Refrigeration & Air Conditioning. (R&AC)

> Refrigeration:

Definition: It is the process of cooling & maintaining the temp. of a substance below that of surrounding atmosphere.

The application of refrigeration are numerous. Application of R&AC play an important role in the development country.

> History:

> In 1748; William Coolen of Glasgow university produced refrigeration by creating partial vacuum over "ethyl ether." But he could not implement in practice.

> In 1834; Parkin's proposed a hand-operated compressor m/c working on ether.

> In 1851; Gorrie's by "air". &

> In 1856; Linde by "Ammonia."

> In 1877, Pictet, Advances in cryogenics - A field of very low temp. refrigeration were registered with the liquefaction of Oxygen.

The working substance being used in ~~R&~~AC is R22. In refrigerators R12 has been used in the year before 2000.

R12 is a CFC (chloro-fluoro carbon). Because of ozone-layer depletion, alternative's such as the following are being used instead of R12.

(i) Refrigerant 290 (or) R290 → Propane (C_3H_8)

(ii) R134a → Tetra-Fluoroethane ($C_2H_2F_4$)

(iii) R600a → Isobutane (C_4H_{10})

> Refrigerators & Heat Pumps:

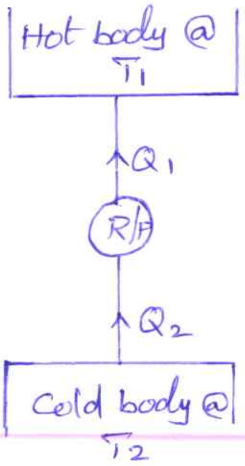
These are m/c's used to extract heat from a body @ low temp, & then rejects this heat to a body @ high temp.

If the purpose of the m/c is to cool some space/room, the m/c is known as "Refrigerator".

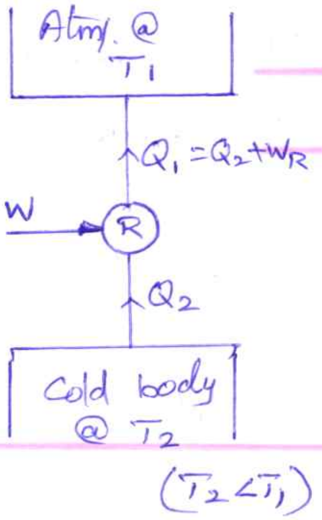
If the purpose of the m/c is to heat/warm some space/room, the m/c is known as "Heat Pump".

Acc. to II law of Thermodynamics ; by clausius statement.

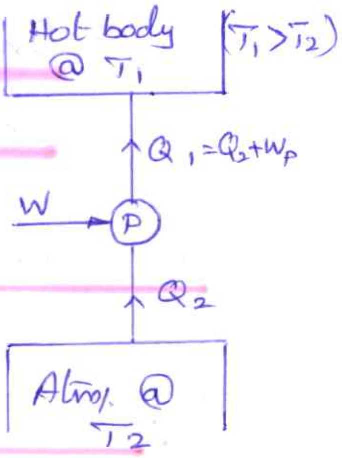
"It is impossible for a self acting m/c, working in a cyclic process, to transfer heat from a body at a lower temp. to a body at a higher temp. without the aid of an external agency."



(a) Perpetual Motion M/c II kind (PMM - II)



(b) Refrigerator



(c) Heat Pump

> Application's of Refrigeration:

- Food Preservation,
- Ice plant,
- Air-Conditioning
- Production Industry
- Photographic Processing (21° to 25°c)
- Liquifaction of gases [Seperating & liquifying of gases & vapour's].
- Food freezing,
- Water cooler's,
- Textile Industry,
- Medical field's

Types of Refrigerators:

- Ice Refrigerator. Ice is kept in a tray @ the top of insulated cabinet. Cold air flows downward from ice compartment & cool the substances kept below in the cabinet.
- Air Refrigerators: Air is used as working substances.
- Dry ice Refrigerators: Dry ice is solid CO_2. It changes directly from solid to vapour & does not go into liquid state. Dry ice blocks are kept around the substance to be cooled.
- Vapour Refrigerators: Vapour's like Ammonia, Freon's, CO_2, SO_2 etc are used as working substances in this type.
  - → Vapour Compression Refrigerators &
  - → Vapour Absorption Refrigerators.

Special Ref. System:
- Adsorption/Cascade/Mixed/Vortex tube/Thermoelectric/Steam Jet.

Refrigeration Effect $\boxed{(N) : (or)\ R_n}$

It is defined as quality of heat extracted in a given time from cold body (or) space (evaporator).

Unit: kJ/sec.

Compressor Work input (W):

The work I/P to the compressor to get the desired refrigerating effect.

Unit: kJ/sec.

Unit of Refrigeration:

It is defined as the quantity of heat required to lower the temp. of 1 kg of water through 1°C in 1 sec. (ie) 4.1868 kJ/sec $\simeq$ 4.19 kW.

* Sp. heat of water, $C_{Pw} = 4.19$ kJ/kg.k.
Sp. heat of Ice, $C_{Pice} = 2.1$ kJ/kg.k.

Tonne of refrigeration [Capacity of refrigerating Plant]:

Unit of refrigeration is practically expressed in terms of 'Tonne of Refrigeration' (TR). (or) Cooling capacity.

Tonne of refrigeration is defined as, the quantity of heat extracted to freeze One tonne (1000 kg) of water from and @ 0°c into ice in a duration of 24 hrs.

$$TR = 336\ kJ/kg$$

{where; 336 kJ/kg → Latent heat of fusion of Ice. (or) Heat extracted from water into ice from & @ 0°c in 24hrs}

$$= \frac{336 \times 1000}{24}$$

$$TR = 14000\ kJ/hr$$

(or)

$$TR = 232.6\ kJ/min \quad (or) \quad 3.89\ kJ/sec.$$

[In actual Practice; TR = 210 kJ/min (or) 3.5 kJ/sec]

> **Co-efficient of Performance [C.O.P]**

It is defined as, the ratio of net refrigerating effect (N) to the work input (W).

(i.e) $C.O.P = \frac{N}{W}$

[COP is always > 1]

> **Relative C.O.P** $= \frac{\text{Actual c.o.p}}{\text{Theoretical c.o.p}}$

⇒ (Theoretical c.o.p ⇒ Values obtained by applying laws of Thermodynamics to the refrigeration cycle)

⇒ (Actual c.o.p ⇒ Value's actually measured during test.)

> **Power Required to Produce refrigeration:**

$$C.O.P = \frac{N}{W} \Rightarrow W = \frac{N}{C.O.P} = \frac{3.5}{C.O.P}\ kW$$

∴ Power required $= \frac{3.5}{C.O.P}$ kW per tonne of refrigeration.

(* 1 kW-hr = 3600 kJ)

Air Refrigeration System:

- It's one of the earliest methods of cooling developed.
- Low co-efficient of performance (C.O.P) & high operating cost.
- Applied to air-craft refrigeration system, where with low equipment weight.
- In this, throughout the cycle the refrigerant remains in gaseous state.

Types of Air-Refrigeration's Systems:

(i) closed system. [Dense Air System].
(ii) Open System.

Open system Air refrigeration cycle:

In this, the air is directly led to the space to be cooled, allowed to circulate thro' the cooler & then returned to the compressor to start another cycle. The pr. of operation in this system is limited to operation at atm. pr. in the refrigerator. Therefore, volume of air handled by the compressor & expander is large.

In this cycle, the moisture is regularly carried away by the air circulated thro' the cooled space. This leads to the formation of frost @ the end of expansion process & clog the line. Thus in an open system, a drier should be used.

Closed (Dense) System Air refrigeration cycle:

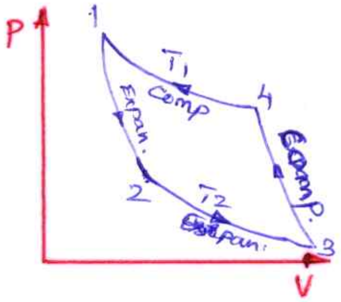
This system of air refrigerant is contained within the piping or components parts of the system @ all times & refrigerator with usually pr. above atm. pr.

Merits:-

It can work at a suction pr. higher than that of atm. pr., therefore the volume of air handled by the compressor & expander are smaller as compared to an open air refrigeration cycle system.

The operating pr. ratio can be reduced, which results in higher C.O.P.

> Air Refrigerator Working on Reversed carnot Cycle

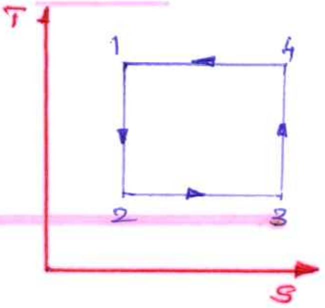


WKT;

Heat engine Working on Carnot Cycle has the highest possible efficiency.

III^{ly}; Refrigeration system working on reversed carnot cycle has the highest possible C.O.P.

In this cycle, using air as working medium (refrigerant).



Stages of the Cycle;

(1-2): Isentropic Expansion:

- No heat is absorbed (or) rejected by the air.

(2-3): Isothermal expansion:

- Heat absorbed by the air (heat extracted from the cold body) during isothermal expansion/kg of air.

$$Q_A = Q_{2-3} = T_2(S_3-S_2) = T_3(S_3-S_2)$$

(3-4): Isentropic Compression: No heat is absorbed (or) rejected by the air.

(4-1): Isothermal Compression:

- Heat rejected by the air during isothermal compression/kg of air.

$$Q_r = Q_{4-1} = T_1(S_4-S_1) = T_4(S_4-S_1)$$

$$= T_1(S_3-S_2) \quad (\because S_1 = S_2 ;\ S_4 = S_3)$$

WKT, W.D during the cycle/kg of air;

$$W_R = Q_r - Q_A = T_1(S_3-S_2) - T_3(S_3-S_2)$$

$$= (T_1-T_3)(S_3-S_2)$$

$$\therefore (C.O.P)_R = \frac{\text{Heat absorbed}}{\text{Workdone}} = \frac{Q_A}{W.D} = \frac{T_3(S_3-S_2)}{(T_1-T_3)(S_3-S_2)}$$

$$\Rightarrow \boxed{(C.O.P)_R = \frac{T_3}{T_1-T_3} = \frac{T_2}{T_1-T_2}}$$

For Heat Pump;

$$(C.O.P)_P = \frac{\text{Heat rejected}}{W.D} = \frac{Q_r}{W.D} = \frac{T_1(S_3-S_2)}{(T_1-T_3)(S_3-S_2)}$$

$$\Rightarrow (C.O.P)_P = \frac{T_1}{T_1-T_3} = \frac{T_1}{T_1-T_2}$$

$$\boxed{(C.O.P)_P = \frac{T_2}{T_1-T_2} + 1 = (C.O.P)_R + 1}$$

For Heat ~~Ef~~ Engine;

$$(C.O.P)_E = \eta_E = \frac{W.D}{Q_r} = \frac{(T_1-T_3)(S_3-S_2)}{T_1(S_3-S_2)}$$

$$\boxed{\eta_E = \frac{T_1-T_3}{T_1} = \frac{T_1-T_2}{T_1}}$$

Note:

In a refrigerating m/c, heat rejected (Q_r) is more than heat absorbed (Q_A). So; $W.D = Q_r - Q_A$

1) A m/c working on Carnot cycle operates b/w. 30°C & −15°C. Determine COP, when it is operated as, (a) Refrigerating m/c, (b) Heat Pump & (c) heat engine.

Soln:

Max. Temp, $T_1 = 30°C = 303\,K$

Min. Temp. $T_2 = -15°C = 258\,K$

(a) $(C.O.P)_R = \dfrac{T_2}{T_1-T_2} = \dfrac{258}{303-258} = 5.73$

(b) $(C.O.P)_P = \dfrac{T_1}{T_1-T_2} = \dfrac{303}{303-258} = 6.73$ (or) $(COP)_R + 1 = 6.73$

(c) $(C.O.P)_E = \eta_E = \dfrac{T_1-T_2}{T_1} = \dfrac{303-258}{303} = 0.1495 = 14.95\,\%$

2) A refrigerating m/c working on Reversed carnot cycle consumes 6kW for producing refrigerating effect of 1000 kJ/min for maintaining region @ $-40^\circ c$. Determine (i) cop of m/c, (ii) Higher temp. of cycle & (iii) Heat delivered in kJ/min, if the device is used as heat pump.

Soln: Power Input (or) Work input, $W = 6\,kW = 360\,kJ/min$

Refrigerating Effect, $N = 1000\,kJ/min.$

Min. Temp, $T_2 = -40^\circ c = 233\,k$

(i) $(C.O.P)_R = \frac{N}{W} = \frac{1000}{360} = \underline{\underline{2.778}}$

(ii) wkt; $(C.O.P)_R = \frac{T_2}{(T_1 - T_2)}$

$$\Rightarrow 2.778 = \frac{233}{T_1 - 233} \Rightarrow T_1 = 316.87\,k$$

$$T_1 = 43.87^\circ c$$

(iii) $(C.O.P)_P = \frac{T_1}{T_1 - T_2} = \frac{316.87}{316.87 - 233} = 3.778$

(or)

$(C.O.P)_P = (C.O.P)_R + 1 = \underline{\underline{3.778}}$

wkt; $(C.O.P)_P = 3.778 = \frac{\text{Heat Delivered}}{\text{WORK INPUT}}$

$\Rightarrow$ Heat Delivered, $Q_R = 3.778 \times 360$

$= \underline{\underline{1360\,kJ/min.}}$

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3) Find the least power of a perfect reversed heat engine that makes 450 kg of ice/hr @ $-3^\circ c$ from water @ $20.5^\circ c$. Assume latent heat of ice as 340 kJ/kg & $C_{P\,ice}$ as 2.1 kJ/kg.K.

Soln: Mass of ice, $m_i = 450\,kg/hr = 0.125\,kg/sec$

Latent heat of Ice, $L = 340\,kJ/kg$

$C_{P\,ice} = 2.1\,kJ/kg.K$

Max. Temp, $T_1 = 20.5^\circ c = 293.5\,k$

Min. Temp, $T_2 = -3^\circ c = 270\,k$

$$(C.O.P)_R = \frac{T_2}{T_1 - T_2} = \frac{270}{293.5 - 270} = \underline{\underline{11.49}}$$

Total heat to be extracted from water @ 20.5°c to produce ice @ −3°c, Refrigerating Effect, $N = Q_1 + Q_2 + Q_3$

where; Q_1 = heat absorbed (extracted) to cool water from 20.5°c to 0°c.

$$\Rightarrow Q_1 = m_i \cdot C_{Pw} (20.5 - 0)$$

$$= 0.125 \times 4.19 \times 20.5 = 10.737 \text{ KJ/sec.}$$

Q_2 = Latent heat extracted to freeze water @ 0°c.

$$= m_i \times L = 0.125 \times 340 = 42.5 \text{ KJ/sec}$$

Q_3 = heat absorbed to cool ice from 0°c to −3°c

$$= m_i \times C_{Pice} (0 - (-3))$$

$$= 0.125 \times 2.1 \times 3 = 0.75 \text{ KJ/sec}$$

$$\therefore N = Q_1 + Q_2 + Q_3 = 53.99 \text{ KJ/sec}$$

$$\Rightarrow COP = \frac{N}{W} \Rightarrow W = \frac{N}{COP} = \frac{53.99}{11.49} = 4.7 \text{ KJ/sec (kW)}$$

∴ Power Required, P = 4.7 kW

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4) A cold storage plant is required to store 20 Tonnes of fish. The temp. of the fish when supplied @ 25°c. storage temp. of fish reqd. @ −8°c. Sp heat of fish above freezing pt. = 2.93 kJ/kg°c. Sp. heat of fish below freezing Pt. = 1.25 kJ/kg°c. Freezing pt. of fish = −3°c. Latent heat of fish = 232 kJ/kg. If the cooling is achieved within 8 hrs, Find out; (i) Capacity of the plant, (ii) Carnot cycle COP, b/t. the temp. range. & (iii) If the actual COP is $\frac{1}{3}$rd of the carnot COP. Find out the Power reqd. to run the plant.

Soln: Heat removed in 8 hr's from each kg of Fish.

$$= [1 \times 2.93 (25 - (-3))] + [232] + [1 \times 1.25 \cdot (-3 - (-8))]$$

$$= 320.29 \text{ kJ/kg}$$

∴ Heat removed by the plant/min $= \frac{320.29 \times 20000}{8}$

$= \underline{\underline{800725 \text{ kJ/hr}}}$

∴ (i) Capacity of the Refrigerating plant:

$= \frac{800725}{14000} = \underline{\underline{57.2}}$ Tonnes of Refrigerant

(∵ 1 TR = 14000 kJ/hr)

$= \underline{\underline{57.2 \text{ TR}}}.$

(ii) COP of Reversed Carnot cycle:

$$(COP)_R = \frac{T_2}{T_1 - T_2} = \frac{265}{298-265} = \underline{\underline{8.03}} \quad \left\{ \begin{array}{l} T_1 = 25^\circ c = 298 k \\ T_2 = -8^\circ c = 265 k \end{array} \right\}$$

(iii) Power Reqd.:-

$$\text{Actual COP} = \frac{N}{W} \qquad \left[\because \text{Actual COP} = \frac{1}{3} \times \text{theo. COP} = \frac{1}{3} \times 8.03 = 2.67 \right]$$

$$\Rightarrow W = \frac{N}{2.67} = \frac{800725}{2.67}$$

$= 299897$ kJ/hr

Power Reqd to run the plant = 83.3 kJ/s (kW)

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5) A heat pump is used for heating the interior of a house in cold climate. The ambient temp. is $-5^\circ c$ & the desired interior temp. is $25^\circ c$. The compressor of heat pump is to be driven by a heat engine working bet. $1000^\circ c$ & $25^\circ c$. Treating both cycles as reversible. Calculate the ratio in which the heat pump & heat engine share the heating load.

Soln: $T_1 = 1000^\circ c = 1273 k$, $T_2 = 25^\circ c = 298 k$, $T_3 = -5^\circ c = 268 k$.

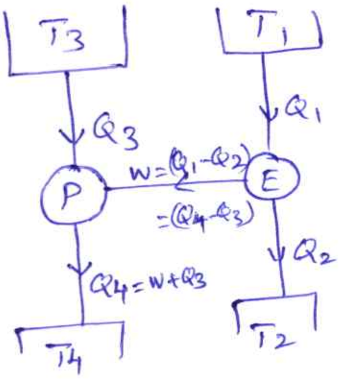
$T_4 = 25^\circ c = 298 k$.

The ratio in which the heat pump & heat engine share the heating load, $\frac{Q_4}{Q_1}$:

Since, Both the cycles are reversible,

$$\therefore \frac{Q_3}{Q_4} = \frac{T_3}{T_4} \quad \& \quad \frac{Q_2}{Q_1} = \frac{T_2}{T_1} = \frac{298}{1273}$$

$$\Rightarrow Q_3 = \frac{268}{298} \cdot Q_4$$



Heat engine drive the heat pump.

$$\therefore\ W = (Q_1 - Q_2) = (Q_4 - Q_3)$$

$$\div^{\text{by}}\ Q_1\ ;\ \Rightarrow 1 - \frac{Q_2}{Q_1} = \frac{Q_4 - Q_3}{Q_1}$$

$$\Rightarrow 1 - \frac{298}{1273} = \frac{Q_4 - \frac{268}{298} \cdot Q_4}{Q_1}$$

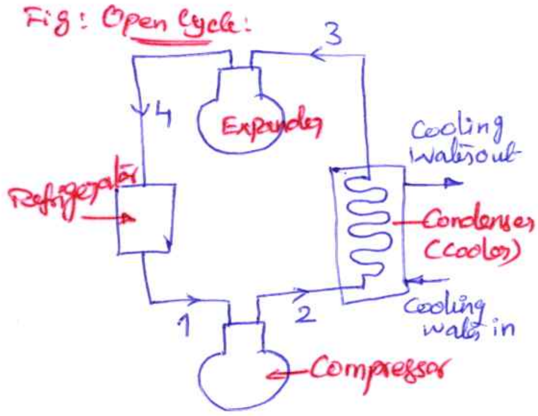
$$\Rightarrow \frac{Q_4}{Q_1} = \underline{\underline{7.61}}$$

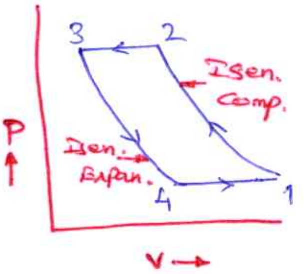
> Air Refrigerator Working on a Reversed Brayton cycle & ~~Reversed~~ (Joule's cycle / Bell-Coleman cycle).

It's a modification of Reversed Carnot cycle.

It was one of the earliest types of refrigerators used in ships carrying frozen meat. It consists of a compressor, cooler, expander & Refrigerator.

Fig: Open cycle:





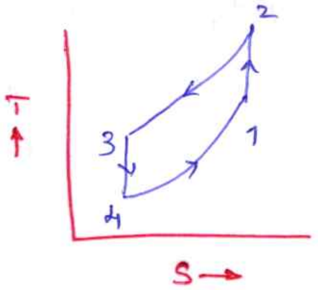


Fig: Closed cycle.

(1-2) - Isentropic Compression.

(2-3) - Cooling @ const. prs. ; $Q_R = Q_{2-3} = C_p(T_2 - T_3)$

(3-4) - Isentropic Expansion.

