REFRIGERATION AND AIR CONDITIONING

Semester: 5TH

STUDY MATERIAL

Refrigeration And Air Conditioning

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Th.5 REFRIGERATION AND AIR CONDITIONING

Chapter -1

1. AIR REFRIGERATION CYCLE.

1.1 Definition of refrigeration and unit of refrigeration.

1.2 Definition of COP, Refrigerating effect (R.E)

1.3 Principle of working of open and closed air system of refrigeration.

1.3.1 Calculation of COP of Bell-Coleman cycle and numerical on it.

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Chapter-1

AIR REFRIGERATION CYCLE

Definition:

The term refrigeration may be defined as the process of removing heat from a substance under controlled conditions. It also includes the process of reducing and maintaining the temperature of its surrounding.

Example:

If some space is to be kept at -20C, we must continuously extract heat which flows into it.

Purpose & Applications of refrigeration & air–conditioning:

Important refrigeration applications are given below

1. Ice making
2. Transportation of foods above and below freezing
3. Industrial air-conditioning
4. Comfort air conditioning
5. Chemical and related industries
6. Medical and surgical aids
7. Processing food products and beverages

Unit of Refrigerating:

The practical unit of refrigeration is expressed in terms of "tone of refrigeration".

One tone of refrigeration is defined as the amount of refrigeration effect produced by the uniform melting of one tonne (1000kg) of ice from and at 00C in 24 hours.

Since the latent heat of ice is 335kj/kg, therefore one tone of refrigeration,

$$1TR = 1000 \times 335 \text{ kj in 24 hours}$$
$$= 232.6 \text{ kj/min}$$

Co-efficient of Performance of a Refrigerator (C.O.P.):

The co-efficient of performance is the ratio of heat extracted in the refrigerator to the work done on the refrigerant.

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Mathematically C.O.P. $= \frac{Q}{W}$

Examples: 1.1 *Find the C.O.P of a refrigeration system if the work input is 80kj/kg and refrigeration effect produced is 160kj/kg of refrigerant following.*

Solution:

Given; $w = 80 kj/kg$; $q = 160 kj/kg$

We know that C.O.P. of a refrigeration system $= \frac{q}{w} = \frac{160}{80} = 2$ *Ans.*

Heat Engine

In heat engine the heat supplied to the engine is converted into useful work. If Q_2 is the heat supplied to the engine and Q_1 is the heat rejected from the engine, the net work done by the engine is given by:

η_E or $(C.O.P)_E = \frac{W_E}{Q_E} = \frac{Workdone}{HeatSupplied} = \frac{Q_2 - Q_1}{Q_2}$

The performance of a heat engine is expressed by its efficiency.

Refrigerator

Refrigerator is a reversed heat engine which either cool or maintain the temperature of a body (T_1) lower than the atmospheric temperature (Ta). This is done by extracting the Heat from a cold body and delivering it to a hot body (Q_2). In doing so, work W_R is required to be done on the system. According to First law of thermodynamics,

$W_R = Q_2 - Q_1$

The performance of a refrigerator is expressed by the ratio of amount of heat taken from the cold body (Q_1) to the amount of work required to be done on the system (W_R). This ratio is called coefficient of performance. Mathematically, coefficient of performance of a refrigerator,

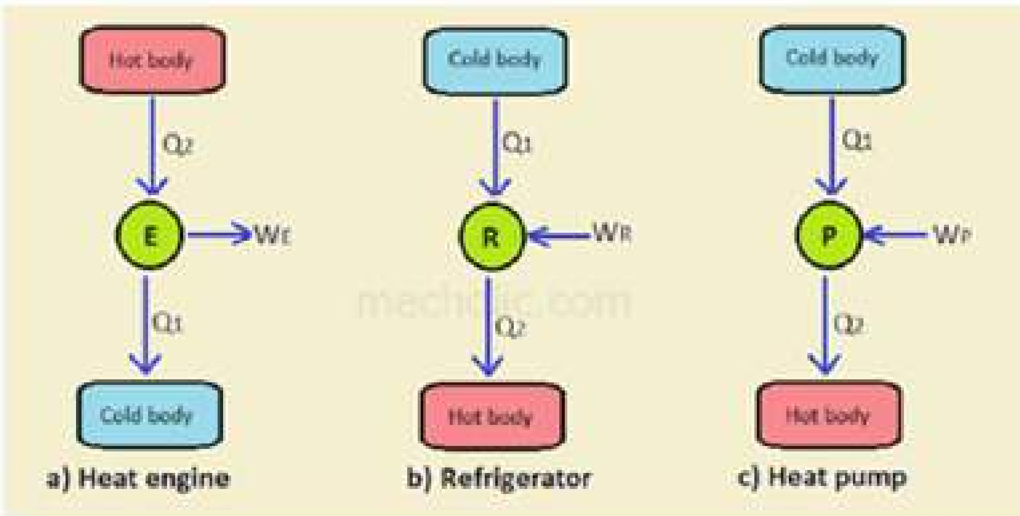
Heat Pump

A refrigerator used for cooling in summer can be used as a heat pump for heating in winter. In the similar way, as discussed for refrigerator, we have

W_p =- $Q_2 - Q_1$

The performance of a heat pump is expressed by the ratio of the amount of the heat delivered to the hot body *(Q2)* to the amount of work required to be done on the system (Wp). This ratio is called coefficient of performance or energy performance ratio (E.P.R.) of a heat pump. Mathematically, coefficient of performance or **energy performance ratio** of a heat pump,

$$(C.O.P)_p = \frac{Q_2}{W_p} = \frac{Q_2}{Q_2 - Q_1} = \frac{Q_1}{Q_2 - Q_1} + 1 = (C.O.P)_R + 1$$



All the components of the heat pumps are same as the refrigerator and even they perform the similar functions; the only difference is that in the heat pumps the components work in a reverse manner. The heat pump is the reverse refrigerator.

Carnot Engine

Carnot engine is a theoretical thermodynamic cycle proposed by **Leonard Carnot**. It gives the estimate of the maximum possible efficiency that a heat engine during the conversion process of heat into work and conversely, working between two reservoirs, can possess. In this section, we will learn about the Carnot cycle and Carnot Theorem in detail.

Air Refrigerator Working on Reversed Carnot Cycle:

If a machine working on reversed Carnot cycle is driven from an external source, it will work or function as a refrigerator. The production of such a machine has not been possible practically because the adiabatic portion of the stroke would need a high speed while during isothermal portion of stroke a very low speed will be necessary. This variation of speed during the stroke, however is not practicable.

- *p-V* and *T-s* diagrams of reversed Carnot cycle are shown in Figs. 1 (*a*) and (*b*).
- Starting from point *l*, the clearance space of the cylinder is full of air, the air is then expanded adiabatically to point *p* during which its temperature falls from T_1 to T_2, and the cylinder is put in contact with a cold body at temperature T_2.
- The air is then expanded isothermally to the point *n*, as a result of which heat is extracted from the cold body at temperature T_2.
- Now the cold body is removed from *n* to *m* air undergoes adiabatic compression with the assistance of some external power and temperature rises to T_1.
- A hot body at temperature T_1 is put in contact with the cylinder. Finally the air is compressed isothermally during which process heat is rejected to the hot body.

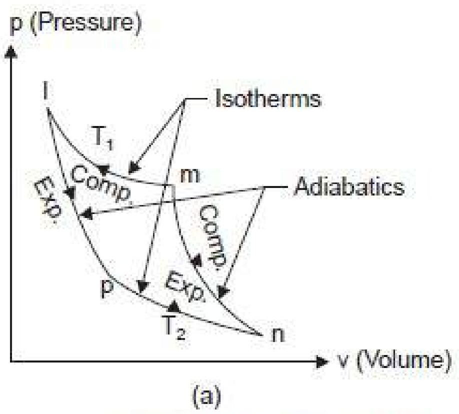


Fig. 1 (*a*) *p-V* diagram for reversed Carnot cycle.

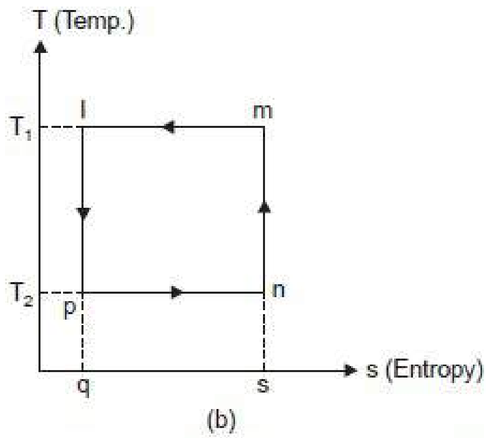


Fig. 1 (*b*) *T-s* diagram for a reversed Carnot cycle.

Heat abstracted from the cold body = Area '*npqs*' = $T_2 \times pn$

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Work done per cycle = Area '*lpnm*'

$= (T_1 - T_2) \times pn$

$$\text{Co-efficient of performance, C.O.P.} = \frac{\text{Heat extracted from the cold body}}{\text{Work done per cycle}} = \frac{Area\ 'npqs'}{Area\ '\text{lpnm}'}$$

$$= \frac{T_2 \times pn}{(T_1 - T_2) \times pn} = \frac{T_2}{(T_1 - T_2)}$$

Since the co-efficient of performance (C.O.P.) means the ratio of the desired effect in kJ/kg to the energy supplied in kJ/kg, therefore C.O.P. in case of Carnot cycle run either as a refrigerating machine or a heat pump or as a heat engine will be as given below :

(i) **For a reversed Carnot cycle 'refrigerating machine':**

$$\text{Co-efficient of performance, C.O.P.} = \frac{\text{Heat extracted from the cold body}}{\text{Work done per cycle}}$$

$$= \frac{Area\ 'npqs'}{Area\ '\text{lpnm}'}$$

$$= \frac{T_2 \times pn}{(T_1 - T_2) \times pn} = \frac{T_2}{(T_1 - T_2)}$$

(ii) **For a Carnot cycle 'heat pump':**

$$\text{Co-efficient of performance, C.O.P.}_{\text{(heat pump)}} = \frac{\text{Heat extracted from the hot body}}{\text{Work done per cycle}}$$

$$= \frac{T_1 \times lm}{(T_1 - T_2) \times pn}$$

$$= \frac{T_1 \times pn}{(T_1 - T_2) \times pn} = \frac{T_1}{(T_1 - T_2)}$$

$$= 1 + \frac{T_2}{(T_1 - T_2)}$$

This indicates that *C.O.P. of heat pump is greater than that of a refrigerator working on reversed Carnot cycle between* the same *temperature* limits $T1$ and $T2$ by *unity*.

(iii) **For a Carnot cycle 'heat engine':**

(iv) $$\text{Co-efficient of performance, C.O.P.}_{\text{(heat pump)}} = \frac{\text{Work obtained/cycle}}{\text{Heat supplied/cycle}}$$

$$= \frac{(T_1 - T_2) \times pn}{T_1 \times lm}$$

$$= \frac{(T_1 - T_2) \times pn}{T_1 \times pn} = \frac{(T_1 - T_2)}{T_1}$$

Examples: 1.2 *A Carnot refrigerator requires 1.3 kW per tonne of refrigeration to maintain a region at low temperature of – 38°C. Determine: (i) C.O.P. of Carnot refrigerator (ii) Higher temperature of the cycle (iii) The heat delivered and C.O.P. when this device is used as heat pump.*

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Solution. $T_2 = 273 - 38 = 235$ K

Power required per tonne of refrigeration = 1.3 kW

(*i*) *C.O.P. of Carnot refrigerator:*

$$(\text{C.O.P.})_{refri} = \frac{\text{Heat extracted from the cold body}}{\text{Work done per cycle}} = \frac{1 tonne}{1.3kW} = \frac{14000kj/hr}{(1.3kW)(3600\sec/hr)} = 2.99 \ Ans$$

(*ii*) *Higher temperature of the cycle:*

$$(\text{C.O.P.})_{refri} = \frac{T_1}{(T_2 - T_1)} \Rightarrow 2.99 = \frac{235}{T_2 - 235} \Rightarrow T_1 = 313.6K \quad Ans$$

(*iii*) *The heat delivered and C.O.P.:*

Heat delivered as Heat Pump:

$$= heat\ absorbed + Work\ done$$

$$= 1 tonne + 1.3kW = \frac{14000kj/hr}{(3600\sec/hr)} + 1.3 = 5.189kj/\sec \quad Ans$$

$$\text{C.O.P.}_{(heat\ pump)} = \frac{\text{Heat extracted from the hot body/Heat delivered}}{\text{Work done per cycle}} = \frac{5.189kj/\sec}{1.3kW} = 3.99 \ Ans$$

Refrigeration-Air Cycles-Open and Closed

Two Ways of Operating Of Bell Coleman Cycle

(i) **Open air refrigeration cycle:** When cooled air from the turbine enters the cabin and comes in physical contact with the occupants. It is not much in use because of moisture added to air in the cabin.

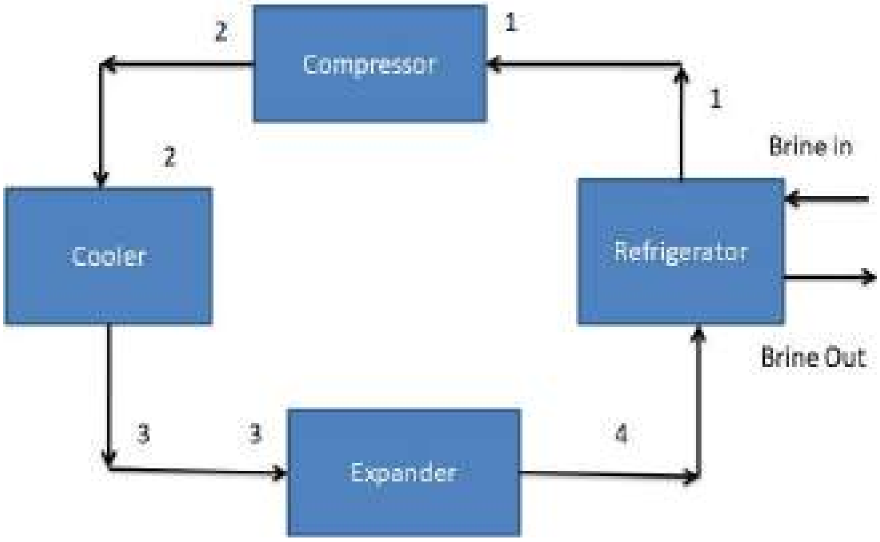
(ii) **Closed air refrigeration cycle or dense cycle:** When cooled air from the turbine passes through the coil and a fan circulates and recirculates cabin air over it. The pressure of cooled air in such systems is much higher than in the open system. Because of high pressure, volume is less and hence density of air is high. It is therefore also called a dense system. It reduces compression ratio and hence COP is high. There is no moisture problem too.

Air Refrigeration System Working On Bell-Coleman Cycle:

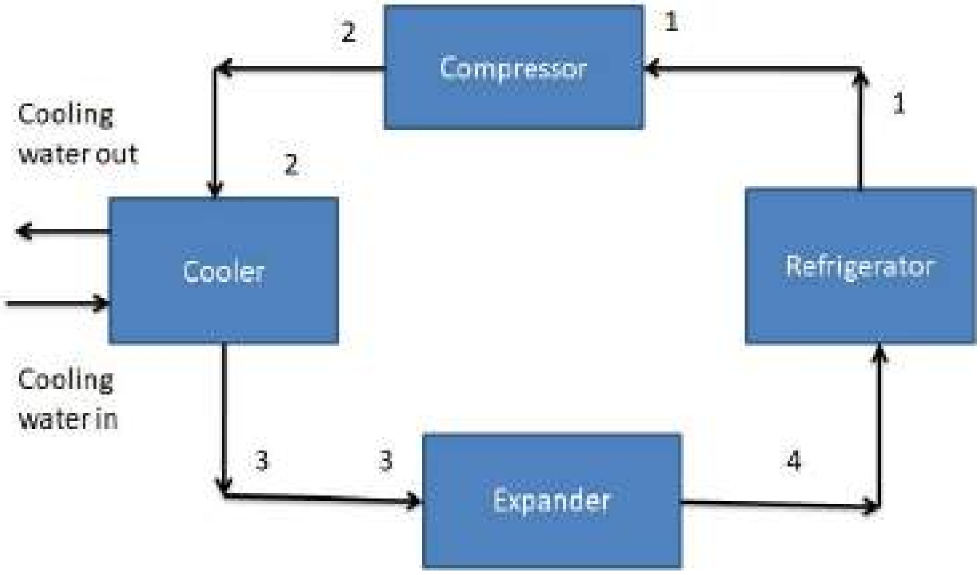
Bell Coleman Cycle also knows as a Reversed Brayton Cycle or the Joule cycle. The fluid is always in a gaseous state which is compressed and expanded. It was one of the most punctual sorts of coolers utilized in boats conveying solidified meat. The cycle uses air as a refrigerant, which is effectively accessible and economical. Used both in cooling and heating effects.

Similarly, refrigeration working on this Cycle has a less COP. The running cost is very high.In air refrigeration system; air is used as the refrigerant which always remains in the gaseous phase. The heat removed consists only of sensible heat and as a result, the coefficient of performance (C.O.P) is low. The various processes are:

Process 1-2: The air leaving the evaporator enters a compressor. Where it is compressed isentropically to higher pressure and temperature.



Closed Bell Coleman Air Cycle



Open Bell Coleman Air Cycle

Process 2-3: This high pressure, high temperature air, then enters a cooler where it is cooled at constant pressure to a low temperature.

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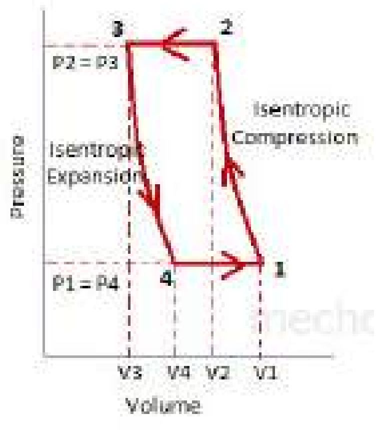
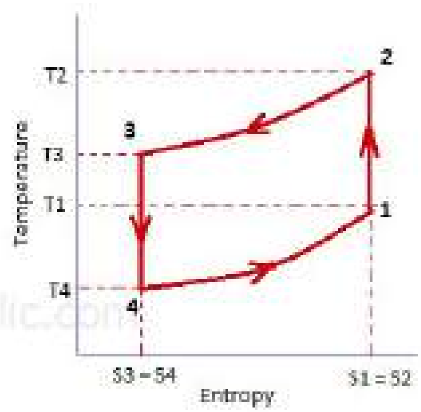
<u>Process 3-4:</u> This high pressure, low temperature air is then expanded in an expander to lower pressure and temperature in a isentropic manner. At point 4, the temperature of the air will be lowest.

<u>Process 4-1</u>: This low temperature air is then passed through the heater coils where it absorbs heat from the space to be cooled namely the refrigerator and the air gets heated back to the initial temperature, but in the process, it cools the refrigerator. And the cycle repeats.

<u>Working of Bell Coleman Cycle :</u>



(a) P-V Diagram



(b) T-S Diagram

Bell Coleman Cycle process is defined below

Let p_1, v_1, and T_1 be the pressure, volume, and temperature of the air.

- **Isentropic Compression**: The Cold air from the fridge is brought into the compressor and compressed isentropically. During this procedure, the pressure increments from P_1 to P_2. The specific volume diminishes from V_1 to V_2 and the temperature increments from T_1 to T_2. During this procedure Entropy‘s’ stays steady (s_1=s_2). No heat is absorbed or rejected by the air.
- **Isothermal compression /Steady Pressure Cooling Process:** The warm air is from the compressor is presently passed into the cooler where it is cooled at consistent pressure, lessening the temperature from T_3 to T_2. Explicit Volume additionally diminishes from v_2 to v_3. Warmth is dismissed by the air during this procedure. Heat dismissed by the air:

$$= q_R = q_{2-3} = c_p(T_2 - T_3)$$

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- **Isentropic Expansion**: Air from the cooler is presently brought into the expander and is extended isentropically. The pressure of the air stays steady during this procedure. Specific volume changes from v_3 to v_4 and the temperature diminishes from T_3 to T_4. No heat is dismissed or consumed by the air.
- **Isothermal expansion/Steady Pressure expansion process**: The cold air from the expander is currently passed into the refrigerator and extended at a consistent pressure. The temperature of the air increments from T_4 to T_1. The specific volume of the air changes from v_4 to v_1.

$$= q_A = q_{4-1} = c_p\left(T_1 - T_4\right)$$

Heat consumed by the air: $q_A = Cp(T_1 - T_4)$

Work done during the cycle per kg of air = Heat rejected – Heat absorbed

$$= q_R - q_A$$

$$= C_p(T_2 - T_3) - C_p(T_1 - T_4)$$

Coefficient of Performance of Bell Coleman Cycle

C.O.P during the cycle per kg of air = $\dfrac{Heat\ absorbed}{workdone}$

$$(\text{C.O.P})_R = \frac{q_A}{q_R - q_A}$$

$$= \frac{C_p\left(T_1 - T_4\right)}{C_p\left(T_2 - T_3\right) - C_p\left(T_1 - T_4\right)} = \frac{\left(T_1 - T_4\right)}{\left(T_2 - T_3\right) - \left(T_1 - T_4\right)}$$

$$= \frac{T_4\left(\frac{T_1}{T_4} - 1\right)}{T_3\left(\frac{T_2}{T_3} - 1\right) - T_4\left(\frac{T_1}{T_4} - 1\right)}$$

Process 1-2,

$$\frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}}$$

Process 3-4,

$$\frac{T_3}{T_4} = \left(\frac{P_3}{P_4}\right)^{\frac{\gamma-1}{\gamma}}$$

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Since , $P_2=P_3$ & $P_1=P_4$,so

$$\frac{T_2}{T_1}=\frac{T_3}{T_4} \text{ or } \frac{T_2}{T_1}=\frac{T_1}{T_4}$$

$$=\frac{T_4}{T_3-T_4}=\frac{1}{\frac{T_3}{T_4}-1}$$

$$=\frac{1}{\left(\frac{P_3}{P_4}\right)^{\frac{\gamma-1}{\gamma}}-1}=\frac{1}{\left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}}-1}$$

$$\text{C.O.P.} =\frac{1}{\left(r_p\right)^{\frac{\gamma-1}{\gamma}}-1}$$

r_p = Compression or Expansion ratio=$\frac{P_1}{P_2}=\frac{P_3}{P_4}$

❖ NOTES:

- ✓ Co-efficient of Performance of a Refrigerator (C.O.P.); C.O.P. $=\frac{Q}{W}$
- ✓ Co-efficient of Performance of a Refrigerator $(C.O.P)_R$=$\frac{T_2}{(T_1-T_2)}$.
- ✓ Co-efficient of Performance of a Pump $(C.O.P)_p$=$(C.O.P)_R$+1==$1+\frac{T_2}{(T_1-T_2)}$
- ✓ Co-efficient of Performance of a Engine $(C.O.P)_E$=

$$=\frac{\text{Work obtained/cycle}}{\text{Heat supplied/cycle}}=\frac{(T_1-T_2)}{T_1}=\frac{1}{(C.O.P)_p}$$

- ✓ Coefficient of Performance **o**f Bell Coleman Cycle=$\frac{T_4}{T_3-T_4}$

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Examples: 1.3 *A refrigeration plant working on the Bell Coleman Cycle, air is compressed to 6 bar from 1 bar. Its starting temperature is 15 ℃. After compression air is cooled to up to 25 ℃ in a cooler before expanding back to 1 bar. Determine the C.O.P of the plant and net refrigerating effect.*

C_p = 1.005 kJ/kg K and C_v= 0.718 kJ/kg K.

Answer

Given: $P_2 = P_3 = 6$ bar, $P1 = P4 = 1$ bar

$$T_1 = 15 + 273 = 288\ K$$

$$T_3 = 25 + 273 = 298\ K$$

$$\gamma = Cp / Cv = 1.005/0.718 = 1.4$$

$$\gamma - 1 = 0.4$$

$$\text{So } \gamma - 1/\gamma = 0.286$$

Therefore

$$T_2/T_1 = (P_2 / P_1)\gamma - 1/\gamma$$

$$\text{So } T_2/T_1 = 1.669$$

Similarly for process 3-4

$$T_4 = \frac{T_3}{1.669} = T_4 = 178.55;$$

$$\text{C.O.P of the cycle is given by, C.O.P} = \frac{T_4}{T_3 - T_4}$$

$$\text{C.O.P} = \frac{178.55}{298 - 178.55} ;\text{C.O.P} = 1.494$$

Net refrigerating effect = Cp ($T_1 - T_4$)

= 1.005(288 – 178.55)

= 109.99 kJ/kg

Advantages of air refrigeration system

1. Air is cheap, easily available.

2. It is not flammable.

3. For a given capacity, weight of air refrigeration system is less compared to other system and hence it is widely used for aircraft cooling.

Disadvantages

1. Since heat removed by air consists only of sensible heat, weight of air required is high.
2. C.O.P of the system is low compared to other systems.

