CHAPTER 43

COMPONENT BALANCING IN REFRIGERATION SYSTEMS

THIS chapter describes methods and components used in balancing a primary refrigeration system. A refrigerant is a fluid used for heat transfer in a refrigeration system. The fluid absorbs heat at a low temperature and pressure and transfers heat at a higher temperature and pressure. Heat transfer can involve either a complete or partial change of state in the case of a primary refrigerant. Energy transfer is a function of the heat transfer coefficients; temperature differences; and amount, type, and configuration of the heat transfer surface and, hence, the heat flux on either side of the heat transfer device.

REFRIGERATION SYSTEM

A typical basic direct-expansion refrigeration system includes an evaporator, which vaporizes incoming refrigerant as it absorbs heat, increasing the refrigerant's heat content or enthalpy. A compressor pulls vapor from the evaporator through suction piping and compresses the refrigerant gas to a higher pressure and temperature. The refrigerant gas then flows through the discharge piping to a condenser, where it is condensed by rejecting its heat to a coolant (e.g., other refrigerants, air, water, or air/water spray). The condensed liquid is supplied to a device that reduces pressure, cools the liquid by flashing vapor, and meters the flow. The cooled liquid is returned to the evaporator. For more information on the basic refrigeration cycle, see Chapter 1 of the 2005 *ASHRAE Handbook—Fundamentals*.

Gas compression theoretically follows a line of constant entropy. In practice, adiabatic compression cannot occur because of friction and other inefficiencies of the compressor. Therefore, the actual compression line deviates slightly from the theoretical. Power to the compressor shaft is added to the refrigerant, and compression increases the refrigerant's pressure, temperature, and enthalpy.

In applications with a large compression ratio (e.g., low-temperature freezing, multitemperature applications), multiple compressors in series are used to completely compress the refrigerant gas. In multistage systems, interstage desuperheating of the lower-stage compressor's discharge gas protects the high-stage compressor. Liquid refrigerant can also be subcooled at this interstage condition and delivered to the evaporator for improved efficiencies.

An intermediate-temperature condenser can serve as a cascading device. A low-temperature, high-pressure refrigerant condenses on one side of the cascade condenser surface by giving up heat to a low-pressure refrigerant that is boiling on the other side of the surface. The vapor produced transfers energy to the next compressor (or compressors); heat of compression is added and, at a higher pressure, the last refrigerant is condensed on the final condenser surface.

Heat is rejected to air, water, or water spray. Saturation temperatures of evaporation and condensation throughout the system fix the terminal pressures against which the single or multiple compressors must operate.

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Generally, the smallest differential between saturated evaporator and saturated condensing temperatures results in the lowest energy requirement for compression. Liquid refrigerant cooling or subcooling should be used where possible to improve efficiencies and minimize energy consumption.

Where intermediate pressures have not been specifically set for system operation, the compressors automatically balance at their respective suction and discharge pressures as a function of their relative displacements and compression efficiencies, depending on load and temperature requirements. This chapter covers the technique used to determine the balance points for a typical brine chiller, but the theory can be expanded to apply to single- and two-stage systems with different types of evaporators, compressors, and condensers.

COMPONENTS

Evaporators may have flooded, direct-expansion, or liquid overfeed cooling coils with or without fins. Evaporators are used to cool air, gases, liquids, and solids; condense volatile substances; and freeze products.

Ice-builder evaporators accumulate ice to store cooling energy for later use. Embossed-plate evaporators are available (1) to cool a falling film of liquid; (2) to cool, condense, and/or freeze out volatile substances from a fluid stream; or (3) to cool or freeze a product by direct contact. Brazed- and welded-plate fluid chillers can be used to improve efficiencies and reduce refrigerant charge.

Ice, wax, or food products are frozen and scraped from some freezer surfaces. Electronic circuit boards, mechanical products, or food products (where permitted) are flash-cooled by direct immersion in boiling refrigerants. These are some of the diverse applications demanding innovative configurations and materials that perform the function of an evaporator.

