large compressors the loss in volumetric efficiency would more than offset any gain, and the capacity would be reduced. Somewhere between these two extremes, the exact point depending on the type of compressor, location of valves, etc., the two factors would offset one another.

Another somewhat unusual subcooling cycle allows cold refrigerant from the downstream side of the expansion valve to cool liquid refrigerant from the condenser down to the evaporator temperature. In such a case the *expansion* valve becomes a simple pressure-reducing valve since there is no vapor formation—hence no expansion—during passage of the liquid refrigerant through it. This system is theoretically identical, from the standpoint of thermodynamic effectiveness, with the simple saturation cycle.

Clearance and Conventional Volumetric Efficiency

Clearance, like displacement, is a characteristic—usually fixed—of a given compressor. In some cases clearance pockets are provided which place within the operator's control the ability to alter the clearance of the machine, but most moderate size compressors are built with fixed clearance. By definition the clearance is the percentage of the volume swept by the piston which is represented by spaces in the end of the cylinder (including valve spaces, etc.) when the piston is at the end of its stroke.

Because of the trapping of high pressure vapor in the clearance space, and its subsequent re-expansion, the suction valves of the compressor do not open until the piston has completed part of its stroke. Hence the volume of fresh vapor introduced into the compressor per stroke is less than the volume swept by the piston. The ratio of actual volume of fresh gas to swept volume is, by definition, the conventional volumetric efficiency, CVE. In equation form,

$$(\mathrm{CVE}) = 100 - V_c\left[\frac{v_s}{v_d} - 1\right] \tag{9}$$

where

CVE = conventional volumetric efficiency.

V_c = clearance, per cent of volume swept by piston, which is contained in spaces at end of cylinder when piston is at end of stroke (clearance includes valve spaces, etc.)

v_s = specific volume of gas at compressor inlet.

v_d = specific volume of gas at compressor discharge.

Values of v_s and v_d can be obtained directly or by calculation from the tables of properties of refrigerants.

Influence of Wiredrawing and of Piping Losses

The pressure drop which occurs during passage of the refrigerant through suction and discharge valves of the compressor is known as wiredrawing. Its effect is equivalent to that of a reduction in the evaporator pressure (but without the disadvantage of reduced refrigerating effect) and an increase in condenser pressure; hence it is undesirable. In analyzing a cycle, wiredrawing can be taken into account since it amounts to the interposition of a small *expansion valve* between the state point of vapor leaving the evaporator and that of vapor at start of the compression process; a similar equivalent valve can be considered as interposed at the end of the compression process and before the state point corresponding to the condition of the superheated vapor in the discharge line.

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,

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

while that of the higher temperature evaporator is,

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

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.