	0.66	—
0	0.82	0.80	0.79	0.78	0.77	0.76	0.75	0.74	0.73	0.71	—
+20	0.86	0.85	0.84	0.83	0.82	0.81	0.79	0.78	0.77	0.76	0.75
+40	0.91	0.90	0.89	0.87	0.86	0.85	0.84	0.83	0.82	0.81	0.80
Sealed Compressors[a]											
–40	0.55	0.54	0.53	0.52	0.51	0.50	0.49	0.47	0.46	0.44	—
–20	0.65	0.64	0.62	0.61	0.60	0.59	0.58	0.55	0.53	0.51	—
0	0.72	0.71	0.70	0.69	0.67	0.66	0.64	0.62	0.60	0.58	—
+20	0.77	0.76	0.75	0.74	0.72	0.71	0.69	0.68	0.66	0.64	0.62
+40	0.81	0.80	0.79	0.78	0.77	0.75	0.74	0.72	0.71	0.70	0.68
+50	0.83	0.82	0.81	0.80	0.79	0.78	0.76	0.75	0.74	0.73	0.72

[a] For R-22, factors are based on 15°F superheat entering the compressor. [b] For ammonia (R-717), factors are based on 10°F superheat.

Notes:
1. These factors should be used only for air-cooled and evaporative condensers connected to reciprocating compressors.
2. Condensing temperature is that temperature corresponding to saturation pressure as measured at compressor discharge.
3. Net refrigeration effect factors are an approximation only. Represent net refrigeration capacity as *approximate only* in published ratings.
4. For more accurate condenser selection and for condensers connected to centrifugal compressors, use total heat rejection of condenser and compressor manufacturer's total rating, which includes heat of compression.

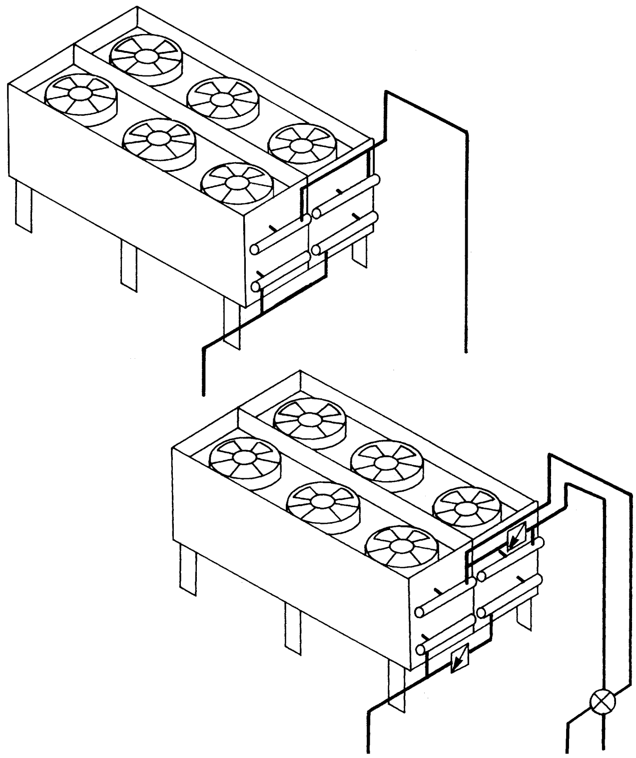


Fig. 7 Equal-Sized Condenser Sections Connected in Parallel and for Half-Condenser Operation During Winter

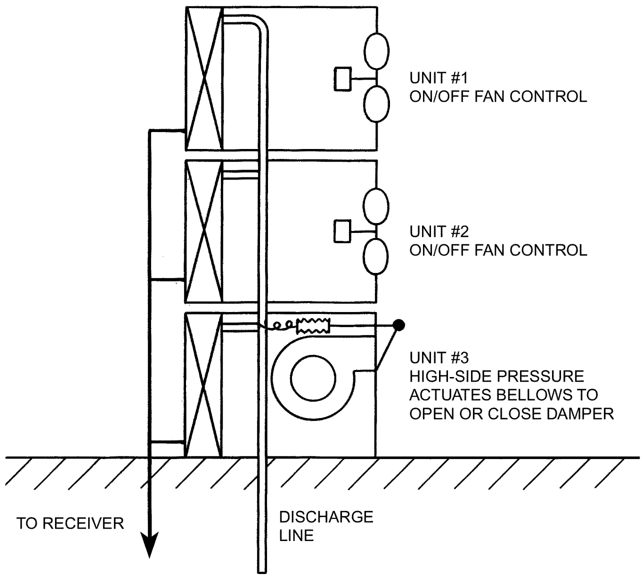


Fig. 8 Unit Condensers Installed in Parallel with Combined Fan Cycling and Damper Control

To prevent excessively low head pressure during winter operation, two basic control methods are used: (1) refrigerant-side control and (2) air-side control.

Refrigerant-Side Control. Control on the refrigerant side may be accomplished in the following ways:

- By modulating the amount of active condensing surface available for condensing by flooding the coil with liquid refrigerant. This method requires a receiver and a larger charge of refrigerant. Several valving arrangements give the required amount of flooding to meet the variable needs. Both temperature and pressure actuation are used.
- By going to one-half condenser operation. The condenser is initially designed with two equal parallel sections, each accommodating 50% of the load during normal summer operation. During winter, solenoid or three-way valves block off one condenser section (as well as its pumpdown to suction). This saves the flooding overcharge and also allows shutdown of fans on the inactive condenser side (Figure 7). Similar splits, such as one-third or two-thirds, are also possible. Anticipated load variations, along with climate conditions, usually dictate the preferred split arrangement for each specific installation.

  This split-condenser design has gained popularity, not only for its cold-weather control, but also for its ability to reduce the refrigerant charge. Part-load operating conditions also benefit when using a split condenser.

Air-Side Control. Control on the air side may be accomplished by one of three methods, or a combination of two of them: (1) fan cycling, (2) modulating dampers, and (3) fan speed control. Any airflow control must be oriented so that the prevailing wind does not cause adverse operating conditions.

