REFRIGERATION AND AIRCONDITIONING

REFRIGERATION AND AIR CONDITIONING

UNIT- 2

Vapour compression system: simple cycle -comparison with Carnot cycle - theoretical, actual and reactive - COP effect of operating parameters on COP - wet, dry and superheated compression - under cooling - actual cycle representation on TS and PH diagrams simple problems. Advanced vapour compression systems - multistage vapour compression systems - flash chamber multiple compression and evaporation systems cascading - simple problems.

UNIT-3

Vapour absorption systems: simple, cycles - actual cycle - ammonia water and lithium bromide water systems - COP - electrolux system. Refrigerant and their properties: Nomenclature - suitability of refrigerants for various applications - unconventional refrigeration methods- Vortex tube, steam-jet, magnetic (cryogenics) refrigeration and thermoelectric refrigeration - applied refrigeration house hold refrigerators - unit air conditioners and water coolers - ice plant cold storage.

UNIT– II

Vapour Compression Refrigeration Systems

2.1 Introduction

A vapour compression refrigeration system is an improved type of air refrigeration system in which a suitable working substance, termed as refrigerant, is used. It condenses and evaporates at temperatures and pressures close to the atmospheric conditions. The refrigerants, usually, used for this purpose are ammonia (NH3), carbon dioxide (CO2) and sulphur dioxide (SO2). The refrigerant used, does not leave the system, but is circulated throughout the system alternately condensing and evaporating. In evaporating, the refrigerant absorbs its latent heat from the brine (salt water) which is used for circulating it around the cold chamber. While condensing, it gives out its latent heat to the circulating water of the cooler. The vapour compression refrigeration system is, therefore a latent heat pump, as it pumps its latent heat from the brine and delivers it to the cooler.

The vapour compression refrigeration system is now-a-days used for all purpose refrigeration. It is generally used for all industrial purposes from a small domestic refrigerator to a big air conditioning plant.

2.2 Advantages and Disadvantages of vapour Compression Refrigeration System over Air Refrigeration System

Following are the advantages and disadvantages of the vapour compression refrigeration system over air refrigeration system:

Advantages

1. It has smaller size for the given capacity of refrigeration.
2. It has less running cost.
3. It can be employed over a large range of temperatures.
4. The coefficient of performance is quite high.

Disadvantages

1. The initial cost is high

2. The prevention of leakage of the refrigerant is the major problem in vapour compression system

2.3 Mechanism of a Simple Vapour Compression Refrigeration System

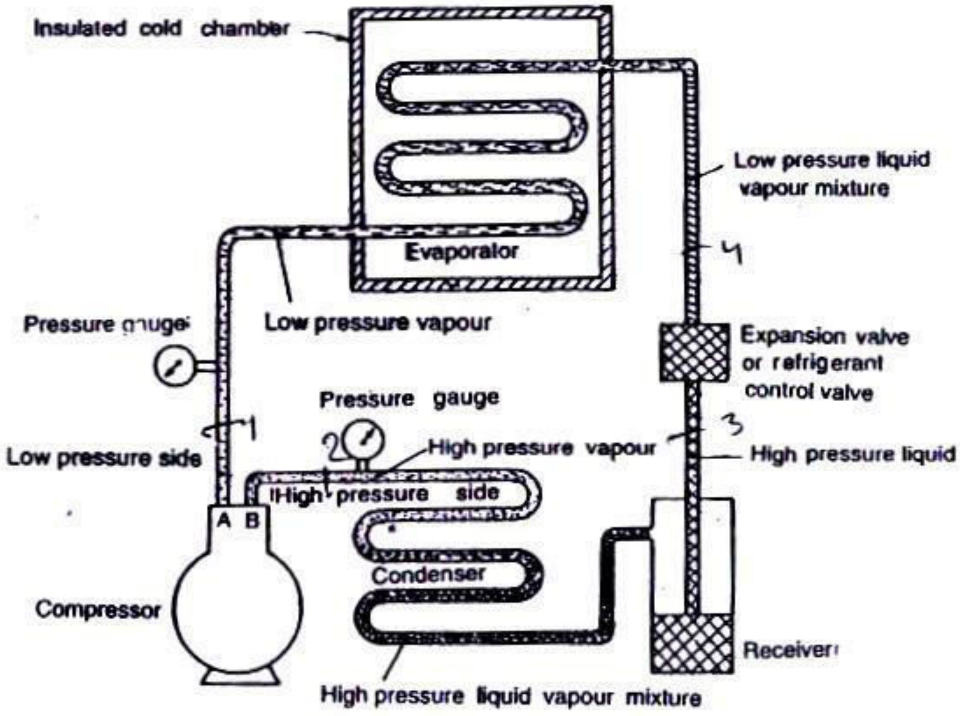


Fig. 2.1 shows the achematic diagram of a simple vapour compression refrigeration system. It consists of the following five essential parts :

1. *Compressor*. The low pressure and temperature vapou refrigerant from evaporator is drawn into the compressor through the inlet or suction valve A, where it is compresses to a high pressure and temperature. This high pressure and temperature vapour refrigerant is discharge the condenser through the delivery or discharge valve B.

2. *Condenser*. The condenser or cooler consists of coils of pipe in which the high pressure and temperature vapour refrigerant is cooled and condensed. The refrigerant, while passing through the condenser, gives up its latent heat to the surrounding condensing medium which is normally air or water.

3. *Receiver*. The condensed liquid refrigerant from the condenser is stored in a vessel known as receiver from where it is supplied to the evaporator through the expansion valve or refrigerant control valve.

4. *Expansion valve*. It is also called throttle valve or refrigerant control valve. The function of the expansion valve is to allow the liquid refrigerant under high pressure and temperature to pass at a controlled rate after reducing its pressure and temperature. Some of the liquid refrigerant evaporates as it passes through the expansion valve, but the greater portion is vaporised in the evaporator at the low pressure and temperature.

5. *Evaporator*. An evaporator consists of coils of pipe in which the liquid-vapour refrigerant at low pressure and temperature is evaporated and changed into vapour refrigerant at low pressure and temperature. In evaporating, the liquid vapour refrigerant absorbs its latent heat of vaporisation from the medium (air, water or brine) which is to be cooled.

Note : In any compression refrigeration system, there are two different pressure conditions. One is called the high pressure side and other is known as low pressure side. The high pressure side includes the discharge line (i.e. piping from the evaporator to the suction valve A).

2.4 Pressure-Enthalpy (p-h) Chart

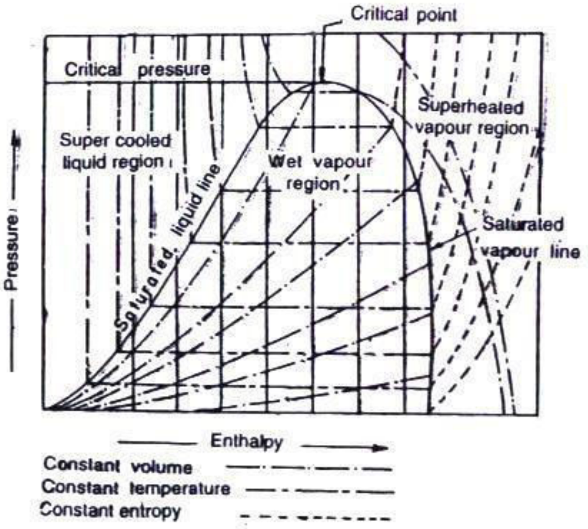


Fig 2.2 pressure enthalpy [p-h] chart

The most convenient chart for studying the behavior of a refrigerant is the p-h chart, in which the vertical ordinates represent pressure and horizontal ordinates represent enthalpy (i.e. total heat). A typical chart is shown in Fig. 2.2, in which a few important lines of the complete chart are drawn. The saturated liquid line and the saturated vapour line merge into one another at the critical point. A saturated liquid is one which has a temperature equal to the saturation temperature corresponding to its pressure. The space to the left of the saturated liquid line will, therefore, be sub-cooled liquid region. The space between the liquid and the vapour lines is called wet vapour region and to the right of the saturated vapour line is a superheated vapour region.

In the following pages, we shall drawn the p-h chart along with the T-s diagram of the cycle.

2.5 Types of Vapour Compression Cycles

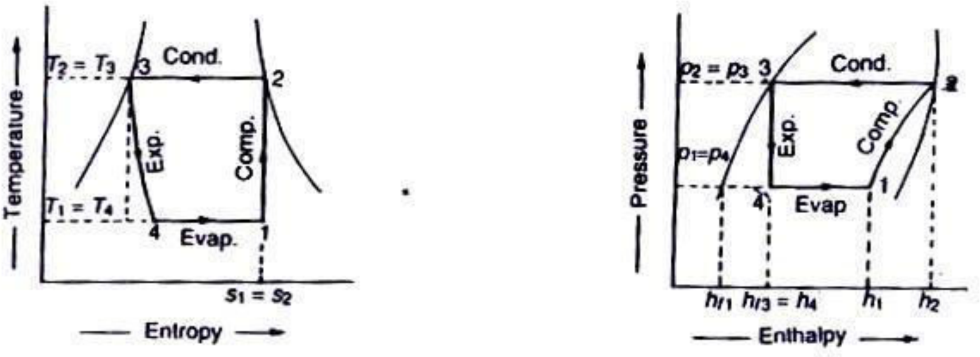
We have already discussed that vapour compression cycle essentially consists of compression, condensation, throttling and evaporation. Many scientists have focussed their attention to increase the coefficient of performance of the cycle. Through there are many cycles, yet the following are important from the subject point of view :

1. Cycle with dry saturated vapour after compression,
2. Cycle with wet vapour after compression,
3. Cycle with superheated vapour after compression,
4. Cycle with superheated vapour before compression, and
5. Cycle with undercooling or subcooling of refrigerant.

Now we shall discuss all the above mentioned cycles, one by one, in the following pages.

2.6 Theoretical Vapour Compression Cycle with Dry Saturated Vapour after Compression

A vapour compression cycle with dry saturated vapour after compression is shown on T-s and p-h diagrams in Fig. 2.3 (a) and (b) respectively. At point 1, let T_1,p_1 and s_1 be the temperature, pressure and entropy of the vapour refrigerant respectively. The four process of the cycle are as follows :



(*a*) *T-s* diagram.

(*b*) *p-h* diagram.

Fig 2.3: theoretical vapour compression cycle with dry saturated vapour after compression

1. *Compression process*. The vapour refrigerent at low pressure p_1 and temperature T_1 is compressed isentropically to dry saturated vapour as shown by the vertical line 1-2 on T-s diagram and by the curve 1-2 on p-h diagram. The pressure and temperature rises from p_1 to p_2 and T_1 to t_2 respectively.

The work done during isentropic compression per kg of refrigerant is given by

$$w = h_2 - h_1$$

where h_1 = Enthalpy of vapour refrigerant at temperature T_1, i.e. at suction of the compressor, and

h_2 = Enthalpy of the vapour refrigerant at temperature T_2, i.e. at discharge of the compressor.

2. *Condensing process*. The high pressure and temperature vapour refrigerant from the compressor is passed through the condenser where it is completely condensed at constant pressure p_2 and temperature T_2, as shown by the

horizontal line 2-3 on T-s and p-h diagrams. The vapour refrigerant is changed into liquid refrigerant. The refrigerant, while passing through the condenser, gives its latent heat to the surrounding condensing medium.

3. *Expansion process.* the liquid refrigerant at pressure $p_3 = p_2$ and temperature $T_3 = T_2$ is expanded by *throttling process through the expansion valve to a low pressure $p_4 = p_1$ and temperature $T_4 = T_1$, as shown by the curve 3-4 on T-s diagram and by the vertical line 3-4 on p-h diagram. We have already discussed that some of the liquid refrigerant evaporates as it passes through the expansion valve, but the greater portion is vaporised in the evaporator. We know that during the throttling process, no heat is absorbed or rejected by the liquid refrigerant.

Notes : (a) In case an expansion cylinder is used in place of throttle or expansion valve to expand the liquid refrigerant, then the refrigerant will expand isentropically as shown by dotted vertical line on T-s diagram in Fig. 2.3 (a). The isentropic expansion reduces the external work being expanded in running the compressor and increases the refrigerating effect. Thus, the net result of using the expansion cylinder is to increase the coefficient of performance.

Since the expansion cylinder system of expanding the liquid refrigerant is quite complicated and involves greater initial cost, therefore its use is not justified for small gain in cooling capacity. Moreover, the flow rate of the refrigerant can be controlled with throttle valve which is not possible in case of expansion cylinder which has a fixed cylinder volume.

(b) In modern domestic refrigerators, a capillary (small bore tube) is used in place of an expansion valve.

4. *Vaporising process.* The liquid-vapour mixture of the refrigerant at pressure $p_4 = p_1$ and temperature $T_4 = T_1$ is evaporated and changed into vapour refrigerant at constant pressure and temperature, as shown by the horizontal line 4-1 on T-s and p-h diagrams. During evaporation, the liquid-vapour refrigerant absorbs its latent heat of vaporisation from the medium (air, water or brine) which is to be cooled. This heat which is absorbed by the refrigerant is called refrigerating effect and it is briefly written as R_E. The process of vaporisation continues upto point 1 which is the starting point and thus the cycle is completed.

We know that the refrigerating effect or the heat absorbed or extracted by the liquid-vapour refrigerant during evaporation per kg of refrigerant is given by

$$R_E = h_1 - h_4 = h_1 - h_{f3} \qquad \ldots (\because h_{f3} = h_4)$$

where h_{f3} = Sensible heat at temperature T_3, i.e. enthalpy of liquid refrigerant leaving the condenser.

It may be noticed from the cycle that the liquid-vapour refrigerant has extracted heat during evaporation and the work will be done by the compressor for isentropic compression of the high pressure and temperature vapour refrigerant.

∴ Coefficient of performance,

$$\text{C.O.P.} = \frac{\text{Refrigerating effect}}{\text{Work done}} = \frac{h_1 - h_4}{h_2 - h_1} = \frac{h_1 - h_{f3}}{h_2 - h_1}$$

Example 2.1 In an ammonia vapour compression system, the pressure in the evaporator is 2 bar. Ammonia at exit is 0.85 dry and at entry its dryness fraction is 0.19. During compression, the work done per kg of ammonia is 150 kJ. Calculate the C.O.P. and the volume of vapour entering the compressor per minute, if the rate of ammonia circulation is 4.5 kg/min. The latent heat and specific volume at 2 bar are 1325 kJ/kg and 0.58 m^3/kg respectively.

Solution. Given : $p_1 = p_4 = 2$ bar ; $x_1 = 0.85$; $x_4 = 0.19$; $w = 150$ kJ/kg ; $m_a = 4.5$ kg/min; $h_{fg} = 1325$ kJ/kg ; $v_g = 0.58$ m^3/kg C.O.P.

The T-s and p-h diagrams are shown in Fig.4.3 (a) and (b) respectively.

