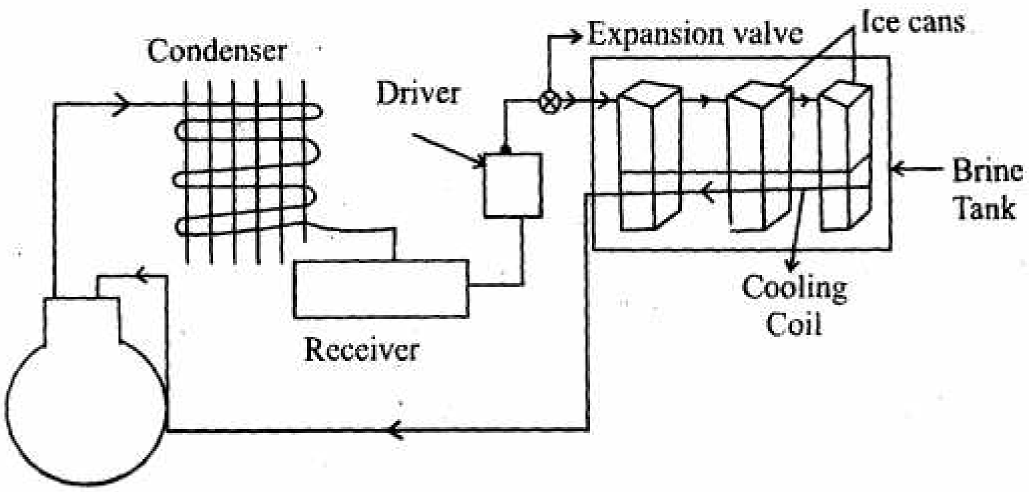
Seventh Semester Examination, Dec., 2007

REFRIGERATION AND AIR CONDITIONING

Note : Attempt any five questions.

Q. 1. (a) Explain with neat sketches the Ice refrigeration and evaporation methods of refrigeration.

Ans. Ice Refrigeration : The commercial ice is produced by freezing portable water in standard cans places in rectangular tanks. The tanks are filled with chilled brine. For increasing the heat transfer from the water in the can to be the chilled brine, the brine solution is kept in constant motion by agitators. The agitators can be either horizontal or vertical and are operated by means of electric motors.



The brine temperature is maintained by the refrigeration plant at –10°C to –11°C. The ammonia gas is used as the refrigerant because of its excellent thermal properties. It also produces very high refrigerating effect per kg of refrigerant and low specific volume of the refrigerant in vapour state. The high temperature, high pressure ammonia vapours are condensed in a condenser which may be of shell and tube type or evaporative type. The condensed liquid ammonia is collected in the receiver and then expanded through the expansion value. Due to the expansion, the pressure of the liquid ammonia is considerably reduces.

Q. 1. (b) Explain in brief which refrigerant(s) would you choose for each of the following applications & why?

(i) A cold storage 100 TR capacity using reciprocating compressor.

(ii) An 800 TR air conditioning plant using centrifugal compressor.

(iii) A small capacity frozen food cabinet to maintain –30°C temperature.

Ans. Type of refrigerant used :

(i) A cold storage 100 TR capacity using reciprocating compressor : These compressors are used for refrigerant which have comparatively low volume perk and a large differential pressure such as ammonia (R-7/7), R-12, R-22 and methyl chloride (R-40). The reciprocating compressors are available in sizes as small as $\frac{1}{12}$ kW which are used in small domestic refrigerators and upto about 150 kW for large capacity installations.

(ii) An 800TR air conditioning plant using centrifugal compressor : This compressor increases the pressure of low pressure vapour refrigerant to a high pressure by centrifugal force. The centrifugal compressor is generally used for refrigerants that requires large displacement and condensing pressure such as R-11 and R-113. However, the refrigerant R-12 is also employed for large capacity applications and low-temperature applications.

Q. 2. A cockpit of a jet plane is cooled by a simple cooling cycle. Assuming the following data, find the quantity or air passed through the cooling turbine and C.O.P. of the cycle. Plane speed 1200 km/h, Ram efficiency 90% Ambient air pressure 0.85 kgf/cm^2, Ambient air temperature 30°C, pressure ratio of the main compressor 4, temperature of the air leaving the heat exchanger 60°C, Pressure drop in the heat exchanger 0.5 Kgf/cm^2, Pressure in the cockpit 1Kgf/cm^2 temperature of air leaving the cockpit 25°C, pressure loss between the cooler turbine and cockpit = 0.20 Kgf/cm^2, Isentropic efficiency of main compressor is 75% and of cooler turbine is 80%, load in the cockpit 10 tons. If the rise in temperature in the cockpit is limited to 10°C and the air coming out from cooler turbine is mixed with the ram air, find the quantity of ram air mixed per kg of air coming from cooler turbine. The ram air is throttled to balance the pressure.

Ans.

$$v = 1200 \text{ km/hr} = 333.3 \text{ m/sec.}$$

$$T_6 = 25°C = 25 + 273 = 298 \text{ k}$$

$$P_6 = 1 \text{ bar}$$

$$P_1 = 0.85 \text{ bar}$$

$$T_1 = 30°C = 30 + 273 = 303 \text{ K}$$

$$\theta = 10\text{TR}, \ \frac{P_3}{P_2'} = 4$$

$$\eta_R = 90\% = 0.9$$

$$T_4 = 60°C = 60 + 273 = 333\text{K}$$

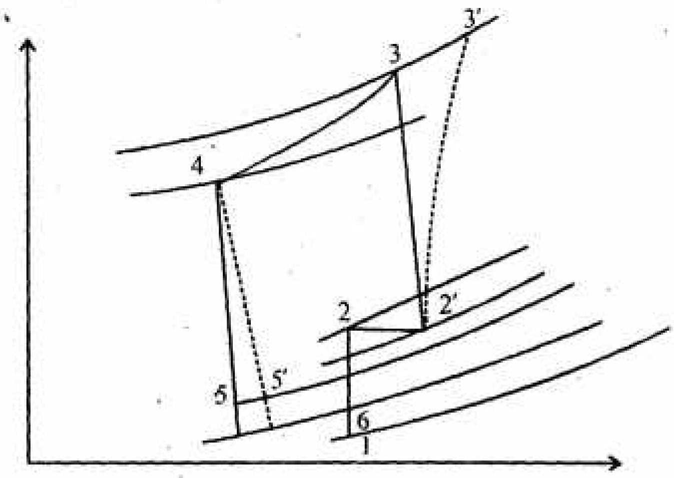
$$P_4 = (P_3' - 0.5)$$

$$P_5 = P_5' = P_6 + 0.2 = 1 + 0.2 = 1.2 \text{ bar}$$

$$\eta_C = \eta_T = 80\% = 0.8$$

$$\nu = 1.4,\ C_P = 1\,\text{kJ/kg K}$$

The T-S diagram for the simple air cooling system with the given conditions is shown in the figure.



