In exactly the same way representation on the ideal cycle can be given to the pressure losses that occur in the connecting piping. In each case the entire loss of a given line can be treated as though it occurred during passage through an equivalent expansion (or pressure reducing) valve located at any convenient place in the line. Data to permit evaluation of line losses are given in a subsequent section of this chapter.

Complex Refrigeration Cycles

The preceding sections have dealt only with refrigeration systems in which there is but one evaporator. When two or more evaporators are required, and the pressures differ in each, much greater opportunity is afforded the engineer for obtaining large economies through selection of one of the more complex cycles. Consider, for example, the refrigeration requirements of an air conditioning system which is to cool a very large volume of 90 F outside air down to a conditioned temperature of 40 F. A simple saturation system, operating with evaporator temperature less than 40 F, would accomplish the desired purpose, but at the expense of excess power requirements since 50 per cent of the sensible cooling load could instead be handled by an evaporator operating at a temperature somewhat less than 65 F. The more effective procedure in this case would be to place two direct expansion coils in series, the first operating at a temperature less than 65 F and the second, at a temperature below 40 F. In this way approximately one half of the total load would be picked up at the higher evaporator pressure and therefore need be raised through a much smaller thermal height. The theoretical advantage of such operation can be visualized from the increase in Carnot effectiveness; assuming that the condenser temperature is 100 F, the coefficient of performance (cop) of the low temperature evaporator is,

$$(\text{cop}) = (35 + 460) \div (100 - 35) = 7.63$$

while that of the higher temperature evaporator is,

$$(\text{cop}) = (60 + 460) \div (100 - 60) = 12.97.$$

Since the load is assumed to be equally distributed between the two evaporators the cop of the series system is the arithmetical average of the values for the two evaporators or, $(7.63 + 12.97) \div 2 = 10.30$. Thus use of the series cycle would afford a theoretical saving in power of approximately one third.

One common fallacy, with respect to complex systems, is the misconception that a high evaporator pressure necessarily means a low power requirement. In many instances operating conditions will require use of series evaporators, but in a cycle for which the vapor leaving the higher pressure evaporator must be throttled to the pressure of the low pressure evaporator before entering the compressor, there obviously is no advantage resulting from the higher operating pressure of the one evaporator since the refrigerant which it handles must be compressed through the same *lift* as though both evaporator pressures were the same. Consideration of this case brings out the fact that the effectiveness of a complete cycle depends upon the possibility of operating the system with suction vapor at different pressures. This can be accomplished through use of more than one compressor, or by means of special individual compressor arrangements which permit use of different suction pressures in the opposite ends of a double-acting machine, or introduction of vapor at two different pressures into the cylinder of a dual-effect compressor.

Compound Compression Cycles

In large systems the compression process can be carried out in steps as the refrigerant passes through a number of cylinder ends arranged for operation in series. Thermodynamically the advantage of compound compression arises from the fact that intercoolers can be placed between the stages of compression to extract heat from the vapor and thereby cause the over-all compression process to more closely approach the ideal condition of isothermal compression. Essentially, such intercoolers—whether of the water or the flash refrigerant type—serve the same purpose as a cooling jacket, but with greater effectiveness because of the more satisfactory heat transfer conditions.

Multiple Expansion Valves

In the simple saturation cycle the saturated liquid entering the expansion valve commences to vaporize as soon as its pressure starts to drop. The vapor produced during the expansion process has no further use, in terms of refrigerating effect, since it has already picked up its latent heat of vaporization as a result of heat which it has extracted from the unvaporized residue. Thus the instant such vapor forms its usefulness is at an end and to allow such material to undergo a further drop in pressure is uneconomical. Unfortunately, however, there is no effective means of extracting vapor continuously during the expansion and re-compressing it. Thus in the simple cycle the flash vapor must necessarily be allowed to drop to evaporator pressure.

When a compound compression cycle is used there is at least one intermediate pressure at which flash vapor can be extracted. In such cases all refrigerant from the condenser can be dropped through a first expansion valve to the higher suction pressure and the flash vapor then extracted and returned to the compressor. Some of the resultant liquid refrigerant then passes through the high-pressure evaporator while the remainder proceeds through a second expansion valve in which its pressure is dropped to the valve corresponding to the low-pressure evaporator.

Pipe Sizes and Friction Losses

The effect on performance of pressure losses in the piping of a refrigerating system has already been discussed. In all cases frictional losses should be kept to a minimum and piping should be selected which will give the smallest loss consistent with over-all economy of the system. Actual losses vary, of course, with the physical characteristics of the particular refrigerants, but, by way of example, data will be given for one of the refrigerants, dichlorodifluoromethane (F-12) which finds wide use in air conditioning applications.

Tables 8, 9, and 10 give the pressure loss per 100 ft of piping (including an *average* number of fittings) for oil-free dichlorodifluoromethane; tabular values should be increased if oil is flowing with the refrigerant. For copper pipe the tables are for type L tubing and are based on the outside diameter.

Common practice fixes suction line velocities between 1,500 and 3,000 fpm while 2,000 to 3,500 fpm is the accepted range for discharged lines. Velocities higher than those indicated will result in noisy operation while lower velocity in the suction line may result in the loss of entrained oil. Refrigeration for air conditioning usually involves wide variation in load,