(4-1) - Expansion (heat Absorbed) @ const. prs. ; $Q_A = Q_{4-1} = C_p(T_1 - T_4)$

WkT;

W.D during the cycle/kg of air $= Q_R - Q_A = C_p(T_2 - T_3) - C_p(T_1 - T_4)$

$$\therefore\ \text{COP} = \frac{Q_A}{W.D} = \frac{C_p(T_1 - T_4)}{C_p(T_2 - T_3) - C_p(T_1 - T_4)}$$

$$\text{C.O.P.} = \frac{T_1 - T_4}{(T_2 - T_3) - (T_1 - T_4)}$$

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

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

$|||^{ly}$ (3-4); $\frac{T_3}{T_4} = \left(\frac{P_3}{P_4}\right)^{\frac{\gamma-1}{\gamma}} = \left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}}$ $(\because P_3 = P_2;\ P_4 = P_1)$

$$\therefore \frac{T_2}{T_1} = \frac{T_3}{T_4} \Rightarrow \frac{T_2}{T_3} = \frac{T_1}{T_4}$$

In Eq.①';

$$\therefore C.O.P = \frac{T_4}{T_3-T_4} = \frac{1}{\left(\frac{T_3}{T_4}\right)-1}$$

$$= \frac{1}{\left(\frac{P_3}{P_4}\right)^{\frac{\gamma-1}{\gamma}}-1} = \frac{1}{(r_p)^{\frac{\gamma-1}{\gamma}}-1} \longrightarrow ②' \left\{ r_p = \text{Comp. (or) Expa. ratio} = \frac{P_2}{P_1} = \frac{P_3}{P_4} \right\}$$

Sometimes, the compression & Expansion processes take place acc. to Polytropic process; $(pv^n = c)$;

WKT; W.D by the compressor during the process (1-2) / kg of air;

$$W_C = \frac{n}{n-1}(P_2V_2 - P_1V_1) = \frac{n}{n-1}(R.T_2 - R.T_1) \quad (\because PV = RT)$$

W.D by the expander during the process (3-4)/kg of air;

$$W_E = \frac{n}{n-1}(P_3V_3 - P_4V_4) = \frac{n}{n-1}(R.T_3 - R.T_4)$$

∴ Net W.D during the cycle/kg of air;

$$W = W_C - W_E = \frac{n}{n-1}.R\left[(T_2-T_1)-(T_3-T_4)\right]$$

WKT; Heat Absorbed during const. Pr. process (4-1);

$$Q_A = C_p(T_1 - T_4)$$

$$\therefore C.O.P = \frac{Q_A}{W} = \frac{C_p(T_1-T_4)}{\frac{n}{n-1}.R\left[(T_2-T_1)-(T_3-T_4)\right]} \longrightarrow ③'$$

WKT; $R = C_p - C_v = C_v(\gamma - 1)$

$$\therefore C.O.P = \frac{C_p(T_1-T_4)}{\frac{n}{n-1}.C_v(\gamma-1)\left[(T_2-T_1)-(T_3-T_4)\right]}$$

$$\Rightarrow \boxed{C.O.P = \frac{(T_1 - T_4)}{\frac{n}{n-1} \cdot \frac{\gamma - 1}{\gamma}\left[(T_2 - T_3) - (T_1 - T_4)\right]}} \qquad \left(\because \frac{C_P}{C_V} = \gamma\right)$$

Note: For; Isentropic compression or Expansion; $n = \gamma$.

$$\therefore \quad C.O.P = \frac{T_1 - T_4}{(T_2 - T_3) - (T_1 - T_4)}$$

> Merits & Demerits of Air-Refrigeration system:

Merits:

- Air is non-flammable.
- It is cheaper & easily available as compared to other refrigerants.
- Compare to other refrigents, weight of air refrigeration system per tonne of refrigeration is quite low. Bcoz of this reason it is employed in air-crafts.

Demerits:

- COP is very low in compare with other systems.
- weight of air reqd. is more.

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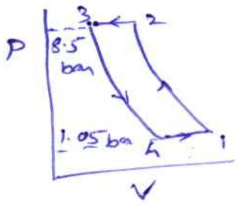
6) A refrigerator working on Bell-Coleman cycle operates b/w. Pr. limits of 1.05 bar & 8.5 bar. Air is drawn from the cold chamber @ 10°c, compressed & then it is cooled to 30°c before entering the expansion cylinder. The expansion & compression follows the law $Pv^{1.3} = c$. Determine the theoretical c.o.p of the system.

Soln: $P_1 = P_4 = 1.05$ bar, $P_2 = P_3 = 8.5$ bar. $T_1 = 10°c = 283K$, $P_3 = 30°c = 303K$, $n = 1.3$.

T_2 & T_4 = Temp. of air the end of compression & Expansion resp.

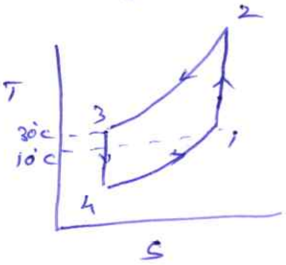


$$\Rightarrow \frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}} = \left(\frac{8.5}{1.05}\right)^{\frac{0.3}{1.3}} = 1.62$$

$$\therefore T_2 = T_1 \times 1.62 = 283 \times 1.62 = \underline{\underline{458.5K}}$$

III^ly;

$$\frac{T_3}{T_4} = \left(\frac{P_3}{P_4}\right)^{\frac{n-1}{n}} \Rightarrow T_4 = \frac{T_3}{1.62} = \frac{303}{1.62} = \underline{\underline{187K}}$$



∴ Wkt;

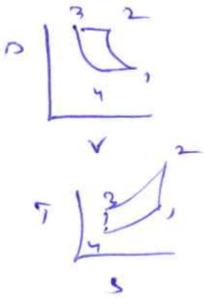
$$C.O.P = \frac{T_1 - T_4}{\frac{n}{n-1} \cdot \frac{\gamma-1}{\gamma}\left[(T_2-T_3)-(T_1-T_4)\right]}$$

$$= \frac{283-187}{\frac{1.3}{0.3} \times \frac{0.4}{1.4}\left[(458.5-303)-(283-187)\right]}$$

$$C.O.P = \underline{\underline{1.3}}$$

---

7) A dense air m/c operates on reversed Braylon cycle & is reqd. for a capacity of 10TR. The cooler pr. is 4.2 bar & the refrigerator pr. is 1.4 bar. The air is cooled in the cooler @ a temp. of 50°C & the temp. of air @ inlet to compressor is −20°C. Determine for the ideal cycle: (i) C.O.P, (ii) mass of air circulated/min., (iii) theoretical piston displacement of compressor, (iv) theor. piston displacement of expander & (v) Net power per tonne of refrigeration.



Soln: $Q = 10\,TR$., $P_2 = P_3 = 4.2\,bar$, $P_1 = P_4 = 1.4\,bar$,

$T_3 = 50°C = 323K$, $T_1 = -20°C = 253K$.

(i) C.O.P: $\frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}} = 1.37 \Rightarrow T_2 = 253 \times 1.37$

$T_2 = \underline{\underline{346K}}$

|||ly; $\frac{T_3}{T_4} = \left(\frac{P_3}{P_4}\right)^{\frac{\gamma-1}{\gamma}} = 1.37 \Rightarrow T_4 = \frac{T_3}{1.37} \Rightarrow T_4 = \underline{\underline{236K}}$

$$\therefore C.O.P = \frac{T_1 - T_4}{(T_2-T_3)-(T_1-T_4)} = \underline{\underline{2.83}}$$

(ii) Mass of Air/min:

Capacity of m/c is 10TR; ∴ Heat extracted/min = 10 × 210 = $\underline{\underline{2100\,kJ/min}}$

(∵ 1TR = 210 kJ/min)

Wkt; Heat extracted from the refrigerator/kg of air $= C_p(T_1 - T_4)$

$= 1(253-236)$

$= 17\,kJ/kg$.

∴ Mass of air circulated per min $= \frac{\text{Heat Extracted/min}}{\text{Heat extracted/kg}}$

$= \frac{2100}{17} = \underline{\underline{123.5\,kg/min}}$

(iii) Theoretical Piston Displacement of Compressor:

Let; $V_1 = \frac{m_a . R_a . T_1}{P_1} = \frac{123.5 \times 287 \times 253}{1.4 \times 10^5} = \underline{\underline{64\ m^3}}$

(iv) Theoretical Piston Displacement of Expander.

Let, $V_4 = \frac{m_a . R_a . T_4}{P_4} = \underline{\underline{60\ m^3}}$ (or) $\underline{\underline{(4-1)\ P=C}}$;

$\frac{V_4}{V_1} = \frac{T_4}{T_1}$

$\Rightarrow V_4 = \frac{64 \times 236}{253} = \underline{\underline{60\ m^3}}$

(v) Net Power / TR :

WKT; Net W.D/min $= \cancel{m_a} (Q_R - Q_A)$

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

$= m_a . C_p [(T_2 - T_3) - (T_1 - T_4)]$

$= 123.5 \times 1 [(346 - 323) - (253 - 236)]$

$= \underline{\underline{741}}\ KJ/min$

∴ Net Power of the m/c $= \frac{741}{60} = 12.35\ KJ/s\ (kW)$

& Net Power Per Tonne of Refrigeration $= \frac{12.35}{10} = \underline{\underline{1.235}}\ kW/TR$

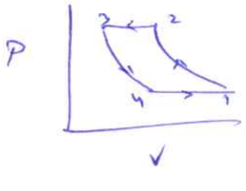
---

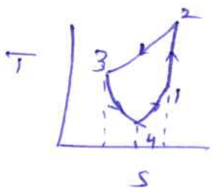
8) A dense air refrigeration system of 10 Tonnes capacity works b/w 4 bar & 16 bar. The air leaves the cold chamber @ 0°c & discharges @ 25°c to the expansion cylinder after air cooler. The expansion & compression cylinder are double acting. The η_{mech} of compressor 85% & expander 80%. The compressor speed is 250 rpm & has a stroke of 250mm. Determine: (i) COP. (ii) Power reqd/. Ep (iii) Bore of compression & expansion cylinder's. Assume; Isentropic Compression & Polytropic expansion. ~~Take PV~~ Take, $n = 1.25$.

Soln:- $Q = 10\ TR$, $P_1 = P_4 = 4$ bar, $P_2 = P_3 = 16$ bar. $T_1 = 0°c = 273k$,

$T_3 = 25°c = 298k$, $\eta_{mech.com.} - \eta_{mc} = 0.85$, $\eta_{me} = 0.8$, $N = 250$ rpm.

$L = 250\ mm = 0.25m$, $\gamma = 1.4$ & $n = 1.25$





& WKT; $\frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}} \Rightarrow T_2 = 405.7K$

|||ly; $\frac{T_3}{T_4} = \left(\frac{P_3}{P_4}\right)^{\frac{\gamma-1}{\gamma}} \Rightarrow T_4 = 225.7K$

(i) C.O.P:

(1-2); W.D by the compressor During $pv^{\gamma} = c$. / kg of air.

$$\therefore W_c = \frac{\gamma}{\gamma-1} \cdot R(T_2 - T_1) \times \frac{1}{\eta_{me}}$$

$$= \frac{1.4}{0.4} \times 0.287(405.7 - 273) \times \frac{1}{0.85}$$

$W_c = 156.8$ KJ/kg

|||ly (3-4); W.D by the expander during $PV^n = c$. / kg of air.

$$W_E = \frac{n}{n-1} \cdot R(T_3 - T_4) \cdot \eta_{me}$$

$$= \frac{1.25}{0.25} \times 0.287(298 - 225.7) \times 0.8$$

$W_E = 83$ KJ/kg

$\therefore$ Net W.D/kg of air, $W = W_c - W_E = 73.8$ KJ/kg

WKT; $Q_A = C_p(T_1 - T_4) = 1.005(273 - 225.7)$

$Q_A = 47.3$ KJ/kg.

$\therefore$ C.O.P $= \frac{Q_A}{W} = \frac{47.3}{73.8} = 0.641$

(ii) Power Reqd! Refrigerating Capacity = 10 TR = 10×210 = 2100 KJ/min

WKT; Refrigerating Effect = Q_A = 47.3 KJ/kg

$\therefore$ Mass of air circulated, $m_a = \frac{2100}{47.3} = 44.4$ kg/min.

& W.D/min $= m_a \cdot W = 44.4 \times 73.8 = 3277$ KJ/min.

$\therefore$ Power $= \frac{3277}{60} = 54.6$ kW

(iii) Bore of compression & Expansion cylinder:-

WKT; $P_1V_1 = m_a \cdot R \cdot T_1$

Piston Displacement of compressor, V_1 $= \frac{44.4 \times 287 \times 273}{4 \times 10^5}$

$= 8.7$ m³/min

$\Rightarrow V_1 = \left(\frac{\pi}{4} D_c^2 \cdot L\right) 2N$ (for Double Acting)

$\Rightarrow D_c = 0.3$ m

|||ly (4-1) P=c; $\frac{V_4}{T_4} = \frac{V_1}{T_1}$

Piston Displacement of Expander, V_4 $= 7.2$ m³/min

|||ly; $D_E = 0.27$ m

Refrigeration Needs of Air-Crafts:

In high-speed Passenger Aircrafts, Jet aircrafts & Missiles has introduced the need for compact & simple refrigeration system capable of high capacity.

- Passenger aircrafts requires → 8 TR capacity of cooling.
- Jet fighter's requires → 10 TR – 20 TR (∵ because of 950 KM/hr)

Method's of (Air-crafts) Air-Refrigeration system's:

- Simple Air Cooling System.
- Simple Air-Evaporative cooling system.
- Boot strap Air cooling system.
- Boot strap Air-Evaporative cooling system.
- Reduced Ambient Air-cooling system. &
- Regenerative Air cooling system.

① Simple Air-Cooling System:

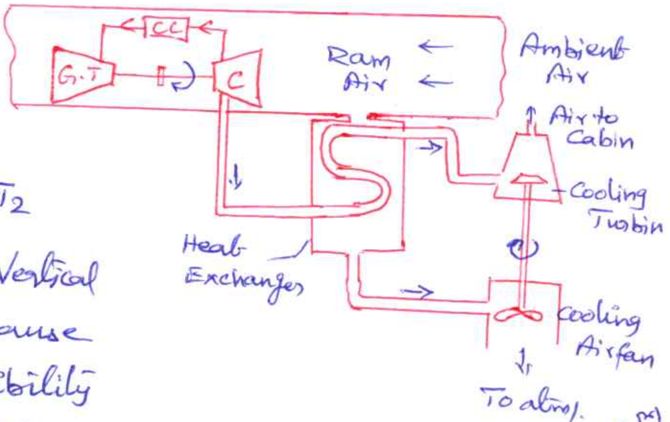
→ (1-2') Ramming Process:

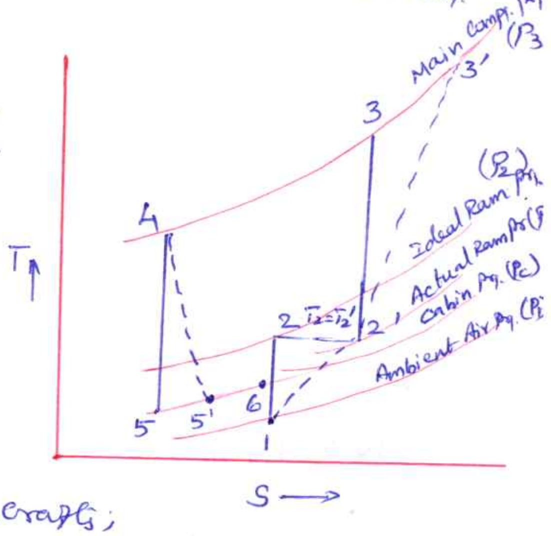
The ambient air rammed isentropically from P_1 & T_1 to P_2 & T_2

The process represented by (1-2) Vertical line. But in actual practice, because of internal friction, due to irreversibility which is adiabatic, not for isentropic. It represented by a curve (1-2').

Now, The Pr. & Temp. of rammed air P_2' & T_2'.

During ideal & actual ramming process, the total energy (or) enthalpy remains Constant. (ie) $h_2 = h_2'$ & $T_2 = T_2'$





If, the velocity of air-craft. V in m/s.

∴ The K.E of outside air relative to aircrafts;

$$K.E = \frac{V^2}{2} \text{ J/kg} = \frac{V^2}{2000} \text{ KJ/kg}$$

From energy eqn/. wkt; $h_2 - h_1 = \frac{V^2}{2000}$

$$C_pT_2 - C_pT_1 = \frac{V^2}{2000} \Rightarrow \frac{T_2}{T_1} = 1 + \frac{V^2}{2000.C_p.T_1} = \frac{T_2'}{T_1}$$

$$\therefore \frac{T_2}{T_1} = \frac{T_2'}{T_1} = 1 + \frac{V^2(\gamma-1)}{2000.\gamma.R.T_1} \qquad \left[\begin{array}{l} \text{wkt; } C_p - C_v = R \\ \therefore \quad C_p = \frac{\gamma.R}{\gamma-1} \end{array}\right]$$

where; R in KJ/kg.k

$$\therefore \frac{T_2}{T_1} = \frac{T_2'}{T_1} = 1 + \frac{V^2(\gamma-1)}{2.\gamma.R.T_1} = 1 + \frac{V^2(\gamma-1)}{2a^2}$$

where; a = Local Sonic velocity @ the ambient air Conditions.

$a = \sqrt{\gamma R T_1}$

$$\Rightarrow \boxed{\frac{T_2}{T_1} = \frac{T_2'}{T_1} = 1 + \frac{(\gamma-1)}{2}.M^2}$$ where; M = Mach No/. = $\frac{\text{Air Craft Velocity}}{\text{Local Sonic Velocity}}$

The temp/. $T_2 = T_2'$ is called stagnation temp/. of ambient air entry thro' the compressor. $\frac{P_2}{P_1} = \left(\frac{T_2}{T_1}\right)^{\frac{\gamma}{\gamma-1}}$

Ram Efficiency, $\eta_{ram} = \frac{\text{Actual pr. rise}}{\text{Isentropic pr. rise}} = \frac{P_2' - P_1}{P_2 - P_1}$

→ (2'-3') Compression Process:

Isentropic compression of air in the main compressor (2'-3). In actual practice, due to irreversibility (2'-3').

W.D during the compression process, $W_c = m_a C_p(T_3' - T_2')$

where; m_a = Mass of air bled from the main compressor for refrigeration Process.

→ (3'-4) Cooling Process: Compressed air is cooled by the ram air in the heat exchanger. Heat rejected in the heat exchanger during cooling; $Q_R = m_a.C_p(T_3' - T_4)$

→ (4-5') Expansion Process: The Cooled air now expanded isentropically in the cooling turbine (4-5). In actual Practice due to irreversibility (4-5').

W.D by the cooling turbine, $W_T = m_a.C_p(T_4 - T_5')$

The turbine work used to drive the cooling air fan.

→ (5'-6) Refrigeration Process: Air from the cooling turbine is sent to the cabin. The refrigeration effect produced (or) heat absorbed is given by, $Q_A = m_a \cdot C_p (T_6 - T_5')$

Where; T_6 - Inside temp. of Cabin, T_5' - Exit temp. of Cooling turbine.

$$C.O.P = \frac{\text{Refrigerating Effect}}{W.D} = \frac{m_a \cdot C_p (T_6 - T_5')}{m_a \cdot C_p (T_3' - T_2')} = \frac{T_6 - T_5'}{T_3' - T_2'}$$

If, 'Q' Tonnes of Refrigeration is the cooling load in the cabin,

$$Q = m_a \cdot C_p (T_6 - T_5')$$

$$\therefore m_a = \frac{210 \cdot Q}{C_p (T_6 - T_5')} \text{ kg/min}$$

$$\text{Power Reqd., } P = \frac{m_a \cdot C_p (T_3' - T_2')}{60} \text{ kW}$$

$$\& (C.O.P)_R = \frac{210 \cdot Q}{m_a \cdot C_p (T_3' - T_2')} = \frac{210 \cdot Q}{P \times 60}$$

Note: The ambient air temp. varies with the altitude of the flight of the aircraft. Generally, the temp. drops by 0.64°C per 100m of height from the sea level temp.

The running cost of air-cycle refrigeration system is not only the consideration for the selection of refrigeration system used in air-crafts. The following loads are considered also.

- Engine load (Alive load), W_E
- Fuel load (W_F) &
- Dead load (or) Pay load (W_P) ['Human load]

Total cost of the plant per TR;

$C_T = C_R + C_E + C_F$

C_R - Running Cost of the plant
C_E - Engine Investment
C_F - Fuel Cost.

A refrigerating plant used for air-crafts works most efficiently if C_T becomes minimum.

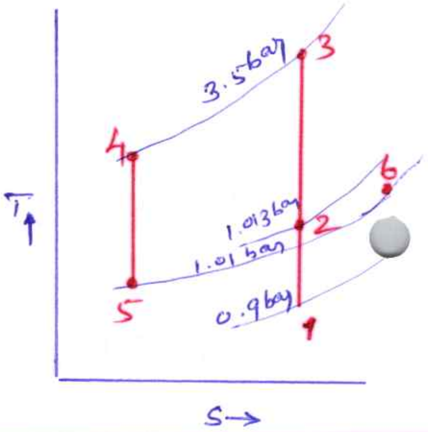
① A simple air-cooled system is used for an aeroplane having a load of 10T. The atm. pr. & temp. are 0.9 bar & 10°c resp. The pr. increases to 1.013 bar due to ramming. The temp. of air is reduced by 50°c in the heat exchanger. The pr. in the cabin is 1.01 bar & the temp. of air leaving the cabin is 25°c. Find,

(i) Power reqd. to take the load of cooling in the cabin, &

(ii) C.O.P of the system.

Assume that all the compression & Expansions are isentropic. The pr. of compressed air is 3.5 bar

Soln:



$Q = 10\,TR$, $P_1 = 0.9$ bar, $T_1 = 10°c = 283k$

$P_2 = 1.013$ bar, $P_5 = P_6 = 1.01$ bar, $(T_3 - T_4) = 50°c$

$T_6 = 25°c = 298K$, $P_3 = 3.5$ bar $= P_4$

(i) Power Reqd:

$$P = \frac{m_a . C_p (T_3 - T_2)}{60}$$

wkt;

$$\frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}} = \left(\frac{1.013}{0.9}\right)^{\frac{0.4}{1.4}} \Rightarrow T_2 = 292.6k$$

llly;

$$\frac{T_3}{T_2} = \left(\frac{P_3}{P_2}\right)^{\frac{\gamma-1}{\gamma}} \Rightarrow T_3 = 417K = 144°c$$

The temp. of air is reduced by 50°c in the heat Exchanger (given)

∴ temp. of air leaving the heat Exchanger, $T_4 = T_3 - 50 = 94°c = 367k$

wkt; The mass of air reqd. for the refrigeration purpose,

$$m_a = \frac{210\,Q}{C_p (T_6 - T_5)}$$

$$\therefore \frac{T_5}{T_4} = \left(\frac{P_5}{P_4}\right)^{\frac{\gamma-1}{\gamma}} \Rightarrow T_5 = 367 \times \left(\frac{1.01}{3.5}\right)^{\frac{0.4}{1.4}} \Rightarrow T_5 = 257k$$

$$\therefore m_a = \frac{210 \times 10}{1(298 - 257)} = 51.2\ kg/min$$

$$\therefore \text{Power, } P = \frac{51.2 \times 1 (417 - 292.6)}{60} = 106\ kW$$

(ii) C.O.P:

$$C.O.P = \frac{210 . Q}{P \times 60} = \frac{210 \times 10}{106 \times 60} = 0.33$$

②. An air-craft moving with speed of 1000km/hr uses simple gas refrg. cycle for air-conditioning. The ambient pr. & temp. are 0.35 bar & −10°c resp. The pr. ratio of compressor is 4.5. The heat exchanger effectiveness is 0.95. The isentropic η's of compressor & expander are 0.8 each. The cabin pr. & temp. are 1.06 bar & 25°c. Find temp's & pr's @ all points of the cycle. Also find the volume flow rate thro' compressor inlet & expander outlet for 100TR. Take, $C_p = 1.005$ kJ/kg.k ; $\gamma = 1.4$ & $R = 0.287$ kJ/kg.k.

