B.Tech 7th Semester Mechanical

Solution of "Refrigeration and Air Conditioning" AS-4215

Section-A ME-471

Multiple type questions

1 (i) (a) compression

(ii) (d) DBT, WBT and air motion

(iii) (a) Increases

(iv) (d) R-113

(v) (d) Ammonia-water

(vi) (c) 4

(vii) (c) Dew point temperature has reached and humidity is 100%

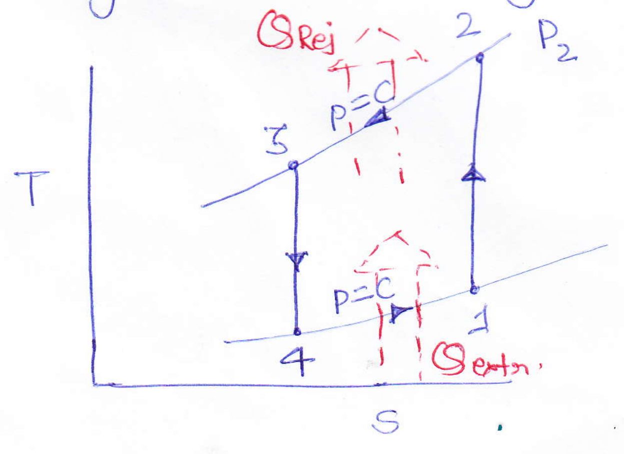
(viii) (d) 1.5

(ix) (a) 4 kW

(x) (b) 205

2. Answer in brief:

(i) Ideal Reversed Brayton Cycle / Bell-Coleman cycle / Joule Cycle



$P_2 > P_1$, $P_2 = P_3$, $P_1 = P_4$

Processes

1-2 → isentropic compression

2-3 → isobaric heat rejection

3-4 → isentropic expansion

4-1 → isobaric heat extraction

We can write the equations for each process by applying SFEE neglecting changes in K.E. and P.E.

$$W_{1-2} = \dot{m}(h_2 - h_1) = \dot{m}C_p(T_2 - T_1)$$

$$Q_{2-3} = \dot{m}(h_2 - h_3) = \dot{m}C_p(T_2 - T_3)$$

$$W_{3-4} = \dot{m}(h_3 - h_4) = \dot{m}C_p(T_3 - T_4)$$

$$Q_{4-1} = \dot{m}(h_1 - h_4) = \dot{m}C_p(T_1 - T_4)$$

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

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

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

$$COP = \frac{Q_{4-1}}{W_{net}} = \frac{\dot{m}C_p(T_1 - T_4)}{\dot{m}C_p(T_2 - T_1) - \dot{m}C_p(T_3 - T_4)} = \frac{T_1 - T_4}{(T_2 - T_1) - (T_3 - T_4)}$$

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

$$= \frac{1}{\left(\frac{T_3}{T_4} - 1\right)} = \frac{1}{\left(\frac{T_2}{T_1} - 1\right)} = \frac{1}{\left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}} - 1}$$

$$COP = \left[(r_p)^{\frac{\gamma-1}{\gamma}} - 1\right]^{-1}$$

$$COP = f(r_p)$$

(ii) Relative Humidity (ϕ or R.H.) :-

It is defined as the ratio of the mass of water vapour m_v in a certain volume of moist air at a given temperature to the mass of water vapour m_{vs} in the same volume of saturated air at the same temperature.

$$\phi = \frac{m_v}{m_{vs}} = \frac{P_v V/\bar{R}T}{P_s V/\bar{R}T} = \frac{P_v}{P_s}$$

$$\phi = \frac{V/v_v}{V/v_s} = \frac{v_s}{v_v}$$

$$\phi = \frac{\omega}{0.622}\frac{P_a}{P_s}$$

$$\mu = \phi\left[\frac{1-P_s/p}{1-P_v/p}\right]$$

Thus R.H. turns out to be the ratio of partial pressure of water vapour in a certain unsaturated moist air at a given temperature T to the saturation pressure of water vapour (or partial pressure of water vapour in saturated air) at the same temperature T. It is usually measured in percentage.

Specific Humidity or Humidity Ratio (ω):-

It is defined as the ratio of mass of water vapour to the mass of dry air (d.a.) in a given volume of the moist air. Thus

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

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

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

(iii) In the ammonia-water system, ammonia is the refrigera[nt] and water is absorbent. Ammonia forms a highly non-ideal solution in water. Hence from the point of view of solubility requirement, it is satisfactory. But the difference in their boiling points is only 138°C. Hence the vapour leaving the generator contains some amount of water.

