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-: HAND WRITTEN NOTES:-

OF 156

MECHANICAL ENGINEERING

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-: SUBJECT:-

REFRIGERATION

& AIR - CONDITIONING

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Refrigeration

[redacted]

Definition:-

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

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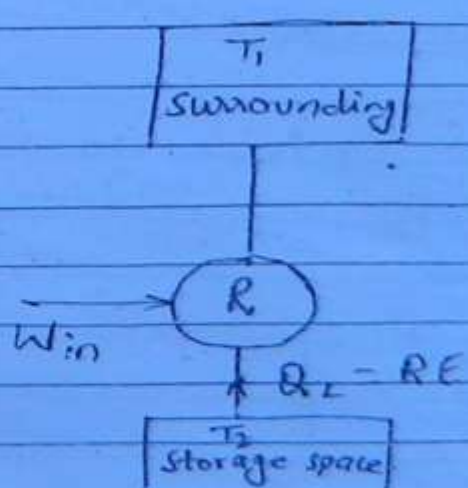
Refrigerant:-

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.



$$COP_R = \frac{Q_2}{W_{in}}$$

$$\boxed{COP_R = \frac{RE}{W_{in}}}$$

Significance of COP :- (4)

COP Represent's Running cost of the system for a given ~~R~~ 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.)

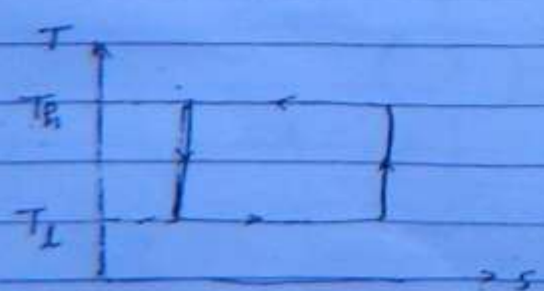
Tonne of Refrigeration Represent's heat transfer rate.

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

$$1\ TR = \frac{310 \times 1000}{3600 \times 24} \quad (L.H.)_{ice} = 310\ KJ$$

Ideal Refrigeration Cycle :-

Reversed Carnot cycle is an ideal refrigeration cycle.



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

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This is vailid for reversible cycle only.

Refrigeration capacity. (RC)

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

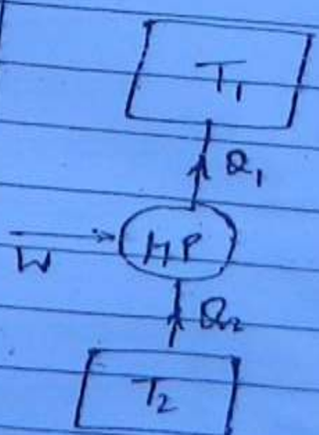
$RC = \dot{m} \times RE$

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

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

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

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



$$COP_{H.P.} = \frac{Q_1}{W}$$

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

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

Solⁿ:-

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

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

$$R.C. = 1\ TR = 3.5\ KW$$

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

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

Solⁿ:-

$$RC = 2\ KW.$$
$$T_l = 200\ K$$
$$T_R = 300\ K$$

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

& we also know that.

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

$$2 = \frac{2\,KW}{P_{in}}$$

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$$P_{in} = 1\ KW. \quad Ans.$$

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 time's the maximum COP then find the power input for a refrigeration capacity of 5KW.

Sol^n:- $T_L = -30°c = 30 + 273 = 243K.$

$T_H = 32°c = 32 + 273 = 305K.$

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

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

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

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

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

$$2.94 = \frac{5KW}{P_{in}}$$

$$P_{in} = \frac{5\ KW}{2.94} = 1.7\ KW$$

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

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. limit's what is it's COP.

Soln:- $$\frac{T_L}{T_H} = 0.8$$

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

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

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

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

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

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.

Sol^n:- $(COP)_R = \dfrac{1-0.4}{0.4} = 1.5$ Ans.

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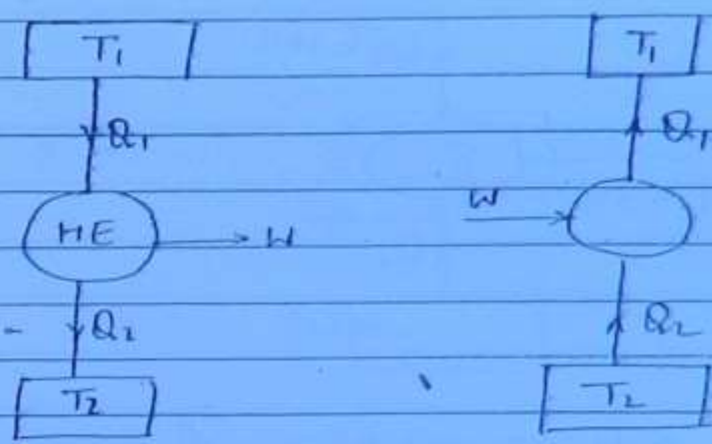
NOTE-



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

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

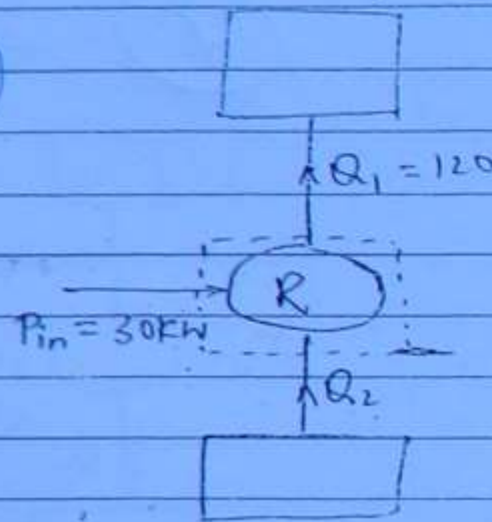
$$(COP)_{H.P.} = 1 + (COP)_{Ref.}$$

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

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

prob. A Condenser of Refrigeration system reject's 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 ~~of~~ on reversed carnot cycle consume 6KW of power for producing a refrigeration capacity of 1000KJ per min. to maintain a region at $-40°C$ then find the higher temp. in °C.

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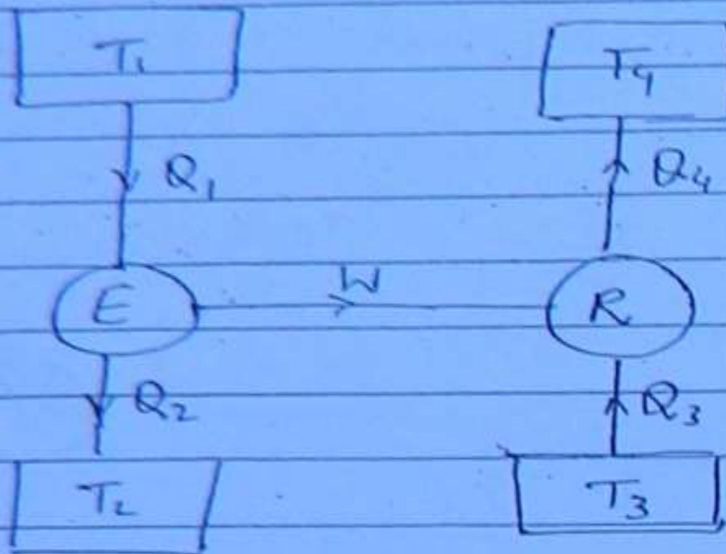

n:

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

$$W = 0.4\,Q_1$$

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

$$W + Q_3 = Q_4$$

$$Q_3 = Q_4 - W$$

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

$$Q_3 = Q_4 - Q_1 + Q_2$$

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

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

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 time's max. COP then find daily electricity consumption in KW-hr.

Soln: $RC = 8000\ KJ/day.$

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$T_l = 2°C = 2+273 = 275K.$

$T_h = 30°C = 30+273 = 303K.$

$$(COP)_{max} = \frac{T_l}{T_h - T_l}$$

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

$$= 9.82$$

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

$$= 1.473$$

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

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

$$P_{in} = 5431\ KJ/day$$

Per day Engine input = 5431 KJ

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

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= 5431 KW - sec.

But 1 hr = 3600 sec.

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

per day consumption = $5431 \text{ KW} \times \frac{1}{3600}$ 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 take's 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:

$T_l = -23°C = 23 + 273 = 250K.$

$T_R = 27°C = 27 + 273 = 300K.$

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

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

$$= 5$$

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

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

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

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

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

$27°C \longrightarrow -23°C$

$27°C \longrightarrow -3°C \rightarrow SH_1 = mC_1(\Delta t)_1$

$-3°C \longrightarrow -3°C \longrightarrow L.H. = m(L.H.)$

$-3°C \longrightarrow -23°C \longrightarrow SH_2 = mC_2(\Delta t)_2$

Total heat removed $= RC = mC_1(\Delta t_1) + m(LH) + mC_2(\Delta t_2)$

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

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

$$RC = \frac{1}{10}\left[60+230+10\right] = 30\ KW$$

$$P_{in} = \frac{30}{2.5}\ KW$$

$$\boxed{P_{in} = 12\ KW \quad Ans.}$$

VCRS

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Vapour Compression Refrigeration System (VC System):-

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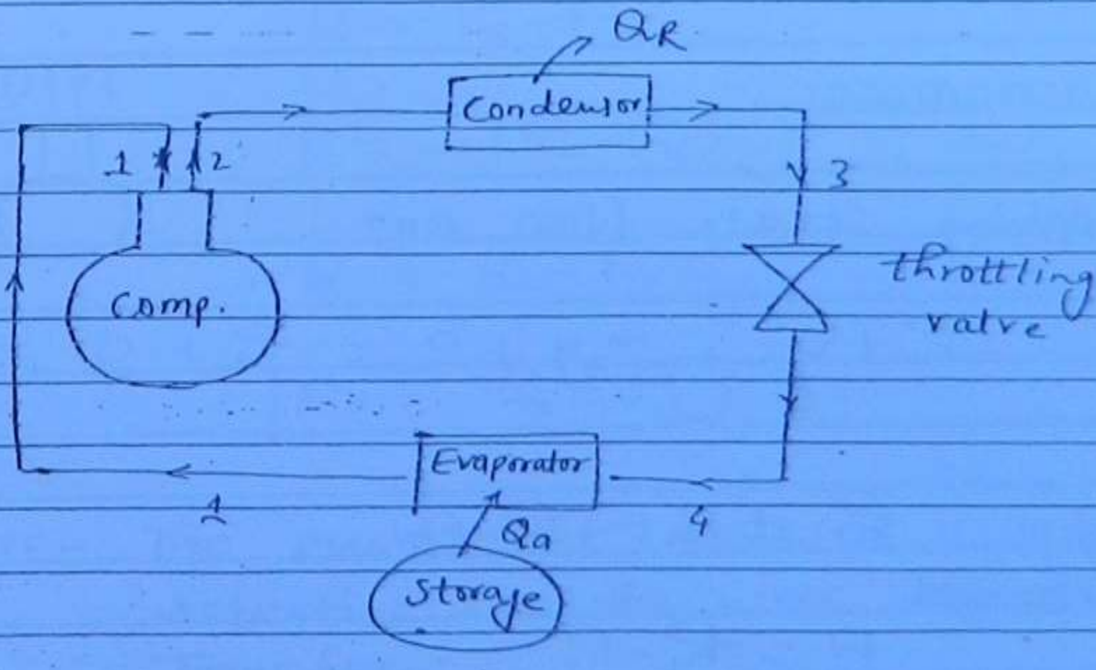
Simple V.C. Cycle or Saturated V.C. cycle:-

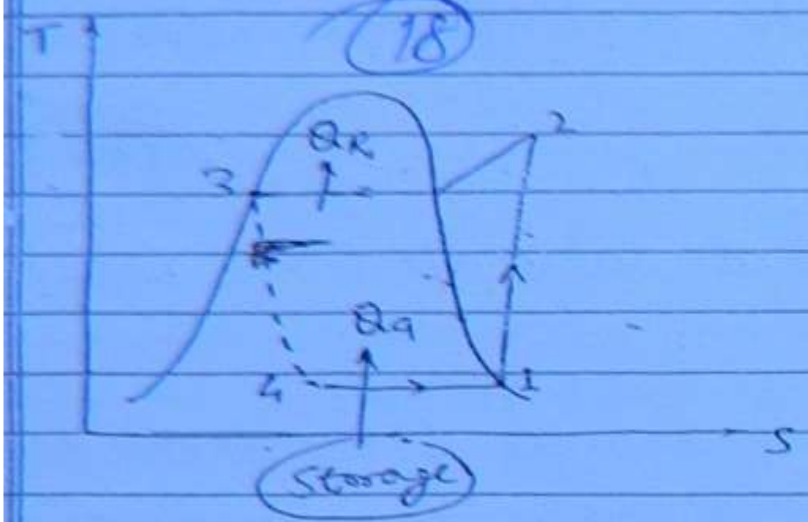
1-2 → Reversible adiabatic (Isentropic) Compressic

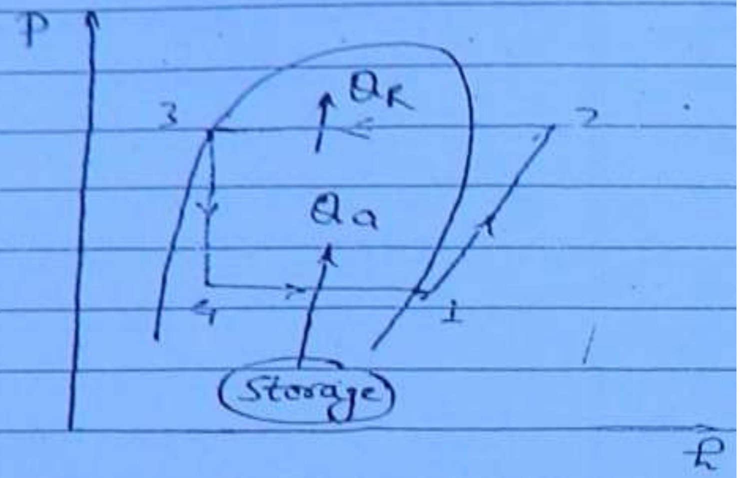
2-3 → Constant pressure heat rejection.

3-4 → throttling (Isenthalpic Expansion)

4-1 → Const. press. heat ~~rejection~~. absorption

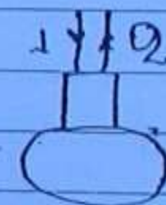






Each device is an open system & operate's ~~tp~~ under steady flow condition and hence steady flow energy eqn. can be applied.

Compressor :-



Applying steady flow eqn.

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

$$h_1 + Q = h_2 + W$$

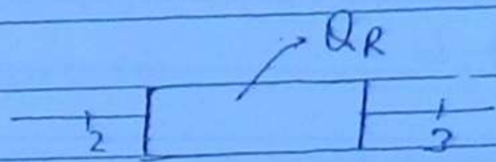
$$W = h_1 - h_2$$

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

$$-W = h_2 - h_1$$

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

(ii) Condensor :- 19



Applying steady flow eqn.

$$h_2 + \frac{C_2^2}{2} + Z_2 g + Q = h_3 + \frac{C_3^2}{2} + Z_3 g + W$$

$$h_2 + Q = h_3 + W$$

$$Q = h_3 - h_2$$

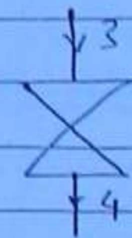
$$Q = -(h_2 - h_3)$$

$$-Q = h_2 - h_3$$

$$\boxed{Q_R = h_2 - h_3}$$

NOTE - The purpose of condeser is only heat rejection not the work ~~rejection alone~~ Transfer.

(iii) Throttling valve or Expansion device



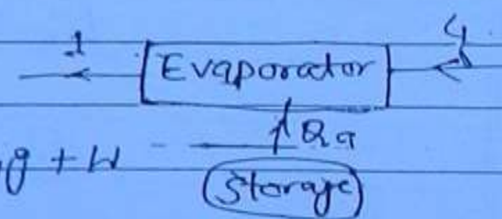
(20)

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

$$h_3 + Q = h_4 + W$$

$$\boxed{h_3 = h_4}$$

(iv) Evaporator :-



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

$$h_4 + Q = h_1 + W$$

$$Q = h_1 - h_4$$

In Evaporator only heat transfer not the work transfer.

$$\boxed{Q_a = h_1 - h_4}$$

$$\boxed{RE = h_1 - h_4}$$

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

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

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

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

(21)

☆ Volumetric efficiency of a Reciprocating Compressor :-

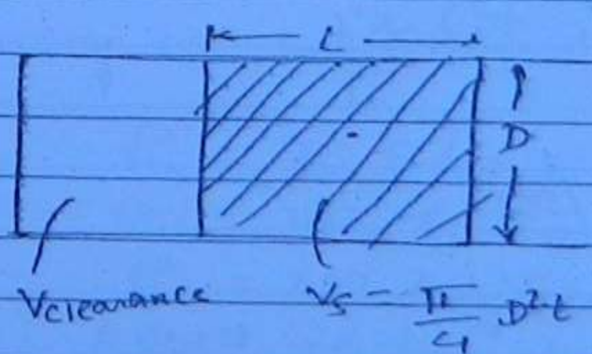
It is the breathing capacity of a compressor & it is defined as the ratio of actual volume entering the compressor to the swept volume.

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

v_1 = sp. volume at the inlet to the comp

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

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



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

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D = Bore dia or piston dia.
L = stroke length.
K = number of compressor.
N = speed in rps

NOTE - $RE = Qa = h_1 - h_4$

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

$$W_{in} = h_2 - h_1$$

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

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

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

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

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

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

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

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,
$h_1 = 180\ KJ/Kg.$
$h_2 = 210\ KJ/Kg.$
$h_3 = 80\ KJ/Kg.$

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

prob. In a 5KW cooling capacity Refrigeration sys operating on V.C. cycle refrigerant enter's evaporator with an enthalpy of 75 KJ/Kg. & leave's with an enthalpy of 183 KJ/Kg. Enthalpy of Refrigerant after compression i 210 KJ/Kg calculate -

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

Soln given,
$h_4 = 75\ KJ/Kg = h_3$
$h_1 = 183\ KJ/Kg$
$h_2 = 210\ KJ/Kg.$
$RC = 5\ KW$

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

$COP = 4$ Ans. 24

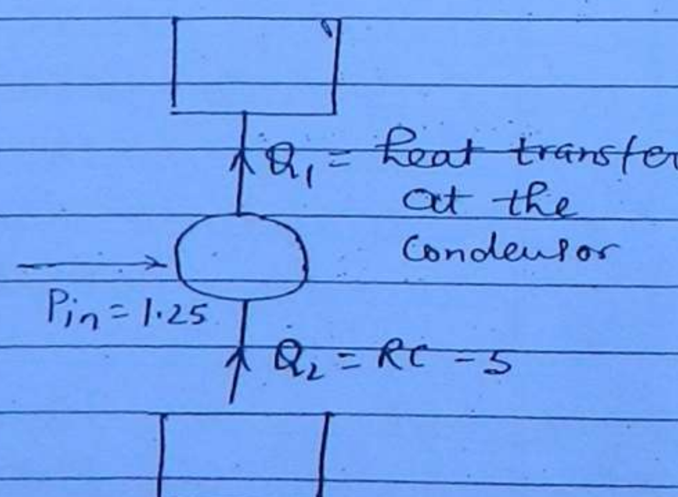
& we also know that,

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

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

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

$1.25 + 5$ = heat transfer at the condensor



NOTE- Higher temp. side condensor heat transfer.

but lower temp. side Evaporator heat transfer.

Imp. prob: In a simple V.C. cycle following's are properties of Refrigerant at various point

Compressor inlet $h_1 = 183.2$ KJ/kg, $v_1 = 0.076$

Compressor discharge $h_2 = 222.6$ KJ/kg, $v_2 = 0.0614$

Condenser Exit $h_3 = 84.9$ KJ/kg, $v_3 = 0.00083$

The swept volume for the compressor is 1.5 lit & it's 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 $= V_s = \frac{\pi}{4} D^2 L = 1.5$ liter

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

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

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

we know that volumetric efficiency.

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

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

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

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

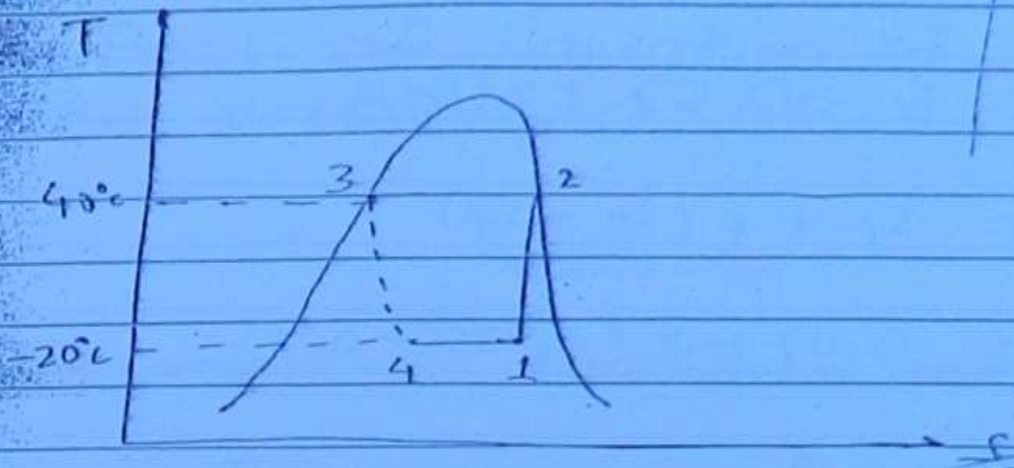
$$= 0.417(183.2 - 84.9)$$

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

2. A Refrigerator based on V.C. Cycle operating b/w temp. limit's of −20°C & 40°C the refrigerant enter's the Condensor as saturated vapour & leave's 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 (°C)	h_f (KJ/Kg)	h_g (KJ/Kg)	s_f (KJ/KgK)	s_g (KJ/KgK)
−20	20	180	0.07	0.7366
40	80	200	0.3	0.67



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

$h_2 = h_g$ at 40°C

$h_2 = 200$ KJ/Kg.

$h_3 = h_f$ at 40°C

$h_3 = 80$ KJ/Kg.

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

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

But 1-2 is isentropic process.

$s_1 = s_2$

$s_2 = s_g$ at 40°C

$s_2 = 0.67$ KJ/Kg K

$S_1 = S_2 = 0.67 \text{ kJ/kg K}$

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

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

$x = 0.9$

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

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

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

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

$= 0.025(164 - 80)$

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

☆ Effect of variation of propertie's on the performance of V.C. cycle

Case - I.

• Decrease in Evaporator pressure :-

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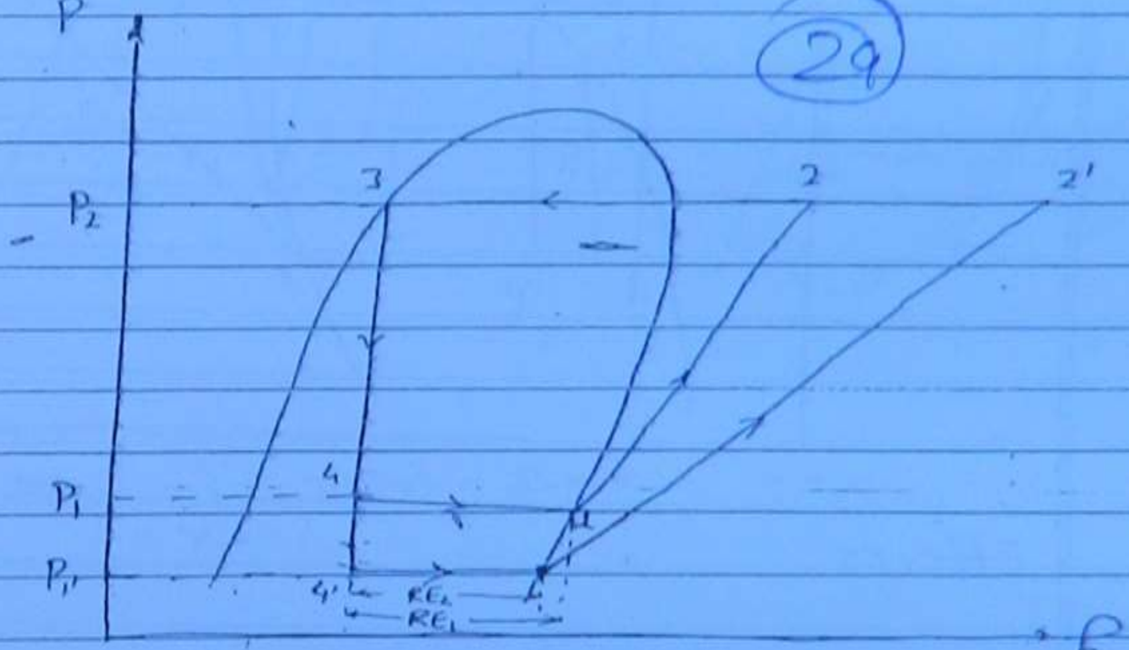

Effect's :-

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

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 - II.