Fan cycling in response to outdoor ambient temperature eliminates rapid cycling, but is limited to use with multiple fan units or is supplemental to other control methods. A common method for control of a two-fan unit is to cycle only one fan. A three-fan unit may cycle two fans. During average winter conditions, large multifan outdoor condensers have electronic controllers programmed to run the first fan (or pair of fans) continuously, at least at low fan speed. The cycle starts with the fan(s) closest to the refrigerant connecting piping as the first fan(s) on, last fan(s) off. The remaining fan(s) may cycle as required on ambient or high-side pressure. Airflow through the condenser may be further reduced by modulating airflow through the uncontrolled fan section, either by controlling the speed of some or all motors, or by dampers on either the air intake or discharge side (Figure 8).

In multiple, direct-drive motor-propeller arrangements, idle (off-cycle) fans should not be allowed to rotate backward; otherwise, air will short-circuit through them. Motor starting torque may be insufficient to overcome this reverse rotation. To eliminate this problem, designs for large units incorporate baffles to separate each fan assembly into individual compartments.

Air dampers controlled in response to either receiver pressure, ambient conditions, or liquid temperatures are also used to control compressor head pressure. These devices throttle airflow through the condenser coil from 100% to zero. Motors that drive propeller fans for such an application should have a flat power characteristic or should be fan-cooled so they do not overheat when the damper is nearly closed.

Fan speed can also be used to control compressor pressure. Because fan power increases in proportion to the cube of fan speed, energy consumption can be reduced substantially by slowing the fan at low ambient temperatures and during part-load operation. Solid-state controls can modulate frequency along with voltage to vary the speed of synchronous motors. Two-speed fan motors also produce energy savings. Chapter 44 has more information on speed control.

Parallel operation of condensers, especially with capacity control devices, requires careful design. Connecting only identical condensers in parallel reduces operational problems. (The section on Multiple-Condenser Installations has further information. Figures 13 and 14 also apply to air-cooled condensers.)

These control methods maintain sufficient condenser and liquid line pressure for proper expansion valve operation. During off cycles, when the outdoor temperature is lower than that of the indoor space to be conditioned or refrigerated, refrigerant migrates from the evaporator and receiver and condenses in the cold condenser. System pressure drops to what may correspond to the outdoor temperature. On start-up, insufficient feeding of liquid refrigerant to the evaporator, because of low compressor head pressure, causes low-pressure cut-out cycling of the compressor until the suction pressure reaches an operating level above the setting of the low pressurestat. At extremely low outdoor temperatures, system pressure may be below the cut-in point of the pressurestat and the compressor will not start. This difficulty is solved by (1) bypassing the low-pressure switch on start-up, and/or (2) using the condenser isolation method of control.

- **Low-pressure switch bypass.** On system start-up, a time-delay relay may bypass the low-pressure switch for 3 to 5 min to allow head pressure to build up and allow uninterrupted compressor start-up operation.
- **Fan cycle switch.** On system start-up, a high-side pressure-sensing switch can delay condenser fan(s) operation until a minimum preset head pressure is reached.
- **Condenser isolation.** This method uses a check valve arrangement to isolate the condenser. This prevents refrigerant from migrating to the condenser coil during the off cycle.

When the liquid temperature coming from the outdoor condenser is lower than the evaporator operating temperature, consider using a noncondensable-charged thermostatic expansion valve bulb. This type of bulb prevents erratic operation of the expansion valve by eliminating the possibility of condensation of the thermal bulb charge on the head of the expansion valve.

INSTALLATION AND MAINTENANCE

Installation and maintenance of remote condensers require little labor because of their relatively simple design. Remote condensers are located as close as possible to the compressor, either indoors or outdoors. They may be located above or below the level of the compressor, but always above the level of the receiver. During manufacturing, tubesheets or endplates are fastened to the finned-tube core at each end to complete the coil assembly. These provide a means of mounting the condenser enclosure or, in themselves, can form a part of the condenser unit enclosure. Center supports of similar design also support the condenser cabinet and, on larger units, allow fact section compartentalizing.

Installation and maintenance concerns include the following:

Indoor Condenser. When condensers are located indoors, conduct warm discharge air to the outdoors. An outdoor air intake opening near the condenser is provided and may be equipped with shutters. Indoor condensers can be used for space heating during winter and for ventilation during summer (Figure 9).

Outdoor Installations. In outdoor installations of vertical-face condensers, ensure that prevailing winds blow toward the air intake, or discharge shields should be installed to deflect opposing winds.

Piping. Piping practice with air-cooled condensers is identical to that established by experience with other remote condensers. Size discharge piping to the condenser for a total pressure drop equivalent no more than about 2°F saturated temperature drop. Liquid line drop leg to the receiver should incorporate an adequately sized check valve. Follow standard piping procedures and good piping practices. For pipe sizing, refer to Chapter 36 of the 2005 *ASHRAE Handbook—Fundamentals*.

A pressure tap valve should be installed at the highest point in the discharge piping run to facilitate removal of inadvertently trapped noncondensable gases. Purging should only be done with the compressor system off and pressures equalized. *Note*: This is only to be done by qualified personnel having the proper reclaim/recovery equipment as mandated by the appropriate agency, such as the Environmental Protection Agency in the United States.

