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BASIC CONCEPTS

Refrigeration :->

Refrigeration is the process of maintaining lower temperature compared to surrounding.

Refrigerant :->

These are the substances which are used for producing lower temperatures eg water, air, NH₃, R-12, R-134 etc.

NOTE :-> As the lower temperatures are to be maintained continuously therefore system must run on a cycle.

Refrigeration Effect :->

It is the amount of heat that is to be extracted from storage space in order to maintain lower temperatures.

$$COP = \frac{\text{Desired effect}}{\text{Energy input}}$$

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

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

Unit of Refrigeration :->

The unit of refrigeration is the tonne of refrigeration. It is the amount of heat that is to be removed from one tonne of water at 0°C in water in order to convert it into ice at 0°C in 24 hours. Therefore Tonn of refrigeration is heat transfer rate

$$1TR = 3.5 \text{ KJ/sec} = 3.5 \text{ kW} = 210 \text{ KJ/min}$$

Ideal Refrigeration cycle $\Rightarrow$

The ideal refrigeration cycle is the reversed Carnot cycle.

$$\text{COP} = \frac{RE}{W} \Rightarrow \text{valid for Rev. \& Irrev. cycle}$$

$$(\text{COP})_{\text{Rev, R}} = (\text{COP})_{\text{ideal}} = (\text{COP})_{\text{max}} = \frac{T_L}{T_H - T_L}$$

Significance of COP $\Rightarrow$

COP represents the running cost of system. Greater the COP, smaller is the work input for a given refrigeration capacity and hence lower is the running cost. Therefore higher COP is always desirable.

$T_H \rightarrow$ Condenser temperature

$T_L \rightarrow$ Evaporator temperature

$$\text{Refrigeration Capacity} = \dot{m} \times RE$$

$$RC = (\dot{m} \times RE) \text{ KJ/sec}$$

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

$$P_{in} = (\dot{m} \times W_{in}) \text{ KJ/sec}$$

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

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

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

Ques 1: → A Carnot refrigerator requires 1.5 kW per tonne of refrigeration to maintain a region at -30°C . Then COP of refrigerator is

Soln: →

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

$$R_C = 1 \text{ TR}$$

$$\text{COP} = \frac{R_C}{P_{in}} = \frac{3.5}{1.5} = 2.33$$

Ques 2: → A refrigerating m/c working on reversed Carnot cycle takes out 2 kW from system while working b/w temp limits of 300 K and 200 K. Then find COP and power input

Soln: →

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

$$\text{COP} = \frac{R_C}{P_{in}} \Rightarrow P_{in} = \frac{R_C}{\text{COP}} = \frac{2}{2} = 1 \text{ kW}$$

Ques 3: → The working temperatures in evaporator & condenser coils are -30°C and 32°C respectively. If the actual refrigerator has a COP of 0.75 times of max COP. Find the power input for R_C of 5 kW.

Soln: →

$$\begin{aligned} \text{COP}_{\max} &= \frac{T_L}{T_H - T_L} \\ &= \frac{243}{305 - 243} = 3.91 \end{aligned}$$

$$\text{COP}_{\text{act}} = 0.75 \times \text{COP}_{\max} = 0.75 \times 3.91 = 2.93$$

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

$$P_{in} = \frac{R_C}{\text{COP}} = \frac{5}{2.93} = 1.7 \text{ kW}$$

Ques 4 :- The ratio of minimum to maximum temp. in a Carnot cycle is 0.8. If a heat pump is operated b/w same temperature limits. Then COP of Heat pump is

a) 2

b) 3

c) 4

d) 5

80% →

$$T_{\min}/T_{\max} = 0.8 T_{\max}$$

$$\text{COP} = \frac{T_h}{T_h - T_l} = \frac{T_h}{T_h - 0.8 T_h} = 5$$

Ques 5 :- A condenser of Refrigeration system rejects heat at the rate of 120 kW and the power input to compressor is 30 kW. Then the COP of refrigerator is

80% →

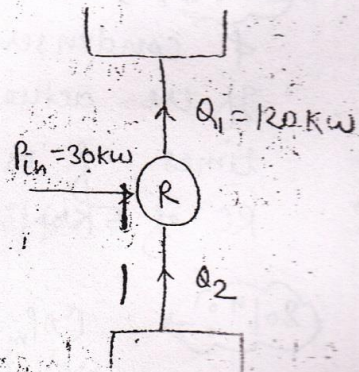
energy conservation

$$30 + Q_2 = 120$$

$$Q_2 = 90 \text{ kW}$$

$$\text{COP} = \frac{Q_c}{P_{\text{in}}} = \frac{90}{30}$$

$$\text{COP} = 3$$



Ques 6 :- A heat engine working b/w T_1 & T_2 has an efficiency of 40%. If this engine is reversed. then what is the COP of the refrigerator.

80% →

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

$$= \frac{1}{W/Q_1} = \frac{1}{\eta}$$

$$(COP)_{HP} = 1 + (COP)_{ref}$$

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

$$(COP)_R = \frac{1}{0.4} - 1 = 1.5$$

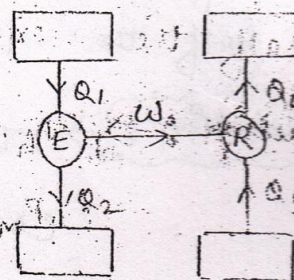
Ques 7: → In the figure shown 'E' is a heat engine with an efficiency of 40% and 'R' is Refrigerator given that $Q_2 + Q_4 = 3Q_1$. Then find the COP of the refrigerator.

Solution: → $COP_R = \frac{Q_3}{P_{in}} = \frac{Q_3}{Q_1 - Q_2}$

$$= \frac{2Q_1}{0.4Q_1} = 5$$

$$\eta_e = \frac{Q_1 - Q_2}{Q_1} \Rightarrow 0.4Q_1 = Q_1 - Q_2$$

$$Q_2 = 0.6Q_1$$



$$W + Q_3 = Q_4, \quad Q_3 = Q_4 - (Q_1 - Q_2) = 2Q_1$$

Ques 8: → A domestic refrigerator set at 2°C handle a refrigeration load of 8000 kJ/day. The ambient temperature is 30°C and the COP of refrigerator is 0.15 times of max. COP. Then find the daily electricity consumption in kWh.

Solution: → $COP_{max} = \frac{273}{303 - 273} = 9.82$

$$COP_{act} = 0.15 \times 9.82 = 1.473$$

$$RC = \frac{8000 \text{ kJ}}{24 \times 3600} = \frac{8000}{86400} = 0.092 \approx 8000 \text{ kJ/day}$$

$$P_{in} = 8000 / 1.473 = 5431 \text{ kJ/day}$$

$$\begin{aligned} \text{per day consumption} &= 5431 \text{ kJ} = 5431 \text{ kJ} \times \frac{1}{3600} \times \text{sec} \\ &= 5431 \text{ kW} \times \frac{\text{hr}}{3600} = 1.5 \text{ kWh} \end{aligned}$$

Ques 9: → A refrigerator storage is supplied with 3600 kg of a substance at a temperature of 27°C . The substance has to be cooled to -23°C . The cooling takes place in 10 hours. The specific Heat of substance is 2 kJ/kgK above freezing point at 0.5 below the freezing point at freezing point is -3°C . The latent heat of freezing is 230 kJ/kg . What is the power required to drive the plant. If the actual COP is half the maximum COP.

Solution: → $T_h = 300 \text{ K}$, $T_L = 250 \text{ K}$

$$\text{COP}_{\text{max}} = \frac{T_L}{T_h - T_L} = \frac{250}{300 - 250} = 5$$

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

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

$$RC = Q = \sum H_1 + LH + \sum H_2$$

$$Q = m [C_1 \Delta t_1 + LH + C_2 \Delta t_2]$$

$$\begin{aligned} &= \frac{3600}{10 \times 3600} [2(27 - (-3)) + 230 + 0.5(-3 - (-23))] \\ &= 30 \text{ kW} \end{aligned}$$

$$P_{in} = \frac{30}{2.5} = 12 \text{ kW}$$

Vapour Compression Refrigeration Cycle

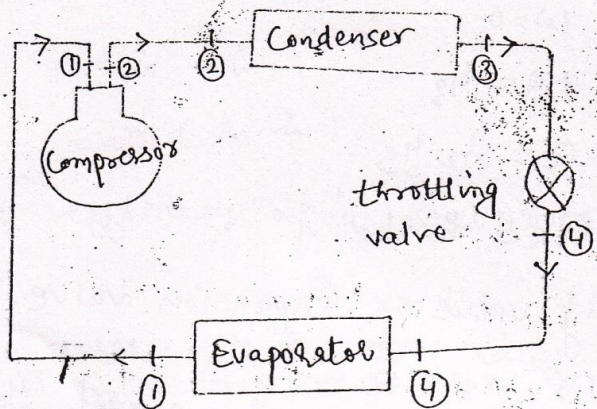
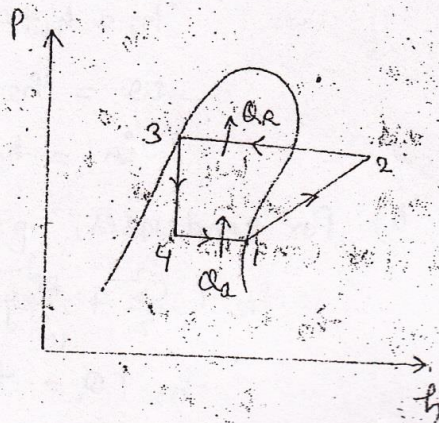
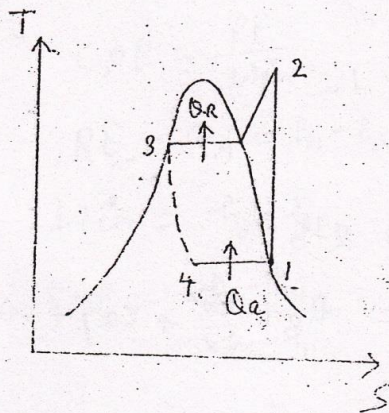
Simple or standard v-c cycle $\rightarrow$

1-2 Isentropic (Rev-adiabatic) Compression

2-3 Constant pressure heat rejection

3-4 Throttling (Isenthalpic) Expansion

4-1 Constant pressure heat absorption



Each Device is an open system and the system run steadily and hence by applying the SFEE, we have,

d) Evaporator, process 4-1

$$h_4 + \frac{C_v^2}{2} + z_4 g + Q = h_1 + \frac{C_1^2}{2} + z_1 g + w$$

$$h_4 + Q = h_1 + w$$

$$w = 0$$

$$h_4 = h_1 - Q$$

$$Q = h_1 - h_4$$

COP of V-C cycle $\Rightarrow$

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

$$RE = Q_A = h_1 - h_4$$

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

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

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

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

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

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

Volumetric efficiency of a Reciprocating Compressor $\Rightarrow$

It is the breathing capacity of a compressor. It is the ratio of actual volume to swept volume. If $\dot{m}$ is the actual mass flow rate of refrigerant and v_1 is actual specific volume at inlet to compressor.

Then actual volume entering to the compressor is $\dot{m} v_1$

$$V_{act} = \dot{m} v_1$$

$$V_{\text{swept}} = \frac{\pi}{4} D^2 L N$$

$$\eta_{\text{vol}} = \frac{\dot{m} v_1}{\frac{\pi}{4} D^2 L N} \quad \text{for single compressor}$$

If there are K number of compressors

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

$$* \text{ If } V_{\text{act}} (\text{m}^3/\text{sec}) \rightarrow V_{\text{swept}} (\text{m}^3/\text{sec})$$

$$V_{\text{act}} (\text{m}^3/\text{min}) \rightarrow V_{\text{swept}} (\text{m}^3/\text{min})$$

Volumetric efficiency is also given by

$$\eta_{\text{vol}} = 1 + C - C \left[\frac{P_2}{P_1} \right]^{1/n}$$

$$\text{Where } C = \text{clearance Ratio} = \frac{V_c}{V_s} = \frac{\text{clearance volume}}{\text{swept volume}}$$

Imp

Ques) $\Rightarrow$ In a V-C refrigeration cycle following data is obtained

$$h_1 = 180 \text{ kJ/kg} \quad \text{entry to compressor}$$

$$h_2 = 210 \text{ kJ/kg} \quad \text{exit to compressor}$$

$$h_3 = 80 \text{ kJ/kg} \quad \text{exit to condenser}$$

Then COP is

$$\begin{aligned} \text{Solution } \Rightarrow \text{COP} &= \frac{h_1 - h_3}{h_2 - h_1} \\ &= \frac{180 - 80}{210 - 180} = 3.33 \end{aligned}$$

Ques 2: $\rightarrow$ In a 6 kW cooling capacity refrigerator system operating on a single simple v.c. cycle. Refrigerant enters the evaporator with an enthalpy of 75 kJ/kg and leaves with an enthalpy of 183 kJ/kg. The enthalpy of the refrigerant after compression is 210 kJ/kg. Find

(i) COP

(ii) power input in kW

(iii) Rate of Heat Transfer at the condenser

Solution $\rightarrow$ As we know

$$Q_R = h_2 - h_3$$

$$= 210 - 75 = 135 \text{ kJ/kg}$$

$$\text{Heat Transfer} = 135 \text{ kJ/kg}$$

$$\text{Power Input} = m \times w_{in}$$

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

$$= 210 - 183 = 27 \text{ kJ/kg}$$

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

$$= \frac{183 - 75}{210 - 183} = 4$$

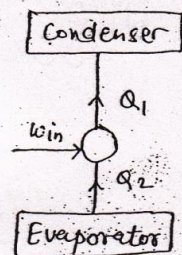
$$P_{in} = \frac{R_C}{\text{COP}} = \frac{5}{4} = 1.25 \text{ kW}$$

$$R_C = m(h_1 - h_4)$$

$$m = \frac{R_C}{h_1 - h_4} = \frac{5}{183 - 75} = 0.046 \text{ kg/s}$$

$$Q_1 = w + Q_2$$

$$= 1.25 + 5 = 6.25 \text{ kJ/kg}$$



Ques 8: In a simple v-c cycle followings are the properties of refrigerant at various points.

Compressor inlet : $h_1 = 183.2 \text{ kJ/kg}$, $v_1 = 0.0767 \text{ m}^3/\text{kg}$

Compressor discharge : $h_2 = 222.6 \text{ kJ/kg}$, $v_2 = 0.0614 \text{ m}^3/\text{kg}$

Condenser exit : $h_3 = 84.9 \text{ kJ/kg}$, $v_3 = 0.00083 \text{ m}^3/\text{kg}$

The swept volume is 1.5 litre. The volumetric efficiency is 80%. speed (N) is 1600 rpm. Find

(i) power input to compressor

(ii) RC in kW.

Solution: $\rightarrow \text{COP} = \frac{h_1 - h_3}{h_2 - h_1} = \frac{183.2 - 84.9}{222.6 - 183.2}$

$\text{COP} = 2.4949$

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

$0.80 = \frac{\dot{m} (0.0767)}{1.5 \times 10^{-3} \times \frac{1600 \times \pi}{60}}$ $\begin{cases} N = 1600 \text{ rpm} \\ K = 1 \end{cases}$

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

$\text{RC} = \dot{m} (h_1 - h_4) = \dot{m} (h_1 - h_3)$

$= 0.417 (183.2 - 84.9)$

$\text{RC} = 40.99 \text{ kW}$

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

$= 0.417 (222.6 - 183.2)$

$P_{\text{in}} = 16.42 \text{ kW}$

Designation of a Refrigerant $\rightarrow$

Case I $\rightarrow$

Saturated Hydrocarbon Refrigerant $\rightarrow$

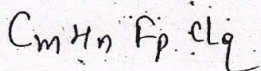
If the chemical formula is $C_m H_n F_p Cl_q$ of a refrigerant. Then such a refrigerant is designated as

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

where

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

eg. CCl_2F_2



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

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

$$R - 012$$

$$R - 12$$

eg. R-11

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

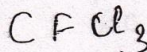
$$m=1,$$

$$n=0$$

$$p=1$$

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

$$q=3$$

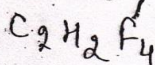


eg. R-134

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

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

$$q=0$$



eg. If designation is given

$$R-22$$

$$R-022$$

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

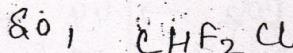
$$m+1=0 \Rightarrow m=-1$$

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

$$p=2 \Rightarrow p=2$$

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

$$q=1$$

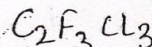


eg. R-113

$$m=2, n=0, p=3$$

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

$$q=3$$



$$R-11 \rightarrow CFCl_3$$

$$R-12 \rightarrow CF_2Cl_2$$

$$R-22 \rightarrow CHF_2Cl$$

$$R-113 \rightarrow C_2F_3Cl_3$$

Case 2: → Unsaturated Hydrocarbon →

If chemical formula of refrigerant is $C_m H_n P_p Cl_q$ then the refrigerant is designated as

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

where $n+p+q = 2m$

eg. C_2H_4

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

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

$$q=0$$

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

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

$$R - 1150$$

Case 3: → Inorganic Refrigerant

$R - 700 + \text{molecular weight}$

$$NH_3 \rightarrow 17 \rightarrow R-717$$

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

$$\text{Water} \rightarrow 18 \rightarrow R-718$$

$$\text{air} \rightarrow 29 \rightarrow R-729$$

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

Gate

Quesy: → An ice making plant using R-12 is having an evaporator saturation temperature of -25°C and Condenser saturation temp of 35°C . The vapour is leaving the compressor at 65°C . Following table shows the properties of refrigerant

$t^{\circ}\text{C}$	$P(\text{kPa})$	$h_f(\text{kJ/kg})$	$h_g(\text{kJ/kg})$
-25	123.7	13.3	176.5
35	850	69.6	201.5

Enthalpy of superheated refrigerant is at 850 kPa & 65°C is 225.5 kJ/kg . Calculate

- (i) COP
(ii) If the capacity of plant is 5 kW . Calculate mass flow rate of refrigerant and power consumption in kW .

Solution: →

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

$$h_1 = h_g \text{ at } -25^{\circ}\text{C} = 176.5\text{ kJ/kg}$$

$$h_3 = h_f \text{ at } 35^{\circ}\text{C} = 69.6\text{ kJ/kg}$$

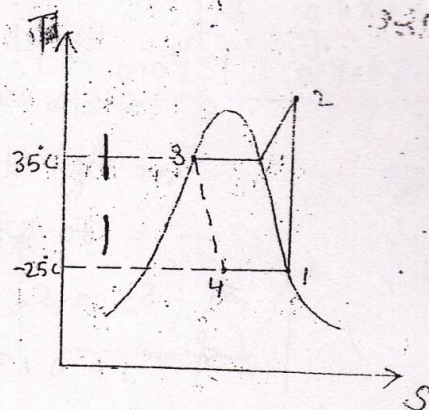
$$\text{COP} = \frac{176.5 - 69.6}{225.5 - 176.5} = 2.18$$

$$\text{RC} = 5\text{ kW}$$

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

$$\dot{m} = \frac{5}{(176.5 - 69.6)} = 0.046\text{ kg/sec}$$

$$\text{COP} = \frac{\text{RC}}{\text{P}_{\text{in}}} \Rightarrow \text{P}_{\text{in}} = \frac{5}{2.18} = 2.29\text{ kW}$$



Ques 5 → A refrigerator based on V-C cycle operates between temperature limits of -20°C and 40°C . The refrigerator enters the condenser as saturated vapour and leaves as saturated liquid. The enthalpy and entropy of saturated liquid and saturated vapour is shown in figure

$t^{\circ}\text{C}$	$h_f (\text{KJ/kg})$	$h_g (\text{KJ/kg})$	$s_f (\text{KJ/kgK})$	$s_g (\text{KJ/kgK})$
-20	20	180	0.07	0.7366
40	80	200	0.3	0.67

If the refrigerant flow rate is 0.025 kg/sec . Find RC in kW and COP of system.

