# ARYAN SCHOOL OF ENGINEERING & TECHNOLOGY

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# LECTURE NOTE

**SUBJECT NAME- REFRIGERATION AND AIR-CONDITIONING**

**BRANCH – MECHANICAL ENGINEERING**

**SEMESTER – 5TH SEM**

**ACADEMIC SESSION - 2022-23**

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# Refrigeration And Air Conditioning

1) Basic Concept (Obj)

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

3) Refrigerant (Theory Obj)

4) Vapour Absorption Refrigeration System (Obj)

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.

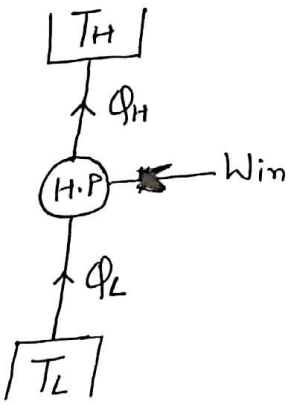
or

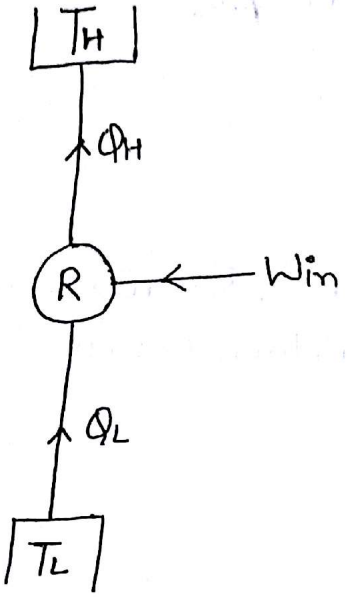
It is defined as Ratio of Refrigeration effect to the Work input.

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

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

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






(Refrigerator)

Objective

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

gmb

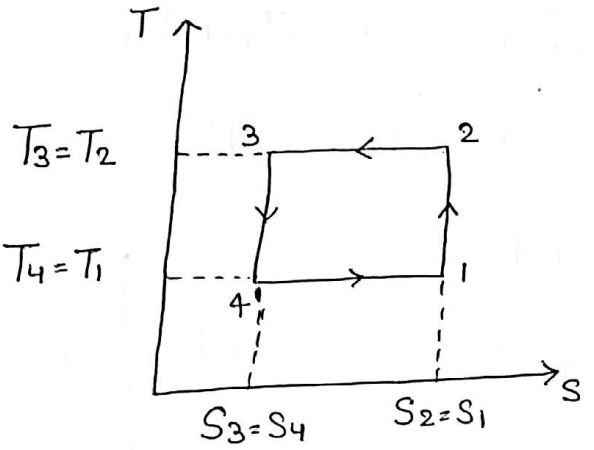
$$(C.O.P.)_{HP} = (C.O.P)_R + 1 = \frac{1}{\eta_E}$$

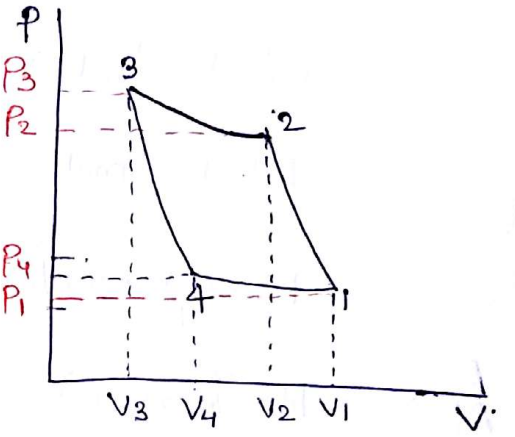
This Relation or expression is applicable b/w the "Same Temperature" limits.

# Ideal Refrigeration Cycle :-

or

# Reversed Carnot Cycle :-




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

$$\boxed{dQ = TdS}$$

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

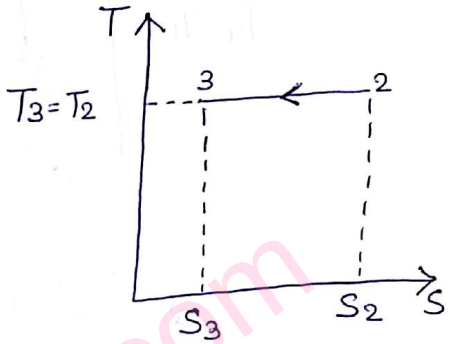
$$C.O.P = \frac{\text{Desired Effect}}{W_{net}}$$

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

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

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

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




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

$dQ_{(2-3)} = -T_H(S_1 - S_4)$ — (2)

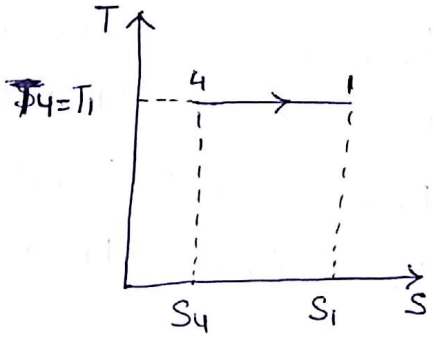
$dQ_{(4-1)} = T(S_F - S_I) = T_L(S_1 - S_4)$ — (3)

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.

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

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

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

Objective

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

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 |
|---|---|
| $T_L = 0°C$ | $T_L = 0°C$ |
| $T_H = 30°C$ | $T_H = 10°C$ |

Ans.

| | |
|---|---|
| $T_H \downarrow$ | COP ↑ |
| $T_L \uparrow$ | COP ↑ |

$T_L$ = Constant

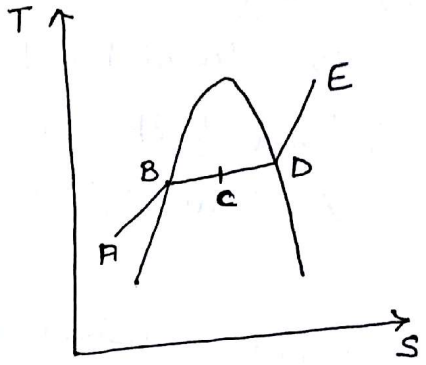
$$(T_H)_S > (T_H)_W$$

$$(COP)_S < (COP)_W$$

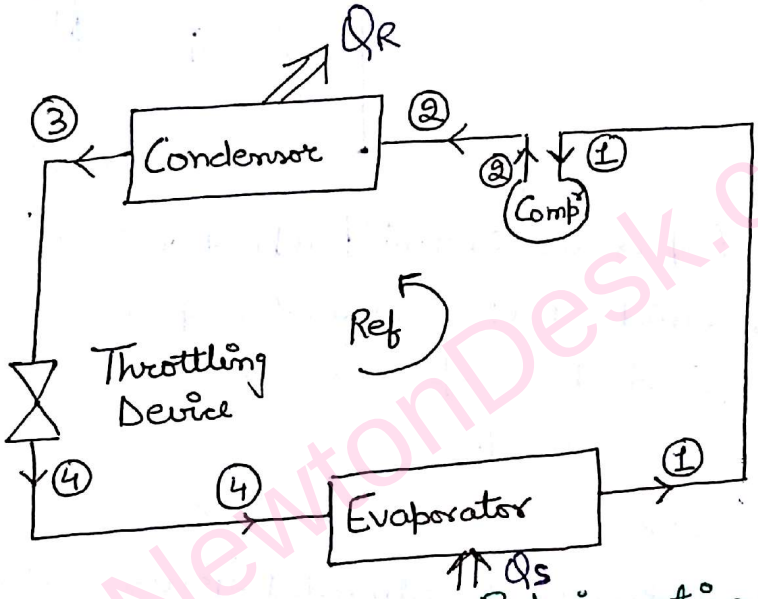
# Pure Substance :-

1) $h_B = h_f$
2) $h_D = h_g$ $(h_g - h_f)$
3) $h_C = h_f + x h_{fg}$
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

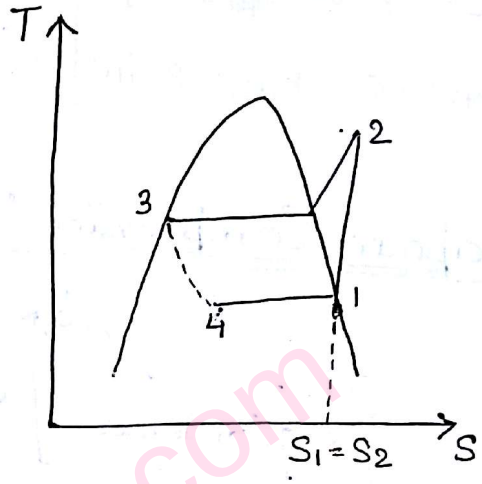
## Assumption :-

1) Entry to the Compresor and exit of evaporator (State 1) is Saturated Vapours.
2) Exit of the Condensor & entry of throttling (State 3) is Saturated liquid

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

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

Compresor is Work producing




**Qn) Why Isentropic Expansion is not preferable in VCRS?**

The State of Working fluid, at the entry of expandor (evaporator) is Saturated liquid, and then expresion 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 Compresior.

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) $\longrightarrow (h_1 - h_4)\ \frac{KJ}{Kg}$

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

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

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

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

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

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

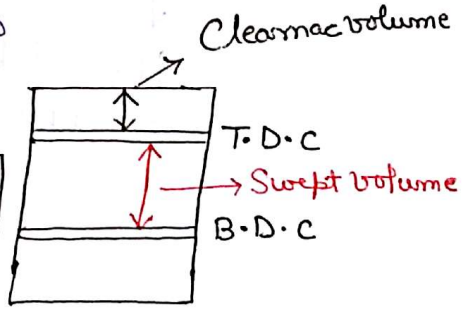
$P_{in} = \dot{m} \times (h_2 - h_1) = \dot{m} \times P_{in}$ ]

## Volumetric Efficiency of Resiprocating Compresor:

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

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

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




Objective

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

$$\eta_v = \frac{\text{Act. Vol entry}}{\text{Swept vol.}}$$

Where,

$\dot{m} \rightarrow$ mass flow rate Kg/Sec

$v \rightarrow$ Specific volume at entry of Compressor $m^3/Kg$

$D \rightarrow$ Bore / Diameter (m)

$L \rightarrow$ Stroke Length (m)

$N \rightarrow$ Speed (rPm)

$K \rightarrow$ No. Of Cylinders

$\dot{m} \rightarrow Kg/sec$

$v \rightarrow m^3/Kg$

$D \rightarrow m$

$L \rightarrow m$

$N \rightarrow rPm$

$K \rightarrow$ 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 $n$ → Polytropic Index

$C$ → Clearnac Ratio

↓

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

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

Objective

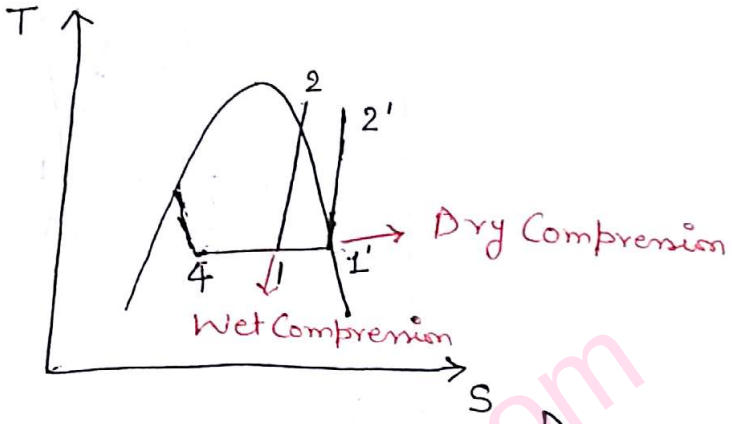
$$\eta_V = \frac{(\dot{m}\, v)_{entry}}{\frac{\pi}{4} D^2 L \times \frac{N}{60} \times K}$$

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

# 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.) ↓.

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

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

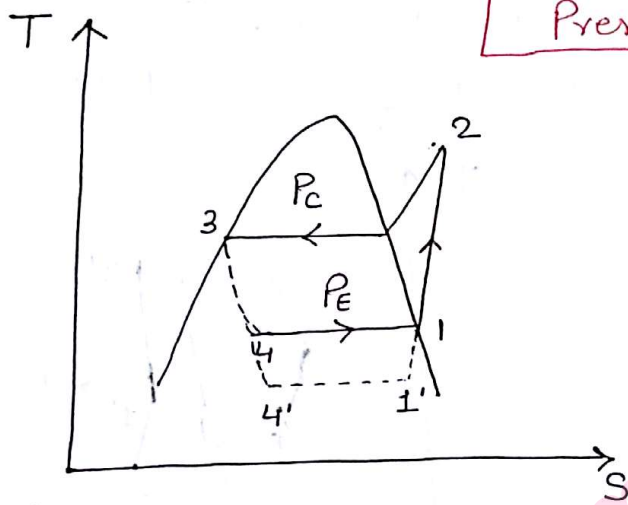
2) The Liquid particle, which is present in the mixture of Refrigerant, may Wash away the lubricant and it ↑ the Chances of Wear & tear & it also damage the Compressor valve & its Body.

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

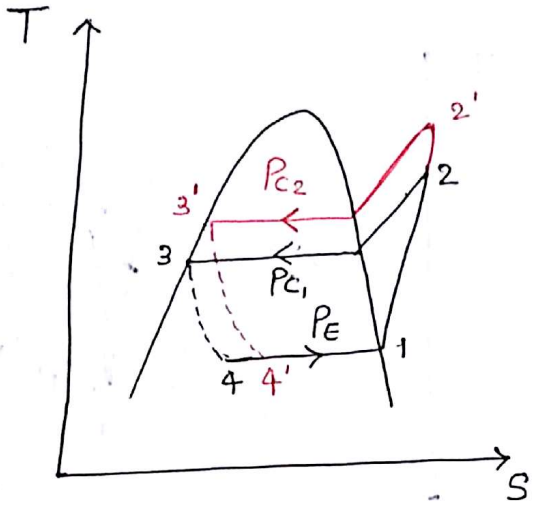
2) $\uparrow W_{in} = h_2 - h_1'$

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

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

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

## 2) Case 2:- Effect of ↑ in Condensor 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) $\frac{P_{C_1}}{P_E}$

### Process → 1-2'-3'-4'-1

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

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

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

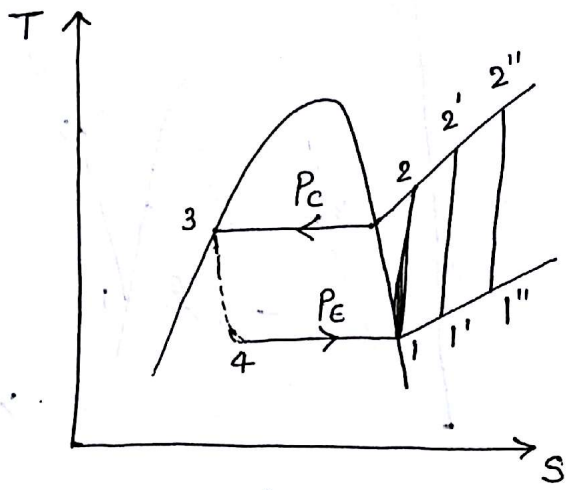
4) $\left(\frac{P_{C_2}}{P_E}\right)\uparrow$, $\eta_v \downarrow$

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

### NOTE:-

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

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




Process → 1-2-3-4-1

1) R.E = $h_1 - h_4$

2) Win = $h_2 - h_1$

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

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

Process → 1'-2'-3-4-1'

1) ↑R.E = $h_1' - h_4$

2) ↑Win = $h_2' - h_1'$

3) C.O.P = $\frac{R.E \uparrow}{Win \uparrow}$

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

Prove that Win is function of $T_{inlet}$

$$\boxed{W_{in} = \eta W_{closed}}$$

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

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

$PV = mRT$

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

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

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

Process 1-2 isentropic / Reversible Adiabatic Compression

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

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

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

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

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

## Effects Of Superheating :-

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

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

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

for example:-

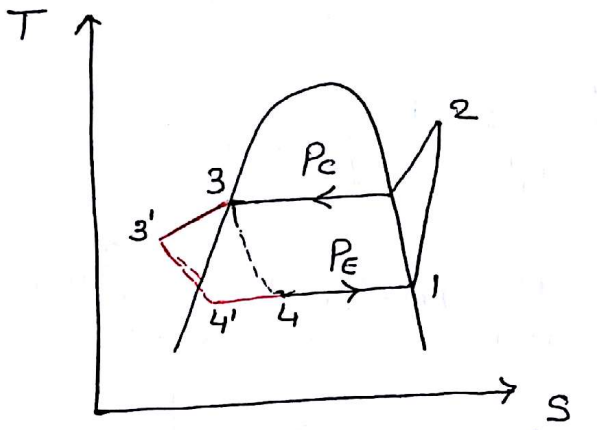
R-12 Refrigerant ↑ (COP)
(Superheating Would Result & ↑ in COP)

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

R12 ↑ COP
NH3 ↓ COP

## 4) Subcooling :-

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) Win = $h_2 - h_1$
3) C.O.P = $\frac{R.E}{Win}$

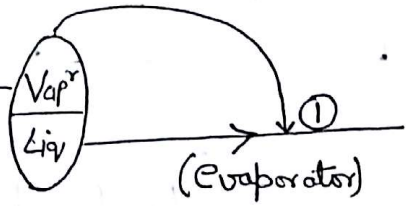
**Process → 1-2-3'-4'-1**

1) ↑R.E = $h_1 - h_{4'}$
2) Win = $h_2 - h_1$ (Const)
3) ↑C.O.P = $\frac{R.E\uparrow}{Win\ (Const)}$

# Use of Flash Chamber in VCRS :-

Flash Chamber is a device, which is used to Sepate 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.



