CHAPTER I

REFRIGERATION CYCLES

1.1 INTRODUCTION

Most practical forms of refrigeration that utilize equipment for the recovery of the refrigerant may be classed broadly as compression systems. Whether this compression is accomplished by reciprocating, rotary, or centrifugal devices, by high-pressure steam ejectors (steam-jet system), or by evaporation from a secondary fluid (absorption system) makes no fundamental difference; all are compression systems basically.

By far the most important systems commercially are those which use a refrigerant alternating between the vapor and the liquid phases. This class also contains the reciprocating, rotary, centrifugal, steam-jet, and absorption systems. These systems, operating cyclically between two pressures and using a two-phase working medium, may, in the broadest terminology, be classed as vapor-compression systems. From a practical standpoint, however, such an all-inclusive classification is undesirable.Therefore the absorption and steam-jet systems are, for clarity, almost always so designated, and the term "vapor-compression system" is reserved for reciprocating and rotary systems using a two-phase refrigerant. The centrifugal system, though differing from the reciprocating and rotary units only in that it is not positive in displacement, is usually classed separately for convenience.

1.2 PRINCIPLES OF REFRIGERATION

All refrigerating mechanisms are designed and built around several basic thermal laws:

1. Fluids absorb heat while changing from a liquid state to a gaseous state.

2. The temperature at which a change of state occurs is constant during the change, but this temperature will vary directly with the pressure.

3. Heat will flow only from a body at a certain temperature to a body at a lower temperature.

4. Inselecting metallic parts of the cooling and condensing units, metals selected should have a high heat conductivity.

5. Heat energy and other forms of energy are mutually convertible.

Refrigeration is the science and art of producing and maintaining temperatures below that of the surrounding atmosphere. Low temperatures can be obtained in various ways, as follow:

1. Allowing a phase change to take place in such a way as to extract heat, for example, the evaporation of a liquid such as water or ammonia or the melting of ice or the solution of a salt.

2. Expanding a compressed gas or vapor so that

it does external work.

3. Expanding a gas in the Joule-Thomson way.

4. Desorption of a gas.

5. Demagnetization of a solid.

6. Passage of an electric current through a bimetallic junction (Peltier effect).

In fact, any reversible change involving the expenditure of work can be utilized to absorb heat and produce low temperatures. Method 1 is that most commonly used for commercial refrigeration. It is also the basis for the use of the large number of cooling mixtures, of which salt and ice is one of the commonest, that are useful for intermittent cooling on a small scale. Methods2 and 3 are used for reaching very low temperatures on an industrial scale. Methods 4 and 5 have been used for the production of very low temperatures in the laboratory; 5, in particular, has been applied to the production of extremely low temperatures approaching absolute zero. Method 6 is mentioned merely as a possible method that has never been applied in practice. Any one of these processes can be considered as occurring (1) in an adiabatic system, in which case the temperature of the system itself will fall; (2) isothermally, with an accompanying influx of heat from the surroundings; or (3) by a combination of both these limiting conditions.

The continuous production of cold by purely mechanical means, such as the use of compression in a cycle involving alternate condensation of a vapor to a liquid and revaporization, is generally known as "mechanical refrigeration." This is by far the most important method of maintaining low temperatures. Methods of producing temperatures just a little under that of the atmosphere by allowing water to evaporate without any mechanical aid from a pump is very important for air conditioning and water cooling. Methods for the production of very low temperatures (say less than -100 F.) are somewhat special.

1.3 THE CARNOT CYCLE

The Carnot cycle is an ideal, thermodynamically reversible cycle, first investigated by Sadi Carnot in 1824 as a measure of the maximum possible conversion of heat energy into mechanical energy. In its reversed form it is used as a measure of the maximum performance of refrigeration equipment. Although it cannot be applied in an actual machine because of the impossibility of obtaining a completely reversible engine, it is nevertheless extremely valuable as a criterion of inherent limitations.