- Increase in Condensor pressure :-

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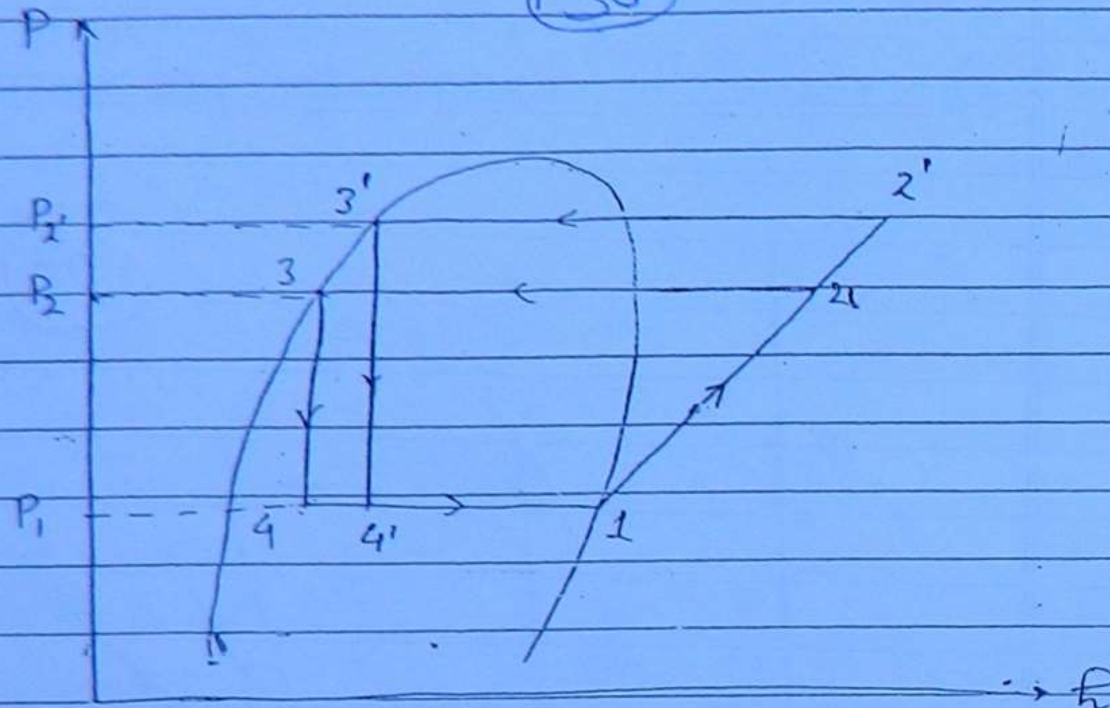

Effect's.

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.

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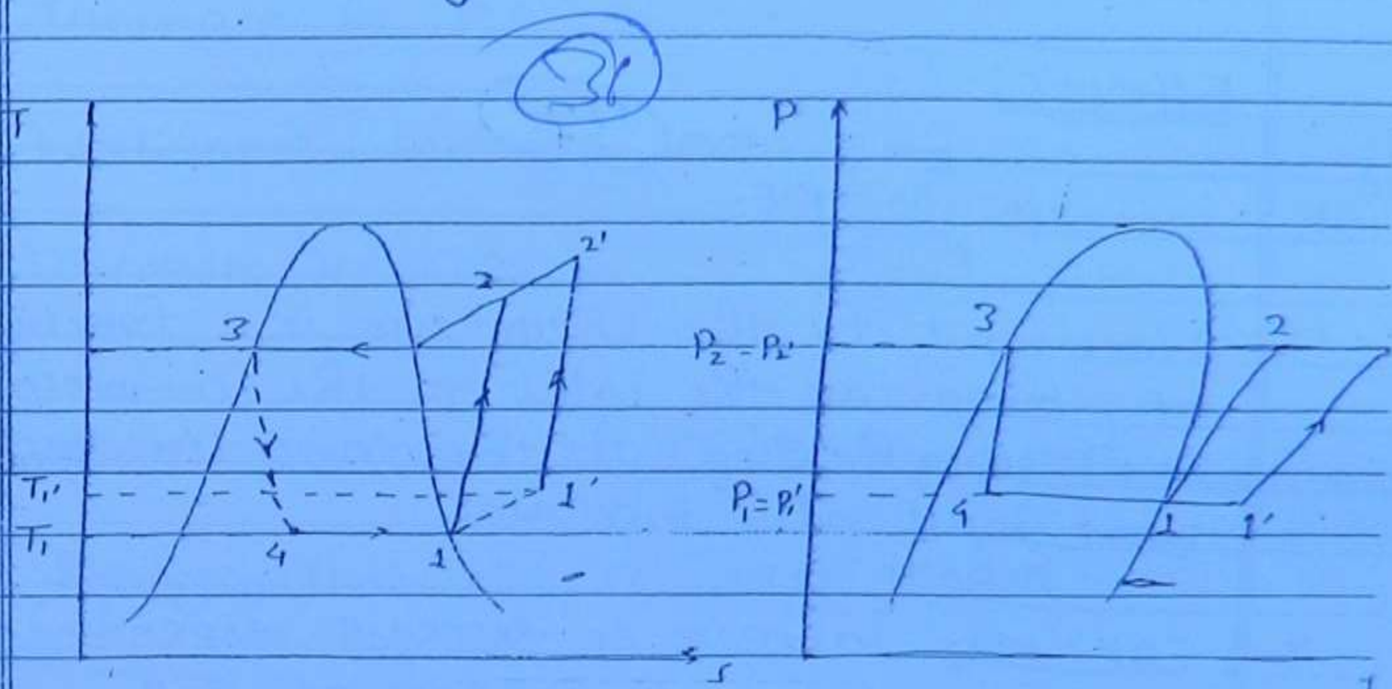
Case - III.

- **Superheating :-**



$$-W = \frac{\gamma}{\gamma-1}(P_1V_1 - P_2V_2)$$

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

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

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

$$W = f(T_1)$$

$W_{12} = f(T_1)$ & $W_{1'2} = f(T_1')$.

$T_{1'} > T_1$

Effect's:-

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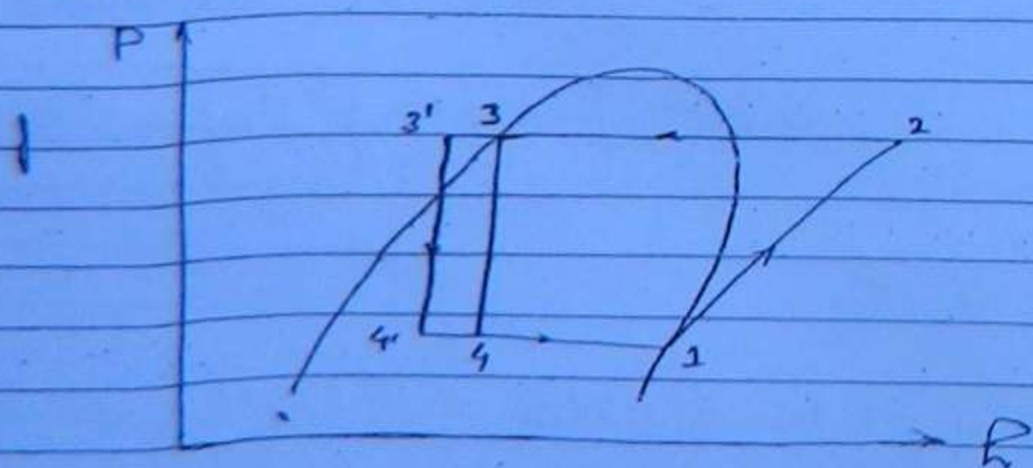
1. increase in R.E.

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 Result's in a increase in COP where as in case ~~of~~ NH_3 superheating result's in decrease in COP.

4. 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

$C_m H_n F_p Cl_q$

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

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

(i) CCl_2F_2

$C_m H_n F_p Cl_q$

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

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

$\boxed{R-12}$

(ii) If $R-11$ 34

$R-011$

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

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

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

$p=1$

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

$1+q = 4$

$q = 3$

then $C_mH_nF_pCl_q$

$\boxed{R-11 \rightarrow CFCl_3}$

(iii) $R-22$

$R-022$

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

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

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

$P = 2$

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

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

$q = 4-3$

$q = 1$

then $C_m H_n F_p Cl_q$

$\boxed{R\text{-}22 \rightarrow CHF_2Cl}$

(iv) R-113

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

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

$P = 3$

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

$q = 3$

$C_m H_n F_p Cl_q$

$\boxed{C_2F_3Cl_3 \longrightarrow R\text{-}113}$

Tri-cloro- Tri-floro ~~Ethe~~ Ethane.

(v) R - 134

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

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

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

$p = 4$

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

$2 + 4 + q = 2(2) + 2$

$q = 0.$

$C_m H_n F_p Cl_q$.

$\boxed{C_2H_2F_4 \rightarrow R-134}$ Tetra floro ethane

Case-II

- Unsaturated hydrocarbon refrigerant's

$C_m H_n F_p Cl_q$.

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

& $n + p + q = 2m$

(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

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Case - III.

÷ <u>In-organic Refrigerant</u> :-

eg. 700 + moleculer weight

NH_3 - 17 → R-700+17 ⇒ R-717

H_2O - 18 → R-718

Air - 29 → R-729

SO_2 - 64 → R-764

CO_2 - 44 → R-744

Air → N_2 - 77% 28
O_2 - 23% 32

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

t (°c)	h_f (kJ/kg)	LH (kJ/kg)	S_f (kJ/kg-K)
25	298.9	1166.94	1.1242
-10	135.37	1297.68	0.5443

Ans- 6.89

soln:

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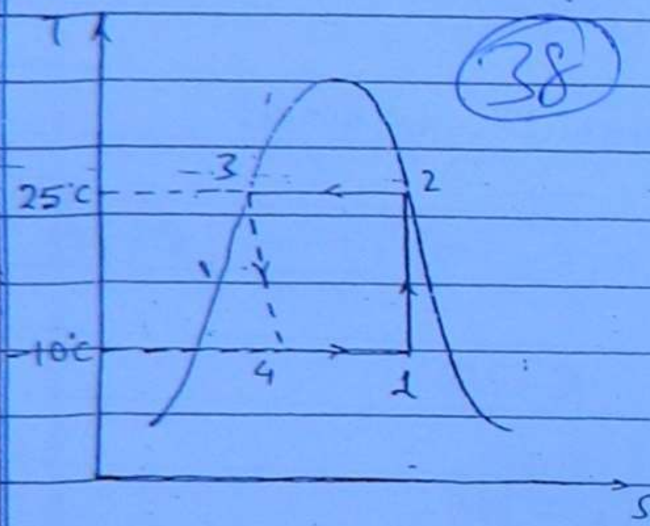

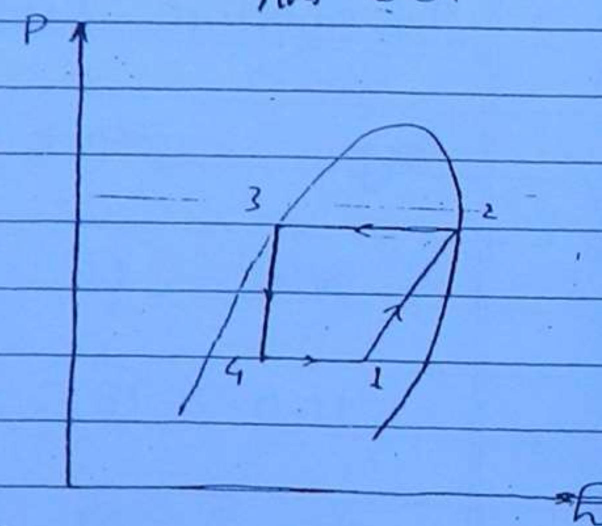

$$COP = \frac{h_1 - h_3}{h_2 - h_1}$$

~~$h_2 = h_g = LH - h_f = 1166.94 - 298.9 = 868.04$ kJ/kg~~

~~$h_3 = h_f = 298.9$~~

~~$h_1 = h_f + x(LH)$~~

$s_r = s_2 = 1.1242 + 1166.94 = 1168.06 \; kJ/kgK$

$s_1 = s_f + x l$

$h_2 = h_{f_2} + LH = 298.9 + 1166.94 = 1465.84 \; kJ$,

$h_3 = h_{f_3} = 298.9 \; kJ/kg$.

$h_1 = h_{f_1} + x_1 (LH) = 135.37 + x_1 (1297.68)$

$$s_2 = s_g = s_{f_2} + \frac{LH}{T_{sat}} = 1.1242 + \frac{1166.94}{298}$$

$s_2 = 5.04 \; kJ/kgK$.

$s_1 = s_2 = 5.04 \; kJ/kgK$ then $h_1 = 1317.55$

$$s_1 = s_{f_1} + x \frac{LH}{T_{sat}}$$

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

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

$\boxed{x = 0.911}$

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(°C)	P(bar)	h_f (KJ/Kg)	h_g (KJ/Kg)	S_f (KJ/KgK)	S_g (KJ/KgK)
48	11.64	82.83	205.83	0.2973	0.682
-20	1.51	17.82	178.74	0.0731	0.7087

Sol^n:

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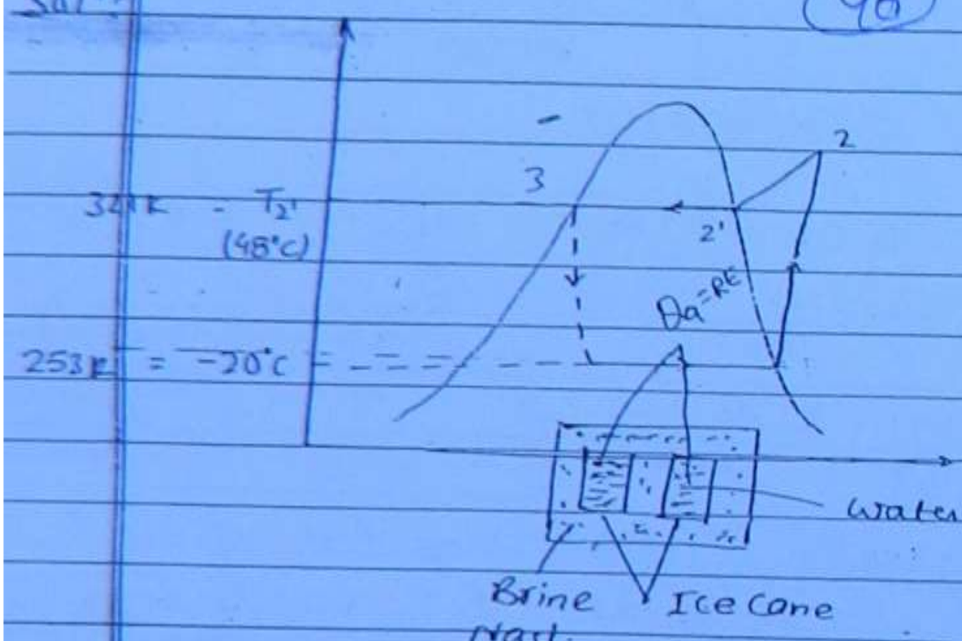

$$P = \dot{m}(h_2 - h_1)$$

$$h_1 = h_g \text{ at } -20°C$$

$$h_1 = 178.74 \text{ KJ/Kg}$$

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

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

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Since the 1-2 is isentropic process.

then. $S_1 = S_2 = S_g$ at $-20°C = 0.7087$ kJ/kgK.

$$S_2 = S_{2'} + C_{pv} \ln \frac{T_2}{T_{2'}}$$

(4)

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

$$T_2 = 331.6K.$$

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

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

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

S.H → $35°C\ H_2O$ —— $0°C\ H_2O \rightarrow mC_w(35-0)$

L.H → H_2O at 0°C —— ice at 0°C → mLH

ice at 0°C —— ice at −8°C → $mC_{ice}[0-(-8)]$

$= mC_{ice} \times 8$

Heat removed $= mC_w \times 35 + mLH + mC_{ice} \times 8$

$$= m\left[C_w \times 35 + LH + C_{ice} \times 8\right]$$

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

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Heat removed = 49.91 KW

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

$\therefore$ 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 6°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.

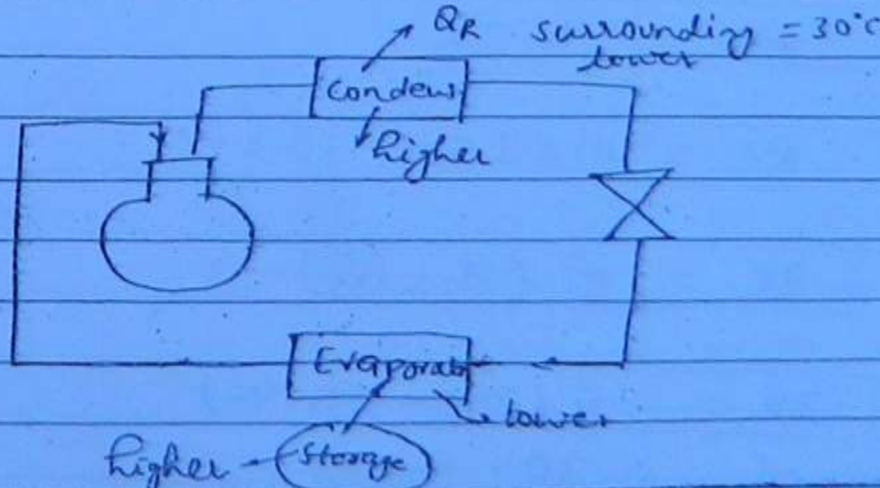
Imp.
(iv) Cylinder dimension's

Assume volumetric efficiency as 90%. &
Compressor's speed is equal to 300 r.p.m.
Take stroke equal to 1.2 time's Bore.

t(°C)	P(bar)	v_g (m^3/Kg)	h_f (KJ/Kg)	h_g (KJ/Kg)	s_f (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(°C)	s_g (KJ/Kg)	$C_{p\,vapour}$ (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's reject heat to surrounding and hence it's temp. must be greater than that of surroundings therefore temp. of refrigerant in condensor is $30+6 = 36°C$.

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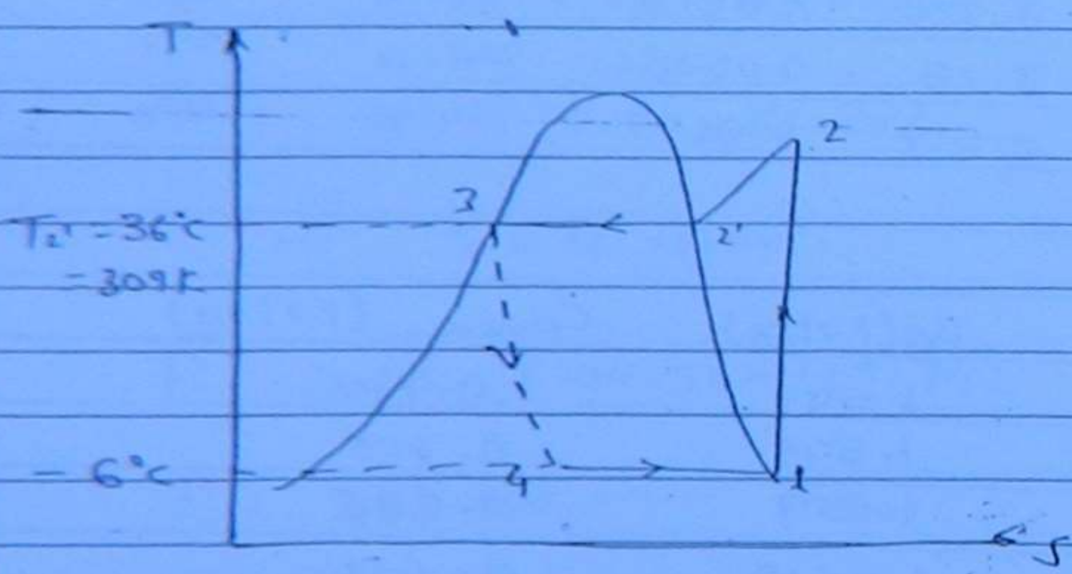
In Evaporator refrigerant absorb heat from storage place and hence it's temp must be lower than that of storage place temp. i.e. temp. of refrigerant in evaporator is $0-6 = -6°C$.



(i) $RC = 10\ TR$

$RC = 10 \times 3.5 = 35\ kW$

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

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

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

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

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

The process 1-2 is isentropic process.

$s_1 = s_2 = s_g$ at $-6°C = 1.5618$ kJ/kgK.

(45)

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

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

$T_2 = 315.2$ K.

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

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

$h_2 = 372.21$ kJ/kg.

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

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

(iii) $COP = \frac{h_1 - h_3}{h_2 - h_1} = \frac{RC}{P_{in}} = \frac{35}{6.75} = 5.185$ Ans

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

$$0.9 = \frac{\dot{m}\, v_1}{\frac{\pi}{4} D^2 L N}$$

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$$\frac{\pi}{4} D^2 L N = \frac{\dot{m}\, v_1}{0.9}$$

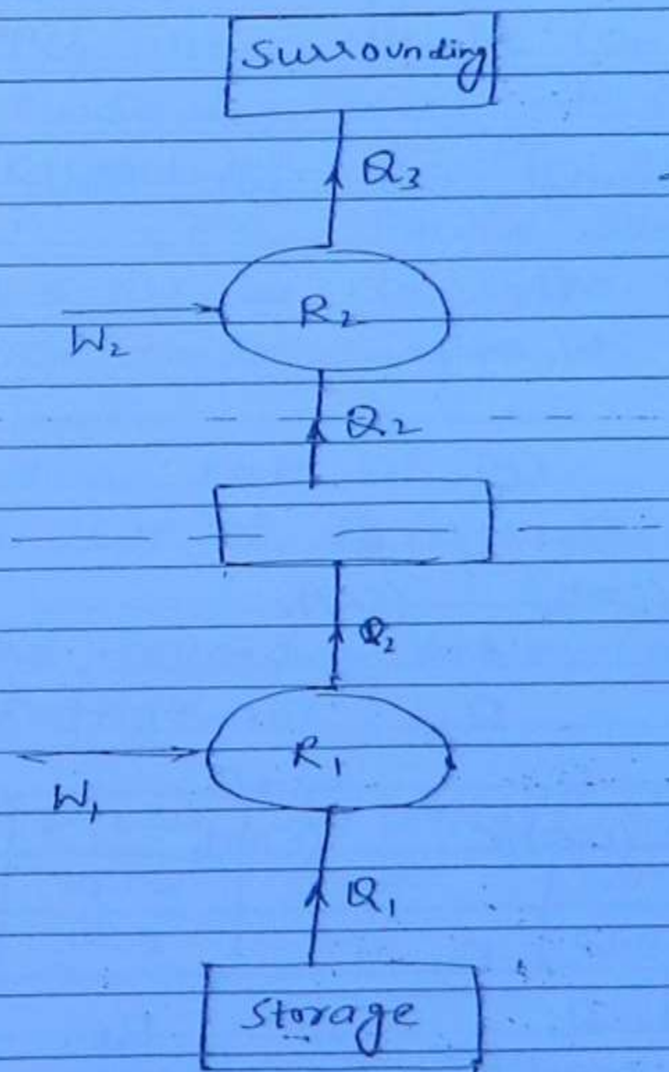
$$\frac{\pi}{4} D^2 (1.2D) \times \frac{300}{60\ sec.} = \frac{0.304 \times 0.0677}{0.9}$$

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

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

CASCADE REFRIGERATION

If very low temp. are desired then the corresponding evaporator pressure is also very low & this result's 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.



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COP of overall plant:

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

COP of 1^{st} system.

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

COP of 2^{nd} system. 98

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

Then from overall COP., we find.

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

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

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

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

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

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

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

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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 $-7°C$. Power supplied to NH_3 compressor is 96.5 KW. CO_2 cycle & NH_3 cycle are assumed to be simple V.C. cycle calculate –

(i) mass flow rate of CO_2.

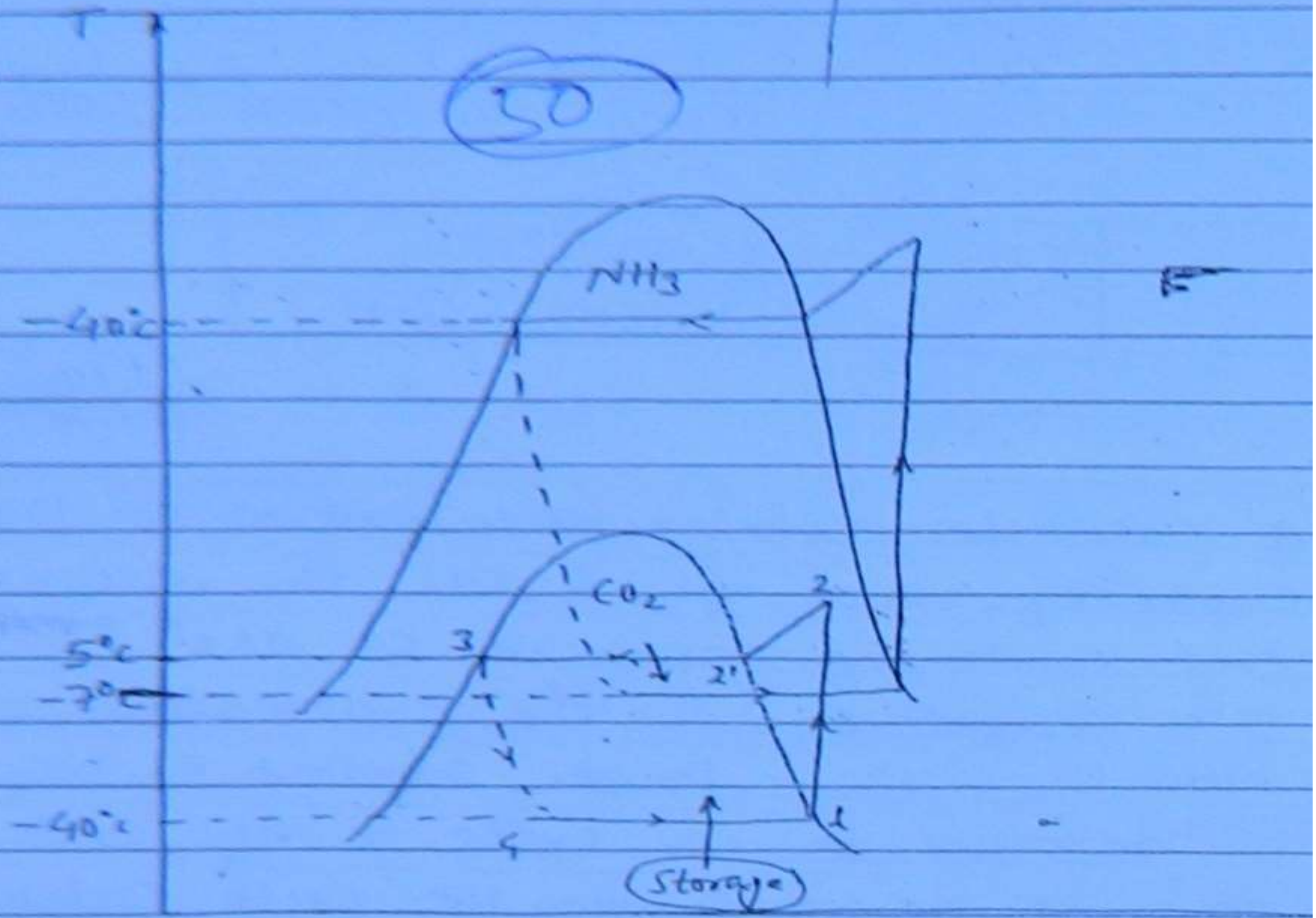
(ii) overall COP of Refrigeration system.

