pressure. This occurs because the liquid line is not a line of equal pressure but of saturation pressures. The error in work per pound of refrigerant figured from total heats, which should be added to the indicated figures is roughly the specific volume of the liquid at the lower pressure and temperature multiplied by the pressure difference in appropriate units. This error may become of some importance in calculations involving carbon dioxide or in problems involving the liquid of any of the refrigerants, as in figuring expansion valve orifices.

Theoretical Work per Pound

The temperature-entropy and pressure-volume diagrams are based on one pound of the refrigerant. Likewise, the theoretical work and the refrigerating effects are conveniently based on a pound of refrigerant. The compression work per pound may be found by several methods.

The temperature-entropy method starts with state point *a*. Since the quality of *a* is known, the heat content of the vapor H_a is known, and also the entropy S_a. Since point *b* lies near the saturation curve it is customary to assume $S_a = S_b$ and with T_2 given, H_b can be determined. If W = work in foot-pounds per pound of refrigerant, then

$$W = (H_b - H_a) \times 778 \tag{1}$$

The pressure-volume method starts with state point *a*, whose pressure and specific volume are known. The work of compression is the adiabatic work of compression from P_1 to P_2, plus the work of expelling the vapor at constant pressure P_2 minus the external work of evaporation of the vapor to volume V_1 at pressure P_1.

$$W = \frac{n}{n-1} \times P_1V_1 \left[\left(\frac{P_2}{P_1} \right)^{\frac{n-1}{n}} - 1 \right] \tag{2}$$

It is frequently helpful to think of the compression of the vapor in terms of head. The head may be likened to a vertical column of vapor in which is located the vapor to be compressed. The compression occurs when the vapor is moved down from a level corresponding to P_1 to a new level corresponding to P_2, in equilibrium with the surrounding vapor. If this process is carried on isentropically, the result will be the same as indicated previously. Then if h is the head in feet,

$$W = h \tag{3}$$

This relationship may easily be seen from the fact that a small difference of head dh divided by the specific volume of the vapor V is equal to the increment of pressure difference dP.

Head is very useful in considering the performance of centrifugal compressors, which merely substitute a centrifugal for the gravity head. It is also useful in considering problems of fluid flow. In these problems, the head per degree can be obtained either by direct calculation or approximately by dividing the total head by the temperature difference $T_2 - T_1$. The velocity head loss can then be calculated in degrees, using the customary formula $V^2 = 2gh$.

Refrigerating Effect per Pound

The refrigerating effect per pound is computed by the same method, regardless of the type of refrigeration system. The solution is indicated on the temperature-entropy diagram of Fig. 2. Assuming that the vapor leaving the evaporator is saturated, the refrigerating effect in Btu per pound is obtained by subtracting from the heat content of the vapor at temperature T_1, the heat content of the liquid at T_2, or if the liquid is sub-cooled, the liquid temperature.

Thus, the refrigerating effect in Btu per pound is equal to

$$H_a - H_c = H_a - H_e \tag{4}$$

If the vapor entering the compressor is superheated or supersaturated, a correction in the heat of the vapor is made accordingly.

The unit of refrigeration is the *ton*, based on the latent heat of fusion of one ton of ice in 24 hr. Thus one ton = 200 Btu per minute = 12,000 Btu per hour.

Coefficient of Performance

The coefficient of performance of a refrigeration system is the ratio of the refrigerating effect to the work of compression, both expressed in the same units.

The ideal or Carnot coefficient of performance depends upon the temperatures T_1 and T_2 in much the same way as the ideal efficiency of a steam engine depends upon its working temperature, with an inverse relationship.

$$\text{Ideal C. of P.} = \frac{T_1}{T_2 - T_1} \tag{5}$$

Evidently the smaller the compression range, the less power will be required to produce a given refrigerating effect.

The theoretical coefficient of performance of actual refrigerants is always less than the ideal due to the tendency of most refrigerants to superheat when compressed, and due to the heat of the liquid which must be removed. The cycle efficiency is the theoretical C. of P. divided by the ideal for the same temperatures. The cycle efficiency usually changes as the compression temperatures change.

Practical Cycle

Fig. 3 illustrates the pressure-volume and temperature-entropy diagrams for an actual cycle. These diagrams are based upon the compressor receiving vapor superheated and upon sub-cooling of the liquid going to the evaporator. The theoretical cycle is $a_1b_1cc_1e_1a_1$. However, the vapor during compression actually follows line a_1b_2 due to superheating as a result of the inefficient work of compression. The theoretical work of compression is $a_1b_1cda_1$. Added to this is the area $b_2b_1g_1h_1b_2$ on the temperature-entropy diagram which represents the inefficient work of compression (assuming no compressor heat losses). The sum of these areas represents the total work of the compressor per pound of refrigerant, and the ratio of theoretical cycle work to the actual work represents the over-all efficiency. It should be noted that area $a_1b_2b_1a_1$ is considered as part