Slove the Problems

1. A heat pump is used for heating the interior of a house in a cold climate. The ambient temperature is –5°C and the desired interior temperature is 25°C. The compressor of the heat pump is to be driven by a heat engine working between 1000°C and 25°C. Treating both cycles as reversible, calculate the ratio in which the heat pump and the heat engine share the heating load. **[Ans. 7]**
2. A refrigerating plant is required to produce 2.5 tonnes of ice per day at – 4°C from water at 20°C. If the temperature range in the compressor is between 25°C and – 6°C, calculate power required to drive the compressor. Latent heat of ice = 335 kJ/kg and specific heat of ice = 2.1 kJ/kg K. **[Ans. 1.437 kW]**
3. A refrigerator using Carnot cycle requires 1.25 kW per tonne of refrigeration to maintain a temperature of – 30°C. Find : 1. C.O.P. of the Carnot refrigerator; 2. Temperature at which heat is rejected; and 3. Heat rejected per tonne of refrigeration. **[Ans. 2.8 ; 55.4°C ; 284 kJ/min]**
4. A Carnot cycle machine operates between the temperature limits of 47°C and –30°C. Determine the C.O.P. when it operates as 1. a refrigerating machine ; 2. a heat pump ; and 3. a heat engine. **[Ans. 3.16 ; 4.16 ; 0.24]**
5. Ten tonnes of fish is frozen to –30°C per day. The fish enters the freezing chamber at 30°C and freezing occurs at –3°C. The frozen fish is cooled to –30°C. The specific heats of fresh and frozen fish are 3.77 kJ/kg K and 1.67 kJ/kg K respectively while latent heat of freezing is 251.2 kJ/kg K. Find the tonnage of the plant which runs for 18 hours per day. The evaporator and condensor temperatures are –40°C and 45°C respectively. If the C.O.P. of the plant is 1.8, determine the power consumption of the plant in kW. Also find the refrigerating efficiency of the plant. **[Ans. 18.6 TR ; 36.1 kW ; 65.7%]**
6. A Carnot refrigeration system has working temperature of –30°C and 40°C. What is the maximum C.O.P. possible ? If the actual C.O.P. is 75% of the maximum, calculate the actual refrigerating effect produced per kilowatt hour. **[Ans. 3.47 ; 0.743 TR]**
7. A refrigerator storage is supplied with 30 tonnes of fish at a temperature of 27°C. The fish has to be cooled to – 9°C for preserving it for long period without deterioration. The cooling takes place in 10 hours. The specific heat of fish is 2.93 kJ/kg K above freezing point of fish and 1.26 kJ/kg K below freezing point of fish which is – 3°C. The latent heat of freezing is 232 kJ/kg. What is the capacity of the plant in tonnes of refrigeration for cooling the fish ? What would be the ideal C.O.P. between this temperature range ? If the actual C.O.P. is 40% of the ideal, find the power required to run the cooling plant. **[Ans. 78 TR ; 7.33 ; 93.3 kW]**

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8. A refrigerating system working on Bell-Coleman cycle receives air from cold chamber at $-5°C$ and compresses it from 1 bar to 4.5 bar. The compressed air is then cooled to a temperature of 37°C before it is expanded in the expander. Calculate the C.O.P. of the system when compression and expansion are (*i*) isentropic ; and (*ii*) follow the law $pv^{1.25}$ = constant. **[Ans. 1.86 ; 1.98]**

9. A Bell-Coleman refrigerator works between 4 bar and 1 bar pressure limits. After compression, the cooling water reduces the air temperature to 17°C. What is the lowest temperature produced by the ideal machine ? Compare the coefficient of performance of this machine with that of the ideal Carnot cycle machine working between the same pressure limits, the temperature at the beginning of compression being $-13°C$. **[Ans. $-78°C$; 2.07, 1.02]**

10. An air refrigerator working on Bell-Coleman cycle takes air into the compressor at 1 bar and 268 K. It is compressed in the compressor to 5 bar and cooled to 298 K at the same pressure. It is further expanded in the expander to 1 bar and discharged to take the cooling load. The isentropic efficiencies of the compressor and expander are 85% and 90% respectively. Determine ; 1. refrigeration capacity of the system if the air circulated is 40 kg / min ; 2. power required for the compressor ; and 3. C.O.P. of the system. **[Ans. 13.14 TR ; 46 kW; 0.812]**

11. An air refrigeration system having pressure ratio of 5 takes air at 0°C. It is compressed and then cooled to 19°C at constant pressure. If the efficiency of the compressor is 95 % and that of expander is 75%, determine: 1. the refrigeration capacity of the system, if the flow of air is 75 kg/min ; 2. the power of the compressor ; and 3. C.O.P. of the system. Assume compression and expansion processes to be isentropic. Take $\gamma = 1.4$; $c_p = 1$ kJ/kg K ; and $c_v = 0.72$ kJ/kg K.

[Ans. 31.68 TR; 106.6 kW; 1.71]

12. A 5 tonne refrigerating machine operating on Bell Coleman cycle has an upper limit of pressure of 12 bar. The pressure and temperature at the start of compression are 1 bar and 17°C respectively. The compressed air cooled at constant pressure to a temperature of 40°C enters the expansion cylinder. Assuming both the expansion and compression processes to be isentropic with $\gamma = 1.4$; Determine : 1. C.O.P.; 2. quantity of air in circulation per minute; 3. piston displacement of compressor and expander; 4. bore of compressor and expansion cylinders. The unit runs at 250 r.p.m. and is double acting. Stroke length is 200 mm ; and 5. power required to drive the unit.Take $c_p = 1$ kJ/kg K ; $c_v = 0.71$ kJ/kg K ; $R = 0.287$ kJ/kg K.

[Ans. 0.952 ; 7.65 kg/min ; 6.37 m³/min, 3.35 m³/min ; 284 mm ; 18.4 kW]

13. An air refrigerator used for food storage, provides 50 TR. The temperature of air entering the compressor is 7°C and the temperature before entering into the expander is 27°C. Assuming a 70% mechanical efficiency, find : 1. actual C.O.P; and 2. the power required to run the compressor.

The quantity of air circulated in the system is 100 kg/min. The compression and expansion follow the law $pv^{1.3}$ = constant.

Take $\gamma = 1.4$; $c_p = 1$ kJ/kg K for air. **[Ans. 1.13 ; 110.6 kW]**

14. A dense air refrigerating system operating between pressures of 17.5 bar and 3.5 bar is to produce 10 tonnes of refrigeration. Air leaves the refrigerating coils at $-7°C$ and it leaves the air cooler at 15.5°C. Neglecting losses and clearance, calculate the net work done per minute and the coefficient of performance. For air $c_p = 1.005$ kJ/kg K and $\gamma = 1.4$. **[Ans. 1237 kJ/min ; 1.7]**

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Th.5 REFRIGERATION AND AIR CONDITIONING

Chapter -2

2. SIMPLE VAPOUR COMPRESSION REFRIGERATION SYSTEM

2.1 Schematic Diagram of Simple Vapors Compression Refrigeration System'

2.2 Types

2.2.1 Cycle with dry saturated vapors after compression.

2.2.2 Cycle with wet vapors after compression.

2.2.3 Cycle with superheated vapors after compression.

2.2.4 Cycle with superheated vapors before compression.

2.2.5 Cycle with sub cooling of refrigerant

2.2.6 Representation of above cycle on temperature entropy and pressure enthalpy diagram

2.2.7 Numerical on above (determination of COP, mass flow)

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Chapter-2

SIMPLE VAPOUR COMPRESSION REFRIGERATION SYSTEM

Simple Vapor Compression Refrigeration System: A simple vapor compression refrigeration system consists of the following equipments: i) Compressor ii) Condenser iii) Expansion valve iv) Evaporator.

Introduction to Vapour Compression System:

- 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 (NH_3), 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.

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.

Mechanism of a Vapour Compression Refrigeration System:

(i) Compression of the vapour, thereby increasing pressure.

(ii) Condensing these vapours and rejecting heating to the cooling medium (usually water or atmospheric air).

(iii) Expanding the condensed liquid refrigerant thereby lowering the pressure and corresponding saturation temperature.

(iv) Evaporating the liquid refrigerant thereby absorbing heat from the body or space to be cooled or refrigerated. It is due to this requirements of compression that the system is called Vapour Compression System and the cycle of operation is called Vapour Compression Cycle of Refrigeration.

Cycle of Vapour Compression Refrigeration System

This cycle, incorporating the compressor and condenser is shown in Fig.2.1. Here the liquid at state D, the discharge of the condenser, is still at the same pressure of compresor discharge. If it were allowed to boil as it is, it will do so at the saturation temperature and will require heat to be supplied to it. Heat could possibly flow only from some temperature higher than this saturation temperature. But the requirement is to achieve a low temperature.

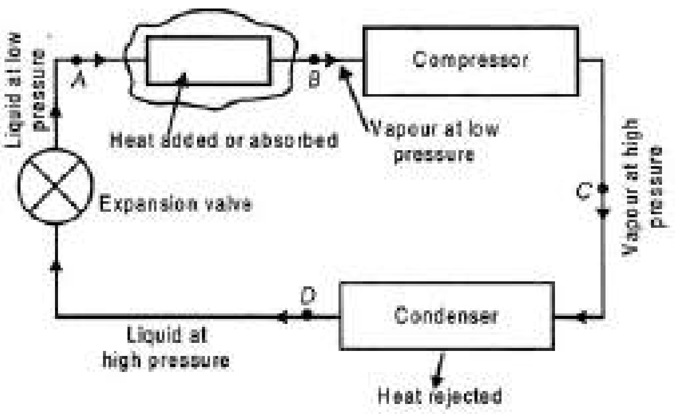
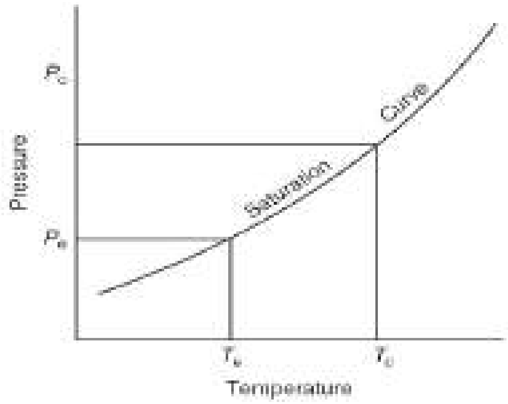


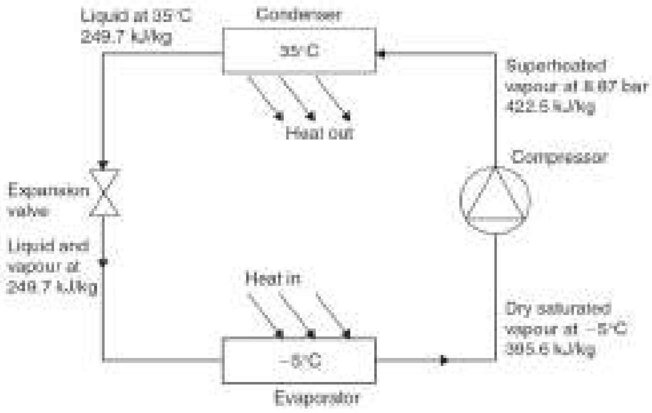
Fig. 2.1 *Schematic of vapour compression cycle*

The *vapour compression cycle* is used for refrigeration in preference to gas cycles; making use of the latent heat enables a far larger quantity of heat to be extracted for a given refrigerant mass flow rate. This makes the equipment as compact as possible.

A liquid boils and condenses – the change between the liquid and the gaseous states – at a temperature which depends on its pressure, within the limits of its freezing point and critical temperature see Figure 2.2. In boiling it must obtain the latent heat of evaporation and in condensing the latent heat is given up.

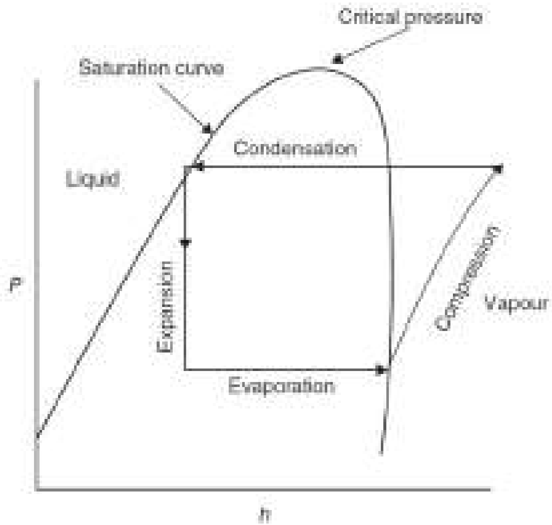


2.2 Evaporation and condensation of a fluid.



2.3 Simple vapour compression cycle with pressure and enthalpy.

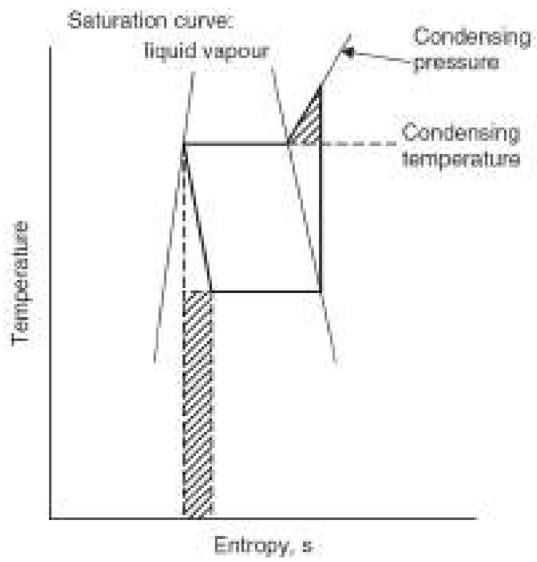
Heat is put into the fluid at the lower temperature and pressure thus providing the latent heat to make it vaporize. The vapour is then mechanically compressed to a higher pressure and a corresponding saturation temperature at which its latent heat can be rejected so that it changes back to a liquid. The cycle is shown in Figure 2.3. The cooling effect is the heat transferred to the working fluid in the evaporation process, i.e. the change in enthalpy between the fluid entering and the vapour leaving the evaporator.



2.4 Pressure–enthalpy, *P*–*h* diagram, showing vapour compression cycle.

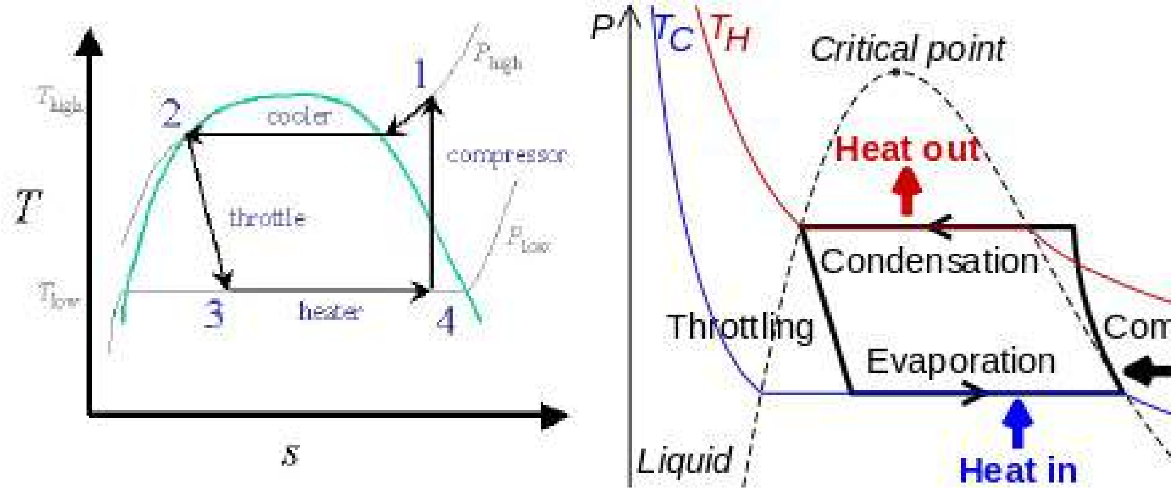
P, and on the horizontal, *h*, enthalpy. The *saturation curve* defines the boundary of pure liquid and pure gas, or vapour. In the region marked vapour, the fluid is superheated vapour. In the region marked liquid, it is sub-cooled liquid. At pressures above the top of the curve, there is no distinction between liquid and vapour. Above this pressure the gas cannot be liquefied. This is called the *critical pressure.* In the region beneath the curve, there is a mixture of liquid and vapour.

The simple vapour compression cycle is superimposed on the *P–h* diagram in Figure 2.4. The evaporation process or vaporization of refrigerant is a constant pressure process and therefore it is represented by a horizontal line. In the compression process the energy used to compress the vapour turns into heat and increases its temperature and enthalpy, so that at the end of compression the vapour state is in the superheated part of the diagram and outside the saturation curve. A process in which the heat of compression raises the enthalpy of the gas is termed *adiabatic compression*. Before condensation can start, the vapour must be cooled. The final compression temperature is almost always above the condensation temperature as shown, and so some heat is rejected at a temperature above the condensation temperature. This represents a deviation from the ideal cycle. The actual condensation process is represented by the part of the horizontal line within the saturation curve.

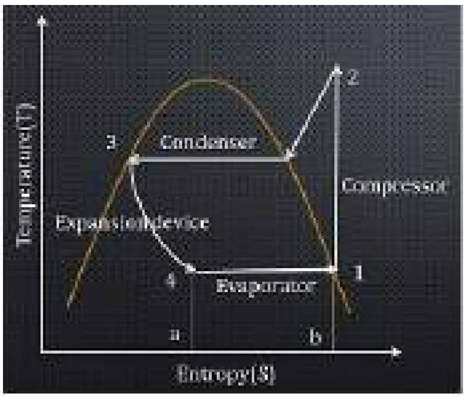


2.5 Temperature–entropy diagram for ideal vapour compression cycle

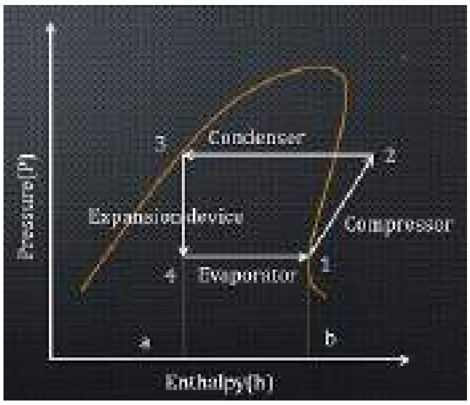
When the simple vapour compression cycle is shown on the temperature–entropy diagram (Figure 2.5), the deviations from the reversed Carnot cycle can be identified by shaded areas. The adiabatic compression process continues beyond the point where the condensing temperature is reached. The shaded triangle represents the extra work that could be avoided if the compression process changed to isothermal (i.e. at constant temperature) at this point, whereas it carries on until the condensing pressure is attained. Expansion is a constant enthalpy process. It is drawn as a vertical line on the *P–h* diagram. No heat is absorbed or rejected during this expansion, the liquid just passes through a valve. Since the reduction in pressure at this valve must cause a corresponding drop in temperature, some of the fluid will flash off into vapour to remove the energy for this cooling. The volume of the working fluid therefore increases at the valve by this amount of flash gas, and gives rise to its name, the expansion valve. No attempt is made to recover energy from the expansion process, e.g. by use of a turbine. This is a second deviation from the ideal cycle. The work that could potentially be recovered is represented by the shaded rectangle in Figure 2.5.



2.6 T-s & P-v diagram of Vapor-Compression Refrigeration Cycle.



T-s diagram



P-h diagram

The various processes of the cycle 1-2-3-4 are as given below:

Process1-2(Isentropic Compression):

- Low pressure and low temperature refrigerant air passes through compressor.
- In compressor it converts into high pressure and high temperature refrigerant air.

- Process shown on T-S and P-H diagram (1-2).
- Refrigerant air entering at compressor is at dry saturated state (1), hence point lies on saturated line after isentropic compression air becomes superheated state (2).

Process 2-3(Constant pressure heat rejection):

- In this process high temperature and high pressurised refrigerant air reject its heat to surrounding by converting into liquid as shown as 2-3 process.
- Condenser rejects heat to atmosphere maintaining constant pressure. In this process high temperature and high pressurised air converted into high temperature and high pressurised liquid refrigerant.

Process 3-4 (Throttling -Expansion):

- In this process high temperature high pressurised liquid refrigerant converted into low temperature low pressurised liquid refrigerant.
- This process is generally carried out in capillary tube, throttling valve etc. Fig shows T-S and P-H diagram showing this process by 3-4.
- At the end of this process that is at state 4 very cold liquid refrigerants is obtained.

Process 4-1(Heat Extraction):

- It is heat absorption process in which heat is absorbed by cold liquid refrigerant and converted into air.
- This process is carried out in Evaporator as shown by 4-1.

Cold refrigerant absorb heat from application and converted into air vapour for compression process in 1-2 and this cycle continues.

Types of Vapour Compression Cycles

We are 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,

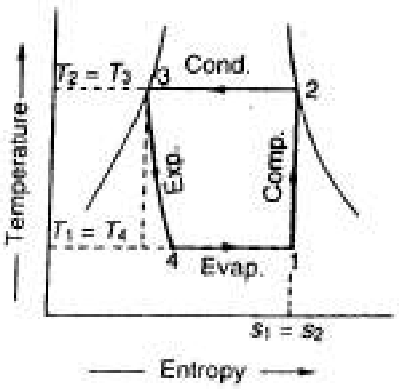
3. Cycle with superheated vapour after compression,

4. Cycle with superheated vapour before compression, and

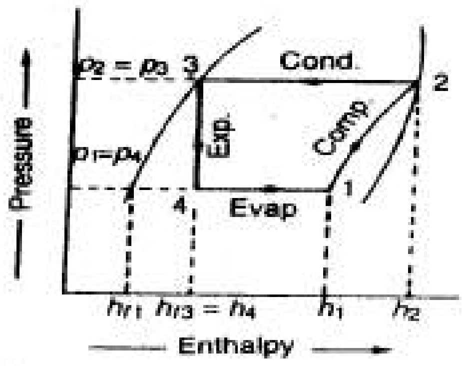
5. Cycle with undercooling or subcooling of refrigerant.

Now we shall discuss all the above mentioned cycles, one by one,

1. Cycle with dry saturated vapour after compression:



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



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

- At point 1, let T_1, p_1 and s_1 be the temperature, pressure and entropy of the vapour refrigerant respectively.

1. *Compression process.*

   The vapour refrigerent at low pressure p1 and temperature T1 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 (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 RE. 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

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

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,

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

Problem 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. (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,

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

$\therefore$ 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

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

Problem 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 : $T_2 = T_3 = 25°C = 25 + 273 = 298$ K ; $T_1 = T_4 = -10°C = 263$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.

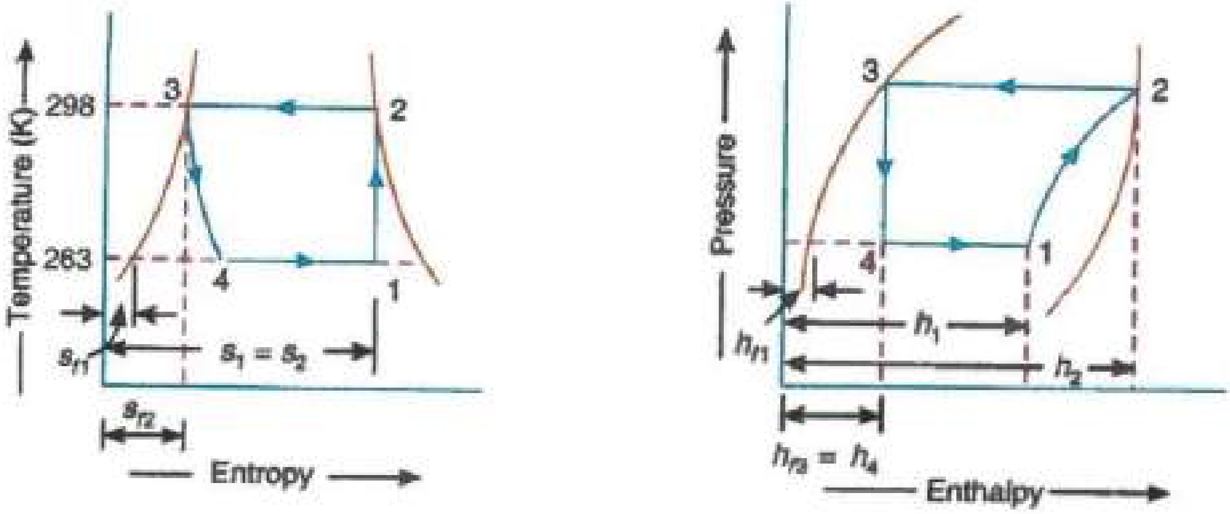
$$s_1 = s_{f1} + \frac{x_1 h_{fg1}}{T_1} = 0.5443 + \frac{x_1 \times 1297.68}{263}$$

$= 0.5443 + 4.934\ x_1$

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 $x_1 = 0.91$



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

We know that enthalpy at point 1,

$$h_1 = hf_1 + x_1\ h_{fg1} = 135.37 + 0.91 \times 1297.68 = 1316.26 \text{ kJ/kg}$$

and enthalpy at point 2,

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

$\therefore$ Coefficient of performance of the cycle

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

Problem 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.*

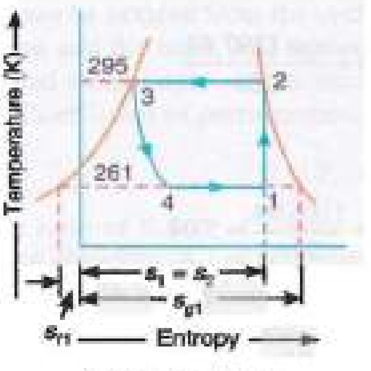
Data :

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*

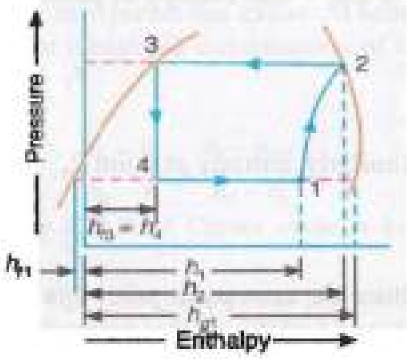
Solution. Given : $p_2 = p_3 = 60$ bar ; $p_1 = p_4 = 25$ bar ; $T_2 = T_3 = 295$ K ; $T_1 = T_4 = 261$ K ; $h_{f3} = h_4 = 151.96$ kJ/kg ; $h_{f1} = 56.32$ kJ/kg ; $h_{g2} = h_2 = 293.29$ kJ/kg ; $h_{g1} = 322.58$ kJ/kg ; $s_{f2} = 0.554$ kJ/kg K ; $s_{f1} = 0.226$ kJ/kg K ; $s_{g2} = s_2 = 1.0332$ kJ/kg K ; $s_{g1} = 1.2464$ kJ/kg K

1. C.O.P. of the cycle

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



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



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

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

We know that entropy at point 1,

$$s_1 = s_{f1} + x_1 s_{fg1} = s_{f1} + x_1(s_{g1} - s_{f1}) \quad \ldots (\because s_{g1} = s_{f1} + s_{fg1})$$

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

and entropy at point 2, $s_2 = s_{g2} = 1.0332$ kJ/kg K ... (Given) ... (*ii*)

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

$$0.226 + 1.0204\, x_1 = 1.0332 \quad \text{or} \quad x_1 = 0.791$$

We know that enthalpy at point 1,

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

$$= 56.32 + 0.791 (322.58 - 56.32) = 266.93 \text{ kJ/kg}$$

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

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

2. Capacity of the refrigerator

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

$$= h_1 - h_{f3} = 266.93 - 151.96 = 114.97 \text{ kJ/kg}$$

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

$$= 5 \times 114.97 = 574.85 \text{ kJ/min}$$

∴ Capacity of the refrigerator

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

Problem 2.4 *28 tonnes of ice from and at 0°C is produced per day in an ammonia refrigerator. The temperature range in the compressor is from 25°C to –15°C. The vapour is dry and saturated at the end of compression and an expansion valve is used. There is no liquid subcooling. Assuming actual C.O.P. of 62% of the theoretical, calculate the power required to drive the compressor. Following properties of ammonia are given:*

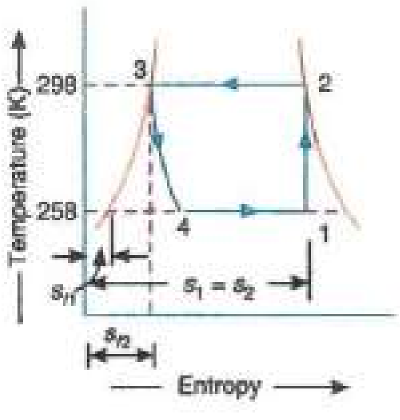
Temperature 0°C	*Enthalpy (kJ/kg)*		*Entropy (kJ/kg K)*	
	Liquid	*Vapour*	*Liquid*	*Vapour*
25	*298.9*	*1465.84*	*1.1242*	*5.0391*
–15	*112.34*	*1426.54*	*0.4572*	*5.5490*

Take latent heat of ice = 335 kJ/kg.