The Carnot cycle, shown graphically in Figure 1.3.1 on temperature-entropy coordinates, consists of an adiabatic expansion and an isothermal expansion followed by an adiabatic compression and an isothermal compression to form a closed cycle. Since the areas on the temperature-entropy diagram represent actual heat quantities supplied or rejected, and since all paths are here parallel to the coordinate lines, it is simplest to analyze the cycle on these coordinates. The Carnot cycle when operating as a heat engine will have the efficiency

$$\text{efficiency} = \frac{\text{net work}}{\text{heat supplied}}$$

$$= \frac{\text{heat supplied} - \text{heat rejected}}{\text{heat supplied}}$$

$$= \frac{(T_a - T_b)(s_b - s_a)}{T_a(s_b - s_a)} = \frac{T_a - T_b}{T_a}$$

where T denotes absolute temperature and s entropy.

Reference to Figure 1.3.1 shows that the refrigeration produced in the reversed Carnot cycle is $T_b(s_b - s_a)$ Btu per pound, and the net work supplied is $(T_a - T_b)(s_b - s_a)$ Btu per pound. Thus the coefficient of performance is

$$\text{COP} = \frac{\text{heat absorbed}}{\text{heat equivalent of net work supplied}}$$

$$= \frac{T_b(s_b - s_a)}{(T_a - T_b)(s_b - s_a)}$$

$$= \frac{T_b}{T_a - T_b}$$

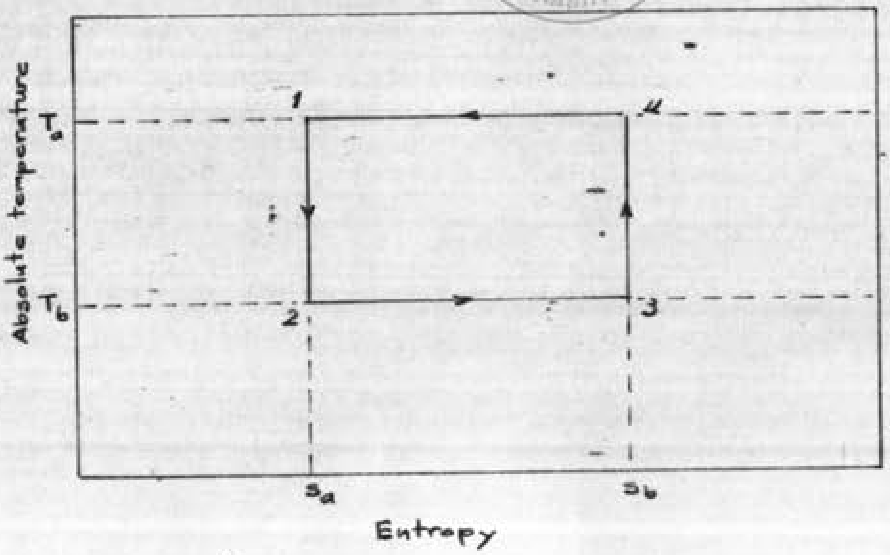


FIGURE 1.3.1[8] Process of Carnot refrigeration cycle.

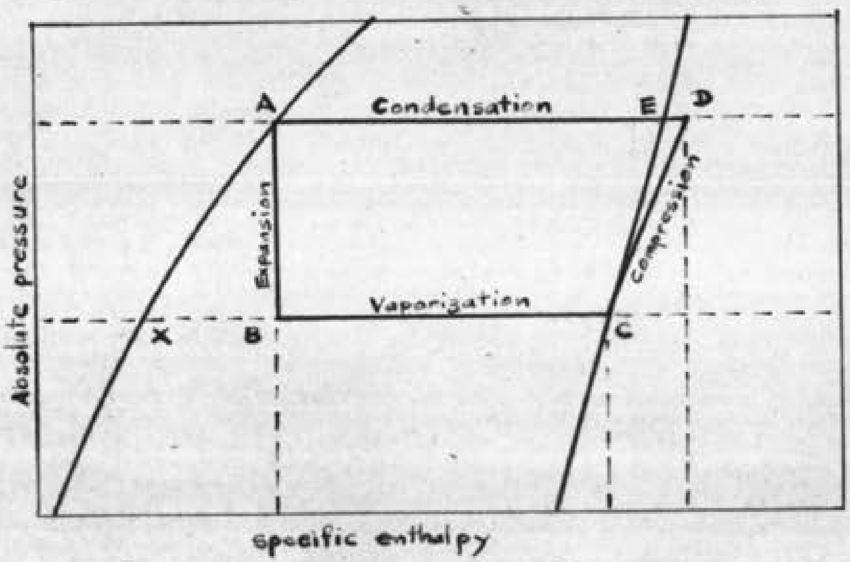


FIGURE 1.4.1[3] Pressure-enthalpy diagram of a simple saturated cycle.