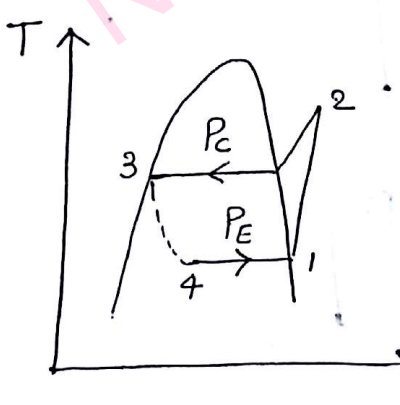

Because density of Water is ↑ and density of Vapour is ↓ low.

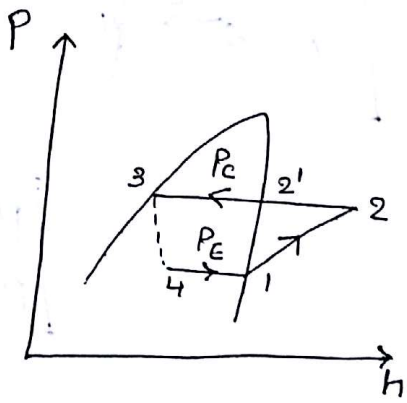
## NOTE :-

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

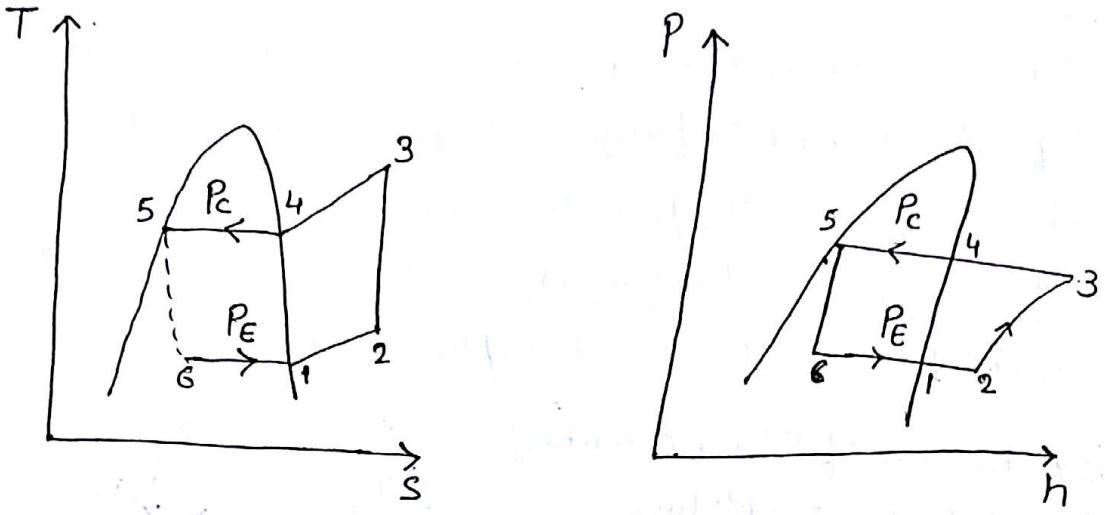



$P_4 = P_E = P_1$

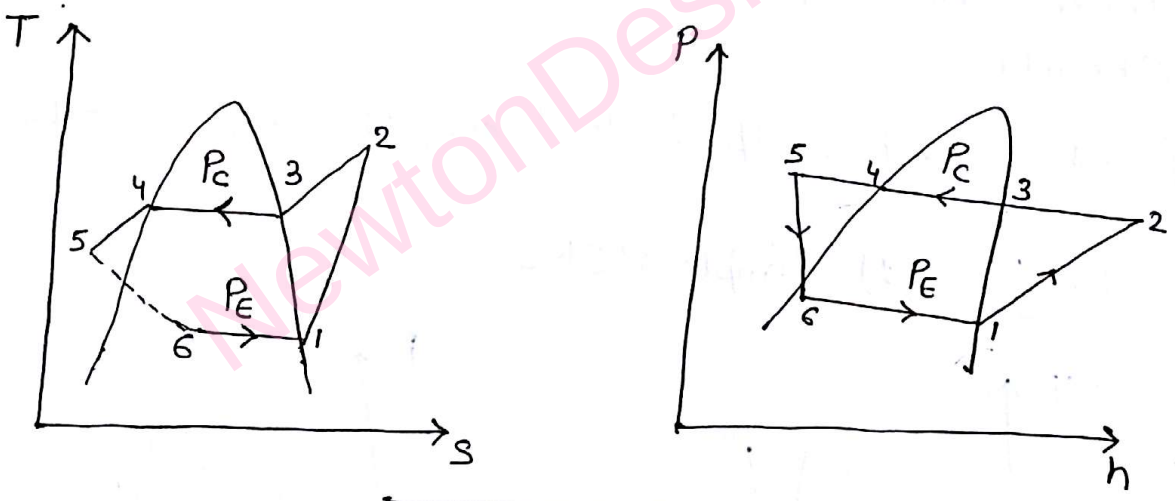
$P_3 = P_2' = P_2 = P_C$

2) Superheating :-




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

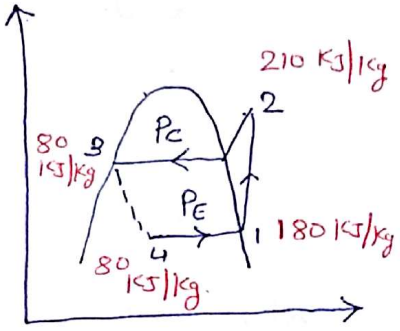
3) Sub Cooling :-




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

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 $$\eta_V = \frac{\dot{m}\, V_{entry\ of\ comp.}}{\frac{\pi}{4} D^2 L \frac{N}{60} \times K}$$

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. Compresion process is isentropic and the enthalpy at the outlet of the Compresor is 210 KJ/Kg Calculate,

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

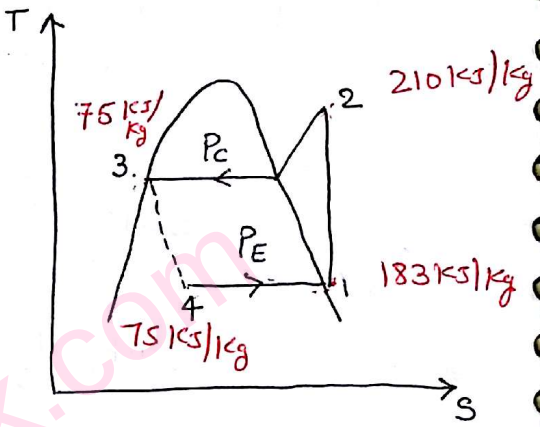
Soln) :

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

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

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

**COP = 4**




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

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

$P_{in} = 0.04629 \times 27$

**$P_{in}$ = 1.25 KW**

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

$5 = \dot{m} \times 108$

$\dot{m} = 0.04629$ Kg/Sec

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

$= 135 \times 0.04629$

**$Q_c$ = 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 $Q_c$ with $\dot{m}$.

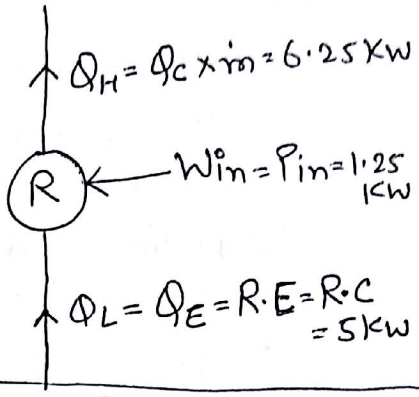
$Q_c = (h_2 - h_3)\ \dot{m}$

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

Another Method to Solve above Question,

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

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

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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

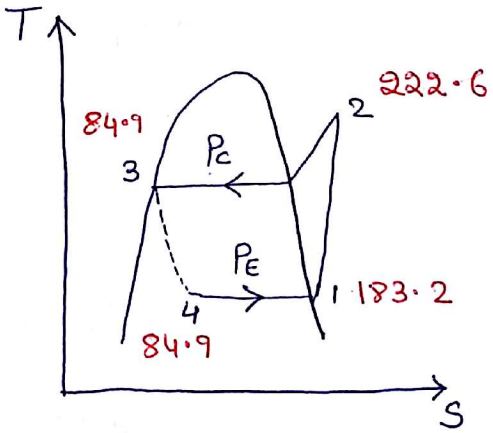
Soln)

$$R.E = h_1 - h_4$$
$$= 183.2 - 84.9$$
$$\boxed{R.E = 98.3\ KJ/kg}$$

$$Win = h_2 - h_1$$
$$= 222.6 - 183.2$$
$$\boxed{Win = 39.4}$$

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

$$Pin = 0.4142 \times 39.4$$
$$\boxed{Pin = 16.319\ KW}$$




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

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

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

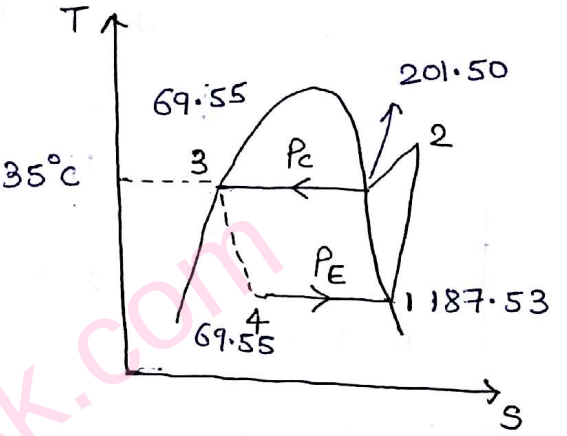
Qn) A Refrigerant operating on Simple VCRS having COP = 6.5. Enthalpy of Saturated liquid & Saturated Vapour Refrigerant at the operating Condensor temperature of 35°C are 69.55 KJ/Kg and 201.50 KJ/Kg. Respectively. The Saturated Refrigerant Vapour leaving the evaporator having enthalpy of 187.53. The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




$R.E = h_1 - h_4$

$= 187.53 - 69.55$

$\boxed{R.E = 117.98 \text{ KJ/Kg}}$

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

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

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

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

$\boxed{C.O.P = \frac{R.E}{W_{in}}}$ $\quad W_{in} = h_2 - h_1$

$18.150 = h_2 - 187.53$

$\boxed{h_2 = 205.680 \text{ KJ/Kg}}$ 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 $41.79$

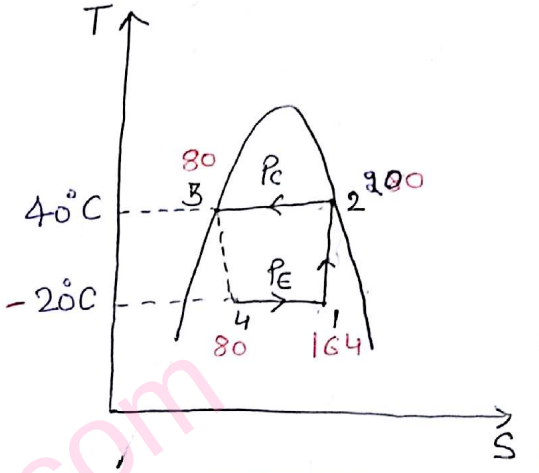
$4.18 = 0.6155(T_2 - 35°)$

$6.7912 = T_2 - 35°$

$\boxed{T_2 = 41.79°C}$ 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 (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

$S_1 = S_2$

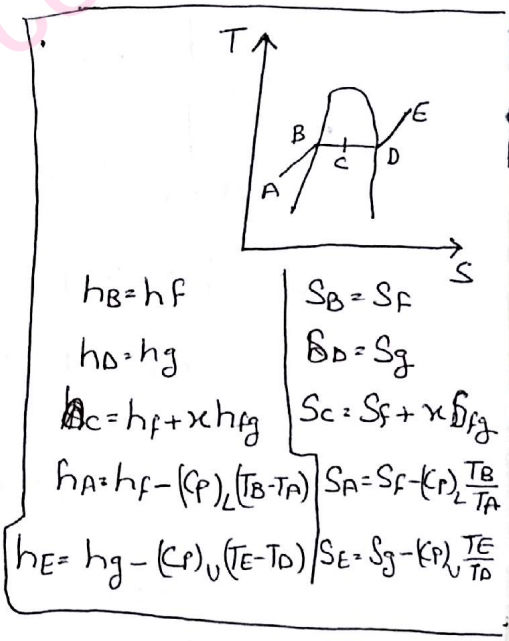
$S_{f1} + x(S_{g1} - S_{f1}) = S_{g2}$

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

$x = 0.9$

Putting value of x in ①

$h_1 = 164$ KJ/kg




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

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

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

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

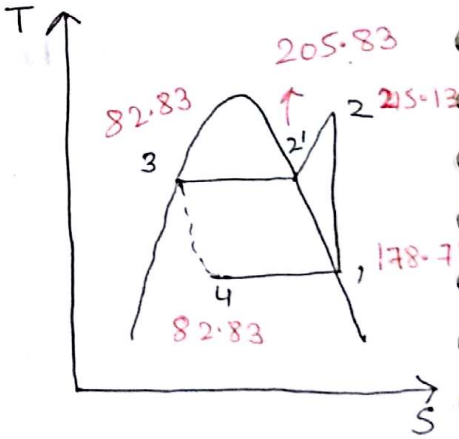
③ $R.C = \dot{m} \times RE$
$= 0.025 \times 84$
RC = 2.1 KW Ans

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 to 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 KJ/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) $\dot{m}_{ice} = 8640\ Kg/day = \frac{8640}{24\times3600} = 0.1\ KJ/sec$




$h_2 = h_{g2'} + (C_p)_v (T_2 - T_2')$

$= 205.83 + 0.82 (T_2 - 321)$ — 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]

$35^\circ C - 0^\circ C - 0^\circ C - 8^\circ C$
(S.H) (L.H) (S.H)

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

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

$R_C = 49.901 kW$

$R_C = \dot{m} R_E$

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

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

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

10% of 49.90
4.999 kW

35°C
-8°C

49.94 + 4.499 = 54.901 kW

$R_C = \dot{m} R_E$

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

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

Putting $\dot{m}$ in ①

$= 0.57 (h_2 - h_1)$

$= 0.57 (205.34 - 178.74)$

$P_{in} = 20.85 kW$

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

$COP = 2.63$

Q. 23 Workbook)

Qn) A Food Storage requires a Refrigeration capacity of 15TR. It Works b/w $-10°C$ & $30°C$. The Temperatures of Refrigerant Superheated as gas in evaporator is $-5°C$ & temperatures 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 (in) of Refrigerant in kg/min
   (c) Theoritical Pistondisplacement
   (d) COP
   (E) Bore and Stroke of the Compressor.

2) if the clearrac volume is 3% of Stroke volume then,
   (a) Determine Volumetric efficiency $\eta_v$.
   (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) | $h_f$ (KJ/kg) | $h_g$ (KJ/kg) | $S_f$ | $S_g$ | V ($m^3$/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 X 3.5 = 52.5 KW

Saturated at $S_g$ Temp. evaporator = $-10°C = 263K$

at $S_f$ Temp Condensor = $30°C = 303$

Temp. of Refrigerant Superheated as gas in evaporator is = $-5°C = 268K$

Temp. of Refrigerant Superheated as liquid in condensor is = $25°C = 298K$

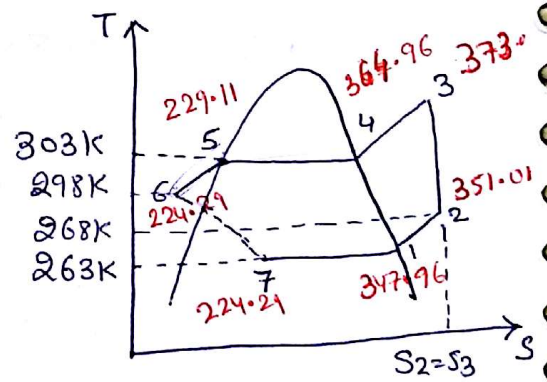
No. of Cylinder (K) = 2

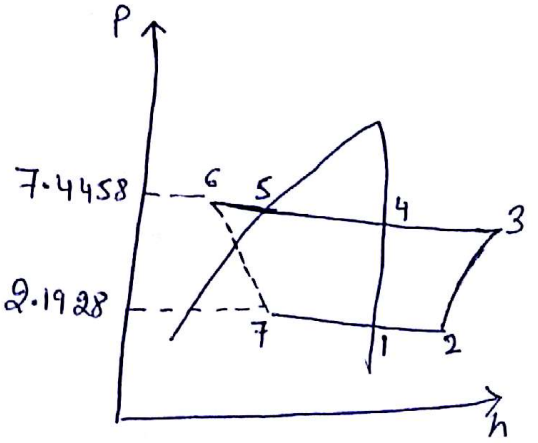
Stroke (L) = 1.5D

Speed (N) = 960 RPM

$(C_p)_V$ = 0.615 KJ/kg-K

$(C_p)_L$ = 0.963 KJ/kg-K



$h_1 = 347.96$

$h_2 = h_{g1} + (C_p)_v (T_2 - T_1)$

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

$h_2 = 347.96 + 3.075$

$h_2 = 351.0 \text{ KJ/kg-K}$

$h_4 = 364.96$

$h_3 = h_{g4} + (C_p)_v (T_3 - T_4)$

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

$S_2 = S_3$

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

~~$0.96561 + 0.615 \ln \frac{268}{263} = +$~~

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

$T_3 = 316.12 K$

On putting value of $T_3$ in eqn (1)

