Related Commercial Resources

CHAPTER 41

LIQUID COOLERS

A LIQUID cooler (hereafter called a cooler) is a heat exchanger in which refrigerant is evaporated, thereby cooling a fluid (usually water or brine) circulating through the cooler. This chapter addresses the performance, design, and application of coolers.

TYPES OF LIQUID COOLERS

Various types of liquid coolers and their characteristics are listed in Table 1 and described in the following sections.

Direct-Expansion

Refrigerant evaporates inside the tubes of a direct-expansion cooler. These coolers are usually used with positive-displacement compressors, such as reciprocating, rotary, or rotary screw compressors, to cool various fluids, such as water, water/glycol mixtures, and brine. Common configurations include shell-and-tube, tube-in-tube, and brazed-plate.

Figure 1 shows a typical **shell-and-tube** cooler. A series of baffles channels the fluid throughout the shell side. The baffles create cross flow through the tube bundle and increase the velocity of the fluid, thereby increasing its heat transfer coefficient. The velocity of the fluid flowing perpendicular to the tubes should be at least 2 fps to clean the tubes and less than the velocity limit of the tube and baffle materials, to prevent erosion.

Refrigerant distribution is critical in direct-expansion coolers. If some tubes are fed more refrigerant than others, refrigerant may not fully evaporate in the overfed tubes, and liquid refrigerant may escape into the suction line. Because most direct-expansion coolers are controlled by an expansion valve that regulates suction superheat, the remaining tubes must produce a higher superheat to evaporate the liquid escaping into the suction line. This unbalance causes poor overall heat transfer. Uniform distribution is usually achieved by adding a distributor, which creates sufficient turbulence to promote a homogeneous mixture so that each tube gets the same mixture of liquid and vapor.

The number of refrigerant passes is another important item in direct-expansion cooler performance. A single-pass cooler must evaporate all the refrigerant before it reaches the end of the first pass; this requires long tubes. A multiple-pass cooler is significantly shorter than a single-pass cooler, but must be properly designed to ensure proper refrigerant distribution after the first pass. Internally and externally enhanced tubes can also be used to reduce cooler size.

A **tube-in-tube** cooler is similar to a shell-and-tube design, except that it consists of one or more pairs of coaxial tubes. The fluid usually flows inside the inner tube while the refrigerant flows in the annular space between the tubes. In this way, the fluid side can be mechanically cleaned if access to the header is provided.

The preparation of this chapter is assigned to TC 8.5, Liquid-to-Refrigerant Heat Exchangers.

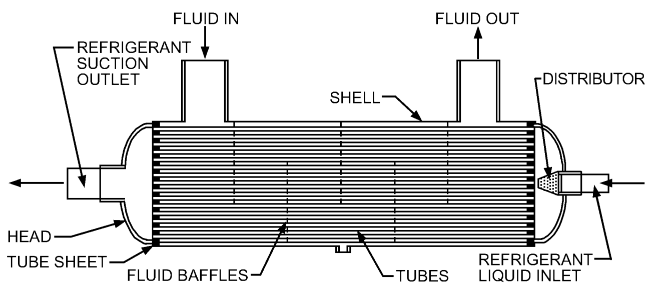

Fig. 1 Direct-Expansion Shell-and-Tube Cooler

Table 1 Types of Coolers

Type of Cooler	Subtype	Usual Refrigerant Feed Device	Usual Capacity Range, tons	Commonly Used Refrigerants
Direct-expansion	Shell-and-tube	Thermal expansion valve	2 to 1000	12, 22, 134a, 404A, 407C, 410A, 500, 502, 507A, 717
		Electronic modulation valve	2 to 1000	
	Tube-in-tube	Thermal expansion valve	5 to 25	12, 22, 134a, 717
	Brazed-plate	Thermal expansion valve	0.6 to 200	12, 22, 134a, 404A 407C, 410A, 500, 502, 507A, 508B, 717, 744
	Semiwelded plate	Thermal expansion valve	50 to 1990	12, 22, 134a, 500, 502, 507A, 717, 744
Flooded	Shell-and-tube	Low-pressure float	25 to 2000	11, 12, 22, 113, 114
		High-pressure float	25 to 6000	123, 134a, 500, 502, 507A, 717
		Fixed orifice(s)	25 to 6000	
		Weir	25 to 6000	
	Spray shell-and-tube	Low-pressure float	50 to 10,000	11, 12, 13B1, 22
		High-pressure float	50 to 10,000	113, 114, 123, 134a
	Brazed-plate	Low-pressure float	0.6 to 200	12, 22, 134a, 500, 502, 507A, 717, 744
	Semiwelded plate	Low-pressure float	50 to 1990	12, 22, 134a, 500, 502, 507A, 717, 744
Baudelot	Flooded	Low-pressure float	10 to 100	22, 717
	Direct-expansion	Thermal expansion valve	5 to 25	12, 22, 134a, 717
Shell-and-coil	—	Thermal expansion valve	2 to 10	12, 22, 134a, 717