Receiver. A condenser may have its receiver installed close by, but most often it will be separate and remotely installed. (Most

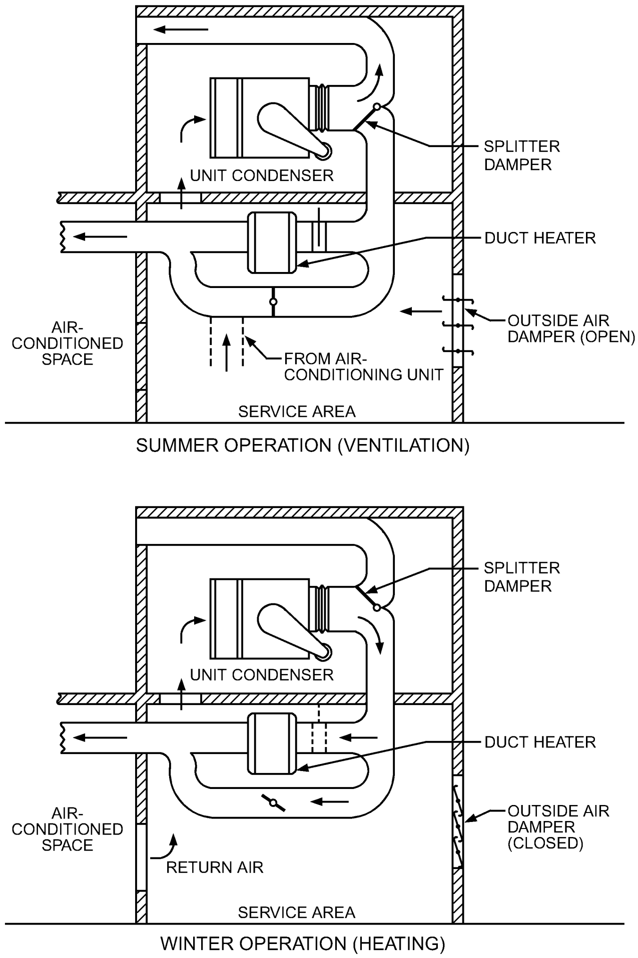


Fig. 9 Air-Cooled Unit Condenser for Winter Heating and Summer Ventilation

water-cooled systems function without a receiver, because it is an integral part of the water-cooled condenser.) If a receiver is used and located in a comparatively warm ambient temperature, the liquid drain line from the condenser should be sized for a liquid velocity of under 100 fpm and designed for gravity drainage as well as for venting to the condenser. The receiver should be equipped with a pressure-relief valve assembly that meets applicable codes and discharge requirements, sight glass(es), and positive-action in-and-out shutoff valves. Requirements for such valving are outlined in ASHRAE *Standard* 15.

The receiver's manufacturer can recommend mechanical subcooling and/or reduced refrigerant charge ideas incorporating improved ambient subcooling for condenser coils.

Maintenance. Schedule periodic inspection and lubrication of fan motor and fan bearings. Lube if required, following the instructions found either on the motor nameplate or in the operation manual. Condensers also require periodic removal of lint, leaves, dirt, and other airborne materials from their inlet coil surface by brushing the air entry side of the coil with a long-bristled brush. Washing with a low-pressure water hose or blowing with compressed air is more effectively done from the side opposite the air entry face. A mild cleaning solution is required to remove restaurant grease and other oil films from the fin surface. Indoor condensers in dusty locations, such as processing plants and supermarkets, should have adequate access for unrestricted coil cleaning and fin surface inspection. In all cases, extraneous material on fins reduces equipment performance, shortens operating life, increases running time, and causes more energy use.

When a condenser coil fails (develops leaks), it is usually at the point where the tube is in contact with one of the supporting tubesheets. If not neutralized in some way, these wear points tend to leak after several years of full-time operation. Air-cooled condenser tube fractures are caused mainly by thermal stresses in the coil core, or by excessive vibration. To address this, some manufacturers of larger coils either minimize or eliminate the tube-to-metal contact wear point at tube traverse points of endplates and center supports by substantially oversizing some or all of the endplate and center support tube holes. This design substantially limits refrigerant tube fractures in the coil.

Other potential failure points are at inlet and outlet manifolds. Failure is likely to occur when the manifold or connecting pipe is not adequately supported and/or discharge pulsing is not properly controlled, or when fan unbalance causes vibration.

EVAPORATIVE CONDENSERS

As with water- and air-cooled condensers, evaporative condensers reject heat from a condensing vapor into the environment. In an evaporative condenser, hot, high-pressure vapor from the compressor discharge circulates through a condensing coil that is continually wetted on the outside by a recirculating water system. As seen in Figure 10, air is simultaneously directed over the coil, causing a small portion of the recirculated water to evaporate. This evaporation removes heat from the coil, thus cooling and condensing the vapor.

Evaporative condensers reduce water pumping and chemical treatment requirements associated with cooling tower/refrigerant condenser systems. In comparison with an air-cooled condenser, an evaporative condenser requires less coil surface and airflow to reject the same heat, or alternatively, greater operating efficiencies can be achieved by operating at a lower condensing temperature.

An evaporative condenser can operate at a lower condensing temperature than an air-cooled condenser because the condensing temperature in an air-cooled condenser is determined by the ambient dry-bulb temperature. In the evaporative condenser, heat rejection is limited by the ambient wet-bulb temperature, which is normally 14 to 25°F lower than the ambient dry bulb. Also, evaporative condensers typically provide lower condensing temperatures than the cooling tower/water-cooled condenser because the heat and mass transfer steps (between refrigerant and cooling water and between water and ambient air) are more efficiently combined in a single piece of equipment, allowing minimum sensible heating of the cooling water. Evaporative condensers are, therefore, the most compact for a given capacity.

HEAT TRANSFER

In an evaporative condenser, heat flows from the condensing refrigerant vapor inside the tubes, through the tube wall, to the water film outside the tubes, and then from the water film to the air. Figure 11 shows typical temperature trends in a counterflow evaporative condenser. The driving potential in the first step of heat transfer is the temperature difference between the condensing refrigerant and the surface of the water film, whereas the driving potential in the second step is a combination of temperature and water vapor enthalpy difference between the water surface and air. Sensible heat transfer at the water/air interface occurs because of the temperature gradient, and mass transfer (evaporation) of water vapor from the water/air interface to the airstream occurs because of the enthalpy gradient. Commonly, a single enthalpy driving force between air saturated at the temperature of the water-film surface and the enthalpy of air in contact with that surface is applied to simplify an analytical approach. Exact formulation of the heat and mass transfer process requires considering the two forces simultaneously.