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

$h_3 = 364.96 + 0.615 (13.12)$

$h_3 = 373.02 \text{ KJ/kg-K}$

$h_5 = 229.11$

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

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

$h_6 = 224.29$

$h_6 = h_7 = 224.29$ Bcz of Throttling

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

$RE = (351.01 - 224.29) \Rightarrow R.E = 126.8 \text{ KJ/kg-K}$

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

$52.5 = \dot{m}_{1 \text{ kg/sec}} \; 126.8 \; \text{KJ/kg-K}$

$\dot{m} = \frac{52.5}{126.8}$ $\quad \dot{m} = 0.414 \text{ kg/sec} \times 60$

$\dot{m} = 24.60 \text{ kg/min}$

(I)(d) $COP = \frac{R.E}{Win} = \frac{126.8}{21.99} = 5.76$

$Win = (h_3 - h_2)$

$= (373 - 351.01)$

$Win = 21.99$ KJ/kg-K

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

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

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

$TPD = 0.416 V_2 - (1)$

$TPD = 0.416 \times 0.07702$

$TPD = 0.326$ m³/sec

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

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

$V_2 = 0.07848$ m³/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

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

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

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

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

$\frac{V_3}{0.02372} = \frac{316}{303} \Rightarrow V_3 = 0.0247$ m³/kg

Putting value of $V_3$ in (1)

$\eta_v = 93.5\%$ Ans

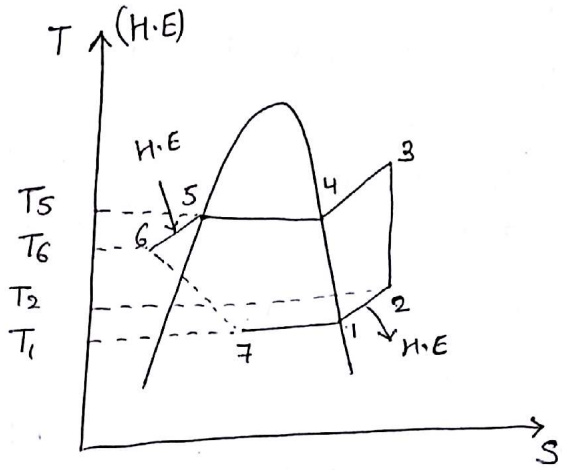
(2)(b) $\eta_v = \frac{\dot{m} v_2}{\frac{\pi}{4} D^2 \frac{L N}{60} \times K}$

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

$D = 0.097$ m

$L = 0.1459$ m. Ans

# Use Of Heat Exchanger In VCRS :-




Evaporator → 7 − 1 → R.E = $h_1 - h_7$

Compressor → 2 − 3 → Win = $h_3 - h_2$

Condensor → 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 50 KW. 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 1000 RPM 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 | Cl | 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 = 50 KW = 50 × 3.5 = ~~150~~ 175 KW

K = 1
L = 1.2D
n = 1000

NOTE :-

As There is no information Provided Regarding clearance Ratio (Cnf) therefore I shall Assume 100% volumetric efficiency.

# CASCADE :-

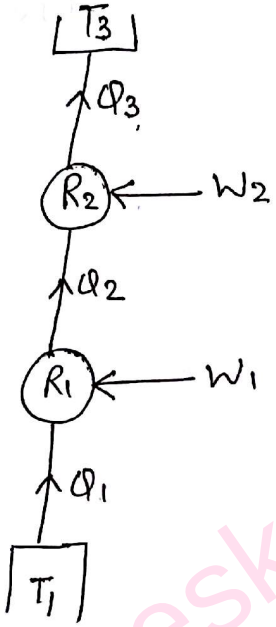


Prove That, COP of Cascade Refrigeration System is

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

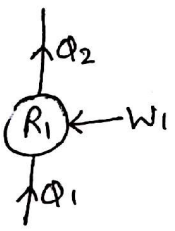
Where $(COP)_1$ is COP of first Refrigeration System i.e $R_1$.

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



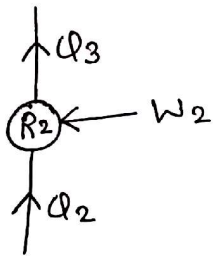

$$Q_2 = Q_1 + W_1$$

$$Q_2 = Q_1 + \frac{Q_1}{COP_1} \quad -(4)$$




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

$$\boxed{W_1 = \frac{Q_1}{COP_1}} \quad -(1)$$




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

$$\boxed{W_2 = \frac{Q_2}{COP_2}} \quad -(2)$$

$$(COP)_{CC} = \frac{(DE)_{LC}}{(Wn)_{LC} + (Win)_{VC}} = \frac{Q_1}{W_1 + W_2} \quad -(3)$$

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

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

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

Ans objective

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

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) | $h_f$ | $h_g$ | $S_f$ | $S_g$ |
|---|---|---|---|---|---|
| -40 | 10.55 | 332.7 | 652.8 | 3.8531 | 5.2262 |
| 5 | 39.77 | 431.0 | 649.8 | 4.2231 | 5.0037 |

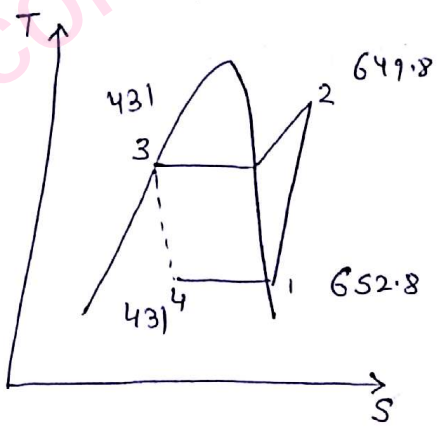
Soln) $RC = 100 TR = 100 \times 3.5 = 350 KW = \dot{m}_{(CO_2)} \times RE = \dot{m}_{CO_2} \times (h_2 - h_1)$

$CO_2 \rightarrow T_E = -40°C$

$T_C = 5°C$

$NH_3 \rightarrow T_E = -7°C$

$P_{NH_3} \rightarrow 96.5 KW$




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

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

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

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

$\dot{m}_{CO_2} = 1.577$ 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)

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

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

$$T_2 = 361.181K$$

using value of $T_2$ in equ (2)

$$h_2 = 672.5 \text{ KJ/Kg}$$

using value of $h_2$ in (1)

$$COP = 1.052$$

# 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 Hydrocarbon. (Halo Carbon)

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

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

$$C_m H_n F_p Cl_q$$

$m \rightarrow$ Represent no. of Carbon element.
$n \rightarrow$ Represent no. of Hydrogen element.
$P \rightarrow$ Represent no. of Flourine element.
$q \rightarrow$ Represent no. of Chlorine element.

1) R-11

R-011

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

$m-1=0 \quad m=1$

$n+1=1 \quad n=0$

$P=1 \quad P=1$

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

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

$1+q = 4$

$\boxed{q=3}$

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

Ans $\rightarrow CFCl_3$

2) R-12

R-012

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

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

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

$P=2 \; ; \; P=2$

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

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

$2+q = 4$

$q=2$

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

$\boxed{CF_2Cl_2}$

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

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

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

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

$P=4$ ; $P=4$

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

$2+4+q=4+2$

$q=6-6$

$\boxed{q=0}$

$C_mH_nF_pCl_q$ $m=2, m=2, P=4$ $q=0$

$\boxed{C_2H_2F_4}$

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 attacts 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.

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

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

$$C_mH_nF_pCl_q$$

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 Condenser 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, Where as 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.

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

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 there, evaporator pressure Should be Kept almost equal to atmospheric pressure. 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. $\downarrow \eta_v$

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

## 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 Latint heat of Vaporization for $NH_3$, Refrigerant is having high value, but it doesn'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 | ✓ | X | $NH_3$, $CO_2$ |
| Rotary or Centrifugal Compressor | X | ✓ | 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 missible with oil. R-11, R-12

R + Fully immissible with oil. $NH_3$, $CO_2$

R + Partially missible with oil. R-22

There are some Refrigerant, fully missible with oil and some Refrigerant fully immissible with oil, doesn't create any Problem but refrigerant which is partially missible with oil like R-22, create problem. Chokking problem occures 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 Wrought iron
Where as Hydrocarbon Compound attacks aluminium (Al)
Prefreable with Copper (Cu)

$NH_3$ Corrodes Copper, Suitable for Wrough iron
Hydrocarbon Corrodes Aluminum 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, Hydrocarbon & Propane | Halide Torch test |
| | 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 :-

$R\text{-}11\ (-23.7^\circ C) > R\text{-}12\ (-29^\circ) > NH_3\ (-33.3^\circ) > R\text{-}22\ (-41^\circ) > CO_2\ (-73.6^\circ)$

Refrigerant in ↑ order of freezing Point :-

$CO_2 < NH_3 < R\text{-}11 < R\text{-}12 < R\text{-}22$

Refrigerant in ↓ order of Critical temp.

$R\text{-}11 > NH_3 > R\text{-}12 > R\text{-}22 > CO_2$

Refrigerant in ↓ order of Critical Pressure.

$NH_3\ (113.86) > CO_2\ (73.8) > R\text{-}22 > R\text{-}11 > R\text{-}12\ (41.2)$

Refrigerant in ↓ of COP :- $R\text{-}11 > NH_3 > R\text{-}12 > R\text{-}22 > CO_2$

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

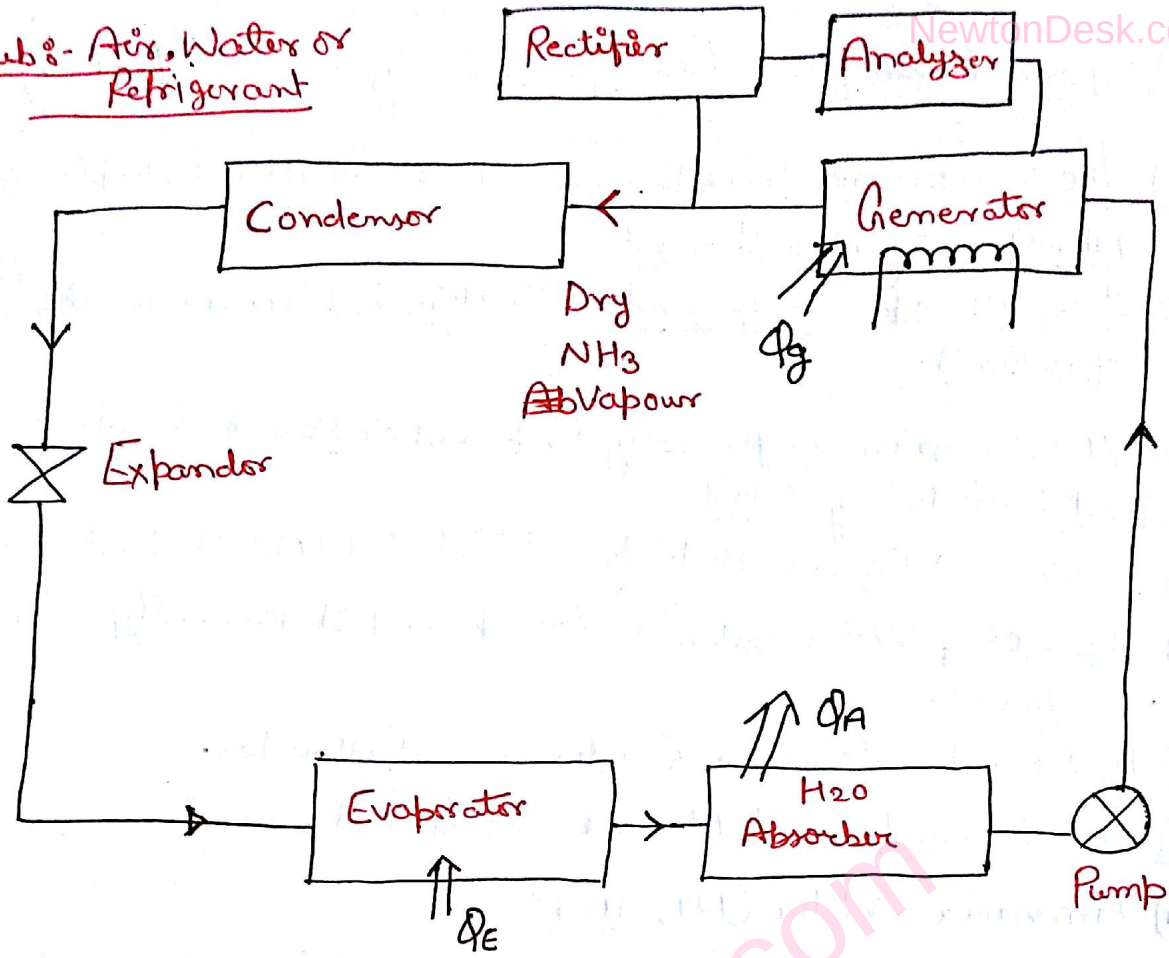
(A) Ammonia & Water ($NH_3-H_2O$) :-

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.

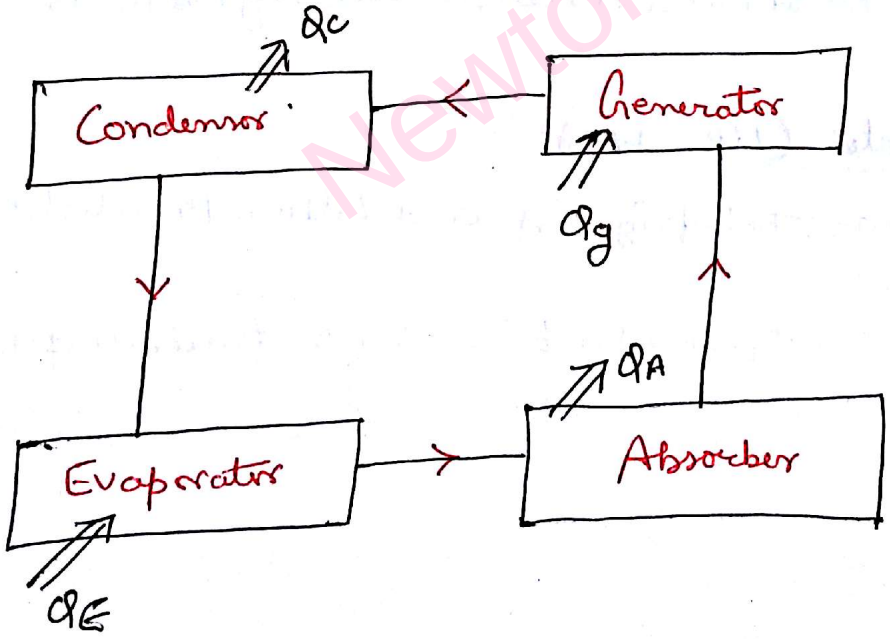
(B) Lithium Bromide & Water ($LiBr-H_2O$) :-

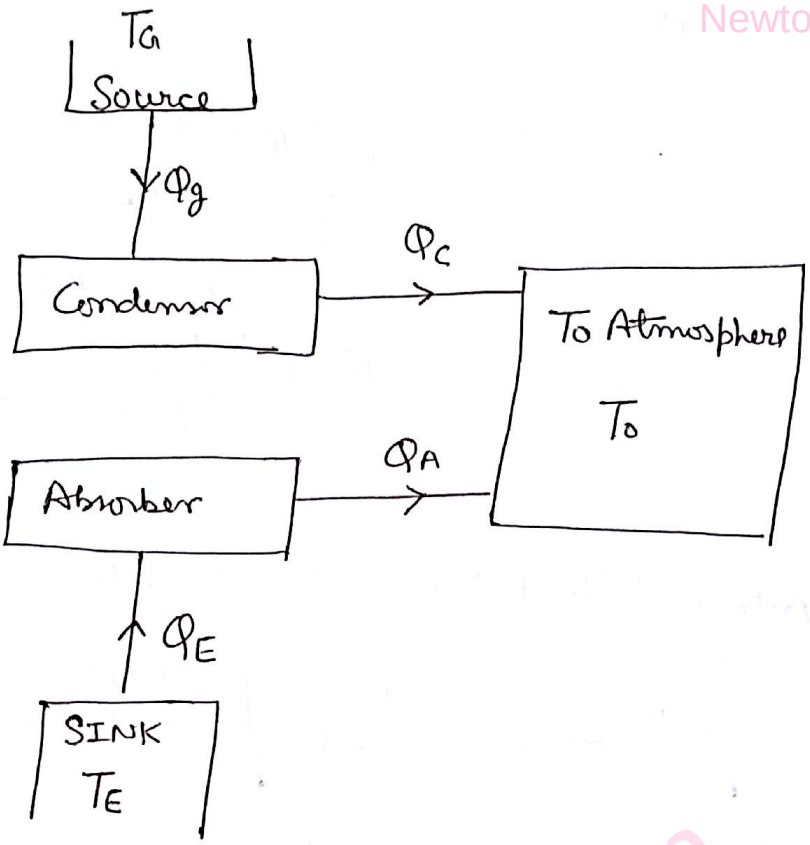
In this Water is used as the Refrigerant and Lithium Bromide (LiBr) is used as the Absorber.
The Above Combination is not prefreable below 0°C (The freezing Pt. of $H_2O$ 0°C)

Working Sub :- Air, Water or Refrigerant




## COP of VARS :-






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

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

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

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

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

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

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

Only objective (Objective)

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

# NOTE:-

1) $T_G$ ] E
$T_o$ ] R
$T_E$

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

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

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

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.]

$$W_P = -V_f \int dP \qquad (V_f \text{ is very Small})$$

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

(Assumption neglect Pump Work)

(if Question Said to neglect Pump Work)

## VCRS

1) Compressor is used.

2) It is a Work operated unit.
Or Runs on high grade energy.

3) Heat Rejection Occours in Condensor only.