Solution →

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

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

$$= 200 \text{ KJ/kg}$$

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

$$= 80 \text{ KJ/kg}$$

$$s_1 = s_2 = s_g \text{ at } 40^{\circ}\text{C}$$

$$s_1 = s_2 = 0.67$$

$$s_1 = s_f + x(s_g - s_f)$$

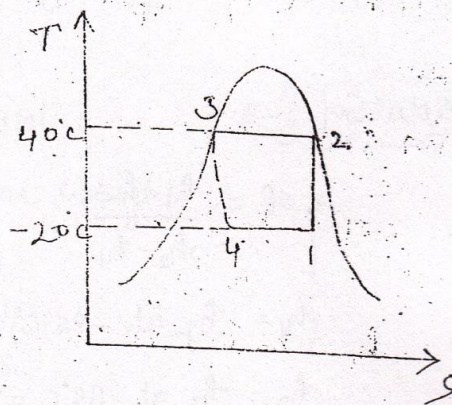
$$0.67 = 0.07 + x(0.7366 - 0.07)$$

$$x = 10.9$$

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

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

$$\text{COP} = \frac{164 - 80}{200 - 164} = 2.33, \quad \text{RC} = \dot{m}(h_1 - h_3) = 2.1 \text{ kW}$$



Ques 6 $\Rightarrow$ A V-C refrigeration using R-12 produces 8640 kg of ice per day. The condensing & evaporating temperatures are 48°C and -20°C . The cycle is simple V-C cycle. Water at 25°C is used to form ice and temp of ice should be -8°C . Heat flow into the brine can from surrounding may be taken as 10% of total heat removed from water to form ice at -8°C . Determine the power required to drive the compressor in kW.

Take specific heat of ice as 2.26 kJ/kgK & latent heat is equal to 334.72 kJ/kg . Specific heat of water 4.18 kJ/kgK and specific heat of vapour refrigerant is equal to 0.82 kJ/kgK .

$t^\circ\text{C}$	$P(\text{bar})$	$h_f(\text{kJ/kg})$	$h_g(\text{kJ/kg})$	$s_f(\text{kJ/kg})$	$s_g(\text{kJ/kg})$
48	11.64	82.83	205.83	0.29173	0.682
-20	1.51	17.82	178.74	0.0731	0.7087

Solution $\Rightarrow$

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

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

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

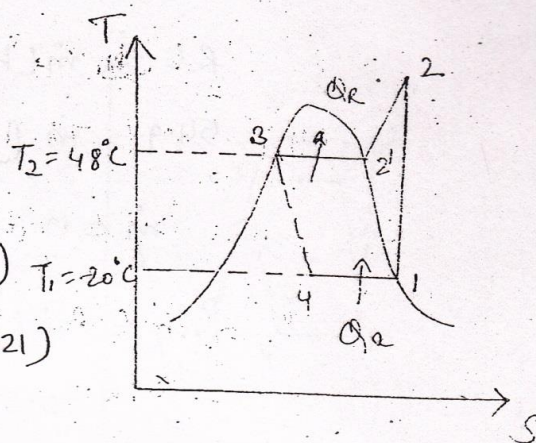
$$h_2 = h_2' + C_{p,vap}(T_2 - T_2')$$

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

$$s_1 = s_2$$

$$s_1 = 0.7087 = s_2$$

$$s_2 = s_2' + C_{p,vap} \ln T_2/T_2'$$



$$0.7087 = 0.682 + 0.82 \times \left(\frac{14}{321} \right)$$

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

$$h_2 = 205.83 + 0.82 (331.6 - 321) \\ = 214.5 \text{ kJ/kg}$$

$$\text{H}_2\text{O at } 35^\circ\text{C} \rightarrow \text{H}_2\text{O at } 0^\circ\text{C} \rightarrow \Sigma H_1$$

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

$$\text{ice at } 0^\circ\text{C} \rightarrow \text{ice at } -8^\circ\text{C} \rightarrow \Sigma H_2$$

$$Q_a = \Sigma H_1 + \text{LH} + \Sigma H_2$$

$$= m [C_1 \Delta T_1 + \text{LH} + C_2 \Delta T_2]$$

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

$$Q_a = 49.94 \text{ kW}$$

$$\text{Heat leak} = \frac{49.94 \times 10}{100} = 4.994 \text{ kJ/sec}$$

$$\text{Total Heat} = 49.94 + 4.994 \\ = 54.9 \text{ kW}$$

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

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

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

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

$$= 0.572 (214.5 - 178.74)$$

$$= 20.5 \text{ kW}$$

Ques: $\rightarrow$ A simple vapour compression system is designed to handle a load of 10 TR. The storage & surrounding temperature are 0°C and 30°C respectively. A minimum temp. difference of 6°C is required in condenser & evaporator for effective heat transfer. Calculate

(i) $\dot{m}$ of refrigerant

(ii) P_{in} in kW

(iii) COP

** (iv) Cylinder dimensions assuming stroke $l = 1.2$ Bore

The compressor speed is 300 rpm

$= 1.2 D$

$$\eta_{vol} = 90\%$$

t_c	P (bar)	v_g (m^3/kg)	h_f (kJ/kg)	h_g (kJ/kg)	s_f (kJ/kg)	s_g (kJ/kg)	C_p
* 36	8.6811	0.02034	234.94	367.2	1.1186	1.547	0.6
30	7.236	0.03124	223.6	361.2	1.0032	1.549	0.6
0	3.28	0.05286	201.32	353	0.9963	1.559	0.6
* -6	2.579	0.0677	194.45	350	0.9796	1.5618	0.6

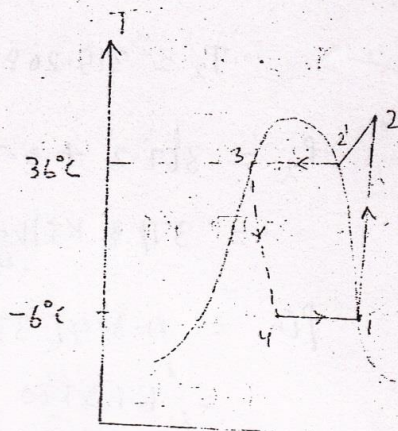
Solution $\Rightarrow$

Refrigeration capacity

$$\begin{aligned} RC &= 10 \text{ TR} \\ &= 10 \times 3.5 \text{ kJ/sec} \\ &= 35 \text{ kJ/sec} \end{aligned}$$

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

$$\begin{aligned} h_1 &= h_g \text{ at } -6^\circ\text{C} \\ &= 350 \text{ kJ/kg} \end{aligned}$$



$$h_3 = h_f \text{ at } 36^\circ\text{C}$$

$$= 234.94 \text{ kJ/kg}$$

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

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

$$\dot{m} = 0.304 \text{ kg/sec}$$

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

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

$$h_2 = h_{g2} + C_{p,vap}(T_2 - T_{sat})$$

$$= h_{g2} + C_{p,vap}(T_2 - T_{sat})$$

Now, $s_1 = s_2$

$$s_1 = 1.5618 \text{ kJ/kg K}$$

$$s_2 = s_{g2} + C_{p,vap} \ln\left(\frac{T_2}{T_{sat}}\right)$$

$$1.5618 = 1.547 + 0.744 \ln\left(\frac{T_2}{309}\right)$$

$$T_2 = 315.208 \text{ K}$$

$$h_2 = 367.2 + 0.744(315.208 - 309)$$

$$= 371.8 \text{ kJ/kg}$$

$$P_{in} = 0.304(371.8 - 350)$$

$$= 6.63 \text{ kW}$$

$$COP = \frac{h_1 - h_3}{h_2 - h_1} = 5.277$$

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

$$v_1 = 0.0677 \text{ m}^3/\text{kg}$$

$$L = 1.2 D$$

$$0.9 = \frac{0.304 \times 0.0677}{\frac{\pi}{4} (D^2) (1.2 D) \times 300 / 6000}$$

$$D = 169 \text{ mm}$$

$$L = 202.8 \text{ mm}$$

Effect of variation of properties on the performance of V.C cycle: $\rightarrow$

(a) Effect of decrease in evaporator pressure: $\rightarrow$

1-2-3-4 original cycle

1'-2'-3-4' new cycle

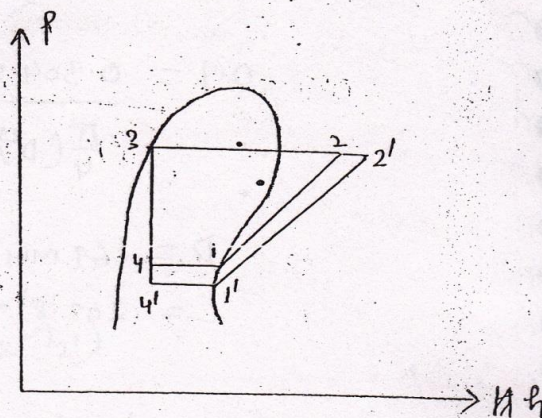
Now decrease in evaporator pressure. $\therefore$ EFFECTS: $\rightarrow$

(i) $RE = h_1 - h_4$
= decreases

(ii) $W_{in} = h_2 - h_1$
= increases

(iii) $COP \rightarrow$ decreases

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



(b) Effect of increase in condenser pressure: $\rightarrow$

1-2-3-4 original cycle

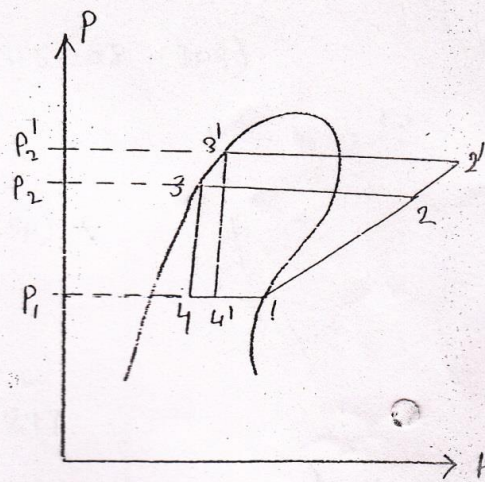
1-2'-3'-4' new cycle

(i) RE decreases

(ii) W_{in} increases

(iii) COP decreases

(iv) η_{vol} decreases



Open system work $\rightarrow$

$$h_1 + \frac{C_1^2}{2} + Z_1 g + Q = h_2 + \frac{C_2^2}{2} + Z_2 g + w$$

Assumption $\rightarrow$

- (1) steady flow
- (2) Neglect K.E. & P.E. changes
- (3) Reversible process

$$h_1 + Q = h_2 + w$$

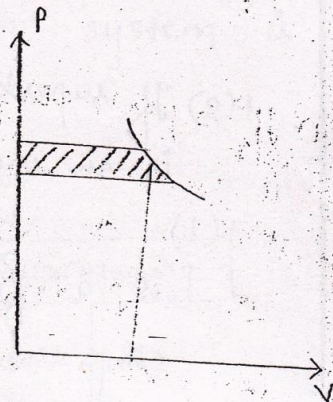
$$Q = h_2 - h_1 + w$$

$$Q = dh + w$$

$$T ds = dh + w \quad \text{--- (i)}$$

also $T ds = dh - v dp \quad \text{--- (ii)}$

$$w = -v dp$$

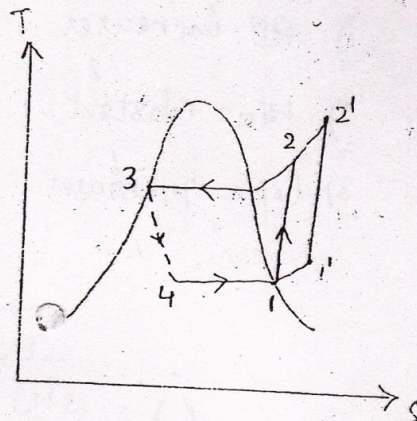
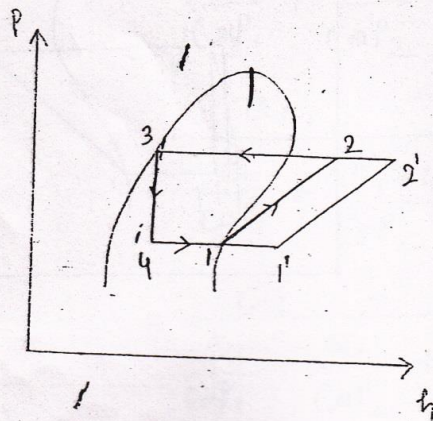


NOTE: Open system work is equal to area under the curve when projected on pressure axis.

NOTE: Open system - adiabatic work is equal to

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

c) Effect of superheating $\rightarrow$



Effects $\rightarrow$

$$\begin{aligned}
 W &= \frac{\gamma}{\gamma-1} (P_1 V_1 - P_2 V_2) \\
 &= \frac{\gamma}{\gamma-1} (MRT_1 - MRT_2) \\
 &= \frac{\gamma}{\gamma-1} \cdot MRT_1 \left(1 - \frac{T_2}{T_1}\right) \\
 W &= \frac{\gamma}{\gamma-1} MRT_1 \left(1 - \left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}}\right)
 \end{aligned}$$

1.(a) If superheating is done in evaporator, then there is an increase in refrigeration effect.

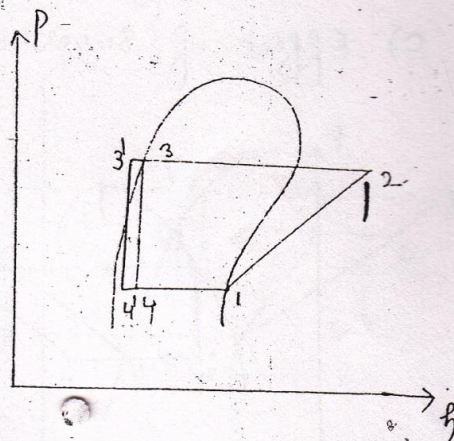
1.(b) We know that work input to the compressor is a function of temperature at the inlet to the compressor. With superheating the temperature increases and hence work input also increases.

1.(c) COP may increase or decrease depending upon the refrigerant. In case of R-12, superheating results in increase in COP (RET). In case of ammonia superheating results in decrease in COP (WinT) ↓

4. Subcooling Effect $\rightarrow$

Effect:-

- 1) RE increases
- 2) Win = constant
- 3) Cop = increases



CASCADE REFRIGERATION :->

To produce ultra low temperature, The evaporator temperature will be low and hence the corresponding pressure is also low. This increases the pressure ratio of v-c cycle and hence this results in decrease in volumetric efficiency. To avoid this cascading refrigeration system is used (Refrigeration in series).

COP of cascading Refrigeration

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

COP of each Refrigerator

$$COP_1 = \frac{Q_1}{W_1}$$

$$COP_2 = \frac{Q_2}{W_2}$$

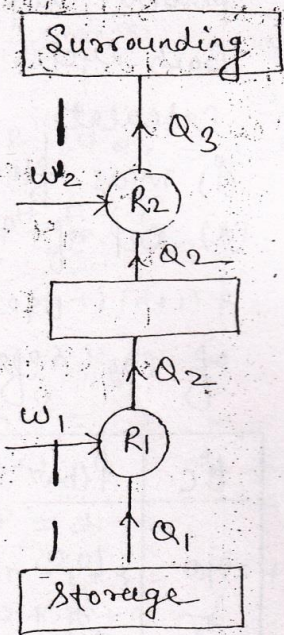
Now,

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

$$= \frac{Q_1}{\frac{Q_1}{COP_1} + \frac{Q_2}{COP_2}}$$

$$= \frac{Q_1}{\frac{Q_1}{COP_1} + \frac{Q_1 + W_1}{COP_2}}$$

$$= \frac{Q_1}{\frac{Q_1}{COP_1} + \frac{Q_1}{COP_2} + \frac{Q_1}{COP_1} \times \frac{1}{COP_2}}$$



$$COP = \frac{1}{\frac{1}{COP_1} + \frac{1}{COP_2} + \frac{1}{COP_1 COP_2}}$$

$$COP = \frac{COP_1 COP_2}{1 + COP_1 + COP_2}$$

Ques: → A cascade refrigeration system 100 TR. Use NH_3 and CO_2 . The evaporating and condensing temperature of CO_2 are $-40^\circ C$ and $5^\circ C$ respectively and evaporating temperature of NH_3 is $-7^\circ C$. Power supplied to NH_3 compressor is 96.5 kW. Both cycles are simple saturated cycle. Calculate,

(1) mass flow rate of CO_2

(2) COP of system

Use the following table as the properties of CO_2 refrigerant.

$$C_{p, \text{vap}} = 0.85 \text{ kJ/kgK}$$

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

Solution: →

$$RC = 100 \text{ TR}$$

$$RC = 100 \times 3.5$$

$$= 350 \text{ kJ/sec}$$

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

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

$$\dot{m}_{CO_2} = 1.577 \text{ kg/sec}$$

$$h_2 = h_2' + C_{p, \text{vap}} (T_2 - T_2')$$

$$S_1 = S_2 = 5.2262$$

$$S_2 = S_2' + C_{p, \text{vap}} \ln \left(\frac{T_2}{T_2'} \right)$$

$$5.2262 = 5.0637 + 0.85 \ln \frac{T_2}{T_2'}$$

$$T_2' = 361.2 \text{ K}$$

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

$$\dot{P}_{\text{CO}_2} = \dot{m}_{\text{CO}_2} (h_2 - h_1)$$

$$= 1.577 (h_2 - h_1)$$

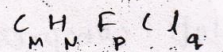
$$= 106.8 \text{ kW}$$

$$\text{COP} = \frac{R_C}{P_{\text{NH}_3} + P_{\text{CO}_2}}$$

$$= \frac{350}{96.5 + 106.8}$$

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

$$R-134a \rightarrow C_2H_2F_4$$



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

$$R = 134a$$

$$m-1 = 1$$

$$m = 2$$

$$n+1 = 3$$

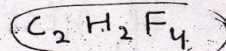
$$n = 2$$

$$p = 4$$

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

$$2+4+q = 6$$

$$q = 0$$



Recent Trends in Refrigerants : $\rightarrow$

The ozone which is situated in stratosphere absorbs harmful ultra-violet radiation and protects us from these radiations. The chlorine element which is present in refrigerant attacks ozone and thus depletes its thickness and hence chlorine element must be removed from refrigerant. So the substitutes are HC, FC, HFC.