Since the ammonia vapour at entry to the evaporator (i.e. at point 4) has dryness fraction (x_4) equal to 0.19, therefore enthalpy at point 4,

$$h_4 = x_4 \times h_{fg} = 0.19 \times 1325 = 251.75 \text{ kJ/kg}$$

Similarly, enthaipy of ammonia vapour at exit i.e. at point 1,

$$h_1 = x_1 \times h_{fg} = 0.85 \times 1325 = 1126.25 \text{ kJ/kg}$$

∴ Heat extracted from the evaporator or refrigerating effect,

$R_E = h_1 - h_4 = 1126.25 - 251.75 = 874.5$ kJ/kg

We know that work done during compression,

$w = 150$ kJ/kg

$\therefore$ C.O.P. $= R_E/w = 874.5/150 = 5.83$ **Ans.**

Volume of vapour entering the compressor per minute

We know that volume of vapour entering the compressor per minute

= Mass of refrigerant / min × Specific volume

$= m_a \times v_g = 4.5 \times 0.58 = 2.61\ m^3$ / min **Ans.**

Example 2.2. The temperature limits of an ammonia refrigerating system are 25°C and -10°C. If the gas is dry at the end of compression, calculate the coefficient of performance of the cycle assuming no undercooling of the liquid ammonia. Use the following table for properties of ammonia:

Temperature (°C)	*Liquid heat (kJ/kg)*	*Latent heat (kJ/kg)*	*Liquid entropy (kJ/kg K)*
25	*298.9*	*1166.94*	*1.1242*
-10	*135.37*	*1297.68*	*0.5443*

Solution. Given : $T_2 = T_3 = 25^{\circ}C = 25 + 273 = 298\ K$; $T_1 = T_4 = -10^{\circ}C = 263 K$;

$h_{f3} = h_4 = 298.9$ kJ/kg ; $h_{fg2} = 1166.94$ kJ/kg ; $s_{f2} = 1.1242$ kJ/kg K ; $h_{f1} = 135.37$ kJ/kg ; $h_{fg1} = 1297.68$ kJ/kg ; $s_{f1} = 0.5443$ kJ/kg K

The T-s and p-h diagrams are shown in Fig. 2. 4 (a) and (b) respectively.

Let x_1 = Dryness fraction at point 1.

We know that entropy at point 1, Fig.

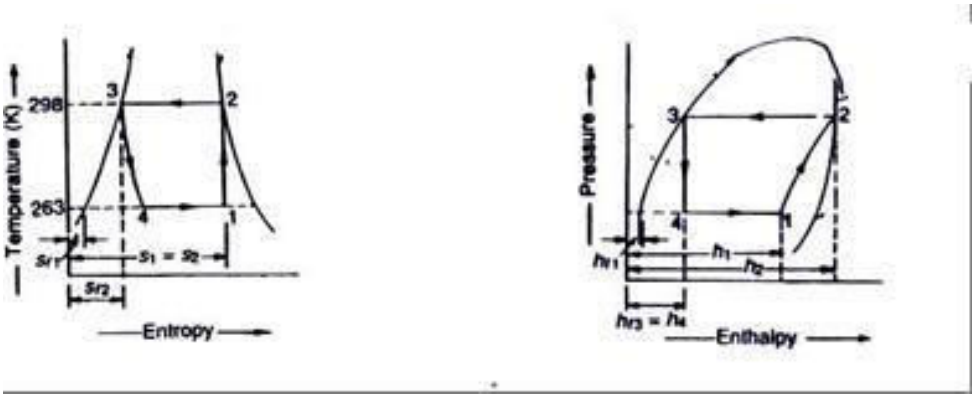
$$s_1 = s_{f1} + \frac{x_1 h_{fg1}}{T_1} = 0.5443 + \frac{x_1 \times 1297.68}{263}$$

$= 0.5443 + 4.934\, x_1$

Similarly, entropy at point 2,

$$s_2 = s_{f2} + \frac{h_{fg2}}{T_2} = 0.5443 + \frac{1166.94}{298} = 5.04$$

Since the entropy at point 1 is equal to entropy at point 2, therefore equating equations (i) and (ii) $0.5443 + 4.934\, x_1 = 5.04$ or $x_1 = 0.91$



We know that enthalpy at point 1,

$h_1 = hf_1 + x_1\, h_{fg1} = 135.37 + 0.91 \times 1297.68 = 1316.26$

kJ/kg and enthalpy at point 2,

$h_2 = h_{f2} + h_{fg2} = 298.9 + 1166.94 = 1465.84$ kJ/kg

∴ Coefficient of performance of the cycle

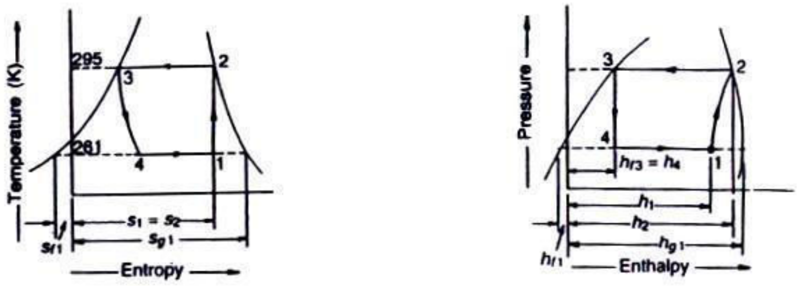
$$= \frac{h_1 - h_{f3}}{h_2 - h_1} = \frac{1316.26 - 298.9}{1465.84 - 1316.26} = 6.8 \textbf{ Ans.}$$

Example 2.3. A vapour compression refrigerator works between the pressure limits of 60 bar and 25 bar. The working fluid is just dry at the end of compression and there is no under-cooling of the liquid before the expansion valve. Determine : 1. C.O.P. of the cycle ; and 2. Capacity of the refrigerator if the fluid flow is at the rate of 5 kg/min.

Data :

Pressure (bar)	*Saturation temperature (K)*	*Enthalpy (kJ/kg)*		*Entropy (kJ/kg K)*	
		Liquid	*Vapour*	*Liquid*	*Vapour*
60	*295*	*151.96*	*293.29*	*0.554*	*1.0332*
25	*261*	*56.32*	*322.58*	*0.226*	*1.2464*

Solution. Given : $p_2 = p_= = 60$ bar ; $p_1 = p_4 = 25$ bar ; $T_2 = T_3 = 295$ K ; $T_1 = T_4 = 261$ K ; $h_{f3} = h_4 = 151.96$ kJ/kg ; $h_{f1} = 56.32$ kJ/kg ; $h_{g2} = h_2$ 293.29 kJ/kg ; $h_{g1} = 322.58$ kJ/kg ; $s_{f2} = 0.554$ kJ/kg K ; $s_{f1} = 0.226$ kJ/kg K ; $s_{g2} = 1.0332$ kJ/kg K; $s_{g1} = 1.2464$ kJ/kg K



(*a*) *T-s* diagram. (*b*) *p-h* diagram.

Fig. 4.5

Fig 2.5

1. *C.O.P. of the cycle*

The T-s and p-h diagrams are shown in Fig. 2.5 (a) and (b) respectively.

Let x_1 = Dryness fraction of the vapour refrigerant entering the compressor at point 1.

We know that entropy at point 1,

$s_1 = s_{f1} + x_1 s_{fg1} = s_{f1} + x_1 \left(s_{g1} s_{f1}\right)$ ...($s_{g1} = s_{f1} + s_{fg1}$)

$= 0.226 + x_1 (1.2464 - 0.226) = 0.226 + 1.0204\, x_1$ ... (i)

and entropy at point 2, $s_2 = 1.0332$ kJ/kg K ... (Given) ... (ii)

Since the entropy at point 1 is equal to entropy at point 2, therefore equating equations (i) and (ii),

$0.226 + 1.0204\, x_1 = 1.0332$ or $x_1 = 0.791$

we know that enthalpy at point 1,

$$h_1 = h_{f1} + x_1 h_{fg1} = h_{f1} + x_1\left(h_{g1} - f_{f1}\right) \qquad \ldots (\because h_{g1} = h_{f1} + h_{fg1})$$

$= 56.32 + 0.791\,(322.58 - 56.32) = 266.93$ kJ/kg

$\therefore$ C.O.P. of the cycle

$$= \frac{h_1 - h_{f3}}{h_2 - h_1} = \frac{266.93 - 151.96}{293.29 - 266.93} = 4.36 \textbf{ Ans.}$$

2. *Capacity of the refrigerator*

We know that the heat extracted or refrigerating effect produced per kg of refrigerant

$= h_1 - h_{f3} = 266.93 - 151.93 = 114.97$ kJ/kg

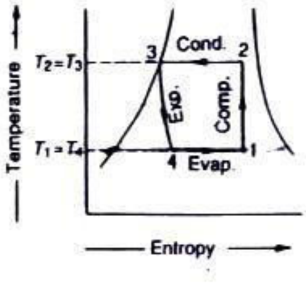
Since the fluid flow is at the rate of 5 kg/min, therefore total heat extracted

$= 5 \times 114.97 = 574.85$ kJ/min

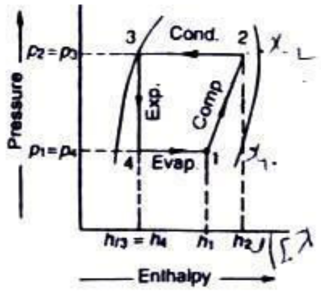
$\therefore$ Capacity of the refrigerator

$$= \frac{574.85}{210} = 2.74 \text{ TR } \textbf{Ans.} \qquad \ldots(\because 1 \text{ TR} = 210 \text{ kJ/min})$$

2.7. Theoretical Vapour Compression Cycle with Wet Vapour after Compression



(a) T-s diagram.



(b) p-h diagram.

A vapour compression cycle with wet vapour after compression is shown on T-s and p-h diagrams in Fig. 2.6 (a) and (b) respectively. In this cycle, the enthalpy at point 2 is found out with the help of dryness fraction at this point. The dryness fraction at points 1 and 2 may be obtained by equating entropies at points 1 and 2.

Now the coefficient of performance may be found out as usual from the relation,

$$\text{C.O.P.} = \frac{\text{Refrigerating effect}}{\text{Work done}} = \frac{h_1 - h_{f3}}{h_2 - h_1}$$

Compound Vapour Compression Refrigeration Systems

Introduction

In the previous chapter, we have discussed the simple vapour compression refrigeration system in which the low pressure vapour refrigerant from the evaporator is compressed in a single stage (or a single compressor) and then delivered to a condenser at a high pressure. But sometimes, the vapour refrigerant is required to be delivered at a very high pressure as in the case of low temperature refrigerating systems. In such cases either we should compress the vapour refrigerant by employing a single stage compressor with a very high pressure ratio between the condenser and evaporator or compress it in two or more compressors placed is series. The compression carried out in two or more compressors is called compound or multistage compression.

In vapour compression refrigeration systems, the major operating cost is the energy input to the system in the form of mechanical work. Thus any method of increasing coefficient of performance is advantageous so long as it does not involve too heavy an increase in other operating expenses, as well as initial plant cost and consequent maintenance.

Since the coefficient of performance of a refrigeration system is the ratio of refrigerating effect to the compression work, therefore the coefficient of performance can be increased either by increasing the refrigerating effect or by decreasing the compression work. A little consideration will show that in a vapour compression system, the compression work is greatly reduced if the refrigerant is compressed very close to the saturated vapour line. This can be achieved by compressing the refrigerant in more stages with intermediate intercooling. But it is economical only where the pressure ratio is considerable as would be the case when very low evaporator temperatures are desired or when high condenser temperature may be required. The compound compression is generally economical in large plants.

The refrigerating effect can be increased by maintaining the condition of the refrigerant in more liquid state at the entrance to the evaporator. This can be achieved by expanding the refrigerant very close to the saturated liquid line. It may be noted that by subcooling the refrigerant and by removing the flashed vapour, as they are during multistage expansion, the expansion can be brought close to the liquid line.

2.2 Advantages of Compound (or Multi-stage) Vapour Compression with Intercooler

Following are the main advantages of compound or multistage compression over single stage compression:

I. The work done per kg of refrigerant is reduced in compound compression with intercooler as compared to single stage compression for the same delivery pressure.

2. It improves the volumetric efficiency for the given pressure ratio.

3. The sizes of the two cylinders (i.e., high pressure and low pressure) may be adjusted to suit the volume and pressure of the refrigerant.

4. It reduces the leakage loss considerably.

5. It gives more uniform torque, and hence a smaller size flywheel is needed.

6. It provides effective lubrication because of lower temperature range.

7. It reduces the cost of compressor.

2.3 Types of Compound Vapour Compression with Intercooler

In compound compression vapour refrigeration systems, the superheated vapour refrigerant leaving the first stage of compression is cooled by suitable method before being fed to the second stage of compression and so on. Such type of cooling the refrigerant is called intercooling. Though there are many types of compound compression with intercoolers, yet the following are important from the subject point of view:

I. Two stage compression with liquid intercooler.

2. Two stage compression with water intercooler.

3. Two stage compression with water intercooler, liquid subcooler and liquid flash chamber.

4. Two stage compression with water intercooler, liquid subcooler and flash intercooler.

5. Three stage compression with flash chambers.

6. Three stage compression with water intercoolers.

7. Three stage compression with flash intercoolers.

The above mentioned types are now discussed, in detail, one by one in the following pages.

2.4 Two Stage Compression with Liquid Intercooler

The arrangement of a two stage compression with liquid intercooler is shown in Fig. 2.7 (a). The corresponding p-h diagram is shown in Fig. 2.7 (b).