Let $T_2' =$ Stagnation temperature of the ambient air entering the main compressor $= T_2$

$P_2 =$ Pressure of air after isentropic ramming

$P_2' =$ Stagnation pressure of air entering the main compressor.

We know that,

$$T_2 = T_2' = T_1 + \frac{V^2}{2000C_P} = 303 + \frac{(333.3)^2}{2000 \times 1}$$

$$= 303 + 55.5 = 358.5\,\text{K}$$

$$\frac{P_2}{P_1} = \left(\frac{T_2}{T_1}\right)^{\frac{\gamma}{\gamma-1}} = \left[\frac{358.5}{303}\right]^{\frac{1.4}{1.4-1}} = (1.183)^{3.5} = 1.8$$

$$P_2 = P_1 \times 1.8 = 0.85 \times 1.8 = 1.53\,\text{bar}$$

We know that ram efficiency

$$\eta_R = \frac{\text{Actual pressure rise}}{\text{Isentropic pressure rise}} = \frac{P_2' - P_1}{P_2 - P_1}$$

$$0.9 = \frac{P_2' - 0.85}{1.53 - 0.85} = \frac{P_2' - 0.85}{0.68} \Rightarrow P_2' = 0.9 \times 0.68 \times 0.85 = 1.46 \text{ bar}$$

Now for the isentropic process, 2'–3

$$\frac{T_3}{T_2'} = \left[\frac{P_3}{P_2'}\right]^{\frac{\gamma-1}{\gamma}} = [4]^{\frac{1.4-1}{1.4}} = (4)^{0.286} = 1.486$$

$$T_3 = T_2' \times 1.486 = 358.5 \times 1.486 = 532.7 \text{ K}$$

And isentropic efficiency of the compressor,

$$\eta_C = \frac{\text{Isentorpic temperature rise}}{\text{Actual temperature rise}} = \frac{T_3 - T_2'}{T_3' - T_2'}$$

$$0.8 = \frac{532 - 358.5}{T_3' - 358.5} = \frac{174.2}{T_3' - 358.5}$$

$$T_3' = \frac{174.2}{0.8} + 358.5 = 576 \text{ K}$$

Since the pressure ratio of the main compressor $\left(\frac{P_3}{P_2'}\right)$ is 4, therefore pressure of air leaving the main compressor,

$$P_3 = P_3' = 4P_2' = 4 \times 1.46 = 5.84 \text{ bar}$$

Pressure drop in the heat exchanger = 0.5bar

∴ Pressure of air after passing through the heat exchanger or at entrance to the cooling turbine,

$$P_4 = P_3' - 0.5 = 5.84 - 0.5 = 534 \text{ bar}$$

Also there is a pressure loss of 0.2 bar between the cooling turbine and the back pit. Therefore, pressure of air leaving the cooling turbine,

$$P_5 = P_5' = P_6 + 0.2 = 1 + 0.2 = 1.02 \text{ bar}$$

Now for the isentropic process 4–5

$$\frac{T_4}{T_5} = \left[\frac{P_4}{P_5}\right]^{\frac{\gamma-1}{\gamma}} = \left[\frac{5.34}{1.2}\right]^{\frac{1.4-1}{1.4}} = [4.45]^{0.286} = 1.53$$

$$\therefore \quad T_5 = \frac{T_4}{1.53} = \frac{333}{1.53} = 217.6\,K$$

We know that isentropic efficiency of the cooling turbine

$$\eta_T = \frac{\text{Actual temperature rise}}{\text{Isentropic temperature rise}} = \frac{T_4 - T_5'}{T_4 - T_5}$$

$$0.8 = \frac{333 - T_5'}{333 - 217.6} = \frac{333 - T_5'}{115.4}$$

$$T_5' = 333 - 0.8 \times 115.4 = 240.7\,k$$

Quantity of air passed through the cooling turbine :

We know that quantity of air forced through the cooling turbines

$$m_a = \frac{210\theta}{C_P(T_6 - T_5')} = \frac{210 \times 10}{1(298 - 240.7)} = 36.6\,kg/min$$

C.O.P. of the system : We know that C.O.P. of the system

$$= \frac{210\theta}{m_a c_p(T_3' - T_2')} = \frac{210 \times 10}{36.5 \times 1(576 - 358.5)} = 0.264$$

Q. 3. (a) Explain the different methods of improving the COP of a simple compression refrigeration cycle.

Ans. Methods of improving the C.O.P. of a simple compression refrigeration cycle :

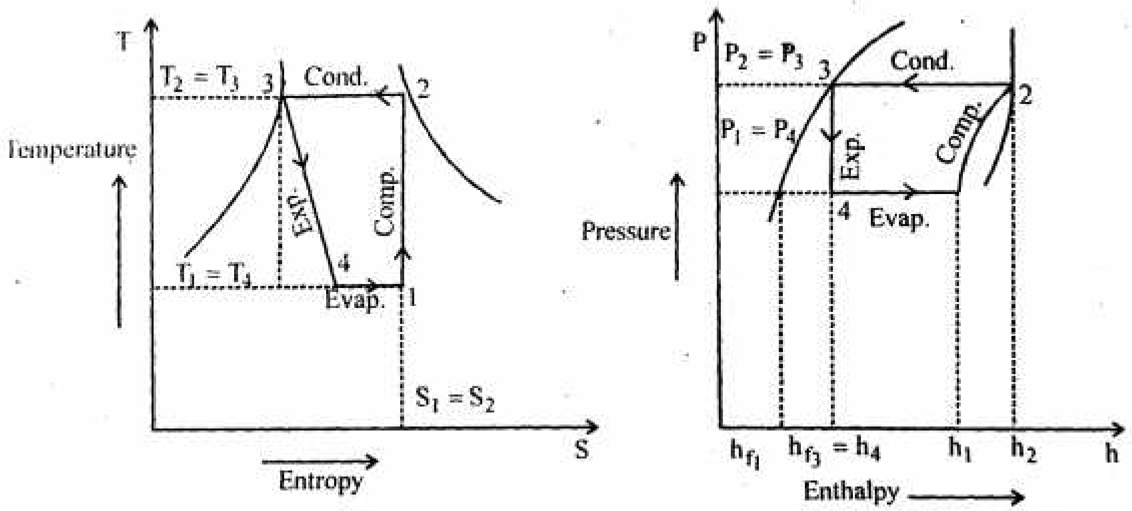
1. Cycle with dry saturated vapour after compression.
2. Cycle with superheated vapour after compression.
3. Cycle with superheated vapour before compression.
4. Cycle with superheated vapour before compression.
5. Cycle with undercooling or subcooling of refrigerant.