1.4 THE SIMPLE SATURATED REFRIGERATING CYCLE

A simple saturated refrigerating cycle is a theoretical cycle wherein it is assumed that the refrigerant vapor leaves the evaporator and enter the compressor as a saturated vapor (at the vaporizing temperature and pressure) and the liquid leaves the condenser and enters the refrigerant control as a saturated liquid (at the condensing temperature and pressure). Although the refrigerating cycle of an actual refrigerating machine will usually deviate somewhat from the simple saturated cycle, the analysis of a simple saturated cycle is nonetheless worthwhile. In such a cycle, the fundamental processes which are the basis of every actual vapor compression refrigerating cycle are easy to identify and understanding. Furthermore, by using the simple saturated cycle as a standard against which actual cycles may be compared, the relative efficiency of actual refrigerating cycles at various operating conditions can be readily determined.

A simple saturated cycle is plotted on a pressure-enthalpy diagram in Figure 1.4.1. The points A, B, C, D, and E on the pressure-enthalpy diagram correspond to points in the refrigerating system as shown on the flow diagram in Figure 1.4.2.

At point A, the refrigerant is a saturated liquid in the condenser at the condensing pressure and temperature, and its properties, the values of pressure,

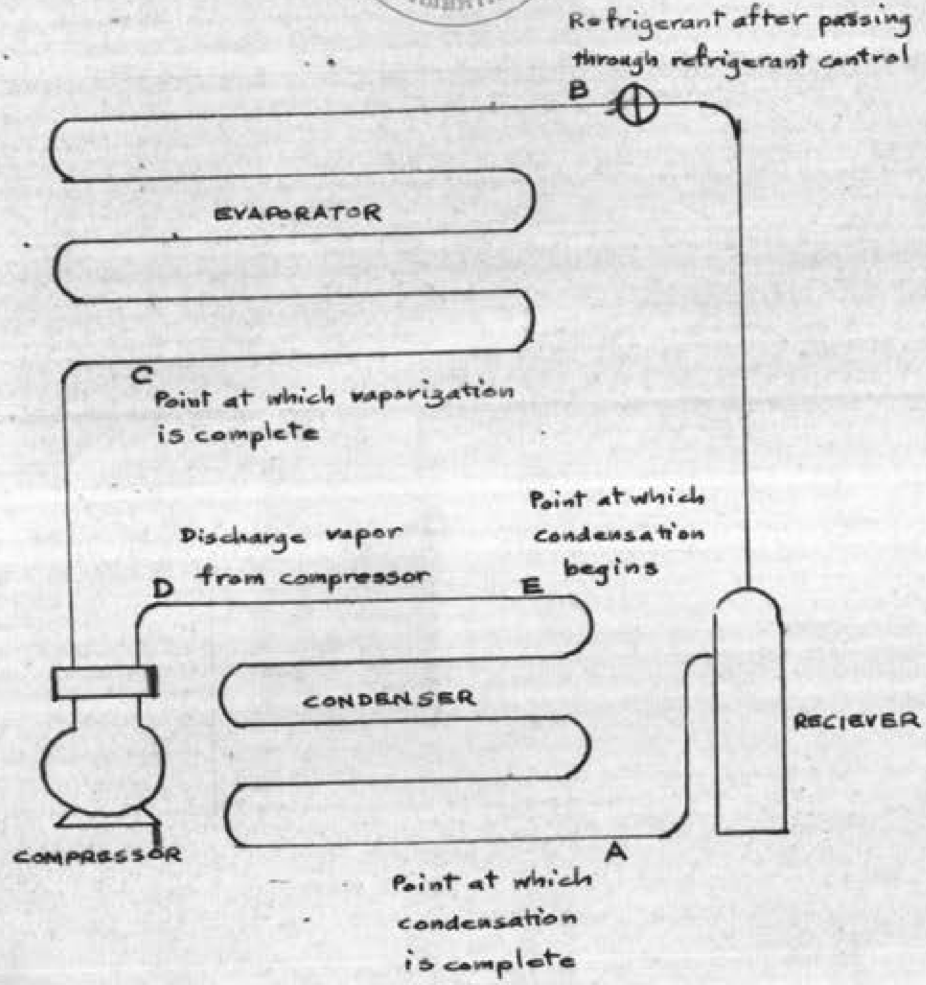


FIGURE 1.4.2[3] Flow diagram of a simple saturated cycle.