Now-a-days environment friendly refrigerants are R-134a ($\text{CCF}_3\text{CH}_2\text{F}$) is used as refrigerant in domestic refrigerator.

Advantages of Dry Compression over Wet Compression : $\rightarrow$

- (1) Wet compression represents incomplete vapourisation of refrigerant and hence there is loss of RE.
- (2) The liquid refrigerant may wash away lubricating oil, thus increasing wear & tear.
- (3) The liquid refrigerant may damage compression valves.

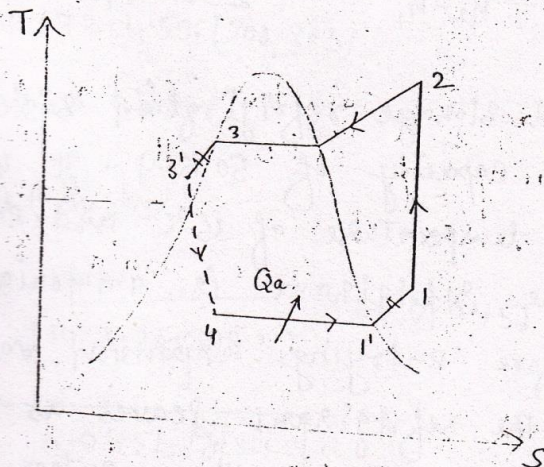
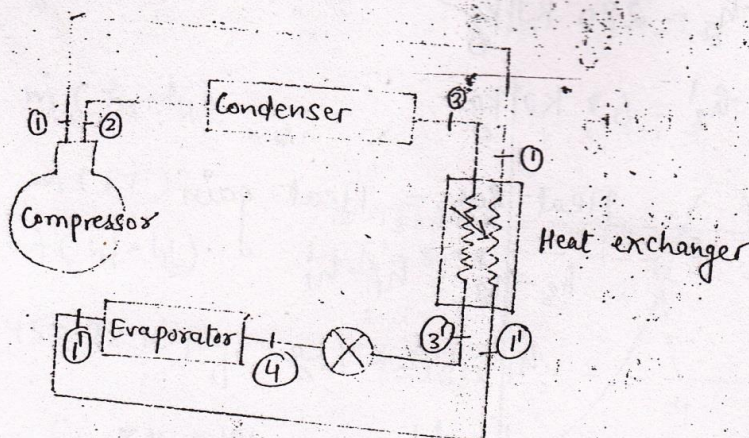
Slight superheat is always desirable.

Use of Heat exchanger in V-C cycle : $\rightarrow$

When we use the heat exchanger in a V-C cycle. Then an additional process like subcooling & superheating occurs, Resulting increase in RE.

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

$$h_3 - h_3' = h_1 - h_1' \Rightarrow C_{p,l} (T_3 - T_3') = C_{p,v} (T_1 - T_1')$$



Though heat loss = heat gain, it does not mean degree of superheat = degree of subcooling, because C_{liq} and C_{vap} are not same.

Ques $\Rightarrow$ A heat exchanger is used in a v-c cycle. The enthalpies, at Condenser outlet and evaporator outlet are 78 kJ/kg and 182 kJ/kg respectively. The enthalpy at the outlet of compressor is 230 kJ/kg , and enthalpy of subcooled liquid is 68 kJ/kg . Then find COP

Soln $\Rightarrow$

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

$$h_1 = 182 \text{ kJ/kg}$$

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

$$h_g' = 68 \text{ kJ/kg}$$

Heat loss = Heat gain

$$h_g - h_g' = h_1 - h_1'$$

$$h_1 = 192 \text{ kJ/kg}$$

$$\text{COP} = \frac{h_1' - h_g'}{h_2 - h_1} = \frac{182 - 68}{230 - 192} = 3$$

Ques → A food storage refrigeration system requires a refrigeration capacity of 50 kW. It works between a condenser temperature of 35°C and evaporator temp of -10°C. The refrigerant is ammonia. It is subcooled by 5°C before entering expansion valve in a heat exchanger. The refrigerant leaves as sat evaporator as saturated vapour and then enters heat exchanger assuming a single cylinder operating at 1000 rpm with stroke is equal to 1.2 bore. determine

- 1) Power required in kW
- 2) Single Cylinder dimensions.

Use the following Table:-

t°C	P(bar)	h _f	h _g	s _f	s _g	sp. vol (m ³ /kg)		sp. heat kJ/kgK	
						Liquid	vapour	Liquid	vapour
-10	2.1957	154.056	1450.22	0.82965	5.755	—	0.41747	—	2.492
35	13.522	366.072	1148.57	1.56605	5.20086	1.7023	0.095629	4.556	2.903

Solution $\rightarrow$

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

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

$$50 = \dot{m} (h_1' - h_2')$$

$$h_1' = 1450.22 \text{ kJ/kg}$$

$$h_3' = h_3 + C_{uq} (T_3 - T_3')$$

$$= 366.072 - 4.556 (308 - 303)$$

$$= 343.3 \text{ kJ/kg}$$

$$50 = \dot{m} (h_1' - h_3')$$

$$\dot{m} = \frac{50}{(1450.22 - 343.3)} = 0.0451 \text{ kg/sec}$$

$$\dot{m} = 0.0451 \text{ kg/sec}$$

Heat loss = Heat gain

$$h_3 - h_3' = h_1 - h_1'$$

$$h_1 = 1472.9 \text{ kJ/kg}$$

Now,

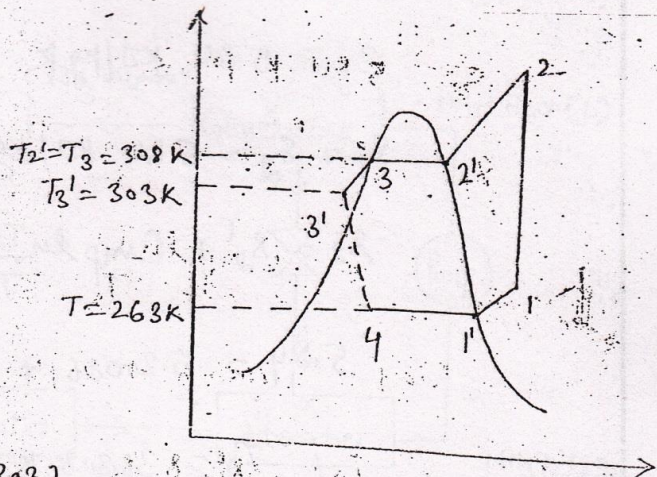
$$h_1 = h_g h_1' + C_{vap} (T_1 - T_1')$$

$$1472.9 = 1450.22 + 2.492 (T_1 - 263)$$

On solving

$$T_1 = 272.1 \text{ K}$$

$$S_1 = S_1' + C_{vap} \left(\ln \frac{T_1}{T_1'} \right)$$



$$S_1 = 5.755 + 2.492 \ln \left(\frac{272.1}{263} \right)$$

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

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

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

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

$$T_2 = 383.8 \text{ K}$$

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

$$h_2 = 1488.57 + 2.903(383.8 - 308)$$

$$= 1708.6 \text{ kJ/kg}$$

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

$$P = 0.0451(1708.6 - 1472.9)$$

$$= 10.6 \text{ kW}$$

2) Dimensions

$$v_1 = ?$$

$$PV = mRT$$

$$v \propto T$$

$$\frac{v_1}{v} = \frac{T_1}{T}$$

$$v_1 = 0.4819$$

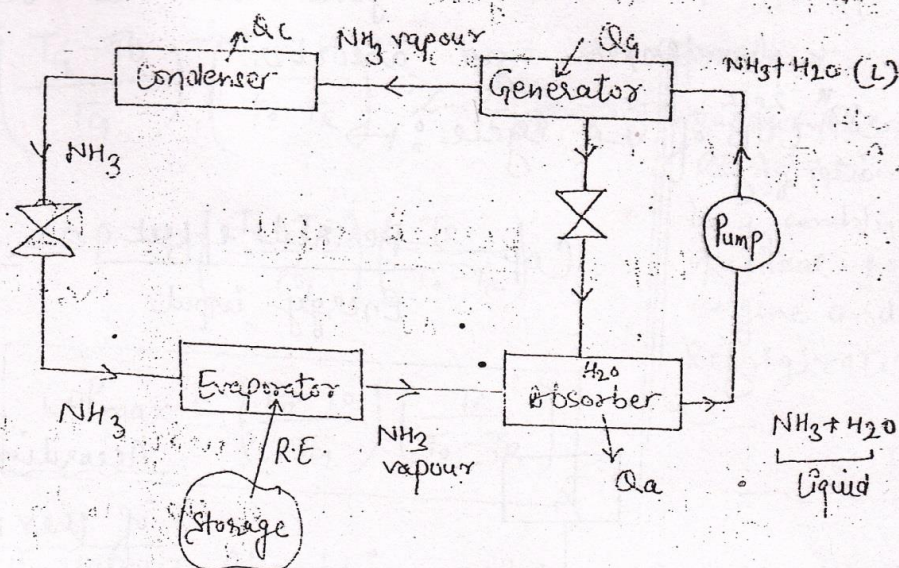
$$\eta_{vol} = \frac{\dot{m} v_1}{\frac{\pi}{4} D^2 L N \cdot K}$$

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

$$D = 0.107 = 10.7 \text{ cm}$$

$$L = 1.2(0.107) = 12.8 \text{ cm}$$

VAPOUR ABSORPTION REFRIGERATION CYCLE



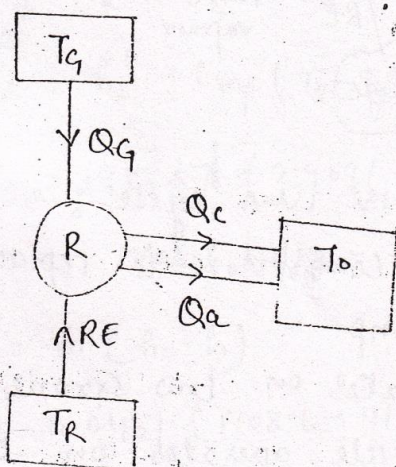
Important points w.r.t V-A cycle :-

- (1) In V-A cycle compressor is replaced with Generator, absorber and pump.
- (2) V-A cycle operates on low grade energy (heat) whereas V-C cycle operates on high grade energy (work).
- (3) This cycle is used where the cost of electricity is very high.
- (4) This cycle is used where large waste heat is available.
- (5) Solar refrigeration systems are based on V-A cycle.
- (6) Pump work in V-A cycle is negligible.
- (7) Most popular VA systems are $\text{NH}_3\text{-H}_2\text{O}$ system where NH_3 is refrigerant and H_2O is absorbent, and $\text{LiBr-H}_2\text{O}$ system, in which water is refrigerant & LiBr is absorbent.

(8) In v-c cycle heat is rejected only in Condenser whereas in v-A cycle heat is rejected in the condenser and absorber.

10* 2012
COP of v-A cycle: $\rightarrow$

$$\text{COP} = \frac{\text{Desired effect}}{\text{Energy input}} = \frac{RE}{Q_G + W_p}$$



According to first law
of thermodynamics

$$Q_G + RE = Q_C + Q_A \quad (1)$$

According to second
law of thermodynamics

$$(\Delta S)_{\text{sys}} + (\Delta S)_{\text{surroundings}} \geq 0$$

$$(\Delta S)_{\text{sys}} = 0$$

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

In v-A cycle work
input is very less
so we can neglect

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

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

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

$$Q_G \left[\frac{1}{T_0} - \frac{1}{T_G} \right] \geq RE \left[\frac{1}{T_R} - \frac{1}{T_0} \right]$$

$$Q_g \left(\frac{T_g - T_o}{T_g} \right) \geq RE \left(\frac{T_o - T_R}{T_R} \right)$$

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

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

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

NOTE: →

Vapour Absorption Refrigeration system is a combination of ideal heat engine and ideal Refrigerator.

Ques: → A vapour Absorption Refrigeration system work with generator, ambient and evaporator Temp as 360 K, 310 K and 260 K respectively. Find the max. COP of the system.

If the evaporator temp falls to 250 K, what should be the generator temp in order to operate the system with same COP.

Ans: →

$$COP = \frac{(360 - 310)}{360} \times \left(\frac{260}{310 - 260} \right) = 0.722$$

$$0.722 = \frac{(T_g - 310)}{T_g} \left(\frac{250}{310 - 250} \right)$$

$$T_g = 315 K$$

Q11C

Ques:- A vapour-Absorption Refrigeration system works with three thermal Reservoir. The Refrigeration effect of 100 W is required at 250 K. The Heat source is available at 400 K. Heat rejection occurs at 300 K. Then find the minimum value of heat required in watt.

Ans:-

$$RE = 100 \text{ W}$$

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

$$(COP)_{\max} = \left(\frac{T_g}{T_g - T_o} \right) \left(\frac{T_R}{T_o - T_R} \right) = 1.25$$

$$(Q_g)_{\min} = \frac{100}{1.25} \text{ W}$$

$$(Q_g)_{\min} = 80 \text{ kW}$$

Ques:- Assertion:- R-11 & R-12 have Greater impact on Ozone depletion than R-22.

Reason:- R-22 has larger number of chlorine

Ans:-

Assertion is right.

Reason is wrong.

Ques:-

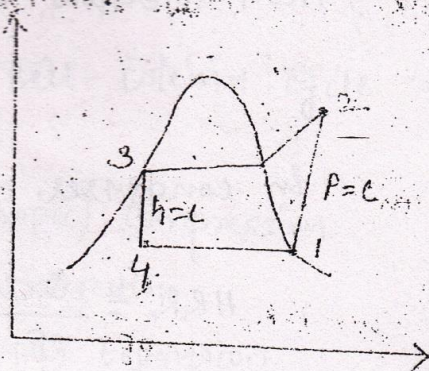
A V-C Refrigeration system is represented as shown in figure, with state 1 as the exit of evaporator. then the coordinate used in the system

a) P-h

b) T-s

c) P-s

d) T-h

Ans: $\rightarrow$ T-hQues: $\rightarrow$ Match list I with list II w.r.t. v-c cycle

LIST I

- Compression
- Heat Rejection
- Expansion
- Heat absorption

LIST II

- Isobaric
- Isothermal
- Isentropic
- Isenthalpic

Ans: a-3, b-1, c-4, d-1

Ques: $\rightarrow$ The ratio of clearance volume to swept volume of a refrigerant reciprocating compressor is 0.05. The specific volume of the refrigerant at inlet and exit of compressor is $0.04 \text{ m}^3/\text{kg}$ and $0.02 \text{ m}^3/\text{kg}$ respectively. Then the volumetric efficiency of compressor is

Ans: $\rightarrow$

$$\begin{aligned}
 \eta_{\text{vol}} &= 1 + C - C \left(\frac{P_2}{P_1} \right)^{1/n} \\
 &= 1 + C - C \left(\frac{V_1}{V_2} \right) \quad (P_1 V_1^n = P_2 V_2^n) \\
 &= 1 + 0.05 - 0.05 \left(\frac{0.04}{0.02} \right) \\
 &= 0.95
 \end{aligned}$$

HEAT REJECTION RATIO OF A CONDENSER: $\rightarrow \frac{(HRR)}{(HRE)}$

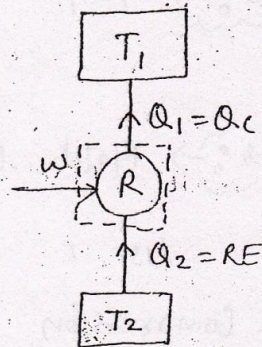
It is the ratio of heat rejected in condenser to the refrigeration effect.

$$HRR = \frac{Q_c}{RE}$$

$$= \frac{W + RE}{RE}$$

$$= \frac{W}{RE} + 1$$

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



If COP is less, HRR will be large
So higher HRR is dangerous.

GAS REFRIGERATION CYCLE

G-R Cycle $\rightarrow$

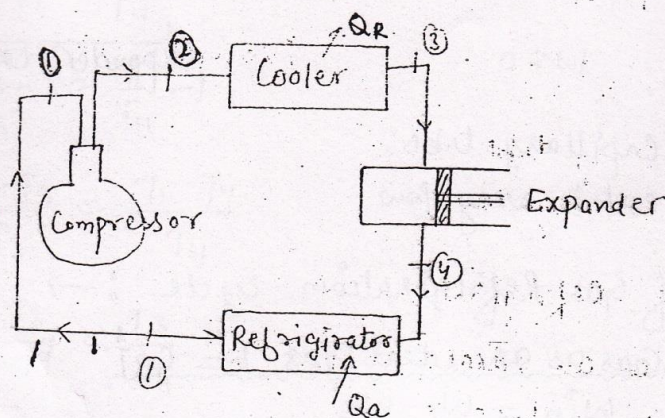
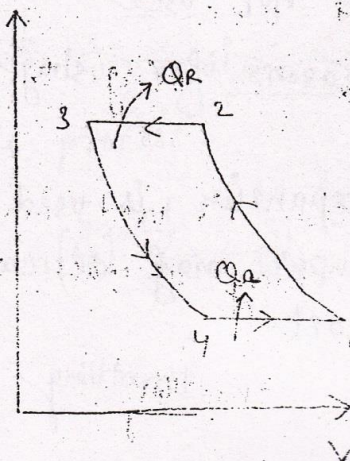
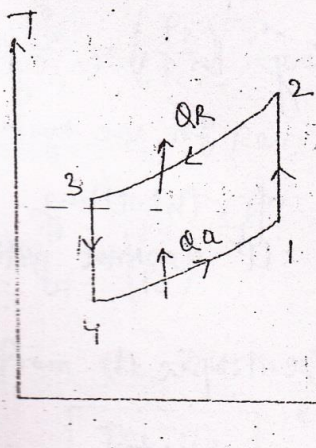
This cycle works on Bell-Coleman cycle or Reversed Brayton cycle.

1-2 $\rightarrow$ Rev. adiabatic (Isentropic) Compression

2-3 $\rightarrow$ Constant pressure heat rejection

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

4-1 $\rightarrow$ Constant pressure heat absorption



Reason for using isentropic expansion instead of throttling in Gas Refrigeration cycle $\rightarrow$

In Gas Refrigeration cycle, the working fluid is gas (ideal gas). During isentropic expansion, temperature of refrigerant drops

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

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

process 1-2 Rev. Adiabatic process $\left[\frac{P_2}{P_1} = \frac{P_3}{P_4} = r_p \right]$

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

process 3-4 Rev. Adiabatic process

$$\frac{T_3}{T_4} = \left(\frac{P_3}{P_4} \right)^{\frac{\gamma-1}{\gamma}} = \left(\frac{P_2}{P_1} \right)^{\frac{\gamma-1}{\gamma}} = (r_p)^{\frac{\gamma-1}{\gamma}} \quad \text{--- (ii)}$$

from (i) & (ii)

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

$$\frac{T_2}{T_3} - 1 = \frac{T_1}{T_4} - 1$$

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

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

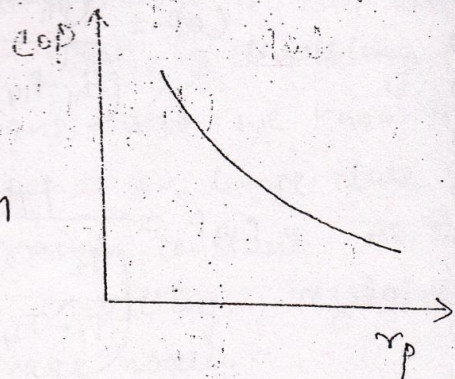
$$\text{COP} = \frac{1}{\frac{T_3}{T_4} - 1}$$

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

This equation is valid only when compression & expansion are isentropic

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

The COP of Gas Refrigeration cycle depends on the pressure ratio ' r_p '.