Theoretical vapour compression cycle with dry saturated vapour after compression :

1. Compression process : The vapour refrigerant at low pressure P_1 and temperature T_1 is compressed isentropically to dry saturated vapour as shown by the vertical line 1–2 on T-S diagram and by the curve 1-2 on p-h diagram. The pressure and temperature rises from P_1 to P_2 and T_1 to T_2 respectively.

The work done during isentropic compression for kg., of refrigerant is given by,

$$w = h_2 - h_1$$



h_1 = Enthalpy of vapour refrigerant at temperature T_1 i.e., at suction of the compressor and

μ_2 = Enthalpy of the vapour refrigerant at temperature T_2, i.e., at discharge of the compressor.

2. Condensing process : The high pressure and temperature vapour refrigerant from the compressed is passed through the condenser where ti is completely condensed at constant pressure P_2 and temperature T_2, as shown by the horizontal line 2-3 on T-S and p-h diagrams. The vapour refrigerant in changed into liquid refrigerant. The refrigerant, while passing through the condenser, gives its latent heat to the surrounding condensing medium.

3. Expansion process : The liquid refrigerant at pressure $P_1 = P_2$ and temperature $T_3 = T_2$ is expanded by throttling process through the expansion value to a low pressure $P_4 = P_1$ and temperature $T_4 - T_1$, as shown by the curve 3-4 on T-S diagram and by the vertical line 3-4 on p-h diagram. We have already discussed that some of the liquid refrigerant evaporator as it passes through the expansion valve, but the greater portion is vaporised in the evaporator. We know that during the throttling process, no heat in absorbed or rejected by the liquid refrigerant.

4. Vapourising process : The liquid-vapour mixture of the refrigerant at pressure $P_4 = P_1$ and temperature $T_4 = T_1$ is evaporated and changed into vapour refrigerant at constant pressure and temperature into vapour refrigerant at constant pressure and temperature, as shown by the horizontal the 4-1 on T-S and P-h diagrams. During evaporation, the liquid-vapour refrigerant absorbs its latent heat of vaporisation from the medium which is to be cooled. This heat which is absorbed by the refrigerant is called refrigerant effect and it is briefly written as R_E. The process of vaporisation continues upto point 1 which is the starting point and thus the cycle is completed.

We know that the refrigerating effect or the heat absorbed or extracted by the liquid-vapour refrigerant during evaporation per kg. of refrigerant is given by

$$R_L = h_1 - h_4 = h_1 = h_{f_3} \quad \left[\because h_{f_3} = h_4\right]$$

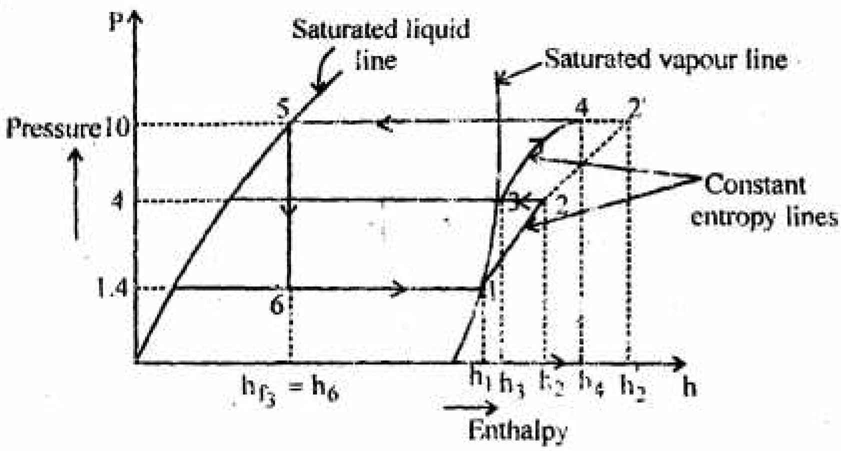
Where, h_{f_3} = Sensible heat at temperature T_3 i.e. enthalpy of liquid refrigerant clearing the condenser.

It may be noticed from the cycle that the liquid-vapour refrigerant has extracted heat during evaporation and the work will be done by the compressor for isentropic compression of the high pressure and temperature vapour refrigerant,

$$\therefore \quad \text{C.O.P.} = \frac{\text{Refrigeranting effect}}{\text{Work done}} = \frac{h_1 - h_4}{h_2 - h_1} = \frac{h_1 - h_{f_3}}{h_2 - h_1}.$$

Q. 3. (b) Calculate the power required to compress 20 Kg/min of NH_3 from saturated vapour at 1.4 bar to condensing pressure of 10 bar by a two stage compression with flash intercooling by liquid refrigerant at 4 bar. Assume the refrigerant leaves evaporator as saturated vapour and leaves condenser as saturated liquid. Also find out the load in evaporator in tonnes of refrigeration. Calculate power required without multistaging.

Ans.



$$m_1 = 20\,kg/min.,\ P_E = 1.4\,bar,\ P_C = 10\,bar,\ P_2 = P_3 = 4\,bar$$

The p-h diagram for a two stage compression with intercooling by liquid refrigerant is shown in figure. The various values of ammonia as read from the p-h diagram are as follows :

Enthalpy of saturated vapour refrigerant entering the law pressure compressor at point 1,

$$h_1 = 1400\,kJ/kg$$

Entropy of saturated vapour refrigerant entering the low pressure compressor at point 1,

$$S_1 = 5.75\,kJ/kg\,k$$

Enthalpy of superheated vapour refrigerant leaving the low pressure compressor at point 2,

$$h_2 = 1527\,kJ/kg$$

Enthalpy of saturated vapour refrigerant leaving the intercooler or entering the high pressure compressor at point 3,

$$h_3 = 1428\,kJ/kg$$

Entropy of saturated vapour refrigerant leaving the intercooler or entering the high pressure compressor at point 3,

$$S_3 = 5.39\,kJ/kg.k$$

Enthalpy of superheated vapour refrigerant leaving the high pressure compressor at point 4,

$$h_4 = 1550\,kJ/kg$$

Enthalpy of saturated liquid refrigerant passing through the condenser at point 5,

$$h_{f_5} = h_6 = 284\,kJ/kg$$

We know that mass of refrigerant passing through the condoners (or high pressure comrpessor),

$$m_2 = \frac{m_1\left(h_2 - h_{f_5}\right)}{h_3 - h_{f_5}} = \frac{20(1527-284)}{1428-284} = 21.73\,kg/min.$$