temperature,and enthalpy may be read directly from the pressure-enthalpy chart. Since the refrigerant is always a saturated liquid at point A, point A will always fall somewhere along the saturated liquid curve and can be located on the pressure-enthalpy chart if either pressure, temperature, or enthalpy is known. Usually in actual practice, either pressure, temperature, or both will be measurable.

1.4.1 THE EXPANSION PROCESS

The process described by the initial and final state points A-B occurs in the refrigerant control when the pressure of the liquid is reduced from the condensing pressure to the evaporating pressure as the liquid passes through the control. When the liquid is expanded into the evaporator through the orifice of the control, the temperature of the liquid is reduced from the condensing temperature to the evaporating temperature by the flashing into vapor of a small portion of the liquid.

Process A-B is a throttling type of adiabatic expansion, frequently called "wire-drawing", in which the enthalpy of the working fluid does not change during the process. This type of expansion occur whenever a fluid is expanded through an orifice from a high pressure to a lower pressure. It is assumed to take place without the gain or loss of heat through the piping or valve and without the performance of work.

1.4.2 THE VAPORIZING PROCESS

The process B-C is the vaporization of the refrigerant in the evaporator. Since vaporization take place at a constant temperature and pressure, B-C is both isothermal and isobaric. Therefore,point C is located on the pressure-enthalpy chart by following the lines of constant pressure and constant temperature from point B to the point where they intersect the saturated vapor curve. At point C the refrigerant is completely vaporized and is a saturated vapor at the vaporizing temperature and pressure.

The enthalpy of the refrigerant increases during process B-C as the refrigerant flow through the evaporator and absorbs heat from the refrigerated space. The quantity of heat absorbed by the refrigerant in the evaporator (refrigerating effect) is the differente between the enthalpy of the refrigerant at point B and C. Thus, if h_a, h_b, h_c, h_d, h_e, and h_x represent the enthalpies of the refrigerant at points A,B,C,D,E,and X, respectively,then

$$q_1 = h_c - h_b$$

where q_1 is the refrigerating effect in Btu/lb.

But since h_b is equal to h_a, then

$$q_1 = h_c - h_a$$

On the pressure- enthalpy diagram, the distance between point X and point C represents the total latent heat of vaporization of 1 lb of refrigerant at the vaporizing pressure. Therefore, since the distance B-C is the useful refrigerating effect, the difference between X-C and B-C, which is the distance X-B, is the loss of the refrigerating effect.

1.4.3 THE COMPRESSION PROCESS

In the simple saturated cycle, the refrigerant undergoes no change in condition while flowing through the suction line from the evaporator to the compressor. Process C-D takes place in the compressor as the pressure of the vapor is increased by compression from the vaporizing pressure to the condensing pressure. For the simple saturated cycle, the compression process, C-D, is assumed to be isentropic. An isentropic compression is a special type of adiabatic process which takes place without friction. It is sometimes described as a "frictionless adiabatic" or "constant-entropy" compression.

Since there is no change in the entropy of the vapor during process C-D, the entropy of the refrigerant at point D is the same as at point C. Therefore, point D can be located on the pressure-enthalpy chart by following the line of constant entropy from point C to the point where the constant entropy line intersects the line of constant pressure corresponding to the condensing pressure.