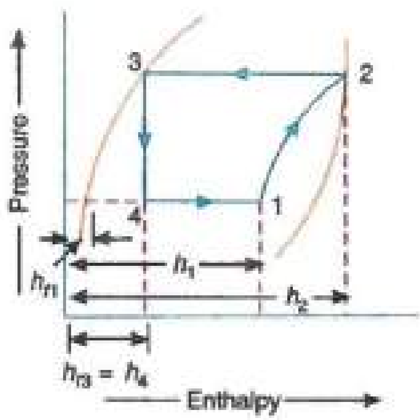
Solution. Given: Ice produced = 28 t/day ; $T_2 = T_3 = 25°C = 25 + 273 = 298K$; $T_1 = T_4 = -15°C = -15 + 273 = 258$ K ; $h_{f3} = h_4 = 298.9$ kJ/kg ; $h_{f1} = 112.34$ kJ/kg ; $h_{g2} = h_2 = 1465.84$ kJ/kg ; $h_{g1} = 1426.54$ kJ/kg ; $s_{f2} = 1.1242$ kJ/kg K ; $s_{f1} = 0.4572$ kJ/kg K ; $s_{g2} = s_2 = 5.0391$ kJ/kg K ; $s_{g1} = 5.5490$ kJ/kg K.

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

First of all, let us find the dryness fraction (x_1) of the vapour refrigerant entering the compressor at point 1.



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



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

We know that entropy at point 1,

$$s_1 = s_{f1} + x_1 s_{fg1} = s_{f1} + x_1(s_{g1} - s_{f1}) \qquad \ldots(\because s_{g1} = s_{f1} + s_{fg1})$$

$$= 0.4572 + x_1(5.5490 - 0.4572)$$

$$= 0.4572 + 5.0918\, x_1 \qquad \ldots(i)$$

and entropy at point 2, $s_2 = s_{g2} = 5.0391$ kJ/kg K ... (Given) ...(ii)

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

$$0.4572 + 5.0918\, x_1 = 5.0391 \quad \text{or} \quad x_1 = 0.9$$

We know that enthalpy at point 1,

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

$$= 112.34 + 0.9\,(1426.54 - 112.34) = 1295.12 \text{ kJ/kg}$$

$$\therefore \quad \text{Theoretical C.O.P.} = \frac{h_1 - h_{f3}}{h_2 - h_1} = \frac{1295.12 - 298.9}{1465.84 - 1295.12} = \frac{996.22}{170.72} = 5.835$$

Since actual C.O.P. is 62% of theoretical C.O.P., therefore

$$\text{Actual C.O.P} = 0.62 \times 5.835 = 3.618$$

We know that ice produced from and at 0°C

Given that

$$= 28 \text{ t/day} = \frac{28 \times 1000}{24 \times 3600} = 0.324 \text{ kg / s}$$

Latent heat of ice = 335 kJ/kg

∴ Refrigeration effect produced

$$= 0.324 \times 335 = 108.54 \text{ kJ/s}$$

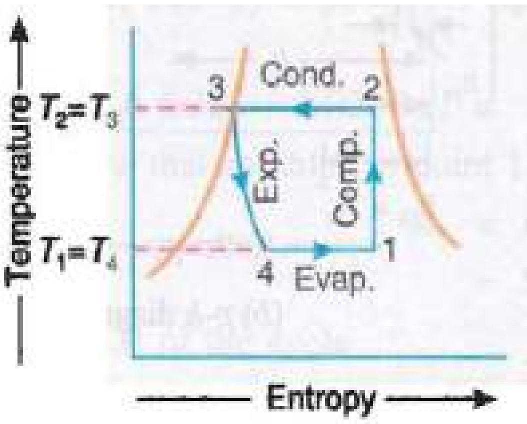
We know that actual C.O.P.,

$$3.618 = \frac{\text{Refrigeration effect}}{\text{workdone}} = \frac{108.54}{\text{workdone}}$$

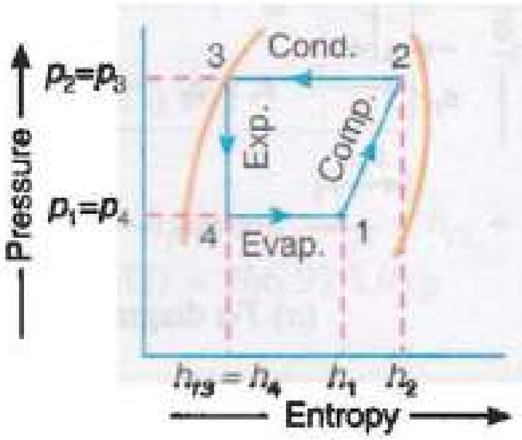
∴ Workdone or power required to drive the compressor

$$= \frac{108.54}{3.618} = 30 \text{ kJ/s or kW } \textbf{Ans.}$$

2. *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. (*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,

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

Problem 2.5 *Find the theoretical C.O.P. for a CO_2 machine working between the temperature range of 25°C and –5°C. The dryness fraction of CO_2 gas during the suction stroke is 0.6. Following properties of CO_2 are given :*

Temperature °C	*Liquid*		*Vapour*		*Latent heat* kJ/kg
	Enthalpy kJ/kg	*Entropy* kJ/kg K	*Enthalpy* kJ/kg	*Entropy* kJ/kg K	
25	164.77	0.5978	282.23	0.9918	117.46
–5	72.57	0.2862	321.33	1.2146	248.76

Solution. Given : $T_2 = T_3 = 25°C = 25 + 273 = 298$ K ; $T_1 = T_4 = -5°C = -5 + 273 = 268$ K ; $x_1 = 0.6$; $h_{f3} = h_{f2} = 164.77$ kJ/kg ; $h_{f1} = h_{f4} = 72.57$ kJ/kg ; $s_{f2} = 0.5978$ kJ/kg K; $s_{f1} = 0.2862$ kJ/kg K ; $h_{2'} = 282.23$ kJ/kg ; $h_{1'} = 321.33$ kJ/kg ; $*s_{2'} = 0.9918$ kJ/kg K ; $*s_{1'} = 1.2146$ kJ/kg K ; $h_{fg2} = 117.46$ kJ/kg ; $h_{fg1} = 248.76$ kJ/kg

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

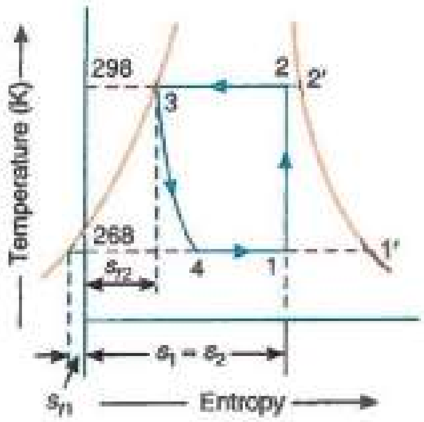
First of all, let us find the dryness fraction at point 2, *i.e.* x_2. We know that the entropy at point 1,

$$s_1 = s_{f1} + \frac{x_1 h_{fg1}}{T_1} = 0.2862 + \frac{0.6 \times 248.76}{268} = 0.8431 \quad \ldots (i)$$

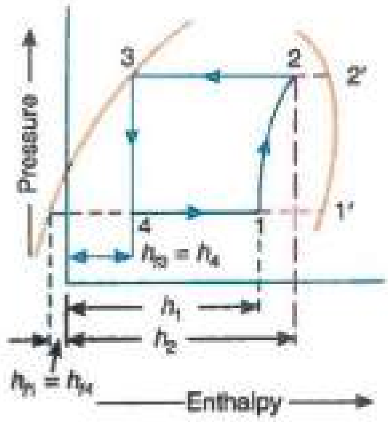
Similarly, entropy at point 2,

$$s_2 = s_{f2} + \frac{x_2 h_{fg2}}{T_2} = 0.5978 + \frac{x_2 \times 117.46}{298}$$

$$= 0.5978 + 0.3941\, x_2 \quad \ldots (ii)$$



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



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

Since the entropy at point 1 (s_1) is equal to entropy at point 2 (s_2), therefore equating equations (*i*) and (*ii*),

$$0.8431 = 0.5978 + 0.3941\, x_2 \quad \text{or} \quad x_2 = 0.622$$

We know that enthalpy at point 1,

$$h_1 = h_{f1} + x_1 h_{fg1} = 72.57 + 0.6 \times 248.76 = 221.83 \text{ kJ/kg}$$

and enthalpy at point 2,

$$h_2 = h_{f2} + x_2 h_{fg2} = 164.77 + 0.622 \times 117.46 = 237.83 \text{ kJ/kg}$$

$$\text{Theoretical C.O.P.} = \frac{h_1 - h_{f3}}{h_2 - h_1} = \frac{221.83 - 164.77}{237.83 - 221.83} = \frac{57.06}{16} = 3.57 \text{ Ans.}$$

Problem 4.6 *An ammonia refrigerating machine fitted with an expansion valve works between the temperature limits of – 10°C and 30°C. The vapour is 95% dry at the end of isentropic compression and the fluid leaving the condenser is at 30°C. Assuming actual C.O.P. as 60% of the theoretical, calculate the kilograms of ice produced per kW hour at 0°C from water at 10°C. Latent heat of ice is 335 kJ/kg. Ammonia has the following properties :*

Temperature °C	*Liquid heat (h_f) kJ/kg*	*Latent heat (h_{fg}) kJ/kg*	*Liquid entropy (s_f)*	*Total entropy of dry saturated vapour*
30	*323.08*	*1145.80*	*1.2037*	*4.9842*
–10	*135.37*	*1297.68*	*0.5443*	*5.4770*

Solution. Given : $T_1 = T_4 = -10°C = -10 + 273 = 263$ K ; $T_2 = T_3 = 30°C = 30 + 273 = 303$ K ; $x_2 = 0.95$; $h_{f3} = h_{f2} = 323.08$ kJ/kg ; $h_{f1} = h_{f4} = 135.37$ kJ/kg ; $h_{fg2} = 1145.8$ kJ/kg ; $h_{fg1} = 1297.68$ kJ/kg , $s_{f2} = 1.2037$; $s_{f1} = 0.5443$; $s_{2'} = 4.9842$; $s_{1'} = 5.4770$

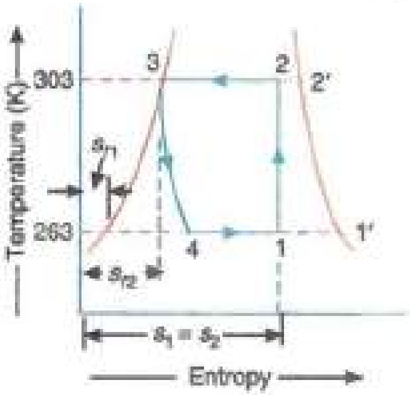
The *T-s* and *p-h* diagrams are shown in Fig. (*a*) and (*b*) respectively.

Let x_1 = Dryness fraction at point 1.

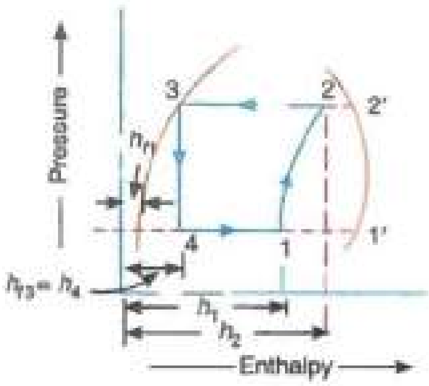
We know that entropy at point 1,

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

$$= 0.5443 + 4.934\, x_1 \qquad \ldots (i)$$



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



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

Similarly, entropy at point 2,

$$s_2 = s_{f2} + \frac{x_2 h_{fg2}}{T_2} = 1.2037 + \frac{0.95 \times 1145.8}{303} = 4.796 \qquad \ldots (ii)$$

Since the entropy at point 1 (s_1) is equal to entropy at point 2 (s_2), therefore equating equations (*i*) and (*ii*),

$$0.5443 + 4.934\, x_1 = 4.796 \qquad \text{or} \qquad x_1 = 0.86$$

$\therefore$ Enthalpy at point 1, $h_1 = h_{f1} + x_1 h_{fg1} = 135.37 + 0.86 \times 1297.68 = 1251.4$ kJ/kg

and enthalpy at point 2, $h_2 = h_{f2} + x_2 h_{fg2} = 323.08 + 0.95 \times 1145.8 = 1411.6$ kJ/kg

We know that theoretical C.O.P.

$$= \frac{h_1 - h_{f3}}{h_2 - h_1} = \frac{1251.4 - 323.08}{1411.6 - 1251.4} = 5.8$$

$\therefore$ Actual C.O.P. $= 0.6 \times 5.8 = 3.48$

Work to be spent corresponding to 1 kW hour,

$$W = 3600 \text{ kJ}$$

$\therefore$ Actual heat extracted or refrigeration effect produced per kW hour

$$= W \times \text{Actual C.O.P.} = 3600 \times 3.48 = 12\ 528 \text{ kJ}$$

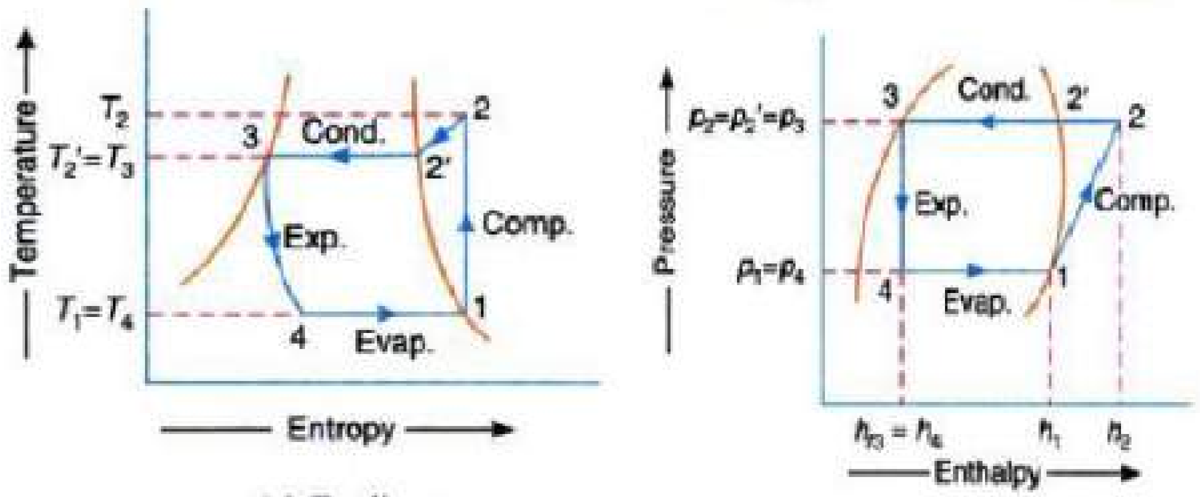
We know that heat extracted from 1 kg of water at 10°C for the formation of 1 kg of ice at 0°C

$$= 1 \times 4.187 \times 10 + 335 = 376.87 \text{ kJ}$$

$\therefore$ Amount of ice produced

$$= \frac{12528}{376.87} = 33.2 \text{ kg / kW hour } \textbf{Ans.}$$

3. Cycle with superheated vapour after compression:



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

Theoretical vapour compression cycle with superheated vapour after compression.

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

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

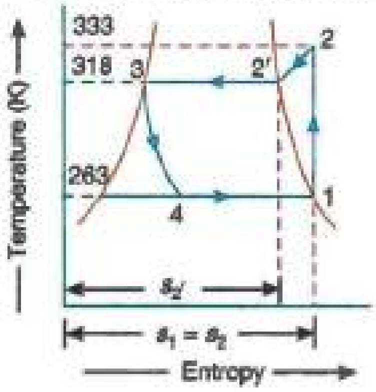
$$\text{C.O.P.} = \frac{\text{Refrigerating effect}}{\text{Work done}} = \frac{h_1 - h_{f3}}{h_2 - h_1}$$

Problem 4.7 *A vapour compression refrigerator uses methyl chloride (R-40) and operates between temperature limits of − 10°C and 45°C. At entry to the compressor, the refrigerant is dry saturated and after compression it acquires a temperature of 60°C. Find the C.O.P. of the refrigerator. The relevant properties of methyl chloride are as follows :*

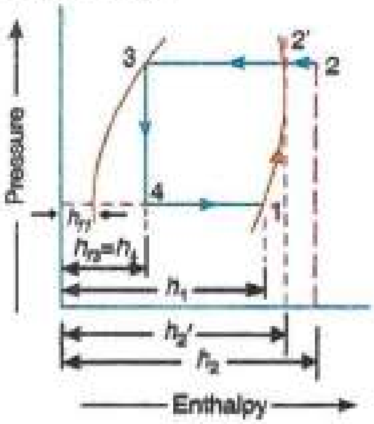
Saturation temperature in °C	*Enthalpy in kJ/kg*		*Entropy in kJ/kg K*	
	Liquid	*Vapour*	*Liquid*	*Vapour*
−10	*45.4*	*460.7*	*0.183*	*1.637*
45	*133.0*	*483.6*	*0.485*	*1.587*

Solution. Given : $T_1 = T_4 = -10°C = -10 + 273 = 263$ K ; $T_{2'} = T_3 = 45°C = 45 + 273 = 318$ K ; $T_2 = 60°C = 60 + 273 = 333$ K ; $*h_{f1} = 45.4$ kJ/kg ; $h_{f3} = 133$ kJ/kg ; $h_1 = 460.7$ kJ/kg ; $h_{2'} = 483.6$ kJ/kg ; $*s_{f1} = 0.183$ kJ/kg K ; $*s_{f3} = 0.485$ kJ/kg K ; $s_1 = s_2 = 1.637$ kJ/kg K; $s_{2'} = 1.587$ kJ/kg K

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



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



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

Let c_p = Specific heat at constant pressure for superheated vapour.

We know that entropy at point 2,

$$s_2 = s_{2'} + 2.3\, c_p \log\left(\frac{T_2}{T_{2'}}\right)$$

$$1.637 = 1.587 + 2.3\, c_p \log\left(\frac{333}{318}\right)$$

$$= 1.587 + 2.3\, c_p \times 0.02 = 1.587 + 0.046\, c_p$$

$$\therefore \quad c_p = 1.09$$

and enthalpy at point 2, $h_2 = h_{2'} + c_p \times$ Degree of superheat $= h_{2'} + c_p (T_2 - T_{2'})$
$= 483.6 + 1.09 (333 - 318) = 500$ kJ/kg

∴ C.O.P. of the refrigerator

$$= \frac{h_1 - h_{f3}}{h_2 - h_1} = \frac{460.7 - 133}{500 - 460.7} = 8.34 \text{ Ans.}$$

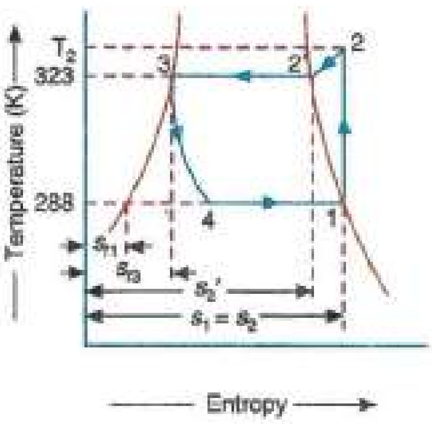
Problem 2.8 *A simple refrigerant 134a (tetrafluroethane) heat pump for space heating, operates between temperature limits of 15°C and 50°C. The heat required to be pumped is 100 MJ/h. Determine : 1. The dryness fraction of refrigerant entering the evaporator; 2. The discharge temperature assuming the specific heat of vapour as 0.996 kJ/kg K; 3. The theoretical piston displacement of the compressor; 4. The theoretical power of the compressor; and 5. The C.O.P.*

The specific volume of refrigerant 134a saturated vapour at 15°C is 0.04185 m³/kg. The other relevant properties of R-134a are given below:

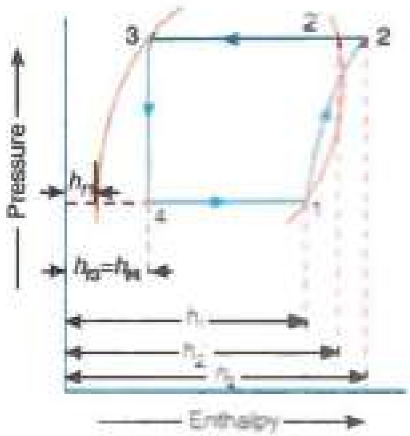
Saturation temperature (°C)	Pressure (bar)	Specific enthalpy (kJ/kg)		Specific entropy (kJ/kg K)	
		Liquid	Vapour	Liquid	Vapour
15	4.887	220.26	413.6	1.0729	1.7439
50	13.18	271.97	430.4	1.2410	1.7312

Solution: Given: $T_1 = T_4 = 15°C = 15 + 273 = 288$ K ; $T_{2'} = T_3 = 50°C = 50 + 273 = 323$ K ; $Q = 100$ MJ/h $= 100 \times 10^3$ kJ/h ; $c_p = 0.996$ kJ/kg K ; $v_1 = 0.04185$ m³/kg ; $h_{f1} = 220.26$ kJ/kg ; $h_{f3} = h_4 = 271.97$ kJ/kg ; $h_1 = 413.6$ kJ/kg ; $h_{2'} = 430.4$ kJ/kg ; $s_{f1} = 1.0729$ kJ/kg K ; $s_1 = s_2 = 1.7439$ kJ/kg K ; $s_{f3} = 1.2410$ kJ/kg K ; $s_{2'} = 1.7312$ kJ/kg K

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



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



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

1. Dryness fraction of refrigerant entering the evaporator

We know that dryness fraction of refrigerant entering the evaporator *i.e.* at point 4.

$$x_4 = \frac{h_4 - h_{f1}}{h_1 - h_{f1}} = \frac{271.97 - 220.26}{413.6 - 220.26} = \frac{51.71}{193.34} = 0.2675 \text{ Ans.}$$

2. Discharge temperature

Let T_2 = Discharge temperature.

We know that entropy at discharge *i.e.* at point 2,

$$s_2 = s_{2'} + 2.3\, c_p \log\left(\frac{T_2}{T_{2'}}\right)$$

$$1.7439 = 1.7312 + 2.3 \times 0.996 \log\left(\frac{T_2}{T_{2'}}\right)$$

$$\log\left(\frac{T_2}{T_{2'}}\right) = \frac{1.7439 - 1.7312}{2.3 \times 0.996} = 0.005\,54$$

$$\frac{T_2}{T_{2'}} = 1.0128 \qquad \text{...(Taking antilog of 0.005 54)}$$

$$T_2 = T_{2'} \times 1.0128 = 323 \times 1.0128 = 327.13 \text{ K} = 54.13°\text{C} \text{ Ans.}$$

3. Theoretical piston displacement of the compressor

We know that enthalpy at discharge *i.e.* at point 2,

$$h_2 = h_{2'} + c_p (T_2 - T_{2'})$$
$$= 430.4 + 0.996\,(327.13 - 323) = 434.5 \text{ kJ/kg}$$

and mass flow rate of the refrigerant,

$$m_R = \frac{Q}{h_2 - h_{f_3}} = \frac{100 \times 10^3}{434.5 - 271.97} = 615.3 \text{ kg/h} = 10.254 \text{ kg/min}$$

∴ Theoretical piston displacement of the compressor

$$= m_R \times v_1 = 10.254 \times 0.4185 = 4.29 \text{ m}^3\text{/min Ans.}$$

4. Theoretical power of the compressor

We know that workdone by the compressor

$$= m_R (h_2 - h_1) = 10.254\,(434.5 - 413.6) = 214.3 \text{ kJ/min}$$

∴ Power of the compressor = 214.3/60 = 3.57 kJ/s or kW **Ans.**

5. C.O.P.

We know that $$\text{C.O.P.} = \frac{h_1 - h_{f3}}{h_2 - h_1} = \frac{413.6 - 271.97}{434.5 - 413.6} = \frac{141.63}{20.9} = 6.8 \text{ Ans.}$$

Problem 4.9 *A refrigeration machine using R-12 as refrigerant operates between the pressures 2.5 bar and 9 bar. The compression is isentropic and there is no undercooling in the condenser.*

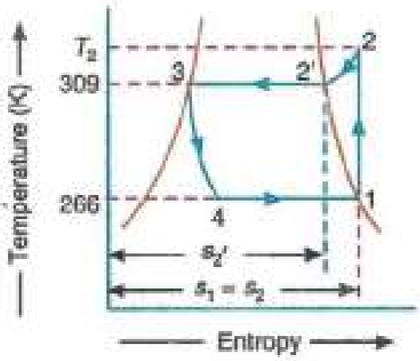
The vapour is in dry saturated condition at the beginning of the compression. Estimate the theoretical coefficient of performance. If the actual coefficient of performance is 0.65 of theoretical value, calculate the net cooling produced per hour. The refrigerant flow is 5 kg per minute. Properties of refrigerant are :

Pressure, bar	*Saturation temperature, °C*	*Enthalpy, kJ/kg*		*Entropy of saturated vapour, kJ/kg K*
		Liquid	*Vapour*	
9.0	36	70.55	201.8	0.6836
2.5	–7	29.62	184.5	0.7001

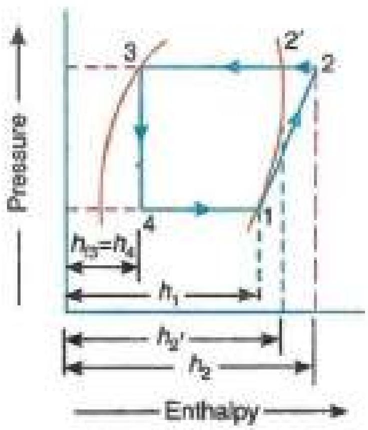
Take c_p for superheated vapour at 9 bar as 0.64 kJ/kg K.

Solution. Given : $T_{2'} = T_3 = 36°C = 36 + 273 = 309$ K ; $T_1 = T_4 = -7°C = -7 + 273 = 266$ K ; $(C.O.P.)_{actual} = 0.65\ (C.O.P.)_{th}$; $m = 5$ kg /min ; $h_{f3} = h_4 = 70.55$ kJ / kg ; $*h_{f1} = h_{f4} = 29.62$ kJ/kg ; $h_2' = 201.8$ kJ / kg ; $h_1 = 184.5$ kJ/kg ; $s_2' = 0.6836$ kJ/kg K ; $s_1 = s_2 = 0.7001$ kJ/kg K ; $c_p = 0.64$ kJ / kg K

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



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



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

Theoretical coefficient of performance

First of all, let us find the temperature at point 2 (T_2).