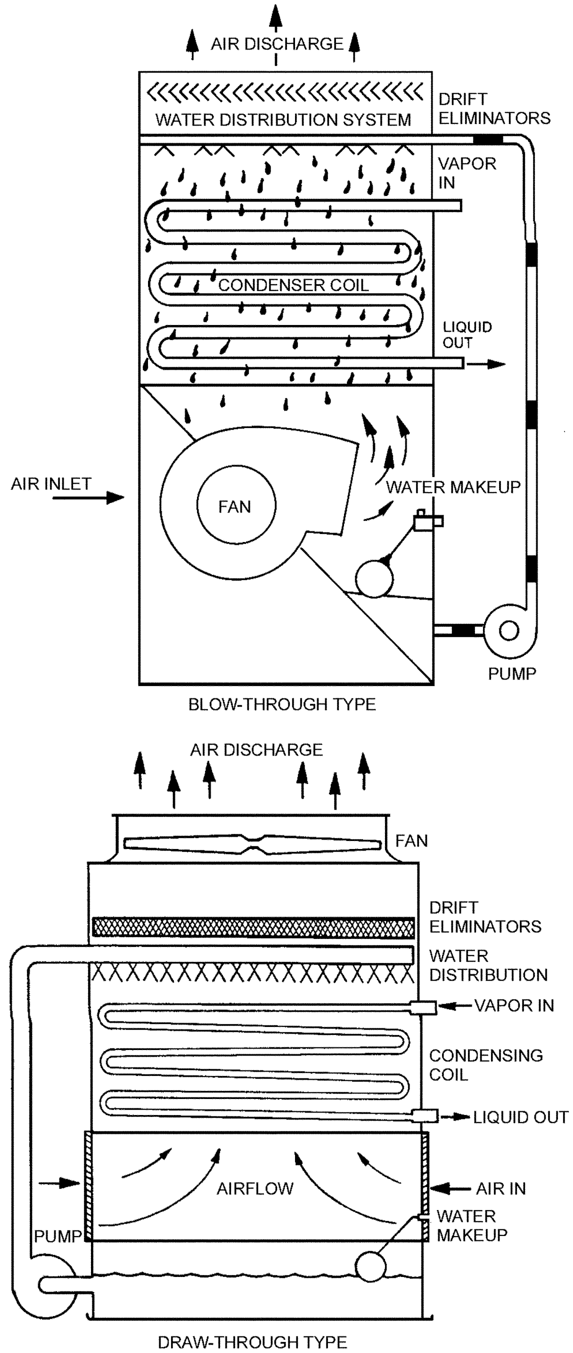


Fig. 10 Functional Views of Evaporative Condenser

Because evaporative condenser performance cannot be represented solely by a temperature difference or an enthalpy difference, simplified predictive methods can only be used for interpolation of data between test points or between tests of different-sized units, provided that air velocity, water flow, refrigerant velocity, and tube bundle configuration are comparable.

The rate of heat flow from the refrigerant through the tube wall and to the water film can be expressed as

$$q = U_s A(t_c - t_s) \tag{8}$$

where

q = rate of heat flow, Btu/h
U_s = overall heat transfer coefficient, Btu/h·ft^2·°F

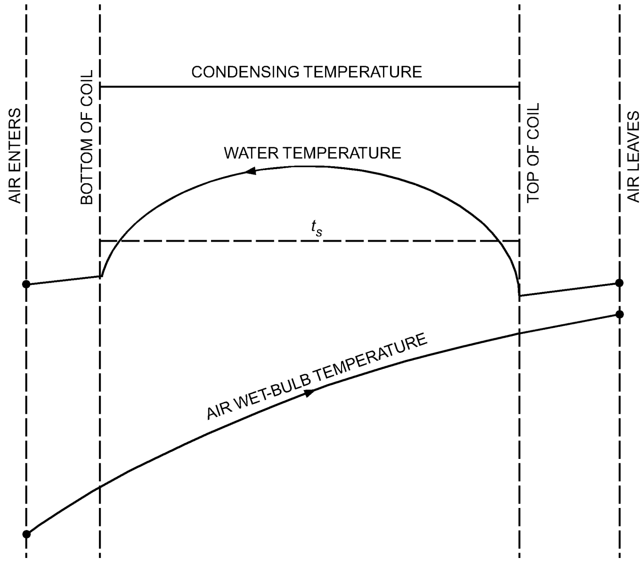


Fig. 11 Heat Transfer Diagram for Evaporative Condenser

t_c = saturation temperature at pressure of refrigerant entering condenser, °F
t_s = temperature of water film surface, °F
A = outside surface area of condenser tubes, ft^2

The rate of heat flow from the water-air interface to the airstream can be expressed as

$$q = U_c A(h_s - h_e) \tag{9}$$

where

q = heat input to condenser, Btu/h
U_c = overall transfer coefficient from water-air interface to airstream, Btu/h·ft^2·°F divided by enthalpy difference Δh in Btu/lb
h_s = enthalpy of air saturated at t_c, Btu/lb
h_e = enthalpy of air entering condenser, Btu/lb

Equations (8) and (9) have three unknowns: U_s, U_c, and t_s (h_s is a function of t_s). Consequently, the solution requires an iterative procedure that estimates one of the three unknowns and then solves for the remaining. This process is further complicated because of some of the more recently developed refrigerant mixtures for which t_c changes throughout the condensing process.

Korenic (1980), Leidenfrost and Korenic (1979, 1982), and Leidenfrost et al. (1980) evaluated heat transfer performance of evaporative condensers by analyzing the internal conditions in a coil. Further work on evaporative condenser performance modeling is under way in the industry.

CONDENSER CONFIGURATION

Principal components of an evaporative condenser include the condensing coil, fan(s), spray water pump, water distribution system, cold-water sump, drift eliminators, and water makeup assembly.