Compressors can be positive-displacement, reciprocating-piston, rotary-vane, scroll, single and double dry and lubricant-flooded screw devices, and single-stage or multistage centrifugals. They can be operated in series or in parallel with each other, in which case special controls may be required.

Drivers for compressors can be direct hermetic, semihermetic, or open with mechanical seals on the compressor. In hermetic and semihermetic drives, motor inefficiencies are added to the refrigerant as heat. Open compressors are driven with electric motors, fuel-powered reciprocating engines, or steam or gas turbines. Intermediate gears, belts, and clutch drives may be included in the drive.

Cascade condensers are used with high-pressure, low-temperature refrigerants (such as R-23) on the bottom cycle, and high-temperature refrigerants (such as R-22, azeotropes, and refrigerant blends or zeotropes) on the upper cycle. Cascade condensers are manufactured in many forms, including shell-and-tube, embossed plate, submerged, direct-expansion double coils, and brazed or welded plate heat exchangers. The high-pressure refrigerant from the compressor(s) on the lower cycle condenses at a given intermediate temperature. A separate, lower-pressure refrigerant evaporates on the other side of the surface at a somewhat lower temperature. Vapor formed from the second refrigerant is compressed by the higher-cycle compressor(s) until it can be condensed at an elevated temperature.

Desuperheating suction gas at intermediate pressures where multistage compressors balance is essential to reduce discharge temperatures of the upper-stage compressor. Desuperheating also helps reduce oil carryover and reduces energy requirements. Subcooling improves the net refrigeration effect of the refrigerant supplied to the next-lower-temperature evaporator and reduces system energy requirements. The total heat is then rejected to a condenser.

Subcoolers can be of shell-and-tube, shell-and-coil, welded-plate, or tube-in-tube construction. Friction losses reduce the liquid pressure that feeds refrigerant to an evaporator. Subcoolers are used to improve system efficiency and to prevent refrigerant liquid from flashing because of pressure loss caused by friction and the vertical rise in lines. Refrigerant blends (zeotropes) can take advantage of temperature glide on the evaporator side with a direct-expansion-in-tube serpentine or coil configuration. In this case, temperature glide from the bubble point to the dew point promotes efficiency and lower surface requirements for the subcooler. A flooded shell for the evaporating refrigerant requires use of only the higher dew-point temperature.

Lubricant coolers remove friction heat and some of the superheat of compression. Heat is usually removed by water, air, or a direct-expansion refrigerant.

Condensers that reject heat from the refrigeration system are available in many standard forms, such as water- or brine-cooled shell-and-tube, shell-and-coil, plate-and-frame, or tube-in-tube condensers; water cascading or sprayed over plate or coil serpentine models; and air-cooled, fin-coil condensers. Special heat pump condensers are available in other forms, such as tube-in-earth and submerged tube bundle, or as serpentine and cylindrical coil condensers that heat baths of boiling or single-phase fluids.

SELECTING DESIGN BALANCE POINTS

Refrigeration load at each designated evaporator pressure, refrigerant properties, liquid refrigerant temperature feeding each evaporator, and evaporator design determine the required flow rate of refrigerant in a system. The additional flow rates of refrigerant that provide refrigerant liquid cooling, desuperheating, and compressor lubricant cooling, where used, depend on the established liquid refrigerant temperatures and intermediate pressures.

For a given refrigerant and flow rate, the suction line pressure drop, suction gas temperature, pressure ratio and displacement, and volumetric efficiency determine the required size and speed of rotation for a positive displacement compressor. At low flow rates, particularly at very low temperatures and in long suction lines, heat gain through insulation can significantly raise the suction temperature. Also, at low flow rates, a large, warm compressor casing and suction plenum can further heat the refrigerant before it is compressed. These heat gains increase the required displacement of a compressor. The compressor manufacturer must recommend the superheating factors to apply. The final suction gas temperature from suction line heating is calculated by iteration.

