HANDWRITTEN NOTES OF

(REFRIGERATION AND AIR CONDITION)

BY

ENGGBUZZ.COM

Refrigeration And Air Conditioning

1) Basic Concept (Obj)

Imp 2) Vapour Compression Refrigeration System (Conv.+Obj)

3) Refrigerant (Theory Obj)

4) Vapour Absorption Refrigeration System (Obj)

Imp 5) Psychrometry (Obj)

6) Summer & Winter A/C (Conv.+Obj)

7) Duct Design (Obj)

Basic Concepts

Refrigeration Effect :-

It is Amount of heat, which is Required to extract in order to provide and maintain Lower temperature than that of Surroundings.

Refrigerant :-

It is Working fluid or Working Substance, that is used to extract the heat from the storage space/system.

C.O.P or E.P.R :-

Coefficient of Performance or Energy Performance Rate :-

It is Ratio of Desired effect to the Work input.

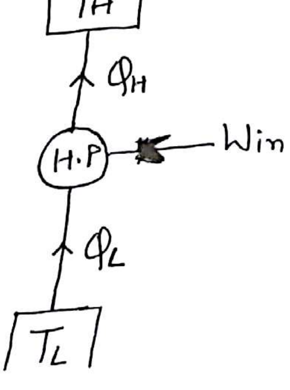
or

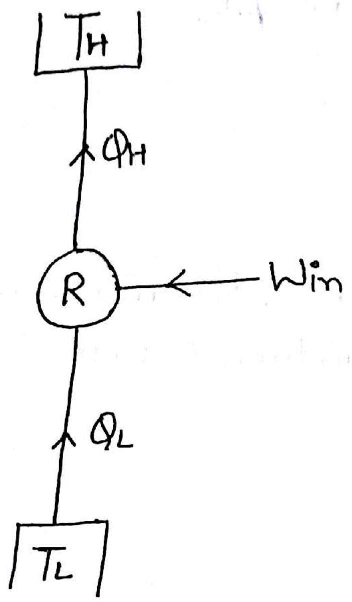
It is defined as Ratio of Refrigeration effect to the Work input.

$$C.O.P = \frac{\text{Desired Effect}}{W_{in}} = \frac{T_H}{T_H - T_L} = \frac{T_L}{T_H - T_L}$$

$\frac{T_H}{T_H - T_L}$ → Heat Pump (D.E. ↑ Temp)

$\frac{T_L}{T_H - T_L}$ → Refrigeration (D.E. ↓ Temp)





(Refrigerator)

Objective

$$(C.O.P)_R = \frac{T_L}{T_H - T_L}$$

gmp

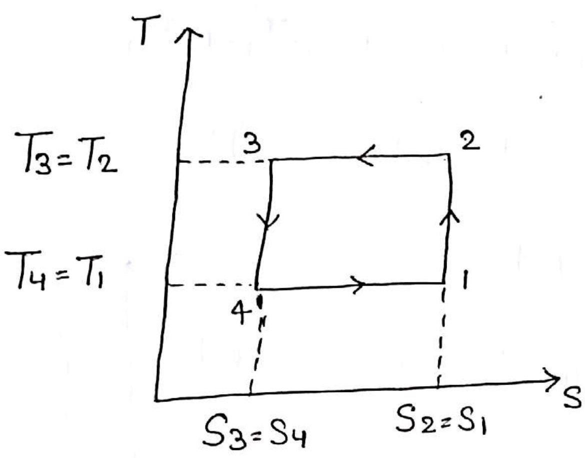
$$(C.O.P.)_{HP} = (C.O.P)_R + 1 = \frac{1}{\eta_E}$$

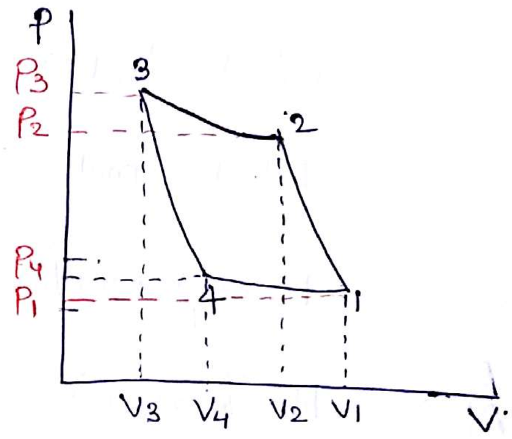
This Relation or expression is applicable b/w the "Same Temperature" limits.

Ideal Refrigeration Cycle :-

or

Reversed Carnot Cycle :-





$$dS = \cancel{\dot{S}_{gen}}^{\,0} + \frac{dQ}{T}$$

$$\boxed{dQ = TdS}$$

Process(1-2) → $dQ = TdS = 0$

Process(2-3) → $dQ = T(S_3 - S_2) = -ve$

Process (1-2) → Reversible Adiabatic or Isentropic Compression.

Process (2-3) → Heat Rejection at Constant Temperature.

Process (3-4) → Isentropic Expansion.

Process (4-1) → Heat Supplied at Constant Temperature

$$C.O.P = \frac{\text{Desired Effect}}{W_{net}}$$

(1-2)	Isentropic Compression
(2-3)	Isothermal Compression
(3-4)	Isentropic Expansion
(4-1)	Isothermal expansion

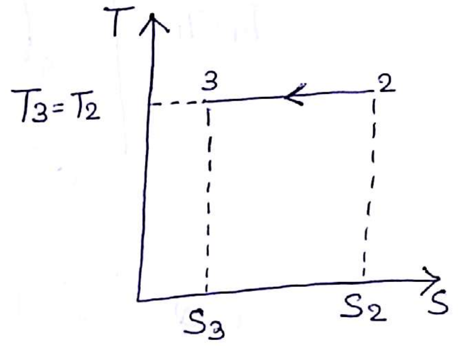
$$C.O.P = \frac{\text{Desired Effect}}{W_{net}}$$

$W_{net} = Q_{net}$ → from I[st] Law of Thermodynamics

$$W_{net} = Q_{net} \Rightarrow \cancel{Q_{1-2}}^{\,0} + Q_{2-3} + \cancel{Q_{3-4}}^{\,0} + Q_{4-1}$$

$$W_{net} = Q_{net} = Q_{2-3} + Q_{4-1} \quad \text{—} \ (1)$$

$$dS = \frac{dQ}{T} \Rightarrow dQ = T\,dS.$$

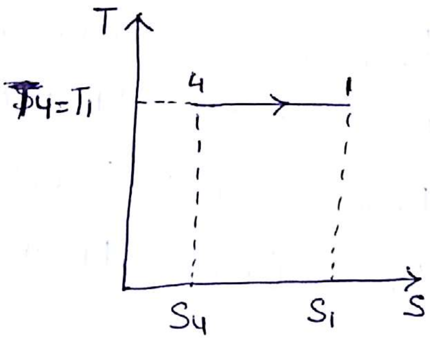


$$dQ = T_H(S_3 - S_2) = -T_H(S_2 - S_3)$$

$$dQ_{(2-3)} = -T_H(S_1 - S_4) \quad \text{—} \ (2)$$

$$dQ_{(4-1)} = T(S_F - S_I) = T_L(S_1 - S_4) \quad \text{—} \ (3)$$

using (2) & (3) in (1)



$$W_{net} = Q_{net} \Rightarrow Q_{2-3} + Q_{4-1}$$

$$-T_H(S_1 - S_4) + T_L(S_1 - S_4)$$

$$= (T_L - T_H)(S_1 - S_4)$$

$$\boxed{W_{net} = -ve}$$

As the Value of Network output is having negative expression, therefore our Assumed System is Work Absorbing device.

$$C.O.P = \frac{\text{Desired Effect}}{W_{in}} = \frac{D.E.}{(T_H - T_L)(S_1 - S_4)} \longrightarrow \text{Process (1-4)}$$

$$C.O.P = \frac{T_L (S_1 - S_4)}{(T_H - T_L)(S_1 - S_4)}$$

IRC → Ideal Refrigeration Cycle.
RCC → Reversed Carnot Cycle.

Objective

$$(C.O.P)_{IRC / RCC} = \frac{T_L}{(T_H - T_L)}$$

If Heat is Rejects by any System, then it must be gain by other System and the System which gains this Rejected Heat, then its entropy must be increase.

NOTE :-

1) Reversed Carnot Cycle (C.O.P) as a function of temperature limits only.
2) If there are "n" no. of Reversible Refrigerator, operating b/w Same Temp. limits, with different working fluids or Refrigerant then the Value of max. Possible (COP) or Reversed Carnot (COP) or Ideal (C.O.P) is having Same Value.
3) Reversed Carnot C.O.P is Independent of Working fluid.

Reversed Carnot — function of — temp. limits
└→ Independent of Working fluid

Unit Of Refrigeration :-

1 Tonne Of Refrigeration

$$1\ T.R = 3.5\ KW = 210\ \frac{KJ}{min} = 50\ \frac{KCal}{min}$$

It is the amount of Heat, Which is Required to extract from 1000 Kg of Water at 0°C in Order to convert it into equivalent ice at 0°C in 24 hours/day.

NOTE :-

Qn) Producing ice at 0°C

a) $(COP)_S > (COP)_W$
b) ✓ $(COP)_S < (COP)_W$
c) $(COP)_S = (COP)_W$
d) Based on given Data

$$\downarrow COP = \frac{T_L}{(\uparrow T_H - T_L)\uparrow}$$

if We ↑ T_H then $(T_H - T_L)$ Will ↑
Such that, T_L = Constant, $T_H \uparrow$] COP↓

Summer	Winter
$T_L = 0°C$	$T_L = 0°C$
$T_H = 30°C$	$T_H = 10°C$

T_L = Constant

$$(T_H)_S > (T_H)_W$$

$$\boxed{(COP)_S < (COP)_W}$$

Bcz.

$T_H \downarrow$ — COP ↑
$T_L \uparrow$ — COP ↑

Pure Substance :-

1) $h_B = h_f$

2) $h_D = h_g$

3) $h_C = h_f + x h_{fg}$ $(h_g - h_f)$

4) $h_E = h_g + (C_p)_{vap}[T_E - T_D]$

5) $h_A = h_f - (C_p)_{liq}(T_B - T_A)$

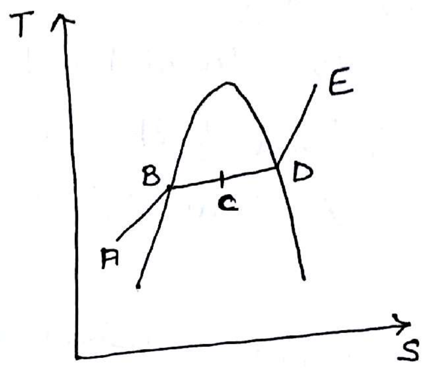
1) $S_B = S_f$

2) $S_D = S_g$

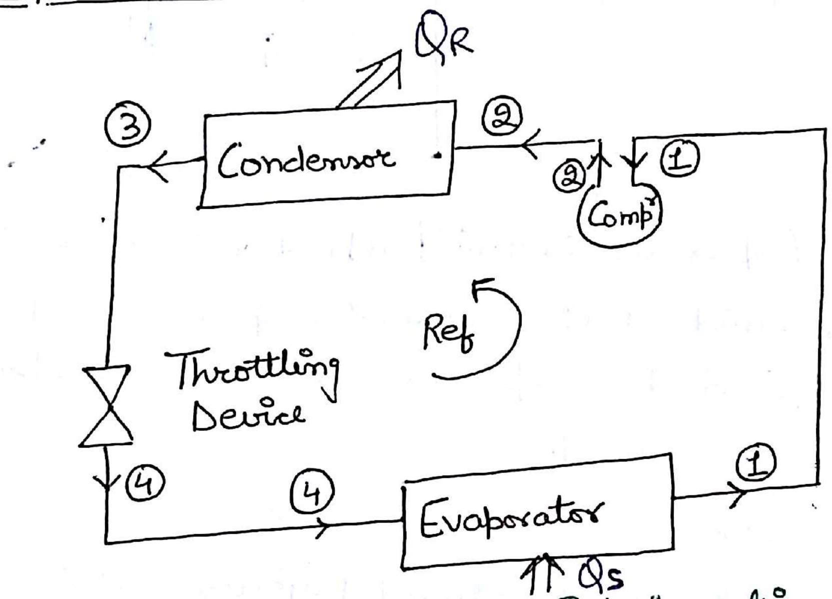
3) $S_C = S_f + x(S_g - S_f)$

4) $S_E = S_g + (C_p)_{vap} \ln\left(\frac{T_E}{T_D}\right)$

5) $S_A = S_f - (C_p)_{liq} \ln\left(\frac{T_B}{T_A}\right)$



Vapour Compression Refrigeration System :- VCRS



Vapour Compression Refrigeration System

Process (1-2) → Isentropic / Reversible Adiabatic Compression

Process (2-3) → Constant Pressure heat Rejection

Process (3-4) → Constant Enthalpy Expansion

Process (4-1) → Constant Pressure heat Addition

Assumption :-

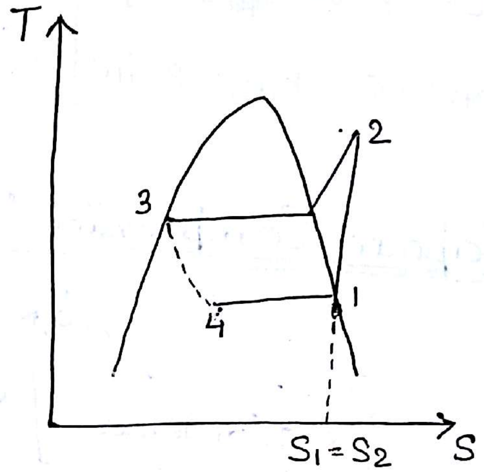
1) Entry to the Compressor and exit of evaporator (State 1) is Saturated Vapour.

2) Exit of the Condensor & entry of throttling (State 3) is Saturated liquid

$$C.O.P = \frac{\text{Desired Effect}}{W_{in}}$$

$$\boxed{C.O.P = \frac{h_1 - h_4}{h_2 - h_1}}$$

Compressor is Work producing



Qn) Why Isentropic Expansion is not prefreable in VCRS?

The State of Working fluid, at the entry of expandor (evaporator) is Saturated liquid, and then expression of Work is given by,

$$W = -\int V_f \, dP$$

$$V_g >>>> V_f$$

Where V_f is Specific volume of Saturated liquid, which is negligible in Comparison to the V_g i.e Specific volume of Saturated Vapour, handelled by the Compressor.

So, the expansion Work is negligible in Comparison to the Compression Work, therefore the use of isentropic expansion will not justify the Cost of expandor.

NOTE:-

1) Refrigeration Effect (R.E) → $(h_1 - h_4)$ $\frac{KJ}{Kg}$

2) Work input (W_{in}) → $(h_2 - h_1)$ $\frac{KJ}{Kg}$

3) Refrigeration Capacity (R.C) → $\dot{m} \times R.E$ (KW)

4) Power input (P_{in}) → $\dot{m} \times W_{in}$ (KW)

$R.E = h_1 - h_4$ KJ/Kg

$W_{in} = h_2 - h_1$ KJ/Kg

$R.C = \dot{m} \times (h_1 - h_4) = \dot{m} \times R.E$] KW

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

Volumetric Efficiency of Resiprocating Compressor:

It is defined as the Ratio of "Actual volume at the entry of Compressor to the theoritical Swept volume."

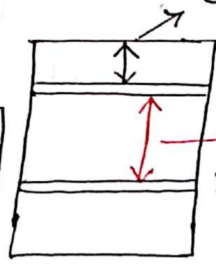
Specific volume → $v = \frac{vol}{mass}$

$\dot{vol} = \dot{m} \times v_{entry}$

Objective

$$\eta_v = \frac{\dot{m}\, v_{entry}}{\frac{\pi}{4} D^2 L \times \frac{N}{60} \times K}$$

$$\eta_v = \frac{Act.\,Vol_{entry}}{Swept\,vol.}$$



Where,

$\dot{m}$ → mass flow rate Kg/sec

v → Specific volume at entry of Compressor m^3/Kg

D → Bore / Diameter (m)

L → Stroke Length (m)

N → Speed (rpm)

K → No. Of Cylinders

$\dot{m}$ → Kg/sec
v → m^3/Kg
D → m
L → m
N → rpm
K → No. of Cylinders

Objective

$$\eta_v = \frac{(\dot{m}\,v)_{entry}}{\frac{\pi}{4} D^2 L \times \frac{N}{60} \times K}$$

Volumetric Efficiency

NOTE :-

1) Volumetric Efficiency is also Calculated by the expression,

$$\eta_v = 1 + C - C\left[\frac{P_{higher}}{P_{Lower}}\right]^{\frac{1}{n}}$$

for RAC

$$\eta_v = 1 + C - C\left[\frac{P_{cond}}{P_{eva}}\right]^{\frac{1}{n}}$$

Where $n \rightarrow$ Polytropic Index

$C \rightarrow$ Clearnac Ratio

↓

It is defined as the Ratio of Clearnac volume to the theoritical Swept volume.

$$C = \frac{V_c}{V_s}$$

Objective

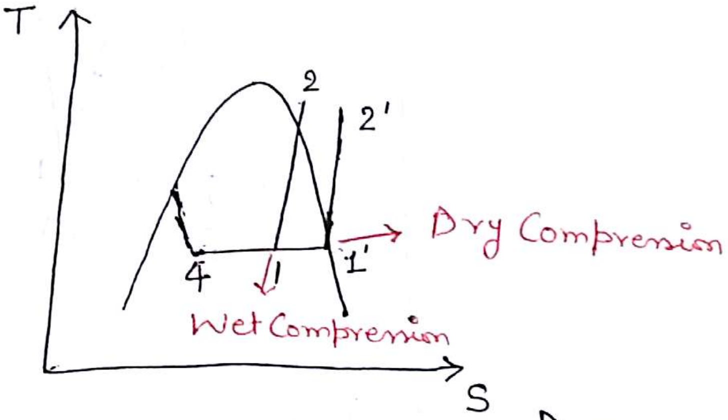
$$\eta_v = \frac{(\dot{m}\,v)_{entry}}{\frac{\pi}{4} D^2 L \times \frac{N}{60} \times K}$$

$$\eta_v = 1 + C - C\left[\frac{P_c}{P_E}\right]^{\frac{1}{n}}$$

Dry Compression v/s Wet Compression :-

Dry Compression means entering point to the Compressor is from Saturated Vapour.

Wet Compression means entering Point to the Compressor is from Wet region (Liquid + vapour).



Disadvantage of Wet Compression over Dry Compression :-

1) Refrigeration Effect (R.E.) ↓.

$$(R.E)_{Dry} = h_{1'} - h_4$$

$$(R.E)_{Wet} = h_1 - h_4$$

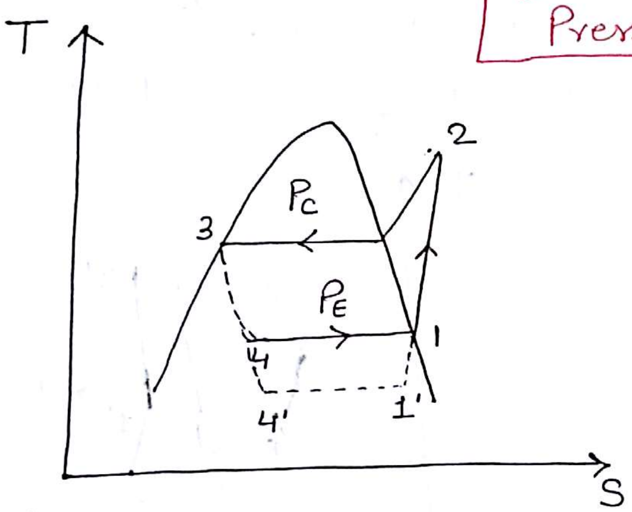
2) The Liquid particle, which is present in the mixture of Refrigerant, may Wash away the lubricant and it ↑ the Chances of Wear & tear & it also damage the Compressor valve & its Body.

3) Wet Compression Represents individually, the incomplete evaporation of Refrigerant.

Effect of Variation in Parameters on Performance of Vapour Compression Refrigeration System (VCRS) :-

1) Case ① → Decrease in evaporator Pressure :-

"Decrease in evaporator Pressure"



Process → 1 – 2 – 3 – 4 – 1

1) $R.E = h_1 - h_4$

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

3) $C.O.P = \frac{R.E}{W_{in}}$

4) $\frac{P_C}{P_E}$

Process → 1' – 2 – 3 – 4' – 1'

1) $R.E\downarrow = h_1' - h_4'$

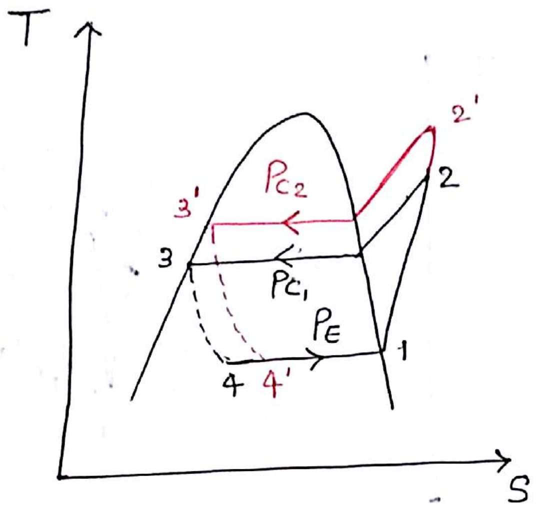
2) $\uparrow W_{in} = h_2 - h_1'$

3) $\downarrow C.O.P = \frac{R.E\downarrow}{W_{in}\uparrow}$

4) $\boxed{\frac{P_C}{P_E\downarrow}}\uparrow$, $\eta_v\downarrow$

$$\eta_v = \left[1 + C - C\left(\frac{P_C}{P_E}\right)^{\frac{1}{n}}\uparrow\downarrow\right]$$

2) Case 2:- Effect of ↑ in Condensor Pressure :-

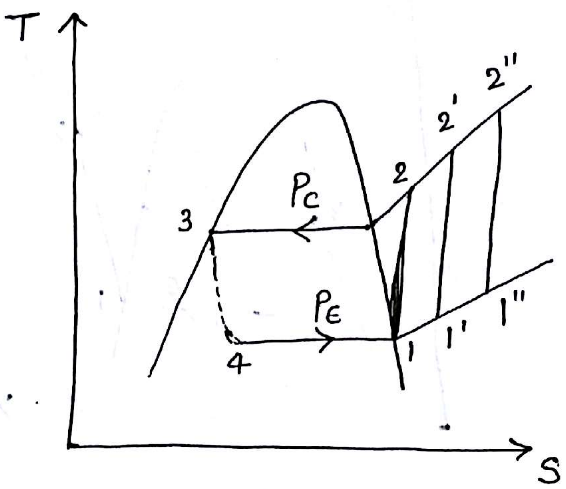


Process → 1-2-3-4-1	Process → 1-2'-3'-4'-1
1) $R.E = h_1 - h_4$	1) $\downarrow R.E = h_1 - h_4'$
2) $W_{in} = h_2 - h_1$	2) $\uparrow W_{in} = h_2' - h_1$
3) $C.O.P = \frac{R.E}{W_{in}}$	3) $\downarrow C.O.P = \frac{R.E\downarrow}{W_{in}\uparrow}$
4) $\frac{P_{C_1}}{P_E}$	4) $\left(\frac{P_{C_2}}{P_E}\right)\uparrow$, $\eta_v \downarrow$ $\rightarrow \eta_v = \left[1 + C - C\left(\frac{P_C}{P_E}\right)^{\frac{1}{n}}\right]$

NOTE :-

1) Effect of ↑ in Condensor pressure and ↓ in evaporator Pressure are exactly Same.

3) Case-③ → Superheating (Within the Evaporator) :-



Process → 1-2-3-4-1

1) $R.E = h_1 - h_4$
2) $W_{in} = h_2 - h_1$
3) $C.O.P = \frac{R.E}{W_{in}}$

Process → 1'-2'-3-4-1'

1) $\uparrow R.E = h_{1'} - h_4$
2) $\uparrow W_{in} = h_{2'} - h_{1'}$
3) $C.O.P = \frac{R.E \uparrow}{W_{in} \uparrow}$

Prove that Win is function of Tinlet

$$W_{in} = n \, W_{closed}$$

$$W_{in} = n \left[\frac{P_I V_I - P_F V_F}{n-1}\right]$$

$$= \frac{n}{n-1}\left[P_1 V_1 - P_2 V_2\right]$$

$PV = mRT$

$$= \frac{n}{n-1}\left(mRT_1 - mRT_2\right)$$

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

$$= \frac{n}{n-1} mRT_1\left(1 - \frac{T_2}{T_1}\right) \quad \text{—— ①}$$

Process 1-2 Isentropic / Reversible Adiabatic Compression

$$PV^n = C \quad \text{(Polytropic)}$$

$$\frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{\left(\frac{n-1}{n}\right)} = \left(\frac{P_C}{P_E}\right)^{\frac{n-1}{n}} \quad \text{—— (2)}$$

$$\left.\begin{aligned} P_3 &= P_2' = P_2 = P_C \\ P_4 &= P_1 = P_1' = P_E \end{aligned}\right] \text{—— (3)}$$

$$W_{in} = \frac{n}{n-1}\, m R T_1 \left[1 - \left(\frac{P_C}{P_E}\right)^{\frac{n-1}{n}}\right]$$

$$\boxed{W_{in} = f(T_{inlet})}$$

Effects of Superheating :-

1) Refrigeration Effect (R.E) ↑, if Superheating occurs in Evaporator.