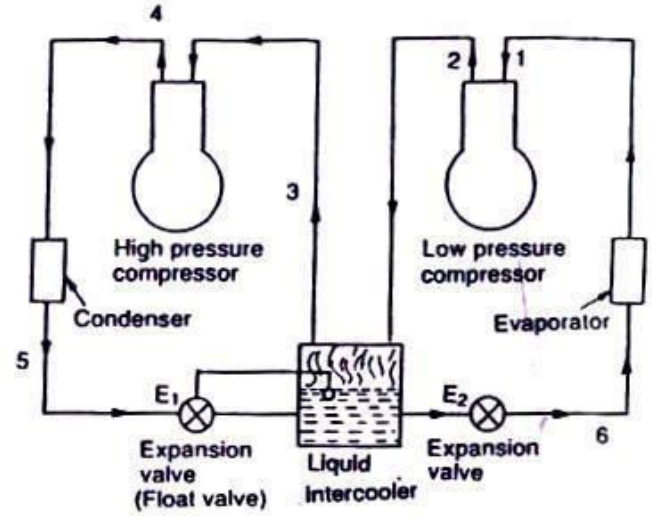
The various points on the p-h diagram are plotted as discussed below:

I. First of all, draw a horizontal pressure line representing the evaporator pressure P_E (or suction pressure of low pressure compressor) which intersects the saturated vapour line at point 1. At this point, the saturated vapour is supplied to the low pressure compressor. Let, at point I, the enthalpy of the saturated vapour is h_1 and entropy s_{v1}

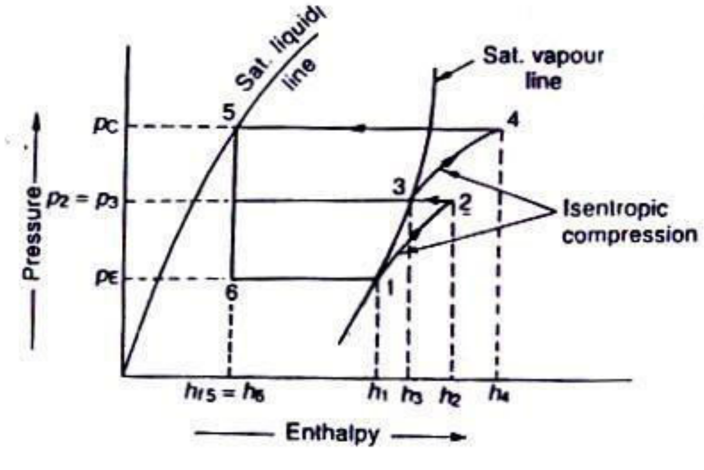
2. The saturated vapour refrigerant admitted at point 1 is compressed isentropically in the low pressure compressor and delivers the refrigerant in a superheated state. The pressure rises from P_E to P_2 The curve 1-2 represents the isentropic compression in the low pressure compressor. In order to obtain point 2, draw a line from point 1, with entropy equal to s_{vl}, along the constant entropy line

intersecting the intermediate pressure line P_2 at point 2. Let enthalpy at this point is h_2.

3. The superheated vapour refrigerant leaving the low pressure compressor at point 2 is cooled (or desuperheated) at constant pressure $P_2 = P_3$ in a liquid intercooler by the liquid refrigerant from the condenser. The refrigerant leaving the liquid intercooler is in saturated vapour state. The line 2-3 represents the cooling or desuperheating process. Let the enthalpy and entropy at point 3 is h_3 and s_{v3} respectively.



(a) Two stage compression with liquid intercooler.



(b) p-h diagram.

Fig. 2.7

4. The dry saturated vapour refrigerant is now supplied to high pressure compressor where it is compressed isentropically from intermediate or interocooler pressure P_2 to condensor pressure p_c. The curve 3-4 represents the isentropic compression in the high pressure compressor. The point 4 on the p-h diagram is obtained by drawing a line of entropy equal to s_{v3} along the constant entropy line as shown in Fig. 2.1 (b). Let the enthalpy of superheated vapour refrigerant at point 4 is h_4.

5. The superheated vapour refrigerant leaving the high pressure compressor at point 4 is now passed through the condenser at constant pressure P_c as shown by a horizontal line 4-5. The condensing process 4-5 changes the state of refrigerant from superheated vapour to saturated liquid.

6. The high pressure saturated liquid refrigerant from the condensor is passed to the intercooler where some of liquid refrigerant evaporates in desuperheating the superheated vapour refrigerant from the low pressure compressor. In order to make up for the liquid evaporated, i.e. to maintain a constant liquid level, an expansion valve E_1 which acts as a float valve, is provided.

7. The liquid refrigerant from the intercooler is first expanded in an expansion valve E_2 and then evaporated in the evaporator to saturated vapour condition, as shown in Fig. 2.1(b).

Let

m_l = Mass of refrigerant passing through the evaporator (or low pressure compressor) in kg/min, and

m_2 = Mass of refrigerant passing through the condenser (or high pressure compressor) in kg/min.

The high pressure compressor in a given system will compress the mass of refrigerant from low pressure compressor (m_l) and the mass of liquid evaporated in the liquid intercooler during cooling or de superheating of superheated vapour refrigerant from low pressure compressor. If m_3 is the mass of liquid evaporated in the intercooler, then

$m_3 = m_2 - m_l$

The value of m_2 may be obtained by considering the thermal equilibrium for the liquid intercooler as shown in Fig. 2.2, i.e .,

Heat taken by the liquid intercooler = Heat given by the liquid intercooler

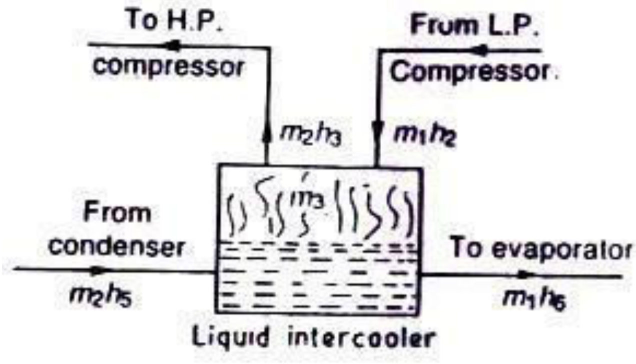


Fig2.8. Thermal equilibrium for liquid intercooler.

or $m_2 h_{f5} + m_1 h_2 = m_1 h_6 + m_2 h_3$

$$\therefore\ m_2 = \frac{m_1(h_2 - h_6)}{h_3 - h_{f5}} = \frac{m_1(h_2 - h_{f5})}{h_3 - h_{f5}} \quad \ldots (\because h_6 = h_{f5})$$

and mass of liquid refrigerant evaporated in the intercooler,

$$^{*}m_3 = m_2 - m_1 = \frac{m_1(h_2 - hf_5)}{h_3 - h_{f5}} - m_1 = \frac{m_1(h_2 - h_3)}{h_3 - h_{f5}}$$

We know that refrigerating effect,

$R_E = m_l (h_l - h_f) = m_l (h_l - h_{f5}) = 210$ Q kJ/min

where Q is the load on the evaporator in tonne of refrigeration.

Total work done in both the compressors,

$W = m_l (h_2 - h_1) + m_2 (h_4 - h_3)$

$\therefore$ Power required to drive the system,

$$P = \frac{m_1(h_2 - h_1) + m_2(h_4 - h_3)}{\text{kW}\ 60}$$

and C.O.P. of the system

$$= \frac{R_E}{W} = \frac{m_1 (h_1 - h_{f5})}{m_1 (h_2 - h_1) + m_2 (h_4 - h_3)} = \frac{210\,Q}{P \times 60}$$

Notes: 1. In case of ammonia, when liquid refrigerant is used for intercooling, the total power requirement will decrease. It is due to the fact that the mass of liquid evaporated during intercooling is extremely small because of its high latent heat of vaporisation and the constant entropy lines of ammonia become very flat in the superheat region. Thus the intercooling by liquid refrigerant is commonly used in multi-stage ammonia plants, because of less power requirement.

2. In case of refrigerant R-12, when liquid refrigerant is used for intercooling, the total power requirements may actually increase. It is due to the fact that the latent heat of vaporisation is small and the constant entropy line of R-12 does not change very much with the temperature. Thus in R-12 systems, the saving in work by performing the compression close to the saturated vapour line does not compensate for the increased mass flow rate through the high stage compressor. Therefore, intercooling by liquid refrigerant in R-12 systems is never employed.

Example 2.1. Calculate the power needed to compress 20 kg / min of ammonia from saturated vapour at 1.4 bar to a condensing pressure of 10 bar by two-stage compression with intercooling by liquid refrigerant at 4 bar. Assume saturated liquid to leave the condenser and dry saturted vapours to leave the evaporator. Use the p-h chart.

Determine, also, the power needed when intercooling is not employed.

Solution. Given : $m_1 = 20$ kg/min ; $P_E = 1.4$ bar ; $p_C = 10$ bar ; $p_2 = p_3 = 4$ bar

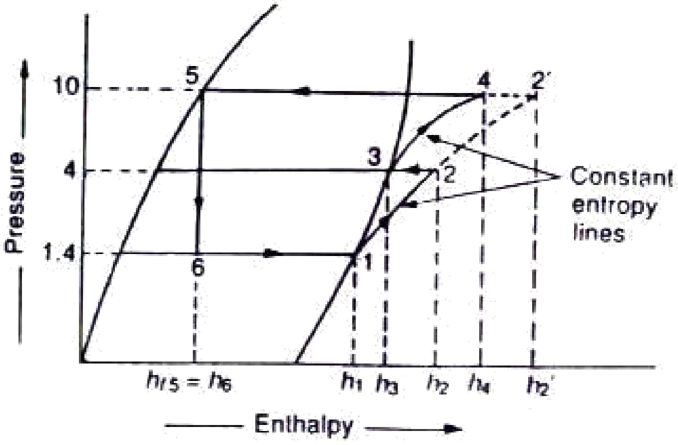


Fig. 2.9

The p-h diagram for a two stage compression with intercooling by liquid refrigerant is shown in Fig. 2.9. The various values for ammonia as read from the p-h diagram are as follows:

Enthalpy of saturated vapour refrigerant entering the low pressure compressor at point 1, h_1 = 1400 kJ/kg

Entropy of saturated vapour refrigerant entering the low pressure compressor at point 1, S_1 = 5.75 kJ/kg K

Enthalpy of superheated vapour refrigerant leaving the low pressure compressor at point 2, h_2 = 1527 kJ/kg

Enthalpy of saturated vapour refrigerant leaving the intercooler or entering the high pressure compressor at point 3,

h_3 = 1428 kJ/kg

Entropy of saturated vapour refrigerant leaving the intercooler or entering the high pressure compressor at point 3,

s_3 = 5.39 kJ/kg K

Enthalpy of superheated vapour refrigerant leaving the high pressure compressor at point 4, h_4 = 1550 kJ/kg

Enthalpy of saturated liquid refrigerant passing through the condenser at point 5, $h_{f5} = h_6$ = 284 kJ/kg

We know that mass of refrigerant passing through the condenser (or high pressure compressor),

$$m2 = \frac{m_1 (h_2 - hf_5)}{h_3 - h_{f\,5}} = \frac{20(1527 - 284)}{1428 - 284} = 21.73 \text{ kg/min}$$

Work done in low pressure compressor,

$W_L = m_1 (h_2 - h_1) = 20 (1527 - 1400) = 2540$

kJ/min Work done in high pressure compressor,

$W_H = m_2 (h_4 - h_3) = 21.73 (1550 - 1428) = 2651$

kJ/min and total work done in both the compressors,

$W = W_L + W_H = 2540 + 2651 = 5191$ kJ/min

$\therefore$ Power needed = 5191/60 = 86.5 kW **Ans.**

Power needed when intercooling is not employed

When intercooling is not employed, the compression of refrigerant will follow the path 1-2 in the low pressure compressor and 2-2' in the high pressure compressor. In such a case,

Work done in the high pressure compressor,

$W_H = m_i (h_2' - h_2) = 20 (1676 - 1527) = 2980$ kJ /min

... (From p-h diagram, $h_2' = 1676$ kJ/kg)

and total work done is both the compressors,

$W = W_L + W_H = 2540 + 2980 = 5520$ kJ/min

$\therefore$ Power needed = 5520/60 = 92 kW **Ans.**

Example 2.2. Calculate the power needed to compress 20 kg/min of R-12 from saturated vapour at 1.4 bar to a condensing pressure of 10 bar by two-stage compression with intercooling by liquid refrigerant at 4 bar. Assume saturated liquid to leave the condenser and dry saturated vapours to leave the evaporator.

Use the p-h chart. Sketch the cycle on a skeleton p-h chart and label the values of enthalpy at salient points.

Solution. Given: $m_1 = 20$ kg/min; $P_E = 1.4$ bar; $P_c = 10$ bar; $P_2 = P_3 = 4$ bar

The p-h diagram for a two-stage compression with intercooling by liquid refrigerant is shown in Fig. 5.4. The various values for R-12 as read from the p-h diagram are as follows :..

Enthalpy of saturated vapour refrigerant entering the low pressure compressor at point 1, $h_1 = 178$ kJ/kg

Entropy of saturated vapour refrigerant entering the low pressure compressor at point 1, $s_1 = 0.71$ kJ/kg K

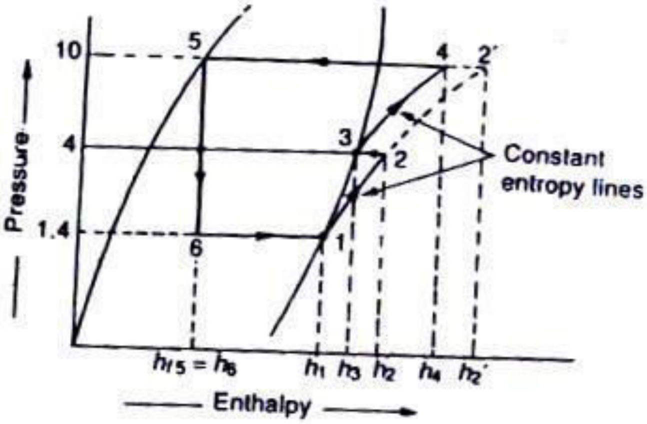


Fig. 2.10

Enthalpy of superheated vapour refrigerant leaving the low pressure compressor at point 2, $h_2 = 195$ kJ/kg

Enthalpy of saturated vapour refrigerant leaving the intercooler or entering the high pressure compressor at point 3,

$h_3 = 191$ kJ/kg

Entropy of saturated vapour refrigerant entering the high pressure compressor at point 3, $s_3 = 0.695$ kJ/kg K

Enthalpy of superheated vapour refrigerant leaving the high pressure compressor at point 4, $h_4 = 210$ kJ/kg K

Enthalpy of saturated liquid refrigerant leaving the condenser at point 5, $h_{f5} = h_6 = 77$ kJ/kg

We know that mass of refrigerant passing through the condenser (or high pressure compressor),

$$m_2 = \frac{m_1(h_2 - h_{f5})}{h_3 - h_{f5}} = \frac{20(195 - 77)}{191 - 77} = 20.7 \text{ kg/min}$$

Work done in low pressure compressor,

$W_L = m_1 (h_2 - h_1) = 20 (195 - 178) = 340$ kJ/min

Work done in high pressure compressor,

$W_H = m_2 (h_4 - h_3) = 20.7 (210 - 191) = 393$

kJ/min We know that refrigerating effect,

$R_E = m (h_1 - h_{f6}) = 210\ Q$ kJ/min

∴ C.O.P. of the system

$$= \frac{R_E}{W} = \frac{m(h_1 - h_{f6})}{\left[(h_2 - h_1) + (h_4 - h_3)\right]} = \frac{210\ Q}{P \times 60}$$

Example 2.3. The following data refer to a two stage compression ammonia refrigerating system with water intercooler.

Condenser pressure = 14 bar; Evaporator pressure = 2 bar; Intercooler pressure = 5 bar; Load on the evaporator = 2 TR

If the temperature of the de-superheated vapour and sub-cooled liquid refrigerant are limited to 30^{o}C, find (a) the power required to drive the system, and (b) CO.P. of the system.