Coils

Generally, evaporative condensers use condensing coils made from bare pipe or tubing without fins. The high rate of energy transfer from the wetted external surface to the air eliminates the need for an extended surface. Bare coils also sustain performance better because they are less susceptible to fouling and are easier to clean. The high rate of energy transfer from the wetted external surface to the air makes finned coils uneconomical when used exclusively for wet operation. However, partially or wholly finned coils are sometimes used to reduce or eliminate plumes and/or to reduce water consumption by operating the condenser dry during off-peak conditions.

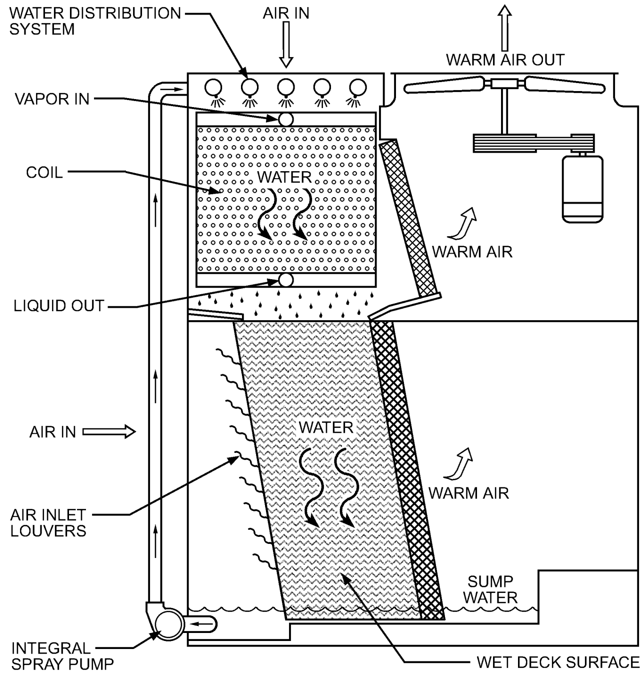


Fig. 12 Combined Coil/Fill Evaporative Condenser

Coils are usually made from steel tubing, copper tubing, iron pipe, or stainless steel tubing. Ferrous materials are generally hot-dip galvanized for exterior protection.

Method of Coil Wetting

The spray water pump circulates water from the cold water sump to the distribution system located above the coil. Water descends through the air circulated by the fan(s), over the coil surface, and eventually returns to the pan sump. The distribution system is designed to completely and continuously wet the full coil surface. Complete wetting ensures the high rate of heat transfer achieved with wet tubes and prevents excessive scaling, which is more likely to occur on intermittently or partially wetted surfaces. Such scaling is undesirable because it decreases heat transfer efficiency (which tends to raise the condensing temperature) of the unit. Water lost through evaporation, drift, and blowdown from the cold water sump is replaced through a makeup assembly that typically consists of a mechanical float valve or solenoid valve and float switch combination.

Airflow

All commercially available evaporative condensers use mechanical draft; that is, fans move a controlled flow of air through the unit. As shown in Figure 10, these fans may be on the air inlet side (forced draft) or the air discharge side (induced draft). The type of fan selected, centrifugal or axial, depends on external pressure needs, energy requirements, and permissible sound levels. In many units, recirculating water is distributed over the condensing coil and flows down to the collecting basin, counterflow to the air flowing up through the unit.

In an alternative configuration (Figure 12) a cooling tower fill augments the condenser's thermal performance. In this unit, one airstream flows down over the condensing coil, parallel to the recirculating water, and exits horizontally into the fan plenum. The recirculating water then flows down over cooling tower fill, where it is further cooled by a second airstream before it is reintroduced over the condenser coil.

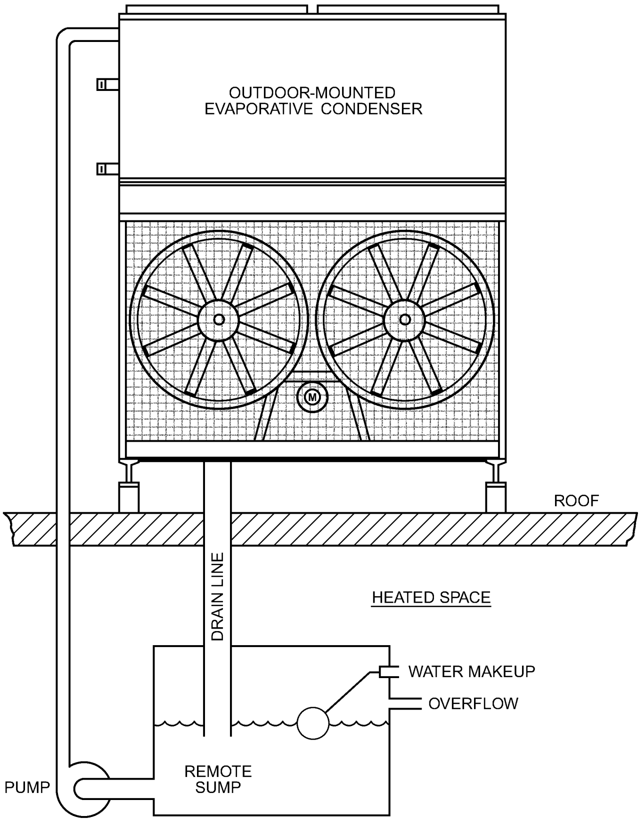
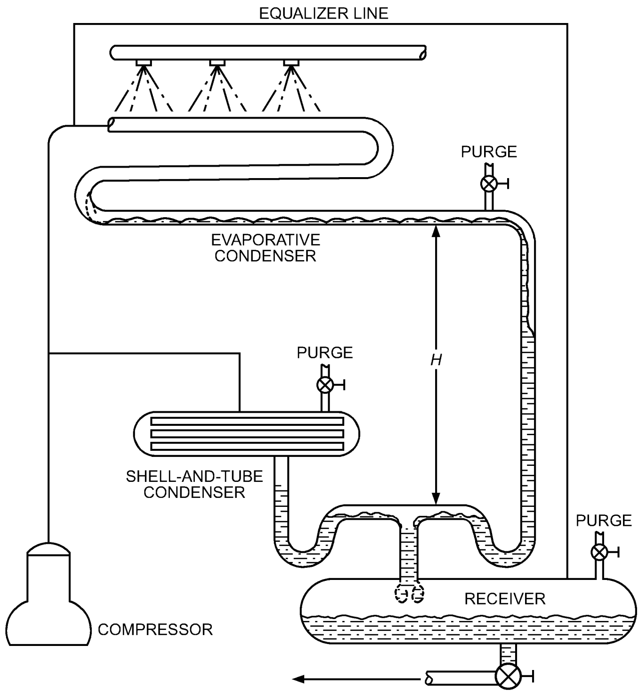


