

-: HAND WRITTEN NOTES:-

OF

156

MECHANICAL ENGINEERING

①

-: SUBJECT:-

REFRIGERATION

& AIR - CONDITIONING

Refrigeration

WWW.GATENOTES.IN

★ Definition:-

It is a process of maintaining lower temp. compare to surrounding. in order to maintain lower temp. continuously. Refrigeration system must run on a cycle.

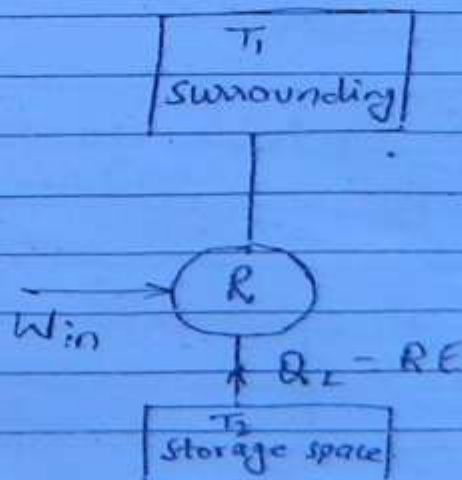
★ Refrigerant:-

Refrigerant is a substance used for producing lower temperature.

E.g. - NH_3 , water, air, R-11, R-12, R-134

★ Refrigeration Effect (RE):-

It is the amount of heat which is to be extracted from storage space in order to maintain lower temp. It is the desired effect of a refrigerator.



$$\text{COP}_R = \frac{Q_2}{W_{in}}$$

$$\text{COP}_R = \frac{\text{RE}}{W_{in}}$$

* Significance of COP:-

(4)

COP Represents Running cost of the system for a given ~~refrigeration~~ Refrigeration Capacity greater the COP lesser is the work input & hence lower is the running cost.

* Unit of Refrigeration:- (TR)

It is the amount of heat that is to be removed from one tonne water at zero (0°C) in order to convert it into ice at 0°C in one day (24 hr.).

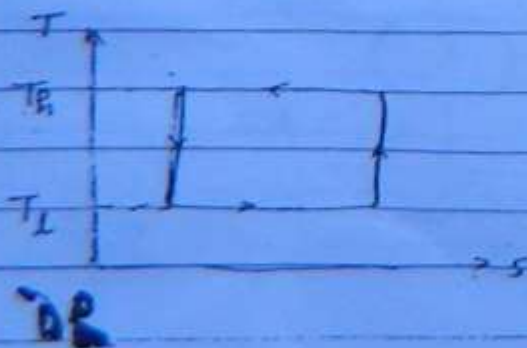
Tonne of Refrigeration Represents Heat transfer rate,

$$1 \text{ T.R.} = 3.5 \frac{\text{KJ}}{\text{sec}} = 3.5 \text{ KW} = 210 \frac{\text{KJ}}{\text{min}}$$

$$1 \text{ TR} = \frac{310 \times 1000}{3600 \times 24} \text{ (L.H.)} = \frac{310 \text{ kg}}{\text{hr}}$$

* Ideal Refrigeration Cycle:-

Reversed Carnot cycle is an ideal refrigeration cycle.



$$\boxed{COP_{max} = \frac{T_L}{T_R - T_L}}$$

This is valid for reversible cycle only. (3)

Refrigeration Capacity. (RC)

$\dot{m}$ = mass flow rate of refrigerant

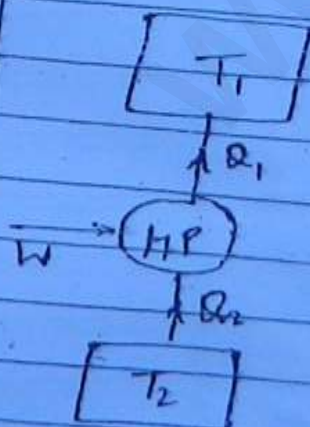
$$RC = \dot{m} \times RE$$

$$\text{Power input} = \dot{m} \times W_{in}$$

$$COP = \frac{RE}{W_{in}}$$

$$COP = \frac{RE \times \dot{m}}{W_{in} \times \dot{m}} = \frac{RC}{P_{in}}$$

$$\boxed{COP = \frac{\text{Refrigerant Capacity}}{\text{Power input}}}$$



$$COP_{HP} = \frac{Q_1}{W}$$

$$COP_{ideal} = \frac{T_L}{T_R - T_L}$$

prob- A Carnot Refrigerator requires 1.5 KW per tonne of Refrigeration to maintain a region at -30°C then the COP is.

Solⁿ:

$$\text{COP} = \frac{R.C.}{P_{in}}$$

(6)

$$P_{in} = 1.5 \text{ KW}$$

$$R.C. = 1 \text{ TR} = 3.5 \text{ KW}$$

$$\text{COP} = \frac{3.5}{1.5} = 2.33 \text{ Ans.}$$

prob- A refrigerating m/c working on reversed Carnot cycle takes out 2 KW from system. While working b/w temp. limit's of 300 K & 200 K. then find the COP & power consumed.

Solⁿ:

$$R.C. = 2 \text{ KW.}$$

$$T_L = 200 \text{ K}$$

$$T_R = 300 \text{ K}$$

$$\text{COP} = \frac{T_L}{T_R - T_L} = \frac{200}{300 - 200} = 2$$

& we also know that.

$$\text{COP} = \frac{R.C.}{P_{in}}$$

$$2 = 2 \text{ kW}$$

$$P_{in}$$

(7)

$$P_{in} = 1 \text{ kW. Ans.}$$

prob- The working temp in evaporator & Condenser coil's of a refrigerator are -30°C & 32°C respectively. If the actual refrigerator has a COP of 0.75 times the maximum COP then find the power input for a refrigerator capacity of 5 kW.

Soln:

$$T_L = -30^{\circ}\text{C} = 30 + 273 = 243 \text{ K.}$$

$$T_H = 32^{\circ}\text{C} = 32 + 273 = 305 \text{ K.}$$

$$(COP)_{max} = \frac{T_L}{T_H - T_L}$$

$$= \frac{243}{305 - 243}$$

$$(COP)_{max} = 3.92$$

$$(COP)_{actual} = 0.75 \times 3.92 = 2.94$$

$$COP = \frac{RC}{P_{in}}$$

$$2.94 = \frac{5 \text{ kW}}{P_{in}}$$

$$P_{in} = \frac{5 \text{ kW}}{2.94} = 1.7 \text{ kW}$$

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$$P_{in} = 1.7 \text{ kW Ans.}$$

prob- The ratio of min. to max. temp. in a Carnot cycle is 0.8. If a heat pump is operated b/w same temp. limits what is its COP.

Solⁿ:- $\frac{T_L}{T_R} = 0.8$

$$COP = \frac{T_R}{T_R - T_L}$$

$$COP = \frac{T_R}{T_R \left(1 - \frac{T_L}{T_R}\right)}$$

$$COP = \frac{1}{\left(1 - \frac{T_L}{T_R}\right)}$$

$$COP = \frac{1}{1 - 0.8}$$

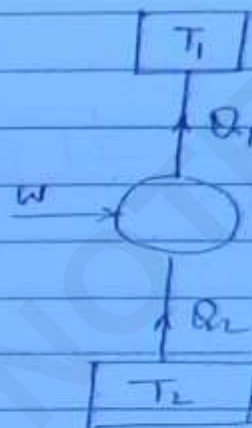
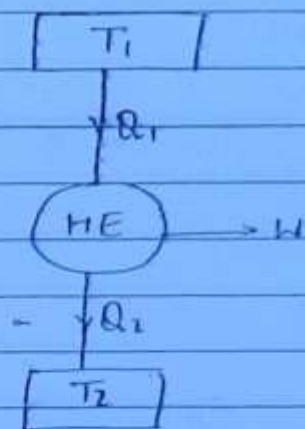
$$COP = 5 \text{ Ans.}$$

prob- A heat engine working b/w T_1 & T_2 has an efficiency of 40%. if the engine is reversed what is the COP of Refrigerator.

Solⁿ:- $(COP)_R = \frac{1 - 0.4}{0.4} = 1.5$ Ans.

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NOTE-



$$\eta = \frac{W}{Q_1}$$

$$(COP)_{HP} = \frac{Q_1}{W} = \frac{1}{W/Q_1}$$

$$(COP)_{H.P.} = \frac{1}{\eta}$$

$$(COP)_{H.P.} = 1 + (COP)_{ref.}$$

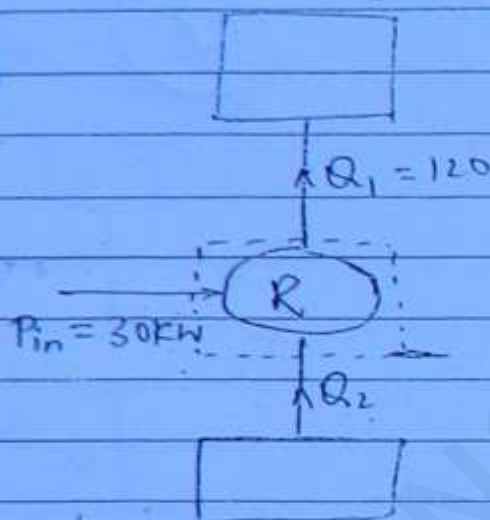
$$\frac{1}{\eta} = 1 + (COP)_{ref.}$$

$$(COP)_{ref.} = \frac{1}{\eta} - 1 = \frac{1 - \eta}{\eta}$$

prob. A Condenser of Refrigeration system rejects heat at the rate of 120kW while the power input is 30kW then what is the COP of a refrigerator.

Soln.

(10)

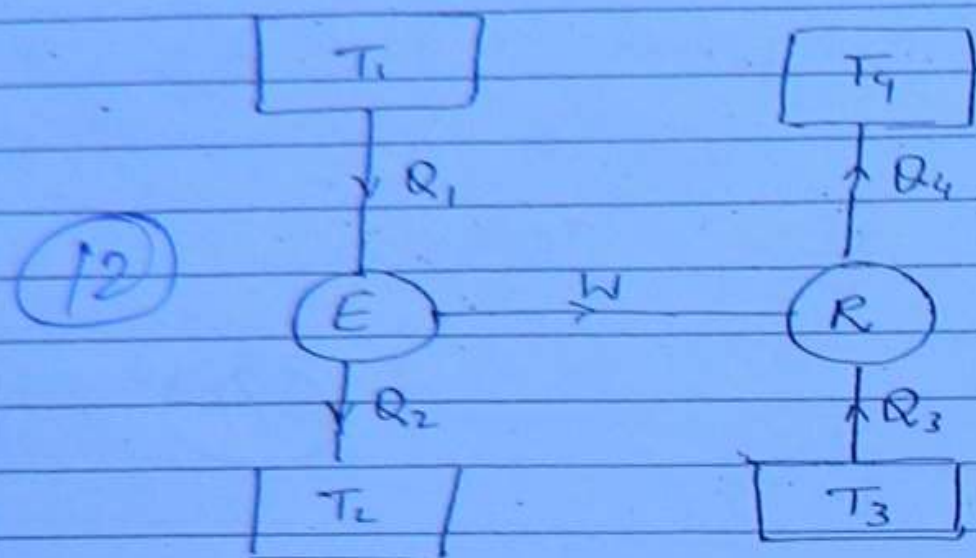


$$30 + Q_2 = 120$$

$$Q_2 = 90$$

$$COP = \frac{R.E.}{P_{in}} = \frac{90}{30} = 3 \text{ Ans.}$$

prob. A refrigerating m/c working on a reversed Carnot cycle consume 6kW of power for producing a refrigeration capacity of 1000kg per min. to maintain a region at -40°C then find the higher temp. in $^\circ\text{C}$.



$$\eta = \frac{W}{Q_1} = 0.4$$

$$W = 0.4 Q_1$$

$$(COP)_R = \frac{Q_3}{W}$$

$$W + Q_3 = Q_4$$

$$Q_3 = Q_4 - W$$

$$Q_3 = Q_4 - (Q_1 - Q_2)$$

$$Q_3 = Q_4 - Q_1 + Q_2$$

$$Q_3 = 3Q_1 - Q_1 = 2Q_1$$

$$(COP)_R = \frac{2Q_1}{0.4 Q_1} = 5$$

Imp.

prob-

A Domestic Refrigerator sit at 2°C handle a refrigeration load of 8000 kJ/day the ambient temp. is 30°C the COP of refrigerator is 0.15 times max. COP then find daily electricity consumption in kW-hr.

Soln:

$$RC = 8000 \text{ kJ/day.}$$

(13)

$$T_L = 2^{\circ}\text{C} = 2 + 273 = 275 \text{ K.}$$

$$T_R = 30^{\circ}\text{C} = 30 + 273 = 303 \text{ K.}$$

$$(COP)_{\max} = \frac{T_L}{T_R - T_L}$$

$$= \frac{275}{303 - 275}$$

$$= 9.82$$

$$(COP)_{\text{act.}} = 0.15 \times 9.82$$

$$= 1.473$$

$$COP = \frac{RC}{P_{\text{in}}}$$

$$1.473 = \frac{8000 \text{ kJ/day}}{P_{\text{in}}}$$

$$P_{\text{in}} = 5431 \text{ kJ/day}$$

Per day Engine input = 5431 KJ

per day consumption = $5431 \frac{\text{KJ}}{\text{sec}} \times \text{sec}$

(14)

= 5431 KW - sec.

But 1 hr = 3600 sec.

$$1 \text{ sec} = \frac{1}{3600} \text{ hr.}$$

per day consumption = $5431 \text{ KW} \times \frac{1}{3600} \text{ hr.}$

= 1.508 KW-hr. Ans.

Q. A refrigerator storage is supplied with 3600 kg of a substance at a temp. of 27°C the substance has to be cooled to -23°C the cooling takes place in 10 hr. the specific heat of substance is 2 KJ/kgK above freezing point & 0.5 KJ/kgK below freezing point & the freezing point is -3°C . latent heat of freezing is 230 KJ/kg .

Find the power required if the actual COP is half the max. COP.

Soln: $T_L = -23^\circ\text{C} = 23 + 273 = 250\text{K}$

$$T_R = 27^\circ\text{C} = 27 + 273 = 300\text{K}$$

$$(\text{COP})_{\max} = \frac{T_L}{T_R - T_L}$$

$$= \frac{250}{300 - 250}$$

$$= 5$$

$$(\text{COP})_{\text{act.}} = \frac{1}{2} \times 5 = 2.5$$

$$\boxed{(\text{COP})_{\text{act.}} = 2.5}$$

$$(\text{COP})_{\text{act.}} = \frac{RC}{P_{\text{in}}}$$

$$2.5 = \frac{RC}{P_{\text{in}}}$$

$$\boxed{P_{\text{in}} = \frac{RC}{2.5}}$$

$$27^\circ\text{C} \longrightarrow -23^\circ\text{C}$$

$$27^\circ\text{C} \longrightarrow -3^\circ\text{C} \longrightarrow SH_1 = mC_1(\Delta t)_1$$

$$-3^\circ\text{C} \longrightarrow -3^\circ\text{C} \longrightarrow L.H. = m(L.H.)$$

$$-3^\circ\text{C} \longrightarrow -23^\circ\text{C} \longrightarrow SH_2 = mC_2(\Delta t)_2$$

$$\text{Total heat removed} = RC = mC_1(\Delta t_1) + mLH + mC_2(\Delta t_2)$$

(16)

$$RC = m(C_1 \Delta t_1 + LH + C_2 \Delta t_2)$$

$$RC = \frac{3600 \text{ kg}}{10 \times 3600 \text{ sec.}} \left[2(27 - (-3)) + 230 + 0.5[73 - (-23)] \right]$$

$$RC = \frac{1}{10} [60 + 230 + 10] = 30 \text{ kW}$$

$$P_{in} = \frac{30 \text{ kW}}{2.5}$$

$$P_{in} = 12 \text{ kW Ans.}$$

Vapour Compression Refrigeration System (VC System):-

(19)

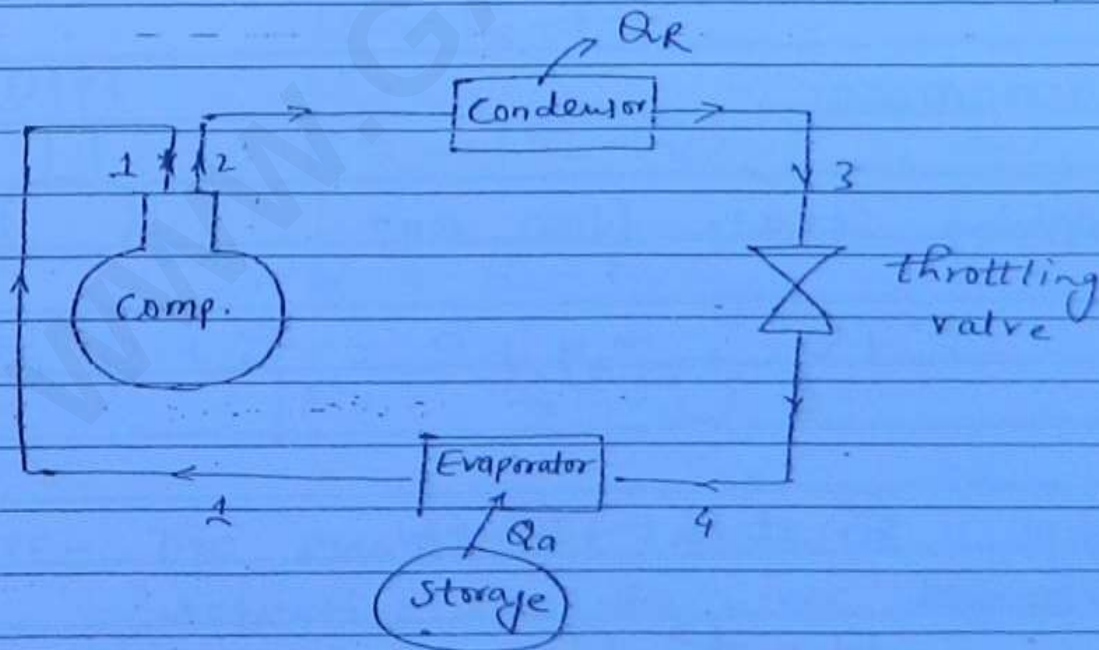
* Simple V.C. cycle or Saturated V.C. cycle:-

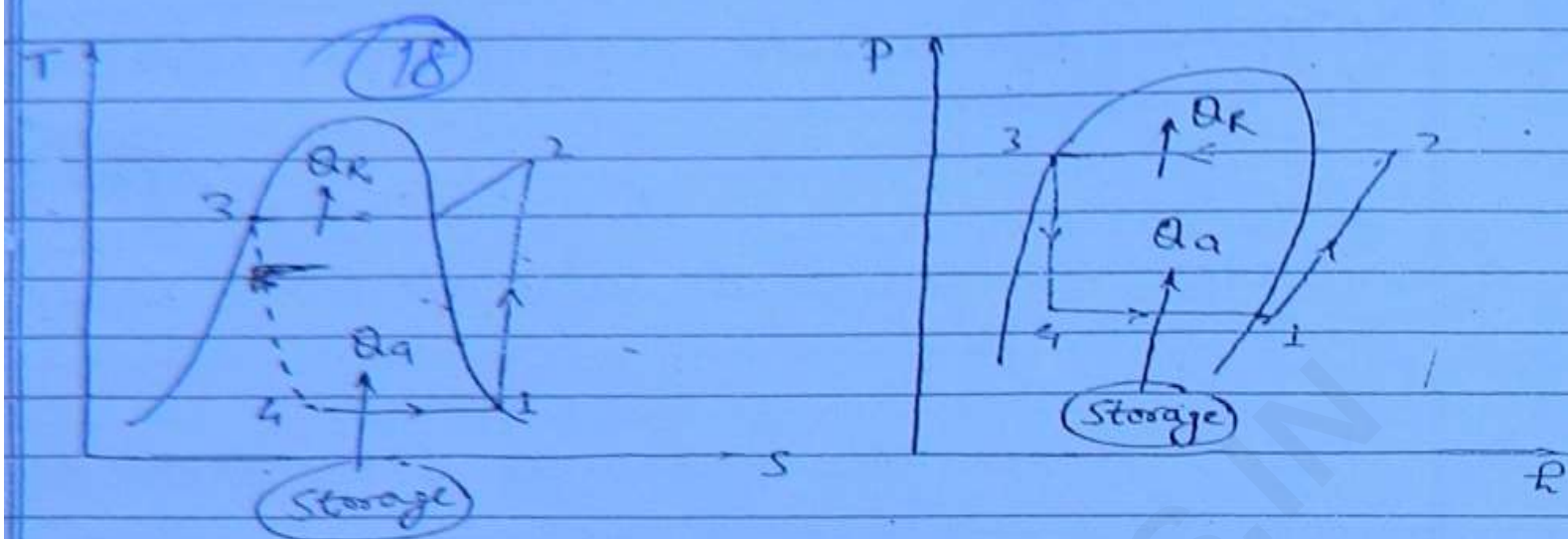
1-2 $\rightarrow$ Reversible adiabatic (Isentropic) compression

2-3 $\rightarrow$ Constant pressure heat rejection.

3-4 $\rightarrow$ throttling (Isenthalpic expansion)

4-1 $\rightarrow$ Const. press. heat rejection, absorption

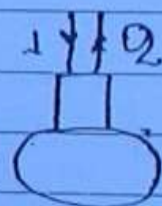




Each device is an open system & operates ~~up~~ under steady-flow condition and hence steady flow energy eqn. can be applied.

Compressor :-

Applying steady flow eqn.



$$h_1 + \frac{C_1^2}{2} + z_1 g + Q = h_2 + \frac{C_2^2}{2} + z_2 g + W$$

$$h_1 + Q = h_2 + W$$

$$W = h_1 - h_2$$

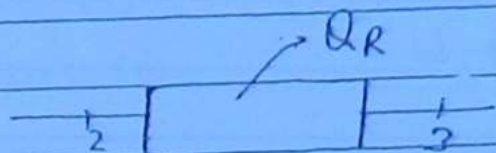
$$W = -(h_2 - h_1)$$

$$-W = h_2 - h_1$$

$$W_{\text{input}} = h_2 - h_1$$

(ii) Condensor:-

(19)



Applying steady flow eqn.

$$h_2 + \frac{c_2^2}{2} + z_2 g + Q = h_3 + \frac{c_3^2}{2} + z_3 g + W$$

$$h_2 + Q = h_3 + W$$

$$-Q = h_3 - h_2$$

$$Q = -(h_2 - h_3)$$

$$-Q = h_2 - h_3$$

$$Q_R = h_2 - h_3$$

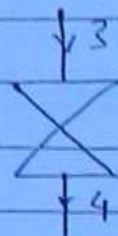
NOTE - The purpose of condenser is only heat rejection not the work rejection and transfer.

(iii) Throttling valve or Expansion device

$$h_3 + \frac{C_3^2}{2} + Z_3 g + Q = h_4 + \frac{C_4^2}{2} + Z_4 g + W$$

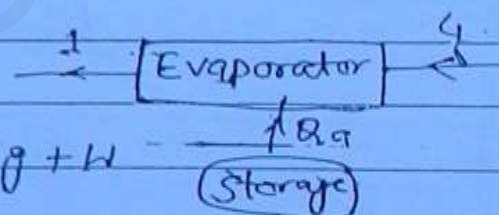
$$h_3 + Q = h_4 + W$$

$$[h_3 = h_4]$$



(iv) Evaporator :-

$$h_4 + \frac{C_4^2}{2} + Z_4 g + Q = h_1 + \frac{C_1^2}{2} + Z_1 g + W$$



$$h_4 + Q = h_1 + W$$

$$Q = h_1 - h_4$$

$$[Q_a = h_1 - h_4]$$

$$[RE = h_1 - h_4]$$

In Evaporator only
Heat transfer not
the work transfer.

$$COP = \frac{h_1 - h_4}{h_2 - h_1}$$

$$COP = \frac{RE}{W_{in}}$$

$$COP = \frac{h_1 - h_3}{h_2 - h_4}$$

NOTE - COP of V.C. Cycle is High & hence this cycle is used mostly in Refrigeration Systems.

(21)

★ Volumetric efficiency of a Reciprocating Compressor :-

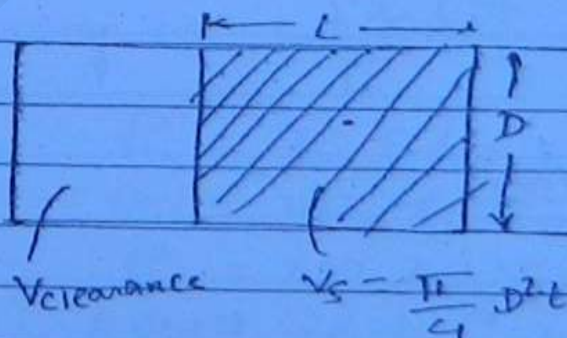
It is the breathing capacity of a compressor & it is defined as the ratio of actual volume entering the compressor to the swept volume.

$\dot{m}$ = mass flow rate of refrigerant (kg/s)

v_1 = sp. volume at the inlet to the compressor

$$V_{act} = \dot{m} v_1 \quad \frac{\text{kg}}{\text{s}} \times \frac{\text{m}^3}{\text{kg}}$$

$$\eta_{vol.} = \frac{V_{act}}{V_{swept}}$$



$$\eta_{\text{vol.}} = \frac{\dot{m} v_1}{\frac{\pi D^2 L}{4} \times N \times K}$$

(22)

D = Bore dia or piston dia.

L = stroke length.

K = number of compressor.

N = speed in rps

NOTE - $RE = Q_a = h_1 - h_4$

- $RE = h_1 - h_4 = h_1 - h_3$

$W_{in} = h_2 - h_1$

$RE = \dot{m} \times RE$

$RC = \dot{m} \times (h_1 - h_4)$

$P_{in} = \dot{m} \times W_{in}$

$P_{in} = \dot{m} (h_2 - h_1)$

$W_{in} = (h_2 - h_1) \frac{KJ}{kg}$

$\dot{m} = \frac{kg}{sec.}$

$P_{in} = \frac{kg}{s} \times \frac{KJ}{kg} = KW$

prob. In V.C. cycle following data is obtained.

Enthalpy at the inlet to the compressor is 180 KJ/kg . & at the exit of compressor is 210 KJ/kg . enthalpy at the exit of condenser is 80 KJ/kg find the COP. (23)

Soln: given, $h_1 = 180 \text{ KJ/kg}$.
 $h_2 = 210 \text{ KJ/kg}$.
 $h_3 = 80 \text{ KJ/kg}$.

$$\text{COP} = \frac{h_1 - h_3}{h_2 - h_1} = \frac{180 - 80}{210 - 180} = 3.33 \text{ Ans.}$$

prob. In a 5 kW cooling capacity Refrigeration system operating on V.C. cycle refrigerant enters evaporator with an enthalpy of 75 KJ/kg & leaves with an enthalpy of 183 KJ/kg . Enthalpy of Refrigerant after compression is 210 KJ/kg calculate -

- (i) COP (ii) Power input to be compressor in KW (iii) Rate of heat transfer at the condenser in KJ/sec .

Soln: given, $h_4 = 75 \text{ KJ/kg} = h_1$
 $h_2 = 183 \text{ KJ/kg}$
 $h_3 = 210 \text{ KJ/kg}$
 $RC = 5 \text{ KW}$

$$COP = \frac{h_1 - h_3}{h_2 - h_1} = \frac{183 - 75}{210 - 183}$$

$$COP = 4 \text{ Ans.}$$

(24)

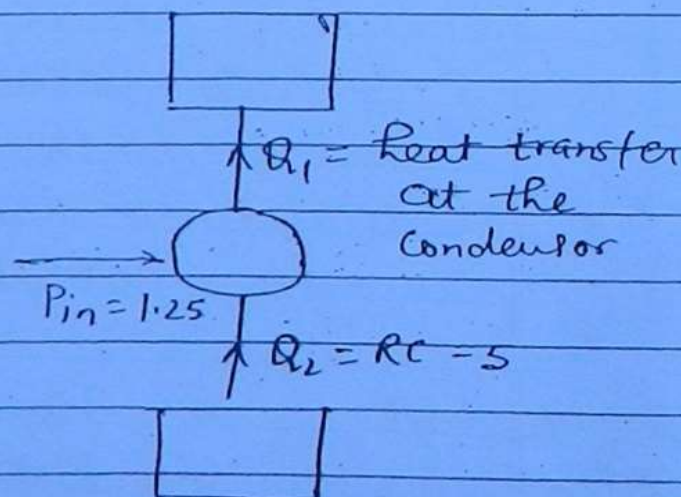
& we also know that,

$$COP = \frac{R_C}{P_{in}}$$

$$4 = \frac{5 \text{ KW}}{P_{in}}$$

$$P_{in} = \frac{5 \text{ KW}}{4} = 1.25 \text{ KW. Ans.}$$

$1.25 + 5 =$ heat transfer
at the
Condensor.



temp.

NOTE- Higher Side Condensor heat transfer.

but lower temp. Side Evaporator
heat transfer.

Imp.

prob. In a simple v.c. cycle followings are properties of Refrigerant at various point

Compressor inlet $h_1 = 183.2 \text{ kJ/kg}$, $v_1 = 0.076$

Compressor discharge $h_2 = 222.6 \text{ kJ/kg}$, $v_2 = 0.0614$

Condenser Exit $h_3 = 84.9 \text{ kJ/kg}$, $v_3 = 0.00083$

The swept volume for the compressor is 1.5 lit & its volumetric efficiency is 80%. Speed of the compressor is 1600 r.p.m. find

- ① Power input to the Compressor in kW.
- ② RC in kW.

Soln: given, Swept volume $= V_s = \frac{\pi D^2 L}{4} = 1.5 \text{ liter}$

but 1 liter $= 10^{-3} \text{ m}^3$

(25)

$$\eta_{\text{vol.}} = 0.8$$

$$N = 1600 \text{ rpm} = \frac{1600}{60} \text{ rps.}$$

We know that volumetric efficiency

$$\eta_{\text{vol.}} = \frac{m v_1}{\frac{\pi D^2 L N}{4}}$$

$$0.8 = \frac{\dot{m} \times 0.0767}{1.5 \times 10^3 \times \frac{1600}{60}}$$

$$\boxed{\dot{m} = 0.417 \text{ kg/sec.}}$$

(26)

$$RC = \dot{m} (h_1 - h_3)$$

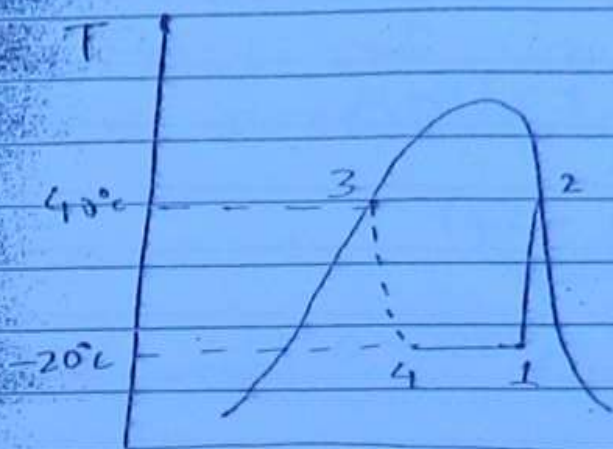
$$= 0.417 (183.2 - 84.9)$$

$$\boxed{RC = 40.98 \text{ kW.}} \text{ Ans.}$$

A Refrigerator based on V.C. Cycle operates b/w temp. limits of -20°C & 40°C the refrigerant enters the condenser as saturated vapour & leaves as saturated liquid the enthalpy & entropy of saturated liquid & saturated vapour at these temp. are given in the table. If the refrigerant flow rate is 0.025 kg/sec find -

RC in kW & COP of the system.