* If absorbent vapour goes with the refrigerant vapour to refrigeration circuit the refrigeration produced will not be isothermal or evaporator temperature doesn't remains constant, refrigeration effect will be reduced.

* COP decreases

* Some water is left behind in the evaporator. This water has to be removed at frequent intervals.

* Condenser temp. doesn't remains constant.

(iv) (a) R-113 — $CCl_2F\,CClF_2$ — $C_2Cl_3F_3$

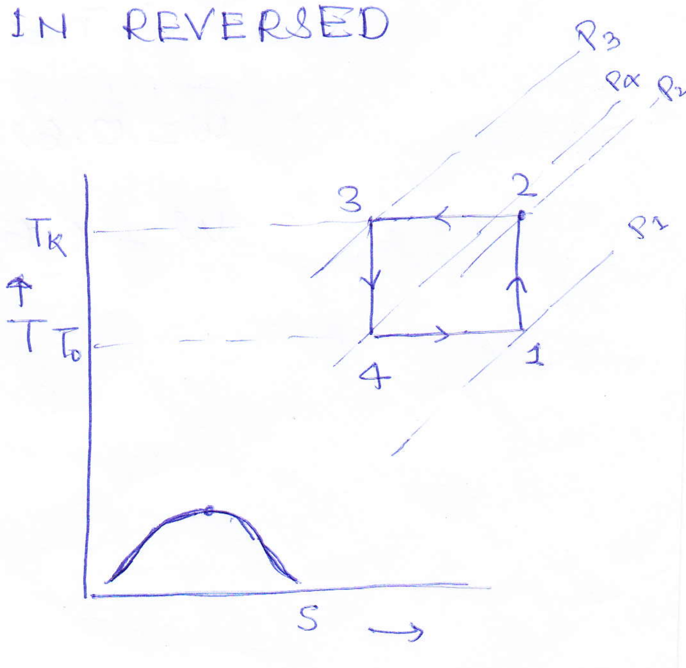
(b) R-134 — $CHF_2\,CHF_2$ — $C_2H_2F_4$

(c) R-11 — CCl_3F

(d) R-13B1 — CF_3Br

(v) GAS AS A REFRIGERANT IN REVERSED CARNOT CYCLE :-

Fig shows the Carnot-cycle 1-2-3-4 with gas as a refrigerant



Answer 2 (V) conti.....

It is found that serious practical difficulties are encountered in the application of Carnot cycle.

* Firstly, it is not possible to devise, in practice, isothermal process of heat absorption and rejection 4-1 and 2-3, with gas as the working substance.
* Secondly, since the volume is changing both during the reversible isothermal and reversible adiabatic processes, the stroke volume of the cylinder is very large. The cycle therefore, suffers from poor actual COP.

SECTION - B

(3)

(i) C = Speed of the air craft = 277.8 m/s

stagnation temperature $T_2 = T_1 + \dfrac{C^2}{2C_p}$

$C_p = 1.005$, $C = 277.8\ m/s$

$T_1 = 258\ K$

$$\boxed{T_2 = 296.5\ K}$$

stagnation pressure $P_2 = P_1 \left(\dfrac{T_2}{T_1}\right)^{\frac{\gamma}{\gamma-1}} = 0.57\ bar$

(ii) discharge pressure from the compressor $P_3 = 3(0.57)$

$P_3 = 1.71\ bar$

discharge temperature from the jet compressor

$T_3 = T_2 \left(\dfrac{P_3}{P_2}\right)^{\frac{\gamma-1}{\gamma}} = 406\ K$

$t_3 = 133°C$

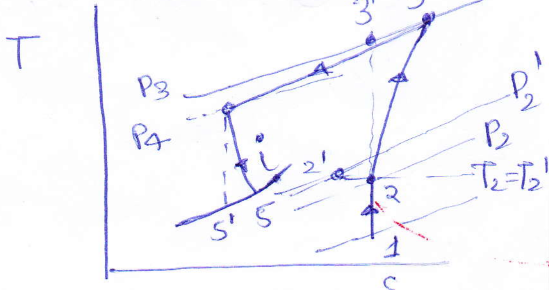
$T_4 = T_2 = 296.5\ K$ (for a perfect heat exchanger)