Fig. 13 Evaporative Condenser Arranged for Year-Round Operation



Pressure created by fluid height (*H*) above trap must be greater than internal resistance of condenser. Note purge connections at both condensers and receiver.

Fig. 14 Parallel Operation of Evaporative and Shell-and-Tube Condenser

Drift eliminators strip most water droplets from the discharge airstream, but some escape as drift. The rate of drift loss from an evaporative condenser is a function of unit configuration, eliminator design, airflow rate through the evaporative condenser, and water flow rate. Generally, an efficient eliminator design can reduce drift loss to a range of 0.001 to 0.2% of the water circulation rate. If the air inlet is near the sump, louvers or deflectors may be installed to prevent water from splashing from the unit.

CONDENSER LOCATION

Most evaporative condensers are located outdoors, frequently on the roofs of machine rooms. They may also be located indoors and ducted to the outdoors. Generally, centrifugal-fan models must be used for indoor applications to overcome the static resistance of the duct system.

Evaporative condensers installed outdoors can be protected from freezing in cold weather by a remote sump arrangement in which the water and pump are located in a heated space that is remote from the condensers (Figure 13). Piping is arranged so that whenever the pump stops, all the water drains back into the sump to prevent freezing. Where remote sumps are not practical, reasonable protection can be provided by sump heaters, such as electric immersion heaters, steam coils, or hot-water coils. Water pumps and lines must also be protected, for example with electric heat-tracing tape and insulation.

Where the evaporative condenser is ducted to the outdoors, moisture from the warm saturated air can condense in condenser discharge ducts, especially if the ducts pass through a cool space. Some condensation may be unavoidable even with short, insulated ducts. In such cases, the condensate must be drained. Also, in these ducted applications, the drift eliminators must be highly effective.

MULTIPLE-CONDENSER INSTALLATIONS

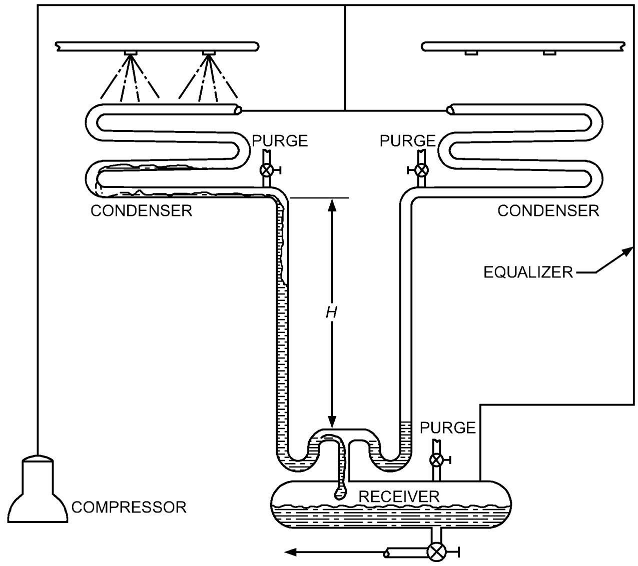
Large refrigeration plants may have several evaporative condensers connected in parallel with each other or with shell-and-tube condensers. In these systems, unless all condensers have the same refrigerant-side pressure drop, condensed liquid refrigerant will flood back into the condensing coils of those condensers with the highest pressure loss. Also, in periods of light load when some condenser fans are off, liquid refrigerant will flood the lower coil circuits of the active condensers.

Trapped drop legs, as illustrated in Figures 14 and 15, provide proper control of such multiple installations. The effective height *H* of drop legs must equal the pressure loss through the condenser at maximum loading. Particularly in cold weather, when condensing pressure is controlled by shutting down some condenser fans, active condensers may be loaded considerably above nominal rating, with higher pressure losses. The height of the drop leg should be great enough to accommodate these greater design loads.

RATINGS

Heat rejected from an evaporative condenser is generally expressed as a function of the saturated condensing temperature and entering air wet bulb. The type of refrigerant has considerable effect on ratings; this effect is handled by separate tables (or curves) for each refrigerant, or by a correction factor when the difference is small.

Evaporative condensers are commonly rated in terms of total heat rejection when condensing a particular refrigerant at a specific condensing temperature and entering air wet-bulb temperature. Many manufacturers also provide alternative ratings in terms of evaporator load for a given refrigerant at a specific combination of saturated suction temperature and condensing temperature. These



Each condenser can operate independently. Height (H) above trap, for either condenser, must be not less than internal resistance of condenser. Note purge connections at both condensers and receiver.

Fig. 15 Parallel Operation of Two Evaporative Condensers

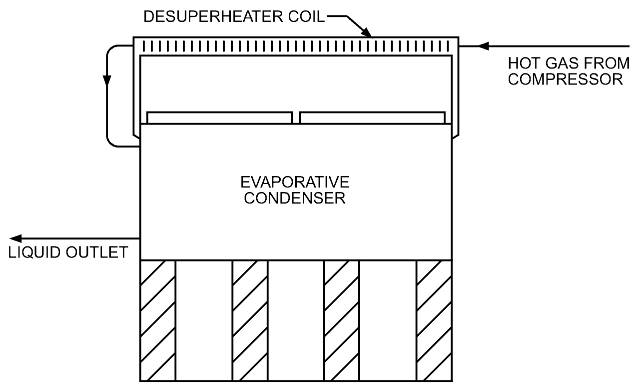


Fig. 16 Evaporative Condenser with Desuperheater Coil

ratings include an assumed value for heat of compression, typically based on an open, reciprocating compressor, and should only be used for that type of unit. Where another type of compression equipment is used, such as a gas-cooled hermetic and semihermetic compressor, the total power input to the compressor(s) should be added to the evaporator load and this value should be used to select the condenser.