Gas Refrigeration cycle is used in air-craft refrigeration system. air is used as a refrigerant because of its low weight per tonne of refrigeration.

Isentropic efficiency of a compressor $\rightarrow (\eta_c)$

Rev. Adiabatic process

$$h_1 + \frac{C_1^2}{2} + z_1 g + Q = h_{2s} + \frac{C_{2s}^2}{2} + z_{2s} g + w$$

$$h_1 + Q = h_{2s} + w$$

$$h_1 = h_{2s} + w$$

$$-w = h_{2s} - h_1$$

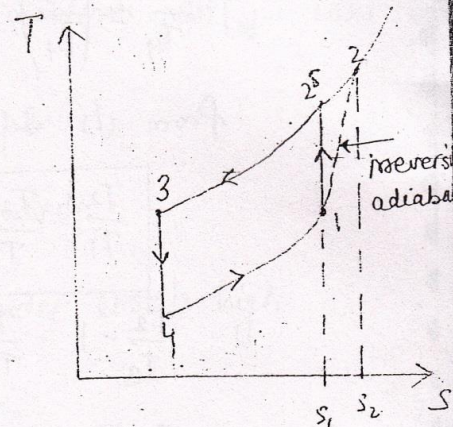
Irrev. Adiabatic process

$$-w = h_2 - h_1$$

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

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

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



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

* If $W_{act} = h_2 - h_1 = C_p(T_2 - T_1)$ is to be found

and C_p, T_1, r_p, η_c is given

** then first find T_{2s} from $\frac{T_{2s}}{T_1} = (r_p)^{\frac{\gamma-1}{\gamma}}$

*** then find T_2 from

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

Isentropic efficiency of Expander or turbine $\rightarrow$

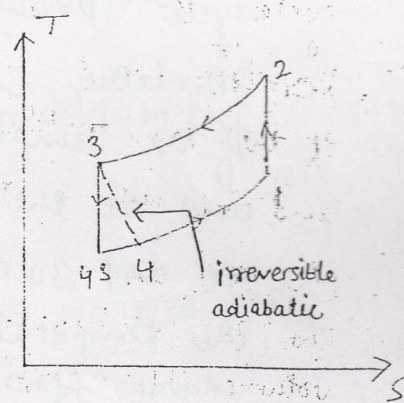
$$W_{isentropic} = h_3 - h_{4s}$$

$$W_{act} = h_3 - h_4$$

$$\eta_t = \frac{W_{act}}{W_{isentropic}}$$

$$\eta_T = \frac{h_3 - h_4}{h_3 - h_{4s}}$$

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



Ques: → In a gas refrigeration cycle temperature at inlet and exit of isentropic expander are 300K and 180K respectively. The compression is assumed to be isentropic. find the COP.

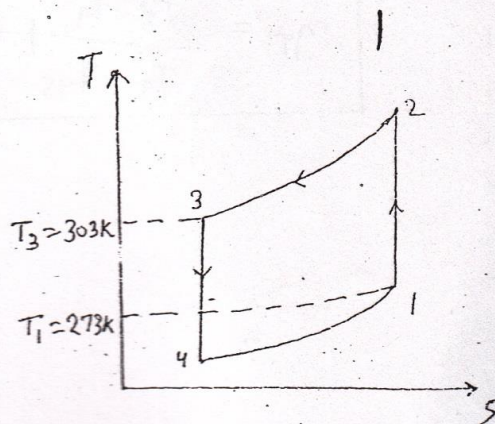
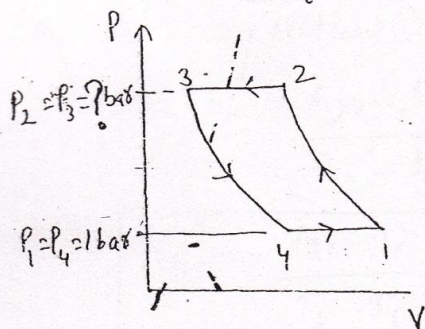
Soln: →

$$\begin{aligned} \text{COP} &= \frac{1}{(r_p)^{\frac{\gamma}{\gamma-1}} - 1} \\ &= \frac{1}{\left[\frac{T_3}{T_4} - 1\right]} \\ &= \frac{1}{\left[\frac{300}{180} - 1\right]} = 1.57 \end{aligned}$$

$$\frac{T_3}{T_4} = \left(\frac{P_3}{P_4}\right)^{\frac{\gamma}{\gamma-1}} = \left(\frac{P_2}{P_1}\right)^{\frac{\gamma}{\gamma-1}}$$

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Ques: → Air is used as refrigerant in a Reversed Brayton cycle. Draw T-S & P-V Diagram for the cycle and derive the COP in terms of pressure ratio 'rp'. If the temperature at the end of heat absorption and at the end of heat rejection are 0°C and 30°C respectively. $r_p = 4$. Determine the temperature at all points and also find volume flow rate at the inlet to the compressor and at the exit of expander. The cooling capacity of the plant is 1TR. and pressure at the inlet to the compressor 1 bar.



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

$$\frac{T_2}{273} = \left(4 \right)^{\frac{1.4-1}{1.4}} \Rightarrow T_2 = 405.6 \text{ K}$$

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

$$\frac{405.6}{273} = \frac{303}{T_4} \Rightarrow T_4 = 204 \text{ K}$$

$$P_1 V_1 = \dot{m} R T_1$$

$$R C = \dot{m} (R E)$$

$$3.5 = \dot{m} (h_1 - h_4)$$

$$\dot{m} = \frac{3.5}{(C_p)(273 - 204)} = 0.05 \text{ Kg/sec}$$

$$V_1 = \frac{0.05 \times 287 \times 273}{100} = 0.039 \text{ m}^3/\text{sec}$$

Now,

$$P_4 V_4 = \dot{m} R T_4$$

$$V_4 = \frac{0.039 \times 287 \times 204}{100} = 0.029 \text{ m}^3/\text{sec}$$

Ques: → In a Bell Coleman Refrigeration plant. The pressure and temperature at the inlet to the compressor is 1 bar and 10°C. The air is compressed to 5 bar. The air is then cooled to 25°C in a cooler. If the compression and expansion follows $PV^{1.35} = C$ and $PV^{1.3} = C$ respectively. Then find the COP of system.

So $\rightarrow$ we know that in polytropic process, there is both heat & work transfer and hence in a steady flow equation Q is not zero for polytropic process, therefore

$$h_1 + Q = h_2 + W, \quad W \neq h_2 - h_1$$

But we know that the compressor is an open system and hence for polytropic process

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

$$T_1 = 283 \text{ K}$$

$$\frac{T_2}{T_1} = \left(\frac{P_2}{P_1} \right)^{\frac{\eta_1 - 1}{\eta_1}}$$

$$\frac{T_2}{T_1} = \left(\frac{5}{1} \right)^{\frac{\eta_1 - 1}{\eta_1}} = \left(5 \right)^{\frac{0.35}{1.35}}$$

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

$$\frac{T_3}{T_4} = \left(\frac{P_3}{P_4} \right)^{\frac{\eta_2 - 1}{\eta_2}} = \left(\frac{P_2}{P_1} \right)^{\frac{\eta_2 - 1}{\eta_2}}$$

$$\frac{T_3}{T_4} = \left(5 \right)^{\frac{0.3}{1.3}}$$

$$T_4 = 205.5 \text{ K}$$

$$W_c = \frac{\eta_1}{\eta_1 - 1} (m R T_1 - m R T_2)$$

$$= \left(\frac{1.35}{1.35 - 1} \right) [1 \times 287 \times 283 - 1 \times 287 \times 429.5]$$

$$W_c = -162 \text{ kJ/kg}$$

$$W_e = \frac{\eta_2}{\eta_2 - 1} (P_3 V_3 - P_4 V_4)$$

$$= \left(\frac{1.3}{1.3 - 1} \right) (m R T_3 - m R T_4)$$

$$W_e = 115 \text{ kJ/kg}$$

$$W_{in} = W_c - W_e = 162 - 115 = 47 \text{ kJ/kg}$$

$$RE = h_1 - h_4$$

$$= c_p [T_1 - T_4] = 1.005 [283 - 205.5]$$

$$= 77.8 \text{ kJ/kg}$$

$$COP = \frac{RE}{W_{in}} = \frac{77.8}{47} = 1.65$$

Ques 3 → In an air-craft refrigeration system air enters the compressor at 0.1 MPa and 4°C and it is compressed to 0.3 MPa with an isentropic efficiency of 72% . It is then cooled to 55°C at constant pressure. The air is then expanded in a expander to 0.1 MPa with an isentropic efficiency of 0.78 . The lower temp air absorbs a cooling load of 3 TR . Assuming air to behave as an ideal gas. Find

- (i) COP
- (ii) Power input in kW
- (iii) mass flow rate

Solution → $\eta_c = 0.72 = \frac{T_{2s} - T_1}{T_2 - T_1}$

$$\left(\frac{T_{2s}}{T_1} \right) = (\gamma_p)^{\frac{\gamma-1}{\gamma}} = 1.368$$

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

$$0.72 = \frac{379.14 - 277}{T_2 - 277} \Rightarrow T_2 = 418.86 \text{ K}$$

$$\frac{T_3}{T_{4s}} = \left(\frac{P_3}{P_4}\right)^{\frac{\gamma-1}{\gamma}} = \left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}} = (r_p)^{\frac{\gamma-1}{\gamma}}$$

$$\frac{T_3}{T_{4s}} = (3)^{\frac{1.4-1}{1.4}}$$

$$T_{4s} = 239.76 \text{ K}$$

$$\eta_e = 0.78 = \frac{T_3 - T_4}{T_3 - T_{4s}} = \frac{328 - T_4}{328 - 239.76}$$

$$T_4 = 259.17 \text{ K}$$

$$RE = 3 \text{ TR}$$

$$= 3 \times 3.5 = 10.5 \text{ kJ/sec}$$

$$\begin{aligned} \text{COP} &= \frac{RE}{W_{in}} = \frac{RE}{W_c - W_e} = \frac{h_1 - h_4}{(h_2 - h_1) - (h_3 - h_4)} \\ &= \frac{T_1 - T_4}{(T_2 - T_1) - (T_3 - T_4)} \end{aligned}$$

$$\text{COP} = \frac{277 - 259.7}{(418.86 - 277) - (328 - 259.7)} = 0.247$$

$$\dot{P}_{in} = \frac{RC}{\text{COP}} = \frac{10.5}{0.247} = 42.6 \text{ kW}$$

$$RC = \dot{m} (h_1 - h_4) = \dot{m} c_p (T_1 - T_4)$$

$$\dot{m}_1 = 0.58 \text{ kg/s}$$

The equation $\text{COP} = \frac{1}{(r_p)^{\frac{\gamma-1}{\gamma}} - 1}$ is valid only when the compression & expansion are reversible adiabatic.

DESIRABLE PROPERTIES OF REFRIGERANT. (SELECTION OF A REFRIGERANT)

properties are divided into thermodynamic, chemical & physical properties.

THERMODYNAMIC PROPERTIES →

a) Critical temperature : →

The critical temperature of a refrigerant must be as high as possible. Certain refrigerant like CO_2 & ethylene are not suitable in tropical climates like India because a critical temperature is below the ambient temperatures in India.

Refrigerant	H_2O	NH_3	R-12	R-22	R-134	CO_2	E
Critical temp	371.5	132.4	111.5	96.5	101.2	31	10

b) Specific Heat : →

We know that $C = T \left(\frac{ds}{dt} \right)$, As liquid refrigerant undergoes throttling process, for minimum irreversibilities ds must be small. Therefore specific heat of liquid must be small.

In case of vapour refrigerant, the degree of superheat must be small for lower work input and hence

specific heat of vapour refrigerant must be large.

Enthalpy of vapourisation $\rightarrow$

Refrigerant with large enthalpy of vapourisation is desirable because for a given capacity of the plant larger enthalpy of vapourisation will result in smaller mass flow rate. $RC = m \dot{h}$

Refrigerant

h_{vap}

Conductivity $\rightarrow$

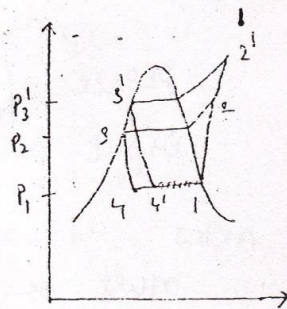
The conductivity of a refrigerant must be as high as possible because conductivity represents the ability to transfer heat.

CONDENSER AND EVAPORATOR PRESSURE $\rightarrow$

The evaporator pressure must be as close to atmospheric pressure as possible and condenser pressure must be moderately higher. Higher condenser pressure results in lower refrigeration effect and larger work input. Low evaporator pressure results in large pressure differential and hence air leakage into the system can occur.

Pressure Ratio $\rightarrow$

Smaller pressure ratios are desirable. Higher pressure ratio results in lower volumetric efficiency and larger work input to the compressor. General compressor ratios in V-C cycle are '4' to '6'.



Freezing point $\rightarrow$

The freezing point of Refrigerant must be as low as possible.

Refrigerant	R22	R12	NH ₃	R134a
Freezing point	-160.5	-157.4	-77.3	-101.2

Specific volume at the inlet to the compressor $\rightarrow$

Volume of the refrigerant represents the size of the compressor. If the refrigerant volume is large then the size of compressor will also be large. Therefore, refrigerant with smaller specific volume are desired.

COP $\rightarrow$

As COP represents the running cost of the system so larger COP is desirable.

Though the hvp of NH₃ refrigerant is very high. It doesnot help anyway in the improvement of COP. Because work input to the NH₃ compressor is high and hence its COP is almost equal to the other common Refrigerants. [γ is very high for NH₃]

Compressor Discharge Temperature (t_2) $\rightarrow$

The Compressor discharge Temperature, must not be very high. The higher discharge temp. results in overheating of compressor.

In case of NH₃, compressor discharge temp is around 120°C (i.e. high) and hence NH₃ compressor are generally water cooled.

Boiling Point : →

Boiling point of the refrigerant must be low. otherwise more heat will be required to convert it into vapour.

CHEMICAL PROPERTIES : →

(1) Toxicity : →

Refrigerant must be non-toxic eg. NH_3 is toxic so it is not used in domestic refrigerator.

(2) Flammability : →

It should be non-flammable. In flammable Ammonia is both toxic and flammable and hence though it is cheap it is not used as refrigerant in domestic refrigerator.

(3) Action with oil : →

In compressor some oil is carried by high temperature refrigerant to the condenser and finally to the evaporator. In evaporator refrigerant vapourises and oil separates from refrigerant, this accumulation of oil in evaporator would result in reduced heat transfer coefficient and oil chocking in evaporator.

Refrigerant that are non-miscible with oil like NH_3 & CO_2 do not present any problems because oil separator is used at the exit of compressor and the separated oil is brought back to the compressor.

Refrigerant that are completely miscible with oil R-11 & R-22 also do not present any

problem because oil which reaches the evaporator is returned to the compressor.

Refrigerant that are partially miscible with oil like R-22 creates problems and hence synthetic oil is used instead of mineral oil.

Action with material of construction : →

Ammonia attacks copper and hence where ammonia is used as Refrigerant wrought iron or steel is used as the material of construction. Similarly, Freons attacks aluminium & therefore for freons generally copper is used as the material of constructions.

PHYSICAL PROPERTIES : →

a) Viscosity : →

The viscosity of a Refrigerant must be very low so that can easily flow.

b) Leak Detection : →

Refrigerant must not leak out from the system. If it all leaks, its detection must be simple. Freons leaks are detected by the halide torch method. In which the colour of the light, changes from Blue to Bluish Green.

Ammonia leaks are detected by sulphur stick method. In presence of NH_3 , white fumes of ammonium sulphide are formed.

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Handwritten notes, mostly illegible due to fading and bleed-through. Some words like "equipment" and "laboratory" are faintly visible.

AIR CONDITIONING

Air conditioning is the simultaneous control of temp. humidity, air velocity and purity of air.

Psychrometry $\rightarrow$

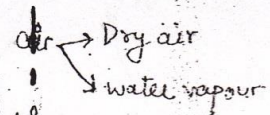
It is a branch of science which deals with the study of properties of moist air.

psychrometry terms $\rightarrow$

Specific Humidity or Humidity Ratio (ω) $\rightarrow$

It is the ratio of mass of vapour to the mass of dry air.

$$\omega = \frac{m_v}{m_a} \rightarrow \text{fixed part, visualization}$$



$$P_t = P_a + P_v$$

$$P_{atm} = P_a + P_v$$

We can assume it as an ideal gas.

Dry air

Vapour

$$P_a V_a = m_a R_a T_a$$

$$P_v V_v = m_v R_v T_v$$

$$P_a V = m_a R_a T = (i)$$

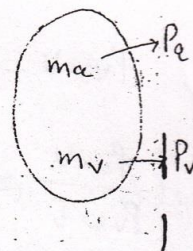
$$P_v V = m_v R_v T = (ii)$$

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

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

$$\Rightarrow \omega = \frac{P_v}{P_a} \times \frac{\bar{R}/29}{\bar{R}/18} = \frac{P_v}{P_a} \times 0.622$$

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



$$R = \frac{\bar{R}}{M}$$

NOTE: $\rightarrow$ w is generally expressed in kg vap / kg of da
 we know that atmospheric pressure P_t is almost constant and hence w is a function of P_v .

$$w = f(P_v) = \text{function of (partial pressure of vapour)}$$

Relative Humidity (ϕ) $\rightarrow$

It is defined as the ratio of mass of vapour to the mass of vapour under saturation condition in the same volume of same temperature

$$\phi = \left(\frac{m_v}{m_{vs}} \right)_{v, T}$$

$$P_v M_v = m_v R_v T_v$$

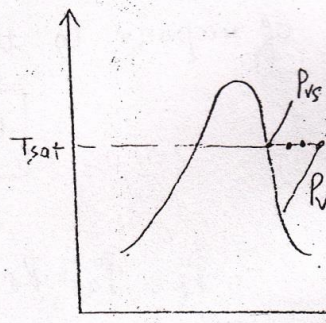
$$P_v \cdot V = m_v R_v T \quad \text{--- (I)}$$

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

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

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

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



for same T_{sat}
 P_{vs} remains same
 for various point
 on vertical Temp
 line

For saturated air relative humidity is 1 or 100%
 w , gives actual mass of vapour,
 ϕ gives moisture absorbing capacity of air

Dry bulb Temperature $\rightarrow$

It is the normal temperature of air measured by an ordinary thermometer.

Wet Bulb Temperature $\rightarrow$

It is temperature recorded by thermometer whose bulb is covered by a wet cloth.