4) Moisture Related Problem is having more Serious impact or dangerous in VCRS.

5) Chances for the leakage of Refrigerant are high.

6) It has higher COP generally Varies from 3-5.

7) Creates More noise Pollution.

Spend Money bcz ↑ COP

## VARS

1) Compressor is Replaced With Absorber, Pump & generator

2) It is a Heat operated unit.
OR
Runs on Low Grade energy.

3) Heat Rejection Occours in Condensor as well absorber.

4) Relatively Lesser Problems.

5) Relatively lesser chances

6) It has Relatively Lower (COP)
Generally Varries from 0.3 - 0.5.

7) Relatively Less noisy.

less money So ↓ COP can be Considered

## Electrolux Refrigerator :- [No use of Pump]

i) 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.

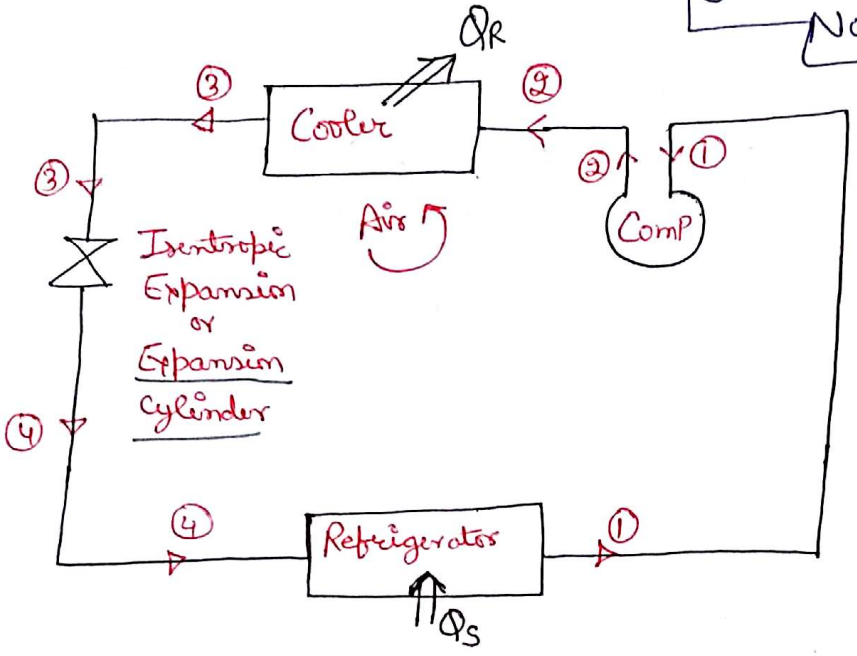
| | |
|---|---|
| $NH_3$ | Refrigerant |
| $H_2O$ | Absorber |
| $H_2$ | (Low Partial Pressure of Ammonia Vapour) used to |

$NH_3$ 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 :- Four Cycle



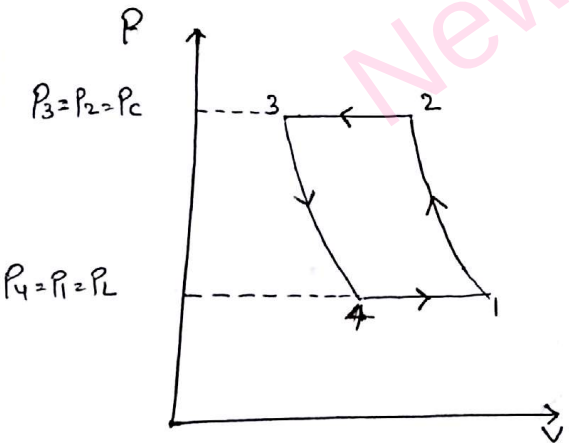
g+is for air only
No Phase change




## Reversed Brayton Cycle / Bell Coleman Cycle

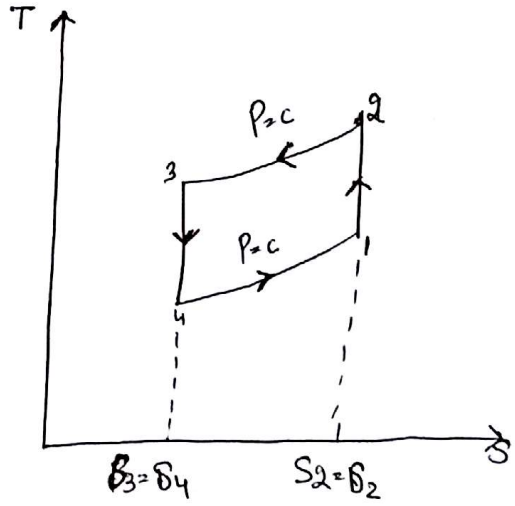
Proces (1-2) → Isentropic / Reversible Adiabatic Compression.
Proces (2-3) → Heat Rejection at Constant Pressure.
Proces (3-4) → Isentropic / Reversible Adiabatic expansion.
Proces (4-1) → Heat addition at Constant Pressure

Processes




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

(P-V curve)

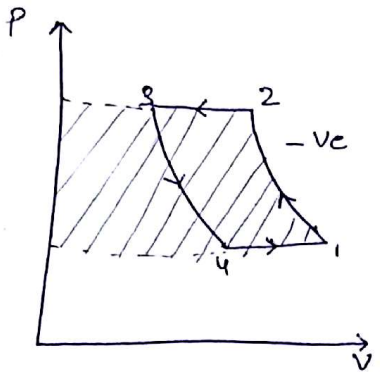


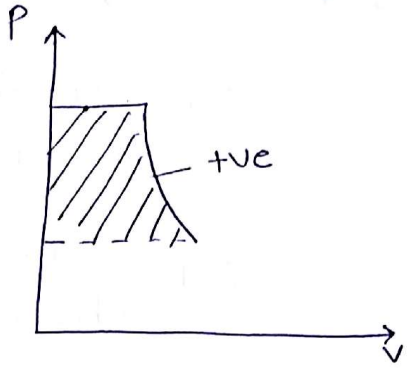

(T-S) Curve

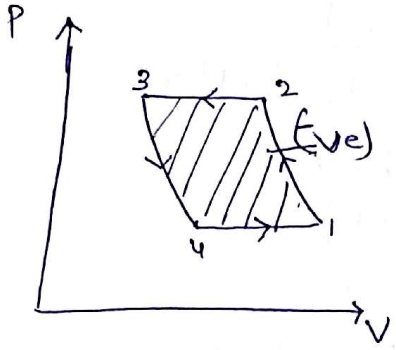
# COP of Reversed Brayton Cycle :-

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

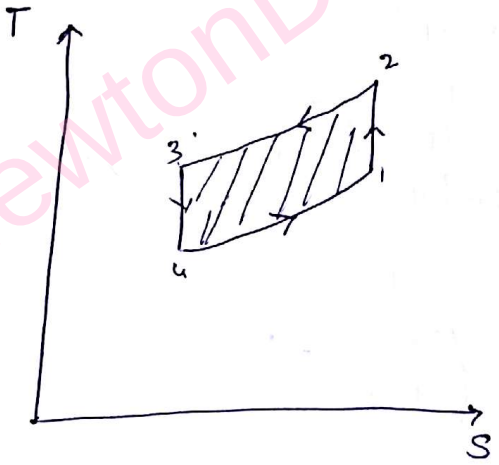
(2-1)




for T-S




$Q_{NET} = -ve$

$W_{Net} = Q_{net}$

$$COP = \frac{R.E}{Win}$$

~~$COP = \frac{R.E}{R_6 = R}$~~ $\quad COP = \frac{R.E}{W_C - W_T}$ — 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 → $\frac{PV = mRT}{h = C_pT}$ enthalpy is fun$^n$ of temp for ideal gas

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

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

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

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 $r_p = \frac{P_1}{P_2}$

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 eqn (4) in (3)

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

using eqn (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 even = odd odd)

grp objective for otto and more cycle. [IC TO PP]

using (6) in eqn (1)

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

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

use eqn 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 $r_p = \frac{P_1}{P_2} = \frac{P_2}{P_1}$

grp Objective

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

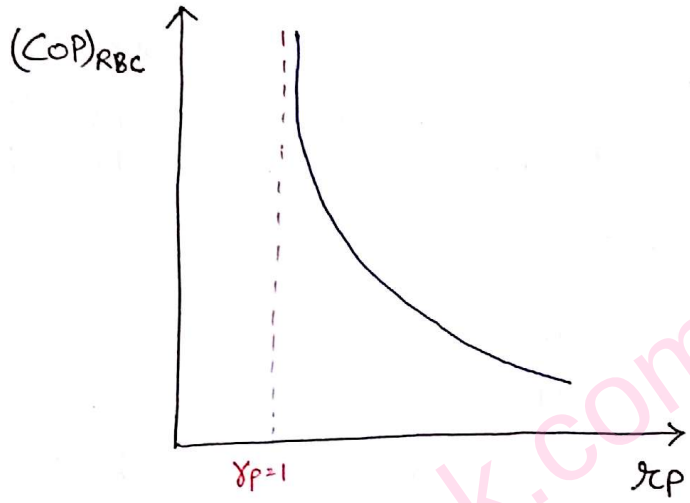
The COP of Reversed Brayton Cycle is a function of Pressure Ratio.

## Assumptions :-

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

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

$r_p = 1 \longrightarrow COP = \infty$




## 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.

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

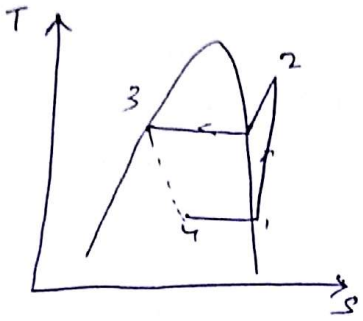
$\int v_g dP \quad \int v_g dP$

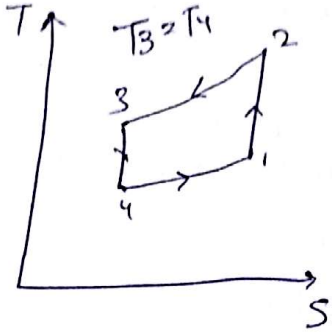
$V_f$ can be neglected but $v_g$ is not

$V_g >>>> V_f$

Qn) Why isenthalpic expansion is not preferable 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. If instead of heat absorbing it will Rejecting Heat. Therefore isentropic expansion is preferable.




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

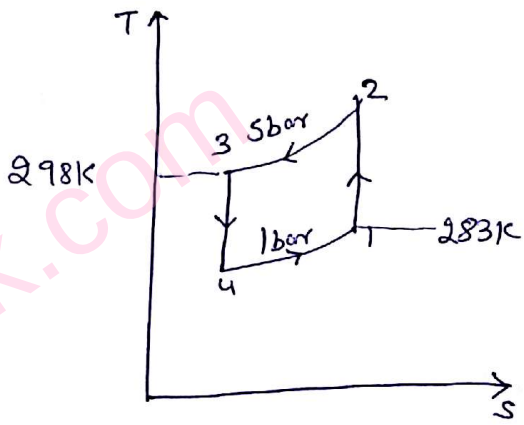
1) Theoritical COP
2) Refrigeration Effect Per Kg, Assumming both Compression & expansion to be ~~isothermal~~ isentropic, having Value of $\gamma = 1.4$, & Specific heat $C_p = 1.004$ kJ/kgK.
3) If Compression follows $PV^{1.35} = C$ (Compressor) and expansion $PV^{1.3} = C$

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




(i) Theoritical COP

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

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

$= C_p(T_1 - T_4)$

$= 1.005(283 - T_4)$ — (1)

$= 1.005(283 - 188.15)$

$= 95$ 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) $PV^{1.35} = C \rightarrow \frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}} \Rightarrow \frac{T_2}{283} = \left(\frac{5}{1}\right)^{\frac{1.35-1}{1.35}} = T_2 = 429.3$K

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

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

$= 1.005(283 - 205)$

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

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

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

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

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.

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

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

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

$PV^{1.3} = C$

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

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

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

$W_{in} = 47\ KJ/kg$

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

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

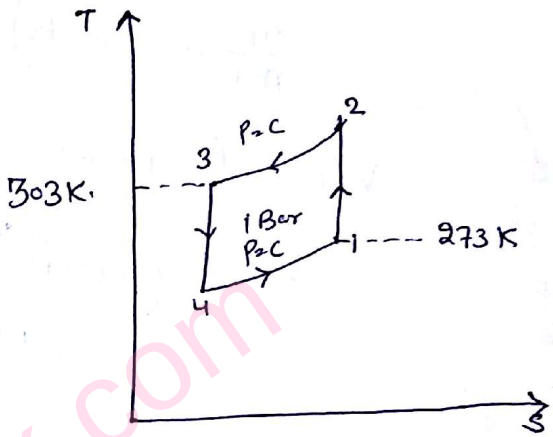
(Qn) Air is used as a Refrigerant in Reversed Brayton Cycle. Draw P-V & T-S diagram for the cycle & derive the expression for COP in terms of Pressure Ratio.

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 expander for 1TN Cooling Capacity.

Assumming inlet pressure ($P_1$) to the Compressor is 1 bar.

Soln $\quad r_p = 4$




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

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

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

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

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

$$\dot{V}_1 = \frac{\dot{m}RT_1}{P_1} \Rightarrow \dot{V}_1 = \frac{\dot{m} \times 0.287 \times 273}{1\times10^5} \quad -(1)$$

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

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

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

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

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

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

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

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

(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 Temperatures. The decrease in higher temperature is equal to ↑ in Lower temperature. Then Determine the new temperatures in Kelvins.

Soln

$T_{L1} = (40^\circ C) - = 233 K$

$T_{L2} = +40^\circ C = 313 K.$

$T_{L2} = T_{L1} + x = 233 + x = 238.875 K$

$T_{H2} = T_{H1} - x = 313 - x = 307.125 K.$

$$3.5 = COP = \frac{T_{L2}}{T_{H2} - T_{L2}} = \frac{T_{L1} + x}{(T_{H1} - x) - (T_{L1} + x)} = \frac{233 + x}{80 - 2x}$$

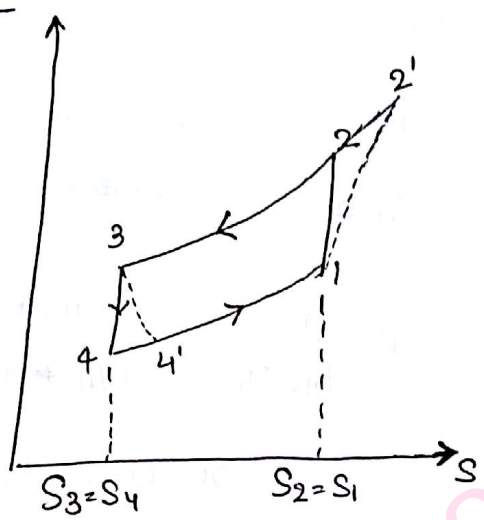
$x = 0.5875$

# 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 & $h = f(T)$

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

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

$\eta_{isentropic}$
↳ Rev. Adiabatic
↳ irr. Adiabatic → $ds = \dot{S}_{gen} + \frac{dQ}{T}$ (dQ = 0)

$\eta \rightarrow$ Loss

## Important Point's -

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

Compressor is used to Compress the gas, so we need that more & more Compression takes Place or ↑ in $W_c$. So we divide by $\eta_c$ and $\eta_c$ is less than 1.

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

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

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

Here multiply With $\eta$ indicates loss.

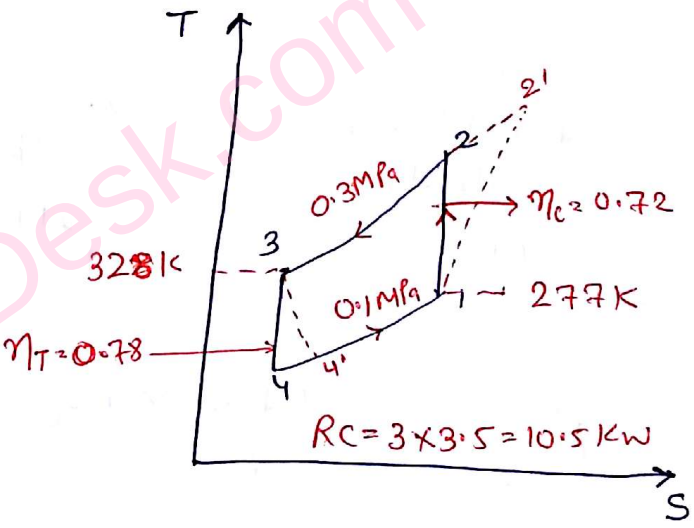
Q.n) In bell-Coleman Refrigeration plant, air enters the Compressor at a pressure of (0.1 MPa) and temperature of 4°C. It is then Compressed to pressure of 0.3 MPa, with an isentropic efficiency of 72%. It is then Cooled in the Cooler to a temperature of 55°C and then expanded to a pressure of 0.1 MPa, 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

As it is isentropic Proces

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




$$RC = \dot{m} \times RE$$
$$RC = \dot{m} \times (h_1 - h_4')$$
$$RC = \dot{m} \times C_p (T_1 - T_4')$$
$$\dot{m} = 0.581 \text{ kg/sec.}$$

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

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

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

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

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

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

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

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

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

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

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

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

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

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

$$P = 0.58 \times 1.005 (418.86 - 297) - (328 - 259)$$

$$= 42.5\ kW$$

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

## 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.
Therefore the Value of Heat Rejction 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.