$t(^{\circ}\text{C})$	$h_f (\text{kJ/kg})$	$h_g (\text{kJ/kg})$	$s_f (\frac{\text{kJ}}{\text{kg K}})$	$s_g (\frac{\text{kJ}}{\text{kg K}})$
-20	20	180	0.07	0.7366
40	80	250	0.3	0.67



$$COP = \frac{h_1 - h_3}{h_2 - h_1}$$

$$h_2 = h_g \text{ at } 40^\circ\text{C}$$

$$h_2 = 200 \text{ kJ/kg}$$

$$h_3 = h_f \text{ at } 40^\circ\text{C}$$

$$h_3 = 80 \text{ kJ/kg}$$

$$h_1 = h_f + x(h_g - h_f)$$

$$h_1 = 20 + x(180 - 20)$$

But 1-2 is isentropic process.

$$s_1 = s_2$$

$$s_2 = s_g \text{ at } 40^\circ\text{C}$$

$$s_2 = 0.67 \text{ kJ/kgK}$$

$$S_1 = S_2 = 0.67 \text{ kJ/kgK}$$

$$S_1 = S_f + x(S_g - S_f)$$

$$0.67 = 0.07 + x(0.7366 - 0.7)$$

$$x = 0.9$$

(28)

$$h_1 = 20 + 0.9(180 - 20)$$

$$h_1 = 164 \text{ kJ/kg}$$

$$\text{COP} = \frac{h_1 - h_3}{h_2 - h_1} = \frac{164 - 80}{200 - 164} = 2.33 \text{ Ans.}$$

$$\begin{aligned} \text{RC} &= \dot{m} \times (h_1 - h_3) \\ &= 0.025(164 - 80) \end{aligned}$$

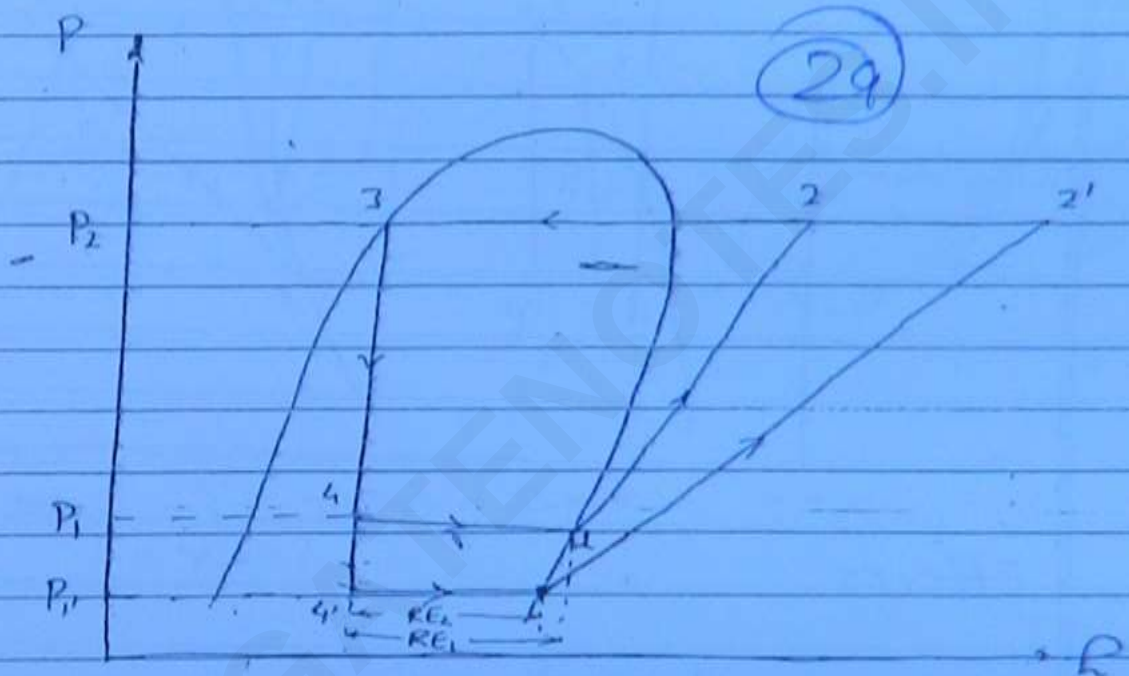
$$\boxed{\text{RC} = 2.1 \text{ kW} \text{ Ans.}}$$

—————> —————> —————>

★ Effect of variation of properties on the performance of V.C. cycle.

Case - I.

- Decrease in Evaporator pressure:



Effect's:

1. Decrease in R.E.

$$\eta_{vol.} = 1 + C - C \left(\frac{P_2}{P_1} \right)$$

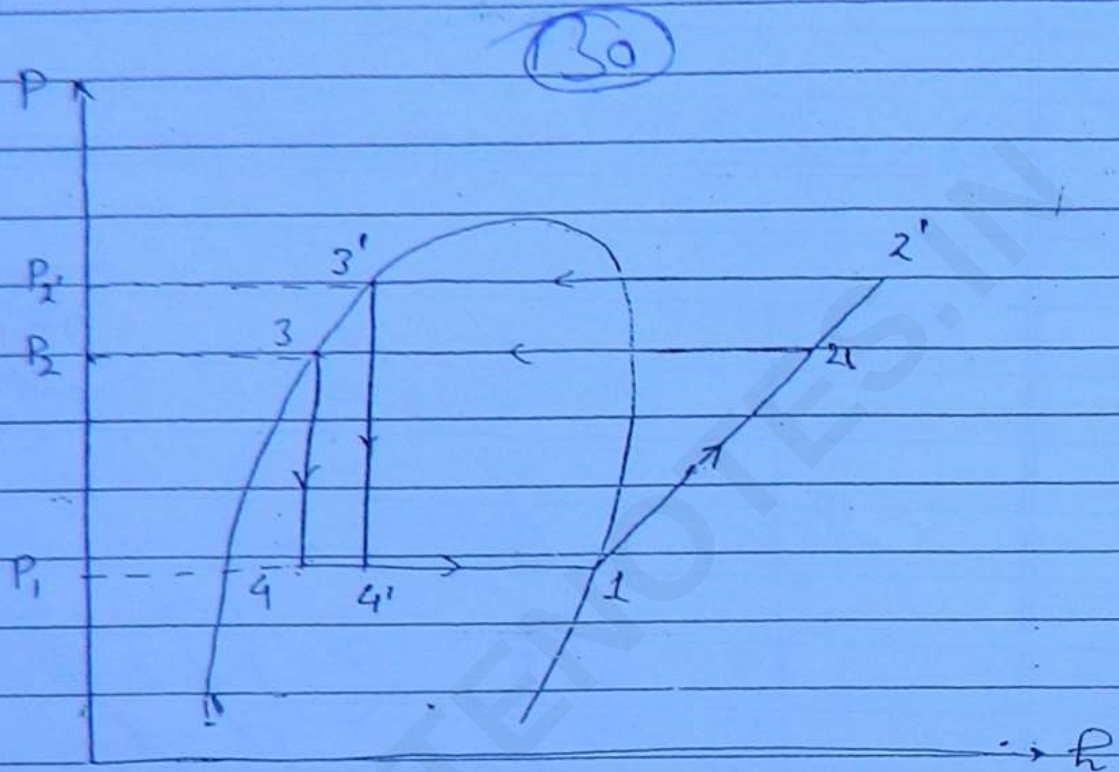
2. Increase in work input.

3. Decrease in COP.

4. Decrease in volumetric efficiency due to increase in pressure ratio.

Case-II.

- Increase in Condensor pressure :-



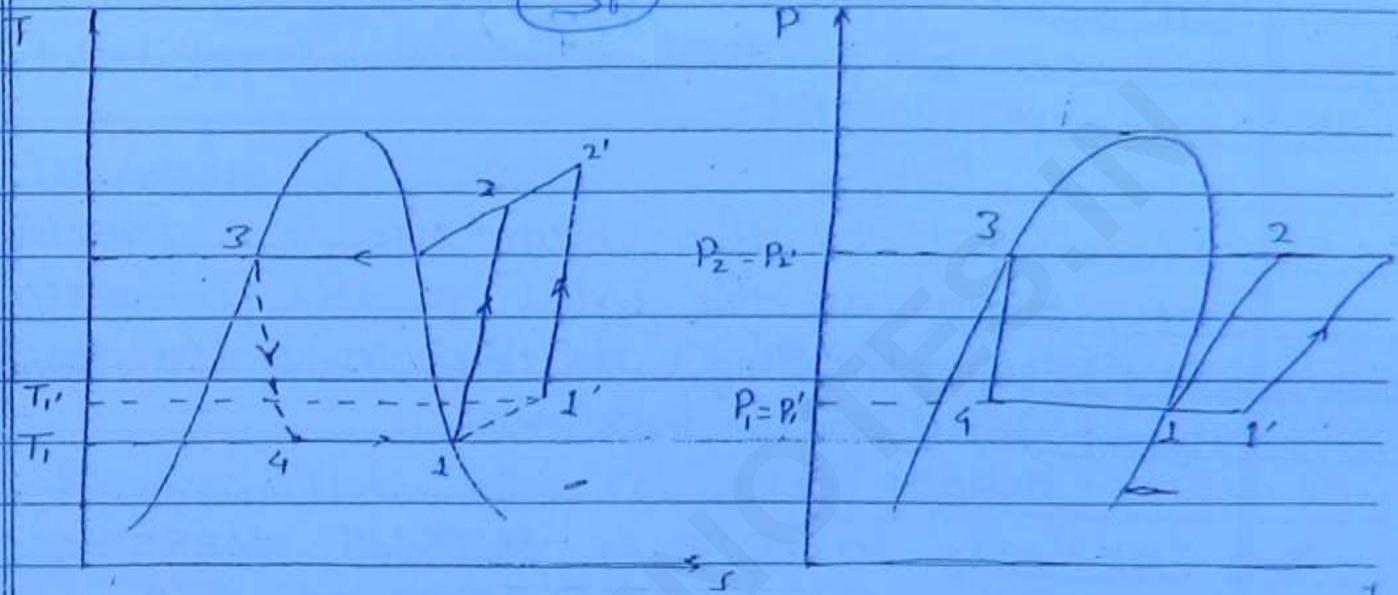
Effect's.

1. Decrease in R.E.
2. Increase in Work input.
3. Decrease in COP.
4. Decrease in volumetric efficiency due to increase in pressure ratio.

Case - III.

• Superheating :-

(3)



$$W = \frac{\gamma}{\gamma-1} (P_1 V_1 - P_2 V_2)$$

$$= \frac{\gamma}{\gamma-1} (mRT_1 - mRT_2)$$

$$= \frac{\gamma}{\gamma-1} mRT_1 \left(1 - \frac{T_2}{T_1}\right)$$

$$= \frac{\gamma}{\gamma-1} mRT_1 \left[1 - \left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}}\right] \quad \text{for adiabatic temp. ratio.}$$

$$W = f(T_1)$$

$$W_{12} = f(T_1) \quad \& \quad W_{12} = f(T_1')$$

$$T_1 > T_1$$

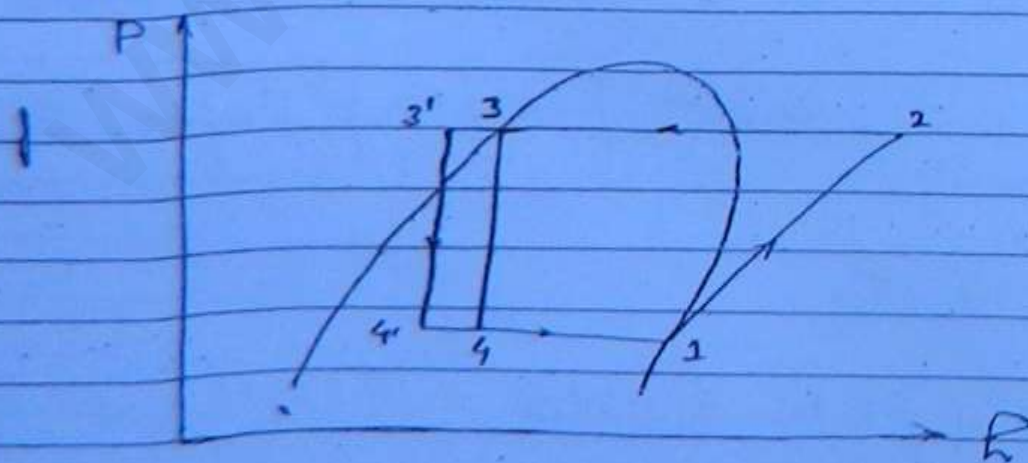
Effect's:-

(32)

1. increase in R.E.
2. Work input to the compressor is a function of temp. at the inlet to the compressor with superheating as the temp. increases work input also increases
3. COP may increase or decrease depending on the refrigerant. in case of R-12 superheating results in increase in COP whereas in case of NH_3 superheating results in decrease in COP.

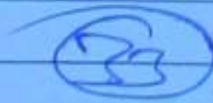
Case - IV

• Sub-cooling :-



Effect's :-

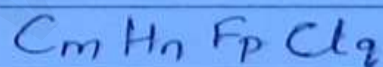
1. Increase in R.E.
2. Work input remain's same.
3. Increase in COP.



* Nomenclature of a refrigerant
or
Designation of a refrigerant

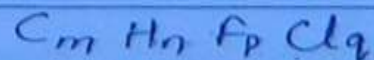
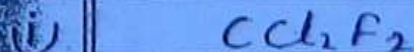
Case-I.

- Saturated hydrocarbon refrigerant



$$R - (m-1)(n+1)P$$

$$\& \quad n + p + q = 2m + 2$$



$$m=1, n=0, p=2, q=2$$

$$R - (1-1)(0+1)^2$$

$$[R - 12]$$

ii)

If

$$R - 11$$

(34)

F

$$R - 011$$

$$R - (m-1)(n+1)p$$

$$m-1 = 0 \Rightarrow m = 1$$

$$n+1 = 1 \Rightarrow n = 0$$

$$p = 1$$

$$0 + 1 + q = 2(1) + 2$$

$$1 + q = 4$$

$$q = 3$$

then

$$C_m H_n F_p Cl_q$$

$$[R - 11 \rightarrow CFCl_3]$$

iii)

$$R - 22$$

$$R - 022$$

$$R - (m-1)(n+1)p$$

$$m-1 = 0 \Rightarrow m = 1$$

$$n+1 = 2 \Rightarrow n = 1$$

$$p = 2$$

(38)

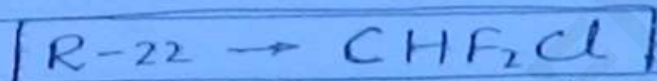
$$n+p+q = 2m+2$$

$$1+2+q = 2(1)+2$$

$$q = 4-3$$

$$q = 1$$

then $C_m H_n F_p Cl_q$



R-113

$$m-1 = 1 \Rightarrow m = 2$$

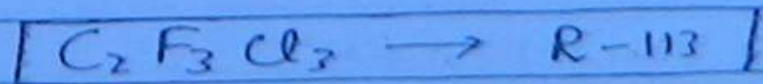
$$n+1 = 1 \Rightarrow n = 0$$

$$p = 3$$

$$0+3+q = 2(2)+2$$

$$q = 3$$

$C_m H_n F_p Cl_q$



Tri-chloro - Tri-fluoro ~~and~~ Ethane.

(v)

R-134

$$R = (m-1)(n+1)P$$

$$m-1 = 1 \Rightarrow m = 2$$

$$n+1 = 3 \Rightarrow n = 2$$

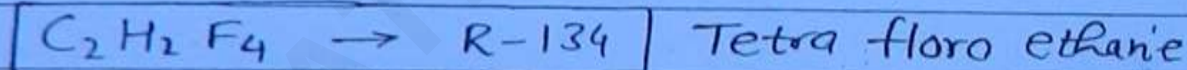
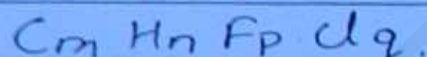
$$P = 4$$

36

$$n + P + Q = 2m + 2$$

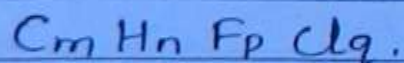
$$2 + 4 + Q = 2(2) + 2$$

$$Q = 0$$



Case-II

- Unsaturated Hydrocarbon refrigerant's



$$R - 1 (m-1)(n+1)P$$

$$Q \quad n + P + Q = 2m$$

(i) C_2H_4 (Ethylene).

Compare to $C_mH_nF_pCl_q$.

$$m=2, n=4, p=0, q=0$$

$$R-1(2-1)(4+1)0$$

$$R-1150$$

(37)

Case - III.

÷ In-organic Refrigerant :-

eg. 700 + molecular weight.

$$NH_3 - 17 \rightarrow R-700+17 \Rightarrow R-717$$

$$H_2O - 18 \rightarrow R-718$$

$$Air - 29 \rightarrow R-729$$

$$SO_2 - 64 \rightarrow R-764$$

$$CO_2 - 44 \rightarrow R-744$$

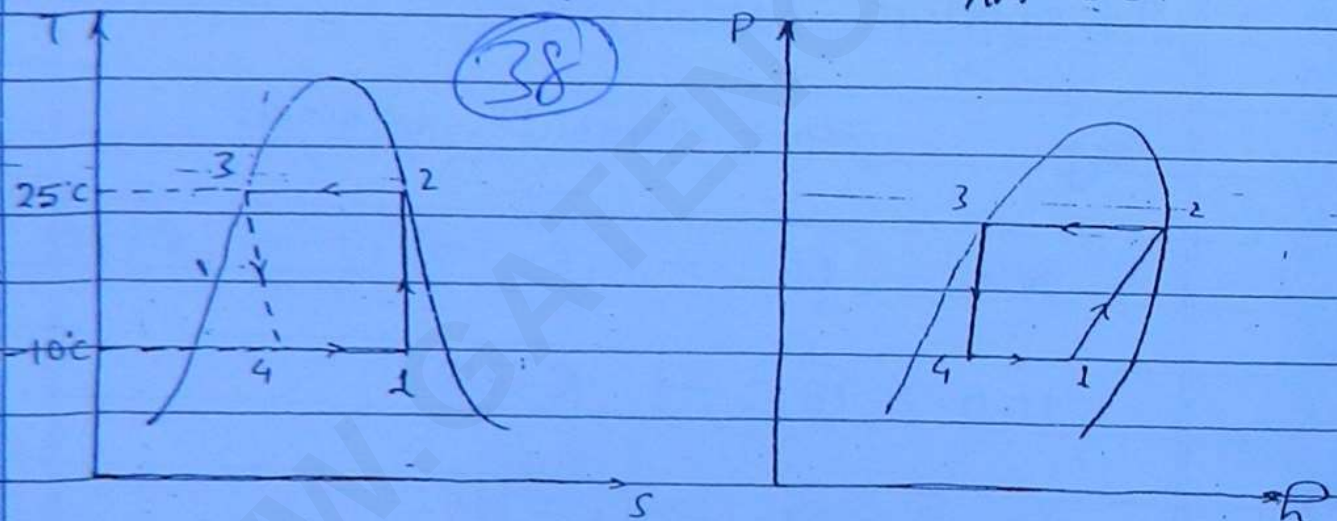
$$Air \rightarrow \left. \begin{array}{ll} N_2 - 77\% & 28 \\ O_2 - 23\% & 32 \end{array} \right\}$$

prob- The temp. limit's of Ammonia Refrigeration system are 25°C & -10°C , Refrigerant enters the condenser as saturated vapour & leaves the condenser as saturated liquid. Assuming compression to be isentropic. find the COP. use the following table. /

$t(^{\circ}\text{C})$	$h_f (\text{KJ/Kg})$	$h_g (\text{KJ/Kg})$	$s_f (\text{KJ/Kg-K})$
25	298.9	1166.94	1.1242
-10	135.37	1297.68	0.5443

Solⁿ:

Ans - 6.89



$$\text{COP} = \frac{h_1 - h_3}{h_2 - h_1}$$

$$h_2 = h_g = h_g - h_f = 1166.94 - 298.9 = 868.04 \text{ KJ/Kg}$$

$$h_3 = h_f = 298.9$$

$$h_1 = h_f + x(h_g - h_f)$$

$$S_1 = S_2 = 11242 + 1166.94 = 1168.06 \text{ kJ/kg}$$

$$S_1 = S_f + xL$$

(59)

$$h_2 = h_{f2} + LH = 298.9 + 1166.94 = 1465.84 \text{ kJ}$$

$$h_3 = h_{f3} = 298.9 \text{ kJ/kg}$$

$$h_1 = h_{f1} + x_1(LH) = 135.37 + x_1(1297.68)$$

$$S_2 = S_g = S_{f2} + \frac{LH}{T_{\text{sat}}} = 11242 + \frac{1166.94}{298}$$

$$S_2 = 5.04 \text{ kJ/kgK}$$

$$S_1 = S_2 = 5.04 \text{ kJ/kgK} \quad \text{then } h_1 = 1377.55$$

$$S_1 = S_{f1} + x \frac{LH}{T_{\text{sat}}}$$

$$\boxed{\text{COP} = 6.8 \text{ Ans.}}$$

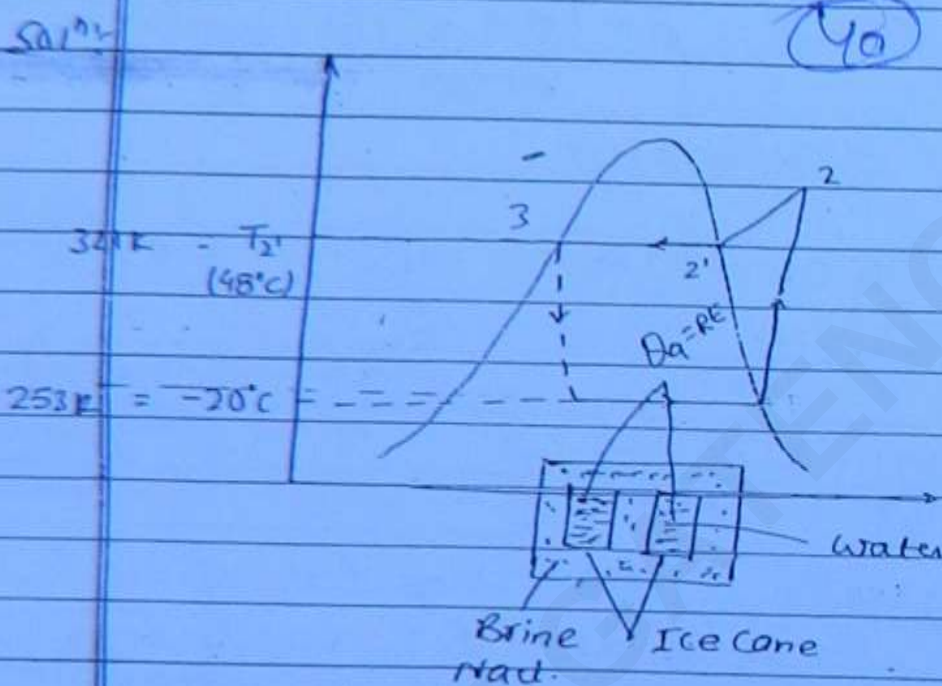
$$5.04 = 0.5443 + x \frac{1297.68}{263}$$

$$\boxed{x = 0.911}$$

A V.C. Refrigeration system using R-12 produces 8640 kg of ice per day the condensing and evaporating temp. are 48°C & -28°C the cycle is simple V.C. cycle water at 35°C is used to form ice & temp. of ice should be -8°C heat flow into the brine tank from surroundings may be taken as 10% of total heat removed from water to form ice at -8°C . Determine the power required to drive the plant take specific heat of ice as 2.26 kJ/kgK latent heat of ice equal to 334.72 kJ/kg & sp. heat

of water 4.18 kJ/kgK & specific heat of vapour refrigerant is equal to 0.82 kJ/kgK . Use the following table.

$t(^{\circ}\text{C})$	$P(\text{bar})$	$h_f (\text{kJ/kg})$	$h_g (\text{kJ/kg})$	$s_f (\text{kJ/kgK})$	$s_g (\text{kJ/kgK})$
48	11.64	82.83	205.83	0.2973	0.682
-20	1.51	17.82	178.74	0.0731	0.7082



$$P = \dot{m} (h_2 - h_1)$$

$$h_1 = h_g \text{ at } -20^{\circ}\text{C}$$

$$h_1 = 178.74 \text{ kJ/kg}$$

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

$$h_2 = 205.83 + 0.82 (T_2 - 321)$$

Since the 1-2 is isentropic process.

then $S_1 = S_2 = S_g \text{ at } -20^\circ\text{C} = 0.7087 \text{ kJ/kgK}$.

$$S_2 = S_2' + C_{pv} \ln \frac{T_2}{T_2'}$$

(41)

$$0.7087 = 0.682 + 0.82 \times \ln \frac{T_2}{321}$$

$$T_2 = 331.6 \text{ K}$$

then $h_2 = 205.83 + 0.82(331.6 - 321)$

$$h_2 = 214.522 \text{ kJ/kg}$$

we know that

$$RC = \dot{m}(h_1 - h_3)$$

$$RC = \dot{m}(178.74 - 82.83)$$

we convert H_2O at 35°C to ice at -8°C

$$\text{SH} \rightarrow 35^\circ\text{C H}_2\text{O} \rightarrow 0^\circ\text{C H}_2\text{O} \rightarrow m C_w (35 - 0)$$

$$\text{H}_2\text{O} \text{ at } 0^\circ\text{C} \rightarrow \text{ice at } 0^\circ\text{C} \rightarrow m L_H$$

$$\text{ice at } 0^\circ\text{C} \rightarrow \text{ice at } -8^\circ\text{C} \rightarrow m C_{ice} [0 - (-8)]$$
$$= m C_{ice} \times 8$$

$$\text{Heat removed} = m C_w \times 35 + m L_H + m C_{ice} \times 8$$

$$= m [C_w \times 35 + L_H + C_{ice} \times 8]$$

$$= \frac{8640 \text{ kg}}{24 \times 3600 \text{ sec}} [4.18 \times 35 + 334.72 + 2.26 \times 8]$$

$$\text{Heat removed} = 49.91 \text{ kW}$$

(99)

As 10% of 49.91 is leak's into tank /

$$\therefore \text{Total heat removed} = 49.91 + \frac{10}{100} \times 49.91$$

$$RC = 54.9 \text{ kW}$$

$$RC = \dot{m} (178.74 - 82.83)$$

$$54.9 = \dot{m} (178.74 - 82.83)$$

$$\dot{m} = 0.572 \text{ kg/sec.}$$

$$P = 0.572 (214.52 - 178.74)$$

$$P = 20.4 \text{ kW.}$$

Prob- A simple v.c. cycle is designed for a load of 10 TR. Storage and surrounding's temp. are 0°C & 30°C respectively. A minimum temp. difference of 5°C is required in condenser and evaporator for effective heat transfer calculate -

- i) mass flow rate of refrigerant.

(ii) Power required in kW.

(iii) COP.

^{Temp.}
(iv) Cylinder dimension's

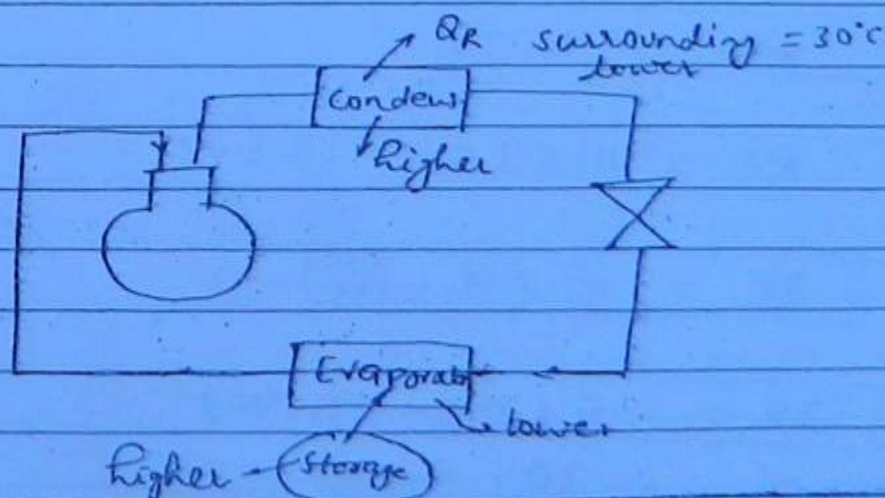
(43)

Assume volumetric efficiency as 90%.
Compressor's speed is equal to 3000 r.p.m.
Take stroke equal to 1.2 times Bore.

$t(^{\circ}\text{C})$	$P(\text{bar})$	$v_g(\text{m}^3/\text{kg})$	$h_f(\text{kJ/kg})$	$h_g(\text{kJ/kg})$	$s_f(\text{kJ/kg})$
36	8.6811	0.02034	234.94	367.6	1.1186
30	7.236	0.03124	223.6	361.2	1.0032
0	3.28	0.05286	201.32	353	0.9963
-6	2.519	0.0677	194.45	350	0.9796

$t(^{\circ}\text{C})$	$s_g(\text{kJ/kg})$	$C_{p\text{vapour}}(\text{kJ/kg})$
36	1.547	0.744
30	1.549	0.719
0	1.559	0.686
-6	1.5618	0.619

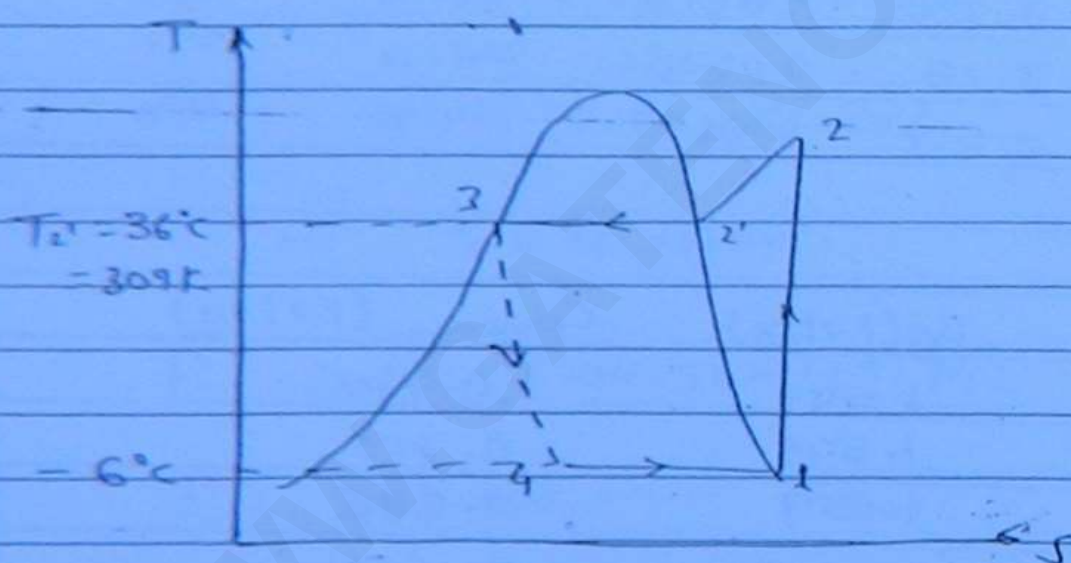
Soln:



In Condensor refrigerant reject heat to surrounding and hence its temp. must be greater than that of surrounding therefore temp. of refrigerant in condensor is $30 + 6 = 36^{\circ}\text{C}$.