Soln: $V = 1000$ km/hr $= 277.8$ m/s , $P_1 = 0.35$ bar, $T_1 = -10°c = 263$K

$\frac{P_3}{P_2} = 4.5$, $\eta_E = 0.95$, $\eta_c = \eta_T = 0.8$, $P_5 = P_5' = P_6 = 1.06$ bar,

$T_6 = 25°c = 298$ K , $Q = 100$ TR .

WKT; $$T_2 = T_1 + \frac{V^2}{2000 \cdot C_p} = 263 + \frac{277.8^2}{2000 \times 1.005} = \underline{\underline{301.4K}} \checkmark$$

& $$\frac{P_2}{P_1} = \left(\frac{T_2}{T_1}\right)^{\frac{\gamma}{\gamma-1}} \Rightarrow P_2 = 0.35 \times \left(\frac{301.4}{263}\right)^{\frac{1.4}{0.4}} = \underline{\underline{0.564 \text{ bar}}} \checkmark$$

Given; $$\frac{P_3}{P_2} = 4.5 \Rightarrow P_3 = 0.564 \times 4.5 = \underline{\underline{2.54 \text{ bar}}} \checkmark$$

(2-3); $$\frac{T_3}{T_2} = \left(\frac{P_3}{P_2}\right)^{\frac{\gamma-1}{\gamma}} \Rightarrow T_3 = 301.4 \times \left(\frac{2.54}{0.564}\right)^{\frac{0.4}{1.4}} = \underline{\underline{463.3K}} \checkmark$$

WKT; (2-3); $$\eta_{com.} = \frac{\text{Isentropic Temp rise}}{\text{Actual temp rise}} = \frac{T_3 - T_2}{T_3' - T_2'} \quad (\because T_2' = T_2)$$

$$\Rightarrow 0.8 = \frac{463.3 - 301.4}{T_3' - 301.4} \Rightarrow T_3' = \underline{\underline{503.8k}} \checkmark$$

Effectiveness of heat Exchanger; $$\eta_E = \frac{T_3' - T_4}{T_3' - T_2} \Rightarrow 0.95 = \frac{503.8 - T_4}{503.8 - 301.4}$$

$$\Rightarrow T_4 = \underline{\underline{311.5 K}} \checkmark$$

WKT; $P_3 = P_4 = \underline{\underline{2.54 \text{ bar}}} \checkmark$

(4-5); $$\eta_T = \frac{\text{Actual temp. rise}}{\text{Isentropic temp rise}} = \frac{T_4 - T_5'}{T_4 - T_5}$$

WKT; $$\frac{T_5}{T_4} = \left(\frac{P_5}{P_4}\right)^{\frac{\gamma-1}{\gamma}}$$

$$\Rightarrow T_5 = 311.5 \times \left(\frac{1.06}{2.54}\right)^{\frac{0.4}{1.4}}$$

$T_5 = \underline{243K} \checkmark$

$$\Rightarrow 0.8 = \frac{311.5 - T_5'}{311.5 - 243}$$

$$\Rightarrow T_5' = \underline{\underline{256.7K}} \checkmark$$

Volume Flow Rate:

WKT, $m_a = \frac{210Q}{c_p(T_6-T_5')} = \frac{210\times100}{1.005(298-256.7)} = 506$ kg/min

$P_2V_2 = m_a \cdot R \cdot T_2$

⇒ Volume flow rate thro' compressor inlet, $V_2 = \frac{506\times287\times301.4}{0.564\times10^5}$

$V_2 = 776$ m³/min ✓

|||ly;

Volume flow rate thro' Expander outlet, $V_5' = \frac{506\times287\times256.7}{1.06\times10^5}$

$V_5' = 351.7$ m³/min ✓

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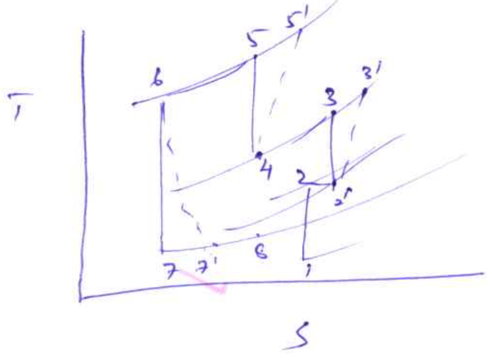
③ A boot-strap cooling system of 10 TR capacity is used in an aeroplane. The ambient air temp. & prs. are 20°c & 0.85 bar resp. The prs. of air increases from 0.85 bar to 1 bar due to ramming action of air. The prs. of air discharged from the main compressor is 3 bar. The discharge prs. of air from the auxiliary compressor is 4 bar. The $\eta_{isen.}$ compressor's (both) are 80%, while that of turbine is 85%. 50% of the enthalpy of air discharged from the main compressor is removed in the first heat exchanger & 30% of the enthalpy of air discharged from the auxiliary compressor is removed in the 2nd heat exchanger using rammed air. Assuming ramming action to be isentropic, the reqd. cabin prs. of 0.9 bar & temp. of air leaving the cabin not more than 20°c. Find: (i) the power reqd. to operate the system & (ii) $(COP)_R$.

Soln:- $Q = 10$ TR, $P_1 = 0.85$ bar, $T_1 = 20°c = 293K$, $P_2 = 1$ bar,

$P_3 = 3$ bar $(= P_3' = P_4)$, $P_5 = 4$ bar $(P_5 = P_5' = P_6)$, $P_8 = 0.9$ bar $(P_8 = P_7 = P_7')$, $T_8 = 20°c = 293K$,

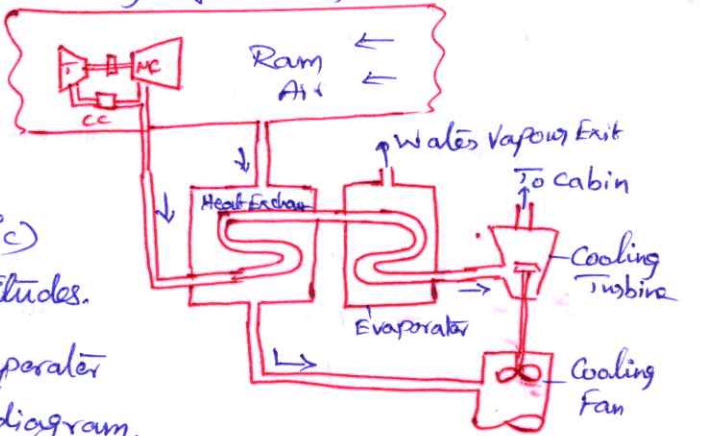
$\eta_{cI} = \eta_{c2} = 0.8$, $\eta_T = 0.85$,



(Continue in P.No: 13) →→

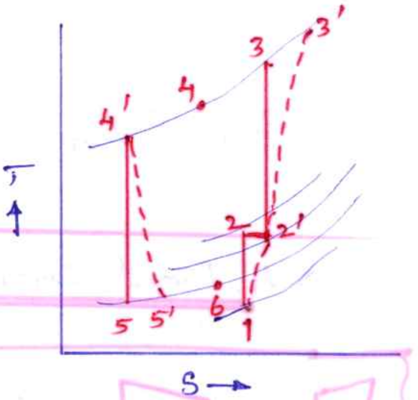
> 2. Simple Air-Evaporative Cooling System:

It's similar to the simple air cooling system & addition to that evaporator is placed in between heat exchanger & cooling turbine.



Generally, water (40°c), Alcohol (9°c) & Ammonia (-70°c) is boils at 20,000m high altitudes.

The cooling process in the evaporator as shown by (4-4') in T-S diagram.



$$m_a = \frac{210.Q}{C_p(T_6 - T_5')} \text{ kg/min}$$

$$P = \frac{m_a.C_p(T_3' - T_2')}{60} \text{ kW}$$

$$\& \ C.O.P = \frac{210.Q}{m_a.C_p(T_3' - T_2')} = \frac{210.Q}{P \times 60}$$

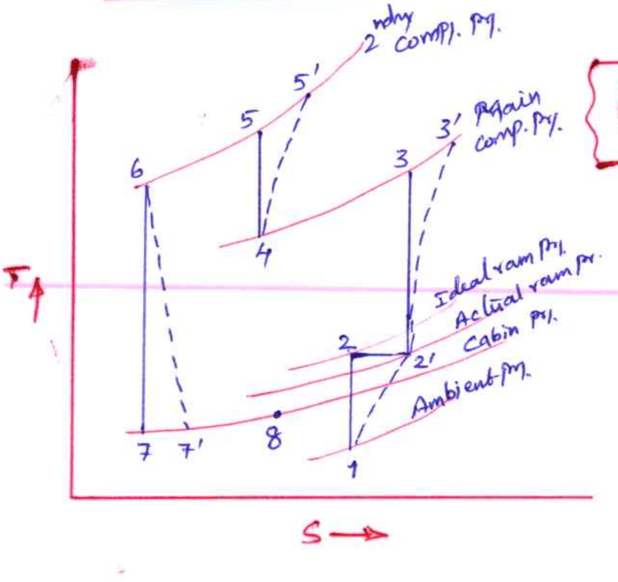
The initial mass of evaporant (m_e) reqd. to be carried for the given flight time is given by,

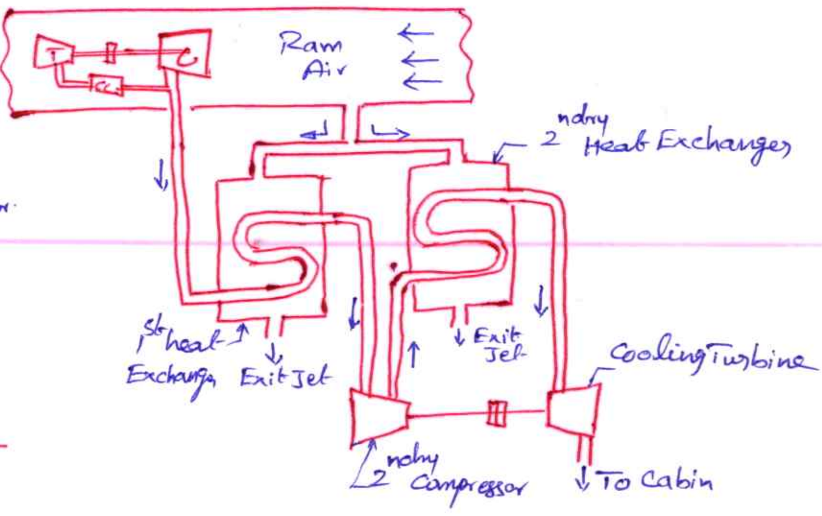
$$m_e = \frac{Q_e.t}{h_{fg}}$$

[∵ Q_e = heat to be removed in evaporation in (kJ/min)
t = Flight time in (min) &
h_{fg} = Latent heat of evaporation of evaporant in (kJ/kg)]

If cooling of ≤ 45 min, Simple air-evaporative cooling system is applicable.

> 3. Boot-Strap Air Cooling System:





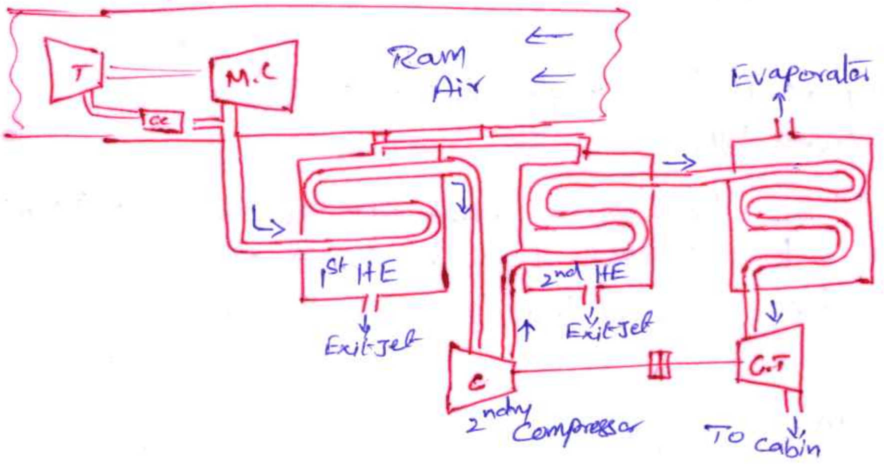
In this two heat exchanger instead of one & cooling turbine drives a secondary compressor. This type is mostly used in transport type aircrafts.

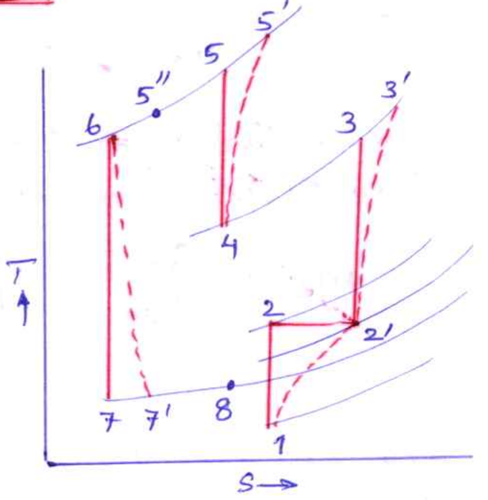
(1-2) – Ramming Action.
(2-3) – Compression in main compressor.
(3-4) – Cooling in primary heat exchanger.
(4-5) – Compression in secondary compressor.
(5-6) – Cooling in secondary heat exchanger.
(6-7) – Expansion in the cooling turbine.
(7-8) – Heat absorbed in cabin.

$$m_a = \frac{210 \cdot Q}{C_p(T_8 - T_7')} \text{ kg/min.} \qquad \text{Power, } P = \frac{m_a \cdot C_p(T_3' - T_2')}{60} \text{ kW}$$

$$\& \; (C.O.P)_R = \frac{210 \cdot Q}{m_a \cdot C_p(T_3' - T_2')} = \frac{210 \cdot Q}{P \times 60}$$

4. Boot-Strap Air-Evaporative Cooling System:



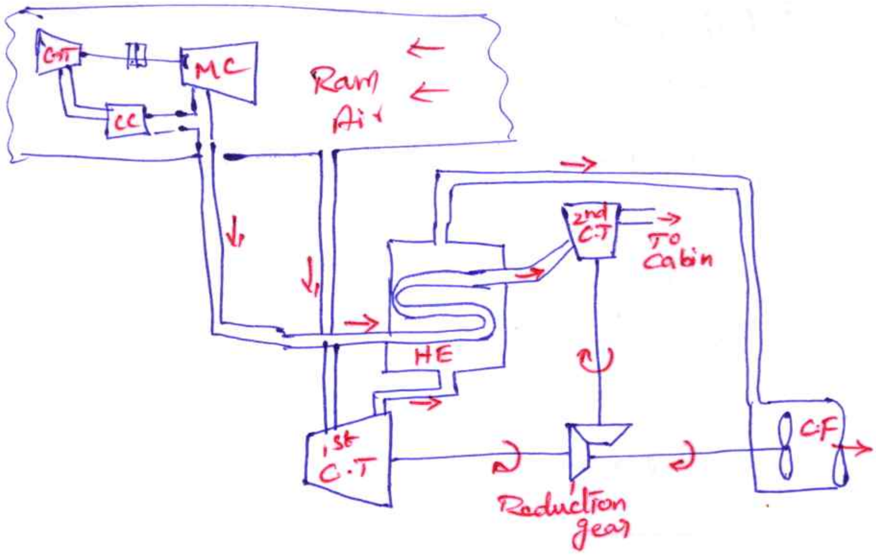


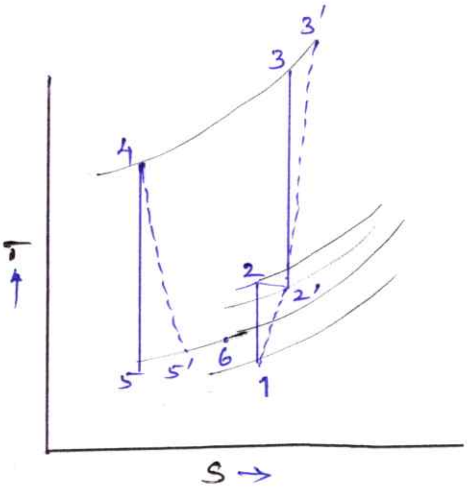
$$m_a = \frac{210 \cdot Q}{C_p(T_8 - T_7')} \text{ kg/min,}$$

$$\text{Power, } P = \frac{m_a \cdot C_p(T_3' - T_2')}{60} \text{ kW}$$

$$\& \; (C.O.P)_R = \frac{210 \cdot Q}{P \times 60}$$

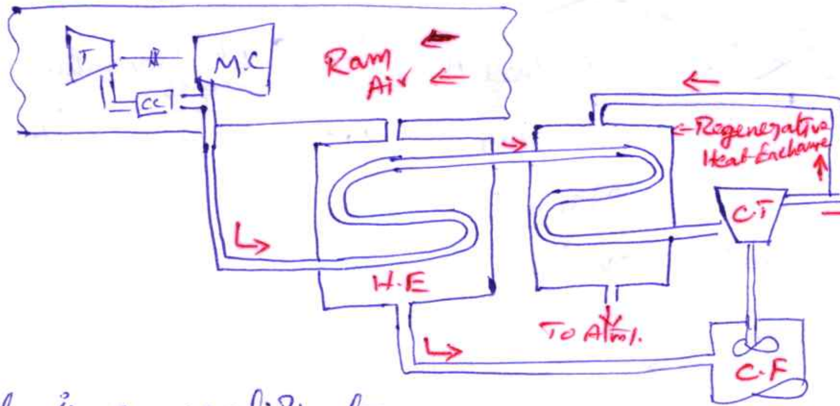
> 5. Reduced Ambient Air Cooling System:

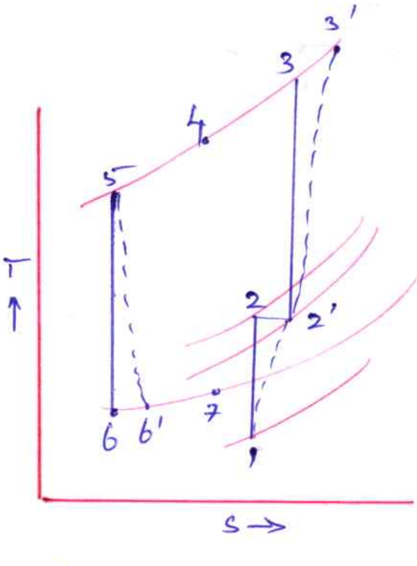




- It includes two cooling turbines & one heat exchanger.
- It is used for very high speed (Supersonic) aircrafts, when the ram temp. is too high.

> 6. Regenerative Air Cooling System:-





It is a modification of simple air cooling system with the addition of regenerative heat exchanger.

Air is cooled by the first heat exchanger by ram air & then further cooled in the regenerative heat exchanger with a portion of the air bled after expansion in the cooling turbine.

It is used in supersonic aircrafts & rockets.

$$m_a = \frac{210 \cdot Q}{C_p(T_7 - T_6')} \text{ kg/min.}$$

For energy balance of Regenerative heat Exchanger, we have;

$$m_2 \cdot C_p(T_8 - T_6') = m_1 \cdot C_p(T_4 - T_5)$$

$$\Rightarrow m_2 = \frac{m_1(T_4 - T_5)}{(T_8 - T_6')}$$

m_a = Mass of air bled from the C_1 to be used for regenerative cooling.

Let; m_1 = Total mass of air bled from the main compressor &

m_2 = Mass of cold air bled from the cooling turbine for regenerative heat exchanger.

T_8 = Temp. of air leaving to atm. from the regenerative heat exchanger.

$(\therefore m_a = m_1 - m_2)$

$$\text{Power, } P = \frac{m_1 \cdot C_p (T_3' - T_2')}{60} \text{ kW}$$

$$\& \ (COP)_R = \frac{210 \cdot Q}{m_1 \cdot C_p (T_3' - T_2')} = \frac{210 \cdot Q}{P \times 60}$$

Comparison of Various Air Cooling Systems Used for Air-Crafts:

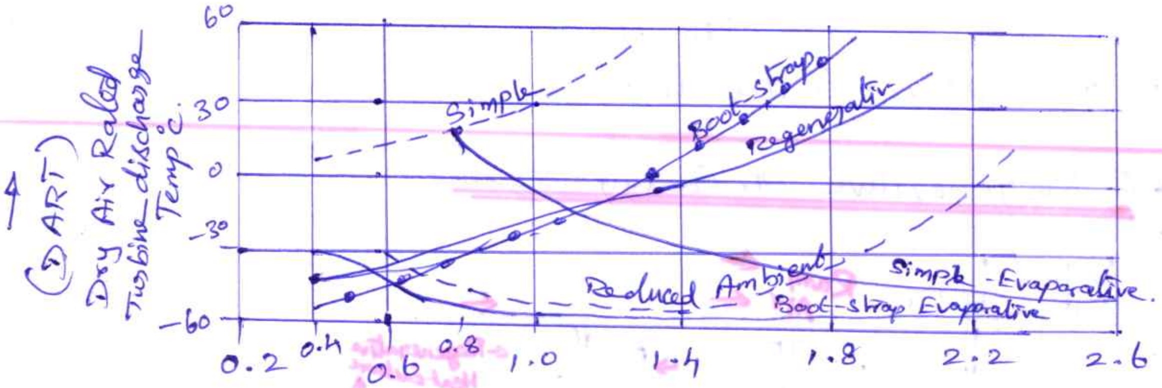


Fig: Performance curve's.

