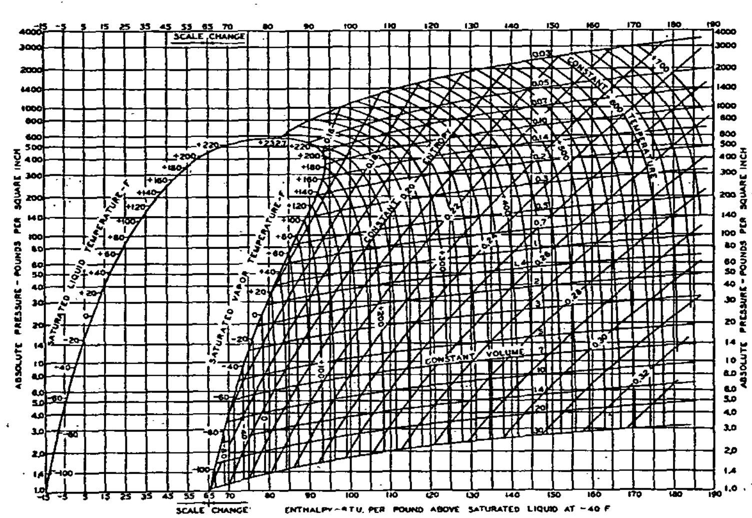
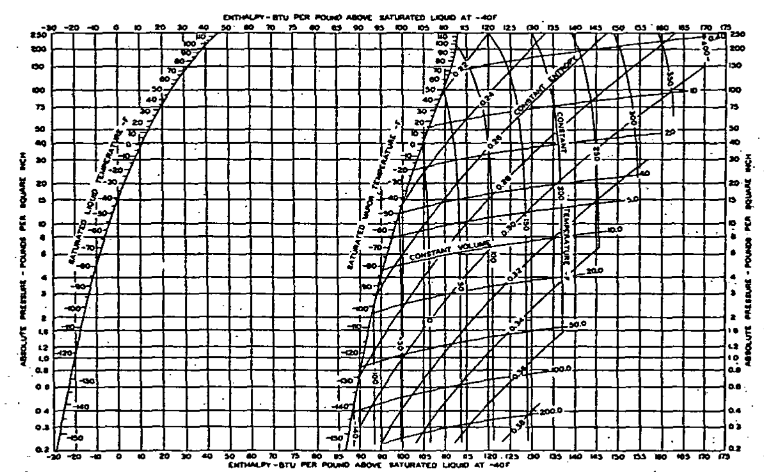


Temperature—F; Volume—Cu Ft per Lb; Entropy—Btu per (Lb) (F deg).

Fig. 1 . . . . Pressure-Enthalpy Diagram for Dichlorodifluoromethane (CCl_2F_2) (Refrigerant 12)



Temperature—F; Volume—Cu Ft per Lb; Entropy—Btu per (Lb) (F deg).

Fig. 2 . . . . Pressure-Enthalpy Diagram for Monochlorodifluoromethane ($CHClF_2$) (Refrigerant 22)

With the ideal Carnot cycle operating as a heat pump, the coefficient of performance is

$$(CP) = \frac{T_c}{T_c - T_e} \qquad (3)$$

The Carnot cycle coefficient of performance for both a refrigerating machine and a heat pump increases as the spread between the evaporator and the condenser temperatures decreases. In general, the same is true for an actual system operating as either a refrigerating machine or a heat pump.

Refrigerants

A desirable refrigerant should possess chemical, physical, and thermodynamic properties that permit its efficient application in refrigerating systems. In addition, when the volume of the charge is large, there should be little or no danger to health or to property in case of its escape.

Thermodynamically, a material for use as a refrigerant should have a large latent heat of vaporization since it is this heat quantity—subject to minor variations—that constitutes the working effectiveness of the refrigerant. Further, since the work required to compress a vapor increases rapidly with the pressure ratio, the thermodynamic characteristics of the fluid should be such that the required low-to-high temperature range can be achieved with only a moderate change in ratio. A further consideration, from the standpoint of practical operating effectiveness, is that the suction pressure should not be below atmospheric (to prevent leakage of air into the refrigerant lines) nor should the condenser pressure be excessively high (to prevent need for extra-heavy construction). The specific volume-specific enthalpy relationship is also important because some materials would have such low density, when in vapor form, that impractical compressor displacements would be needed to handle the suction vapor.