(44)

In Evaporator refrigerant absorb heat Q from storage place and hence its temp. must be lower than that of storage place temp. i.e. temp. of refrigerant in evaporator is $0 - 6 = -6^{\circ}\text{C}$.



ii)

$$RC = 10 TR$$

$$RC = 10 \times 3.5 = 35 \text{ kW}$$

$$RC = \dot{m} (h_1 - h_3)$$

$$35 = \dot{m} (350 - 234.94)$$

$$\dot{m} = 0.394 \text{ kg/sec.}$$

$$(ii) \quad P_{in} = \dot{m} (h_2 - h_1)$$

$$P_{in} = 0.304 (h_2 - 350)$$

The process 1-2 is isentropic process.

$$s_1 = s_2 = s_g \text{ at } -6^\circ\text{C} = 1.5618 \text{ kJ/kgK.}$$

(15)

$$s_2 = s_{2'} + C_{pv} \ln \frac{T_2}{T_{2'}}$$

$$1.5618 = 1.547 + 0.744 \ln \frac{T_2}{309}$$

$$T_2 = 315.2 \text{ K.}$$

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

$$h_2 = 367.6 + 0.744 (315.2 - 309)$$

$$h_2 = 372.21 \text{ kJ/kg}$$

$$P_{in} = 0.304 (372.21 - 350)$$

$$P_{in} = 6.75 \text{ kW. Ans.}$$

$$(iii) \quad \text{COP} = \frac{h_1 - h_2}{h_2 - h_1} = \frac{RC}{P_{in}} = \frac{35}{6.75} = 5.185 \text{ Ans}$$

(iv)

$$\eta_{vol.} = \frac{m \dot{V}_1}{\frac{\pi}{4} D^2 L N}$$

$$0.9 = \frac{m \dot{V}_1}{\frac{\pi}{4} D^2 L N}$$

(96)

$$\frac{\pi}{4} D^2 L N = \frac{m \dot{V}_1}{0.9}$$

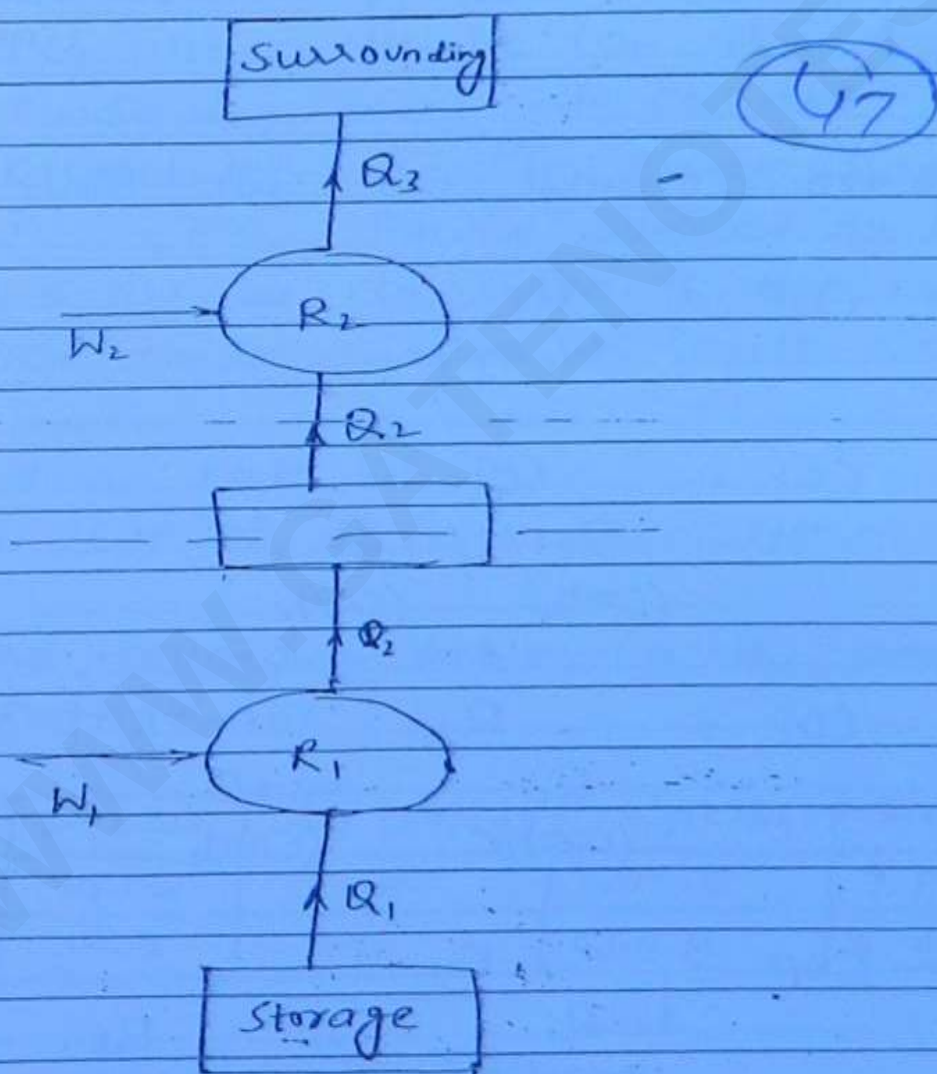
$$\frac{\pi}{4} D^2 (1.2D) \times \frac{350}{60 \text{ sec.}} = \frac{0.304 \times 0.0677}{0.9}$$

$$D = 0.169 \text{ m or } 16.9 \text{ cm. Ans.}$$

$$L = 1.2D = 1.2 \times 16.9 = 20.3 \text{ cm. Ans.}$$

CASCADE REFRIGERATION

If very low temp. are desired then the corresponding evaporator pressure is also very low & this results in higher pressure ratio & due to this higher pressure ratio there is a reduction in volumetric efficiency. To avoid this cascade refrigeration system is used as shown in figure.



COP of overall plant:

$$\boxed{\text{COP} = \frac{Q_1}{W_1 + W_2} = \frac{\text{Desired effect}}{\text{Work input}}}$$

COP of 1st system.

$$(COP)_1 = \frac{Q_1}{W_1} \Rightarrow W_1 = \frac{Q_1}{(COP)_1}$$

COP of 2nd system.

(98)

$$(COP)_2 = \frac{Q_2}{W_2} \Rightarrow W_2 = \frac{Q_2}{(COP)_2}$$

Then from overall COP, we find.

$$COP = \frac{Q_1}{W_1 + W_2}$$

$$COP = \frac{Q_1}{\frac{Q_1}{(COP)_1} + \frac{Q_2}{(COP)_2}}$$

$$COP = \frac{Q_1}{\frac{Q_1}{(COP)_1} + \frac{W_1 + Q_1}{(COP)_2}}$$

$$COP = \frac{Q_1}{\frac{Q_1}{(COP)_1} + \frac{W_1}{(COP)_2} + \frac{Q_1}{(COP)_2}}$$

$$COP = \frac{Q_1}{\frac{Q_1}{(COP)_1} + \frac{1}{(COP)_2} \times \frac{Q_1}{(COP)_1} + \frac{Q_1}{(COP)_2}}$$

$$\text{COP} =$$

$$\frac{1}{(\text{COP})_1} + \frac{1}{(\text{COP})_1 (\text{COP})_2} + \frac{1}{(\text{COP})_2}$$

$$\text{COP} = \frac{(\text{COP})_1 (\text{COP})_2}{1 + (\text{COP})_1 + (\text{COP})_2}$$

(49)

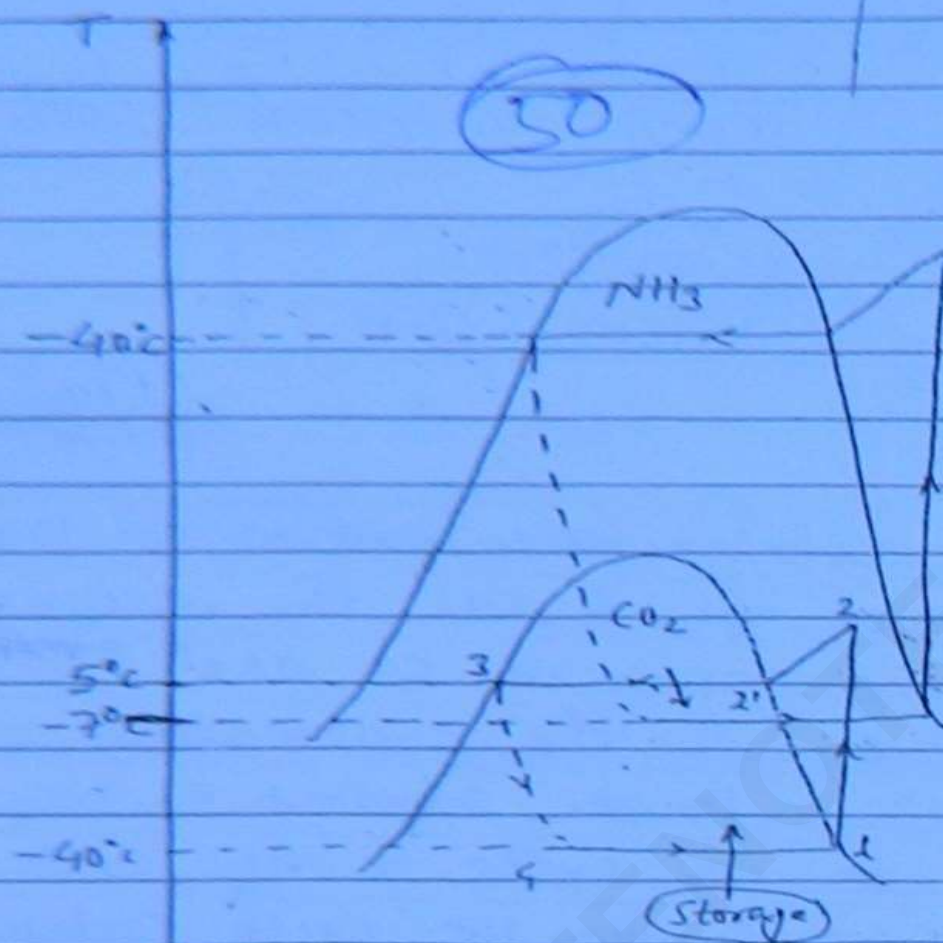
b A Cascade Refrigeration system of 100 TR capacity uses NH_3 & CO_2 the evaporating and condensing temp. of CO_2 are -40°C & 5°C respectively. & the evaporating temp. of NH_3 is -70°C . Power supplied to NH_3 compressor is 96.5 kW. ~~in~~ CO_2 cycle & NH_3 cycle are assumed to be simple V.C. cycle Calculate -

- (i) mass flow rate of CO_2 .
- (ii) overall COP of Refrigeration system.

Use the following table for the properties of CO_2 Refrigerant.

$t(^\circ\text{C})$	$P(\text{bar})$	$h_f(\text{kJ/kg})$	$h_g(\text{kJ/kg})$	$s_f(\text{kJ/kgK})$	$s_g(\text{kJ/kgK})$
-40	10.05	332.7	652.8	3.8531	5.2262
5	39.7	431	649.8	4.2231	5.0037

$$C_{p \text{ vapour}} = 0.85 \text{ kJ/kg-K}$$



$$\dot{R}_C = 100 \text{ TR}$$

$$1 \text{ TR} = 3.5 \text{ kW}$$

$$\dot{R}_C = 350 \text{ kW}$$

$$\dot{R}_C = \dot{m}_{\text{CO}_2} (h_1 - h_3)$$

$$350 = \dot{m}_{\text{CO}_2} (652.8 - 431) \quad |$$

$$\dot{m}_{\text{CO}_2} = 1.578 \text{ kg/s. Ans.}$$

$$COP = \frac{RC}{P_{in}}$$

$$RC = 350 \text{ kW}$$

$$P_{in} = P_{CO_2} + P_{NH_3}$$

(51)

$$P_{NH_3} = 96.5 \text{ kW}$$

$$P_{CO_2} = \dot{m}_{CO_2} (h_2 - h_1)$$

$$h_1 = 652.8 \text{ kJ/kg}$$

$$\dot{m}_{CO_2} = 1.57 \text{ kg/s}$$

We know that 1-2 is isentropic process.

$$S_1 = S_2 = S_g \text{ at } -40^\circ\text{C} = 5.2262 \text{ kJ/kg}$$

$$\text{But } S_2 = S_2' + C_{pv} \ln \frac{T_2}{T_2'}$$

$$5.2262 = 5.0037 + 0.85 \ln \frac{T_2}{278}$$

$$T_2 = 361.18 \text{ K}$$

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

$$h_2 = 649.8 + 0.85 (361.18 - 278)$$

$$h_2 = 720.5 \text{ kJ/kg}$$

$$P_{CO_2} = \dot{m}_{CO_2} (h_2 - h_1)$$

$$= 1.57 (720.5 - 652.8)$$

$$= 106.3 \text{ KW}$$

(52)

$$COP = \frac{RC}{P_{CO_2} + P_{NH_3}}$$

$$COP = \frac{350}{106.3 + 96.5}$$

$$COP = 1.72 \text{ Ans.}$$

* Advantages of dry compression over wet compression or the disadvantages of wet compression:

1. Wet compression represents incomplete vaporization of refrigerant and thus there is reduction in refrigeration effect.
2. The liquid refrigerant may wash away lubricating oil thus increasing wear & tear.
3. Liquid refrigerant may damage compressor valves.

$$\text{COP} = \frac{RE}{W_{in}} = \frac{h_{1'} - h_4}{h_2 - h_4} = \frac{h_{1'} - h_{3'}}{h_2 - h_4}$$

Heat exchanger concept.

Heat loss = Heat gain.

(54)

$$h_3 - h_{3'} = h_1 - h_{1'}$$

$$C_{\text{liquid}} (T_3 - T_{3'}) = C_{\text{vap.}} (T_1 - T_{1'})$$

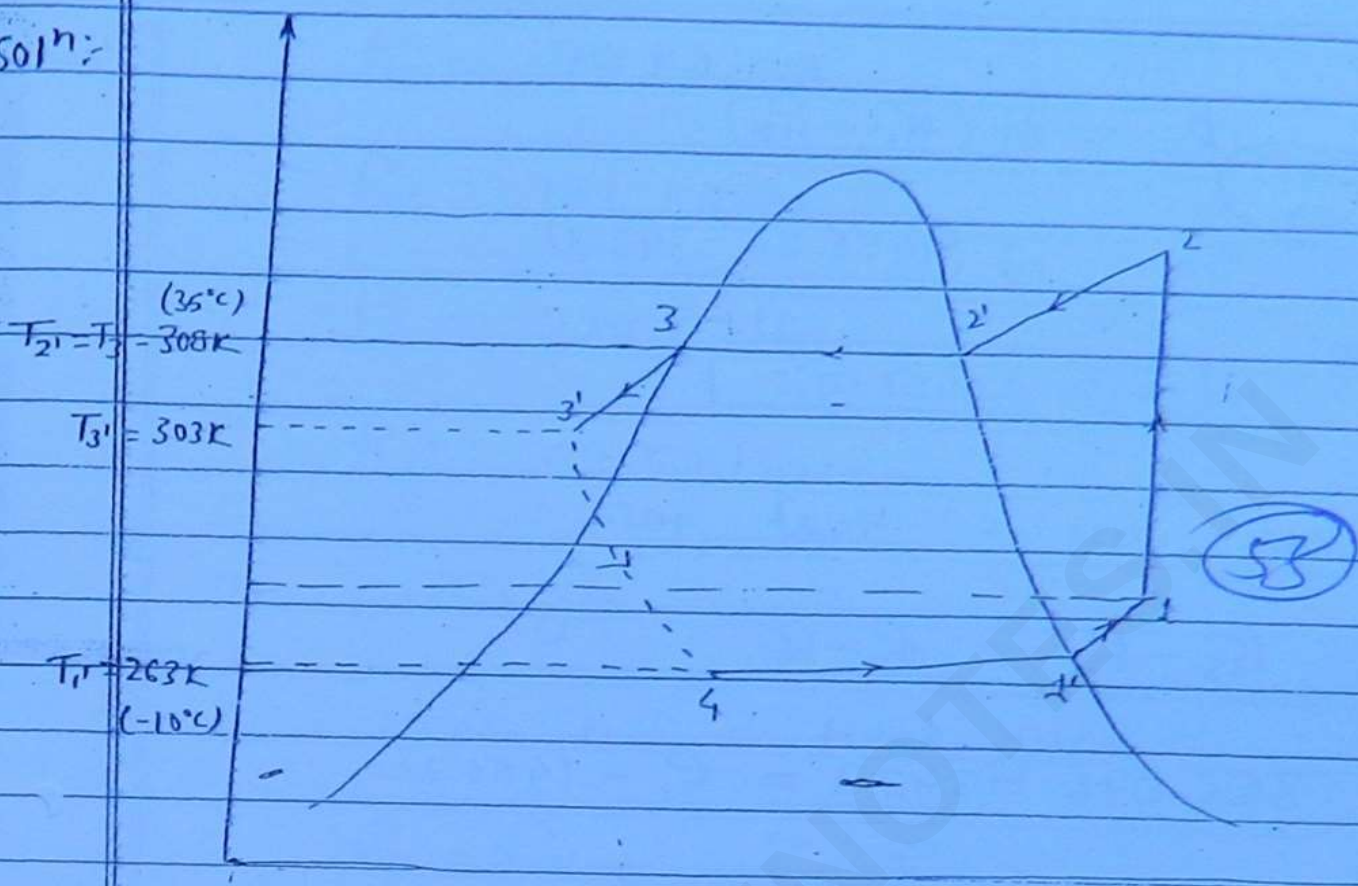
$T_3 - T_{3'}$ = Degree of subcooling.

$T_1 - T_{1'}$ = Degree of superheating.

The degree of superheat is not equal to degree of subcooling because specific heat of liquid and vapour refrigerant are not same.

26- A heat exchanger is used in a refrigeration cycle. Enthalpy at condenser outlet & evaporator outlet are 78 kJ/kg & 182 kJ/kg respectively. The enthalpy at the outlet of compressor is 230 kJ/kg & the enthalpy of sub cooled liquid is 68 kJ/kg . Then find the COP of the cycle.

Soln:



$$P = \dot{m} (h_2 - h_1)$$

$$R_C = 50 \text{ kW}$$

$$R_C = \dot{m} \times R_E$$

$$R_C = \dot{m} (h_1 - h_{3'})$$

$$h_1 = 1450.22$$

$$h_{3'} = h_3 - C_{\text{liquid}} (T_3 - T_{3'})$$

$$h_{3'} = 366.072 - 4.556 (308 - 303)$$

$$h_{3'} = 343.3 \text{ kJ/kg}$$

$$\textcircled{56} \quad R_c = \dot{m} (h_1 - h_3)$$

$$50 = \dot{m} (1450.22 - 343.3)$$

$$\boxed{\dot{m} = 0.0451 \text{ kg/s}}$$

Heat loss = Heat gain

$$h_3 - h_{3'} = h_1 - h_1$$

$$366.072 - 343.3 = h_1 - 1450.22$$

$$\boxed{h_1 = 1473 \text{ kJ/kg}}$$

$$S_1 = S_1' + C_p \ln \frac{T_1}{T_1'}$$

$$h_1 = h_1' + C_{\text{vap.}} (T_1 - T_1')$$

$$1473 = 1450.22 + 2.492 (T_1 - 263)$$

$$\boxed{T_1 = 272.14 \text{ K}}$$

$$S_1 = 5.755 + 2.492 \ln \frac{272.14}{263}$$

$$S_1 = 5.84 \text{ kJ/kgK}$$

But 1-2 process is isentropic process.

$$S_1 = S_2 = 5.84 \text{ kJ/kgK}$$

Soln:

$$h_3 = 78 \text{ kJ/kg}$$

$$h_{1'} = 182 \text{ kJ/kg}$$

$$h_2 = 230 \text{ kJ/kg}$$

$$h_{3'} = 68 \text{ kJ/kg}$$

(57)

In Exchanger,

$$\text{Heat loss} = \text{Heat gain.}$$

$$h_3 - h_{3'} = h_{1'} - h_1$$

$$78 - 68 = h_{1'} - 182$$

$$h_{1'} = 192 \text{ kJ/kg}$$

$$\text{COP} = \frac{h_{1'} - h_{3'}}{h_2 - h_1}$$

$$= \frac{182 - 68}{230 - 192}$$

$$\text{COP} = 3 \text{ / Ans.}$$

(58)

Ques.
Imp.
Prob.

A food storage refrigeration system requires a refrigeration capacity of 50 kW. It works b/w a condenser temp. of 35°C & evaporator temp. of -10°C . The refrigerant is Ammonia and it is subcooled by 5°C before entering expansion valve. Refrigerant leaves evaporator as saturated vapour and then it enters in heat exchanger. The refrigerant leaves condenser as saturated liquid assuming a single cylinder operating at 1000 r.p.m. with stroke equal to 1.2 times Bore. Determine power input to the compressor in kW & cylinder dimension. Take volumetric efficiency as 100%.

$t(^{\circ}\text{C})$	$P(\text{bar})$	$h_f (\text{kJ/kg})$	$h_g (\text{kJ/kg})$	$s_f (\text{kJ/kgK})$
-10	2.9157	154.056	1450.22	0.82965
35	13.522	366.072	1488.57	1.56605

$s_g (\text{kJ/kg})$	Sp. vol (m^3/kg)		Sp. heat (kJ/kgK)	
	liquid	vapour	liquid	vapour
5.755	—	0.41747	—	2.492
5.20086	—	0.095629	4.556	2.903

$$S_2 = S_2' + C_{vap} \ln \frac{T_2}{T_2'}$$

~~5.84 = 5.20086 + 2.903 \times \ln \frac{T_2}{308}~~

$$5.84 = 5.20086 + 2.903 \times \ln \frac{T_2}{308}$$

$$T_2 = 383.85 \text{ K}$$

(59)

$$h_2 = h_2' + C_{pvap} (T_2 - T_2')$$

$$h_2 = 1488.57 + 2.903 (383.85 - 308)$$

$$h_2 = 1708.8 \text{ kJ/kg}$$

$$P_{in} = \dot{m} (h_2 - h_1)$$

$$P_{in} = 0.0451 (1708.8 - 1473.3)$$

$$P_{in} = 10.64 \text{ kW}$$

$$\eta_{vol.} = \frac{\dot{m} v_1}{\frac{\pi D^2 L N}{4}}$$

$$1 = \frac{\dot{m} v_1}{\frac{\pi D^2 L N}{4}}$$

$$\frac{\pi D^2 L N}{4} = \dot{m} v_1$$

$$L = 1.2 D$$

$$N = 1000 \text{ r.p.m.}$$

$$\frac{\pi D^2 (1.2 D) 1000}{4 \times 60} = 0.0451 \times v_1$$

1-2 is a constant pressure process & the refrigerant can be assumed as an ideal gas in superheated region therefore

$$\frac{v_1}{T_1} = \frac{v_{1'}}{T_{1'}}$$

GG

$$v_1 = \frac{v_{1'}}{T_{1'}} \times T_1$$

$$v_1 = \frac{0.41747}{263} \times 272.14$$

$$v_1 = 0.43 \text{ m}^3/\text{kg}$$

then,

$$\frac{\pi}{4} D^3 \times 1.2 \times \frac{1000}{60} = 0.0451 \times 0.43$$

$$D = 0.107 \text{ m} = 10.7 \text{ cm. Ans.}$$

$$L = 1.2 \times 10.7 = 12.8 \text{ cm.}$$

Gas Refrigeration Cycle

This cycle works on Bell-Coleman Cycle or Reversed Brayton Cycle.

Air is used as a refrigerant and this cycle is used in aircraft refrigeration system because of its low weight per tonne of Refrigeration.

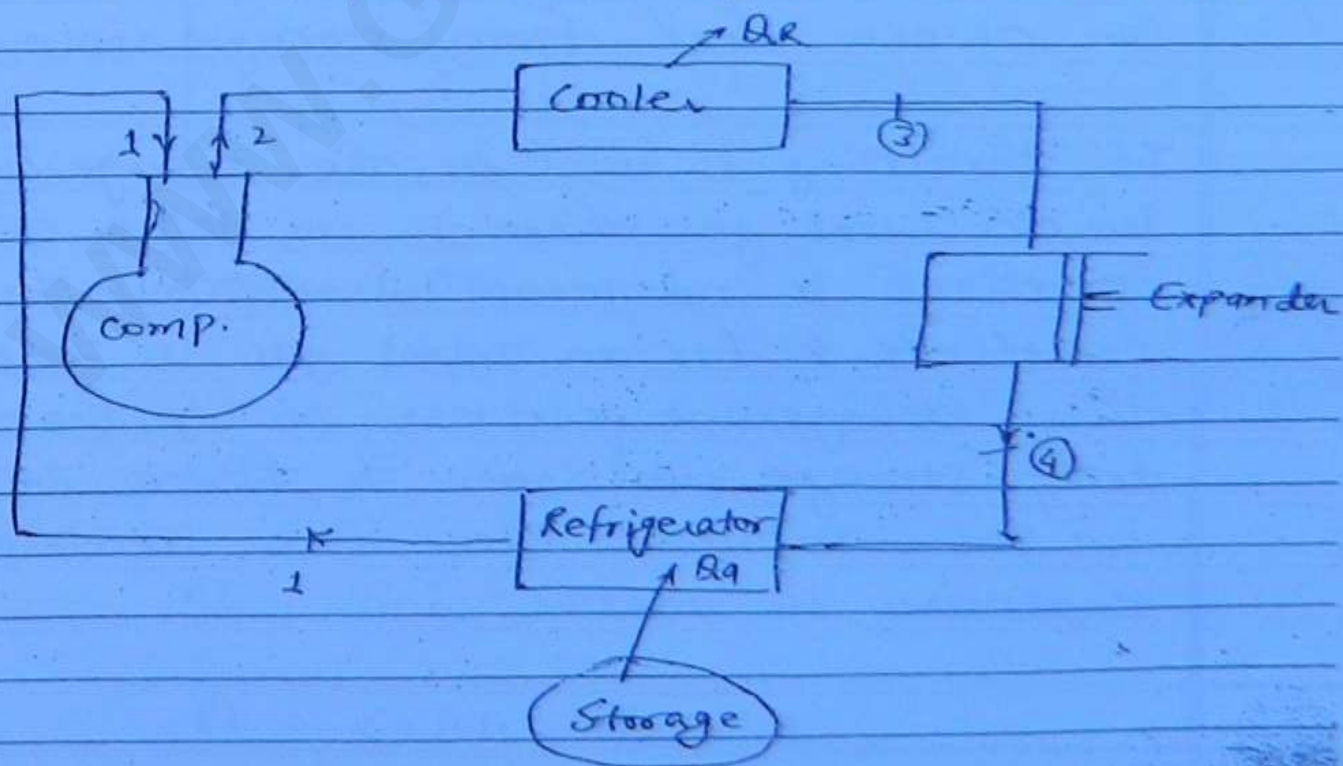
(6)

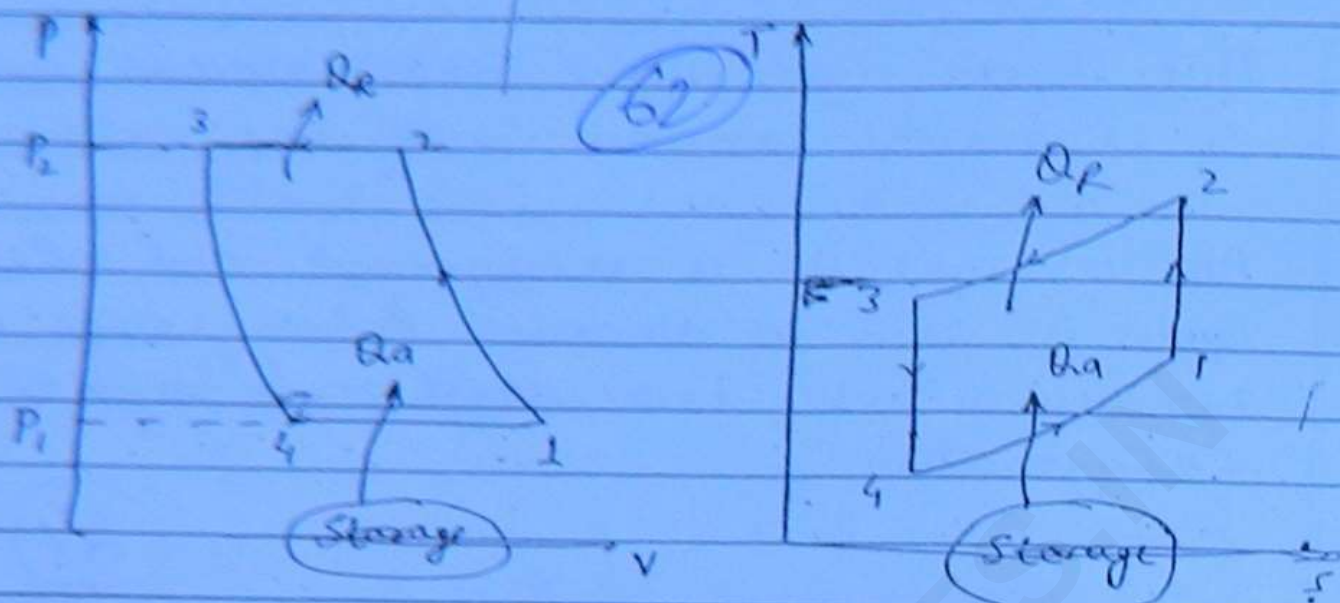
1-2 $\rightarrow$ Reversible adiabatic compression (Isentropic).

2-3 $\rightarrow$ Constant pressure heat rejection.

3-4 $\rightarrow$ Isentropic (rev. adiabatic) Expansion.

4-1 $\rightarrow$ ~~Comp~~ Constant press. heat absorption.





Reason's for using isentropic Expansion instead of throttling in gas Refrigeration Cycle

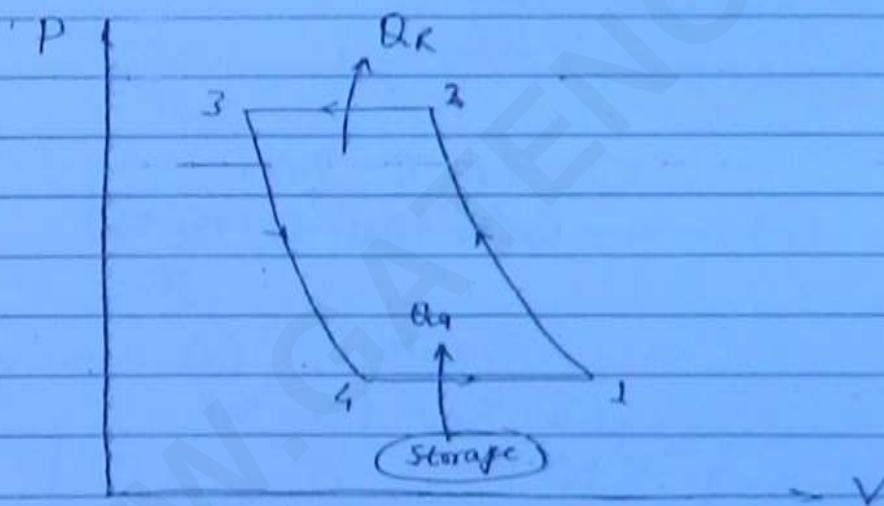
In Isentropic expansion there is a temp. drop & hence during this expansion as the refrigerant temp. is low it can absorb heat from storage space.

If throttling is used as the refrigerant is an Ideal gas there is no temp. change i.e. there is no temp. drop (Joule Thomson coefficient for an Ideal gas is zero) thus Refrigerant can not pickup heat from storage space & hence R.E. is not obtained there fore isentropic expansion is used.

Reasons for using throttling in V.C. cycle:-

Though in isentropic expansion some positive work is obtained this work is less because liquid is expanding. The specific volume of liquid is less & for this small work output the cost of isentropic expander does not justify its use. hence

COP of Gas Refrigeration Cycle:-



$$COP = \frac{RE}{W_{in.}}$$

$$RE = h_1 - h_4$$

$$RE = C_p T_1 - C_p T_4$$

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

$$RC = \dot{m} \times RE$$

$$RC = \dot{m} \times C_p (T_1 - T_4) \quad ***$$

$$\& \quad W_{in} = W_C + W_E$$

(64)

$$W_{in} = (h_2 - h_1) - (h_3 - h_4)$$

$$W_{in} = C_p (T_2 - T_1) - C_p (T_3 - T_4)$$

$$\text{then.} \quad COP = \frac{RE}{W_{in}}$$

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

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

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

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

$$\boxed{COP = \frac{1}{\frac{T_2 - T_3}{T_1 - T_4} - 1}} \quad (1)$$

$$\text{Pressure ratio } r_p = \frac{P_2}{P_1} = \frac{P_3}{P_4}$$

1-2 Reversible adiabatic process (Isentropic)

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

$$\frac{T_2}{T_1} = (r_p)^{\frac{\gamma-1}{\gamma}} \quad \text{--- (II)} \quad \text{65}$$

3-4 Reversible adiabatic process.

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

$$\frac{T_3}{T_4} = (r_p)^{\frac{\gamma-1}{\gamma}} \quad \text{--- (III)}$$

from eqn. (II) & (III)

$$\boxed{\frac{T_2}{T_1} = \frac{T_3}{T_4} = (r_p)^{\frac{\gamma-1}{\gamma}}}$$

Objective
Point of
view.

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

This eqn. is valid only when
compression & expansion both reversible
adiabatic.