$P_4 = 1.71 - 0.1 = 1.61\ bar$

$P_5 = 1.06\ bar$

$T_5 = T_4 \left(\dfrac{P_5}{P_4}\right)^{\frac{\gamma-1}{\gamma}} = 263\ K$

$t_5 = -10°C$



Refrigeration effect $q_0 = 1.005(25+10) = 35.18$ kJ/kg

mass flow rate of air

$$\dot{m} = \frac{58.05 \times 3600}{35.18} = 5950 \text{ kg/hr}$$

1.65 kg/sec

(iii) volume handled by the compressor

$$\dot{V}_c = \frac{mRT_2}{P_2} = \frac{5950(0.286 \times 10^3)(296.5)}{0.57 \times 10^5}$$

$$\dot{V}_c = 9050 \text{ m}^3/\text{hr}$$

$= 2.59$ m³/sec

(iv) Ram work $W_R = C_p(T_2 - T_1) = 38.7$ kJ/kg

Compressor work $W_c = C_p(T_3 - T_2) = 111$ kJ/kg

Expander work $W_E = C_p(T_4 - T_5) = 33.7$ kJ/kg

Net work $W = W_R + W_c - W_E = 116$ kJ/kg

(v) $$COP = \frac{35.18}{116} = 0.3$$

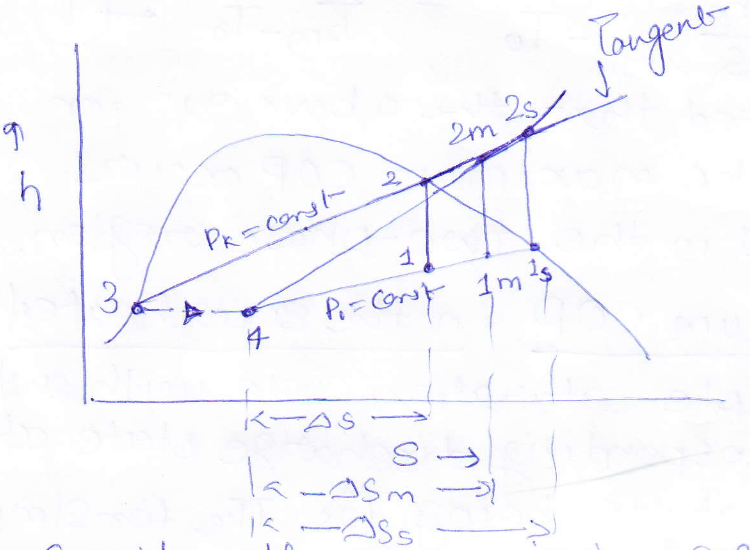
(4) **EWING'S Construction :-**

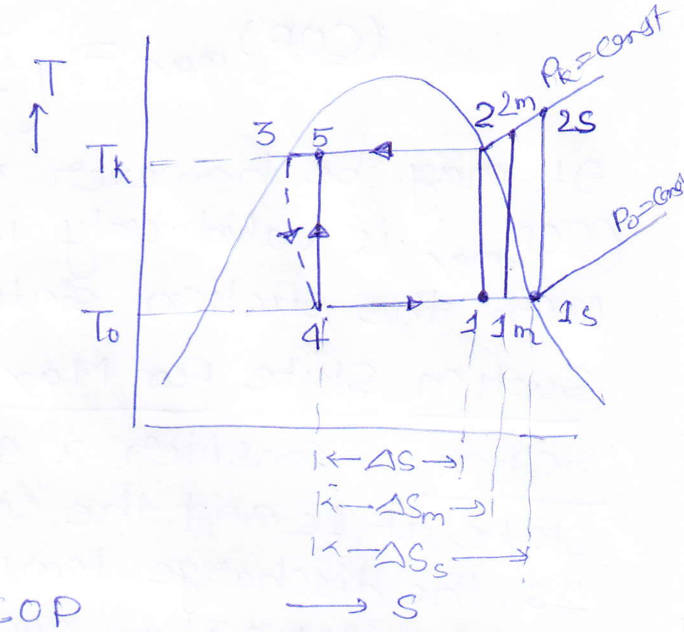
During superheating we have found that superheating increases

* Refrigeration effect
* Work of compression
* Specific volume

So finally COP will increase, decrease or not, that depends upon nature of the refrigerant. There is a method by which we can find a criteria for the suction state for maximum COP. It is suggested by EWING, known as EWING'S construction.

For some refrigerants and certain operating conditions the maximum COP occurs with the suction state 1m in the two phase region and for some others in the superheated region.