Another concern is that more energy is required to compress refrigerant to a given condenser pressure as the suction gas gains more superheat. This can be seen by examining a pressure-enthalpy diagram for a given refrigerant such as R-22, which is shown in Figure 2 in Chapter 20 of the 2005 *ASHRAE Handbook—Fundamentals*. As suction superheat increases along the horizontal axis, the slopes of the constant entropy lines of compression decrease. This means that a greater enthalpy change must occur to produce a given pressure rise. For a given flow, then, the power required for compression is increased. With centrifugal compressors, pumping capacity is related to wheel diameter and speed, as well as to volumetric flow and acoustic velocity of the refrigerant at the suction entrance. If the thermodynamic pressure requirement becomes too great for a given speed and volumetric flow, the centrifugal compressor experiences periodic backflow and surging.

Figure 1 shows an example system of curves representing the maximum refrigeration capacities for a brine chilling plant. The example shows only one type of positive-displacement compressor using a water-cooled condenser in a single-stage system operating at a steady-state condition. The figure is a graphical method of expressing the first law of thermodynamics with an energy balance applied to a refrigeration system.

One set of nearly parallel curves (A) represents cooler capacity at various brine temperatures versus saturated suction temperature (a pressure condition) at the compressor, allowing for suction line pressure drops. The (B) curves represent compressor capacities as the saturated suction temperature varies and the saturated condenser temperature (a pressure condition) varies. The (C) curves represent heat transferred to the condenser by the compressor. It is calculated by adding the heat input at the evaporator to the energy imparted to the refrigerant by the compressor. The (D) curves represent condenser performance at various saturated condenser temperatures as the inlet temperature of a fixed quantity of cooling water is varied.

The (E) curves represent the combined compressor and condenser performance as a “condensing unit” at various saturated suction temperatures for various cooling water temperatures. These curves were cross plotted from the (C) and (D) curves back to the set of brine cooler curves as indicated by the dashed construction lines for the 80 and 92°F cooling water temperatures. Another set of construction lines (not shown) would be used for the 86°F cooling water. The number of construction lines used can be increased as necessary to adequately define curvature (usually no more than three per condensing-unit performance line).

The intersections of curves (A) and (E) represent the maximum capacities for the entire system at those conditions. For example, these curves show that the system develops 150 tons of refrigeration when cooling the brine to 44°F at 36.4°F (saturated) suction and using 80°F cooling water. At 92°F cooling water, capacity drops to 134.5 tons if the required brine temperature is 42°F and the required saturated suction temperature is 35°F. The corresponding saturated condensing temperature for 42°F brine with an accompanying suction temperature of 36.4°F and using 80°F water is graphically projected on the brine cooler line with a capacity of 150 tons of refrigeration to meet a newly constructed 36.4°F saturated suction temperature line (parallel to the 34°F and 37°F lines). At this junction, draw a horizontal line to intersect the vertical saturated condensing temperature scale at 93.5°F. The condenser heat rejection is apparent from the (C) curves at a given balance point.

The equation at the bottom of Figure 1 may be used to determine the shaft horsepower (BHP) required at the compressor for any given balance point. A sixth set of curves could be drawn to indicate the power requirement as a function of capacity versus saturated suction and saturated condensing temperatures.

The same procedure can be repeated to calculate cascade system performance. Rejected heat at the cascade condenser would be treated as the chiller load in making a cross plot of the upper-cycle, high-temperature refrigeration system.

For cooling air at the evaporator(s) and for condenser heat rejection to ambient air or evaporative condensers, use the same procedures. Performance of coils and expansion devices such as thermostatic expansion valves may also be graphed, once the basic concept of heat and mechanical energy input equivalent combinations is recognized. Chapter 1 of the 2005 *ASHRAE Handbook—Fundamentals* has further information.