Now from eqn. (I), we get.

$$\text{COP} =$$

$$\frac{T_2 (1 - T_3/T_2)}{T_1 (1 - T_4/T_1)} - 1$$

$$COP = \frac{1}{\frac{T_2}{T_1} - 1}$$

$$\left[\because \frac{T_4}{T_1} = \frac{T_3}{T_2} \right]$$

Objective point of view

$$COP = \frac{1}{(\gamma_p)^{\frac{\gamma-1}{\gamma}} - 1}$$

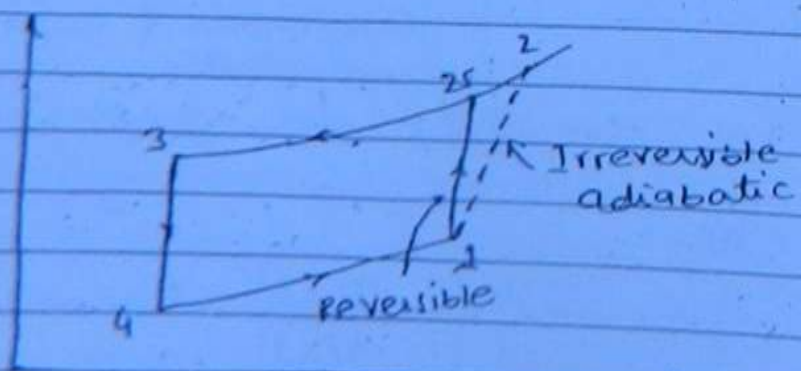
66

It is valid only when the compression & expansion process are Reversible adiabatic process.

COP of a gas Refrigeration cycle depends on pressure ratio only



* Isentropic Efficiency of A Compressor:



It is the ratio of isentropic work to the actual work.

$$(ds)_{\text{universe}} > 0 \quad (\text{Irrev. Adiabatic})$$

$$(ds)_{\text{system}} + (ds)_{\text{surrounding}} > 0.$$

$$\boxed{(ds)_{\text{system}} > 0}$$

(62)

$$\eta_c = \frac{\text{Isentropic Work}}{\text{Actual Work.}}$$

$$\eta_c = \frac{h_{2s} - h_1}{h_2 - h_1} = \frac{C_p (T_{2s} - T_1)}{C_p (T_2 - T_1)}$$

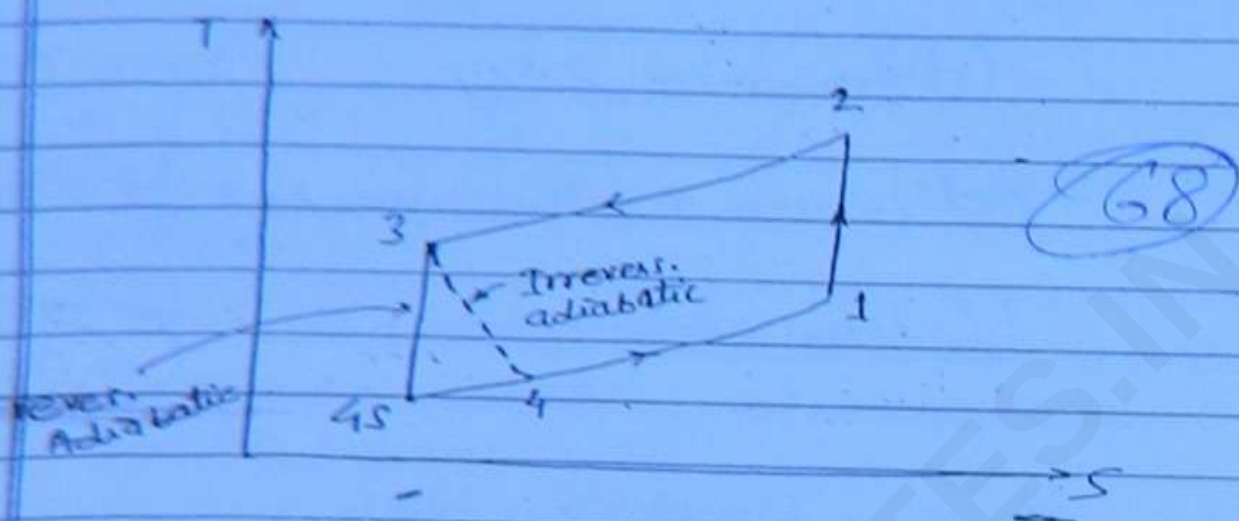
$$\boxed{\eta_c = \frac{T_{2s} - T_1}{T_2 - T_1}}$$

$$\frac{T_{2s}}{T_1} = \left(\frac{P_{2s}}{P_1} \right)^{\frac{\gamma-1}{\gamma}}$$

$$\boxed{\frac{T_{2s}}{T_1} = (r_p)^{\frac{\gamma-1}{\gamma}}}$$

NOTE - Applying steady flow eqn. & we get
Isentropic & Actual work.

* Isentropic efficiency of Expander or Turbine -



$$\eta_T = \frac{\text{Actual work}}{\text{Isentropic work}}$$

$$\eta_T = \frac{h_3 - h_4}{h_3 - h_{4s}} = \frac{c_p(T_3 - T_4)}{c_p(T_3 - T_{4s})}$$

$$\eta_T = \frac{T_3 - T_4}{T_3 - T_{4s}}$$

$$\frac{T_3}{T_{4s}} = \left(\frac{P_3}{P_{4s}} \right)^{\frac{\gamma-1}{\gamma}}$$

$$\frac{T_3}{T_{4s}} = (\delta_p)^{\frac{\gamma-1}{\gamma}}$$

NOTE. - $PV^\gamma = c$ is valid only for the reversible adiabatic process.

prob- In a Gas Refrigeration cycle the temp. at the inlet & exit of isentropic expander are 300K & 180K respectively. & compression is assumed to be isentropic then find the COP.

Soln: $T_3 = 300K$ & $T_4 = 180K$.

$$COP = \frac{1}{(\gamma_p)^{\frac{\gamma-1}{\gamma}} - 1}$$

(69)

$$\frac{T_2}{T_1} = \frac{T_3}{T_4} = (\gamma_p)^{\frac{\gamma-1}{\gamma}}$$

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

$$COP = \frac{1}{\frac{300}{180} - 1}$$

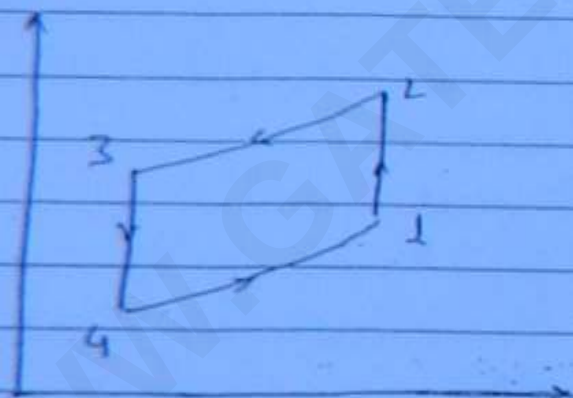
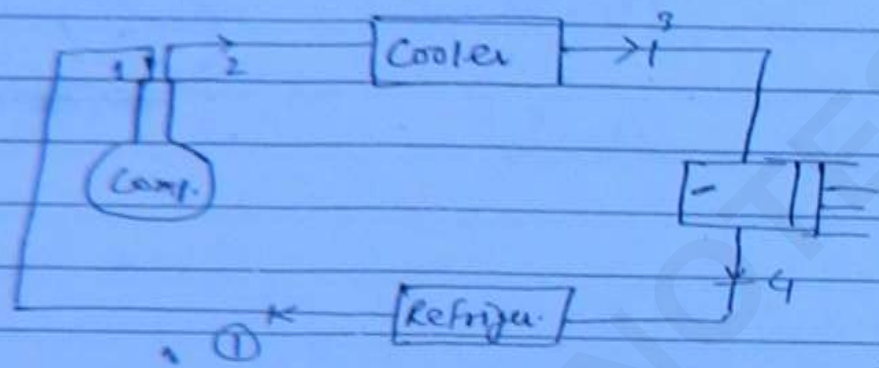
$$COP = 1.5 \text{ Ans.}$$

prob- A Gas Refrigeration system working on Reversed Brayton cycle has a temp. of 250K at the inlet to the compressor. the temp. at the end of constant pressure cooling is 300K & the rise in temp. of air in refrigerator is 50K then find the net work input. take $C_p = 1.005 \text{ kJ/kgK}$.

Solⁿ: given, $T_1 = 250\text{ K}$
 $T_3 = 300\text{ K}$
 $C_p = 1\text{ kJ/kg K}$

The live ~~in~~ temp. of air in refrigerator is

$$T_1 - T_4 = 50\text{ K}$$



$$W_{in} = W_c - W_E$$

$$W_{in} = (h_2 - h_1) - (h_3 - h_4)$$

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

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

$$W_{in} = 1(375-250) - 1(300-200)$$

$$= 125 - 100 = 25$$

$$T_4 = 200K.$$

$$T_2 = 375K.$$

(71)

prob. Air is used as Refrigerant in brayton cycle. draw T-s. & P-v diagram for this cycle & derive the expression for COP. in terms of pressure ratio (r_p).

If the temp. at the end of heat absorption & at the end of heat rejection. are 0°C & 30°C respectively. pressure ratio = 4 find temp. at all points & also find volume flow rate at the inlet to the compressor and at the exit of expander. The cooling capacity of the plant is 1 TR & pressure at the inlet to the compressor is 1 bar. assume Compression & expansion to be isentropic.

Solⁿ:-

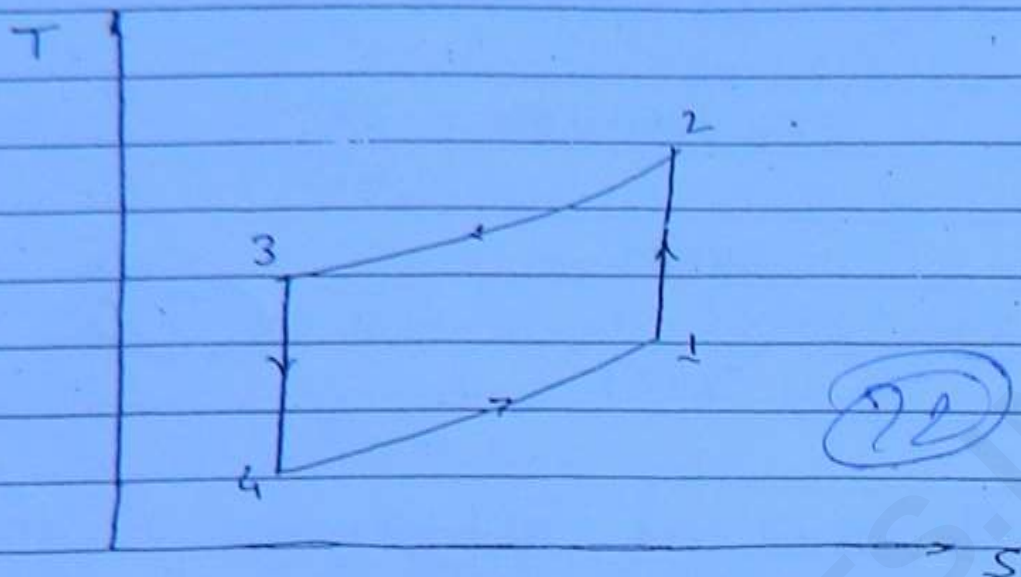
$$T_1 = 0^\circ C$$

$$T_3 = 30^\circ C$$

$$(\gamma_p) = 4 = \frac{T_2}{T_1} = \frac{T_3}{T_4}$$

$$RC = 1 TR = 1 \times 3.5 kW = 3.5 kW.$$

$$P_1 = 1 bar.$$



1-2 is Reversible adiabatic process.

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

$$\frac{T_2}{273} = (4)^{\frac{1.4-1}{1.4}}$$

$$T_2 = 405.67 \text{ K.}$$

The Compressor and expansion is isentropic process.

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

$$\frac{405.67}{273} = \frac{303}{T_4}$$

$$\boxed{T_4 = 203.9 \text{ K}}$$

$$RC = \dot{m} \times RE$$

$$RE = h_1 - h_4 = C_p (T_1 - T_4)$$

then $RC = \dot{m} C_p (T_1 - T_4)$

$$3.5 = \dot{m} \times 1.005 (273 - 203.9)$$

$$\dot{m} = 0.05 \text{ kg/s.}$$

(73)

∴ Air is treated as ideal gas. Then.

$$PV = mRT.$$

$$P_1 V_1 = mRT_1$$

$$V_1 = \frac{mRT_1}{P_1}$$

∴ $R = 0.287 \text{ kJ/kgK}$ & $P_1 = 100 \text{ kPa.}$

$$V_1 = \frac{mRT_1}{P_1} = \frac{0.05 \times 0.287 \times 273}{100}$$

$$\boxed{V_1 = 0.039 \text{ m}^3/\text{sec.}} \text{ Ans.}$$

& $P_4 V_4 = mRT_4$

$$V_4 = \frac{mRT_4}{P_4} = \frac{0.05 \times 0.287 \times 203.9}{100}$$

$$V_4 = 0.029 \text{ m}^3/\text{sec.}$$

Q6. In a bell-Coleman Refrigeration plant air is used as a refrigerant the pressure & temp. at the beginning of comp. are 1 bar & 10°C respectively. Air is compressed to 5 bar & is then cooled to 25°C in a cooler if the compression & expansion follows $PV^{1.35} = C$ & $PV^{1.3} = C$. Find the COP. take $\gamma = 1.4$ & $C_p = 1.009 \text{ kJ/kgK}$.

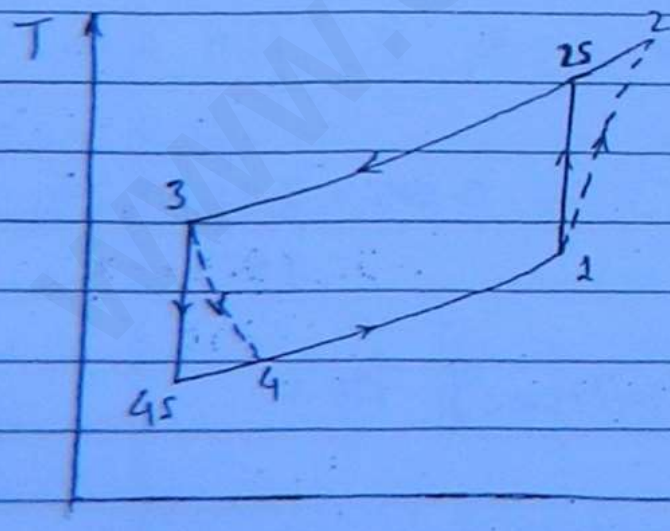
Q17: $T_1 = 10^\circ\text{C} = 10 + 273 = 283 \text{ K.}$

$$P_1 = 1 \text{ bar.}$$

$$P_2 = 5 \text{ bar.}$$

$$T_3 = 25^\circ\text{C} = 25 + 273 = 298 \text{ K.}$$

74

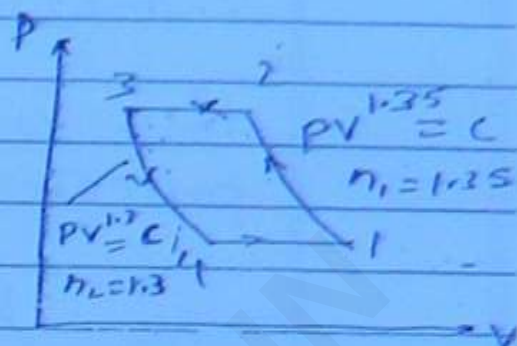


$$\frac{P_1}{P_2} = \left(\frac{T_1}{T_2} \right)^{\frac{\gamma}{\gamma-1}}$$

$$\frac{T_2}{T_1} = \left(\frac{P_2}{P_1} \right)^{\frac{n_1-1}{n_1}}$$

$$\frac{T_2}{T_1} = \left(\frac{P_2}{P_1} \right)^{\frac{n_1-1}{n_1}}$$

$$\frac{T_2}{283} = 5^{\frac{1.35-1}{1.35}}$$



$$T_2 = 429.5 \text{ K}$$

$$\frac{T_3}{T_4} = \left(\frac{P_3}{P_4} \right)^{\frac{n_2-1}{n_2}}$$

$$\frac{298}{T_4} = \left(5 \right)^{\frac{1.3-1}{1.3}}$$

$$T_4 =$$

$$T_4 =$$

(TS)

$$\text{COP} = \frac{RE}{W_{in}}$$

$$RE = h_1 - h_4 = C_p (T_1 - T_4)$$

$$RE = 1.009 (283 - 205.8)$$

$$RE = 78.14 \text{ kJ/kg}$$

∴ 1-2 is a polytropic compression process.

then compression work in polytropic process.

$$W = \frac{n}{n-1} (P_1 V_1 - P_2 V_2)$$

$$W_c = \frac{n_1}{n_1-1} (P_1 V_1 - P_2 V_2)$$

(76)

$$W_c = \frac{n_1}{n_1-1} (m R T_1 - m R T_2)$$

$$= \frac{1.35}{1.35-1} (m R (T_1 - T_2))$$

$$= \frac{1.35}{0.35} \times 1 \times 0.287 (283 - 429.5)$$

$$= -162.17 \text{ KW.}$$

Expansion work in polytropic process.

$$W_E = \frac{n_2}{n_2-1} (P_3 V_3 - P_4 V_4)$$

$$= \frac{n_2}{n_2-1} (m R T_3 - m R T_4)$$

$$W_e = \frac{1.3}{1.3-1} \times 1 \times 0.287 (298 - 205.5)$$

$$W_e = 115$$

$$W_{in} = W_c - W_e$$

$$= 162.17 - 115$$

$$= 47.17$$



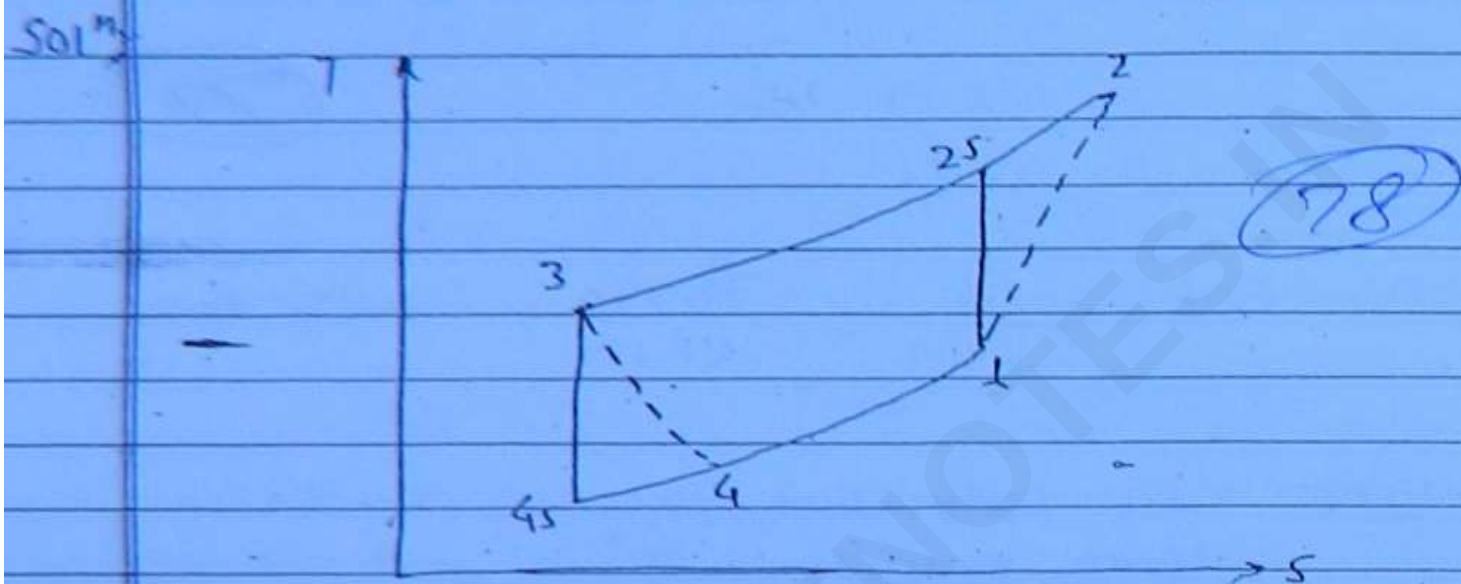
$$COP = \frac{R.E.}{W_{in}} = \frac{78.14}{47.17} = 1.65 \text{ Ans.}$$

NOTE - In polytropic process as there is heat transfer Q is not equal to zero is ~~a~~ S.F.E.E. & hence work is not equal to $(h_2 - h_1)$

prob - In a air craft Refrigeration system air enters the compressor at 0.18 MPa, 4°C & air is compressed to 0.3 MPa with an isentropic efficiency of 72%. Air is then cooled to 55°C at constant press. & then it is expanded in a turbine to 0.1 MPa ~~with~~ with an isentropic efficiency of 78%. The lower temp. air absorbs a cooling load of 3TR at constant pressure before returning to the compressor assuming air to be an ideal gas find. -

(i) COP.

(ii) NET power input in kW & mass flow rate is kg/s.



$$P_1 = 0.1 \text{ MPa.}$$

$$T_1 = 4^\circ\text{C} = 4 + 273 = 277\text{K}$$

$$P_2 = 0.3 \text{ MPa.}$$

$$h_c = 0.72$$

$$T_3 = 55^\circ\text{C} = 55 + 272 = 328\text{K}$$

$$P_4 = 1 \text{ MPa.}$$

$$h_T = 0.78$$

$$RC = 3TR = 3 \times 3.5 = 10.5 \text{ kW.}$$

$$r_p = \frac{P_2}{P_1} = \frac{0.3}{0.1} = 3$$

$$\eta_c = \frac{\text{Isentropic Work}}{\text{Actual Work.}}$$

$$= \frac{h_{2s} - h_1}{h_2 - h_1}$$

$$= \frac{C_p (T_{2s} - T_1)}{C_p (T_2 - T_1)}$$

$$\Rightarrow 0.72 = \frac{T_{2s} - T_1}{T_2 - T_1}$$

(79)

$$\frac{T_{2s}}{T_1} = (r_p)^{\frac{\gamma-1}{\gamma}}$$

$$\frac{T_{2s}}{277} = (3)^{\frac{1.4-1}{1.4}}$$

$$T_{2s} = 379.14 \text{ K.}$$

$$\text{then } 0.72 = \frac{379.14 - 277}{T_2 - 277.}$$

$$\boxed{T_2 = 418.88 \text{ K.}}$$

$$\frac{T_3}{T_{4s}} = (r_p)^{\frac{\gamma-1}{\gamma}}$$

$$\frac{328}{T_{4s}} = (3)^{\frac{1.4-1}{1.4}}$$

$$T_{45} = 239.6 \text{ K}$$

$$\eta_{LT} = \frac{T_3 - T_4}{T_3 - T_{45}}$$

$$0.78 = \frac{328 - T_4}{328 - 239.6}$$

$$T_4 = 259 \text{ K}$$

80

$$COP = \frac{R_E}{W_{in}}$$

$$COP = \frac{h_1 - h_4}{W_C - W_E}$$

$$COP = \frac{h_1 - h_4}{(h_2 - h_1) - (h_3 - h_4)}$$

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

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

$$= \frac{277 - 259}{(418.86 - 277) - (328 - 259)}$$

$$= 0.29$$

$$\text{COP} = \frac{RC}{P_{in}}$$

$$0.24 = \frac{10.5}{P_{in}}$$

$$[P_{in} = 43.75 \text{ kW}] \text{ Ans.}$$

$$RC = \dot{m} \times RE = \dot{m} \times (h_1 - h_4)$$

$$RC = \dot{m} C_p (T_1 - T_4)$$

$$10.5 = \dot{m} \times 1.005 (277 - 259)$$

$$[\dot{m} = 0.58 \text{ kg/sec.}] \text{ Ans.}$$

(81)

Vapour absorption Refrigeration Cycle

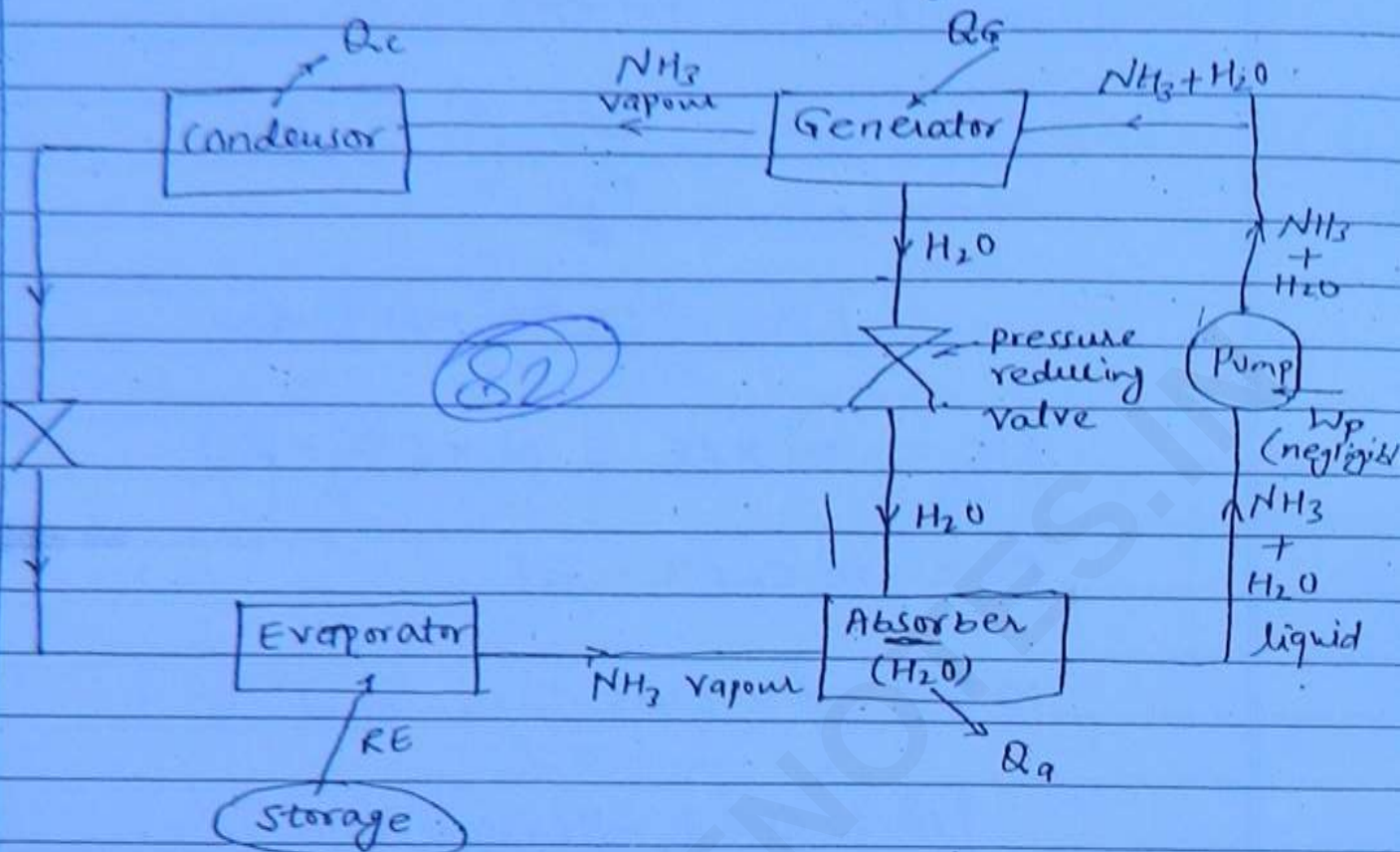


fig- $\text{NH}_3\text{-H}_2\text{O}$ VARS System.

Pump $\rightarrow$ liquid
Compressor $\rightarrow$ gas.

$$W = - \int v dp$$

$$W = - \int v_l dp$$

$$W = - \int v_g dp$$

Volume of liquid is less than the volume of gas. then:

$$[W_c > W_p]$$

Important points

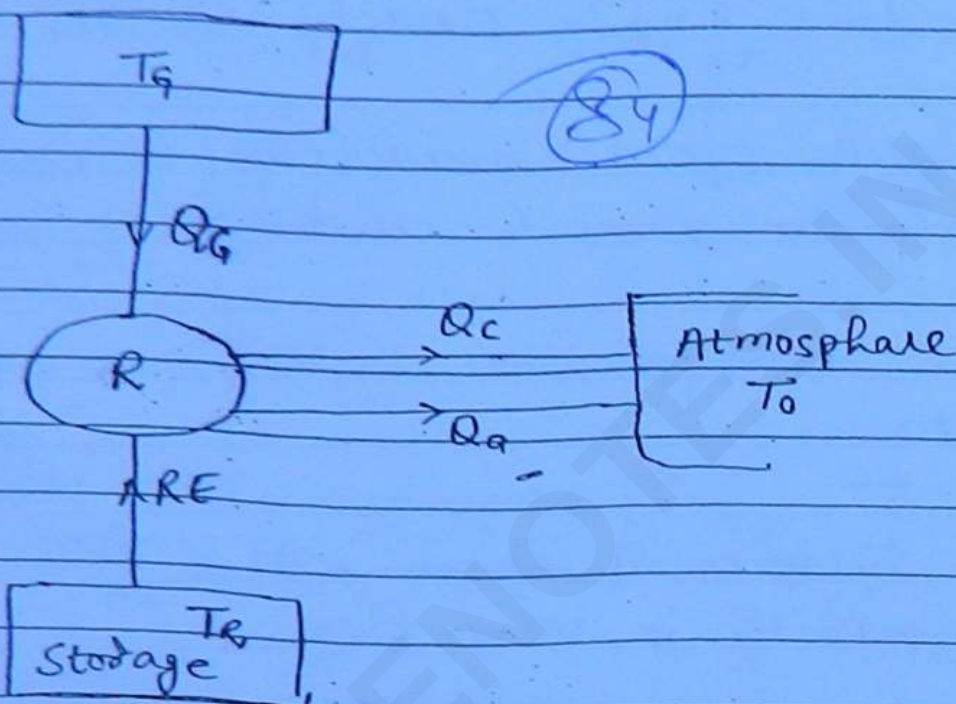
1. Vapour compression Refrigeration system operates on high grade energy (work) whereas Vapour absorption Refrigeration cycle operates on low grade energy (Heat).
2. In V.A. cycle compressor is replaced with generator pump & absorber.
3. Solar refrigeration systems are based on V.A. cycle. (80)
4. V.A. system is used where the cost of electricity is very high.
5. Most popular V.A. system is NH_3 - H_2O V.A. system in which NH_3 is refrigerant & H_2O is absorbent.
6. In V.A. system Refrigerant rejects heat to surrounding in Condenser & absorber.

★ COP of V.A. Cycle:-

$$\text{COP} = \frac{\text{desired effect}}{\text{Work input}} = \frac{RE}{Q_G + W_P}$$

∴ Pump work is very small i.e. pump work is negligible.

$$\text{COP} = \frac{RE}{Q_G}$$



from 1st law of thermodynamics

$$Q_G + RE = Q_c + Q_a \quad \text{--- (1)}$$

From 2nd law of thermodynamics

$$(\Delta S)_{\text{universe}} \geq 0$$

$$(\Delta S)_{\text{system}} + (\Delta S)_{\text{surrounding}} \geq 0$$

As system is undergoing cycle, $(\Delta S)_{\text{system}} = 0$, because entropy is a property.

$$(\Delta S)_{\text{surrounding}} \geq 0$$

$$-\frac{Q_G}{T_g} - \frac{RE}{T_R} + \frac{Q_c}{T_0} + \frac{Q_a}{T_0} \geq 0$$

$$-\frac{Q_G}{T_G} - \frac{RE}{T_R} + \frac{Q_G + RE}{T_0} \geq 0$$

$$-\frac{Q_G}{T_G} - \frac{RE}{T_R} + \frac{Q_G + RE}{T_0} \geq 0$$

$$-\frac{Q_G}{T_G} - \frac{RE}{T_R} + \frac{Q_G}{T_0} + \frac{RE}{T_0} \geq 0$$

$$Q_G \left(\frac{1}{T_0} - \frac{1}{T_G} \right) \geq RE \left(\frac{1}{T_R} - \frac{1}{T_0} \right)$$

$$Q_G \left(\frac{T_G - T_0}{T_0 T_G} \right) \geq RE \left(\frac{T_0 - T_R}{T_0 T_R} \right)$$

$$\frac{\frac{T_G - T_0}{T_G}}{\frac{T_0 - T_R}{T_R}} \geq \frac{RE}{Q_G}$$

$$\left(\frac{T_G - T_0}{T_G} \right) \left(\frac{T_R}{T_0 - T_R} \right) \geq \text{COP}$$

$$\text{COP} \leq \left(\frac{T_G - T_0}{T_G} \right) \left(\frac{T_R}{T_0 - T_R} \right)$$

$$\boxed{(\text{COP})_{\max} = \left(\frac{T_G - T_0}{T_G} \right) \left(\frac{T_R}{T_0 - T_R} \right)} \quad \text{Imp. - 2011}$$

Imp.
prob

A vapour absorption Refrigeration cycle works with generator, ambient & evaporator temp. as 360 K, 310 K & 260 K respectively. Find the max. COP. If the evaporator temp. falls to 250 K what should be the generator temp. in order to operate the system with same COP.