2) Win. in the Compressor ↑, because it is ~~in~~ the function Of inlet temperature to the Compressor. (function Of T_{inlet})

3) (C.O.P) may be ↑ or ↓, dependending on Refrigerant.

for example:-

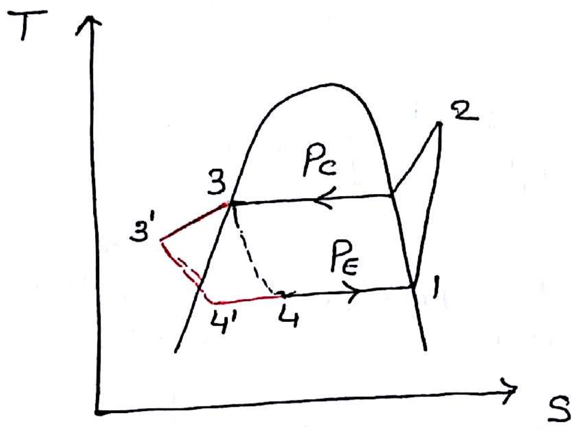
R-12 Refrigerant ↑ (COP)
(Superheating Would Result & ↑ in COP)

NH_3 Ammonia Refrigerant ↓ (COP)
(Superheating Would Result in ↓ in C.O.P)

R12 ↑ COP
NH_3 ↓ COP

4) Sub Cooling :-

It is the Process of ↓ the temperature at Constant Pressure below Saturated Liquid.



Process → 1-2-3-4-1

1) $R.E = h_1 - h_4$

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

3) $C.O.P = \frac{R.E}{W_{in}}$

Process → 1-2-3'-4'-1

1) $\uparrow R.E = h_1 - h_4'$

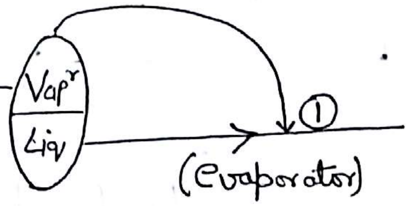
2) $W_{in} = h_2 - h_1$ (Const)

3) $\uparrow C.O.P = \frac{R.E \uparrow}{W_{in} \text{ (Const)}}$

Use Of Flash Chamber in VCRS :-

Flash Chamber is a device, Which is used to Seprate liquid Refrigerant from the Vapour, at the entry of evaporator and, it allows only the Liquid refrigerant to enter into evaporator Which results in absorption of heat.

By the use of Flash chamber Size of evaporator Reduce Which is good for industrial Purpose.



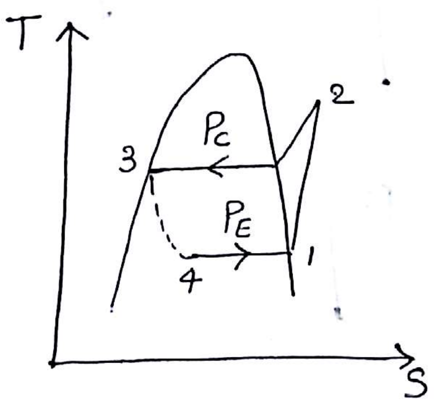
Because density of Water is ↑ and density of Vapour is ↓ low.

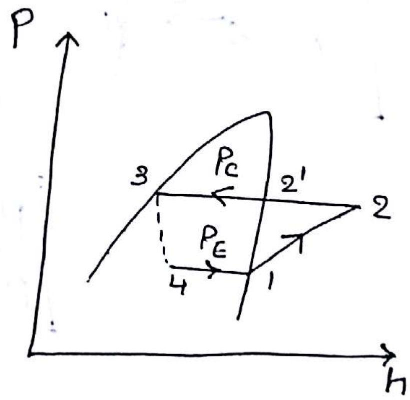
NOTE :-

1) There is no impact on COP With the use of flash Chamber.

Imp 2) Flash Chamber helps in reducing the Size of evaporator.

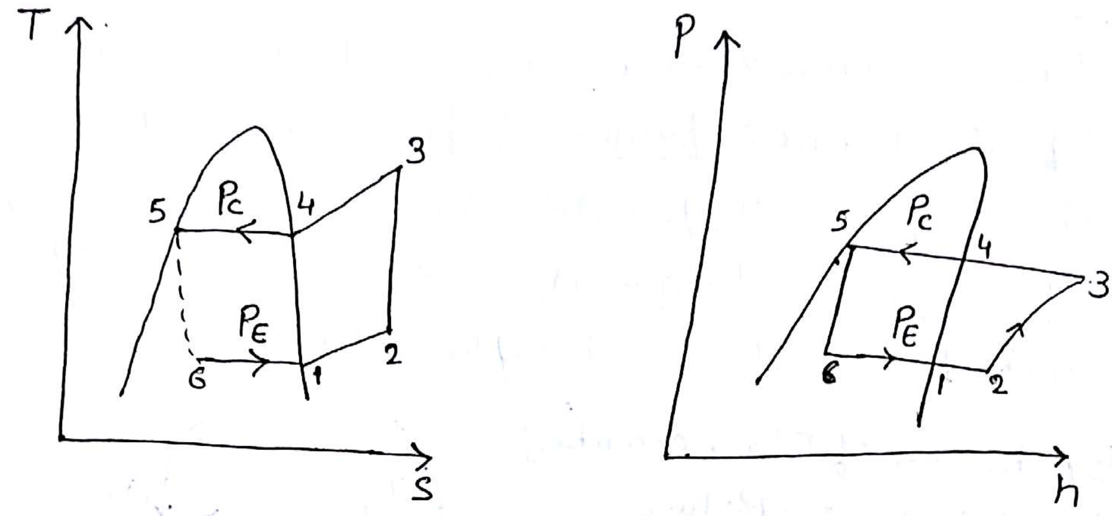
NOTE :- (1) Simple VCRS





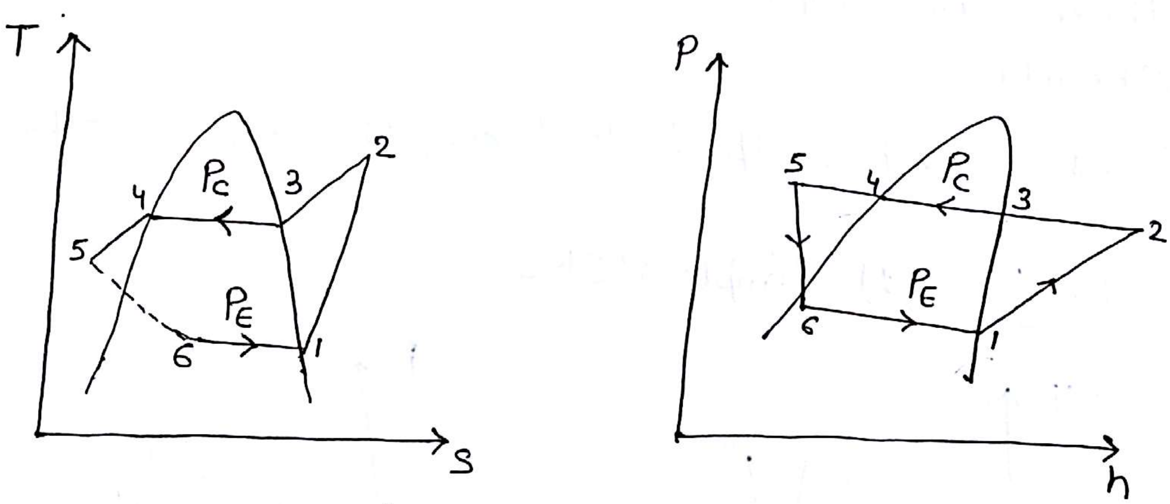
$$P_4 = P_E = P_1$$
$$P_3 = P_2' = P_2 = P_C$$

2) Superheating :-



$$P_6 = P_1 = P_2 = P_E$$
$$P_5 = P_4 = P_3 = P_C$$

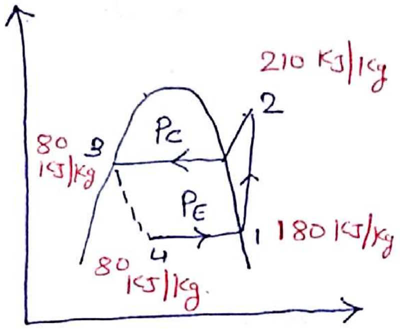
3) Subcooling :-



$$P_5 = P_4 = P_3 = P_2 = P_C$$
$$P_6 = P_1 = P_E$$

Qn) A Refrigerant opening/operating on Simple VCRS, having enthalpy at the entry of evaporator is 80 KJ/Kg & leaving the evaporator with enthalpy of 180 KJ/Kg. Enthalpy at entry of Condensor is 210 KJ/Kg.

find COP?



Soln) $R.E = h_1 - h_4$

$= 180 - 80$

$= 100$

$W_{in} = h_2 - h_1$

$= 210 - 180$

$= 30$

$$COP = \frac{R.E}{W_{in}}$$

$$COP = \frac{100}{30}$$

$$\boxed{COP = 3.33}$$ Ans

Methods To find mass flow Rate $\dot{m}$

① $R.C = \dot{m} \times R.E$

② $P_{in} = \dot{m} \times W_{in}$

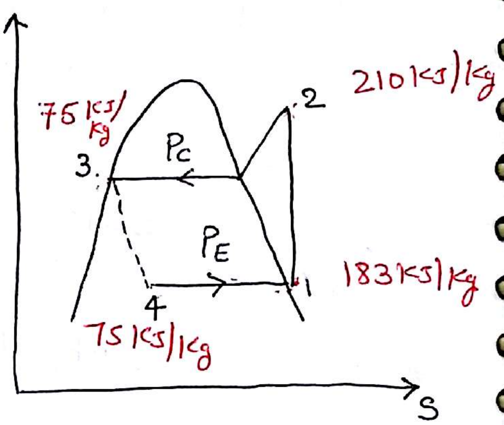
③ $$\eta_v = \frac{\dot{m}\, v_{\text{entry of Comp.}}}{\frac{\pi}{4} D^2 L \frac{N}{60} \times K}$$

if there is no value given in Data of K then it must be understood to taken it as 1

Qn) In 5 KW Cooling Capacity, Refrigeration System, the Refrigerant enter in evaporator with the enthalpy of 75 KJ/Kg. & leaves with the enthalpy of 183 KJ/Kg. Compression process is isentropic and the enthalpy at the outlet of the Compressor is 210 KJ/Kg Calculate,

1) COP.
2) Power Consumption in KW.
3) Rate of Heat Rejection across Condensor in KW.

Soln) 1



$R.E = h_1 - h_4$
$= 183 - 75$
$= 108$

$W_{in} = h_2 - h_1$
$= 210 - 183$
$= 27$

$COP = \frac{R.E}{W_{in}} = \frac{108}{27} = 4$

$\boxed{COP = 4}$

2)

$R.C = \dot{m} \times R.E$
$5 = \dot{m} \times 108$
$\dot{m} = 0.04629 \ Kg/sec$

$P_{in} = \dot{m} \times W_{in}$
$P_{in} = \dot{m} \times 27$
$P_{in} = 0.04629 \times 27$

$\boxed{P_{in} = 1.25 \ KW}$

3)

$Q_c = (h_2 - h_3) \times \dot{m}$
$= 135 \times 0.04629$

$\boxed{Q_c = 6.24915 \ KW}$

$Q_c = (h_2 - h_3)$
But it Result Ans in KJ/Kg so to Convert it into KW We must multiply Qc with $\dot{m}$.

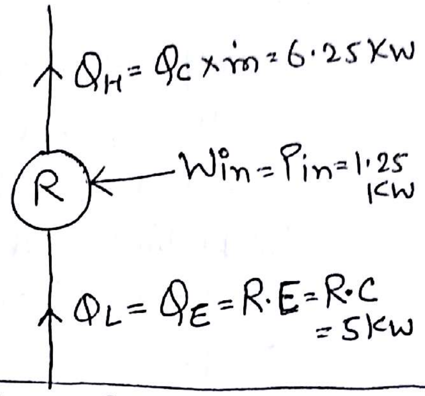
$Q_c = (h_2 - h_3)\ \dot{m}$

$\boxed{\frac{KJ}{Kg} \times \frac{Kg}{Sec} \quad \frac{KJ}{Sec} = KW}$

Another Method to Solve above Question.

$$C.O.P = \frac{RE}{Win} = \frac{RE \times \dot{m}}{Win \times \dot{m}} = \frac{RC}{Pin}$$

$$P = \frac{RC}{C.O.P} = \frac{5}{4} = 1.25\ KW$$



Qn) A Refrigeration operating on Simple VCRS has Piston displacement vol (Swept vol.) of 1.5 L, having 80% volumetric efficiency and 1600 R.P.m. the following Data's are Provided,

	h KJ/Kg	v m^3/kg
Compressor Inlet	183.2	0.0767
Compressor discharge	222.6	0.0614
Condensor exit	84.9	0.00083

1600 rpm / 80% / 1.5 L

Then find,

1) Refrigeration effect in KW
2) Power input in KW

Soln) $R.E = h_1 - h_4$
$= 183.2 - 84.9$

R.E = 98.3 KJ/Kg

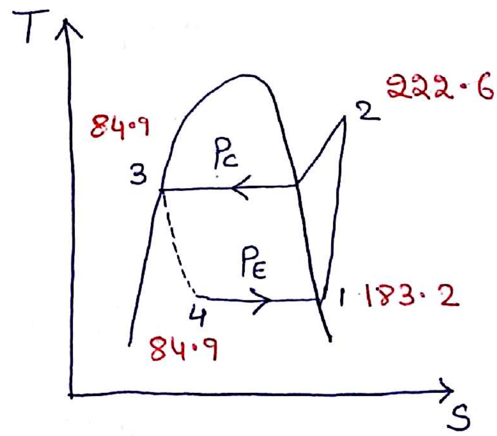
$Win = h_2 - h_1$
$= 222.6 - 183.2$

Win = 39.4

Power input = $Pin = \dot{m} \times Win$

$Pin = 0.4142 \times 39.4$

Pin = 16.319 KW



$$\eta_v = \frac{\dot{m}\ v_{entry}}{\frac{\pi}{4} D^2 L \frac{N}{60} \times K}$$

$$0.80 = \frac{\dot{m} \times 0.0767}{1.5 \times 10^{-3} \times \frac{1600}{60} \times 1}$$

$\dot{m} = 0.4142$ Kg/sec

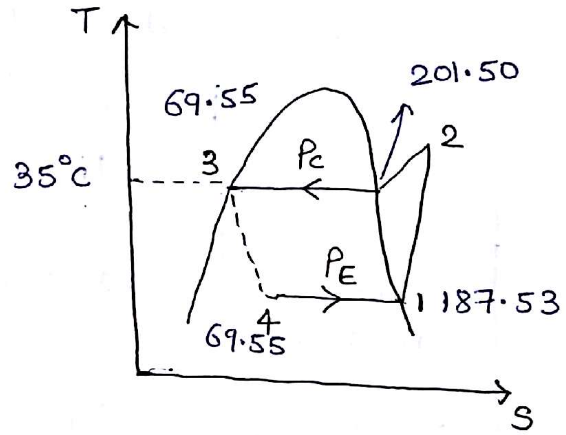
Qn) A Refrigerant operating on Simple VCRS having COP = 6.5. Enthalpy of Saturated liquid & Saturated Vapour Refrigerant at the operating Condensor temperature of 35°C are 69.55 KJ/Kg and 201.50 KJ/Kg. Respectively. The Saturated Refrigerant Vapour leaving the evaporator having enthalpy of 187.53. The Specific heat of Vapour refrigerant is 0.6155 KJ/Kg-K.

Find the Compressor discharge temp in °C.

Soln) Given Data,

$COP = 6.5$

$(C_p)_{vap} = 0.6155\ KJ/Kg\text{-}K$



$R.E = h_1 - h_4$

$= 187.53 - 69.55$

$R.E = 117.98\ KJ/Kg$

$C.OP = \frac{R.E}{W_{in}}$

$6.5 = \frac{117.98}{W_{in}}$

$W_{in} = \frac{117.98}{6.5}$

$W_{in} = 18.150\ KJ/Kg$

$C.OP = \frac{R.E}{W_{in}}$

$W_{in} = h_2 - h_1$

$18.150 = h_2 - 187.53$

$h_2 = 205.680\ KJ/Kg$ Ans

$h_E = h_{g_2} + (C_p)_{vap}(T_2 - T_2')$

$h_2 = h_{g_2} + (C_p)_{vap}(T_2 - T_2')$

$205.680 = 201.50 + 0.6155(T_2 - 35°)$

$4.0152 = T_2 - 35°$

$T_2 = 39.01°C$ or 41.79

$4.18 = 0.6155(T_2 - 35°)$

$6.7912 = T_2 - 35°$

$T_2 = 41.79°C$ Ans

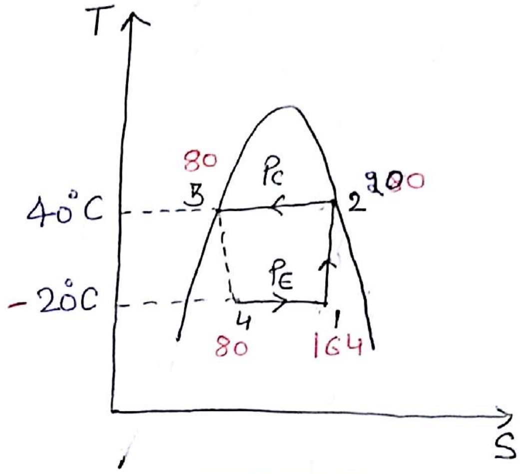
Qn) A Refrigerant based on ideal VCRS. Operates b/w a temp limits of $-20°C$ & $40°C$.
The Refrigerant Enters the Condensor as Saturated Vapour and leaves the Condensor as Saturated Liquid. Then find,

1) COP
2) Refrigeration ~~effect~~ capacity in KW, if $\dot{m} = 0.025$ Kg/sec.

Soln) Given,

Temp. limits = $-20°$ & $40°C$

$\dot{m} = 0.025$ Kg/Sec



$t°C$	h_f	h_g	S_f	S_g
$-20°$	20	180	0.07	0.7366
$40°$	80	200	0.03	0.67

$h_1 = h_{f1} + x(h_{g1} + h_{f1})$

$h_1 = 20 + x_1(180 - 20)$

$h_1 = 20 + x_1(160)$ —①

$S_1 = S_2$

$S_{f1} + x(S_{g1} - S_{f1}) = S_{g2}$

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

$x = 0.9$

Putting Value of x in ①

$h_1 = 164$ KJ/Kg

① $R.E = h_1 - h_4$
$= 164 - 80$
$= 84$

$R.E = 84$

② $W_{in} = h_2 - h_1$
$= 200 - 164 = 36$

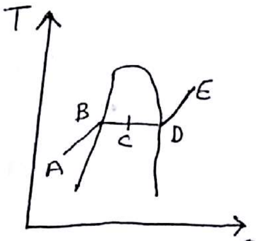
$W_{in} = 36$

$C.O.P = \frac{84}{36} = 2.33 = 2.33$

$C.O.P = 2.33$ Ans

③ $R.C = \dot{m} \times RE$
$= 0.025 \times 84$

$RC = 2.1$ KW Ans



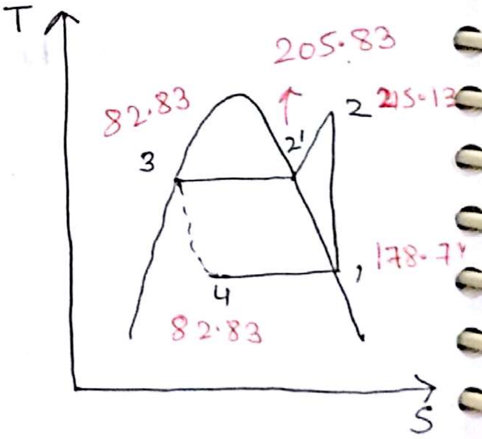
$h_B = h_f$	$S_B = S_f$
$h_D = h_g$	$S_D = S_g$
$h_C = h_f + x h_{fg}$	$S_C = S_f + x S_{fg}$
$h_A = h_f - (C_P)_L (T_B - T_A)$	$S_A = S_f - (C_P)_L \frac{T_B}{T_A}$
$h_E = h_g - (C_P)_V (T_E - T_D)$	$S_E = S_g - (C_P)_V \frac{T_E}{T_D}$

Q22) A VCRS System using R-12 is employed to Produce 8640 Kg of ice/day. The Condensing & evaporator of Refrigerant are 48°C & -20°C. Saturated liquid leaves the Condensor and Saturated Vapour leaves the evaporator. The Compression is isentropic and Water at 35°C is used to form ice and the temp of the ice should be 8°C. Heat flow into the brine tank from Surrounding which is 10% of total heat removed from Water into form ice. Determine the total power required to drive the Compressor in KW.

Assume Specific heat of ice 2.26 KJ/KgK, Latent heat of ice is 334.72 Kg/Kg & Specific heat of Vapor Refrigerant $(C_P)_V = 0.82$ KJ/KgK

t°C	P(bar)	h_f (KJ/Kg)	h_g (KJ/Kg)	S_f	S_g
48°	11.64	82.83	205.83	0.2973	0.6802
-20°	1.51	17.82	178.74	0.0731	0.7087

Solⁿ) $\dot{m}_{ice} = 8640\ Kg/day = \frac{8640}{24 \times 3600} = 0.1\ KJ/sec$



$h_2 = h_{g2}' + (C_P)_V (T_2 - T_2')$

$= 205.83 + 0.82 (T_2 - 321)$ —①

$S_1 = S_2$

$S_{g1} = S_{g2}' + (C_P)_V \ln \frac{T_2}{T_1}$

$0.7087 = 0.6802 + 0.82 \ln \frac{T_2}{321}$

$\boxed{T_2 = 332.35K}$

using Value of T_2 in ①

$h_2 = 215.139\ KJ/Kg$

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

$P_{in} = \dot{m} \times h_2 - h_1 \Rightarrow P_{in} = \dot{m} \times 36.34$

[S.H → Temp. Change]
[L.H → Phase Change]

$$35°C - 0°C - 0°C - 8°C$$
(SH) (LH) (SH)

$$R_C = \dot{m} C_{r_{fg}} (T_H - T_L) + \dot{m} (LH) + \dot{m} C_{<fg} (T_H - T_L)$$

$$= 0.1 \times 4.187 (35-0) + 0.1 \times 334.72 + 0.1 \times 2.26 \times (0-(-8))$$

$$R_C = 49.90 \text{ kW}$$

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

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

$$49.90 = \dot{m}\, 95.91$$

$$\dot{m} = 0.520 \text{ kg/sec}$$

10% of 49.90
4.999 kW

[35°C / -8°C] → 49.94 + 4.499 = 54.90 kW

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

$$54.90 = \dot{m}\, 95.91$$

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

Putting $\dot{m}$ in ①

$$= 0.57 (h_2 - h_1)$$

$$= 0.57 (205.34 - 178.74)$$

$$P_{in} = 20.85 \text{ kW}$$

$$C.O.P = \frac{RE}{W_{in}} = \frac{h_1 - h_4}{h_2 - h_1} = \frac{95.91}{36.39}$$

$$\boxed{COP = 2.63}$$

Q. 23 Workbook)

Qn) A Food Storage requires a Refrigeration capacity of 15TR. It Works b/w −10° & 30°C. The Temperature of Refrigerant Superheated as gas in evaporator is −5°C & temperature of Refrigerant Subcooled as liquid in the ~~compressor~~ Condensor is 25°C. No. of cylinder are equal to 2, Stroke is 1.5 times the bore & Speed is 960 RPM. Determine,

1) (a) R.E/kg
   (b) mass flow Rate ($\dot{m}$) of Refrigerant in kg/min
   (c) Theoritical Piston displacement
   (D) COP
   (E) Bore and Stroke of the Compressor.

2) if the Clearnac volume is 3% of Stroke volume then,
   (a) Determine Volumetric efficiency η_v.
   (b) Bore & Stroke of the Compressor.