Problem: ③

WKT; (1-2); $\frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}} \Rightarrow T_2 = 306.8\text{ K} = 33.8°C = T_2'$

(2-3); $\frac{T_3}{T_2} = \left(\frac{P_3}{P_2}\right)^{\frac{\gamma-1}{\gamma}} \Rightarrow T_3 = 420.3\text{ K} = 147.3°C$

WKT; $\eta_{c_1} = \frac{\text{Isentropic temp rise}}{\text{Actual temp. rise}} = \frac{T_3 - T_2}{T_3' - T_2'} \Rightarrow 0.8 = \frac{420.3 - 306.8}{T_3' - 306.8}$

$\Rightarrow T_3' = 448.7\text{ K} = 175.7°C$

Since; 50% of enthalpy of air discharged from the C_1 in the first H Exchanger,

(3'-4); $T_4 = 0.5 \times 175.7 = 87.85°C = 360.85\text{ K}$

Now: (4-5); $\frac{T_5}{T_4} = \left(\frac{P_5}{P_4}\right)^{\frac{\gamma-1}{\gamma}} \Rightarrow T_5 = 391.5\text{ K} = 118.5°C$

WKT; $\eta_{c_2} = \frac{T_5 - T_4}{T_5' - T_4} \Rightarrow 0.8 = \frac{391.5 - 360.85}{T_5' - 360.85} \Rightarrow T_5' = 399.16\text{ K} = 126.16°C$

III^{ly} Since 30% of enthalpy of air removed from C_2 in the 2nd H Exchanger,

(5'-6): $T_6 = 0.7 \times 126.16 = 88.3°C = 361.3\text{ K}$

$\therefore$ (6-7): $\frac{T_7}{T_6} = \left(\frac{P_7}{P_6}\right)^{\frac{\gamma-1}{\gamma}} \Rightarrow T_7 = 236\,K = -37^\circ C$

we'; $\eta_T = \frac{\text{Actual temp, increases}}{\text{Isentropic temp, increases}} = \frac{T_6 - T_7'}{T_6 - T_7} \Rightarrow 0.85 = \frac{361.3 - T_7'}{361.3 - 236}$

$\Rightarrow T_7' = 254.8\,K$
$= -18.2^\circ C$

(ii) Power, $P = \frac{m_a . C_p (T_3' - T_2)}{60}$ $m_a = \frac{210 . Q}{C_p (T_8 - T_7')} = 55\,kg/min$

$\therefore P = \frac{55 \times 1 (448.7 - 306.8)}{60} = 130\,kW$

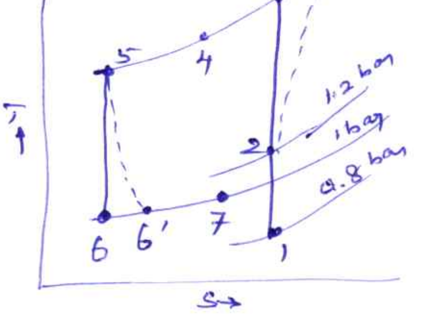
(iii) (C.O.P) $= \frac{210 . Q}{m_a . C_p (T_3' - T_2)} = 0.27$

---

④ A regenerative air cooling system is used for an Aeroplane to take 20TR load. The ambient air @ pr. & temp of 0.8 bar & 10°C resp. is rammed isentropically till the pr. rises to 1.2 bar. The air bled off the main compressor @ 4.5 bar is cooled by the ram air in the heat exchanger whose effectiveness is 60%. The air from the heat exchanger is further cooled to 60°C in the regenerative heat exchanger with a portion of the air bled after expansion in the cooling turbine. The cabin is to be maintained @ a temp. & pr. of 25°C & 1 bar resp. If the $\eta_{isen.}$ of the compressor & turbine are 90% & 80% resp. Find: (i) Mass of the air bled from cooling turbine to be used for regenerative cooling, (ii) Power, (iii) (COP) of the system. Assume the temp. of air leaving to atm. from the regenerative heat exchanger as 100°C.

Soln:- $Q = 20\,TR$, $P_1 = 0.8\,bar$, $T_1 = 10^\circ C = 283\,K$, $P_2 = 1.2\,bar$,
$P_3 = 4.5\,bar$, Effectiveness of HE, $\eta_{HE} = 0.6$, $T_5 = 60^\circ C = 333\,K$
$P_7 = P_6 = P_6' = 1\,bar$, $T_7 = 25^\circ C = 298\,K$, $\eta_c = 0.9$, $\eta_T = 0.8$, $T_8 = 100^\circ C = 373\,K$

(We ki;)
(1-2); $\frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}} \Rightarrow T_2 = 317.8\,K.$



(2-3); $T_3 = 464\,K$

$\eta_c = \frac{T_3 - T_2}{T_3' - T_2} \Rightarrow T_3' = 480\,K$

$\eta_H = \dfrac{T_3' - T_4}{T_3 - T_2} \Rightarrow T_4 = 382.7\,K$, $T_5 = 333\,K$ (given)

(5-6); $T_6 = 216\,K.$

$\eta_T = \dfrac{T_5 - T_6'}{T_5 - T_6} \Rightarrow T_6' = 239.4\,K$, $T_7 = 298\,K$ (given)

(i) Mass of air bled from the cooling turbine to be used for regenerative (m_2) cooling.

WKT; $m_a = m_1 - m_2 = \dfrac{210.Q}{C_p(T_7 - T_6')} = \dfrac{210(20)}{1(298 - 239.4)}$

$\Rightarrow m_a = m_1 - m_2 = 71.7$ kg/min

from energy eqn. ; $m_2 = \dfrac{m_1(T_4 - T_5)}{(T_8 - T_6')} = 0.372\,m_1$

WKT; $m_1 - m_2 = 71.7 \Rightarrow m_1 - 0.372\,m_1 = 71.7$

$\Rightarrow \underline{\underline{m_1 = 113.4\,\text{kg/min.}}}$

$\therefore m_2 = 0.372 \times 113.4 = \underline{\underline{42.2\,\text{kg/min.}}}$

(ii) Power, $P = \dfrac{m_1 . C_p(T_3' - T_2)}{60} = \dfrac{113.4 \times 1(480 - 317.8)}{60}$

$= \underline{\underline{307}}\,\text{kW}$

(iii) $(C.O.P)_R = \dfrac{210.Q}{P \times 60} = \underline{\underline{0.23}}$

Vapour Refrigeration.

Now adays it's universally used for all Purpose refrigeration. It is used from a small domestic unit of 0.5T capacity to 200T capacity.

Merits:
- > Smaller size for given capacity,
- > Less Running cost,..
- > COP is quite high....

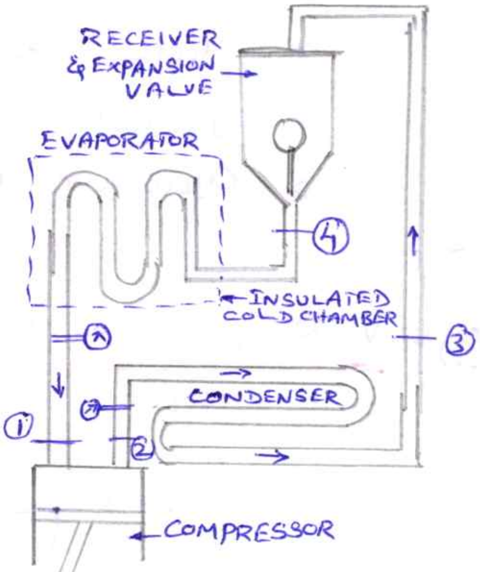
Demerits:
- > Greater safety,
- > Prevention of leaks,..
- > Inflammability & toxicity....

Vapour Compression Refrigeration System:-

VCR as compared to the air refrigeration systems is that, the vapour alternately undergoes a change of phase from vapour to liquid & liquid to vapour during the completion of a cycle. The latent heat of vaporization is utilised for carrying heat from the refrigerator which is quite high compared with the air-cycle, which depends only upon the sensible heat of the air.

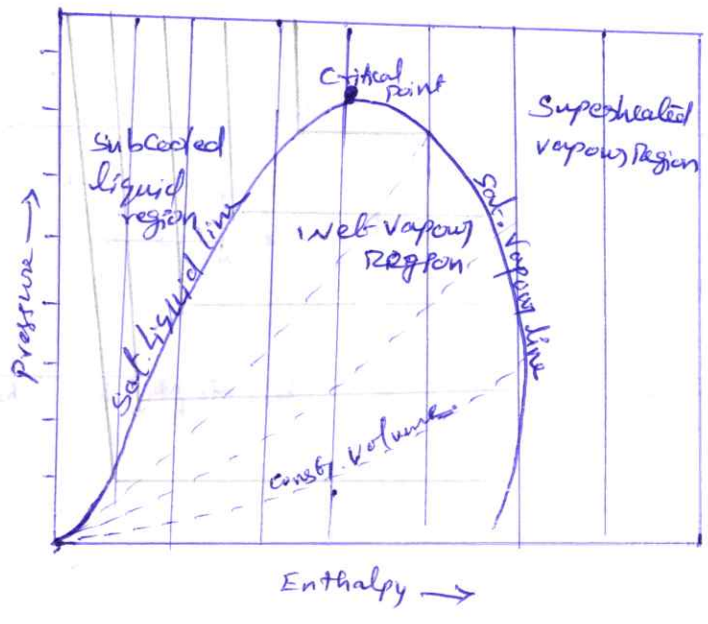
In this the various fundamental processes are completed in One Cycle.

1. Compression, — In Compressor.
2. Condensation. — In Condenser.
3. Receiving & Expansion &p — In Expansion valve
4. Vapourisation. — In Evaporator.



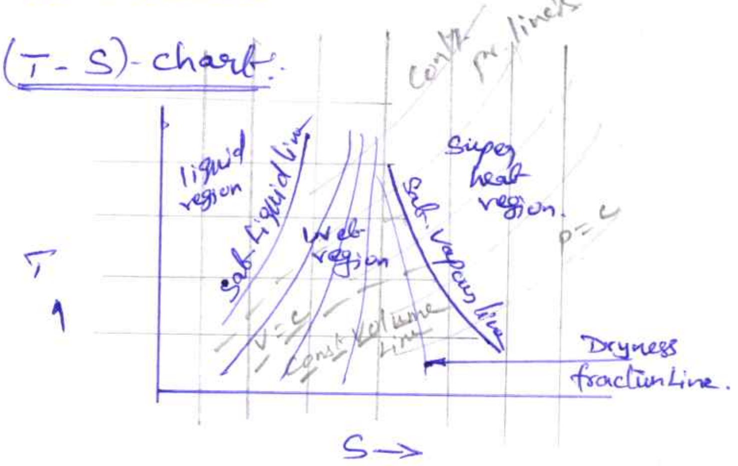
~~(1-2)~~ :

Pressure - Enthalpy (P-h) chart:



(p-h) chart: Co-ordinating the values of the fluid properties, pressure Vs heat content results in the construction of p-h diagram. A valuable instrument for solving design problems. This chart is recommended by the refrigeration committee of the "Institution of Mechanical Engineers". It can be constructed with the help of T-S diagram by changing the co-ordinates.

(T-S)-chart:



(h-S) chart:

The dgn, problem can be easily solved with the help of h-S diagram, it's similar manner as the Mollier dig.

Types of VCRS:

To increase the COP of the Cycle, there are many cycles as follows;

- > Cycle with dry saturated vapour after compression.
- > Cycle with Wet vapour " ".
- > cycle with Superheated vapour " "
- > Cycle with superheated vapour before ",
- > Cycle with Under (Sub) cooling of refrigerant.

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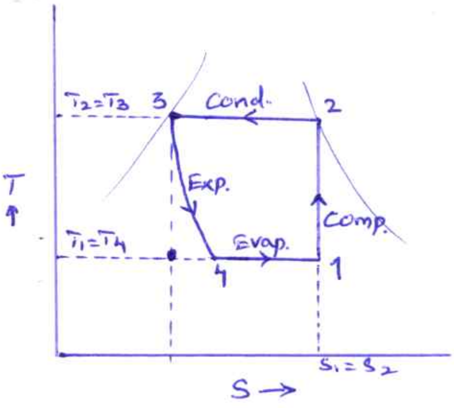
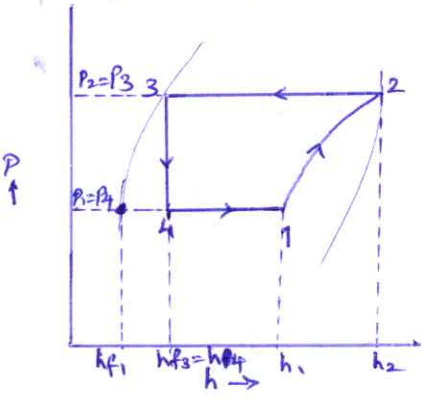
(1)> VCR Cycle with "Dry Saturated Vapour After Compression:





1. Compression process:

W.D during isentropic Compression / kg of refrigerant;

$$W = h_2 - h_1$$

2. Condensing process: – phase change @ $P = C$ & $T = C$.
3. Expansion Process: (Throttling – (Irreversible Process))
4. Vaporising process: During evaporation, the liquid-vapour refrigerant absorbs its latent heat of vaporisation from the medium which is to be cooled. This heat which is absorbed by the refrigerant is called Refrigerating effect.

$$R_E = h_1 - h_4 = h_1 - h_{f_3}$$

$$C.O.P = \frac{R_E}{W} = \frac{h_1 - h_4}{h_2 - h_1} = \frac{h_1 - h_{f_3}}{h_2 - h_1}$$

1.> A vapour compression refrigerator works b/w. the pr. limits of 60 bar & 25 bar. The working fluid is just dry at the end of compression & there is no under cooling of the liquid before the expansion valve. Determine: (i) cop of the cycle, & (ii) capacity of the refrigerator, if the fluid flow is at the rate of 5 kg/min.

Pr. (bar)	Saturation Temp. (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

Soln: $P_2 = P_3 = 60$ bar, $P_1 = P_4 = 25$ bar, $T_2 = T_3 = 295$ K,

$T_1 = T_4 = 261$ K, $h_{f_3} = h_4 = 151.96$ kJ/kg, $h_{f_1} = 56.32$ kJ/kg,

$h_{g_2} = h_2 = 293.29$ kJ/kg, $h_{g_1} = 322.58$ kJ/kg.K,

$S_{f_2} = 0.554$ kJ/kg.K [Extra data], $S_{f_1} = 0.226$ kJ/kg.K,

$S_{g_2} = S_2 = 1.0332$ kJ/kg.K, $S_{g_1} = 1.2464$ kJ/kg.K

Let; x_1 = Dryness fraction of the vapour refrigerant entering to the Compressor at ①.

WKT; entropy @ ①; $S_1 = S_{f_1} + x_1 . S_{fg_1} = S_{f_1} + x_1 (S_{g_1} - S_{f_1})$

$$\Rightarrow S_1 = 0.226 + x_1 (1.2464 - 0.226)$$

$$S_1 = 0.226 + 1.0204 x_1 \longrightarrow ①$$

$S_2 = S_{g_2} = 1.0332$ kJ/kg.K (given).

Since; $S_1 = S_2$ $\therefore 0.226 + 1.0204\, x_1 = 1.0332$

$\Rightarrow \underline{x_1 = 0.791}$

WKT; Enthalpy @ ①;

$$h_1 = h_{f_1} + x_1 . h_{fg_1} = h_{f_1} + x_1 (h_{g_1} - h_{f_1})$$

$$\Rightarrow h_1 = 56.32 + 0.791 (322.58 - 56.32)$$

$$h_1 = 266.93 \text{ kJ/kg}.$$

(i)

$$\therefore C.O.P = \frac{h_1 - h_{f_3}}{h_2 - h_1} = \frac{266.93 - 151.96}{293.29 - 266.93} = \underline{4.36}$$

(ii) Capacity:-

WKT; $R_E = h_1 - h_{f_3} = 266.93 - 151.96 = 114.97$ kJ/kg.

The fluid flow is @ the rate of 5 kg/min.

$\therefore$ Total heat extracted $= (114.97 \times 5) = 574.85$ kJ/min.

$\therefore$ Capacity of the refrigerator $= \frac{574.85}{210} = \underline{2.74 \text{ TR}}$

---

(2).> A refrigeration plant using CO_2 as refrigerant works btw. 25°C & −5°C. The dryness of CO_2 is 0.6 @ the entry of Compressor. Find the ice formed per day, if the ice is formed @ 0°C from water @ 10°C, quantity of CO_2 circulated = 10 kg/min.

Take, $C_{pw} = 4.187$ kJ/kg.°C, h_{fg} (ice) = 335 kJ/kg. $\eta_{relative} = 60\%$

Temp. °C	Liquid heat (kJ/kg)	Latent heat (kJ/kg)	Liquid entropy (kJ/kg.K)
25	h_{f_2} 81.25	h_{fg_2} 121.6	0.2513
−5	h_{f_1} −7.53	h_{fg_1} 245.8	−0.0419

Soln:

$h_1 = -7.53 + (0.6 \times 245.8) = 140$ kJ/kg.

$h_3 = 81.25$ kJ/kg $= h_4$

$h_2 = 81.25 + (1 \times 121.6) = 202.85$ kJ/kg.

$$\therefore COP = \frac{h_1 - h_4}{h_2 - h_1} = \frac{140 - 81.25}{202.85 - 140}$$

$COP = \underline{0.935}$

$$\eta_{relative} = \text{Relative (COP)} = 0.6 = \frac{\text{Actual COP}}{\text{Theoretical COP}}$$

$\Rightarrow$ Actual (COP) $= 0.6 \times 0.935 = \underline{0.56}$

Actual cooling $= 0.6 (h_1 - h_4) \times m_r = 0.6 (140 - 81.25) \times 10 = 352.5$ kJ/min.

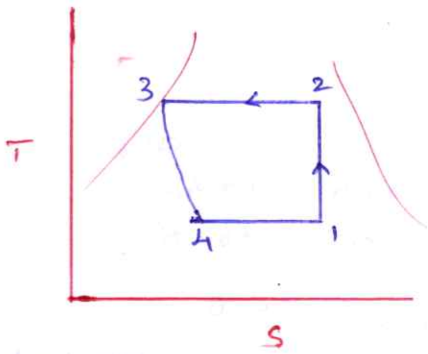
The amt. of heat taken to form 1 kg of ice from 1 kg of water @ 10°C,

$= 4.187 (10) + 335 = 376.87$ kJ/kg

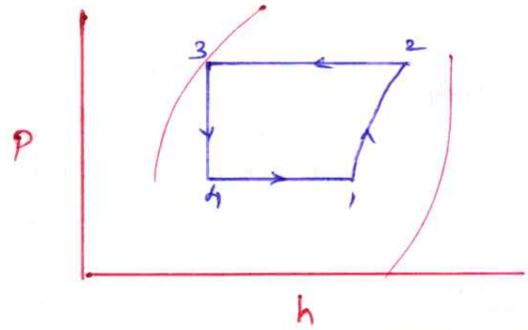
$\therefore$ Amt. of Ice formed per min. $= \frac{352.5}{376.87} = 0.935$ kg/min $= 0.935 \times 60 \times 24$

$= \underline{1347 \text{ kg/day}}$

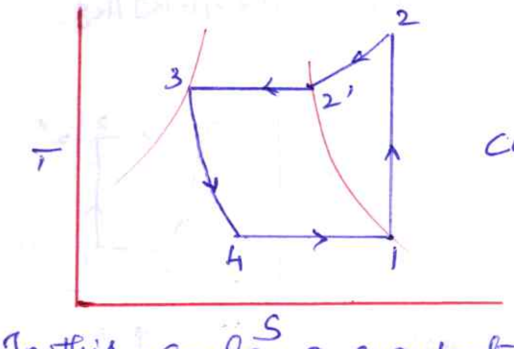
(2) > VCR Cycle with "Wet Vapour After Compression":



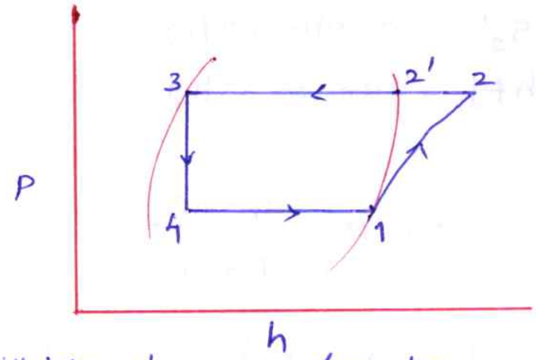
$$C.O.P = \frac{h_1 - h_{f_3}}{h_2 - h_1}$$



(3) > VCR Cycle with "Superheated Vapour After Compression":

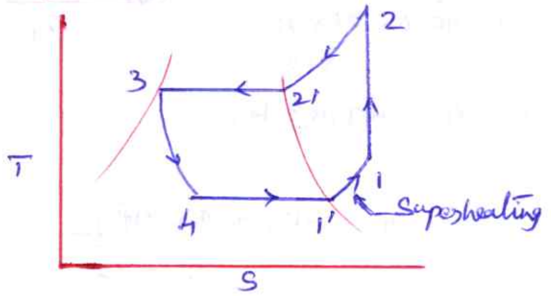


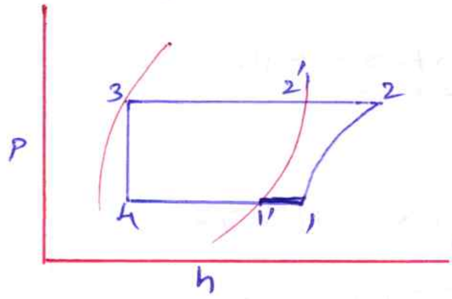
$$COP = \frac{h_1 - h_{f_3}}{h_2 - h_1}$$



In this, cooling of superheated vapour will take place in two stages. (2-2') @ Condensed P=C & (2'-3) @ T=C.

(4) > VCR Cycle with "Superheated Vapour Before Compression":

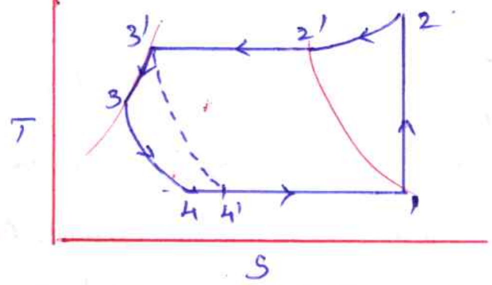




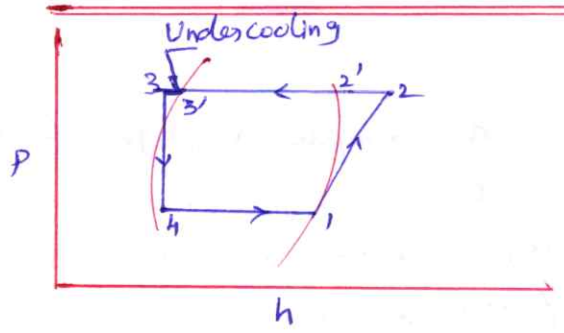
Heat absorbed in two stages. (4-1') & (1'-1)

$$cop = \frac{h_1 - h_{f_3}}{h_2 - h_1}$$

(5) > VCR Cycle with "Under (or) Subcooling of Refrigerant":



(3-3')
↓
Undercooling.



The refrigerant, after condensation process (2'-3'), is cooled below the saturation temp (T_3') before expansion by throttling. This process is called Under cooling.

$C.O.P = \dfrac{h_1 - h_{f_3}}{h_2 - h_1}$ where; $h_{f_3} = h_{f_3'} - (C_{P,liq} \times \text{Degree of undercooling})$

3. Find the theoretical C.O.P for a CO_2 m/c working bet. the temp. range of 25°C & −5°C. The dryness fraction of CO_2 gas during the suction stroke is 0.6. Following properties of CO_2 are given.