Soln: (i) $T_g = 360 \text{ K}$
 $T_o = 310 \text{ K}$
 $T_R = 260 \text{ K}$

(86)

$$COP_{max} = \left(\frac{T_g - T_o}{T_g} \right) \left(\frac{T_R}{T_o - T_R} \right)$$

$$= \left(\frac{360 - 310}{360} \right) \left(\frac{260}{310 - 260} \right)$$

$$\boxed{(COP)_{max} = 0.72 \text{ Ans.}}$$

(ii) $T_R = 250 \text{ K}$
 $T_o = 310 \text{ K}$
 $T_g = ?$
 $COP = 0.72$

$$COP = \left(\frac{T_g - T_o}{T_g} \right) \left(\frac{T_R}{T_o - T_R} \right)$$

$$0.72 = \left(\frac{T_g - 310}{T_g} \right) \left(\frac{250}{310 - 250} \right)$$

$$\boxed{T_g = 375 \text{ K} / \text{Ans.}}$$

Imp.

prob- A VARS works with 3 thermal reservoirs. A refrigeration effect of 100 Watt is required at 250 K. Heat source is available at 400 K. Heat rejection occurs at 300 K then find the min. value of heat required in watt.

Soln: given, $RE = 100 \text{ W}$
 $T_R = 250 \text{ K}$
 $T_g = 400 \text{ K}$
 $T_o = 300 \text{ K}.$

(87)

$$COP = \frac{RE}{Q_g}$$

$$(COP)_{\max} = \frac{RE}{(Q_g)_{\min.}}$$

$$\left(\frac{T_g - T_o}{T_g} \right) \left(\frac{T_R}{T_o - T_R} \right) = \frac{RE}{(Q_g)_{\min.}}$$

$$\left(\frac{400 - 300}{400} \right) \left(\frac{250}{300 - 250} \right) = \frac{100}{(Q_g)_{\min.}}$$

$$\boxed{(Q_g)_{\min} = 80 \text{ W} / \text{Ans.}}$$

Prob In $\text{NH}_3\text{-H}_2\text{O}$ Absorption Refrigeration system heat is supplied to the generator by condensing steam at 0.2 MPa & 0.9 dryness fraction. Steam reaches saturated liquid state after condensation the temp. to be maintained in refrigerator is -10°C & ambient temp. is 30°C . Find the max. COP of system if the actual COP is 40% of maximum COP & Refrigeration load is 20 TR. What will be the required steam flow rate at 0.2 MPa.

$$t_{\text{sat}} = 120.2^\circ\text{C}$$

$$h_g - h_f = 2201.9 \text{ kJ/kg}$$

88

Solⁿ

$$T_R = -10^\circ\text{C} = -10 + 273 = 263 \text{ K}$$

$$T_0 = 30^\circ\text{C} = 30 + 273 = 303 \text{ K}$$

$$R.C. = 20 \text{ TR} = 20 \times 3.5 = 70 \text{ kW}$$

$$T_g = 273 + 120.2 = 393.2 \text{ K}$$

~~Refrigerator~~

$$(\text{COP})_{\text{max}} = \left(\frac{T_g - T_0}{T_g} \right) \left(\frac{T_R}{T_0 - T_R} \right)$$

$$= \left(\frac{393.2 - 303}{393.2} \right) \left(\frac{263}{303 - 263} \right)$$

$$(\text{COP})_{\text{max}} = 1.5$$

$$(\text{COP})_{\text{act}} = 1.5 \times 0.4 = 0.6$$

$$h_1 = h_f + x(h_g - h_f)$$

$$h_2 = h_f$$

$$Q_g = h_1 - h_2$$

$$= h_f + x(h_g - h_f) - h_f$$

$$= x(h_g - h_f)$$

$$= 0.9(2201.9)$$

$$Q_g = 1981.7 \text{ kJ/kg}$$

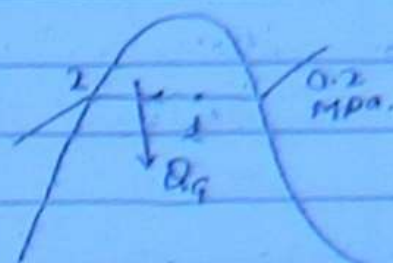
(89)

$$Q_g = m(1981.7)$$

$$\text{COP} = \frac{R_E}{Q_g} = \frac{20 \times 3.5}{m(1981.7)} \quad \text{KW}$$

$$0.6 = \frac{20 \times 3.5}{m(1981.7)}$$

$$\boxed{m = 0.0588 \text{ kg/sec.}} \quad \text{Ans.}$$



prob Match the following - with respect to V.C. Cycle.

list - I

list - II

A. Compression

1. Iso-baric

B. Heat rejection

2. Iso-thermal

C. Expansion

3. Isentropic

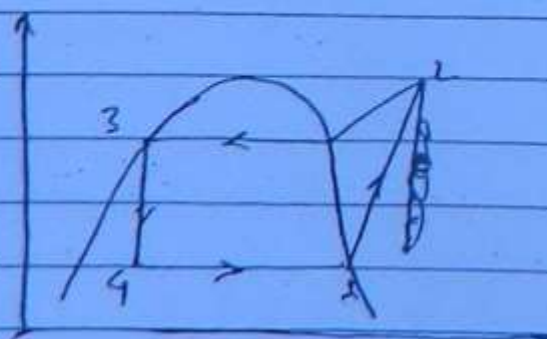
D. Heat absorption

4. Isenthalpic

	A	B	C	D
Ans. (i)	3	1	4	2
(ii)	2	3	4	1
(iii)	3	1	4	1
(iv)	3	2	1	4

90

prob A vapour compression cycle is represented as shown in figure with state one being exit of evaporator. The co-ordinate system used in the figure is



(i) P-R (ii) T-S (iii) P-S (iv) T-R

prob- Heat rejection ratio of a condensor.

Heat rejection ratio or Heat rejection factor in condensor.

It is defined as the ratio of Heat rejecte in condensor to the refrigeration effect.

$$HRR = \frac{Q_{\text{condensor}}}{RE}$$

It is an indication of condensor size

Q1

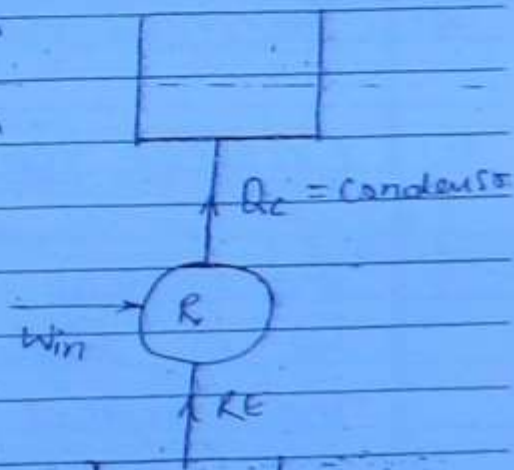
$$Q_c = W_{in} + RE$$

$$HRR = \frac{W_{in} + RE}{RE}$$

$$HRR = \frac{W_{in}}{RE} + 1$$

$$HRR = 1 + \left(\frac{RE}{W_{in}} \right)$$

$$HRR = \frac{1}{COP} + 1$$



* Desirable property of Refrigerant or Selection of a Refrigerant :-

The properties of Refrigerants are classified into -

1. Thermodynamic properties
2. Chemical properties
3. Physical properties

(92)

1. Thermodynamic properties :-

(i) Critical temperature :-

The critical temp. of refrigerant should be as high as possible.

The critical temp. of CO_2 & Ethylene are very low & hence these refrigerants are not suitable in tropical condition like India because a critical temp. of these refrigerants is less than surrounding temp. in summer.

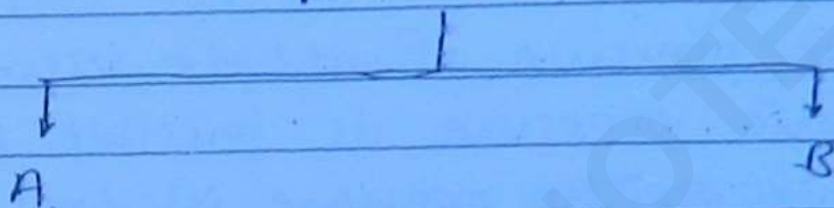
Refrigerant	H_2O	NH_3	R-12	R-22	R-134	CO_2
Critical temp. $^{\circ}\text{C}$	374.5	132.5	111.5	96.5	101.2	31

Ethylene - 10.6

(ii) Enthalpy of vaporization:

Larger Enthalpy of vaporization is desirable because for a given refrigeration capacity, larger Enthalpy of vaporization results in lower mass flow rate.

NOTE- $RC = 1500 \text{ KW}$



(93)

Refrigerant	Enthalpy of vap. (KJ/kg)
H ₂ O	2261
R-22	234.7
NH ₃	1369
R-12	165.7
R-134	197.3

NOTE- OF all the common Refrigerant's NH₃ Refrigerant has large enthalpy of vaporization (large RE).

iii) Conductivity :-

For faster heat transfer higher conductivity is desired.

(94)

iv) Evaporator & Condensor pressure :-

Evaporator pressure should be as close to atmospheric pressure as possible because if the evaporator pressure is low then atmospheric air can leak into the system.

Condensor pressure must be moderate (not very high) because higher condensor pressure results in lower R.E. & higher work input.

v) Pressure Ratio :- (P_2/P_1)

Lower pressure ratio is desirable because higher pressure ratio results in larger work input & smaller volumetric efficiency.

used

We generally use pressure ratio in V.C. cycle 3 to 5.

vi) Freezing point :-

The freezing point of the refrigerant must be low.

(ix) Compressor discharge temperature:

Compressor discharge temp. should not be very high if it is high then it's result in overheating of compressor.

NH_3 compressor's are generally water cooled because the compressor discharge temp. for ammonia refrigeration system is high (around 120°C).

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* Recent trends in Refrigerant's:-

The chlorine element which is present in refrigerant attacks ozone which is situated in stratosphere & hence Cl element must be eliminated from refrigerant therefore the suitable substitutes are-

- (i) Hydro-carbon's
- (ii) Hydro-fluoro-carbon's
- (iii) Fluoro-carbon's.

Now day's environment friendly refrigerant used in domestic refrigerator is R-134a ($\text{CF}_3\text{CH}_2\text{F}$)

2. Chemical properties:-

(i) Toxicity:- (NH_3 is the toxic)

Refrigerant must be non-toxic

(ii) Flammability:-

(99)

The Refrigerant must be non-flammable.

NOTE - Though NH_3 is a cheap Refrigerant it is not used in domestic Refrigerator because of its flammable & toxic nature.

(iii) Action With Oil:-

In compressor's some oil is carried by high temp. refrigerant to the condenser & finally to the evaporator in evaporator Refrigerant vaporizes and oil separates from Refrigerant thus oil gets accumulate in evaporator this accumulation of oil result in reduce heat transfer coefficient.

Refrigerant that are not miscible with oil such as NH_3 & CO_2 do not present any problem's because oil separator is used & separated oil is broad back to the compressor.

Refrigerant that are completely miscible with oil like R-11 & R-12 also do not present any problem because oil which reaches evaporator is broad to the compressor.

Refrigerant's that are partially miscible with oil like R-22 create's problem & hence synthetic oil is used instead of mineral oil.

(98)

(iv) Action with material of construction:-

Ammonia (NH_3) attacks Cu & hence when ammonia is used as a refrigerant wrought iron or steel is used as the material of construction similarly freon's attacks aluminium & hence for freon's Cu is used as the material of construction.

3. Physical property:-

(i) viscosity:- It should be less.

(ii) Leak detection:-

Refrigerant should not leak out from the system & if at all it leaks its detection must be simple. Freon leaks

are detected by Hallide torch method in presence of freaon the colour of the light changes from blue to bluish green. Similarly Ammonia leak's are detected by sulphur stick method. In presence of NH_3 white fumes of Ammonium sulphide are formed.

99

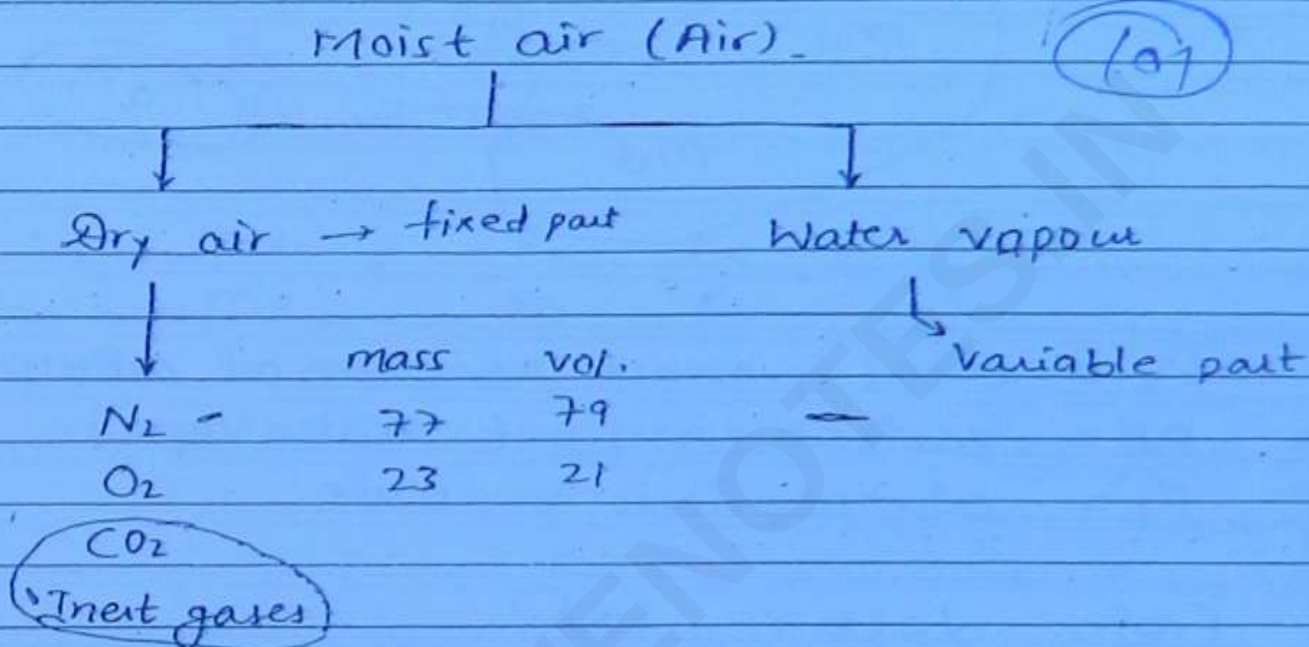
100

Air-CONDITIONING

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Air - Conditioning

It is the simultaneous control of temp., humidity, Air velocity & purity of air



Psychrometry :- It is the science which deals with the study of properties of moist air.

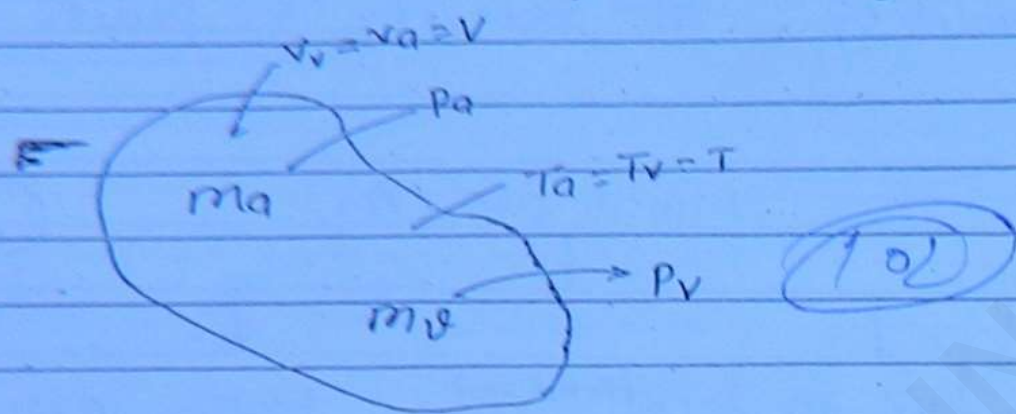
* Psychrometry term *

(i) Specific humidity or Humidity Ratio (ω):-

It is defined as the ratio of mass of vapour to the mass of dry air i.e.

$$\omega = m_v / m_a$$

It is expressed in kg vapour / kg of dry air.



P_t = total pressure of atmosphere.

P_v = partial press. of water vapour.

P_a = atmospheric pressure.

From dalton's law.

$$P_t = P_a + P_v$$

$$P_a = P_t - P_v$$

$$P_v = m R T$$

$$P_a v_a = m_a R_a T_a$$

$$P_a v = m_a R_a T \quad \text{--- (1)}$$

~~$P_a v$~~

$$P_v v_v = m_v R_v T_v$$

$$P_v v = m_v R_v T \quad \text{--- (2)}$$

Dividing eqn. (ii) to (i).

$$\frac{P_v V}{P_a V} = \frac{m_v R_v T}{m_a R_a T}$$

$$\frac{P_v}{P_a} = \frac{m_v R_v}{m_a R_a}$$

(103)

$$\frac{m_v}{m_a} = \frac{P_v}{P_a} \cdot \frac{R_a}{R_v}$$

$$W = \frac{P_v}{P_a} \cdot \frac{\bar{R}}{29}$$

$$R_v = \frac{\bar{R}}{M_v} = \frac{\bar{R}}{18}$$

$$\frac{\bar{R}}{18}$$

$$R_a = \frac{\bar{R}}{M_a} = \frac{\bar{R}}{29}$$

$$W = \frac{18}{29} \times \frac{P_v}{P_a}$$

$$W = 0.622 \frac{P_v}{P_a}$$

$$W = 0.622 \frac{P_v}{P_t - P_v}$$

NOTE- As total atmospheric pressure P_t is almost constant W is a function of partial pressure of vapour.

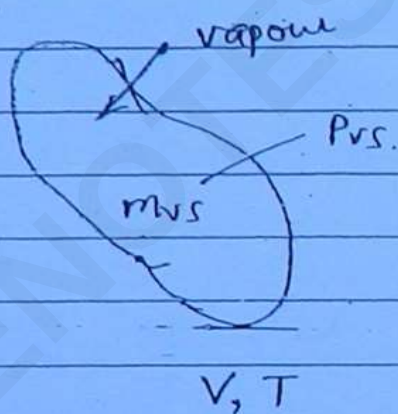
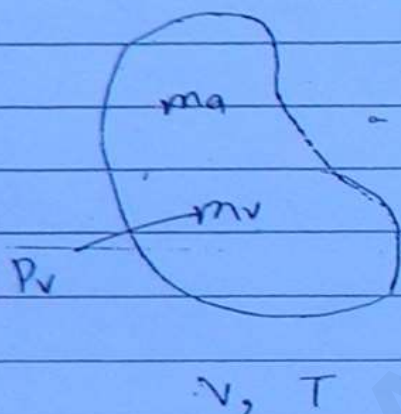
$$W = f(P_v)$$

ii Relative Humidity (ϕ):-

It is defined as the ratio of mass of vapour to the ~~mass~~ mass of vapour under saturation condition's in the same volume & at the same temp.

$$\phi = \frac{m_v}{m_{vs}}$$

(104)



$$P_v V = m_v R_v T$$

$$P_v V = m_v R_v T \quad \text{--- (I)}$$

$$P_{vs} V = m_{vs} R_v T$$

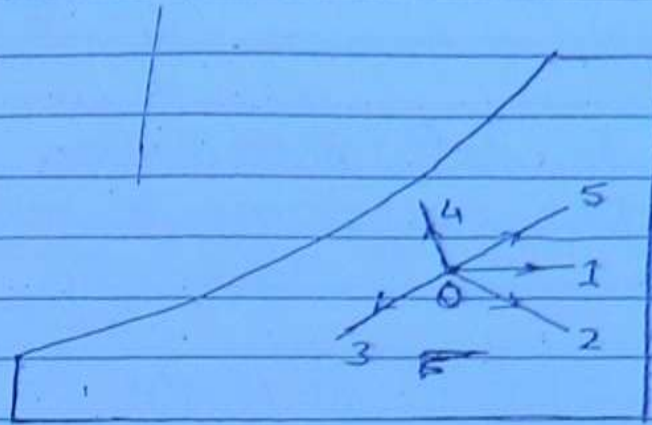
$$P_{vs} V = m_{vs} R_v T \quad \text{--- (II)}$$

Dividing eqn. (II) to (I)

$$\frac{P_v V}{P_{vs} V} = \frac{m_v R_v T}{m_{vs} R_v T}$$

$$\frac{m_v}{m_{vs}} = \frac{P_v}{P_{vs}}$$

$$\boxed{\phi = P_v / P_{vs}}$$



105

Answer →

P	Q	R	S	T
2	1	3	5	4

prob-

A LIST-I

LIST-II

- | | | |
|-----|---|----------------------------------|
| (A) | Steam spray in air | 1. Sensible cooling |
| (B) | Air passing over a coil carrying steam. | 2. cooling and dehumidification. |
| (C) | Air passing over a coil having temp. less than dew point temp. | 3. Heating & humidification. |
| (D) | Air passing over a coil having temp. above dew point temp. but below dry bulb temp. | 4. sensible heating. |

Ans-

A	B	C	D
3	4	2	1

NOTE- Cooling & dehumidification occurs when the coil temp. is less than coil temp.

(106)

Q6- Air at 20°C DBT & 40% RH is heated to 40°C using an electric heater the surface temp. of heater is 45°C then find the bypass factor of heater.

Soln:

$$\begin{array}{ccc} t_1 & & t_2 & \text{---} & t_3 \\ 20^{\circ}\text{C} & & 40^{\circ}\text{C} & & 45^{\circ}\text{C} \end{array}$$

$$\text{BPF} = \frac{t_3 - t_2}{t_3 - t_1} = \frac{45 - 40}{45 - 20} = 0.2$$

Q7- In an A/C process 5 kW/min of heat is extracted from room if the sensible heat factor is 0.8 then find the latent heat load.

Soln:

$$\text{TH} = 5$$

$$\text{SHF} = 0.8$$

$$\text{SHF} = \frac{\text{SH}}{\text{TH}}$$

$$0.8 = \frac{\text{SH}}{5}$$

$$\text{SH} = 4$$

$$0.6 = \frac{P_{v1}}{0.05732}$$

$$P_{v1} = 0.03498 \text{ bar.}$$

$$\omega_1 = 0.622 \left(\frac{0.03498}{1.013 - 0.03498} \right)$$

$$\omega_1 = 0.02186 \text{ kg vapour / kg dry air.}$$

(187)

$$t_2 = 25^\circ\text{C}$$

$$P_t = 1.013 \text{ bar.}$$

$$P_{v2} = 0.03229$$

$$\omega_2 = 0.02186 - 5 \times 10^{-3}$$

$$\omega_2 = 0.01686 \text{ kg vapour / kg dry air.}$$

$$\phi_2 = \frac{P_{v2}}{P_{vs2}}$$

$$\omega_2 = 0.622 \frac{P_{v2}}{P_t - P_{v2}}$$

$$0.01686 = 0.622 \times \frac{P_{v2}}{1.013 - P_{v2}}$$

$$P_{v2} = 0.026 \text{ bar.}$$

$$\phi_2 = \frac{P_{v2}}{P_{s2}} = \frac{0.026}{0.03229} = 0.827$$

$$\phi_2 = 82.7\%$$

(108)

$$h_2 = C_{pa} + W_2 (2500 + 1.88 t_2)$$

$$h_2 = 1.005 \times 25 + 0.01686 (2500 + 1.88 (25))$$

$$h_2 = 68.06 \text{ kJ/kg}$$

Prob. Match group I with group-II

Group-I

Group-II

(Process in figure)

(Name of process)

P	0-1	1. Adiabatic chemical De-Humidification.
Q	0-2	2. Sensible heating.
R	0-3	3. Cooling & De-Humidification.
S	0-4	4. Humidification with steam injection.
T	0-5	5. Humidification with water injection.

NOTE - ϕ is generally expressed in %

For saturated air ϕ is equal to 1 or 100%.

A moist air at 101.3 kPa contains 10 gram of water vapour per kg of dry air. Assuming air and water vapour mixture to behave as an ideal gas then find the relative humidity of air take saturation pressure of vapour as 3.167 kPa.

$$P_t = 101.3 \text{ kPa.}$$

$$m_v = 10 \text{ gram.}$$

$$P_{vs} = 3.167 \text{ kPa.}$$

$$m_a = 1 \text{ kg of dry air.}$$

100%

$$\omega = \frac{m_v}{m_a} = \frac{10 \times 10^{-3}}{1} = 10^{-2} \text{ kg vapour/kg dry air.}$$

$$\omega = 0.622 \frac{P_v}{P_t - P_v}$$

$$0.01 = 0.622 \frac{P_v}{101.3 - P_v}$$

$$P_v = 1.6 \text{ kPa.}$$

$$\phi = \frac{P_v}{P_{vs}} = \frac{1.6}{3.167} \times 100 = 50.6 \%$$

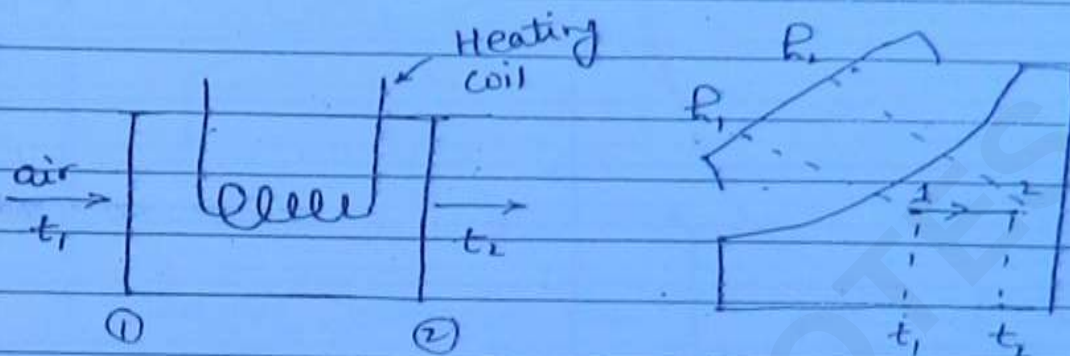
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Basic Psychrometric process

1. SENSIBLE HEATING:-

(111)

It is the process of heating air at constant w .



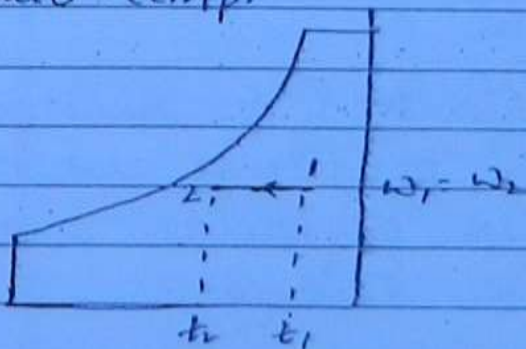
Applying steady flow eqn.

$$h_1 + Q_1 = h_2 + W_2$$

$$Q_1 = h_2 - h_1$$

2. SENSIBLE COOLING:-

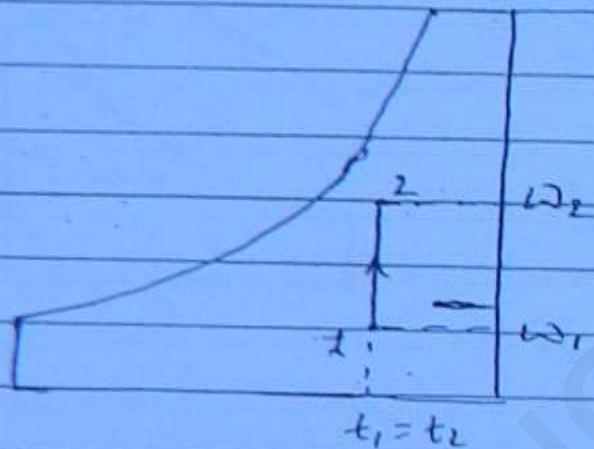
It is the process of cooling air at constant w . For sensible cooling the coil temp. must be greater than dew point temp. & less than dry bulb temp.



3. Humidification:

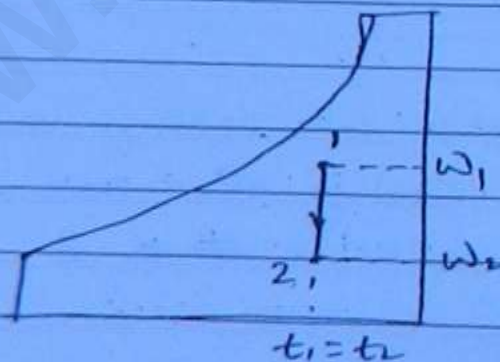
(1/2)

It is the process of increasing ω at constant dry bulb temp.



4. De-humidification:

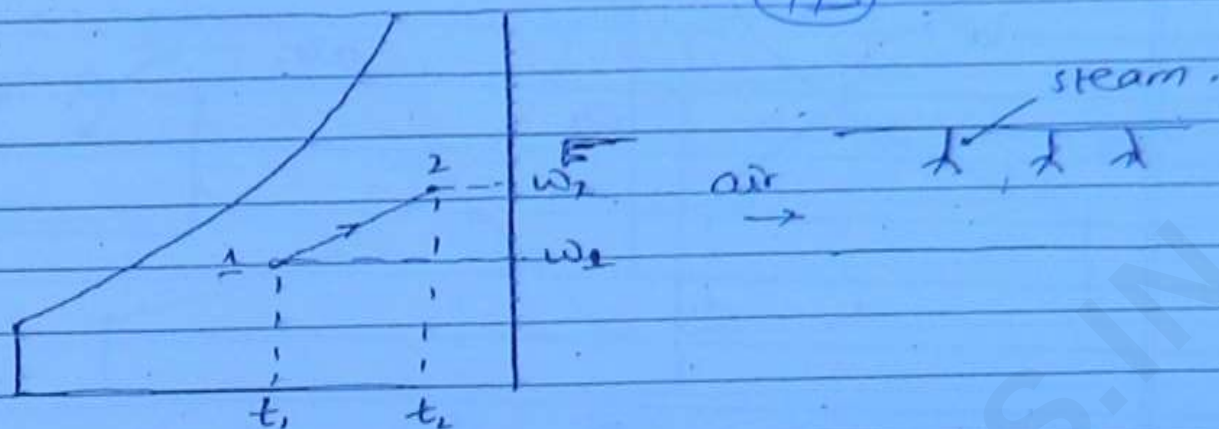
It is the process of removing moisture at constant dry bulb temp.



NOTE: Pure humidification & pure de-humidification processes are not possible because these processes are associated with temp. change.