Brazed- or **semiwelded-plate** coolers are constructed of plates brazed or laser-welded together to make an assembly of separate channels. Semiwelded designs have the refrigerant side welded and the fluid side gasketed and allow contact of the refrigerant with the fluid-side gaskets. These designs can be disassembled for inspection and mechanical cleaning of the fluid side. Brazed types do not have gaskets, cannot be disassembled, and are cleaned chemically. Internal leaks in brazed plates typically cannot be repaired. This type of evaporator is designed to work in a vertical orientation. Uniform distribution in direct-expansion operation is typically achieved by using a special plate design or distributor insert; flooded and pumped overfeed operations do not require distribution devices. Plate coolers are very compact and require minimal space.

Most tubular direct-expansion coolers are designed for horizontal mounting. If they are mounted vertically, performance may vary considerably from that predicted because two-phase flow heat transfer is a direction-sensitive phenomenon and dryout begins earlier in vertical upflow.

Flooded

In a flooded **shell-and-tube** cooler, refrigerant vaporizes on the outside of the tubes, which are submerged in liquid refrigerant in a closed shell. Fluid flows through the tubes as shown in Figure 2. Flooded coolers are usually used with rotary screw or centrifugal compressors to cool water, water/glycol mixtures, or brine.

A refrigerant liquid/vapor mixture usually feeds into the bottom of the shell through a distributor that distributes the mixture equally under the tubes. The relatively warm fluid in the tubes heats the refrigerant liquid surrounding the tubes, causing it to boil. As bubbles rise through the space between tubes, the liquid surrounding the tubes becomes increasingly bubbly (or foamy, if much oil is present).

The refrigerant vapor must be separated from the mist generated by the boiling refrigerant. The simplest separation method is provided by a dropout area between the top row of tubes and the suction connections. If this dropout area is insufficient, a coalescing filter may be required between the tubes and connections. Perry and Green (2007) give additional information on mist elimination. Another approach is to add another vessel, or "surge drum," above the suction connections. The diameter of this vessel is selected so that the velocity of the liquid droplets slows to the point where they fall back to the bottom of the surge drum. This liquid is then drained back into the flooded cooler.

The size of tubes, number of tubes, and number of passes should be determined to maintain fluid velocity typically between 3 and 10 fps for copper alloy tubing. Velocities beyond these limits may be used if the fluid is free of suspended abrasives and fouling substances (Ayub and Jones 1987; Sturley 1975) or if the tubing is manufactured from special alloys, such as titanium and stainless steel, that have better resistance to erosion. In some cases, the minimum velocity may be determined by a lower Reynolds number limit.

One variation of this cooler is the **spray shell-and-tube** cooler. In large-diameter coolers where the refrigerant's heat transfer coefficient is adversely affected by the refrigerant pressure, liquid can be sprayed to cover the tubes rather than flooding them. A mechanical pump circulates liquid from the bottom of the cooler to the spray heads.

Flooded shell-and-tube coolers are generally unsuitable for other than horizontal orientation.

In a flooded **plate** cooler (Figure 3), refrigerant vaporizes in vertical channels between corrugated plates with the liquid inlet at the bottom and the vapor outlet at the top (i.e., vertical upflow). The warm fluid flow may be either counter or parallel to the refrigerant flow. Both thermosiphon (gravity feed) and pumped overfeed operation are used. Surge drums are required for pumped overfeed operation but usually not for thermosiphon operation because the corrugated plates demist flow under most conditions.

Baudelot

Baudelot coolers (Figure 4) are used to cool a fluid to near its freezing point in industrial, food, and dairy applications. In this cooler, fluid circulates over the outside of vertical plates, which are easy to clean. The inside surface of the plates is cooled by evaporating the refrigerant. The fluid to be cooled is distributed uniformly along the top of the heat exchanger and then flows by gravity to a collection pan below. The cooler may be enclosed by insulated walls to avoid unnecessary loss of refrigeration.

R-717 (ammonia) is commonly used with flooded Baudelot coolers using conventional gravity feed with a surge drum. A low-pressure float valve maintains a suitable refrigerant liquid level in the surge drum. Baudelot coolers using other common refrigerants are generally direct-expansion, with thermostatic expansion valves.