Solution. Given : $P_c = 14$ bar ; $P_E = 2$ bar ; $P_2 = P_3 = 5$ bar ; $Q = 10$ TR ; $t_3 = t_6 = 30°$ C

The p-h diagram for a two stage compression system with water intercooler is shown in Fig. 2.11. The various values as read from the p-h diagram for ammonia are as follows:

Enthalpy of saturated vapour refrigerant entering the low pressure compressor at point 1. h_1 = 1420 kJ/kg

Entropy of saturated vapour refrigerant at point 1, s_1 = 5.6244 kJ/kg K

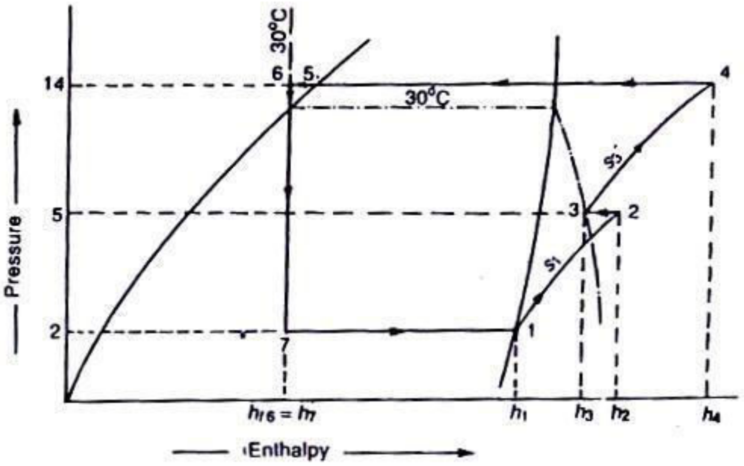


Fig. 2.11

Enthalpy of superheated vapour refrigerant leaving the water intercooler at point 3. h_3= 1510 kJ/kg

Entropy of superheated vapour refrigerant at point 3,

s_3 = 5.424 kJ/kg K

Enthalpy of superheated vapour refrigerant leaving the high pressure compressor at point 4.

h_4 = 1672 kJ/kg

Enthalpy of liquid refrigerant leaving the liquid sub-cooler, $h_{f6} = h_7$ = 323 kJlkg

The points 2 and 4 on the p-h diagram are obtained in the similar way as discussed in Art. 5.3.

From the p-h diagram, we find that enthalpy of superheated vapour refrigerant at point 2, h_2 = 1550 kJ/kg.

(a) Power required to drive the system

We know that mass of refrigerant circulating through the system,

$$m = \frac{210\,Q}{h_1 - h_{f6}} = \frac{210 \times 10}{1420 - 323} = 1.91 \text{kg/min}$$

Total work done in both the compressors,

$w = m\,[(h_2 - h_1) + (h_4 - h_3)]$

$= 1.91\,[(1550 - 1420) + (1672 - 1510) = 557.7$ kJ/min

$\therefore$ Power required to drive the system,

(b) *CO.P. of system*

We know that refrigerating effect of the system,

$R_E = 210Q = 210 \times 10 = 2100$ kJ/min

:. C.O.P. of the system

$$= \frac{R_E}{W} = \frac{2100}{557.7} = 3.76$$ **Ans.**

2.6 Two Stage Compression with Water Intercooler, Liquid Sub-cooler and Liquid Flash Chamber

The arrangement of a two stage compression with water intercooler, liquid sub-cooler and liquid flash chamber is shown in Fig. 2.12 (a). The corresponding p-h diagram is shown in Fig. 2.7 (b). The various processes, in this system, are as follows:

I. The saturated vapour refrigerant at the evaporator pressure P_E is admitted to low pressure compressor at point I. In this compressor, the refrigerant is compressed isentropically from evaporator pressure P_E to water intercooler (or flash chamber)

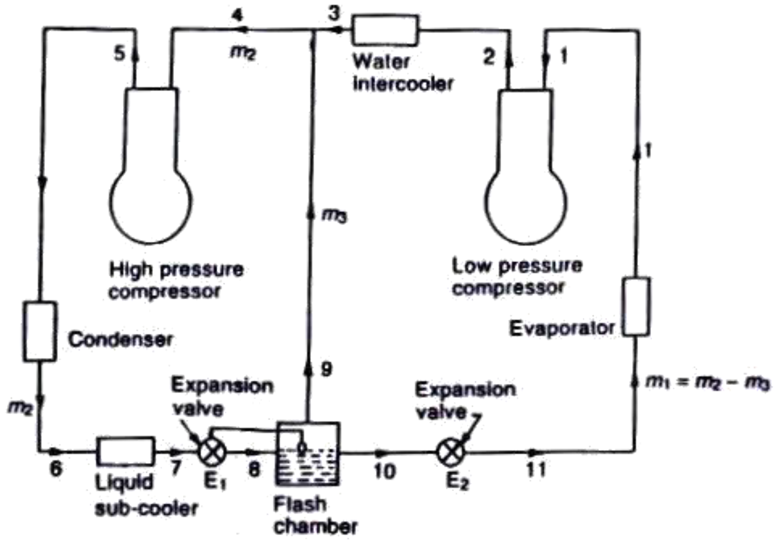
pressure P_F as shown by the curve 1-2 in Fig. 2.12 (b).

2. The superheated vapour refrigerant leaving the low pressure compressor at point 2 is now passed through the water intercooler at constant pressure P_F in order to reduce the degree of superheat (i.e., from temperature t_2 to t_3). The line 2-3 represents the water intercooling or de-superheating process.

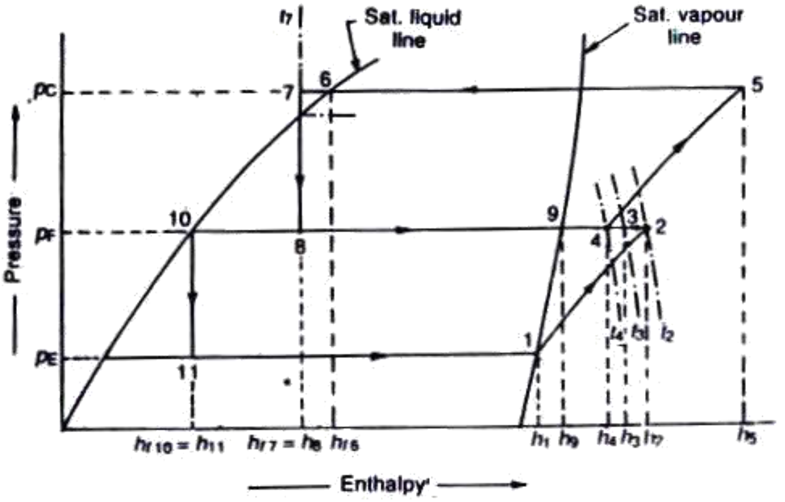
3. The superheated vapour refrigerant leaving the water intercooler at point 3 is mixed with the vapour refrigerant supplied by the flash chamber at point 9. The condition of refrigerant after mixing is shown by point 4 which is in superheated state. Let the temperature at this point is t_4.

4. The superheated vapour refrigerant admitted at point 4 to the high pressure compressor is compressed isentropically from the intercooler or flash chamber pressure P_F to condenser pressure P_c as shown by the curve 4-5. The temperature rises from t_4 to t_5.

5. The superheafed vapour leaving the high pressure compressor at pressure P_c is passed through a condenser at constant pressure as shown by a horizontal line 5-6. The condensing process 5-6 changes the state of refrigerant from superheated vapour to saturated liquid.



(*a*) **Two stage compression with water intercooler, liquid sub-cooler and liquid flash chamber.**



(*b*) *p-h* **diagram.**
Fig. 5.7

Fig 2.12

6. The saturated liquid refrigerant from the condenser is now cooled in liquid sub-cooler to a temperature, say t_7. The line 6-7 represents a sub-cooling process.

7. The liquid refrigerant leaving the sub-cooler at pressure P_c is expanded in an expansion valve E_1 to a pressure equal to the flash chamber pressure P_F, as shown by vertical line 7-8. The expanded refrigerant which is a mixture of vapour and liquid refrigerants is admitted to a flash chamber at point 8. The flash chamber separates the

vapour and liquid refrigerants at pressure P_F. The vapour refrigerant from the flash chamber at point 9 is mixed with the refrigerant from the water intercooler. The liquid refrigerant from the flash chamber at Point 10 is further expanded in an expansion valve E_2 as shown by the vertical line 10-11.

8. The liquid refrigerant leaving the expansion valve E_2 is evaporated in the evaporator at the evaporator pressure P_E (usually 2 bar) as shown by the horizontal line 11-1 in Fig. 2.12 (b).

Let m2 = Mass of refrigerant passing through the condenser (or high pressure compressor), and

m3 = Mass of vapour refrigerant formed in the flash chamber.

:. Mass of refrigerant passing through the evaporator (or low pressure compressor),

$m_1 = m_2 - m_3$

If Q tonne of refrigeration is the load on the evaporator, then the mass of refrigerant passing through the evaporator,

$$m1 = \frac{210\,Q}{h_1 - h_{11}} = \frac{210\,Q}{h_1 - h_{f10}} \text{ kg/min} \qquad \ldots(\because h_{11} = h_{f10})$$

Now let us consider the thermal equilibrium of the flash chamber. Since the flash chamber is an insulated vessel, therefore there is no heat exchange between the flash chamber and atmosphere. In other words, the heat taken and given by the flash chamber are same. Mathematically,

Heat taken by the flash chamber

= Heat given by the flash chamber

$m_2 h_8 = m_3 h_9 + m_1 h_{f10}$

$= m_3 h_9 + (m_2 - m_3) h_{f10} \qquad \ldots(\because m_1 = m_2 - m_2)$

$m_2 (h_8 - h_{f10}) = m_3 (h_9 - h_{f10})$

$$\therefore m_3 = m_2 \left(\frac{h_8 - h_{f10}}{h_9 - h_{f10}} \right) = m_2 \left(\frac{h_{f7} - hf_{10}}{h_9 - h_{f10}} \right) \qquad \ldots(\because h_8 = h_{f7}) \ldots \text{(i)}$$

The vapour refrigerant from the water intercooler (represented by point 3) is mixed with vapour refrigerant m_3 from the flash chamber (represented by point 9) at the same pressure before entering the high pressure compressor. The enthalpy of the mixed refrigerant (represented by point 4) may be calculated by using the equation,

$$m_2 h_4 = m_3 h_9 + m_1 h_3$$

$$= m_3 h_9 + (m_2 - m_3) h_3$$

We know that refrigerating effect of the system,

$$R_E = m_1 (h_1 - h_{11}) = 210\ Q \text{ kJ/min}$$

Work done in low pressure compressor,

$$W_L = m_1 (h_2 - h_1)$$

Work done in high pressure compressor,

$$W_H = m_2 (h_5 - h_4)$$

Total workdone in both the compressors,

$$W = W_L + W_H = m_1 ((h_2 - h_1) + m_2 (h_5 - h_4)$$

∴ Power required to drive the system,

$$p = \frac{m_1(h_2 - h_1) + m_2(h_5 - h_4)}{60} \text{kW}$$

and C.O.P. of the system

$$= \frac{R_E}{W} = \frac{m_1(h_2 - h_{11})}{m_1(h_2 - h_1) + m_2(h_5 - h_4)} = \frac{210\ Q}{p \times 60}$$

Note: Since the mass of vapour refrigerant m, is cooled in the water intercooler from condition 2 to 3, therefore cooling capacity of the intercooler

$$= m_1 (h_2 - h_3)$$

Example 2.4. A two stage compression ammonia refrigeration system operates between overall pressure limits of 14 bar and 2 bar. the temperature of the desuperheated vapour and subcooled liquid refrigerant are limited to 30°C. The flash tank separates dry vapour at 5 bar pressure and the liquid refrigerant then expands to 2 bar.

Estimate the C.O.P. of the machine and power required to drive the compresor, if the mechanical effficiency of the drive is 80% and load on the evaporator is 10 TR.

Solution. Given :P_c = 14 bar; P_E = 2 bar; P_F = 5 bar; $t_3 = t_7 = 30^{o}$ C ; η_m = 80% = 0.8 ; Q = 10TR

The p-h diagram for a two-stage compression system with given conditions is shown in Fig. 5.8. The values as read from p-h diagram for ammonia, are as follows:

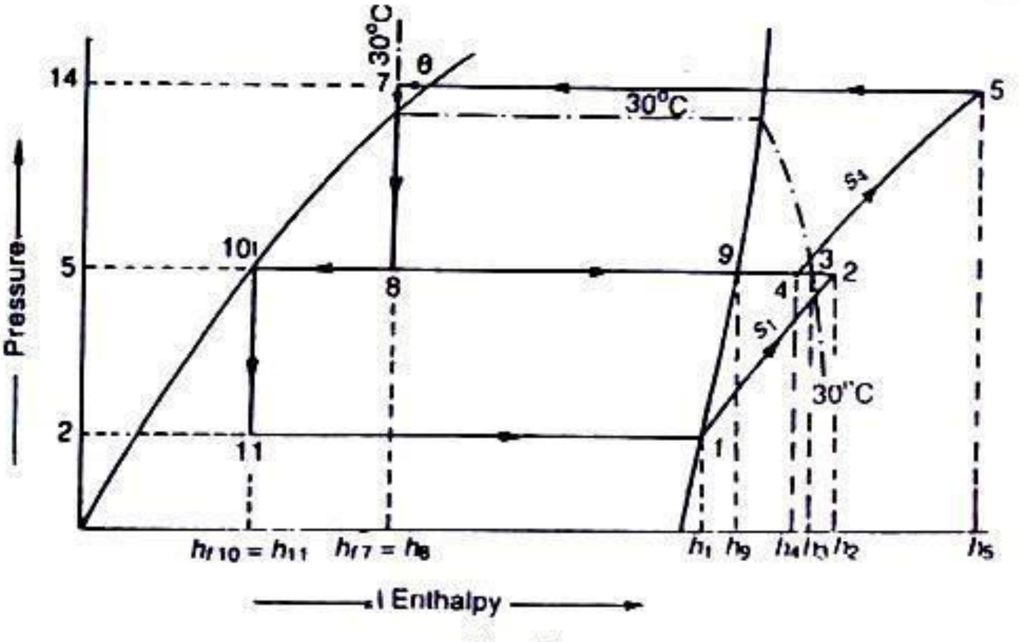


Fig 2.13

Enthalpy of saturated vapour refrigerant entering the low pressure compressor at point 1. h_i = 1420 kJ/kg

Entropy of saturated vapour refrigerant entering the low pressure compressor at point 1. s_1 = 5.6244 kJ/kg K

Enthalpy of superheated vapour refrigerant leaving the low pressure compressor at point 2, h_2 =1550 kJ/kg

Enthalpy of superheated vapour refrigerant leaving the water intercooler at point 3. $h_3 = 1510$ kJ/kg

Enthalpy of saturated vapour refrigerant leaving the flash tank at point 9,

$h_9 = 1432$ kJ/kg

Enthalpy of liquid refrigerant leaving the subcooler at point 7,

$h_{f7} = h_8 = 323$ kJ/kg

Enthalpy of saturated liquid refrigerant leaving the second expansion valve at point 10, $h_{f10} = h_{11} = 198$ kJ/kg

Let m_2 = Mass of refrigerant passing through the condenser.