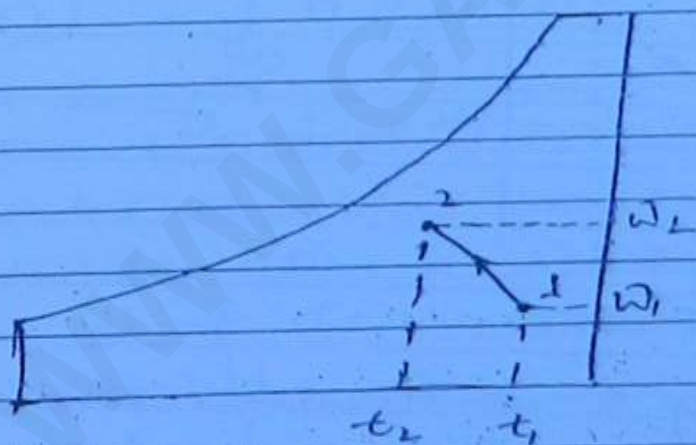
Heating & Humidification.

(1/2)



Steam spray in air example of heating & humidification

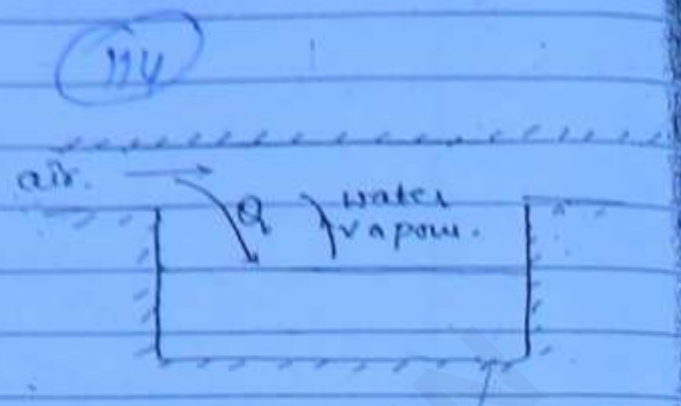
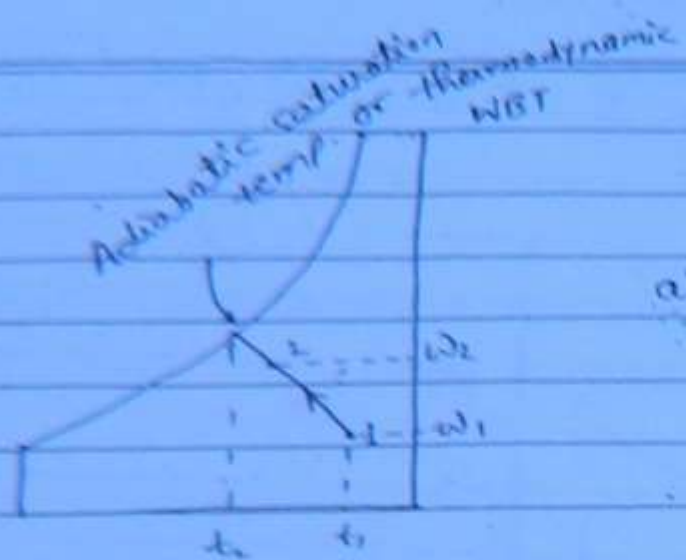
Cooling & Humidification:-



NOTE: On psychrometric chart adiabatic process are represented along const. enthalpy line.

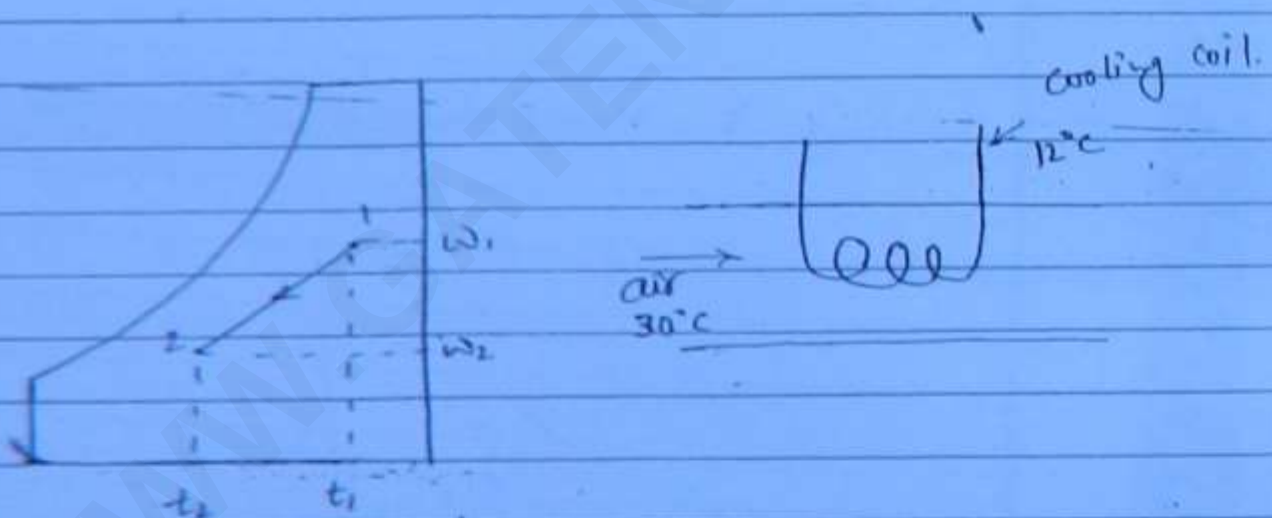
$$Q = h_2 - h_1$$

$$0 = h_2 - h_1 \quad [h_2 = h_1 = \text{const.}]$$



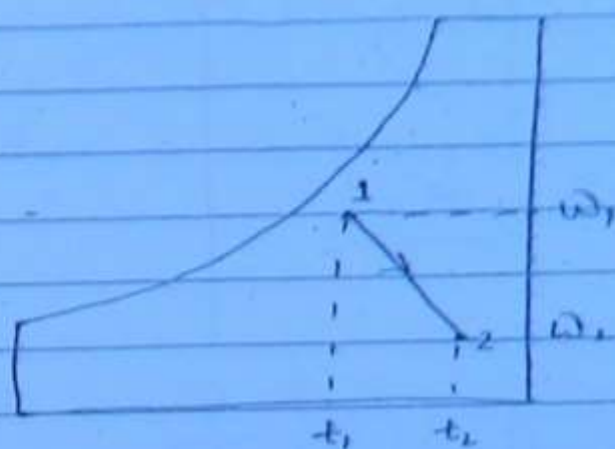
7. Cooling & De-humidification:-

Cooling & De-humidification occurs when the coil temp. is less than dew point temp.

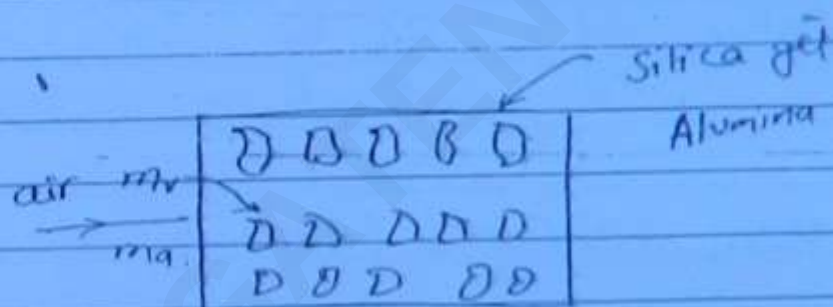


For cooling & de-humidification the coil temp. must be less than dew point temp. this process is generally followed in summer.

8. Heating & De-humidification :-



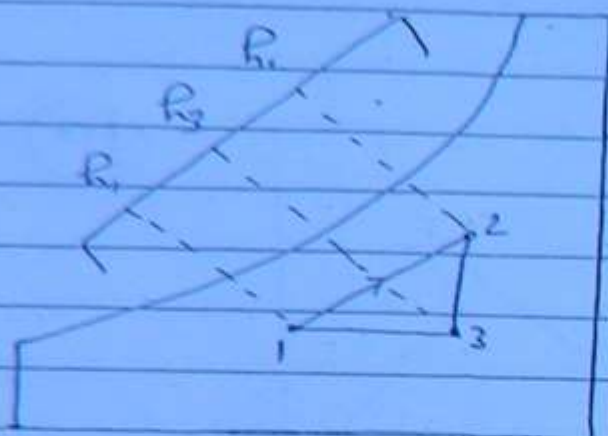
Example:- Special case :- Adiabatic chemic dehumidification.



Certain chemical like silica gel, Alumina are used for absorbing moisture as latent heat of condensation is released the dry bulb temp. of air increases.

9. SENSIBLE HEAT FACTOR SHF :-

It is the ratio of sensible heat to the total heat.



1/16

$$TH = h_2 - h_1$$

$$SH = h_3 - h_1$$

$$LH = h_2 - h_3$$

$$TH = SH + LH$$

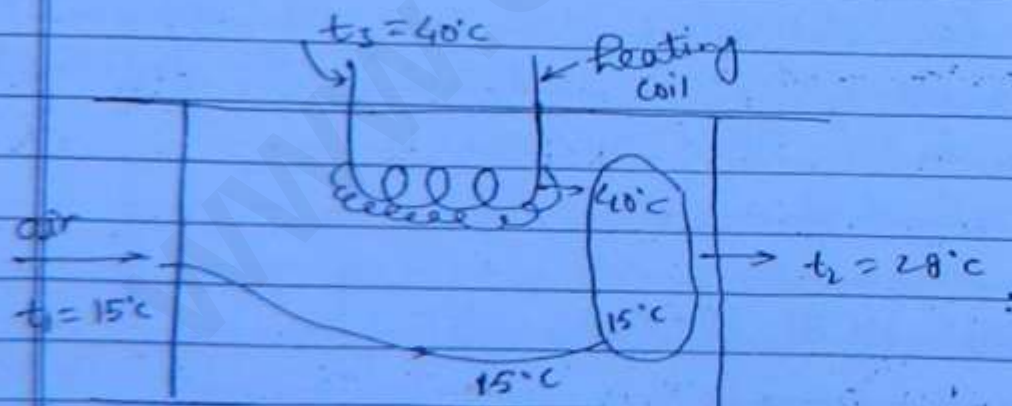
$$= h_2 - h_1 + h_2 - h_3$$

$$TH = h_2 - h_1$$

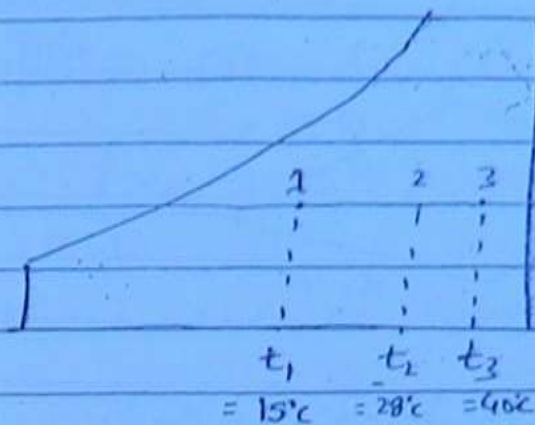
$$SHF = \frac{SH}{TH} = \frac{h_3 - h_1}{h_2 - h_1}$$

* By Pass factor :- (BPF) :-

It is also known as Uncontacted factor or loss factor.



NOTE- For moist air 3 parameters are required to fix the state but on psychrometric chart only 2 parameters are required to locate the state because the chart is drawn for a fixed pressure i.e. the third parameter is fixed.



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$$BPF = \frac{t_3 - t_2}{t_3 - t_1}$$

$$\eta = \frac{t_2 - t_1}{t_3 - t_1}$$

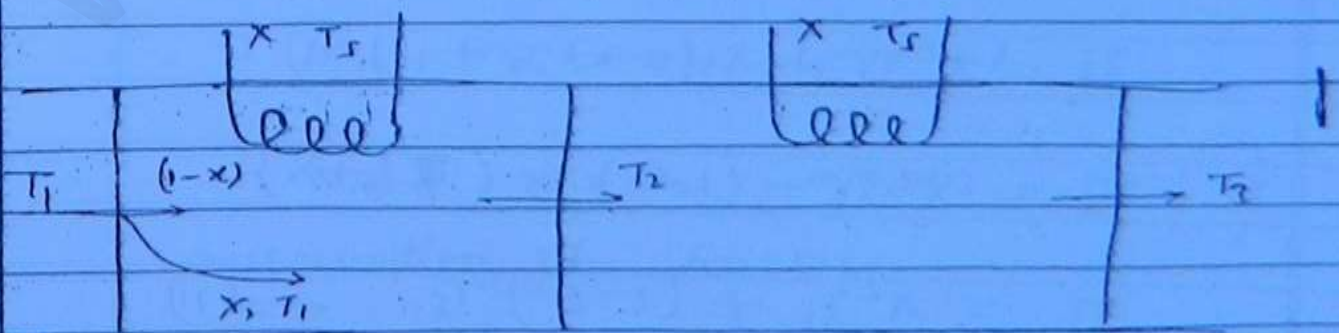
$$\eta + BPF = \frac{t_2 - t_1}{t_3 - t_1} + \frac{t_3 - t_2}{t_3 - t_1} = 1$$

$$\boxed{\eta = 1 - BPF}$$

Imp.

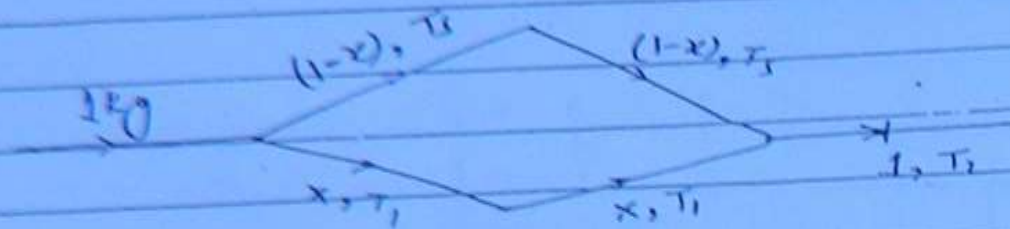
★

Effective By pass factor or
Equivalent By-pass factor.



$$\boxed{h = \text{CPT}}$$

$$\dot{m} h_1 + \dot{m} h_2 = \dot{m} h_3$$



$$h = C_p T$$

$$x C_p T_1 + (1-x) C_p T_3 = 1 \times C_p T_2$$

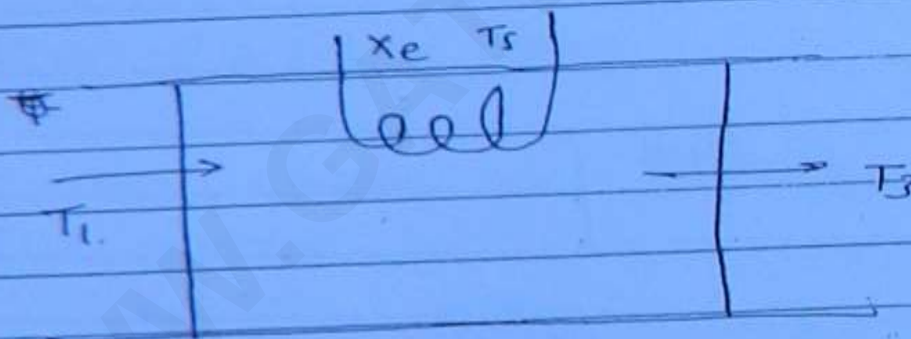
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$$T_2 = x T_1 + (1-x) T_3$$

Similarly,

$$T_3 = x T_2 + (1-x) T_5$$

$$T_3 = x [x T_1 + (1-x) T_5] + (1-x) T_5$$



$$T_3 = x_e T_1 + (1-x_e) T_5 \quad \text{--- (1)}$$

$$T_3 = x^2 T_1 + x (1-x) T_5 + (1-x) T_5$$

$$T_3 = x^2 T_1 + (1-x) T_5 (1+x)$$

$$T_3 = x^2 T_1 + (1-x^2) T_5 \quad \text{--- (ii)}$$

Compare ① & ②

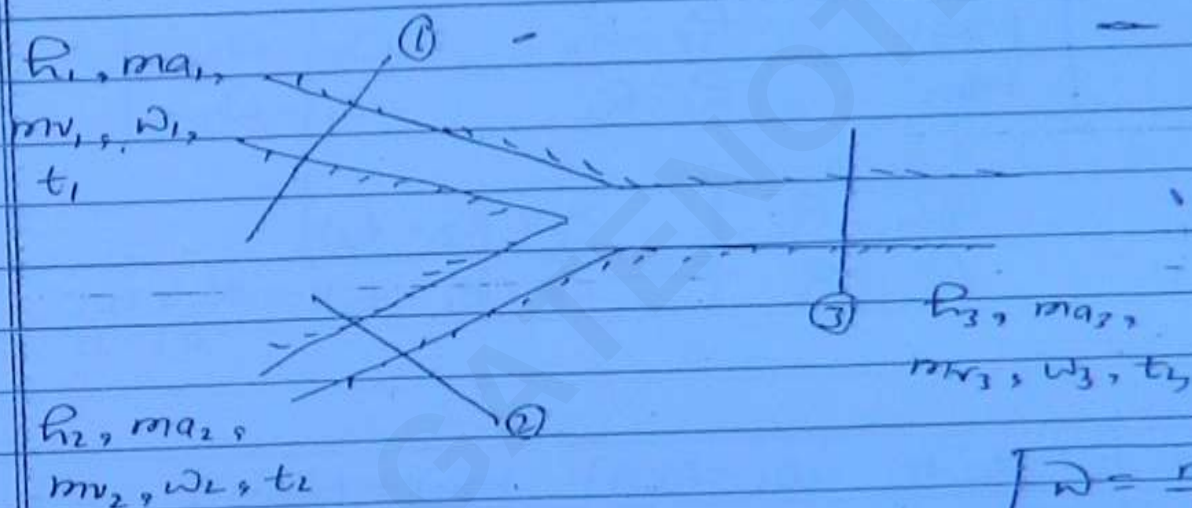
$$[X_e = X^2]$$

If there are n number of similar coil.

$$[X_e = X^n]$$

(119)

* Mixing of air stream *



Conservation of mass.

$$ma_1 + ma_2 = ma_3 \rightarrow (\text{dry air}) \text{ --- ①}$$

$$mv_1 + mv_2 = mv_3 \text{ (vapour)}$$

$$ma_1 w_1 + ma_2 w_2 = ma_3 w_3 \text{ --- ②}$$

Conservation of Energy

$$ma_1 R_1 + ma_2 R_2 = ma_3 R_3 \text{ --- ③}$$

from eqn (i) put the value of m_{13} in eqn. (ii)

$$m_{a1}h_1 + m_{a2}h_2 = (m_{a1} + m_{a2})h_3$$

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$$m_{a1}h_1 + m_{a2}h_2 = m_{a1}h_3 + m_{a2}h_3$$

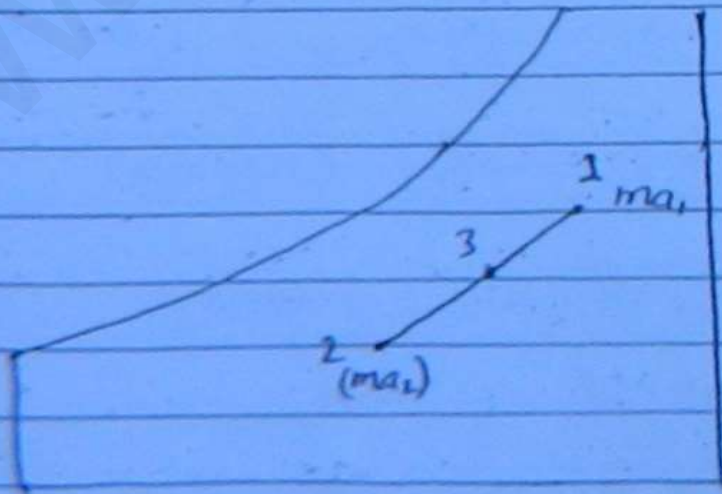
$$m_{a1}h_1 - m_{a1}h_3 = m_{a2}h_3 - m_{a2}h_2$$

$$m_{a1}(h_1 - h_3) = m_{a2}(h_3 - h_2)$$

$$\frac{m_{a1}}{m_{a2}} = \frac{h_3 - h_2}{h_1 - h_3} = \frac{w_3 - w_2}{w_1 - w_3}$$

$$\frac{h_3 - h_2}{h_1 - h_3} = \frac{C_p(t_3 - t_2)}{C_p(t_1 - t_3)}$$

$$\frac{m_{a1}}{m_{a2}} = \frac{t_3 - t_2}{t_1 - t_3} = \frac{w_3 - w_2}{w_1 - w_3}$$

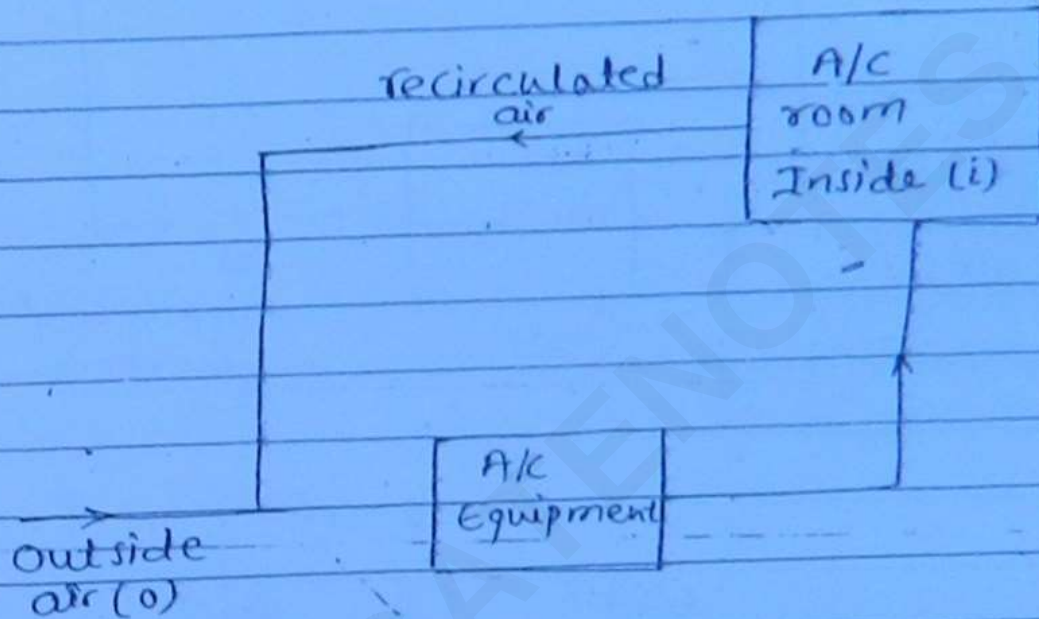


$$m_{a1} = t_3 - t_2 \quad \& \quad m_{a2} = t_1 - t_3$$

The Resultant point 3 device the line joining 1 & 2 in the inverse ratio of their masses.

(12)

* Ventilation Air *



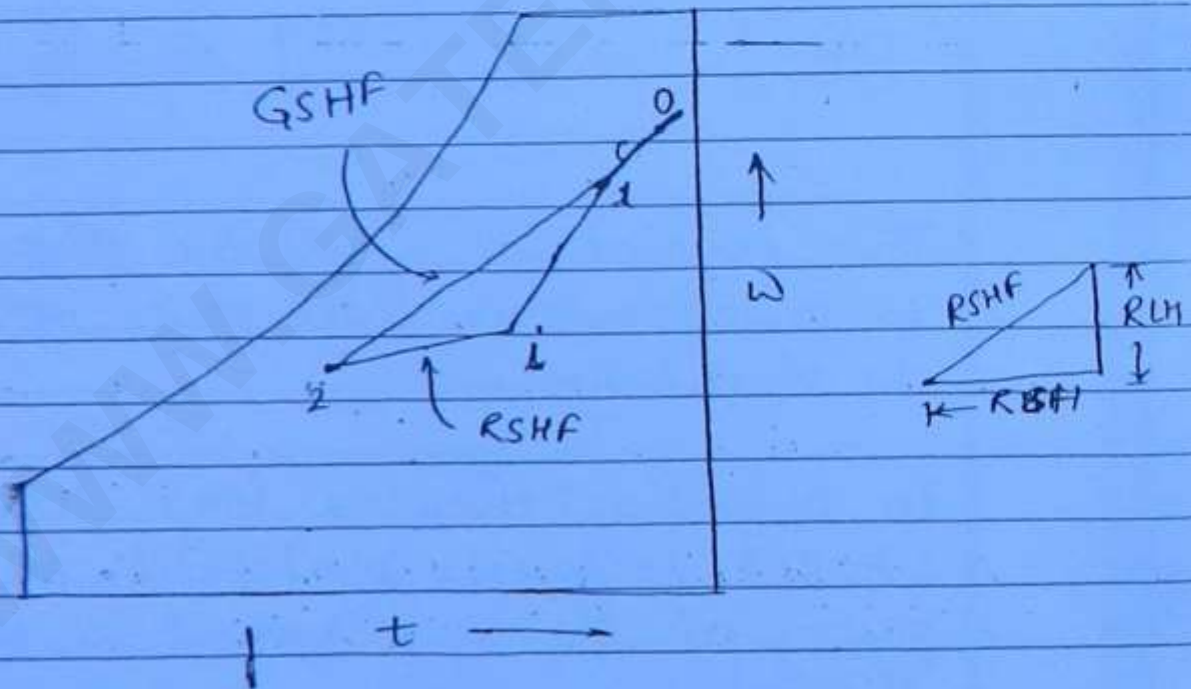
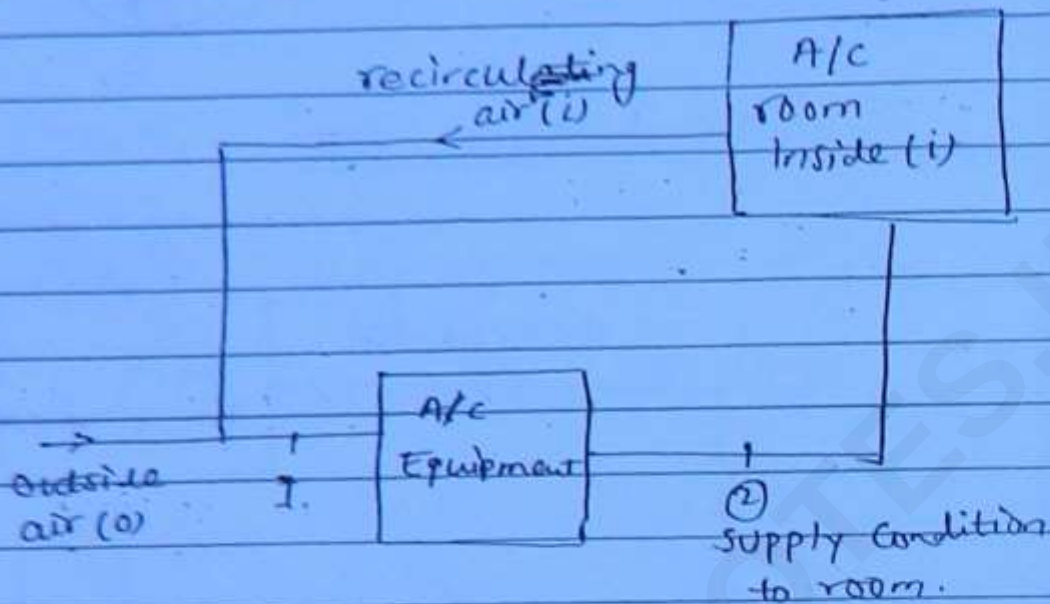
The amount of fresh air that is supplied in order to maintain purity of air is known as ventilation air.

In operation theater 100% fresh air is supplied because availability of oxygen & purity of air is important in operation theater.

Generally recirculated air is 60 to 70% & fresh air is 30 to 40%.

* Summer air-conditioning with ventilation air *

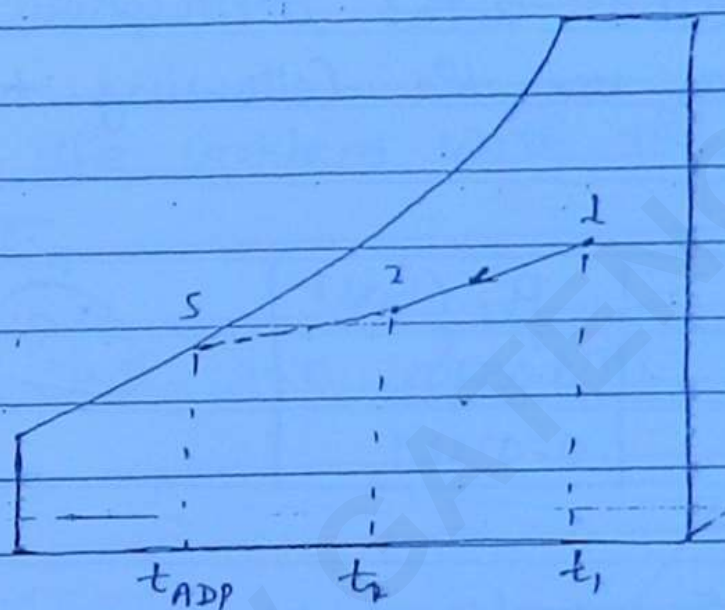
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The line which joins inlet & exit condition of A/c equipment is known as Grand Sensible heat factor line.

- The line which joint's supply condition's with inside condition's is known as room sensible heat factor line. The point of intersection of GSHE & RSHE will give supply condition's to room.

Apparatus Dew point (ADP)



It is the temp. at which cooling & dehumidification line cut's saturation curve's.

$$BPF = \frac{t_2 - t_s}{t_1 - t_s}$$

Imp

prob-

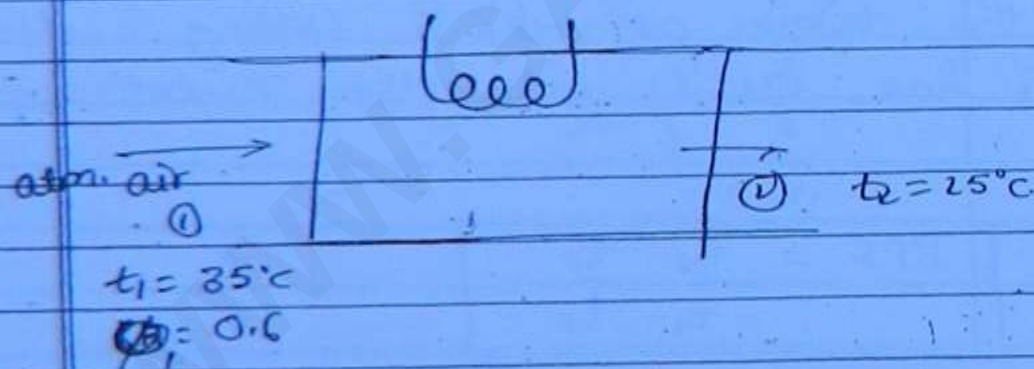
5 gram of water vapour per kg of dry air from atmospheric air is removed & the temp. of air after removing water vapour becomes 25°C dry bulb temp. find the relative humidity & enthalpy of moist air after removal of moisture. Assume atmospheric air at 35°C DBT & 60% RH take total atmospheric pressure as 1.013 bar use the following table.