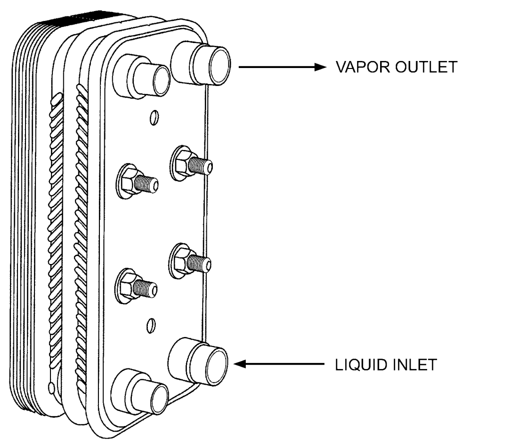

Fig. 3 Flooded Plate Cooler

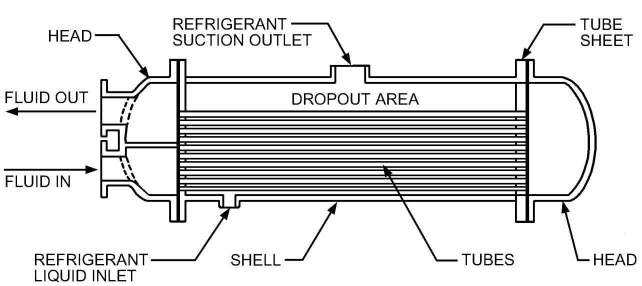

Fig. 2 Flooded Shell-and-Tube Cooler

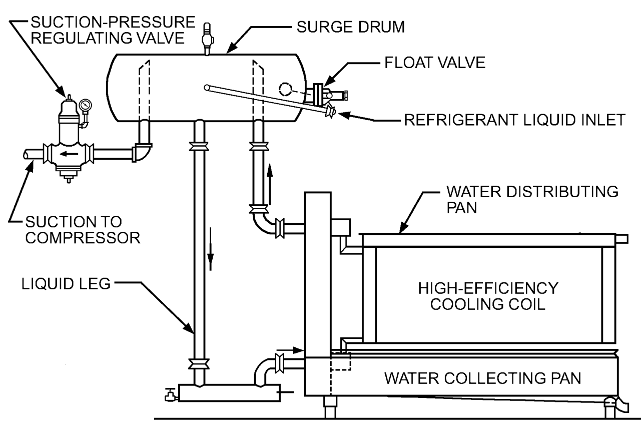

Fig. 4 Baudelot Cooler

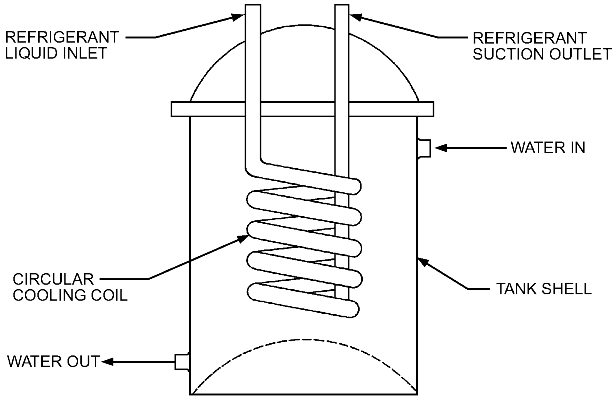

Fig. 5 Shell-and-Coil Cooler

Shell-and-Coil

A shell-and-coil cooler is a tank containing the fluid to be cooled with a simple coiled tube used to cool the fluid. This type of cooler has the advantage of cold fluid storage to offset peak loads. In some models, the tank can be opened for cleaning. Most applications are at low capacities (e.g., for bakeries, for photographic laboratories, and to cool drinking water).

The coiled tube containing the refrigerant can be either inside the tank (Figure 5) or attached to the outside of the tank in a way that allows heat transfer.

HEAT TRANSFER

Heat transfer for liquid coolers can be expressed by the following steady-state heat transfer equation:

$$q = UA\Delta t_m \tag{1}$$

where

q = total heat transfer rate, Btu/h
Δt_m = mean temperature difference, °F
A = heat transfer surface area associated with U, ft²
U = overall heat transfer coefficient, Btu/h·ft²·°F

The area A can be calculated if the geometry of the cooler is known. Chapter 3 of the 2005 *ASHRAE Handbook—Fundamentals* describes the calculation of the mean temperature difference.

This chapter discusses the components of U, but not in depth. U may be calculated by one of the following equations.

Based on inside surface area

$$U = \frac{1}{1/h_i + [A_i/(A_o h_o)] + (t/k)(A_i/A_m) + r_{fi}} \tag{2}$$

Based on outside surface area

$$U = \frac{1}{[A_o/(A_i h_i)] + 1/h_o + (t/k)(A_o/A_m) + r_{fo}} \tag{3}$$

where

h_i = inside heat transfer coefficient based on inside surface area, Btu/h·ft²·°F
h_o = outside heat transfer coefficient based on outside surface area, Btu/h·ft²·°F
A_o = outside heat transfer surface area, ft²
A_i = inside heat transfer surface area, ft²
A_m = mean heat transfer area of metal wall, ft²
k = thermal conductivity of heat transfer material, Btu/h·ft·°F
t = thickness of heat transfer surface (tube wall thickness), ft
r_{fi} = fouling factor of fluid side based on inside surface area, ft²·h·°F/Btu
r_{fo} = fouling factor of fluid side based on outside surface area, ft²·h·°F/Btu

Note: If fluid is on inside, multiply r_{fi} by A_o/A_i to find r_{fo}.
If fluid is on outside, multiply r_{fo} by A_i/A_o to find r_{fi}.