This method finds the natural balance points of compressors operating at their maximum capacities. For multiple-stage loads at several specific operating temperatures, the usual way of controlling compressor capacities is with a suction pressure control and

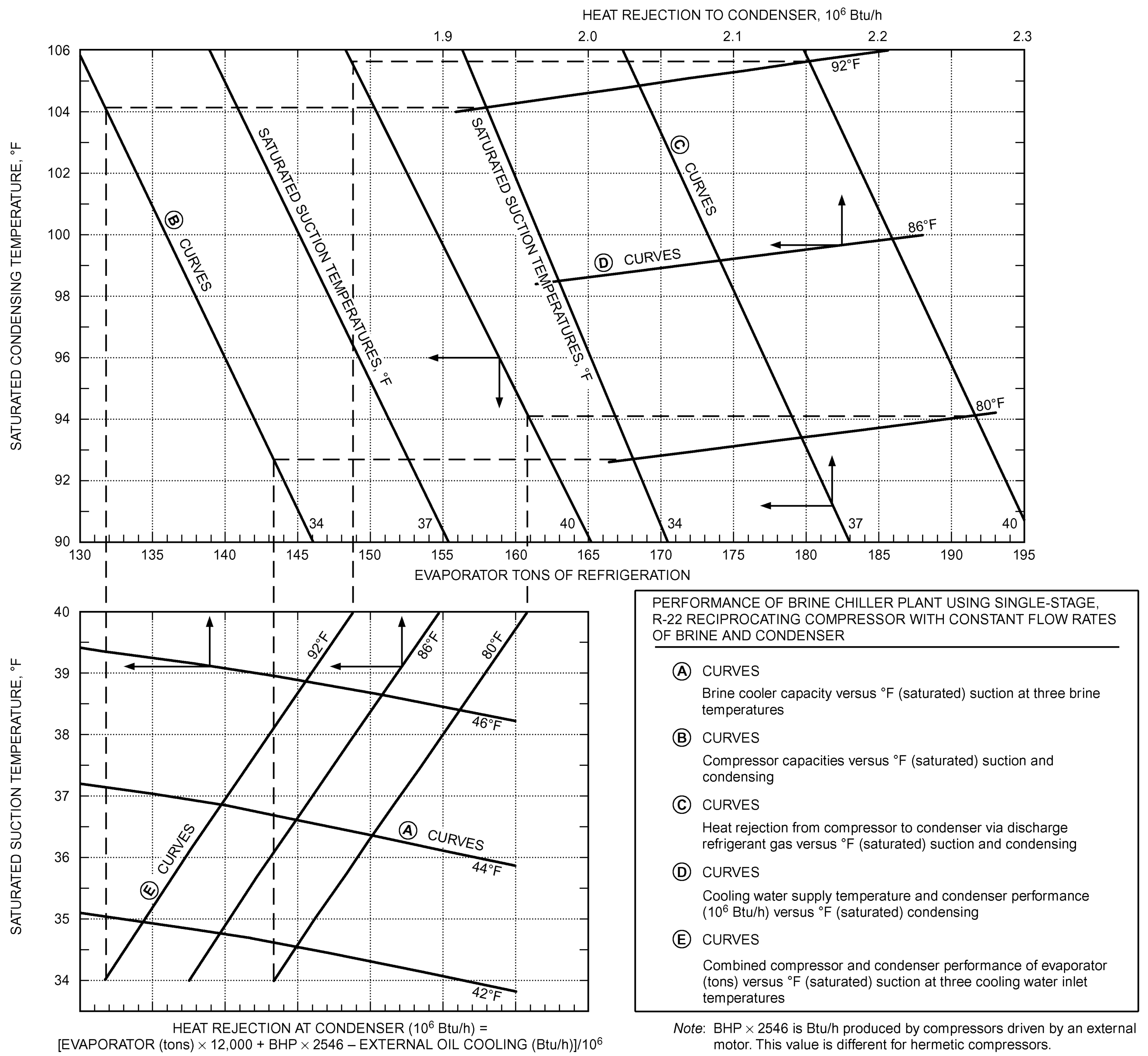

Fig. 1 Brine Chiller Balance Curve

compressor capacity control device. This control accommodates any mismatch in pumping capabilities of multistage compressors, instead of allowing each compressor to find its natural balance point.

Computer programs could be developed to determine balance points of complex systems. However, because applications, components, and piping arrangements are so diverse, many designers use available capacity performance data from vendors and plot balance points for chosen components. Individual computer programs may be available for specific components, which speeds the process.

