handled. Assuming a volumetric efficiency of 90 percent, the compressor must then displace 20.49 ÷ 0.9 = 22.8 cfm. The speed is given as 500 rpm and, as the unit is known to be double-acting, the displacement is therefore (22.8 × 1728) ÷ (2 × 500) = 39.4 cu in. If the unit were designed so that bore $d$ and stroke were the same,

$$(\pi d^3) \div 4 = 39.4 \qquad d = 3.69 \text{ in.}$$

$$(g)\ (\text{CP}) = (h_{vs} - h_{fc}) \div (h_d - h_{vs})$$

$$= (82.82 - 29.68) \div (89.34 - 82.82) = 8.17$$

where $h_{fc}$ is the specific enthalpy of liquid at discharge from the condenser.

The coefficient of performance of *Example* 1 may be compared with that of an ideal system operating on the Carnot cycle between the same temperature limits. Then $T_s$ = 501 F (which is 41 F + 460) and $T_c$ = 554 F (which is 94 F + 460) and,

$$(\text{CP}) = \frac{501}{554 - 501} = 9.6$$

The actual cycle is therefore 8.17 ÷ 9.6 or 85 percent as effective as a Carnot cycle between the same temperature limits.

## Influence of Suction Pressure

Brief consideration of the analytical procedure used in discussion of the simple saturation cycle will bring out the need for maintaining the suction pressure on any refrigeration system as high as the load will permit. As the suction pressure increases, for fixed discharge pressure, the enthalpy of refrigerant entering the evaporator remains unchanged, but the leaving enthalpy increases and, hence, the refrigerating effect increases. Further, compressor energy input is reduced not merely because of the greater enthalpy of the gas at suction, but also because of a reduction in the enthalpy of the superheated gas at discharge. Since the refrigerating effect is greater and the work less, it is obvious that there will be a substantial gain in the coefficient of the performance. See Chapter 43, Fig. 1.

The actual value of suction pressure on any system is obviously determined by the required temperature which must be maintained in the conditioned space. For a direct expansion system the evaporator can be held at a temperature not much less than that of the conditioned enclosure, except in cases where lower temperatures may be needed in order to establish a desired ratio of dehumidifying to cooling load. When dehumidification requirements dictate the use of unusually low evaporator temperatures, the increased operating cost should properly be charged against the dehumidification rather than the sensible cooling.

## Influence of Discharge Pressure

In contrast to the suction pressure, the compressor discharge pressure should be kept as low as operating conditions will allow. This pressure must be high enough to provide a saturation temperature of refrigerant within the condenser which is greater than the exit temperature of the cooling water. The discharge pressure therefore is a direct function of the temperature of the cooling fluid, and will automatically rise whenever the temperature of cooling water (or air) rises; it will also rise when the flow rate of the cooling medium is decreased.

Increase in discharge pressure (for fixed suction pressure) raises the enthalpy of the gas leaving the compressor; hence, increases the work of



compression. Further, as the enthalpy of saturated liquid leaving the condenser increases with pressure, the refrigerating effect must decrease. Thus the effect of such a pressure rise is to require more work per pound of refrigerant handled, and at the same time to necessitate an increase in the refrigerant flow rate. See Chapter 43, Fig. 1.

## Influence of Water Jacket

The preceding discussion has, in every case, assumed isentropic compression. Where exact performance data are not available, this assumption is a desirable one since it leads to a conservatively large determination of the power required. In most actual systems, the compression process departs from isentropic due to irreversible heat transfers which occur between the vapor in the cylinder and the cylinder wall, and also because of intentional heat dissipation from the outside of the cylinder walls to the surroundings, or to a cooling fluid passing through a water jacket around the cylinder. Compressor cooling is highly desirable as a method of reducing power consumption.

## Influence of Superheating and Subcooling

The most common departure from conditions of the simple saturation cycle is that resulting from admission of superheated vapor to the compressor. Thermodynamically, superheat is undesirable because the enthalpy increase required to compress a vapor through a given pressure range increases with superheat. Further, superheated vapor leaving an evaporator is usually an indication that the suction pressure is lower than necessary. Under practical operating conditions, however, superheat is almost universally used as a means of assuring complete vaporization of the refrigerant going to the compressor. With modern compressors operating at high speed, and with relatively small clearance space, it is particularly necessary to avoid admission through the suction valves of liquid refrigerant.

Another common departure of actual systems from the simple saturation cycle occurs because of subcooling of refrigerant in the condenser. Thermodynamically, such subcooling is advantageous since it increases the refrigerating effect without affecting the unit energy requirements of the compressor. Further, it can be shown that for a fixed ratio of condenser cooling water to refrigerant circulating rate, the total compressor power requirements will be greater when operating at simple saturation than when operating with maximum subcooling. What is even more surprising is that condenser pressure may be lower for the subcooling cycle than for the saturation cycle. This condition results from the fact that, for the same capacity on a heavily loaded condenser, the refrigerant flow rate is less when there is subcooling.

Because of the advantages attendant upon the use of subcooling, many methods are in use for obtaining some subcooling effect outside of the condenser. One common procedure is to use the cold vapor leaving the evaporator to cool the liquid flowing from condenser to expansion valve.

Another somewhat unusual subcooling cycle allows cold refrigerant from the downstream side of the expansion valve to cool liquid refrigerant from the condenser down to the evaporator temperature. Fig. 5 shows the pressure-enthalpy diagram for a typical refrigeration cycle operating with both subcooling of the refrigerant from the condenser and superheating of the refrigerant leaving the evaporator.