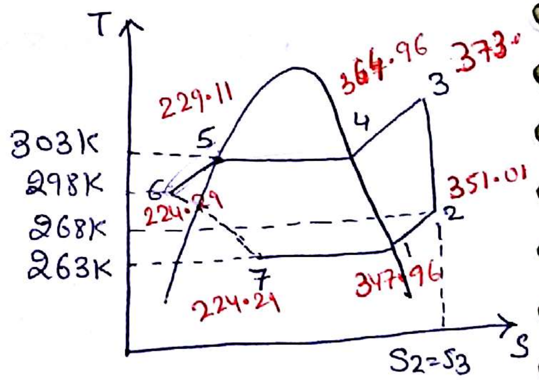
The Specific heat of liquid is taken as 0.963 KJ/kgK & Specific heat of Vapour is 0.615 KJ/kg K

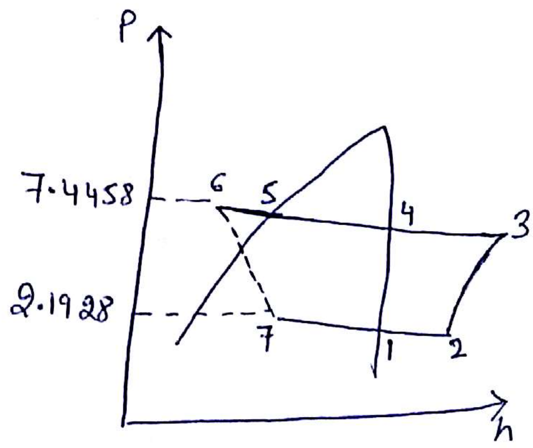
t°C	Pc(bar)	hf (KJ/kg)	hg (KJ/kg)	Sf	Sg	V (m³/kg)
−10	2.1928	190.72	347.96	0.96561	1.5632	0.07702
30	7.4458	229.11	364.96	1.0999	1.5481	0.02372

Soln Given Data, R.C = 15TR
R.C = 15 × 3.5 = 52.5 KW

Saturated
at Sg Temp. evaporator = −10°C = 263K
at Sl Temp Condenser = 30°C = 303
Temp. of Refrigerant Superheated as gas in evaporator is = −5°C = 268K
Temp. of Refrigerant Superheated as liquid in Condenser is = 25°C = 298K

No. of cylinder (K) = 2
Stroke (L) = 1.5D
Speed (N) = 960 RPM
$(C_P)_V = 0.615$ KJ/kg-K
$(C_P)_L = 0.963$ KJ/kg-K





$h_1 = 347.96$

$h_2 = h_{g1} + (C_P)_V (T_2 - T_1)$

$h_2 = 347.96 + 0.615 (268 - 263)$

$h_2 = 347.96 + 3.075$

$h_2 = 351.0$ KJ/Kg-K

$h_4 = 364.96$

$h_3 = h_{g4} + (C_P)_V (T_3 - T_4)$

$h_3 = 364.96 + 0.615 (T_3 - 303)$ — (1)

$S_2 = S_3$

$S_{g1} + (C_P)_V \ln \frac{T_2}{T_1} = S_{g4} + (C_P)_V \ln \frac{T_3}{T_4}$

~~0.96561 + 0.615 ln 268/263 = +.~~

$1.5632 + 0.615 \ln \frac{268}{263} = 1.5481 + 0.615 \ln \frac{T_3}{303}$

$T_3 = 316.12K$

On putting value of T_3 in eqn (1)

$h_3 = 364.96 + 0.615 (316.12 - 303)$

$h_3 = 364.96 + 0.615 (13.12)$

$h_3 = 373.02$ KJ/Kg-K

$h_5 = 229.11$

$h_6 = h_{f5} - (C_P)_L (T_5 - T_6)$

$h_6 = 229.11 - 0.963 (303 - 298)$

$h_6 = 224.29$

$h_6 = h_7 = 224.29$ Bcz of Throttling

(1) R.E = $h_2 - h_7$

$RE = (351.01 - 224.29) \Rightarrow R.E = 126.8$ KJ/Kg-K

$R.C = \dot{m}\, R.E$

$52.5 = \dot{m}\; 126.8$ — KJ/Kg-K
KJ/sec

$\dot{m} = \frac{52.5}{126.8}$ KJ/sec

$\dot{m} = 0.414$ Kg/sec × 60

$\dot{m} = 24.60$ Kg/min

1 (d) $COP = \frac{R.E}{W_{in}} = \frac{126.8}{21.99} = 5.76$

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

$= (373 - 351.01)$

$W_{in} = 21.99$ KJ/kg-K

1(c) Theoritical Piston displacement means 100% volumetric efficiency.

$\eta_v = 100\% = 1 = \frac{\dot{m} v_2}{\frac{\pi}{4} D^2 L \frac{N}{60} K}$

$TPD = \dot{m} v_2 = \frac{\pi}{4} D^2 L \frac{N}{60} K$

$TPD = 0.416 \, v_2$ — (1)

$TPD = 0.416 \times 0.07702$

$TPD = 0.326 \ m^3/sec$

$Pv = mRT \quad P\left(\frac{V}{m}\right) = RT \quad V \propto T$

$\frac{V_2}{V_1} = \frac{T_2}{T_1} = \frac{v_2}{0.07702} = \frac{268}{263}$

$V_2 = 0.07848 \ m^3/kg$

$TPD = 0.0326 = \frac{\pi}{4} D^2 L \frac{N}{60} \times K$

$0.0326 = \frac{\pi}{4} D^2 \, 1.5D \times \frac{960}{60} \times 2$

$\frac{m^3}{sec} = \frac{D^2}{sec}$

$D = 0.095 \ m$

$L = 0.1429 \ m$

2(a) $V_c = 3\% \ V_s$

$C = \frac{V_c}{V_s} = 0.03$

$V_c = 0.03 V_s$

$P_3 = P_4 = P_c$

$P_2 = P_1 = P_7 = P_E \rightarrow Pv^n = C$

$\frac{T_3}{T_2} = \left(\frac{P_3}{P_2}\right)^{\frac{n-1}{n}}$

$\eta_v = 1 + C - C\left(\frac{P_3}{P_2}\right)^{\frac{1}{n}}$ or $\eta_v = 1 + C - C\left(\frac{T_3}{T_2}\right)^{\frac{1}{n-1}}$

$\left(\frac{T_3}{T_2}\right)^{\frac{1}{n-1}} = \left(\frac{P_3}{P_2}\right)^{\frac{1}{n}}$

or

$\eta_v = 1 + C - C\left(\frac{V_2}{V_3}\right)$

$\left(\frac{P_3}{P_2}\right)^{\frac{n-1}{n}} = \left(\frac{V_2}{V_3}\right)^{n-1} \Rightarrow \left(\frac{P_3}{P_2}\right)^{\frac{1}{n}} = \frac{V_2}{V_3}$

$\eta_v = 1 + 0.03 - 0.03\left(\frac{0.7848}{v_3}\right)$ — (1)

$V \propto T = \frac{V_3}{V_4} = \frac{T_3}{T_4}$

$\frac{V_3}{0.02372} = \frac{316}{303} \Rightarrow V_3 = 0.0247 \ m^3/kg$

Putting value of V_3 in (1)

$\eta_v = 93.5\%$ Ans

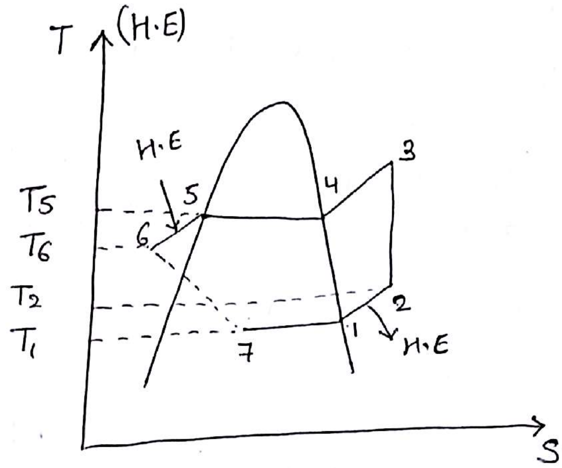
(2)(b) $\eta_v = \frac{\dot{m} V_2}{\frac{\pi}{4} D^2 L \frac{N}{60} \times K}$

$\eta_v = .935 = \frac{0.416 \times 0.07848}{\frac{\pi}{4} D^2 \times 1.5D \ \frac{960}{60} \times 2}$

$D = 0.097 \ m$

$L = 0.1459 \ m.$ Ans

Use Of Heat Exchanger In VCRS :-



Evaporator → 7 – 1 – ~~Q~~ → $R.E = h_1 - h_7$

Compressor → 2 – 3 → $W_{in} = h_3 - h_2$

Condenser → 3 – 4 – 5 – ~~6~~ → $Q_c = h_3 - h_5$

Throttling → 6 – 7 → $h_6 = h_7$

Heat Exchanger → 1–2, 5–6

Qn) A Food Storage Requires a refrigeration Capacity of 50KW. It Works b/w a Condenser temperature of 35°C & evaporator temperature of -10°C. It is Sub Cooled by 5°C before entering the expansion valve. by the dry saturated Vapour leaving the evaporator. The Refrigerant Ammonia is Assuming a Single cylinder, Single acting Compressor, operating at 1000RPM With Stroke is equal to 1.2 time the Bore. Determine,

1) Power Required
2) Cylinder dimension

t°C	P (bar)	h (KJ/Kg)		S (KJ/Kg-K)		v (m³/Kg)		Sp. Heat (Cp) KJ/Kg K	
		hf	hg	Sf	Sg	Vf	Vg	Liq	V
-10	2.9157	154.05	1450.22	0.82965	5.7550	-	0.41747	-	2.492
35	13.522	366.07	1488.57	1.56605	5.2086	1.7023	0.09562	4.556	2.903

Soln)
$R.C = 50KW = 50 \times 3.5 = 175 KW$

$K = 1$

$L = 1.2D$

$n = 1000$

NOTE:-
As There is no information Provided Regarding clearance Ratio (Imp) therefore i will Assumme 100% Volumetric efficiency.

CASCADE :-

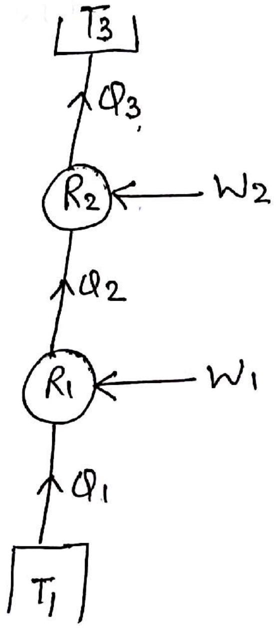
Conventional

Prove That, COP of Cascade Refrigeration System is

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

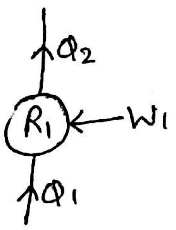
Where $(COP)_1$ is COP of first Refrigeration System i.e R_1.

$(COP)_2$ is COP of Second Refrigeration System i.e R_2.



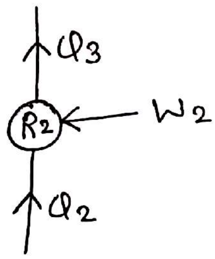
$\rightarrow \quad Q_2 = Q_1 + W_1$

$$Q_2 = Q_1 + \frac{Q_1}{COP_1} \quad \text{—} ④$$



$$(COP)_1 = \frac{DE}{W_{in}} = \frac{Q_1}{W_1}$$

$$\boxed{W_1 = \frac{Q_1}{COP_1}} \quad \text{—} ①$$



$$(COP)_2 = \frac{DE}{W_{in}} = \frac{Q_2}{W_2}$$

$$\boxed{W_2 = \frac{Q_2}{COP_2}} \quad \text{—} ②$$

$$(COP)_{cc} = \frac{(DE)_{LC}}{(W_{in})_{LC} + (W_{in})_{VC}} = \frac{Q_1}{W_1 + W_2} \quad \text{—} ③$$

using ① & ② in equation ③

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

$$(COP)_{cc} = \frac{Q_1}{\frac{Q_1}{(COP)_1} + \frac{Q_1 + \frac{Q_1}{(COP_1)}}{(COP)_2}} = \frac{Q_1}{\frac{Q_1}{COP_1} + \frac{Q_1}{COP_2} + \frac{Q_1}{COP_1 \times COP_2}}$$

Imp objective

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

Hence Proved.

Q13) A Cascade Refrigeration System of 100 TR Capacity uses ammonia & CO_2 Refrigerant. The evaporating & Condensing temp of CO_2 are $-40°C$ & $5°C$. The evaporating temp of NH_3 is $-7°C$. The Power Supplied to the NH_3 Compressor is 96.5 KW. In the CO_2 Circuit the liquid leaving the Condensor as Saturated liquid, the Vapour leaving the evaporator is dry & Saturated & the Compression is isentropic. Calculate the mass flow rate of CO_2 & COP of the total System. use the following table for CO_2 having $(C_P)_v = 0.85\ KJ/Kg\,K$

t°C	P(bar)	hf	hg	Sf	Sg
-40	10.55	332.7	652.8	3.8531	5.2262
5	39.77	431.0	649.8	4.2231	5.0037

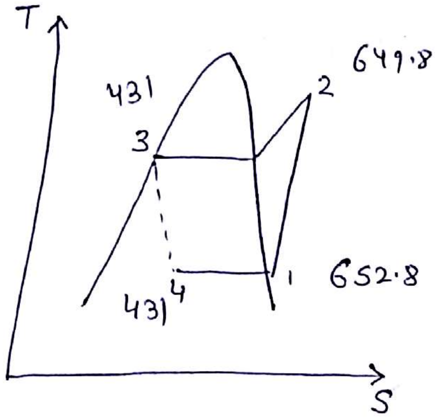
Soln) $RC = 100\,TR = 100 \times 3.5 = 350\,KW = \dot{m}_{(CO_2)} \times RE = \dot{m}_{CO_2} \times (h_2 - h_1)$

$CO_2 \rightarrow T_E = -40°C$

$T_C = 5°C$

$NH_3 \rightarrow T_E = -7°C$

$P_{NH_3} \rightarrow 96.5\,KW$



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

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

$350 = \dot{m}_{CO_2}\ 221.8$

$\dot{m}_{CO_2} = \frac{350}{221.8}$

$\dot{m}_{CO_2} = 1.577\ Kg/sec.$

$P_{CO_2\,in} = \dot{m} \times W_{in}$

$= 1.577 \times (h_2 - h_1)$

$= 1.577 \times (h_2 - 652.8)$ — (1)

$(COP)_{CC} = \frac{350}{1.577(h_2 - 652.8) + 96.5}$

$h_2 = h_{g2'} + (C_P)_v\,(T_2 - T_2')$

$h_2 = 649.8 + 0.85\,(T_2 - 278)$ — (2)

$S_{g1} = S_{g2'} + (C_p)_v \ln\left(\frac{T_2}{T_1}\right)$

$5.2262 = 5.0037 + 0.85 \ln\left(\frac{T_2}{278}\right)$

$T_2 = 361.18 K$

using value of T_2 in equ (2)

$h_2 = 672.5$ KJ/Kg

using value of h_2 in (1)

COP = 1.52

Refrigerants

Type of Refrigerant :-

Primary Refrigerant

Secondary Refrigerant

1) **Primary Refrigerant :-**

Primary Refrigerant are the Working fluid, that undergoing the Cyclic process and extract the heat on the System. There is a Latent heat of Transformation for the Refrigeration.

e.g:- R-11, R-12, R-22, R-134.

2) **Secondary Refrigerant :-**

Secondary Refrigerants are the Working fluid, which are first Cooled by primary Refrigerant & then used for the Cooling at desired place.

e.g:- H_2O, Brine.

Designation of Refrigerant :-

1) CASE I :- When The Refrigerant is Saturated ~~Hydro Carbon~~ Halo Carbon.

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

$$n+P+q = 2m+2$$

$$C_m H_n F_p Cl_q$$

m → Represent no. of Carbon element.

n → Represent no. of Hydrogen element.

P → Represent no. of Flourine element.

q → Represent no. of Chlorine element.

1) R-11

R-011

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

$m-1=0$ $m=1$

$n+1=1$ $n=0$

$P=1$ $P=1$

$n+P+q=2m+2$

$0+1+q=2\times1+2$

$1+q=4$

$\boxed{q=3}$

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

Ans → $CFCl_3$

2) R-12

R-012

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

$m-1=0$; $m=1$

$n+1=1$; $n=0$

$P=2$; $P=2$

$n+P+q=2m+2$

$0+2+q=2\times1+2$

$2+q=4$

$q=2$

$C_mH_nF_pCl_q$ $(m=1, n=0, P=2, q=2)$

$\boxed{CF_2Cl_2}$

3) R-22

R-022

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

$m-1=0$; $m=1$

$n+1=2$; $n=1$

$P=2$; $P=2$

$n+P+q=2m+2$

$1+2+q=2+2$

$3+q=4$

$q=1$

$C_mH_nF_PCl_q$ ($m=1$, $n=1$, $P=2$, $q=1$)

Ans → CHF_2Cl

4) R-134

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

$m-1=1$; $m=2$

$n+1=3$; $n=2$

$P=4$; $P=4$

$n+P+q=2m+2$

$2+4+q=4+2$

$q=6-6$

$q=0$

$C_mH_nF_PCl_q$ $m=2$, $n=2$, $P=4$ $q=0$

$C_2H_2F_4$

NOTE :-

R-11 → $CFCl_3$

R-12 → CF_2Cl_2

R-22 → CHF_2Cl_2

R-134 → $C_2H_2F_2$

R-134 :→ $C_2H_2F_2$

R-134, which is $C_2H_2F_2$, is known as ecofriendly Refrigerant. The chlorine element which are present in commonly used Refrigerant attacks the Ozone layer, which is situated in Stratosphere, which prevents us or filter the harmful UV radiation which is emitted by Sun. The chlorine (Cl) element present in the commonly use refrigerant, attacks the Ozone layer & reduce Ozone layer thickness. Therefore we are use such refrigerant which have the minimum tendency at reduction in Ozone layer.

2) CASE II :- When the Refrigerant is unsaturated Hydrocarbon.

$$R\text{-}1\,(m-1)\,(n+1)\,P$$

$$n+P+q = 2m$$

$$C_mH_nF_pCl_q$$

3) Case III :- When Refrigerant is inorganic Compound.

$$R-(700 + \text{molecular Weight})$$

NH_3 (717)

CO_2 (744)

SO_2 (764)

H_2O (718)

Air (729)

Selection Of Refrigerant (Desirable Property Of Refrigerant) :-

A — Thermodynamic Property

B — Chemical Property

C — Physical Property

A) Thermodynamic Property :-

(i) Critical Temperature :- must be ↑ as possible

The Critical Temperature of Refrigerants Should be as high as possible for above the Condensor pressure & Temperature.

NOTE :-

The Critical Temperature of CO_2 & ethylene are almost. For the indian Summer ambient Condition.

H_2O — 314°C.

SO_2 — 156.5°C

NH_3 — 132.4°C

R-12 — 111.5°C

R-22 — 96.5°C

R-134 — 101.2°C

ii) Specific Heat :-

The Specific Heat of Vapours Should be high, in order to limit the degree of Superheated, Whereas the specific heat of liquid Should be low in order to limit the degree of irreversible. (Low value of ds).

bcz $C_p \propto ds$

iii) Enthalpy of Vaporization :-

It Should be as high as possible because the same Refrigerant Capacity. The mass flow rate ($\dot{m}$) Reduce.

$R_C = \downarrow \dot{m} \times R.E \uparrow$

NOTE :-

Among The Commonly used Refrigerants NH_3 have high Value of the enthalpy of Vaporization.

↑ H_2O — 2261

NH_3 — 1369

R-11 — 234.7

R-12 — 165.7

R-134 — 197.3

$R_C \rightarrow \downarrow \dot{m} \times R.E. \uparrow$

iv) Thermal Conductivity (K) :- must be as high possible.

It Should be high, Because it help to Reducing the Size of evaporator and Condensor.

$$Q = KA\frac{dT}{\partial x}$$

$$K \propto \frac{1}{Area}$$

V) Evaporator and Condensor Pressure :-

Both should be Positive (+ve). If the evaporator pressure is less than atmospheric pressure, then there is probability for the leakage of air, so thus, evaporator pressure should be kept almost equal to atmospheric pressure. {Imp}
On the other side, the condensor pressure should be kept at some moderate value.

VI) Compression Ratio :-

It is Defined as the Ratio of volume before Compression to the volume after Compression.
Low Compression Ratio is desirable because the high Compression Ratio results in increase in Work input to the Compressor and ↓ in volumetric efficiency.

VII) Freezing Point :-

Low Freezing Point is desirable.
Freezing Point of NH_3 → (−77°C)
freezing Point of H_2O → (0°C)

NOTE :-

Freezing Point of Water is 0°C. below 0°C it Convert into Solid State and its flow is not possible.

R-22 → −160.5°C
R-12 → −157.4°C
NH_3 → −77°C
R-134 → −101.2°C

viii) Compressor Discharge Temperature :-

NH_3 Compressor are Water Cooled Compressor.
Because of its high Compressor discharge temperature
Where as

R-11
R-12

Compressor are air Cooled.

NH_3 — Water Cooled Comp^r
R-11, R-12 — Air Cooled Comp^r

$\downarrow \eta_v$

ix) Coefficient of Performance (COP) :-

It represent the Running Cost of the equipment.
Higher the COP, Lower Will be the Running Cost.

$\uparrow$ COP = $\downarrow$ Running Cost

NOTE :-

1) Almost all of the Refrigerant are having Similar Value of COP, when operating between Same temperature limit.

2) Even though, the Latent heat of Vaporization for NH_3 Refrigerant is having high value, but it dosn't help to improve in COP because of it, high Work input to the Compressor.

R-11 → 4.04
NH_3 → 4.06
R-12 → 4.12

3)

Compressor	Pressure	volume	Refrigerant
Screw or Resiprocating Compressor	✓	✗	NH_3, CO_2
Rotary or Centrifugal Compressor	✗	✓	R-11, R-113

B) Chemical Property :-

1) Toxicity :-

Refrigerant Should be non-Toxic.

NOTE :-

NH_3 is Non Toxic in Nature.

2) Flamability :-

Refrigerant Should be non flamable in nature.

NOTE :-

NH_3 is both Non-Toxic and non flamable in nature.

3) Action With oil :-

R + fully missicible with oil. R-11, R-12

R + Fully immisible with oil. NH_3, CO_2

R + Partially missible with oil. R-22

There are Some Refrigerant, fully misible with oil and some Refrigerant fully immisible with oil, doesn't create any Problem but refrigerant which is partially miscible with oil like R-22, create problem. Chokking problem occurs in Condensor.

Therefore Synthetic oil is used in Case of

NOTE :-

1) oil seprator is install b/w Compressor & Condensor.
2) oil seprator, which no Requirement to install, when Refrigerant & oil are immisible at Condensor pressure & Temp.
3) Sensing bulb is placed at the exit of evaporator.

4) Action With material of Construction :-

NH_3 attacks Copper (Cu), so preferable Wrough iron

Where as Hydrocarbon Compound attacks aluminium (Al)

Prefreable with Copper (Cu)

> NH_3 Corrodes Copper, Suitable for Wrough iron
> Hydrocarbon Corrodes Aluminium Suitable for Copper

C) Physical Property :-

(i) Cost :-

It should be Low.

Viscocity ↑

ii) Leak Detection :-

First of All it Should be not be leakage of ~~Rego~~ Refrigerant at any cost but if it leaks out, then its detection should be as faster as possible by the simplest Methods.

For example :

1) Feron leak or Halocarbon leak :-

(I.E.S) Objective

i) Halide Torch Method :-

If Feron leaks then Colour of light changes from blue to bluish green.

(ii) Soap bubble Method.

2) NH_3 leaks :- (Burning Sulphur Candle)

i) Sulphur stick Method :-

The Pressence of NH_3 leaks, White fumes of ammonium Sulphides are formed When Burning Sulphur Candle.

ii) SO_2 leaks :-

NH_3 Swab Test.

Leakage	Detected by
NH_3	Burning Sulphur Candle
Halocarbon, "	Halide Torch test
Hydrocarbon & Propane	Soap and Water Test
SO_2	NH_3 Swab Test
Refrigiration equipment	Electronic Leak detector (high Sensivity)

Azeotropes :-

It is mixture of Refrigerants, which behaves like a Pure System.
There designation are Started with R-500.

Azeotropes → R-500

Mix of Refrigerant behave like pure system

Refrigerant And there Application :-

R-11 → Central Air Conditioning

R-12 → Domestic Refrigerator, Water Cooler

R-22 → Window A/c

NH_3 → Cooling Storage Plant

CO_2 → Direct Contact freezing of food. used as dry ice in transport.