ENERGY AND MASS BALANCES

A systematic, point-to-point flow analysis of the system (including piping) is essential in accounting for pressure drops and heat gains, particularly in long suction lines. Air-cooled condensers, in particular, can have large pressure drops, which must be included in the analysis to estimate a realistic balance. Making a flow diagram of the system with designated pressures and temperatures, loads, enthalpies, flow rates, and energy requirements helps identify all important factors and components.

An overall energy and mass balance for the system is also essential to avoid mistakes. The overall system represented by the complete flow diagram should be enclosed by a dotted line envelope. Any energy inputs to or outputs from the system that directly affect the heat content of the refrigerant itself should cross the dotted envelope line and must enter the energy balance equations. Accurate estimates of the ambient heat gains through insulation and heat losses from discharge lines where they are significant improve the comprehensiveness of the energy balance and accuracy of equipment selections.

Cascade condenser loads and subcooler or desuperheating loads carried by a refrigerant are internal to the system and thus do not enter into the overall energy balance. The total energy entering the system equals the total energy leaving the system. If calculations do not show an energy balance within reasonable tolerances for the accuracy of data used, then an omission occurred or a mistake was made and should be corrected.

The dotted envelope technique can be applied to any section of the system, but all energy transmissions must be included in the equations, including the enthalpies and mass flow rates of streams that cross the dotted line.

SYSTEM PERFORMANCE

Rarely are sufficient sensors and instrumentation devices available, nor are conditions proper at a given job site to allow calculation of a comprehensive, accurate energy balance for an operating system. Water-cooled condensers and oil coolers for heat rejection and the use of electric motor drives, where motor efficiency and power factor curves are available, offer the best hope for estimating the actual performance of the individual components in a system. Evaporator heat loads can be derived from the measured heat rejection and derived mechanical or measured electrical energy inputs. A comprehensive flow diagram assists in a field survey.

Various coolant flow detection devices are available for direct measurement inside a pipe and for measurement from outside the pipe with variable degrees of accuracy. Sometimes flow rates may be estimated by simply weighing or measuring an accumulation of coolant over a brief time interval.

Temperature and pressure measurement devices should be calibrated and be of sufficient accuracy. Calibrated digital scanning devices for comprehensive simultaneous readings are best. Electrical power meters are not always available, so voltage and current at each leg of a motor power connection must be measured. Voltage drops for long power leads must be calculated when the voltage measurement points are far removed from the motor. Motor load versus efficiency and power factor curves must be used to determine motor output to the system.

Gears and belt or chain drives have friction and windage power losses that must be included in any meaningful analysis.

Stack gas flows and enthalpies for engine or gas turbine exhausts as well as air inputs and speeds must be included. In this case, performance curves issued by the vendor must be heavily relied on to estimate the energy input to the system.

Calculating steam turbine performance requires measurements of turbine speed, steam pressures and temperatures, and condensate mass flow coupled with confidence that the vendor's performance curves truly represent the current mechanical condition. Plant personnel normally have difficulty in obtaining operating data at specified performance values.

Heat rejection from air-cooled and evaporative condensers or coolers is extremely difficult to measure accurately because of changing ambient temperatures and the extent and scope of airflow measurements required. Often, one of the most important issues is the wide variation or cycling of process flows, process temperatures, and product refrigeration loads. Hot-gas false loading and compressor continuous capacity modulations complicate any attempt to make a meaningful analysis.

Prediction and measurement of performance of systems using refrigerant blends (zeotropes) are especially challenging because of temperature variations between bubble points and dew points.

Nevertheless, ideal conditions of nearly steady-state loads and flows with a minimum of cycling sometimes occur frequently enough to permit a reasonable analysis. Computer-controlled systems can provide the necessary data for a more accurate system analysis. Several sets of nearly simultaneous data at all points over a short time enhance the accuracy of any calculation of performance of a given system. In all cases, properly purging condensers and eliminating excessive lubricant contamination of the refrigerant at the evaporators are essential to determine system capabilities accurately.