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,

$$(\text{cop}) = (h_{vs} - h_{lc}) \div (h_d - h_{vs}) \qquad (6)$$

where

cop = coefficient of performance.
h_{vs} = specific enthalpy of saturated vapor entering compressor.
h_{lc} = specific enthalpy of liquid at discharge from condenser.
h_d = specific enthalpy of vapor at discharge from compressor.

The subscripts vs, d, and lc 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 ÷ 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 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, the enthalpy of saturated liquid leaving the condenser increased with pressure so 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.

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.

Exact measurement of the heat carried away in the jacket cooling water requires facilities which frequently are not available on field installations, but a reasonable close approximation to both the heat loss and the work requirement can be obtained from theory, providing the temperature of the gas leaving the compressor is experimentally determined. Knowing the temperature and pressure at both suction and discharge, the actual state points can be readily determined from the tables of refrigerant properties and the entropies and enthalpies thereby evaluated. Then the energy dissipation to cooling water can be calculated approximately from the equation,

$$Q_j = W_r\,(\Delta s)\,T_{avg} \qquad (7)$$

where

Q_j = energy dissipated to cooling water, Btu per minute.
W_r = weight of refrigerant, pounds per minute.
Δs = entropy change between suction and discharge.
T_{avg} = average temperature of gas passing through compressor, Fahrenheit degrees, absolute.