Use the following table for the properties of CO_2 Refrigerant.

$t(°C)$	P (bar)	h_f (KJ/Kg)	h_g (KJ/kg)	s_f (KJ/kgK)	s_g (KJ/kg
−40	10.05	332.7	652.8	3.8531	5.2262
5	39.7	431	649.8	4.2231	5.0037

$C_{p\,vapour} = 0.85$ KJ/Kg-K.

(50)



$R\dot{C} = 100\,TR$

$1\,TR = 3.5\,KW$

$RC = 350\,KW$

$RC = \dot{m}_{CO_2}(h_1 - h_3)$

$350 = \dot{m}_{CO_2}(652.8 - 431)$

$\dot{m}_{CO_2} = 1.578\ kg/s.$ Ans.

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

$RC = 350\ KW$

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

5)

$P_{NH_3} = 96.5\ KW$

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

$h_1 = 652.8\ KJ/Kg.$

$\dot{m}_{CO_2} = 1.57\ Kg/S.$

We know that 1-2 is isentropic process.

$S_1 = S_2 = S_g \text{ at } -40°C = 5.2262\ KJ/Kg$

But $S_2 = S_{2'} + C_{pv} \ln \frac{T_2}{T_{2'}}$

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

$T_2 = 361.18\ K.$

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

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

$h_2 = 720.5\ KJ/Kg.$

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

$= 1.57\,(720.5 - 652.8)$

$= 106.3$ KW

(52)

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

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

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

* Advantage's of dry compression over wet compression or ~~Thi~~ dis-advantage's of wet compression :-

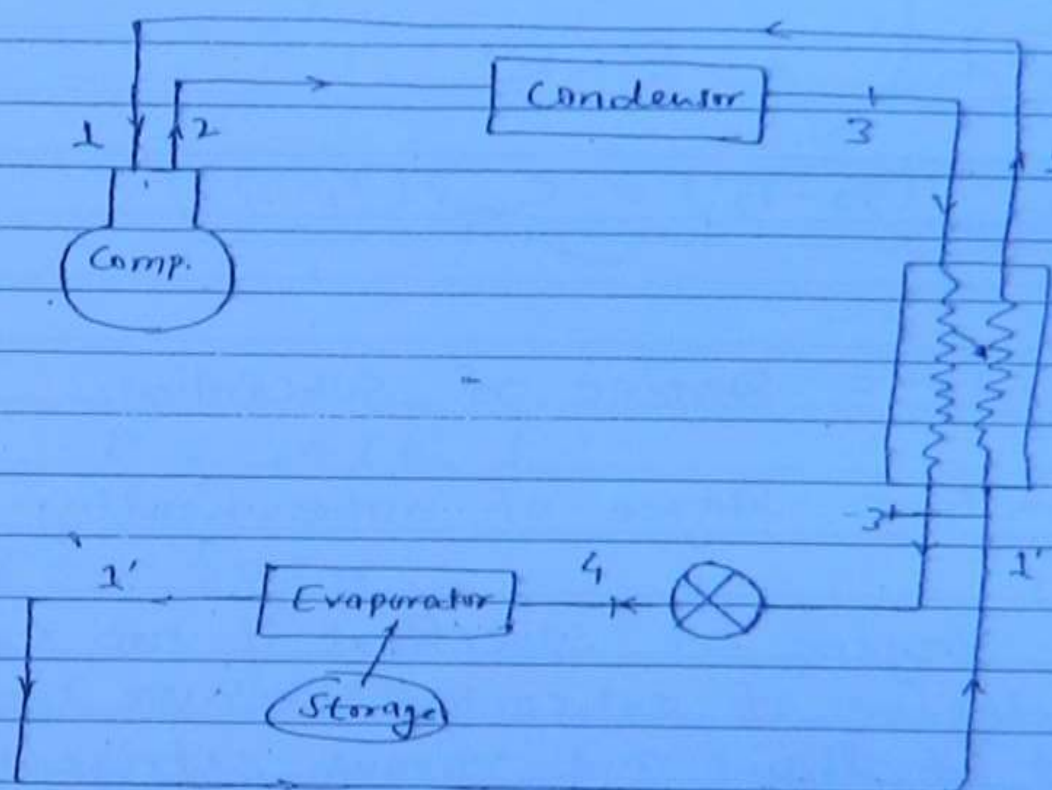
1. Wet compression Represent's incomplete vaporization of Refrigerant and thus there is reduction in ~~R~~ Refrigeration effect

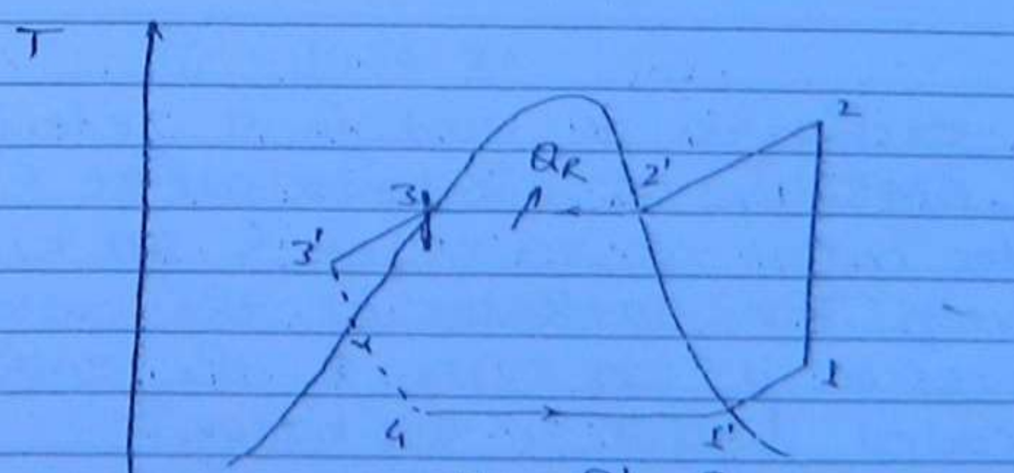
2. The liquid refrigerant may wash away lubricating oil thus increasing wear & tear.

3. Liquid refrigerant may damage compressor valve's.

NOTE- Slight superheat is desirable, excess superheat result's in increase in work input to the compressor.

** Used of Heat Exchanger in V.C. Cycle :- ***





$RE = h_1' - h_4$

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

Heat exchanger concept.

Heat loss = Heat gain. (54)

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

$$\boxed{C_{liquid} (T_3 - T_3') = C_{vap.} (T_1 - T_1')}$$

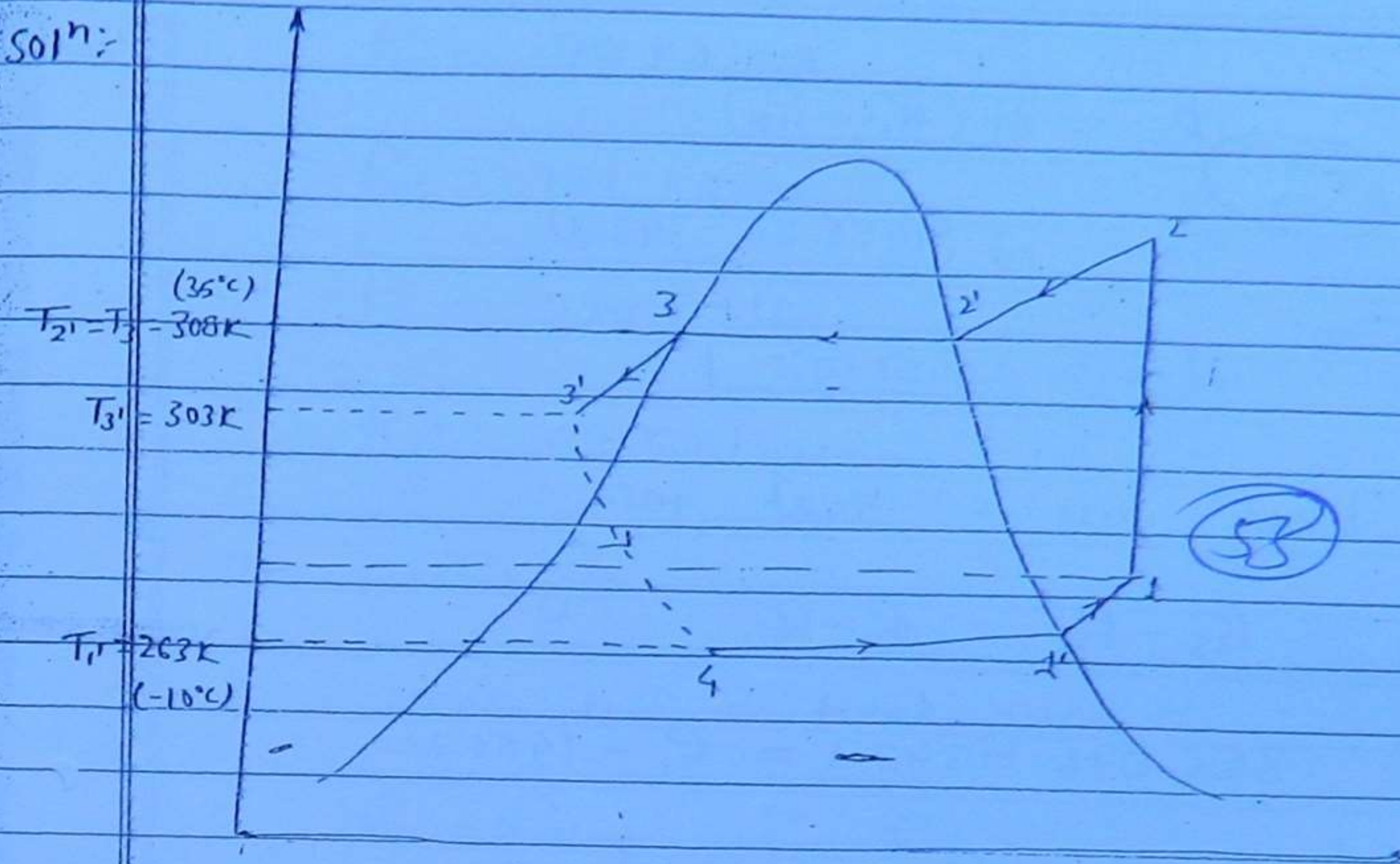
$T_3 - T_{3'}$ = Degree of subcooling.

$T_1 - T_{1'}$ = 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.

Q6- A heat exchanger is used in a refrigeration cycle. Enthalpy at condensor 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.

Sol^n:-



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

$$RC = 50\ kW$$

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

$$RC = \dot{m}(h_{1'} - h_{3'})$$

$$h_{1'} = 1450.22$$

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

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

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

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$RC = \dot{m}(h_{1'} - h_{3'})$

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

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

Heat loss = Heat gain

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

$366.072 - 343.3 = h_1 - 1450.22$

$\boxed{h_1 = 1473 \text{ KJ/kg.}}$

$S_1 = S_{1'} + C_p \ln \frac{T_1}{T_{1'}}$

$h_1 = h_{1'} + C_{vap.}(T_1 - T_{1'})$

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

$\boxed{T_1 = 272.14 K}$

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

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

But 1-2 process is isentropic process.

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

Sol^n: $h_3 = 78$ kJ/kg

$h_{1'} = 182$ kJ/kg.

$h_2 = 230$ kJ/kg.

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$h_{3'} = 68$ kJ/kg.

In Exchanger,

Heat loss = Heat gain.

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

$78 - 68 = h_1 - 182$

$h_1 = 192$ kJ/kg

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

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

$\boxed{COP = 3}$ Ans.

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(58)

~~Sold~~
Imp.

Prob A food storage refrigeration system require's a refrigeration capacity of 50 KW It work's b/w a condensor temp. of 35°C & evaporator temp. of -10°C. The refrigerant is Amonia and it is subcooled by 5°C before entering expansion valve. Refrigerant leave's evaporator as saturated vapour and then it enter's in heat exchanger The refrigerant leave's condensor as saturated liquid assuming a single cylinder operating at 1000 r.p.m. with stroke equal to 1.2 time's Bore Determine power input to the compressor in KW & cylinder dimenssion. Take volumetric efficiency as 100%.

t (°C)	P (bar)	h_f (KJ/Kg)	h_g (KJ/Kg)	s_f (KJ/KgK)
-10	2.9157	154.056	1450.22	0.82965
35	13.522	366.072	1488.57	1.56605

s_g (KJ/Kg)	Sp. Vol (m³/Kg)		Sp. heat (KJ/KgK)	
	Liquid	Vapour	liquid	Vapour
~~5.9955~~ 5.755	–	0.41747	–	2.492
5.20086	–	0.095629	4.556	2.903

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$$S_2 = S_{2'} + C_{vap.} \ln \frac{T_2}{T_{2'}}$$

~~5.84 = 2.903 +~~

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

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

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

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

$$\boxed{h_2 = 1708.8 \text{ KJ/Kg}}$$

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

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

$$\boxed{P_{in} = 10.64 \text{ KW}}$$

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

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

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

$L = 1.2 D$

$N = 1000$ r.p.m

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

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

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

(60)

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

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

$$v_1 = 0.43 \ m^3/kg$$

then,

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

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

$$L = 1.2 \times 10.7 = 12.8 \ cm.$$

Gas Refrigeration Cycle

This cycle work's 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 it's low weight per tonne of Refrigeration.

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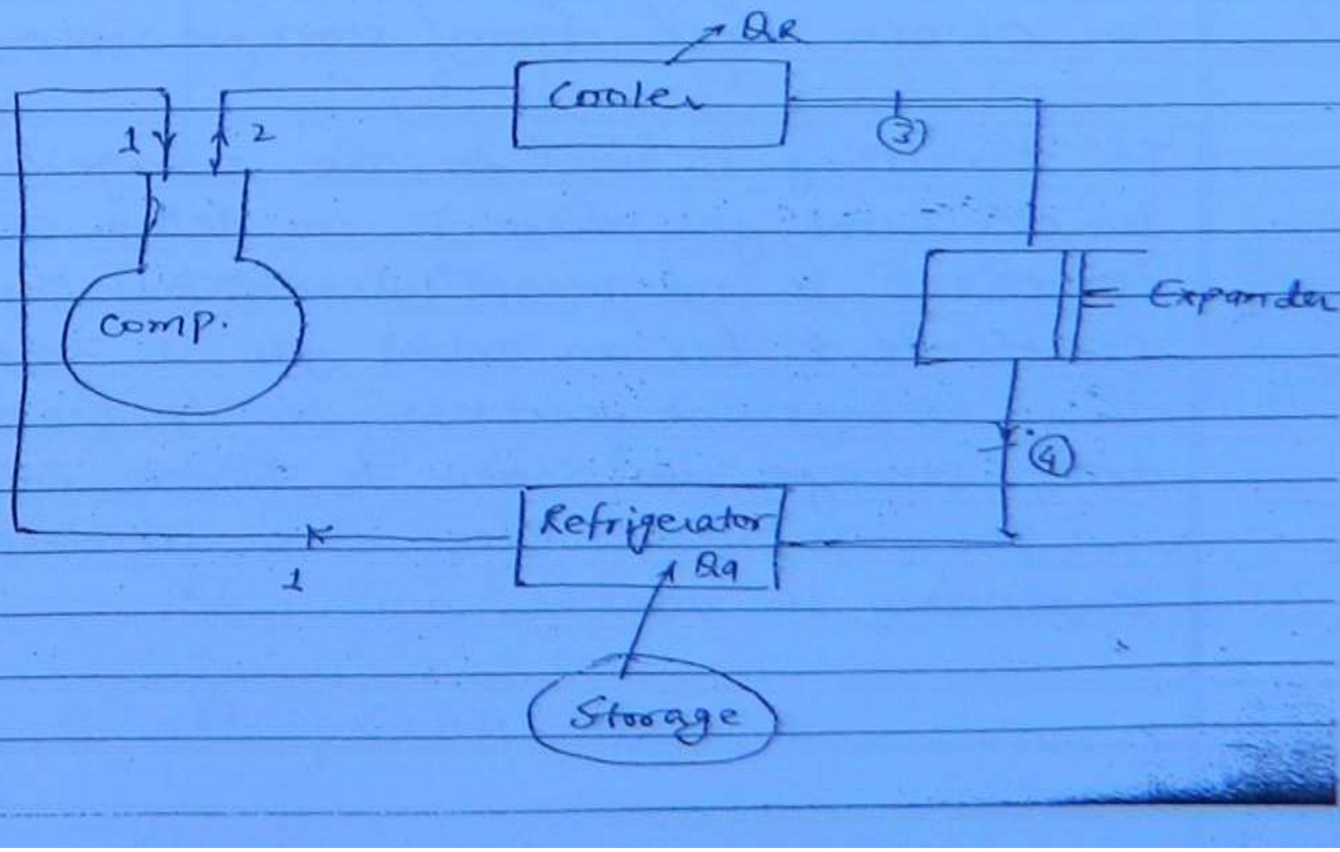
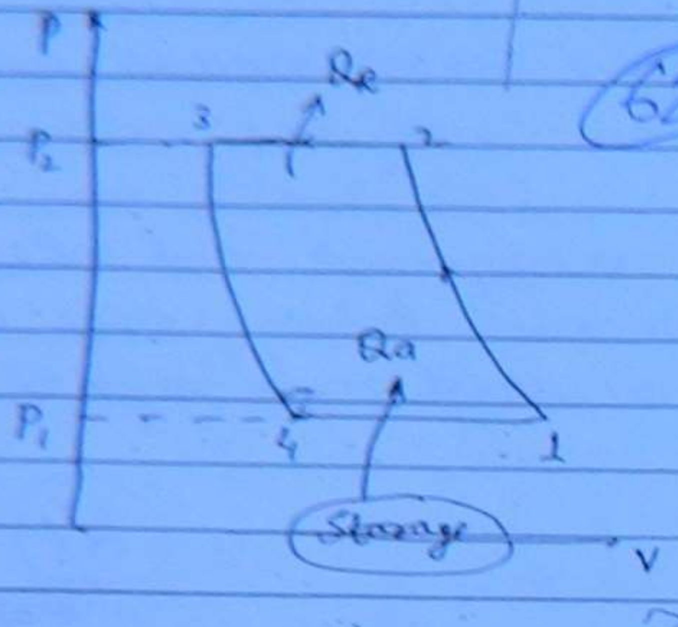
1-2 → Reversible adiabatic compression (Isentropic).

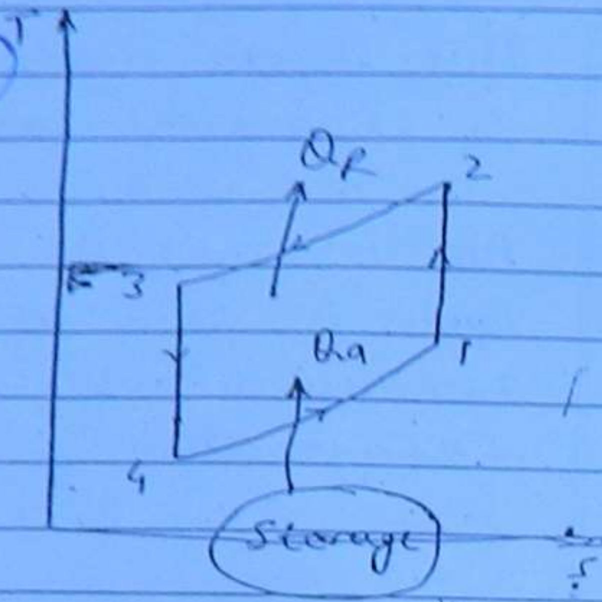
2-3 → Constant pressure heat rejection.

3-4 → Isentropic (Rev. adiabatic) Expansion.

4-1 → ~~Comp.~~ Constant press. heat absorption.







Reason's for using isentropic Expansion istead of trottling 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 ~~pta~~ space & hence R.E. is not obtained there fore isentropic expansion is used.

I.V.
Imp.

Reason's for using throttling in V.C. cycle:-

Though in isentropic expansion some positive work is obtained this work is less becaus ~~as~~ liquid is expanding The specific volume of liquid is less & for this small work output the cost of isentropic expander does not justify it's use. ~~reference~~

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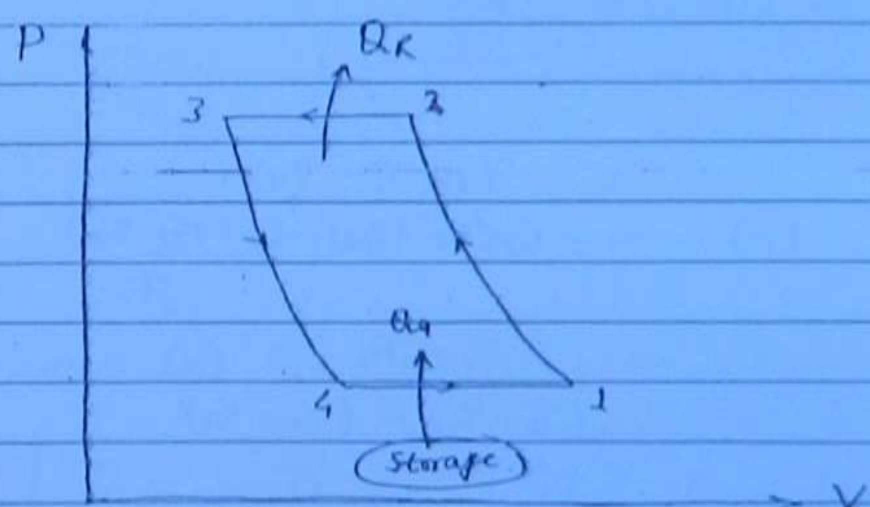
Imp.

COP of Gas Refrigeration Cycle:-



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

$$RE = h_1 - h_4$$

$$RE = C_p T_1 - C_p T_4$$

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

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

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

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& $W_{in} = W_C + W_E$

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

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

then. $COP = \dfrac{RE}{W_{in}}$

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

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

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

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

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

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

1-2 Reversible adiabatic process (Isentropic)

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

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

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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.

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

***

This eqn. is vailid only when compression & expansion both ~~rever~~ adiabatic.

Now from eqn. (I), we get.

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

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$$COP = \frac{1}{\frac{T_2}{T_1} - 1} \qquad \left[\because \frac{T_4}{T_1} = \frac{T_3}{T_2}\right]$$

objective point of view

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

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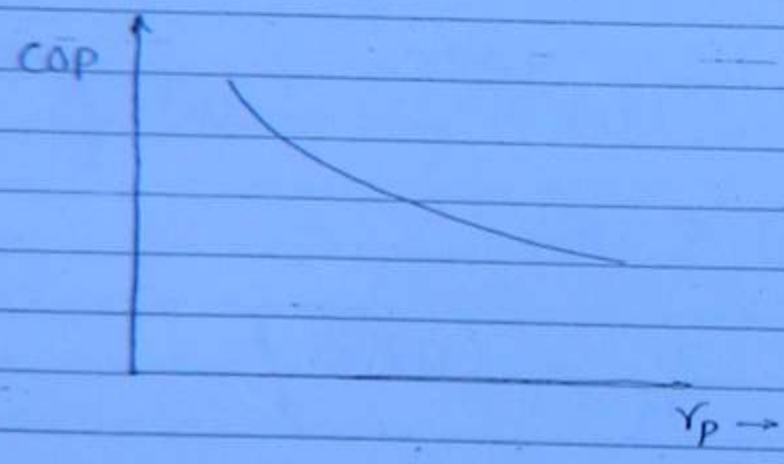
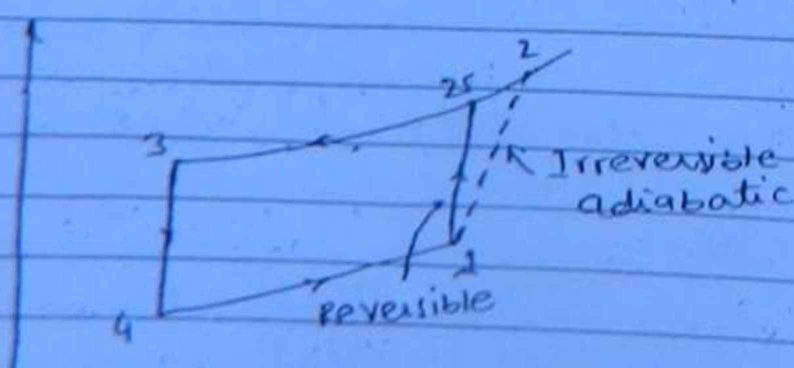
It is vailid only when the compression & expansion proces are Reversible adiabatic process.