Brine → Milk Chilling Plants

Air → Gas liquidification, Air craft Refrigeration System

Refrigerant in ↓ order of Normal Boiling Point :-

R-11 (-23.7°C) > R-12 (-29°) > NH_3 (-33.3°) > R-22 (-41°) > CO_2 (-73.6°)

Refrigerant in ↑ order of freezing Point :-

CO_2 < NH_3 < R-11 < R-12 < R-22

Refrigerant in ↓ order of Critical temp.

R-11 > NH_3 > R-12 > R-22 > CO_2

Refrigerant in ↓ order of Critical Pressure.

NH_3 (113.86) > CO_2 (73.8) > R-22 > R-11 > R-12 (41.2)

Refrigerant in ↓ of COP :- R-11 > NH_3 > R-12 > R-22 > CO_2

Vapour Absorption Refrigeration System :- (VARS)

1) The Compressor which is used in (VCRS) is Replaced With Absorber, Pump & Generator.
2) Solar Absorption Refrigeration System is Working on the Principal Of (VARS).
3) VARS System is generally Preferred in Remote locations & where the Cost of electricity is high.
4) Waste heat Can be effectively utilized in (VARS) System.
5) The COP of VARS System is Low ↓ and it generally lies b/w 0.3 to 0.5.

imp
6) Heat Rejection Occurs in Condensor and Absorber.
7) The Commonly used Absorber Refrigerant pair is,

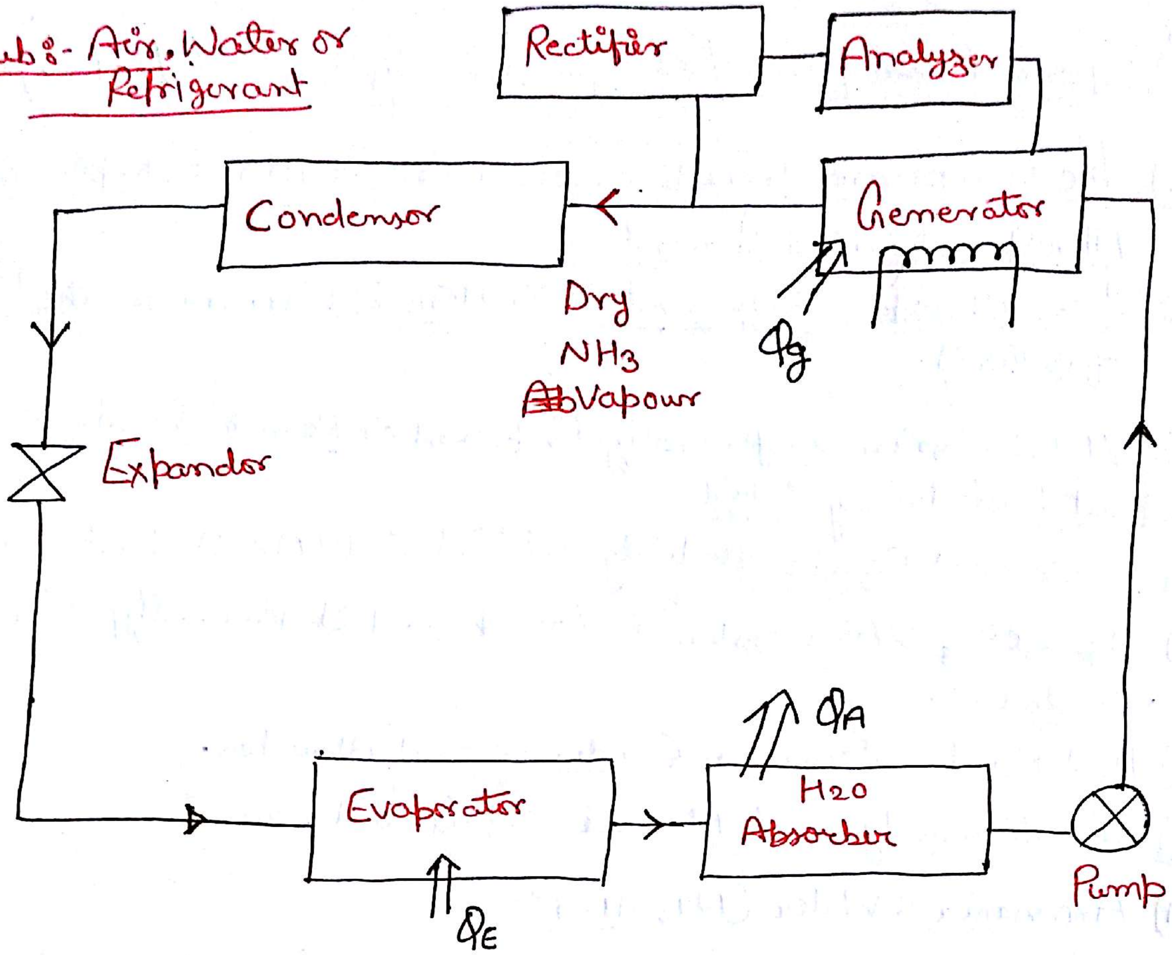
(A) Ammonia & Water (NH_3-H_2O) :-

In this Ammonia (NH_3) is used as the Refrigerant & Water is used as a Absorber. Inorder to Remove Water Particles from the ammonia Vapour Analyser & Rectifier Assembly is used.
Here, Water is Removed in two Stages. The Complete elimination Of Water Particles occurs in Rectifier & it Will produce dry Ammonia Vapour.

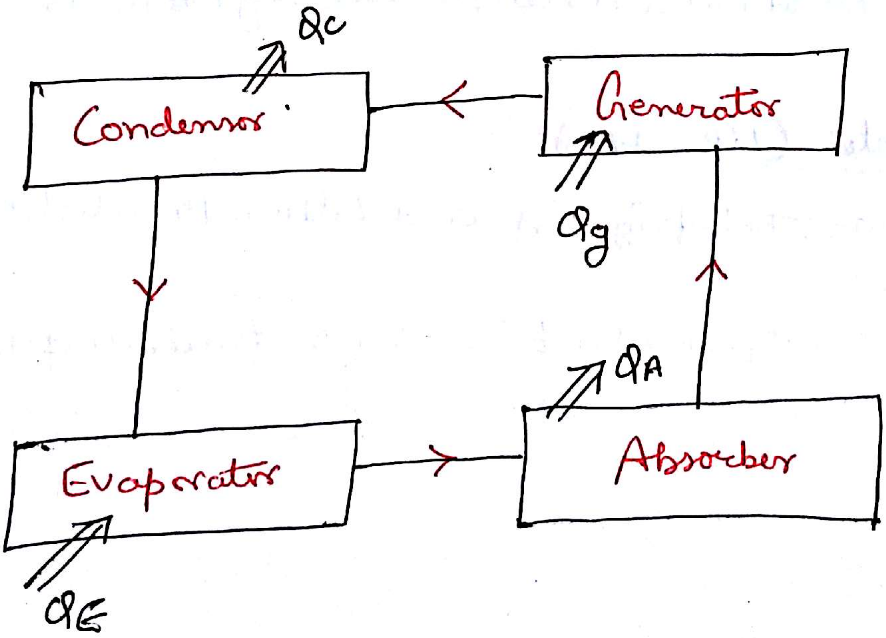
(B) Lithium Bromide & Water ($LiBr-H_2O$) :-

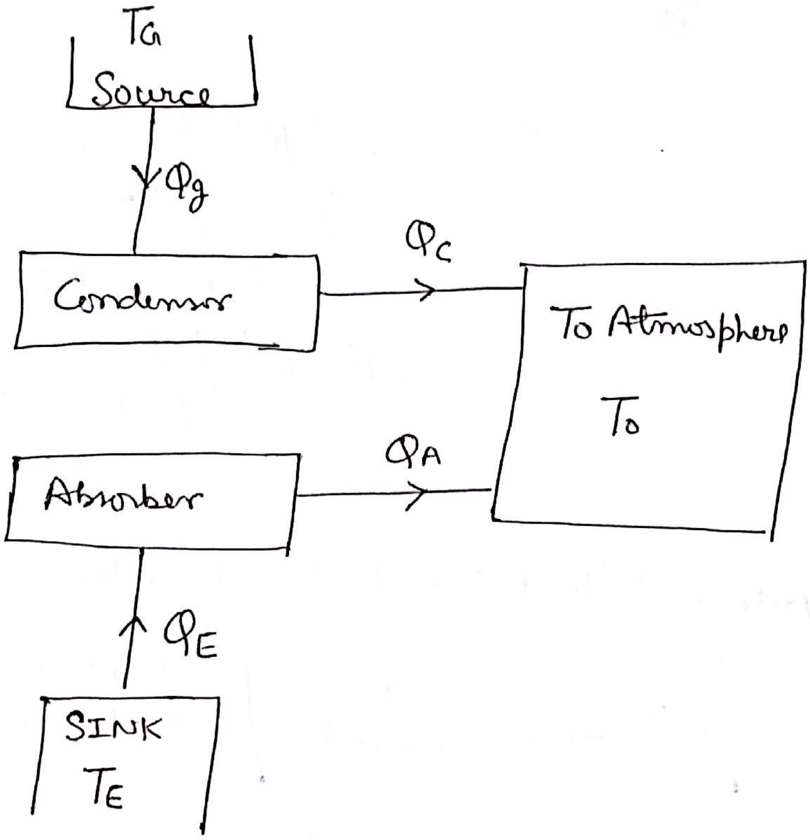
In this Water is used as the Refrigerant and Lithium Bromide (LiBr) is used as the Absorber.
The Above Combination is not prefreable below 0°C (The freezing Pt. of H_2O 0°C)

Working Sub :- Air, Water or Refrigerant



COP of VARS :-





$$Q_E + Q_G = Q_C + Q_A \quad \text{— ①} \quad (\text{from } I^{st} \text{ Law of Thermodynamics})$$

$$\frac{Q_E}{T_E} + \frac{Q_G}{T_G} = \frac{Q_C}{T_0} + \frac{Q_A}{T_0} \quad \text{— ②}$$

$$\frac{Q_E}{T_E} + \frac{Q_G}{T_G} = \frac{Q_C + Q_A}{T_0} = \frac{Q_E + Q_G}{T_0} = \frac{Q_E}{T_0} + \frac{Q_G}{T_0}$$

$$\frac{Q_E}{T_E} - \frac{Q_E}{T_0} = \frac{Q_G}{T_0} - \frac{Q_G}{T_G}$$

$$Q_E\left(\frac{1}{T_E} - \frac{1}{T_0}\right) = Q_G\left(\frac{1}{T_0} - \frac{1}{T_G}\right)$$

$$Q_E\left(\frac{T_0 - T_E}{\cancel{T_0} T_E}\right) = Q_G\left(\frac{T_G - T_0}{\cancel{T_0} T_G}\right)$$

grp objective (Objective)

$$COP = \frac{Q_E}{Q_G} = \frac{T_G - T_0}{T_G} \times \frac{T_E}{T_0 - T_E} \rightarrow$$

$$(COP)_{VARS} = \frac{T_E}{T_G} \times \frac{T_G - T_0}{T_0 - T_E}$$

NOTE :-

1) $T_G \rceil E$, $T_0 \rceil R$, T_E

$$(COP)_{VARS} = \eta_E \times (COP)_R$$

$$= \left(1 - \frac{T_0}{T_G}\right)\left(\frac{T_E}{T_0 - T_E}\right)$$

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

2) $(COP)_{Actual}$, Actual COP of (VARS) System is,

$$\boxed{(COP)_{Actual} = \frac{Q_E}{W_P + Q_G}}$$

[if Question Said to Consider Pump Work.]

$W_P = -V_f \int dP$ (V_f is very Small)

$$\boxed{(COP)_{VARS} = \frac{Q_E}{Q_G}}$$

(Assumption neglect Pump Work)

(if Question Said to neglect Pump Work)

VCRS	VARS
1) Compressor is used.	1) Compressor is Replaced With Absorber, Pump & generator
2) It is a Work Operated Unit. Or Runs on high grade energy.	2) It is a Heat Operated unit. or Runs on Low Grade energy.
3) Heat Rejection Occurs in Condensor only.	3) Heat Rejection Occurs in Condensor as Well absorber.
4) Moisture Related Problem is having more Serious impact or dangerous in VCRS.	4) Relatively Lesser Problem.
5) Chances for the leakage of Refrigerant are high.	5) Relatively lesser chances.
6) It has higher COP, generally Varies from 3-5.	6) It has Relatively Lower (COP) Generally Varries from 0.3-0.5.
7) Creates More noise Pollution.	7) Relatively Less noisy.
Spend Money bcz ↑ COP	less money so ↓ COP can be Considered

Electrolux Refrigerator :- [No use of Pump]

1) The main Aim, of using this Refrigerator System, is to Create noiseless Operation. (i.e) no use of Pump. **

ii) It is a three fluid System. i.e Ammonia, Water & Hydrogen.

NH_3	Refrigerant
H_2O	Absorber
H_2	(Low Partial Pressure of Ammonia Vapour) used to

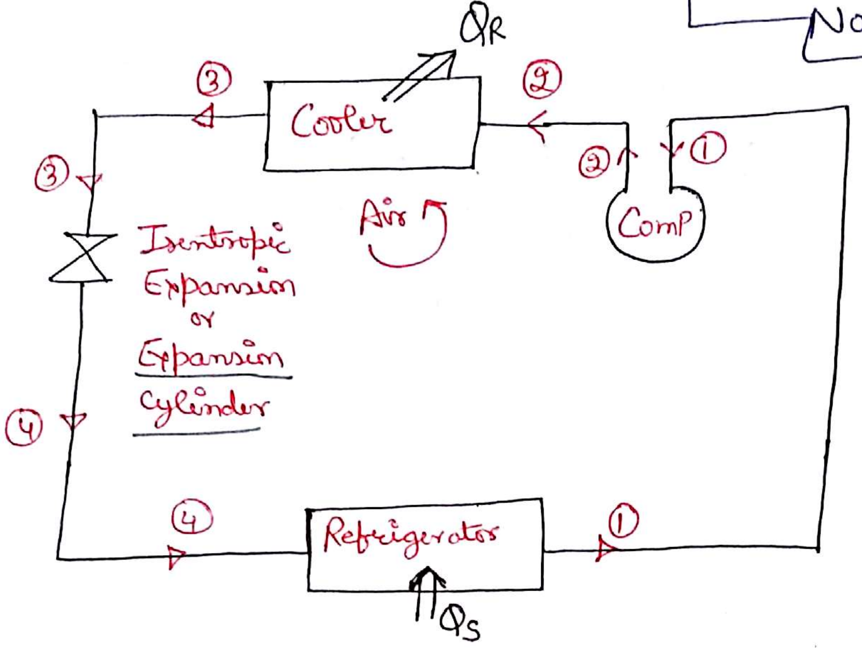
NH_3 is used as the Refrigerant.

H_2O is used as the Absorber.

H_2 is used to Create Low Partial Pressure of NH_3 Vapour

Reversed Brayton Cycle / Bell-Coleman Cycle :- ~~Joule Cycle~~

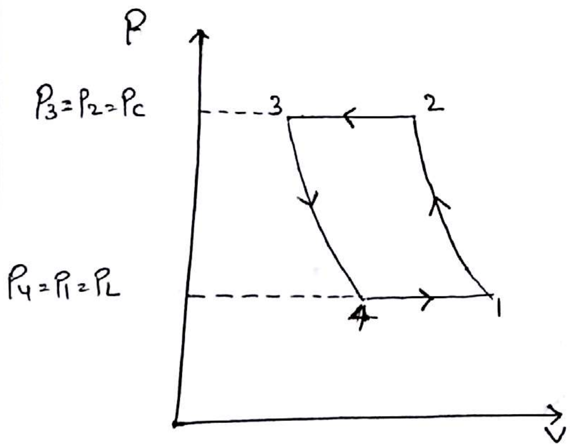
It is for air only
No Phase change



Reversed Brayton Cycle / Bell Coleman Cycle

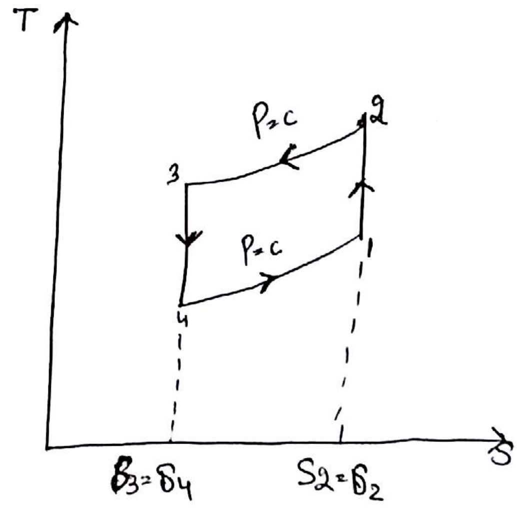
Process (1-2) → Isentropic / Reversible Adiabatic Compression.
Process (2-3) → Heat Rejection at Constant Pressure.
Process (3-4) → Isentropic / Reversible Adiabatic expansion.
Process (4-1) → Heat addition at Constant Pressure

} Processes



Let rp be Pressure Ratio :- $\frac{P_H}{P_L}$

(P-V curve)

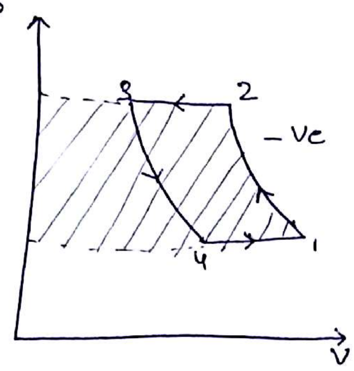


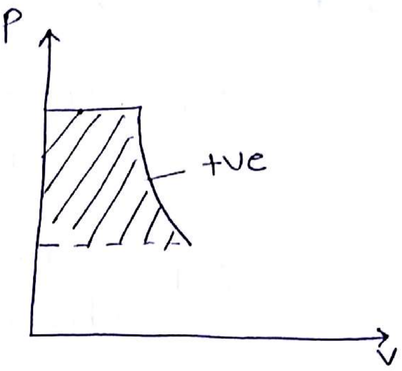
(T-S) curve

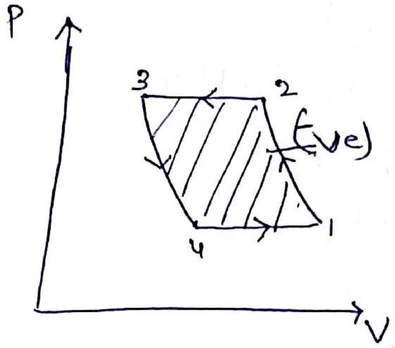
COP of Reversed Brayton Cycle :-

$$COP = \frac{R.E}{W_{net}}$$

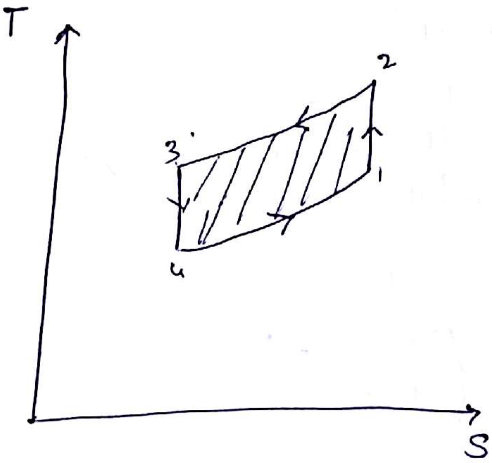
(2-1)







for T-S



$Q_{NET} = -ve$

$W_{NET} = Q_{net}$

$$COP = \frac{R.E}{W_{in}}$$

~~$COP = \frac{R.E}{R_E - R}$~~ $COP = \frac{R.E}{W_C - W_T}$ — Reason bcz R.E is taken as (-ve)

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

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

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

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

$$COP = \frac{1}{\left(\frac{h_2 - h_3}{h_1 - h_4}\right) - 1}$$

Air → $Pv = mRT$, $h = C_pT$ enthalpy is funⁿ of temp for ideal Gas

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

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

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

Process 1-2 $PV^{\gamma} = C$

$$\frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{x} \quad \text{—— (2)}$$

Assuming $\frac{\gamma - 1}{\gamma} = x$ & Pressure Ratio $r_p = \frac{P_H}{P_L}$

Process - 3-4 — $PV^{\gamma} = C$ b/c (Adiabatic)

$$\frac{T_3}{T_4} = \left(\frac{P_3}{P_4}\right)^{x} \quad \text{—— (3)}$$

Process (2-3) :— $P_2 = P_3$
Process (4-1) :— $P_1 = P_4$] — (4) →

using equ^n (4) in (3)

$$\frac{T_3}{T_4} = \left(\frac{P_2}{P_1}\right)^{x} \quad \text{—— (5)}$$

using equ^n (2) & (5)

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

Objective

$$\boxed{T_2 T_4 = T_3 T_1} \quad \text{—— (6)}$$

even   odd
even   odd

Imp Objective for otto and more cycle. [IC TD PP]

using (6) in equ^n (1)

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

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

use equ^n No (2) :- $\frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}} = (r_p)^{\frac{\gamma-1}{\gamma}}$

Let $r_p = \frac{P_H}{P_L} = \frac{P_2}{P_1}$

Imp Objective

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

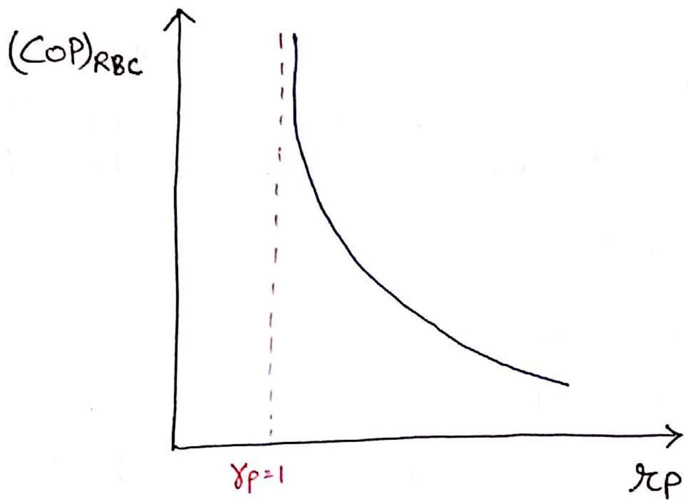
The COP of Reversed Brayton Cycle is a function of Pressure Ratio.

Assumptions:- ☆☆☆

1) There is only one Assumption taken in deriving the expression for COP of Reversed Brayton Cycle is Both Compression & expansion are isentropic.

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

$r_p = 1 \rightarrow COP = \infty$



NOTE:-

1) Air → is the Working fluid used because of Low Wt/TON of Refrigeration.

2) The Expansion Work is not Negligible in Comparison to the Compression Work. Because Both equipment (Compressor & turbine) are handling Same state of the Working fluid that is gaseous Phase.

$$COP = \frac{R.E}{W_C - W_E}$$

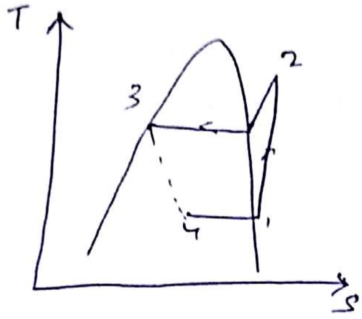
$\int V_g dP \quad \int V_g dP$

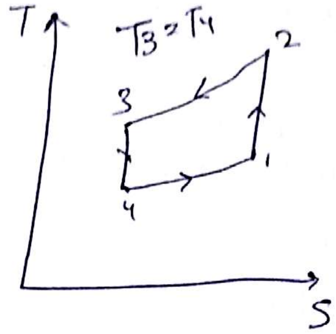
V_f can be neglected but V_g is not

$V_g >>>> V_f$

Qn) Why isenthalpic expansion is not prefreable in Reversed Brayton Cycle?

Ans) Air is Treated As ideal gas, & for an ideal gas enthalpy is funct^n of temp So in the Case of isenthalpic expansion, there is no drop in temp. taken place & instead of heat absorbing it will Rejecting Heat. therefore isentropic expansion is prefrable.





Air = $Pv = mRT$

$h = C_p T$

$h_3 = h_4$

$C_p T_3 = C_p T_4$

$\boxed{T_3 = T_4}$

Qn) A Bell-Coleman Refrigeration Plant operating with Air as a Working fluid, having 1 bar pressure and 10°C temperature is Compressed to a Pressure of 5 bar. Air is then Cooled in the Cooler to a temperature of 25°C. Before expanding in the expansion cylinder. Where the Cold Pressure of 1 bar is maintain. Determine

1) Theoritical COP
2) Refrigeration Effect Per Kg, Assumming both Compression & expansion to be ~~isotrop~~ isentropic, having value of $\gamma = 1.4$, & Specific heat $C_p = 1.004$ kJ/kg K.
3) If Compression follows $PV^{1.35} = C$ (Compressor) and expansion $PV^{1.3} = C$

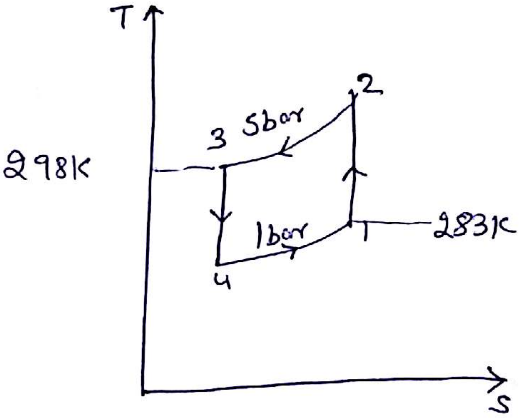
Then Calculate

(a) COP
(b) Refrigeration Effect Per Kg.