These equations can be applied to incremental sections of the heat exchanger to include local effects on the value of U, and then the increments summed to obtain a more accurate design.

Heat Transfer Coefficients

The refrigerant-side coefficient usually increases with (1) an increase in cooler load, (2) a decrease in suction superheat, (3) a decrease in oil concentration, or (4) an increase in saturated suction temperature. The amount of increase or decrease depends on the type of cooler. Schlager et al. (1989) and Zürcher et al. (1998) discuss the effects of oil in direct-expansion coolers. Flooded coolers have a relatively small change in heat transfer coefficient as a result of a change in load, whereas a direct-expansion cooler shows a significant increase with an increase in load. A Wilson plot of test data (Briggs and Young 1969; McAdams 1954) can show actual values for the refrigerant-side coefficient of a given cooler design. Collier and Thome (1994), Thome (1990, 2003), and Webb (1994) provide additional information on predicting refrigerant-side heat transfer coefficients.

The fluid-side coefficient is determined by cooler geometry, fluid flow rate, and fluid properties (viscosity, specific heat, thermal conductivity, and density) (Palen and Taborek 1969; Wolverine Tube 1984). For a given fluid, the fluid-side coefficient increases with fluid flow rate because of increased turbulence and with fluid temperature because of improvement of fluid properties as temperature increases.

The heat transfer coefficient in direct-expansion and flooded coolers increases significantly with fluid flow. The effect of flow is smaller for Baudelot and shell-and-coil coolers.

An enhanced heat transfer surface can increase the heat transfer coefficient of coolers in the following ways:

- It increases heat transfer area, thereby increasing the overall heat transfer rate and reducing the thermal resistance of fouling, even if the refrigerant-side heat transfer coefficient is unchanged.
- Where flow of fluid or refrigerant is low, it increases turbulence at the surface and mixes fluid at the surface with fluid away from the surface. For stratified internal flows of refrigerants, it may convert the flow to complete wetting of the tube perimeter.
- In flooded coolers, an enhanced refrigerant-side surface may provide more and better nucleation points to promote boiling of refrigerant.

Pais and Webb (1991) and Thome (1990) describe many enhanced surfaces used in flooded coolers. The enhanced surface geometries provide substantially higher boiling coefficients than integral finned tubes. Nucleate pool boiling data are provided by Webb and Pais (1991). The boiling process in the tube bundle of a flooded cooler may be enhanced by forced-convection effects. This is basically an additive effect, in which the local boiling coefficient is the sum of the nucleate boiling coefficient and the forced-convection effect. Webb et al. (1989) describe an empirical method to predict flooded cooler performance, recommending row-by-row calculations.

Based on this model, Webb and Apparao (1990) present the results of calculations using a computer program. The results show some performance differences of various internal and external surface geometries. As an example, Figure 6 shows the contribution of nucleate pool boiling to the overall refrigerant heat transfer coefficient for an integral finned tube and an enhanced tube as a function of the tube row. Forced convection predominates with the integral finned tube.

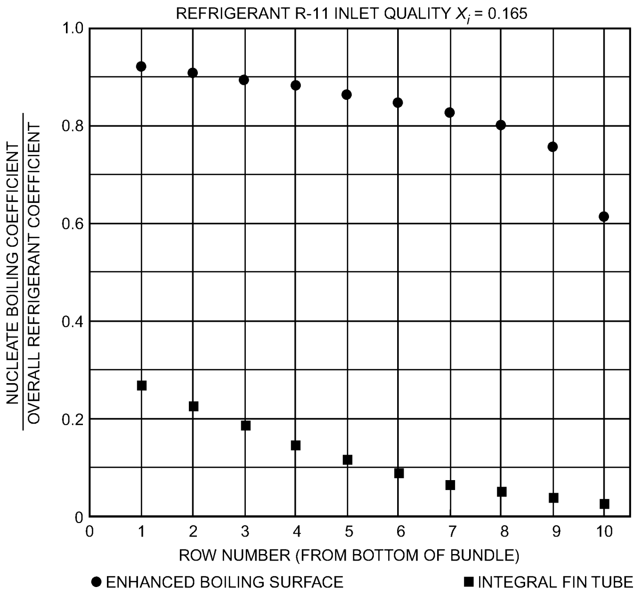

Fig. 6 Nucleate Boiling Contribution to Total Refrigerant Heat Transfer

ASHRAE research projects RP-725 (Chyu 1995) and RP-668 (Moeykens et al. 1995) studied spray evaporation performance in ammonia and halocarbon refrigerant systems. Moeykens et al. investigated shell-side heat transfer performance for commercially available enhanced surface tubes in a spray evaporation environment. The study determined that spray evaporation heat transfer can yield shell-side heat transfer coefficients equal to or greater than those found with enhanced nucleate boiling surface tubes in the flooded boiling environment. Moeykens and Pate (1996) describe an enhancement to shell-side heat transfer performance generated with small concentrations of oil (<2.5%) in spray evaporation. They attribute the improvement to foaming, which enhances heat transfer performance in the upper rows of large tube bundles operating in flooded boiling mode.