COP of a gas Refrigeration cycle depend's on pressure ratio only



Isentropic Efficiency of A Compressor:



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

$$(ds)_{Universe} > 0. \qquad \text{(Irrev. Adiabatic)}$$

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

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

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

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

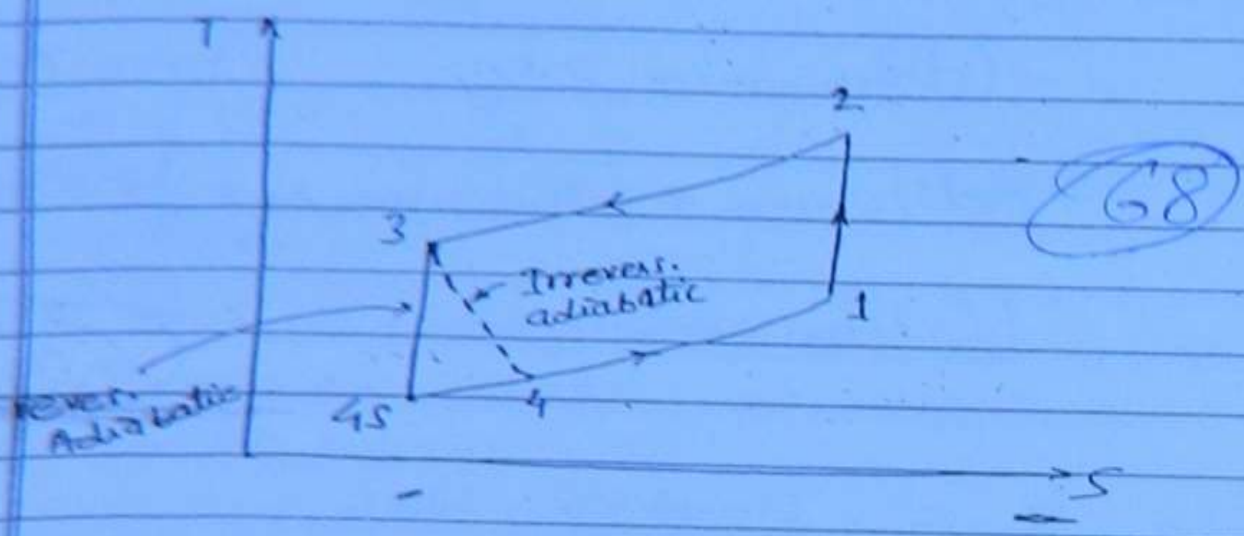
$$\boxed{\eta_c = \frac{T_{2s} - T_1}{T_2 - T_1}}$$

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

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

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

* Isentropic efficiency of Expander or Turbine –



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

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

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

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

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

NOTE. – $PV^\gamma = C$ is vailid only for the reversible adiabatic process.

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

Sol^n: $T_3 = 300K$ & $T_4 = 180K$.

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

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

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

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

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

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 $C_p = 1. KJ/kgK$.

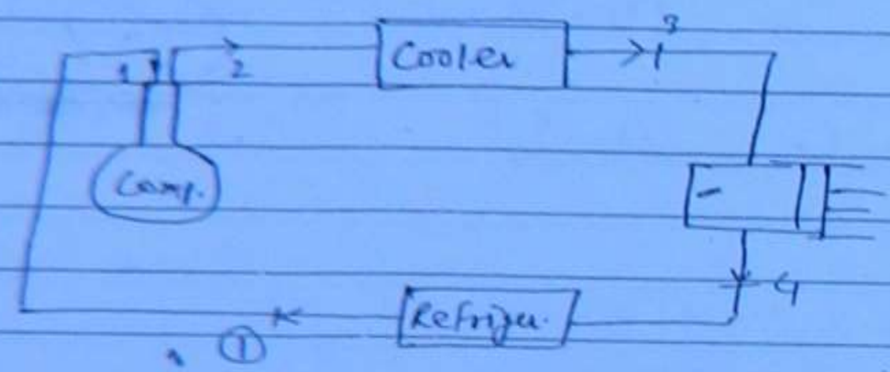
Sol^n: given, $T_1 = 250 K$

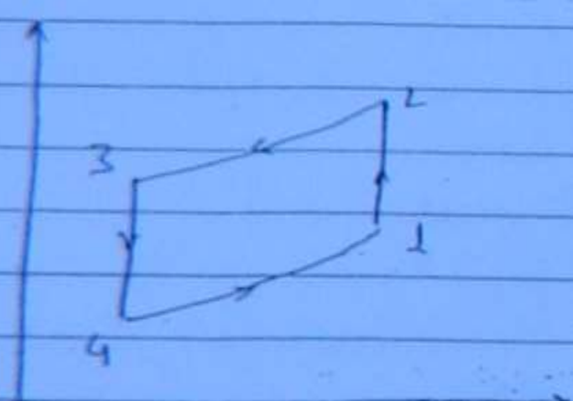
$T_3 = 300 K$

$C_p = 1 \ kJ/kgK.$

The rise in temp. of air in refrigerator is

$T_1 - T_4 = 50K.$





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

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

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

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

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

$= 125-100 = 25$

$T_4 = 200K.$

$T_2 = 375K.$

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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).

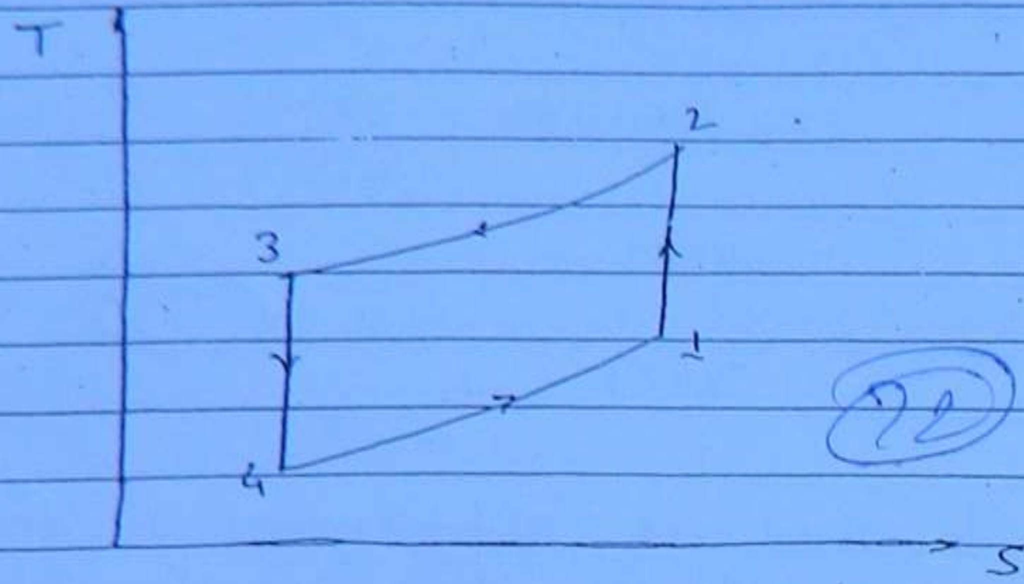
If the temp. at the end of heat absorption & at the end of heat rejection are 0°C & 30°C respectively. pressure ratio = 4 find temp. at all point's & also find volume flow rate at the inlet to the compressor and at the exit of expander. The cooling capacity of the plant is 1 TR & pressure at the inlet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

Solⁿ:- $T_1 = 0°C$ $\qquad (r_p) = 4 = \frac{T_2}{T_1} = \frac{T_3}{T_4}$

$T_3 = 30°C$

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

$P_1 = 1\ bar.$



1-2 is Reversible adiabatic process.

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

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

$$T_2 = 405.67\,K.$$

The compressor and expansion is isentropic process.

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

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

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

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

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

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

$$3.5 = \dot{m} \times 1.005 \ (\underbrace{273 - 203.9}_{73})$$

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

$\because$ Air is treated as ideal gas. Then.

$$PV = mRT.$$

$$P_1 V_1 = mRT_1$$

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

$\because$ $R = 0.287 \ kJ/kgK$ & $P_1 = 100 \ kPa.$

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

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

& $$P_4 V_4 = mRT_4$$

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

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

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 cold to 25°c in a cooler if the compression & expansion follows $PV^{1.35} = C$ & $PV^{1.3} = C$. find the COP. take $\gamma = 1.4$ & $C_P = 1.009$ KJ/kgK.

Soln:-

$T_1 = 10°C = 10 + 273 = 283K.$

$P_1 = 1\ bar.$

$P_2 = 5\ bar.$

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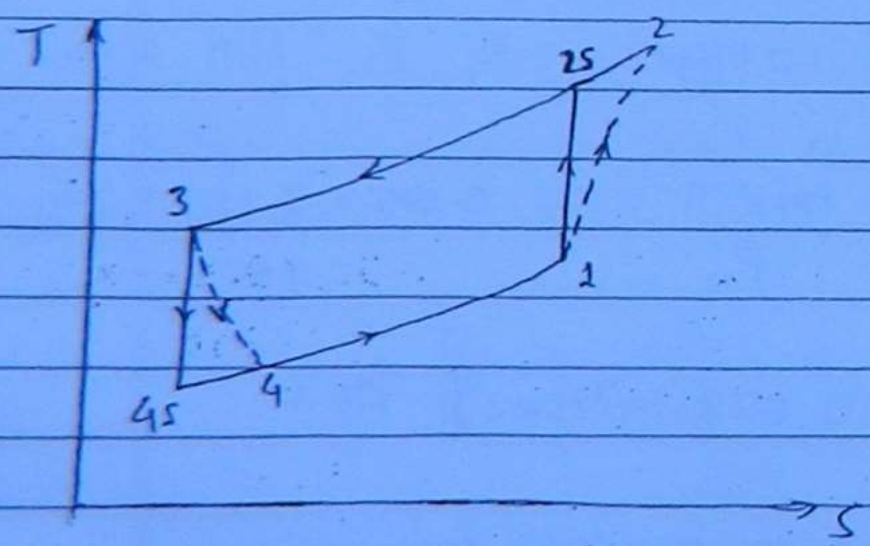
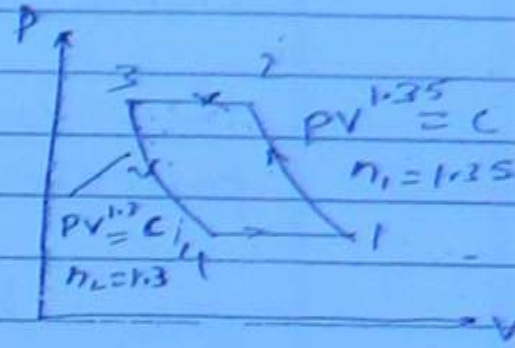
$T_3 = 25°C = 25 + 273 = 298K.$



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

$$\cancel{= \left(\frac{283}{T_2}\right)^{\frac{1.4}{1.4-1}}}$$



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

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

$$T_2 = 429.5\ K$$

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

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

TS

$$T_4 =$$

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

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

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

$$RE = 78.14\ KJ/Kg.$$

$\because$ 1-2 is a polytropic compression process.

then compression work in ~~play~~ polytropic process.

$$W = \frac{n}{n-1}(P_1V_1 - P_2V_2)$$

$$W_c = \frac{n_1}{n_1-1}(P_1V_1 - P_2V_2)$$

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$$W_c = \frac{n_1}{n_1-1}(mRT_1 - mRT_2)$$

$$= \frac{1.35}{1.35-1} mR\,(\cancel{T_1 - T_2})$$

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

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

Expansion work in polytropic process.

$$W_E = \frac{n_2}{n_2-1}(P_3V_3 - P_4V_4)$$

$$= \frac{n_2}{n_2-1}(mRT_3 - mRT_4)$$

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

$$W_e = 115.$$

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

$$= 162.17 - 115$$

$$= 47.17$$

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

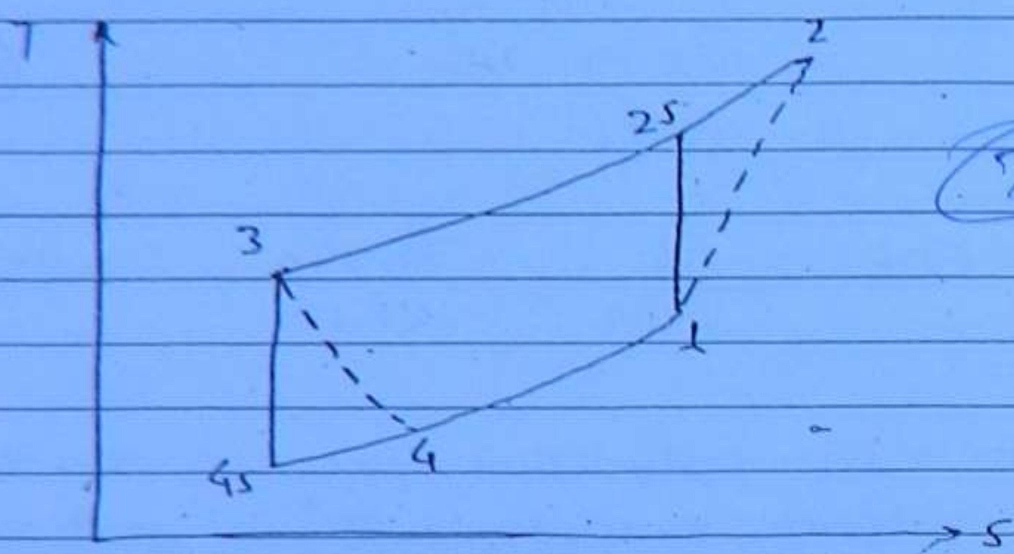
NOTE- In polytropic process as there is heat transfer Q is not is equal to zero is S.F.E.E & hence work is not equal to $(h_2 - h_1)$

Prob- In a air craft Refrigeration system air enter's 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 an isentropic efficiency of 78%. The lower temp. air absorb's 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.

Solⁿ:



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$P_1 = 0.1$ MPa.

$T_1 = 4°C = 4 + 273 = 277 K$

$P_2 = 0.3$ MPa.

$\eta_c = 0.72$

$T_3 = 55°C = 55 + 273 = 328 K$.

$P_4 = 1$ MPa.

$\eta_T = 0.78$

$RC = 3TR = 3 \times 3.5 = 10.5$ kW.

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

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

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

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

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

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

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

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

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

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

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

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

$$T_{4s} = 239.6\,K.$$

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

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

$$\boxed{T_4 = 259\,K}$$

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

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

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

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

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

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

$$= 0.24$$

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

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

$$\boxed{P_{in} = 43.75 \text{ KW}} \text{ Ans.}$$

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

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

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

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

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Vapour absorption Refrigeration cycle

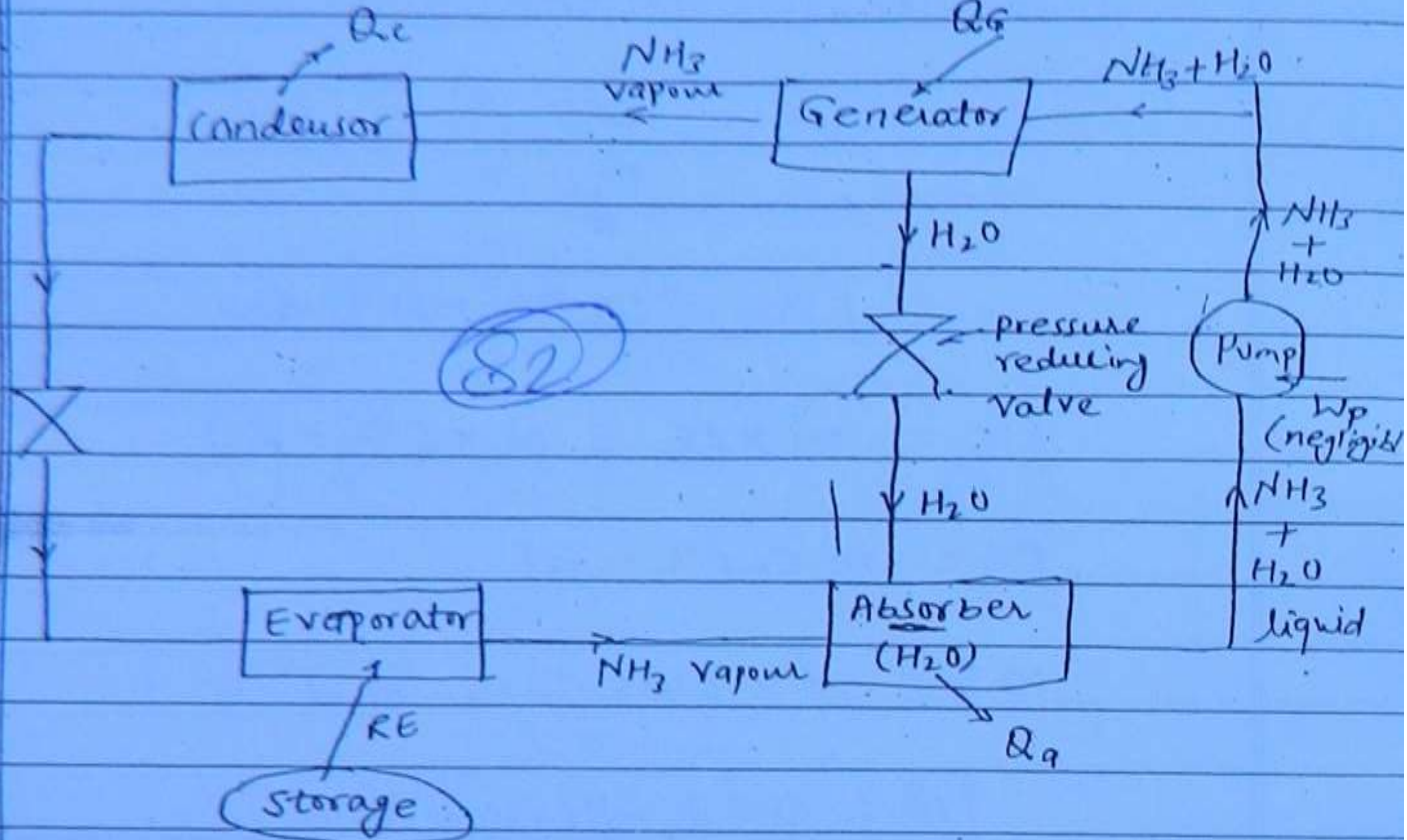

fig:- $NH_3 - H_2O$ VARS System.

Pump → liquid
Compressor → ~~air~~ 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.

$$\boxed{W_c > W_p}$$

Important point's

1. Vapour compression Refrigeration system operate's on high grade energy (work) whereas vapour absorption Refrigeration cycle operates on low grade energy (Heat)

2. In V.A. cycle compressor is replace with generator pump & absorber.

3. Solar refrigeration system are based on V.A. cycle

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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 reject's heat to surrounding in condensor & absorber.

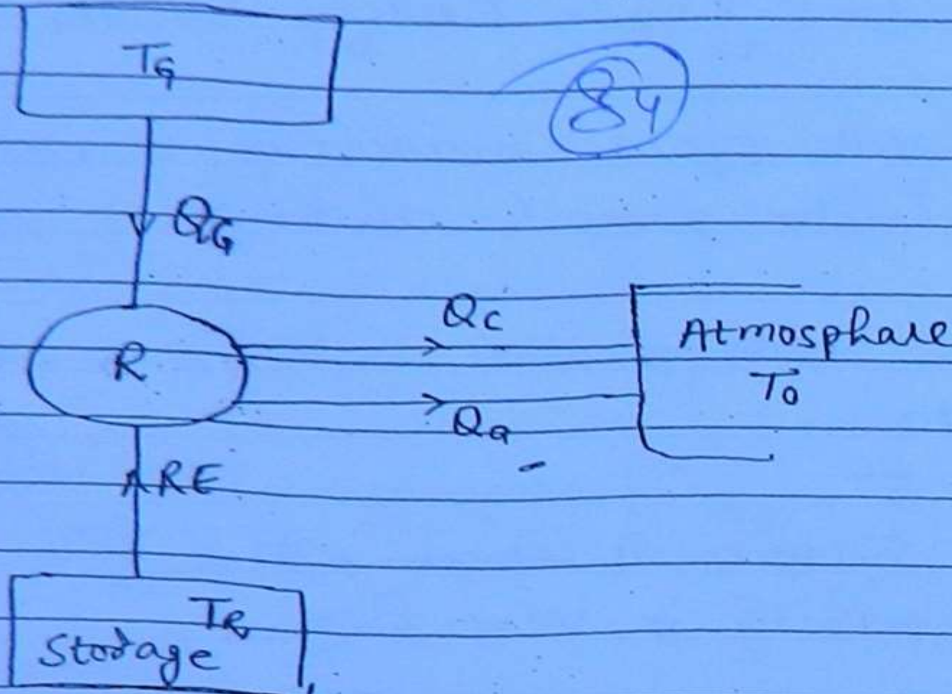
COP of V.A. Cycle :-

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

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

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

84



from 1st law of thermodynamic's.

$$Q_G + RE = Q_C + Q_a \quad \text{——} \; ①$$

From 2nd law of thermodynamic's.

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

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

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

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

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

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

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

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

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

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

85

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

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

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

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

*** Imp. - 2011

Imp.

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

Soln. (i) $T_G = 360 K$

$T_0 = 310 K$

$T_R = 260 K$

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$$COP_{max.} = \left(\frac{T_G - T_0}{T_G}\right)\left(\frac{T_R}{T_0 - T_R}\right)$$

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

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

(ii) $T_R = 250 K.$

$T_0 = 310 K$

$T_G = ?$

$COP = 0.72$

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

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

$$\boxed{T_G = 375\,K} \text{ Ans.}$$

Imp. prob– A VARS work's with 3 thermal reservoir's a refrigeration effect of 100 Watt is required at 250 K. Heat source is available at 400K. heat rejection occur's at 300k then find the min. value of heat required in watt.

soln:– given,

$RE = 100\,W$

$T_R = 250\,K$

$T_G = 400K$

$T_0 = 300K.$

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

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

$$\left(\frac{T_G - T_0}{T_G}\right)\left(\frac{T_R}{T_0 - T_R}\right) = \frac{RE}{(Q_G)_{min.}}$$

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

$$\boxed{(Q_G)_{min} = 80\,W} \text{ Ans.}$$

Prob In $NH_3 - H_2O$ 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.

$$t_{sat} = 120.2\,°C$$

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

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Soln:

$$T_R = -10°C = -10 + 273 = 263 \text{ K.}$$

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

$$RC = 20 \text{ TR} = 20 \times 3.5 = 70 \text{ KW.}$$

$$T_G = 273 + 120.2 = 393.2 \text{ K.}$$

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

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

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

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

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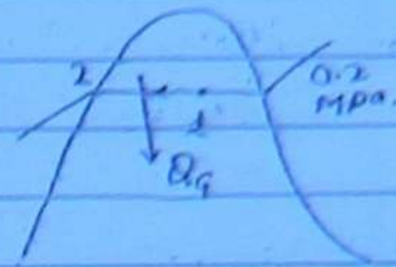
$$h_1 = h_f + x(h_g - h_f)$$



$$h_2 = h_f$$

$$Q_G = h_1 - h_2$$

$$= \cancel{h_f} + x(h_g - h_f) - \cancel{h_f}$$

$$= x(h_g - h_f)$$

$$= 0.9(2201.9)$$

$$Q_G = 1981.7 \text{ KJ/kg}$$

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$$Q_G = m(1981.7)$$

$$COP = \frac{RE}{Q_G} = \frac{20 \times 3.5}{m(1981.7)} \text{ KW}$$

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

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

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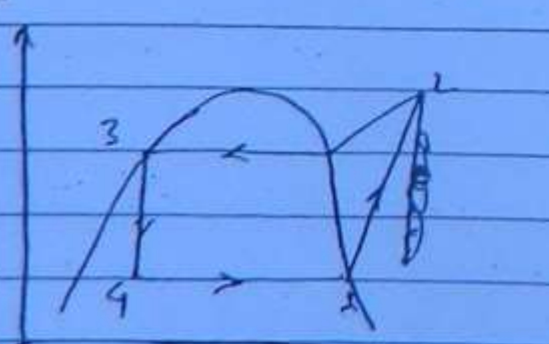
Prob. A vapour compression cycle is represented as shown in figure with state one being exit of evaporator. The the co-ordinate system used in the figure is



(i) P-h (ii) T-S. (iii) P-S और T-h.

prob- Heat rejection ratio of a condensor.

Heat rejection ratio or Heat rejection factor in condensor.

It is defined as the ratio of heat rejected in condensor to the refrigeration effect.