Consider the expression for COP

$$COP = \frac{h_1 - h_4}{h_2 - h_1} = \frac{(h_1 - h_4)/\Delta s}{(h_2 - h_4)/\Delta s - (h_1 - h_4)/\Delta s}$$

It can be noted from enthalpy-entropy diagram that the term $(h_1 - h_4)/\Delta s$ is the gradient of the evaporator pressure P_0 = const. line and is equal to T_0, as long as point 1 is in the two-phase region. The expression for COP, with the suction state in the two-phase region, is then

$$COP = \frac{T_0}{(h_2 - h_4)/\Delta s - T_0}$$

The numerical value of the term $(h_2 - h_4)/\Delta s$, however changes as point 2 shifts along with point 1. It can be seen that the COP is maximum when this gradient is minimum, i.e. when a line drawn from 4 to P_k = const line makes a tangent at 2m. The corresponding suction state for maximum COP is obtained on the isentropic line at 1m. Using the thermodynamic relation

$$\left(\frac{\Delta h}{\Delta s}\right)_p = T \quad \text{we find that} \quad \left[\frac{h_{2m} - h_4}{\Delta s_m}\right]_{P_K = Const} = T_{2m}$$

so that the slope of the tangent $\Delta h/\Delta s$ from point 4 to the P_k = const. line is equal to the discharge temperature T_{2m}. The expression for max COP is then

$$(COP)_{max} = \frac{T_0}{\left(\frac{h_2 - h_4}{\Delta S}\right)_{min} - T_0} = \frac{T_0}{T_{2m} - T_0} \quad -(1)$$

It may be however noted that the above eq^n for $(COP)_{max}$ is valid only if the maximum COP occurs with the suction state 1 in the two-phase region.

<u>Suction State for Maximum COP in the superheated region</u> :- Consider a simple saturation cycle with suction state at 1s and the corresponding discharge state at 2s, the discharge temperature being T_{2s}, The for simple ~~COP~~ saturation cycle

$$COP = \frac{T_0}{\frac{h_{2s} - h_4}{\Delta S_s} - T_0} \quad -(2)$$

from fig slope of tangent at 2m < slope of line 4-2s < slope of tangent at ~~2s~~ 2s. It implies that

$$T_{2m} < \frac{h_{2s} - h_4}{\Delta S_s} < T_{2s} \quad -(3)$$

Comparing eq^n (1), (2) and (3), we see that

$$(COP)_{max} > (COP)_{sat} > \frac{T_0}{T_{2s} - T_0}$$

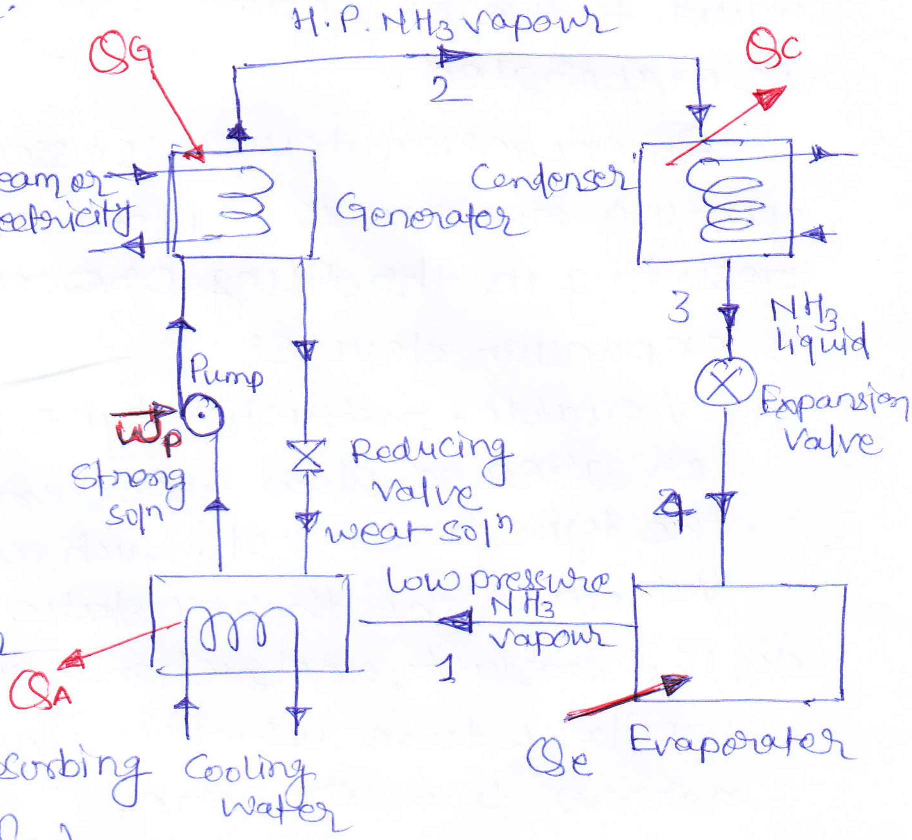
In similar manner, it can be shown that for max COP to occur with the suction state in the super heat region

$$\boxed{(COP)_{sat} < \frac{T_0}{T_{2s} - T_0}}$$

Answer (5)

5. It consist of an absorber, a pump, a generator and a pressure reducing valve to replace the compressor of vapour compression system. The other component of the system and and condenser, expansion valve and evaporator as in the vapour compression system.

