

Fig. 17 .... Required Head vs Flow for Water Chiller with Cooling Tower

requirements of Fig. 17, and this is why it has been most universally adopted in recent years.

Where more stringent requirements for variation must be met, hot gas bypassing is sometimes used. Hot gas bypassing is inefficient, however, and suitable provisions must be made to desuperheat the bypass gas to prevent excessive discharge temperatures at low loads. Automatic liquid injection into the bypass gas stream may be provided for this purpose, or the gas stream may be brought into contact with cooling refrigerant liquid by other arrangements.

In most water-chilling machines, capacity control is obtained by varying the inlet guide vane setting in response to a thermostat in the chilled water leaving the evaporator, in order to maintain the temperature of that water as nearly constant as possible. In practice, the temperature of the chilled water leaving the evaporator, is allowed to vary slightly, as a function of the load, in order to obtain a suitable signal to operate the guide vanes. This provides a simple and stable control scheme. Either pneumatic, electro-pneumatic, or electronic controls are available for this purpose, and all are widely used. Other methods are available for automatically readjusting the set point of the basic capacity control in response to load, thereby eliminating drift, and holding the temperature more nearly constant. This feature is commonly referred to as *reset*.

For further information on automatic controls, see Chapter 13.

Safety Controls

In addition to the basic capacity controls, centrifugal refrigeration machines normally require a number of other controls, which are generally classed as *safety controls*. Some of these are limiting controls to keep the compressor within bounds during operation, and others are designed to shut down the machine without damage in the event of various possible malfunctions. Typical controls of this type are: (1) a load-limiting control which overrides the capacity control during start-up, during pull-down, or at other times when conditions would overload the motor, (2) a safety relief valve or valves on the condenser or evaporator, (3) a low oil pressure cutout switch for shutting down the unit as a result of lubrication failure, (4) a high condenser pressure cutout, (5) a low evaporator water (or brine) temperature cutout, (6) a low refrigerant temperature or pressure cutout, and sometimes (7) a chilled water flow cutout. Still other controls may be provided for special situations.

MULTIPLE MACHINE OPERATION

When more than one machine is installed to chill a common load, choices must be made concerning the piping of the chillers and condensers, i.e., whether the chilled water should be piped through the multiple evaporators in series or parallel and whether the condenser water should be piped through the multiple condensers in series or parallel. In addition, choices must be made of the methods of chilled-water temperature control. Each method has advantages and disadvantages, and selections will be based on the best match between the characteristics of the equipment, the initial cost, and the operating cost. Sometimes the overall system requirements, including consideration of the air-side equipment, will dictate the most economical arrangement.

Series operation of the chillers is desirable primarily because one machine can be operated at a higher evaporator temperature, with improved cycle efficiency. Water-side pressure drop is higher if the same number of passes are used as in parallel operation. Total full-load horsepower, including pump horsepower, can be less for series than for parallel arrangement of the chillers. However, this advantage becomes questionable if extensive partial load operation is expected. Much the same trend exists with regard to series arrangement of the condenser. The water-side pressure drop in series operation of either the condensers or the evaporators can be minimized by reducing the number of passes. Reduction of the number of passes usually lowers heat transfer performance somewhat and tends to cancel the performance gain obtained by series operation.

A single-control thermostat may be used to control all machines, or each machine may have its own thermostat. If a single-control thermostat is used in parallel operation of the chillers, overriding thermostats on each machine are usually required to prevent freeze-up of one operating machine when the other is shut down at partial load. Controls can be arranged to reduce load on all machines simultaneously, or to reduce load on one machine at a time, with others remaining at full load until the unloading machine is shut down. Controls are usually arranged to shut down machines as load reduces, in order to operate the remaining machines at higher load and better efficiency. Arrangements usually permit selection of which machine or machines will remain in operation, so that machine operating time can be equalized.

SELECTION OF CENTRIFUGAL REFRIGERATION MACHINES

In the design or selection of a centrifugal refrigeration machine for a given situation, the following minimum information is required:

1. *System Load in Tons of Refrigeration.* The system load required for selection or specification is determined by careful consideration of all the individual or zone loads, and the diversity of factors entering into the summation of these loads. This is discussed in Chapter 27 of this volume and Chapter 19 of the 1964 Guide And Data Book.

2. *Design Leaving Chilled-Water Temperature.* The leaving chilled-water temperature is determined by the type of equipment selected to absorb the system load, balancing adequate humidity control with economical selection of coil surface. If all other conditions remain constant, a higher leaving chilled-water temperature, will result in increased refrigeration unit capacity for a given kilowatt input, and a more economical selection of refrigeration equipment.

3. *Design Chilled-Water Rate.* Since the chilled-water temperature drop of the refrigeration unit is normally determined by the adsorbing equipment selection, the chilled-water rate is a function of chilled-water temperature drop and the refrigeration load (in tons) in accordance with Equation 11.

$$w = \frac{24q}{\Delta t_w} \qquad (11)$$

where

w = water flow rate, gallons per minute.
q = load, tons.
Δt_w = water temperature drop in the chiller, degrees.