Work is done on the vapor during the compression process, C-D, and the enthalpy of the refrigerant is increased by an amount equal to the heat energy equivalent of the mechanical work done onthe vapor. The heat energy equivalent of the work done during the compression is often referred to as the heat of compression and is equal to the difference in the enthalpy of the refrigerant at point D and C. Thus, where q_2 is the heat of compression per pound of refrigerant circulated,

$$q_2 = h_d - h_c$$

The mechanical work done on the vapor by the piston during the compression may be calculated from the heat of compression. If w is the work done in foot-pounds per pound of refrigerant circulated and J is the mechanical energy equivalent of heat, then

$$w = q_2 \times J$$

or

$$w = J(h_d - h_c)$$

As a result of absorbing the heat of compression, the hot vapor discharged from the compressor is in a superheated condition, that is, its temperature is greater than the saturation temperature corresponding to its pressure. Thus, before the vapor can be condensed, the superheat must be removed and the temperature of the vapor lowered from the discharge temperature to the saturation temperature corresponding to its pressure.

1.4.4 THE CONDENSING PROCESS

Usually, both processes D-E and E-A take place in the condenser as the hot gas discharged from the compressor is cooled to the condensing temperature and condensed. Process D-E occurs in the upper part of the condenser and to some extent in the hot gas line. It represents the cooling of the vapor from the discharge temperature to the condensing temperature as the vapor rejects heat to the condensing medium. During process D-E, the pressure of the vapor remains constant and point E is located on the pressure-enthalpy chart by following a line of constant pressure from point D to point where the constant pressure line intersects the saturated vapor curve.

The quantity of sensible heat (superheat) removed from 1 lb of vapor in the condenser in cooling the vapor from the discharge temperature to the condensing temperature is the difference between the enthalpy of the refrigerant at point D and the enthalpy at point E (h_d-h_e).

Process E-A is the condensation of the vapor in the condenser. Since condensation takes place at a constant temperature and pressure, process E-A follows along lines of constant pressure and temperature from point E to point A. The heat rejected to the condensing medium during process E-A is the difference between the enthalpy of the refrigerant at points E and A (h_e-h_a).

On returning to point A, the refrigerant has completed one cycle and itsproperties are the same as those previously described for point A.

Since both processes D-E and E-A occur in the condenser, the total amount of heat rejected by the refrigerant to the condensing medium in the condenser is the sum of the heat quantities rejected during processes D-E and E-A. The total heat given up by the refrigerant at the condenser is the difference between the enthalpy of the superheated vapor at point D and the saturated liquid at point A. Hence,

$$q_3 = h_d - h_a$$

where q_3 = the heat rejected at the condenser per pound of refrigerant circulated.

If the refrigerant is to reach point A at the end of the cycle in the same condition as it left point A at the beginning of the cycle, the total heat rejected by the refrigerant to the condensing medium in the condenser must be exactly equal to the heat absorbed by the refrigerant at all other points in the cycle. In a simple saturated cycle, the refrigerant is heated at only two points in the cycle: (1) in the evaporator by absorbing heat from the refrigerated space (q_1) and (2) in the compressor by the heat of compression (q_2). Therefore,

$$q_3 = q_1 + q_2$$

1.5 THE ACTUAL REFRIGERATING CYCLE

The actual refrigerating cycle as applied in practice differs in several ways from thetheoretical cycle as follows:

1. Frequently the liquid refrigerant is subcooled before it is entered the expansion valve.

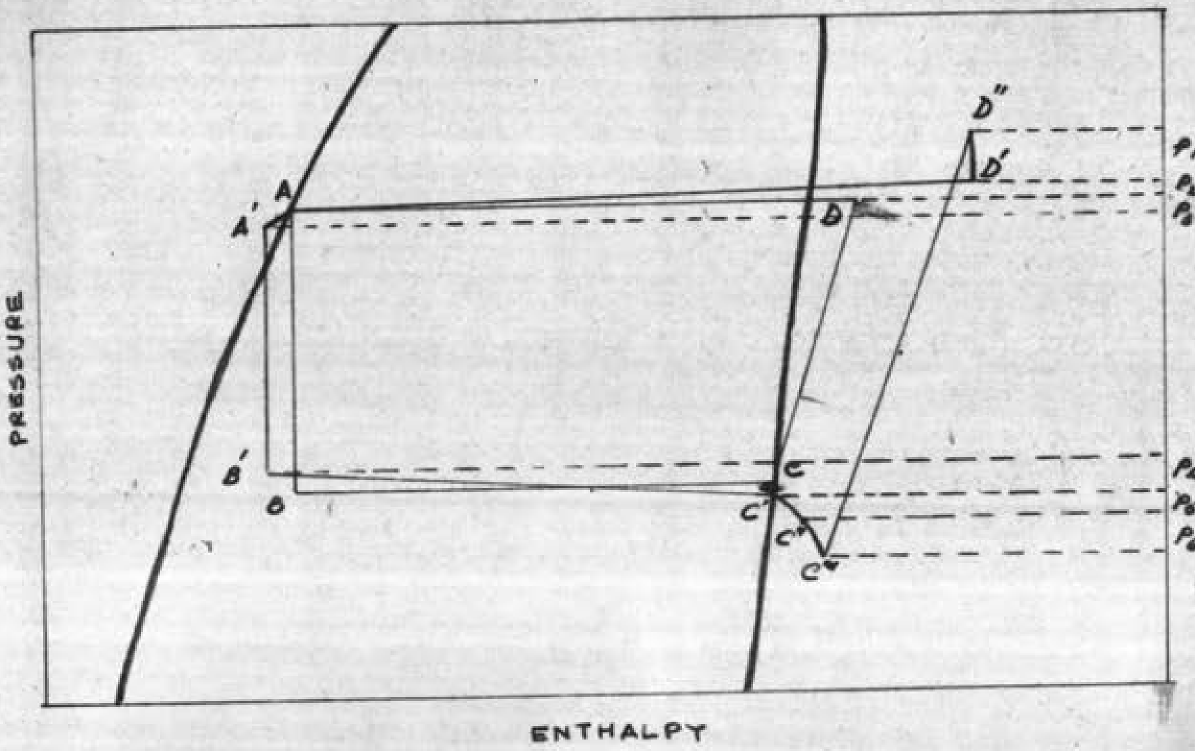
2. Usually the gas leaving the evaporator is superheated a few degrees before it enters the compressor. This superheating may occur as an result of the type of expansion control used or through a pickup of heat in the suction line between the evaporator and compressor.

3. Compression, although usually assumed to be isentropic, may actually prove to be neither isentropic nor polytropic.

007046 4. Both the compressor suction and discharge valves are actuated by pressure difference, and this process requires the actual suction pressure inside the compressor to be slightly below that of the evaporator and the discharge pressure to be above that of the condenser.

5. Although isentropic compression assumes no transfer of heat between the refrigerant and the cylinder walls, actually the cylinder walls are hotter than the incoming gases from the evaporator and colder than the compressed gases discharged to the condenser.

There are numerous other deviations from the theoretical cycle, but most of these are small and consist primarily of heat exchanges, either positive or ne-



Pressure drop

$D''D'$ - compressor discharge valves = p_1

$D'A$ - discharge line and condenser = p_2

AA' - liquid line = p_3

$B'C'$ - evaporator = p_4

$C'C''$ - suction line = p_5

$C''C'''$ - compressor suction valves = p_6

FIGURE 1.5.1[3] Pressure-enthalpy diagram of an actual refrigeration cycle illustrating effect of subcooling, superheating, and losses in pressure. A simple saturated cycle is drawn in for comparison.

gative, between parts of the system and the surrounding air. Two additional deviations, not negligible, are the pressure drop in long suction and liquid line piping and any vertical differences in head created by locating the evaporator and condenser at different elevations.

Figure 1.5.1 shows a pressure-enthalpy diagram of a typical refrigeration cycle, which illustrated the combined effects of pressure drop, subcooling, and superheating, is compared to the pressure-enthalpy diagram of the simple saturated cycle.

Line B'-C' represents the vaporizing process in the evaporator during which the refrigerant undergoes a drop in pressure of p_4. Line C'-C" represents the drop in pressure p_5 experienced by the suction vapor in flowing through the suction line from the evaporator to the compressor inlet. Line C"-C" represents the drop in pressure p_6 that the suction vapor undergoes in flowing through the suction valves and passages of the compressor into the cylinder. Line C"-D" represents the compression process for the cycle undergoing the pressure drops. Notice that the vapor in the cylinder is compressed to a pressure considerably above the average condensing pressure. This is necessary in order to force the vapor out of the cylinder through the discharge valves against the condensing pressure and against the additional pressure occasioned by the spring-loading of the discharge valves. Line D"-D' represents the drop in pressure p_1 required to force the discharge valves open against the spring-loading and to force the vapor out through the discharge valves and passages of the compressor into the discharge line. Line D'-A represents the drop in pressure p_2, resulting from the flow of the refrigerant through the discharge line and condenser. Line A-A' represents the pressure drop p_3, resulting from the flow of the refrigerant through the receiver tank and liquid line, and line also shows the removal of heat of liquid or subcooling.