↗ Means R.E

$H.RR = 1.2$

$R.E = 2000$ KJ/min

$Q_c = ?$

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

$$Q_c = 2520 \text{ KJ/min}$$

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

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

↑ $H.RR = \frac{Q_c}{R.E}$ ↓

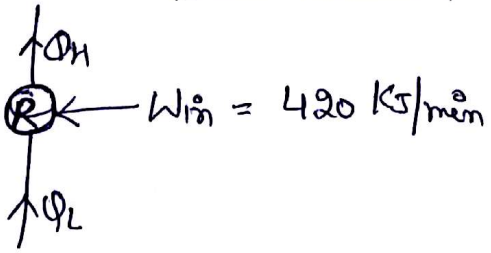
↓ So first two are neglected

~~A) 2100,4~~

~~B) 2100,5~~

C) 2520,4

D) 2520,5




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

$$COP = 5$$

Qn) Prove that Heat Rejection Ratio,

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

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

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

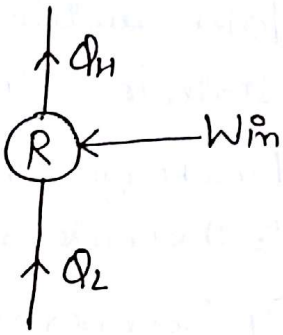
$$= \frac{1}{HRR - 1}$$

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

Objective

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

Objective



# Refrigeration Equipment



## Function of Compressor :-

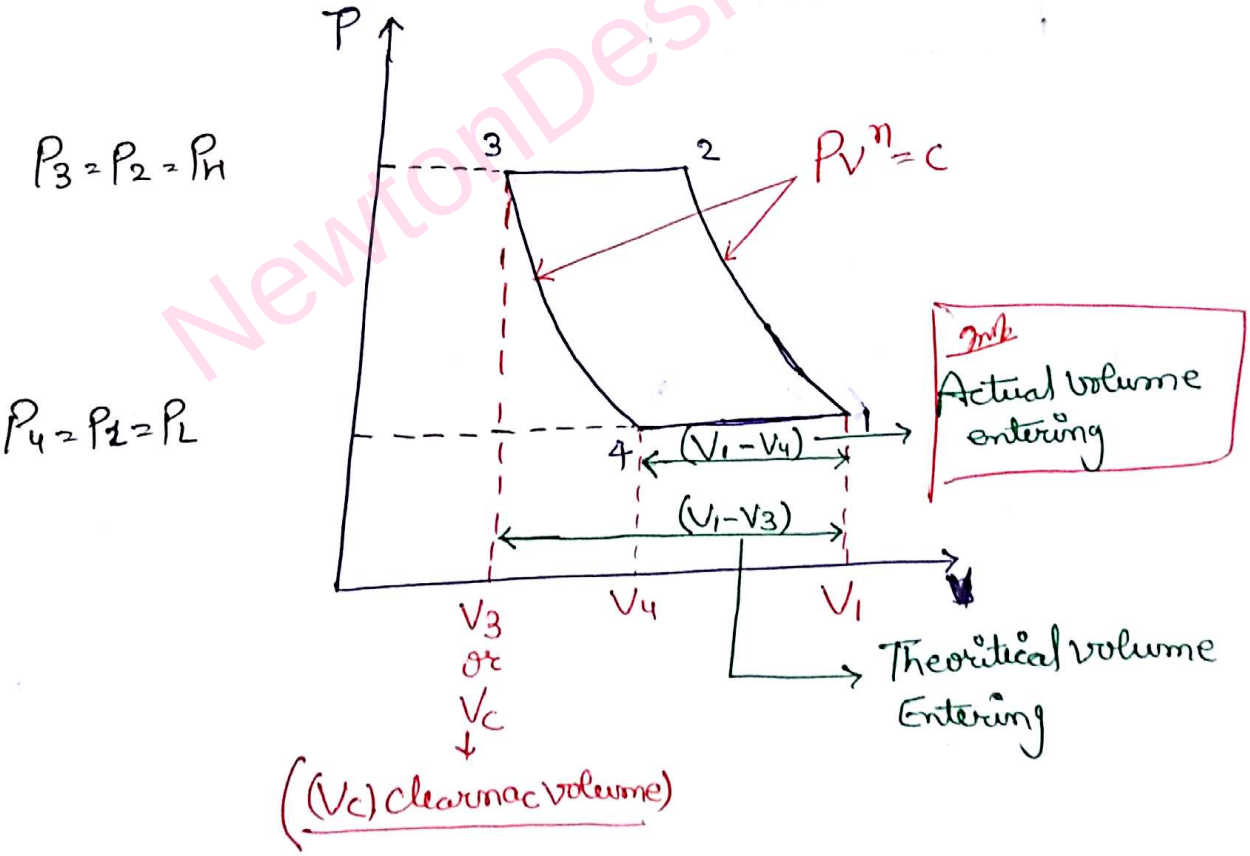
i) Take Suction of Refrigerant generally in Saturated Vapour State, from the evaporator.

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,

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



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

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

$$\eta_V = \frac{V_1 - V_4 - V_3 + V_3}{V_1 - V_3}$$ multiply & 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

$$PV^n = C$$

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

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

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

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

Putting (2) & (1)

Objective

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

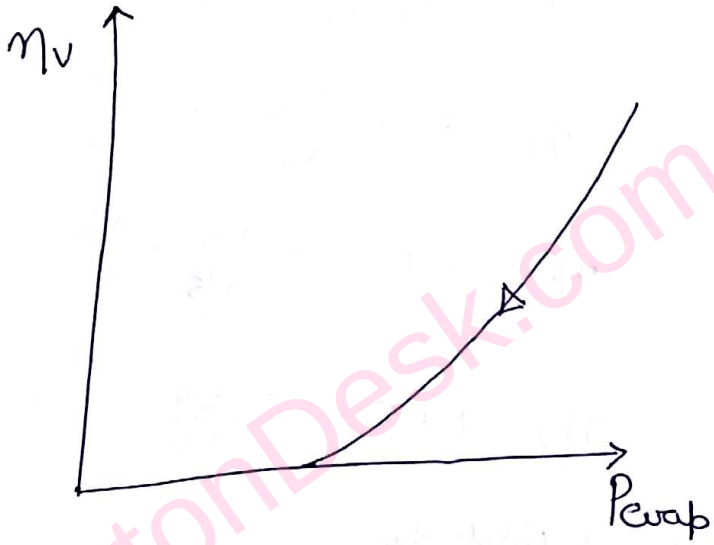
# Performance Parameters:-

## 1) Effect of Evaporator Pressure:-

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

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

Reason for mass flow Rate



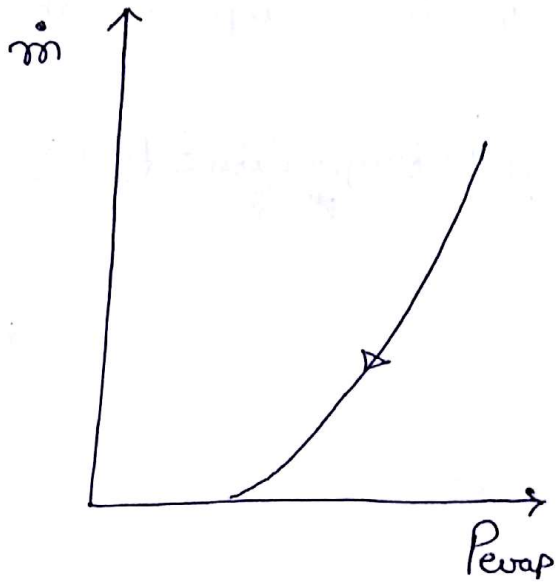

Volumetric Efficiency V/s evaporator Pressure

Here graph is Plot from $P_2 = 1$

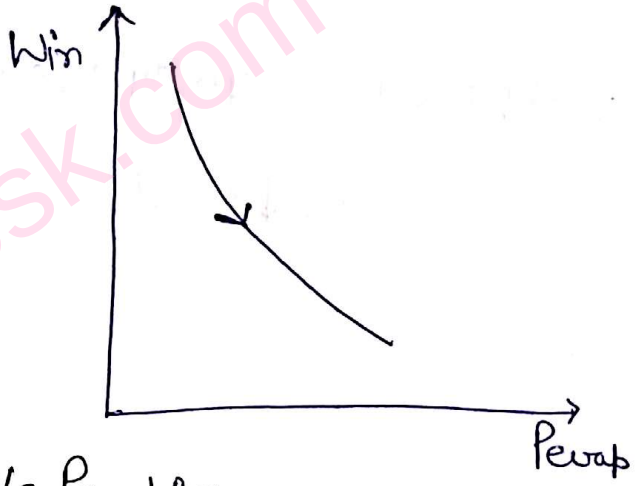
Bcz Pevaporator must be Same or equal to evaporator Pressure bcz, Below Pevap. it becomes Vaccume

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

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




## 3) Workin to Compressor V/s Pevap :-

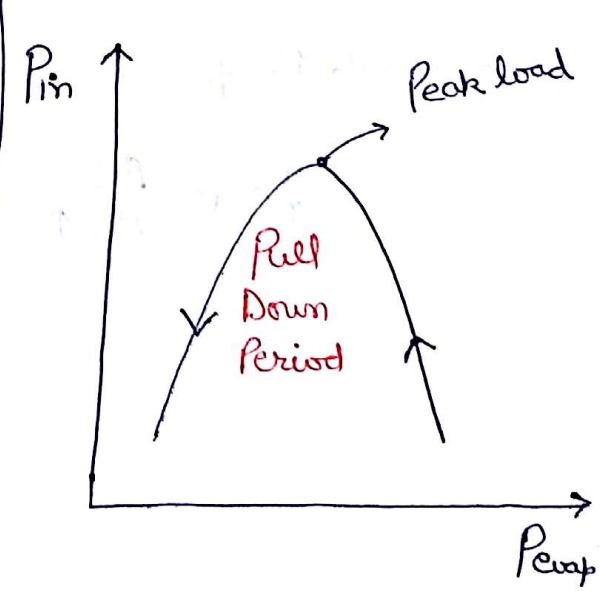



## 4) Power Input to the Compressor V/s Pevap :-

With the ↓ Pevap., 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

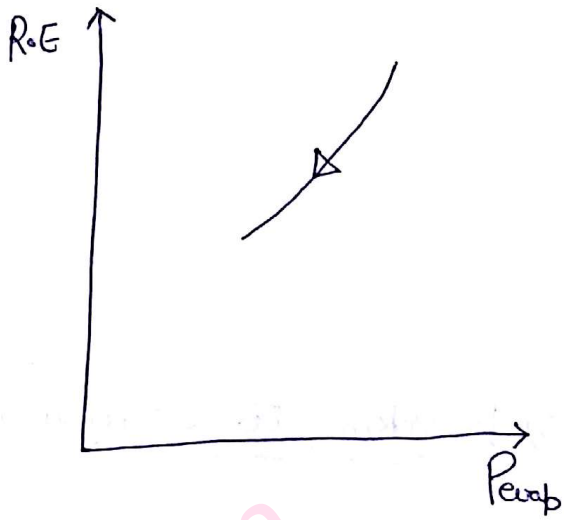
$$P_{in} = \downarrow \dot{m} \times Win \uparrow$$



## NOTE:-

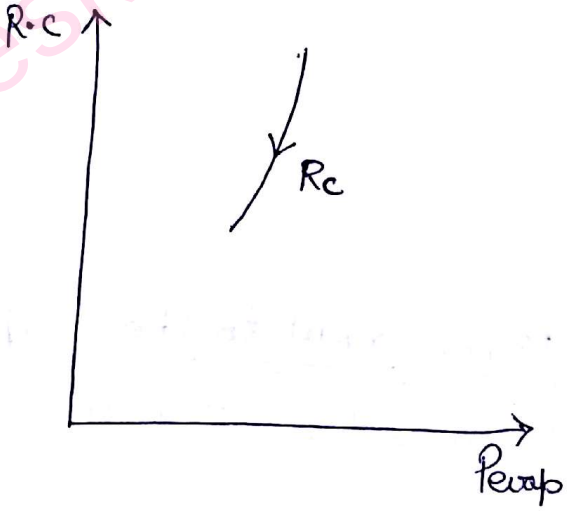
1) The Compressor Should be design for Peak loads/Peak hours.

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



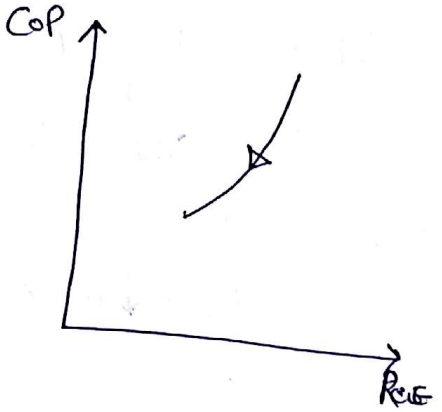

6) Refrigeration Capacity V/s evaporator Pressure :-

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




7) COP V/s Pevap

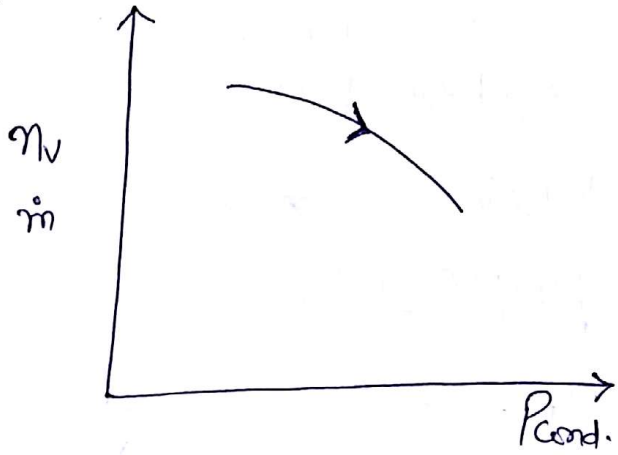
$\downarrow COP = \frac{\downarrow R.E}{R.E\uparrow}$

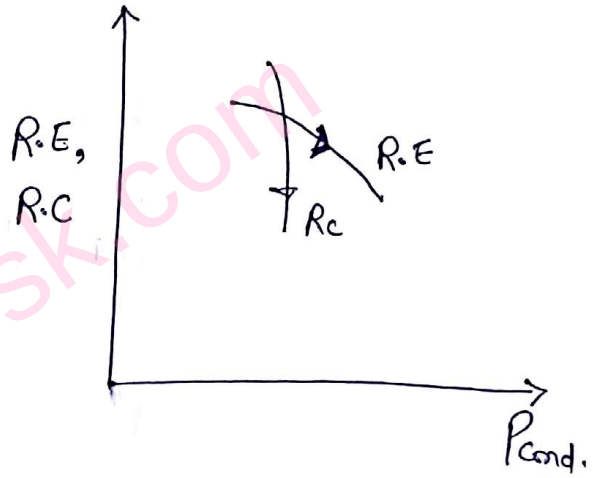


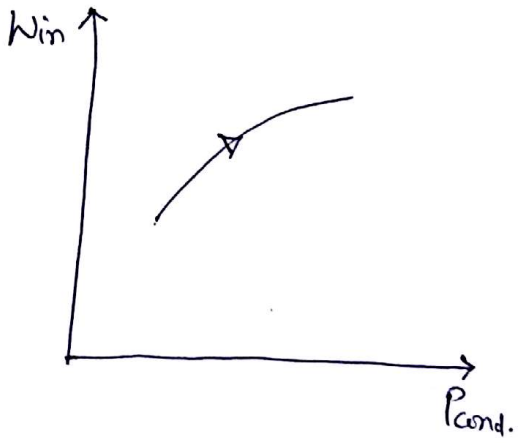
## * Effect of Evaporator Pressure :-

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

$\longrightarrow \eta_V, \dot{m}$ 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 Compresor :-

## 1) Open Type :-

In this Type both Compresor 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 beet its maintenance is Very easy. Because the Compresor and Motor the Seprate units.

## 2) Semi hermitically Sealed :-

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

## 3) Hermitically Sealed :-

In this, the Compresor and motor are placed in the Welded Steel Shell.
The Chances for the leakage of Refrigerant are negligible beet its maintenance is Complex.

## Disadvantage of Hermitically Sealed Compresor :-

- 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 Condensor. 3TR $C_p = 1.005$ KJ/Kg-K

2) H2O/Water Cooled Condensor. $C_p = 4.187$ KJ/Kg-K

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

Air Cooled → 3TR
H2O Cooled. →
- Double tube → 10TR
- Shell & Coil → 50TR
- Shell & tube → 1000TR

## (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 Condenser 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 Proptional to the length of Capillary tube and inversely Proptional to the diamiter of Capillary tube. The pressure drop in the Capillary tube is achieve through frictional Resistance and accularation of the fluid in the tube.

### 2) Constant volume Type :-

#### (1) Automatic Expansion Device :-

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

#### (2) Thermo Static :

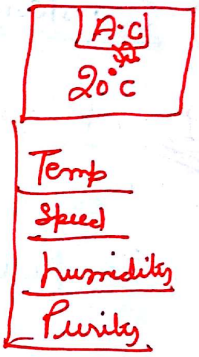
It is used to main, Constant degree of Superheat in the evaporator irrespective of Load.

# Air Conditioning

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

**Air Conditioner**

use → to control/maintain lower Temperatures
→ 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)

Mole. Wt. Vapour = 18

Molecular Wt. air = 29

[which is for dry as well atmospheric air]

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

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

$$P_v = mRT$$

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

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

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

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

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

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

GATE

1 gm/Kg of d.A = $10^{-3}$ Kg/Kg of d.A

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

## 2) Relative humidity $\phi$ :-($\phi$)

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 Dippression :-

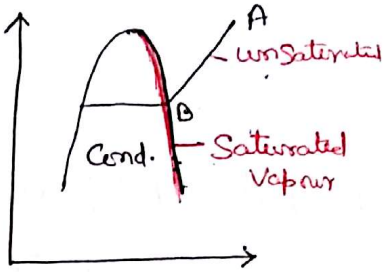
It is the difference b/w DBT & WBT.