4. *Design Temperature of the Cooling Water Entering the Condenser.* The design temperature of the cooling water entering the condenser is dependent primarily on the source of the cooling water. Because there are limitations on water usage in most areas, the cooling tower has become the most common source of condenser cooling water. Normal tower design can provide cooling water at a temperature 7 F deg above the design wet-bulb temperature, and, in the case of a 78 F design outdoor wet-bulb temperature, the cooling water leaving the tower and entering the condenser will be 85 F.

5. *Design Condenser Water Rate.* The condenser water rate is usually specified on the basis of a 10 F deg water temperature rise through the condenser. The water rate can be related to the temperature rise by Equation 12.

$$w = \frac{24(q + 0.212\,P)}{\Delta t_c} \qquad (12)$$

where

P = power required to pump refrigerant through the compressor, horsepower.
Δt_c = water temperature rise in condenser, degrees.

For water chilling applications, a good approximation is given by Equation 13.

$$w = \frac{30q}{\Delta t_c} \qquad (13)$$

All other conditions remaining constant, the lower the leaving condenser water temperature, the higher will be the refrigeration unit capacity for a given kilowatt input and the more economical will be the selection of refrigeration equipment.

With this information, a first approximation of the condensing and evaporating temperatures can be made. An *approach* of 7–8 F deg can be assumed for the evaporator, based on a water-side fouling factor of 0.0005, if water or a fluid having similar heat transfer coefficient is used. Approach refers to the temperature difference between refrigerant and leaving water. If a fluid having a low heat transfer coefficient is to be chilled, a higher approach should be assumed in order to prevent the evaporator from becoming uneconomically large. An approach of 7–8 F deg can also be assumed for the condenser, based on a fouling factor of 0.0005. If dirty water is expected, with a still higher fouling factor, *ARI Standard* 555-63 (Application and Ratings of Centrifugal Liquid Chilling Packages) allows for the selection of water chilling packages for 2 F deg lower leaving chilled water temperature for each increment of 0.0005 fouling allowance in the cooler, and for 2.5 F deg higher leaving condenser water temperature for each increment of 0.0005 fouling allowance in the condenser. Alternately, approaches can be calculated using the information given in Chapters 42 and 43.

With these approximate values of evaporating and condensing temperatures, the flow and head for a given refrigerant can be calculated. In doing this, it is usually assumed that very little subcooling or superheating occurs in the normal cycle and that condenser liquid and evaporator vapor can therefore be assumed to be at saturation.

For a simple single-stage compression cycle, as shown in Fig. 9, the flow volume is calculated by the following equation:

$$Q = \frac{200\,q}{h_1 - h_2}\,v_1 \qquad (14)$$

where

Q = quantity of refrigerant at compressor intake, cubic feet per minute.
v_1 = specific volume of vapor at the compressor inlet, cubic feet per pound.

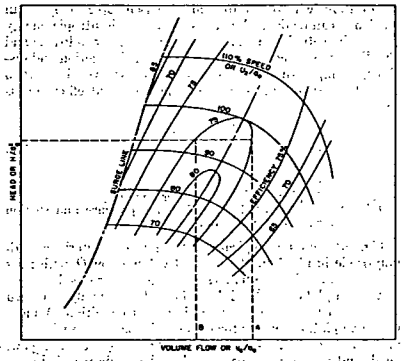


Fig. 18 .... Typical Performance—Single Stage Centrifugal Compressor

h_1 = enthalpy of gas at saturated suction temperature, Btu per pound.
h_2 = enthalpy of liquid at saturated condensing temperature, Btu per pound.

Although a polytropic process can be thought of as *any* process which is *non-isentropic*, as commonly used with ordinary centrifugal refrigeration compressors (where there is little external heat transfer during compression), the polytropic process is generally referred to as an *irreversible adiabatic process*. It can be shown that the polytropic head requirement for such a process may be calculated using the following equation:

$$H = 144 \log_e \frac{p_2}{p_1} \left[\frac{p_2 v_2 - p_1 v_1}{\log_e \dfrac{p_2 v_2}{p_1 v_1}} \right] \qquad (15)$$

where

H = polytropic head requirement.
p_1 = pressure at stage inlet, pounds per square inch absolute.
p_2 = pressure at stage discharge, pounds per square inch absolute.
v_1 = specific volume at stage inlet, cubic feet per pound.
v_2 = specific volume at stage discharge, cubic feet per pound.

Polytropic analysis is usually preferred over isentropic analysis because it tends to stay invariant over a wider range of compressors, fluids, and operating conditions. The polytropic or *hydraulic* efficiency is commonly *defined* as the ratio of the Polytropic Head to the actual enthalpy rise of the fluid, otherwise known as the Input Head:

$$\eta_p = \frac{H_i}{H_a} = \frac{H}{778(h_d - h_s)} \qquad (16)$$

where

η_p = polytropic efficiency.
H_i = Input Head, foot pound force per pound mass.
h_d = enthalpy at discharge pressure and temperature, Btu per pound.
h_s = enthalpy at suction pressure and temperature, Btu per pound.

With flow quantity and head requirement as calculated in Equations 14–16, the compressor performance characteristic may be used to determine the efficiency and operating speed requirement.