We know that entropy at point 2,

$$s_2 = s_2' + 2.3\ c_p \log\left(\frac{T_2}{T_{2'}}\right)$$

$$0.7001 = 0.6836 + 2.3 \times 0.64 \log\left(\frac{T_2}{309}\right)$$

$$\log\left(\frac{T_2}{309}\right) = \frac{0.7001 - 0.6836}{2.3 \times 0.64} = 0.0112$$

$$\frac{T_2}{309} = 1.026 \quad \text{... (Taking antilog of 0.0112)}$$

$$\therefore \quad T_2 = 1.026 \times 309 = 317 \text{ K}$$

We know that enthalpy of superheated vapour at point 2,

$$h_2 = h_{2'} + c_p (T_2 - T_{2'})$$
$$= 201.8 + 0.64\,(317 - 309) = 206.92 \text{ kJ/kg}$$

∴ Theoretical coefficient of performance,

$$(\text{C.O.P.})_{th} = \frac{h_1 - h_{f3}}{h_2 - h_1} = \frac{184.5 - 70.55}{206.92 - 184.5} = 5.1 \quad \textbf{Ans.}$$

Net cooling produced per hour

We also know that actual C.O.P. of the machine,

$$(\text{C.O.P.})_{actual} = 0.65 \times (\text{C.O.P.})_{th} = 0.65 \times 5.1 = 3.315$$

and actual work done, $\quad w_{actual} = h_2 - h_1 = 206.92 - 184.5 = 22.42$ kJ/kg

We know that net cooling (or refrigerating effect) produced per kg of refrigerant

$$= w_{actual} \times (\text{C.O.P.})_{actual} = 22.42 \times 3.315 = 74.3 \text{ kJ/kg}$$

∴ Net cooling produced per hour

$$= m \times 74.3 = 5 \times 74.3 = 371.5 \text{ kJ/min}$$

$$= \frac{371.5}{210} = 1.77 \text{ TR} \quad \textbf{Ans.} \qquad \text{... } (\because 1 \text{ TR} = 210 \text{ kJ/min})$$

Problem 2.10 *A simple saturation cycle using R-12 is designed for taking a load of 10 tonnes. The refrigerator and ambient temperature are –0°C and 30°C respectively. A minimum temperature difference of 5°C is required in the evaporator and condenser for heat transfer. Find : 1. Mass flow rate through the system ; 2. Power required in kW ; 3. C.O.P. ; and 4. Cylinder dimensions assuming L / D = 1.2, for a single cylinder, single acting compressor if it runs at 300 r.p.m. with volumetric efficiency of 90%.*

Solution. Given : Q = 10TR = 10 × 210 = 2100 kJ/min

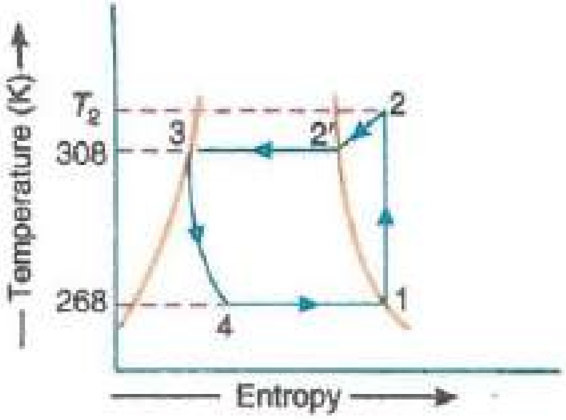
Since a minimum temperature difference of 5°C is required in the evaporator and condenser, therefore evaporator temperature would be

$$T_1 = T_4 = 0 - 5 = -5°\text{C} = -5 + 273 = 268 \text{ K}$$

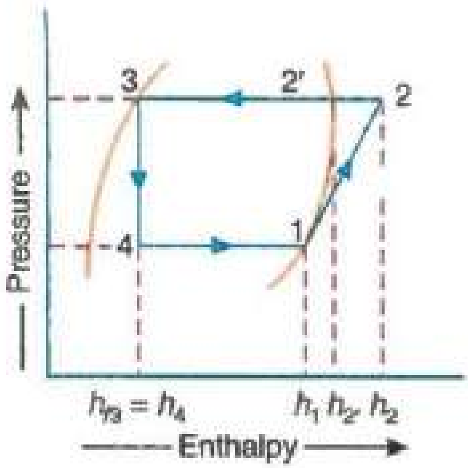
and condenser temperature,

$$T_{2'} = T_3 = 30 + 5 = 35°\text{C} = 35 + 273 = 308 \text{ K}$$

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



(a) T-s diagram.



(b) p-h diagram.

From p-h diagram, we find that enthalpy of dry saturated vapour at –5°C (268 K) *i.e.* at point 1,

$$h_1 = 185 \text{ kJ/kg}$$

Enthalpy of superheated vapour at point 2,

$$h_2 = 206 \text{ kJ/kg}$$

Enthalpy of saturated liquid at 35°C (308 K) *i.e.* at point 3,

$$h_{f3} = h_4 = 70 \text{ kJ/kg}$$

and specific volume of dry saturated vapour at –5°C (268 K) *i.e.* at point 1,

$$v_1 = 0.065 \text{ m}^3\text{/kg}$$

1. Mass flow rate through the system

We know that refrigerating effect per kg of the refrigerant

$$= h_1 - h_{f3} = 185 - 70 = 115 \text{ kJ/kg}$$

$\therefore$ Mass flow rate, $m_R = \dfrac{\text{Refrigerating capacity}}{\text{Refrigerating effect}} = \dfrac{10 \times 210}{115} = 18.26 \text{ kg / min}$ **Ans.**

2. Power required

We know that workdone during compression of the refrigerant

$$= m_R (h_2 - h_1) = 18.26 (206 - 185) = 383.46 \text{ kJ/min}$$

$\therefore$ Power required $= 383.46/60 = 6.4$ kJ/s or kW **Ans.**

3. C.O.P.

We know that C.O.P. $= \dfrac{h_1 - h_{f3}}{h_2 - h_1} = \dfrac{185-70}{206-185} = \dfrac{115}{21} = 5.476$ **Ans.**

4. Cylinder dimensions

Let D = Bore of cylinder,

L = Length of stroke = 1.2 D ...(Given)

N = Speed of compressor = 300 r.p.m. ...(Given)

η_v = Volumetric efficiency = 90% = 0.9 ...(Given)

We know that theoretical suction volume or piston displacement per minute

$$= m_R \times v_1 \times \frac{1}{\eta_v} = 18.26 \times 0.065 \times \frac{1}{0.9} = 1.32 \text{ m}^3/\text{min} \quad ...(i)$$

We also know that suction volume or piston displacement per minute

= Piston area × Stroke × R.P.M.

$$= \frac{\pi}{4} \times D^2 \times L \times N = \frac{\pi}{4} \times D^2 \times 1.2D \times 300 = 282.8D^3 \text{ m}^3/\text{min} \quad ...(ii)$$

Equating equations (*i*) and (*ii*),

D^3 = 1.32 / 282.8 = 4.667 × 10^{-3} or D = 0.167m = 167 mm **Ans.**

and L = 1.2 D = 1.2 × 167 = 200.4 mm **Ans.**

Problem 2.11 *A water cooler using R-12 works on the condensing and evaporating temperatures of 26°C and 2°C respectively. The vapour leaves the evaporator saturated and dry. The average output of cold water is 100 kg / h cooled from 26°C to 6°C.*

Allowing 20% of useful heat into water cooler and the volumetric efficiency of the compressor as 80% and mechanical efficiency of the compressor and the electric motor as 85% and 95% respectively, find :1. volumetric displacement of the compressor ; and 2. power of the motor. Data for R-12 is given below :

Temperature °C	*Pressure bar*	*Enthalpy, kJ/kg*		*Entropy, kJ/kg K*		*Specific heat kJ/kg K*		*Specific volume of vapour m³/kg*
		Liquid	*Vapour*	*Liquid*	*Vapour*	*Liquid*	*Vapour*	
26	*6.69*	*60.64*	*198.10*	*0.2270*	*0.6865*	*0.996*	*0.674*	*0.026*
2	*3.297*	*37.92*	*188.39*	*0.1487*	*0.6956*	*1.067*	*0.620*	*0.052*

Solution. Given : $T_{2'} = T_3 = 26°C = 26 + 273 = 299$ K ; $T_1 = T_4 = 2°C = 2 + 273 = 275$ K ; $m_w = 100$ kg/h ; $T_{w1} = 26°C = 26 + 273 = 299$ K ; $T_{w2} = 6°C = 6 + 273 = 279$ K ; $\eta_v = 80\% = 0.80$; $\eta_{m1} = 85\% = 0.85$; $\eta_{m2} = 95\% = 0.95$; $h_{f3} = 60.64$ kJ/kg ; $^*h_{f1} = 37.92$ kJ/kg ; $h_{2'} = 198.10$ kJ/kg ; $h_1 = 188.39$ kJ/kg ; $^*s_{f3} = 0.2270$ kJ/kg K ; $^*s_{f1} = 0.1487$ kJ/kg K ; $^*s_{2'} = 0.6865$ kJ/kg K ; $s_1 = s_2 = 0.6956$ kJ/kg K ; $^*c_{p3} = 0.996$ kJ/kg K ; $^*c_{p4} = 1.067$ kJ/kg K; $^*c_{p2'} = 0.674$ kJ/kg K ; $c_{p1} = 0.620$ kJ/kg K ; $^*v_{2'} = 0.026$ m³ / kg ; $v_1 = 0.052$ m³ / kg

1. *Volumetric displacement of the compressor*

The *T-s* and *p-h* diagrams are shown in Fig. 4.15 (*a*) and (*b*) respectively. Since 20% of the useful heat is lost into water cooler, therefore actual heat extracted from the water cooler,

$$h_R = 1.2\, m_w \times c_w (T_{w1} - T_{w2})$$
$$= 1.2 \times 100 \times 4.187\,(299 - 279) = 10\,050 \text{ kJ/h} = 167.5 \text{ kJ/min}$$

... ($\because$ Sp. heat of water, $c_w = 4.187$ kJ/kg K)

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

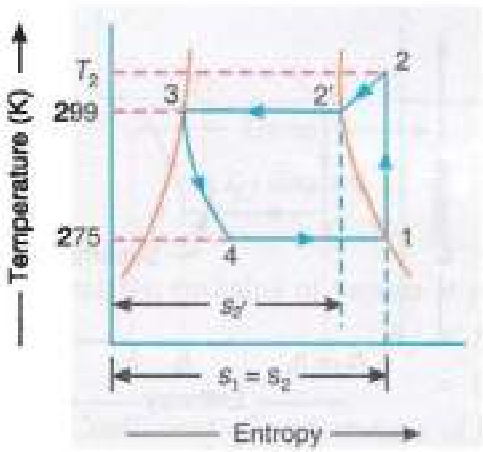
$$= h_1 - h_{f3} = 188.39 - 60.64 = 127.75 \text{ kJ/kg}$$

$\therefore$ Mass flow of the refrigerant,

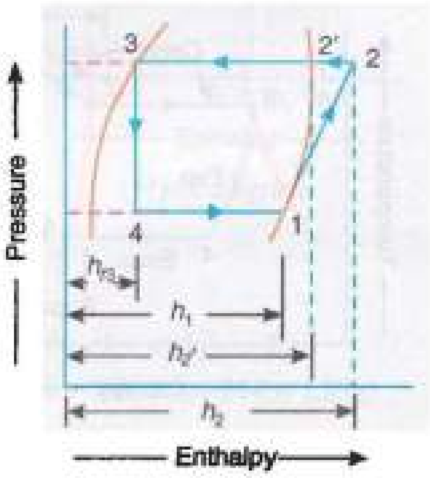
$$m_R = \frac{167.5}{127.75} = 1.3 \text{ kg/min}$$

and volumetric displacement of the compressor

$$= \frac{m_R \times v_1}{\eta_v} = \frac{1.3 \times 0.052}{0.80} = 0.085 \text{ m}^3\text{/min } \textbf{Ans.}$$



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



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

2. Power of the motor

First of all, let us find the temperature at point 2 (T_2). We know that entropy at point 2,

$$s_2 = s_2' + 2.3\, c_{p2}' \log\left(\frac{T_2}{T_2'}\right)$$

$$0.6956 = 0.6865 + 2.3 \times 0.674 \log\left(\frac{T_2}{299}\right)$$

$$\log\left(\frac{T_2}{299}\right) = \frac{0.6956 - 0.6865}{2.3 \times 0.674} = 0.005\ 87$$

or $$\frac{T_2}{299} = 1.0136 \qquad \text{... (Taking anti-log of 0.005 87)}$$

$\therefore$ $$T_2 = 299 \times 1.0136 = 303 \text{ K}$$

We know that enthalpy at point 2,

$$h_2 = h_2' + c_{p2}' (T_2 - T_2')$$

$$= 198.10 + 0.674\,(303 - 299) = 200.8 \text{ kJ/kg}$$

We also know that work done by the compressor per kg of the refrigerant

$$= h_2 - h_1 = 200.8 - 188.39 = 12.41 \text{ kJ/kg}$$

and work done per minute $$= m_R \times 12.41 = 1.3 \times 12.41 = 16.133 \text{ kJ/min}$$

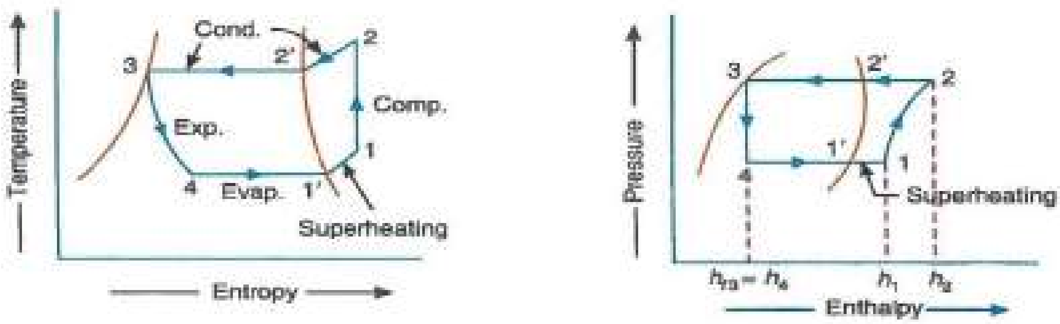
$$= 0.27 \text{ kJ/s} = 0.27 \text{ kW}$$

$\therefore$ Power required for the compressor

$$= \frac{0.27}{\eta_{m1}} = \frac{0.27}{0.85} = 0.317 \text{ kW}$$

and power of the motor $$= \frac{0.317}{\eta_{m2}} = \frac{0.317}{0.95} = 0.334 \text{ kW}$$ **Ans.**

4. Cycle with superheated vapour before compression:



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

Theoretical vapour compression cycle with superheated vapour before compression.

A vapour compression cycle with superheated vapour before compression is shown on T-s and p-h diagrams in Fig. (a) and (b) respectively. In this cycle, the evaporation starts at point 4 and continues upto point 1′, when it is dry saturated. The vapour is now superheated before entering the compressor upto the point 1.

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

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

Problem 2.11 *A vapour compression refrigeration plant works between pressure limits of 5.3 bar and 2.1 bar. The vapour is superheated at the end of compression, its temperature being 37°C. The vapour is superheated by 5°C before entering the compressor.*

If the specific heat of superheated vapour is 0.63 kJ/kg K, find the coefficient of performance of the plant. Use the data given below :

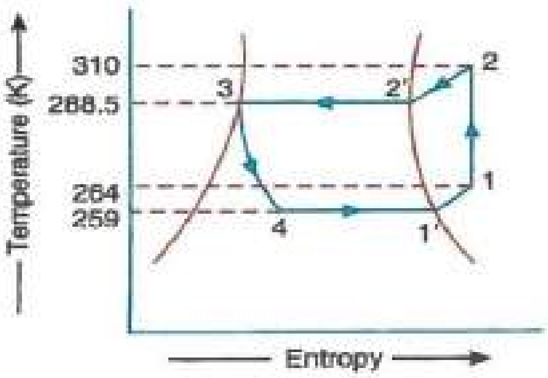
Pressure, bar	*Saturation temperature, °C*	*Liquid heat, kJ/kg*	*Latent heat, kJ/kg*
5.3	*15.5*	*56.15*	*144.9*
2.1	*–14.0*	*25.12*	*158.7*

Solution. Given : $p_2 = 5.3$ bar ; $p_1 = 2.1$ bar ; $T_2 = 37°\text{C} = 37 + 273 = 310$ K ; $T_1 - T_1' = 5°\text{C}$; $c_p = 0.63$ kJ/kg K ; $T_2' = 15.5°\text{C} = 15.5 + 273 = 288.5$ K ; $T_1' = -14°\text{C} = -14 + 273 = 259$ K ; $h_{f3} = h_{f2}' = 56.15$ kJ/kg ; $h_{f1}' = 25.12$ kJ/kg ; $h_{fg2}' = 144.9$ kJ/kg ; $h_{fg1}' = 158.7$ kJ/kg

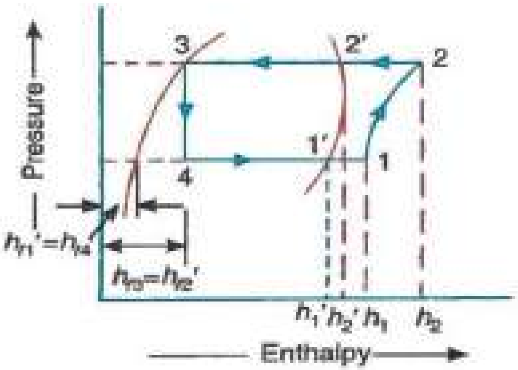
The T-s and p-h diagrams are shown in Fig. (a) and (b) respectively.

We know that enthalpy of vapour at point 1,

$$h_1 = h_1' + c_p (T_1 - T_1') = (h_{f1}' + h_{fg1}') + c_p (T_1 - T_1')$$
$$= (25.12 + 158.7) + 0.63 \times 5 = 186.97 \text{ kJ/kg}$$



(a) T-s diagram.



(b) p-h diagram.

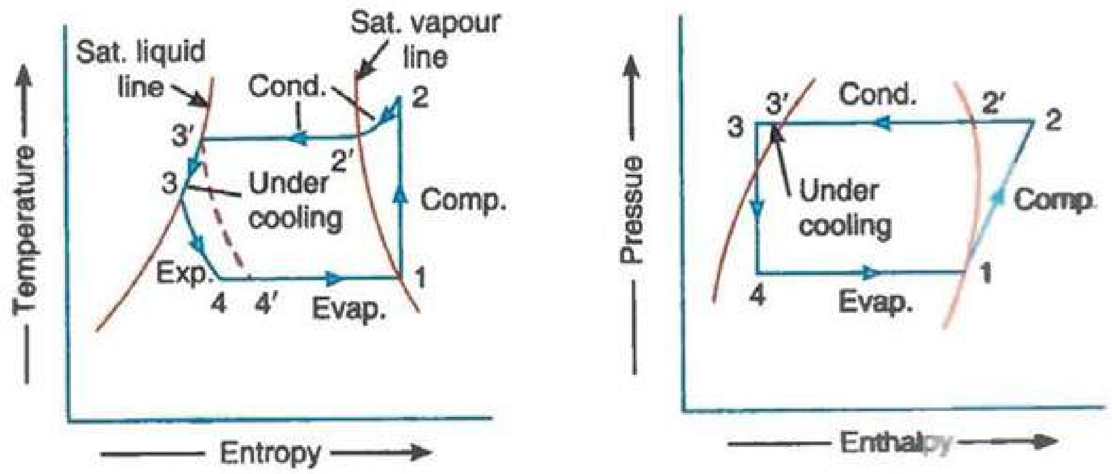
Similarly, enthalpy of vapour at point 2,

$$h_2 = h_{2'} + c_p (T_2 - T_{2'}) = (h_{f2'} + h_{fg2'}) + c_p (T_2 - T_{2'})$$
$$= (56.15 + 144.9) + 0.63 (310 - 288.5) = 214.6 \text{ kJ/kg}$$

∴ Coefficient of performance of the plant,

$$\text{C.O.P.} = \frac{h_1 - h_{f3}}{h_2 - h_1} = \frac{186.97 - 56.15}{214.6 - 186.97} = \frac{130.82}{27.63} = 4.735 \textbf{ Ans.}$$

5. *Cycle with under cooling or sub-cooling of refrigerant:*



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

Theoretical vapour compression cycle with undercooling or subcooling of the refrigerant

Sometimes, the refrigerant, after condensation process 2'-3', is cooled below the saturation temperature ($T_{3'}$) before expansion by throttling. Such a process is called ***undercooling*** or ***subcooling*** of the refrigerant and is generally done along the liquid line as shown in Fig. (*a*) and (*b*). The ultimate effect of the undercooling is to increase the value of coefficient of performance under the same set of conditions.

The process of undercooling is generallly brought about by circulating more quantity of cooling water through the condenser or by using water colder than the main circulating water. Sometimes, this process is also brought about by employing a heat exchanger. In actual practice, the refrigerant is superheated after compression and undercooled before throttling, as shown in Fig. (*a*) and (*b*) . A little consideration will show, that the refrigerating effect is increased by adopting both the superheating and undercooling process as compared to a cycle without them, which is shown by dotted lines in Fig. (*a*).

In this case, the refrigerating effect or heat absorbed or extracted,

$$R_E = h_1 - h_4 = h_1 - h_{f3}$$

and work done,

$$w = h_2 - h_1$$

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

Note: The value of h_{f3} may be found out from the relation,

$$h_{f3} = h_{f3'} - c_p \times \text{Degree of undercooling}$$

Problem 2.12 *A vapour compression refrigerator uses R-12 as refrigerant and the liquid evaporates in the evaporator at – 15°C. The temperature of this refrigerant at the delivery from the compressor is 15°C when the vapour is condensed at 10°C. Find the coefficiet of performance if (i) there is no undercooling , and (ii) the liquid is cooled by 5°C before expansion by throttling.*

Take specific heat at constant pressure for the superheated vapour as 0.64 kJ/kg K and that for liquid as 0.94 kJ/kg K. The other properties of refrigerant are as follows :

Temperature in °C	*Enthalpy in kJ/ kg*		*Specific entropy in kJ / kg K*	
	Liquid	*Vapour*	*Liquid*	*Vapour*
–15	*22.3*	*180.88*	*0.0904*	*0.7051*
+10	*45.4*	*191.76*	*0.1750*	*0.6921*

Solution. Given : $T_1 = T_4 = -15°C = -15 + 273 = 258$ K ; $T_2 = 15°C = 15 + 273 = 288$K ; $T_{2''} = 10°C = 10 + 273 = 283$ K ; $c_{pv} = 0.64$ kJ/kg K ; $c_{pl} = 0.94$ kJ/kg K ; $h_{f1} = 22.3$ kJ/kg; $h_{f3'} = 45.4$ kJ/kg ; $h_{1'} = 180.88$ kJ/kg ; $h_{2'} = 191.76$ kJ/kg ; $s_{f1} = 0.0904$ kJ/kg K ; $s_{f3} = 0.1750$ kJ/kg K ; $s_{g1} = 0.7051$ kJ/kg K ; $s_{2'} = 0.6921$ kJ/kg K

(i) Coefficient of performance if there is no undercooling

The *T-s* and *p-h* diagrams, when there is no undercooling, are shown in Fig. (a) and (b) respectively.

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

We know that entropy at point 1,

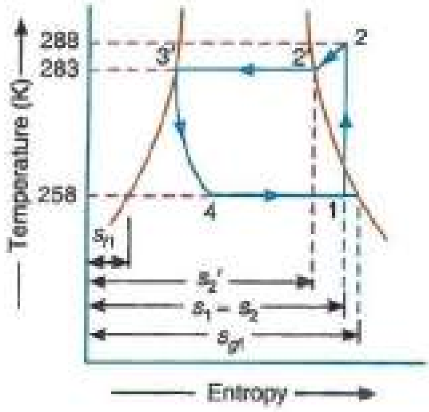
$$s_1 = s_{f1} + x_1 s_{fg1} = s_{f1} + x_1 (s_{g1} - s_{f1})$$
$$= 0.0904 + x_1 (0.7051 - 0.0904) = 0.0904 + 0.6147\, x_1 \quad \text{------}(i)$$

and entropy at point 2,

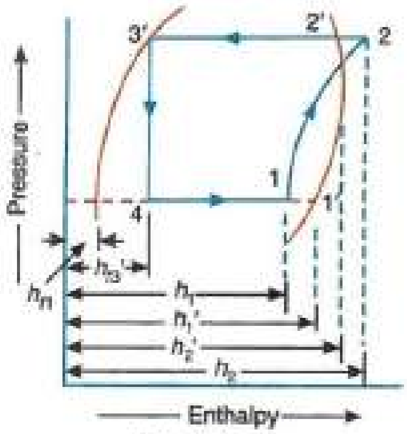
$$s_2 = s_{2'} + 2.3\, c_{pv} \log\left(\frac{T_2}{T_{2'}}\right)$$
$$= 0.6921 + 2.3 \times 0.64 \log\left(\frac{288}{283}\right)$$
$$= 0.6921 + 2.3 \times 0.64 \times 0.0077 = 0.7034 \quad \text{... }(ii)$$

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

$$0.0904 + 0.6147\, x_1 = 0.7034 \quad \text{or} \quad x_1 = 0.997$$



(a) T-s diagram.



(b) p-h diagram.

We know that the enthalpy at point 1,

$$h_1 = h_{f1} + x_1\, h_{fg1} = h_{f1} + x_1\,(h_{g1} - h_{f1})$$
$$= 22.3 + 0.997\,(180.88 - 22.3) = 180.4 \text{ kJ/kg}$$
$$\dots (\because h_{g1} = h_1')$$

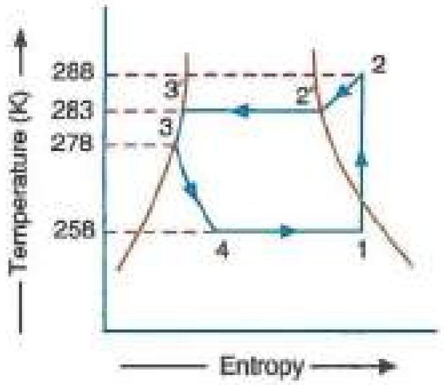
and enthalpy at point 2,

$$h_2 = h_{2'} + c_{pv}\,(T_2 - T_{2'})$$
$$= 191.76 + 0.64\,(288 - 283) = 194.96 \text{ kJ/kg}$$

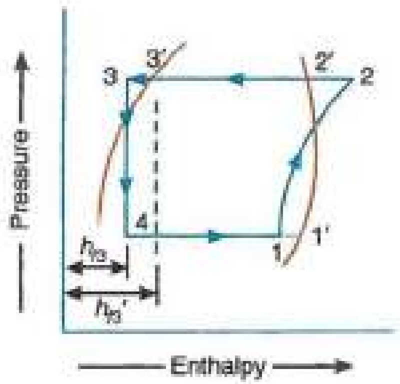
$$\therefore \qquad \text{C.O.P.} = \frac{h_1 - h_{f3'}}{h_2 - h_1} = \frac{180.4 - 45.4}{194.96 - 180.4} = 9.27 \textbf{ Ans.}$$

(ii) Coefficient of performance when there is an undercooling of 5°C

The *T-s* and *p-h* diagrams, when there is an undercooling of 5°C, are shown in Fig. (*a*) and (*b*) respectively.



(a) T-s diagram.



(b) T-s diagram.

We know that enthalpy of liquid refrigerant at point 3,

$$h_{f3} = h_{f3'} - c_{pl} \times \text{Degree of undercooling}$$
$$= 45.4 - 0.94 \times 5 = 40.7 \text{ kJ/kg}$$

$$\therefore \quad \text{C.O.P.} = \frac{h_1 - h_{f3}}{h_2 - h_1} = \frac{180.4 - 40.7}{194.96 - 180.4} = 9.59 \text{ Ans.}$$

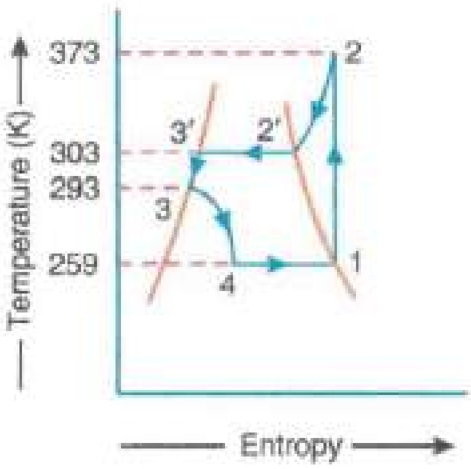
Problem: 2.13 *A simple NH_3 vapour compression system has compressor with piston displacement of 2 m³/min, a condenser pressure of 12 bar and evaporator pressure of 2.5 bar. The liquid is sub-cooled to 20°C by soldering the liquid line to suction line. The temperature of vapour leaving the compressor is 100°C, heat rejected to compressor cooling water is 5000 kJ/hour, and volumetric efficiency of compressor is 0.8.*

Compute : Capacity ; Indicated power ; and C.O.P. of the system.