For unsaturated air wet bulb temperature is less than Dry Bulb temperature

$$WBT < DBT$$

(i) For saturated air $DBT = WBT$

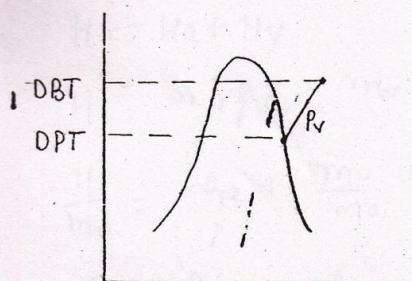
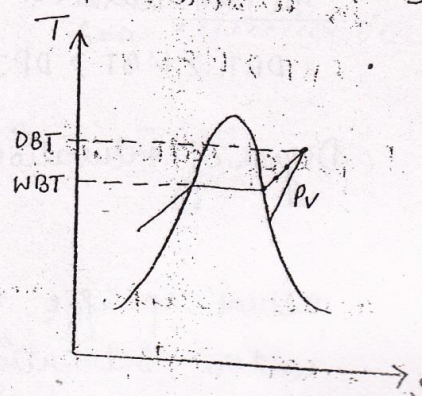
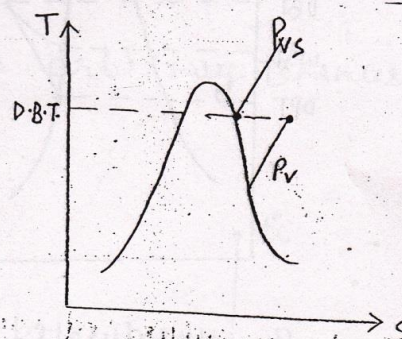
(ii) Wet Bulb Depression,

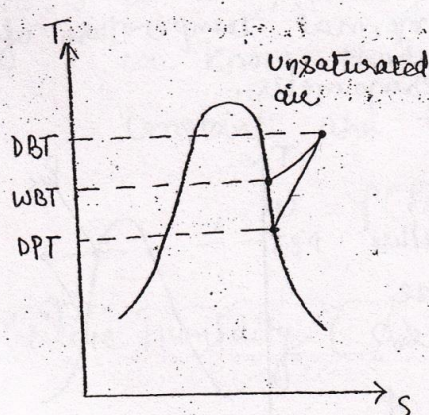
$$WBD = DBT - WBT$$

*** For saturated air $WBD = 0$

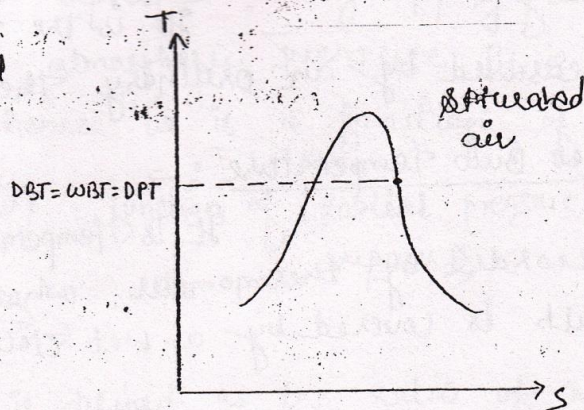
DEW POINT TEMPERATURE $\rightarrow$ (DPT)

It is the temperature at which water vapour in air starts condensing.





In unsaturated air
 $DBT > WBT > DPT$



In saturated air
 $DBT = WBT = DPT$

Degree of saturation $\Rightarrow (\mu)$ (% humidity)

It is defined as the ratio of actual specific humidity to specific humidity under saturation conditions.

$$\mu = \frac{0.622 \frac{P_v}{P_t - P_v}}{0.622 \frac{P_{vs}}{P_t - P_{vs}}}$$

$$= \frac{P_v}{(P_t - P_v)} \times \frac{P_t - P_{vs}}{P_{vs}}$$

$$\mu = \phi \frac{(P_t - P_{vs})}{(P_t - P_v)}$$

Enthalpy of Moist Air $\Rightarrow$

$$H = H_a + H_v$$

Enthalpy of ~~dry~~ moist air = enthalpy of dry air + enthalpy of vapour

Enthalpy of Dry air (h_a) : →

$$h_a = \frac{H_a}{m_a} \Rightarrow H_a = h_a m_a$$

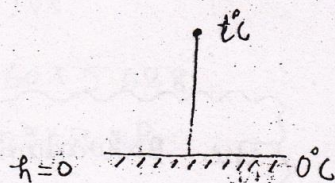
enthalpy of dry air at 0°C is arbitrary taken as zero.

$$h_a = C_{p_a}(t - 0)$$

$$h_a = C_{p_a} \cdot t \quad \text{KJ/kg da}$$

where

$t \rightarrow ^\circ\text{C}$ Centigrade



Enthalpy of water vapour : →

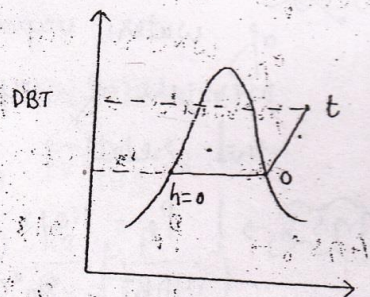
$$h_v = \frac{H_v}{m_v}$$

$$H_v = m_v h_v$$

Enthalpy of water vapour at 0° is taken as zero, $h=0$

$$h_v = LH + C_{p_v}(t - 0)$$

$$h_v = (2500 + 1.88t) \quad \text{KJ/kg v}$$



$$H = H_a + H_v$$

$$H = m_a h_a + m_v h_v$$

$$\frac{H}{m_a} = h_a + \frac{m_v}{m_a} h_v$$

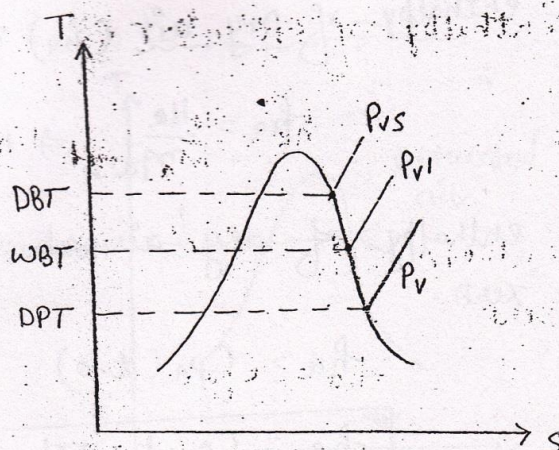
$$h = h_a + w h_v$$

$$h = C_{p_a} t + w(2500 + 1.88t) \quad \text{KJ/kg da}$$

DBT $\rightarrow P_{vs}$

WBT $\rightarrow P_{vl}$

DPT $\rightarrow P_v$



The saturation pressure corresponding to Dry bulb Temperature is P_{vs} .

The saturation pressure corresponding to WBT is P_{vl} .

The saturation pressure corresponding to DPT is P_v .

Ques: moist air at 1.013 bar at 30°C contains 10 gram of water vapour per kg of dry air. The saturation pressure of vapour at 30°C is 3.167 kPa. Then find the relative humidity of air.

Ans: $P_t = 1.013 = 101.3 \text{ kPa}$

DBT = 30°C

$$w = \frac{10 \text{ gram kgv}}{1 \text{ kg of da}} = 10^{-2} \frac{\text{kgv}}{\text{kg da}}$$

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

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

$$P_v = 1.61 \text{ kPa}$$

$$\phi = \frac{P_v}{P_{vs}} = \frac{1.61}{3.167} \times 100 = 50.6\%$$

Ques: 2 Atmospheric air at 1.013 bar at 35°C as a relative humidity of 60%. The saturation pressure of vapour at 35°C is 5.628 kPa. Then find the specific humidity?

Ans:

$$\phi = 0.60$$

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

$$\Rightarrow P_v = \phi \cdot P_{vs}$$

$$= 0.60 \times 5.628$$

$$= 3.3768 \text{ kPa}$$

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

$$= 0.622 \frac{3.3768}{1.013 - 3.3768} = 0.0214$$

$$\omega = 0.0214 \frac{\text{kgv}}{\text{kgda}}$$

Ques: 3 Match the following

List I

- Specific Humidity
- Relative Humidity
- Specific volume of DA.

List II

- P_v / P_{vs}
- $\frac{R_a}{R_v} \left(\frac{P}{P_t - P_v} \right)$
- $\frac{P_v}{P_{vs}} \left(\frac{P_t - P_{vs}}{P_t - P_v} \right)$
- $\frac{R_a T}{P_t - P_v}$

Ans: A-2, B-1, C-4

$$\omega = \frac{R_a}{R_v} = \frac{R/29}{R/18} = 0.622$$

$$v/m \text{ } ^\circ = \frac{RT}{P} = \frac{R_a T}{P_a} = \frac{R_a T}{P_t - P_v}$$

Ques: Atmospheric air at 100 kPa and 30°C, as a relative humidity of 70%. The saturation pressure of vapour at 30°C is 4.25 kPa. Then find the partial pressure of dry air.

Ans: →

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

$$0.70 = \frac{P_v}{4.25}$$

$$P_v = 2.975 \text{ kPa}$$

$$P_a = P_t - P_v$$

$$= 100 - 2.975$$

$$= 97.025 \text{ kPa}$$

$$P_a = 0.97 \text{ bar}$$

Ques: → 5 If the volume of moist air with 50% Relative humidity is isothermally reduced to half its volume, then the relative humidity of moist air becomes — a) 25% b) 50% c) 75% d) 100%

Ans: →

$$\phi_1 = 0.5 = \frac{P_{v1}}{P_{vs}}$$

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

$$P_{v1} \cdot V_{v1} = P_{v2} \cdot V_{v2}$$

$$P_{v1} \cdot V_{v1} = P_{v2} \cdot \frac{V_{v1}}{2}$$

$$P_{v2} = 2 \cdot P_{v1}$$

$$\phi_2 = \frac{2 \cdot P_{v1}}{P_{vs}}$$

$$= 2 \times 0.5$$

$$= 1.00$$

$$= 100\%$$

Ques: 6 The degree of saturation of air at 30°C is 24% and total atmospheric pressure is 100 kPa. The saturation pressure of vapour at 30° is 4 kPa. Then find relative humidity & specific humidity.

Ans: $\rightarrow$

$$P_t = 100 \text{ kPa}$$

$$P_{vs} = 4 \text{ kPa}$$

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

$$0.24 = \frac{P_v}{4} \left[\frac{100 - 4}{100 - P_v} \right]$$

$$P_v = 0.99 \text{ kPa}$$

$$\phi = \frac{P_v}{P_{vs}} = \frac{0.99}{4}$$

$$\phi = 0.247$$

$$\boxed{\phi = 24.7\%}$$

$$w = 0.622 \left(\frac{P_v}{P_t - P_v} \right)$$

$$= 0.622 \left(\frac{0.99}{100 - 0.99} \right)$$

$$w = 6.21 \times 10^{-3} \frac{\text{kg v}}{\text{kg da}}$$

Ques: 7 Dew point temperature of air at ~~103 kPa~~ 101.3 kPa at 18°C . The psych bulb temp is 30°C . The saturation pressure of vapour at 18°C and 30°C are 2.062 kPa & 4.241 kPa respectively. Then find the enthalpy of moist air.

Ans: $\rightarrow$

$$h_{\text{moist air}} = C_p t + w \cdot (2500 + 1.88t)$$

$$= 1.005 \times 30 + \left(0.622 \frac{P_v}{P_t - P_v} \right) (2500 + 1.88 \times 30)$$

$$= 30.15 + 32.977$$

$$h_{ma} = 63.127 \text{ kJ/kg da}$$

Ques: 3 moist air with a DBT of 40°C has a relative humidity of 50%. Atmospheric pressure is 1.04 bar the saturation pressure of vapour at 40°C is 7.38 kPa and the saturation pressure of vapour at 15°C is 4.758 bar. Find

i) specific humidity

ii) moist air as stated above is compressed to 5.05 bar and its corresponding DBT is 15°C . Then find the relative humidity of compressed air.

Ans: $\rightarrow$ Case 1 $\rightarrow$

$$w = 0.622 \left(\frac{P_v}{P_t - P_v} \right)$$

$$\phi = \frac{P_v}{P_{vs}} \Rightarrow P_v = 0.5 \times 7.38 = 3.69 \text{ kPa}$$

$$w = 0.622 \left(\frac{3.69}{1.04 - 3.69} \right) = 0.0235$$

Case 2 :-

$$w_1 = w_2$$

$$w = 0.622 \left(\frac{P_{v2}}{P_{t2} - P_{v2}} \right) \text{ at } 15^{\circ}\text{C}$$

$$P_{v2} = 0.184 \text{ bar}$$

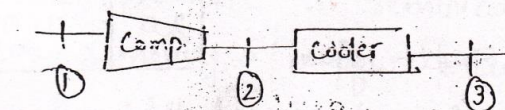
$$\phi_c = \frac{P_{v2}}{P_{vs2}}$$

$$\phi_c = \frac{0.184}{4.758} = 3.5\%$$

*** Ques 9

moist air at a pressure of 100 kPa is compressed to 500 kPa and it then cooled to 35°C in a cooler at constant pressure. Then air at the exit to the cooler is unsaturated and becomes just saturated at the exit of cooler. The saturation pressure of water at 35°C is 5.628 kPa. Then find the partial pressure of vapour entering the compressor.

Ans →



$$P_1 = 100 \text{ kPa} \quad P_2 = 500 \text{ kPa}$$

$$t = 35^\circ\text{C}$$

$$P_t = 500$$

$$\phi = 100\% \text{ (just saturated)}$$

$$P_{vs} = 5.628 \text{ kPa}$$

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

$$P_v = P_{vs} = 5.628 \text{ kPa}$$

$$\omega = 0.622 \left(\frac{P_v}{P_t - P_v} \right)$$

$$= 7.08 \times 10^{-3}$$

$$\omega_3 = \omega_1$$

$$7.08 \times 10^{-3} = 0.622 \times \left(\frac{P_v}{100 - P_v} \right)$$

$$P_v = 1.125 \text{ kPa}$$

Ques 10

Atmospheric air at DBT of 40°C at WBT of 20°C undergoes a process in which its WBT remains constant. The wet bulb depression at the exit is 25% of that at inlet. Then find the DBT at the exit.

$$WBD_2 = 0.25 WBD_1$$

(25°C) A

$$(DBT_2 - WBT_2) = 0.25 (DBT_1 - WBT_1)$$

$$DBT_2 - 20 = 0.25 \times 20 \Rightarrow DBT_2 = 25^\circ\text{C} \quad \underline{A}$$

Ques: 11 The Dry bulb & Wet bulb temperature of moist air are 30° & 20°C respectively. The atmospheric pressure is 0.9865 bar . Find

- i) Relative humidity
- ii) Specific humidity
- iii) Degree of saturation
- iv) Vapour density
- v) enthalpy of moist air

$t^\circ\text{C}$	P_{vs}
30	0.04242
20	0.0237

Use the following table given and also use following empirical relationship

$$P_v = P_v' - \frac{1.8 P_t (\text{DBT} - \text{WBT})}{2700}$$

P_v' is the saturation vapour pressure corresponding to WBT.

Ans: $\rightarrow P_v = P_v' - \frac{1.8 P_t (\text{DBT} - \text{WBT})}{2700}$

$$P_v = 0.0237 - \frac{1.8 \times 0.9865 (30 - 20)}{2700}$$

$$= 0.0171\text{ bar}$$

$$W = 0.622 \left(\frac{P_v}{P_t - P_v} \right) = 0.622 \left(\frac{0.0171}{0.9865 - 0.0171} \right)$$

$$= 0.0109\text{ kg/kg da}$$

$$\phi = \frac{P_v}{P_{vs}} = 0.402$$

$$\mu = 0.402 \left(\frac{0.9865 - 0.04242}{0.9865 - 0.0171} \right) = 0.39 = 39\%$$

$$h = c_{p,a} t + W(2500 + 1.88 t) = 58.27\text{ kJ/kg da}$$

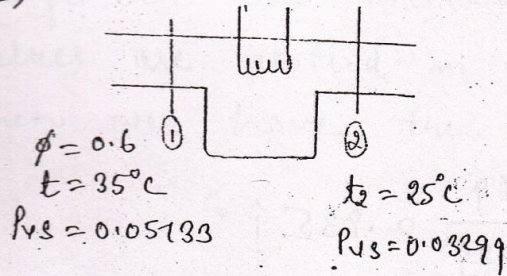
$$P_v = mRT$$

$$\frac{m_v}{v_v} = \frac{P_v}{R_v T_v}$$

$$P_v = \frac{P_v}{R_v T_v} = \frac{0.0171 \times 10^3}{\frac{8.134}{18} \times (30 + 273)} = 0.0122 \text{ kg/m}^3$$

Ques: ¹² 5 gram of water vapour per kg of dry air at atmospheric pressure is removed and temperature of air after removing vapour becomes 25°C. Find the relative humidity and enthalpy of moist air after the removal of moisture. Assume atmospheric air at 35°C and 60% RH.