We know that mass of the vapour refrigerant formed in the flash tank,

$$m_3 = m_2 \left(\frac{h_8 - h_{f10}}{h_9 - h_{f10}} \right) = m_2 \left(\frac{323 - 198}{1432 - 198} \right) = 0.1\, m_2 \qquad \text{... (i)}$$

and mass of refrigerant passing through the evaporator,

$$m_1 = m_2 - m_3 = \frac{210\, Q}{h_1 - h_{f10}} = \frac{210 \times 10}{1420 - 198} = 1.72 \text{ kg/min} \qquad \text{... (ii)}$$

From equations (i) and (ii),

$m_2 - 0.1\, m_2 = 1.72$ or $m_2 = 1.9$ kg/min

$m_3 = 0.1\, m_2 = 0.1 \times 1.9 = 0.19$ kg/min

The desuperheated vapour refrigerant $(m_2 - m_3)$ as represented by point 3 is mixed with the vapour refrigerant from the flash tank as represented by point 9. The enthalpy of the mixed refrigerant entering the high pressure compressor as represented by point 4 is given by

$m_2 h_4 = m_3 h_9 + (m_2 - m_3) h_3$

$= 0.1\, m_2 h_9 + (m_2 - 0.1\, m_2) h_3$ ... [From equation (i)]

or $h_4 = 0.1 \times h_9 + 0.9 \times h_3 = 0.1 \times 1432 + 0.9 \times 1510 = 1502$ kJ/kg

We see from p-h diagram that at point 4 (intersection of pressure 5 bar and enthalpy 1502 kJ/kg), the entropy is $s_4 = 5.51$ kJ/kg K. Now from point 4, draw a line of entropy equal to 5.51 kJ/kg K along the constant entropy line which intersects the condenser pressure (14 bar) line at point 5. Thus, the point 5 is located. From p-h diagram, we find that enthalpy of refrigerant leaving the high pressure compressor at point 5 is

$h_5 = 1650$ kJ/kg

C.O.P. of the machine

We know that refrigerating effect

$= m_l (h_l - h_{f10}) = 210\ _Q = 210 \times 10 = 2100$

kJ/min Work done in both the compressors

$= m_l (h_2 - h_1) + m_2 (h_5 - h_4)$

$= 1.72 (1550 - 1420) + 1.9 (1650 - 1502)$

$= 223.6 + 281.2 = 504.8$ kJ/min

Since the mechanical efficiency of the drive is 80%, therefore actual work done in both the compressors

$= 504.8/0.8 = 631$ kJ/min

$\therefore$ Actual C.O.P = $\dfrac{\text{Refrigerating effect}}{\text{Actual work done}} = \dfrac{2100}{631} = 3.32$ **Ans.**

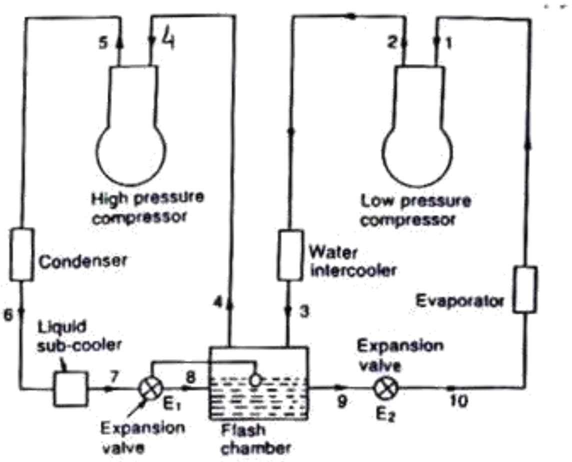
Power required to drive the compressors

We know that power required to drive the compressors

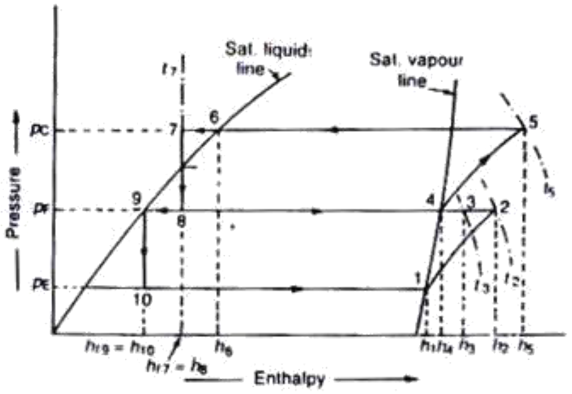
$= \dfrac{\text{Actual work done}}{60} = \dfrac{631}{60} = 10.5$ kW **Ans.**

2.7 Two Stage Compression with Water Intercooler, Liquid Sub-cooler and Flash Intercooler

A two stage compression with water intercooler, liquid sub-cooler and flash intercooler is shown in Fig. 2.9 (a). The corresponding p-h diagram is shown in Fig. 5.9 (b).



(a) Two stage compression with water intercooler, liquid sub-cooler and flash intercooler.



(b) p-h diagram.

Fig 2.14

We have seen in the previous article that when the vapour refrigerant from the low pressure compressor is passed through the water intercooler, its temperature does not reduce to the saturated vapour line or even very near to it, before admitting it to the high pressure compressor [Refer point 4 of Fig. 2.12 (b)]. In fact, with water cooling there may be no saving of work in compression. But the improvement in performance and the reduction in compression work may be achieved by using a flash chamber as an intercooler as well as flash separator, as shown in Fig. 2.14 (a). The corresponding p-h diagram is shown in Fig. 2.14 (b). The various processes, in this system, are as follows:

1. The saturated vapour refrigerant at the evaporator pressure P_E is admitted to the low pressure compressor at point 1. In this compressor, the refrigerant is

compressed isentropically from evaporator pressure P_E to the flash intercooler pressure P_F,as shown by the curve 1-2 in Fig2.14 (b).

2. The superheated vapour refrigerant leaving the low pressure compressor at point 2 is now passed through the water intercooler at constant pressure P_F, in order to reduce the degree of superheat (i.e. from temperature t_2 to t_3). The line 2-3 represents the water intercooling or desuperheating process.

3. The superheated vapour refrigerant leaving the water intercooler at point 3 is passed through a flash intercooler which cools the superheated vapour refrigerant to saturated vapour refrigerant as shown by the line 3-4. The cooling of superheated vapour refrigerant is done by the evaporation of a part of the liquid refrigerant from the flash intercooler placed at point 8.

4. The saturated vapour refrigerant leaving the flash intercooler enters the high pressure compressor at point 4 where it is compressed isentropically from flash intercooler pressure P_F to condenser pressure P_C, as shown by the curve 4-5.

5. The superheated vapour refrigerant leaving the high pressure compressor at pressure P_c is passed through a condenser at constant pressure. The condensing process as shown by line 5-6 changes the state of refrigerant from superheated vapour to saturated liquid.

6. The saturated liquid refrigerant leaving the condenser at point 6 is now cooled at constant pressure P_c in the liquid sub-cooler to a temperature t_7 as shown in Fig. 2.9 (b). The line 6-7 shows the sub-cooling process.

7. The liquid refrigerant leaving the sub-cooler at point 7 is expanded in an expansion valve E_1 to a pressure equal to the flash intercooler pressure P_F , as shown by the vertical line 7-8. The expanded refrigerant (which is a mixture of vapour and liquid refrigerant) is admitted to flash intercooler at point 8 which also acts as a flash separator.

8. The liquid refrigerant leaving the flash intercooler at point 9 is passed through the second expansion valve E_2 (process 9-10) and then evaporated in the evaporator as shown by the horizontal line 10-1.

Let m_1 = Mass of the refrigerant passing through the evaporator or low pressure compressor), and

m_2 = Mass of the refrigerant passing through the condenser (or high pressure compressor).

If Q tonne of refrigeration is the load on the evaporator, then the mass of refrigerant passing through the evaporator is given by,

$$m_1 = \frac{210\,Q}{h_1 - h_{10}} = \frac{210\,Q}{h_1 - h_{f9}}\ \text{kg/min} \qquad \dots (\because h10 = hf9)$$

Now for the thermal equilibrium of the flash intercooler, Heat taken by the flash intercooler

= Heat given by the flash intercooler

$m_2\,h_8 + m_1\,h_3 = m_2\,h_4 + m_1\,h_{f9}$

$m1(h3 - hf9) = m2\,(h4 - h8)$

$$\therefore m_2 = m_1\left(\frac{h_3 - h_{f9}}{h_4 - h_8}\right) = m_1\left(\frac{h_3 - h_{f9}}{h_4 - h_{f7}}\right)\ \text{kg/min}$$

We know that refrigerating effect,

$R_E = m_1\,(h_1 - h_{10}) = m_1\,(h_1 - h_{f9}) = 210\,Q$ kJ/min

and work done in both the compressors,

W = Work done in L.P. compressor + Work done in H.P. compressor

$= m_1\,(h_2 - h_1) + m_2\,(h_5 - h_4)$

$\therefore$ Power required to drive the system,

$$p = \frac{m_1(h_2 - h_1) + m_2(h_5 - h_4)}{\text{kW}\ 60}$$

and coefficient of performance of the system,

$$\text{C.O.P.} = \frac{R_E}{W} = \frac{m_1(h_1 - h_{f9})}{m_1(h_2 - h_1) + m_2(h_5 - h_4)} = \frac{210\,Q}{P \times 60}$$

MODULE – III

Vapour Absorption Refrigeration Systems

3.1 Introduction

The vapour absorption refrigeration system is one of the oldest method of producing refrigerating effect. The principle of vapour absorption was first discovered by Michael Faraday in 1824 while performing a set of experiments to liquify certain gases. The first vapour absorption refrigeration machine was developed by a French scientist Ferdinand Carre in 1860. This system may be used in both the domestic and large industrial refrigerating plants. The refrigerant, commonly used in a vapour absorption system, is ammonia.

The vapour absorption system uses heat energy, instead of mechanical energy as in vapour compression systems, in order to change the conditions of the refrigerant required for the operation of the refrigeration cycle. We have discussed in the previous chapters that the function of a compressor, in a vapour compression system, is to withdraw the vapour refrigerant from the evaporator. It then raises its temperature and pressure higher than the cooling agent in the condenser so that the higher pressure vapours can reject heat in the condenser. The liquid refrigerant leaving the condenser is now ready to expand to the evaporator conditions again.

In the vapour absorption system, the compressor is replaced by an absorber, a pump, a generator and a pressure reducing valve. These components in vapour absorption system perform the same function as that of a compressor in vapour compression system. In this system, the vapour refrigerant from the evaporator is drawn into an absorber where it is absorbed by the weak solution of the refrigerant forming a strong solution. This strong solution is pumped to the generator where it is heated by some external source. During the heating process, the vapour refrigerant is driven off by the solution and enters into the condenser where it is liquefied. The liquid refrigerant then flows into the evaporator and thus the cycle is completed.

3.2 Simple Vapour Absorption System -

The simple vapour absorption system, as shown in Fig. 3.1, consists of an absorber, a pump, a generator and a pressure reducing valve to replace the compressor of vapour compression system. The other components of the system are condenser, receiver, expansion valve and evaporator as in the vapour compression system.

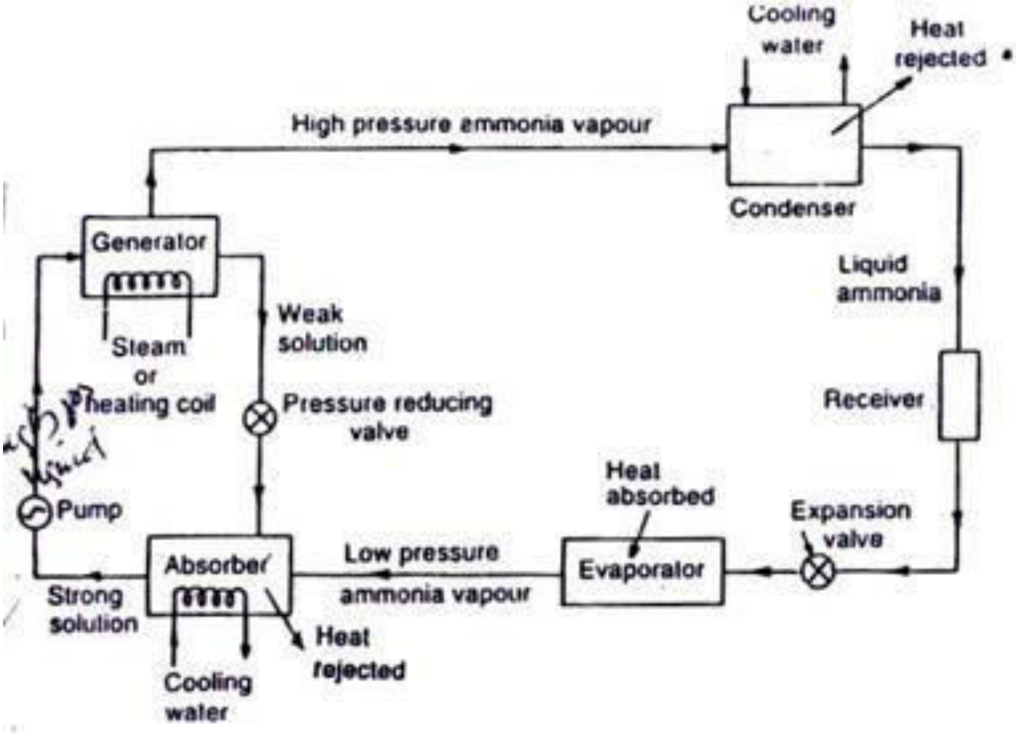


Fig 3.1 Simple vapour absorbtion system

. In this system, the low pressure ammonia vapour leaving the evaporator enters the absorber where it is absorbed by the cold water in the absorber. The water has the ability to absorb very large quantities of ammonia vapour and the solution thus formed, is known as aqua-ammonia. The absorption of ammonia vapour in water lowers the pressure in the absorber which in turn draws more ammonia vapour from the evaporator and thus raises the temperature of solution. Some form of cooling arrangement (usually water cooling) is employed in the absorber to remove the heat of solution evolved there. This is necessary in order to increase the absorption capacity of water, because at higher temperature water absorbs less ammonia vapour. The strong solution thus formed in the absorber is pumped to the generator by the liquid pump. The pump increases the pressure of the solution upto 10 bar.