Soln)

$$r_p = \frac{P_H}{P_L} = \frac{P_2}{P_1} = \frac{5}{1} = 5$$

Pressure Ratio



(1) Theoritical COP

$$\frac{1}{(5)^{\frac{0.4}{1.4}} - 1} = 1.71$$

(ii) R.E/kg = $h_1 - h_4$ enthalpy is fnⁿ of temp.

$$= C_p(T_1 - T_4)$$

$$= 1.005(283 - T_4) \quad \text{--- (1)}$$

$$= 1.005(283 - 188.15)$$

$$= 95 \text{ kJ/kg.}$$

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

$$\frac{298}{T_4} = (5)^{\frac{0.4}{1.4}} \Rightarrow T_4 = 188.15\,K$$

iii) $PV^{1.35} = C \longrightarrow \frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}} \Rightarrow \frac{T_2}{283} = \left(\frac{5}{1}\right)^{\frac{1.35-1}{1.35}} = T_2 = 429.3\,K$

$PV^{1.3} = C \longrightarrow \frac{298}{T_4} = \left(\frac{5}{1}\right)^{\frac{1.35-1}{1.35}} \Rightarrow T_4 = 205\,K.$

(b) $RE/kg = h_1 - h_4 = C_p(T_1 - T_4)$

$$= 1.005(283 - 205)$$

$$= 77.8 \frac{kJ}{kg}.$$

$W_{1-2} = n\, W_{closed}$

$\boxed{PV^{1.35} = c} \quad = m\left(\frac{P_1V_1 - P_2V_2}{n-1}\right)$

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

As it is case of ideal gas

$PV = mRT$

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

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

W_{1-2} = (Compressor) must be −ve.

$W_{1-2} = \left(\frac{1.35}{1.35-1}\right) \times 1 \times 0.287 (283 - 429)$

$= -162 \frac{KJ}{Kg}$

$W_{3-4} = \left(\frac{n}{n-1}\right) mR (T_3 - T_4)$

$PV^{1.3} = C$

$= \left(\frac{1.3}{1.3-1}\right) \times 1 \times 0.287 (298 - 205)$

$= +114.98 \frac{KJ}{Kg}$

$W_{net} = -162 + 114.98 = -47 \; KJ/Kg$

$W_{in} = 47 \; KJ/Kg$

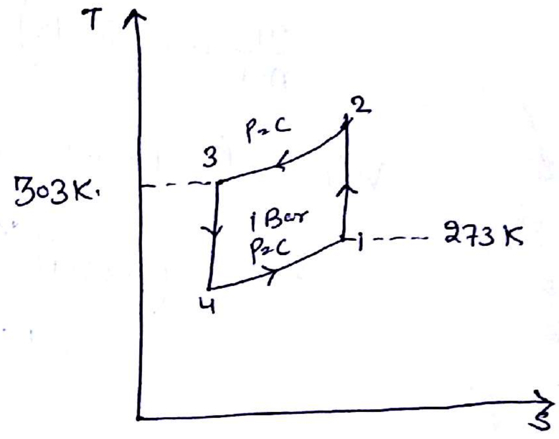
$COP = \frac{R.E}{W_{in}}$

$\boxed{(COP) = \frac{77.8}{47} = 1.64}$

Qn) Air is used as a Refrigerant in Reversed Brayton Cycle. Draw P-V & T-S diagram for the cycle & derive the expression for COP in terms of Pressure Ratio.
If the temperature at the end of Heat absorption & Heat Rejection are 0°C and 30°C Respectively. and pressure ratio (r_P) = 4. Then Determine the temperature of all other Point. and volume flow rate of the inlet of Compressor & exit of expandor for 1TN Cooling Capacity. Assumming inlet pressure (P_1) to the Compressor is 1 bar.

Soln

$r_P = 4$



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

$$\frac{T_2}{273} = (4)^{\frac{0.4}{1.4}} \rightarrow T_2 = 405K.$$

$$\frac{303}{T_4} = (4)^{\frac{0.4}{1.4}} \rightarrow T_4 = 203.9 K.$$

~~$P_1\dot{V}_1 = P_2\dot{V}_2$~~

$$P_1\dot{V}_1 = \dot{m}RT_1$$

$$\dot{V}_1 = \frac{\dot{m}RT_1}{P_1} \Rightarrow \dot{V}_1 = \frac{\dot{m} \times 0.287 \times 273}{1 \times 10^5} \quad -(1)$$

$$= 0.039\ m^3/sec$$

$$RC = \dot{m} \times R.E \Rightarrow 3.5 = \dot{m}(h_1 - h_4)$$

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

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

$$\dot{m} = 0.05\ Kg/sec$$

$$\dot{V} \propto T$$

$$\frac{\dot{V}_4}{\dot{V}_1} = \frac{T_4}{T_1}$$

$$\frac{\dot{V}_4}{0.039} = \frac{203.9}{273} \qquad \dot{V}_4 = 0.29\ m^3/S \text{ Ans}$$

Qn) It is desire to make the COP of a Carnot Refrigerator System Operating b/w $-40^{\circ}C$ & $+40^{\circ}C$ is equal to 3.5, by changing the Temperature. The decrease in higher temperature is equal to ↑ in Lower temperature. Then Determine the new temperatures in Kelvin.

Soln

$T_{L_1} = (40^{\circ}C) - = 233K$

$T_{L_2} = +40^{\circ}C = 313K.$

$T_{L_2} = T_{L_1} + x = 233 + x = 238.875\ K$

$T_{H_2} = T_{H_1}$ [redacted] x = 313 [redacted] x = $307.125K.$

$$3.5 = COP = \frac{T_{L_2}}{T_{H_2} - T_{L_2}} = \frac{T_{L_1} + x}{(T_{H_1} - x) - (T_{L_1} + x)} = \frac{233 + x}{80 - 2x}.$$

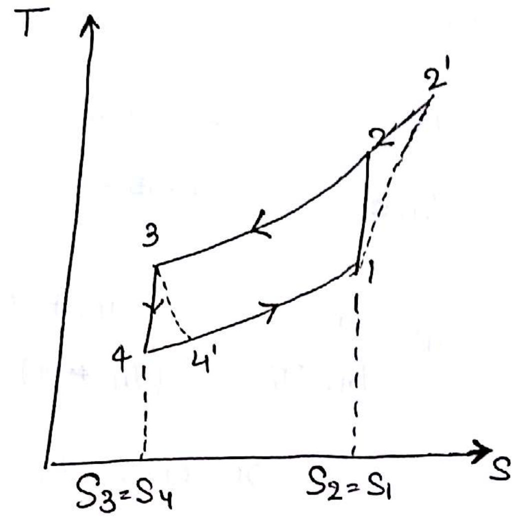
$x = 0.5875$

Isentropic Efficiency of Compressor and Turbine:-

Isentropic Efficiency of Compressor is defined as the Ratio of Ideal enthalpy rise to the actual enthalpy Rise.
Whereas, isentropic efficiency of turbine is defined as the Ratio of Actual enthalpy drop to that of ideal enthalpy drop.

$(\eta_{is})_c = \frac{\text{Ideal}}{\text{Actual}}$

$(\eta_{is})_T = \frac{\text{Actual}}{\text{Ideal}}$



Use only when Ideal gas b(3 h = f(T)

$$(\eta_{is})_C = \frac{h_2 - h_1}{h_2' - h_1} = \frac{C_p (T_2 - T_1)}{C_p (T_2' - T_1)} = \frac{T_2 - T_1}{T_2' - T_1}$$

$$(\eta_{is})_T = \frac{h_3 - h_4'}{h_3 - h_4} = \frac{C_p (T_3 - T_4')}{C_p (T_3 - T_4)} = \frac{T_3 - T_4'}{T_3 - T_4}$$

$\eta_{isentropic}$
→ Rev. Adiabatic
→ irr. Adiabatic → $ds = S_{gen} + \frac{dQ}{T}$ ↗ 0

η → Loss

Important Point :-

$$\text{Actual } W_c = \frac{\text{Ideal } W_c}{\eta_c}$$

Compressor is used to Compress the gas, So We need that more & more Compression takes Place or ↑ in W_c. So We divide by η_c and η_c is less than 1.

$$\text{Actual } W_T = \text{Ideal } W_T \times \eta_T$$

$$\underset{(BP)}{\text{Actual Power output}} = \underset{(IP)}{\text{Ideal Power output}} \times \eta_{mech.}$$

$$\text{Actual } Q_s = \text{Ideal } Q_s \times \eta_{cc}$$

Here multiply With η indicates Loss.

Qn) In bell-Coleman Refrigeration plant, air enters the Compressor at a pressure of (0.1 MPa) and temperature of 4°C. It is then Compressed to pressure of 0.3 MPa, with an isentropic efficiency of 72%. It is then Cooled in the Cooler to a temperature of 55°c and then expanded to a pressure of 0.1 MPa, With an isentropic efficiency of 78%. Assumming Air to be an Ideal Gas and the Lower temp air, Absorbs Cooling load of (3 TR).

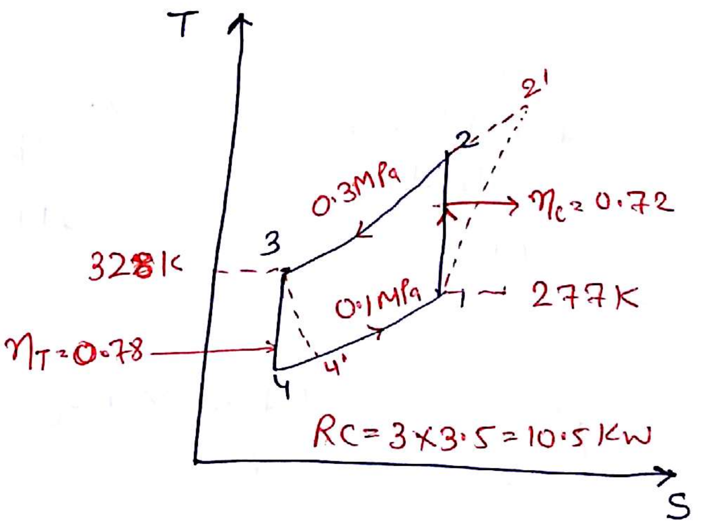
Determine;

1) mass flow Rate in Kg/sec.
2) Power Consumption in KW.
3) COP

Soln

Ideal → 1-2-3-4-1
Actual → 1-2'-3-4'-1

As it is isentropic Process



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

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

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

$\dot{m} = 0.58 \, Kg/sec.$

$$\frac{T_2}{277} = \left(\frac{0.3}{0.1}\right)^{\left(\frac{0.4}{1.4}\right)}$$

$$\boxed{T_2 = 379 K}$$

$$\eta_c = \frac{T_2 - T_1}{T_2' - T_1}$$

$$0.72 = \frac{379 - 277}{T_2' - 277}$$

$$\boxed{T_2' = 418.86 K}$$

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

$$\frac{328}{T_4} = \left(\frac{0.3}{0.1}\right)^{\frac{0.4}{1.4}}$$

$$\boxed{T_4 = 239\,K}$$

$$\eta_t = \frac{T_3 - T_4'}{T_3 - T_4}$$

$$0.78 = \frac{328 - T_4'}{328 - 239}$$

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

$$P = \dot{m}(W_C - W_T)$$

$$P = \dot{m}(h_2' - h_1) - (h_3 - h_4')$$

$$P = \dot{m}\, C_p (T_2' - T_1) - (T_3 - T_4')$$

$$P = 0.58 \times 1.005 (418.86 - 277) - (328 - 259)$$

$$= 42.5\ KW$$

$$\boxed{COP = \frac{R.C}{P_{in}} = \frac{10.5}{42.5} = 0.247}$$

Heat Rejection Ratio :-

It is defined as the Ratio of Heat Rejected across condensor to the Refrigeration Effect.

If the Value of Heat Rejection Ratio = 1 then it violets clausis statmenent Of Thermodynamic.
There fore the Value of Heat Rejection Ratio is always greather than one.

$$HRR = \frac{Q_c}{R.E}$$

Qn) If the Value of Heat Rejection Ratio is 1.2 & the Heat absorbed across evaporator is 2000 KJ/min, then Calculate the Heat Rejected across Condensor in KJ/min & COP.

(Annotation: the Heat absorbed → Means R.E)

$H.RR = 1.2$

$R.E = 2000$ KJ/min

$Q_c = ?$

$\uparrow H.RR = \frac{Q_c}{R.E \downarrow}$

So first two are neglated

~~A) 2100, 4~~

~~B) 2100, 5~~

C) 2520, 4

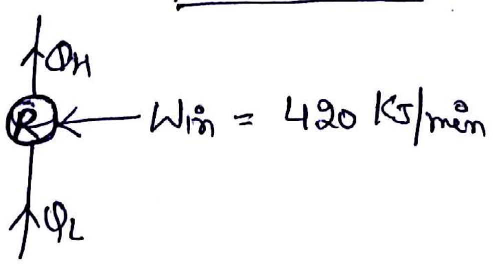
D) 2520, 5

$1.2 = \frac{Q_c}{2100}$

$Q_c = 2520$ KJ/min

$1.2 = \frac{Q_c}{\quad}$

$$H.RR = \frac{Q_c}{R.E}$$

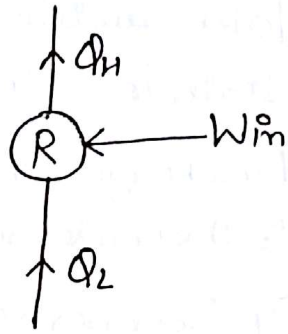


$COP = \frac{2100}{420}$

$COP = 5$

Q) Prove that Heat Rejection Ratio,

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



$$COP = \frac{R.E}{W_{in}} = \frac{Q_L}{Q_H - Q_L}$$

$$= \frac{1}{\left(\frac{Q_H}{Q_L} - 1\right)}$$

$$= \frac{1}{HRR - 1}$$

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

Objective

$$H.R.R = \frac{1}{COP} + 1$$

Objective

Refrigeration Equipment

Function Of Compressor :-

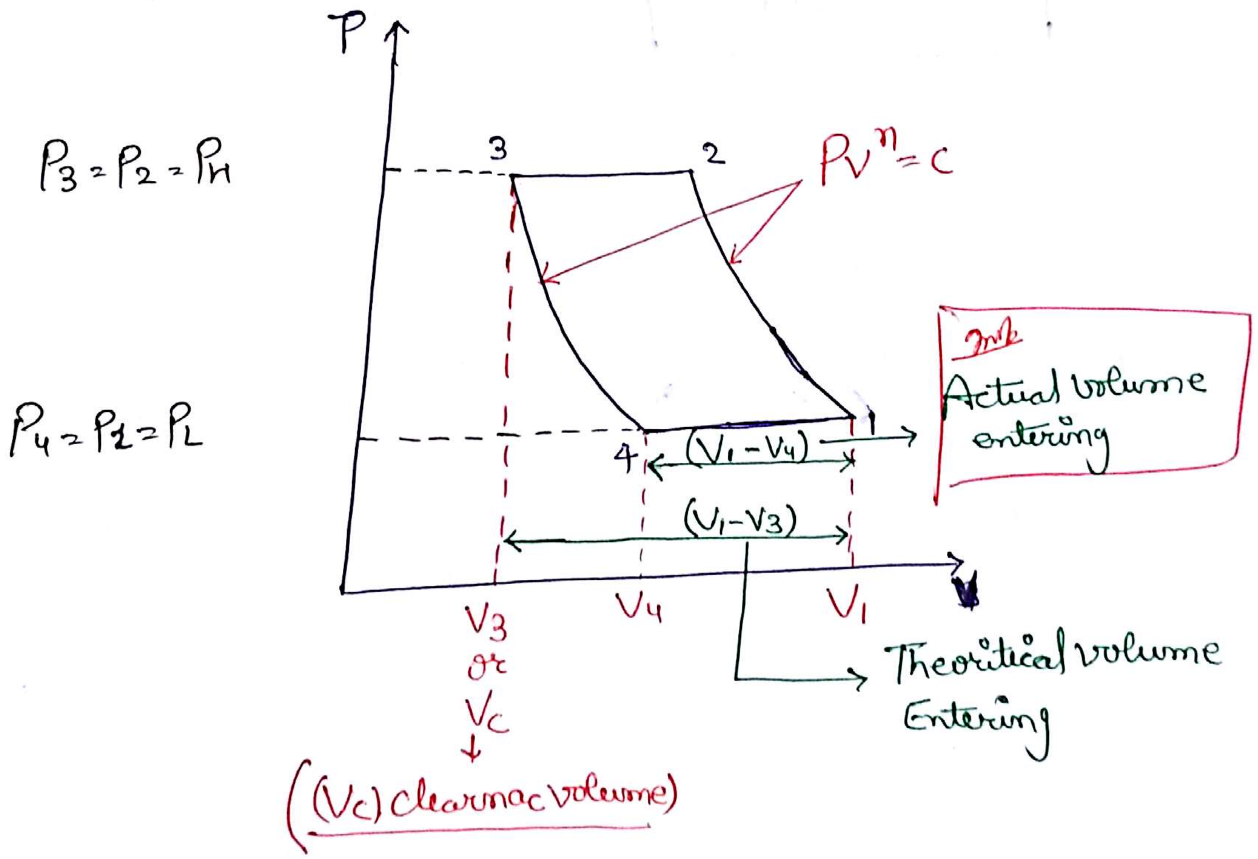
i) Take Suction of Refrigerant generally in Saturated Vapour State, from the evaporator.

ii) Discharge the Refrigerant generally in Superheated State to the Condensor.

iii) It increases the pressure & temperature of the Refrigerant.

Qn) Prove That Volumetric Efficiency is,

$$\eta_v = 1 + C - C\left(\frac{P_H}{P_L}\right)^{\frac{1}{n}}$$



$$\eta_v = \frac{(\text{Actual volume})_{\text{entering}}}{\text{Swept volume.}}$$

$$\eta_v = \frac{V_1 - V_4}{V_1 - V_3}$$

$$\eta_v = \frac{V_1 - V_4 - V_3 + V_3}{V_1 - V_3}$$ multipply & by $-V_3 + V_3$

$$\eta_v = \frac{V_1 - V_3 + V_3 - V_4}{V_1 - V_3}$$

$$\eta_v = \frac{V_1 - V_3}{V_1 - V_3} + \frac{V_3}{V_1 - V_3} - \frac{V_4}{V_1 - V_3}$$

$V_3 = V_c$
$V_S = V_1 - V_3$

$$\eta_v = 1 + \frac{V_3}{V_S} - \frac{V_4}{V_S}$$

$$\eta_v = 1 + C - \frac{V_4}{V_S} \times \frac{V_c}{V_c}$$

$$\eta_v = 1 + C - C\frac{V_4}{V_c} \quad \text{—— (1)}$$

Process (3-4) Polytropic

$$PV^n = C$$

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

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

$P_3 = P_2 = P_H$
$P_4 = P_1 = P_L$

$$\frac{V_4}{V_3} = \left(\frac{P_3}{P_4}\right)^{1/n} \Rightarrow \left(\frac{P_H}{P_L}\right)^{\frac{1}{n}} \quad \text{—— (2)}$$

Putting (2) & (1)

Objective

$$\eta_v = 1 + C - C\left(\frac{P_H}{P_L}\right)^{\frac{1}{n}}$$

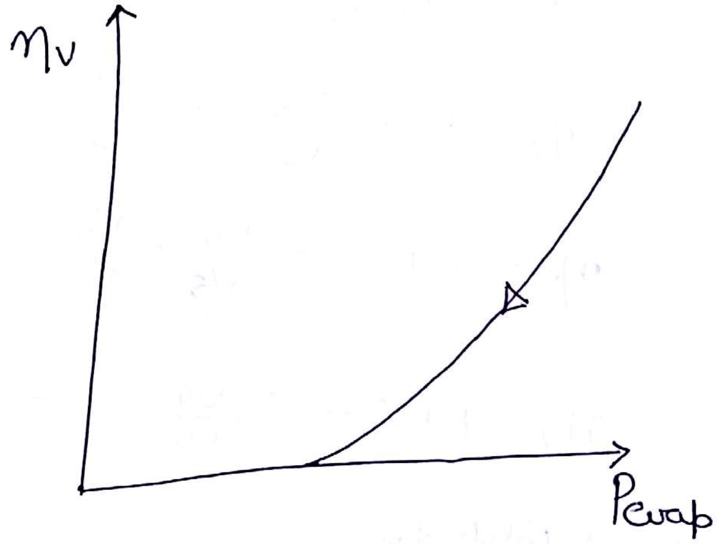
Performance Parameters:-

1) Effect of evaporator Pressure:-

$\downarrow P_E - \downarrow R.E - \uparrow W_{in} - \downarrow COP - \uparrow Pr. Ratio - \downarrow \eta_v - \downarrow \dot{m}$

$$\downarrow \eta_v = \frac{\downarrow \dot{m}\, v_{entry}}{\frac{\pi}{4} D^2 L \times \frac{N}{60} \times K}$$

Reason for mass flow Rate



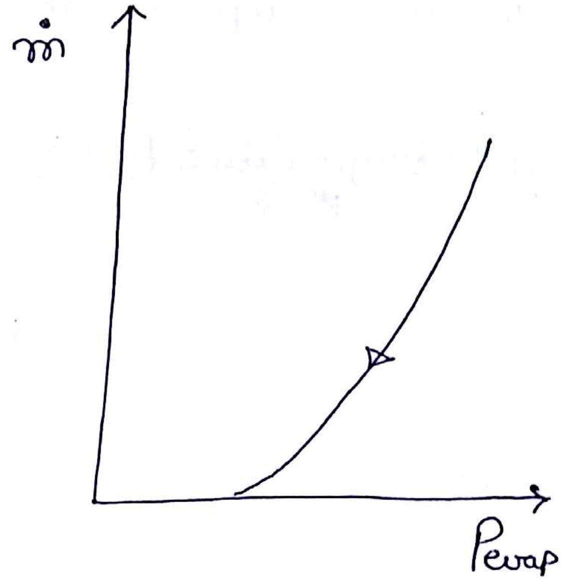
Volumetric Efficiency V/s evaporator Pressure

Here graph is Plot from P=1

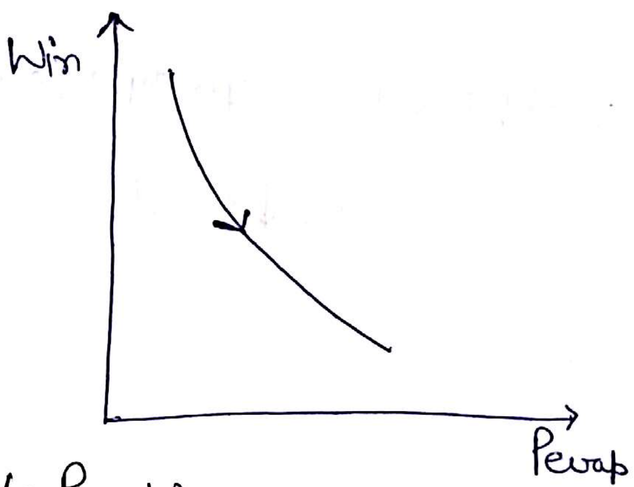
Bcz Pevaporator must be Same or equal to evaporator Pressure. bcz, Below Pevp. it becomes Vaccume

2) Mass flow Rate V/s evaporator pressure :-

$$\downarrow \eta_V = \frac{\downarrow \dot{m}\ v_{entry}}{\frac{\pi}{4} D^2 L \frac{N}{60} \times K}$$



3) Workin to Compressor V/s Pevap :-

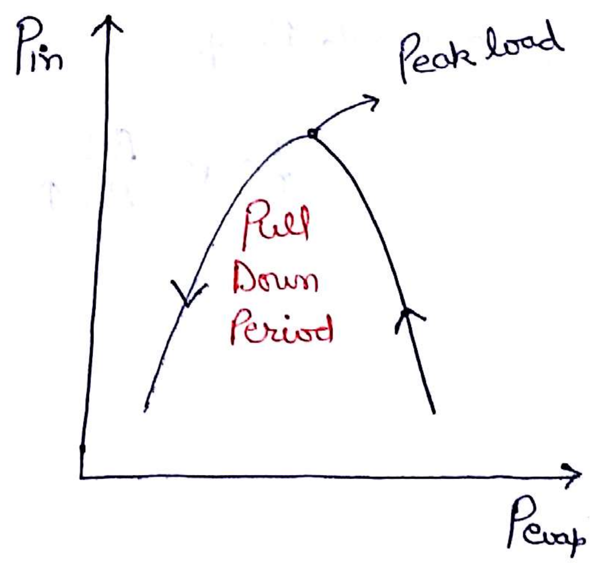


4) Power Input to the Compressor V/s Pevap :-

With the ↓ Peva., the mass flow Rate ↓ (ṁ) & (Win) to the Compressor increases.