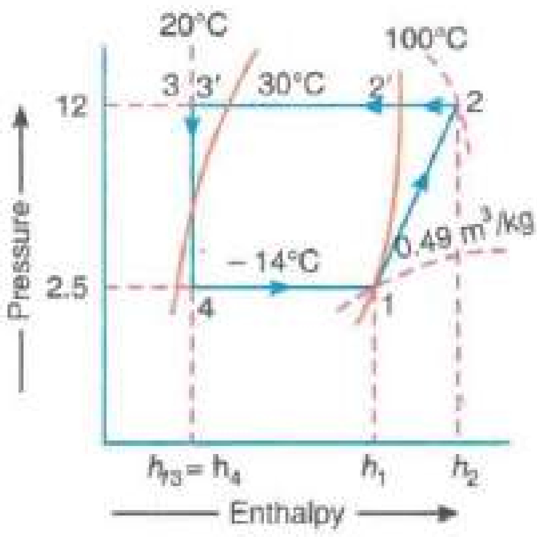
Solution. Given : $v_p = 2$ m³/min ; $p_2 = p_{2'} = p_{3'} = p_3 = 12$ bar ; $p_1 = p_4 = 2.5$ bar ; $T_3 = 20°C = 20 + 273 = 293$ K ; $T_2 = 100°C = 100 + 273 = 373$ K ; $\eta_v = 0.8$

Capacity of the system

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



(a) T-s diagram.



(b) p-h diagram.

From p-h diagram, we find that the evaporating temperature corresponding to 2.5 bar is

$$T_1 = T_4 = -14°C = -14 + 273 = 259 \text{ K}$$

Condensing temperature corresponding to 12 bar is

$$T_{2'} = T_{3'} = 30°C = 30 + 273 = 303 \text{ K}$$

Specific volume of dry saturated vapour at 2.5 bar (*i.e.* at point 1),

$$v_1 = 0.49 \text{ m}^3\text{/kg}$$

Enthalpy of dry saturated vapour at point 1,

$$h_1 = 1428 \text{ kJ/kg}$$

Enthalpy of superheated vapour at point 2,

$$h_2 = 1630 \text{ kJ/kg}$$

and enthalpy of sub-cooled liquid at 20°C at point 3,

$$h_{f3} = h_4 = 270 \text{ kJ/kg}$$

Let m_R = Mass flow of the refrigerant in kg/min.

We know that piston displacement,

$$v_p = \frac{m_R \times v_1}{\eta_v} \quad \text{or} \quad m_R = \frac{v_p \times \eta_v}{v_1} = \frac{2 \times 0.8}{0.49} = 3.265 \text{ kg / min}$$

We know that refrigerating effect per kg of refrigerant

$$= h_1 - h_{f3} = 1428 - 270 = 1158 \text{ kJ/kg}$$

and total refrigerating effect

$$= m_R (h_1 - h_{f3}) = 3.265 (1428 - 270) = 3781 \text{ kJ/min}$$

$\therefore$ Capacity of the system $= 3781 / 210 = 18$ TR **Ans.**

Indicated power of the system

We know that work done during compression of the refrigerant

$$= m_R (h_2 - h_1) = 3.265 (1630 - 1428) = 659.53 \text{ kJ/min}$$

Heat rejected to compressor cooling water

$$= 5000 \text{ kJ/h} = 5000/60 = 83.33 \text{ kJ/min} \qquad \text{... (Given)}$$

$\therefore$ Total work done by the system

$$= 659.53 + 83.33 = 742.86 \text{ kJ/min}$$

and indicated power of the system

$$= 742.86/60 = 12.38 \text{ kW}$$ **Ans.**

C.O.P. of the system

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

$$= \frac{\text{Total refrigerating effect}}{\text{Total work done}} = \frac{3781}{742.86} = 5.1$$ **Ans.**

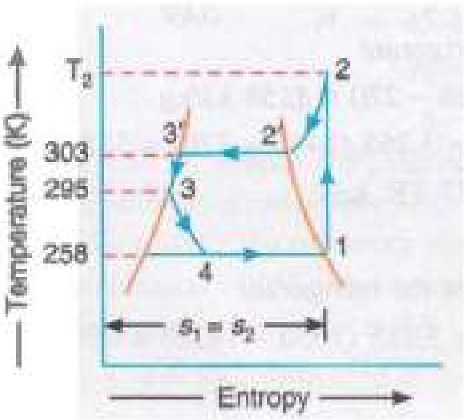
Problem:2.14 *Saturated ammonia at 2.5 bar enters a 160mm × 150mm (bore × stroke) twin cylinder, single acting compressor whose volumetric efficiency is 79% and speed is 250 r.p.m. The head pressure is 12 bar. The subcooled liquid ammonia at 22°C enters the expansion valve. For a standard refrigeration cycle, find: 1. The ammonia circulated in kg / min.; 2. The refrigeration in TR ; and 3. The C.O.P. of the refrigeration cycle. Refer to the following table for the properties of ammonia:*

Pressure (bar)	*Saturation temperature (°C)*	*Specific volume of vapour (m³/kg)*	*Specific enthalpy (kJ/kg)*		*Specific entropy (kJ/kg K)*	
			Liquid	*Vapour*	*Liquid*	*Vapour*
2.5	–15	0.5098	112.4	1426.58	0.4572	5.5497
12	30	0.1107	323.08	1468.87	1.2037	4.9842

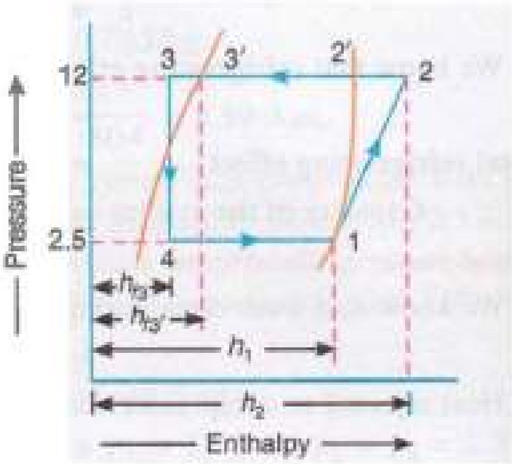
Assume specific heat at constant pressure for liquid ammonia as 4.606 kJ/kg K and for superheated ammonia vapour as 2.763 kJ/kg K.

Solution. Given : $p_1 = p_4 = 2.5$ bar ; $D = 160$ mm $= 0.16$ m ; $L = 150$ mm $= 0.15$ m ; No. of cylinders = 2 ; $\eta_v = 79\% = 0.79$; $N = 250$ r.p.m. ; $p_2 = p_3 = 12$ bar ; $T_3 = 22°C = 22 + 273 = 295$ K ; $T_1 = T_4 = -15°C = -15 + 273 = 258$ K ; $T_{2'} = T_{3'} = 30°C = 30 + 273 = 303$ K ; $v_1 = 0.5098$ m³/kg; $*v_{2'} = 0.1107$ m³/kg $*h_{f1} = 112.4$ kJ/kg ; $h_{f3'} = 323.08$ kJ/kg ; $h_1 = 1426.58$ kJ/kg ; $h_{2'} = 1468.87$ kJ/kg ; $s_{f1} = 0.4572$ kJ/kg K ; $s_{f3} = 1.2037$ kJ/kg K ; $s_1 = s_2 = 5.5497$ kJ/kg K ; $s_{2'} = 4.9842$ kJ/kg K ; $c_{pl} = 4.606$ kJ/kg K ; $c_{pv} = 2.763$ kJ/kg K

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



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



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

1. Ammonia circulated in kg/min

Let m_R = Mass flow rate of ammonia in kg/min.

We know that suction volume or piston displacement per minute

= Piston area × Stroke × R.P.M × No. of cylinders

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

$$= \frac{\pi}{4}(0.16)^2 0.15 \times 250 \times 2 = 1.508 \text{ m}^3\text{/min} \text{ --------}(i)$$

We also know that piston displacement per minute

$$= m_R \times v_1 \times \frac{1}{\eta_v} = m_R \times 0.5098 \times \frac{1}{0.79} = 0.6453\, m_R \text{ ---------}(ii)$$

Equating equations (*i*) and (*ii*),

m_R = 1.508 / 0.6453 = 2.34 kg / min **Ans.**

$$s_2 = s_{2'} + 2.3\, c_{pv} \log\left(\frac{T_2}{T_{2'}}\right)$$

$$5.5497 = 4.9842 + 2.3 \times 2.763 \log\left(\frac{T_2}{T_{2'}}\right)$$

$$\log\left(\frac{T_2}{T_{2'}}\right) = \frac{5.5497 - 4.9842}{2.3 \times 2.763} = 0.089$$

$$\frac{T_2}{T_{2'}} = 1.227 \qquad \text{...(Taking antilog of 0.089)}$$

$$T_2 = T_{2'} \times 1.227 = 303 \times 1.227 = 371.78 \text{ K}$$

∴

= 371.78 – 273 = 98.78°C

We know that enthalpy at point 2,

$$h_2 = h_{2'} + c_{pv}(T_2 - T_{2'})$$

= 1468.87 + 2.763 (371.78 – 303) = 1658.9 kJ/kg

∴ C.O.P. of the refrigeration cycle

$$= \frac{h_1 - h_{f3}}{h_2 - h_1} = \frac{1426.58 - 286.23}{1658.9 - 1426.58} = \frac{1140.35}{232.32} = 4.91 \text{ Ans.}$$

.....

Th.5 REFRIGERATION AND AIR CONDITIONING
Chapter -3

3.0 VAPOUR ABSORPTION REFRIGERATION SYSTEM

3.1 Simple vapor absorption refrigeration system

3.2 Practical vapor absorption refrigeration system

3.3 COP of an ideal vapor absorption refrigeration system

3.4.Numerical on COP

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CHAPTER – 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.

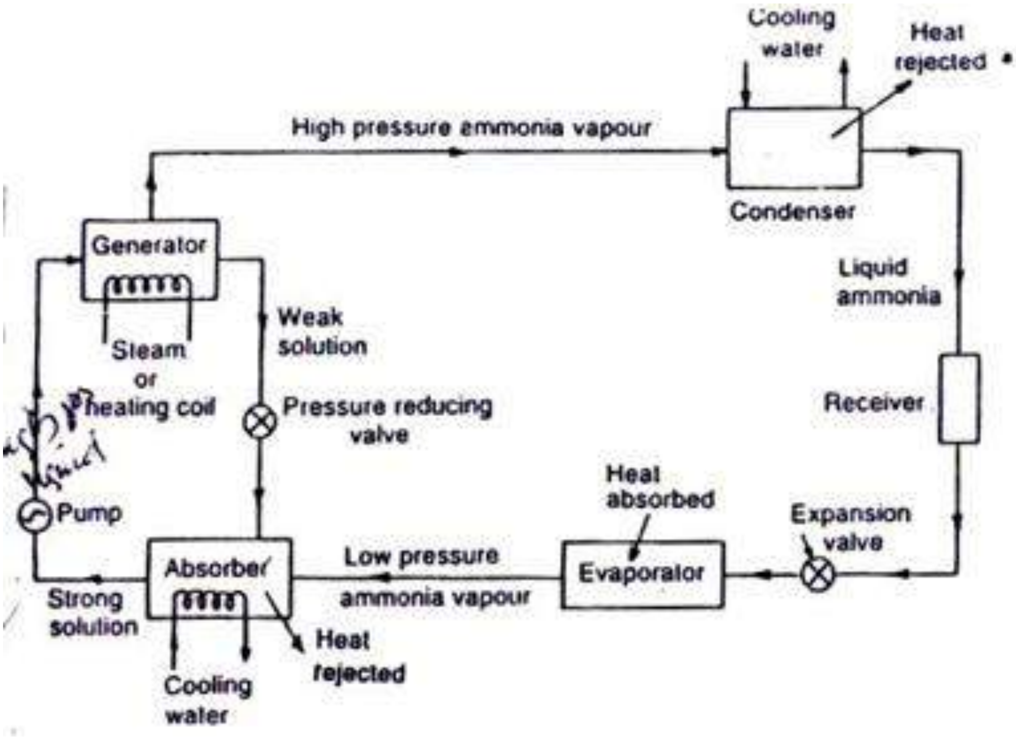

Fig 3.1 Simple vapour absorbtion system

. 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.

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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 :-

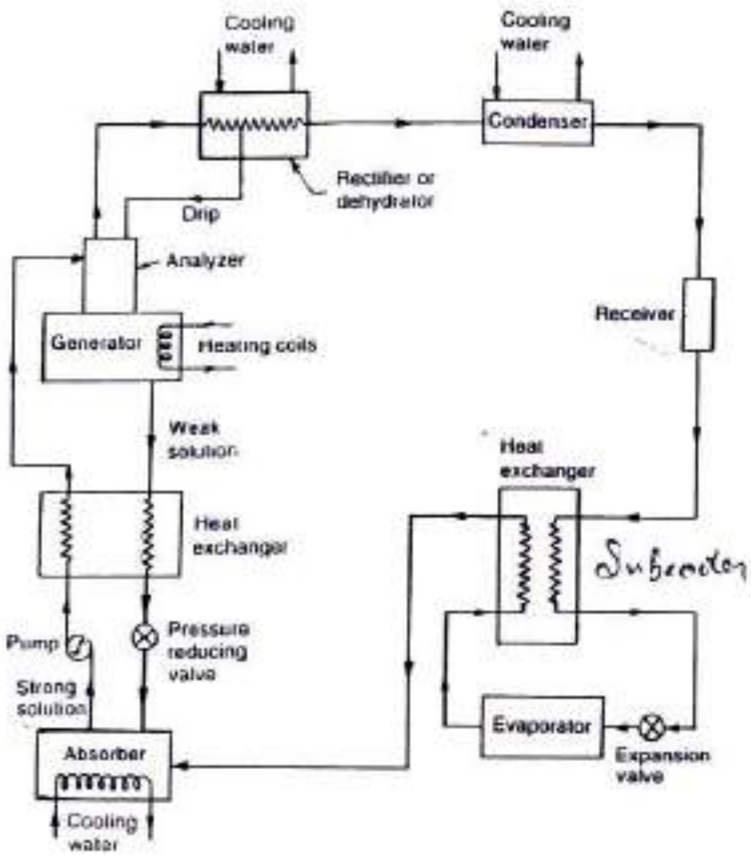

Fig. 3.2. Practical vapour absorption system.

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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

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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.

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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 (Qp) is added to the refrigerant due to pump work.

Neglecting the heat due to pump work (Qp), 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,

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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,

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I. 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

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

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

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

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^oC = 100 + 273 = 373 K ; Tc = 20^oC = 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.

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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^{o}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.

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

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

$\therefore$ 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

$\therefore$ Mass of steam required per hour

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

Th.5 REFRIGERATION AND AIR CONDITIONING

Chapter -2

4.0 REFRIGERATION EQUIPMENTS

4.1 REFRIGERANT COMPRESSORS

4.1.1 Principle of working and constructional details of reciprocating and rotary compressors.

4.1.2 Centrifugal compressor only theory

4.1.3 Important terms.

4.1.4 Hermetically and semi hermetically sealed compressor.

4.2 CONDENSERS

4.2.1 Principle of working and constructional details of air cooled and water cooled condenser

4.2.2 Heat rejection ratio.

4.2.3 Cooling tower and spray pond.

4.3 EVAPORATORS

1.6.1 Principle of working and constructional details of an evaporator.

1.6.2 Types of evaporator.

1.6.3 Bare tube coil evaporator, finned evaporator, shell and tube evaporator.

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MODULE IV

Refrigeration System Components

4.1 Introduction

A refrigerant compressor, as the name indicates, is a machine used to compress the vapour refrigerant from the evaporator and to raise its pressure so that the corresponding saturation temperature is higher than that of the cooling medium. It also continually circulates the refrigerant through the refrigerating system. Since the compression of refrigerant requires some work to be done on it, therefore, a compressor must be driven by some prime mover.

Note : Since the compressor virtually takes the heat at a low temperature from the evaporator and pumps it at the high temperature to the condenser. therefore it is often referred to as a heat pump.

4.2 Classification of Compressors

The compressors may be classified in many ways, but the following are important from the subject point of view:

1. *According to the method of compression*

(a) Reciprocating compressors.

(b) Rotary compressors, and

(c) Centrifugal compressors.

2. *According to the number of working strokes*

(a) Single acting compressors, and

(b) Double acting compressors.

3. *According to the number of stages*

(a) Single stage (or single cylinder) compressors, and

(b) Multi-stage (or multi-cylinder) compressors.

4. *According to the method of drive employed*

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(a) Direct drive compressors. and

(b) Belt drive compressors.

5. *According to the location of the prime mover*

(a) Semi-hermetic compressors (direct drive, motor and compressor in separate

housings), and

(b) Hermetic compressors (direct drive, motor and compressor in same housings).

4.3 Important Terms

The following important terms, which will be frequently used in this chapter, should be clearly understood at this stage:

1. *Suction pressure*. It is the absolute pressure of refrigerant at the inlet of a compressor.

2. *Discharge pressure*. It is the absolute pressure of refrigerant at the outlet of a compressor.

3. *Compression ratio (or pressure ratio).* It is the ratio of absolute discharge pressure to the absolute suction pressure. Since the absolute discharge pressure is always more than the absolute suction pressure, therefore, the value of compression ratio is more than unity.

Note: The compression ratio may also be defined as the ratio of total cylinder volume to the clearance volume.

4. *Suction volume*. It is the volume of refrigerant sucked by the compressor during its suction stroke. It is usually denoted by v_s.

5. *Piston displacement volume or stroke volume or swept volume*. It is the volume swept by the piston when it moves from its top or inner dead position to bottom or outer dead centre position. Mathematically, piston displacement volume or stroke volume or swept volume,

$$v_p = \frac{\pi}{4} \times D^2 \times L$$

where D = Diameter of cylinder. and

L = Length of piston stroke .

6. *Clearance factor*. It is the ratio of clearance volume (v_c) to the piston displacement volume (v_p). Mathematically, clearance factor

$$C = \frac{v_c}{v_p}$$

7. *Compressor capacity*. It is the volume of the actual amount of refrigerant passing through the compressor in a unit time. It is equal to the suction volume (v_s). It is expressed in m3/s.

8. *Volumetric efficiency*. It is the ratio of the compressor capacity or the suction volume (v_s) to the piston displacement volume (v_p). Mathematically, volumetric efficiency,

$$\eta_v = \frac{v_s}{v_p}$$

4.4 Reciprocating Compressors

The compressors in which the vapour refrigerant is compressed by the reciprocating (i.e. back and forth) motion of the piston, are called reciprocating compressors. These compressors are used for refrigerants which have comparatively low volume per kg and a large differential pressure, such as ammonia (R-717), R-12, R-22, and methyl chloride (R-40). The reciprocating compressors are available in sizes as small as 1/12 kW which are used in small domestic refrigerators and up to about 150 kW for large capacity installations.

The two types of reciprocating compressors in general use are single acting vertical compressors and double acting horizontal compressors. The single acting compressors usually have their cylinders arranged vertically, radially or in a V or W form. The double acting compressors usually have their cylinders arranged horizontally.

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Fig. 4.1 shows a single stage single acting reciprocating compressor in its simplest form.

The principle of operation of the compression cycle is as discussed below:

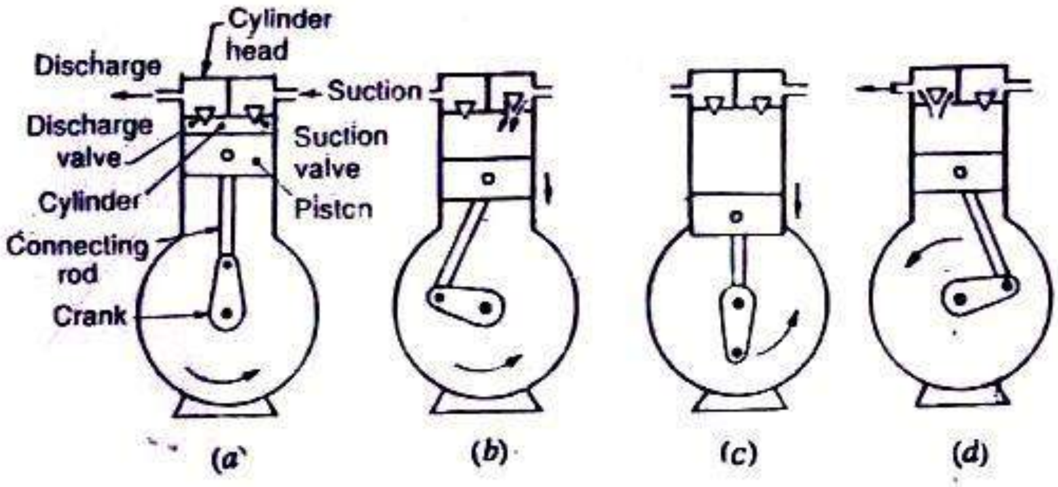

Fig. 4.1. Principle of operation of a single stage, single acting reciprocating compressor.

Let us consider that the piston is at the top of its stroke as shown in Fig. 4.1 (a). This is called top dead centre position of the piston. In this position, the suction valve is held closed because of the pressure in the clearance space between the top of the piston and the cylinder head. The discharge valve is also held closed because of the cylinder head pressure acting on the top of it.

When the piston moves downward (i.e. during suction stroke) as shown in Fig. 4.1 (b), the refrigerant left in the clearance space expands. Thus the volume of the cylinder (above the piston) increases and the pressure inside the cylinder decreases. When the pressure becomes slightly less than the suction pressure or atmospheric pressure, the suction valve gets opened and the vapour refrigerant flows into the cylinder. This flow continues until the piston reaches the bottom of its make (I.e. bottom dead centre). At the bottom of the stroke, as shown in Fig. 4.1 (c), the suction valve closes because of spring action. Now when the piston moves upward (i.e. during compression stroke) as shown in Fig. 4.1 (d), the volume of the cylinder decreases and the pressure inside the cylinder Increases. When the pressure inside the

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cylinder becomes greater than that on the top of the discharge valve, the discharge valve gets opened and the vapour refrigerant is discharged into the condenser and the cycle is repeated.

It may be noted that in a single acting reciprocating compressor, the suction, compression and discharge of refrigerant takes place in two strokes of the piston or in one revolution of the crankshaft.

Notes: 1. In a double acting reciprocating compressor, the suction and compression takes place on both sides of the piston. It is thus obvious, that such a compressor will supply double the volume of refrigerant than a single acting reciprocating compressor (neglecting volume of piston rod).

2. There must be a certain distance between the top of the piston and the cylinder head when the piston is on the top dead centre so that the piston does not strike the cyclinder head. This distance is called clearance space and the volume therein is called the clearance volume. The refrigerant left in this space is at discharge pressure and its pressure must be reduced below that of suction pressure (atmospheric pressure) before any vapour refrigerant flows into the cylinder. The clearance space should be a minimum.

3. The low capacity compressors are air cooled. The cylinders of these compressors usually have fins to provide better air cooling. The high capacity compressors are cooled by providing water jackets around the cylinder.

4.5 Work Done by a Single Stage Reciprocating Compressor

We have already discussed that in a reciprocating compressor, the vapour refrigerant is first sucked, compressed and then discharged. So there are three different operations of the compressor. Thus we see that work is done on the piston during the suction of refrigerant. Similarly, work is done by the piston during compression as well as discharge of the refrigerant. A little consideration will show, that the work done by a reciprocating compressor is mathematically equal to the work done by the compressor during compression as well as discharge minus the work done on the compressor during suction.

Here we shall discuss the following two important cases of work done:

1. When there is no clearance volume in the cylinder, and

2. When there is some clearance volume.

4.6 Work Done by a Single Stage, Single Acting Reciprocating Compressor without Clearance Volume

Consider a single stage, single acting reciprocating compressor without clearance as shown by the p-v and T-s diagrams in Fig. 9.2.

Let P_1 = Suction pressure of the refrigerant (before compression),

V_1 = Suction volume of the refrigerant (before compression),

T_1 = Suction temperature of the refrigerant (before compression),

P_2 ,v_2 and T_2 = Corresponding values for the refrigerant at the discharge point i.e. after compression, and

r = Compression ratio or pressure ratio, P_2/P_1.

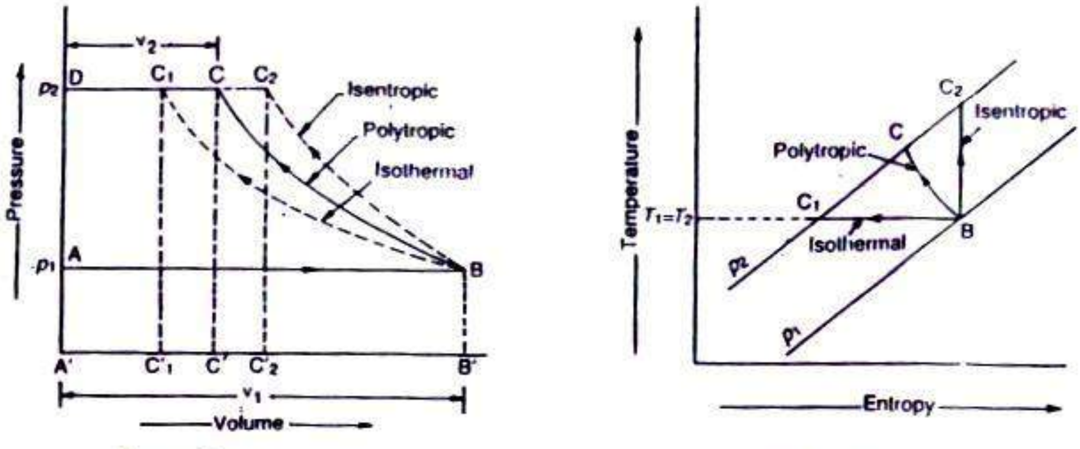
As a matter of fact, the compression of refrigerant may be isothermal, polytropic, or isentropic (reversible adiabatic). Now we shall find out the amount of work done in compressing the refrigerant in all the above mentioned three cases.

1. *Work done during isothermal compression*

We have already discussed that when the piston moves from the top dead centre (point A), the refrigerant is admitted into the compressor cylinder and it continues till the piston reaches at its bottom dead centre (point B). Thus the line A B represents suction stroke and the area below this line (i.e. area AB B' A') represents the work done during suction stroke. From the figure, we find that work done during suction stroke,

WI = Area ABB' A' = $P_l v_l$

The refrigerant is compressed during the return stroke (or compression stroke BC_1) of the piston at constant temperature. The compression continues till the pressure (P_2) in the cylinder is sufficient to force open the discharge valve at C_1. After that no compression takes place with the inward movement of the piston. Now during the remaining part of the compression stroke, the compressed refrigerant is discharged to condenser till the piston reaches its top dead centre.