The *strong solution of ammonia in the generator is heated by some external source such as gas or steam. During the heating process, the ammonia vapour is driven off the solution at high pressure leaving behind the hot weak ammonia solution in the generator. This weak ammonia solution flows back to the absorber at low pressure after passing through the pressure reducing valve. The high pressure ammonia vapour from the generator is condensed in the condenser to a high pressure liquid ammonia. This liquid ammonia is passed to the expansion valve through the receiver and then to the evaporator. This completes the simple vapour absorption cycle.

3.3 actical Vapour Absorption System

The simple absorption system as discussed in the previous article is not very economical. In order to make the system more practical, it is fitted with an analyser, a rectifier and two heat exchangers as shown in Fig. 3.2. These accessories help to improve the performance and working of the plant, as discussed below :-

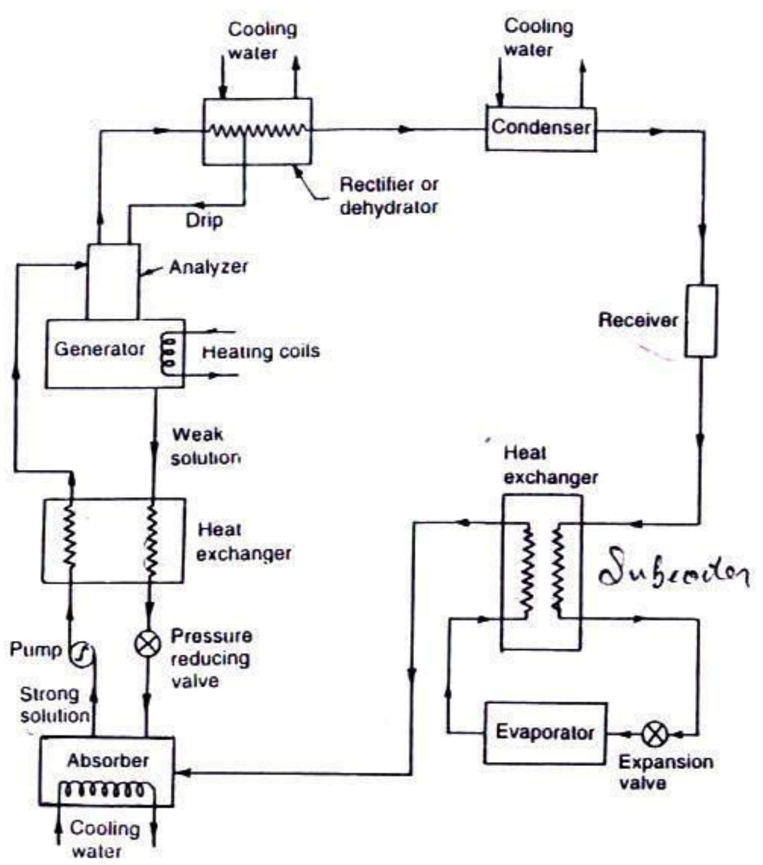


Fig. 3.2. Practical vapour absorption system.

1. *Analyser.* When ammonia is vaporised in the generator, some water is also vaporised and will flow into the condenser along with the ammonia vapours in the simple system. If these unwanted water particles are not removed before entering into the condenser, they will enter into the expansion valve where they freeze and choke the pipe line. In order to remove these unwanted particles flowing to the condenser, an analyser is used. The analyser may be built as an integral part of the generator or made as a separate piece of equipment. It consists of a series of trays mounted above the generator. The strong solution from the absorber and the aqua from the rectifier are introduced at the top of the analyser and flow downward over the trays and into the generator. In this way, considerable liquid surface area is exposed to the vapour rising from the generator. The vapour is cooled and most of the water vapour condenses, so that mainly ammonia vapour leaves the top of the analyser. Since the aqua is heated by the vapour, less external heat is required in the generator.

2. *Rectifier.* In case the water vapours are not completely removed in the analyser, a closed type vapour cooler called rectifier (also known as dehydrator) is used. It is generally water cooled and may be of the double pipe, shell and coil or shell and tube type. Its function is to cool further the ammonia vapours leaving the analyser so that the remaining water vapours are condensed. Thus, only dry or anhydrous ammonia vapours flow to the condenser. The condensate from the rectifier is returned to the top of the analyser by a drip return pipe.

3. *Heat exchangers*. The heat exchanger provided between the pump and the generator is used to cool the weak hot solution returning from the generator to the absorber. The heat removed from the weak solution raises the temperature of the strong solution leaving the pump and going to analyser and generator. This operation reduces the heat supplied to the generator and the amount of cooling required for the absorber. Thus the economy of the plant increases.

The heat exchanger provided between the condenser and the evaporator may also be called liquid sub-cooler. In this heat exchanger, the liquid refrigerant leaving the condenser is sub- cooled by the low temperature ammonia vapour from the evaporator as shown in Fig. 7.2. This sub-cooled liquid is now passed to the expansion valve and then to the evaporator.

In this system, the net refrigerating effect is the heat absorbed by the refrigerant in the evaporator. The total energy supplied to the system is the sum of work done by the pump and the heat supplied in the generator. Therefore, the coefficient of performance of the system is given by

$$\text{C.O.P.} = \frac{\text{Heat absorbed in evaporator}}{\text{Work done by pump + Heat supplied in generator}}$$

3.4 Advantages of Vapour Absorption Refrigeration System over Vapour Compression Refrigeration System

Following are the advantages of vapour absorption system over vapour compression system:

1. In the vapour absorption system, the only moving part of the entire system is a pump which has a small motor. Thus, the operation of this system is essentially quiet and is subjected to little wear.

The vapour compression system of the same capacity has more wear, tear and noise due to moving parts of the compressor.

2. The vapour absorption system uses heat energy to change the condition of the refrigerant from the evaporator. The vapour compression system uses mechanical energy to change the condition of the refrigerant from the evaporator.

3. The vapour absorption systems are usually designed to use steam, either at high pressure or low pressure. The exhaust steam from furnaces and solar energy may also be used. Thus this system can be used where the electric power is difficult to obtain or is very expensive.

4. The vapour absorption systems can operate at reduced evaporator pressure and temperature by increasing the steam pressure to the generator, with little decrease in capacity. But the capacity of vapour compression system drops rapidly with lowered evaporator pressure.

5. The load variations does not effect the performance of a vapour absorption system. The load variations are met by controlling the quantity of aqua circulated and the quantity of steam supplied to the generator.

The performance of a vapour compression system at partial loads is poor.

6. In the vapour absorption system, the liquid refrigerant leaving the evaporator has no bad effect on the system except that of reducing the refrigerating effect. In the vapour compression system, it is essential to superheat the vapour refrigerant leaving the evaporator so that no liquid may enter the compressor.

7. The vapour absorption systems can be built in capacities well above 1000 tonnes of refrigeration each which is the largest size for single compressor units.

8. The space requirements and automatic control requirements favour the absorption system more and more as the desired evaporator temperature drops.

3.5 Coefficient of Performance of an Ideal Vapour Absorption Refrigeration System

We have discussed earlier that in an ideal vapour absorption refrigeration system,

(a) the heat (Q_G) is given to the refrigerant in the generator,

(b) the heat (Q_c) is discharged to the atmosphere or cooling water from the condenser and absorber.

(c) the heat (Q_E) is absorbed by the refrigerant in the evaporator, and

(d) the heat (Q_p) is added to the refrigerant due to pump work.

Neglecting the heat due to pump work (Q_p), we have according to First Law of Thermodynamics,

$$Q_C = Q_G + Q_E \qquad \text{... (i)}$$

Let T_G = Temperature at which heat (Q_G) is given to the generator,

T_c = Temperature at which heat (Q_C) is discharged to atmosphere or cooling water from the condenser and absorber, and

T_E = Temperature at which heat (Q_E) is absorbed in the evaporator.

Since the vapour absorption system can be considered as a perfectly reversible system, therefore the initial entropy of the system must be equal to the entropy of the system after the change in its condition.

$$\therefore \frac{Q_G}{T_G} + \frac{Q_E}{T_E} = \frac{Q_C}{T_C} \qquad \text{...(ii)}$$

$$= \frac{Q_G + Q_E}{T_C} \qquad \text{... [From equation (i)]}$$

$$\text{or } \frac{Q_G}{T_G} - \frac{Q_G}{T_C} = \frac{Q_E}{T_C} - \frac{Q_E}{T_E}$$

$$Q_G\left(\frac{T_C + T_G}{T_G \times T_C}\right) = Q_E\left(\frac{T_E - T_C}{T_C \times T_E}\right)$$

$$Q_G = Q_E\left[\frac{T_E - T_C}{T_C \times T_E}\right]\left[\frac{T_G \times T_C}{T_C - T_G}\right]$$

$$= Q_E\left[\frac{T_C - T_E}{T_C \times T_E}\right]\left[\frac{T_G \times T_C}{T_G - T_C}\right]$$

$$= Q_E\left(\frac{T_C - T_E}{T_E}\right)\left(\frac{T_G}{T_G - T_C}\right) \qquad \text{... (iii)}$$

Maximum coefficient of performance of the system is given by

$$(\text{C.O.P.})_{max} = \frac{Q_E}{Q_G} = \frac{Q_E}{Q_E\left(\frac{T_C - T_E}{T_E}\right)\left(\frac{T_G}{T_G - T_C}\right)}$$

$$= \left(\frac{T_E}{T_C - T_E}\right)\left(\frac{T_G - T_C}{T_G}\right) \qquad \text{...(iv)}$$

It may noted that,

1. The expression $\left(\frac{T_E}{T_C - T_E}\right)$ is the C.O.P. of a Carnot refrigerator working between the temperature limits of T $_E$ and T $_C$.

2. The expression $\left(\frac{T_G - T_C}{T_G}\right)$ is the efficiency of a Carnot engine working between the temperature limits of T_G and T_C.

Thus an ideal vapour absorption refrigeration system may be regarded as a combination of a Carnot engine and a Carnot refrigerator. The maximum C.O.P. may be written as

$$(\text{C.O.P.})_{max} = (\text{C.O.P})_{carnot} \times \eta_{carnot}$$

In case the heat is discharged at different temperatures in condenser and absorber, then

$$(\text{C.O.P.})_{max} = \left[\frac{T_E}{T_C - T_E}\right]\left[\frac{T_G - T_C}{T_G}\right]$$

where T_A= Temperature at which heat is discharged in the absorber.

Example 3.1. In a vapour absorption refrigeration system, heating, cooling and refrigeration takes place at the temperatures of 100^{O} C, 20^{O} C and _5° C respectively. Find the maximum C.O.P. of the system.

Solution. Given: TG = 100^{O}C = 100 + 273 = 373 K ; Tc = 20^{O}C = 20 + 273 = 293 K ; T E = - 5° C = - 5 + 273 = 268 K

We know that maximum C.O.P. of the system

$$= \left(\frac{T_E}{T_C - T_E}\right)\left(\frac{T_G - T_C}{T_G}\right) = \left(\frac{268}{293 - 268}\right)\left(\frac{373 - 293}{373}\right) = 2.3 \textbf{ Ans.}$$

Example 3.2. In an absorption type refrigerator, the heat is supplied to NH_3 generator by condensing steam at 2 bar and 90% dry. The temperature in the refrigerator is to be maintained at - 5° C. Find the maximum C.O.P. possible.

If the refrigeration load is 20 tonnes and actual C.O.P. is 10% of the maximum C.O.P ., find the mass of steam required per hour. Take temperature of the atmosphere as 30°C.

Solution. Given: p = 2 bar; x = 90% = 0.9; T_E = -5°C = -5 + 273 = 268 K ; Q = 20 TR; Actual C.O.P. = 70% of maximum C.O.P. ; T_C = 30° C = 30 + 273 = 303 K

Maximum C.O.P.

From steam tables, we find that the saturation temperature of steam at a pressure of 2 bar is

T_G = 120.2° C = 120.2 + 273 = 393.2 K

We know that maximum C.O.P.

$$= \left[\frac{T_E}{T_C - T_E}\right]\left[\frac{T_G - T_C}{T_G}\right] = \left[\frac{268}{303 - 268}\right]\left[\frac{393 - 303}{393.2}\right] = 1.756 \textbf{ Ans.}$$

Mass of steam required per hour

We know that actual C.O.P.

= 70% of maximum C.O.P. = 0.7 × 1.756 = 1.229

∴ Actual heat supplied

$$= \frac{\text{Refrigeration load}}{\text{Actual C.O.P.}} = \frac{20 \times 210}{1.229} = 3417.4 \text{ kJ/min}$$

Assuming that only latent heat of steam is used for heating purposes, therefore from steam tables, the latent heat of steam at 2 bar is

h_{fg} = 2201.6 kJ/kg

∴ Mass of steam required per hour

$$= \frac{\text{Actual heat supplied}}{x \times h_{fg}} = \frac{3417.4}{2201.6} = 1.552 \text{ kg/min} = 93.12 \text{ kg/h } \textbf{Ans.}$$

3.6 Domestic Electrolux (Ammonia Hydrogen) Refrigerator

The domestic absorption type refrigerator was invented by two Swedish engineers Carl Munters and Baltzer Von Platan in 1925 while they were studying for their under-graduate course of Royal Institute of Technology in Stockholm. The idea was first developed by the 'Electrolux Company' of Luton, England.

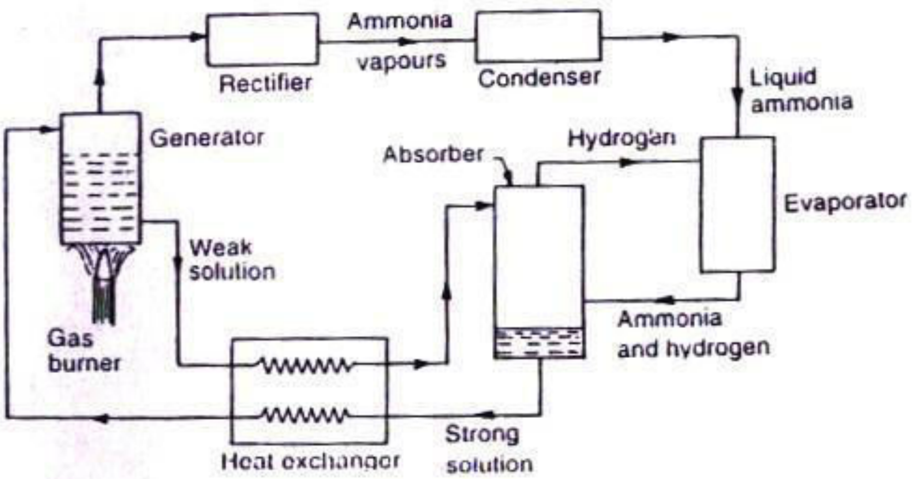


Fig. 3.3. Domestic electrolux type refrigerator.