The ammonia-vapour leaving the evaporator at point 1 is absorbed in the low temperature hot solution in the absorber, releasing the latent heat of condensation. The temp. of the solution tends to rise, while the absorber is cooled by the circulating water, absorbing the heat of solution (Q_A) and maintaining a constant temperature.



Strong solution rich in ammonia, is pumped (increasing the pressure of the solution upto condenser pressure) to the generator where heat (Q_G) is supplied from an external source. Since the boiling point of ammonia is less than that of water, the ammonia vapour is given off from the aqua-ammonia solution at high pressure, and the weak solution returns to the absorber through a pressure reducing valve.

The high pressure ammonia vapour from the generator is condensed in the condenser to a high pressure liquid NH_3. This liquid NH_3 is throttled by the expansion valve, and then evaporates, absorbing the heat of evaporation from the surroundings.

6. An expansion device in a refrigeration system normally serves two purposes. One is the thermodynamic function of expanding the liquid refrigerant from the condenser pressure to evaporator pressure. The other is control function which may involve the supply of the liquid to the evaporator at the rate at which it is evaporated.

An expansion device is essentially a restriction offering resistance to flow so that the pressure drops, resulting in throttling process. There are two types of expansion devices :-

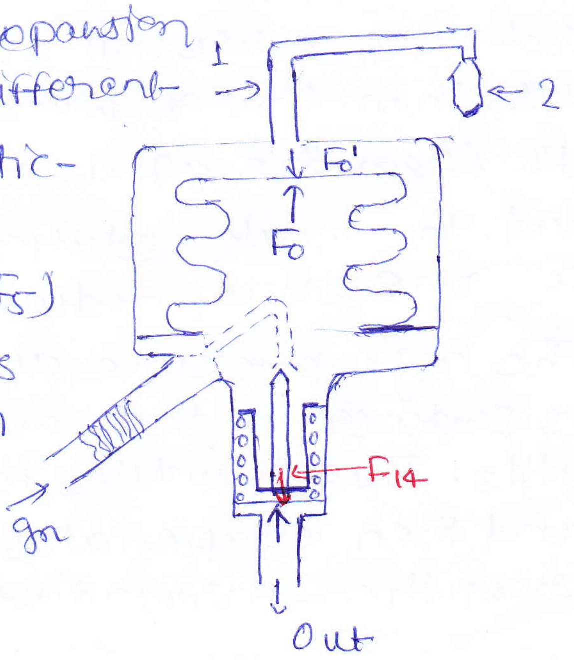
(i) Variable-restriction type :- the extent of opening or area of flow keeps on changing depends on the type of control. Such as automatic expansion valve and the thermostatic expansion valve.

(ii) The constant restriction type device is the capillary tube which is merely a long tube with a narrow diameter bore.

Thermostatic Expansion Valve :-

It is a throttling device which works automatically, maintaining proper and correct liquid flow as per the requirements of the load on the evaporator.