Gupte and Webb (1995a, 1995b) investigated convective vaporization in triangular enhanced tube bundles. They proposed a modified Chen superposition model that predicts the overall convective/vaporization coefficient as the sum of the single tube nucleate pool boiling coefficient and a weighted contribution of a single-phase convective coefficient.

Casciaro and Thome (2001a, 2001b) provide a recent comprehensive review of thermal design methods for flooded evaporators. For direct-expansion evaporators, Kattan et al. (1998) proposed a flow-pattern-based method that includes effects of flow stratification at low flow rates and onset of dryout at high vapor qualities. Zürcher et al. (1998) in ASHRAE research project RP-800 added a method to predict the adverse effect of oil on local flow boiling heat transfer coefficients to the Kattan-Thome-Favrat model.

Fouling Factors

Over time, most fluids foul the fluid-side heat transfer surface, reducing the cooler's overall heat transfer coefficient. If fouling is expected to be a problem, a mechanically cleanable cooler should be used, such as a flooded, Baudelot, or cleanable direct-expansion tube-in-tube cooler. Direct-expansion shell-and-tube, shell-and-coil, and brazed-plate coolers can be cleaned chemically. Flooded coolers and direct-expansion tube-in-tube coolers with enhanced fluid-side heat transfer surfaces tend to be self-cleaning because of high fluid turbulence, so a smaller fouling factor can probably be used for these coolers. Water quality in closed chilled-water loops has been studied as part of ASHRAE-sponsored research (RP-560). Haider et al. (1991) found little potential for fouling in such systems in a field survey. Experimental work with various tube geometries by Haider et al. (1992) confirmed that negligible fouling occurs in closed-loop evaporator tubes at 3 to 5 fps and 7 fps water velocities. ARI *Standard* 480 discusses fouling calculations.

The refrigerant side of the cooler is not subject to fouling, and a fouling factor need not be included for that side.

Wall Resistance

Typically, the t/k term in Equations (2) and (3) is negligible. However, with low thermal conductivity material or thick-walled tubing, it may become significant. Refer to Chapter 3 of the 2005 *ASHRAE Handbook—Fundamentals* and to Chapter 38 of this volume for further details.

PRESSURE DROP

Fluid Side

Pressure drop is usually minimal in Baudelot and shell-and-coil coolers but must be considered in direct-expansion and flooded coolers. Both direct-expansion and flooded coolers rely on turbulent fluid flow to improve heat transfer. This turbulence is obtained at the expense of pressure drop.

For air-conditioning, pressure drop is commonly limited to 10 psi to keep pump size and energy cost reasonable. For flooded coolers, see Chapter 38 for a discussion of pressure drop for flow in tubes. Pressure drop for fluid flow in shell-and-tube direct-expansion coolers depends greatly on tube and baffle geometry. The following equation projects the change in pressure drop caused by a change in flow:

$$\text{New pressure drop} = \text{Original pressure drop}\left[\frac{\text{New rate}}{\text{Original rate}}\right]^{1.8} \quad (4)$$

Refrigerant Side

The refrigerant-side pressure drop must be considered for direct-expansion, shell-and-coil, brazed-plate, and (sometimes) Baudelot coolers. When there is a pressure drop on the refrigerant side, the refrigerant inlet and outlet pressures and corresponding saturated temperature are different. This difference changes the mean temperature difference, which affects the total heat transfer rate. If pressure drop is high, expansion valve operation may be affected because of reduced pressure drop across the valve. This pressure drop varies, depending on the refrigerant used, operating temperature, and type of tubing. For flooded evaporators, Casciaro and Thome (2001b) summarize prediction methods as part of ASHRAE research project RP-1089. For direct-expansion evaporators, Ould Didi et al. (2002) describe seven prediction methods, compare them to data for five refrigerants, and provide recommendations on the best choice.

VESSEL DESIGN

Mechanical Requirements

Pressure vessels must be constructed and tested under the rules of national, state, and local codes. The ASME *Boiler and Pressure Vessel Code*, Section VIII, gives guidance on rules and exemptions.

The more common applicable codes and standards are as follows:

1. ARI *Standard* 480 covers industry criteria for standard equipment, standard safety provisions, marking, and recommended rating requirements.
2. ASHRAE *Standard* 24 covers recommended testing methods for measuring liquid cooler capacity.