Therefore, initially with the ↓ in Pevap., the Power input to the Compressor is initially increases, then reaches to its peak value & finally decreases. This Period is Known as Pull Down Period

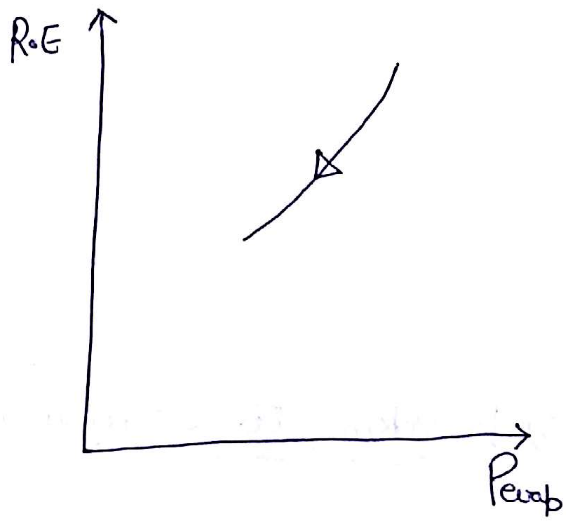
$$P_{in} = \downarrow \dot{m} \times W_{in} \uparrow$$



NOTE:-

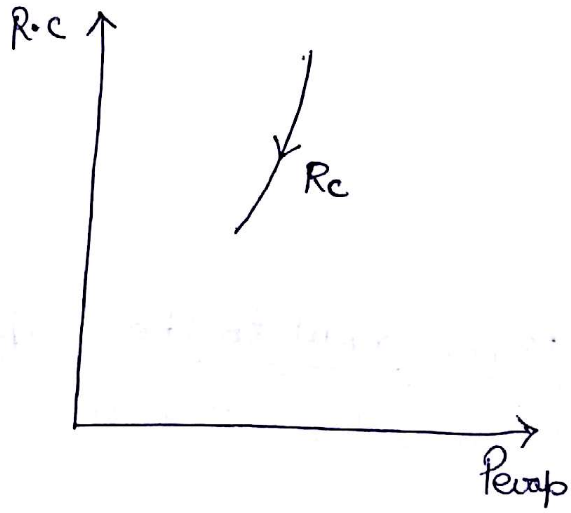
1) The Compressor should be design for Peak loads/Peak hours.

5) Refrigeration Effect V/s evaporation (Evaporator) Pressure:



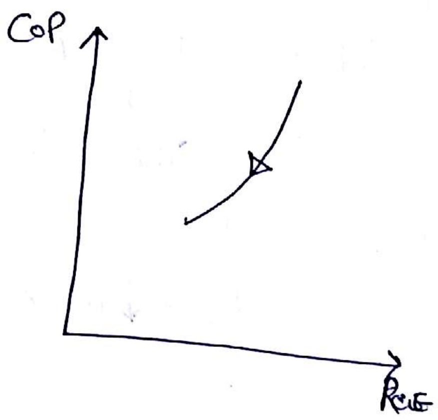
6) Refrigeration Capacity V/s evaporator Pressure:-

$Rc = \downarrow \dot{m} \times \downarrow R.E$



7) COP V/s Pevap

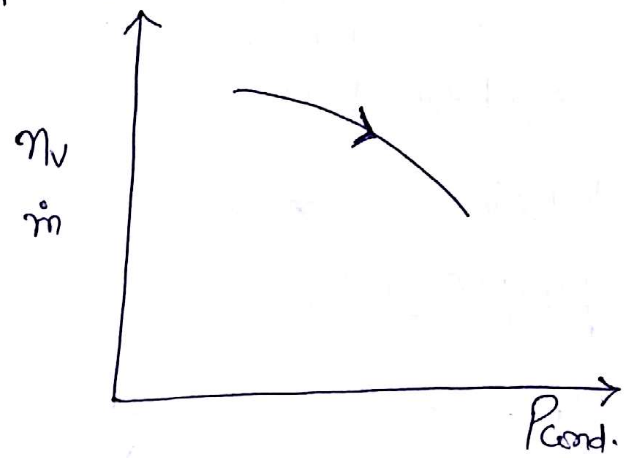
$\downarrow COP = \frac{R.E}{R.E \uparrow}$

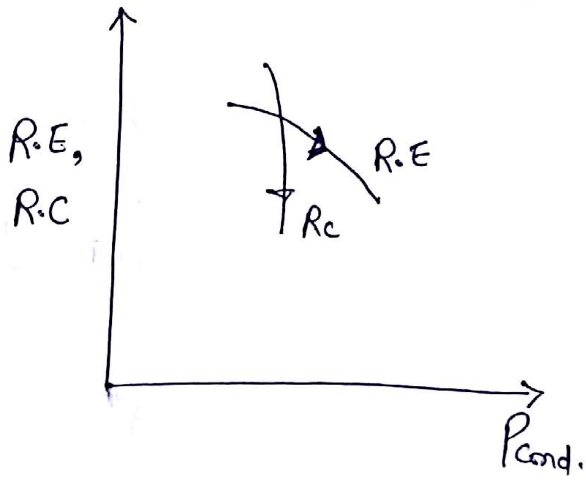


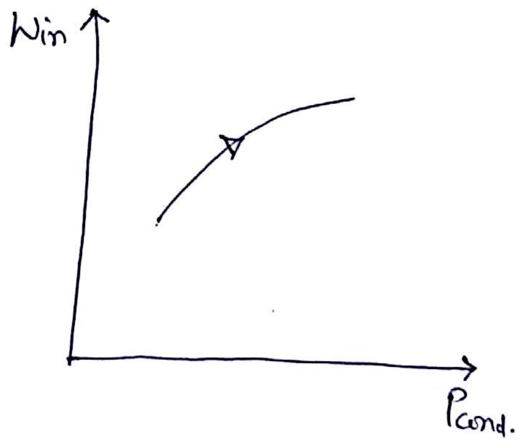
* Effect of Evaporator Pressure:-

$\downarrow P_E \rightarrow \uparrow P_c \rightarrow \downarrow RE - \uparrow W_{in} - \downarrow COP - \uparrow Pr\ Ratio - \downarrow \eta_v - \dot{m}\downarrow$

$\longrightarrow \eta_v, \dot{m}$ v/s P_{cond}







NOTE:

1) Effect of ↑ in P_{cond} and ↓ in P_{evap} are adjactly Same. but We are more Sensitive towards evaporator pressure because of Desired Condition which is Refrigeration.

NOTE:-

PURGING:

It is the Removal of Air, from the Condensor. Air is non Condensible Gas. and it has poor heat Transfer. Coefficient. Therefore it offers more thermal Resistance and hence Reduces the Performance of VCRS.

$$\downarrow Q = \downarrow KA \frac{dT}{dx}$$

$$\uparrow R_{Th} = \frac{1}{\downarrow KA}$$

Types of Resiprocating Compressor :-

1) Open Type :-

In this Type both Compressor and motor are the Seprate units and are Connected by the ~~main source~~ means of open belt drive. The Chances for the leakage of Refrigerant are high but its maintenance is Very easy. Because the Compressor and Motor the Seprate units.

2) Semi hermitically Sealed :-

In this, the Compressor and motor are placed in a Cylinder Shell With a flexible or removable Cover.

3) Hermitically Sealed :-

In this, the Compressor and motor are placed in the Welded Steel Shell.

The Chances for the leakage of Refrigerant are negligible but its maintenance is Complex.

Disadvantage of Hermitically Sealed Compressor :-

- Its heat Rejection Ratio is high $\boxed{HRR = 1 + \frac{1}{COP}}$ because of its Lower COP.

$$\boxed{HRR\uparrow = 1 + \uparrow\left(\frac{1}{COP\downarrow}\right)}$$

Condensor And Evaporator :-

→ Both are the Heat exchanger, with the Same Refrigerant as a Common medium.

→ In Condensor, Refrigerants Rejects it heat. Whereas, In evaporator Refrigerants absorbs the heat.

Types of Condensor :-

1) Air Cooled ~~Compressor~~ Condensor. 3 TR $C_p = 1.005$ KJ/kg-K

② H_2O/Water Cooled ~~Compressor~~ Condensor. $C_p = 4.187$ KJ/kg-K

- Double Tube Type — 10 TR
- Shell & Coil Type — 50 TR
- Shell & tube — 1000 TR

Air Cooled → 3 TR

H_2O Cooled →
- Double tube → 10 TR
- Shell & Coil → 50 TR
- Shell & tube → 1000 TR

(capacity increasing ↑)

③ Evaporative Type Condensor :-

These are generally Preferred When Water is not available in Large Quantity. Or When there is Scarcity of Water.

In this type, First Absorbs Heat from Refrigerant & then Water in turn Rejects its heat to Air.

For example → Cooling Tower.

Expansion Device :-

→ The Function Of the Expansion Device, is to Reduce the Pressure from Condensor to evaporator.

→ It will supplies the flow of Refrigerant to the evaporator as per desired capacity.

Types of Expansion Device :-

1) Constant Area Type :-

Example; Capillary Tube.

It is a narrow tube of Constant Crossection Area.
It is used for Low Capacity Application.
e.g; Domestic Refrigerator, Water Cooler & Window AC.

The Pressure drop in the Capillary tube is directly Propotional to the Length of Capillary tube and inversely Propotional to the diameter of Capillary tube. The pressure drop in the Capillary tube is achieve through frictional Resistance and accelaration of the fluid in the tube.

2) Constant volume Type :-

(1) Automatic Expansion Device :-

It is used to maintain Constant pressure in the evaporator irrespective of Load.

(2) Thermostatic :

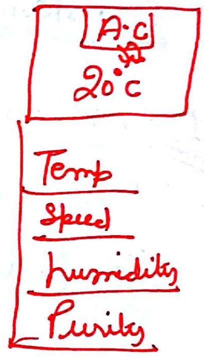
It is used to main, Constant degree of Superheat in the evaporator irrespective Of Load.

Air Conditioning

1. Temperature of Air
2. Purity
3. velocity
4. humidity

Air Conditioner

use → to Control / maintain lower Temperature
→ Purity of Air
→ velocity of Air
→ humidity



Cooler → temp maint ✗↓
→ Purity ✗↓
→ accel ✗↑
→ hum ✓↑

Air Conditioning :- New Topic

It is the Simultaneous Control of the temperature of Air, Purity of air, velocity of air and humidity of air.

Psychrometry :-

It is the Branch of the Science, which deals with the Study, Properties of Moist air.

Moist Air is the Composition of Dry Air and Water Vapour.

Dry Air is the Pure Substance. But Moist air is impure Substance. Because the % of Water Vapour Content Varies from place to place. As we have seen that at some places, there is high humidity and at some places there is Low humidity.

NOTE :-

Generally moist air is in Superheated State.

Various Psychrometric Terms :-

1) Specific humidity :- / Humidity Ratio :- (ω)

→ It is defined As the Ratio of Mass of Water Vapour per kg of dry Air in a given volume and at a Same Temperature.

(kg/kg of d.A) Imp

$$\omega = \frac{m_v}{m_a} = \frac{0.622 P_v}{P - P_v}$$

Mole. Wt. Vapour = 18
Molecular Wt. air = 29
[Which is for dry as well atmospheric air]

Imp GATE

gm/kg of d.A = 10^{-3} kg/kg of d.A

$$\frac{m_v}{m_a} = \quad V_v = V_a = V, \quad T_v = T_a = T$$

$$PV = mRT$$

$$\frac{P_v V_v}{P_a V_a} = \frac{m_v R_v T_v}{m_a R_a T_a}$$

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

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

$$\frac{m_v}{m_a} = \frac{P_v}{P_a} \times \left(\frac{\bar{R}/\text{mole. Wt}}{\bar{R}/\text{mole. Wt}}\right)$$

$$\frac{m_v}{m_a} = \frac{P_v}{P_a}\left[\frac{\text{mol. wt } a}{\text{mole. wt. } v}\right]$$

$$\frac{m_v}{m_a} = \frac{18}{29} \times \frac{P_v}{P_a}$$

$(P = P_a + P_v)$ dalton's Law of Partial Pressure.

2) Relative humidity Φ :- (Φ)

It is defined as the Ratio of Mass of Water Vapour to the mass of Water Vapour under Saturated Condition in a given volume and at a same temperature.

Imp objective

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

NOTE :-

The Specific humidity indicates the actual amount of Water Vapour present in the air.

Whereas

Relative humidity indicates indirectly the moisture absorption Capacity of the Present air.

3) Dry Bulb Temperature :-

It is the temperature of moist air, measured by ordinary Thermometer.

4) Wet Bulb Temperature :-

It is the temperature shown by Thermometer whose bulb is Covered with wet cloth.

5) Wet Bulb Depression :-

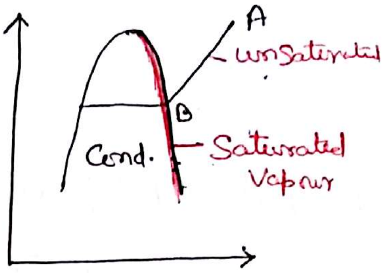
It is the difference b/w DBT & WBT.

6) Dew Point Temperature :-

It is the Saturation temperature, Corresponding to the initiation of Condensation or Water Particles Just Start to Condense.

OR

It is the Saturation Temperature Corresponding to the Partial Pressure of Water Vapour.



NOTE :-

i) In Case of unsaturated air.

$$DBT > WBT > DPT$$

ii) In case of Saturated air, all the temperature are equal.

$$DBT = WBT = DPT$$

$\phi = 1$ or 100%

iii) When the air is fully Saturated, the Value of Relative humidity is 1 or 100%.

iv) In Case of Saturated air, Value of Wet Bulb depression is zero.

~~v) Sling~~

v) Sling Pyschrometer measures Both DBT as well as WBT.

7) Degree of Saturation / Percent humidity : (μ)

Objective

$$\mu = \frac{W}{W_s} = \frac{P_v}{P_{vs}}\left(\frac{P - P_{vs}}{P - P_v}\right)$$

$$\mu = \frac{W}{W_s} = \frac{\frac{0.622 P_v}{P - P_v}}{\frac{0.622 P_{vs}}{P - P_{vs}}} = \frac{P_v}{P_{vs}}\left(\frac{P - P_{vs}}{P - P_v}\right)$$

8) <u>Enthalpy of Moist Air</u>:-

$$h_{mA} = 1.005t + \omega(2500 + 1.88T) \text{ KJ/kg of d.a.}$$

$t \rightarrow$ DBT (°C)

$\omega \rightarrow$ Kg/Kg of d.A

9) <u>APJON Formula</u>:-

It is used to Calculate the Partial pressure of Water Vapour.

<u>Objective</u>

$$P_v = P'_v - \frac{1.8P(t - t')}{2700}$$

<u>NOTE</u>:-

1) $t \rightarrow$ Dry Bulb temperature taken in °C.

2) $t' \rightarrow$ Wet Bulb Temperature taken in °C

3) $P_v \rightarrow$ Partial Pressure of Water Vapour corresponding to P_v, the Saturation temperature, Provides the values of <u>Dew Point temperature</u>.

4) $P_v' \rightarrow$ It is the Saturation Pressure Corresponding to <u>Wet Bulb temperature</u>.

5) $P_{vs} \rightarrow$ Partial Pressure of Water Vapour under Saturated Condition, Corresponding to P_{vs}, the Saturation Temperature Provides the Value of <u>dry Bulb temperature</u>.

6) $P \rightarrow$ It is the total Pressure or if total Pressure not provided then taken as atmospheric Pressure.

P = Total Pressure

$P_{atm} = 1.01325$ bar

101.325 KPa

$1.01325 \times 10^5 \frac{N}{m^2}$ or Pascal

Qn) The Dry Bulb Temperature and Wet Bulb temperature of Air are 30°C and 20°C Respectively. The atmospheric Pressure is 740 mm of Mercury (Hg). Determine, (2 Mark fixed on Gate)

i) Partial Pressure of Water Vapour
ii) Specific humidity (ω)
iii) Relative humidity (ϕ)
iv) Degree of Saturation (H)
v) Enthalpy of Moist Air (h_{ma})
vi) Vapour density.

740 of Hg.

	Sat. Temp	Sat. Pressure (Bar)
D.BT	30°C	0.04242
WBT	20°C	0.02337

$$\omega = \frac{m_v}{m_a} = \frac{0.622 P_v}{P - P_v}$$

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

$$H = \frac{W}{W_s} = \frac{P_v}{P_{vs}}\left(\frac{P - P_{vs}}{P - P_v}\right)$$

$$P_v = P_v' + \frac{1.8 P (t - t')}{2700}$$

$$h_{mA} = 1.005 t + \omega(2500 + 1.88 T)$$

$t \rightarrow$ Dry Bulb temp
$t' \rightarrow$ Wet Bulb Tem
$P_v \rightarrow$ Partial Pressure $\rightarrow$ DPT
$P_v' \rightarrow$ Saturation Pressure $\rightarrow$ WBT
$P_{vs} \rightarrow$
$P =$ TP.

Soln)
DBT = 30°C
WBT = 20°C
Patm = 740 mm

$$P_v = 0.02337 - \frac{1.8 \times P(30° - 20°)}{2700}$$

$$P_v = 0.02337 - \frac{1.8 \times 0.9875 (10°)}{2700}$$

$$= 0.02337 - 0.006583$$

$$\boxed{P_v = 0.0167 \text{ bar.}}$$

$P = \rho g h$
$= 13.6 \times 1000 \times 9.81 \times 740 \times 10^{-3}$
$= 0.9875 \times 10^5 \ N/m^2$
$= 0.9875$ bar

ii) Specific humidity

$$\omega = \frac{m_v}{m_a} = \frac{0.622 P_v}{P - P_v}$$

$$\omega = \frac{0.622 \times 0.0167}{0.9875 - 0.0167} = \frac{0.01044}{0.9708} = 0.01075$$

$$\boxed{\omega = 0.01075}$$

iii Relative humidity $\phi = \frac{P_v}{P_{vs}}$

$$\phi = \frac{0.0167}{0.04242}$$

$$\boxed{\phi = 0.3936}$$

iv) Degree of Saturation → H

$$H = \phi\left(\frac{P - P_{vs}}{P - P_v}\right)$$

$$H = 0.3936\left(\frac{0.9875 - 0.04242}{0.9875 - 0.0167}\right)$$

$$H = 0.385$$

v) Enthalpy of Moist air, (KJ/kg)

$$h = 1.005t + \omega(2500 + 1.88t)$$

$$= 1.005 \times 30 + 0.01075(2500 + 1.88 \times 30)$$

$$= 54.14 \text{ KJ/kg}$$

vi) Vapour Density :-

$$\omega = \frac{m_v/V}{m_a/V}$$

$$\boxed{\omega = \frac{\rho_v}{\rho_a}}$$

$$\boxed{\rho_v = \omega \rho_a} \text{ imp}$$

$$P_V = mRT$$

$$P = \left(\frac{m}{V}\right)RT$$

$$\boxed{P = \rho RT}$$

$$\rho = \frac{P}{RT}$$

$$\rho_v = \omega\left(\frac{P_a}{R_a T_a}\right)$$

T → in Kelvin

N/m^2

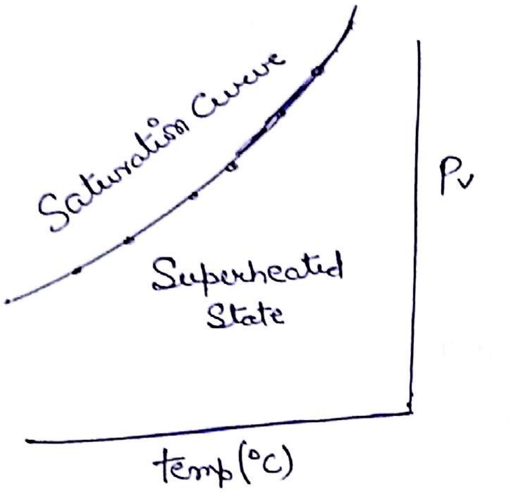
Dalton's law

$$P = P_a + P_v$$

$$\rho_v = \omega\frac{(P - P_v)}{R_a T_a} = 0.0107\frac{(0.9875 - 0.0167)}{0.287 \times 10^3 \times (273 + 30)} = 0.0119 \text{ kg/m}^3$$

Psychrometry Chart :-

We Know that, as a temperature (Saturation) ↑, the Saturation Pressure also ↑. So the Plot b/w Saturation temperature & Pressure is,



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

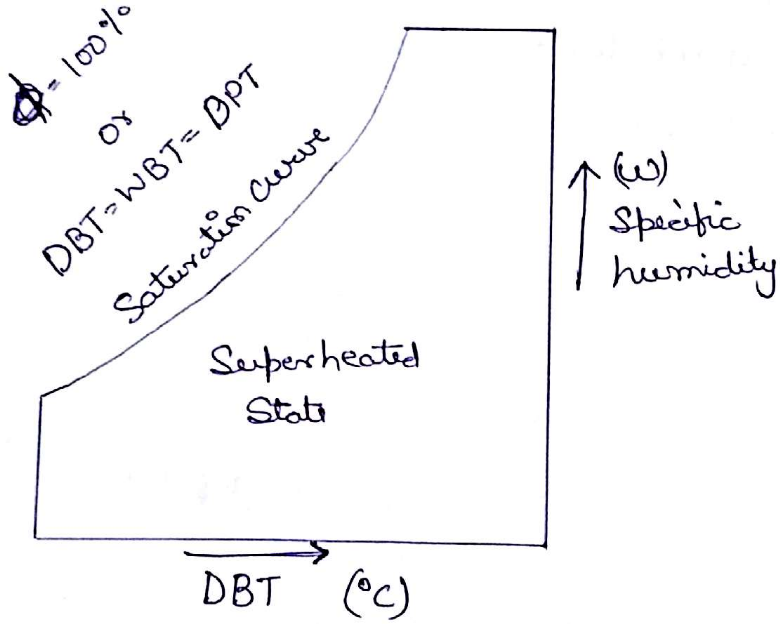
$$W = F(P_v)$$

For Superheated

$PV = mRT$

$P \propto T$

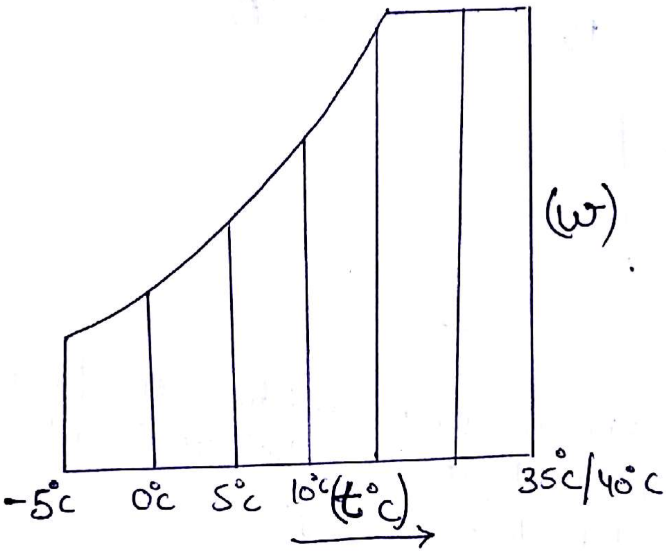
Later on, We found that the Specific humidity is the function of Partial pressure of Water Vapour, therefore in the original Psychrometry Chart (P_v) is Replaced with (ω) Specific humidity.



Kg / Kg of dry air

Representation of Different Constant Parameters on Psychrometric Chart :-

1) Constant Dry Bulb Temperature Line DBT :-



→ These are the vertical lines.

→ Increasing order is (+x) direction.