Work done in low pressure compressor,

$$w_L = m_1(h_2 - h_1) = 20(1527-1400) = 2540\,kJ/min.$$

Work done in high pressure comrpessor,

$$w_H = m_2(h_4 - h_3) = 21.73(1550-1428) = 2651\,kJ/min.$$

And total work done in both the compressors,

$$w = w_L + w_H = 2540 + 2651 = 5191\,kJ/min.$$

$\therefore$ Power needed $= 5191/60 = 86.5$ kW

Power needed when intercooling is not employed : When intercooling is not employed, the compression of refrigerant will follow the path 1-2 in the low pressure compressor and 2-2' in the high pressure compression. In such as case,

Work done in high compressor,

$$w_H = m_1(h_2' - h_2) = 20(1676 - 1527) = 2980 \text{ kJ/min.} \quad [\text{From p-n diagram, } h_2' = 1676 \text{ kJ/kg}]$$

And total workdone in both the compressors,

$$w = w_L + w_H = 2540 + 2980 = 5520 \text{ kJ/min.}$$

$\therefore$ Power needed $= 5520/60 = 92$ kW.

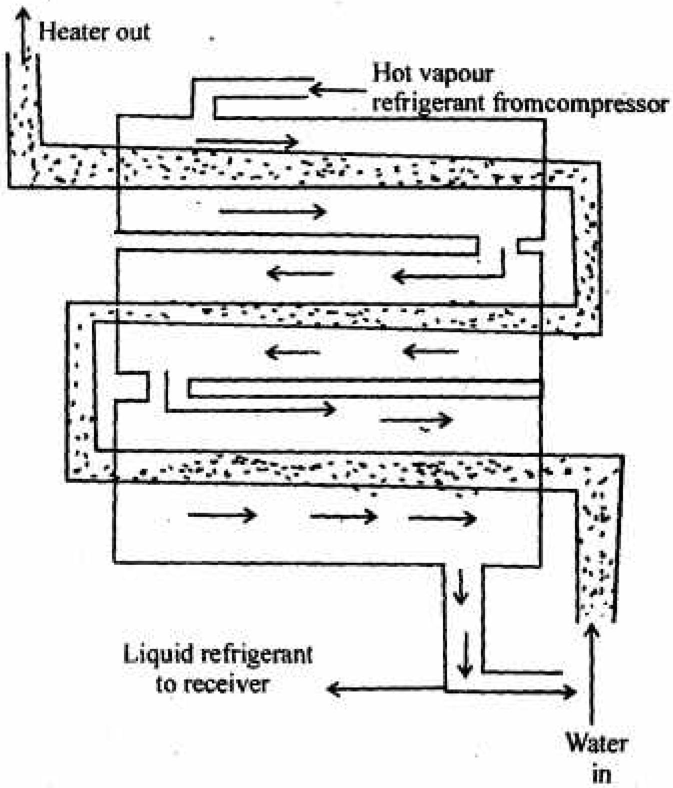
Q. 4. Explain the following with neat diagram :

(a) Double pipe water cooled condenser

(b) Flooded evaporator

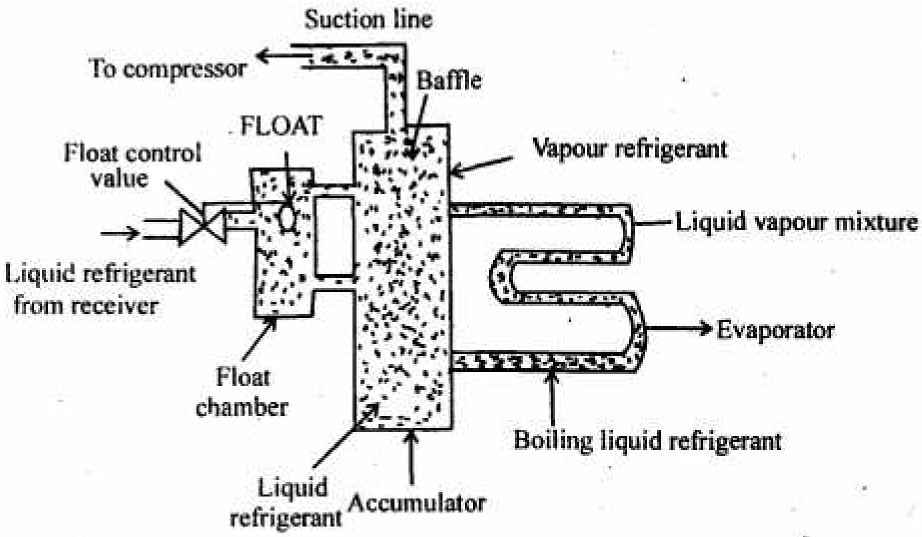
(c) Thermostatic expansion valve.

Ans. (a) Double pipe water cooled condenser :



It consists of a water tube inside a large refrigerant tube. In this type of condenser, the hot vapour refrigerant enters at the top of condenser. The water absorbs the heat from the refrigerant and the condensed liquid refrigerant flows at the bottom. Since the refrigerant flows at the bottom. Since the refrigerant tubes are exposed to ambient air, therefore some of the heat is also absorbed by ambient air by natural convection.

(b) Flooded evaporator :



In a flooded evaporator, a constant liquid refrigerant is always maintained. A float control valve is used as an expansion device which maintains constant liquid level in the evaporator. The liquid refrigerant from the receiver passes through a low side float control valve and accumulator before entering the evaporator coil. The accumulator serves as a storage tank for the liquid refrigerant. It maintain a constant liquid level in the evaporator and helps to separate the liquid refrigerant from the vapour returning to the compressor. Due to the heat supplied by the substance to be cooled, the liquid refrigerants in the evaporator coil vaporisen and thus the liquid level falls down. The accumulator supplies more liquid to the evaporator in order to keep the liquid refrigerant in the evaporator at proper level. In this way, the level of liquid refrigerant in the accumulator also falls down. Since the float within the float chamber rests on liquid refrigerant at the same level as that in the accumulator, therefore the float also falls down and open the float valve. Now the liquid refrigerant from the receiver is admitted into the accumulator. As the liquid level in the accumulator rises and reaches to the constant leave, the float also rises with it until the float control valve closes.

