the space (or in some cases from a secondary cooling fluid such as brine or cold water) to the refrigerant. The vapor formed in the evaporator is then raised in pressure (by a compressor, or by the absorber-generator combination of the absorption system) until its new boiling temperature exceeds the temperature of the available cooling medium; under these conditions heat transfer is established from the refrigerant vapor to the cooling medium with resultant condensation of the refrigerant. When condensed, the high-pressure liquid refrigerant is reduced in pressure and again allowed to *boil* in the evaporator.

In order to permit evaluation of the effectiveness with which any given cycle operates, some term is desirable which would be comparable to the *efficiency* that is used for heat engines. In refrigeration the desired effect is heat extraction and the cost of achieving this extraction is the amount of energy which must be supplied as shaft work. Thus the ratio of refrigerating effect to the heat equivalent of the compressor work is used as a measure of effectiveness and is defined as the coefficient of performance.

If the desired effect is the rejection of heat through the condenser instead of heat extraction through the evaporator, the refrigeration system is then termed a heat pump. In this case the coefficient of performance is the ratio of the heat rejected from the condenser to the heat equivalent of the compressor work. The coefficient of performance for the heat pump is greater than that for a system operating as a refrigerating machine because all mechanical shaft work required to operate the compressor is dissipated as useful heat through the condenser.

The Carnot cycle, an ideal, thermodynamically reversible cycle consisting of an adiabatic expansion and an isothermal expansion followed by an adiabatic compression and an isothermal compression to form a closed cycle, may be shown to be a measure of the maximum possible conversion of heat energy into mechanical energy. In its reversed form it is a measure of the maximum performance possible for any refrigeration cycle operating either as a refrigerator or as a heat pump. Although it cannot be applied in an actual machine because of the impossibility of obtaining complete reversibility, it is, nevertheless, extremely valuable as a criterion of inherent limitations. The coefficient of performance (CP) of a reversed Carnot cycle system operating as a refrigeration system is:

$$(CP) = \frac{T_e}{T_c - T_e} \tag{2}$$

where

T_e = evaporator temperature, Fahrenheit degrees, absolute.
T_c = condenser temperature, Fahrenheit degrees, absolute.

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

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

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 which 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—which 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 is that linear distances on the chart correspond to energy gains or losses and the two types of processes, constant-pressure and constant-enthalpy, which occur most frequently in refrigeration cycles, can both be represented by straight vertical or horizontal lines. Figures 1 and 2 present pressure-enthalpy charts for ammonia and dichlorodifluoromethane (Freon-12). 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: dichlorodifluoromethane (Freon-12), monochlorodifluoromethane (Freon-22), ammonia, and monofluorotrichloromethane (Freon-11); the first three of these materials are commonly used in reciprocating compressors; the last refrigerant is 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 F-12 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 sub-cooled liquids. With an accuracy sufficient for all practical purposes the specific volume and the enthalpy of any sub-cooled refrigerant can be taken as equal to the values read from the tables for a saturated liquid at the same temperature. Thus if F-12 at 121 psia and