(a) p-v diagram. (b) T-s diagram.

Fig. 4.2. p-v and T-s diagrams for a single stage reciprocating compressor.

From the figure, we find that Work done during compression.

$$W2 = \text{Area } BC_1C_1'B' = p_1v_1 \log_e\left(\frac{v_1}{v_2}\right)$$

and work done during discharge,

$$W3 = \text{Area } C_1DA'C_1' = p_2v_2$$

$\therefore$ Work done by the compressor per cycle,

$$W = \text{Area } ABC_1D$$

$$= \text{Area } C_1DA'C_1' + \text{Area } BC_1C_1'B' - \text{Area } ABB'A'$$

$$= W3 + W2 - W1$$

$$= p_2v_2 + p_1 \log_e\left(\frac{v_1}{v_2}\right) - p_1v_1 = p_1v_1 \log_e\left(\frac{v_1}{v_2}\right) \qquad \ldots (\because p_1v_1 = p_2v_2)$$

$$= 2.3p_1v_1\log\left(\frac{v_1}{v_2}\right) = 2.3\ p_1v_1 \log r \qquad \left(\because \frac{v_1}{v_2} = \frac{p_2}{p_1} = r\right)$$

$$= 2.3\ m\ RT_1 \log r \qquad \ldots (\because p_1v_1 = mRT_1)$$

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4.7 Hermetic Sealed Compressors

When the compressor and motor operate on the same shaft and are enclosed in a common casing, they are known as hermetic sealed compressors. These types of compressors eliminate the use of crank shaft seal which is necessary in ordinary compressors in order to prevent leakage of refrigerant. These compressors may operate on either reciprocating or rotary principle and may be mounted with the shaft in either the vertical or horizontal position. The hermetic units are widely used for small capacity refrigerating systems such as in domestic refrigerators, home freezers and window air conditioners.

The hermetic sealed compressors have the following advantages and disadvantages:

Advantages

1. The leakage of refrigerant is completely prevented.

2. It is less noisy.

3. It requires small space because of compactness.

4. The lubrication is simple as the motor and compressor operate in a sealed space with the lubricating oil.

Disadvantages

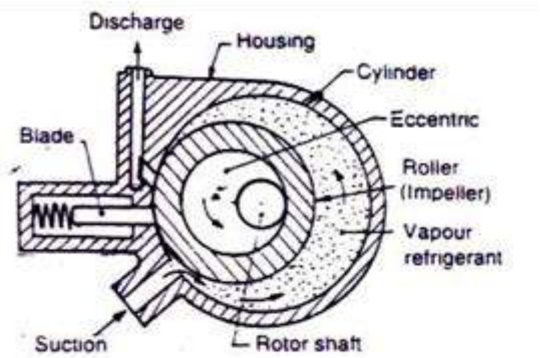
I. The maintenance is not easy because the moving parts are inaccessible.

2. A separate pump is required for evacuation and charging of refrigerant.

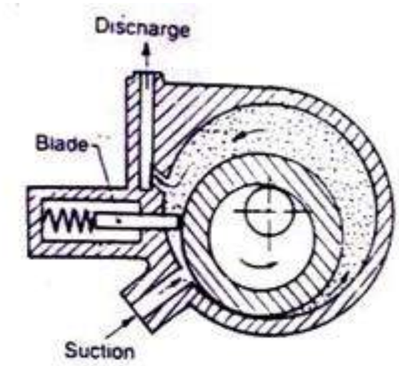
4.8 Rotary Compressors

In rotary compressors, the vapour refrigerant from the evaporator is compressed due to the movement of blades. The rotary compressors are positive displacement type compressors. Since the clearance in rotary compressors is negligible, therefore they have high volumetric efficiency. These compressors may be used with refrigerants R-12, R-22, R-114 and ammonia. Following are the two basic types of rotary compressors:

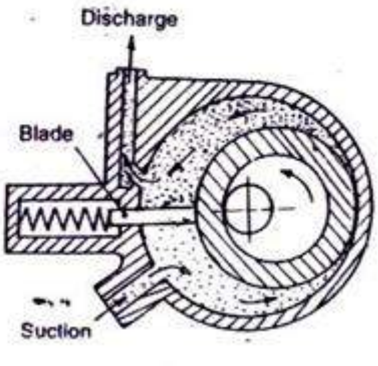
1. *Single stationary blade type rotary compressor*. A single stationary blade type rotary compressor is shown in Fig. 9.14. This consists of a stationary cylinder, a roller (or impeller) and a shaft. The shaft has an eccentric on which the roller is mounted. A blade is set into the slot of a cylinder in such a manner that it always maintains contacts with the roller by means of a spring. The blade moves in and out of the slot to follow the rotor when it rotates. Since the blade separates the suction and discharge ports as shown in Fig. 9.14, therefore it is often called a sealing blade. When the shaft rotates, the roller also rotates so that it always touches the cylinder wall.



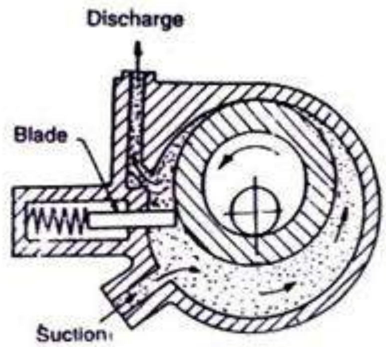
(a)Comletion of intake strokeand Beginning of compression



(b) Compression stroke continued and new intake stroke started



(c) compression continued and new intake stroke continued



(d) Compressed vapour discharge to condenser and new intake stroke continued

Fig. 4.3 Stationary single blade rotary compressor.

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Fig. 4.3(a) to (d) shows the various positions of roller as the vapour refrigerant is compressed. Fig. 4.3(a) shows the completion of intake stroke (i.e. the cylinder is full of low pressure and temperature vapour refrigerant) and the beginning of compression stroke. When the roller rotates, the vapour refrigerant ahead of the roller is being compressed and the new intake from the evaporator is drawn into the cylinder, as shown in Fig.4.3 (b). As the roller turns towards mid position as shown in Fig. 4.3 (c), more vapour refrigerant is drawn into the cylinder while the compressed refrigerant is discharged to the condenser. At the end of compression stroke, as shown in Fig. 4.3 (d), most of the compressed vapour refrigerant is passed through the discharge port to the condenser. A new charge of refrigerant is drawn into the cylinder. This, in turn, is compressed and discharged to the condenser. In this way, the low pressure and temperature vapour refrigerant is compressed gradually to a high pressure and temperature.

2. *Rotating blade type rotary compressor*. The rotating blade type rotary compressor is shown in Fig. 4.4. This consists of a cylinder and a slotted rotor containing a number of blades. The centre of the rotor is eccentric with the centre of the cylinder.

The blades are forced against the cylinder wall by the centrifugal action during the rotation of the motor.

The low pressure and temperature vapour refrigerant from the evaporator is drawn through the suction port. As the rotor turns, the suction vapour refrigerant entrapped between the two adjacent blades is compressed. The compressed refrigerant at high pressure and temperature is discharged through the discharge port to the condenser.

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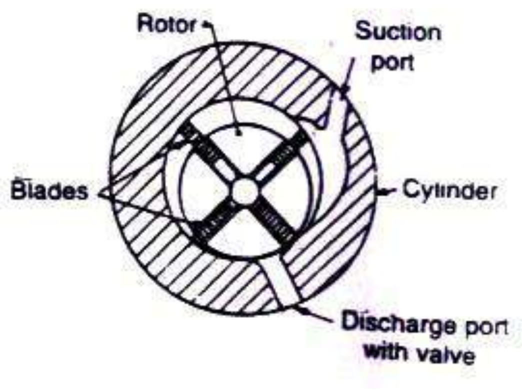

Fig. 4.4. Rotating blade type rotary compressor.

Note: The whole assembly of both the types of rotary compressors is enclosed in a housing which is filled with oil. When the compressor is working, an oil film forms the seal between the high pressure and low pressure side. But when the compressor stops, this seal is lost and therefore high pressure vapour refrigerant will flow into low pressure side. In order to avoid this, a check valve is usually provided in the suction line. This valve prevents the high pressure vapour refrigerant from flowing back to the evaporator.

4.9 Centrifugal Compressors

The centrifugal compressor for refrigeration systems was designed and developed by Dr.Willis H. Carrier in 1922. This compressor increases the pressure of low pressure vapour refrigerant to a high pressure by centrifugal force. The centrifugal compressor is generally used for refrigerants that require large displacement and low condensing pressure, such as R-11 and R-113. However, the refrigerant R-12 is also employed for large capacity applications and low-temperature applications.

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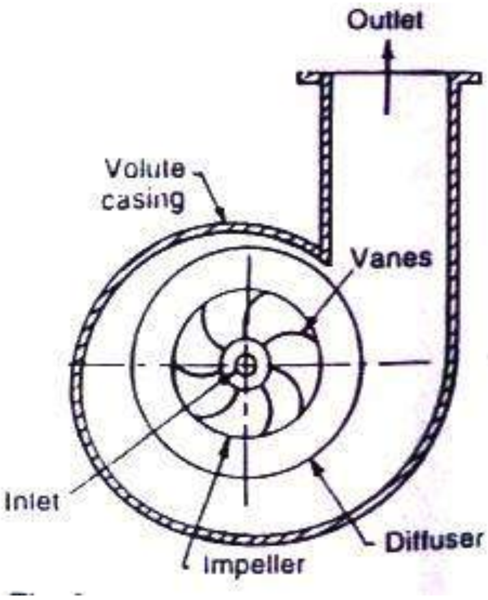

Fig. 4.5

A single stage centrifugal compressor, in its simplest form, consists of an impeller to which a number of curved vanes are fitted symmetrically, Inlet as shown in Fig.. 4.5. The impeller rotates in an air tight volute casing with inlet and outlet points.

The impeller draws in low pressure vapour refrigerant from the evaporator. When the impeller rotates, it pushes the vapour refrigerant from the centre of the impeller to its periphery by centrifugal force. The high speed of the impeller leaves the vapour refrigerant at a high velocity at the vane tips of the impeller. The kinetic energy thus attained at the impeller outlet is converted into pressure energy when the high velocity vapour refrigerant passes over the diffuser. The diffuser is normally a vaneless type as it permits more efficient part load operation which is quite usual in any air-conditioning plant. The volute casing collects the refrigerant from the diffuser and it further converts the kinetic energy into pressure energy before it leaves the refrigerant to the evaporator.

Notes: 1. In case of a single stage centrifugal compressor, the compression ratio that an impeller can develop is limited to about 4.5. But when high compression ratio is desired, multi- stage centrifugal compressors with intercoolers are employed.

2. The centrifugal compressors have no valves, pistons and cylinders. The only wearing parts are the main bearings.

4.10 Advantages and Disadvantages of Centrifugal Compressors over Reciprocating Compressors

Following are the advantages and disadvantages of centrifugal compressors over reciprocating compressors:-

Advantages

I. Since the centrifugal compressors have no valves, pistons. cylinders, connecting rod etc., therefore the working life of these compressors is more as compared to reciprocating compressors.

2. These compressors operate with little or no vibration as there are no unbalanced masses.

3. The operation of centrifugal compressors is quiet and calm.

4. The centrifugal compressors run at high speeds (3000 r.p.m. and above), therefore these can be directly connected to electric motors or steam turbines.

5. Because of the high speed, these compressors can handle large volume of vapour refrigerant, as compared to reciprocating compressors.

6. The centrifugal compressors are especially adapted for systems ranging from 50 to 5000 tonnes. They are also used for temperature ranges between - 90° C and + 10° C.

7. The efficiency of these compressors is considerably high.

8. The large sizes centrifugal compressors require less floor area as compared to reciprocating compressors.

Disadvantages

I. The main disadvantage in centrifugal compressors is •surging. It occurs when the refrigeration load decreases to below 35 percent of the rated capacity and causes severe stress conditions in the compressor.

2. The increase in pressure per stage is less as compared to reciprocating compressors.

3. The centrifugal compressors are practical below 50 tonnes capacity load.

4. The refrigerants used with these compressors should have high specific volume.

4.11 Capacity Control of Compressors

There are many refrigeration applications in which the refrigeration load is not constant. It is, therefore, necessary to provide some means to control the capacity of a compressor according to the load. It may be noted that the compressors operating under partial loads and low back pressure creates a condition where the coil may freeze or damage may result.

4.2 Condensers

4.12 Introduction

The condenser is an important device used in the high pressure side of a refrigeration system. Its function is to remove heat of the hot vapour refrigerant discharged from the compressor. The hot vapour refrigerant consists of the heat absorbed by the evaporator and the heat of compression added by the mechanical energy of the compressor motor. The heat from the hot vapour refrigerant in a condenser is removed first by transferring it to the walls of the condenser tubes and then from the tubes to the condensing or cooling medium. The cooling medium may be air or water or a combination of the two.

The selection of a condenser depends upon the capacity of the refrigerating system, the type of refrigerant used and the type of cooling medium available.

4.13 Working of a Condenser

The working of a condenser may be best understood by considering a simple refrigerating system as shown in Fig. 4.6 (a). The corresponding p - h diagram showing three stages of a refrigerant cooling is shown in Fig. 4.6 (b) . The compressor draws in the superheated vapour refrigerant that contains the heat it absorbed in the evaporator. The compressor adds more bell (i. e. the heat of compression) to the superheated vapour. This highly superheated vapour from the compressor is pumped to the condenser through the discharge line. The condenser cools the refrigerant in the following three stages :-

I. First of all, the superheated vapour is cooled to saturation temperature (called desuperheating) corresponding to the pressure of the refrigerant. This is shown by the line 2-3 in Fig. 4.6 (b). The desuperheating occurs in the discharge line and in the first few coils of the condenser.

2. Now the saturated vapour refrigerant gives up its latent heat and is condensed to a saturated liquid refrigerant. This process, called condensation, is shown by the line 3-4.

3. The temperature of the liquid refrigerant is reduced below its saturation temperature (i.e. sub-cooled) in order to increase the refrigeration effect. This process is shown by the line 4-5.

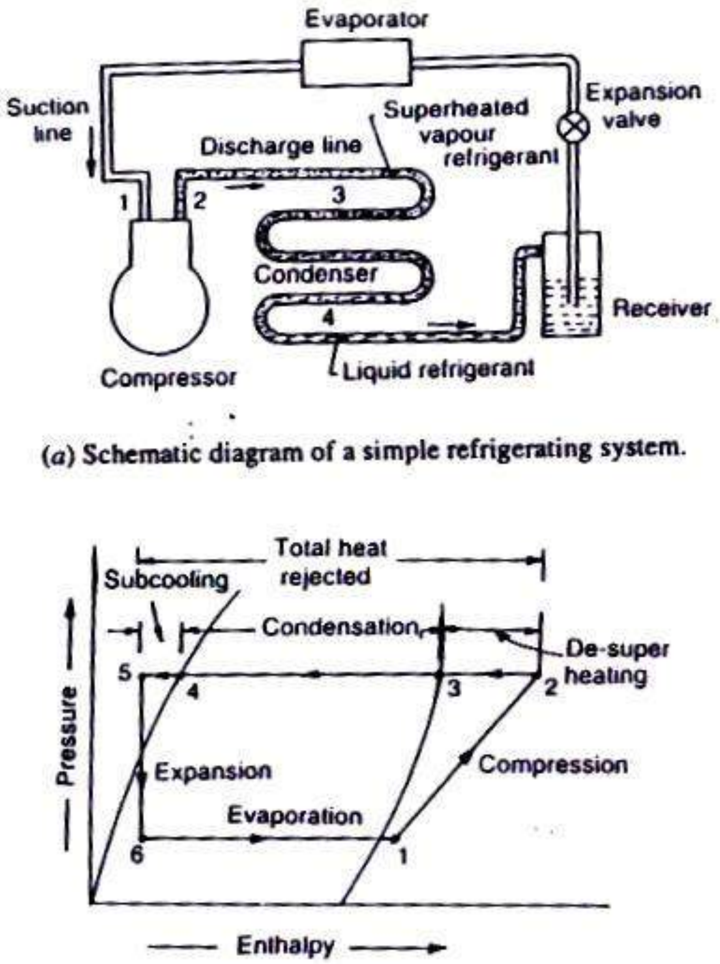

Fig. 4.6

4.14 Factors Affecting the Condenser Capacity

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The condenser capacity is the ability of the condenser to transfer heat from the hot vapour refrigerant to the condensing medium. The heat transfer capacity of a condenser depends upon the following factors :

1. *Material*. Since the different materials have different abilities of heat transfer, therefore the size of a condenser of a given capacity can be varied by selecting the right material. It may be noted that higher the ability of a material to transfer heat, the smaller will be the size of condenser.

2. *Amount of contact*. The condenser capacity may be varied by controlling the amount of contact between the condenser surface and the condensing medium. This can be done by varying the surface area of the condenser and the rate of flow of the condensing medium over the condenser surface. The amount of liquid refrigerant level in the condenser also affects the amount of contact between the vapour refrigerant and the condensing medium. The portion of the condenser used for liquid sub-cooling can not condense any vapour refrigerant.

3. *Temperature difference*. The heat transfer capacity of a condenser greatly depends upon the temperature difference between the condensing medium and the vapour refrigerant. As the temperature difference increases, the heat transfer rate increases and therefore the condenser capacity increases. Generally, this temperature difference cannot be controlled. But when the temperature difference becomes so great that it becomes a problem, devices are available that will change the amount of condensing surface and the air flow rate to control condenser capacity.

Note: Most air-cooled condensers are designed to operate with a temperature difference of 14° C.

4.15 Heat Rejection Factor

We have already discussed that in a vapour compression refrigeration system, the heat is rejected in a condenser. The load on the condenser per unit of refrigeration capacity is known as heat rejection factor. The load on the condenser(Q_C) is given by

Q_C = Refrigeration capacity + Work done by the compressor = $R_E + W$

:. Heat rejection factor,

$$HRF = \frac{Q_c}{R_E} = \frac{R_E + W}{R_E} = 1 + \frac{W}{R_E} = 1 + \frac{1}{COP} \qquad \ldots \left(\because COP = \frac{R_E}{W} \right)$$

From above, we see that the heat rejection factor depends upon the coefficient of performance (COP) which in turn depends upon the evaporator and condenser temperatures.

In actual air-conditioning applications for R-12 and R-22, and operating at a condenser temperature of 40^{o} C and an evaporator temperature of 5° C, the heat rejection factor is about 1.25.

4. 16 Classification of Condensers

According to the condensing medium used, the condensers are classified into the following

three groups :

1. Air cooled condensers,

2. Water cooled condensers, and

3. Evaporative condensers.

These condensers are discussed, in detail, in the following pages.

4. 17 Air Cooled Condensers

An air-cooled condenser is one in which the removal of heat is done by air. It consists of steel or copper tubing through which the refrigerant flows. The size of tube usually ranges from 6 mm to 18 mm outside diameter, depending upon the size of condenser. Generally copper tube used because of its excellent heat transfer ability. The condensers with steel tubes are- used in ammonia refrigerating systems. The tubes are usually provided with plate type fins to increase the surface area for heat transfer, as shown in Fig. 4.7. The fins are usually made from aluminium because of its light weight. The fin spacing is quite wide to reduce dust clogging.

The condensers with single row of tubing provides the most efficient heat transfer. This is because the air temperature rises at it passes through each row of tubing. The temperature difference between the air and the vapour refrigerant decreases in each row of tubing and therefore each row becomes less effective.

However, single row condensers require more space than multi-row condensers. The single row condensers are usually used in small capacity refrigeration systems such as domestic refrigerators. freezers, water coolers and room air conditioners.

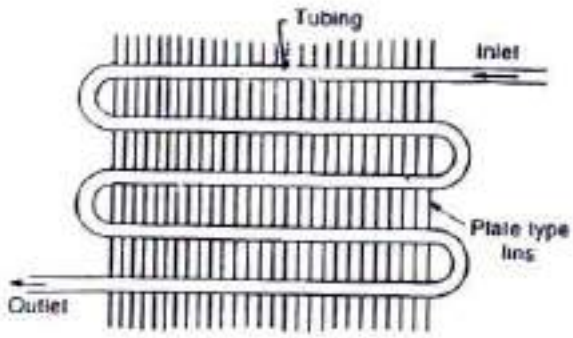

Fig. 4.7. Air cooled condenser.

The air cooled condensers may have two or more rows of tubing, but the condensers with up to six rows of tubing are common. Some condensers have seven or eight rows. However more than eight rows of tubing are usually not efficient. This is because the air temperature will be too close to the condenser temperature to absorb any more heat after passing through eight rows of tubing.

Note: The main disadvantage of an air cooled condenser is that it operates at a higher condensing temperature than a water cooled condenser. The higher condensing temperature causes the compressor to work more.

4.18 Types of Air-Cooled Condensers

Following are the two types of air-cooled condensers:

1. *Natural convection air-cooled condensers*. In natural convection air-cooled condenser, the heat transfer from the condenser coils to the air is by natural convection. As the air comes in contact with the warm condenser tubes, it absorbs heat from the refrigerant and thus the temperature of air increases. The warm air being lighter, rises up and the cold air from below rises to take away the heat from the condenser. This cycle continues in natural convection air-cooled condensers. Since the rate of heat transfer in natural convection condenser is slower, therefore they require a larger surface area as compared to forced convection condensers. The

natural convection air-cooled condensers are used only in small capacity applications such as domestic refrigerators, freezers, water coolers and room air- conditioners.

2. *Forced convection air-cooled condensers*. In forced convection air-cooled condensers, the fan (either propeller or centrifugal) is used to force the air over the condenser coils to increase its heat transfer capacity. The forced convection condensers may be divided into the following two groups:

(a) Base mounted air-cooled condensers, and (b) Remote air-cooled condensers.

The base mounted air-cooled condensers, using propeller fans, are mounted on the ,same base of compressor, motor, receiver and other controls. The entire arrangement is called a condensing unit. In small units, the compressor is belt driven from the motor and the fan required to force the air through the condenser is mounted on the shaft of the motor. The use of this type of compressor for indoor units is limited up to 3 kW capacity motor only. These condensing units are used on packaged refrigeration systems of 10 tonnes or less.

The remote air-cooled condensers are used on systems above 10 tonnes and are available up to 125 tonnes. The systems above 125 tonnes usually have two or more condensers. These condensers may be horizontal or vertical. They can be located either outside or inside the building.

The remote condensers located outside the building can be mounted on a foundation on the ground, on the roof or on the side of a building away from the walls. These condensers usually use propeller fans because they have low resistance to air flow and free air discharge. They require 18 to 36 m3/min of air per tonne of capacity. The propeller fans can move this volume of air as long as the resistance to air flow is low. To prevent any resistance to air flow, the fan intake on vertical outdoor condensers usually faces the prevailing winds. If this is not possible, the air discharge side is usually covered with a shield to deflect opposing winds.

The remote condensers located inside the building usually require duct work to carry air 10 and from the unit. The duct work restricts air flow to and from the condenser and causes high air pressure d op. Therefore, inside condensers usually

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use centrifugal fans which can move the necessary volume of air against the resistance to air flow.

4.19 Water Cooled Condensers

A water cooled condenser is one in which water is used as the condensing medium. They are always preferred where an adequate supply of clear inexpensive water and means of water disposal are available. These condensers are commonly used in commercial and industrial refrigerating units. The water cooled condensers may use either of the following two water systems:

I. Waste water system, or 2. Recirculated water system.

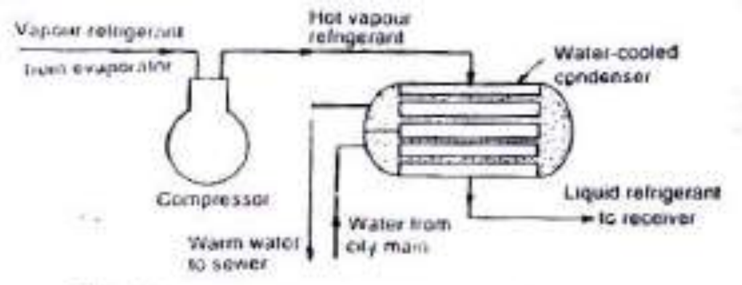

Fig. 10. 3. Water cooled condenser using waste water system.

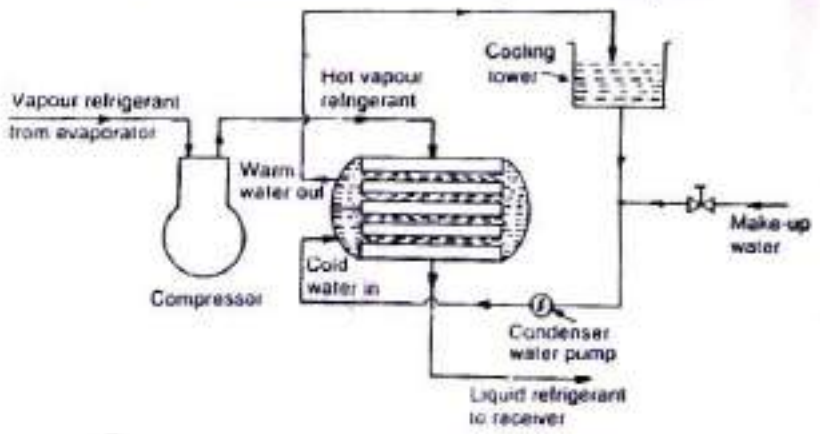

Fig. 4.8. Waler cooled condenser with recirculating water system.

In a *waste water system*, the water after circulating in the condenser is discharged to a sewer, as shown in Fig. 4.8 This system is used on small units and in locations where large quantities of fresh inexpensive water and a sewer system large enough to handle the waste water are available. The most common source of fresh water supply is the city main.

In a *recirculated water system* as shown in Fig. 10.4, the same water circulating in the condenser is cooled and used again and again. Thus this system requires some type of water cooling device. The cooling water towers and spray ponds are the most common cooling devices used in a recirculated water system. The warm water from the condenser is led to the cooling tower where it is cooled by self evaporation into a stream of air. The water pumps are used to circulate the water through the system and then to the cooling tower which is usually located on the roof. Once a recirculated water system is filled with water, the only additional water required is make-up water. The make up water simply replaces the water that evaporates from the cooling tower or spray pond.

Note: The water cooled condensers operate at a lower condensing temperature than an air-cooled condenser. This is because the supply water temperature is normally lower than the ambient air temperature, but the difference between the condensing and cooling medium temperatures is normally the same (i.e. 14° C). Thus, the compressor for a water cooled condenser requires less power for the same capacity.

4.20 Types of Water Cooled Condensers

The water cooled condensers are classified, according to their construction, into the following three groups:

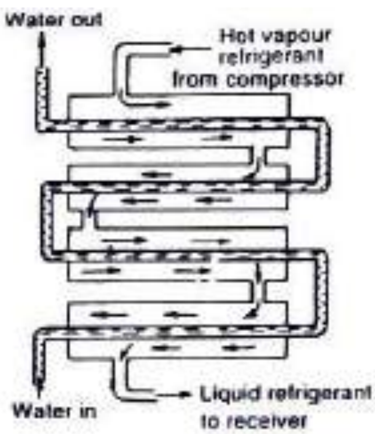

Fig. 4.9 Tube-in-tube condenser.