Temp. °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

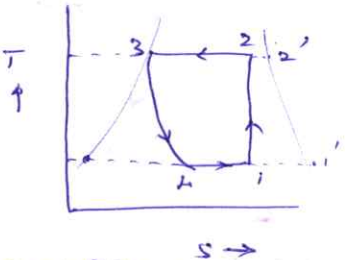
Soln:- $T_2 = 25°C = T_3$, $T_1 = -5°C = T_4$, $x_1 = 0.6$,

$h_{f_3} = h_{f_2} = 164.77$ kJ/kg, $h_{f_1} = h_{f_4} = 72.57$ kJ/kg, $s_{f_2} = 0.5978$ kJ/kg.K,

$s_{f_1} = 0.2862$ 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, $h_{fg_2} = 117.46$ kJ/kg.

$h_{fg_1} = 248.76$ kJ/kg.



$$C.O.P = \frac{h_1 - h_{f_3}}{h_2 - h_1}$$

$\Rightarrow h_1 = h_{f_1} + x_1 . h_{fg_1} = 72.57 + (0.6 \times 248.76) = \underline{\underline{221.83}}$ kJ/kg.

$h_2 = h_{f_2} + x_2 . h_{fg_2}$

$\therefore h_2 = 164.77 + (0.622 \times 117.46)$

$h_2 = \underline{237.83}$ kJ/kg

$$\therefore C.O.P = \frac{221.83 - 164.77}{237.83 - 221.83}$$

$C.O.P = \underline{3.57}$

$$\Rightarrow \left\{ \begin{array}{l} s_1 = s_{f_1} + x_1 . s_{fg_1} \\ s_1 = 0.2862 + \left(0.6 \times \frac{248.76}{268}\right) \quad \left(\because s_{fg_1} = \frac{h_{fg_1}}{T_1}\right) \\ s_1 = 0.8431 \text{ kJ/kg.K.} \\ \text{|||}^{ly};\quad s_2 = s_1 = s_{f_2} + x_2 . \frac{h_{fg_2}}{T_2} \\ 0.8431 = 0.5978 + \left(x_2 \times \frac{117.46}{298}\right) \\ \Rightarrow \underline{\underline{x_2 = 0.622}} \end{array} \right.$$

---

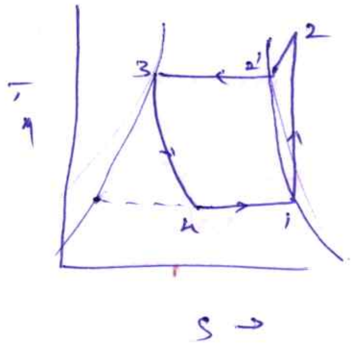
4. A simple refrigerant 134a (Tetrafluroethane) heat Pump for space heating, operates bet. temp. limits of 15°C & 50°C. The heat reqd. to be pumped is 100 MJ/hr. Determine: (i) The dryness fraction of refrigerant entering the evaporator, (ii) The discharge temp., assuming the sp. heat of vapour is 0.996 kJ/kg.K. (iii) The theoretical piston displacement of the compressor &* (iv) Theoretical power of the compressor & (v) C.O.P.

The sp. vol. of R134a saturated vapour @ 15°C is 0.04185 m^3/kg. The other values are given below.

Satn. temp (°C)	Pr. (bar)	Sp. Enthalpy (kJ/kg)		Sp. Entropy (kJ/kg.k)	
		Liquid	Vapour	Liquid	Vapour
$T_1=T_4$ 15	4.887	h_{f1} 220.26	h_1 413.6	s_{f1} 1.0729	$s_1=s_2$ 1.7439
$T_2'=T_3$ 50	13.18	$h_{f3}=h_4$ 271.97	h_2' 430.4	s_{f3} 1.2410	s_2' 1.7312

Soln: $Q = 100\times10^3$ kJ/hr, $C_p = 0.996$ kJ/kg.K, $v_1 = 0.04185\ m^3/kg$,

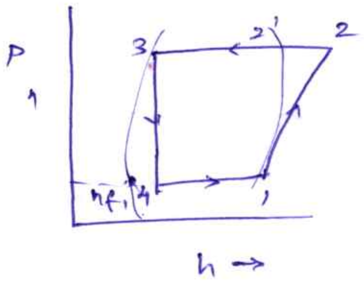
(i) Dryness fraction (x_4):



$$x_4 = \frac{h_4 - h_{f1}}{h_1 - h_{f1}} = \frac{271.97 - 220.26}{413.6 - 220.26}$$

$$x_4 = \underline{\underline{0.2675}}$$

(ii) Discharge Temp. (T_2):



$$S_2 = S_2' + C_p \ln\left(\frac{T_2}{T_2'}\right) = \cdot$$

$$\Rightarrow 1.7439 = 1.7312 + 2.3\times0.996 \log\left(\frac{T_2}{T_2'}\right)$$

$$\Rightarrow \frac{T_2}{T_2'} = 1.0128 \quad \Rightarrow T_2 = \underline{\underline{327.13k}} = \underline{\underline{54.13°c}}$$

(iii) Theoretical piston Displacement.

wkt, @ 2; $h_2 = h_2' + C_p(T_2 - T_2') = 430.4 + 0.996(327.13 - 323)$

$$\Rightarrow h_2 = \underline{434.5}\ kJ/kg$$

Mass flow rate of the refrigerant, $m_R = \frac{Q}{h_2 - h_{f3}} = \frac{100\times10^3}{434.5 - 271.97} = 615.3\ kg/hr$

$= \underline{\underline{10.254\ kg/min}}$

∴ Theoretical piston displacement of the Compressor $= m_R \times V_1$

$= 10.254 \times 0.04185$

$= \underline{\underline{4.3\ m^3/min}}$

(iv) Theoretical Power:

WKT, $W.D = m_R(h_2 - h_1) = 10.254(434.5 - 413.6)$

$= 214.3$ kJ/min

$\therefore$ Power $= \frac{214.3}{60} = 3.57$ kW

(v) $COP = \frac{h_1 - h_{f_3}}{h_2 - h_1} = 6.8$

---

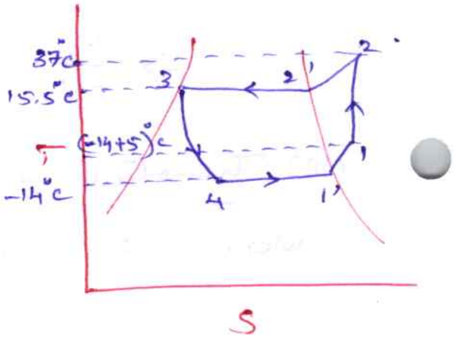
5) A vapour compression refrigeration plant works b/t. pr. limits of 5.3 bar & 2.1 bar. The vapour is superheated @ the end of compression, its temp. being 37°c. The vapour is superheated by 5°c before entering the compressor. If the sp. heat of superheated vapour is 0.63 kJ/kg.k. Find the C.O.P of the plant. Use the data given below.

Pr. (bar)	Sat. temp (°c)	Liquid heat (kJ/kg)	Latent heat (kJ/kg)
P_2 5.3	T_2' 15.5	h_{f_3} 56.15 $= h_{f_2}'$	144.9 h_{fg_2}'
P_1 2.1	T_1' −14.0	h_{f_1}' 25.12	158.7 h_{fg_1}'

Soln: $T_2 = 37°c = 310K$, $(T_1 - T_1') = 5°c \overset{=264k}{}$, $C_p = 0.63$ kJ/kg.k.

$T_2' = 15.5°c = 288.5K$, $T_1' = -14°c = 259K$.

$$C.O.P = \frac{h_1 - h_{f_3}}{h_2 - h_1}$$



$\Rightarrow h_1 = h_1' + C_p(T_1 - T_1')$

$= (h_{f_1}' + h_{fg_1}') + C_p(T_1 - T_1')$

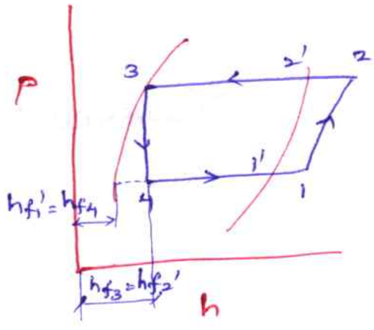
$= (25.12 + 158.7) + (0.63 \times 5)$

$h_1 = 186.97$ kJ/kg

III^{ly} $h_2 = h_2' + C_p(T_2 - T_2')$

$= (56.15 + 144.9) + 0.63(310 - 288.5)$

$h_2 = 214.6$ kJ/kg



$$\therefore C.O.P = \frac{186.97 - 56.15}{214.6 - 186.97} = 4.74$$

(ii) WKT; Referigerating Effect/kg of refrigerant;

$= h_1 - h_{f_3} = 1428 - 270 = 1158$ kJ/kg.

∴ Total Refrigerating Effect, $N = m_R \times (h_1 - h_{f_3})$

$= 3.265 \times 1158$

$N = \underline{\underline{3781}}$ kJ/min.

(ii) Plant capacity $= \dfrac{3781}{210} = \underline{\underline{18\ TR}}$

(iii) WKT; W.D during compression of the refrigerant;

$= m_R(h_2 - h_1) = 3.265(1630 - 1428)$

$= 659.53$ kJ/min.

(given) Heat rejected to compressor by cooling water $= 5000$ kJ/hr

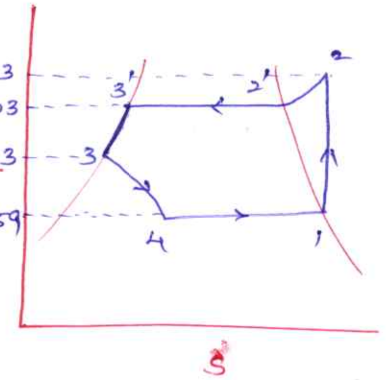
$= 83.33$ kJ/min

∴ Total W.D by the system $= 659.53 + 83.33 = 742.86$ kJ/min

∴ Indicated Power of the system $= \dfrac{742.86}{60} = \underline{\underline{12.38}}$ kW

(iv) $C.O.P = \dfrac{N}{W.D} = \dfrac{3781}{742.86} = \underline{\underline{5.1}}$

6) A simple NH_3 vapour compression system has compressor with piston displacement of $2 m^3/min$, a condenser pr. of 12 bar & evaporator pr. of 2.5 bar. The liquid is sub-cooled to 20°c by soldering the liquid line to suction line. The temp. of vapour leaving the compressor is 100°c, heat rejected to compressor cooling water is 5000 kJ/hr & η_{vol} of compressor is 0.8. Calculate: (i) plant capacity, (ii) Indicated Power, (iii) C.O.P & (iv) mass flow rate in kg/min.



Soln:- $V_p = 2 m^3/min.$,

$P_2 = P_2' = P_3' = P_3 = 12$ bar,

$P_1 = P_4 = 2.5$ bar,

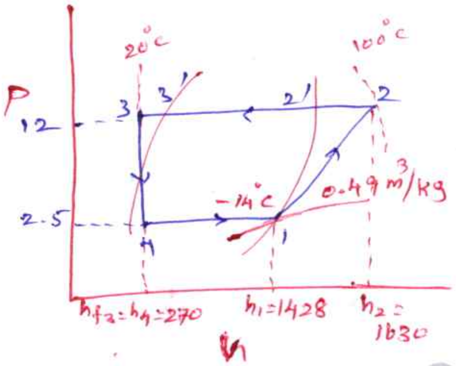
$T_3 = 20°c = 293 K$

$T_2 = 100°c = 373K$, $\eta_v = 0.8$,

From (p-h) diagram:

> we find that the evaporating temp. corresponding to 2.5 bar is;

$T_1 = T_4 = -14°c = 259 K$



> Condensing temp. corresponding to 12 bar is;

$T_2' = T_3' = 30°c = 303K$

> Sp. vol. of dry saturated vapour @ 2.5 bar (@ Point 1);

$v_1 = 0.49 m^3/kg.$

> Enthalpy of dry saturated vapour at point 1;

$h_1 = 1428$ kJ/kg.

> Enthalpy of superheated vapour at point 2; $h_2 = 1630$ kJ/kg.

> Enthalpy of sub-cooled liquid @ 20°c @ Point 3;

$h_{f3} = h_4 = 270$ kJ/kg.

wkt; Piston displacement; $V_p = \frac{m_R \cdot v_1}{\eta_v}$

(i) => Mass flow rate of the refrigerant $\Big\}$. $m_R = \frac{V_p \cdot \eta_v}{v_1}$

$= \frac{2 \times 0.8}{0.49}$

$m_R = \underline{\underline{3.265}}$ kg/min

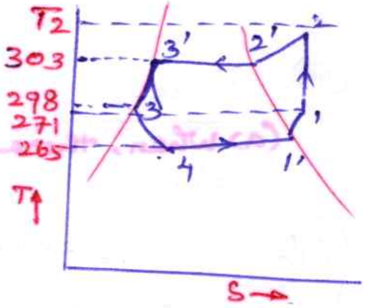
A food storage lockers requires a refrigeration capacity of 12TR & works btw. the evaporating temp. of −8°c & condensing temp. of 30°c. The refrigerant R-12 is subcooled by 5°c before entry of expansion valve & the vapour is superheated to −2°c before leaving the evaporator. Assuming a two cylinder, single acting compressor operating @ 1000 RPM with stroke equal to 1.5 times the bore. Determine: (i) C.O.P, (ii) Power/TR & (iii) Bore & stroke of compressor, when (a) there is no clearance & (b) there is a clearance of 2%. Use the data given below.

Sat. Temp.(°c)	Pr. (bar)	Sp. Enthalpy (kJ/kg)		Sp. Entropy (kJ/kg.k)		Sp. vol. of Vapour (m^3/kg)
		Liquid	Vapour	Liquid	Vapour	
$T_1' = T_4$ −8	2.354	h_{f_1} 28.72	h_1' 184.07	*s_{f_1} 0.1149	s_1' 0.7007	v_1' 0.0790
$T_2' = T_3'$ 30	7.451	$h_{f_3'}$ 64.59	h_2' 199.62	*s_{f_3} 0.240	s_2' 0.6853	v_2' 0.0235

Take; $C_{pl} = 1.235$ kJ/kg.k & $C_{pv} = 0.733$ kJ/kg.k.

Soln:-

$Q = 12TR$, $T_3' - T_3 = 5°c$, $T_1 = -2°c$,

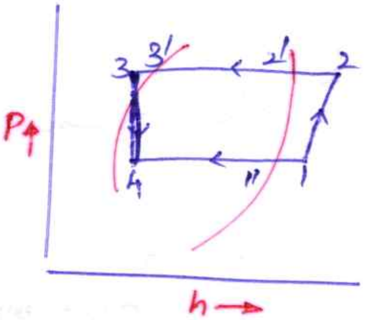


(i) $C.O.P = \dfrac{h_1 - h_{f_3}}{h_2 - h_1}$

$\Rightarrow h_1 = h_1' + C_{pv}(T_1 - T_1')$

$= 184.07 + 0.733(271 - 265)$

$h_1 = \underline{188.47}$ kJ/kg



$\Rightarrow h_2 = h_2' + C_{pv}(T_2 - T_2')$

when; $s_1 = s_1' + 2.3 \cdot C_{pv} \log\left(\frac{T_1}{T_1'}\right)$

$= 0.7007 + 2.3 \times 0.733 \log\left(\frac{271}{265}\right)$

$\underline{s_1} = 0.7171$ kJ/kg.k

Illy; $s_2 = s_1 = s_2' + 2.3\, C_{pv} \log\left(\frac{T_2}{T_2'}\right)$

$\Rightarrow 0.7171 = 0.6853 + 2.3 \times 0.733 \log\left(\frac{T_2}{303}\right)$

$\Rightarrow T_2 = \underline{316.4\text{K}}$

$\therefore h_2 = 199.62 + 0.733(316.4 - 303) = \underline{\underline{209.44 \text{ kJ/kg}}}$

WKT; $h_{f_3} = h_{f_3}' - C_{Pl}(T_3' - T_3)$

$$\left[\because h_{f_3} = h_{f_3}' - (C_p \times \text{Degree of undercooling})\right]$$

$\Rightarrow h_{f_3} = 64.59 - (1.235 \times 5)$

$= \underline{58.42 \text{ kJ/kg}}$

$$\therefore C.O.P = \frac{188.47 - 58.42}{209.44 - 188.47} = \underline{6.2}$$

(ii) Theoretical Power per TR:

WKT; Heat ~~to~~ extracted / kg of refrigerant;

$Q_A = R_E = (h_1 - h_{f_3}) = 188.47 - 58.42$

$R_E = \underline{130.05 \text{ kJ/kg}}$

Refrigerating capacity, $Q = 12\text{ TR} = 12 \times 210 = 2520$ kJ/min

$$\therefore \text{Mass flow rate, } m_R = \frac{Q}{R_E} = \frac{2520}{130.05} = \underline{19.4 \text{ kg/min}}$$

$\therefore W.D = m_R(h_2 - h_1) = 19.4(209.44 - 188.47) = 406.82$ kJ/min

$$\therefore \text{Power/TR} = \frac{406.82}{60 \times 12} = \underline{0.565} \text{ kW/TR}$$

(iii) Bore & Stroke of Compressor:

(a) when, there is "no clearance":

WKT; Theoretical suction volume/min; $= m_R \cdot V_1$

$= 19.4 \times 0.081$

$= 1.57 \text{ m}^3/\text{min}$

$= \underline{0.785 \text{ m}^3/\text{min}}$ (by $\div 2$, $\because$ because of two no. of cylinders)

(1'-1) $p = c$; (charle's law)

$$\therefore \frac{V_1}{T_1} = \frac{V_1'}{T_1'} \Rightarrow V_1 = \frac{V_1' \cdot T_1}{T_1'}$$

$$V_1 = 0.079 \times \frac{271}{265}$$

$V_1 = 0.081 \text{ m}^3/\text{kg}$

Also; theoretical suction volume/min $= \frac{\pi}{4} D^2 \cdot L \cdot N$

$\Rightarrow 0.785 = \frac{\pi}{4} D^2 \times 1.5D \times 1000$

$\therefore L = 1.5D$

$\Rightarrow D = 0.087 \text{ m} = \underline{87 \text{ mm}}$

$L = \underline{130.5 \text{ mm}}$

(b) when, there is a clearance of 2%;

$$\text{Piston displacement (or) theoretical suction vol./min} = \frac{m_R \cdot V_1}{2 \times \eta_v} = \frac{19.4 \times 0.081}{2 \times 0.954}$$

$= \underline{0.8236 \text{ m}^3/\text{min.}}$

where; $\eta_v = 1 + C - C\left(\frac{V_1}{V_2}\right)$ ($\because C = 2\% = 0.02$)

WKT; $\frac{V_2}{T_2} = \frac{V_2'}{T_2'}$

$\Rightarrow V_2 = 0.0235 \times \frac{316.4}{303}$

$V_2 = 0.0245 \text{ m}^3/\text{kg}$

$$\therefore \eta_v = 1 + 0.02 - 0.02\left(\frac{0.081}{0.0245}\right)$$

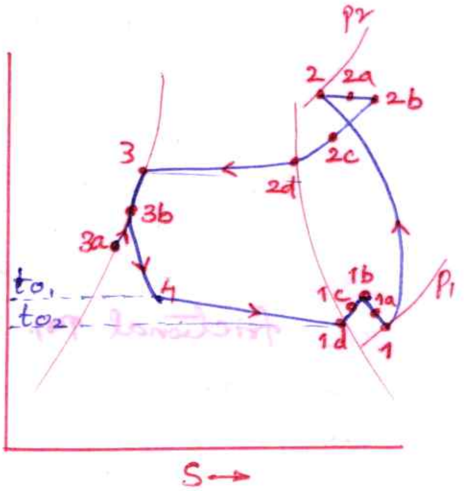
$\eta_v = \underline{0.954}$

$|||^{ly}$ $\Rightarrow 0.8236 = \frac{\pi}{4} D^2 \times 1.5D \times 1000$

$\Rightarrow D = 0.0887 \text{ m} = \underline{88.7 \text{ mm}}$

$\therefore L = 1.5D = \underline{133 \text{ mm}}$

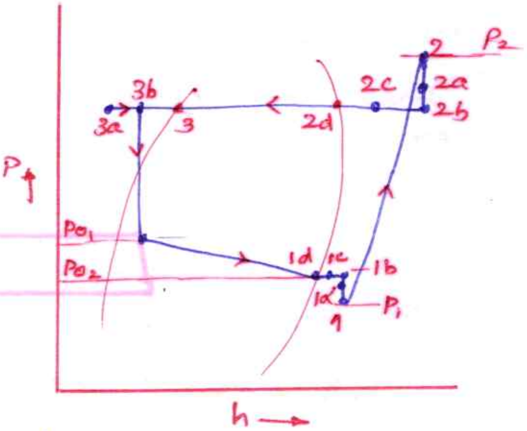
> ACTUAL VAPOUR COMPRESSION CYCLE:



Actual vapour compression cycle differs from theoretical vapour compression cycle, due to flow of the refrigerant thro' the condenser, evaporator & piping, there will be drops in pressure.

In addition, there will be heat losses (or) gained depends on the temp. difference b/w. the refrigerant & the surroundings.

Compression will be polytropic with friction & heat transfer instead of isentropic. Fig. shows;



- >(1d - 1c) - Superheating of the vapour in the evaporator.
- >(1c - 1b) - Heat gain & superheating of the vapour in the suction line.
- >(1b - 1a) - Pr. drop in the suction line.
- >(1a - 1) - Pr. drop due to wire drawing @ the compressor suction valve.
- >(1 - 2) - Polytropic compression istead of isentropic.
- >(2 - 2a) - Pr. drop at the compressor delivery valve.
- >(2a - 2b) - Pr. drop in the delivery line.
- >(2b - 2c) - Heat loss & De-superheating of the vapour in the delivery line.
- >(2c - 2d) - De-superheating of the vapour in the condenser.
- >(2b - 3) - Pr. drop in the condenser.
- >(3 - 3a) - Sub-cooling of the liquid in the condenser (or) Sub-cooler.
- >(3a - 3b) - Heat gain in the liquid line.

The lines (3 - 3a) & (3a - 3b) are along the saturated liquid line on the (T-S) dig. as the constant pr. lines in the liquid region.

- >(3b - 4) - Expansion of Sub-cooled liquid refrigerant by throttling from the condenser pr. to the evaporator pr.
- >(4 - 1d) - Pr. drop in the evaporator.

Pr. drop in the evaporator is large. This is due to the cumulative effect of two factors.

> 1st : Pr. drop in evaporator due to friction [Frictional Pr. Drop].

> 2nd : Volume increases & hence the velocity (K.E) must also increase. The increase in K.E comes from a decrease in enthalpy & further Pr. drop also. [Momentum Pr. Drop].

In the Condenser, the pr. drop is not significant. Since, the frictional pr. drop is positive & the momentum pr. drop is negative. In Condenser pr. drop is not very critical & it's very much useful for design of evaporator.