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

It is an indication of condensor ~~side~~ size

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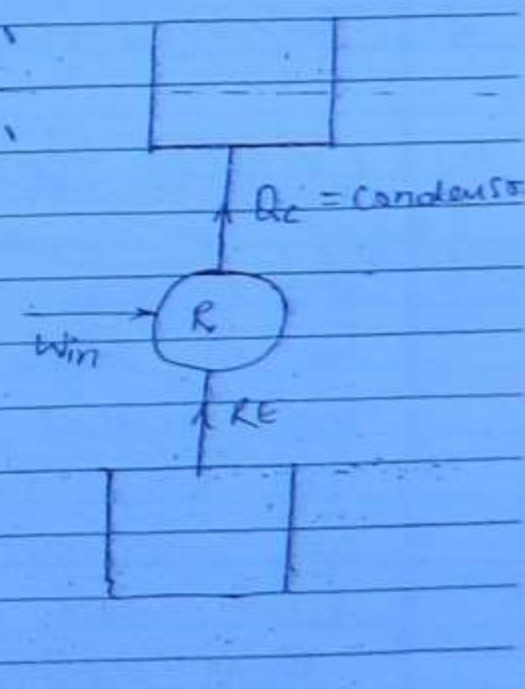

$$Q_C = W_{in} + RE$$

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

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

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

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

Desirable property of Refrigerant or selection of a refrigerant :-

The propertie's of Refrigerants are classified into -

1. Thermodynamic propertie's
2. Chemical propertie's.
3. Physical propertie's

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1. Thermodynamic propertie's :-

(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 refrigerant's are not suitable in tropical condition like india because a critical temp. of these refrigerant's is less than surrounding temp. in summer.

Refrigerant	H_2O	NH_3	R-12	R-22	R-134	CO_2
Critical temp. °C	371.5	132.5	111.5	96.5	101.7	31

Ethylene - 10.6

(ii) Enthalpy of vaporization:-

Larger Enthalpy of vaporization is desirable because for a given refrigeration capacity larger Enthalpy of vaporization result's in lower mass flow rate.

NOTE-

RC = 1000 KW

A B

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Refrigerant	Enthalpy of vap. (KJ/kg)
H_2O	2261
R-22	234.7
NH_3	1369
R-12	165.7
R-134	197.3

NOTE- Of all the common Refrigerant's NH_3 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 leake into the system.

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

(v) Pressure Ratio :- (P_2/P_1)

Lower pressure ratio is desirable because higher pressure ratio result's in larger work input & smaller volumetric efficiency

We general used pressure ratio in V.C. cycle 3 to 5.

(vi) Freezing point :-

The freezing point of the Retrigerant 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).

Recent trend's in Refrigerant's:-

The clorine element which is present in Refrigerant attac's 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 - flouro- Carbon's

(iii) Flouro- carbon's.

Now day's environment fraindly refrigerant used in domestic refrigerator. is R-134a (CF_3CH_2F)

2. Chemical propertie's :-

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

Refrigerant must be non-toxic

(ii) Flammability :-

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The Refrigerant must be non-flammable.

NOTE - though NH_3 is a cheap Refrigerant it is not used in domestic Refrigerator because of it's flamable & 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 separate's from Refrigerant thus oil get's accumulate in evaporator this accumulation of oil result in reduce heat transfer coefficient.

Refrigerant that are not miscible with oil such as NH_2 & CO_2 do not present any problem's because oil separator is used & separated oil is broad back to the compressor.

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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 Compresser

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

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(iv) Action with material of construction:-

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

3. Physical property:-

(i) Viscosity:- It should be less.

(ii) Leak detection:-

Refrigerant should not leak out from the system. & if at all it leak's it's detection must be simple Freon leak's are detected by Hailide torch method in presence of freaon the colour of the light change's from blue to bluish green similarly Ammonia leak's are detected by Sulphur stick method In presence of NH_3 white fume's of Ammonium sulphide are formed.

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Air-CONDITIONING

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Air - Conditioning

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

Moist air (Air) (07)

- Dry air → fixed part
- Water vapour → Variable part

	mass	vol.
N_2	77	79
O_2	23	21
CO_2		
Inert gases		

Psychrometry :- It is the science which deal's with the study of propertie's of moist air.

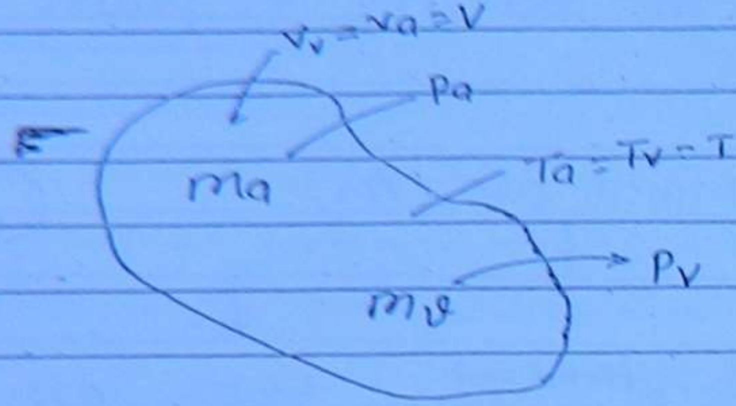
* Psychrometry term *

(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.

$$\boxed{\omega = m_v/m_a}$$

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



(10)

P_t = total pressure of atmosphere.
P_v = partial press. of water vapour.
P_a = atmospheric pressure.

From dalton's law.

$$P_t = P_a + P_v$$

$$P_a = P_t - P_v$$

$$PV = mRT$$

$$P_a v_a = m_a R_a T_a$$

$$P_a V = m_a R_a T \quad \text{——} \quad (1)$$

$$P_v v_v = m_v R_v T_v$$

$$P_v V = m_v R_v T \quad \text{——} \quad (ii)$$

Dividing eqn. (ii) to (i).

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

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

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

$$\omega = \frac{P_v}{P_a} \cdot \frac{\bar{R}/29}{\bar{R}/18}$$

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

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

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

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

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

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

$$\boxed{\omega = f(P_v)}$$

ii. Relative humidity (ϕ):-

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

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

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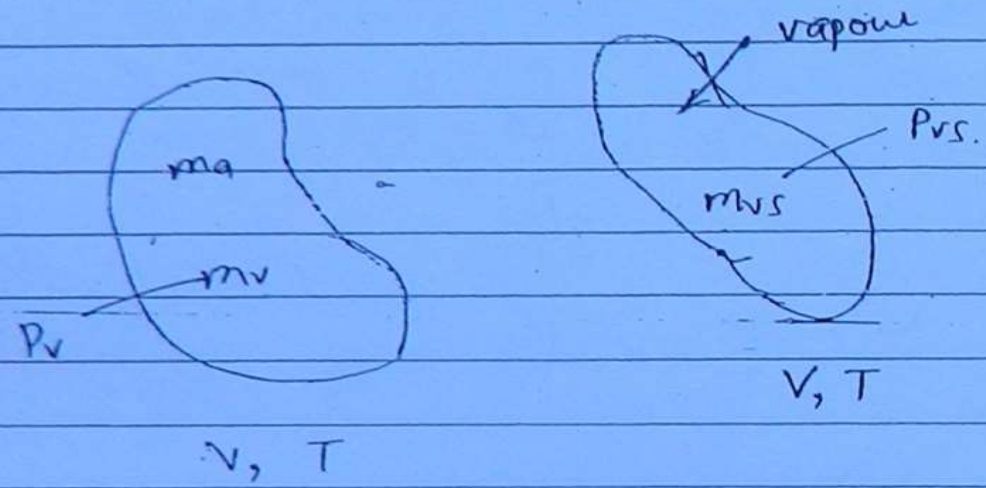


$PV = mRT$

$P_v V = m_v R_v T$ —— (1)

$PV = mRT$

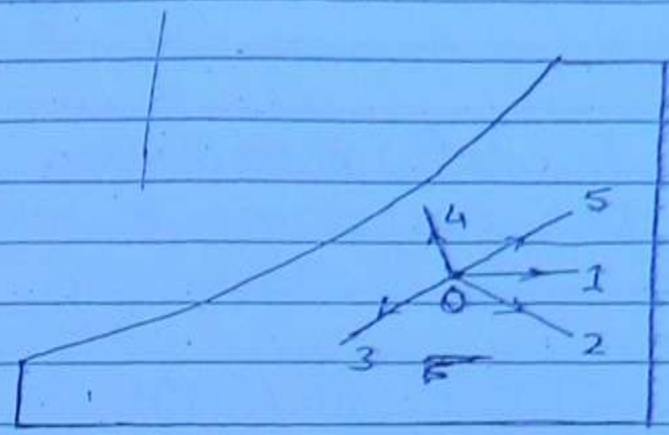
$P_{vs} V = m_{vs} R_v T$ —— (11)

Dividing eqn. (11) to (1)

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

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

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



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Answer → P Q R S T

P	Q	R	S	T
2	1	3	5	4

prob- LIST – I LIST – II

	LIST – I		LIST – II
(A)	Steam spray in air	1.	Sensible cooling
(B)	Air passing over a coil carrying steam.	2.	Cooling and dehumification.
(C)	Air passing over a coil having temp. less than due point temp.	3.	Heating & humidification.
(D)	Air passing over a coil having temp. above due 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.

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

Solⁿ:

t_1	t_2	t_3
20°C	40°C	45°C

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

Prob- In an A/c process 5 KJ/min of heat is extracted from room if the sensible Heat factor is 0.8 then find the latent heat load.

Solⁿ:

$TH = 5$

$SHF = 0.8$

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

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

$SH = 4$

$$0.6 = \frac{P_{v_1}}{0.05733}$$

$P_{v_1} = 0.03498$ bar.

$$\bar{\omega}_1 = 0.622 \left(\frac{0.03498}{1.013 - 0.03498} \right)$$

$\omega_1 = 0.02186$ kg vapour/ kg dry air.

(b)

$t_2 = 25°C$

$P_t = 1.013$ bar.

$P_{vs_2} = 0.03229$

$\omega_2 = 0.02186 - 5\times10^{-3}$

$\omega_2 = 0.01686$ kg vapour/kg dry air.

$$\phi_2 = \frac{P_{v_2}}{P_{vs_2}}$$

$$\omega_2 = 0.622 \frac{P_{v_2}}{P_t - P_{v_2}}$$

$$0.01686 = 0.622 \times \frac{P_{v_2}}{1.013 - P_{v_2}}$$

$P_{v_2} = 0.026$ bar.

$$\phi_2 = \frac{P_{v_2}}{P_{vs_2}} = \frac{0.026}{0.03229} = \cancel{0.828}\ 0.827$$

$$\phi_2 = 82.7\%$$

(198)

$$h_2 = C_{pa} + \omega_2 (2500 + 1.88 t_2)$$

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

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

Q6: Math group 1 with group-II

~~Group~~

Group-I (Process in figure)		Group-II (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 behaves as an ideal gas then find the relative humidity of air take saturation pressure of vapour as 3.167 kPa.

n:-

$P_t = 101.3$ kPa.

$m_v = 10$ gram.

$P_{vs} = 3.167$ kPa.

$m_a = 1$ kg of dry air.

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

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

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

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

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

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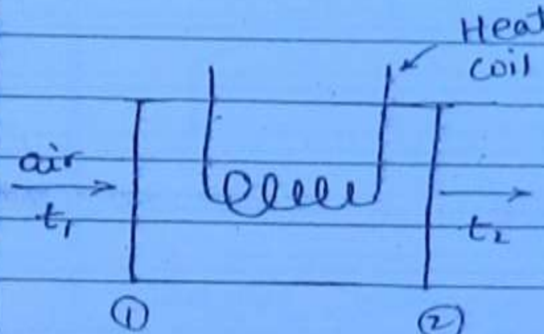
DATE : / /

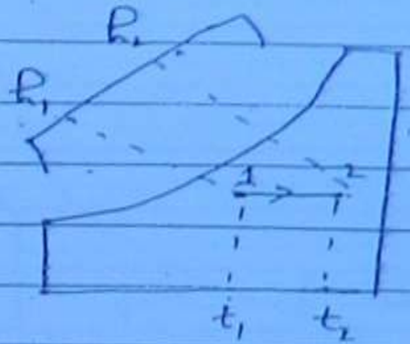
Basic Psychrometric process

1/1

1. SENSIBLE HEATING :-

It is the process of heating air at constant ω.





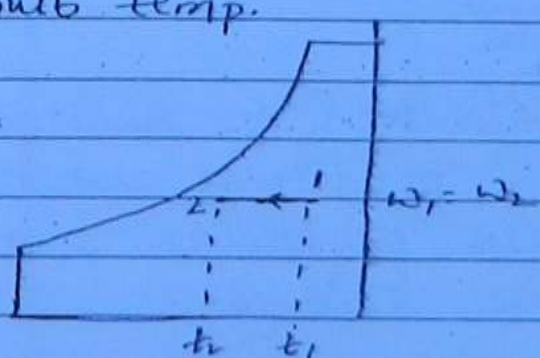
Applying steady flow eqn.

$$h_1 + Q_1 = h_2 + W_1$$

$$\boxed{Q_1 = h_2 - h_1}$$

2. SENSIBLE COOLING :-

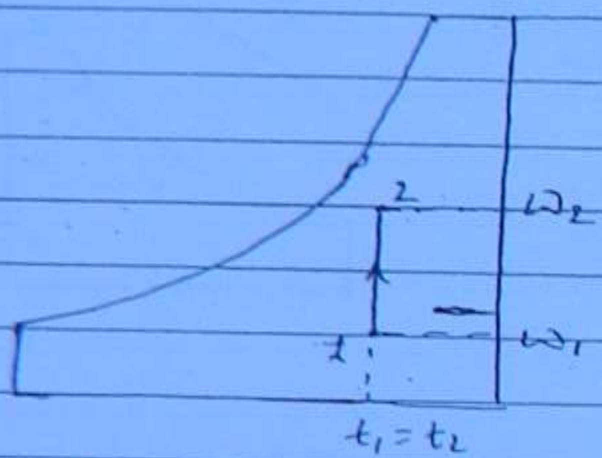
It is the process of cooling air at constant ω. for sencible cooling the coil temp. must be greater than due point temp. & less than dry bulb temp.



3. Humidification:-

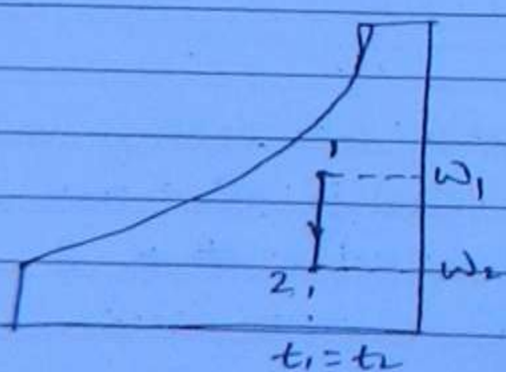
(112)

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.

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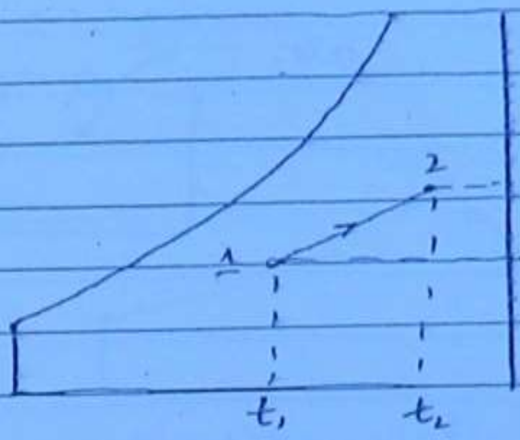
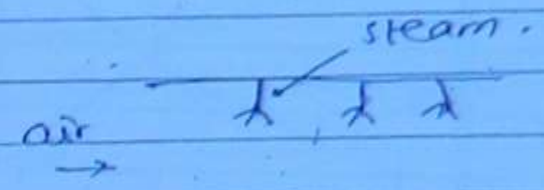
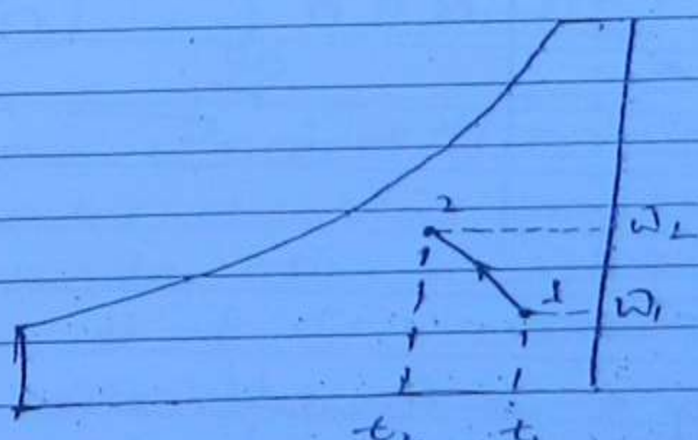




Steam spray in air example. of ~~heam~~ heating & humidification

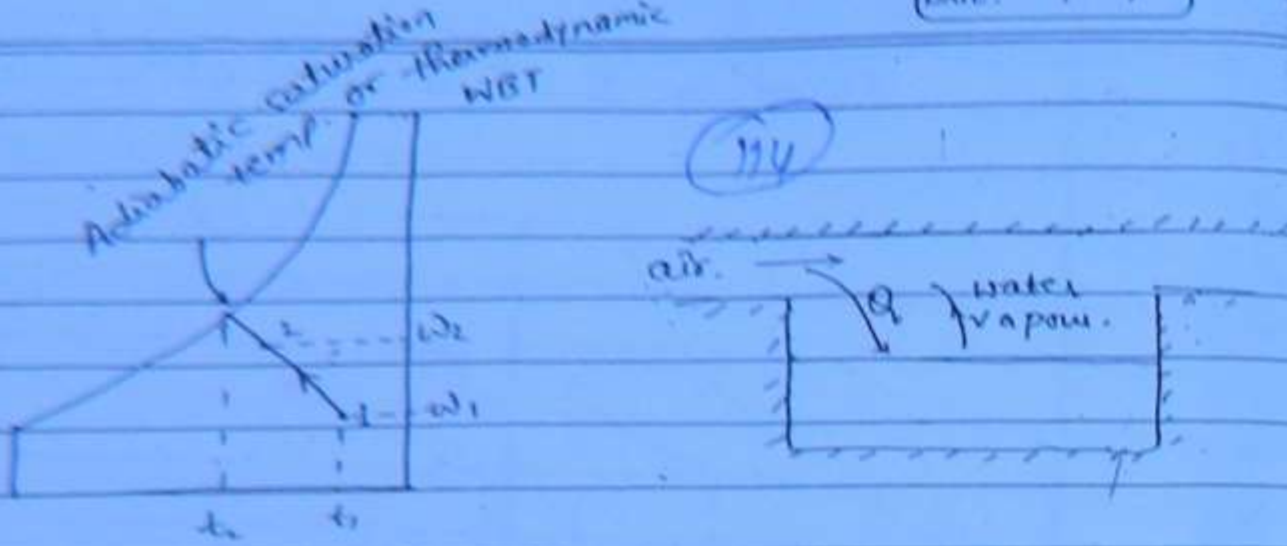
Cooling & humidification :-



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

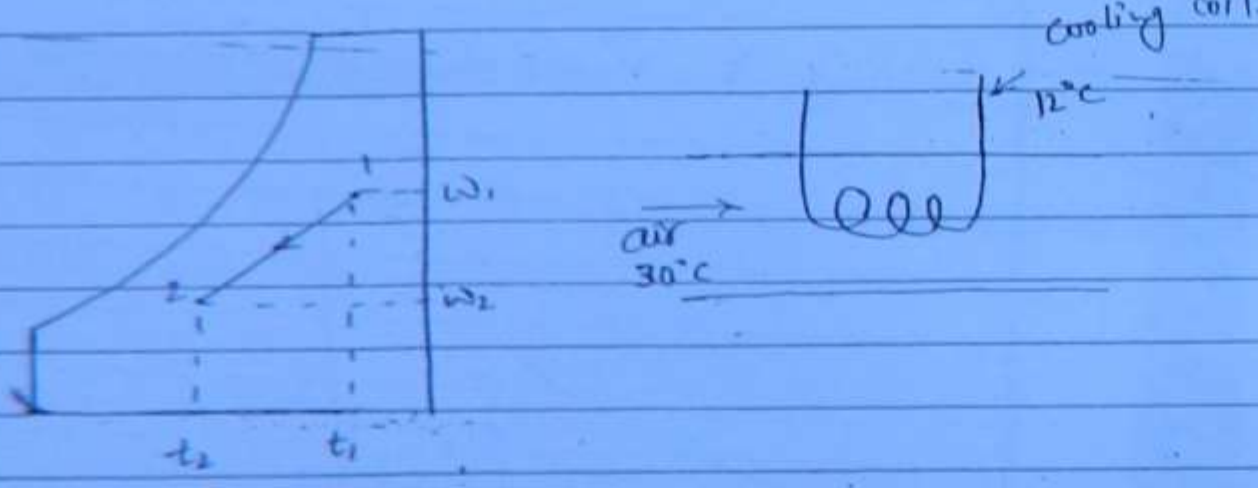
$Q = h_2 - h_1$

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



7. <u>Cooling & De-humidification</u>:-

Cooling & De-humidification occurs when the coil temp. Is <u>less than due point temp.</u>



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

8. Heating & De-humidification :-

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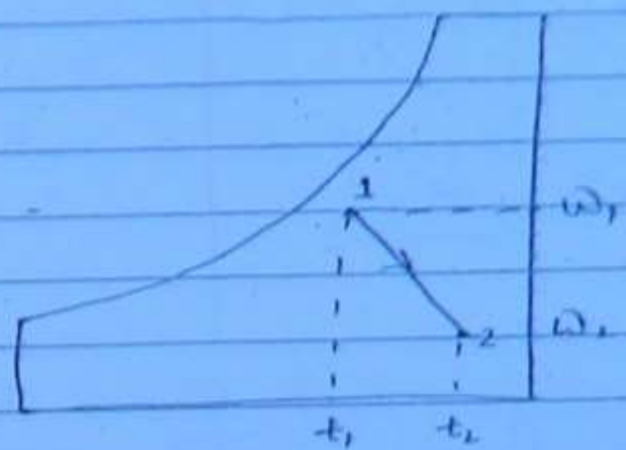
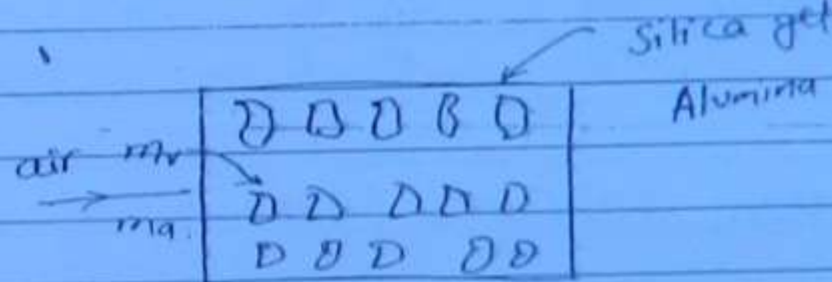


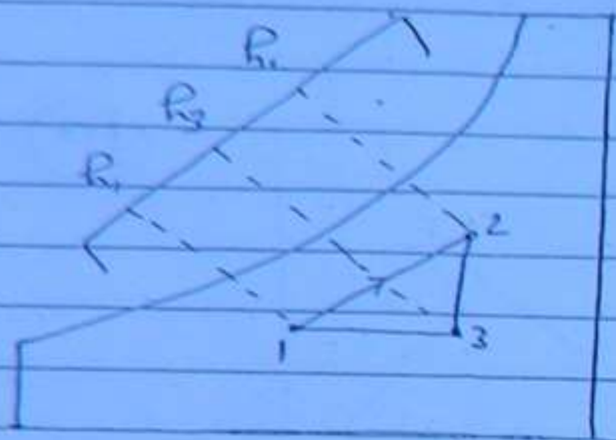
Example :- Special case :- Adiabatic chemic dehumidification.



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

9. SENSIBLE HEAT FACTOR SHF :-

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



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$TH = h_2 - h_1$

$SH = h_3 - h_1$

$LH = h_2 - h_3$

$TH = SH + LH$

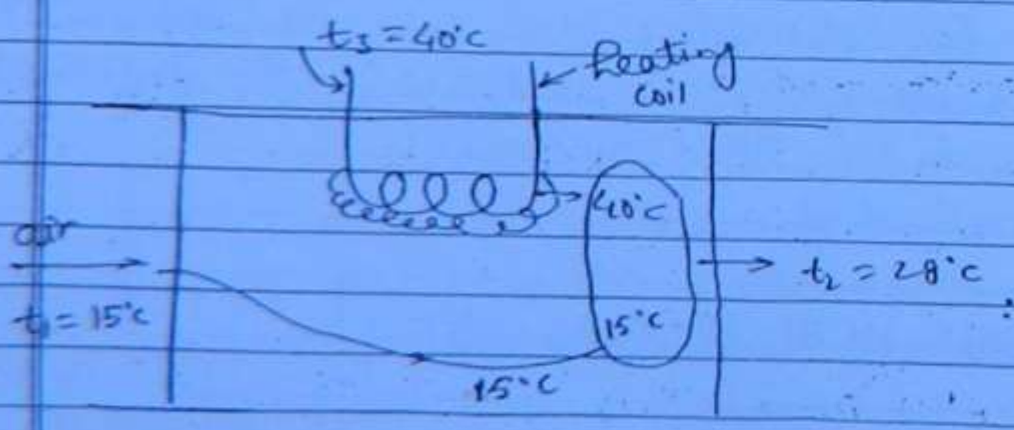
$= h_3 - h_1 + h_2 - h_3$

$TH = h_2 - h_1$

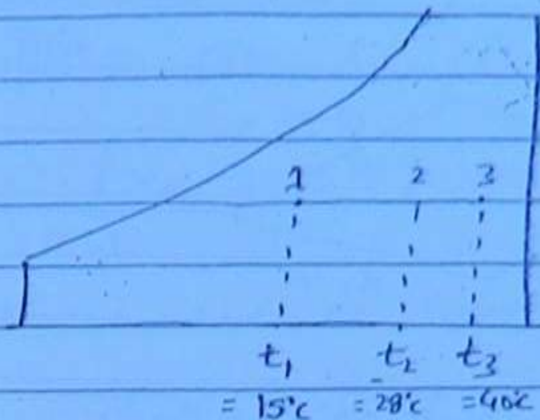
$$SHF = \frac{SH}{TH} = \frac{h_3 - h_1}{h_2 - h_1}$$

☆ By pass factor :- (BPF) :-

It is also known as Uncontacted factor or loss factor.