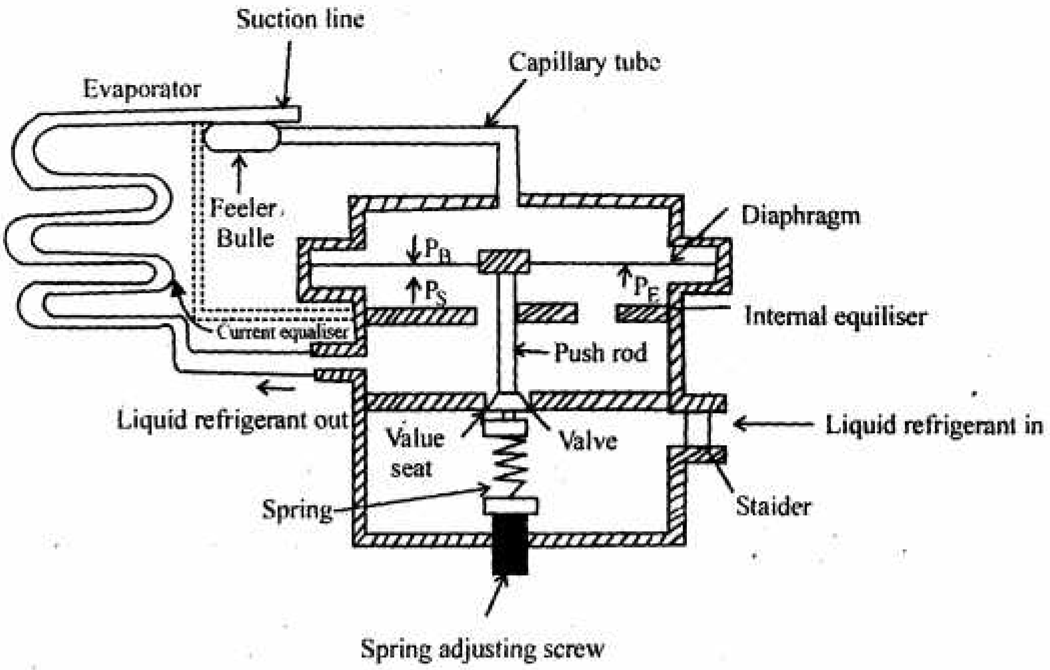
Since the evaporator is almost completely filled with liquid refrigerant, therefore the vapour refrigerants from the evaporator is not superheated but it is in a saturated condition. In order to prevent liquid refrigerant to entire into the compressor, an accumulator is generally used with the folded evaporations. The liquid refrigerant trapped in the accumulator is re-circulated through the evaporator.

(c) Thermostatic expansion valve : The thermostatic expansion valve consists of a needle valve and a seat, a metallic diaphragm, spring and an adjusting screw. In addition to this, it has a feeder or thermal bulb which is mounted on the suction line near the outlet of the evaporator coil. The feeler bulb is partly filled with the same liquid refrigerant as used in the refrigeration system. The opening and closing of the valve depends upon the following forces acting on the diaphragm :

1. The spring pressure (P_s) acting on the bottom of the diaphragm.

2. The evaporator pressure (P_E) acting on the bottom of the diaphragm.

3. The feller bulb pressure (P_B) acting on the top of the diaphragm.



Q. 5. The following data refers to steam jet refrigeration system. The steam supplied to the nozzle = 7 bar (abs) and dry evaporator (Flash chamber) temperature = 4°C, temperature of make up water to flash chamber = 17°C, condenser pressure 0.06 bar, Nozzle efficiency = 0.9, entrainment efficiency = 0.6, compression efficiency = 0.7. Determine the following :

(a) Mass of motive steam/Kg of flash vapour.

(b) Refrigeration capacity/Kg of flash chamber.

(c) Mass flow rate of motive steam per ton of refrigeration.

(d) Volume of vapour leaving the flash chamber/ton refrigeration.

(e) Dryness fraction of steam from flash chamber. Assume the quality of steam at the entry to the thermo-compressor = 0.92.

Ans. $P_B = 7\ \text{bar}$

$t_W = 4°C$

$t_{mw} = 17°C$

$P_C = 0.06\ \text{bar}$

$\eta_V = 90\% = 0.9$

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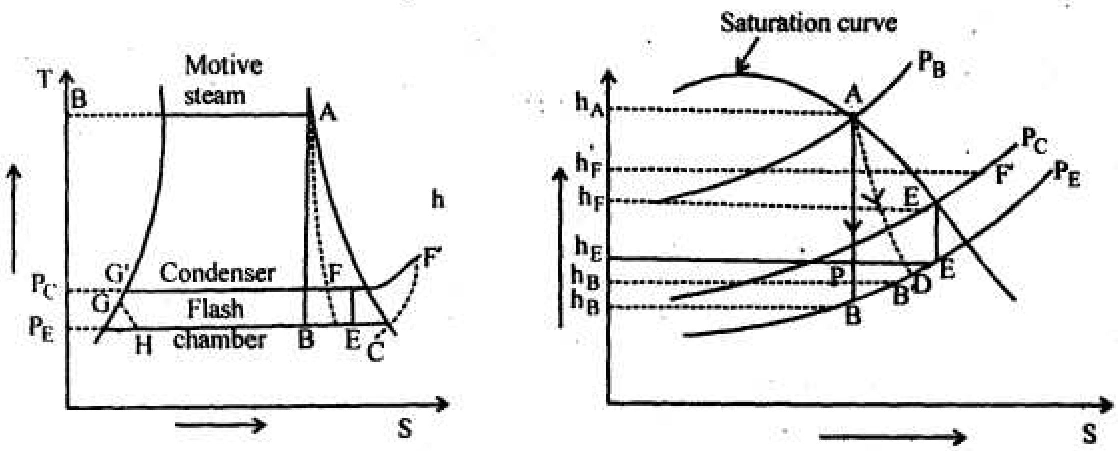
$\eta_C = 0.7$

$x_E = 0.92$

The T-S and h-s diagram for the steam electro refrigeration system is shown in the figure.



From steam taken of dry saturated steam, corresponding to a pressure of 7 bar, we find that

$h_A = 2762\ \text{kJ/kg}$

$S_A = 6.705\ \text{kJ/kg}$

$t_A = 165°C$

And corresponding to a temperature of 4°C, we find that

$$h_{f_B} = 18.9 \text{ kJ / kg.}$$

$$h_{f_{gB}} = 2490.9 \text{ kJ / kg}$$

$$S_{fB} = 0.0685 \text{ kJ / kgk}$$

$$S_{f_{gB}} = 8.9715 \text{ kJ / kg.K}$$

First of all, let us find the dryness fraction of the steam at point. $B(\text{i.e.} x_B)$.