## 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 Presure 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 ~~Bulb~~ Wet Bulb diffression is Zero.

~~v) Sling~~

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

## 7) Degree of Saturation / Percent humidity : ($\mu$)

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) Enthalpy of Moist Air :-

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

$t \rightarrow$ DBT (°C)

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

## 9) APJON Formula :-

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

Objective

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

## NOTE :-

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 Dew Point temperature.
4) $P_v' \rightarrow$ It is the Saturation Pressure Corresponding to Wet Bulb temperature.
5) $P_{vs} \rightarrow$ Partial Pressure of Water Vapour under Saturated Condition, Corresponding to $P_{vs}$, the Saturation Temperature Provides the Value of dry Bulb temperature.
6) $P \rightarrow$ It is the total Pressure or if total Pressure not provided then taken its 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 note)

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 | (Bar) Sat. Pressure |
|---|---|---|
| DBT | 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 → Dry Bulb temp
t' → Wet Bulb Tem
$P_v$ → Partial Pressure → DPT
$P_v'$ → Saturation Pressure → WBT
$P_{vs}$ →
P = TP.

Sol$^n$) DBT = 30°C
WBT = 20°C
$P_{atm}$ = 740 mm

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

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

$$= 0.02337 - 0.006583$$

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

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

ii) $\omega = \frac{m_v}{m_a} = \frac{0.622 P_v}{P - P_v}$ Specific humidity

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

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

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 :-

$$P_v = mRT$$

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

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

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

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

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

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

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

T → in Kelvin

$N/m^2$

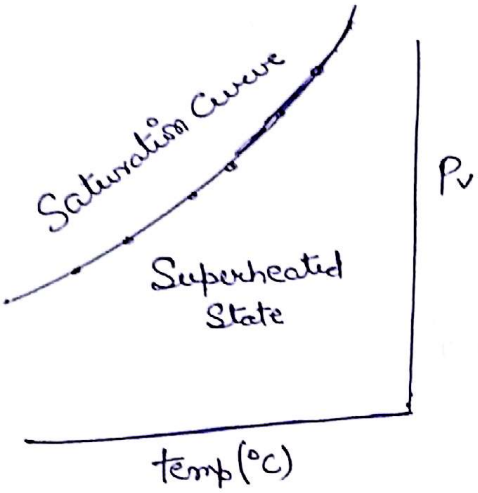
Dalton's law

$$P = P_a + P_v$$

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

# Psychrometry Chart :-

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




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

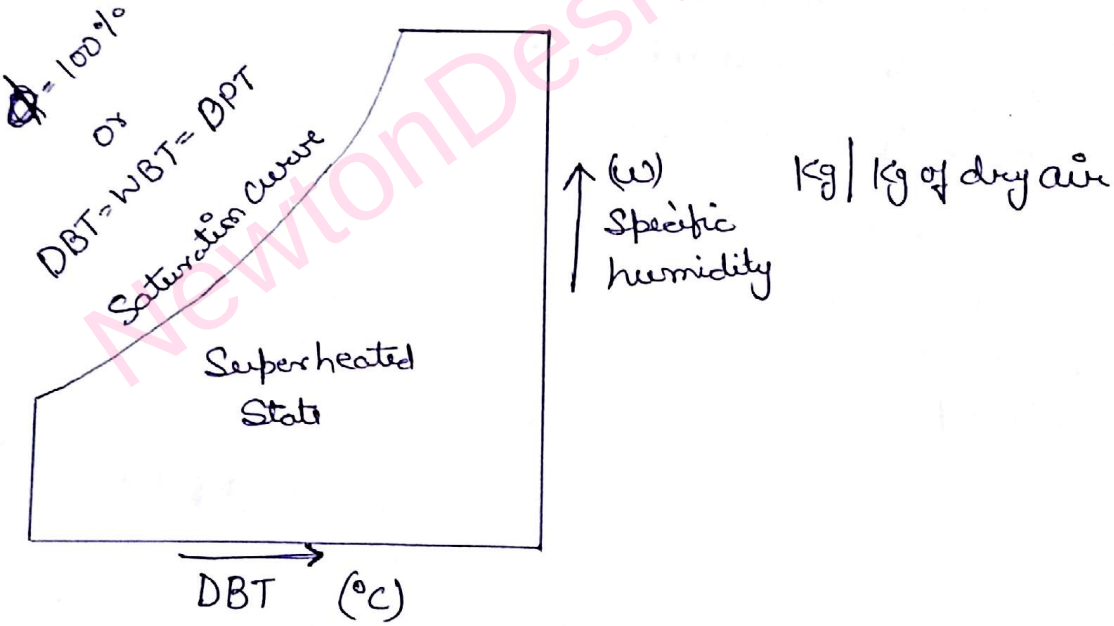
$$\omega = F(P_v)$$

For Superheated

$PV = mRT$

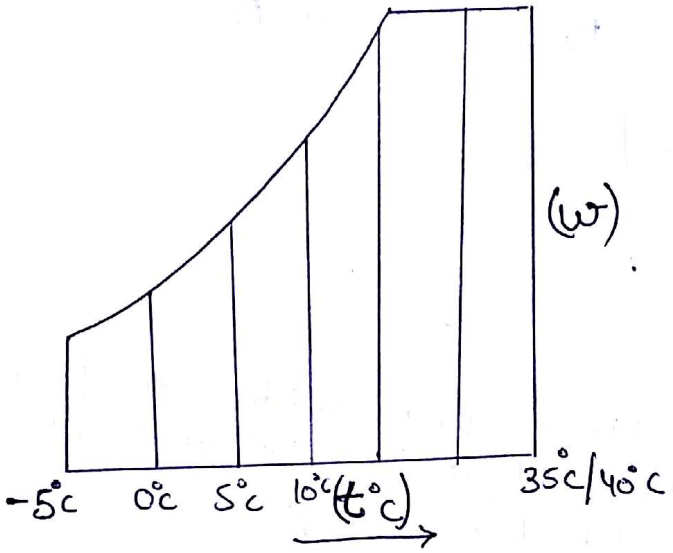
$P \propto T$

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 ($\omega$) Specific humidity.



# Representation of Different Constant Parameters on Psychrometric Chart :-

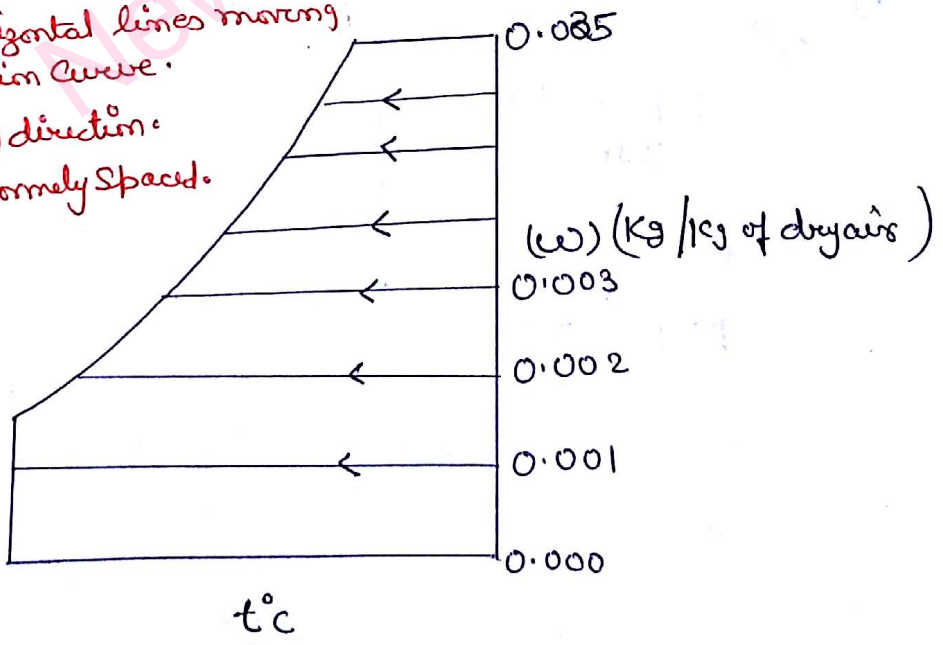
## 1) Constant Dry Bulb Temperature Line DBT :-




→ These are the vertical lines.
→ Increasing order is (+x) direction.
→ These are uniformly spaced.

## 2) Constant Specific humidity lines :- (w)

→ These are Horizontal lines moving towards Saturation Curve.
→ ↑ order in (+y) direction.
→ These are uniformly Spaced.

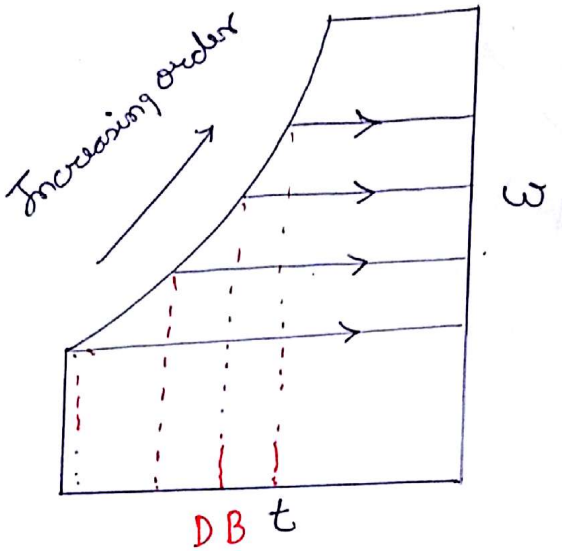


## 3) Constant Dew Point Temperature Lines DPT :-

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

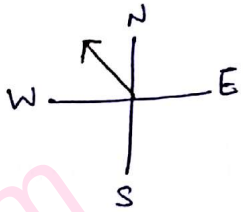
→ These are non uniformly Spaced.

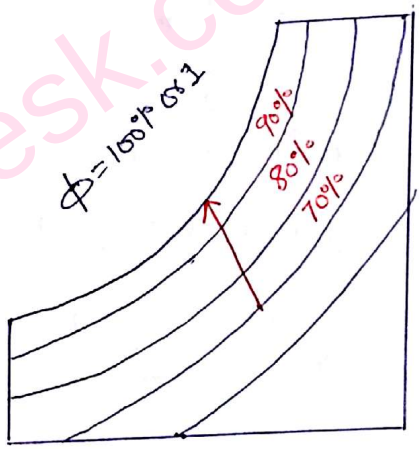
→ Increasing order




## 4) Constant Relative humidity Curve :-

1) These are Parallel to Saturation Curve.
2) increasing order in North-West direction.

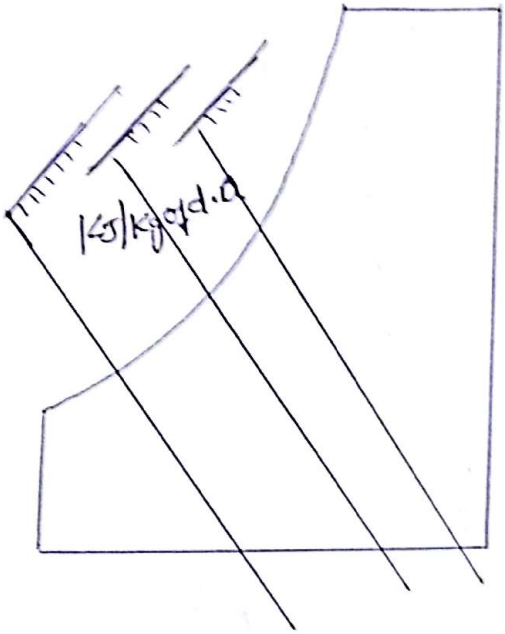


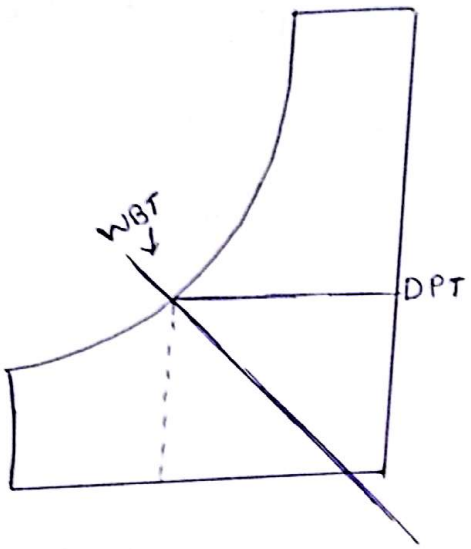

## 5) Constant Enthalpy lines, Constant Wet Bulb temperature line, Constant Specific Volume lines :-

1) h / WBT / v
↓ ↓ ↓
kJ/kg °C $m^3/kg$

② h




③ W.BT

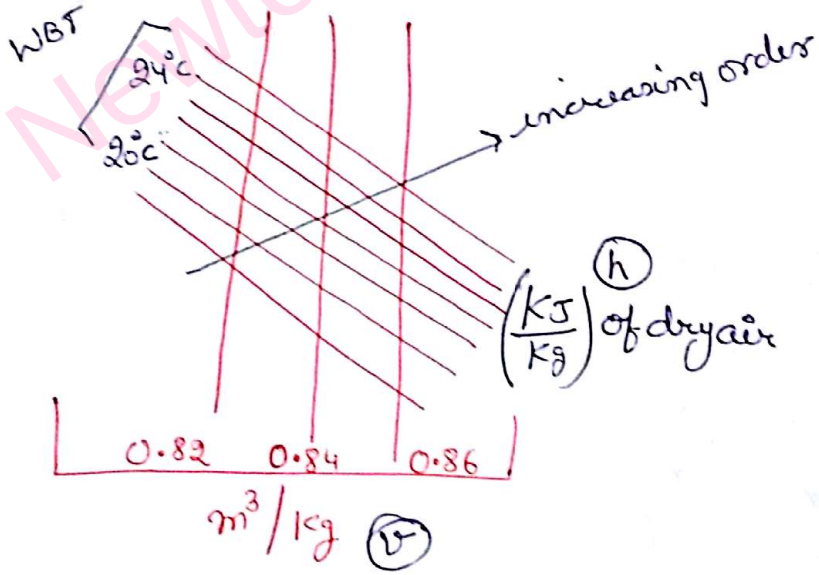



④ h, WBT → Same degree of Inclination

V = highest degree of Inclination

⑤ h → uniformly Spaced.

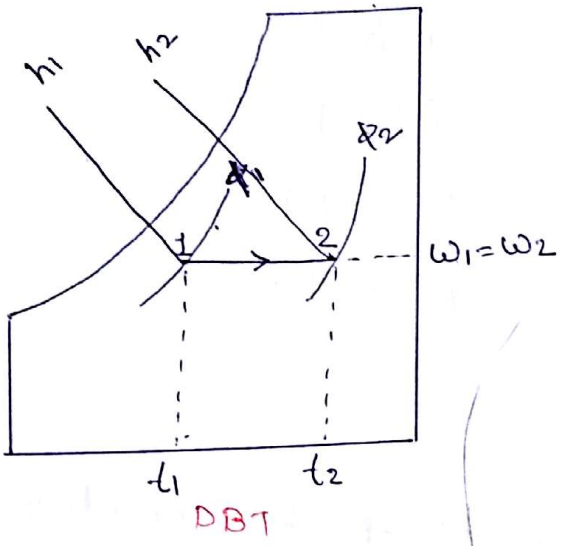
WBT → Non uniformly Spaced.



# Basic Psychrometry Process :-

## 1) Sensible Heating :-

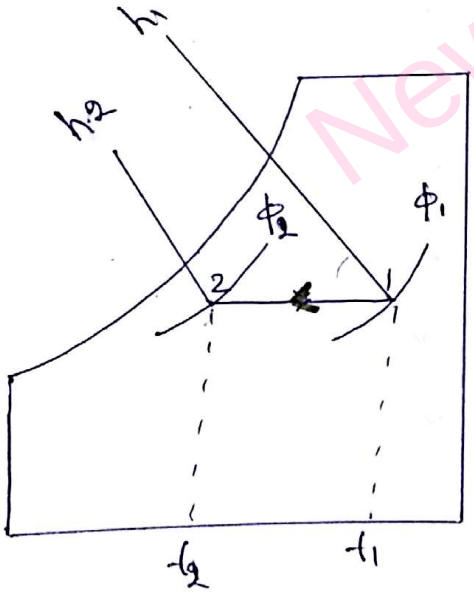
It is the Process of ↑ the dry bulb temperature at Constant Specific humidity.




① t ↑
② W Const
③ DPT Const
④ Φ ↓
⑤ h ↑
⑥ WBT ↑
⑦ V ↑

## 2) Sensible Cooling :- (DBT)

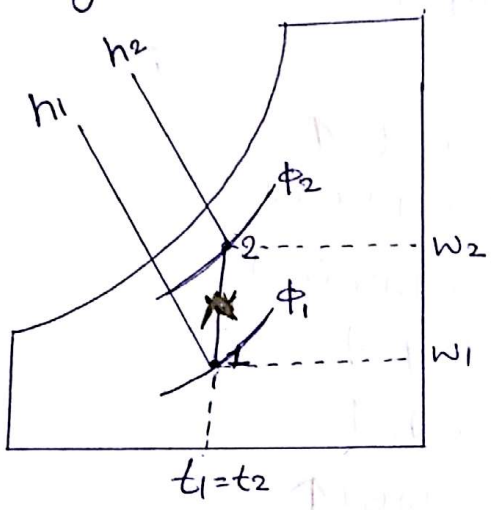
It is the Process of ↓ the temperature at Constant Specific humidity.