NOTE- For moist air 3 parameter's are required to fixed the state but on psychrometric chart only 2 parameter's 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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$$BPF = \frac{t_3 - t_2}{t_3 - t_1}$$

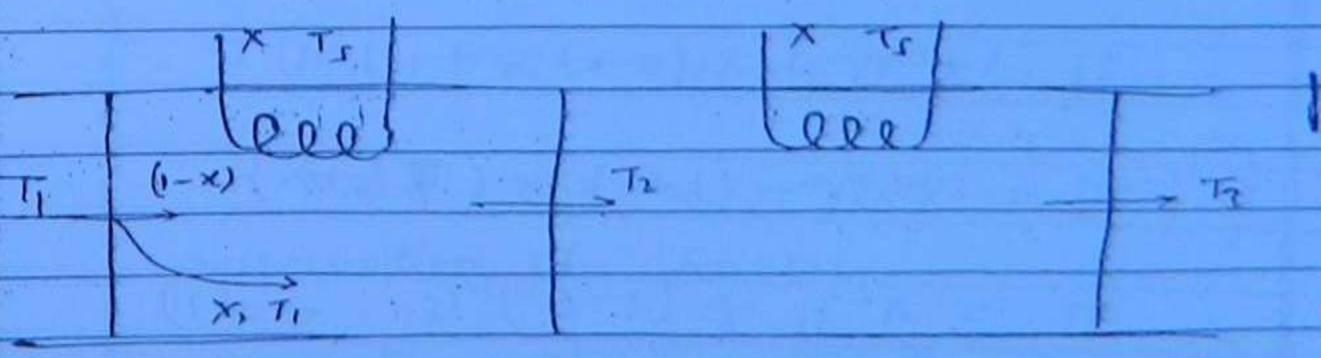
$$\eta = \frac{t_2 - t_1}{t_3 - t_1}$$

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

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

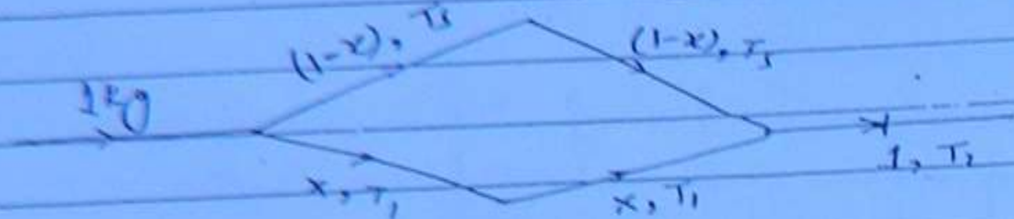
Imp.

☆ Effecitive By pass factor or Equivalent By-pass factor.



$$\boxed{h = c_p T}$$

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



$(h = C_p T)$

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

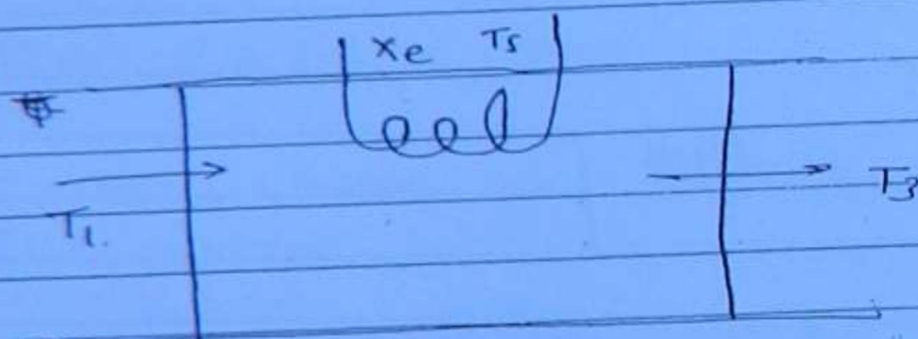
HB

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

Similarly,

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

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



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

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

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

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

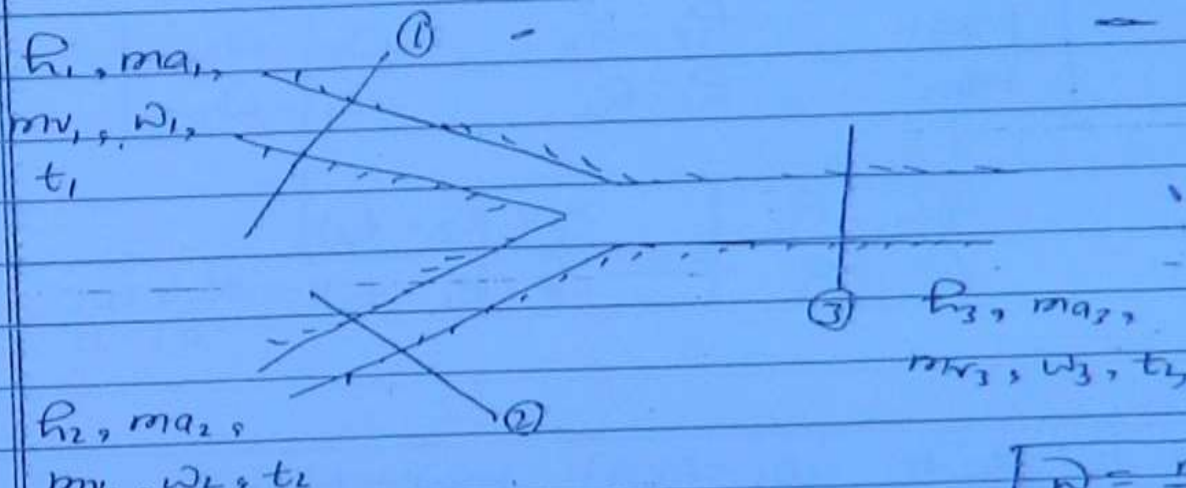
Compare ① & ②

$$\boxed{x_e = x^2}$$

If there are n number of similar coil.

$$\boxed{x_e = x^n}$$ (119)

* Mixing of air stream *



$$\boxed{\omega = \frac{mv}{ma}}$$

Conservation of mass.

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

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

$$ma\omega_1 + ma_2\omega_2 = ma_3\omega_3 \quad \text{—— (II)}$$

conservation of Energy

$$ma_1 h_1 + ma_2 h_2 = ma_3 h_3 \quad \text{—— (III)}$$

from eqn (i) put the value of m_{a3}. in eqn (iii)

$$m_{a1}h_1 + m_{a2}h_2 = (m_{a1} + m_{a2})h_3$$ 120

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

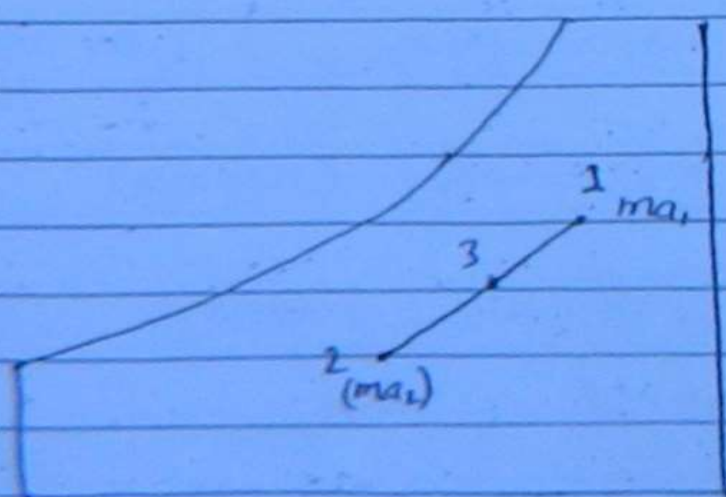
$$m_{a1}h_1 - m_{a1}h_3 = m_{a2}h_3 - m_{a2}h_2$$

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

$$\boxed{\frac{m_{a1}}{m_{a2}} = \frac{h_3 - h_2}{h_1 - h_3} = \frac{\omega_3 - \omega_2}{\omega_1 - \omega_3}}$$

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

$$\boxed{\frac{m_{a1}}{m_{a2}} = \frac{t_3 - t_2}{t_1 - t_3} = \frac{\omega_3 - \omega_2}{\omega_1 - \omega_3}}$$



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

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

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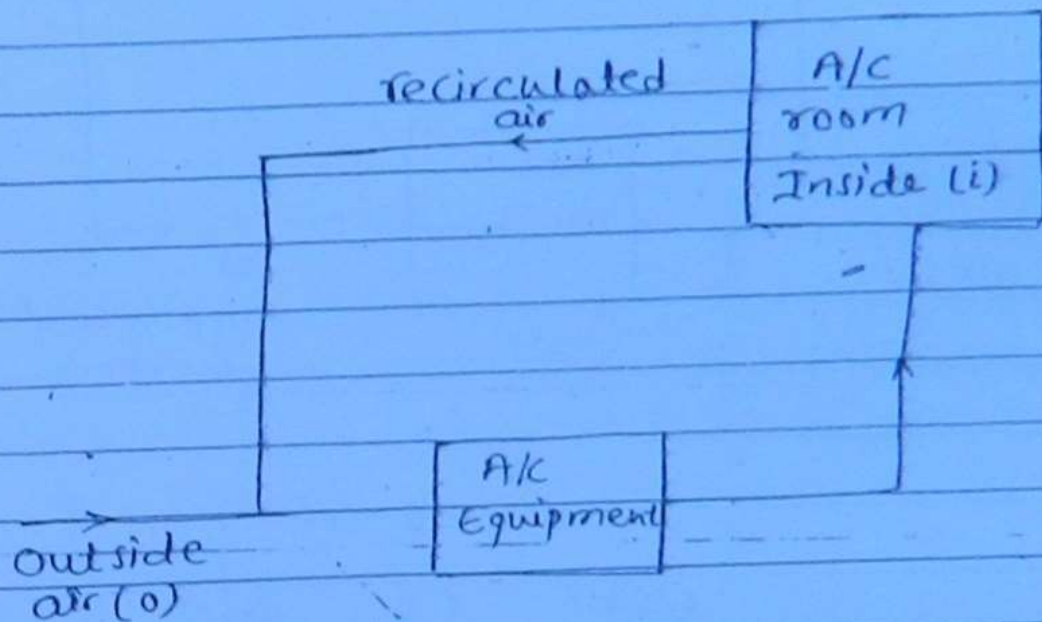
* Ventilation Air *



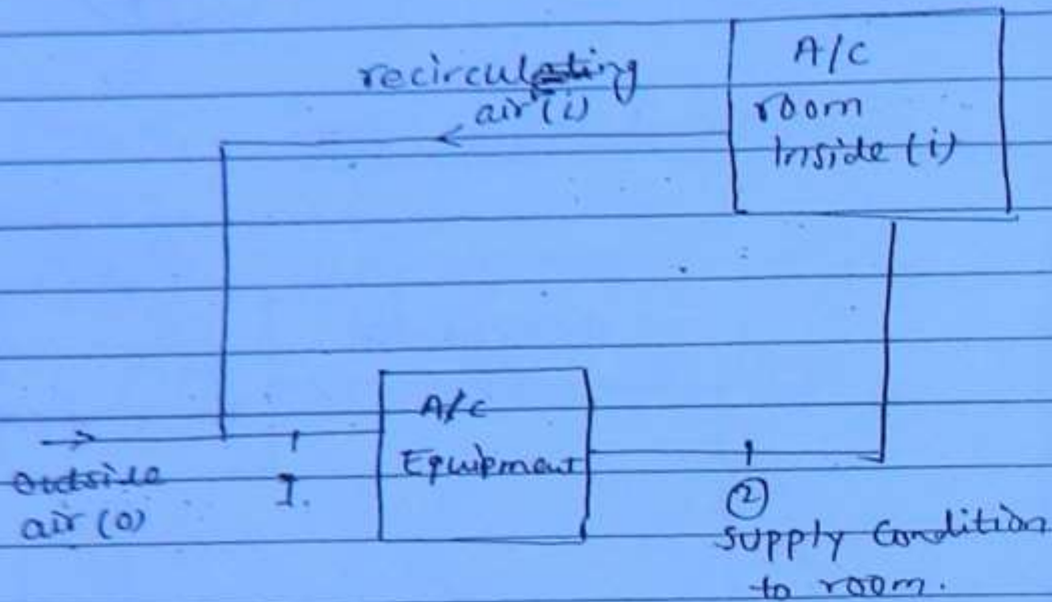
The amount of fresh air that is supplied in order to maintain purity of air is know as ventilation air.

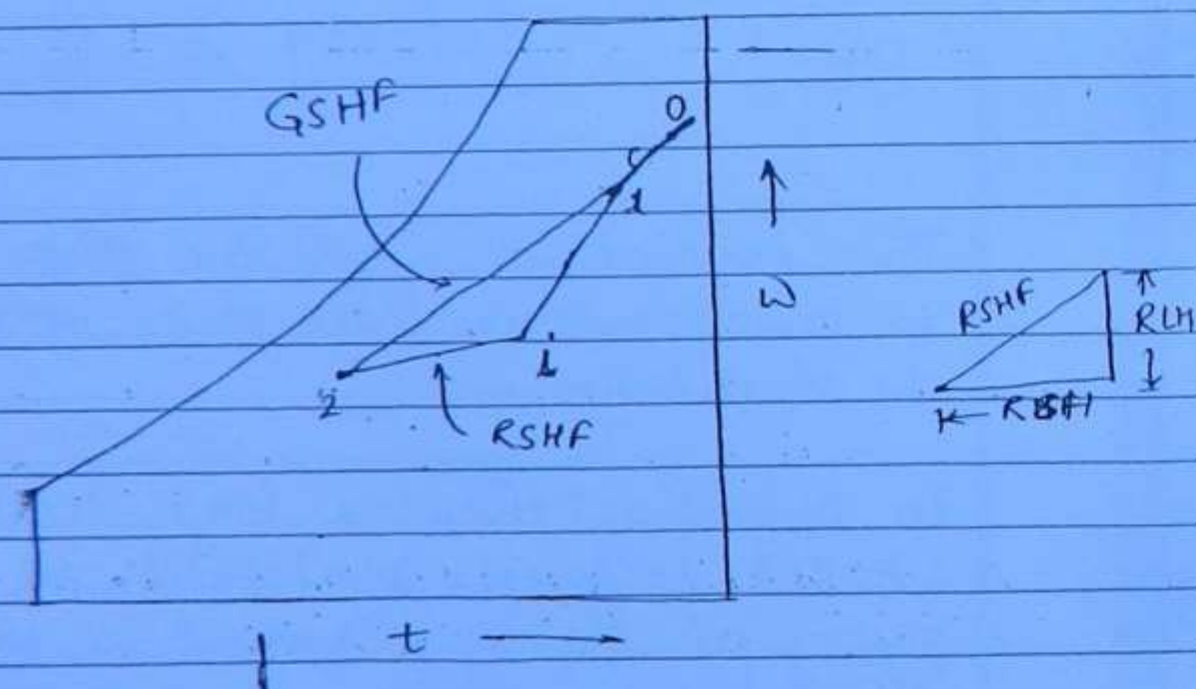
In operation theater 100% fresh air is supplied because availability of oxygen & purity of air is important in operation theater.

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

* Summer air-conditioning with ventilation air *

(122)

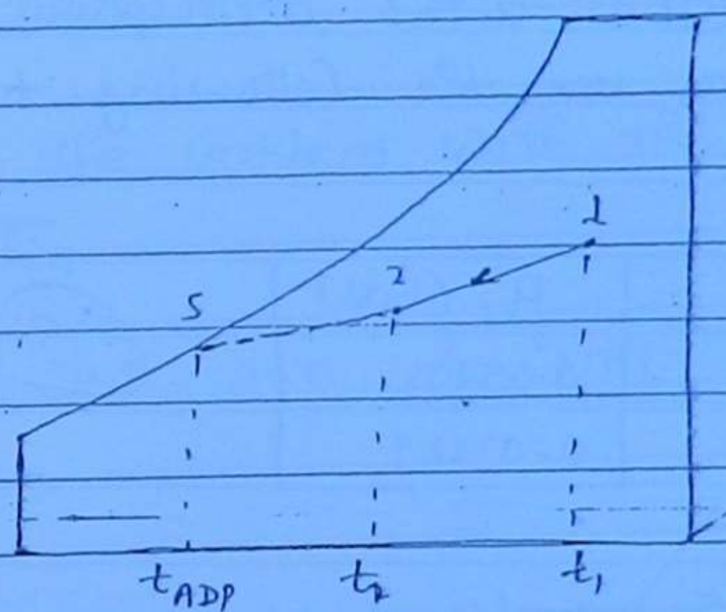




The line which join's inlet & exit condition of A/c equipment is known as Grand sensible heat factor line.

- 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 GSHF & RSHF will give supply condition's to room.

Apparatus Dew point (ADP)



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

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

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 become's 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.

$t°C$	P_{vs} (bar)
35	0.05733
25	0.03229

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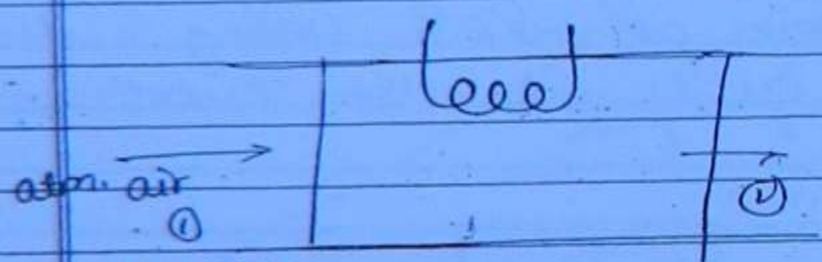
Soln:- m_v = 5 gram/kg of dry air.



$t_2 = 25°C$

$t_1 = 35°C$

$\phi_1 = 0.6$

$$\omega_1 = 0.622 \frac{P_{v_1}}{P_t - P_{v_1}}$$

$$\phi_1 = \frac{P_{v_1}}{P_{vs_1}}$$

$P_{vs_1} = 0.05733$

$$TH = SH + LH$$
$$5 = 4 + LH$$

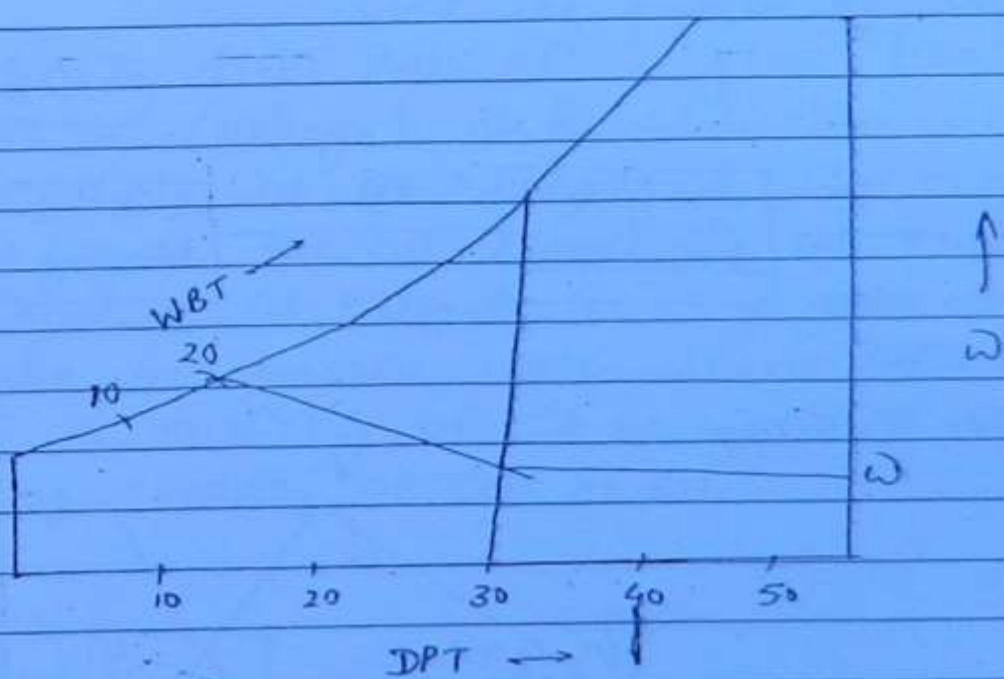
$$\boxed{LH = 1}$$ Ans.

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

(i) Relative humidity (ii) Enthalpy
(iii) Due point temp. (iv) Specific volume

Solve the problem with the help of psychromet. chart.

Soln:- DBT = 30°C & WBT = 20°C



RH = 40% , H = 57 , DPT = 15°C

ω = 0.0105 to 0.0109 , ϕ = 39 to 40%

v = 0.87 to 0.876

25-98

Prob- A small oditorium 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 m^3/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 -

(i) Cooling capacity of the coil in TR & it's Bypass factor.

(ii) Amount of water vapour removed by cooling coil in kg vapour per hour.

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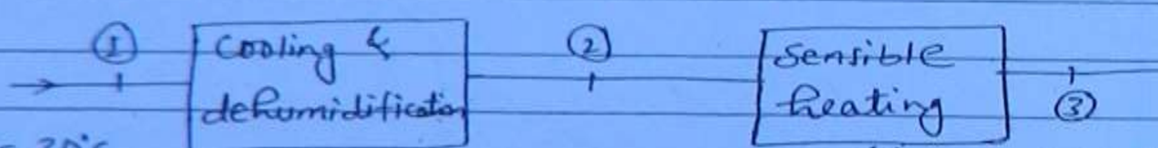
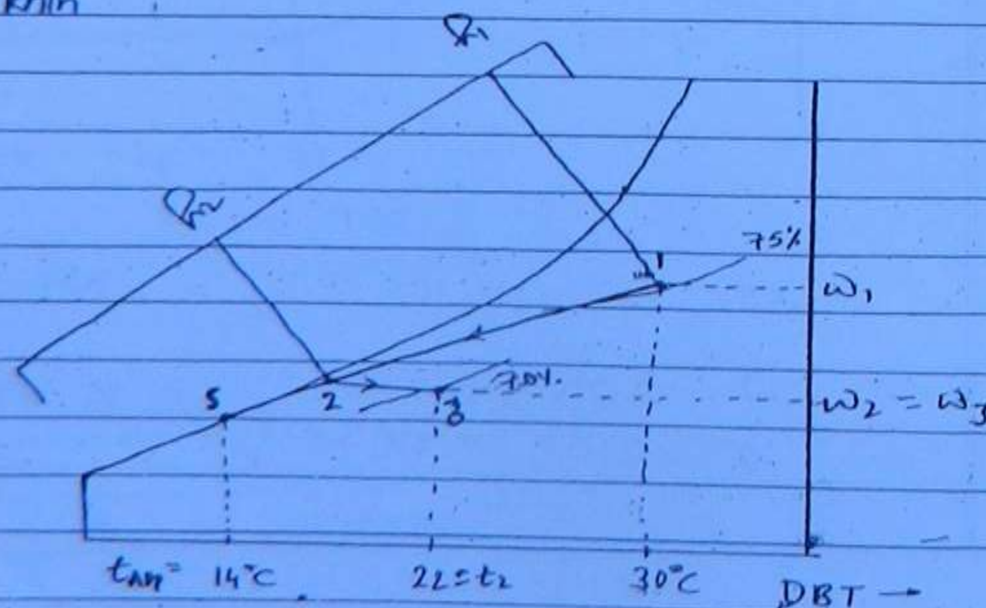
Soln:



$t = 30°C$
$\phi = 75\%$
$V_1 = 200\ m^3/min$

$t = 22°C$
$\phi = 70\%$



$\vartheta = \frac{V}{m}$ & $m = \frac{V_1}{\vartheta_1} =$ 127

Cooling load = $(h_2 - h_1)$ KJ/kg da. =

Total cooling load = $m(h_2 - h_1) \rightarrow \frac{kg\,da}{min} \times \frac{KJ}{kg\,da}$

= ~~37 to 38~~

moisture removed = $(\omega_1 - \omega_2)$ kg v/kg da.

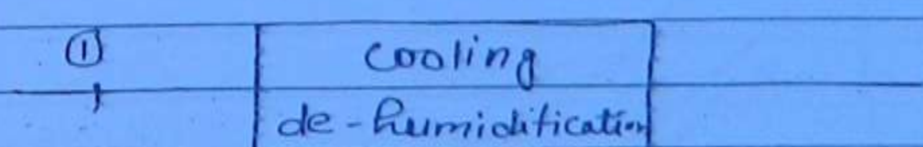
Total moisture removed = $m(\omega_1 - \omega_2) \rightarrow \frac{kg\,da}{min} \times \frac{kg\,v}{kg\,da}$

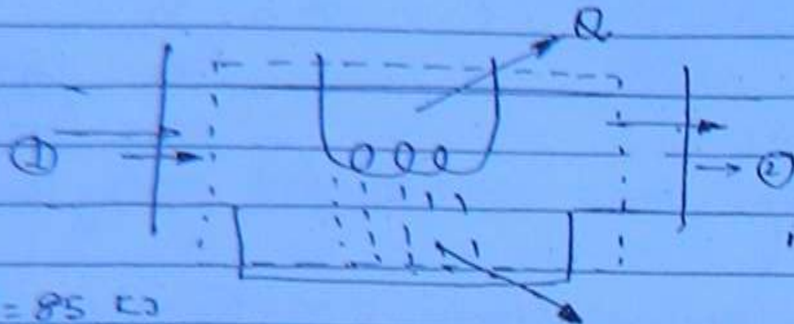
= 106 to 140

By pass factor = 0.12 to 0.2

Q6- 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 & ω = 19 gram of vapour/kg da the air leaves the coil with an enthalpy of 43 KJ/kg da & ω = 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.