3. ASHRAE *Standard* 15 involves specific 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. This code requires pressure-limiting and pressure-relief devices on refrigerant-containing systems, as applicable, and defines setting and capacity requirements for these devices.
4. ASME *Boiler and Pressure Vessel Code*, Section VIII, Unfired Pressure Vessels, covers safety aspects of design and construction. Most states require coolers to meet ASME requirements if they fall within the scope of the ASME code. Some of the exceptions from meeting the ASME requirements are as follows:
   - Cooler shell inner diameter (ID) is 6 in. or less.
   - Pressure is 15 psig or less.
   - The fluid portion of the cooler need not be built to the requirements of the ASME code if the fluid is water, the design pressure does not exceed 300 psig, and the design temperature does not exceed 210°F.

   Coolers meeting ASME code requirements have an ASME stamp, which is a U or UM inside a four-leaf clover. The U can be used for all coolers, and the UM can be used for small coolers.
5. Underwriters Laboratories (UL) *Standard* 207 covers specific design criteria, use of materials, testing, and initial approval by Underwriters Laboratories. A cooler with the ASME U or UM stamp does not require UL approval.

Design Pressure. On the refrigerant side, design pressure should be chosen per ASHRAE *Standard* 15. Standby temperature and the temperature encountered during shipping of chillers with a refrigerant charge should also be considered.

Required fluid-side (water-side) pressure varies, depending largely on (1) static pressure, (2) pump pressure, (3) transients caused by pump start-up, and (4) valve closing.

Chemical Requirements

The following chemical requirements are given by NACE (1985) and Perry and Green (2007).

R-717 (Ammonia). Carbon steel and cast iron are the most widely used materials for ammonia systems. Stainless steel alloys are satisfactory but more costly. Copper and high-copper alloys are avoided because they are attacked by ammonia when moisture is present. Aluminum and aluminum alloys may be used with caution.

Halocarbon Refrigerants. Almost all common metals and alloys are used satisfactorily with these refrigerants. Exceptions where water may be present include magnesium and aluminum alloys containing more than 2% magnesium. Zinc is not recommended for use with R-113; it is more chemically reactive than other common construction metals and, therefore, it is usually avoided when other halogenated hydrocarbons are used as refrigerants. Under some conditions with moisture present, halocarbon refrigerants form acids that attack steel and even nonferrous metals. This problem does not commonly occur in properly cleaned and dehydrated systems. ASHRAE *Standard* 15, paragraph 9.1.2, states that “aluminum, zinc, magnesium, or their alloys shall not be used in contact with methyl chloride,” or magnesium alloys with any halogenated refrigerant.

Water. Relatively pure water can be satisfactory with both ferrous and nonferrous metals. Brackish water, seawater, and some river water are quite corrosive to iron and steel and also to copper, aluminum, and many alloys of these metals. A reputable water consultant who knows the local water condition should be contacted. Chemical treatment by pH control, inhibitor applications, or both may be required. Where this is not feasible, more noble construction material or special coatings must be used. Aluminum should not be used in the presence of other metals in water circuits.

Brines. Ferrous metal and a few nonferrous alloys are almost universally used with sodium chloride and calcium chloride brines. Copper alloys can be used if adequate quantities of sodium dichromate are added and caustic soda is used to neutralize the solution. Even with ferrous metal, brines should be treated periodically to hold pH near neutral.

Ethylene glycol and propylene glycol are stable compounds that are less corrosive than chloride brines.

Electrical Requirements

When the fluid being cooled is electrically conductive, the system must be grounded to prevent electrochemical corrosion.

APPLICATION CONSIDERATIONS

Refrigerant Flow Control

Direct-Expansion Coolers. The constant superheat thermal expansion valve is the most common control used, located directly upstream of the cooler. A thermal bulb strapped to the suction line leaving the cooler senses refrigerant temperature. The valve can be adjusted to produce a constant suction superheat during steady operation. If refrigerant pressure drop between the expansion valve and the thermal bulb is significant, an externally equalized expansion valve should be used.

The thermal expansion valve adjustment is commonly set at a suction superheat of 8 to 10°F, which is sufficient to ensure that liquid is not carried into the compressor. Direct-expansion cooler performance is affected greatly by superheat setting. Reduced superheat improves cooler performance; thus, suction superheat should be set as low as possible while avoiding liquid carryover to the compressor.

Flooded Coolers. As the name implies, flooded coolers must have good liquid refrigerant coverage of the tubes to achieve good performance. Liquid level control in a flooded cooler becomes the principal issue in flow control. Some systems are designed critically charged, so that when all the liquid refrigerant is delivered to the cooler, it is just enough for good tube coverage. In these systems, an orifice is often used as the throttling device between condenser and cooler.