1.6 METHEMATICAL ANALYSIS OF VAPOR COMPRESSION REFRIGERATION

1.6.1 REFRIGERATING EFFECT

Refrigerating effect is the amount of heat absorbed by the refrigerant in its travel through the evaporator. In Figure 1.4.1, this effect is represented by the equation

$$q_1 = h_c - h_a$$

In addition to the latent heat of vaporization it may include any heat of superheat absorbed in the evaporator.

1.6.2 REFRIGERANT FLOW RATE

Weight of refrigerant circulated per minute per ton of refrigeration may be determined by dividing the amount of heat in Btu absorbed per minute per ton of refrigeration by the refrigerating effect. Thus

$$m = \frac{200}{q_1}$$

or $$m = \frac{200}{h_c - h_a}$$

where m is the weight of refrigerant to be circulated per minute per ton.

1.6.3 THEORETICAL HORSEPOWER

The theoretical horsepower per ton of refrigeration is the horsepower theoretically required to compress the refrigerant. It does not take into consideration the mechanical efficiency or the volumetric efficiency of the compressor. Compression of the refrigerant is usually considered to be isentropic, since the degree of deviation is usually not great unless the compressor is cooled by external means.

If isentropic compression is assumed,

$$hp = m(h_d - h_c)\left(\frac{778}{33000}\right)$$

or $$hp = \frac{m(h_d - h_c)}{42.42}$$

where hp is theoretical horsepower per ton, 778 isthe mechanical equivalent of heat and 33000 is the foot-pounds per minute horsepower.

1.6.4 PISTON DISPLACEMENT

The theoretical piston displacement per minute per ton of refrigeration may be found by multiplying the weight of refrigerant to be circulated per minute per ton of refrigeration by the specific volume of the refrigerant gas, $(v_g)_c$, at its entrance to the compressor, thus

$$V_{pt} = \frac{200}{(h_c - h_a)}(v_g)_c$$

where V_{pt} is the theoretical piston displacement in cubic feet per minute per ton.

The piston displacement of a reciprocating compressor is the total cylinder volume swept through by the piston in any certain time interval and is usually expressed in cubic feet per minute. For any single-acting, reciprocating compressor, the piston displacement is computed as follow:

$$V_p = \frac{\pi D^2 \times L \times N \times n}{4 \times 1728}$$

where V_p = the piston displacement in cubic feet per minute

D = the diameter of the cylinder(bore) in inches

L = the length of stroke in inches

N = revolutions of the crankshaft per minute (rpm)

n = the number of cylinders

1.6.5 COEFFICIENT OF PERFORMANCE

In reference to power-generation equipment, efficiency is defined as the work or useful output divided by the thermal energy required to produce that output. In a refrigeration system, the useful output is the heat absorbed at refrigeration temperature, and the input is the work required to produce that refrigeration. The ratio of these values in appropriate energy or power units is called the coefficient of performance,and is a real measure of the effectiveness of the system.

$$COP = \frac{\text{Refrigerating effect}}{\text{Work input}}$$

1.6.6 VOLUMETRIC EFFICIENCY

This term is of significance with positive displacement machinery such as reciprocating and rotary compressor. It is defined as the ratio of actual weight of vapor handled per unit time by the compressor compared to the weight which could be handled by the piston displacement of the compressor computed at the temperature and pressure conditions the vapor possesses inthe suction line of the compressor. Volumetric efficiency is affected by many factors, of which clearance is perhaps the most significant.

The actual volume of suction vapor compressed per minute is the actual displacement of the compressor. The ratio of the actual displacement of the compressor to its piston displacement is known as the total or real volumetric efficiency of the compressor. Thus

$$E_v = \frac{V_a}{V_p} \times 100$$

where E_v = the total volumetric efficiency

V_a = actual volume of suction vapor compressed per minute

V_p = the piston displacement of the compressor