Soln:-





$m_a = 5$ kg da/sec.

$h_1 = 85 \frac{KJ}{kg\,da}$ $\qquad$ $h_2 = 43$ KJ/kg da

$\omega_1 = 19 \frac{gm\,v}{kg\,da}$ $\qquad$ $\omega_2 = 8$ gm v/kg da

$m = 3$ kg da/sec

Condensate H_2O $\downarrow$ $h = 67 \frac{KJ}{kg}$

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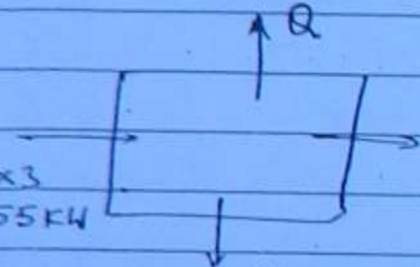


$85 \times 3 = 255$ KW $\qquad$ $43 \times 3 = 129$

$19 - 8 = 11 \frac{gm\,v}{kg\,da} = 11 \times 10^{-3}$ kg v/kg da

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

Heat Carrying by H_2O $= 33 \times 10^{-3} \times 67 = 2.211$ KW

$255 = Q + 2.211 + 129$

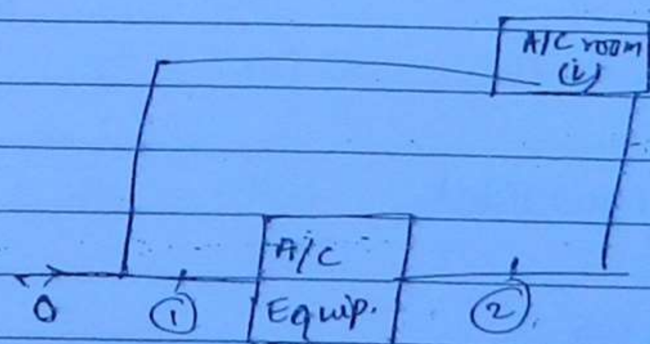
$Q = 123.79$ KW.

prob- In an A/C. Space 15 kg/sec. of fresh air at 45°C DBT and 30% RH is introdu in room at 25°C DBT & 50% RH 450 kg/sec. of recirculated air is mixe with out side air and this mixed air then flow's over a cooling coil with ADP 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 psychro metric chart ⇒ use the following table

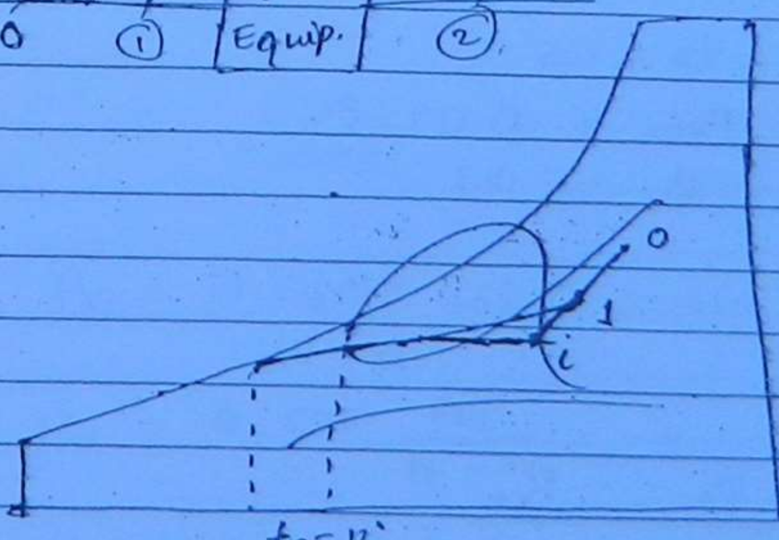
$t°C$	12	25	45
P_{vs} (bar)	0.014016	0.03166	0.09584

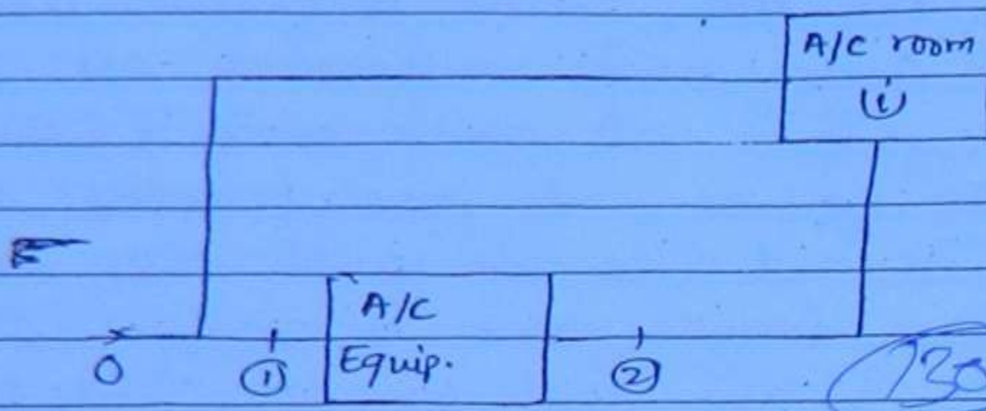
& $P_t = 1.0$

Soln:



T24





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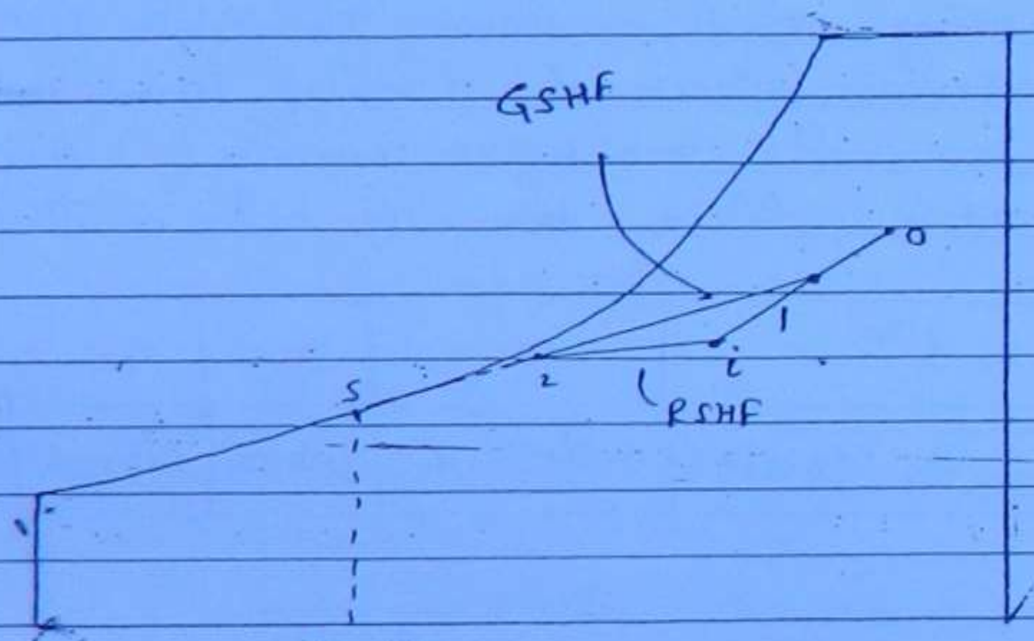

Point 0 (outside).

$t_o = 45$

$P_{vso} = 0.09584$

$\phi_o = 0.3$

$$W_o = 0.622 \frac{P_{vo}}{P_t - P_{vo}}$$

$$\phi_o = \frac{P_{vo}}{P_{vso}}$$

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

$$P_{v_o} = 0.0287 \qquad 131$$

$$\omega_o = 0.622 \times \frac{0.0287}{1.013 - 0.0287}$$

$$\omega_o = 101816$$

$$h_o = C_{pa}t + \omega_o(2500 + 1.88t)$$
$$h_o = 1.005 \times 45 + 0.01816\,(2500 + 1.88 \times \cancel{45}$$
$$h_o = 46.87 \text{ KJ/kg da.}$$

Point (i) —

$t_i = 25$, $\phi_i = 0.5$, $P_{vsi} = 0.03166$
$P_t = 1.013$ bar.

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

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

$$P_{vi} = 0.0158$$

$$\omega_i = 0.622\,\frac{P_{vi}}{P_t - P_{vi}}$$

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

$$\omega_i = 0.00987$$

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$$h_i = C_{pa} t_i + \omega_i (2500 + 1.88 t_i).$$

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

$$= 50.27 \text{ KJ/kg da}$$

Point (1),

$$m_0 \omega_0 + m_i \omega_i = m_1 \omega_1$$

$m_0 = 50$ fresh air given.

$m_i = 450$ recirculated given.

$$m_1 = 50 + 450 = 500$$

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

$$\omega_1 = 0.0107$$

$$m_0 h_0 + m_i h_i = m_1 h_1$$

$$50 \times 92.2 + 450 \times 50.2 = 500 \times h_1$$

$$h_1 = 54.4$$

$$h_1 = C_{pa} t_1 + \omega_1 (2500 + 1.88 t_1)$$

$$t_1 = 27°C$$

Point (S),

$t_s = 12°c$, $\phi_s = 1$, $P_{vs} = 0.014016$

$\phi_s = \frac{P_v}{P_{vs}}$

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$1 = \frac{P_v}{P_{vs}}$

$\Rightarrow \quad P_v = P_{vs} = 0.014016$

~~$P_v = 0.0$~~

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

$= 0.622 \times \frac{0.014016}{1.013 - 0.014016}$

$\omega_s = 0.00872$

$h_s = C_{pa} t_s + \omega_s (2500 + 1.88 t_s)$

$= 1.005 \times 12 + 0.00872 (2500 + 1.88 \times 12]$

$= 34.07$ KJ/kg da.

$BPF = \frac{T_2 - T_s}{T_1 - T_s}$

$0.15 = \frac{t_2 - 12}{27 - 12}$ $\quad t_2 = 14.25°C$

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

$$BPF = \frac{h_2 - h_s}{h_1 - h_s}$$ 134

$$0.15 = \frac{h_2 - 34.07}{54.4 - 34.07}$$

$$h_2 = 37.1 \text{ KJ/Kg da.}$$

$$h_2 = C_{pa} t_2 + \omega_2 (2500 + 1.88 t_2)$$

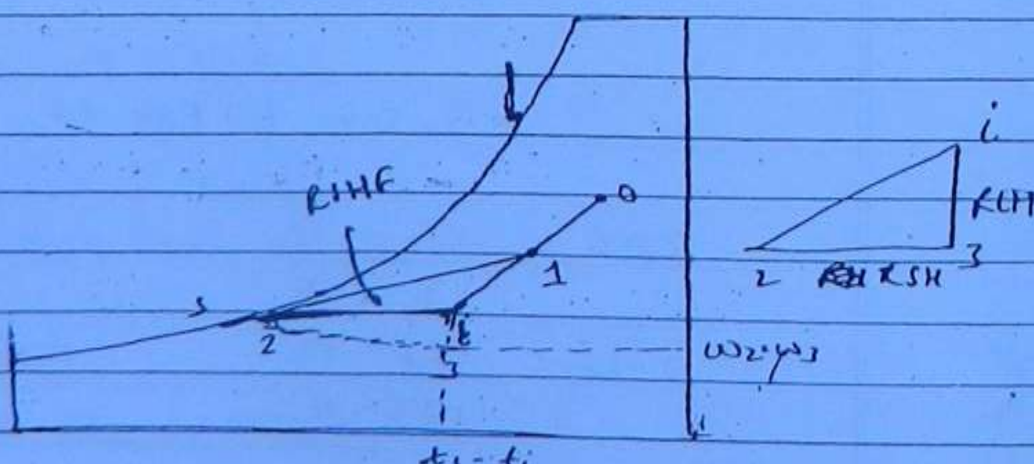
$$37.1 = 1.005 \times 14.25 + \omega_2 (2500 + 1.88 \times 14.25)$$

$$\omega_2 = 0.00902$$

i) Cooling load $= m (h_1 - h_2)$

$$= 500 (54.4 - 37.1) \quad \frac{Kg}{sec} \times \frac{KJ}{Kg}$$

$$= 8650 \text{ KW}$$



$$h_3 = C_{pa} t_3 + \omega_3 (2500 + 1.88 t_3)$$

$$= 1.005 \times 25 + 0.00902 (2500 + 1.88 \times 25)$$

$$= 48.1 \text{ kJ/kg da.}$$

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$$RSH = m(h_3 - h_2)$$

$$RSH = 500 (0.481 - 37.1)$$

$$RSH = 5480 \text{ kW.}$$

$$RLH = m(h_1 - h_3)$$

$$= 500 (50.27 - 48.1)$$

$$= 1090 \text{ kW.}$$

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. include's temp. (DB), Humidity and air velocity.

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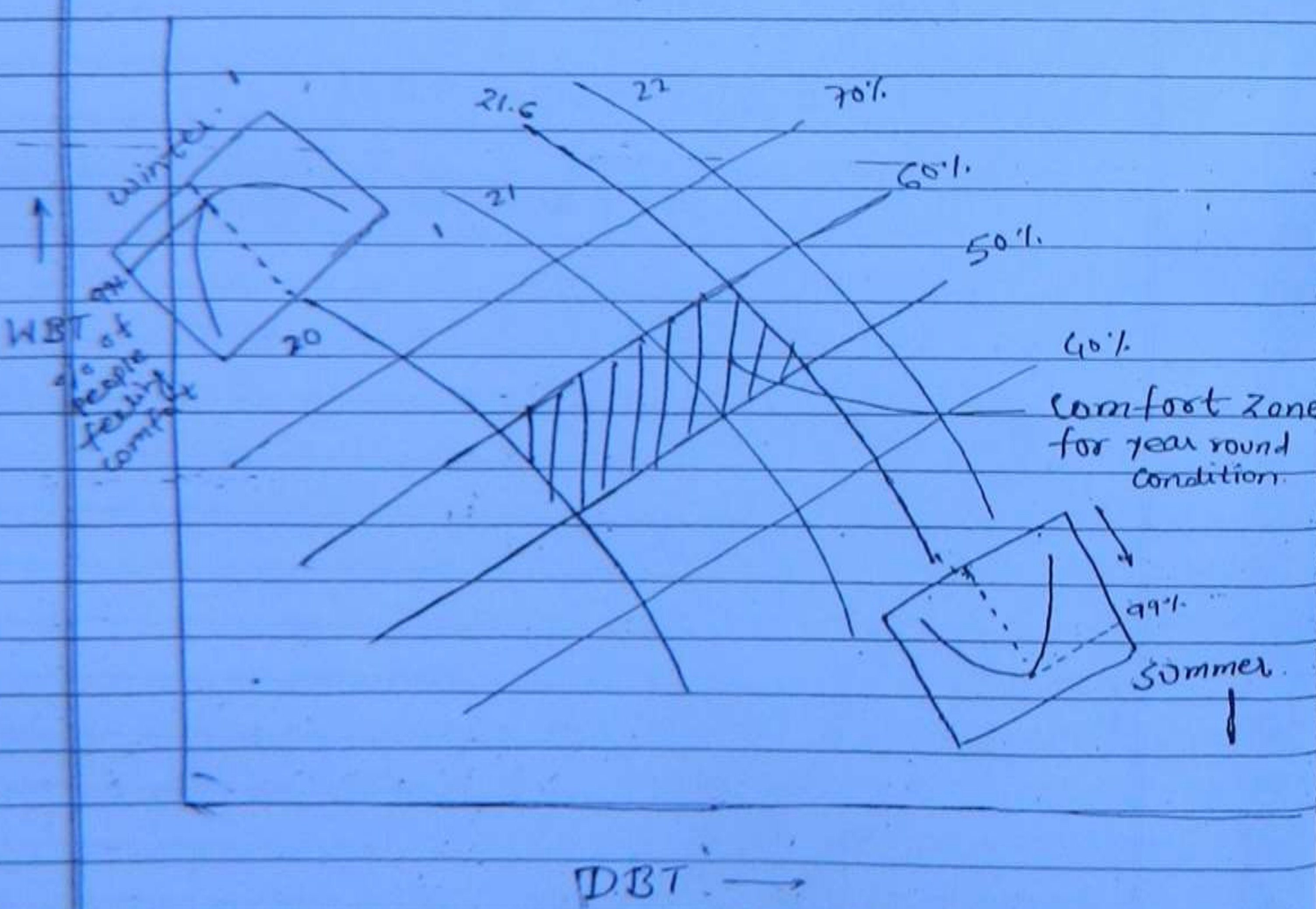
Comfor Chart



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 ~~d~~ dry bulb temp. is taken on x-axis & wet bulb temp. is taken on y-axis Relative humidity line's are drawn from psychrometric chart it is found from ~~sea~~ research at an effective temp. about 20°C, 97% ~~pepo~~ people were comfortable in winter condition & in summer it is 21.6°C.
The sadded area show's comfort zone for year round conditioning.

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Refrigeration Equipment

The principle component's are, Compressor, condensor, Expansion device & Evaporator.

A Compressor :

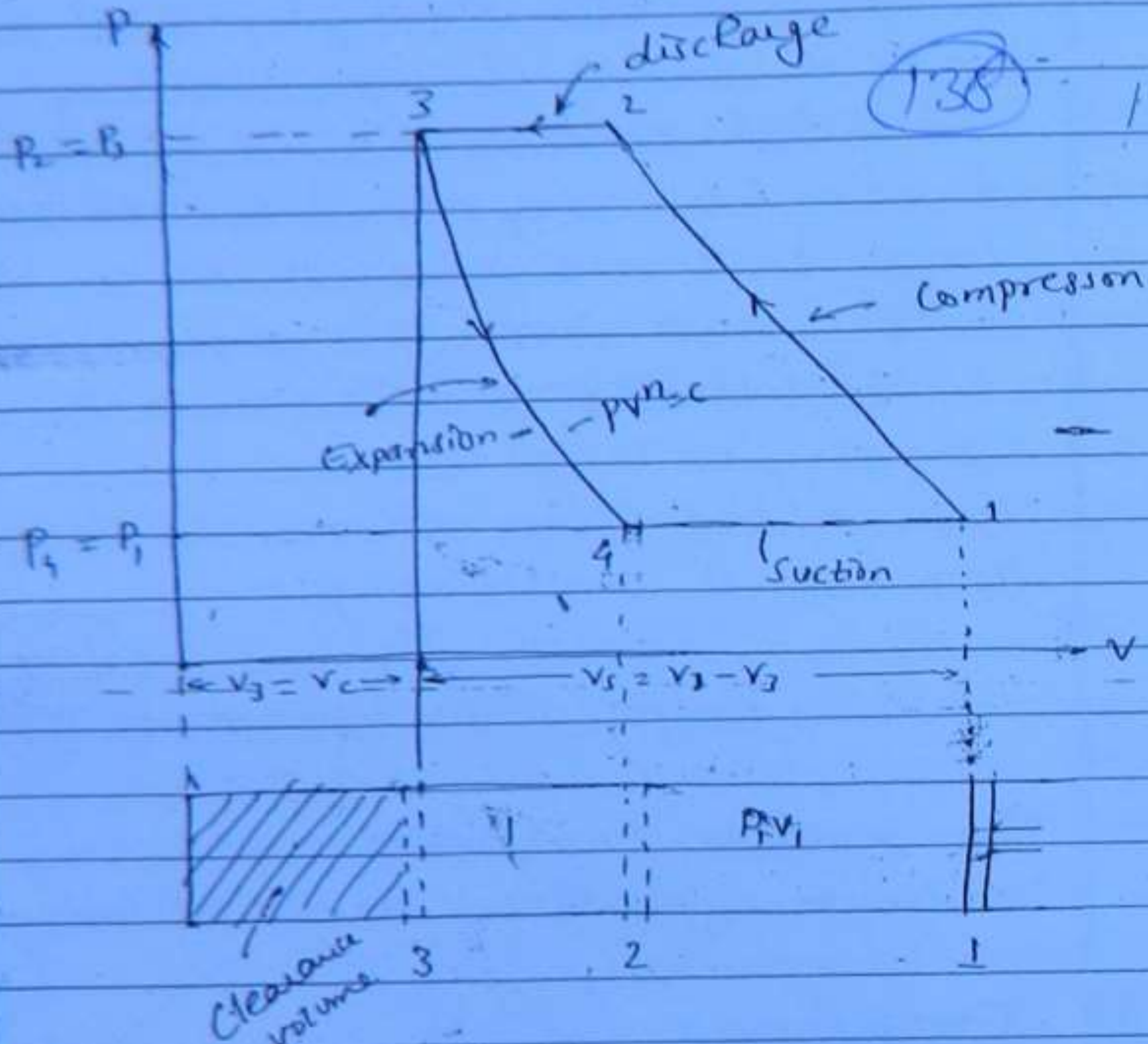
Function's of Compressor.

1. To draw refrigerant from evaporator.

2. To compress the refrigerant

3. To discharge high pressure refrigerant to Condensor.

* Volumetric efficiency of a Reciprocating compressor :-



$$\eta_{vol} = \frac{V_{act}}{V_{swp}} = \frac{V_1 - V_4}{V_1 - V_3}$$

$$C = \frac{V_c}{V_s} = \frac{V_3}{V_1 - V_3}$$

$$\eta_{vol} = \frac{V_1 - V_4 + V_3 - V_3}{V_1 - V_3}$$

$$C = \frac{V_3}{V_1 - V_3}$$

$$\eta_{vol} = \frac{V_1 - V_3 - (V_4 - V_3)}{V_1 - V_3}$$

$$\eta_{vol} = 1 - \frac{(V_4 - V_3)}{V_1 - V_3}$$

$$\eta_{vol} = 1 - \frac{V_3 \left(V_4/V_3 - 1 \right)}{V_1 - V_3}$$

$$\eta_{vol} = 1 - C\left(\frac{V_4}{V_3} - 1 \right)$$

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$$\eta_{vol} = 1 + C - C\left(\frac{V_4}{V_3} \right)$$

Expansion is taken as polytropic.

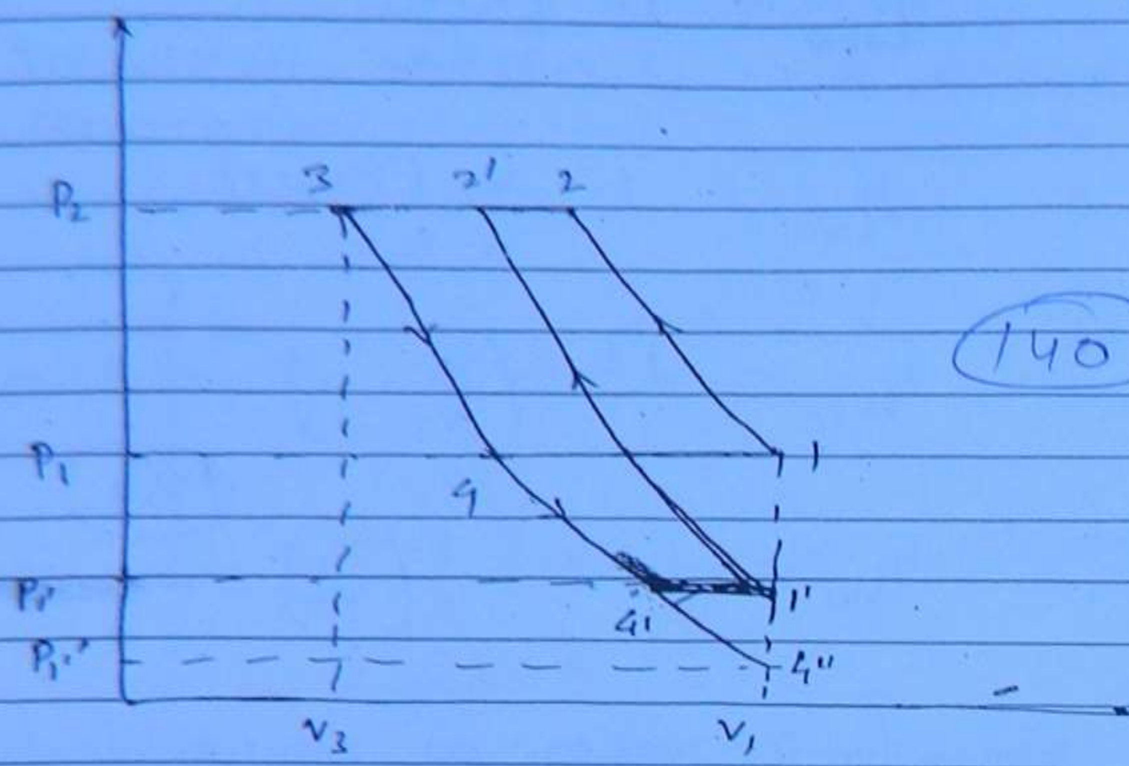
$$P_3 V_3^n = P_4 V_4^n$$

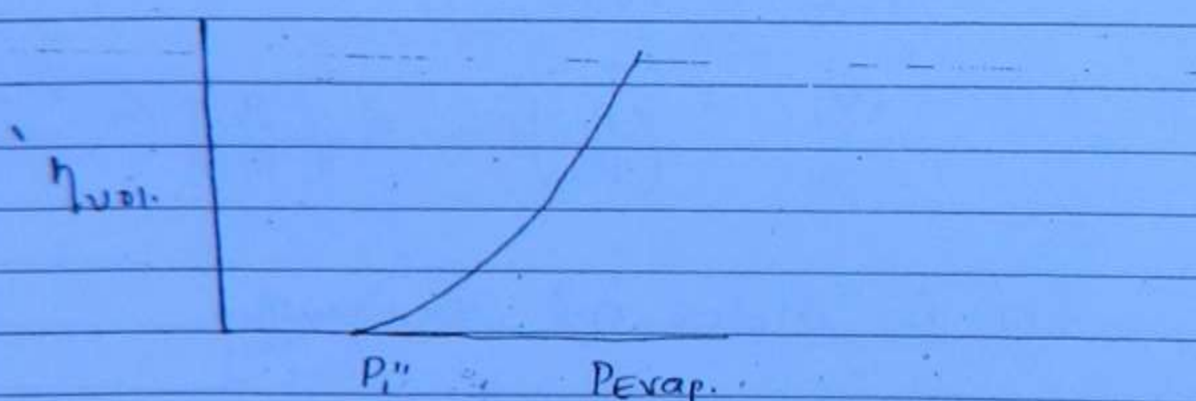
$$\frac{V_4}{V_3} = \left(\frac{P_3}{P_4} \right)^{1/n} = \left(\frac{P_2}{P_1} \right)^{1/n}$$

n is index of expansion.