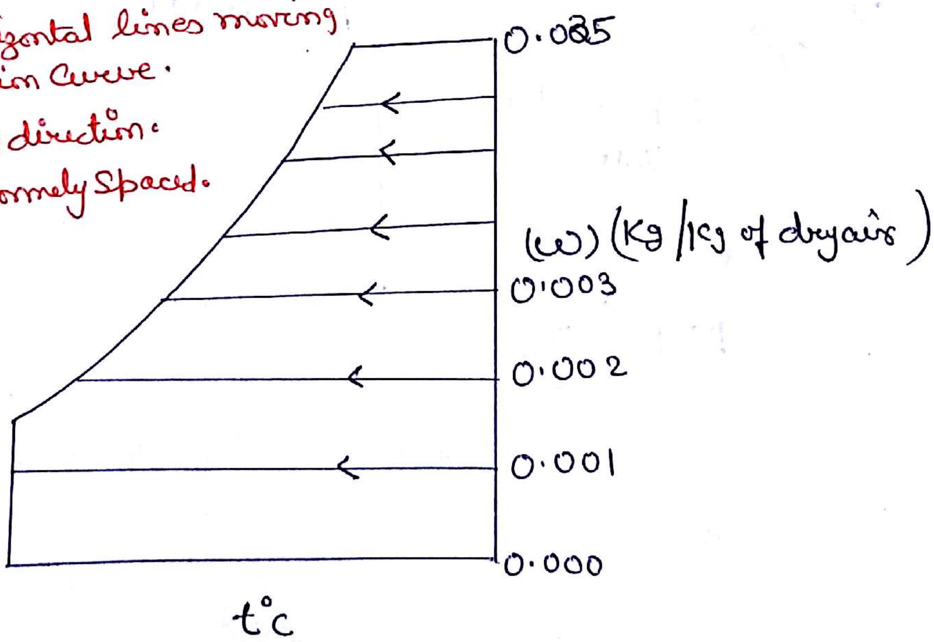
→ These are uniformly spaced.

2) Constant Specific humidity lines :- (ω)

→ These are Horizontal lines moving towards Saturation Curve.

→ ↑ order in (+y) direction.

→ These are uniformly spaced.

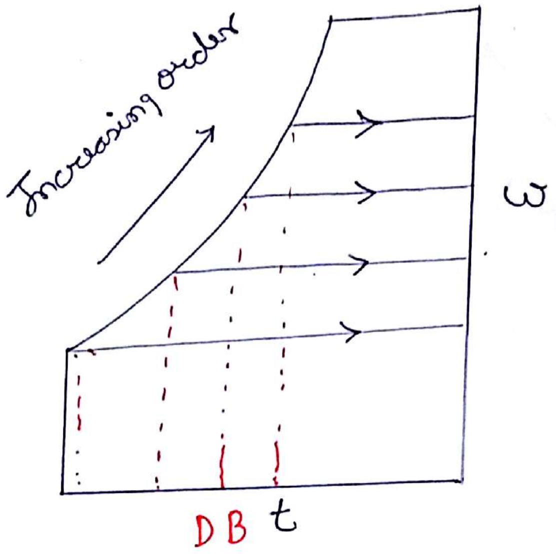


3) Constant Dew Point Temperature Lines DPT :-

→ These are the Horizontal lines moving away from Saturation Curve.

→ These are non uniformly Spaced.

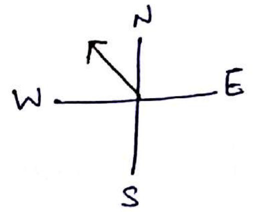
→ Increasing order

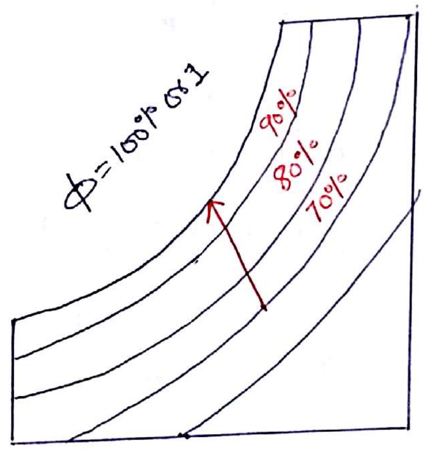


4) Constant Relative humidity Curve :-

1) These are Parallel to Saturation Curve.

2) increasing order in North-West direction.





5) Constant Enthalpy lines, Constant Wet Bulb temperature line, Constant Specific Volume lines :-

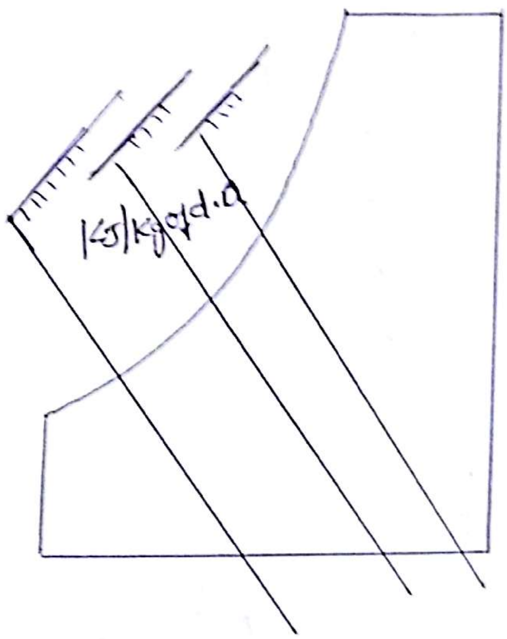
1) h / WBT / v

h ↓ KJ/Kg

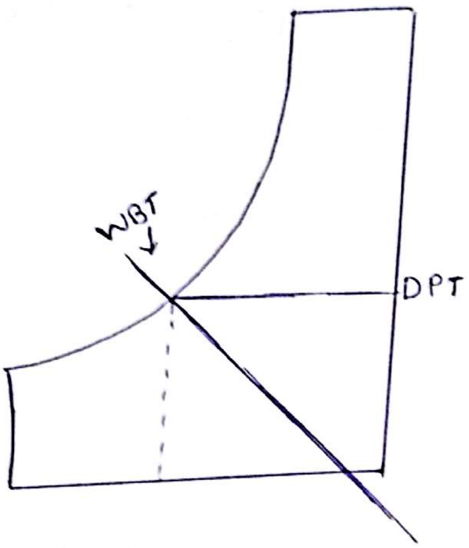
WBT ↓ °C

v ↓ m^3/Kg

② h



③ W.BT

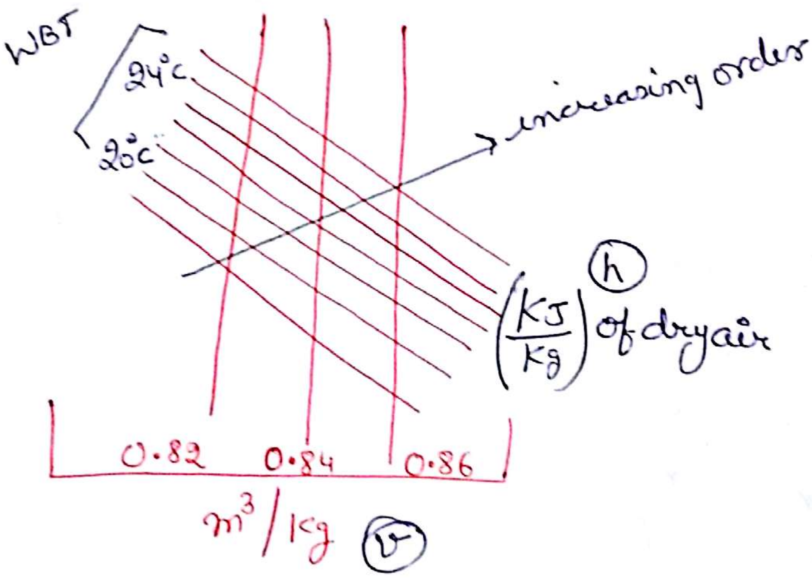


④ h, WBT → Same degree of Inclination

v = highest degree of Inclination

⑤ h → uniformly spaced.

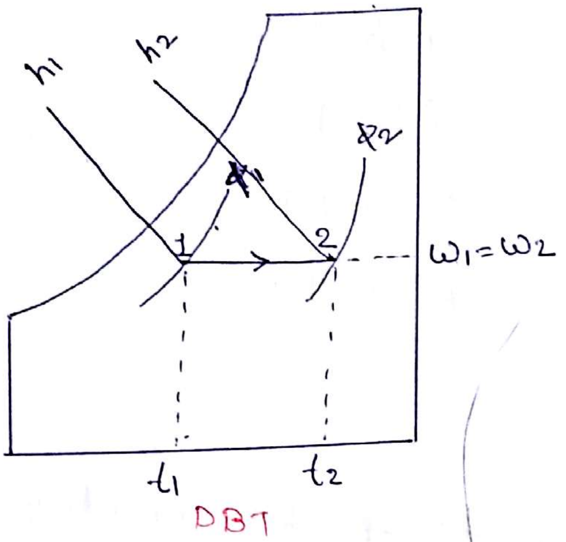
WBT → Non uniformly spaced.



Basic Psychrometry Process :-

1) Sensible Heating :-

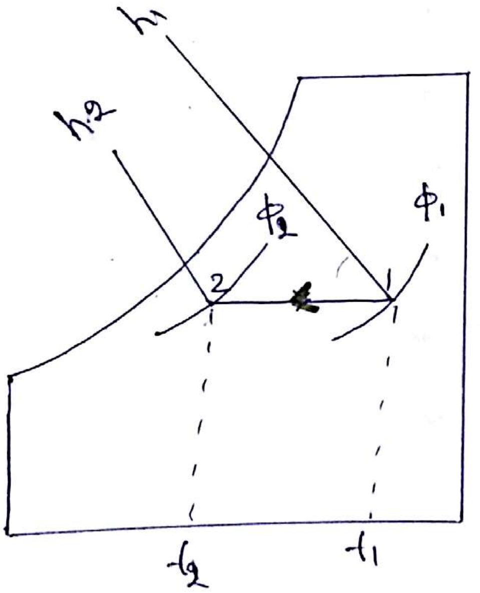
It is the Process of ↑ the dry bulb temperature at Constant Specific humidity.



1. t ↑
2. W Const
3. DPT Const
4. φ ↓
5. h ↑
6. WBT ↑
7. v ↑

2) Sensible Cooling :-

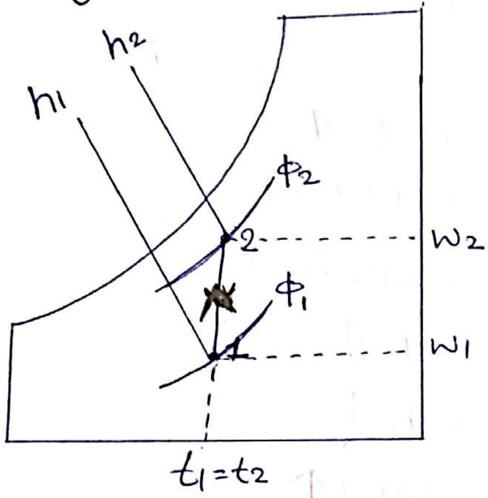
It is the Process of ↓ the temperature (DBT) at Constant Specific humidity.



1. t ↓ (DBT)
2. W Const
3. DPT Const
4. φ ↑
5. h ↓
6. WBT ↓
7. v ↓

③ Humidification:-

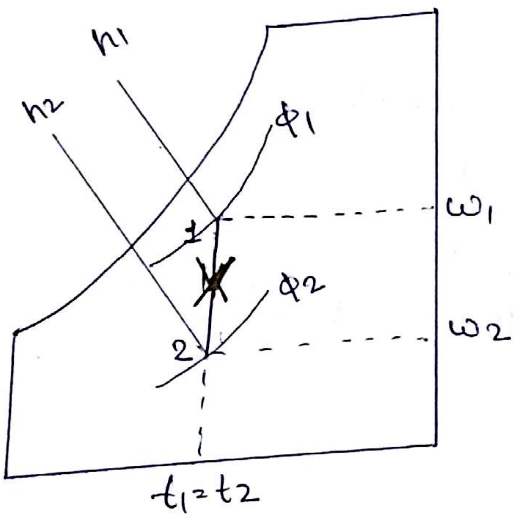
It is the Process of ↑ the Specific humidity at Constant dry Bulb temperature.



① t Const
② ω ↑
③ DPT ↑ Dew Point Temp
④ φ ↑
⑤ h ↑
⑥ WBT ↑
⑦ v ↑

④ Dehumidification:-

It is the Process of decreasing the Specific humidity at Constant dry Bulb temperature.

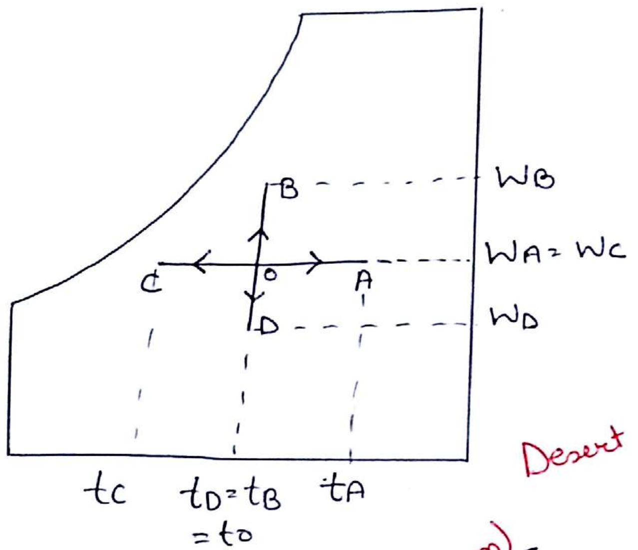


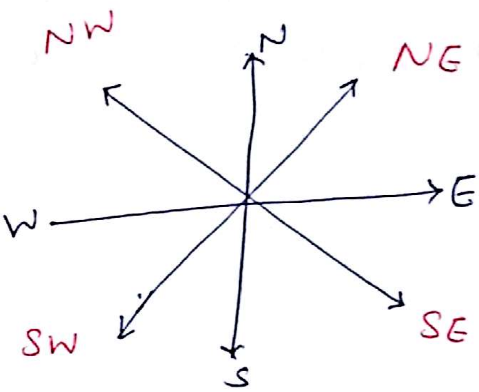
① t Constant
② ω ↓
③ DPT ↓
④ φ ↓
⑤ h ↓
⑥ WBT ↓
⑦ v ↓

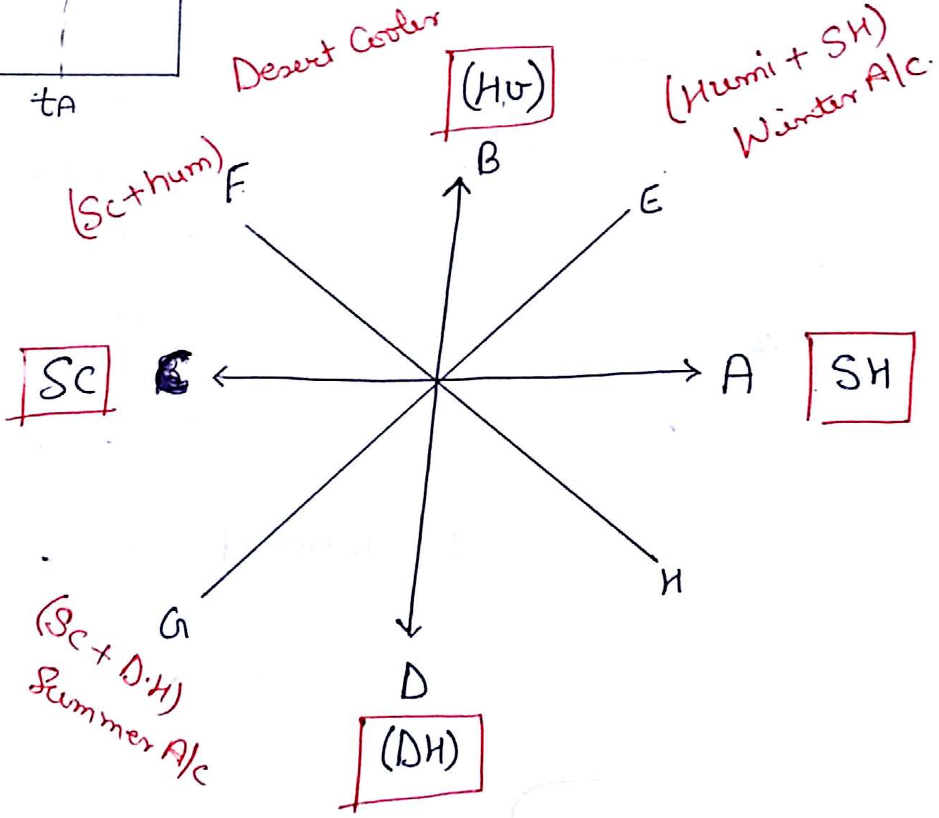
NOTE :-

Pure humidification & Dehumidification are impossible to achieve Practically.
Therefore these are Combined either with Sensible heating or Sensible Cooling.

Representation of all the above Process on Same Psychrometric chart :-







NOTE :-

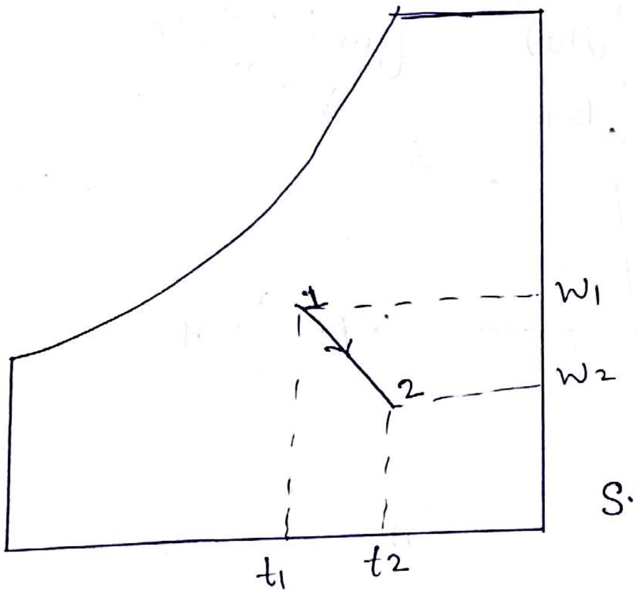
In Case of Summer air Conditioning the process of Cooling & Dehumidification.

Whereas in case of Winter air Conditioning, The Process of heating & humidification Occurs.

→ In Case of desert Cooler the process of Cooling & humidification or Adiabatic Saturation process occurs

★★ → Desert Coolers are most effective when the Value of Wet bulb depression is high. (important in terms of question / Interview)

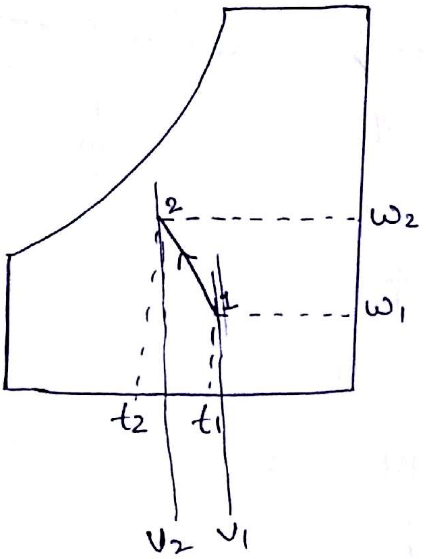
→ 4. Point



① t ↑
② ω ↓
③ DPT ↓
④ φ ↓
⑤ h
⑥ WBT → We Can't Say
⑦ v

S.H + Dehumidification

Adiabatic or chemical humidification :-

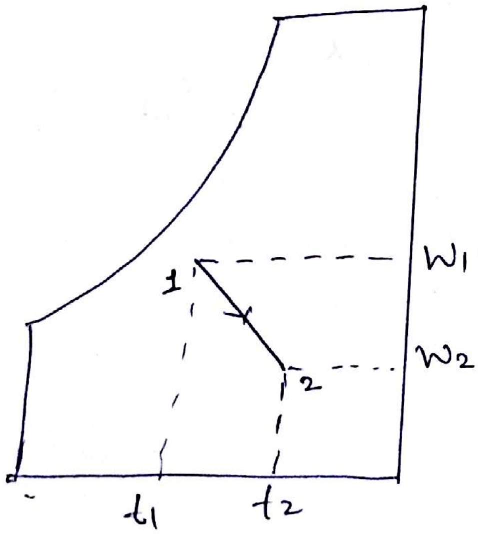


Adiabatic humidification → h → Constant

hu/DH → Decides direction

1. t ↓
2. W ↑
3. DPT ↑
4. ϕ ↑
5. h Const
6. WBT Const
7. v ↓

Adiabatic chemical Dehumidification :-

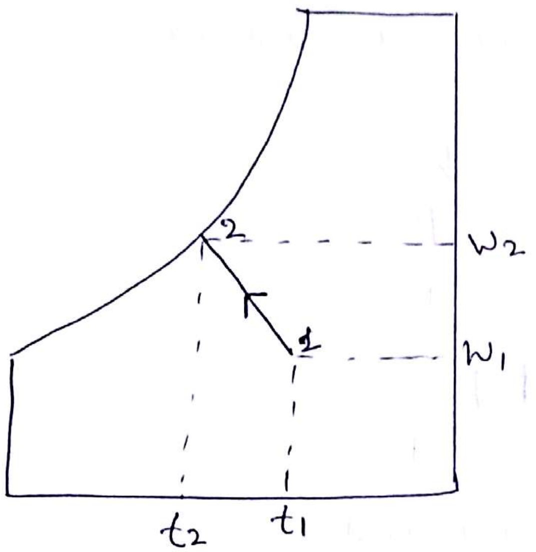


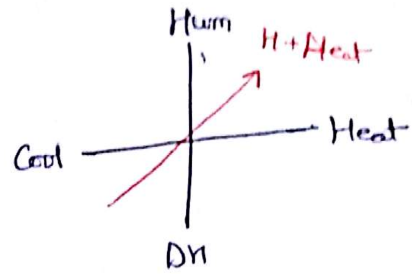
Ad/DH → h → Constant

DH → decides Direction

1. t ↑
2. W ↓
3. DPT ↓
4. ϕ ↓
5. h Const
6. WBT Const
7. v ↑

Adiabatic Saturation :-

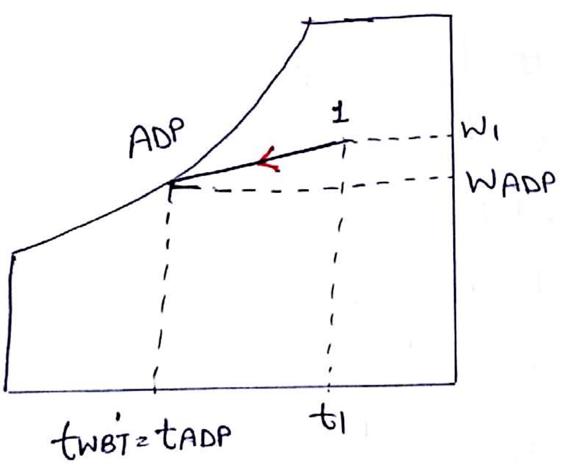




1. t ↓
2. w ↑
3. DPT ↑
4. φ ↑
5. h Const
6. WBT Const
7. v ↓

APPratus Dew Point :- (ADP)

It is the Point obtained by the intersection of Cooling and dehumidification with the Saturation curve.



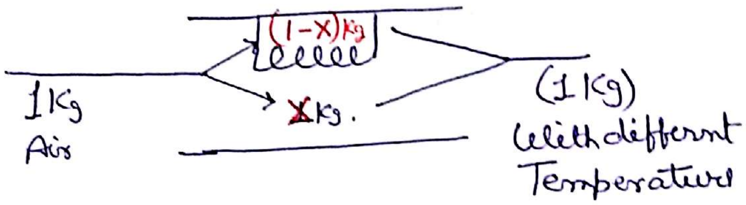
Three Cases are formed

1. Cooling & Dehumidification
2. Saturation curve touch and at Saturation curve
3. $\phi_{ADP} = 1$

By-Pass Factor :- (x)

It Simply Represent the loss.

It Represent the uncontanctid air / it respresents the fractional Part of inlet air which is not Comming in Contact with the Coil.

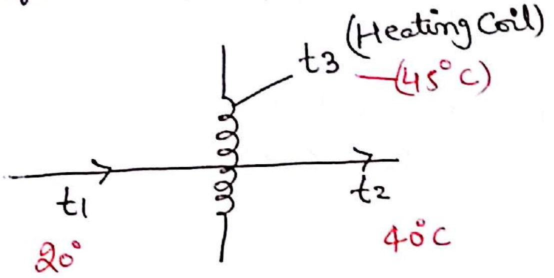


$\eta = 1$

$\uparrow \eta = 1 - BPF \downarrow$

By-Pass factor Of Heating Coil :-

Let, t_1, be the inlet temp. of air, t_2 be the outlet temp. of air and t_3 be the Surface temp. of heating coil.

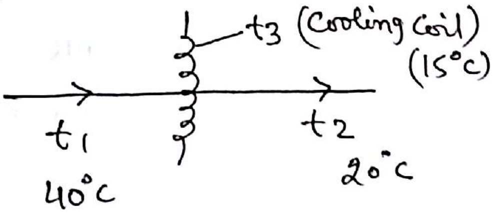


$$(BPF)_{HC} = \frac{t_3 - t_2}{t_3 - t_1} \quad \begin{matrix} \text{— loss} \\ \text{— diff} \end{matrix}$$

$$(\eta)_{HC} = 1 - (BPF)_{HC}$$

By Pass factor Of Cooling Coil :-

Let t_3 be Surface temp. of Cooling Coil.