Properties of refrigerants are usually given either in tabular or graphic form. In contrast to the temperature-entropy plotting which is used almost exclusively in steam-power work, refrigeration problems are usually referred to a pressure-enthalpy chart. The advantage of pressure-enthalpy plotting in that linear distances on the chart correspond to energy gains or losses, and the two types of processes, constant-pressure and constant-enthalpy, that occur most frequently in refrigeration cycles, can both be represented by straight vertical or horizontal lines. Figs. 1 and 2 present pressure-enthalpy charts for dichlorodifluoromethane, CCl_2F_2, and monochlorodifluoromethane, $CHClF_2$, respectively. Although tabular arrangements of refrigerant properties require interpolation between values, they have the advantage of an accuracy greater than that obtainable from a chart. Tables 1, 2, 3, and 4 give the thermodynamic properties of four of the more common refrigerants used in air-conditioning installations: Refrigerant* 12, dichlorodifluoromethane; Refrigerant 22, monochlorodifluoromethane; Refrigerant 11, trichloromonofluoromethane; and Refrigerant 113, trichlorotrifluoroethane. The first two of these are commonly used in reciprocating compressors while the last two are commonly used in centrifugal machines.

Referring to Table 1, the first column gives the range of saturation temperatures likely to occur in practice. The second column gives the saturation pressure expressed in pounds per square inch absolute corresponding to a given temperature, while the next six columns give the three fundamental specific properties, volume, enthalpy, and entropy, of the saturated liquid and saturated vapor, respectively. The last four columns give values of enthalpy and entropy for gases with 25 deg and with 50 deg of superheat; note particularly that the column heading *50 F superheat* means, not that the gas is at a temperature of 50 F, but that its temperature exceeds by 50 deg the saturation temperature corresponding to its actual pressure. Thus, CCl_2F_2 vapor at 38.0 psig and 91 F possesses 50 deg of superheat, since its saturation temperature corresponding to 52.7 psia is 41 F.

The tabular arrangements of refrigerant properties are literally for saturated or superheated materials only. In many cases, however, the engineer must work with subcooled liquids. With an accuracy sufficient for all practical purposes, the specific volume and the enthalpy of any subcooled refrigerant can be taken as equal to the values read from the tables for a saturated liquid at the same temperature. Thus, if CCl_2F_2 at 121 psia and 40 F is passing through a pipe, its volume and enthalpy can be determined from Table 1 as 0.0116 cu ft per pound and 17.0 Btu per pound.

Frequently it is necessary to determine the properties of a *wet* vapor or of a mixture of liquid with some added vapor, such as is found at the discharge from an expansion valve. This can be done from the tables by noting that the specific enthalpy of the mixture must be equal to that of the saturated liquid, plus a fraction of the latent heat of vaporization equal to the fraction of refrigerant that is present in vapor form. Consider, for example, CCl_2F_2 with a quality (the percent in vapor form) of 30 percent; the enthalpy of this material would be equal to

$$h_m = h_f + 0.30\,(h_v - h_f) \qquad (4)$$

where

h_m = specific enthalpy of the mixture.
h_f = specific enthalpy of the liquid.
h_v = specific enthalpy of the saturated vapor.

Values of h_f and h_v for use in Equation 4 can be obtained directly or by calculation from the tables of properties of refrigerants.

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_e 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_{fe}, and can therefore be read from the table. Thus,

$$h_{fe} = h_m = h_{vd} - (1 - x)\,(h_{vd} - h_{fd}) \qquad (5)$$

or,

$$x = (h_m - h_{fd}) \div (h_{vd} - h_{fd}) \qquad (6)$$

where

h_{fe} = 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.

Vapor Compression Refrigeration Cycle

Simple Cycle. The refrigerant *cycle* is the series of state changes (which occur in the conditioning processes) needed

* ASRE designations for these refrigerants.