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

☆ Effect of variation of pressure ratio on volumetric efficiency :-



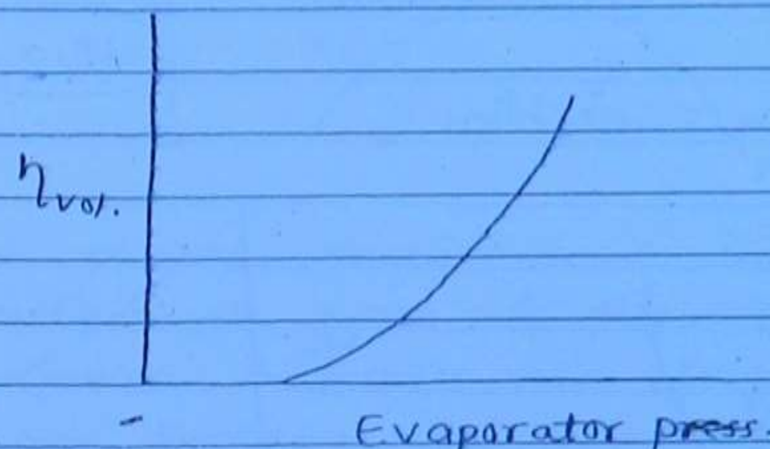


$$\eta_{vol} = 1 + C - C\left(\frac{P_2}{P_1}\right)^{1/n}$$

With decrease in evaporator pressure the volumetric efficiency decreases because the actual section volume decreases & at one perticular very low evaporator pressure (P_1'') the volumetric efficiency became zero.

Performance parameters :

1. Volumetric Efficiency v/s Evaporator pressure.



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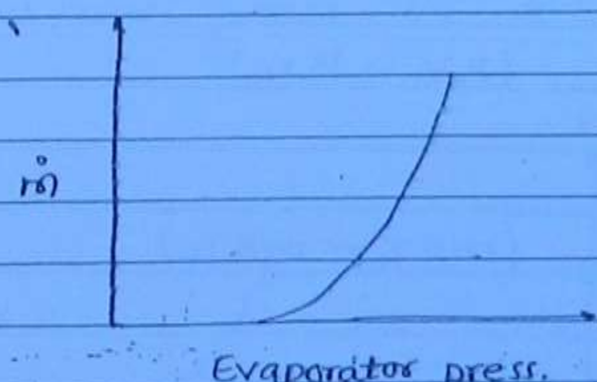
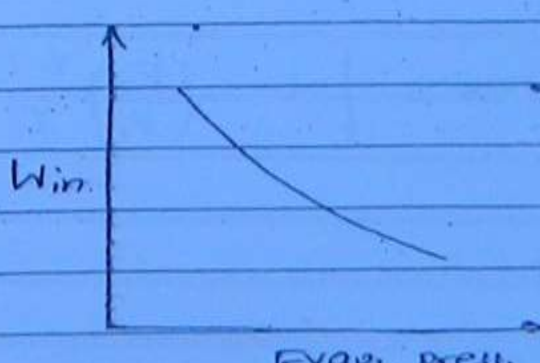
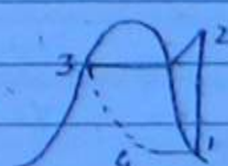
2. Evaporator pressure v/s mass flow rate.



3. Evaporator press. v/s work input.





$W_{in} = h_2 - h_1$

4. Evaporator press. v/s power input.

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

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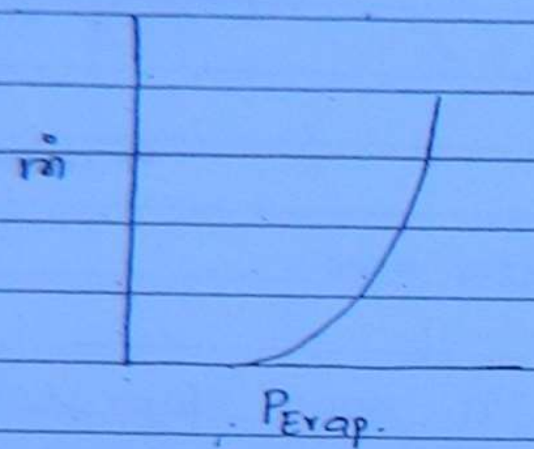
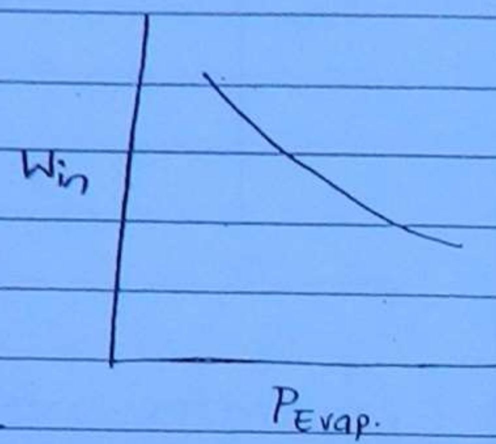
$$P_{in} = \dot{m}\,W_{in}$$



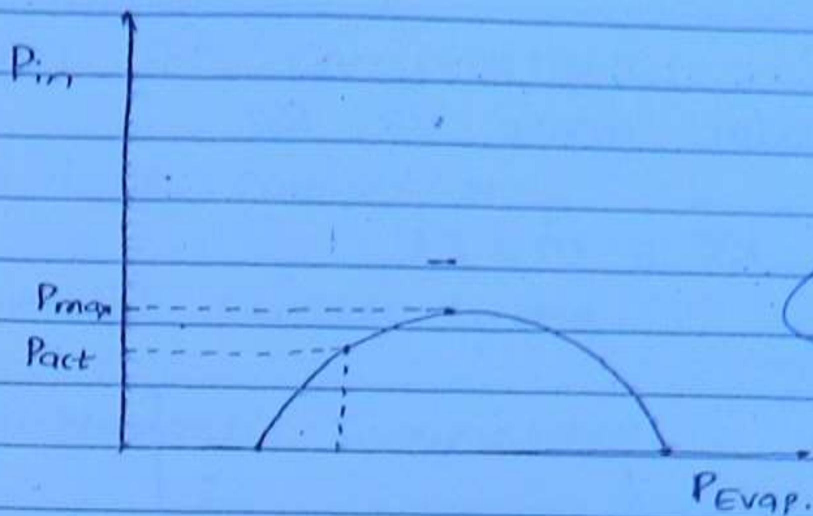


$$W = \frac{\gamma}{\gamma - 1}\,(P_1V_1 - P_2V_2)$$

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

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

$$= \frac{\gamma}{\gamma - 1}\,mRT_1\left[1 - \left(\frac{P_2}{P_1}\right)^{\frac{\gamma - 1}{\gamma}}\right]$$



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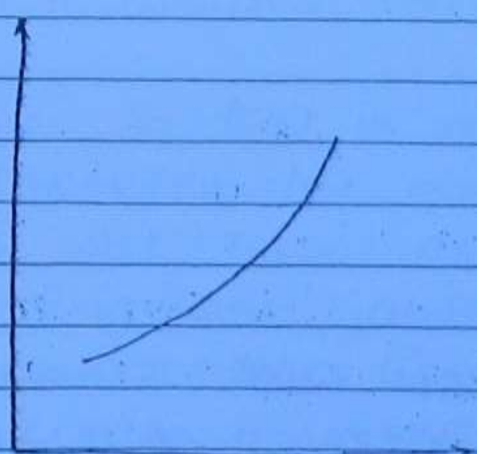
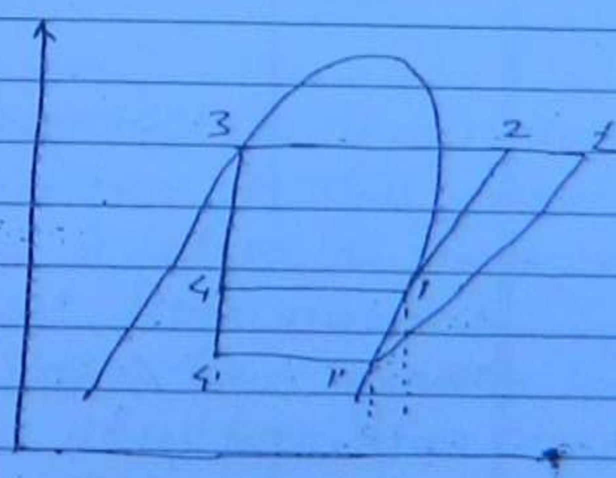
The power requirement initially increases. reaches it's peak & then it decreases. Compressor motor's are design for peak power because to reach the operating condition it should pass through peak power

5.4 Evaporator press. v/s R.E :-





6. Evaporator press. V/s RC.

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

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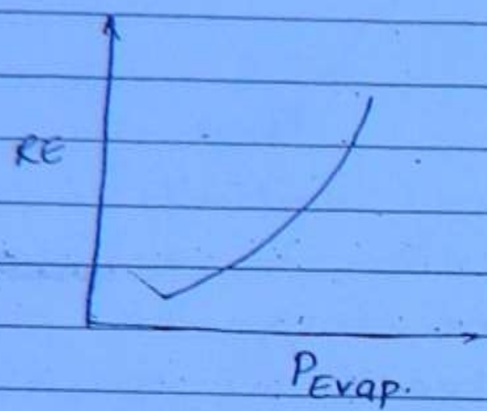

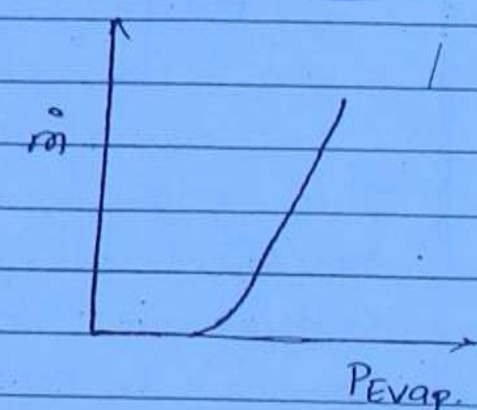

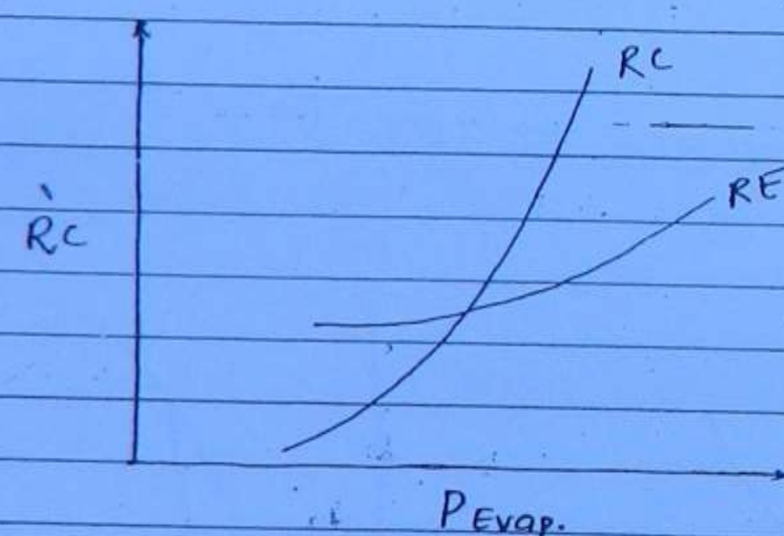

Though RE & RC increase with increase in Evaporator press. RC increases at a faster rate.

Type's of Compressor's :-

1. Open type compressor.

2. Hermetic Compressor's.

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3. Semi-Hermetic Compressor's.

In open type compressor's the compressor run's with the help of motor by mean's 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.

Hermitcally sealed compressor are used in domestic refrigerator & window A/c

DATE: / /

2. Condensor's & Evaporator :-

Both are heat Exchanger's in condensor's refrigerant reject heat to ~~s~~surrounding & in Evaporator it absorb's heat from ~~E~~ storage space.

Type's of Condensor's :- (146)

1. Air Cooled.

2. water cooled.

3. Evaporative Condensors

NOTE- Capacity of domestic Refrigerator is 0.1 to 0.25 TR

Air cooled Condensor's are used for small capacity plant's (Upto 3TR Capacity).

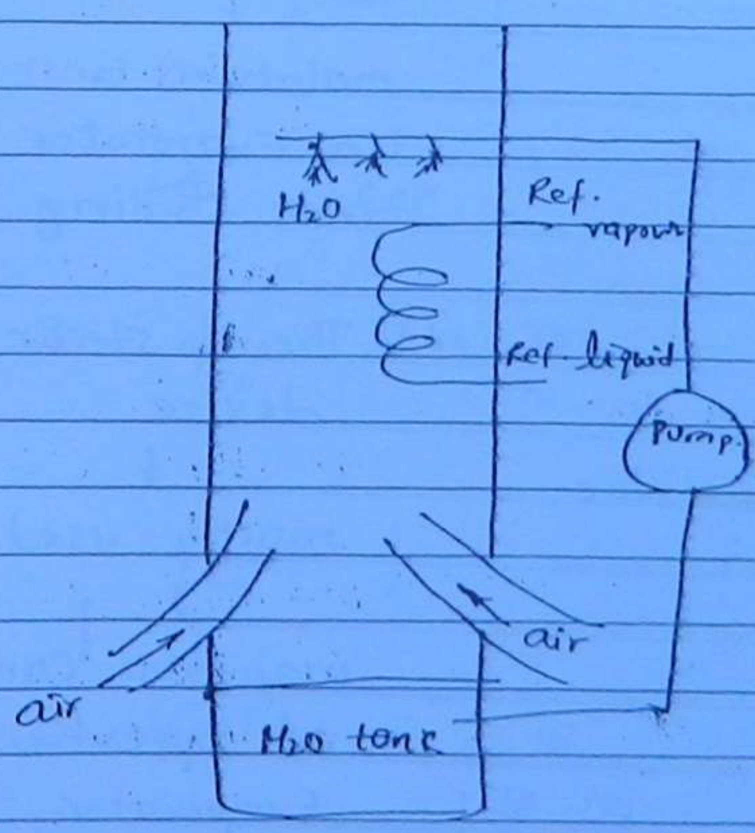
In water cooled condensor water is the cooling medium & it is used for upto 1000 TR capacity.

In shell & tube type condensor's refrigerant flow's in shell & water flow's in tube's if refrigerant flow's 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 condensors:- (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 tonne then reject heat to air and thus same water is recirculated again and again.



Imp.
* Plate type of Evaporator are used in domestic Refrigerator's.

Imp.
* Direct contact ~~typ~~ coil type Evaporator are used in air conditioner's.

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3. <u>Expansion Device's</u> :-

Const. area type
↓
Capillary ~~type~~ tube
↓
Domestic ref.
window A/C
water cooled
↓
$\Delta P \propto \dfrac{l}{d}$

pressure occure due to (i) friction
(ii) acceleration

Variable area type.

① Automatic Exp. device
↓
maintain's const. press. in Evaporator used in milk chilling center's.

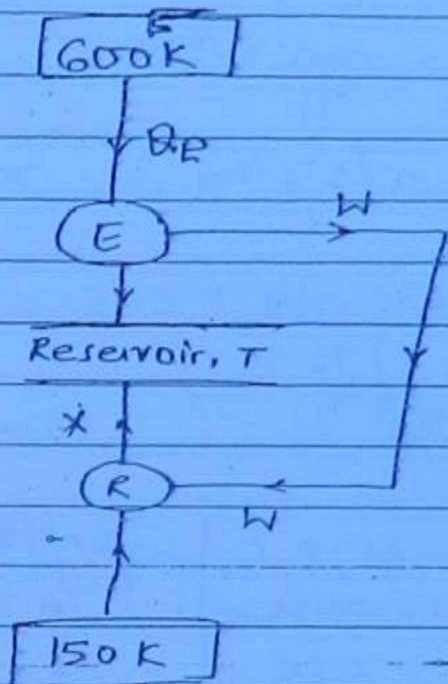
② Thermostatic Expansion device
↓
mostly used
↓
maintain's const. degree of superheat in Evaporator.

WORK-BOOK

CHAPTER- I

1. (b)

2. (a)



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$$\eta_E = \frac{1}{COP} \qquad \eta_E = \frac{T_R - T_l}{T_R}$$

$$\eta_E = \frac{600 - 300}{600} = 0.5 = \frac{1}{2}$$

$$\eta_E = \frac{1}{2}$$

$$(COP)_{HP} = \frac{T_R}{T_R - T_l} = \frac{300}{300-150} = 2$$

$$(COP)_{Ref.} = \frac{T_l}{T_R - T_l} = \frac{150}{300-150} = 1$$

3. (a) All though heat pump & Refrigerator

4. (a) $COP = \frac{T_l}{T_h - T_l}$ $\quad T_h = T_2$ & $T_l = T_1$

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$$COP = \frac{T_1}{T_2 - T_1} = \frac{\cancel{T_1}}{\cancel{T_1}\left(\frac{T_2}{T_1} - 1\right)}$$

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

5. (b)

ice	comfort
assume $T_h = 40^\circ C = 313\,K$	$T_h = 313\,K$
$T_l = 0^\circ C = 273\,K$	$T_l = 20^\circ C = 293\,K$
$(COP)_{ice} = \frac{T_l}{T_h - T_l}$	$(COP)_{comfort} = \frac{T_l}{T_h - T_l}$
$(COP)_{ice} = max$	$COP/comfort = less$

6. (c) Carnot refrigerator depend's only temp. limit.

7. (d) If nothing is given then take reversible

8. (c) 9. (c) 10. (a)

11. (c) $1\ TR = 3.5\ KW = 3.5\ kJ/s$.

$t_i = 35^\circ$, $t_f = 20^\circ c$

$C = 4.18$

$$Q = mc\,(\Delta t)$$

$$3.5 = m \times 4.18\,(15)$$

$$m = 0.055 \text{ Kg/s.}$$

(157)

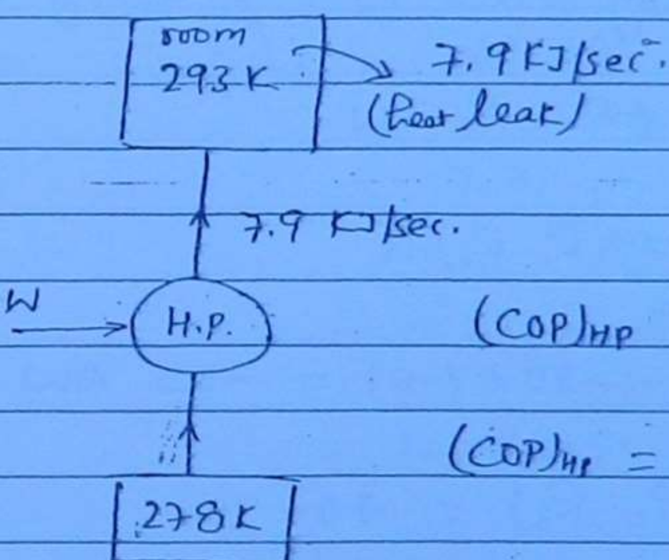
$$m = 0.055 \times 3600 \text{ Kg/hr.}$$

$$m = 200.9 \text{ Kg/hr}$$ $\rho_{water} = 1000 \text{ Kg/m}^3$

$$m = 200.9 \text{ liter/hr.}$$

12. ~~(b)~~ 13. (c) 14. (d)

15. (b)



$$(COP)_{HP} = \frac{T_H}{T_H - T_L} = \frac{293}{293-278}$$

$$(COP)_{HP} = \frac{7.9}{W}$$

$$\frac{7.9}{W} = \frac{293}{293-278}$$

$$W = 0.405 \text{ KJ/sec.}$$

$$= 0.405 \text{ KW}$$

$$= 405 \text{ W.}$$

(16) (d) (17) (d) 18. (b) 19. (c)

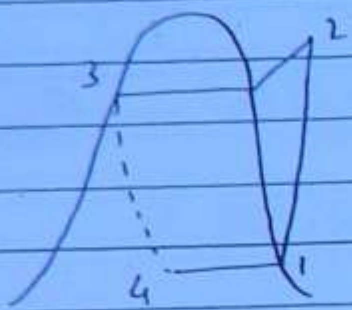
20. (d)

CHAPTER-2

1. (c) 2. (c) 3. (a) 4. (c)

5. (b) 6. (c) 7. (c) (152)

8. (c)



$$h_1 - h_4 = h_1 - h_3 = 70$$

$\&$ $h_2 - h_3 = 90$

$Q_{comp} = 5$

$$h_1 + \cancel{\frac{c_1^2}{2}} + \cancel{z_1 g} + Q = h_2 + \cancel{\frac{c_2^2}{2}} + \cancel{z_2 g} + W$$

$$h_1 + Q = h_2 + W$$

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

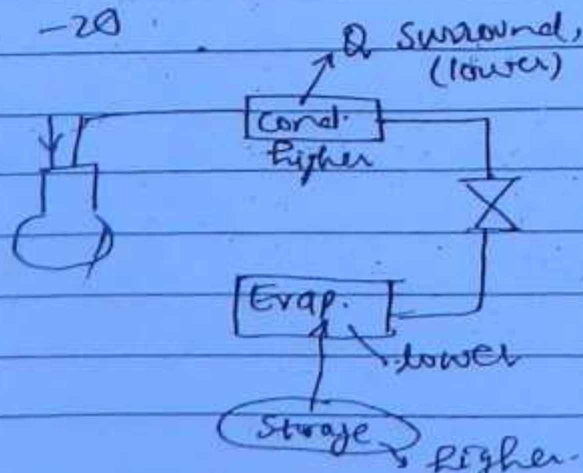
$$W = -20 + (-5) = -25 \text{ Ans.}$$

$$(h_1 - h_3) - (h_2 - h_3) = 70 - 90$$

$$h_1 - \cancel{h_3} - h_2 + \cancel{h_3} = -20$$

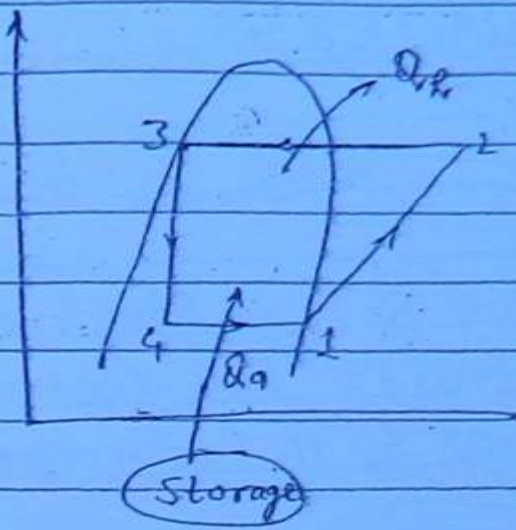
$$h_1 - h_2 = -20$$

9. (b) 10. (c)



11. (d)

11. (d)



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$h_1 = 195$, $h_2 = 210$, $h_3 = h_4 = 90$

$m = 0.5$

- In heat pump. –

heat supplied $= h_2 - h_3$

$= 210 - 90$

$= 120 \text{ KJ/kg}$

$= \dot{m} \times (h_2 - h_3)$

$= 120 \times 0.5 = 60 \text{ KW}$ Ans.

12. (a) & (b) 13. (d)

CHAPTER - 3

1. (c) 2. (d) 3. (b) 4. (d) 154

NOTE - Electrolux Refrigerator work's on three fluid's NH_3 (Refrigerant), water (Absorbent) & H_2 to create low NH_3 pressure's.

This system work's on dalton's law of ~~part~~ partial press. in this system the total press. is kept as 14 bar & in Evaporator hydrogen is maintain at a pressure at 12 bar once NH_3 enter's in Evaporator from condensor it's press. drop to 2 bar. i.e. in electrolux refrigerator no expansion device is used.

In electrolux refrigerator no pump is used because the fluid flow's by gravity.

5. (d) 6. (a) Former → V.A. System.

latter → V.C. System

7. (d) 8. (c) 9. (b) 10. (c)

11. (d) 12. (a)