$t^{\circ}\text{C}$	P_{vs} (bar)
35	0.05733
25	0.03229

(124)

Soln:

$m_v = 5 \text{ gram/kg of dry air.}$



$$W_1 = 0.022 \quad \frac{P_{v1}}{P_t - P_{v1}}$$

$$\phi_1 = \frac{P_{v1}}{P_{vs1}}$$

$$P_{vs1} = 0.05733$$

$$TH = SH + LH$$

$$5 = 4 + LH$$

(125)

$$[LH = 1] \text{ Ans.}$$

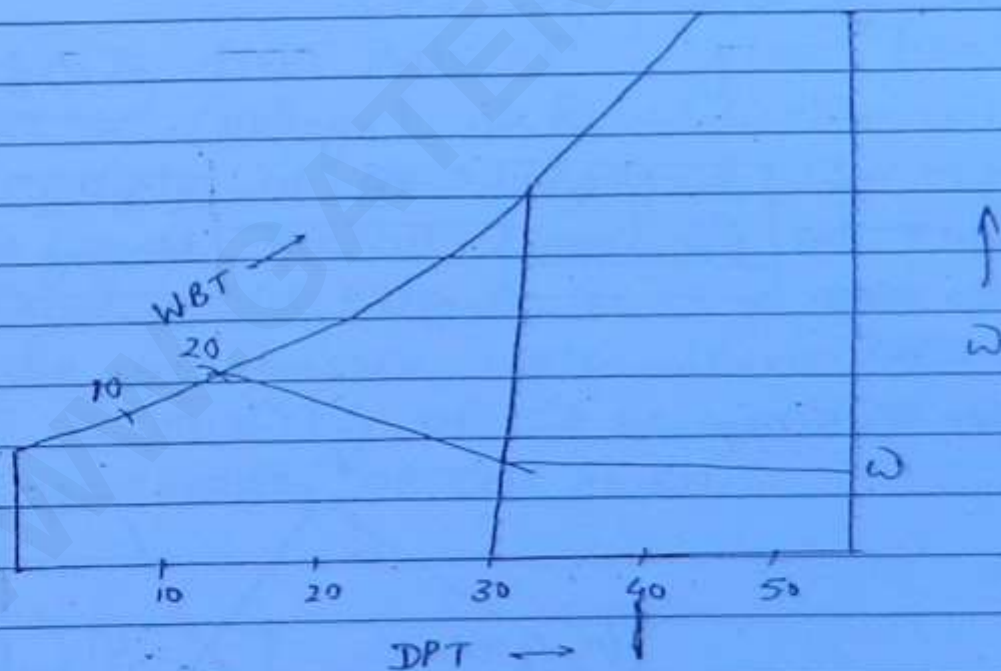
prob- Air at a DBT 30°C has a WBT of 20°C find

- (i) Relative humidity (ii) $\Rightarrow$ Enthalpy
(iii) Dew point temp. (iv) Specific volume

Solve the problem with the help of psychrometric chart.

Soln:-

$$DBT = 30^\circ\text{C} \quad \& \quad WBT = 20^\circ\text{C}$$



$$RH = 40\% \quad , \quad H = 57 \quad , \quad DPT = 15^\circ\text{C}$$

$$W = 0.0105 \text{ to } 0.0107 \quad , \quad \phi = 39 \text{ to } 40\%$$

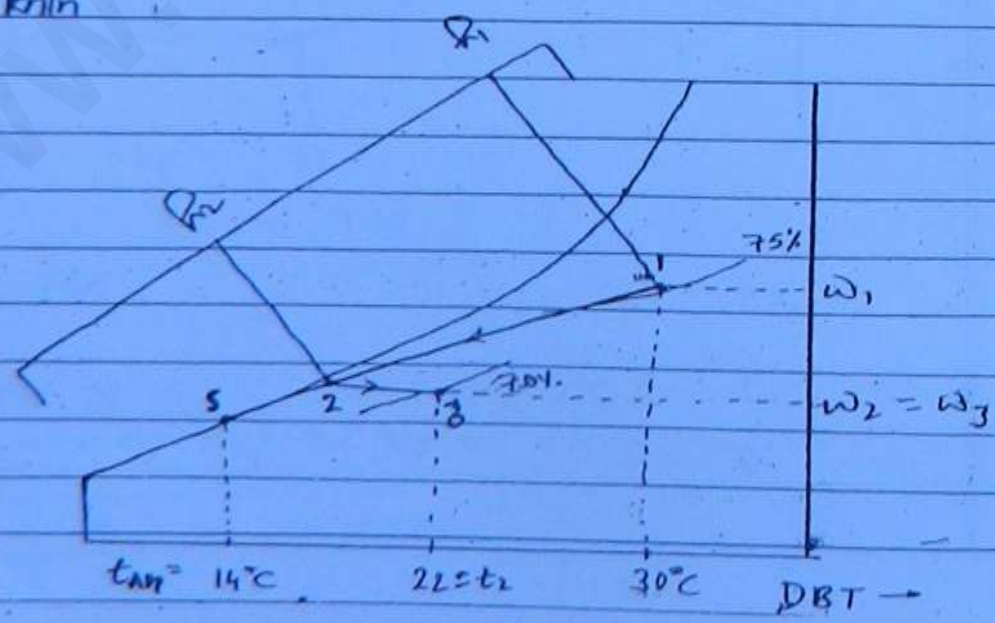
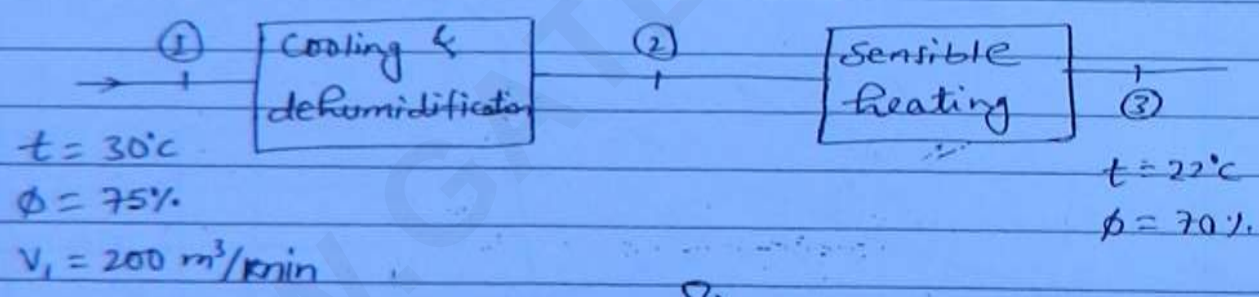
$$v = 0.87 \text{ to } 0.876$$

prob- A small auditorium is required to be maintain at 22°C DBT & 70% RH. The ambient condition's are 30°C DBT & 75% RH. The amount of free air circulated is $200\text{ m}^3/\text{min}$. The required condition's are acheived first by cooling & dehumidification through a coil having ADP of 14°C & then by sensible heating with the help of psychrometric chart find -

- ① Cooling capacity of the coil in TR & it's Bypass factor.
- ② Amount of water vapour removed by cooling coil in kg vapour per hour.

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501 m³



$$v = \frac{V}{m} \quad \& \quad m = \frac{V_1}{v_1} = \frac{127}{v_1}$$

$$\text{Cooling load} = (h_2 - h_1) \text{ kJ/kg da.} =$$

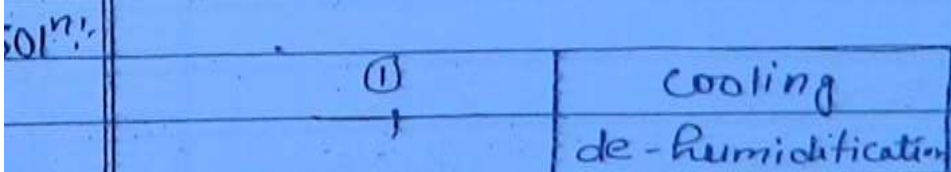
$$\text{Total cooling load} = m (h_2 - h_1) \rightarrow \frac{\text{kg da}}{\text{min}} \times \frac{\text{kJ}}{\text{kg da}} = 37 \text{ to } 38$$

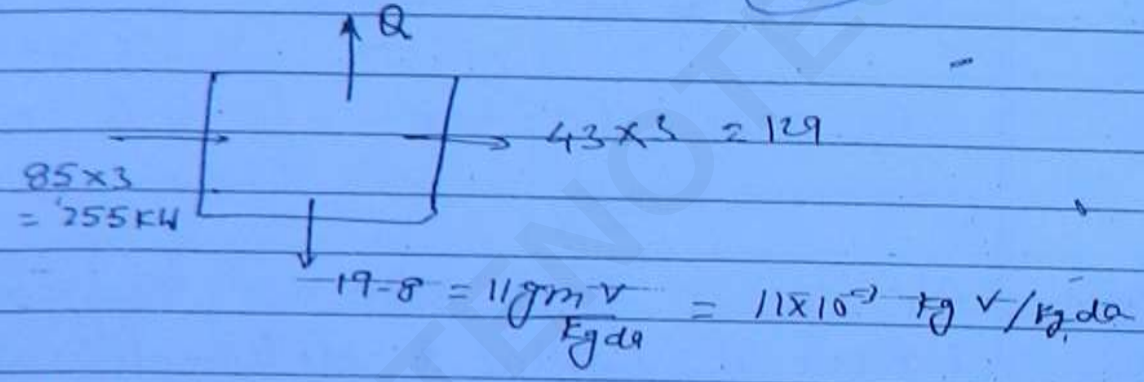
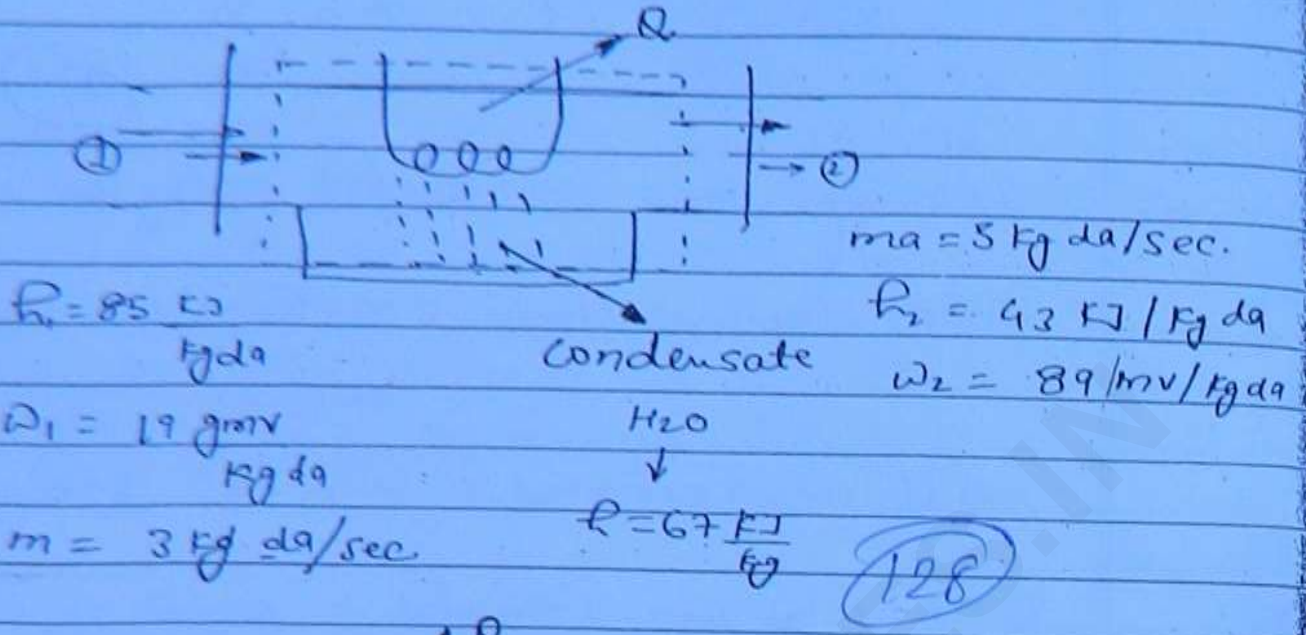
$$\text{moisture removed} = (w_1 - w_2) \text{ kg v/kg da.}$$

$$\text{Total moisture removed} = m (w_1 - w_2) \rightarrow \frac{\text{kg da}}{\text{min}} \times \frac{\text{kg v}}{\text{kg da}} = 106 \text{ to } 140$$

$$\text{By pass factor} = 0.12 \text{ to } 0.2$$

26- Air at a flow rate of 3 kg/sec. (Dry air basis) Enters a cooling & dehumidification coil with an enthalpy of 85 kJ/kg da & $w = 19$ gram of vapour/kg da the air leaves the coil with an enthalpy of 43 kJ/kg da & $w = 8$ gram of vapour/kg da. If the condensate water leaves the coil with an enthalpy of 67 kJ/kg then find cooling capacity of coil in kW.





Total H_2O removed = $3 \times 11 \times 10^{-3} = 33 \times 10^{-3} \text{ kg}$

Heat Carrying by $\text{H}_2\text{O} = 33 \times 10^{-3} \times 67 = 2.211 \text{ kW}$

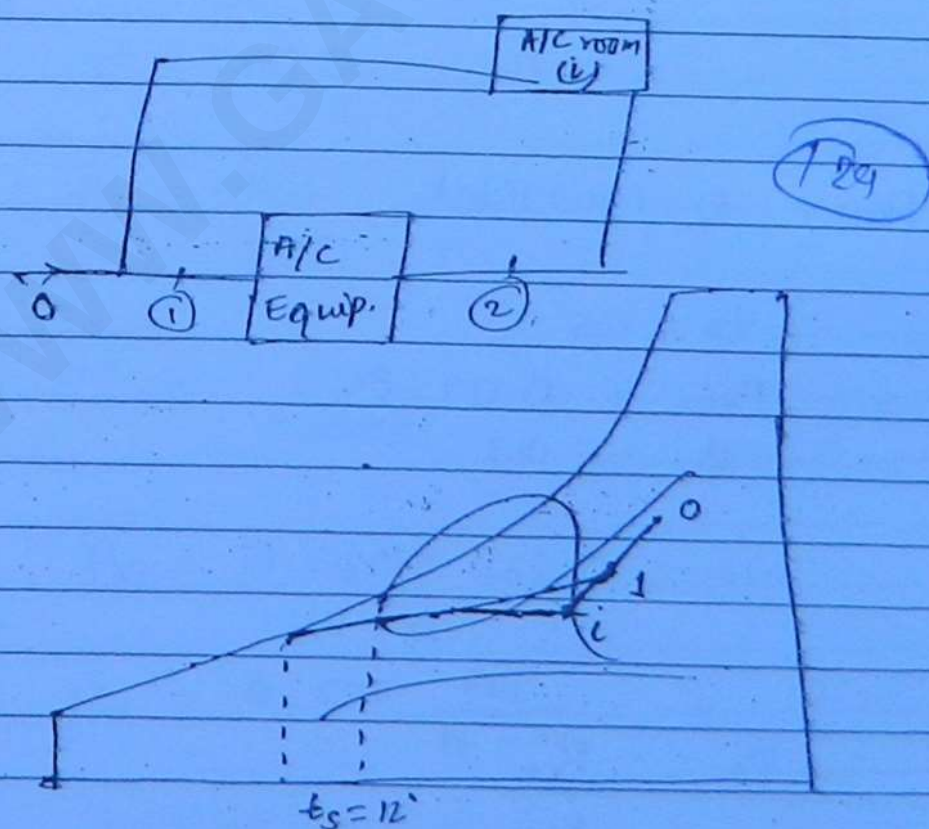
$255 = Q + 2.211 + 129$

$Q = 123.79 \text{ kW}$

prob- In an A/C. Space 15 kg/sec. of fresh air at 45°C DBT and 30% RH is introduced in room at 25°C DBT & 50% RH. 450 kg/sec. of recirculated air is mixed with outside air and this mixed air then flows over a cooling coil with ADE 12°C & the coil by pass factor is 0.15. Determine the condition at outlet of cooling coil, Room Sensible Heat, Room latent heat & cooling load of the coil. solve the problem without using psychrometric chart ~~use~~ use the following table.

$t^{\circ}\text{C}$	12	25	45	$P_t = 1.0$
P_v (bar)	0.014016	0.03166	0.09589	

Soln:



$$0.3 = \frac{P_{v0}}{0.09584}$$

$$P_{v0} = 0.0287$$

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$$W_0 = \frac{0.62 \times 0.0287}{1.013 - 0.0287}$$

$$W_0 = 0.01816$$

$$h_0 = C_{pat} + W_0 (2500 + 1.88t)$$

$$h_0 = 1.005 \times 45 + 0.01816 (2500 + 1.88 \times 45)$$

$$h_0 = 45.87 \text{ kJ/kg da.}$$

Point (i) —

$$t_i = 25, \quad \phi_i = 0.5, \quad P_{vsi} = 0.03166$$

$$P_t = 1.013 \text{ bar.}$$

$$\phi_i = \frac{P_{vi}}{P_{vsi}}$$

$$0.5 = \frac{P_{vi}}{0.03166}$$

$$P_{vi} = 0.0158$$

$$W_i = 0.622 \frac{P_{vi}}{P_t - P_{vi}}$$

$$\omega_i = \frac{0.622 \times 0.0158}{1.013 - 0.0158}$$

$$\omega_i = 0.00987$$

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$$h_i = c_{p0} t_i + \omega_i (2500 + 1.88 t_i)$$

$$= 1.005 \times 25 + 0.00987 (2500 + 1.88 \times 25)$$

$$= 50.27 \text{ kJ/kg dry}$$

Point (1),

$$m_o \omega_o + m_i \omega_i = m_1 \omega_1$$

$m_o = 50$ fresh air given.

$m_i = 450$ recirculated given.

$$m_1 = 50 + 450 = 500$$

$$50 \times 0.01816 + 450 \times 0.00987 = 500 \omega_1$$

$$\omega_1 = 0.0107$$

$$m_o h_o + m_i h_i = m_1 h_1$$

$$50 \times 92.2 + 450 \times 50.2 = 500 \times h_1$$

$$h_1 = 54.4$$

$$h_1 = c_{p0} t_1 + \omega_1 (2500 + 1.88 t_1)$$

$$t_1 = 27^\circ\text{C}$$

Point (S),

$$t_s = 12^\circ\text{C}, \quad \phi_s = 1, \quad P_{vs} = 0.014016$$

$$\phi_s = \frac{P_v}{P_{vs}}$$

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$$1 = \frac{P_v}{P_{vs}}$$

$$\Rightarrow P_v = P_{vs} = 0.014016$$

$$P_v = 0.0$$

$$\omega_s = 0.622 \frac{P_{vs}}{P_t - P_v}$$

$$\approx 0.622 \times \frac{0.014016}{1.013 - 0.014016}$$

$$\omega_s = 0.00872$$

$$h_s = c_{p9} t_s + \omega_s (2500 + 1.88 t_s)$$

$$= 1.005 \times 12 + 0.00872 (2500 + 1.88 \times 12)$$

$$= 34.07 \text{ kJ/kg da.}$$

$$BPF = \frac{T_2 - T_s}{T_1 - T_s}$$

$$0.15 = \frac{t_2 - 12}{27 - 12}$$

$$t_2 = 14.25^\circ\text{C}$$

$$BPF = \frac{t_2 - t_s}{t_1 - t_s} \times \frac{C_p}{C_p}$$

$$BPF = \frac{h_2 - h_s}{h_1 - h_s}$$

(134)

$$0.15 = \frac{h_2 - 34.07}{54.4 - 34.07}$$

$$h_2 = 37.1 \text{ kJ/kg da.}$$

$$h_2 = C_{p0} t_2 + W_2 (2500 + 1.88 t_2)$$

$$37.1 = 1.005 \times 14.25 + W_2 (2500 + 1.88 \times 14.25)$$

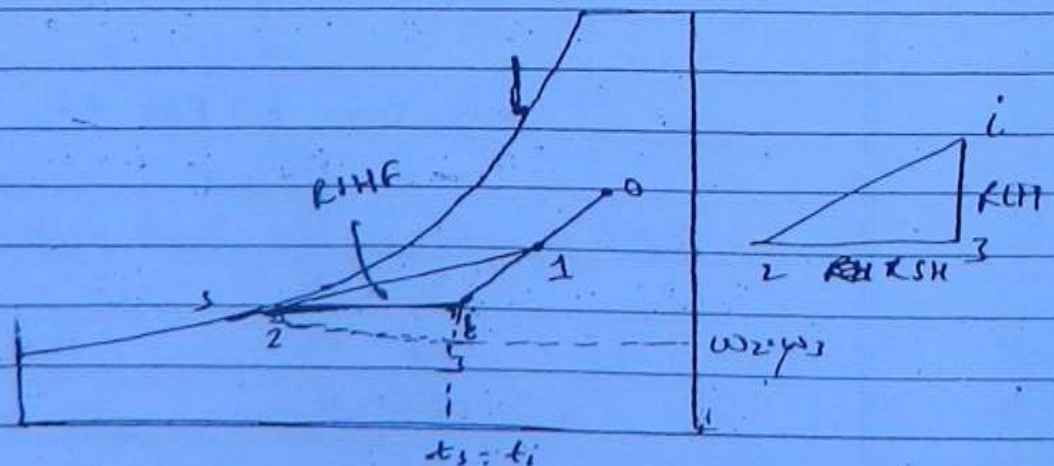
$$W_2 = 0.00902$$

i) Cooling load = $m (h_1 - h_2)$

$$= 500 (54.4 - 37.1)$$

$$\frac{\text{kg}}{\text{sec}} \times \frac{\text{kJ}}{\text{kg}}$$

$$= 8650 \text{ kW}$$



$$h_3 = C_p a t_3 + w_3 (2500 + 1.88 t_3)$$

$$= 1.005 \times 25 + 0.00902 (2500 + 1.88 \times 25)$$

$$= 48.1 \text{ kJ/kg da.}$$

(135)

$$R_{SH} = m (h_3 - h_L)$$

$$R_{SH} = 500 (0.481 - 37.1)$$

$$R_{SH} = 5480 \text{ kW.}$$

$$R_{LH} = m (h_i - h_3)$$

$$= 500 (50.27 - 48.1)$$

$$= 1090 \text{ kW.}$$

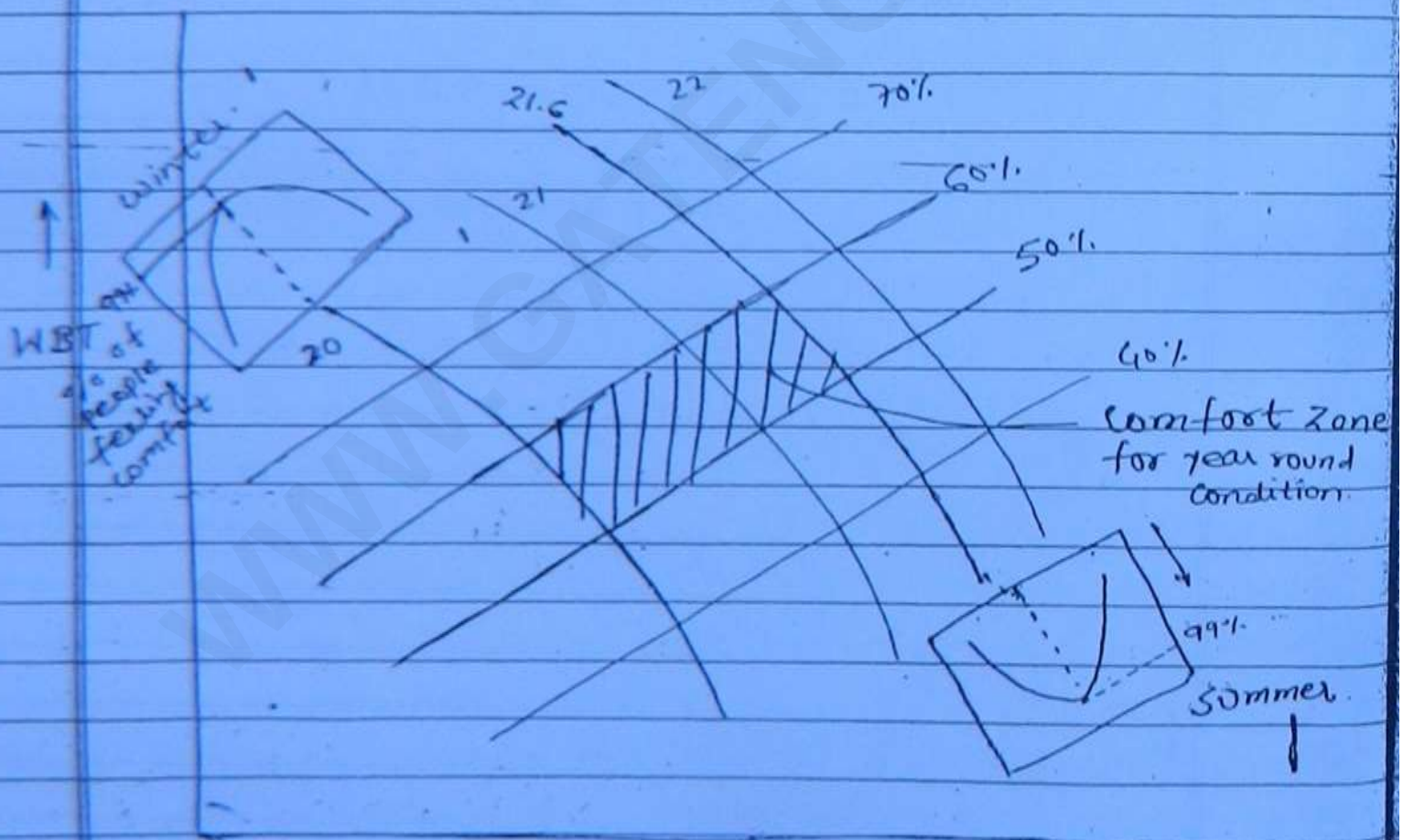
Effective temperature

It is the temp. of saturated air at which a person would experience same feeling of comfort as in unsaturated environment.

Effective temp. includes temp. (DB), Humidity and air velocity.

Comfor chart

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DBT →

Winter → ET → 20°C → 99% people comfort feel.

Summer → ET → 21.5°C → 99% " " " "

This chart was developed by ASHRAE i.e. American Society for Heating, Refrigeration & air-conditioning engineers by conducting research on different people throughout the world & in this chart ~~dry~~ bulb temp. is taken on x-axis & wet bulb temp. is taken on y-axis. Relative humidity lines are drawn from psychrometric chart. It is found from sea research that an effective temp. about 20°C , 99% ~~pepo~~ people were comfortable in winter condition & in summer it is 21.5°C .

The shaded area shows comfort zone for year round conditioning.

Refrigeration Equipment

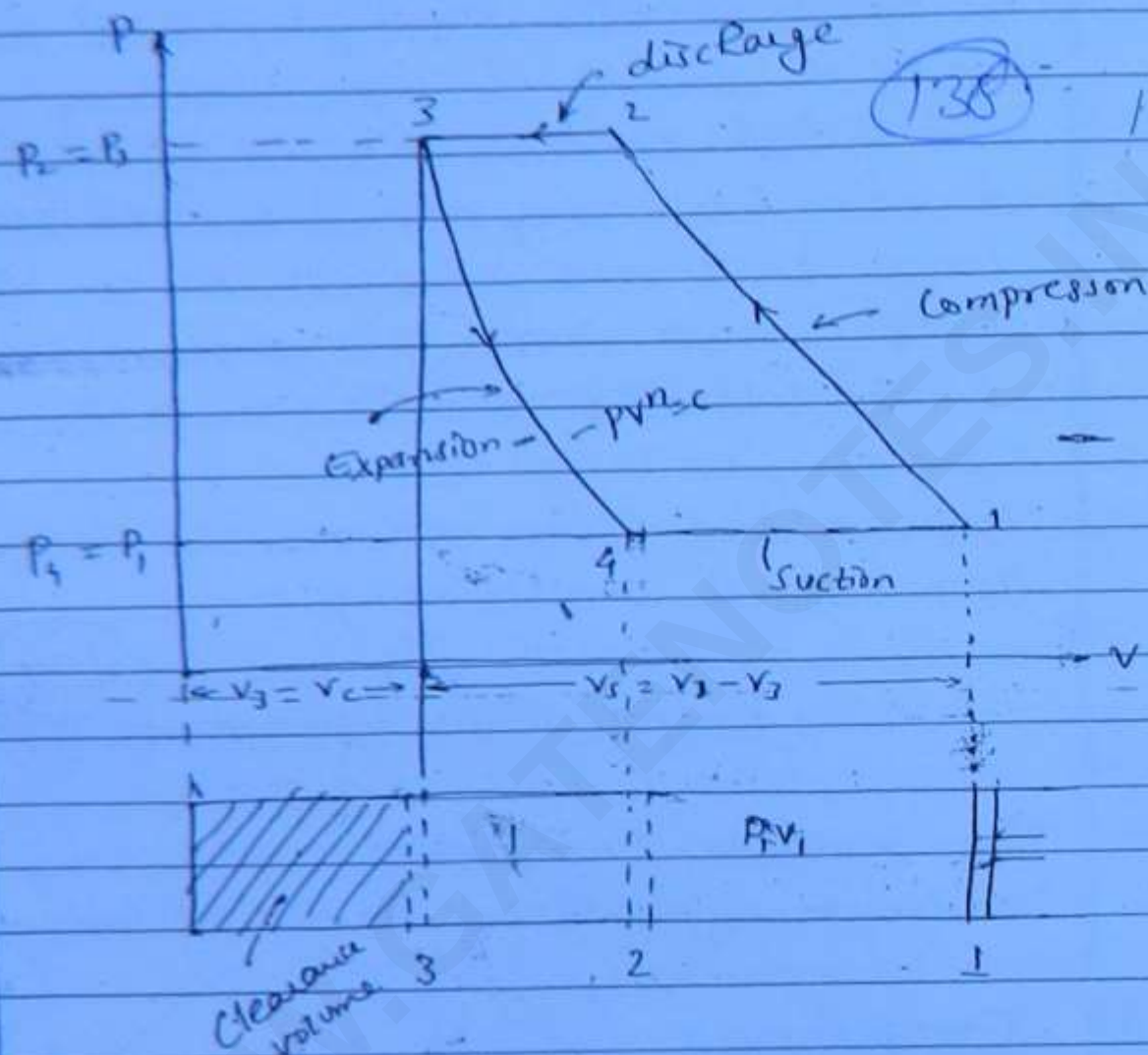
The principle components are, Compressor, Condensor, Expansion device & Evaporator.

Compressor :-

Functions of Compressor.

1. To draw refrigerant from evaporator.
2. To compress the refrigerant.
3. To discharge high pressure refrigerant to Condenser.

* Volumetric efficiency of a Reciprocating Compressor :-



$$\eta_{vol} = \frac{V_{act}}{V_{swp}} = \frac{V_1 - V_4}{V_1 - V_3}$$

$$C = \frac{V_c}{V_s} = \frac{V_3}{V_1 - V_3}$$

$$\eta_{vol} = \frac{V_1 - V_4 + V_3 - V_3}{V_1 - V_3}$$

$$C = \frac{V_3}{V_1 - V_3}$$

$$\eta_{vol} = \frac{V_1 - V_3 - (V_4 - V_3)}{V_1 - V_3}$$

$$\eta_{\text{OVDI}} = 1 - \frac{(V_4 - V_3)}{V_1 - V_2}$$

$$\eta_{\text{VOI}} = 1 - \frac{V_3 (V_4/V_3 - 1)}{V_1 - V_2}$$

$$\eta_{\text{VOI}} = 1 - C \left(\frac{V_4}{V_3} - 1 \right)$$

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$$\eta_{\text{VOI}} = 1 + C - C \left(\frac{V_4}{V_3} \right)$$

Expansion is taken as polytropic.