This type of refrigerator is also called three-fluids absorption system. The main purpose of this system is to eliminate the pump so that in the absence of moving parts. the machine becomes noise-less. The three fluids used in this sytem are ammonia, hydrogen and water. The ammonia is used as a refrigerant because it possesses most of the desirable properties. It is toxic. but due to absence of moving parts, there is very little changes for the leakage and the total amount of refrigeration used is small. The hydrogen being the lightest gas. is used to increase the rate of evaporation of the liquid ammonia passing through the evaporator. The hydrogen is also non-corrosive and insoluble in water. This is used in the low-pressure side of the system. The water is used as a solvent because it has the ability to absorb ammonia readily. The principle of operation of a domestic electolux type refrigerator. as shown in Fig. 3.3. is discussed below:

The strong ammonia solution from the absorber through heat exchanger is heated in the generator by applying heat from an external source usually a gas burner. During this heating process, ammonia vapours are removed from the solution and passed to the condenser. A rectifirer or a water separator fitted before the condenser removes water vapour carried with the ammonia vapours. so that dry ammonia vapours are supplied to the condenser. These water vapours,˙ f not removed, they will enter into the evaporator causing freezing and choking of the machine . The hot weak solution left behind in the generator flow to the absorber through the heat excl anger. This hot weak solution while passing through the exchanger is cooled. The heat removed by the weak solution is utilised in raising the temperature of strong solution passing through the heat exchanger. In this way, the absorption is accelerated and the improvement in the performance of a plant is achieved.

The ammonia vapours in the condenser are condensed by using external cooling source. The liquid refrigerant leaving the condenser flows under gravity to the evaporator where it meets the hydrogen gas. The hydrogen gas which is being fed to the evaporator permit the liquid ammonia to evaporate at a low pressure and temperature according to Dalton's principle. During the process of evaporation, the ammonia absorbs latent heat from the refrigerated space and thus produces cooling effect.

The mixture of ammonia vapour and hydrogen is passed to the absorber where ammonia is absorbed in water while the hydrogen rises to the top and flows hack to the evaporator. This completes the cycle. The coefficient of performance of this refrigerator is given by :

$$\text{C.O.P.} = \frac{\text{Heat absorbed in the evaporator}}{\text{Heat supplied in the generator}}$$

Notes: 1. The hydrogen gas only circulates from the absorber to the evaporator and back.

2. The whole cycle is carried out entirely by gravity flow of the refrigerant.

3. It can not be used for industrial purposes as the C.O.P. of the system is very low.

3.7 Lithium Bromide Absorption Refrigeration System

The lithium-bromide absorption refrigeration system uses a solution of lithium bromide in water. In this system, the *water is being used as a refrigerant whereas lithium bromide, which is a highly hydroscopic salt, as an absorbent. The lithium bromide solution has a strong affinity for water vapour because of its very low vapour pressure. Since lithium bromide solution is corrosive, therefore inhibitors should be added in order to protect the metal parts of the system against corrosion. Lithium chromate is often uged as a corrosion inhibitor. This system is very popular for air conditioning in which low refrigeration temperatures (not below 0° C)** are required.

Fig. 3.4 shows a lithium bromide vapour absorption system. In this system. the absorber and the evaporator are placed in one shell which operates at the same low pressure of the system. The generator and condenser are placed in another shell which operates at the same high pressure of the system. The principle of operation of this system is discussed below :

The water for air-conditioning coils or process requirements is chilled as it is pumped through the chilled-water tubes in.the evaporator by giving up heat to the refrigerant water sprayed over the tubes. Since the pressure inside the evaporator is maintained very low, therefore, the refrigerant waterevaporates. The water vapours thus formed will be absorbed by the strong lithium-bromide solution which is sprayed in the absorber. In absorbing the water vapour, the lithium bromide solution helps in maintaining very low pressure (high vacuum) needed in the evaporator, and the solution becomes weak. This weak solution is pumped by a pump to the generator where it is heated up by using steam or hot water in the heating coils. A portion of water is evaporated by the heat and the solution now becomes more strong. This strong solution is passed through the heat exchanger and then sprayed in the absorber as discussed above. The weak solution of lithium bromide from the absorber to the generator is also passed through the heat exchanger. This weak solution gets heat from the strong solution in the heat exchanger, thus reducing the quantity of steam required to heat the weak solution in the generator.

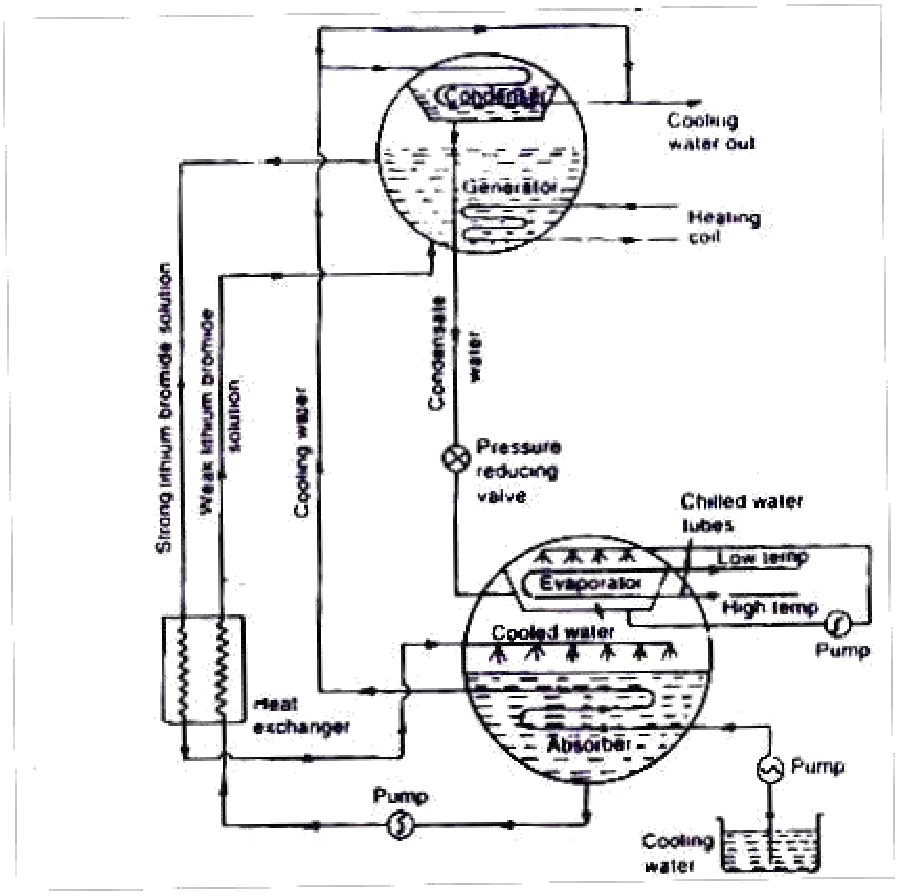


Fig. 3.4. Lithium-Bromide absorption refrigeration system.

The refrigerant water vapours formed in the generator due to heating of solution are passed to the condenser where they are cooled and condensed by the cooling water flowing through the condenser water tubes. The cooling water for condensing is pumped from the cooling water pond or tower. This cooling water first enters the absorber where it takes away the heat of condensation and dilution. The condensate from the condenser is supplied to the evaporator to compensate the water vapour formed in the evaporator. The pressure reducing valve reduces the pressure of condensate from the condenser pressure to the evaporator pressure. The cooled water from the evaporator is pumped and sprayed in the evaporator in order to cool the water for air conditioning flowing through the chilled tubes. This completes the cycle.

Note: The pressure difference between the generator and the absorber and the gravity due to the height difference of the two shells is utilised to create the pressure for the spray.

6.8 Steam Jet Refrigeration System:

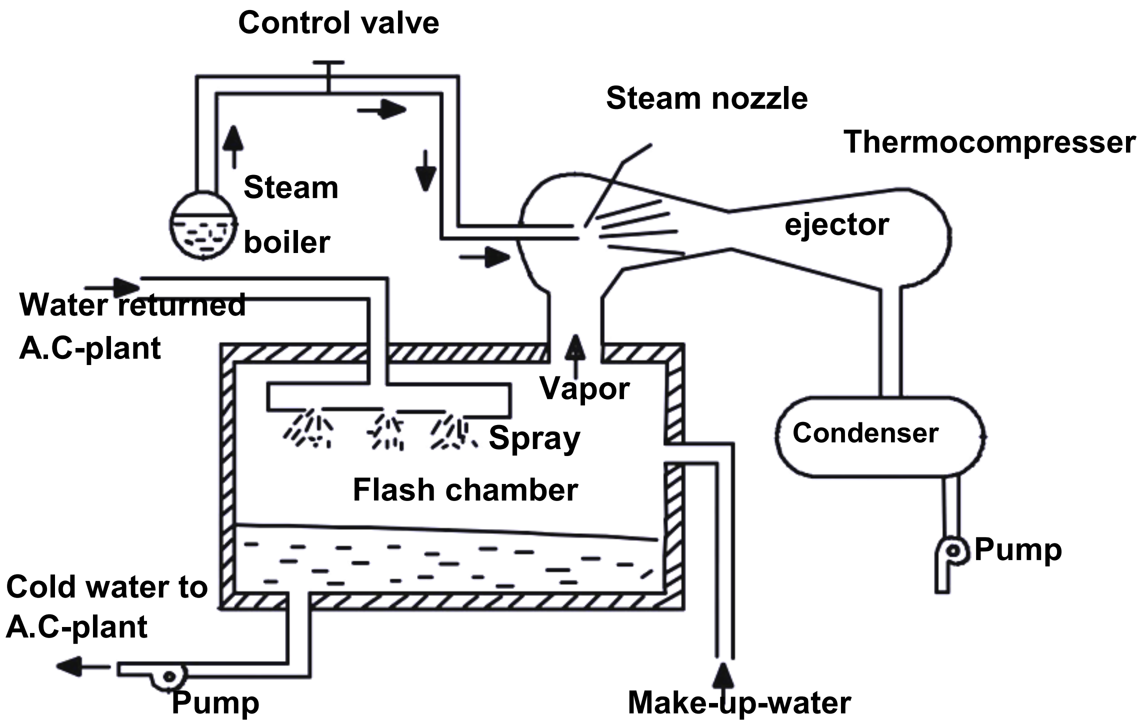


Fig.6.8. Steam jet refrigeration system

This system uses the principle of boiling the water below 100^{0}C. If the pressure on the surface of the water is reduced below atmospheric pressure, water can be made boil at low temperatures. Water boils at 6^{0}C, when the pressure on the surface is 5 cm of Hg and at 10^{0}C, when the pressure is 6.5 cms of Hg. The very low pressure or high vacuum on the surface of the water can be maintained by throttling the steam through jets or nozzles. The general arrangement of the system is shown in the Fig.6.8.

Consider a flash chamber contains 100 kg of water. If suddenly 1 kg of water is removed by boiling, as pressure is reduced due to throttling of steam through nozzles. Approximately 2385 kJ of heat will be removed from the water, which is equivalent to heat of evaporation of water. The fall in temperature of the remaining water will be

$$Q = m\,C_p dT$$

$$dT = \frac{2385}{99 * 4.187} = 5.7^0\,C$$

Evaporating one more kg of water reduces the remaining water temperature by 5.7^{0}C further. Thus by continuing this process, the remaining water can be made to freeze. Water is the refrigerant used in the steam jet refrigeration system. As water freezes at 0^{0}C, then either refrigeration has to be stopped or some device is required to pump the ice.

Operation:

High pressure steam is supplied to the nozzle from the boiler and it is expanded. Here, the water vapor originated from the flash chamber is entrained with the high velocity steam jet and it is further compressed in the thermo compressor. The kinetic energy of the mixture is converted into static pressure and mass is discharged to the condenser. The condensate is usually returned to the boiler. Generally, 1% evaporation of water in the flash chamber is sufficient to decrease the temperature of chilled water to 6^{0}C. The chilled water in the flash chamber is circulated by a pump to the point of application. The warm water from the load is returned to the flash chamber. The water is sprayed through the nozzles to provide maximum surface area for cooling. The water, which is splashed in the chamber and any loss of cold water at the application, must be replaced by makeup water added to the cold water circulating system.

Advantages:

a) *It is flexible in operation; cooling capacity can be easily and quickly changed.*
b) *It has no moving parts as such it is vibration free.*
c) *It can be installed out of doors.*
d) *The weight of the system per ton of refrigerating capacity is less.*
e) *The system is very reliable and maintenance cost is less.*
f) *The system is particularly adapted to the processing of cold water used in rubber mills,, distilleries, paper mills, food processing plants, etc.*
g) *This system is particularly used in air-conditioning installations, because of the complete safety of water as refrigerant and ability to adjust quickly to load variations and no hazard from the leakage of the refrigerant.*

Disadvantages:

a) *The use of direct evaporation to produce chilled water is usually limited as tremendous volume of vapor is to be handled.*
b) *About twice as much heat must be removed in the condenser of steam jet per ton of refrigeration compared with the vapor compression system.*
c) *The system is useful for comfort air-conditioning, but it is not practically feasible for water temperature below* 4^0C

Thermoelectric Refrigeration

Thermoelectric cooling uses the Peltier effect to create a heat flux between the junctions of two different types of materials.

This effect is commonly used in camping and portable coolers and for cooling electronic components and small instruments.

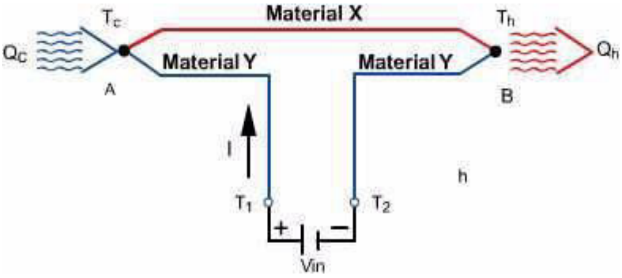
Applying a DC voltage difference across the thermoelectric module, an electric current will pass through the module and heat will be absorbed from one side and released at the opposite side. One module face, therefore, will be cooled while the opposite face simultaneously is heated.

On the other hand, maintaining a temperature difference between the two junctions of the module, a voltage difference will be generated across the module and an electrical power is delivered.

Thermoelectricity is based upon following basic principles:

1. SEEBECK EFFECT
2. PELTIER EFFECT
3. THOMSON EFFECT
4. JOULE EFFECT
5. FOURIER EFFECT

Peltier effect



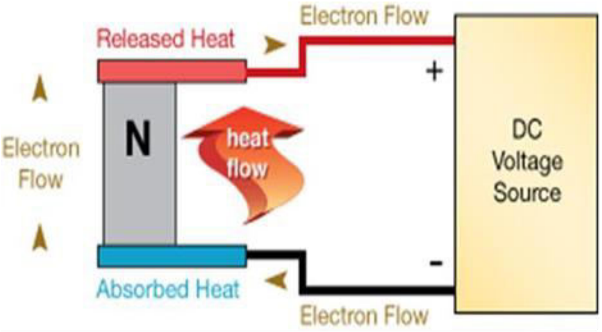
When a current is made to flow through a junction between two conductors A and B, heat may be generated (or removed) at the junction. The Peltier heat generated at the junction per unit time, Q , is equal to;

$$Q \alpha I$$
$$Q = \pi abI$$
$$\pi ab = \pi a - \pi b$$

where (πa & πb) is the Peltier coefficient of conductor A & B, and I is the electric current (from A to B).