① t ↓ (DBT)
② W Const
③ DPT Const
④ Φ ↑
⑤ h ↓
⑥ WBT ↓
⑦ V ↓

## ③ Humidification :-

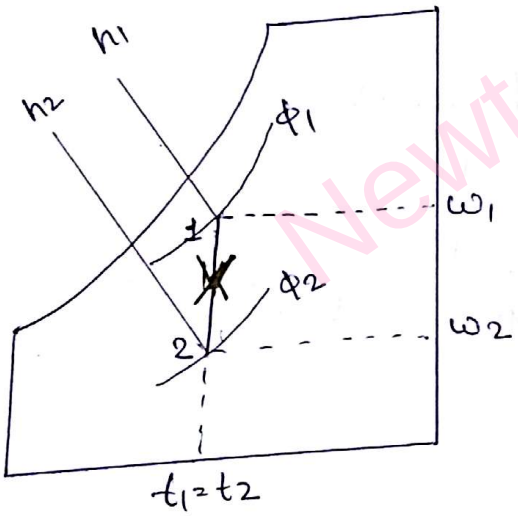
It is the Process of ↑ the Specific humidity at Constant dry Bulb temperature.




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.

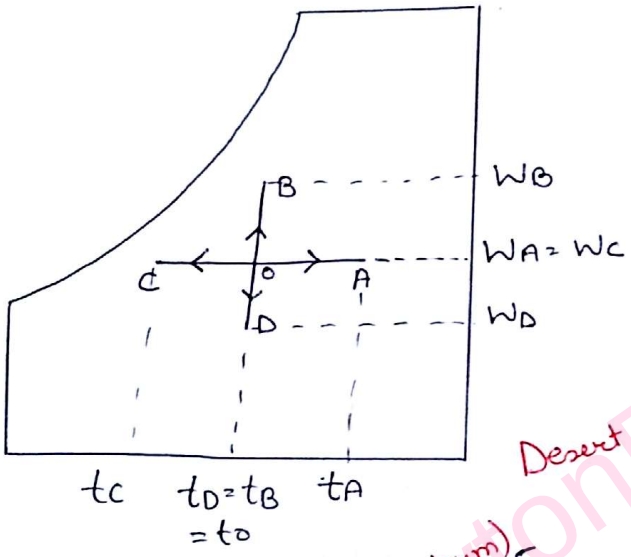


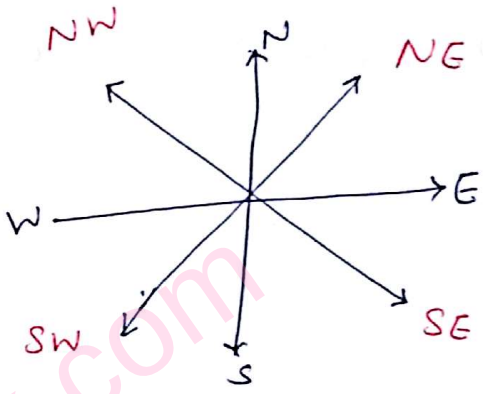

① t Constant
② ω ↓
③ DPT ↓
④ ϕ ↓
⑤ h ↓
⑥ WBT ↓
⑦ v ↓

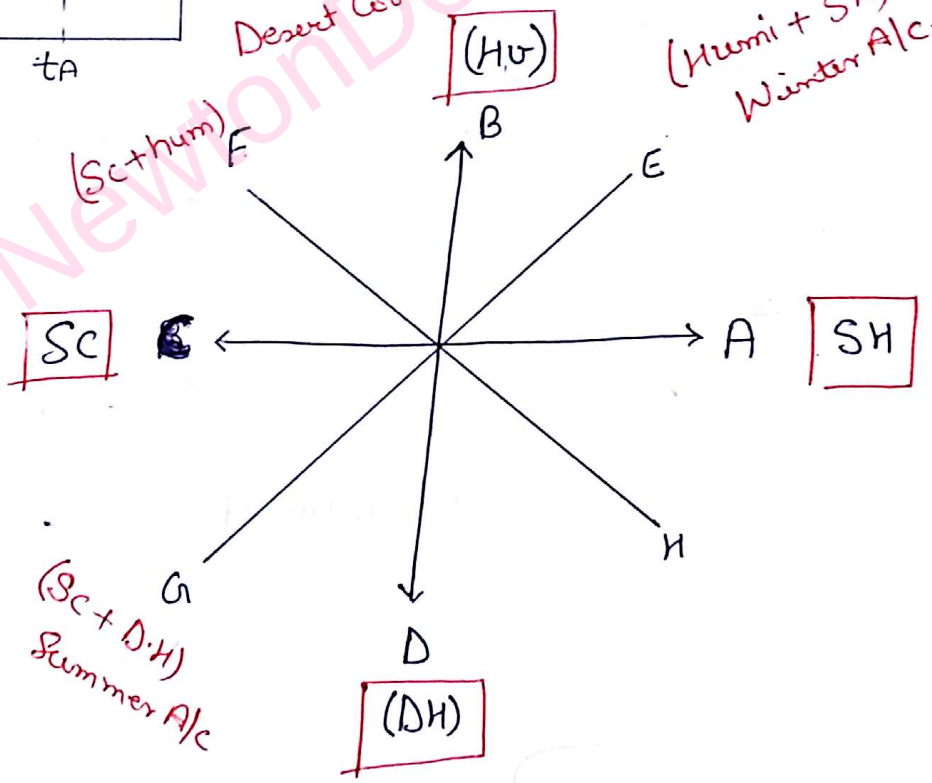
## NOTE :-

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

## Representation of all the above Proces on Same Psychrometric chart :-



## NOTE :-

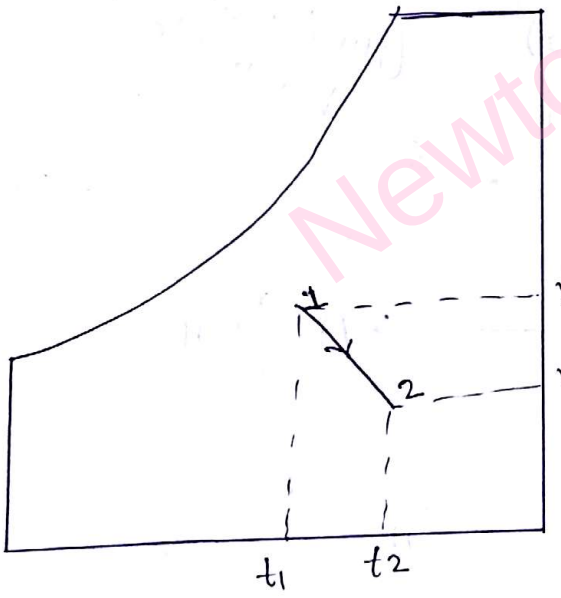
In Case of Summer air Conditioning the process of Cooling & Dehumidification.

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 diffression is, high. (important in terms of question / Interview)

→ 4. Point

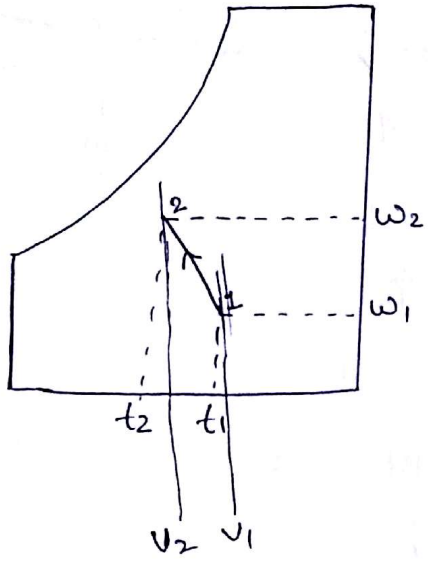



S.H + Dehumidification

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

We Can't say

# Adiabatic or Chemical humidification :-

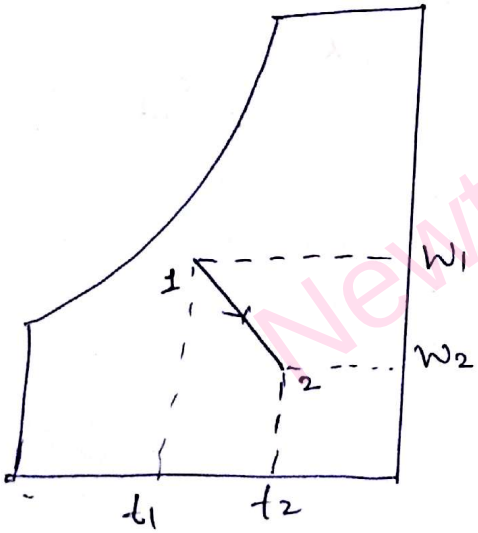



Adiabatic humidification → h → Constant

hu/DH → Decides direction

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

# Adiabatic chemical Dehumidification :-

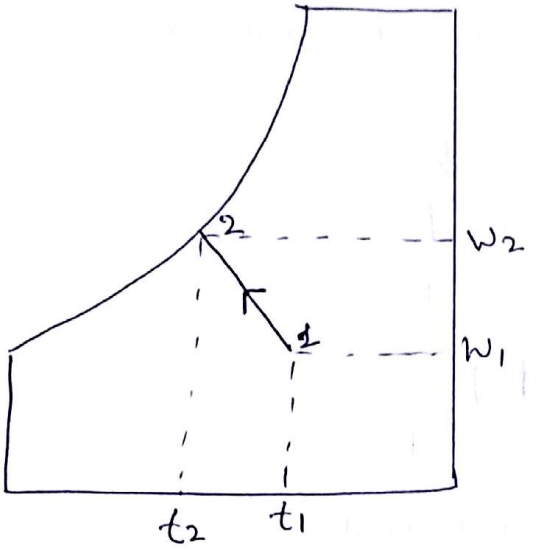


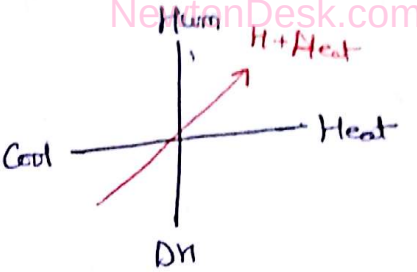

Ad/DH → h → Constant

DH → decides Direction

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

# Adiabatic Saturation :-




(1) $t \downarrow$

(2) $w \uparrow$

(3) DPT $\uparrow$

(4) $\phi \uparrow$

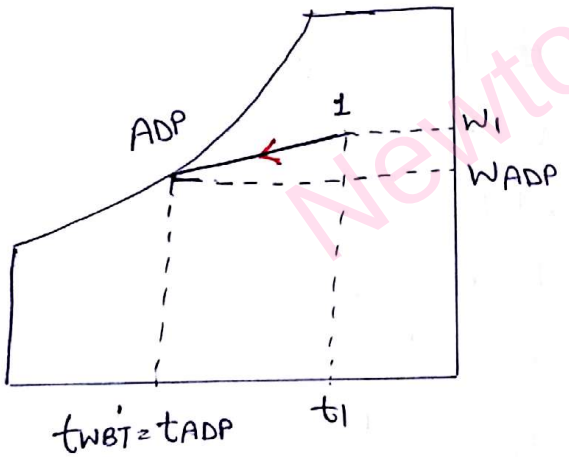
(5) h Const

(6) WBT Const

(7) $v \downarrow$

## 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

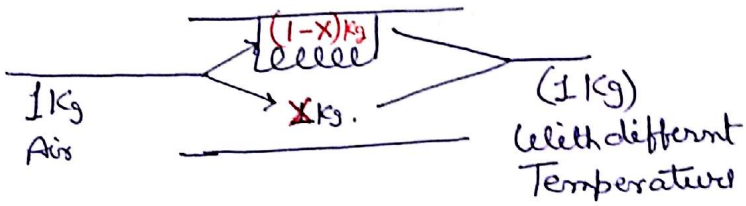
(2) Saturation Curve touch and at Saturation Curve

(3) $\phi_{ADP} = 1$

## By-Pass Factor :- (X)

It Simply Represent the loss.

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

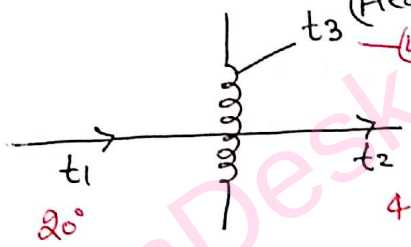



$\eta = 1$

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

## 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.

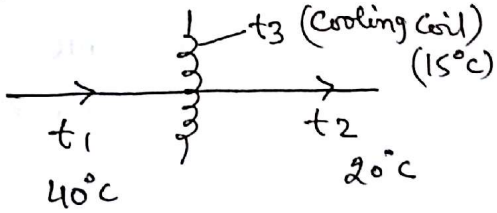



$$(BPF)_{HC} = \frac{t_3 - t_2}{t_3 - t_1}$$ (loss / diff)

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

## 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)

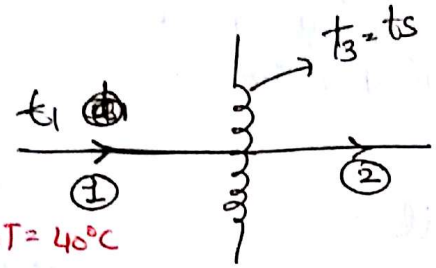
[No such case]

~~test test test~~

$X^N$ = N – Total no. of Coil

Because 1 Heat ② Cool ③ Heat

→




DBT = 40°C

DPT = 20°C

SH, SC → DBT

Humidification → $H_2O$

dehumidification → DPT



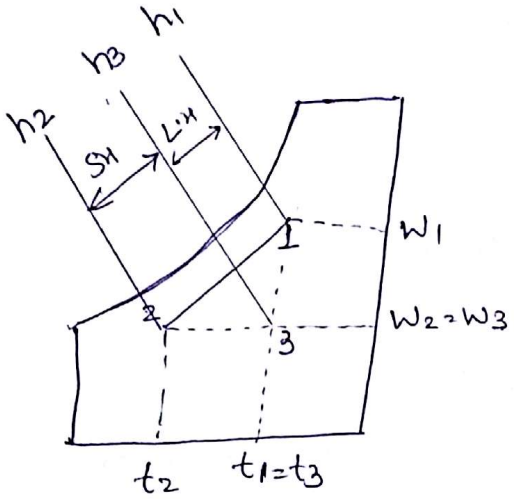
| | ts | outlet temp |
|---|---|---|
| ① | 50°C | SH |
| ② | 50°C $H_2O$ | SH + Hu |
| ③ | Steam (100°C) | SH + Hu |
| ④ | 30°C | SC |
| ⑤ | 20°C or 19°C | ~~SC + Hu~~ SC + DH |

## 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).




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

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

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

## Valve of Sensible Heat factor for different Places :-

| | |
|---|---|
| Residence & PVT. office | 0.9 |
| Resturant & Busy office | 0.8 |
| Auditorium & Cinema hall | 0.7 |
| Dancehall 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, Accaleration 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 2-3°C higher effective temperature than Adults.
Similarly in the case with womens. which need 2-3°C ↑.

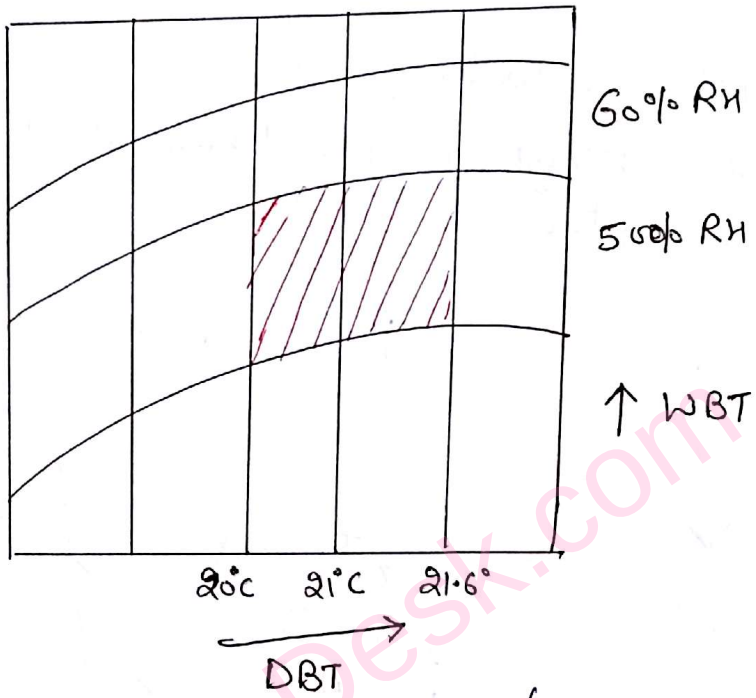
### 3) Kind of Activity :-

If a Person is involved in Activities like dancing, foundary Shop & 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 60%, 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 :-