Rotary screw compressors use oil to lubricate moving parts and provide a seal between the rotors and the compressor housing. This oil is subsequently cooled either in a separate heat exchanger or by refrigerant injection. In the former case, the amount of heat removed from the oil in the heat exchanger is subtracted from the sum of the refrigeration load and compressor brake power to obtain the total heat rejected by the evaporative condenser. When liquid injection is used, the total heat rejection is the sum of the refrigeration load and brake horsepower. Heat rejection rating data together with any ratings based on refrigeration capacity should be included.

DESUPERHEATING COILS

A desuperheater is an air-cooled finned coil, usually installed in the discharge airstream of an evaporative condenser (Figure 16). Its

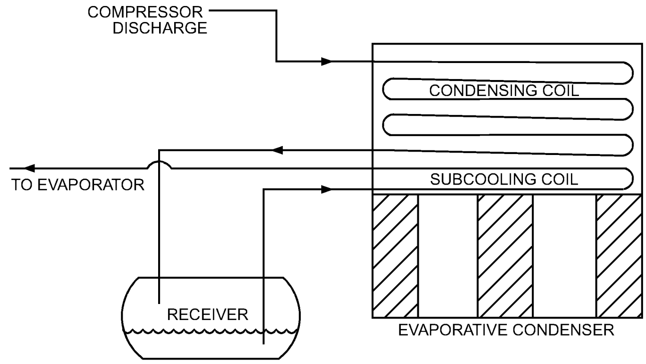


Fig. 17 Evaporative Condenser with Liquid Subcooling Coil

primary function is to increase condenser capacity by removing some of the superheat from the discharge vapor before the vapor enters the wetted condensing coil. The amount of superheat removed is a function of the desuperheater surface, condenser airflow, and temperature difference between the refrigerant and air. In practice, a desuperheater is limited to reciprocating compressor ammonia installations where discharge temperatures are relatively high (250 to 300°F).

REFRIGERANT LIQUID SUBCOOLERS

The refrigerant pressure at the expansion device feeding the evaporator(s) is lower than that in the receiver because of the pressure drop in the liquid line. If the liquid line is long or if the evaporator is above the receiver (which further decreases refrigerant pressure at the expansion device), significant flashing can occur in the liquid line. To avoid flashing where these conditions exist, the liquid refrigerant must be subcooled after it leaves the receiver. The minimum amount of subcooling required is the temperature difference between the condensing temperature and saturation temperature corresponding to the saturation pressure at the expansion device. Subcoolers are often used with halocarbon systems but seldom with ammonia systems for the following reasons:

- Because ammonia has a relatively low liquid density, liquid line static pressure loss is small.
- Ammonia has a very high latent heat; thus, the amount of flash gas resulting from typical pressure loss in the liquid line is extremely small.
- Ammonia is seldom used in a direct-expansion feed system where subcooling is critical to proper expansion valve performance.

One method commonly used to supply subcooled liquid for halocarbon systems places a subcooling coil section in the evaporative condenser below the condensing coil (Figure 17). Depending on the design wet-bulb temperature, condensing temperature, and subcooling coil surface area, the subcooling coil section normally furnishes 10 to 15°F of liquid subcooling. As shown in Figure 17, a receiver must be installed between the condensing coil and subcooling coil to provide a liquid seal for the subcooling circuit.

MULTICIRCUIT CONDENSERS AND COOLERS

Evaporative condensers and evaporative fluid coolers, which are essentially the same, may be multicircuited to condense different refrigerants or cool different fluids simultaneously, but only if the difference between the condensing temperature and leaving fluid temperature is small. Typical multicircuits (1) condense different refrigerants found in food market units, (2) condense and cool different fluids in separate circuits (e.g., condensing refrigerant from a screw compressor in one circuit and cooling water or glycol for its oil cooler in another circuit), and (3) cool different fluids in separate circuits such as in many industrial applications where it is necessary to keep the fluids from different heat exchangers in separate circuits.

A multicircuit unit is usually controlled by sensing sump water temperature and using capacity control dampers, fan cycling, or a variable-speed drive to modulate airflow and match unit capacity to the load. Two-speed motors or separate high- and low-speed motors are also used. Sump water temperature has an averaging effect on control. Because all water in the sump is recirculated once every minute or two, its temperature is a good indicator of changes in load.

WATER TREATMENT

As recirculated water evaporates in an evaporative condenser, the dissolved solids in the makeup water continually increase as more water is added. Continued concentration of these dissolved solids can lead to scaling and/or corrosion problems. In addition, airborne impurities and biological contaminants are often introduced into recirculated water. If these impurities are not controlled, they can cause sludge or biological fouling. Simple blowdown (discharging a small portion of the recirculating water to a drain) may be adequate to control scale and corrosion on sites with good-quality makeup water, but it does not control biological contaminants such as *Legionella*. All evaporative condensers should be treated to restrict biological growth and many benefit from chemical treatment to control scale and corrosion. Chapter 48 of the 2007 *ASHRAE Handbook—HVAC Applications* covers water treatment in more detail. Also see ASHRAE *Guideline* 12 for specific recommendations on *Legionella* control. Specific recommendations on water treatment can be obtained from water treatment suppliers.

