

(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 horse power 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,

$$(\text{hp}) = W_r (h_d - h_{vs}) \div 42.42 \quad (5)$$

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 I 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 state 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 into Equation 5,

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

(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 I, 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 per cent 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,

$$(\pi d^3) \div 4 = 39.4$$

$$d = 3.69 \text{ in.}$$

Coefficient of Performance

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, thus,

$$(\text{cop}) = (h_{vs} - h_{fc}) \div (h_d - h_{vs}) \quad (6)$$

where

cop = coefficient of performance.

h_{vs} = specific enthalpy of saturated vapor entering compressor.

h_{fc} = specific enthalpy of liquid at discharge from condenser.

h_d = specific enthalpy of vapor at discharge from compressor.

The subscripts vs, d, and fc represent state points at suction and discharge of the compressor and at discharge from the condenser. Thus for the conditions of the simple saturation cycle which was used in Example 1,

$$(\text{cop}) = (82.82 - 29.68) \div (89.34 - 82.82) = 8.17.$$

This coefficient can be compared with that which would exist if the system were to operate on an ideal Carnot cycle for which the coefficient of performance would be,

$$(\text{cop}) = \frac{T_s}{T_c - T_s}$$

where

T_s = evaporator temperature, Fahrenheit degrees, absolute.

T_c = condenser temperature, Fahrenheit degrees, absolute.

In problem 1, $T_s = 501$ F (which is 41 F + 460) and $T_c = 554$ F (which is 94 F + 460) and,

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

The actual cycle is therefore $8.17 \div 9.6$ or 85 per cent 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 performance.

The actual value of suction pressure on any system is obviously determined by the required temperature which must be maintained in