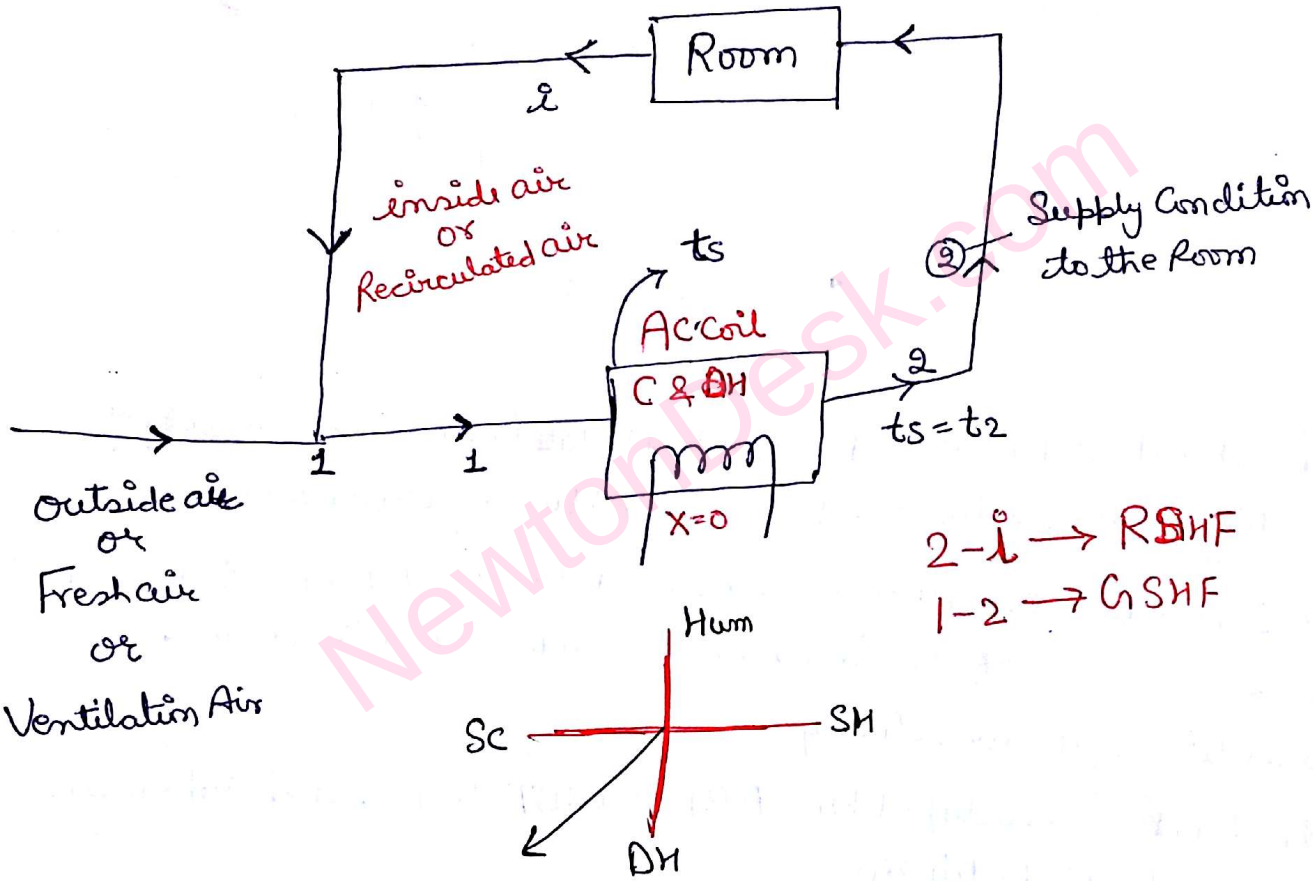
It is the amount of fresh air which is supplied to the A/c Coil, inorder to maintain its purity.

**NOTE:-**

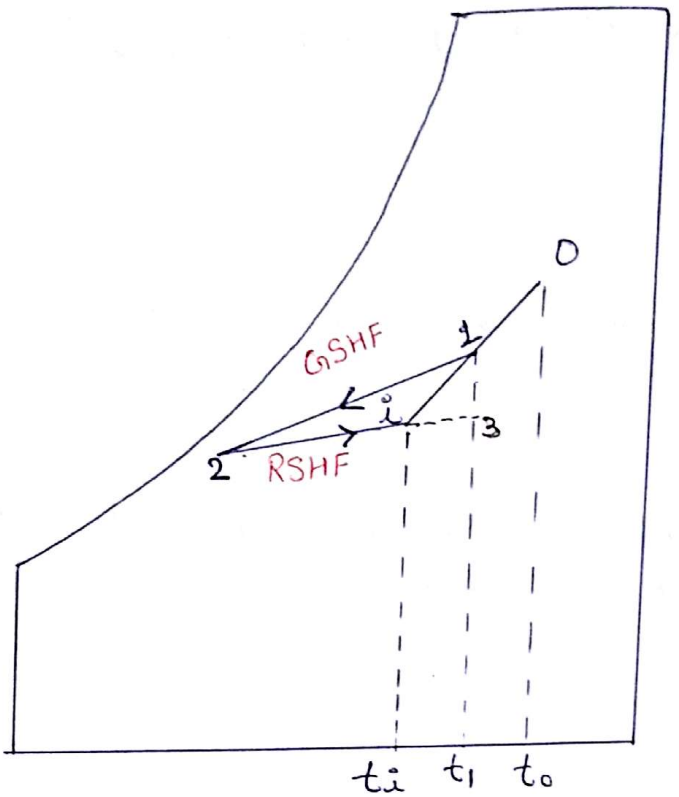
In Case of operation theatre / ICU, 100% outside air is supplied.

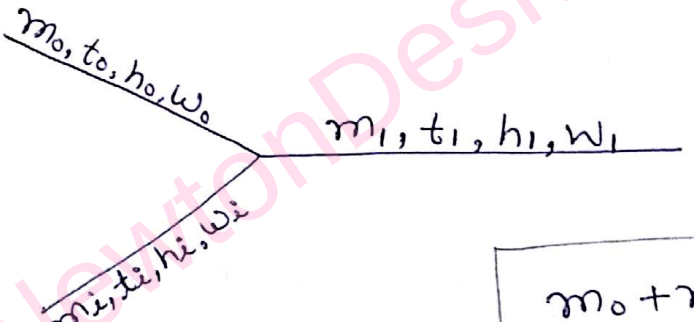
## Summer Air Conditioning :

Air is Passing through cooling & dehumidification coil with (BPF) 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 :

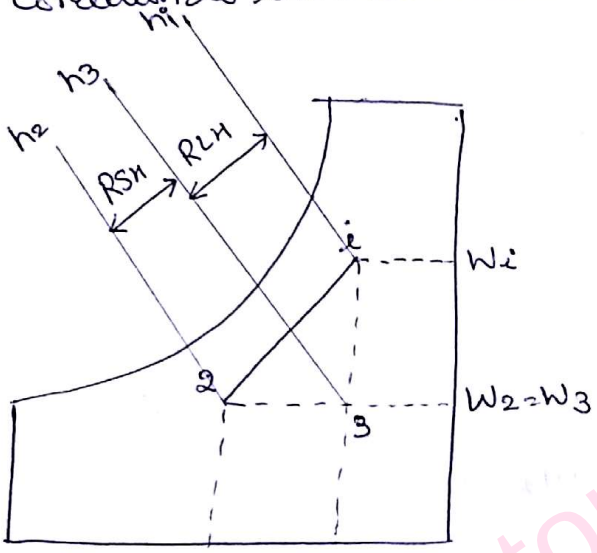
It is the line or curve obtain by the joining of inlet & outlet of the coil.

## 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.




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

$$\boxed{RSHF = \frac{h_3-h_2}{h_i-h_2}}$$

## NOTE :

(Room Sensible Heat)

① $RSH = 0.0204\ Cmm\ \Delta t\ (KW)$

② $RLH = 50\ Cmm\ \Delta\omega\ (KW)$

Cmm = _____ $m^3/min$ ans it must be in this unit

③ No. of Air flow changes/hr

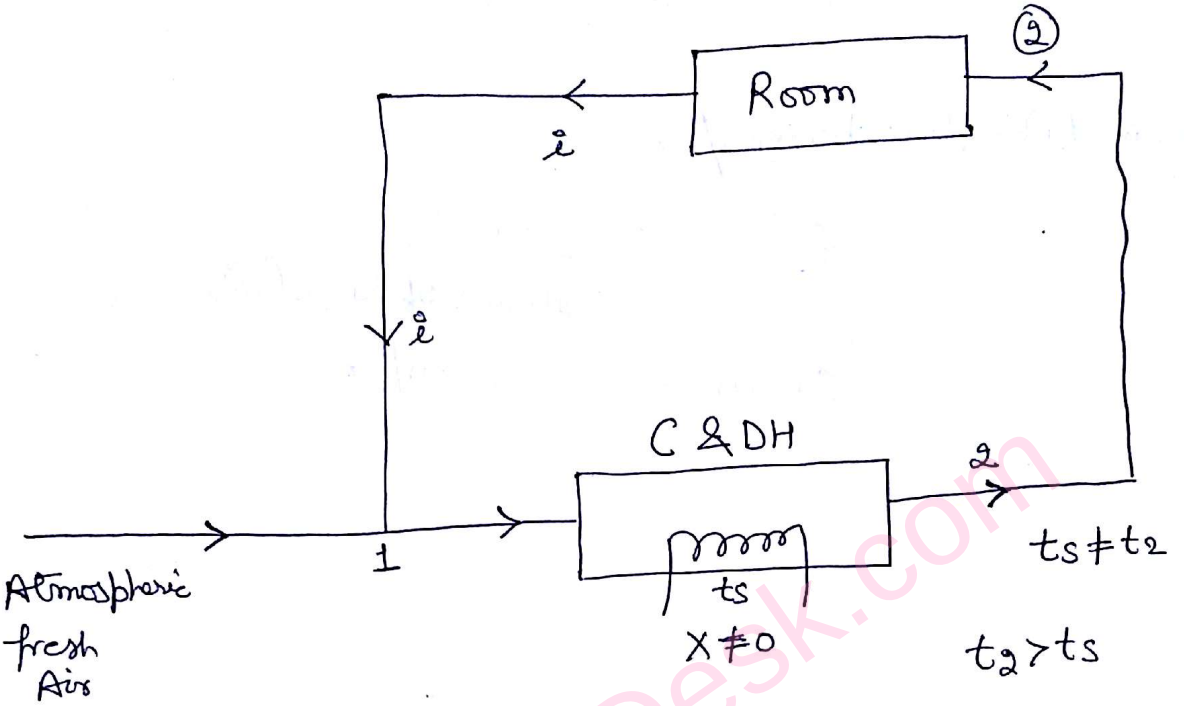
$$= \frac{Cmm}{\text{Volume of Room } (m^3)}$$

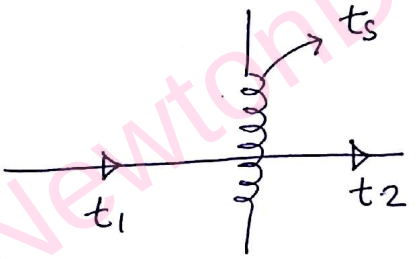
Cmm = _____ $m^3/hr$

Case 2 :-

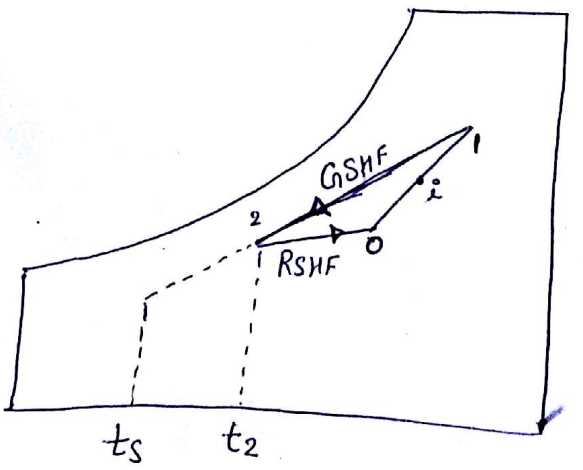
## Summer Air Conditioning :-

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



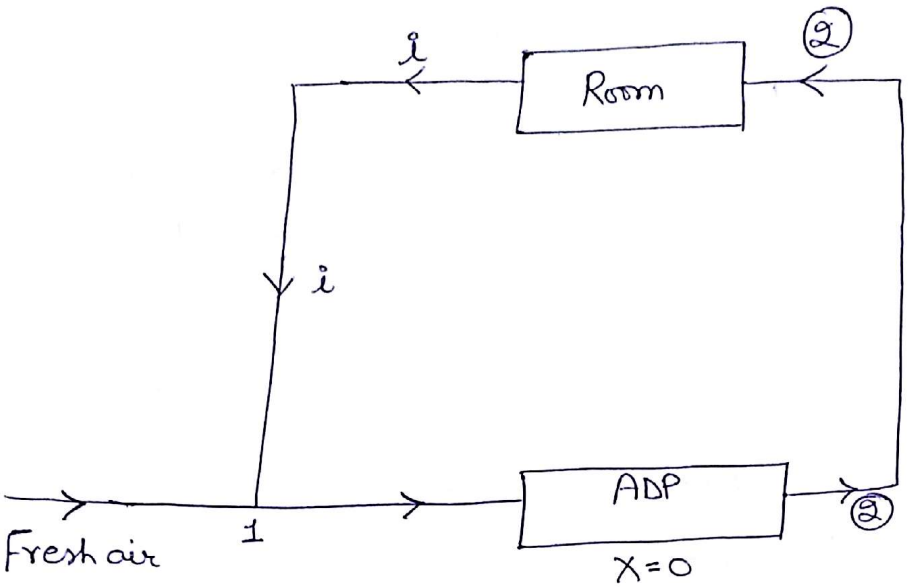




$$X = \frac{t_2 - t_s}{t_1 - t_s} = \frac{h_2 - h_s}{h_1 - h_s} = \frac{\omega_2 - \omega_s}{\omega_1 - \omega_s}$$



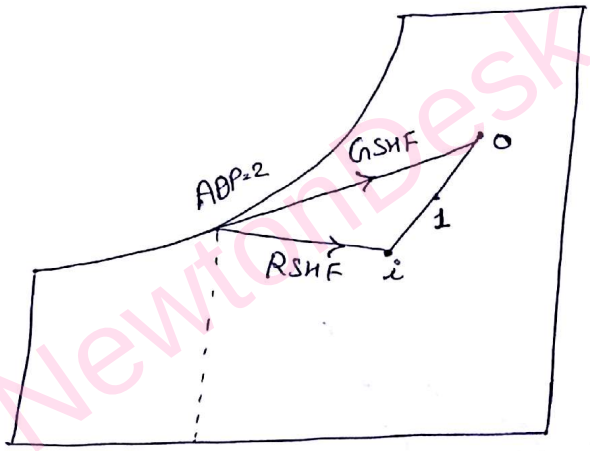
## Case-3

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




Summary

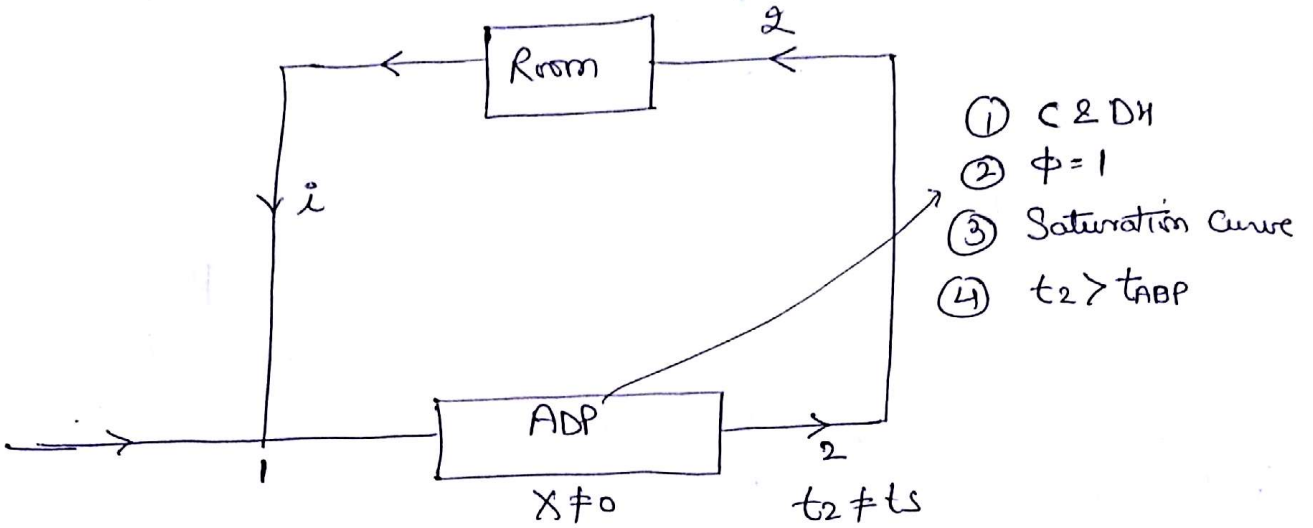
1. C & DH
2. Saturation Curve (ADP)
   Apparatus Dew Pt.
3. $\phi_{ADP} = 1$
4. $t_2 = t_{ADP}$

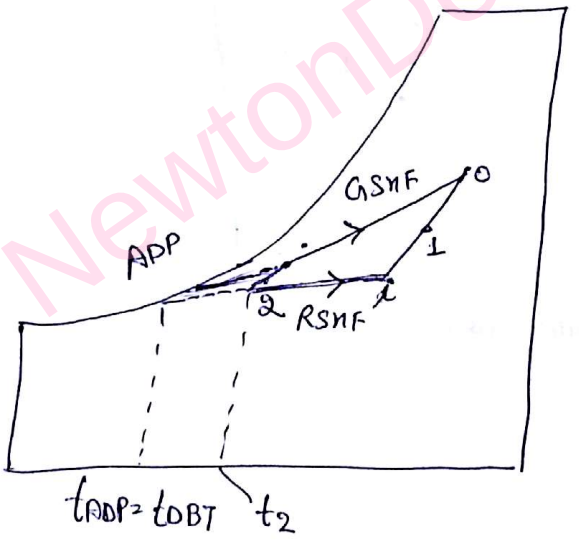



$t_2 = t_{ADP} = t_{DBT}$

Case 4

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



# 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 Can be used as humidifier, Dehumidifier and filter.
+
filter