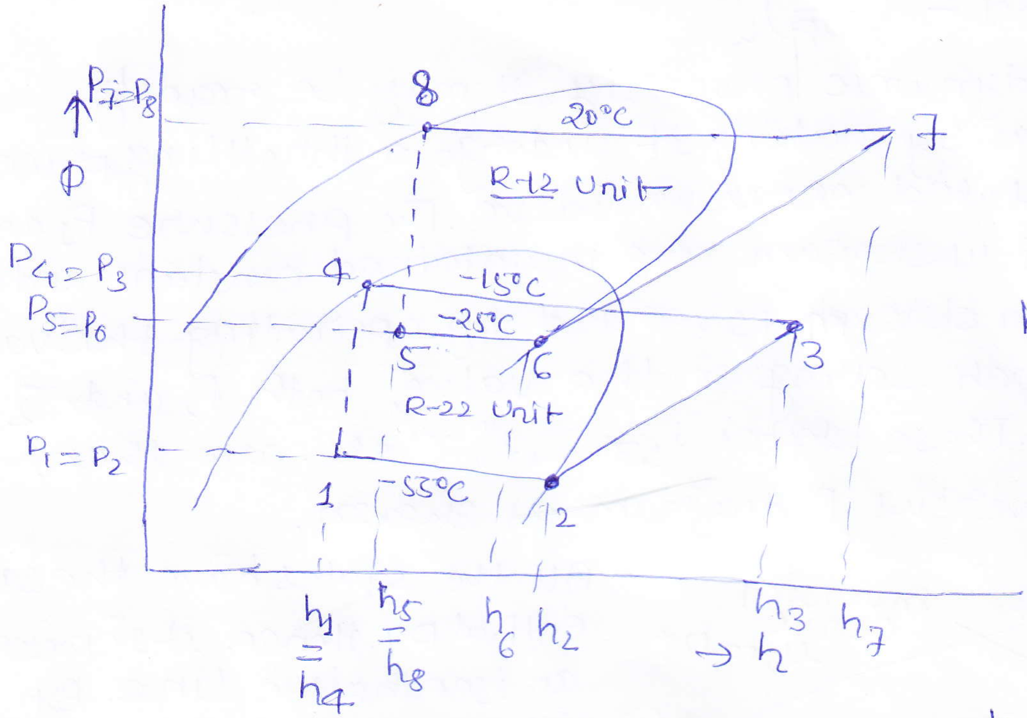
A typical design of thermostatic-expansion valve is shown. It is only slightly different in construction from the automatic-expansion valve. In this case the downward force, on the bellows (F_7+F_5) which tends to open the valve is replaced by a force F_0' exerted on the bellows through the capillary tube (1) which is connected to the bulb (2). The bulb in turn fixed at the outlet end of the evaporator



Answer (7)

(7.) $Q = 9$ TR, $t_E(R\text{-}22) = -55°C$, $t_E(R\text{-}12) = -25°C$

$t_C(R\text{-}22) = -15°C$, $t_C(R\text{-}12) = 20°C$



For R-22 from p-h diagram

$P_1 = P_2 = 0.496$ bar

$P_3 = P_4 = 2.969$ bar

$h_{f4} = h_4 = 27.8$ kJ/kg

$h_2 = 226.2$ kJ/kg

$h_3 = 270$ kJ/kg

For R-12

$P_5 = P_6 = 1.238$ bar

$P_7 = P_8 = 5.674$ bar

$h_5 = 55.1$ kJ/kg $= h_8 = h_{f8}$

$h_6 = 176.5$ kJ/kg

$h_7 = 215$ kJ/kg

Compression Ratio

for R-22 unit $= \dfrac{2.969}{0.496} = 6.33$

For R-12 Unit $= \dfrac{5.674}{1.238} = 4.58$

Quantity of Refrigerant Circulated

For R-22 unit $\dot{m}_1 = \dfrac{14000 \times 9}{60(226.2 - 27.8)} = 10.58$ kg/min

For R-12 unit $\dot{m}_2 = \dfrac{\dot{m}_1(h_3 - h_4)}{(h_6 - h_5)} = \dfrac{10.58(270 - 27.8)}{(176.5 - 55.1)} = 21.10$ kg/min

COP for each unit

$$(COP)_{R\text{-}22} = \frac{\dot{m}_1(h_2 - h_1)}{\dot{m}_1(h_3 - h_2)} = \frac{226.2 - 27.8}{270 - 226.2} = 4.53$$

$$(COP)_{R\text{-}12} = \frac{\dot{m}_2(h_6 - h_5)}{\dot{m}_2(h_7 - h_6)} = 3.15$$

COP of the Whole System $= \dfrac{14000\, Q}{60[\dot{m}_1(h_3 - h_2) + \dot{m}_2(h_7 - h_6)]}$