Generally condensers are oversized, however in the case of evaporator's both undersizing & oversizing are harmful.

The actual work of compression can be determined by knowing the initial & final states & polytropic index (n). The heat rejected in the polytropic process can be obtained by applying the steady flow energy eqn.

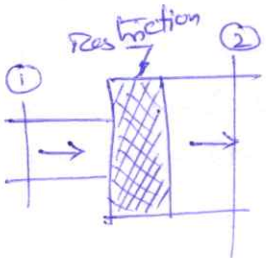
$$q = (h_2 - h_1) + W$$

$$\Rightarrow \boxed{-W = (h_2 - h_1) - q}$$ → It represents the energy balance of the compressor.

(i.e) Work of Compression = Increase in enthalpy of gas } + Heat lost in cooling.

Throttling:

It is an irreversible adiabatic process. It is employed to reduce the pr. of a fluid by introducing a restriction in the flow passage.



As the process is adiabatic & no external work is done. we have,

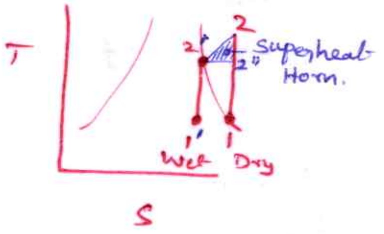
$$h_1 + \frac{V_1^2}{2} = h_2 + \frac{V_2^2}{2}$$

Since, K.E change is negligible; we find that in a throttling process,

$$h_1 = h_2$$

Accordingly, throttling is also referred to as an "Isenthalpic-Expansion Process".

> Wet Versus Dry Compression:



The compression process as shown in fig.

The compression of wet refrigerant vapour @ 1' to dry saturated vapour @ 2'. It is called "wet compression".

with a reciprocating compressor, wet compression is not found suitable, because of;

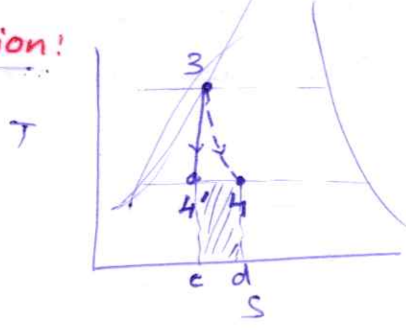
-> The liquid refrigerant may be trapped in the cylinderhead & may demage the compressor valves & the cylinder. Even through the state of vapour @ the end of wet compression is theoretically dry saturated, it is normal to expect some liquid droplets to remain suspended in the gas, as the time taken by the compression process is quite small compared to the time needed for evaporation of droplets.

-> Liquid refrigerant droplets may wash away the lubricating oil from the walls of the cylinder in compressor, thus increasing wear.

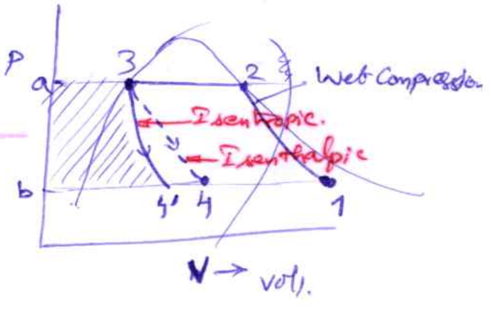
Thus, the compression with vapour initially dry saturated at 1. Such compression is known as "dry compression".

The increased work of the cycle due to the substitution of wet compression by dry compression appears as the area 2-2'-2", is known as "Superheat Horn".

Throttling Versus Isentropic Expansion:



Isentropic expasion work by area; (3-a-b-4') -> positive work is even smaller, as compared to the isentropic compression work by area (1-2-a-b) -> negative work.

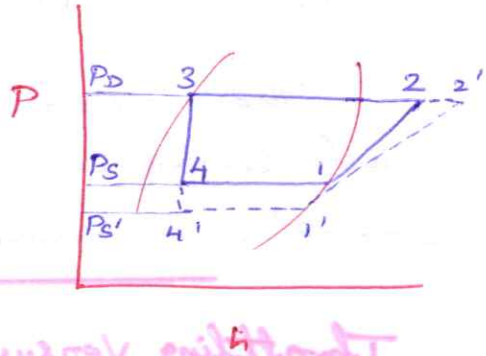


In the expander, the refrigerant is in the liquid state, whereas, in the compressor, it is in gaseous state. The volume of the vapour is very large when compared to the vol. of liquid. ($v_g >> v_f$). Hence, the +ve work of isentropic expansion is seldom large enough to justify the cost of an expander and the thermodynamic & friction losses are employed. may even exceed the gain in work.

Acc. to the isentropic, expansion process of the carnot cycle may be replaced by a simple throttling process (Isenthalpy) by the use of an expansion device such as a throttle valve or a capillary tube.

The process is an ir-reversible & is accompained by increase of entropy as shown by line (3-4) on the (T-s) dig. Thus, the (3-4) Isenthapic process substitute, instead of (3-4') Isentropic process, would theoretically result in a loss of work represented by area 3-a-b-4' on the (p-v) dig. & a decrease in the refrigerating effect represented by area 4'-c-d-4 on the (T-s) dig. It can be shown that both these areas are equal.

> Effect of Suction (Evaporator) pr.:



The suction pr. decreases due to the Frictional resistance of flow of the refrigerant.

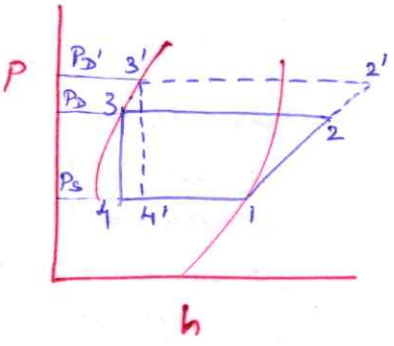
From Ps to Ps' (i.e) from Cycle (1-2-3-4) to (1'-2'-3-4').

The decrease in suction pr. gives;

- Decrease in refrigerating effect from $(h_1 - h_4)$ to $(h_1' - h_4')$ &
- Increase in Compressor work from $(h_2 - h_1)$ to $(h_2' - h_1')$

Since, the C.O.P is to decrease for the same amount of refrigerant flow & increases the cost of refrigeration for required capacity of refrigerating.

> Effect of Discharge (condenser) Psr.



The discharge Psr. increases due to frictional resistance of flow of the refrigerant.

From P_D to P_D' from cycle $(1-2-3-4)$ to $(1^*-2'-3'-4')$.

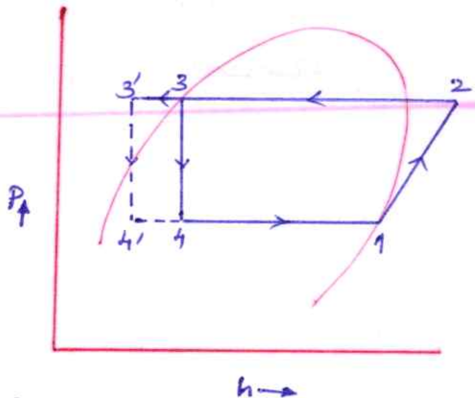
The increase in discharge Psr. gives;

- Decrease the refrigerating effect from (h_1-h_4) to (h_1-h_4') &
- Increase the work of compressor from (h_2-h_1) to $(h_2'-h_1)$

That, the effect of increase in discharge Psr. is similar to the effect of decrease in suction Psr.

NOTE: * The effect of increase in condenser Psr. is not as severe, on the refrigerating capacity & power consumption per TR, as that of the decrease in evaporator pressure.

> Effect of Liquid Sub-cooling:



It is possible to reduce the temp. of the liquid refrigerant to within a few degree's of the temp. of the water entering the condenser, in some condenser designs by installing a "Subcooler" btw. the condenser & the expansion valve.

Subcooling reduces flashing of the liquid during expansion & increases the "RE".

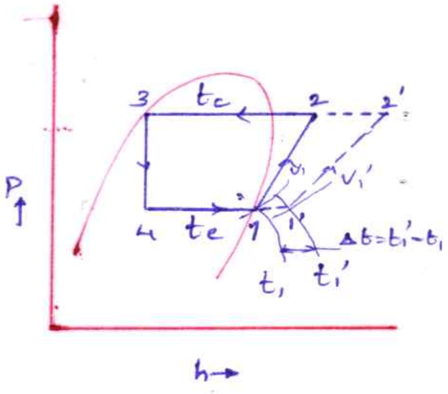
Normally, cooling water first passes thro' the subcooler & then thro' the condenser. Thus, the coolest water comes in contact with the liquid being subcooled.

The use of cooling water effectively in the condenser itself to keep the condensing temp. as near to the temp. of the cooling water inlet as possible.

In general, the functions of the condenser as well as the subcooler can be combined in the condenser itself by slightly oversizing the condenser.

> Effect of Suction Vapour Superheat.

Superheating of the suction vapour is advisable in practice because it ensures complete vaporization of the liquid in the evaporator before it enters the compressor. Also, in most R & AC systems, the degree of superheat serves as a means of actuating & modulating the capacity of the expansion valve.



[(1-2) steeper Isentropic,
(1'-2') Flatter Isentropic]

The effect of superheating of the vapour from $t_1 = t_e$ to t_1' is as follows.

- Increase in sp. vol. of suction vapour from v_1 to v_1'.
- Increase in R_E from $(h_1 - h_4)$ to $(h_1' - h_4)$
- Increase in sp. work from $(h_2 - h_1)$ to $(h_2' - h_1')$.

As both the Refrigerating effect (R_E) & the work (W) increase, the numerical value of C.O.P may increase (or) decrease (or) remain the same.

Ex: Freon 12 System, Superheating increases the C.O.P, whereas; Freon 22 & NH_3 System, it decreases the COP.

Influence of Various Parameters on improving the C.O.P.

The Working of Simple compression m/c is not meet the different requirements of the industry.

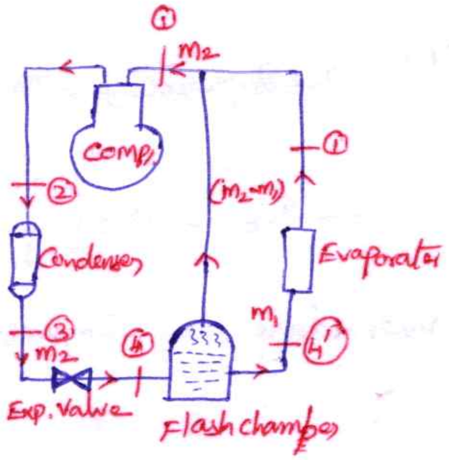
So, it is necessary to modify the simple working cycle in order to improve its performance, @ the same time, it must meet the different requirements of the industry such as reduction in plant size, different loads @ different temp/. without involving the considerable increase in other operating expenses, the investment cost & maintanance expenses.

Methods of Improving COP!

> with "FLASH CHAMBER"

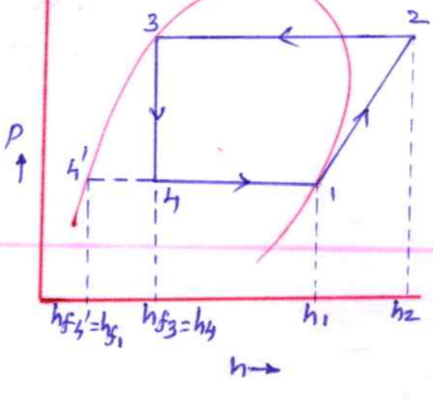
"Flash" is defined as the mass of vaporized refrigerant/kg after leaving the throttle valve. →

("Partial evaporation of the liquid refrigerant is known as Flash.")



↳ This formed vapour refrigerant does not take part in the refrigerating effect as compared with the liquid refrigerant which carries the heat in the form of latent heat. This vapour which is incapable of producing refrigeration can be bypassed around the evaporator & supplied directly to the compressor as shown in fig.

This is done by using a tank before evaporator which is known as flash chamber. This seperates liquid & vapour & thus reduces the mass of refrigerant passing thro' the evaporator.



This arrangement has no effect on the thermodynamic cycle. The power & COP will remain as a simple saturation cycle (Theoretically). It reduces the size of evaporator used providing better conditions for heat transfer.

Let; m_1 = mass of liquid refrigerant supplied to the evaporator &

m_2 = mass of refrigerant (liq + vap.) circulating thro' the condenser (or) leaving the expansion value.

∴ mass of vapour refrigerant flowing directly from the flash chamber to the suction of the compressor $= (m_2 - m_1)$.

Considering the thermal equilibrium of the flash chamber;

$$\left.\begin{array}{c}\text{Heat taken by the flash}\\ \text{chamber}\end{array}\right\} = \begin{array}{c}\text{Heat given out by the flash}\\ \text{chamber.}\end{array}$$

$$m_2 \cdot h_4 = m_1 \cdot h_{f_4'} + (m_2 - m_1) h_1$$

$$\therefore m_2 (h_1 - h_4) = m_1 (h_1 - h_{f_4'})$$

$$\therefore m_1 = m_2 \left[\frac{h_1 - h_4}{h_1 - h_{f_4'}} \right] = m_2 \left[\frac{h_1 - h_{f_3}}{h_1 - h_{f_4'}} \right]$$

WKT; Refrigerating effect; $R_E = m_1 (h_1 - h_{f_4'})$

$$\Rightarrow R_E = m_2 \left[\frac{h_1 - h_{f_3}}{h_1 - h_{f_4'}} \right] (h_1 - h_{f_4'}) = m_2 (h_1 - h_{f_3})$$

Workdone in Compressor; $W = m_2 (h_2 - h_1)$

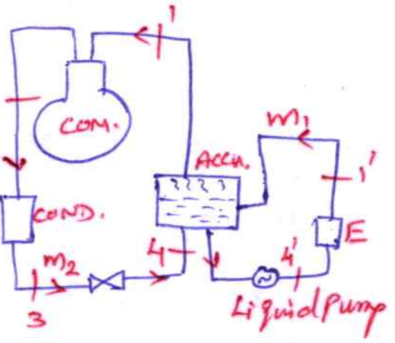
$$\therefore C.O.P = \frac{R_E}{W} = \frac{h_1 - h_{f_3}}{h_2 - h_1}$$

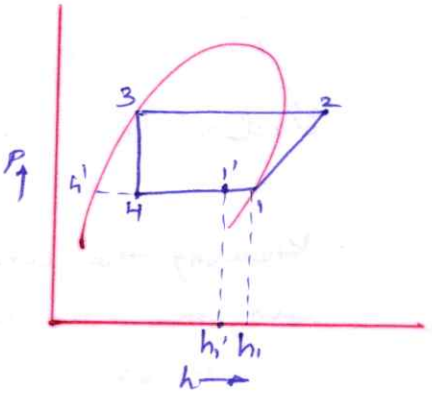
& Power reqd. to drive the compressor;

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

> With " ACCUMULATOR (or) PRE-COOLER".

Sometimes, the liquid refrigerant passing thro' the evaporator is not completely evaporated. If the compressor is supplied with liquid along with vapour refrigerant, then the compressor has to do an additional work of evaporating & raising the temp. of liquid refrigerant. It will also upset the normal working of the compressor which is meant only for compressing the pure vapour refrigerant. In order to avoid this difficulty, an insulated vessel, known as accumulator is used in the system.





Heat taken in by the accumulator = Heat given out by the accumulator.

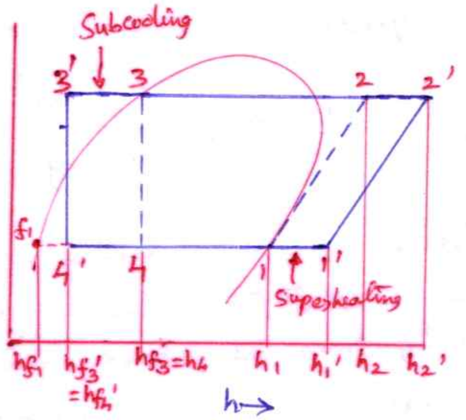
$$m_2 h_4 + m_1 . h_1' = m_2 h_1 + m_1 h_{f_4}'$$

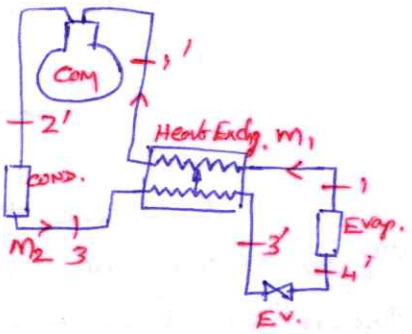
$$\Rightarrow m_1 = m_2 \left[\frac{h_1 - h_4}{h_1' - h_{f_4}'} \right] \quad ; \quad cop = \frac{h_1 - h_{f_3}}{h_2 - h_1}$$

111^{ly} to the flash chamber.

$$\therefore P = \frac{m_2 (h_2 - h_1)}{60} \text{ kW.}$$

> With " SUBCOOLING OF LIQUID REFRIGERANT by VAPOUR REFRIGERANT".





The temp. of the vapour refrigerant coming out of the evaporator is less than the temp. of the liquid refrigerant coming out of the condenser. The liquid refrigerant can be cooled giving its heat to the vapour.

3-3' → Subcooling the liquid refrigerant.

1-1' → Super-heating the vapour refrigerant.

Considering the energy balance in the heat exchanger;

⇒ Heat lost by liquid refrigerant = Heat gained by vapour refrigerant.

$$m_2 \times C_{pl} (T_3 - T_3') = m_1 \times C_{pv} (T_1' - T_1)$$

Since; $m_2 = m_1$ & $C_{pl} > C_{pv}$

$$\therefore (T_3 - T_3') < (T_1' - T_1)$$

∴ For energy balance; $(h_{f_3} - h_{f_3'}) = (h_1' - h_1)$

Under Ideal conditions of heat transfer in heat exchanger;

$$\therefore T_3' = T_1' = T_m$$

Knowing the conditions of points 1 & 3; conditions of points 1' & 3' can be obtained by trial & error procedure on p-h chart.

Mass of refrigerant reqd. to be passed in the evaporator of 'Q' tonnes load for sub-cooled cycle is given by;

$$m_R = \frac{210.Q}{h_1 - h_{f_3}'} \text{ kg/min.} = \frac{210.Q}{h_1 - h_{f_4}'}$$

$$\therefore \text{Power of compressor, } P_1 = \frac{m_R (h_2' - h_1')}{60} \text{ kW} = \frac{210.Q}{60}\left[\frac{h_2' - h_1'}{h_1 - h_{f_3}'}\right] \text{kW}$$

& Power reqd. to drive the compressor without heat exchanger;

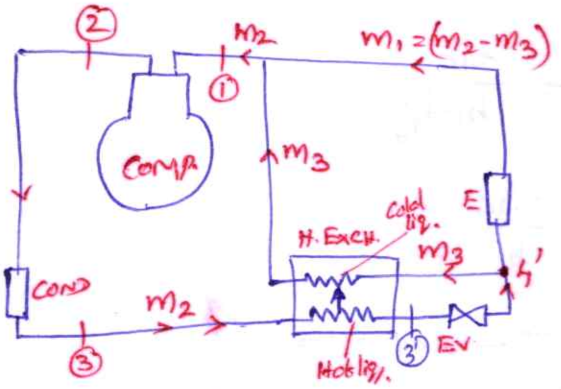
$$P_2 = \frac{m_R (h_2 - h_1)}{60} \text{ kW} = \frac{210.Q}{60}\left[\frac{h_2 - h_1}{h_1 - h_{f_3}}\right] \text{kW}$$

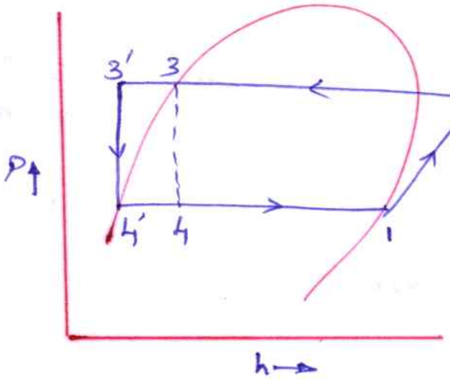
∴ Excess power required in sub-cooled cycle; $P_{excess} = P_1 - P_2$

$$\therefore P_{Excess} = \frac{210.Q}{60}\left[\frac{h_2' - h_1'}{h_1 - h_{f_3}'} - \frac{h_2 - h_1}{h_1 - h_{f_3}}\right] \text{kW}$$

This shows that, by sub-cooling the liquid refrigerant by vapour refrigerant, the COP of the cycle is reduced. Even with theoretical loss resulting from the above type of sub-cooling, there are many actual installations which adopt this process.

➤ With" SUB-COOLING OF LIQUID REFRIGERANT by LIQUID REFRIGERANT.





The temp. of the liquid refrigerant coming out of expansion valve is less than the temp. of the liquid refrigerant coming out of condenser, so that cooling of liquid coming out from condenser is possible by using the liquid coming out of expansion (throttle) valve.

let; m_3 = Mass of liquid refrigerant supplied to the heat exchanger, from the expansion valve.

Considering the energy balance in the heat exchanger;

⇒ Heat lost by hot liquid = Heat gained by cold liquid.

$$m_2(h_{f_3} - h_{f_3'}) = m_3(h_1 - h_{f_4'})$$

$$\Rightarrow m_3 = m_2\left[\frac{h_{f_3} - h_{f_3'}}{h_1 - h_{f_4'}}\right] = m_2\left[\frac{h_{f_3} - h_{f_3'}}{h_1 - h_{f_3'}}\right]$$

$$(\because h_{f_4'} = h_{f_3'})$$

W.K.T;

$$R_E = m_1(h_1 - h_{f_4'}) = (m_2 - m_3)(h_1 - h_{f_4'})$$

$$\therefore R_E = \left[m_2 - m_2\left(\frac{h_{f_3} - h_{f_3'}}{h_1 - h_{f_4'}}\right)\right](h_1 - h_{f_4'})$$

$$= m_2 h_1 - m_2 h_{f_4'} - m_2 h_{f_3} + m_2 h_{f_3'}$$

$$R_E = m_2(h_1 - h_{f_3})$$

& W.D in compressor; $W = m_2(h_2 - h_1)$

$$\therefore COP = \frac{R_E}{W} = \frac{h_1 - h_{f_3}}{h_2 - h_1}$$

If, Q Tonnes of refrigeration is the load on the evaporator then the mass of refrigerant (m_1) reqd. to be circulated thro' the evaporator is given by;

$$m_1 = \frac{210 \cdot Q}{h_1 - h_{f_4}'} \Rightarrow (m_2 - m_3) = \frac{210 \cdot Q}{h_1 - h_{f_4}'}$$

$$\Rightarrow m_2 - m_2\left(\frac{h_{f_3} - h_{f_3}'}{h_1 - h_{f_4}'}\right) = \frac{210 \cdot Q}{h_1 - h_{f_4}'}$$

$$\Rightarrow \therefore m_2 = \frac{210 \cdot Q}{h_1 - h_{f_3}}$$

wkt;

Power reqd, to drive the compressor;

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

$$\therefore P = \frac{210 \cdot Q}{60}\left(\frac{h_2 - h_1}{h_1 - h_{f_3}}\right) \text{ kW.}$$

From the Power & COP expression, it is obvious that there is no advantages with the use of this arrangement as C.O.P of the new system remains unchanged. The arrangement is thermodynamically identical with simple cycle.