We know that for isentropic expansion AB,

Entropy before expansion (S_A) = Entropy after expansion (S_B)

$$6.705 = S_{fB} + x_B \times S_{f_{gB}} = 0.0685 + x_B \times 8.9715$$

$$\therefore \quad x_B = \frac{6.705 - 0.0685}{8.9715} = 0.74$$

And enthalpy at B,

$$h_B = h_{fB} + x_B \times h_{f_{gB}} = 18.9 + 0.74 \times 2490.9$$

$$= 1862.16 \text{ kJ / kg}$$

We know that nozzle efficiency, η_n

$$0.9 = \frac{h_A - h_B'}{h_A - h_B} = \frac{2762 - h_B'}{2762 - 1862.16}$$

$$\therefore \quad h_B' = 2762 - 0.90(2762 - 1862.16) = 1970.14 \text{ kJ / kg}$$

Now, let us find the dryness fraction of steam at point B' (i.e. x_B'). Since the points B, B', D and E lie on the some pressure line, therefore,

$$h_{fB} = h_{fB'} = h_{fD} = h_{fE} = 18.9 \text{ kJ / kg}$$

$$\& \quad h_{f_{gB}} = h_{f_{gB'}} = h_{f_{gE}} = 2490.9 \text{ kJ / kg}$$

We know that enthalpy at B',

$$h_B' = h_{fB'} + x_B' + h_{f_{gB'}}$$

$$1970.18.9 + x_B' \times 2490.9$$

$$\therefore \quad x_B' = \frac{1970.14 - 18.9}{2490.9} = 0.78$$

Let h_D = enthalpy of steam at D &

x_D = dryenss fraction of steam at D

We know that entrainment efficiency (η_E),

$$0.60 = \frac{h_A - h_D}{h_A - h_B'} = \frac{2762 - h_D}{2762 - 1970.14}$$

$$\therefore \quad h_D = 2762 - 0.60(2762 - 1970.14) = 227.3\,\text{kJ/kg}$$

We also know that enthalpy at point D (h_D),

$$2247.3 = h_{fD} \times x_D \times h_{f_{gD}} = 18.9 + x_D \times 2490.9$$

$$\therefore \quad x_D = \frac{2247.3 - 18.9}{2490.9} = 0.894$$

Enthalpy at point E,

$$h_E = h_{f_E} + x_E \times h_{fgE} = 18.9 + 0.92 \times 2490.9$$

$$= 2310.5\,\text{kJ/kg}$$

Now let us find the dryness fraction of the mixture of the motive steam and water vapour after isentropic compression at point F,

Let x_F = Dryness fraction at point F.

We know that entropy at point E,

$$S_E = S_{fE} \times x_E \times S_{f_{gE}} = 0.0685 + 0.92 \times 8.9715$$

$$= 8.3223\,\text{kJ/kg K}.$$

From the steam tables, corresponding to a condenser pressure of 0.06 bar, we find that

$$h_{fF} = 148.86\,\text{kJ/kg},$$

$$h_{f_{gE}} = 2417.5 \text{kJ/kg}$$

$$S_{f_E} = 0.512 \text{kJ/kg k m}$$

$$S_{f_{gE}} = 7.831 \text{kJ/kg.k}$$

Since the compression of the mixture is isentropic, therefore entropy before compression (S_{fC}) = Entropy after compression (S_F).

$$8.2332 = S_{gF} \times x_F \times S_{f_{gF}} = 0.512 + x_F \times 7.831$$

$$x_F = \frac{8.3223 - 0.512}{7.831} = 0.997$$

We know that at point F,

$$h_F = h_{f_F} + x_F \times h_{f_{gF}} = 148.86 + 0.997 + 2417.5 = 2559.1 \text{kJ/kg}$$

We also know that compression efficiency, (η_c),

$$0.7 = \frac{h_F - h_E}{h_F' - h_E} = \frac{2559.1 - 2310.5}{h_F' - 2310.5}$$

$$\therefore \quad h_F' = \frac{2559.1 - 2310.5}{0.7} + 2310.5 = 2621.2 \text{kJ/kg}$$

1. Mass of motive steam required per kg. of the flash vapour we know that mass of motive steam required per kg. of the flash vapour,

$$\frac{m_s}{m_v} = \frac{h_F - h_E}{(h_A - h_B)\eta_N \eta_E \eta_C - (h_F - h_E)}$$

$$= \frac{2559.1 - 2310.5}{(2762.1862.16)0.90 \times 0.60 \times 0.70 - (2559.1 - 2310.5)}$$

$$= \frac{248.6}{411.8 - 248.6} = 1.523 \text{kg/kJ of flash vapour.}$$

2. Quality of vapour flashed from the flash chamber.

Let x_c = Dryness fraction of the vapour flashed from the flash chamber.

First all. let us find enthalpy at point C, we know that,

$$m_v h_c + m_s h_D = (m_s + m_v) h_E$$

$$h_c + \frac{m_s}{m_v} \times h_D = \left[\frac{m_s}{m_v} + 1\right] h_E$$

$$h_c + 1.523 \times 2247.3 = (1.523 + 1) 2310.5$$

$$h_c + 3422.6 = 5829.4$$

$$\therefore \quad h_c = 2406.8 \text{ kJ/kg}$$

We also know that enthalpy at point (h_c),

$$2406.8 = h_{f_c} + x_c + h_{fB} = 18.9 + x_c \times 2490.9$$

$$x_c = \frac{2406.8 - 18.9}{2490.9} = 0.96$$

3. Refrigerating effect per kg of flash vapour.

We know that refrigerating effect per kg. of flash vapour,

$$R_E = h_c - h_{fG} = 2406.8 - 75.5 = 2331.3 \text{ kJ/kg}$$

4. Mass of motive steam required per hour per tonne of refrigeration.

We know that mass of motive steam required per hour per tonne of refrigeration.

$$= \frac{210\theta}{h_c - h_{fG}} \times \frac{m_g}{m_v} = \frac{210 \times 1}{2406.8 - 75.6} \times 1.523$$

$$= 0.133 \text{ kg/min/TR}$$

$$= 0.133 \times 60 = 7.98 \text{ kg}/\mu/\text{TR}$$

5. Volume of vapour removed from the flash chamber per have per tonne of refrigeration.

We know that volume of vapour (per kg.) removed from the flash chamber,

V_c = Volume of liquid at $c + x_c$ (Volume of saturated vapour-volume of liquid)

$$= 1 + 0.95(152.22 - 1) = 144.66 \text{ m}^3/\text{kg}$$

$$= v_c \times \frac{210\theta}{h_c - h_{pG}} \times 60 = 144.66 \times \frac{210 \times 1}{2406.8 - 75.5} \times 60$$

$$= 782\,m^3/h/TR$$

6. Coefficient of performance of the system.

From he steam tables, corresponding to a condenser pressure (0.06 bar), we find that enthalpy of liquid at point G'.

$$h_{fG'} = 148.8\,kJ/kg$$

We know that coefficient of performance of the system,

$$C.O.P. = \frac{m_v(h_c - h_{fG})}{m_s(h_A - h_{fG'})} = \frac{1(2406.8 - 75.5)}{1.523(2762 - 148.8)} = 0.586.$$

Q. 6. (a) The DBT and WBT of atmospheric air are 35°C and 23°C respectively when the barometer reads 750 mm of Hg. Determine :

(i) Relative humidity, (ii) Humidity ratio, (iii) dew point temperature, (iv) density, (v) enthalpy of air. Without the help of psychometric chart.