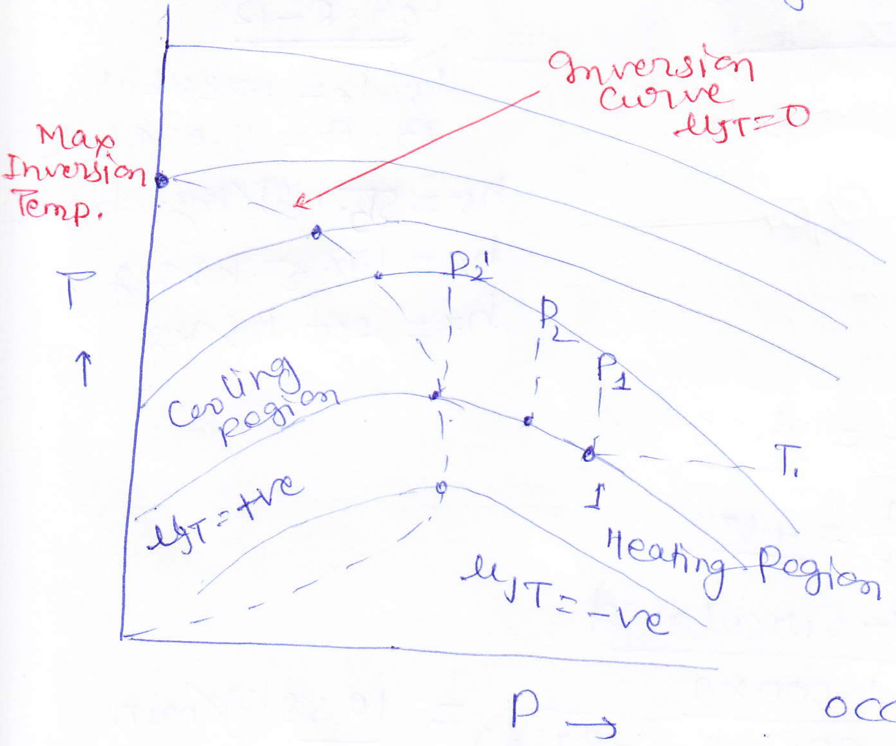
$= 1.69$ Ans

(8.) Joule-Thomson Expansion :-

The joule-thomson coefficient is defined by the expression

$$\mu_J = \left(\frac{\partial T}{\partial P}\right)_h$$

It is a thermodynamic property. It may be readily obtained for a gas by making it undergo a throttling process through an isolated expansion valve. The pressure P_1 and temperature T_1 upstream are maintained constant and the pressure P_2 down stream is varied by operating the valve manually, for each setting of the valve, both P_2 and T_2 are measured. These states 1, 2, 2', 2'', 2''' etc. are then plotted on a T versus P diagram as shown.



All the states have the same enthalpy. Hence it represents an isenthalpic line. By changing the initial state P_1, T_1 and hence h_1, a group of such isenthalpic lines can be drawn. The locus of the maxima of these isenthalpics is called "inversion curve". The point at which this maximum occurs, i.e the point at which the slop of T-p curve changes from positive to negative is called the inversion point. The slope of these isenthalpics $(\partial T/\partial P)_h$ is equal to the joule thomson coefficient. It is seen that μ_J is positive for the region inside the inversion ~~point~~ curve. Hence it represents the cooling region. on the other hand μ_J is negative for the region outside the inversion curve. Hence it represents the heating region. Inversion curve represents the locus of initial states of a gas corresponding to which there is neither cooling nor heating on Joule-Thomson Expansion.

* If the intial condition is ~~below~~ above the max. inversion temp. upon throttling cooling will never be obtained.

(9.) Q $P = 680$ mm of Hg $= \frac{680}{760} \times 1.0132 = 0.9065$ bar

(i) Specific humidity

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

Carrier's Equation

$$P_v = P_{wb} - \frac{(P - P_{wb})(T_{db} - T_{wb})}{1547 - 1.44\, T_{wb}}$$

$T_{db} = 35°C$, $T_{wb} = 28°C$

$P_{wb} = 0.03169$ bar (at 28°C)

$P_v = 0.0259$ bar

$$\therefore w = \frac{0.622 \times 0.0259}{0.9065 - 0.0259} = 0.01829 \text{ kg of w.v./kg of d.a.}$$

(ii) Specific Volume

$$v = \frac{R_a T_{db}}{(P - P_{db}) \times 10^5} = \frac{287.2 \times 308}{(0.9065 - 0.0259) \times 10^5}$$

$= 1.0045\ m^3/kg$ of d.a.

(iii) $h = 1.004 \times 35 + 0.01829(2501.4 + 1.88 \times 35)$

$= 82.094$ KJ/kg of d.a.

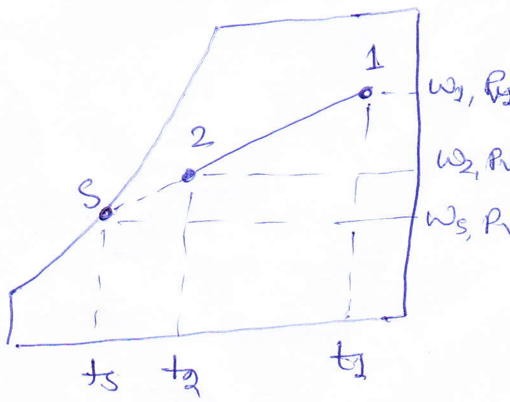
(iv) $R.H. = \frac{P_v}{P_s} \times 100 = \frac{0.0259}{0.05628} \times 100 = 46.02\%$

P_s = Sat. pressure of water vapour at $35°C = 0.05628$ bar

(10.) At the apparatus dew point

$w_s = 5.25$ g/kg d.a.