Ans: →



$t^\circ\text{C}$	$P_{vs} \text{ (bar)}$
35°C	0.05133
25°C	0.03229

At point 1: →

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

$$P_v = 0.6 \times P_{vs}$$

$$= 0.6 \times 0.05133$$

$$= 0.030798 \text{ bar}$$

$$\omega = 0.622 \left(\frac{P_v}{P_t - P_v} \right)$$

$$= 0.622 \left(\frac{0.030798}{1.013 - 0.030798} \right) = 0.022 \text{ kg/kg air}$$

Now, specific humidity after passing through cooling effect, where 5 gram of water vapour per kg of dry air is removed.

$$\begin{aligned}w_2 &= 0.022 - 5 \times 10^{-3} \\&= 0.017 \text{ kg/kg da}\end{aligned}$$

Now,

$$h_2 = C_p t + w(2500 + 1.88t)$$

$$= (1.005 \times 25) + 0.017(2500 + 1.8 \times 25)$$

$$= 68.4 \text{ kJ/kg da}$$

$$w_2 = 0.622 \left(\frac{P_v}{P_t - P_v} \right)_2 \text{ at point 2}$$

$$P_{v2} = 0.0269 \text{ bar}$$

$$\begin{aligned}\phi_2 &= \frac{P_{v2}}{P_{vs2}} = \frac{0.0269}{0.03229} = 0.835 \\&= 83.5 \%\end{aligned}$$

DEVELOPMENT OF PSYCHROMETRIC CHART

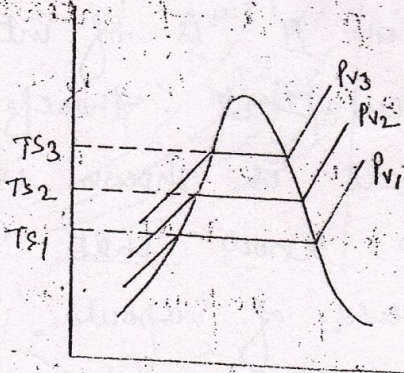


Figure 1

From the figure 1, it can be observed that the increase in P_v , the corresponding saturation temperature also increases. If these saturation values are plotted on the P_v vs T diagram, then we have the figure 2, below

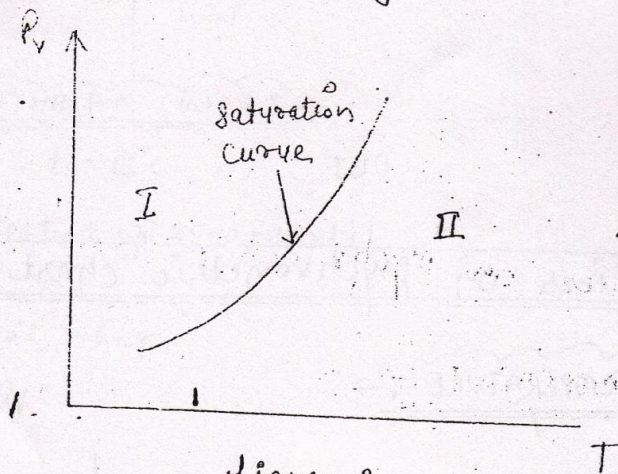
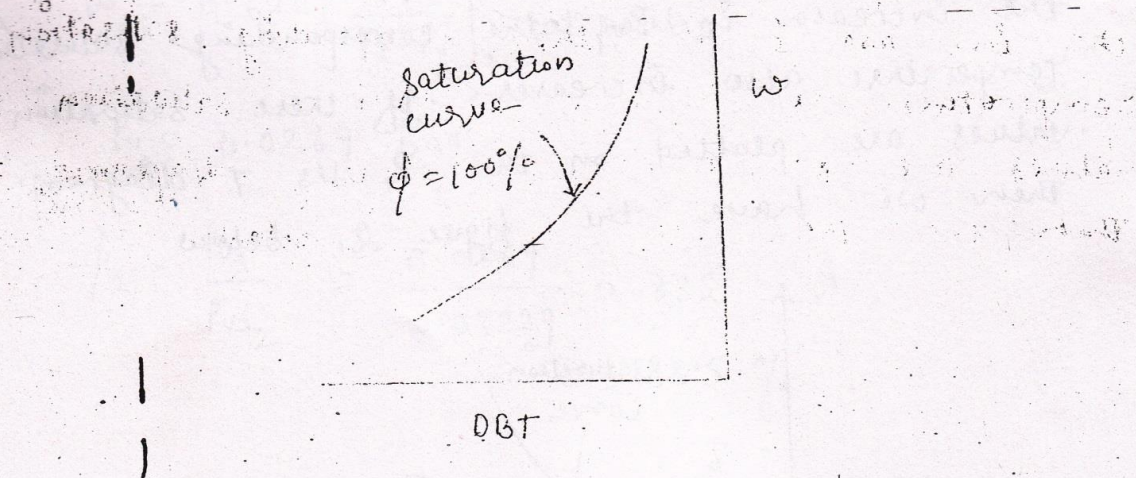


Figure 2

We know that in region 'II', the actual temperature is greater than saturation temperature corresponding to that pressure. therefore region

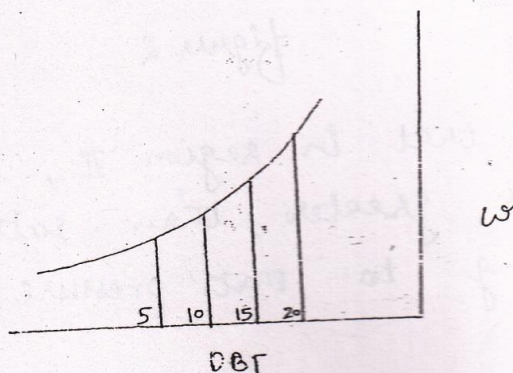
II is a superheated region and also know that water vapour in air exists in superheated region and hence region 'II' is of interest in air conditioning calculations. Therefore the original psychrometric chart is shown in figure,

We also know that 'w' is a function of partial pressure of vapour in new psychrometric chart 'P_v' is replaced with 'w'. Therefore new psychrometric chart is shown

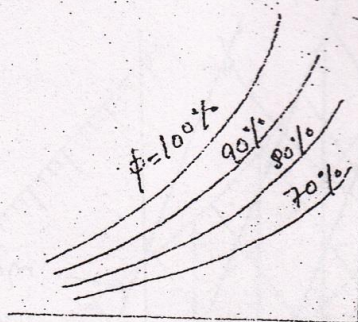


Various lines on psychrometric chart :->

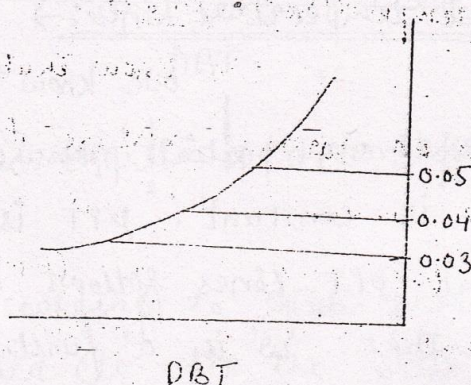
DRY BULB TEMPERATURE :->



CONSTANT RELATIVE HUMIDITY LINES: →



CONSTANT SPECIFIC HUMIDITY LINES: →



CONSTANT ENTHALPY LINES: →

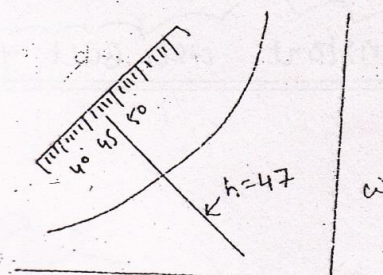
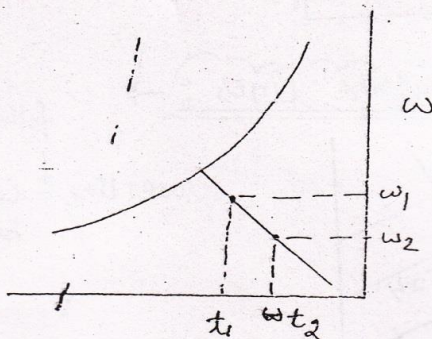
Let $h = 50$, $t_1 = 15^\circ\text{C}$

$$h = C_p a t_1 + w_1 [2500 + 1.88 t_1]$$

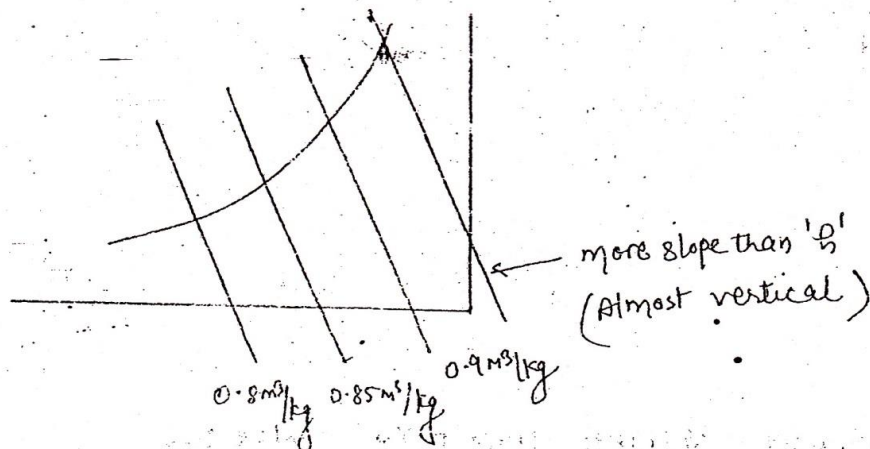
$w_1 = ?$

$t_2 = 20^\circ\text{C}$, $h = 50$

$w_2 = ?$

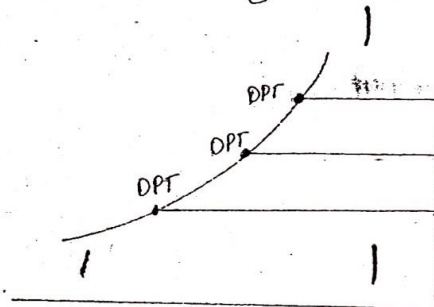


CONSTANT SPECIFIC VOLUME LINES: →

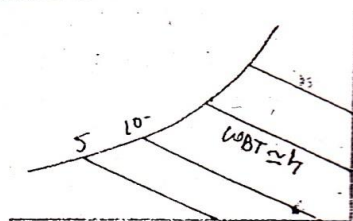


Constant Dew point temperature lines: →

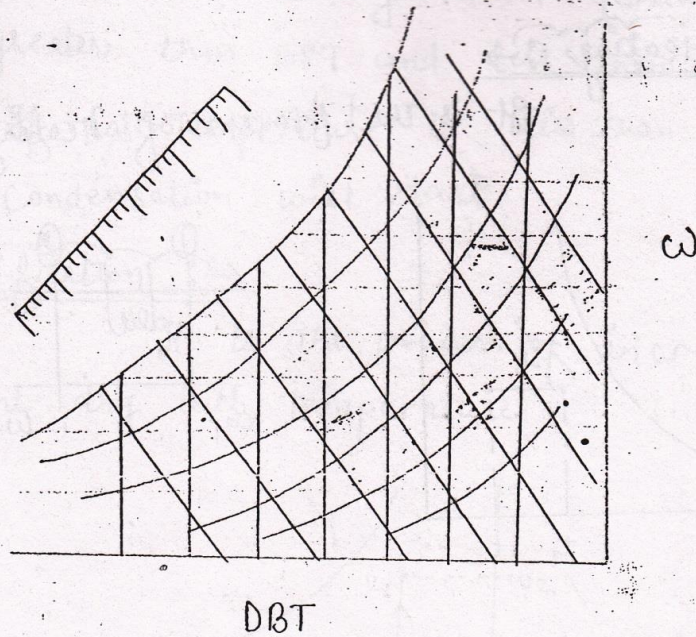
We know that dew point temperature depends on partial pressure of vapour ' p_v ' as long as ' p_v ' is constant, DPT is also constant. Therefore constant DPT lines follow constant p_v lines. But we know that ' w ' is a function of ' p_v '. Therefore when p_v is constant ' w ' is also constant and hence constant dpt lines follow constant ' w ' lines.



Constant wet Bulb temperature lines: →



follows
WBT Line $\downarrow$ = Constant Enthalpy Line



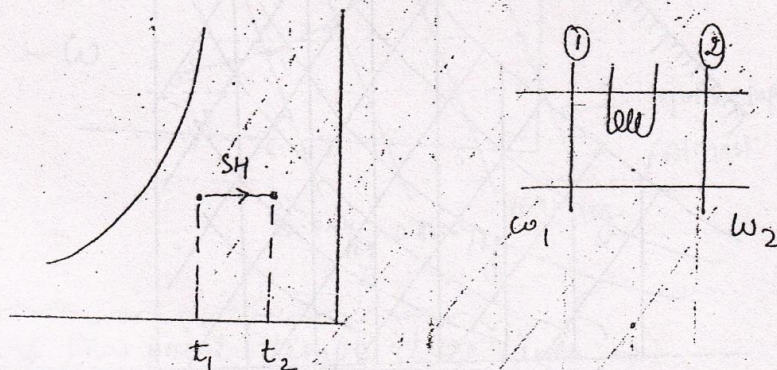
Psychrometric chart is drawn for a fixed atmospheric pressure.

NOTE According to Gibbs's phase Rule 3 parameters are required to fix the state of moist air. But on psychrometric chart only two parameters are required because the chart is drawn for a fixed pressure, i.e. the third parameter is fixed.

BASIC PSYCHROMETRIC PROCESSES

A) Sensible Heating $\rightarrow$

It is the process of Heating air at const w .



SFEE

$$h_1 + \frac{C^2}{2} + Zg + Q = h_2 + \frac{C^2}{2} + Zg + w$$

$$h_1 + Q = h_2 + w, \quad w = 0$$

$$Q = h_2 - h_1 = \text{change in enthalpy}$$

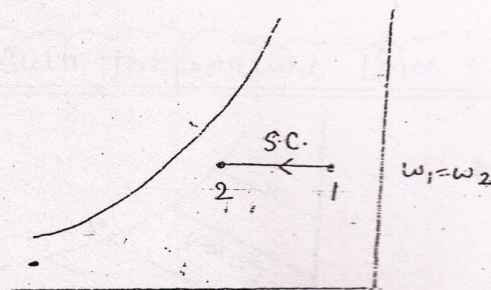
Adiabatic process follows constant enthalpy lines because $Q = 0$ for adiabatic process

$$0 = h_2 - h_1$$

$$h_1 = h_2 = \text{Constant}$$

B) SENSIBLE COOLING $\rightarrow$

process of cooling air at constant w .

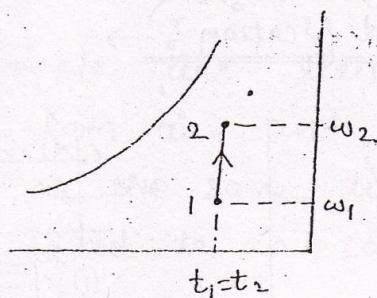


For sensible cooling the temperature of coil must be greater than DPT and less than DBT.

If coil temperature is less than DPT, then the condensation will start.

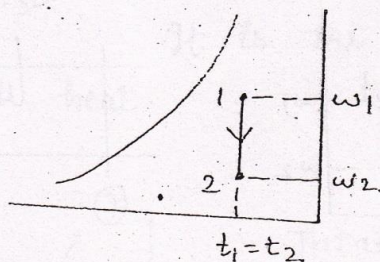
3) Humidification :-

It is the process of increasing w at constant dry bulb temperature.



4) Dehumidification :-

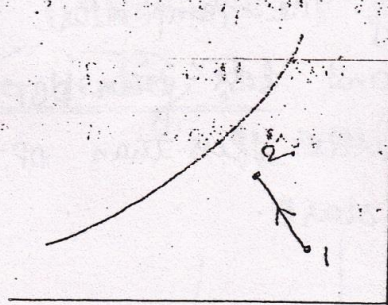
It is the process of decreasing w at constant DBT.



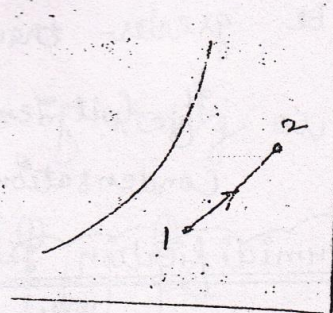
pure humidification and dehumidification processes are not possible because they are associated with temperature change.

5) Heating & Humidification :-

Example :- steam spray in air

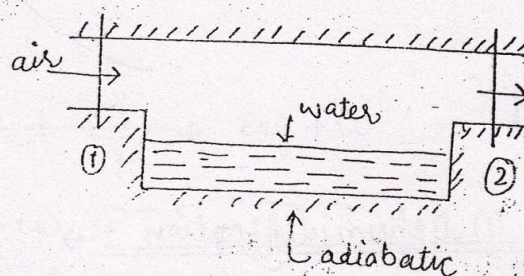
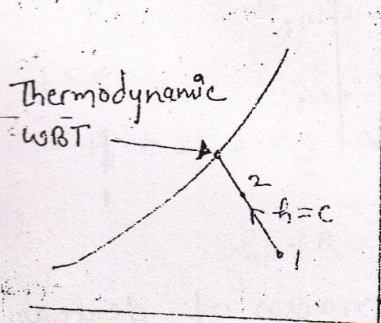


Cooling & Humidification

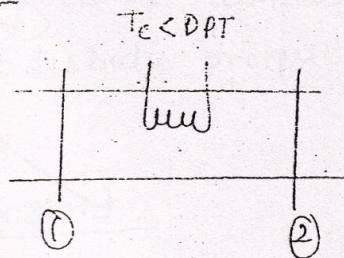
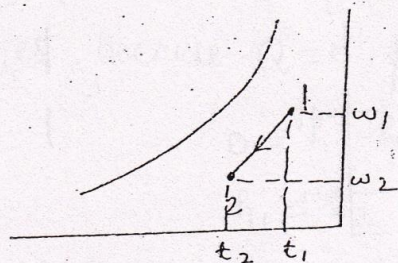


Heating & Humidification

Cooling & Humidification :-



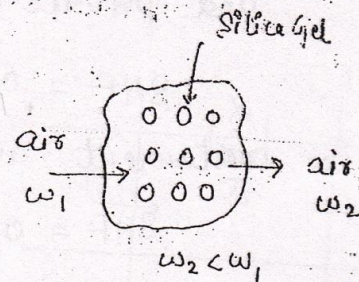
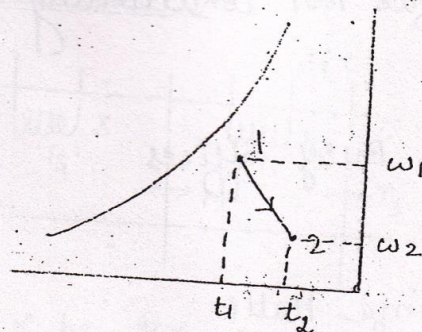
Cooling & Dehumidification :-



This process is possible when the coil temperature is less than dew point temperature of air. This process is generally followed in summer air conditioning.

6) Heating & Dehumidification :->

504



obj

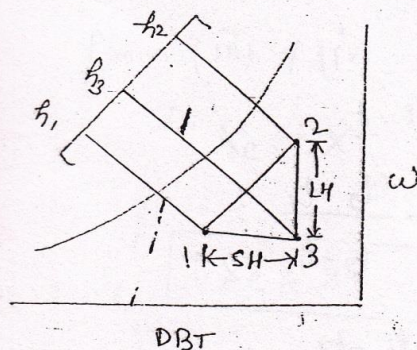
Special Case :-> Adiabatic chemical Dehumidification

Certain chemicals like silica Gel and alumina absorb moisture from air and hence its w value decreases, but at the same time latent heat of condensation is added to air and hence its dry bulb temperature increases.