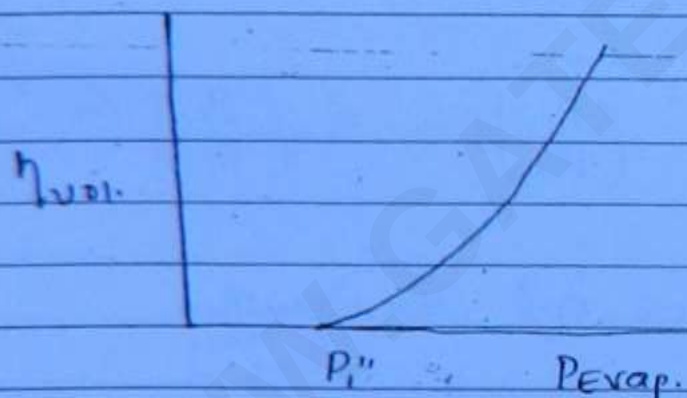
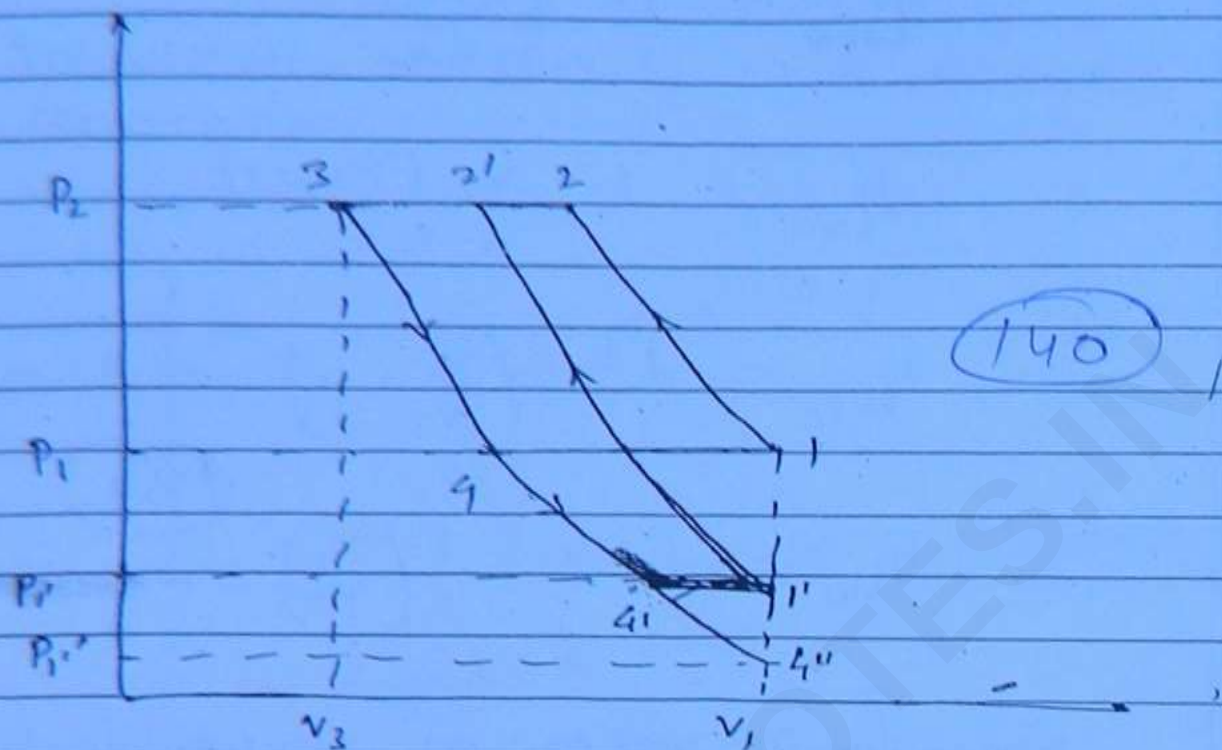
$$P_3 V_3^n = P_4 V_4^n$$

$$\frac{V_4}{V_3} = \left(\frac{P_3}{P_4} \right)^{1/n} = \left(\frac{P_2}{P_1} \right)^{1/n}$$

n is index of expansion.

$$\eta_{\text{VOI}} = 1 + C - C \left(\frac{P_2}{P_1} \right)^{1/n}$$

★ Effect of variation of pressure ratio on volumetric efficiency :-

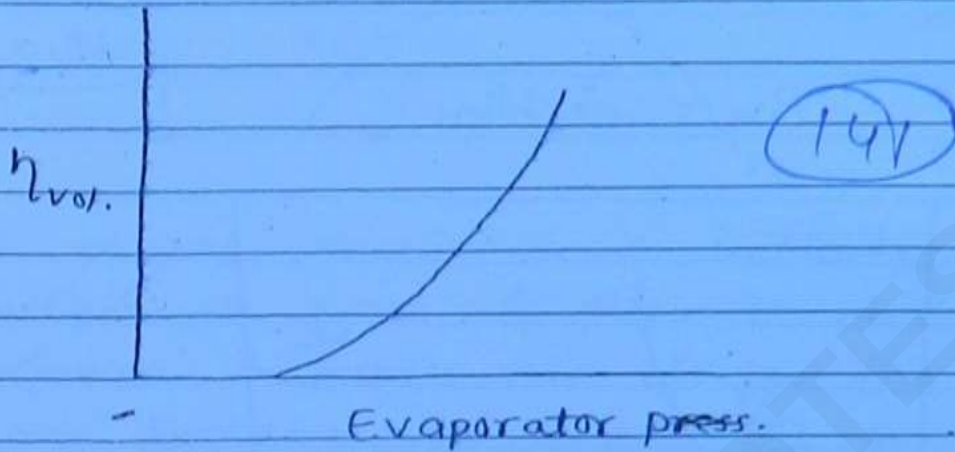


$$\eta_{vol} = 1 + C - C \left(\frac{P_2}{P_1} \right)^{1/n}$$

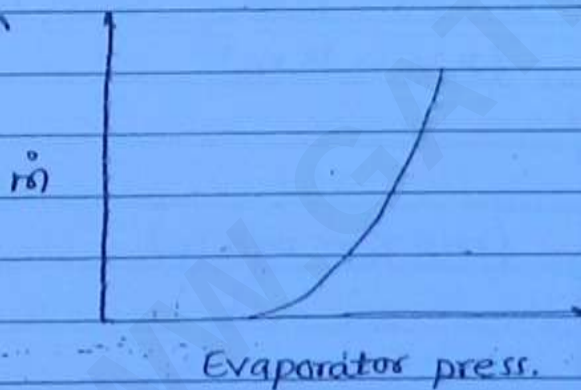
With decrease in evaporator pressure the volumetric efficiency decreases because the actual suction volume decreases & at one particular very low evaporator pressure (P_1'') the volumetric efficiency became zero.

Performance parameters:

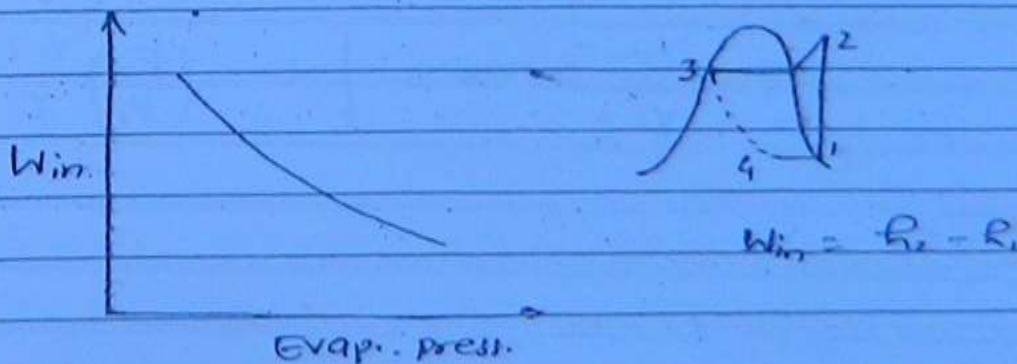
1. Volumetric Efficiency vs Evaporator pressure.



2. Evaporator pressure vs mass flow rate.



3. Evaporator pressure vs work input.

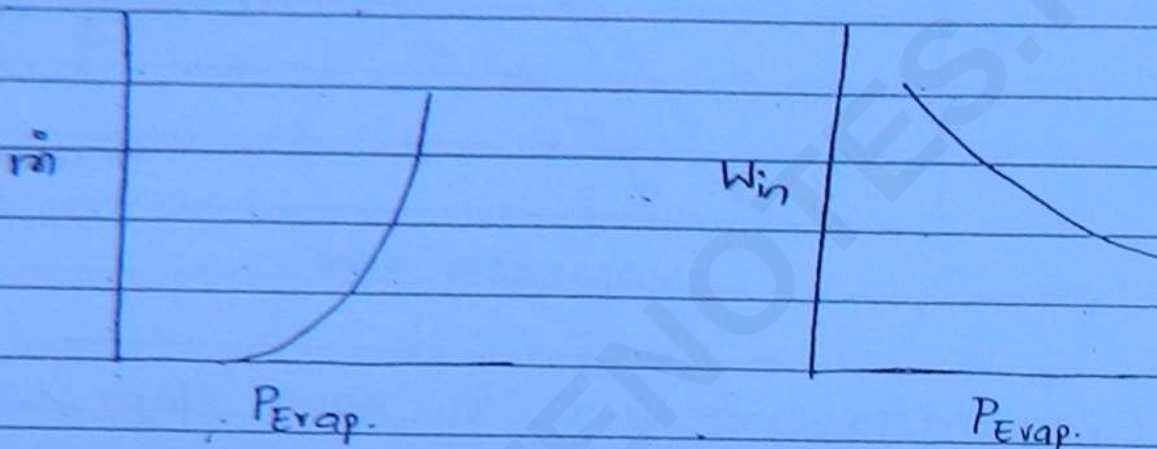


4. Evaporator press. v/s power input.

$$\text{Pin} = \dot{m} (h_2 - h_1)$$

$$\text{Pin} = \dot{m} W_{in}$$

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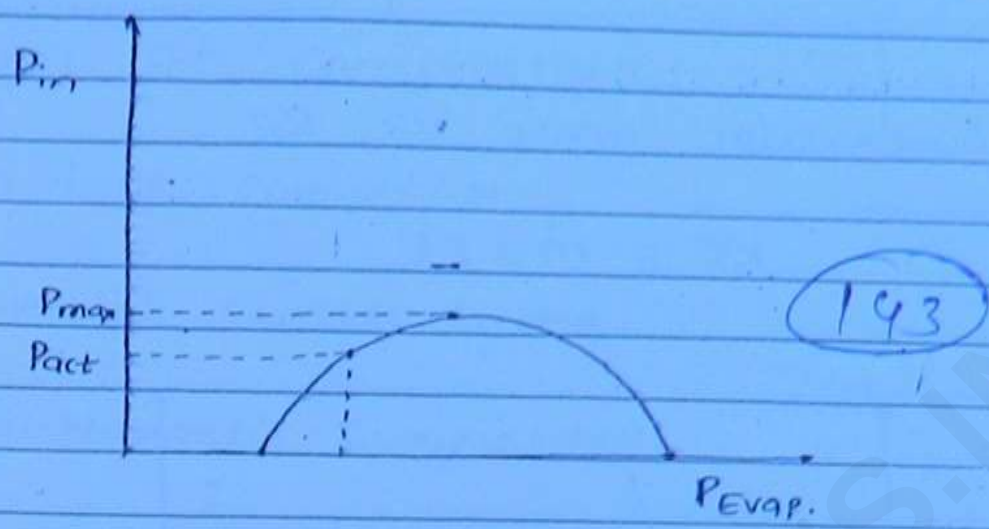


$$W = \frac{\gamma}{\gamma-1} (P_1 V_1 - P_2 V_2)$$

$$= \frac{\gamma}{\gamma-1} (m R T_1 - m R T_2)$$

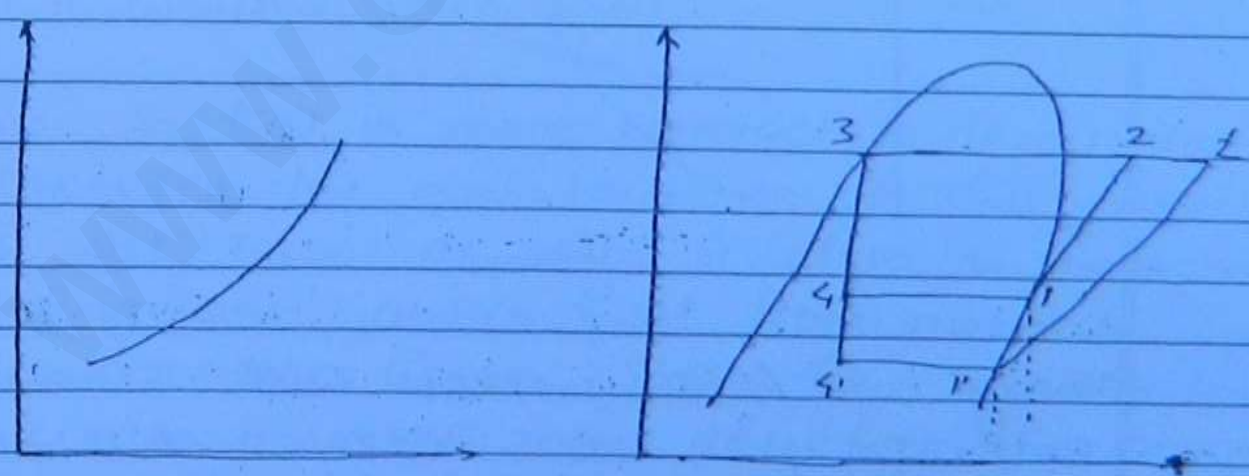
$$= \frac{\gamma}{\gamma-1} m R T_1 \left(1 - \frac{T_2}{T_1} \right)$$

$$= \frac{\gamma}{\gamma-1} m R T_1 \left[1 - \left(\frac{P_2}{P_1} \right)^{\frac{\gamma-1}{\gamma}} \right]$$



The power requirement initially increases reaches its peak & then it decreases. Compressor motor's are design for peak power because to reach the operating condition it should pass through peak power.

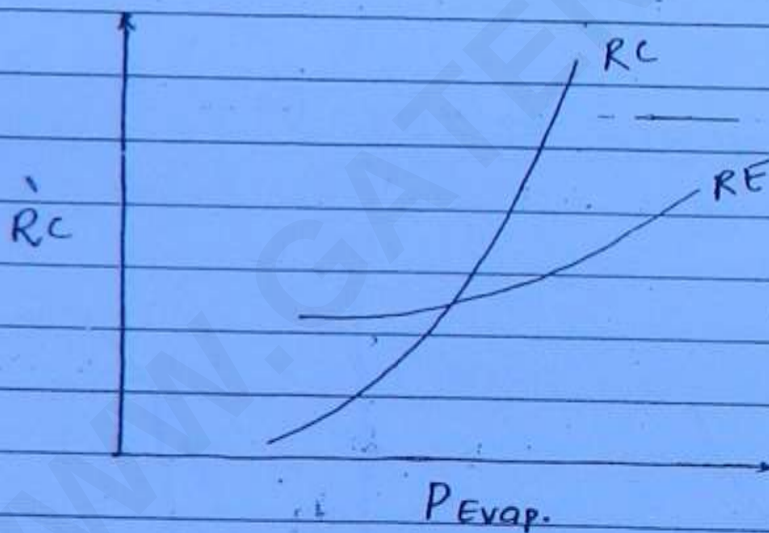
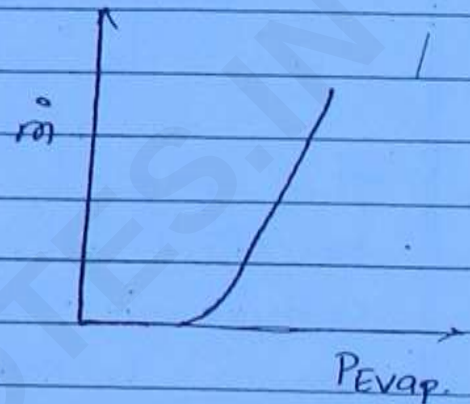
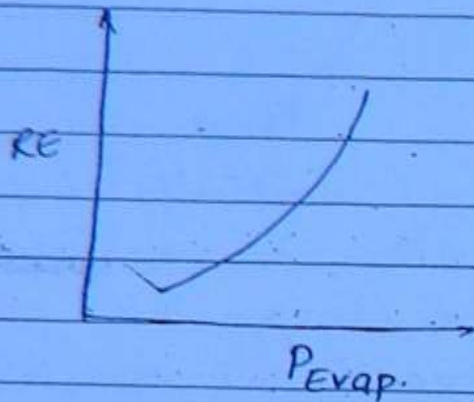
5.4 Evaporator press. v/s R.E :-



G. Evaporator press. v/s RC.

$$RC = \dot{m} \times RE.$$

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Though RE & RC increase with increase in Evaporator press. RC increases at a faster rate.

Type's of Compressor's :-

1. Open type Compressor.
2. Hermetic Compressor's.
3. Semi-Hermetic Compressor's.

(145)

In Open type compressor's the compressor runs with the help of motor by means of belt pulley system this system occupied large space & in this system huge noise is generated but the maintainance of the system is Easy.

In Hermitically sealed Compressor Compressor is mounted in motor shaft & is enclosed in a welded steel shell.

This system is more compact & less noise generated the maintainance of this system is not Easy. As motor is close to compressor Refrigerant absorb heat from motor & Reache's super heated state this result's in increase in work input therefore the energy consumption of Refrigeration system's using Hermetical seal compressor is high.

Hermitically sealed Compressor are used in domestic refrigerator & window AC

2. Condensers & Evaporator:-

Both are Heat Exchangers in Condensers refrigerant reject heat to surrounding & in Evaporator it absorbs heat from storage space.

Types of Condensers:-

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1. Air Cooled.
2. Water Cooled.
3. Evaporative Condensers

NOTE- Capacity of domestic Refrigerator is 0.1 to 0.25 TR

Air Cooled Condensers are used for small capacity plants (upto 3TR capacity).

In water cooled Condenser water is the cooling medium & it is used for upto 1000 TR capacity.

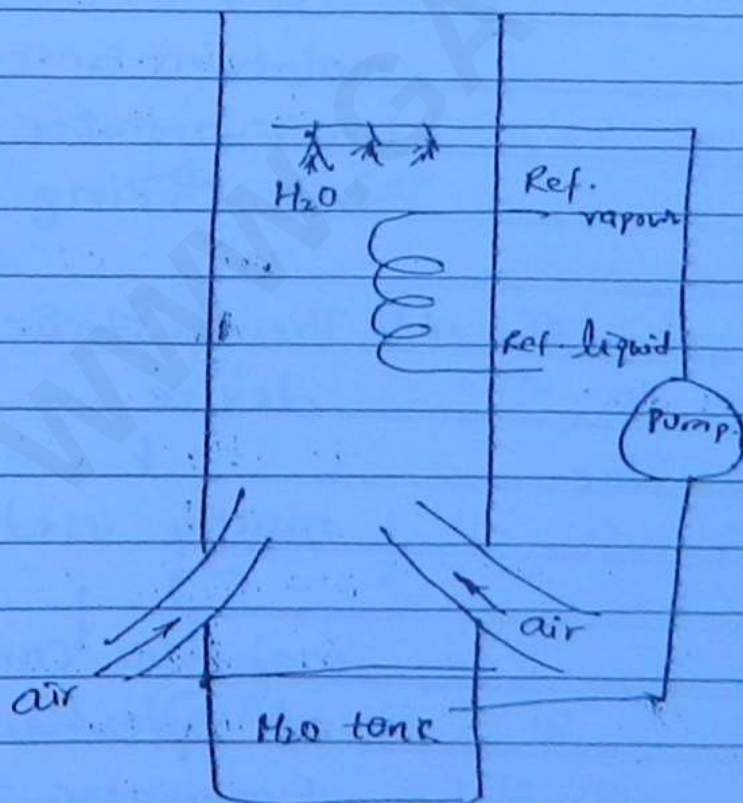
In shell & tube type Condensers refrigerant flows in shell & water flows in tube's if refrigerant flows in tube there will be a pressure drop in refrigerant & hence it is not allowed to flow in tube's

Shell & coil type condensor's are used when pure water is available because if water is not pure scale's are formed & cleaning of scale's is not easy in coil's.

Evaporative Condensor's:-

(147)

This type of condensor is used where there is large shortage of water. In this condensor Refrigerant first reject heat to water & water in turn then reject heat to air and thus same water is recirculated again and again.

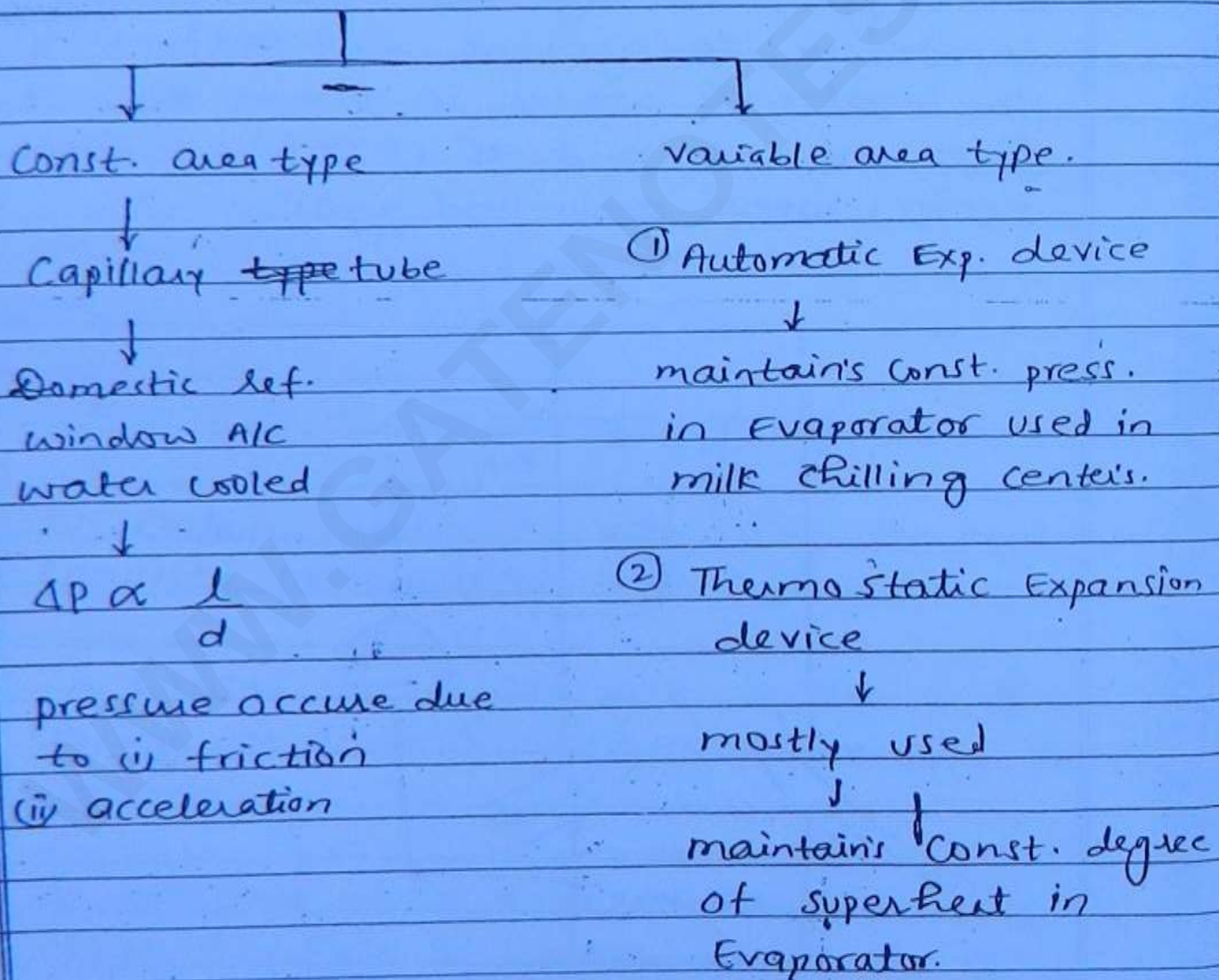


Imp. * Plate type of Evaporator are used in Domestic Refrigerators.

Imp. * Direct Contact ~~type~~ Coil type Evaporator are used in air conditioners.

3. Expansion Device's :-

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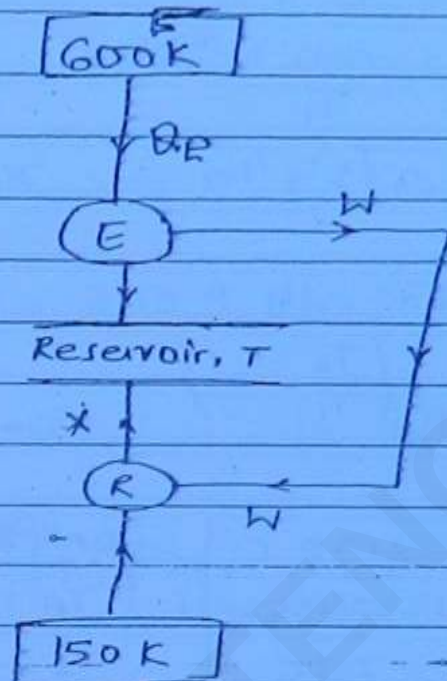


Work-Book

CHAPTER-I

1. (b)

2. (a)



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$$\eta_E = \frac{1}{COP}$$

$$\eta_E = \frac{T_R - T_L}{T_R}$$

$$\eta_E = \frac{600 - 300}{600} = 0.5 = \frac{1}{2}$$

$$\eta_E = \frac{1}{2}$$

$$(COP)_{HP} = \frac{T_R}{T_R - T_L} = \frac{300}{300 - 150} = 2$$

$$(COP)_{Ref.} = \frac{T_L}{T_R - T_L} = \frac{150}{300 - 150} = 1$$

3. (a) All though Heat pump & Refrigerator

4. (a)
$$\text{COP} = \frac{T_1}{T_R - T_1} \quad T_R = T_L \quad \& \quad T_1 = T_1$$

$$\text{COP} = \frac{T_1}{T_2 - T_1} = \frac{T_1}{T_1 \left(\frac{T_2}{T_1} - 1 \right)}$$

$$\text{COP} = \frac{1}{\left(\frac{T_2}{T_1} - 1 \right)}$$

5. (b) ice

assume $T_R = 40^\circ\text{C} = 313\text{K}$

$T_L = 0^\circ\text{C} = 273\text{K}$

$$(\text{COP})_{\text{ice}} = \frac{T_L}{T_R - T_L}$$

$$(\text{COP})_{\text{ice}} = \text{max}$$

comfort

$T_R = 313\text{K}$

$T_L = 20^\circ\text{C} = 293\text{K}$

$$(\text{COP})_{\text{comfort}} = \frac{T_L}{T_R - T_L}$$

$$\text{COP}_{\text{comfort}} = \text{less}$$

6. (c) Carnot refrigerator depends only temp. limit.

7. (d) If nothing is given then take reversible

8. (c)

9. (c)

10. (a)

11. (c) $1 \text{ TR} = 3.5 \text{ kW} = 3.5 \text{ kJ/s.}$

$t_i = 35^\circ, \quad t_f = 20^\circ\text{C}$

$C = 4.18$

$$Q = mc(\Delta t)$$

$$3.5 = m \times 4.18 (15)$$

$$m = 0.055 \text{ Kg/s.}$$

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$$m = 0.055 \times 3600 \text{ Kg/hr.}$$

$$m = 200.9 \text{ Kg/hr}$$

$$\rho_{\text{water}} = 1000 \text{ Kg/m}^3$$

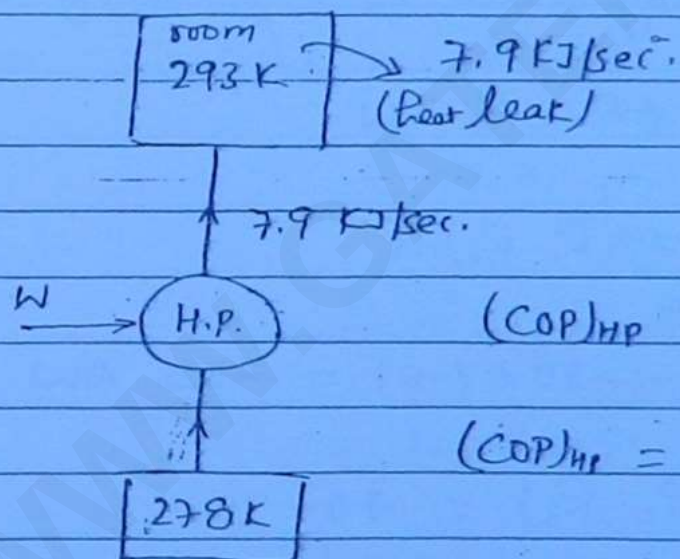
$$m = 200.9 \text{ liter/hr.}$$

12. (b)

13. (c)

14. (d)

15. (b)



$$(COP)_{HP} = \frac{T_R}{T_R - T_L} = \frac{293}{293 - 278}$$

$$(COP)_{HP} = \frac{7.9}{W}$$

$$\frac{7.9}{W} = \frac{293}{293 - 278}$$

$$W = 0.405 \text{ KJ/sec.}$$

$$\begin{aligned} &= 0.405 \text{ kW} \\ &= 405 \text{ W.} \end{aligned}$$

16. (d)

17. (d)

18. (b)

19. (c)

20. (d)

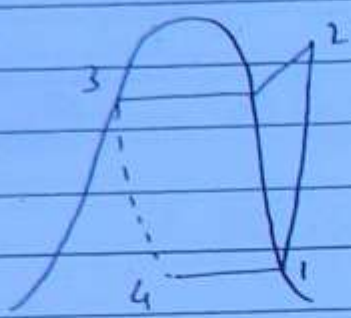
CHAPTER-2

1. (C) 2. (C) 3. (a) 4. (C)

5. (b) 6. (C) 7. (C)

152

8. (C)



$$h_1 - h_4 = h_1 - h_3 = 70$$

$$\& \quad h_2 - h_3 = 90$$

$$Q_{\text{comp}} = 5$$

$$h_1 + \frac{C_p}{2} + \cancel{Z_1 g} + Q = h_2 + \frac{C_p}{2} + \cancel{Z_2 g} + W$$

$$h_1 + Q = h_2 + W$$

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

$$W = -20 + (-5) = -25 \text{ A.W.}$$

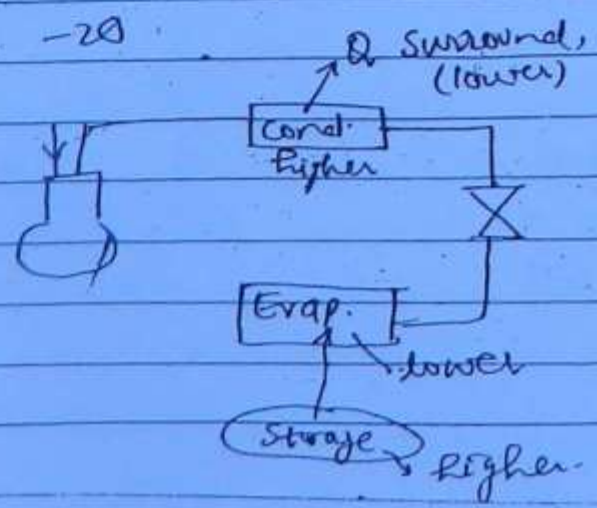
$$(h_1 - h_3) - (h_2 - h_3) = 70 - 90$$

$$h_1 - \cancel{h_3} - h_2 + \cancel{h_3} = -20$$

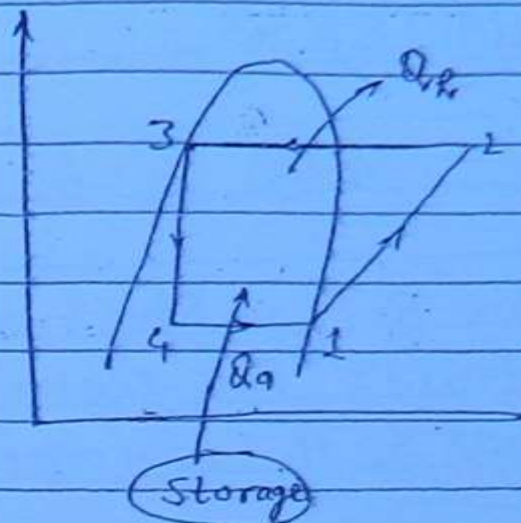
$$h_1 - h_2 = -20$$

9. (b) 10. (C)

11. (d)



11. (d)



$$h_1 = 195, \quad h_2 = 210, \quad h_3 = h_4 = 90$$

$$m = 0.5$$

- In heat pump. -

$$\begin{aligned} \text{Heat supplied} &= h_2 - h_3 \\ &= 210 - 90 \\ &= 120 \frac{\text{kJ}}{\text{kg}} \\ &= \dot{m} \times (h_2 - h_3) \end{aligned}$$

$$= 120 \times 0.5 = 60 \text{ kW Ans.}$$

12. (a) & (b) 13. (d)

CHAPTER - 3

1. (C) 2. (d) 3. (b) 4. (d) (154)

NOTE- Electrolux Refrigerator works on three fluids NH_3 (Refrigerant), water (Absorbent) & H_2 to create low NH_3 pressure's.

This system works on Dalton's law of partial pressure. In this system the total pressure is kept as 14 bar & in Evaporator Hydrogen is maintained at a pressure of 12 bar once NH_3 enters in Evaporator from condenser its pressure drops to 2 bar. i.e. in electrolux refrigerator no expansion device is used.

In electrolux refrigerator no pump is used because the fluid flows by gravity.

5. (d) 6. (a) Former $\rightarrow$ V.A. System.
latter $\rightarrow$ V.C. System

7. (d) 8. (C) 9. (b) 10. (C)

11. (d) 12. (a)