$h_s = 17.7$ KJ/kg d.a.



State of entering air

$w_1 = 8.2$ g/kg d.a.

$v_1 = 0.872\ m^3/kg$ d.a.

$h_1 = 52.5$ KJ/kg d.a.

mass flow rate of dry air

$$\dot{m}_a = \frac{Q_v}{v} = \frac{39.6}{0.872} = 44.41 \text{ kg d.a./min}$$

Cooling load per kg of dry air

$$h_1 - h_2 = \frac{\dot{Q}}{\dot{m}_a} = \frac{(12.5)(60)}{44.41} = 16.89 \text{ kJ/kg d.a.}$$

Enthalpy of air leaving the coil $h_2 = 52.5 - 16.89$

$$h_2 = 35.61 \text{ kJ/kg of d.a.}$$

Equation of the condition line

$$\frac{h_1 - h_2}{h_1 - h_s} = \frac{w_1 - w_2}{w_1 - w_s}$$

$$\frac{52.5 - 35.61}{52.5 - 17.7} = \frac{8.2 - w_2}{8.2 - 5.25}$$

$$w_2 = 6.77 \text{ g w.v./kg d.a.}$$

dry bulb temp. and wet-bulb temp. of air leaving the coil for calculated values of h_2 and w_2 from psychrometric chart

$$t_2 = 18.6°C$$

$$t_2' = 12.5°C$$

Coil bypass factor $X = \dfrac{h_2 - h_s}{h_1 - h_s} = 0.515$

11. (i) when the duct is circular

cross-section area, $A = \frac{\pi}{4} D^2 = 0.06157 m^2$

$\therefore$ velocity of air $C = \frac{Q}{A} = \frac{1.2}{0.06157} = 19.49 \ m/s$

wetted perimeter, $P = \pi D = 0.8796 \ m$

hydraulic mean depth, $m = \frac{A}{P} = 0.07 \ m$

Pressure Drop $P_f = \frac{fL}{m} \left(\frac{C}{4.04}\right)^2$ mm of water

$P_f = 19.15$ mm of water

(ii) when the duct is Square

cross section area $A = 0.28 \times 0.28 = 0.0784 \ m^2$

velocity of air $C = \frac{Q}{A} = 15.31 \ m/s$

wetted perimeter $P = 2(a+b) = 1.12 \ m$

$m = \frac{A}{P} = 0.07 \ m$

Pressure Drop. $P_f = \frac{fL}{m} \left(\frac{C}{4.04}\right)^2$ mm of water

$P_f = 11.82$ mm of water

12. Transport Refrigeration:-

Most significant application of refrigeration is in food preservation, whether it is by way of processing or for storage. Processing is done by heating, heat drying etc. and by refrigeration such as in chilling, freezing or freeze-drying. Storage may be of either chilled or frozen product. Some of the important products involve in processing are candy, beverages, meat, poultry, fish bakery. and dairy products, fruits and vegetables etc. An interesting feature of the chilled and frozen food industry is the cold chain that must be maintained from the farm to the consumer. An important link in this chain is that of transport refrigeration.

Refrigerated Trucks and Trailers:-

These vehicles are refrigerated to maintain temperatures of either 1.5 to 4°C for cold foods or −18°C for frozen foods. The types of refrigeration systems used are given below:-

Using water ice:- The top of the product can be suitably iced. Again it is a satisfactory method for short distances and for some products only. The refrigerating effect produced by the melting of ice is 335.4 kJ/kg.

Using Dry Ice:- Dry ice is used in many small retail trucks for the delivery of frozen food, such as ice-cream. The usual positioning of the dry ice blocks is in the ceiling. The cooling is by natural convection. The refrigerating effect produced by the sublimation of dry ice, which takes place at a temp. of −78.5°C, is 605.5 kJ/kg.

Water Ice in Bunker with Forced Air Circulation:- Fig. shows the sketch of an ice bunker that is fitted in front of an insulated vehicle. Air is sucked over ice by the blower taking it drive from the engine. A $\frac{1}{2}$ HP blower will add a heat equivalent of 0.37 kW. A mixture of ice and salt can also be used for lower temperature upto −9°C.