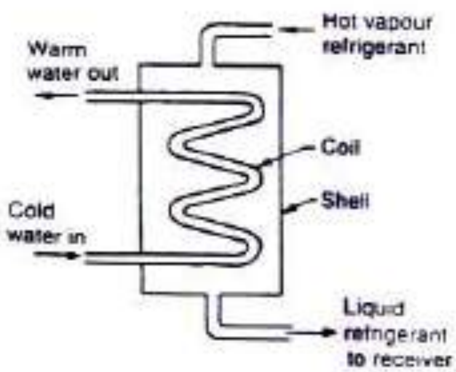

Fig. 4.10. Shell and coil condenser.

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1. *Tube -in-tube or double tube condensers*.The tube-in-tube or double tube condenser, as shown in Fig. 4.9, consists of a water tube inside a large refrigerant tube. In this type of condenser, the hot vapour refrigerant enters at the top of the condenser. The water absorbs the heat from the refrigerant and the condensed liquid refrigerant flows at the bottom. Since the refrigerant tubes are exposed to ambient air, therefore some of the heat is also absorbed by ambient air by natural convection.

The cold water in the inner tubes may flow in either direction. When the water enters at the bottom and flows in the direction opposite to the refrigerant, it is said to be a counter-flow system. On the other hand, when the water enters at the top and flows in the same direction as the refrigerant, it is said to be a *parallel flow system*.

The counter-flow system, as shown in Fig. 10.5, is preferred in all types of water cooled condensers because it gives high rate of heat transfer. Since the coldest water is used for final cooling of the liquid refrigerant and the warmest water absorbs heat from the hottest vapour refrigerant, therefore the temperature difference between the water and refrigerant remains fairly constant throughout the condenser. In case of parallel flow system, as the water and refrigerant flow in the same direction, therefore the temperature difference between them increases. Thus the ability of water to absorb heat decreases at it passes through the condenser.

2. *Shell and coil condensers*. A shell and coil condenser, as shown in Fig. 4.10 consists of one or more water coils enclosed in a welded steel shell. Both the finned and bare coil types are available.

The shell and coil condenser, may be either vertical (as shown in the figure) or horizontal. In this type of condenser, the hot vapour refrigerant enters at the top of the shell and surrounds the water coils. As the vapour condenses, it drops to the bottom of the shell which often serves as a receiver. Most vertical type shell and coil condensers use counter flow water system as it is more efficient than parallel flow water system. In the shell and coil condensers, coiled tubing is free to expand and contract with temperature changes because of its spring action and can withstand any strain caused by temperature changes. Since the water coils are enclosed in a welded steel shell, therefore the mechanical cleaning of these coils is not possible. The coils are cleaned with chemicals. The shell and coil condensers are used for units upto 50 tonnes capacity.

3. *Shell and tube condensers*. The shell and tube condenser, as shown in Fig. 4.11, consists of a cylindrical steel shell containinga number of straight water tubes. The tubes are expanded into grooves in the tube sheet holes to form a vapour- tight fit. The tube sheets are welded to the shell at both the ends. The removable water boxes are bolted to the tube sheet at each end to facilitate cleaning of the condenser. The intermediate supports are provided in the shell to avoid sagging of the tubes.

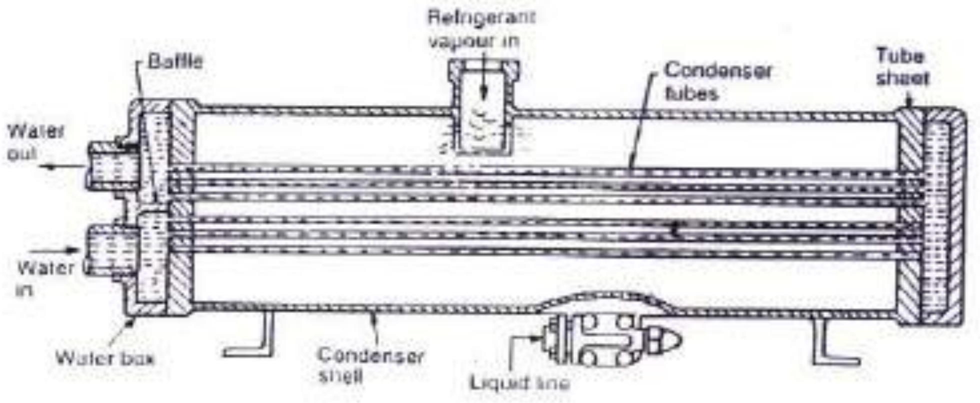

Fig. 4.11 Shell and tube condenser.

The condenser tubes are made either from steel or copper, with or without fins. The steel tubes without fins are usually used in ammonia refrigerating systems because ammonia corrodes copper tubing.

In this type of condenser, the bot vapour refrigerant enters at the top of the shell and condenses as it comes in contact with water tubes. The condensed liquid refrigerant drops to the bottom of the shell which often serves as a receiver. However, if the maximum storage capacity for the liquid refrigerant is less than the total charge of the system, then a receiver of adequate capacity has to be added in case the pump down facility is to be provided as in ice plants, cold storages etc. In some condensers, extra rows of water tubes are provided at the lower end of the condenser for sub-cooling of the liquid refrigerant below the condensing temperature.

4.21 Comparison of Air-Cooled and Water Cooled Condensers

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Following are the comparison.between air-cooled and water cooled condensers:

S.No.	Air-cooled condenser	Water cooled condenser
1.	Since the construction of air cooled condenser is very simple, therefore the initial cost is less. The maintenance cost is also low.	Since the construction of water cooled condenser is complicated, therefore the initial cost is high. The maintenance cost is also high.
2.	There is no handling problem with air cooled condensers.	The water cooled condensers are difficult to handle.
3.	The air cooled condensers do not require piping arrangement for carrying the air.	The pipes are required to take water to and from the condenser.
4.	There is no problem in disposing of used air.	There is a problem of disposing the used water unless a recirculatiqn system is provided.
5.	Since there is no corrosion, therefore fouling effect is low.	Since corrosion occurs inside the tubes carrying the water, therefore fouling effects are high.
6.	The air-cooled condensers have low heat transfer capacity due to low thermal conductivity of air.	The water cooled condensers have high heat transfer capacity due to high thermal conductivity of water.

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7.	These condensers are used for low capacity plants (less than 5 TR).	These condensers are used for large capacity plants.
8.	Since the power required to drive the fan is excessive, therefore, the fan noise becomes objectionable.	There is no fan noise.
9.	The distribution of air on condenser surface is not uniform.	There is even distribution of water on the condensing surface.
10.	The air-cooled condensers have high flexibility.	The water cooled condensers have low flexibility.

4.22 Fouling Factor

The water used in water cooled condensers always contain a certain amount of minerals and other foreign materials, depending upon its source. These materials form deposits inside the condenser water tubes. This is called water fouling. The deposits insulate the tubes, reduce their heat transfer rate and restricts the water flow.

The fouling factor is the reciprocal of heat transfer coefficient for the material of scale.

The following are the recommended fouling factors:

I. For copper tubes used for R-12 and R-22 condensers, the fouling factor is 0.000 095 m^2 s K/J.

2. For steel tubes used in ammonia condensers, the fouling factor is 0.000 18 m^2 s K/J

4.23 Heat Transfer in Condensers

The heat transfer (Q) in water cooled condensers is given by

$$Q = UA\ \Delta T = \frac{\Delta T}{R}$$

where U = Overall heat transfer coefficient,

A = Surface area of the condenser,

ΔT = Overall temperature difference, and

R = Overall thermal resistance of the condenser = l/U A

In order to find the overall thermal resistance, let us consider that the water is flowing inside the tube and the refrigerant outside the tube in a shell of a condenser, as shown in Fig. 4.12. When steady state is reached, there is a film of water inside the tube over the scale formed due to hardness of water. Another layer of film over the tube is formed by the refrigerant. The heat transfer m. the vapour refrigerant to the water in tubes takes place in the following manner.

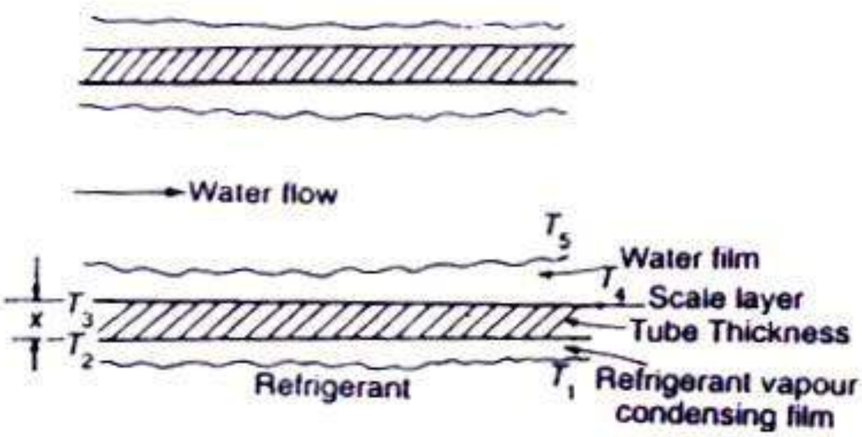

Fig. 4.12. Heat transfer in condensers.

1. The heat transfer takes place from the vapour refrigerant to the outside of the tube through the condensing film. The value of this heat transfer is given by :

$$Q = h_0 A_0 (T_1 - T_2)$$

or $\quad T1-T2 = \dfrac{Q}{h_0 A_0}$

where $\quad T_1$ = Temperature of the refrigerant vapour condensing film,

T_2 = Temperature at the outside surface of the tube,

h_0 = Coefficient of heat transfer for the refrigerant vapour condensiq film, and

A_0 = Condensing area.

2. The heat transfer takes place from the outside surface to the inside surface of the tube.

The value of this heat transfer is given by

$$Q = \frac{k A_m (T_2 - T_3)}{x}$$

or $T_2 - T_3 = \dfrac{Qx}{k A_m (T_2 - T_3)}$ ... (ii)

whereT3 = Temperature at the inside surface of the tube,

x = Thickness of the tube,

k = Thermal conductivity of the tube material, and

A_m = Mean surface area of the tube.

3 EVAPORATORS

4.24 Bare Tube Coil Evaporaters

The simplest type of evaporator is the bare tube coil evaporator, as shown in Fig. 4.13

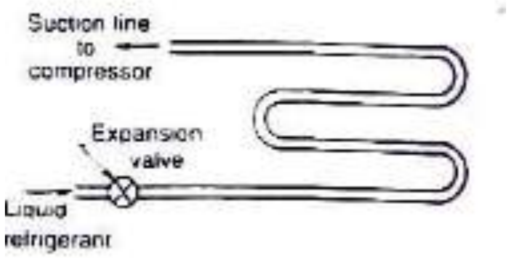

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Fig.4.13 Bare tube coil evapourator

The bare tube coil evaporators are also known as prime- surface evaporators. Because of its simple construction, the bare tube coil is easy to clean and defrost. A little consideration will show that this type of evaporator offers relatively little surface contact area as compared to other types of coils. The amount of surface area may be increased by simply extending the length of the tube, but there are disadvantages of excessive tube length. The effective length of the tube is limited by the capacity of expansion valve. If the tube is too long for the valve's capacity, the liquid refrigerant will tend to completely vaporise early in its progress through the tube, thus leading to excessive superheating at the outlet. The long tubes will also cause considerably greater pressure drop between the inlet and outlet of the evaporator. This results in a reduced suction line pressure.

The diameter of the tube in relation to tube length may also be critical. If the tube diameter is too large, the refrigerant velocity will be too low and the volume of refrigerant will be too great in relation to the surface area of the tube to allow complete vaporisation. This, in turn, may allow liquid refrigerant to enter the suction line with possible damage to the compressor (i.e., slugging). On the other hand, if the diameter is too small, the pressure drop due to friction may be too high and will reduce the system efficiency.

The bare tube coil evaporates may be used for any type of refrigeration requirement. Its use is, however, limited to applications where the box temperatures are under 0^{o}C and in liquid cooling, because the accumulation of ice or frost on these evaporates has less effect on the heat transfer than on those equipped with fins. The bare tube coil evaporators are also extensively used in house-hold refrigerators because they are easier to keep clean.

4.25 Finned Evaporators

The finned evaporator, as shown in Fig. 4.14, consists of bare tubes or coils over which the metal plates or fins are fastened.

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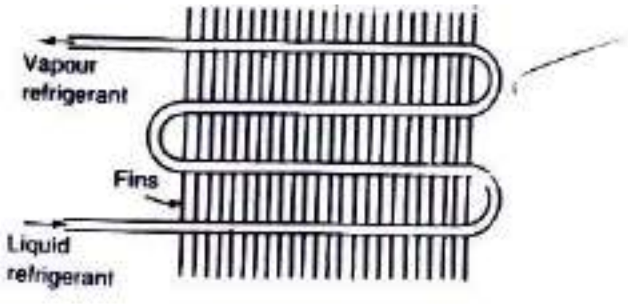
The metal fins are constructed of thin sheets of metal having good thermal conductivity. The shape, size or spacing of the fins can be adapted to provide best rate of heat transfer for a given application. Since the fins greatly increases the contact surfaces for heat transfer, therefore the finned evaporators are also called extended surface evaporators.



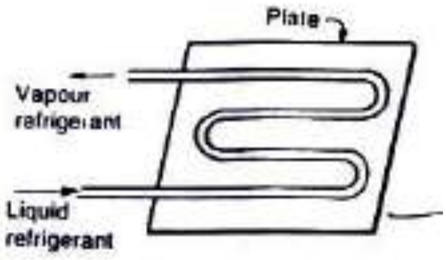


Fig. 4.14. Finned evaporator. Fig. 4.15. Plate evaporator.

The finned evaporators are primarily designed for air conditioning applications where the refrigerator temperature is above $0^{o}C$. Because of the rapid heat transfer of the finned evaporator, it will defrost itself on the off cycle when the temperature of the coil is near $0^{o}C$. A finned coil should never be allowed to frost because the accumulation of frost between the fins reduces the capacity. The air conditioning coils which operate at suction temperatures which are high enough so that frosting never occurs, have fin spacing as small as 3 mm. The finned coils which frost on the on cycle defrost on the off cycle have wider fin spacing.

4.26 Plate Evaporators

A common type of plate evaporator is shown in Fig. 11.6. In this type of evaporator, the coils are either welded on one side of a plate or between the two plates which are welded together at the edges. The plate evaporators are generally used in house-hold refrigerators, home freezers, beverage coolers, ice cream cabinets, locker plants etc.

4.3.4 Shell and Tube Evaporators

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The shell and tube evaporator, as shown in Fig. 4.16, is similar to a shell and tube condenser. It consists of a number of horizontal tubes enclosed in a cylindrical shell. The inlet and outlet headers with perforated metal tube sheets are connected at each end of the tubes.

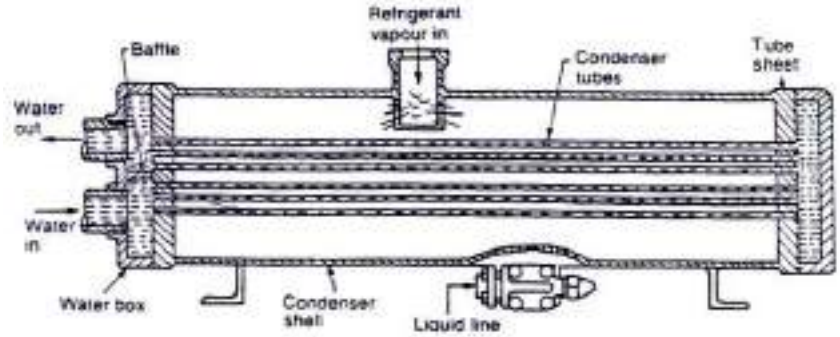

Fig. 4.16. Shell and tube evaporator.

These evaporators are generally used to chill water or brine solutions. When it is operated as a dry expansion evaporator, the refrigerant circulates through the tubes and the liquid to be cooled fills the space around the tubes within the shell. The dry expansion shell and tube evaporators are used for refrigerating units of 2 to 250 TR capacity. When it is operated as a flooded evaporator, the water or brine flows through the tubes and the refrigerant circulates around the tubes. The flooded shell and tube evaporators are used for refrigerating units of 10 to 5000 TR capacity.

4.27 Shell and Coil Evaporators

The shell and coil evaporators, as shown in Fig. 4.17, are generally dry expansion evaporators to chill water. The cooling coil is a continuous tube that can be in the form of a single or double spiral. The shell may be sealed or open. The sealed shells are usually found in shell and coil evaporators used to cool drinking water. The evaporators having flanged shells are often used to chill water in secondary refrigeration systems.

Another type of shell and coil evaporator is shown in Fig. 4.18 Both types of evaporators are usually used where small capacity (2 to 10 TR) liquid cooling is required. It may be noted that the shell and coil evaporator is restricted to operation above 5°C in order to prevent the freezing problems.

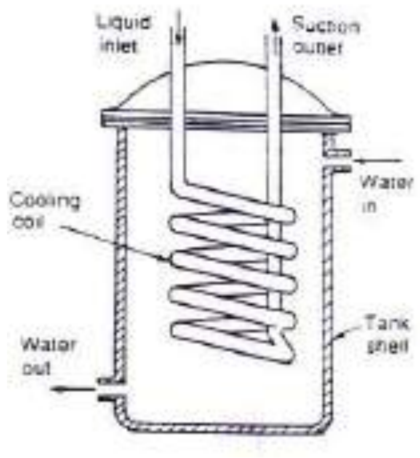

Fig. 4.17. Shell and coil evaporator.

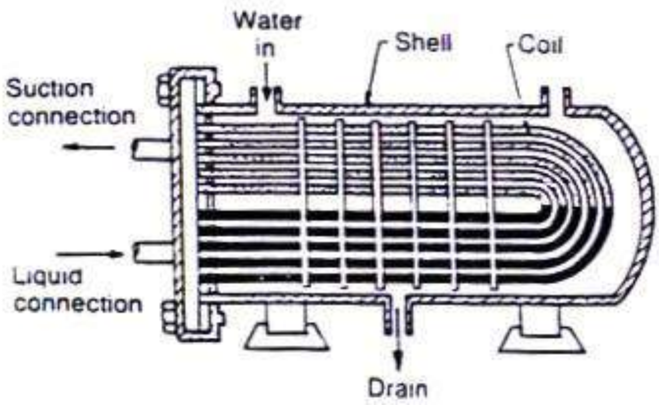

Fig. 4.18. Shell and coil evaporator.

4.28 Tube-in-Tube or Double Tube Evaporators

The tube-in-tube evaporator (or double tube evaporator) as shown in Fig. 4.19 consists

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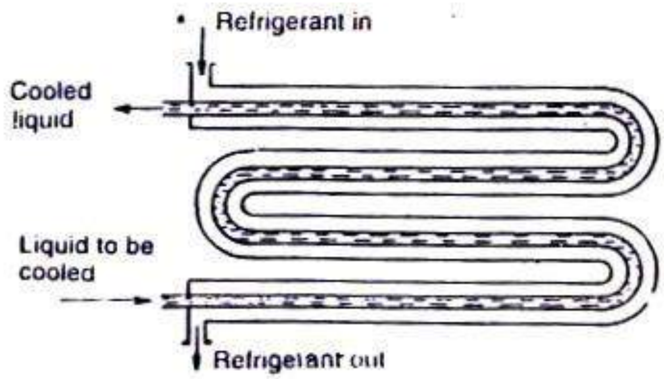

Fig. 4.19. Tube-in-tube or double tube evaporator.

of one tube inside another tube. The liquid to be cooled flows through the inner tube while the primary refrigerant or secondary refrigerant (i.e. water, air or brine) circulates in the space between the two tubes. The tube- in-tube evaporator provides high heat transfer rates. However, they require more space than shell and tube evaporators of the same capacity. These evaporators are used for wine cooling and in petroleum industry for chilling of oil.

4.29 Flooded Evaporators

In a flooded evaporator, as shown in Fig. 4.20, a constant liquid refrigerant level is always , maintained. A float control valve is used as an expansion device which maintains constant liquid level in the evaporator. The liquid refrigerant from the receiver passes through a low side float

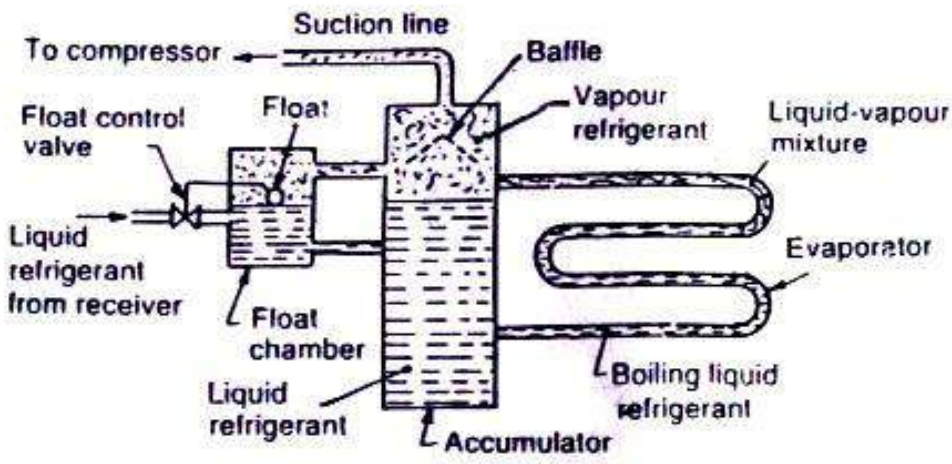

Fig. 4.11. Flooded evaporator.

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control valve and accumulator before entering the evaporator coil. The accumulator (also called a surge drum or surge tank) serves as a storage tank for the liquid refrigerant. It maintains a constant liquid level in the evaporator and helps to separate the liquid refrigerant from the vapour returning to the compressor. Due to the heat supplied by the substance to be cooled. the liquid refrigerant in the evaporator coil vaporises and thus the liquid level falls down. The accumulator supplies more liquid to the evaporator in order to keep the liquid refrigerant in the evaporator at proper level. In this way, the level of liquid refrigerant in the accumulator also falls down. Since the float within the float chamber rests on liquid refrigerant at the same level as that in the accumulator, therefore the float also falls down and open the float valve. Now the liquid refrigerant from the receiver is admitted into the accumulator. As the liquid level in the accumulator rises and reaches to the constant level, the float also rises with it until the float control valve closes.

Since the evaporator is almost completely filled with liquid refrigerant, therefore the vapour refrigerant from the evaporator is not superheated but it is in a saturated condition. In order to prevent liquid refrigerant to enter into the compressor, an accumulator is generally used with the flooded evaporators. The liquid refrigerant trapped in the accumulator is re-circulated through the evaporator. The evaporator coil is connected to the accumulator and the liquid flow from the accumulator to the evaporator coil is generally by gravity. The vapour formed by vaporising the liquid in the coil being lighter, rises up and passes on to the top of the accumulator from where it is supplied to the suction side of the compressor. The baffle plate arrests any liquid present in the vapour.

The advantage of the flooded evaporator is that the whole surface of the evaporator coil is in contact with the liquid refrigerant under all the load conditions. Thus, it gives high heat transfer rates (i.e., more efficient cooling) than a dry expansion evaporator of the same size. However, the flooded evaporator is more expensive to operate because it requires more refrigerant charge.

The flooded evaporators have many industrial applications, especially in the chemical and food processing industries. These evaporators used in comfort and process air cooling installations may of the bare coil type or finned type. Another type of the flooded evaporator is the plate evaporator which is found in cold storage boxes and freezers

4.30 Dry Expansion Evaporators

The dry expansion evaporators are not really dry at all. They simply use relatively little refrigerant as compared to flooded evaporators having the same coil volume. The dry expansion evaporators are usually only one-fourth to one-third filled with liquid refrigerant. A simple bare-tube dry expansion evaporator is shown in Fig. 4.21. The finned coil dry expansion evaporators are also available.

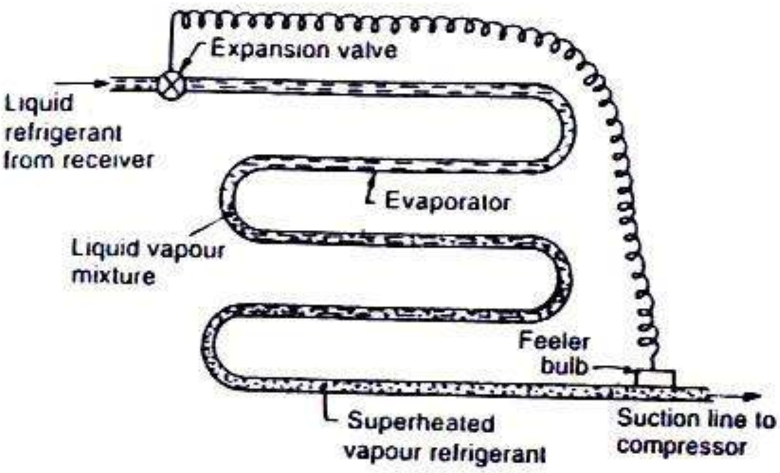

Fig. 4.21 Dry expansion evaporator.

In dry expansion evaporators, the liquid refrigerant from the receiver is fed by the expansion valve to the evaporator coil. The expansion valve controls the rate of flow of liquid refrigerant in such a way that all the liquid refrigerant is vaporised by the time it reaches at the end of the evaporator coils or the suction line to the compressor. The vapour is also superheated to a limited extent. It may be noted that in a dry expansion system, the refrigerant does not recirculate within the evaporator as in flooded type evaporator.

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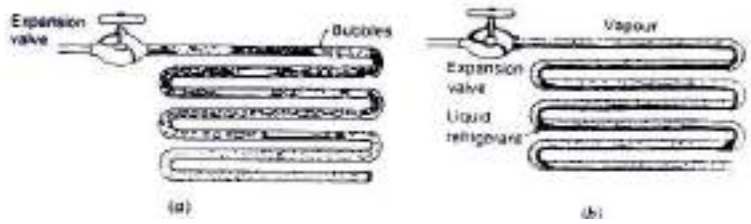

Fig. 4.22

The rate at which the liquid refrigerant is fed to the evaporator generally depends upon the rate of vaporisation and increases or decreases as the load on the evaporator increases or decreases. When the cooling load on the evaporator is light, the quantity of liquid refrigerant in the evaporator is small. We know that when liquid refrigerant passes through the expansion valve, some vapour (or flash gas) is formed. The flash gas causes bubbles in the evaporator. As more refrigerant vaporises, the bubbles get larger. If the coil diameter is small, the bubbles can cause dry areas on the interior walls of the coil, as shown in Fig. 4.22 (a). These dry areas reduce the rate of heat transfer. Thus, the evaporator efficiency decreases as dry areas increase, i.e., when the load on the evaporator is light. If the cooling load on the evaporator is heavy, the expansion valve allows a larger quantity of liquid refrigerant into the evaporator coil in order to accommodate the heavy load. In this case, the liquid and vapour separate. The liquid refrigerant flows along the bottom of the coil and vapour rises towards the top as shown in Fig. 4.22 (b). Thus, when the evaporator operates in this way, its efficiency is greatest. However, this efficiency depends upon the diameter of evaporator tubes, quantity of refrigerant in the evaporator and the velocity of the liquid refrigerant within the evaporator coil.

Note: Since t medium to be cooled comes in direct contact with the evaporator surfaces, in flooded and dry expansion evaporators, therefore they are called as direct expansion evaporators.