WATER CONSUMPTION

For the purpose of sizing makeup water piping, all heat rejected by an evaporative condenser is assumed to be latent heat (approximately 1050 Btu/lb of water evaporated). The heat rejected depends on operating conditions, but can range from 14,000 Btu/h per ton of air conditioning to 17,000 Btu/h per ton of freezer storage. The evaporated water ranges from about 1.6 to 2 gph per ton of refrigeration. In addition, a small amount of water can be lost in the form of drift through the eliminators. With good-quality makeup water, bleed rates may be as low as one-quarter to one-half the evaporation rate, and total water consumption would range from 2 gph/ton for air conditioning to 3 gph/ton for refrigeration.

CAPACITY MODULATION

To ensure operation of expansion valves and other refrigeration components, extremely low condensing pressures must be avoided. Capacity can be controlled in response to condensing pressure on single-circuit condensers and in response to the temperature of the spray water on multicircuited condensers. Means of controlling capacity include (1) intermittent fan operation; (2) a modulating damper in the airstream to reduce the airflow (centrifugal fan models only); and (3) fan speed control using variable-speed motors and/or drives, two-speed motors, or additional lower-power pony motors (small modular motors on the same shaft). Two-speed fan motors usually operate at 100 and 50% fan speed, which provides 100% and approximately 60% condenser capacity, respectively. Pony motors also provide some redundancy in case one motor fails. With the fans off and the water pump operating, condenser capacity is approximately 10%.

Often, a two-speed fan motor or supplementary (pony) motor that operates the fan at reduced speed can provide sufficient capacity control, because condensing pressure seldom needs to be held to a very tight tolerance other than to maintain a minimum condensing pressure to ensure refrigerant liquid feed pressure for the low-pressure side and/or sufficient pressure for hot-gas defrost requirements. For applications requiring close control of condensing pressure, virtually infinite capacity control can be provided by using frequency-modulating controls to control fan motor speed. These controls may also be justified economically because they save energy and extend the life of the fan and drive assembly compared to fan cycling with two-speed control. However, special concerns and limitations are associated with their use, which are discussed in Chapter 44.

Modulating air dampers also offer closer control on condensing pressure, but they do not offer as much fan power reduction at part load as fan speed control. Water pump cycling for capacity control is not recommended because the periodic drying of the tube surface promotes scale build-up.

The designer should research applicable codes and standards when applying evaporative condensers. Most manufacturers provide sound level information for their equipment. Centrifugal fans are inherently quieter than axial fans, but axial fans generally have a lower power requirement for a given application. Low-speed operation and multistaging usually lowers sound levels of condensers with axial fans. In some cases, factory-supplied sound attenuators may need to be installed.

PURGING

Refrigeration systems operating below atmospheric pressure and systems that are opened for service may require purging to remove air that causes a high condensing pressure. With the system operating, purging should be done from the top of the condensing coil outlet connection. On multiple-coil condensers or multiple-condenser installations, one coil at a time should be purged. Purging two or more coils at one time equalizes coil outlet pressures and can cause liquid refrigerant to back up in one or more of the coils, thus reducing operating efficiency.

Noncondensables can be removed from the condenser using a mechanical, automatic operational purge unit for halogenated refrigerants. Additional information on purge units is located in Chapter 42. Manual purging, which involves opening a manual valve and releasing noncondensables (and refrigerant) to the atmosphere, is not acceptable for halogenated refrigerants because environmental laws prohibit intentional venting of refrigerant from the system.

Purging may also be done from the high point of the evaporative condenser refrigerant feed, but it is only effective when the condenser is not operating. During normal operation, noncondensables are dispersed throughout the high-velocity vapor, and excessive refrigerant would be lost if purging were done from this location (see Figures 14 and 15). All codes and ordinances governing discharge, recovery, and recycling of refrigerants must be followed.

MAINTENANCE

Evaporative condensers are often installed in remote locations and may not receive the routine attention of operating and maintenance personnel. Programmed maintenance is essential, however, and the manufacturer's operation and maintenance guidelines (often downloadable from their Web sites) should be consulted. Table 2 shows a typical checklist for evaporative condensers. Chapter 39 also has information on maintenance that applies to evaporative condensers.

CODES AND STANDARDS

If state or local codes do not take precedence, design pressures, materials, welding, tests, and relief devices should be in accordance with the ASME *Boiler and Pressure Vessel Code*, Section VIII, Division 1. Evaporative condensers are typically exempt, however, from the ASME Code on the basis of Item (c) of the Scope of the Code, which states that if the inside diameter of the condenser shell is 6 in. or less, it is not governed by the Code. (Coils can be built in

Table 2 Typical Maintenance Checklist

Maintenance Item	Frequency
1. Check fan and motor bearings, and lubricate if necessary. Check tightness and adjustment of thrust collars on sleeve-bearing units and locking collars on ball-bearing units.	Q
2. Check belt tension.	M
3. Clean strainer. If air is extremely dirty, strainer may need frequent cleaning.	W
4. Check, clean, and flush sump, as required.	M
5. Check operating water level in sump, and adjust makeup valve, if required.	W
6. Check water distribution, and clean as necessary.	W
7. Check bleed water line to ensure it is operative and adequate as recommended by manufacturer.	M
8. Check fans and air inlet screens and remove any dirt or debris.	D
9. Inspect unit carefully for general preservation and cleanliness, and make any needed repairs immediately.	R
10. Check operation of controls such as modulating capacity control dampers.	M
11. Check operation of freeze control items such as pan heaters and their controls.	Y
12. Check the water treatment system for proper operation.	W
13. Inspect entire evaporative condenser for spot corrosion. Treat and refinish any corroded spot.	Y

D = Daily; W = Weekly; M = Monthly; Q = Quarterly;
Y = Yearly; R = As required.

compliance with ASME *Code* B31.5, however.) Other rating and testing standards include

- ASHRAE *Standard* 15, Safety Code for Mechanical Refrigeration.
- ASHRAE *Standard* 64, Methods of Testing Remote Mechanical-Draft Evaporative Refrigerant Condensers.
- ARI *Standard* 490, Remote Mechanical-Draft Evaporative Refrigerant Condensers.

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