Objective DBT $\uparrow \rightarrow w \downarrow$

Sensible Heat factor :-> (SHF)

It is the ratio of Sensible Heat to the total heat.



$$\text{Total Heat} = h_2 - h_1$$

$$SH = h_3 - h_1$$

$$LH = h_2 - h_3$$

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

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

a) For Residence SHF FOR VARIOUS APPLICATIONS
and private office Air conditioning

$$SHF = 0.9$$

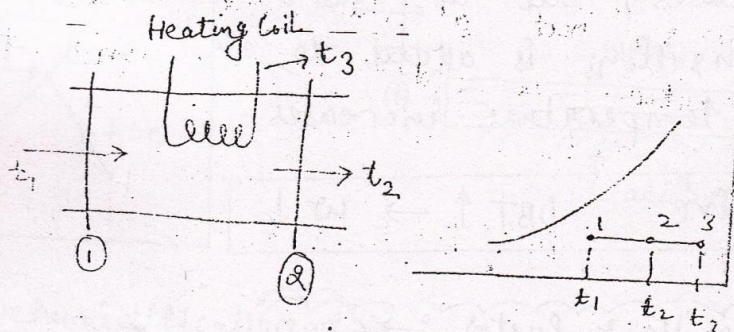
b) For restaurant and Busy offices

$$SHF = 0.8$$

c) Auditorium & Cinema Halls

$$SHF = 0.7$$

By Pass-Factor : $\rightarrow$ Uncontacted factor (BPF) : $\rightarrow$



$$BPF = \frac{\text{loss}}{\text{overall change}}$$

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

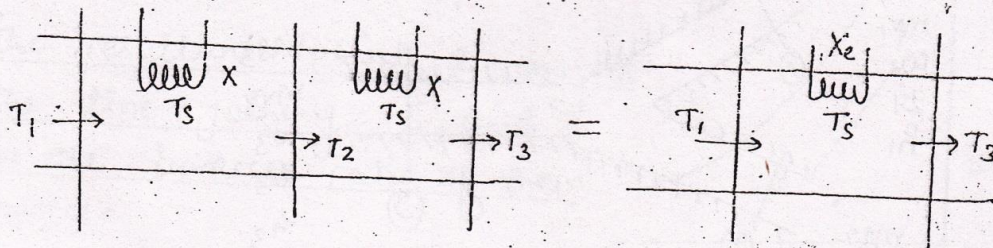
$$\eta = \frac{\text{actual gain}}{\text{ideal gain}} = \frac{t_2 - t_1}{t_3 - t_1}$$

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

$$\eta + BPF = 1$$

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

Effective By-Pass Factor with more than one coils
(Similar coils) $\therefore \rightarrow$



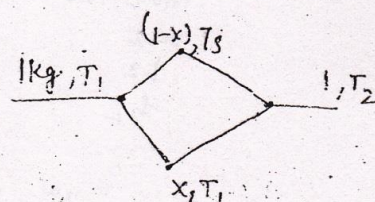
Let X be the bypass factor

Then,

$$(1-X)C_p T_s + X C_p T_1 = 1 X C_p T_2$$

$$(1-X) T_s + X T_1 = T_2 \quad \text{--- (i)}$$

$$(1-X) T_s + X T_2 = T_3 \quad \text{--- (ii)}$$



For figure 2

$$(1-X_e) T_s + X T_1 = T_3 \quad \text{--- (iii)}$$

$$(1-X) T_s + X [(1-X) T_s + X T_1] = T_3$$

$$(1-X) T_s [1 + X] + X^2 T_1 = T_3$$

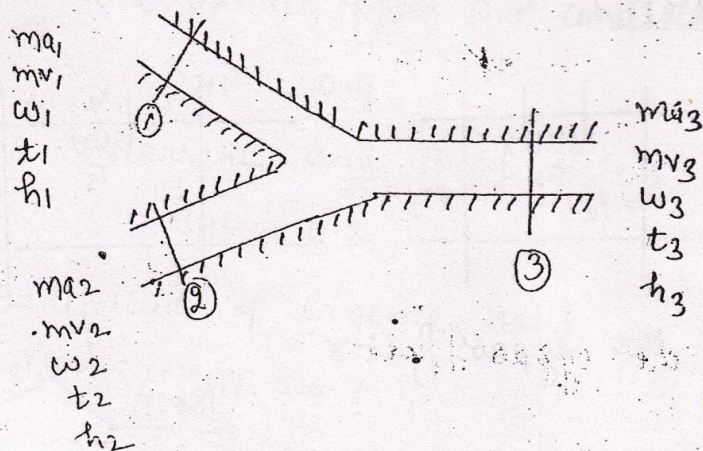
$$[1 - X^2] T_s + X^2 T_1 = T_3 \quad \text{--- (iv)}$$

from (iii) & (iv)

$$X_e = X^2 \quad \text{for 2 coils}$$

$$X_e = X^n \rightarrow \text{for } n \text{ coils}$$

Mixing of two streams →



I) Conservation of mass

$$a) m_1 + m_2 = m_3 \quad \text{--- (i)}$$

$$b) mv_1 + mv_2 = mv_3$$

$$m_1 w_1 + m_2 w_2 = m_3 w_3 \quad \text{--- (ii)}$$

$$w = \frac{mv}{m}$$

II) Conservation of Energy

$$m_1 h_1 + m_2 h_2 = m_3 h_3 \quad \text{--- (iii)}$$

$$m_1 h_1 + m_2 h_2 = (m_1 + m_2) h_3$$

$$= m_1 h_3 + m_2 h_3$$

$$m_1 h_1 + m_2 h_2 = m_1 h_3 + m_2 h_3$$

$$m_1 [h_1 - h_3] = m_2 [h_3 - h_2]$$

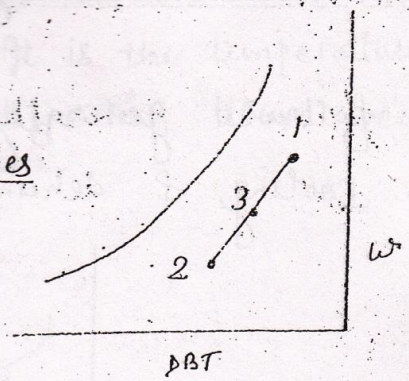
$$\frac{m_1}{m_2} = \frac{h_3 - h_2}{h_1 - h_3} = \frac{c_p(t_3 - t_2)}{c_p(t_1 - t_3)}$$

Similarly,

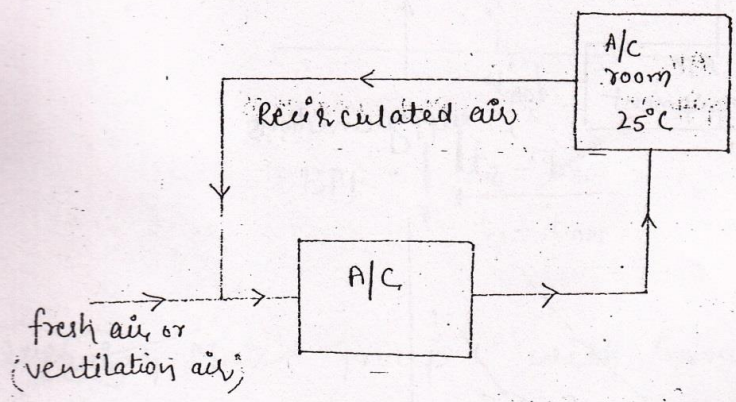
$$\frac{m_1}{m_2} = \frac{w_3 - w_2}{w_1 - w_3}$$

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

Note The resultant point '3' divides the line joining '1' and '2' in the inverse ratio of their masses.



Ventilation Air $\therefore \rightarrow$

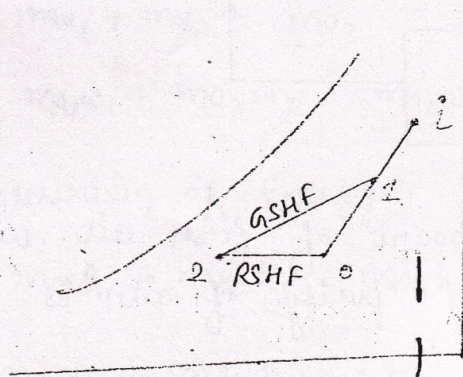
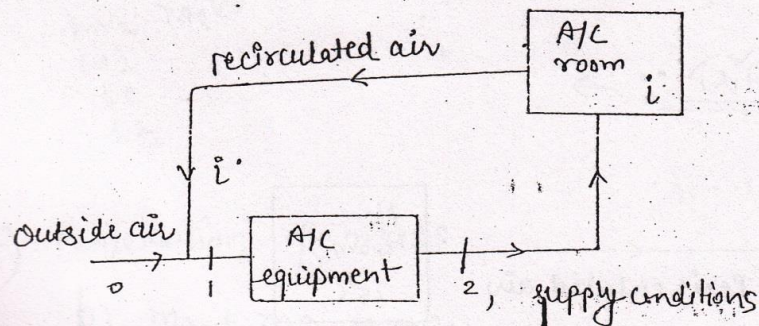


The amount of fresh air which is required to maintain purity of air is known as ventilation air.

Note In operation theatre air conditioning 100% fresh air or ventilation air is supplied, because in operation theatre purity of air is important.

Summer Air Conditioning with Ventilation Air : →

The process that is followed generally in summer air conditioning is cooling & dehumidification.



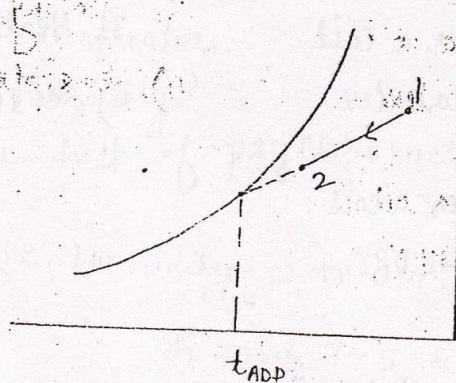
The line joining the inlet & exit conditions of a/c equipment is known as Grand Sensible Heat Factor (GSHF) line.

The line joining supply conditions with inside conditions is known as Room Sensible Heat Factor (RSHF) line.

The point of intersection of GSHF & RSHF will give supply conditions to the room.

Apparatus Dew point (ADP) :-

It is the temperature at which cooling & dehumidification line cut the saturation curve.



$$BPF = \frac{t_2 - t_{ADP}}{t_1 - t_{ADP}}$$

Ques :- Match Group 1 with Group 2 and select the correct answer

Group 1
(process in fig)

P) 0-1

Q) 0-2

R) 0-3

S) 0-4

T) 0-5

Group 2

(Name of the process)

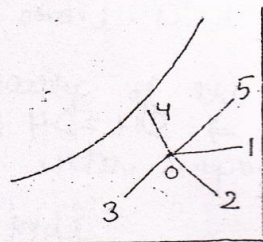
1) Chemical dehumidification

2) Sensible cooling & heating

3) Cooling & dehumidification

4) Humidification with steam injection

5) Humidification with water injection



Ans :- P-2, Q-1, R-3, S-5, T-4

Ques 2 → Match the following

List I

- Steam spray in Air
- Air passing over a coil carrying steam
- Air passing over a coil having temperature less than DPT
- Air passing over a coil having temp above DPT but below DBT

List II

- Sensible Cooling
- Cooling & Dehumidification
- Heating & Humidification
- Sensible Heating

Ans: - A-3, B-4, C-2, D-1

Ques 3 → Air at 20°C DBT and 40% RH is heated to 40°C using electric heater. The temperature of heater is 45°C. Then the bypass factor of heater is

Ans: →

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

$$= \frac{45 - 40}{45 - 20} = 0.2$$

Ques 4 → In an air conditioning process 5 kJ/min of heat is extracted from room. If SHF is 0.8, find the Latent Heat load.

Ans: →

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

$$0.8 = \frac{SH}{5} \Rightarrow SH = 4 \text{ kJ/min}$$

$$TH = SH + LH$$

$$LH = TH - SH = 5 - 4 = 1 \text{ kJ/min}$$

Ques: Air at a dry Bulb Temperature of 30°C has a WBT of 20°C . Then find

- 1) specific humidity, w
- 2) Enthalpy
- 3) specific volume
- 4) Dew point temperature
- 5) Relative humidity

with the help of psychrometric chart.

Soln: $\rightarrow$ Specific Humidity = 0.0105

$$h = 57.5 \text{ kJ/kgda}$$

$$\text{Specific volume} = 0.875 \text{ m}^3/\text{kg}$$

$$\text{DPT} = 15^{\circ}\text{C}$$

$$\text{RH} = 40\%$$

Ques: A small auditorium is required to be maintained at 22°C DBT and 70% RH. The atmospheric conditions are 30°C DBT and 75% RH. The amount of free air circulated is $200 \text{ m}^3/\text{min}$. The required conditions are achieved: first by cooling & dehumidification through a coil having ADP of 14°C and then by sensible heating with the help of psychrometric chart. Find -

- i) Cooling capacity of the coil in TR, and BPF
- ii) Amount of water vapour removed by cooling coil in kg/kgda

free air

$$V = 200 \text{ m}^3/\text{min}$$

$$t_1 = 30^\circ\text{C}$$

$$\phi_1 = 75\%$$

(Solution) $\rightarrow$

$$\psi = 0.887 \text{ m}^2/\text{kg}$$

$$\dot{m} = \frac{V_1}{\psi_1}$$

$$= 225.47 \text{ kg da/min}$$

$$\dot{m} = 3.75 \text{ kg da/sec}$$

$$CC_1 = \dot{m} (h_2 - h_1)$$

$$= 32.5 \text{ kJ/kg da} \times (3.75) \text{ kg da/sec}$$

$$CC_1 = 121.87 \text{ kJ/sec}$$

$$\text{Capacity in TR} = \frac{121.87}{3.5} = \underline{34.82 \text{ TR}}$$

$$\text{moisture Removed} = \omega_1 - \omega_2$$

$$= (0.0198 - 0.0115) = \omega$$

$$= 0.0198 - 0.0115 = 0.0083$$

$$\text{Total moisture} = \dot{m} (\omega_1 - \omega_2)$$

$$= 112.28 \text{ kg v/hr}$$

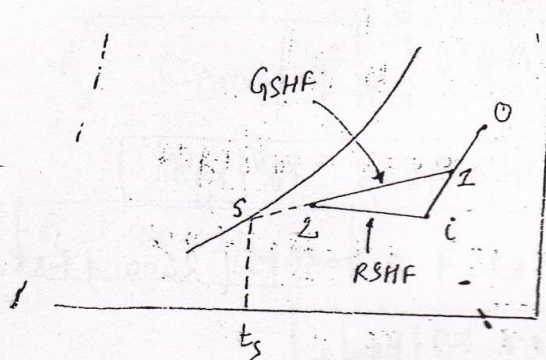
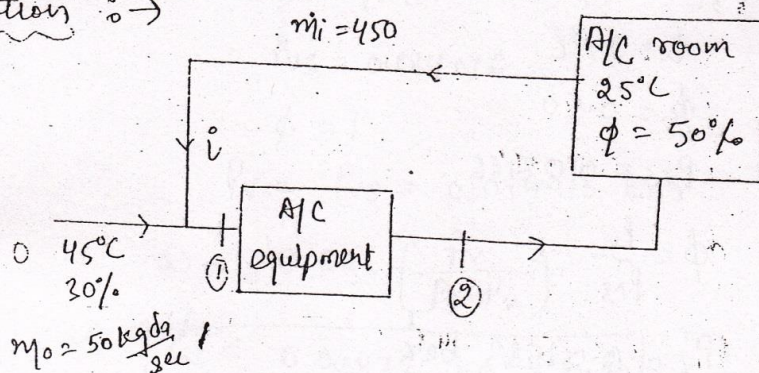
$$\text{BPF} = \frac{T_2 - T_3}{T_1 - T_3} = \frac{22 - 18}{30 - 18} = 0.33$$

Ques 7% → In an air condition space 50 kg/sec of fresh air (dry air basis) at 45°C DBT and 30% RH is introduced in room at 25°C DBT and 50% RH. 450 kg/sec (dry air basis) of recirculated air is mixed with outside air and this mixed air flows over a cooling coil with ADP of 12°C and the coil BPF is 0.15. Determine the conditions at the outlet of cooling coil and also find RSH, RCH and condensate Rate. Solve the problem without using psychrometric chart. And use the following table

$t^{\circ}\text{C}$	$P_{vs} (\text{bar})$
12	0.014016
25	0.03166
45	0.09584

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

Solution →



At point o,

$$t = 45^{\circ}\text{C}$$

$$\phi = 30\% = 0.3$$

$$P_{vs} = 0.09584 \text{ bar}$$

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

$$P_v = 0.02875 \text{ bar}$$

$$\omega_o = 0.622 \left(\frac{P_v}{P_t - P_v} \right) = 0.622 \left(\frac{0.02875}{1.013 - 0.02875} \right)$$

$$\boxed{\omega_o = 0.0182 \text{ kgv/kgda}}$$

$$h_o = C_p t + \omega (2500 + 1.88t)$$

$$= 1.005 \times 45 + 0.0182 (2500 + 1.88 \times 45)$$

$$\boxed{h_o = 92.2 \text{ kJ/kgda}}$$

At point i,

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

$$\phi = 0.50$$

$$P_{vs} = 0.03166$$

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

$$P_v = 0.01583 \text{ bar}$$

$$\omega_i = 0.622 \left(\frac{P_v}{P_t - P_v} \right)$$

$$\boxed{\omega_i = 0.009875 \text{ kgv/kgda}}$$

$$h_i = 1.005 \times 25 + 0.009875 (2500 + 1.88(25))$$

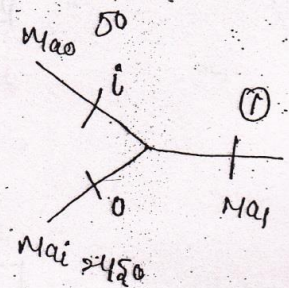
$$\boxed{h_i = 50.27 \text{ kJ/kgda}}$$

Now, apply mass conservation

$$m_{ao} \cdot \omega_o + m_{ai} \omega_i = m_{ai} \omega_i$$

$$50 \times 0.0182 + 450 \times 0.009875 = 500 \times \omega_i$$

$$\omega_i = 0.0107 \text{ kgv/kgda}$$



$$m_{ao} h_o + m_{ai} h_i = m_{ai} h_i$$

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

$$h_i = 54.46 \text{ kJ/kgda}$$

$$h_i = c_p t_i + \omega_i (2500 + 1.88 t_i)$$

$$t_i = 27^\circ\text{C}$$

At point S;

$$t = 12^\circ\text{C}$$

$$P_{vs} = 0.014016$$

$$\phi = 1$$

$$P_v = P_{vs} = 0.014016 \text{ bar}$$

$$\omega = 0.622 \left(\frac{P_v}{P_t - P_v} \right)$$

$$\omega_s = 0.00872 \text{ kgv/kgda}$$

$$h_s = c_p t_s + \omega (2500 + 1.88 t_s)$$

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

$$h_s = 34 \text{ kJ/kgda}$$

At point 2 $\rightarrow$

$$BPF = \frac{t_2 - t_8}{t_1 - t_5} = 0.15$$

$$\frac{t_2 - 12}{27 - 12} = 0.15$$

$$t_2 = 14.25^\circ\text{C}$$

$$\frac{C_p(t_2 - t_s)}{C_p(t_1 - t_s)} = 0.15$$

$$\frac{h_2 - h_s}{h_1 - h_s} = 0.15$$

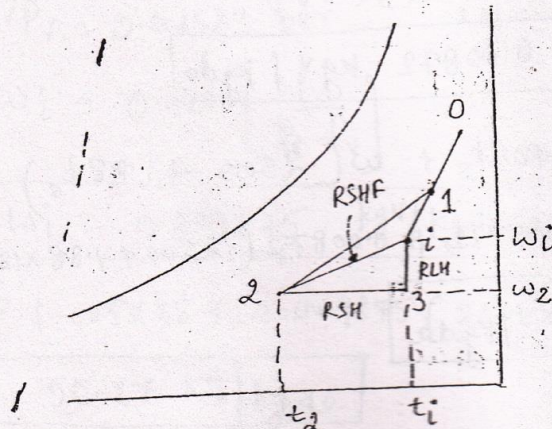
$$\frac{h_2 - 34}{54.4 - 34} = 0.15$$

$$h_2 = 37 \text{ kJ/kg da}$$

$$h_2 = C_p a t_2 + \omega_2 (2500 + 1.88 t_2)$$

$$37 = 1.005 \times 14.25 + w_2 [2500 + 1.88 (14.25)]$$

$$\omega_2 = 0.009 \text{ kg}^{\text{v}}/\text{kgda}$$



mass flow rate from 1 to 2 to i is 500 kg dry/sec

$$RSH = (h_3 - h_2) \text{ KJ/kgda}$$

$$\text{Total RSH} = m(h_3 - h_2) \cdot \text{KJ/sec}$$

$$h_3 = C_p t_3 + w_3 (2500 + 1.88 t_3)$$

$$= C_p t_i + w_2 (2500 + 1.88 t_i)$$

$$h_3 = 48 \text{ KJ/kgda}$$

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

$$RSH = 5500 \text{ KJ/sec}$$

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

$$= 500(50.27 - 48)$$

$$RLH = 1135 \text{ kW}$$

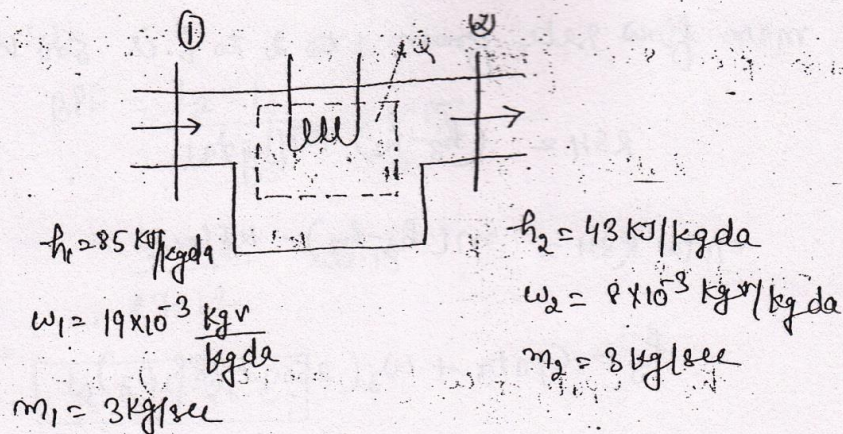
$$\text{condensation Rate} = m(w_1 - w_2)$$

$$= 500(0.0107 - 0.004)$$

$$CR = 0.85 \text{ Kg}$$

Ques 8 → Air at a flow rate of 3 kg/sec (dry air basis) enters a cooling & dehumidification coil with an enthalpy of 85 KJ/kgda and $w = 19$ gram of vap/kgda. The air leaves the coil with an enthalpy of 43 KJ/kgda and $w = 8 \text{ g/kgda}$. If the condensate water leaves the coil with an enthalpy of 67 KJ/kg, find cooling capacity of coil.