Important

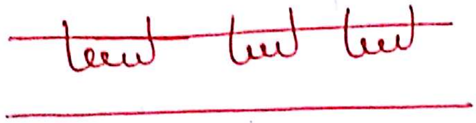
$$(BPF)_{CC} = \frac{t_2 - t_3}{t_1 - t_3}$$

$$\eta_{CC} = 1 - (BPF)_{CC}$$

NOTE

→ By Pass factor in case of Combined Coil (when there is more than one Rows of Coil)

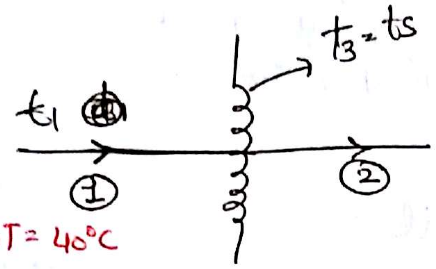
[No such case]



X^N = N – Total no. of Coil

Becaus 1 Heat ② Cool ③ Heat

→



DBT = 40°C
DPT = 20°C

SH, SC → DBT

Humidification → H_2O
dehumidification → DPT

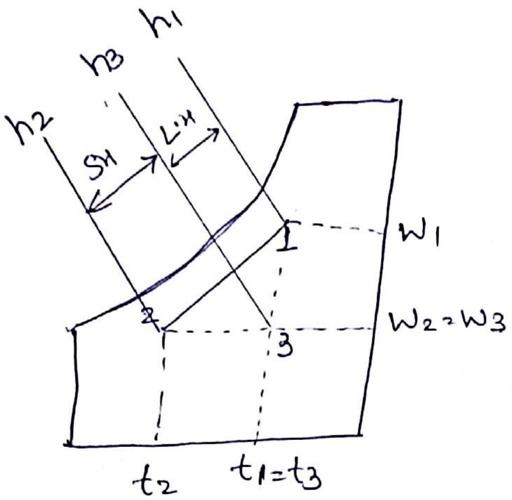
	ts	outlet temp
		SH
①	50°C	SH + Hu
②	50°C H_2O	SH + Hu
③	Steam (100°C)	SC
④	30°C	~~SC + Hu~~
⑤	20°C or 19°C	SC + DH

Sensible Heat Factor :-

It is defined as the Ratio of Sensible heat to the total heat.

~~Total is the~~

Total Heat is the Summission of (SH) & (LH).



$$SHF = \frac{SH}{LH} = \frac{SH}{SH+LH}$$

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

$$SHF = \frac{h_3 - h_2}{h_1 - h_2}$$ Imp

Value of Sensible Heat Factor for different Places :-

Residence & PVT. office	0.9
Resturant & Busy office	0.8
Auditorium & Cinema hall	0.7
Dance hall room	0.6

Effective Temperature :-

It is the temperature of Saturated Air at which human being Can/Would feel Same level of Comfort as in Actual environment. It includes Comfort temperature, humidity, Accelaration of air & velocity.

~~Psychrometry~~

Factors Affecting effective Temperature :-

(1) Climatic and Seasonal Differences :-

Peoples living in Colder climate feeling Comfortable at lower effective temperature than the people living in Warmer Region.

In Summer optimum effective temperature is 21.6°C

Where as in Winter effective temp is 20°C

(2) Age and Gender :-

Childrens and old aged Persons needs ~~to~~ 2–3°C higher effective temperature than Adults.

Similarly in the Case With Womens. Which need 2–3°C ↑.

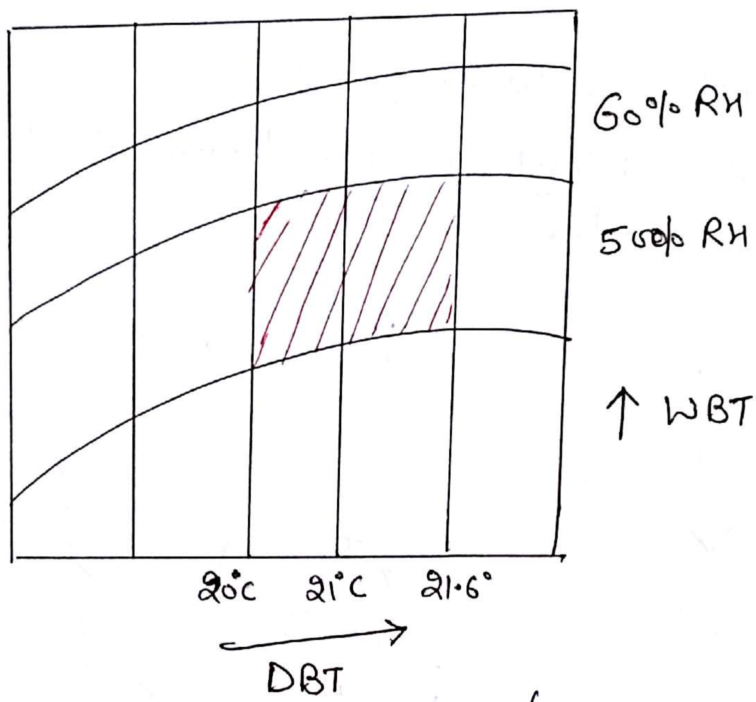
3) Kind Of Activity :-

If a Person is involved in Activities like dancing, foundary Shop ~~B~~ near Boiler furnac etc. means/needs lower effective temp. than the person who are in rest Condition.

4) Density of Occupants:-

Highly Density occupied areas needs lower effective temp. than the less density occupied area.

5) Comfort Chart:-



This chart is Developed by ASHRAE (American Society of Heating, Refrigeration and Air conditioning Engineers).

By Conducting a survey on different kinds of People Subjected to wide range of environmental temperature Condition, humidity and air velocity.

This Chart is developed b/w DBT & WBT which are taken on X & Y axis Respectively.

→ If the Value of Relative humidity is above 50%, then there is tendency of Sticky Sensation develops.

Where as if the Value of Relative humidity is below 50% then the skin is too dry.

Ventilation Air :-

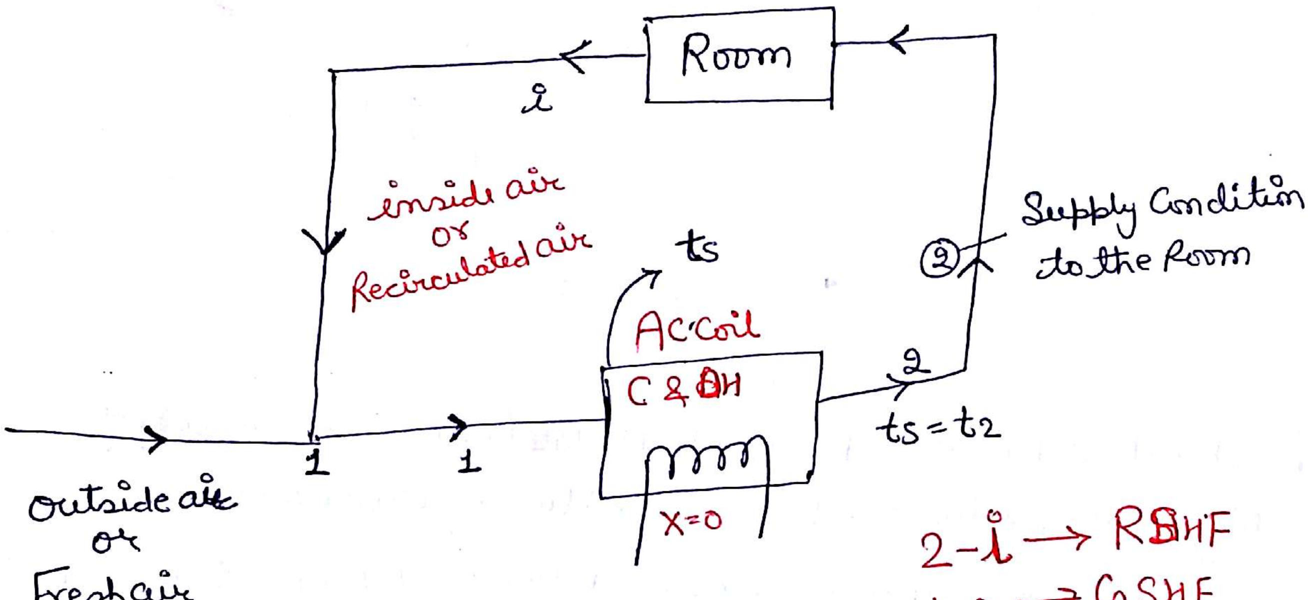
It is the amount of fresh air which is supplied to the A/c Coil, inorder to maintain its ~~its~~ purity.

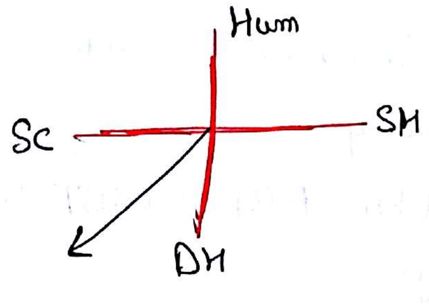
NOTE:-

In Case of operation theatre / ICU, 100% outside air is supplied.

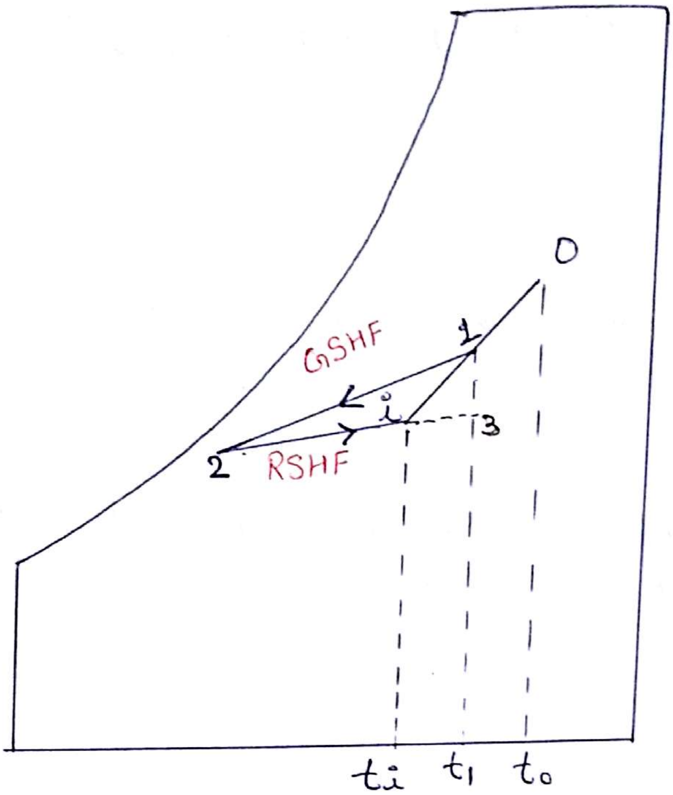
Summer Air Conditioning :

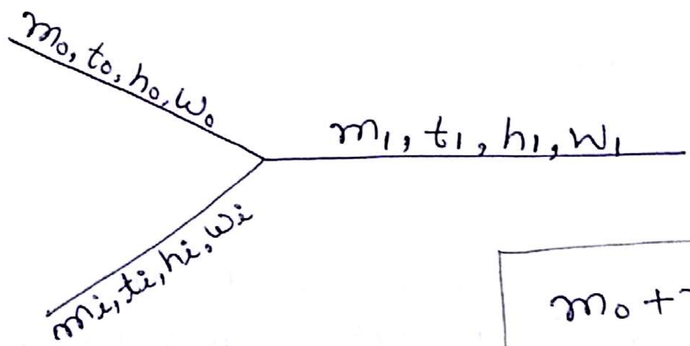
Air is Passing through cooling & dehumidification coil with (0BPF) Zero By-Pass factor.





in Summer
outer temp ↑





$$m_o + m_i = m_1$$
$$m_o h_o + m_i h_i = m_1 h_1$$
$$m_o t_o + m_i t_i = m_1 t_1$$
$$m_o W_o + m_i W_i = m_1 W_1$$

GSHF :

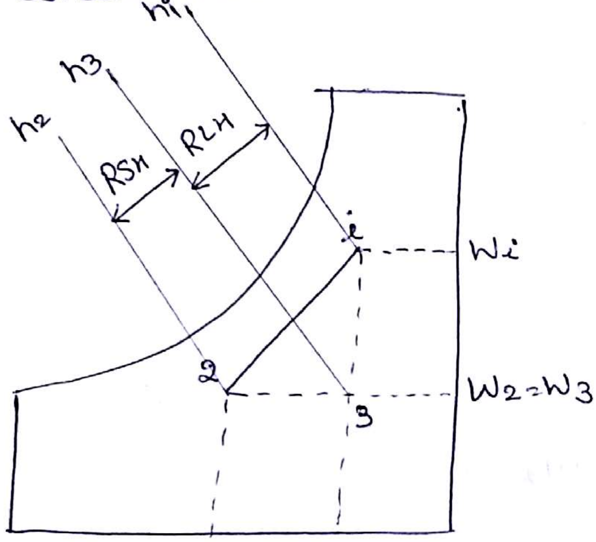
It is the line or Curve obtain by the joining of inlet & outlet of the Coil.

RSHF :-

It is defined as the Ratio of room sensible heat to the room total heat.

or

It is the line or Curve which is obtained by joining the Supply Condition to the room with the inside Condition.



$$R_{SHF} = \frac{R_{SH}}{R_{TH}} = \frac{R_{SH}}{R_{SH}+R_{LH}} = \frac{h_3-h_2}{(h_3-h_2)+(h_i-h_3)}$$

$$\boxed{R_{SHF} = \frac{h_3-h_2}{h_i-h_2}}$$

NOTE :

① $RSH = 0.0204 \, Cmm \, \Delta t$ (KW) (Room Sensible Heat)

② $RLH = 50 \, Cmm \, \Delta \omega$ (KW)

imp $Cmm =$ ______ m^3/min imp it must be in this unit

③ No. of Air flow changes/hr

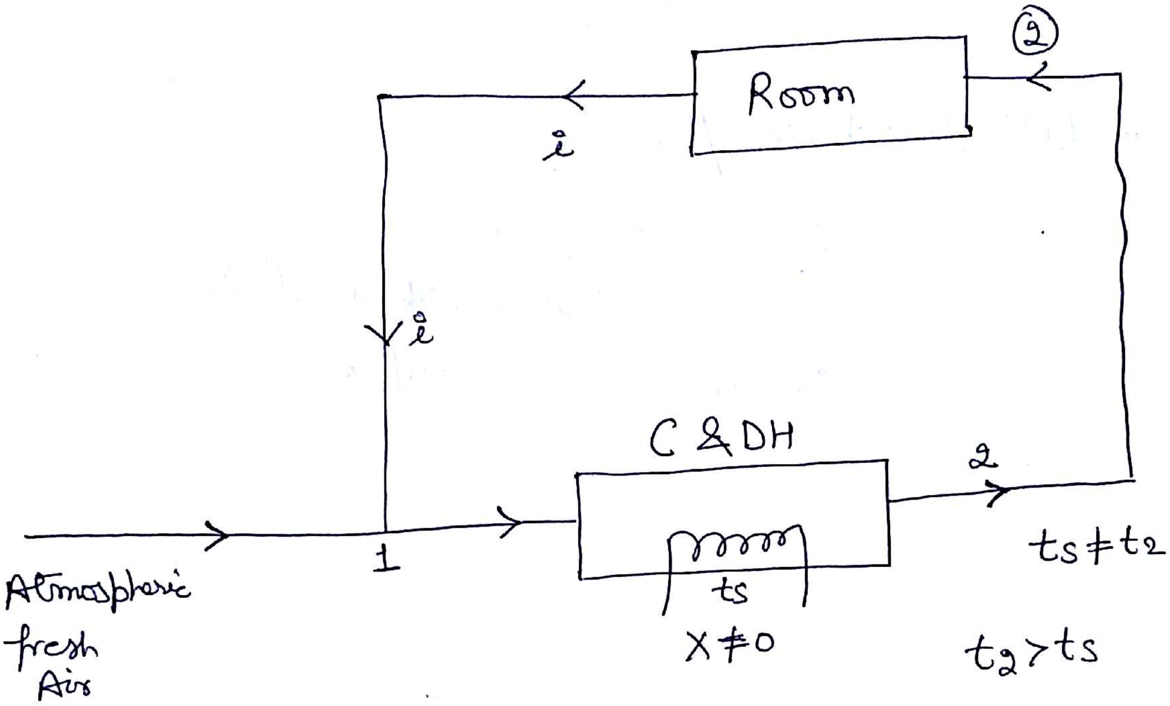
$$= \frac{Cmm}{\text{Volume of Room } (m^3)}$$

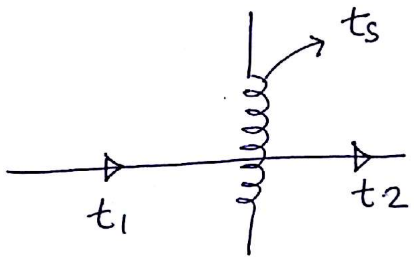
$Cmm =$ ______ m^3/hr

Case 2 :-

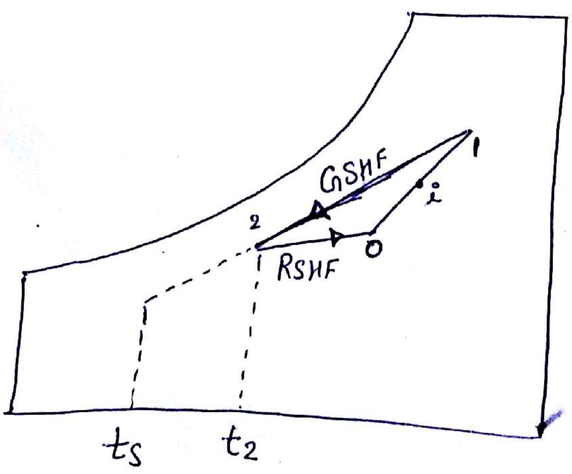
Summer Air Conditioning :-

Air is Passing Through a Cooling and dehumidification Coil with non zero ByPass factor.



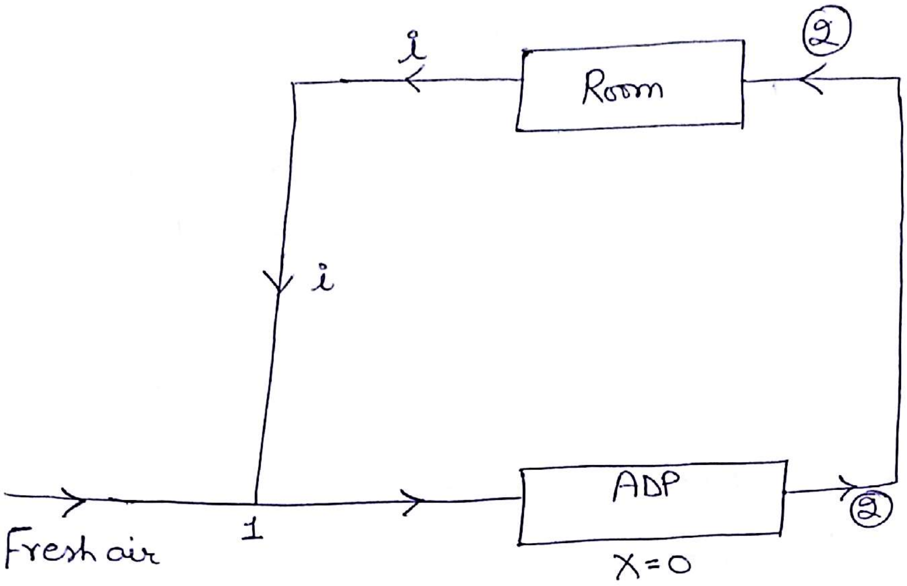


$$X = \frac{t_2 - t_s}{t_1 - t_s} = \frac{h_2 - h_s}{h_1 - h_s} = \frac{\omega_2 - \omega_s}{\omega_1 - \omega_s}$$



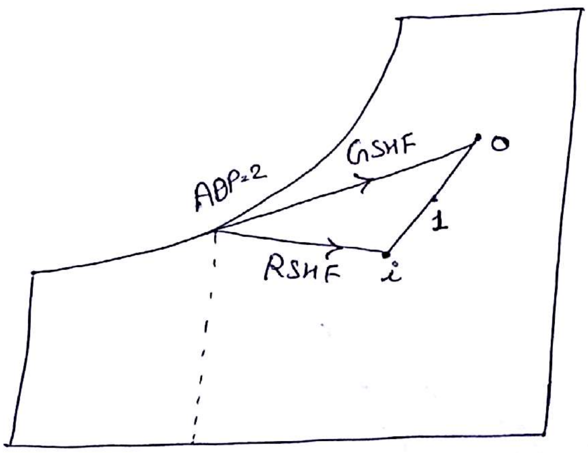
Case-8

Air Is Passing Through a Coil having Some Given Value of ADP (Apparatus dew Point) With Zero By Pass factor:-



Summary

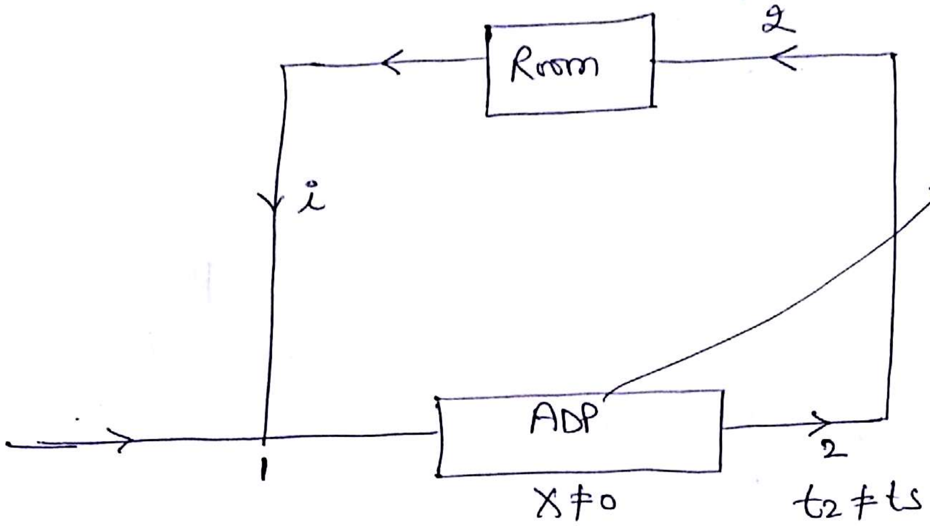
1. C & DH
2. Saturation Curve (ADP) Apparatus Dew Pt.
3. $\phi_{ADP} = 1$
4. $t_2 = t_{ADP}$



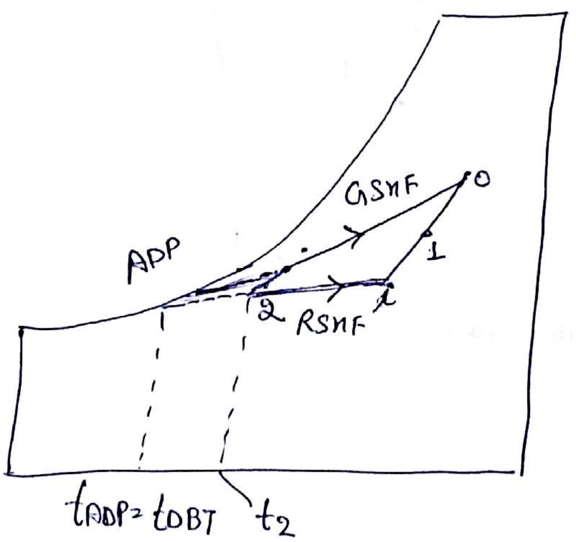
$t_2 = t_{ADP} = t_{DBT}$

Case 4

Air is Passing through a coil having some given value of ADP with non zero by Pass factor :-



1. C & DH
2. $\phi = 1$
3. Saturation Curve
4. $t_2 > t_{ADP}$



NOTE :-

→ Human beings are feeling Comfort b/w 24 - 26°C DBT and 50 - 60% Relative humidity.

→ The Degree of Freedom of moist air is,

$$P + F = C + 2$$

$$1 + F = 2 + 2$$

$$\boxed{F = 3}$$

The degree of freedom of moist air is 3, but we can locate or fix the state of moist air on the chart by using two Variables, Because the chart is developed for the Particular pressure, that is atmospheric pressure.

→ During the Compression of moist air or when moist air is heated in a air tight vessel, then the Specific humidity Remains Constant.

→ Air Washer (+ Filter) Can be used as humidifier, Dehumidifier and Filter.