Ans.

$$t_d = 35°C,$$

$$t_w = 23°C,$$

$$P_b = 750\,mm \text{ of Hg}.$$

1. Dew Point Temperature :

From steam tables, we find that saturation pressure corresponding to wet bulb temperature 23°C is,

$$P_w = 0.02337\,bar$$

We know that barometer pressure,

$$P_b = 750\,mm \text{ of Hg}.$$

$$= 750 \times 133.3 = 98642\,N/m^2$$

$$= 0.98642\,bar$$

∴ Partial pressure of water vapour,

$$P_v = P_w - \frac{(P_b - P_w)(t_d - t_w)}{1544 - 1.44\,t_w}$$

$$= 0.02337 - \frac{(0.98642 - 0.02337)(35 - 23)}{15440 - 1.44 \times 23}$$

$$= 0.02337 - 0.00636 = 0.01701 \text{ bar}$$

Since the dew point temperature is the saturation temperature corresponding to the partial pressure of water vapour (P_v), therefore from steam tables, we find that corresponding to a pressure of 0.01701 bar, the dew point temperature is,

$$t_{dP} = 15°C.$$

2. Relative humidity : From steam tables, we find that the saturation pressure of vapour corresponding to dry bulb temperature 35°C is,

$$P_s = 0.04242 \text{ bar}$$

We know relative humidity,

$$\phi = \frac{P_v}{P_s} = \frac{0.01701}{0.04242} = 0.40 \text{ or } 40\%$$

3. Specific humidity : We know that specific humidity

$$w = \frac{0.622 P_v}{P_b - P_v} = \frac{0.622 \times 0.01701}{0.98642 - 0.01701}$$

$$= \frac{0.01058}{0.96941} = 0.010914 \text{ kg/kg of dry air.}$$

$$= 10.914 \text{ g/kg of dry air.}$$

4. Degree of saturations : We know that specific humidity of saturated air,

$$w_s = \frac{0.622 P_s}{P_b - P_s} = \frac{0.622 \times 0.04242}{0.98642 - 0.04242}$$

$$= \frac{0.02638}{0.944} = 0.02745 \text{ kg/kg of dry air.}$$

We know that degree of saturations,

$$\mu = \frac{w}{w_s} = \frac{0.010914}{0.027945} = 0.391 \text{ or } 39.1\%$$

5. Vapour density : We know that vapour density

$$\rho_v = \frac{w(P_b - P_v)}{R_a T_d} = \frac{0.010914(0.98642 - 0.01701) \times 10^5}{287(273 + 30)}$$

$$= 0.01216 \text{kg/m}^3 \text{ of dry air.}$$

6. Enthalpy of mixture per kg of dry air.

From steam tables, we find that the latent heat of vapoirsation of water at dew point temperature of 15°C is,

$$h_{fgdp} = 2466.1 \text{kJ/kg}$$

∴ Enthalpy of mixture per kg of dry air,

$$h = 1.022 t_d + w\left[h_{fgdp} + 2.3 t_{dp}\right]$$

$$= 1.022 \times 30 + 0.010914[2466.1 + 2.3 \times 15]$$

$$= 30.66 + 27.29 = 57.95 \text{kJ/kg of dry air.}$$

Q. 6. (b) Air at 31°C DBT and 18.5°C WBT is passed through a cooling coil maintained at 4.4°C. The heat extracted by the cooling coil from air is 12.5 KW and air flow rate is 39.5m^3/min. Determine DBT and WBT of the air leaving the coil and coil bypass factor.

Ans. Refer to fig in psychrometry chart

At the apparatus dew point,

$$w_s = 5.25 \text{g/kgd.a}$$

$$h_s = 17.7 \text{kJ/kgd.a}$$

State of entering air,

$$w_1 = 8.2 \text{g kg/d.a}$$

$$v_1 = 0.872 \text{m}^3/\text{kgd.a}$$

$$h_1 = 52.5 \text{kJ/kgd.a}$$

Mass flow rate of dry air

$$\dot{m}_a = \frac{\dot{Q}_v}{v} = \frac{39.6}{0.872} = 44.41 \text{kgd.a/min}$$

Cooling load per kg of dry air,

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

Enthalpy of air leaving the coil,

$$h_2 = 52.5 - 16.89 = 35.61 kJ / kg d.a$$

Equation for hte condition line

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

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

We get $$w_2 = 6.77 g.w.v / kg / d.a$$

Dry and wet bulb temperature of air leaving the coil for calculated values of h_2, w_2 from psychrometric chart,

$$t_2 = 18.6°C$$

$$t_2' = 12.5°C$$

Coil bypass factor

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

$$= \frac{35.61 - 17.7}{52.5 - 17.7} = 0.515 \text{ (very high).}$$

Q. 7. (a) An air conditioning system is to be designed for a small restaurant when the following data is available. Heat flow through walls, roof and floor = 22000 KJ/h. Solar heat gain through glass=7000 KJ/h, equipment semible and latent heat gain = 10,500KJ/h, 2500KJ/h respectively. Amount of fresh air supplied 1600m^3/h, Infiltratd load 400m^3/h, the hall seating capacity 50, Servants serving the meals 5, outside design conditions 35°C DBT and 26°C WBT Inside design conditions 27°C DBT and 55% RH.