4.31 Natural Convection Evaporators

The natural convection evaporators are used where low air velocity and minimum hydration of the product is desired. The domestic refrigerators, water coolers and small freezers have natural convection evaporators. The circulation of air in a domestic refrigerator by natural convection is shown in Fig. 4.23 The evaporator coil should be placed as high as possible in the refrigerator because the cold air falls down as it leaves the evaporator.

(a) Air circulation without baffles.

(b) Air circulation with baffles.

Fig. 4.23. Circulation of air in natural convection evaporator.

The velocity of air over the evaporator coil considerably affects the capacity. In natural convection, the velocity of air depends upon the temperature difference between the evaporator and the space to be cooled. When the temperature difference, using a natural convection evaporator, is low (less than 8° C), the velocity of air is too low for satisfactory circulation. A lower temperature difference than 8°C may even cause slime on certain products such as meat or poultry. On the other hand, too great a temperature difference causes excessive dehydration.

(a) Air circulation with incorrect shape of coil.

(b) Air circulation with correct shape of coil

Fig 4.24

The circulation of air around the coil depends upon its size, shape and location. A small compact coil in a large box will cause only a small portion of air in the box to come in contact with the coil. The remainder of the air in the box will then be cooled by induced currents install of by direct contact. This will result in large variations of temperature in various parts of the box. The effect on air distribution with incorrect shape of coil is shown in Fig. 11.15. The coil should occupy at least 2/3rd of the width of the path of the air circulation and 3/4th the length of box. The natural convection can be improved by the use of baffles as shown in Fig. 4.23 and 4.24 These are simply sheet metal plates which guide the ascending and descending air currents in their proper channels.

4.32 Forced Convection Evaporators

In forced convection evaporators, the air is forced over the refrigerant cooled coils and fins. This is done by a fan driven by an electric motor. The fins are provided to increase the heat transfer rate.

The forced convection evaporators are more efficient than natural convection evaporators because they require less cooling surface and high evaporator pressures can be used which save considerable power input to the compressor. These types of evaporators are suited for air cooling units as well as for refrigerator cabinets used to store bottled beverages or foods in sealed containers.

The forced circulation air cooling units may be divided into the following three groups according to the velocity of air leaving the unit.

1. *Low velocity units*. These units have a discharge air rate from 60 m/min to 90 m/min. The low velocity cooling units are used in comfort air conditioning where low noise and low air velocity rates are needed. Both centrifugal and propeller type fans are used with low velocity cooling units.

2. *Medium velocity cooling units*. These units have an exit velocity of air from 150 m/min to 240 m/min. They are frequently used in refrigerators and freezers where drafts and noise are not a problem. The propeller fans are usually the source of air circulation in these units.

3. *High velocity cooling units*. These units have a discharge air rate above 240 m/min. They are used principally in blast freezers in special product refrigerators requiring quick pull-down of temperature. The high velocity cooling units usually use centrifugal fans as the source of air circulation.

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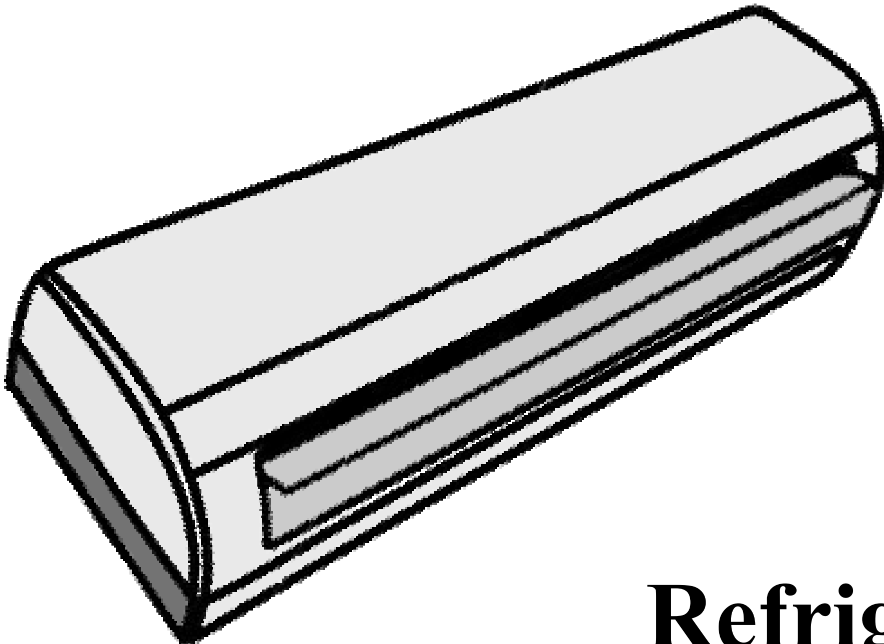
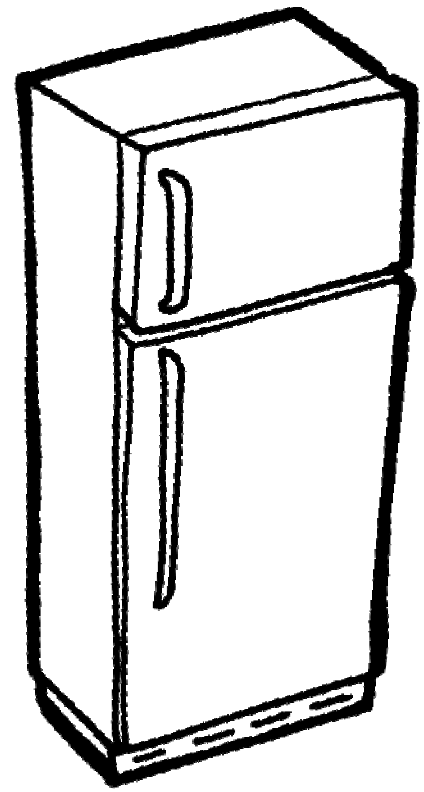


Refrigeration & air conditioning



SYLLABUS

Chapter:1

AIR REFRIGERATION CYCLE

1.1 Definition –refrigeration effect & Units of refrigeration.

1.2 Definition of COP, refrigeration effect (R.E)

Principle and working of

open air system refrigeration.

closed air system refrigeration.

Calculation of COP of Bell-Coleman cycle and numerical on it.

DEFINITIONS

REFRIGERATION:

It is a method of reducing the temperature of system below that of the surroundings and maintain it at the lower temperature by continuously abstracting the heat from it.

AIR CONDITIONING:

Providing a cool indoor atmosphere at all times regardless of weather conditions needed either for human comfort or industrial purposes by artificially cooling, humidifying or dehumidifying ,cleaning and recirculation the surrounding air is called air conditioning.

CONCEPTS OF REFRIGERATION

- Heat transfer is possible from a high temperature region to a low temperature region.
- Heat transfer is possible from a lower temperature system to higher temperature surroundings by some external means as per the 2nd law of thermodynamics.
- The working fluid changes from vapour phase to liquid phase after heat rejection and from liquid phase to vapour phase after heat absorption.
- The change of phase of the working fluid from liquid phase to vapour phase results in cooling effect.

- # Unit of Refrigerating:

The practical unit of refrigeration is expressed in terms of "tone of refrigeration".One tone of refrigeration is defined as the amount of refrigeration effect produced by the uniform melting of one tonne (1000kg) of ice from and at 00C in 24 hours. $\frac{Q}{W}$

Since the latent heat of ice is 335kj/kg, therefore one tonne of refrigeration,

$$1TR = 1000 \text{ x } 335 \text{ kj in 24 hours}$$

$$= 232.6 \text{ kj/min}$$

- **Co-efficient of Performance of a Refrigerator (C.O.P.):**

The co-efficient of performance is the ratio of heat extracted in the refrigerator to the work done on the refrigerant.

- Mathematically C.O.P. $=\frac{Q}{W}$

- **Examples:1** *Find the C.O.P of a refrigeration system if the work input is 80kj/kg and refrigeration effect produced is 160kj/kg of refrigerant following.*

$$w = 80kj / kg$$

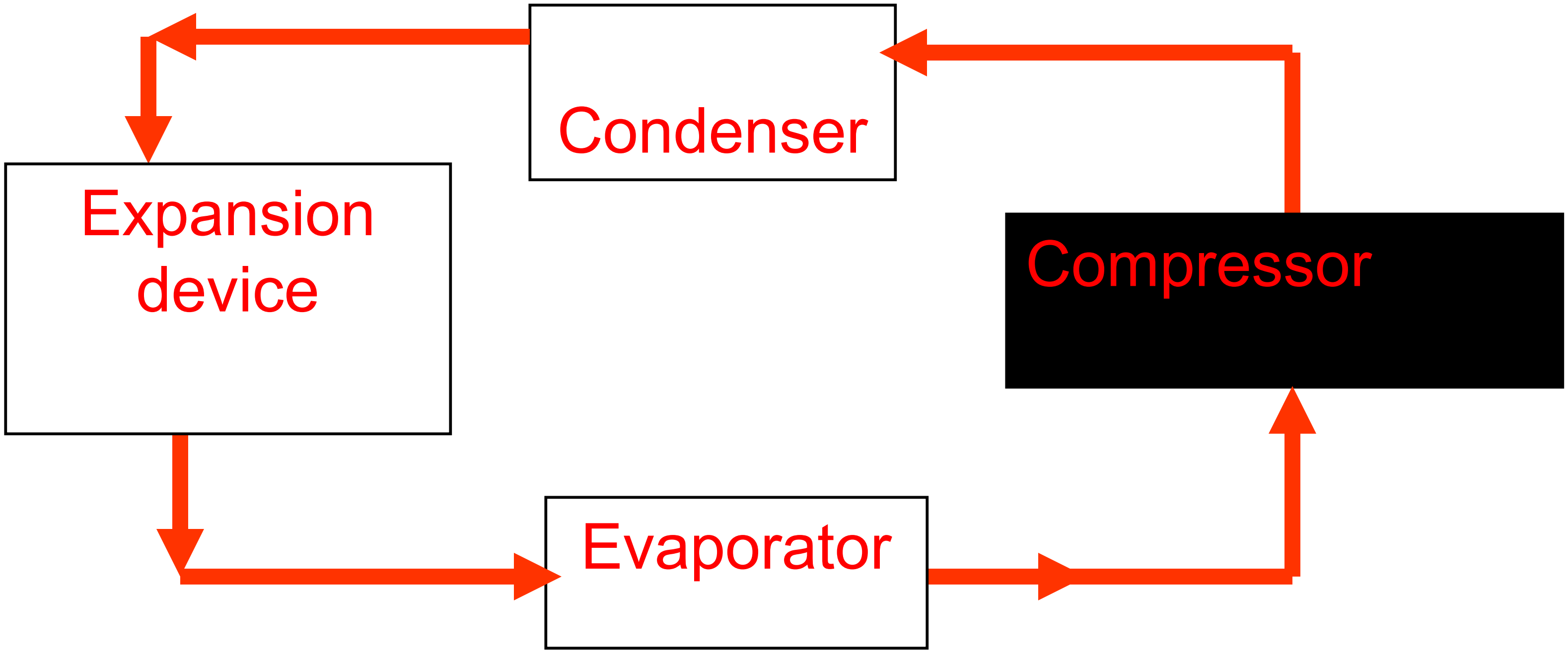
$$q = 160kj / kg$$

- *Solution: Given.*

We know that C.O.P. of a refrigeration system = $\frac{q}{w}$

$$\frac{160}{80} = 2$$

PARTS OF A REFRIGERATOR



1. COMPRESSOR (OR) PUMP:

To compress and circulate the low temperature and low pressure working fluid into high temperature and high-pressure vapour.

They are power absorbing mechanical devices and need input power. An electrical motor supplies power to these drives.

2. EVAPORATOR

It has cooling coils arranged in form of U – tubes.

The function of the evaporator is to reduce the temperature of the refrigerator cabinet.

The low temperature two phase mixture of refrigerant passing through the evaporator coils absorbs heat from the cabinet and changes into vapour phase.

This effect of cooling is also known as refrigerating effect

3. CONDENSER

Condenser consists of a series of coils in the form of U – tubes.

The high pressure, high temperature refrigerant from the compressor enters condenser

Where the refrigerant rejects its heat to the surrounding atmosphere.

The latent heat of the refrigerant is given to the surrounding atmosphere, which results in change of phase of the refrigerant.

4. EXPANSION VALVE:

The high pressure and temperature liquid refrigerant expands in the expansion valve to low pressure & low temperature two-phase mixture.

The temperature of the refrigerant drops in the expansion valve due to partial evaporation

REFRIGERATION EFFECT

In a refrigeration system ,the rate at which the heat is absorbed in a cycle from the interior space to be cooled is called refrigerating effect.

Capacity of refrigeration system is expressed in ton of refrigeration

- **Heat Engine**

✓ In heat engine the heat supplied to the engine is converted into useful work. If Q_2 is the heat supplied to the engine and Q_1 is the heat rejected from the engine, the net work done by the engine is given by:

$$\eta_E = \left(C.O.P\right)_E = \frac{W_E}{Q_E} = \frac{workdone}{HeatSupplied} = \frac{Q_2 - Q_1}{Q_2}$$

- **Refrigerator**

✓ Refrigerator is a reversed heat engine which either cool or maintain the temperature of a body (T_1) lower than the atmospheric temperature (Ta). This is done by extracting the Heat from a cold body and delivering it to a hot body (Q_2).

✓ In doing so, work W_R is required to be done on the system. According to First law of thermodynamics, $W_R = Q_2 - Q_1$

✓ The performance of a refrigerator is expressed by the ratio of amount of heat taken from the cold body (Q_1) to the amount of work required to be done on the system (W_R).

✓ This ratio is called coefficient of performance. Mathematically, coefficient of performance of a refrigerator,

- **Heat Pump**

✓ A refrigerator used for cooling in summer can be used as a heat pump for heating in winter. In the similar way, as discussed for refrigerator, we have W_p =- $Q_2 - Q_1$

✓ The performance of a heat pump is expressed by the ratio of the amount of the heat delivered to the hot body *(Q2)* to the amount of work required to be done on the system (Wp).

✓ This ratio is called coefficient of performance or energy performance ratio (E.P.R.) of a heat pump. Mathematically, coefficient of performance or **energy performance ratio** of a heat pump,

$$(C.O.P)_p = \frac{Q_2}{W_p} = \frac{Q_2}{Q_2 - Q_1} = \frac{Q_1}{Q_2 - Q_1} + 1 = (C.O.P)_R + 1$$

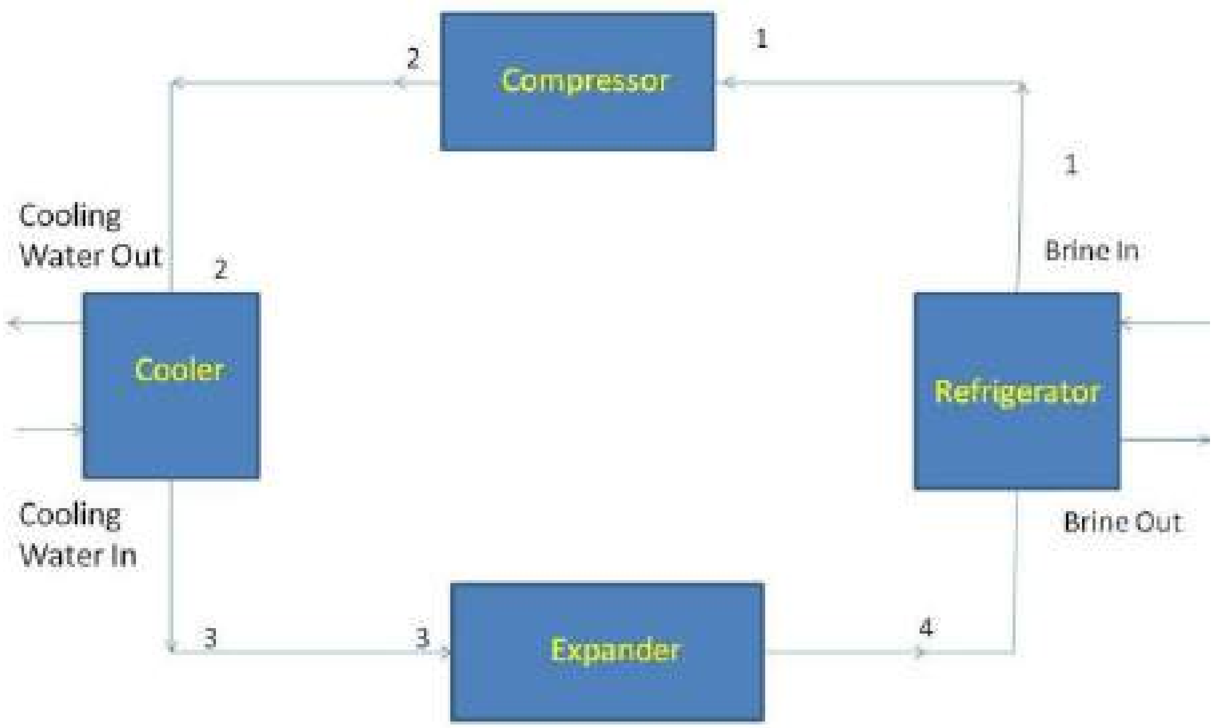
Refrigeration-Air Cycles-Open and Closed

TWO WAYS OF OPERATING OF BELL COLEMAN CYCLE

- Two Ways of Operating Of Bell Coleman Cycle
- Open air refrigeration cycle
- Closed air refrigeration cycle or dense cycle

Air Refrigeration System Working On Bell-Coleman Cycle

Closed Bell Coleman Air Cycle



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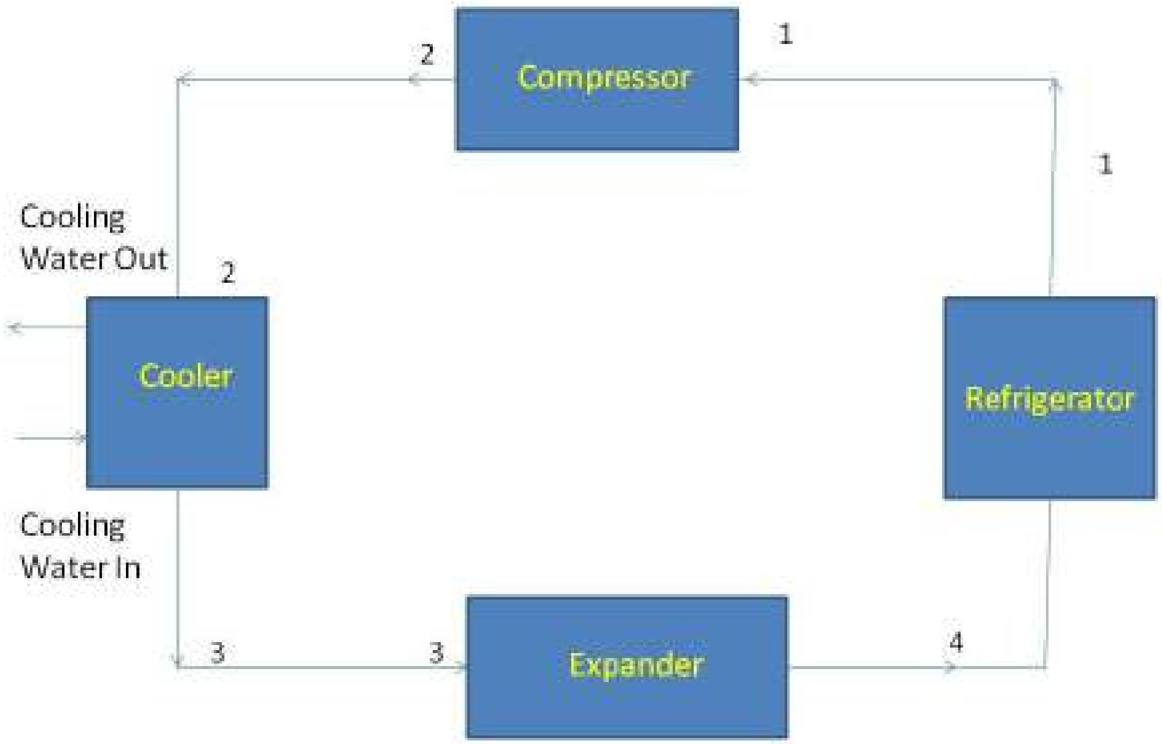
Air Refrigeration System Working On Bell-Coleman Cycle

Opened Bell Coleman Air Cycle



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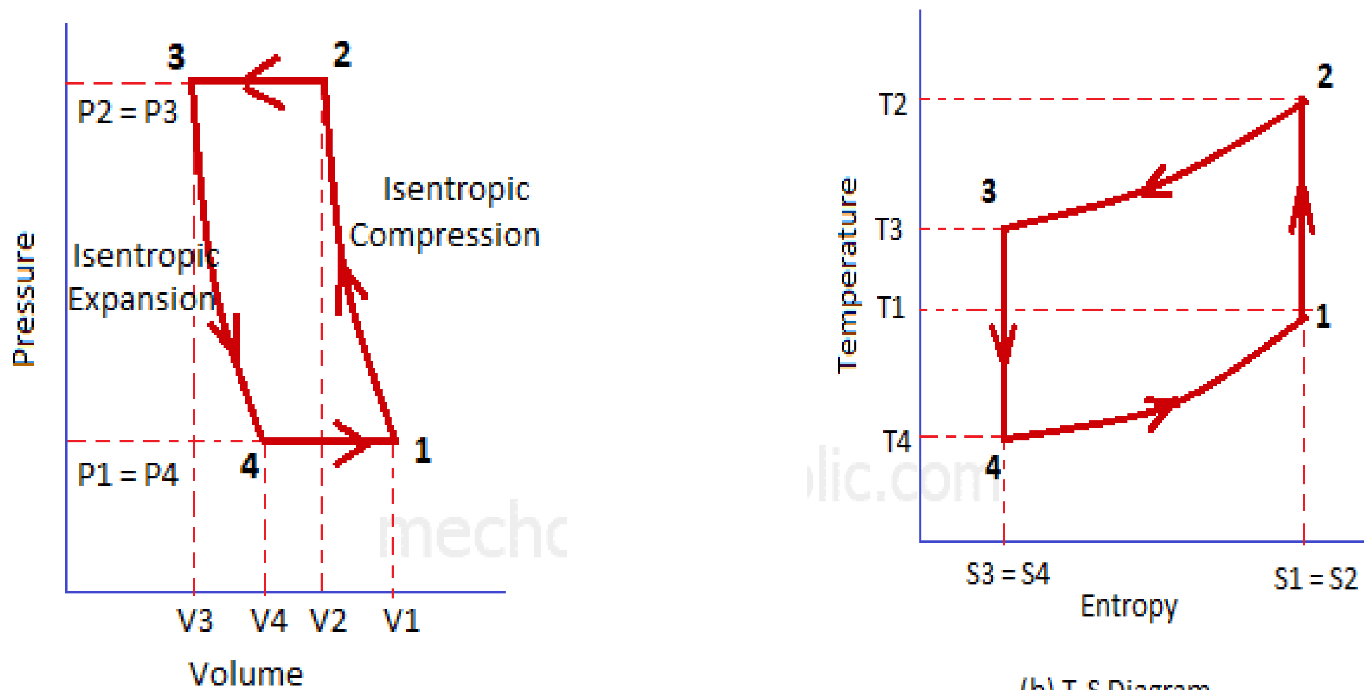
Working of Bell Coleman Cycle



(a) P-V Diagram

(b) T-S Diagram

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Bell Coleman Cycle process

- Isentropic Compression
- Isothermal compression /Steady Pressure Cooling Process: $q_R = q_{2-3} = c_p(\mathrm{T}_2 - \mathrm{T}_3)$
- Isentropic Expansion
- Isothermal expansion/Steady Pressure expansion process: $q_A = q_{4-1} = c_p(T_1 - T_4)$

$=q_A = Cp(T_1 - T_4)$

- Workdone during the cycle per kg of air =
  =Heat rejected – Heat absorbed

$$= q_R - q_A$$

$$= C_p(T_2 - T_3) - C_p(T_1 - T_4)$$

- Coefficient of Performance of Bell Coleman Cycle

- C.O.P during the cycle per kg of air $= \dfrac{Heat\ absorbed}{workdone}$

$$= \frac{q_A}{q_R - q_A} = \frac{C_p(T_1 - T_4)}{C_p(T_2 - T_3) - C_p(T_1 - T_4)}$$

$$= \frac{(T_1 - T_4)}{(T_2 - T_3) - (T_1 - T_4)} = \frac{T_4\left(\dfrac{T_1}{T_4} - 1\right)}{T_3\left(\dfrac{T_2}{T_3} - 1\right) - T_4\left(\dfrac{T_1}{T_4} - 1\right)}$$

- Process 1-2 $$\frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}}$$

- Process 3-4, $$\frac{T_3}{T_4} = \left(\frac{P_3}{P_4}\right)^{\frac{\gamma-1}{\gamma}}$$

- Since ,P_2=P_3 & P_1=P_4,so

$$\frac{T_2}{T_1} = \frac{T_3}{T_4} \quad or \quad \frac{T_2}{T_3} = \frac{T_1}{T_4}$$

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$$C.O.P. = \frac{T_4}{T_3 - T_4} = \frac{1}{\frac{T_3}{T_4} - 1}$$

$$= \frac{1}{\left(\frac{P_3}{P_4}\right)^{\frac{\gamma-1}{\gamma}} - 1} = \frac{1}{\left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}} - 1}$$

$$C.O.P. = \frac{1}{\left(r_p\right)^{\frac{\gamma-1}{\gamma}} - 1}$$

$$r_p = Compression\ or Expansion\ ratio = \frac{P_2}{P_1} = \frac{P_3}{P_4}$$

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- *A refrigeration plant working on the Bell Coleman Cycle, air is compressed to 6 bar from 1 bar. Its starting temperature is 15 °C. After compression air is cooled to up to 25 °C in a cooler before expanding back to 1 bar. Determine the C.O.P of the plant and net refrigerating effect.*

C_p = 1.005 kJ/kg K and C_v= 0.718 kJ/kg K.

Given: $P_2 = P_3 = 6$ bar, $P1 = P4 = 1$ bar

$$T_1 = 15 + 273 = 288\ K$$

$$T_3 = 25 + 273 = 298\ K$$

$$\gamma = Cp / Cv = 1.005/0.718 = 1.4$$

$$\gamma - 1 = 0.4$$

$$\text{So } \gamma - 1/\gamma = 0.286$$

Therefore

$$T_2/T_1 = (P_2 / P_1)\gamma - 1/\gamma$$

$$\text{So } T_2/T_1 = 1.669$$

Similarly for process 3-4

- $T_4 = \dfrac{T_3}{1.669} = T_4 = 178.55;$
- C.O.P of the cycle is given by, C.O.P $= \dfrac{T_4}{T_3 - T_4}$
- C.O.P $= \dfrac{178.55}{298 - 178.55}$ C.O.P = 1.494
- Net refrigerating effect = Cp ($T_1 - T_4$)
- = 1.005(288 – 178.55)
- = 109.99 kJ/kg

Example 2.11. *In a refrigeration plant working on Bell Coleman cycle, air is compressed to 5 bar from 1 bar. Its initial temperature is 10°C. After compression, the air is cooled up to 20°C in a cooler before expanding back to a pressure of 1 bar. Determine the theoretical C.O.P. of the plant and net refrigerating effect. Take* c_p = 1.005 kJ/kg *K* and c_v = *0.718* kJ/kg *K.*