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 pressure of 52.7 psia and 121 psia with dichlorodifluoromethane, CCl_2F_2, 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; (*g*) coefficient of performance.

Solution: (*a*) Saturated liquid CCl_2F_2 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.

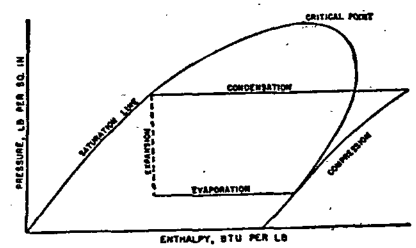

Fig. 4. Pressure-Enthalpy Diagram for Simple Saturation Cycle

(*b*) The refrigerant circulating rate is equal to the total heat to be picked up in unit time, divided by the pick-up per pound of refrigerant or,

$$W_r = (7 \text{ ton} \times 200) \div 53.14 = 26.3 \text{ lb per minute.}$$

(*c*) The horsepower required is equal to the increase in energy of the refrigerant passing through the compressor (expressed in Btu per minute) divided by the conversion factor 42.42, which is the number of Btu per minute corresponding to 1 hp,

$$(\text{hp}) = W_r\,(h_d - h_{vs}) \div 42.42 \qquad (7)$$

where

- hp = horsepower.
- W_r = refrigerant circulating rate in pounds per minute.
- h_d = enthalpy of vapor at condition of discharge from compressor.
- h_{vs} = enthalpy of saturated vapor entering compressor.

W_r is known from (*b*) and h_{vs} is the enthalpy of refrigerant as it enters the compressor in a saturated vapor state at 52.7 psia; thus h_{vs} = 82.82.

In order to determine h_d, the state of the refrigerant must first be determined at the compressor discharge. At the known suction state the entropy (from Table 1 for saturated vapor at 52.7 psia) is 0.16828 and, since the compression is assumed to occur isentropically, it therefore follows that the discharge stage must have the same entropy at 121 psia. From the table the entropy of vapor superheated 25 deg is

0.17330, so the superheat, t_{sd}, possessed by the actual gas discharged from this compressor can be obtained by interpolation as,

$$\frac{t_{sd}}{25} = \frac{0.16828 - 0.16608}{0.17330 - 0.16608}$$

from which t_{sd} = 7.6 deg.

As the saturation temperature at 121 psia is 94 F the actual temperature, t_d, of the vapor leaving the compressor is, $t_d = 94 + t_{sd} = 94 + 7.6 = 101.6$ F. By the same kind of interpolation the enthalpy of the discharged vapor can be determined from the enthalpies given for vapor superheated 25 F and for saturated vapor,

$$\frac{(h_d - 88.10)}{(92.16 - 88.10)} = \frac{(0.16828 - 0.16608)}{(0.17330 - 0.16608)}$$

from which, h_d = 89.34 Btu per pound.

Then substituting in Equation 7,

$$(\text{hp}) = 26.3\,(89.34 - 82.82) \div 42.42 = 4.03.$$

(*d*) The rate of heat loss from the condenser, Q_c, must be equal to the sum of the energies picked up by the refrigerant in the evaporator and the compressor,

$Q_c = 53.14 + (89.34 - 82.82) = 53.14 + 6.52 = 59.66$ Btu per pound or $26.3 \times 59.66 = 1569$ Btu per minute. This same figure can, of course, be determined more directly by subtraction of the enthalpy of liquid leaving the condenser from the enthalpy of superheated vapor going into it, thus,

$$Q_c = 26.3\,(89.34 - 29.68) = 1569 \text{ Btu per minute.}$$

(*e*) The cooling water rate (based on a gallon as 8.34 lb) is $1569 \div (8 \times 8.34) = 23.5$ gpm.

(*f*) The compressor size is fixed by the volume of gas which must be drawn into the machine per unit time. Saturated vapor at 52.7 psia has a specific volume, from Table 1, of 0.779 cu ft per pound, hence $26.3 \times 0.779 = 20.49$ cfm of gas must be handled. Assuming a volumetric efficiency of 90 percent, the compressor must then displace $20.49 \div 0.9 = 22.8$ cfm. The speed is given as 500 rpm and, as the unit is known to be double-acting, the displacement is therefore $(22.8 \times 1728) \div (2 \times 500) = 39.4$ 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