Guide And Data Book. It is important to note that, at lower evaporating temperatures, moisture is an even more serious problem so that thorough dehydration is of greater importance than for the normal ranges of operation. Since very low absolute evaporator pressures often result in operation below atmospheric pressure, the possibility of moisture entering with low side leaks is present. Tightness of the system is, therefore, even more significant than for systems where the refrigerant pressure is always above atmospheric pressure.

Because of the great importance of eliminating moisture from low temperature systems the value of liquid line sight glasses with moisture indicators is greater for low temperature systems than for the ordinary systems.

INSULATION AND VAPOR BARRIERS

Insulation and vapor barriers are discussed in Chapters 22 and 23 of the 1961 Guide And Data Book. Particular mention should be made of the greater importance of insulation and effective vapor barriers on low temperature systems. As the low temperatures are continually decreasing, condensation in the insulation is becoming of greater importance. One should keep in mind that gases other than water vapor may condense in the insulation, especially in industrial atmospheres or where temperatures go as low as −300 F. Where conditions of this type exist the partial pressure differential across the vapor barrier will be much larger than normal and very special types of vapor barriers may be necessary.

Since the capacities of refrigeration compressors do decrease rapidly with increases in specific volume of the suction refrigerant and since this specific volume increases quite rapidly with temperature at the low absolute suction pressures, a lesser degree of heat influx in the low temperature suction lines can be justified than for the higher temperature applications. Thus, both the thermal insulating effectiveness as well as the effectiveness of the vapor barrier are more important considerations for these systems than for the normal systems.

CHAPTER 73

WATER AND BRINE PIPING

Water Piping, Classification of Piping Systems, Design Water Quantity, Pipe Friction, Hydrostatic Pressure, Pipe Sizing, Pump Selection, Expansion Tanks, Installation Practices, Water Treatment, Piping Arrangements, Direct and Reversed-Return Systems, Loop-Zone Systems, Open Systems with Multiple Reservoirs, Brine Piping

THIS chapter will discuss piping primarily for chilled water, condenser water, and brine systems. Piping for hot water systems is discussed in Chapters 8 and 9. Systems for building services are covered in Chapter 75.

WATER PIPING

For every piping system there is an optimum combination of operating performance, ease of maintenance, and initial cost, and the design of the system should satisfy the relative importance of these items. This section will present basic information and current practices for piping of normal installations. The system designer should familiarize himself with local codes and restrictions and any special or unusual job requirements before undertaking a specific design.

Classification of Piping Systems

Piping systems are either the once-through type, where water flows in series from a reservoir through a pump and piping system to equipment and is relieved to a different reservoir; or the recirculation type, where the same water is returned to the original reservoir and recirculated continuously.

Systems are further classified as open (if water is continuously brought in close contact with air at some point in the system) or closed (if water does not come in close contact with air at any point in the system). An open expansion tank is not construed as providing close contact with the air because the surface for exposure is very limited. For example, a city water system is an open nonrecirculating system; a condenser water system with cooling tower and a spray-type dehumidifier chilled water system are open recirculating systems, and a coil-type dehumidifier chilled water system is a closed recirculating system.

Design Water Quantity

The relationships of water temperature rise (temperature difference Δt), water quantity (gpm) and heat carrying capacity Btuh are given by Equations 1, 2, 3, and 4.

$$\text{Btuh} = \text{gpm} \times \Delta t \times 500 \quad (1)$$

or, for *chilled water* systems

$$\text{gpm} = \frac{\text{Chiller Tons} \times 24}{\Delta t} \quad (2)$$

If a heat rejection factor of 1.25 is assumed for *compression refrigeration*, the gpm for *condenser systems* is

$$\text{gpm} = \frac{\text{Chiller Tons} \times 30}{\Delta t} \quad (3)$$

If a heat rejection factor of 2.6 is assumed for *absorption refrigeration*, the gpm for *condenser systems* is

$$\text{gpm} = \frac{\text{Chiller Tons} \times 62}{\Delta t} \quad (4)$$

Normal design practice usually results in a chilled water temperature rise of between 10 and 14 F deg, and a condensing water rise of 10 F deg for reciprocating or centrifugal refrigeration machines or 17 F deg for absorption refrigeration machines. If city water is used for condensing, a rise of 20 deg or higher in the reciprocating or centrifugal machine condensing water temperature is normal.

An economic evaluation of a proposed system should be made to insure that the desired balance between installation cost and operating cost has been achieved. In many cases the factors which determine water quantity can be varied, to provide a better economic balance. Table 1 shows the effect of coil circuiting and chilled water temperature on water quantity and rise. The coil rows, fin spacing, air side performance, and cost are identical for all selections. Detailed studies[1] have been published showing the effect of such changes on the overall system. Considering the investment cost of piping and insulation versus the operating cost of refrigeration and pumping motors, it is generally agreed that 1.0 to 1.5 gpm per ton is preferred on chilled water systems with long piping runs, and 1.8 to 2.4 gpm per ton for close-coupled systems.

The type of control influences the design water quantity. For systems using multiple air handling units, diversity factors may be applied to water quantities before sizing pump and piping mains if, because of exposure or usage, the unit design loads do not occur simultaneously and if water side throttling control is used. If air side (face-and-bypass, or fan-cycling) or constant volume (three-way valve) water control is used, diversity should not be considered.

Pipe Friction

The friction loss resulting from water flow through a pipe can be obtained by use of the Darcy formula:

$$h_f = \frac{L}{D} \frac{V^2}{2g}$$

where

h_f = Friction loss, feet of liquid.
f = Dimensionless friction factor.
L = Length of pipe, feet.
D = Inside Diameter of pipe, feet.
V = Fluid velocity, feet per second.
g = Acceleration of gravity, 32.174 feet per second.

The friction factor f is a function of Reynolds number and pipe surface roughness. The surface roughness is dependent upon pipe material and, in the case of steel pipe, upon the surface corrosion and scale, which is influenced by the amount of air dissolved in the water, which is in turn influenced by whether the piping system is an open or closed type.

Friction loss for water flow in copper tube and steel pipe can be obtained from Figs. 3, 4, and 5 of Chapter 75.

Losses in valves, fittings, or obstructions can be obtained by