

Consider, for example, F-12 with a quality (the per cent in vapor form) of 30 per cent; the enthalpy of this material would be equal to:

$$h_m = h_l + 0.30 (h_v - h_l) \quad (2)$$

where

h_m = specific enthalpy of the mixture.

h_l = specific enthalpy of the liquid.

h_v = specific enthalpy of the saturated vapor.

Values of h_l and h_v are obtained from Table 1 for the actual pressure of the mixture.

By a reversal of this same procedure the tabular data can be used to determine the state of a mixture leaving an expansion valve. Consider a valve to which saturated liquid at pressure p_a is admitted and a mixture of saturated liquid and vapor at pressure p_d is discharged. The quality of the material at discharge is then determined by making use of the fact that the expansion process is completely irreversible, is a throttling process, and hence occurs without change in enthalpy. Thus the enthalpy of the mixture, h_m , is equal to the enthalpy of the saturated liquid at the entrance state, h_{fa} , and can therefore be read from the table.

Thus,

$$h_{fa} = h_m = h_{vd} - (1 - x) (h_{vd} - h_{fd}) \quad (3)$$

or,

$$x = (h_m - h_{fd}) \div (h_{vd} - h_{fd}) \quad (4)$$

where

h_{fa} = enthalpy of saturated liquid at entrance to expansion valve.

h_m = enthalpy of mixture.

h_{vd} = enthalpy of saturated vapor at discharge.

h_{fd} = enthalpy of liquid at discharge.

x = proportion of liquid in the mixture, decimal.

The refrigerant *cycle* is the series of state changes which occur in the conditioning processes needed to restore the refrigerant to a condition in which it will possess the ability to extract heat from the space to be cooled. For all compression-type systems the cycle consists of four processes: heat gain in the evaporator; pressure rise in the compressor; heat loss in the condenser; pressure loss in the expansion valve. The compression process is accomplished at the expense of energy added to the compressor in the form of shaft work and the expansion process could be carried out, if the economics of the system would permit, in an expanding engine with consequent release of energy as shaft work. In ordinary systems, however, the additional first cost and maintenance costs of an expanding engine so greatly exceed the advantage resulting from the work realized that such engines are not used and the pressure reduction is allowed to occur irreversibly in an expansion valve. Basically, then, a refrigeration cycle consists of two heat transfer processes and two pressure change processes, no work entering into the heat transfer processes and—in the simple cycle—no heat transfer occurring during the pressure-change processes.

Simple Refrigeration Cycles

The most common and least complicated type of refrigeration cycle is shown in Fig. 1 and is called the *simple saturation cycle*. For this system saturated vapor flows without gain or loss of heat from the

evaporator to the suction of the compressor. During passage through the compressor the energy added as shaft work goes entirely to increase the enthalpy of the refrigerant, and the compression process, which is assumed to occur irreversibly and without external heat transfer, is characterized by constant entropy. Thus the state of the superheated vapor leaving the compressor can be determined from the tables of thermodynamic properties by noting the discharge pressure and fixing also the entropy of the saturated vapor at entrance to the compressor.

Superheated vapor from the compressor flows to the condenser where desuperheating and condensation take place. From the condenser the refrigerant flows to the expansion valve, undergoes a constant-enthalpy pressure reduction and returns to the evaporator where it again removes a quantity of undesired heat. When the evaporator is arranged to permit direct cooling of room air by the refrigerant, the system is said to be

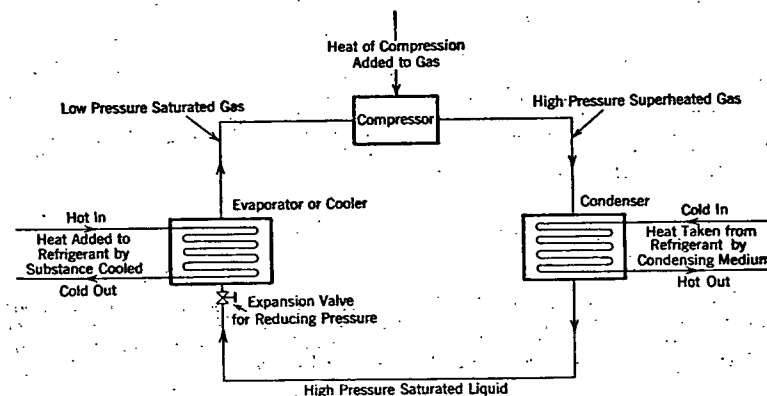


FIG. 1. MECHANICAL REFRIGERATION SYSTEM

of the *direct expansion* type, while a system in which the evaporating refrigerant cools water or brine, which in turn cools the air, is said to be *indirect*. Although many differences exist between most actual systems and that of the simple saturation cycle this latter is nonetheless of great value in that it provides an extremely simple method of rapidly achieving an approximate analysis of probable power requirements, compressor size, etc. Further, the equations used in analysis of a simple saturation cycle form the basis of the more complex treatments required for compound refrigeration cycles. For these reasons a typical simple saturation problem will be worked in detail.

Example 1. A simple saturation cycle carries a 7 ton load when operating between suction and discharge pressures of 52.7 psia and 121 psia with F-12 as the refrigerant. Determine: (a) the cooling effect provided by each pound of refrigerant, (b) the refrigerant circulating rate, (c) the horsepower required, (d) the quantity of heat to be dissipated from the condenser, (e) the required condenser cooling water, in gallons per minute, if temperature rise of water passing through the condenser is 8 deg, (f) the bore and stroke of a double acting cylinder (neglecting the effect of the piston rod) if speed of compressor is 500 revolutions per minute.

Solution. (a) Saturated liquid F-12 at 121 psia leaves the condenser and enters the expansion valve. The enthalpy of this material (from Table 1) is 29.68 Btu per pound and this must also be its enthalpy at entrance to the evaporator. Leaving the evaporator as a saturated vapor at 52.7 psia, its enthalpy is 82.82 so the refrigerating effect must be $82.82 - 29.68 = 53.14$ Btu per pound.