(80170) →



$$\text{moisture Removed per kg} = (19-8) \times 10^{-3}$$

$$= 11 \times 10^{-3} \frac{\text{kgv}}{\text{kgda}}$$

$$\text{Total moisture Removed}$$

$$= 3 \times 11 \times 10^{-3}$$

$$w_3 = 33 \times 10^{-3} \frac{\text{kgv}}{\text{kgda}}$$

Energy Conservation

$$\text{Entry Energy} = 85 \frac{\text{kJ}}{\text{kgda}} \times 3 \frac{\text{kgda}}{\text{sec}}$$

$$= 255 \text{ kW}$$

$$\text{Leaving Energy} = Q + (43 \times 3) + (33 \times 10^{-3} \times 67)$$

$$= (Q + 129 + 2.21) \text{ kW}$$

$$255 = Q + 129 + 2.21$$

$$Q = 123.8 \text{ kW}$$

Ques 9) An air conditioning system is designed for restaurant with following data

Total Heat flow through walls, roof, floor = 6.2 kW

Solar Heat gain through Glass = 2 kW

Equipment Sensible Heat gain = 2.9 kW

Equipment latent Heat gain = 0.7 kW

Total infiltration air = 400 m³/hr

The outdoor conditions 35°C DBT, 26° WBT
The inside design conditions are 27°C DBT and 55% RH. Total Number of seating chair is 50. The employee serving meals is equal to 5. Sensible Heat gain per person (Both seating & employee) is equal to 58 W. Latent Heat gain per person (seating person is equal to 44 W).

Latent Heat for employee is equal to 76 W

Sensible Heat added from meal is equal to 0.1 kW

Latent Heat added from meal is equal to 0.3 kW

Find the Sensible Heat factor.

Ans: →

SH

$$= 6.2 \text{ kW}$$

$$= 2.0 \text{ kW}$$

$$= 2.9 \text{ kW}$$

$$\frac{58 \times 55}{1000} = 3.19 \text{ kW}$$

$$= 0.1 \text{ kW}$$

LH

$$= 0.7 \text{ kW}$$

$$\frac{44 \times 50}{1000} = 2.2 \text{ kW}$$

$$\frac{76 \times 5}{1000} = 0.38 \text{ kW}$$

$$= 0.3 \text{ kW}$$

$$\text{SH Gain (KJ/kgda)} = h_1 - h_c$$

$$\text{SH} = \dot{m} (h_1 - h_c) = 0.126 \times (68 - 59) = 1.134 \text{ KJ/sec}$$

$$\text{LH Gain (KJ/kgda)} = (h_o - h_i)$$

$$\text{LH} = \dot{m} (h_o - h_i) = 0.126 (81 - 68) = 1.64 \text{ KJ/sec}$$

$$\text{SHF} = \frac{\text{SH}}{\text{SH} + \text{LH}} = \frac{15.54}{15.54 + 5.24} = \boxed{0.74}$$

Refrigeration Equipment: →

* Compressor

* Condenser

* Expansive Device

* Evaporator

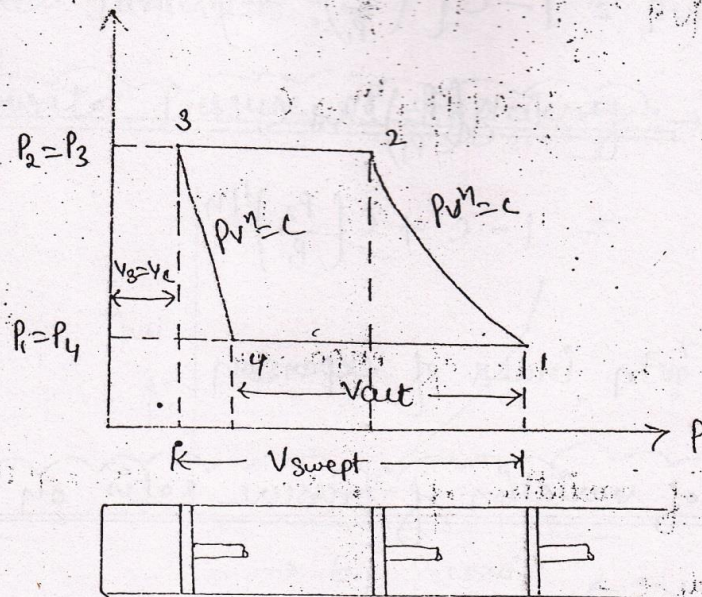
functions of Compressor: →

- (i) To Draw the refrigerant from evaporator.
- (ii) To increase the pressure of Refrigerant
- (iii) To discharge high pressure refrigerant to Condenser.

Volumetric efficiency of Reciprocating compressor: →

$$\eta_{vol} = \frac{V_{net}}{V_{swept}}$$

$$C = \frac{V_{clearance}}{V_{swept}}$$



$$\eta_{vol} = \frac{v_{out}}{v_{swept}}, \quad c = \frac{v_2}{v_3} = \frac{v_s}{v_1 - v_3}$$

$$= \frac{v_1 - v_4}{v_1 - v_3}$$

$$= \frac{v_1 - v_4 + v_3 - v_3}{v_1 - v_3}$$

$$= \frac{(v_1 - v_3) - (v_4 - v_3)}{v_1 - v_3}$$

$$= 1 - \frac{v_3}{v_1 - v_3} \left(\frac{v_4}{v_3} - 1 \right)$$

$$= 1 - c \left(\frac{v_4}{v_3} - 1 \right)$$

$$P_3 v_3^\gamma = P_4 v_4^\gamma$$

$$\frac{v_4}{v_3} = \left(\frac{P_4}{P_3} \right)^{1/\gamma} = \left(\frac{P_2}{P_1} \right)^{1/\gamma}$$

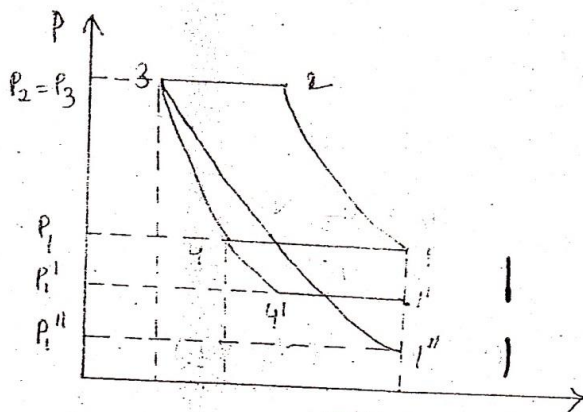
$$\begin{aligned}
 \eta_{vol} &= 1 - C \left[\left(\frac{P_2}{P_1} \right)^{1/\eta} + C \right] \\
 &= 1 - C \left(\frac{P_2}{P_1} \right)^{1/\eta} + C \\
 &= 1 - C + C \left(\frac{P_2}{P_1} \right)^{1/\eta}
 \end{aligned}$$

where, η index of expansion

Effect of variation of pressure ratio on the volumetric

Efficiency $\rightarrow$

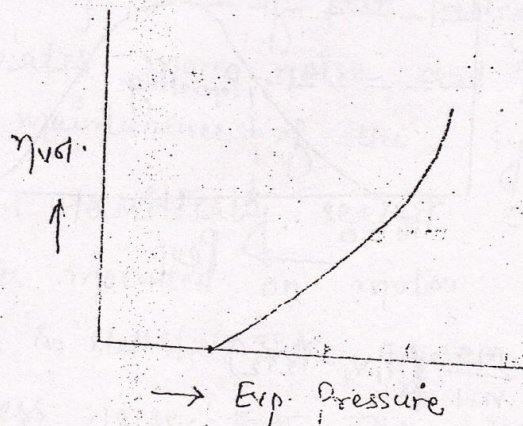
$$\eta_{vol} = \frac{V_{act}}{V_{swept}}$$



With decrease in evaporator pressure, the pressure ratio increases and hence the actual volume of the refrigerant entering the compressor decreases and hence volumetric efficiency also decreases. It is found that at certain low evaporator pressure, the volumetric the η_{vol} becomes zero.

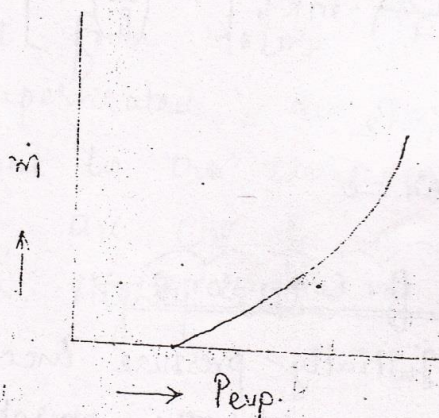
Performance Parameters : $\rightarrow$

(i) Evaporator Pressure vs Efficiency : $\rightarrow$

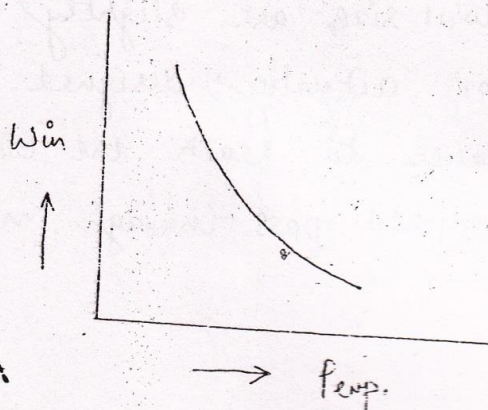


Evap Press $\downarrow \Rightarrow \eta_{\text{ref}} \downarrow$

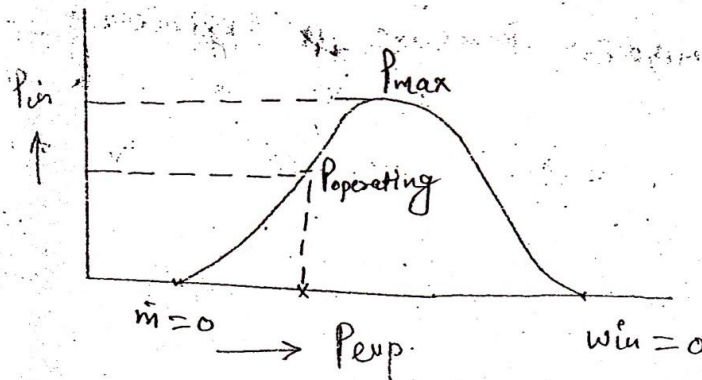
(ii) Peep vs mass flow rate ($\dot{m}$) : $\rightarrow$



(iii) Peep vs Work Input : $\rightarrow$



(iv) P_{exp} vs. P_{in} : $\rightarrow$



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

$$= \left(\frac{\eta}{n-1} \right) \cdot m R T_1 \left(1 - \frac{T_2}{T_1} \right)$$

$$= \left(\frac{\eta}{n-1} \right) \cdot m R T_1 \left[1 - \left(\frac{P_2}{P_1} \right)^{\frac{n-1}{n}} \right]$$

Initially $P_1 = P_2$
 $w_{in} = 0$

Power Requirement for Compressor : $\rightarrow$

Initially pressure increases its peak and hence then it decreases. The operating power is less than maximum power or peak power. Actually the compressor motor size are slightly oversized i.e. the compressor motor actually designed for peak power conditions because to reach the actual operating conditions, it should pass through maximum power conditions.

In open type compressor, the motor & comp^r are separate ^{unit} and the compressor runs with the help of motor by means of belt pulley system. The system generates more noise and occupies large space. The maintenance of the system is easy.

In Hermitically sealed compressor, the compressor is mounted on motor and it enclosed in a welded in steel shell. This system is compact and generates less noise. But the maintenance of system is not easy. As compressor and motor are close to each other. Refrigerant in compressor absorbs heat from motor and thus the refrigerant is superheated, due to this superheating the work input to the compressor increases. and hence the COP of a hermitically sealed compressor is slightly low.

Hermetically Sealed Compressor : →

In open type compressor, the motor and compressor units are separate and condenser runs with the help of motor by means of belt pulley system. This system generates more noise and occupies large space. The maintenance of system is easy.

In Hermetically sealed compressor, the compressor is mounted on motor shaft & is enclosed in a welded steel shell. This system is compact & it generates less noise but the maintenance of system is not easy. As compressor & motor are close to each other refrigerant in compressor absorbs heat from motor and thus the refrigerant is superheated. Due to superheating the work input to the compressor increases. Hence COP of Hermetically sealed compressor is slightly low.

CONDENSER AND EVAPORATOR : →

The function of condenser is to reject heat to the surrounding. Condensers are of three types

- a) Air cooled Condenser
- b) Water cooled Condenser
- c) Evaporative Condenser.

Air cooled condenser are used for small capacity plants like

Water cooled Condensers are used for large capacity plants upto 1000 TR capacity. Water cooled Condenser are of two types

- 1) Shell & tube type
- 2) Shell & coil type

Shell & coil type condensers are used when pure water is available. If water is not pure & scales will be formed and cleaning of coils is not easy.

Evaporative condensers are used where

There is large scarcity of water, in evaporative condensers refrigerant first rejects heat to water and in then then rejects heat to air. Thus same water can be recirculated.

In window air conditioners direct contact type evaporator are used whereas in domestic refrigerator plate type evaporator are used.

EXPANSION VALVE :-

Constant Area $\rightarrow$ Capillary Tube (pressure drop occurs due to friction)

eg Domestic Refrigerator
window A/c

Water Cooler

Variable Area $\rightarrow$ Automatic Exp. device (maintain constant pressure)

$\rightarrow$ thermostatic Exp. device (maintain constant degree of superheat)

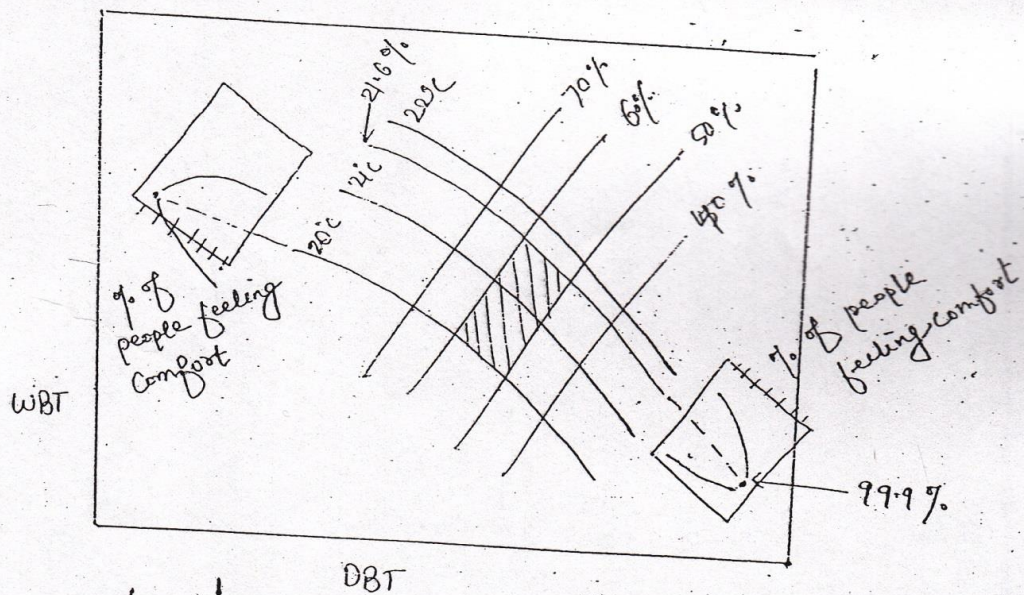
Effective Temperature : $\rightarrow$

It is the temperature of saturated air at which a person experiences same feeling of comfort as in unsaturated environment. Effective temperature includes temperature, Humidity and Air velocity.

$$\text{Effective Temperature} = T_s H, V$$

Effective Temperature depends on age, sex, climatic conditions, Activity.

*** Comfort Chart : $\rightarrow$



The chart is prepared by ASHRAE —
(American Society for Heating, Refrigeration and air conditioning Engineer)
By conditeling research on different

people in this chart drybulb temperature is taken
 on x-axis and WBT on y-axis. Relative
 humidity lines are taken from Psychrometric
 chart. It is found that at an eff. temp.
 20°C 99% of the people feel comfortable
 and in summer at an effective temp of
 21.6°C 99% of people were comfort. the
 shaded region in this chart shows the comfort
 zone for year round conditioning.