DOMESTIC ELECTROLUX [NH_3-H_2] REFRIGERATOR:

[THREE FLUID ABSORPTION SYSTEM].

Electrolux Principle works on 3-Fluid System. There is no solution circulation pump. Total prs. is same throughout system. The third fluid (H_2) remains mainly in the evaporator thus reducing partial prs. of refrigerant to enable it to evaporate @ low prs. & hence low temp.

Hydrogen is choosen as it is non-corrosive & insoluble in water. A thermosyphon bubble pump is used to lift the weak-aqua from the generator to the seperator. The discharge tube from the generator is extended down below the liquid level in the generator. The bubbles rise & carry slugs of weak H_2O-NH_3 solution into the seperator.

Two U-bends are provided as vapour-locks to prevent H_2 from getting into the high side (or) solution circuits.

Partial prs. of H_2 provides the prs. difference of NH_3 between the condenser & the evaporator.

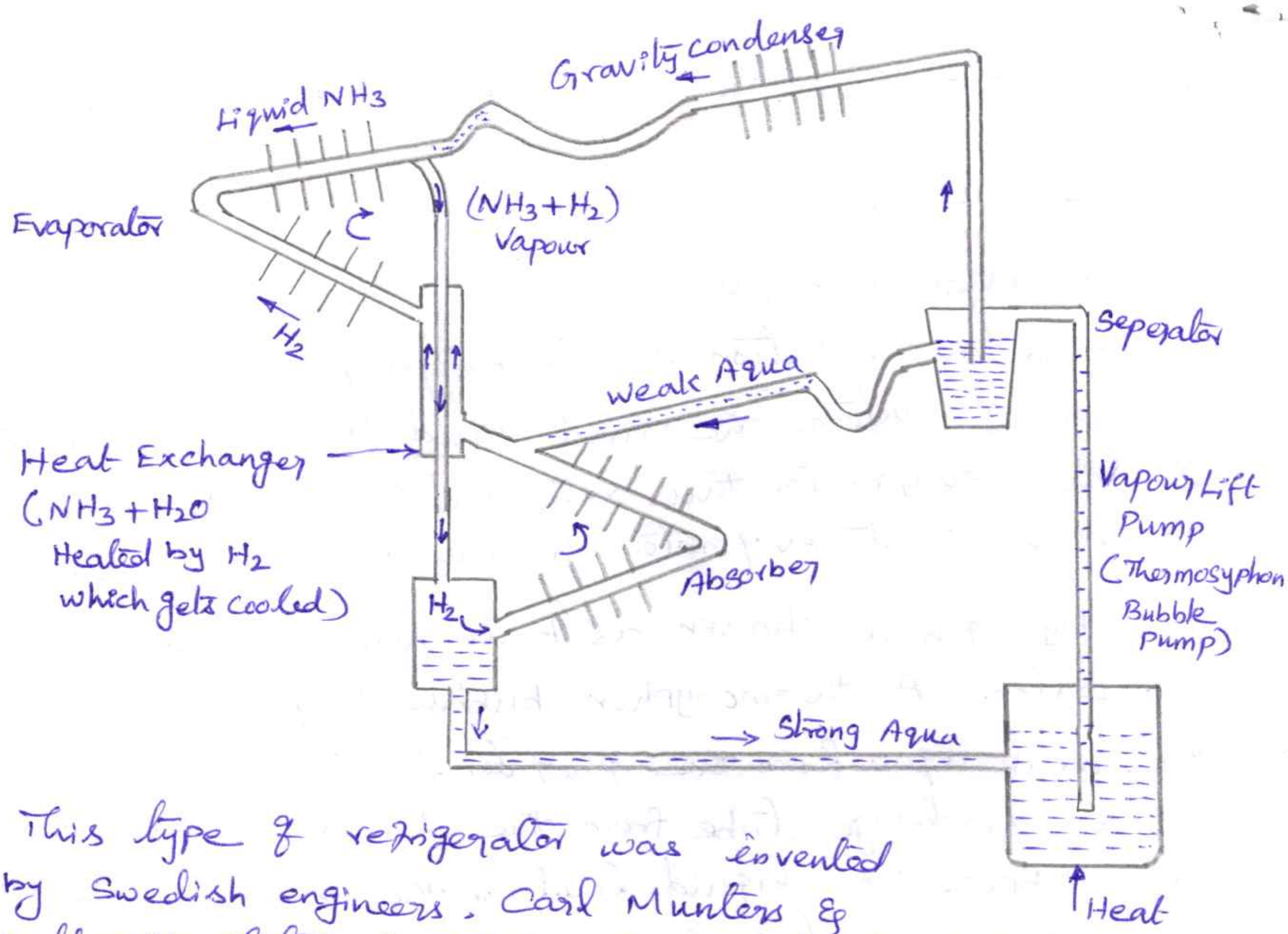
In Condenser pure NH_3 Vapour pressure = Total Pressure.

In evaporator NH_3 Vapour pressure = Total Prs. - Partial Prs. of H_2.

The main advantage of this system is, there is no noise due to absence of mechanical machinery & no mechanical troubles also. It has not been used for industrial applications as the C.O.P is very low.

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

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



This type of refrigerator was invented by Swedish engineers, Carl Munters & BaltzerVonPlatan in 1925. It was developed by the "Electrolux company" of Luton, England.

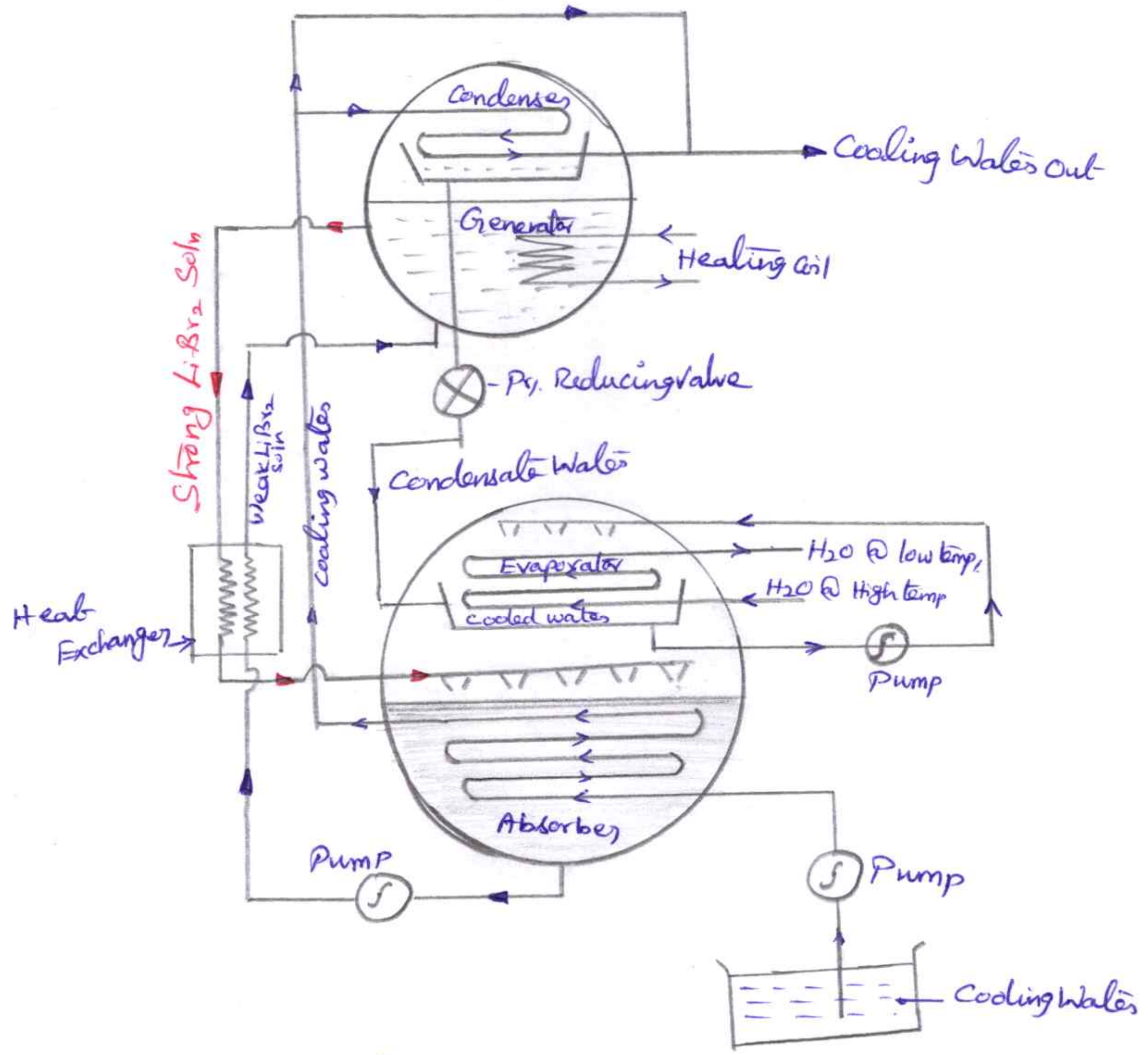
LITHIUM-BROMIDE ($LiBr_2$) ABSORPTION REFRIGERATION SYSTEM:

[$H_2O-LiBr_2$ System]

In Aqua-Ammonia refrigeration system, water as the absorbent & NH_3 used as refrigerant. But in this, water itself works as a refrigerant & $LiBr_2$ salt solution as an absorbent.

In this system, the absorber & the evaporator are placed in one shell which operates at the same low pr. of the system. The generator & condenser are placed in another shell at the same high pressure of the system.

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



The Li-Br$_2$ solution has a strong affinity for water vapour, because of its very low vapour pr. Since Li-Br$_2$ solution is corrosive, therefore, inhibitors should be added in order to protect the metal parts. Lithium-chromate is often used as a corrosion inhibitor. This system is very popular for air-conditioning in which low refrigeration temp. (Not below 0°c) are required.

Types of Water Cooled Condensers:

- Tube-in-tube
- Shell & Coil
- Shell & Tube.

REFRIGERANTS.

The first refrigerant used was ether. employed by "Parkins" in the "hand-operated vapour compression machine.

In the earlier days, ethyl chloride (C_2H_5-Cl) - 1875.
Sulphur dioxide (SO_2) in 1874,
Methyl chloride (CH_3Cl) in 1878,
Carbon dioxide (CO_2) in 1881.

During (1910-1930) many new refrigerants, such as N_2O_3, CH_4, C_2H_6, C_2H_4, C_3H_8 were employed for medium & low-temperature refrigeration.

Hydrocarbons were, however, found extremely inflammable. Dichloromethane (CH_2Cl_2), Dichloroethylene ($C_2H_2Cl_2$) & Mono-bromomethane (CH_3Br) were also used as refrigerants for centrifugal machines.

1930s - Developement of Freons (a trade Name).
It is a great breakthrough occured in the field of refrigeration. Freons are a series of Fluorinated hydrocarbons, generally known as "Fluoro carbons" derived from methane, ethane etc.. as bases.

Presently, the most commonly used organic refrigerants are the Chloro-fluoro derivatives of CH_4 & C_2H_6. The fully halogenated ones with chlorine in their molecules are "chloro-fluorocarbons", referred to as CFCs. Those containing H atom's in the molecules along with Cl & F atoms are referred to as "Hydro-chloro-fluoro carbons" (HCFCs) & those having no chlorine atoms in the molecules are "hydro-fluorocarbon's (HFCs) & simply hydro carbons are HCs. Thus, we have HCs, HFCs, HCFCs & CFCs.

CFCs & HCFCs were identified as high COP refrigerants, with many favourable properties (e.g): Non-flammability, low toxicity & material compatibility that led to their widespread use as refrigerants

Properties of an Ideal refrigerants:

- > Low boiling point & freezing point.
- > low sp. heat of liquid & high sp. heat of vapour,
- > low sp. volume of vapour.
- > low cost,
- > High critical prs. & temp,
- > High latent heat of vapourisation,
- > High thermal conductivity, & high COP,
- > Non-toxic, Non-corrosive to metal, Non-inflammable & Non-explosive,
- > easily & regularly available,
- > easy to liquid at less prs. & temp.
- > Mixes well with oil,
- > Ozone friendly.

Designation of Refrigerants:

The international designation of refrigerants uses 'R' (or alternatively CFC, HCFC, HFC & HC as the case may be) as the designation followed by certain numericals.

⟹ Thus, for a compound derived from a saturated HC denoted by the chemical formula: $C_mH_nCl_pF_q$ in which $(n+p+q) = 2m+2$, the complete designation is $R(m-1)(n+1)(q)$.

A refrigerant's followed by a 2 digits no. – Methane Base, 3 digits no. – Ethane Base,

The no. assigned to HCs & Halo-Carbon refrigerants have a special meaning.

where, m = No. of 'Carbon' atoms,
n = No. of 'Hydrogen' atoms,
p = No. of 'Cl' atoms,
q = No. of 'F' atoms.

Chemical Formula & the Number:

Example:

> Dichloro - difluoro Methane (CCl_2F_2)

No/. of Chlorine atoms, $p = 2$

No/. of Fluorine atoms, $q = 2$

No/. of Hydrogen atoms, $n = 0$

w/ci; $n+p+q = 2m+2 \Rightarrow 0+2+2 = 2m+2$

$\Rightarrow m = 1$

(i.e) No/. of carbon atom's, $m = 1$

∴ The number of Refrigerants becomes,

$R(m-1)(n+1)q \Rightarrow R(1-1)(0+1)(2)$

$\Rightarrow R_{012} = R_{12}$

$\Rightarrow \boxed{CCl_2F_2 \text{ is } R\text{-}12}$ $\boxed{\text{\& Vice versa}}$ - To find chemical Formula.

> Dichloro - tetrafluoro ethane. ($C_2Cl_2F_4$)

$P = 2$, $q = 4$, $n = 0$, $m = 2$ (or) $\left\{\begin{matrix} n+p+q = 2m+2 \\ 0+2+4 = 2m+2 \\ \Rightarrow m = 2. \end{matrix}\right\}$

∴ The no/. of Refrigerants becomes,

$R(2-1)(0+1)4 \Rightarrow R\text{-}114$

$\boxed{C_2Cl_2F_4 \text{ is } R\text{-}114}$

> Trichloro - monofluoro methane (CCl_3F)

$P = 3$, $q = 1$, $n = 0$, $m = 1$.

∴ $R(1-1)(0+1)1 = R\text{-}011$

$\boxed{C \cdot Cl_3F \text{ is } R\text{-}11}$

⇒ The compound derived from a Un-Saturated compound's for which, $(n+p+q) = 2m$, are distinguished by Putting the digit 1 before $(m-1)$.

> Ethylene (C_2H_4) : $P = 0, q = 0, n = 4, m = 2$

(or) $\left\{ \begin{array}{l} n+p+q = 2m \\ 0+0+4 = 2m \Rightarrow m = 2 \end{array} \right\}$

$\therefore R-1\,(m-1)(n+1).q$

$\Rightarrow R-1(2-1)(4+1)0 \Rightarrow R-1150$

$\Rightarrow \boxed{C_2H_4 \text{ is } R\text{-}1150}$

> Propylene (C_3H_6) : $P = 0, q = 0, n = 6, m = 3.$

$\therefore R-1(3-1)(6+1)\,0 \Rightarrow R-1270$

$\Rightarrow \boxed{C_3H_6 \text{ is } R\text{-}1270}$

⇒ In the case of common "In-organic Refrigerants", numerical designations have been given according to their molecular weight added to 700.

Thus, Ammonia : $NH_3 \Rightarrow (N = 14, H_3 = 3 \Rightarrow 17)$

$\therefore \boxed{NH_3 = R\text{-}717}$

Carbon dioxide : $CO_2 \Rightarrow (C = 12, O_2 = 32 \Rightarrow 44)$

$\therefore \boxed{CO_2 = R\text{-}744}$

Sulphur dioxide : $SO_2 \Rightarrow (S = 32, O_2 = 32 \Rightarrow 64)$

$\therefore \boxed{SO_2 = R\text{-}764}$

Water; $H_2O \Rightarrow (H_2 = 2, O = 16 \Rightarrow 18)$ $\therefore \boxed{H_2O = R718}$

Note: Problem arises in the case of "Butane : C_4H_{10}" & higher Hydro-carbon's. In the case of Butane, $n = 10$, a double digit figure. Hence, the prescribed method of designation cannot be used. Accordingly, n-butane & Isobutane are assigned the designations arbitrarily as (R600 & R600a) respectively.

Classification of Refrigerants:

Refrigerants

- > Primary Refrigerants (which directly take part in the refrigeration system.)
  - Halo-Carbon [Organic] Refrigerants.
  - InOrganic Refrigerants.
  - Azetrope "
  - Hydro-Carbon ".
- > Secondary Refrigerants (The refrigerants which are first cooled by primary refrigerants & then used for cooling purpose.)
  - Water
  - Brines
    - Sodium chloride
    - Calcium chloride
    - Propylene Glycol.

> HALO-CARBON REFRIGERANTS:

The "American Society of Heating, Refrigeration & Air Conditioning Engineers" (ASHRAE) identifies 42-50 halo carbon compounds as refrigerants.

This groups of refrigerants was invented & developed by "charles kettering & Dr. Thomas Migely in 1928. These refrigerants are sold in the market under trade names as "Freon", "Genetron", "Isotron" & "Arcton". This group includes refrigerants which contain one (or) more of three halogens, Chlorine, Fluorine & Bromine.

The most of the refrigerants used for domestic, Commercial & Industrial purposes are selected from this group due to their outstanding advantages over the refrigerants from other groups.

Methane-Base

(R-11 - Trichloro-Monofluoro-Methane - CCl_3F

R-12 - Dichloro - Difluoro - methane - CCl_2F_2

R-13 - Monochloro - Trifluoro - methane - $CClF_3$

- Carbon Tetra Fluoride
R-14 - CF_4

- Di-chloro-Monofluoro methane
R-21 - $CHCl_2F$

- Monochloro-Di fluoro methane
R-22 - $CHClF_2$)

- Methylene chloride
R-30 - CH_2Cl_2

- Methyl chloride
R-40 - CH_3Cl

- Ethyl chloride.
R-100 - C_2H_5Cl

(R-113 - $C_2Cl_3F_3$
Trichloro-Trifluoro ethane

- Di-chloro-Tetra fluoro ethane
R-114 - $C_2Cl_2F_4$

- Monochloro-Penta Fluoro Ethane
R-115 - C_2ClF_5

- Di-chloro-Tri fluoro ethane
R-123 - CF_3CHCl_3

R-124 - $CF_3CHCl.F$

- Tetra Fluoro ethane
R-134a - CF_3CH_2F

- Di fluoro ethane
R-152a - CH_3CHF_2)

Ethane-Base

- **Inorganic Refrigerants:** The refrigerants under this group were universally used for all purposes before the introduction of halocarbon group. They are still used for different purposes due to their inherent thermodynamic & physical properties.

R-717 - Ammonia - NH_3 - In Ice plants
R-729 - Air - - - In Aircrafts
R-118 - Water - H_2O - In steam refrig. system.
R-744 - Carbondioxide - CO_2 - In ship.
R-764 - Sulphur dioxide - SO_2 - -

- **Azeotrope Refrigerants:** The refrigeration under this group Consists of mixture of refrigerants whose vapour & liquid phase retain identical compositions over a wide range of temperatures (or) pressures (or) both. They have fixed thermodynamic properties.

R-500 - 73.8% R12 & 26.2% R152 - CCl_2F_2 / CH_3CHF_2
R-502 - 48.8% R22 & 51.2% R115 - $CHClF_2 / CClF_2CF_3$
R-503 - 40.1% R23 & 59.9% R-13 - $CHF_3 / CClF_3$
R-504 - 48.2% R32 & 51.8% R115 - $CH_2F_2 / CClF_2CF_3$

- **Hydro-Carbon Refrigerants:-** Most of the HC refrigerants are successfully used in industrial & commercial installations. They possess satisfactory thermodynamic properties but are highly flammable & explosive. Since the HC refrigerants are not commonly used now-a-days.

R-170 - Ethane (C_2H_6)
R-290 - Propane (C_3H_3)
R-600 - Butane (C_4H_{10})
R-600a - Isobutane (C_4H_{10})
R-1120 - Trichaloro ethylene ($C_2H_4Cl_3$)
R-1130 - Dichloro ethylene ($C_2H_4Cl_2$)

R-1150 - Ethylene (C_2H_4)
R-1270 - Propylene (C_3H_6)

Properties of an ideal Refrigerants:

(a) Thermodynamic Properties.
(b) Safe working ,, (chemical)
(c) Physical ,,
(d)

Thermodynamic Properties:

- > Normal Boiling Point (temperature).
- > Freezing Point
- > Evaporator & condenser pressure.
- > Critical temp. & Pr.
- > Latent heat of refrigerant.
- > Volume of Suction vapour
- > Isentropic discharge temp.
- > COP & horsepower per ton (Power requirements)

chemical Properties:

- > Toxicity
- > Flammability
- > Corrosive property
- > Chemically stability
- > Effect on stored product
- > Action of refrigerant with water (Solubility of water)
- > Action with oil. ((i) Immiscible, (ii) Miscible & (iii) Partially miscible)
- > Action with materials of construction (Effect on perishable materials)

Physical Properties:

- > Dielectric strength.
- > Thermal conductivity
- > Viscosity
- > Heat capacity
- > Surface Tension
- > Leak Tendency
- > Specific volume
- > Sp. heat of liquid & vapour
- > Stability & inertness
- > Cost of refrigerant & availability
- > odour.

SECONDARY REFRIGERANTS:

Under many circumstances, it is not desirable to carry the heat from heat generating source directly by refrigerants, then it is carried by using the 2ndry refrigerant as " Air (or) water (or) brine (or) glycols & sometimes even halocarbons.

The heat carried by the 2ndry refrigerant from the generating source is given to the refrigerant in the evaporator & recirculated again & again.

> Isothermal Refrigeration: Pumping of heat from a constant low temperature.

> Non-Isothermal Refrigeration: Removal of heat @ varying temperature.

> Space Refrigeration: The cold body is at a const. low temp., such as in domestic refrigerator, cold storage etc

> Process Refrigeration: A body is cooled thro' a range of temp. Such as in food freezing, chilling, most chemical processes & even cooling of air for air-conditioning.