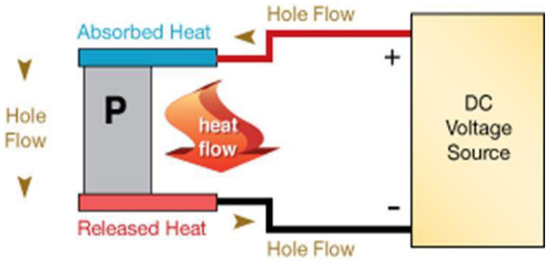
BASIC MECHANISM OF THERMOELECTRICS

Bismuth telluride (a semiconductor), is sandwiched between two conductors, usually copper. A semiconductor (called a pellet) is used because they can be optimized for pumping heat and because the type of charge carriers within them can be chosen. The semiconductor in this examples N type (doped with electrons) therefore, the electrons move towards the positive end of the battery. The semiconductor is soldered to two conductive materials, like copper. When the voltage is applied heat is transported in the direction of current flow



N-TYPE SINGLE SEMICONDUCTOR PELLET

When a p type semiconductor (doped with holes) is used instead, the holes move in a direction opposite the current flow. The heat is also transported in a direction opposite the current flow and in the direction of the holes. Essentially, the charge carriers dictate the direction of heat flow.



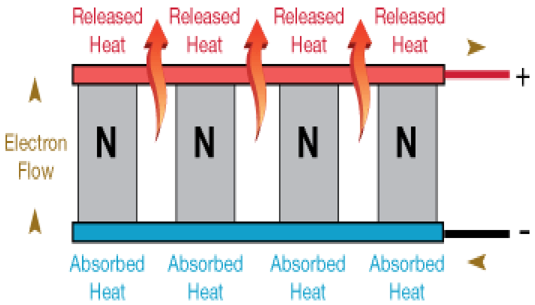
P-TYPE SINGLE SEMICONDUCTOR PELLET

Method of Heat Transport:

Electrons can travel freely in the copper conductors but not so freely in the semiconductor.As the electrons leave the copper and enter the hot-side of the p-type, they must fill a "hole" in order to move through the p-type. When the electrons fill a hole, they drop down to a lower energy level and release heat in the process.Then, as the electrons move from the p-type into the copper conductor on the cold side, the electrons are bumped back to a higher energy level and absorb heat in the process. Next, the electrons move freely through the copper until they reach the cold side of the n-type semiconductor. When the electrons move into the n-type, they must bump up an energy level in order to move through the semiconductor. Heat is absorbed when this occurs.Finally, when the electrons leave the hot-side of the n-type, they can move freely in the copper. They drop down to a lower energy level and release heat in the process.

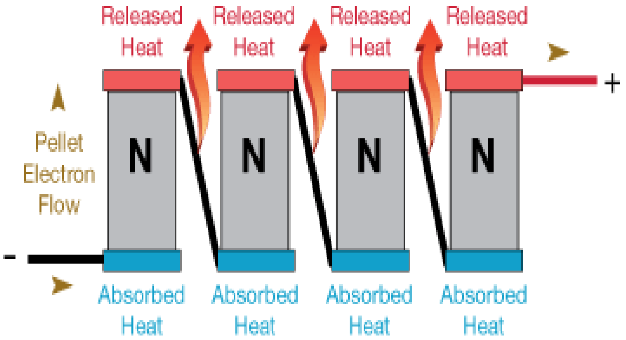
ELECTRICALLY AND THERMALLY PARALLEL MULTIPLE PELLETS

To increase heat transport, several p type or n type thermoelectric(TE) components can be hooked up in parallel. However, the device requires low voltage and therefore, a large current which is too great to be commercially practical.



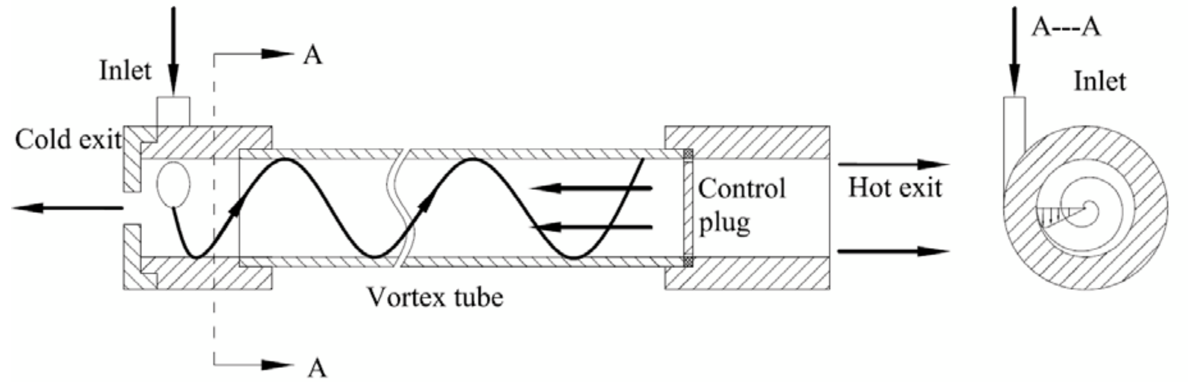
THERMALLY PARALLEL AND ELECTRICALLT IN SERIES MULTIPLE PELLETS

The TE components can be put in series but the heat transport abilities are diminished because the interconnecting between the semiconductors creates thermal shorting.



Vortex tube refrigeration system

A vortex tube is a thermal device, which generates two streams at different temperature from a single injection. Injected into the vortex tube tangentially, the compressed gas is then divided into two parts and exhausted from the exits at temperatures lower and higher than the inlet gas, respectively. In this way, cold and hot streams are generated by only the vortex tube without any additional components. Figure 1 shows the structure of a counter-flow vortex tube as well as the proposed flow behaviour inside the tube. Importantly, as the vortex tube contains no other part inside the tube, the separation of two streams at different temperature by the vortex tube can only be attributed by the effects of fluid dynamics. On comparison with other industry-based technologies, the significant advantages of the vortex tube, such as having no moving parts, being small, low in cost, maintenance free and having adjustable instant cold and hot streams, encourage the on-going investigations into the mechanism of this simple device, with the objective of improving the tube performance and identifying of the primary factors underlying its operation.



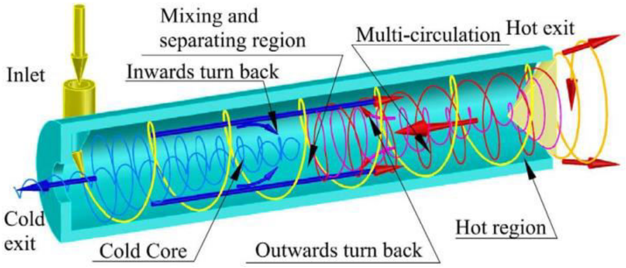
Flow structure in a counter-flow vortex tube

Several hypothesises have been proposed for the basic of the temperature separation, but a well-accepted explanation has not been forthcoming due to the complex flow mechanisms inside a vortex tube. The adiabatic compression and expansion caused by the turbulent eddies, form the basic for the temperature separation, and provide a theoretical prediction of the temperatures based on the pressures at the exits. However, compression of the working fluid cannot be considered as the reason for the temperature rise, because the pressure inside a vortex tube is always lower than the inlet pressure.

The generation of hot and cold streams in a vortex tube and is based on partial stagnation and mixture due to the nature of the multi-circulation occurring in the rear part of the tube, and the pressure gradient near the injection port.

Visualizations of theflow structure,and measurements ofthe velocity componentsinbothair-andwater-operatedvortextubes,showthattheflow to be divided into two streams at different temperatures.

The cooling effect in a vortex tube



Flow structure inside a counter-flow vortex tube

The cooling effect of a vortex tube is the result of the sudden expansion of the working fluid near the injection port. When the fluid is injected into the vortex tube, the main part of the fluid rotates and moves along the periphery towards the hot end. Near the injection point, the inner part of the peripheral flow turns back and moves towards the cold exit. A cold core is formed near the injection due to the pressure gradient of the forced vortex, and the temperature drops due to the decreased pressure of the working fluid in this cold core. The flow behaviour in the cold part of a vortex tube can be seen in Figure , which shows the inwards turn back of the inner flow and the cold core.

Pulse tube refrigerator

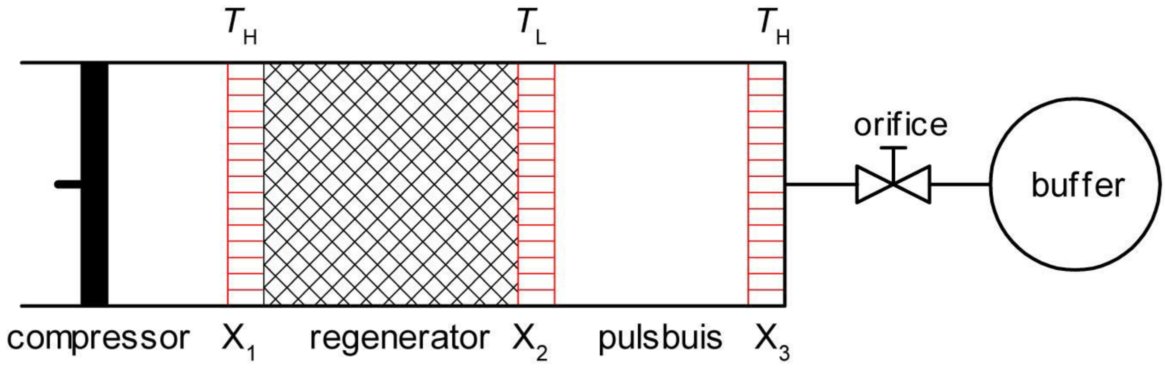
The pulse tube refrigerator (PTR) or pulse tube cryocooler is a developing technology that emerged largely in the early 1980s with a series of other innovations in the broader field of thermoacoustics. In contrast with other cryocoolers (e.g. Stirlingcryocooler and GM-refrigerators), this cryocooler can be made without moving parts in the low temperature part of the device, making the cooler suitable for a wide variety of applications.

Uses

Pulse tube cryocoolers are used in industrial applications such as semiconductor fabrication and in military applications such as for the cooling of infrared sensors. Pulse tubes are also being developed for cooling of astronomical detectors where liquid cryogens are typically used, such as the Atacama Cosmology Telescope or the Qubic experiment (an interferometer for cosmology studies). PTRs are used as precoolers of dilution refrigerators. Pulse tubes will be particularly useful in space-based telescopes where it is not possible to replenish the cryogens as they are depleted. It has also been suggested that pulse tubes could be used to liquefy oxygen on Mars.

Here the so-called Stirling-type single-orifice pulse-tube refrigerator will be treated operating with an ideal gas (helium) as the working fluid. Figure 1 represents the Stirling-type single-orifice Pulse-Tube Refrigerator (PTR). From left to right the components are:

- a compressor, with a piston moving back and forth at room temperature TH;
- a heat exchanger X_1 where heat is released to the surroundings;
- a regenerator consisting of a porous medium with a large specific heat,The porous medium can be stainless steel wire mesh, copper wire mesh, phosphor bronze wire mesh or lead balls or lead shot or (rarely) earthen materials to produce very low temperature;
- a heat exchanger X_2 where the useful cooling power is delivered at the low temperature T_L;
- a tube, often called "the pulse tube";
- a heat exchanger X_3 at room temperature where heat is released to the surroundings;
- a flow resistance (often called orifice);
- a buffer volume (a large closed volume at practically constant pressure).



The part in between X_1 and X_3 is thermally insulated from the surroundings, usually by vacuum. The cooler is filled with helium at a pressure ranging from 10 to 30 bar. The pressure varies gradually and the velocities of the gas are low. So the name "pulse" tube cooler is misleading, since there are no pulses in the system.

Operation

The piston moves periodically from left to right and back. As a result, the gas also moves from left to right and back while the pressure within the system increases and decreases. If the gas from the compressor space moves to the right it enters the regenerator with temperature T_H and leaves the regenerator at the cold end with temperature T_L, hence heat is transferred into the regenerator material. On its return the heat stored within the regenerator is transferred back into the gas.

The thermal environment of a gas element near X_2 that moves back and forth in the system changes when it passes the heat exchanger. In the regenerator and in the heat exchanger the heat contact between the gas and its surrounding material is good. Here the temperature of the gas is practically the same as of the surrounding medium. However, in the pulse tube the gas element is thermally isolated (adiabatic), so, in the pulse tube, the temperature of the gas elements varies with the pressure.

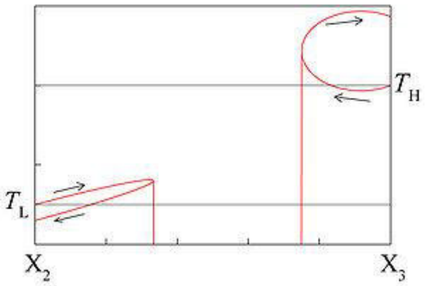


Figure 2: Left: (near X_2): a gas element enters the pulse tube with temperature T_L and leaves it with a lower temperature. Right: (near X_3): a gas element enters the tube with temperature T_H and leaves it with a higher temperature.

Look at figure 1 and consider gas molecules close to X_3 (at the hot end) which move in and out of the pulse tube. Molecules flow into the tube when the pressure in the tube is low (it is sucked into the tube via X_3 coming from the orifice and the buffer). At the moment of entering the tube it has the temperature T_H. Later in the cycle the same mass of gas is pushed out from the tube again when the pressure inside the tube is high. As a consequence its temperature will be higher than T_H. In the heat exchanger X_3 it releases heat and cools down to the ambient temperature T_H.

At the cold end of the pulse tube there is the opposite effect: here gas enters the tube via X_2 when the pressure is high with temperature T_L and return when the pressure is low with a temperature below T_L. They take up heat from X_2: this gives the desired cooling effect.

Thermionic refrigeration system

Production of low temperature dry ice refrigeration system

Raw Materials

The only raw material used in the manufacture of dry ice is carbon dioxide. This raw material is the byproduct of the refinement of gases emitted during the manufacture or refinement of other products. Most carbon dioxide used in the manufacture of dry ice in the United States is derived from refinement of gases given off during the refinement of petroleum and ammonia. The carbon dioxide emitted during these processes is sucked off and "scrubbed" to remove impurities for food grade carbon dioxide that will eventually become dry ice.