Level-sensing devices are another option. Typically, a high-side float valve or level-sensing system can meter flow to the cooler at a controlled rate based on the condenser liquid level. For more direct control of liquid level in the cooler, a low-side float valve or level-sensing system can be used to control flow of entering refrigerant based on the level in the cooler itself. Because of the volatile nature of boiling inside the flooded cooler and the potential for false level readings, consideration must be given to design and installation of a low-side liquid level sensing device.

Freeze Prevention

Freeze prevention must be considered for coolers operating near the fluid’s freezing point. In some coolers, freezing causes extensive damage. Two methods can be used for freeze protection: (1) hold saturated suction pressure above the fluid freezing point or (2) shut the system off if fluid temperature approaches the freezing point.

A suction-pressure regulator can hold the saturated suction pressure above the fluid’s freezing point. A low-pressure cutout can shut the system off before the saturated suction pressure drops to below the freezing point of the fluid. The leaving fluid temperature can be monitored to cut the system off before a danger of freezing, usually about 4°F above the fluid freezing temperature. It is recommended that both methods be used.

Baudelot, shell-and-coil, brazed-plate, and direct-expansion shell-and-tube coolers are all somewhat resistant to damage caused by freezing and ideal for applications where freezing may be a problem.

If a cooler is installed in an unconditioned area, possible freezing caused by low ambient temperature must be considered. If the cooler is used only when ambient temperature is above freezing, the fluid should be drained from the cooler for cold weather. Alternatively, if the cooler is used year-round, the following methods can be used to prevent freezing:

- Heat tape or other heating device to keep cooler above freezing
- For water, adding an appropriate amount of ethylene glycol
- Continuous pump operation

Oil Return

Most compressors discharge a small percentage of oil in the discharge gas. This oil mixes with condensed refrigerant in the condenser and flows to the cooler. Because the oil is nonvolatile, it does not evaporate and may collect in the cooler.

In direct-expansion coolers, gas velocity in the tubes and suction gas header is usually sufficient to carry oil from the cooler into the suction line. From there, with proper piping design, it can be carried back to the compressor. At light load and low temperature, oil may gather in the superheat section of the cooler, detracting from performance. For this reason, operating refrigerant circuits at light load for long periods should be avoided, especially under low-temperature conditions. Some oil hold-up threshold measurements were presented by Zürcher et al. (1998) in ASHRAE research project RP-800.

In flooded coolers, vapor velocity above the tube bundle is usually insufficient to return oil up the suction line, and oil tends to accumulate in the cooler. With time, depending on the compressor oil loss rate, oil concentration in the cooler may become large. When concentration exceeds about 1%, heat transfer performance may be adversely affected if enhanced tubing is used.

It is common in flooded coolers to take some oil-rich liquid and return it to the compressor on a continuing basis, to establish a rate of return equal to the compressor oil loss rate.

Maintenance

Cooler maintenance centers around (1) safety and (2) cleaning the fluid side. The cooler should be inspected periodically for any weakening of its pressure boundaries. Visual inspection for corrosion, erosion, and any deformities should be included, and any pressure relief device should also be inspected. The insurer of the cooler may require regular inspection. If the fluid side is subjected to fouling, it may require periodic cleaning by either mechanical or chemical means. The manufacturer or a service organization experienced in cooler maintenance should have details for cleaning.

Insulation

A cooler operating at a saturated suction temperature lower than the ambient-air dew point should be insulated to prevent condensation. Direct-expansion coolers installed where the ambient temperature may drop below the process fluid's freezing point should also be insulated and wrapped with heat tape, to prevent the fluid from freezing and damaging the cooler during *off* periods. Chapter 23 of the 2005 *ASHRAE Handbook—Fundamentals* describes insulation in more detail.

REFERENCES

ARI. 2001. Remote type refrigerant-cooled liquid coolers. *Standard* 480-01. Air-Conditioning and Refrigeration Institute, Arlington, VA.

ASHRAE. 2000. Methods of testing for rating liquid coolers. ANSI/ASHRAE *Standard* 24-2000.

ASHRAE. 2007. Safety standards for refrigeration systems. ANSI/ASHRAE *Standard* 15-2007.

ASME. 2007. Rules for construction of pressure vessels. ANSI/ASME *Boiler and Pressure Vessel Code*, Section VIII. American Society of Mechanical Engineers, New York.

Ayub, Z.H. and S.A. Jones. 1987. Tubeside erosion/corrosion in heat exchangers. *Heating/Piping/Air Conditioning* (December):81.

Briggs, D.E. and E.H. Young. 1969. Modified Wilson plot techniques for obtaining heat transfer correlations for shell and tube heat exchangers. *Chemical Engineering Symposium Series* 65(92):35-45.

Casciaro, S. and J.R. Thome. 2001a. Thermal performance of flooded evaporators, Part 1: Review of boiling heat transfer studies. *ASHRAE Transactions* 107(1):903-918.