SECONDARY REFRIGERANTS:

In large refrigeration plant's, 2ndry refrigerant (or) coolant such as water, brines, glycols & sometimes even halocarbons. This is used instead of directly obtaining it by the evaporating refrigerant at the place of application. This is done in order to reduce the quantity of the refrigerant charge in the system & to reduce pr. losses in lines. The desirable properties of 2ndry refrigerants are low freezing point, low viscosity, non-flammability, good stability & low vapour pr.

Chilled water is used as a 2ndry refrigerant in air-conditioning applications.

For low temp. applications, brines / glycols & halocarbons are used.

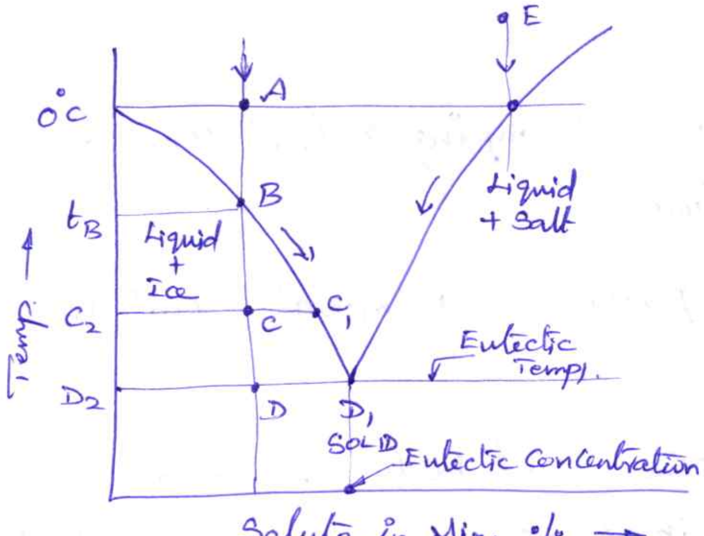


Fig: Phase Diagram for brine.

BRINES:

Brines are formed by dissolving salt in water.

The fun/. of salt in water is to depress its freezing point. The temp/. at which the freezing pt/. is lowest is called the eutectic temp/., & the composition at this temp/. is called the "eutective composition".

If a brine soln/. has less than eutectic composition such as at 'A', its crystallization temp/. (or) freezing pt/. will be lowered to 't_B'. If this soln/. at 'A' is cooled, ice crystals will begin to form at 'B'. As a result, the soln/. will become richer in salt content.

At any point 'C', the mixture will consist of ice at C_2 & soln/. at C_1. At point D, the soln/. will have the composition to D_1. On futher cooling the soln/. will freeze as a whole without any seperation of ice crystals.

Similarly, if a brine soln/. of more than eutectic composition such as at 'E' is cooled, it will first seperate into salt & soln/., untill it reaches pt/. 'D_1' again, whereafter the soln/. will freeze as a whole.

Brines are mainly used in industrial ice plants, cold storages, skating rinks, etc. Common brines are water soln/. of calcium chloride ($CaCl_2$) & Sodium chloride ($NaCl$). Corrosion is, however, a serious prob/. in the use of $CaCl_2$ brine. To reduce corrosion, excessive contact of air with brine should be avoided.

Corrosion inhibitors such as "Sodium Dichromate" are added to keep alkaline conditions. (i.e) A value of pH : 7 to 8.5.

Steam Jet Refrigeration system:

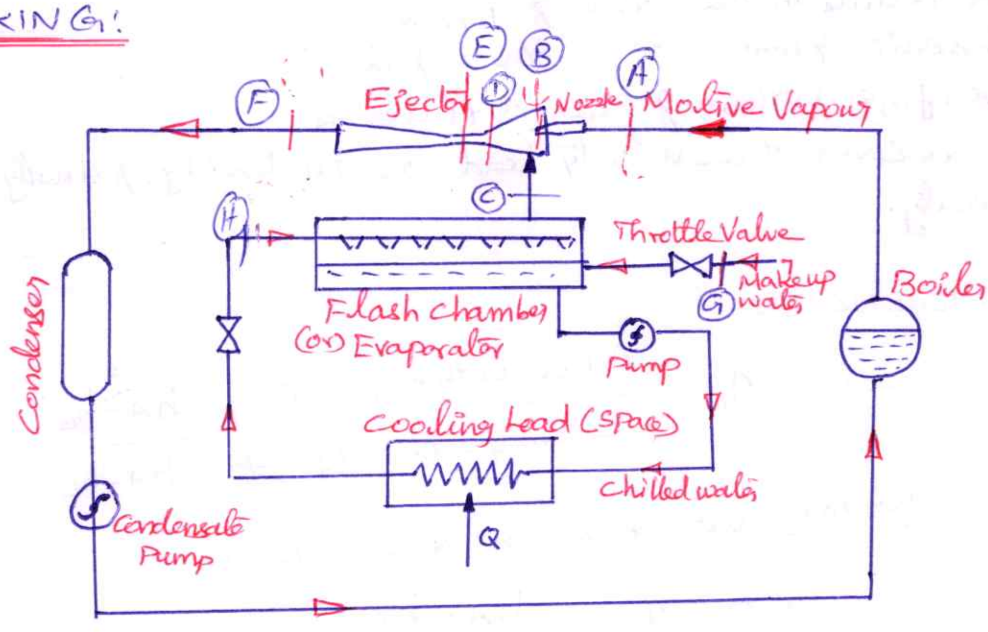
It's oldest method of refrigeration system. In this system the Ejector is used instead of Compressor. In this, water is used as the refrigerant. Since, the freezing point of water is 0°C. Therefore, it can't be used for applications below 0°C. It is used in food processing plants for pre-cooling of vegetables, gas plant, Paper mills, concentrating fruit juice ..etc.

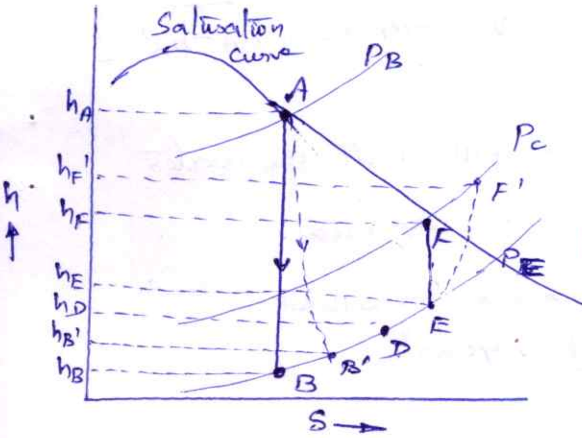
PRINCIPLE:

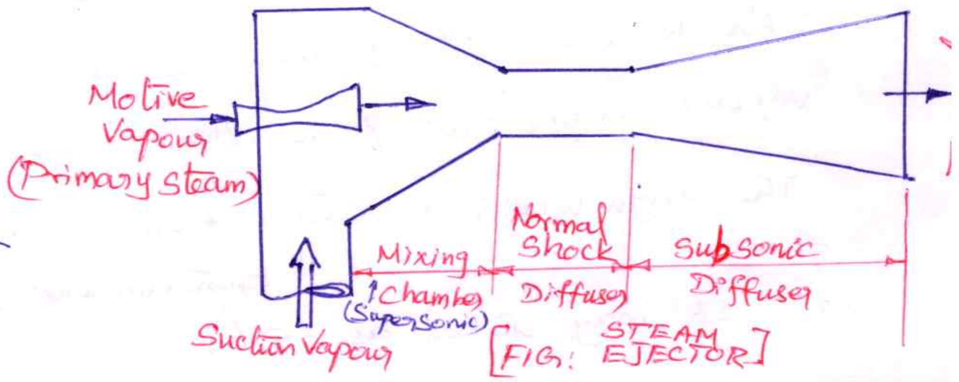
The boiling point of a liquid change with change in external pressure.

@ 1.01325 bar → Boiling pt. of Pure water is 100°C
@ 0.014 bar → " 12°C
@ 0.01 bar → " 7°C.

WORKING:





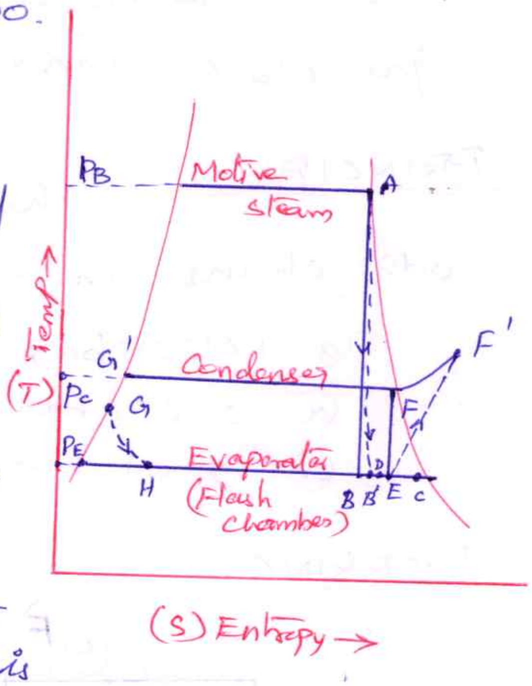


[FIG: STEAM EJECTOR]

(The high pr. steam from the boiler is passed thro' the steam nozzles, thereby increasing the velocity. The high velocity steam in the ejector would entrain the water vapours from the flash chamber which would result in further formation of vapours. The mixture of steam & water vapour passes thro' the venturi tube of the ejector & gets compressed.)

In steam ejector, the velocity of steam 1000 m/s to 1350 m/s

Nozzle pressure ratio : < 200.



The water vapours from the flash chamber are entrained by the high velocity steam & both are mixed in the mixing section @ p=c. The mean velocity of the mixture will be supersonic, after the mixing is complete. This supersonic steam gets a normal shock, in the constant area throat of the diffuser. This results in the rise of pr. & subsonic flow. The fun. of the diverging portion of the diffuser is to recover the velocity head as pr. head by gradually reducing the velocity.

EFFICIENCIES :

> η Nozzle : $$\eta_N = \frac{\text{Actual Enthalpy Drop}}{\text{Isentropic Enthalpy Drop}} = \frac{h_A - h_{B'}}{h_A - h_B}$$

It may vary from 85% to 90%.

> η Entrainment : $$\eta_E = \frac{h_A - h_D}{h_A - h_{B'}} \qquad \left[\because V = 44.72\sqrt{h_A - h_{B'}}\right]$$

It's kinetic energy given the reqd. momentum to the water vapour coming out of the flash chamber (or Evaporator.

"The high velocity of steam from flash chamber is called "Entrainment of vapour".

It may taken as 65%.

∴ $\eta_{compression}$ (or) $\eta_{Diffuser}$,

$$\eta_c = \frac{\text{Isentropic Enthalpy Increase}}{\text{Actual Enthalpy Increase}} = \frac{h_F - h_E}{h_F' - h_E}$$

It may be taken as 75% to 80%.

Mass of Motive steam Required!

Acc. to law of conservation of energy, the available energy for compression must be equal to the energy required for compression.

Let; m_s = Mass of motive steam supplied in kg/min.
m_v = " of water vapour " "
$m = (m_s + m_v)$ = mass of mixture for compression.

wkt,
Available energy for compression $= m_s(h_A - h_D)$
& Energy reqd. for compression $= m(h_F' - h_E)$
$= (m_s + m_v)(h_F' - h_E)$

Now, Acc. to law of Conversation of energy;

$$m_s(h_A - h_D) = (m_s + m_v)(h_F' - h_E) \longrightarrow ①$$

wkt;
$$\eta_N = \frac{h_A - h_B'}{h_A - h_B} \Rightarrow (h_A - h_B') = \eta_N(h_A - h_B)$$

$$\eta_E = \frac{h_A - h_D}{h_A - \eta_B'} \Rightarrow (h_A - h_D) = \eta_E(h_A - h_B')$$
$$= \eta_E \cdot \eta_N(h_A - h_B)$$

$$\eta_c = \frac{h_F - h_E}{h_F' - h_E} \Rightarrow (h_F' - h_E) = \frac{h_F - h_E}{\eta_c}$$

∴ Eqn. ① becomes;

$$m_s \cdot \eta_E \cdot \eta_N(h_A - h_B) = (m_s + m_v)\left(\frac{h_F - h_E}{\eta_c}\right)$$

÷ by (m_v) on both sides. $\Rightarrow$
$$\boxed{\frac{m_s}{m_v} = \frac{(h_F - h_E)}{(h_A - h_B) \cdot \eta_N \cdot \eta_E \cdot \eta_c - (h_F - h_E)}}$$

where; $\frac{m_s}{m_v}$ = Mass of motive steam reqd. / kg of water vapour produced in the flash chamber.

The makeup water is supplied @ pt. 'G' whose temp. is slightly lower than the condenser temp. & is throttled to pt. 'H' in the flash chamber & leaves it corresponding to the condition @ pt. 'C'. Since the enthalpy of water at point 'G' is equal to the enthalpy of water @ pt. 'H'.

∴ For each kg of water vapour formed, heat absorbed is $(h_c - h_{f_G})$ kJ/kg. In other words,

Net Refrigerating Effect, $R_E = m_v(h_c - h_{f_G})$ kJ/min.

If 'Q' TR is the refrigerating load, then the heat absorbed (or) $R_E = 210.Q$ kJ/min.

$$\therefore \boxed{m_v = \frac{210.Q}{h_c - h_{f_G}} \text{ kg/min.}}$$

Since 1 kg of water vapour requires 'm_s' kg of motive steam.

∴ Mass of motive steam required / Q TR load,

= (Mass of water vapour/min) × (Motive steam reqd per kg of vapour)

$= m_v \times m_s$

$$\boxed{m = \frac{210.Q}{h_c - h_{f_G}} \times m_s}$$

& also, C.O.P of the system $= \dfrac{m_v (h_c - h_{f_G})}{m_s (h_A - h_{f_G'})}$

> A steam ejector refrigeration system is supplied with motive steam @ 7 bar saturated with the water in the flash chamber @ 4.5°C. The make up water is supplied to the cooling system @ 18°C & the condenser is operated @ 0.058 bar. The $\eta_N = 88\%$, $\eta_E = 65\%$ & $\eta_c = 80\%$. The quality of steam & flash vapour @ the beginning of compression is 92%. Determine: (i) Mass of motive steam reqd. / kg of flash vapour, (ii) Quality of vapour flashed from the flash chamber, (iii) RE per kg of flash vapour, (iv) mass of motive steam reqd. / hr per TR, (v) Volume of vapour removed from the flash chamber per hr per TR & (vi) COP of the system.

Soln! $P_B = 7$ bar, $t_w = 4.5°C$, $t_{mw} = 18°C$, $P_c = 0.058$ bar, $\eta_N = 0.88$, $\eta_E = 0.65$, $\eta_c = 0.8$, $x_E = 0.92$.

From steam tables of dry saturated steam,

@ $P_B = 7$ bar; $h_A = 2762$ kJ/kg, $S_A = 6.705$ kJ/kg.K, $t_A = 165°c$

@ 4.5°C; $h_{fB} = 18.9$ kJ/kg, $h_{fgB} = 2490.9$ kJ/kg, $S_{fB} = 0.0685$ kJ/kg.

$S_{fgB} = 8.9715$ kJ/kg.K

(i) Mass of motive steam reqd./kg of the flash vapour:

$$\frac{m_s}{m_v} = \frac{(h_F - h_E)}{(h_A - h_B)\eta_N \cdot \eta_E \cdot \eta_c - (h_F - h_E)}$$

where; $\eta_N = \dfrac{h_A - h_B'}{h_A - h_B}$ $\quad \because h_B = h_{fB} + x_B \cdot h_{fgB}$ $\quad (\because S_A = S_B)$

$S_B = S_{fB} + x_B \cdot S_{fgB}$

$\Rightarrow h_B = 18.9 + (0.74 \times 2490.9)$

$\therefore h_B = \underline{\underline{1862.16}}$ kJ/kg

$\Rightarrow 6.705 = 0.0685 + x_B (8.9715)$

$\Rightarrow \underline{\underline{x_B = 0.74}}$

$\Rightarrow 0.88 = \dfrac{2762 - h_B'}{2762 - 1862.16}$

$\Rightarrow h_B' = \underline{\underline{1970.14}}$ kJ/kg

Now, let us find the dryness fraction of steam @ pt. (B') [(i.e) $x_{B'}$]. Since the points B, B', C, D & E lies on the same pr. line

$\therefore h_{fB} = h_{fB'} = h_{fD} = h_{fE} = 18.9$ kJ/kg $= h_{fc}$

$h_{fgB} = h_{fgB'} = h_{fgD} = h_{fgE} = 2490.9$ kJ/kg $= h_{fgc}$

wkT; Enthalpy of B' ; $h_{B'} = h_{fB'} + x_{B'} \cdot h_{fg_{B'}}$

$$1970.14 = 18.9 + (x_{B'} \times 2490.9)$$

$$\Rightarrow x_{B'} = \underline{\underline{0.78}}$$

wkT; $\eta_E = \dfrac{h_A - h_D}{h_A - h_{B'}} \Rightarrow 0.65 = \dfrac{2762 - h_D}{2762 - 1970.14} \Rightarrow h_D = \underline{\underline{2247.3 \text{ kJ/kg}}}$

[Enthalpy of steam@D]

wkT; $h_D = h_{fD} + x_D \cdot h_{fg_D} \Rightarrow 2247.3 = 18.9 + (x_D \times 2490.9)$

$$\Rightarrow x_D = \underline{\underline{0.894}}$$

∴ Enthalpy @ pt. E, $h_E = h_{fE} + x_E \cdot h_{fg_E}$

$$h_E = 18.9 + (0.92 \times 2490.9) = \underline{\underline{2310.5 \text{ kJ/kg}}}$$

Now, let us find the dryness fraction of the mixture of the motive steam & water vapour after isentropic compression @ pt. F. $S_E = S_{fE} + x_E \cdot S_{fg_E} = 0.0685 + (0.92 \times 8.9715)$

$$S_E = \underline{\underline{8.3223}} \text{ kJ/kg.K}$$

From steam table @ $P_c = 0.058$ bar;

$h_{fF} = 148.86$ kJ/kg , $h_{fg_F} = 2417.5$ kJ/kg.

$S_{fF} = 0.512$ kJ/kg.k , $S_{fg_F} = 7.831$ kJ/kg.k.

Since, the compression of mixture is isentropic.

∴ Entropy before compression (S_E) = Entropy after compression (S_F)

$$\Rightarrow 8.3223 = S_F = S_{fF} + x_F \cdot S_{fg_F}$$

$$= 0.512 + x_F \cdot 7.831$$

$$\Rightarrow x_F = \underline{\underline{0.997}}$$

wkT;

$$h_F = h_{fF} + x_F \cdot h_{fg_F} = 148.86 + (0.997 \times 2417.5)$$

$$\Rightarrow h_F = \underline{\underline{2559.1 \text{ kJ/kg}}}$$

wkT; $\eta_c = \dfrac{h_F - h_E}{h_{F'} - h_E} \Rightarrow 0.8 = \dfrac{2559.1 - 2310.5}{h_{F'} - 2310.5} \Rightarrow h_{F'} = \underline{\underline{2621.2 \text{ kJ/kg}}}$

∴ (i) Mass of motive steam reqd./kg of the flash vapour:

$$\Rightarrow \frac{m_s}{m_v} = \frac{(2559.1 - 2310.5)}{(2762 - 1862.16) \times 0.88 \times 0.65 \times 0.8 - (2559.1 - 2310.5)}$$

$$\boxed{\frac{m_s}{m_v} = 1.523 \text{ kg/kg of flash vapour.}}$$

(ii) Quality of vapour flashed from the flash chamber: (x_c)

$$m_v \cdot h_c + h_D \cdot m_s = (m_s + m_v) h_E$$

$$h_c + \frac{m_s}{m_v} \cdot h_D = \left(\frac{m_s}{m_v} + 1\right) h_E$$

$$\Rightarrow h_c + (1.523 \times 2247.3) = (1.523 + 1)\, 2310.5$$

$$\Rightarrow h_c = \underline{2406.8} \text{ kJ/kg.}$$

WKT;

$$h_c = h_{fc} + x_c \cdot h_{fg_c} \Rightarrow 2406.8 = 18.9 + (x_c \times 2490.9)$$

$$\Rightarrow \boxed{x_c = 0.96}.$$

(iii) Refrigerating Effect/kg of flash vapour:

WKT;

$$R_E = (h_c - h_{fG})$$

$$= 2406.8 - 75.5$$

$$\boxed{R_E = 2331.3 \text{ kJ/kg}}$$

From S.Table; @ $t_G = 18°C$; $h_{fG} = 75.5$ kJ/kg

(iv) Mass of motive steam reqd./hr/TR.

$$m = \frac{210 \cdot Q}{h_c - h_{fG}} \cdot \frac{m_s}{m_v} = \frac{210 \times 1}{2406.8 - 75.5} \times 1.523$$

$$\Rightarrow m = 0.133 \text{ kg/min/TR}$$

$$\boxed{m = 7.98 \text{ kg/hr/TR}}$$

(v) Volume of vapour removed from the flash chamber/hr/TR.

w.k.t; Vol. of vapour (per kg) removed from the flash chamber,

V_c = vol. of liquid at c + x_c(vol. of saturated vapour − vol. of liquid)

(from S.Table @ 4.5°C, vol. of saturated vapour = 152.22 m³/kg)

$= 1 + 0.96[152.22 - 1]$

$\Rightarrow V_c = 144.66 \text{ m}^3/\text{kg}.$

∴ Vol. of vapour removed from the flash chamber/hr/TR $\Big\} = V_c \cdot \frac{210 \cdot Q}{h_c - h_{fG}} \times 60$

$= 144.66 \cdot \frac{210 \times 1}{2406.8 - 75.5} \times 60$

$\boxed{= 782 \text{ m}^3/\text{hr}/\text{TR}}$

(vi) C.O.P:

From S.Table; @ $P_c = 0.058$ bar,

$h_{fG}' = 148.8$ kJ/kg.

$$\therefore \text{C.O.P} = \frac{m_r(h_c - h_{fG})}{m_s(h_A - h_{fG}')} = \frac{1(2406.8 - 75.5)}{1.523(2762 - 148.8)} = 0.586$$

$\boxed{\text{C.O.P} = 0.586}$

Merits of Steam Jet Refrigeration System:

- Simple in construction & rigidly designed.
- It's vibration free system as pumps are the only moving parts.
- Low maintenance cost, low production cost & high reliability.
- Water as a refrigerant. So it is very safe, non-poisonous & non-inflammable.
- It is an ability to adjust quickly to load variations.
- Running cost is quite low.

Demerits:

- Not suitable for water below 4°C.
- For proper function, maintenance of high vaccum in the evaporator is necessary.