The work of compression is then given by the equation:

$$(\text{hp}) = [W_r (h_d - h_{vs}) + Q_j] \div 42.42 \qquad (8)$$

where

hp = horsepower.
h_d = specific enthalpy of vapor at compressor discharge.
h_{vs} = specific enthalpy of saturated vapor entering compressor.

From the form of Equation 8 it would appear that an increase in the energy loss to the jacket would result in a greater power requirement since the Q_j term is additive; actually, however, computation will readily show that the effect of an increase in the Q_j term is to *decrease* the horse power since the rate of decrease of enthalpy difference will, in every case, exceed the increase in heat loss. Thus compressor cooling is highly desirable as a method of reducing power consumption.

Influence of Superheating and Subcooling

The most common departure from conditions of the simple saturation cycle is that resulting from admission of superheated vapor to the compressor. Thermodynamically, superheat is undesirable since the enthalpy increase required to compress a vapor through a given pressure range increases with superheat. Further, superheated vapor leaving an evaporator is usually an indication that the suction pressure is lower than necessary. Under practical operating conditions, however, superheat is almost universally used as a means of assuring complete vaporization of the refrigerant going to the compressor. With modern compressors operating at high speed and with relatively small clearance space it is particularly necessary to avoid admission through the suction valves of liquid refrigerant.

Another common departure of actual systems from the simple saturation cycle occurs because of subcooling of refrigerant in the condenser. Thermodynamically such subcooling is advantageous since it increases the refrigerating effect without affecting the unit energy requirements of the compressor. Further, it can be shown that for a fixed ratio of condenser cooling water to refrigerant circulating rate the total compressor power requirements will be greater when operating simple saturation than when operating with maximum sub-cooling. What is even more surprizing is that the condenser pressure is lower for the sub-cooling cycle than for the saturation cycle; this condition results from the fact that, for the same load, the refrigerant flow rate is less when there is sub-cooling.

Because of the advantages attendant upon the use of sub-cooling, many methods are in use for obtaining some sub-cooling effect outside of the condenser. One common procedure is to use the cold vapor leaving the evaporator to cool the liquid flowing from condenser to expansion valve. In this case subcooling is realized at the expense of superheating the suction vapor and thermodynamic analysis will show at once that a definite loss will accrue. The only conditions under which such a system can be justified are those for which the vapor would, in any event, pick up a comparable degree of superheat while en route to the compressor; in this event the deliberate superheating, with useful subcooling as a result, is preferable to the useless superheating which would otherwise occur. In small compressors having large surface area in relation to volume, a definite gain is usually accomplished by such interchange, because for such units there would be considerable heating in any event. On very 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,

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

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 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.