Casciaro, S. and J.R. Thome. 2001b. Thermal performance of flooded evaporators, Part 2: Review of void fraction, two-phase pressure drop and flow pattern studies. *ASHRAE Transactions* 107(1):919-930.

Chyu, M. 1995. Nozzle-sprayed flow rate distribution on a horizontal tube bundle. *ASHRAE Transactions* 101(2):443-453.

Collier, J.G. and J.R. Thome. 1994. *Convective boiling and condensation*, 3rd ed. Oxford University Press.

Gupte, N.S. and R.L. Webb. 1995a. Shell-side boiling in flooded refrigerant evaporators—Part I: Integral finned tubes. *International Journal of HVAC&R Research* (now *HVAC&R Research*) 1(1):35-47.

Gupte, N.S. and R.L. Webb. 1995b. Shell-side boiling in flooded refrigerant evaporators—Part II: Enhanced tubes. *International Journal of HVAC&R Research* (now *HVAC&R Research*) 1(1):48-60.

Haider, S.I., R.L. Webb, and A.K. Meitz. 1991. A survey of water quality and its effect on fouling in flooded water chiller evaporators. *ASHRAE Transactions* 97(1):55-67.

Haider, S.I., R.L. Webb, and A.K. Meitz. 1992. An experimental study of tube-side fouling resistance in water-chiller-flooded evaporators. *ASHRAE Transactions* 98(2):86-103.

Kattan, N., J.R. Thome, and D. Favrat. 1998. Flow boiling in horizontal tubes, Part 3: Development of a new heat transfer model based on flow patterns. *Journal of Heat Transfer* 120(1):156-165.

McAdams, W.H. 1954. *Heat transmission*, 3rd ed. McGraw-Hill, New York.

Moeykens, S.A., B.J. Newton, and M.B. Pate. 1995. Effects of surface enhancement, film-feed supply rate, and bundle geometry on spray evaporation heat transfer performance. *ASHRAE Transactions* 101(2):408-419.

Moeykens, S.A. and M.B. Pate. 1996. Effects of lubricant on spray evaporation heat transfer performance of R-134a and R-22 in tube bundles. *ASHRAE Transactions* 102(1):410-426.

NACE. 1985. *Corrosion data survey—Metal section*, 6th ed. D.L. Graver, ed. National Association of Corrosion Engineers, Houston.

Ould Didi, M.B., N. Kattan, and J.R. Thome. 2002. Prediction of two-phase pressure gradients of refrigerants in horizontal tubes. *International Journal of Refrigeration* 25(7):935-947.

Pais, C. and R.L. Webb. 1991. Literature survey of pool boiling on enhanced surfaces. *ASHRAE Transactions* 97(1):79-89.

Palen, J.W. and J. Taborek. 1969. Solution of shell side pressure drop and heat transfer by stream analysis method. *Chemical Engineering Progress Symposium Series* 65(92).

Perry, R.H. and D.W. Green. 2007. *Perry's chemical engineers handbook*, 8th ed. McGraw-Hill, New York.

Schlager, L.M., M.B. Pate, and A.E. Bergles. 1989. A comparison of 150 and 300 SUS oil effects on refrigerant evaporation and condensation in a smooth tube and a micro-fin tube. *ASHRAE Transactions* 95(1).

Sturley, R.A. 1975. Increasing the design velocity of water and its effect on copper tube heat exchangers. *Paper* 58, International Corrosion Forum, Toronto, Canada.

Thome, J.R. 1990. *Enhanced boiling heat transfer*. Hemisphere, New York.

Thome, J.R. 2003. Boiling. In *Heat transfer handbook*, pp. 635-717. A. Bejan and A.D. Krause, eds. Wiley Interscience, New York.

UL. 2001. Refrigerant-containing components and accessories, nonelectrical. *Standard* 207-01. Underwriters Laboratories, Northbrook, IL.

Webb, R.L. 1994. *Principles of enhanced heat transfer*. John Wiley & Sons, New York.

Webb, R.L. and T. Apparao. 1990. Performance of flooded refrigerant evaporators with enhanced tubes. *Heat Transfer Engineering* 11(2):29-43.

Webb, R.L. and C. Pais. 1991. Pool boiling data for five refrigerants on three tube geometries. *ASHRAE Transactions* 97(1):72-78.

Webb, R.L., K.-D. Choi, and T. Apparao. 1989. A theoretical model for prediction of the heat load in flooded refrigerant evaporators. *ASHRAE Transactions* 95(1):326-338.

Wolverine Tube, Inc. 2001. *Wolverine engineering data book II*. http://www.wlv.com/products/databook/databook.pdf.

Zürcher, O., J.R. Thome, and D. Favrat. 1998. Intube flow boiling of R-407C and R-407C/oil mixtures, Part II: Plain tube results and predictions (RP-800). *International Journal of HVAC&R Research* (now *HVAC&R Research*) 4(4):373-399.