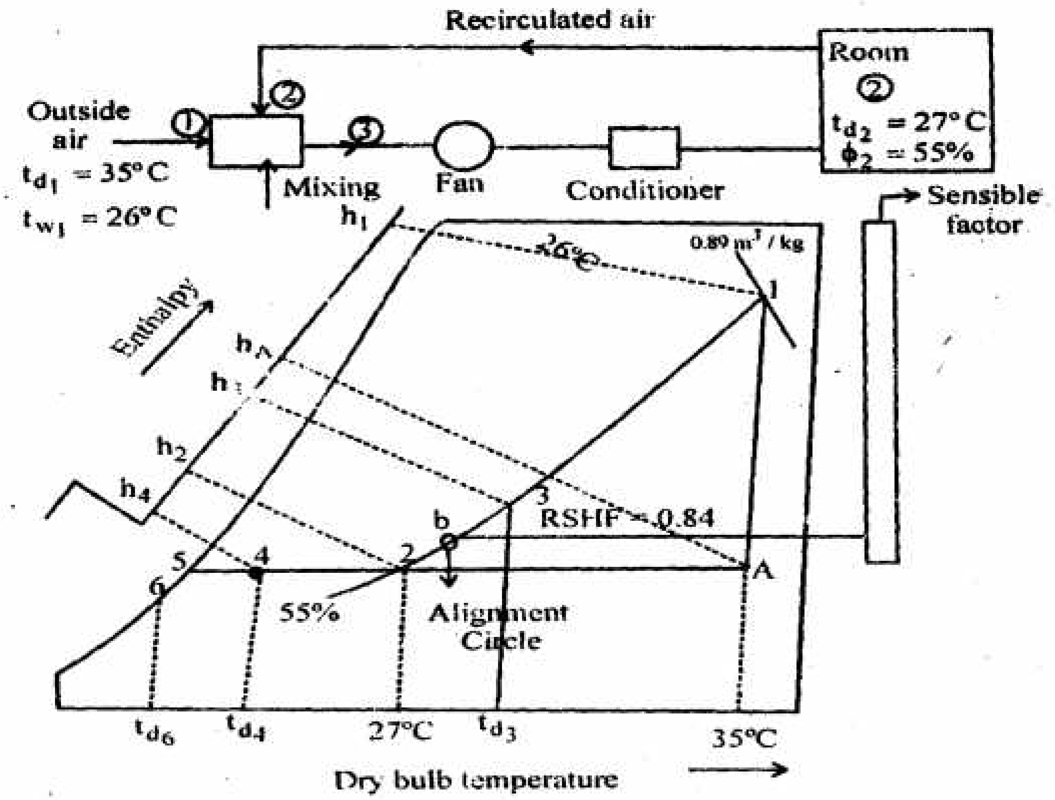
The temperature of air supplied to the dining hall should not fall below 17°C. The fan in the system is fixed before air-conditioning system. The power of the motor connecting the fan is 10kw. Find the following : (i) Amount of air delivered to the dining hall in m^3/h

(ii) Percentage of recirculated air

(iii) Refrigeration load on the cooling coil in tons of refrigeration (iv) ADP of the cooling coil and its bypass factor.

Ans. Given latent heat for or through walls, roofs floor

$$Q_{W,R,F} = 22000 kJ / hr$$



Solar heat through glass $= 7000\ kJ/hr = Q_{SG}$

Equipment sensible heat $= Q_{SE} = 10{,}500\ kJ/hr$

Equipment latent heat $= Q_{LE} = 2500\ kJ/hr$

From the psychrometric chart we find that the specific volume of air at point 1

$V_{s_1} = 0.89\ m^3/kg$ of dry air seating capacity = 50

Servants serving meal = 5.

$$t_{d_1} = 35°$$

$$t_{w_1} = 26°C$$

$$t_{d_2} = 27°C$$

$$\phi_2 = 55\%.$$

Enthalpy of air at point 1

$$h_1 = 81\,kJ/kg \text{ of dry air}$$

Enthalpy of air at point 2

$$h_2 = 57\,kJ/kg \text{ of dry air}$$

And enthalpy of air at point A

$$h_A = 68\,kJ/kg \text{ of dry air}$$

We know that mass of infiltrated air at point 1

$$m_1 = \frac{v_1}{v_{s_1}} = \frac{400}{0.89} = 449.43\,kg/min.$$

$\therefore$ Sensible heat gain due to infiltrated air

$$= m_1(h_A - h_2) = 449.43(68 - 57)$$

$$= 4943.82\,kJ/kg$$

And latent heat gain due to infiltrated air

$$= m_1(h_1 - h_A)$$

$$= 449.43(81.68) = 5842.59\,kJ/hr$$

Total sensible heat gain in the room

$$RSH = 22000 + 7000 + 10500 + 4943.82$$

$$= 44443.82\,kJ/hr$$

Total latent heat gain in the room

$$RLH = 2500 + 5842.59$$

$$= 8342.59\,kJ/hr$$

$\therefore$ Room sensible heat factor

$$RSHF = \frac{RSH}{RSH + RLH} = \frac{44443.82}{44443.82 + 8342.59}$$

From the psychrometric chart, we can find that dry bulb temperature of air at point 3

$$t_{d_3} = 29°C$$

And enthalpy of air at point 3

$$h_3 = 58kJ / kg \text{ of dry air.}$$

Let for this problem the BPF (By-Pass-Filter) be 0.2.

$$0.2 = \frac{t_{d_4} - t_{d_6}}{t_{d_3} - t_{d_6}} = \frac{t_{d_4} - t_{d_6}}{29 - t_{d_6}}$$

By trail& error method, we find that

$$t_{d_4} = 15°C \ \& \ t_{d_6} = 10.9°C$$

Enthalpy at point 4

$$h_4 = 37kJ / kg \text{ of dry air}$$

& specific volume of air at point 4

$$V_{s_4} = 0.82m^3 / kg \text{ of dry ari}$$

(i) Amount of air delivered to the dining hall in m^3 / hr

Let V_a = Amount of total air required in m^2 / hr

$$\text{Total air required } m_a = \frac{\text{Total room heat}}{\text{Total heat removed}}$$

$$m_a = \frac{RSH + RLH}{h_2 - h_4} = \frac{44443.82 + 8342.59}{57 - 37}$$

$$= 2639.32 \, kg / hr$$

$$V_a = m_a \times v_{s_4} = 2639.32 \times 0.82 = 2164.24 \, m^3 / hr$$

(ii) Percentage of recirculated air

Mass of air supplied $\quad m_F = \frac{v_F}{v_{s_1}} = \frac{1600}{0.82} = 1951.22 \, kg / hr$

Mass of recirculated air $\quad = m_a - m_F$

$$= 2639.32 - 1951.22 = 688.10 \, kg / hr$$

Percentage of recirculated air, $= \frac{688.10}{2639.32} = 26.07\%$

(iii) Refrigeration load on the cooling coil in TR.

$\therefore$ Refrigeration load on the coil.

$$= m_a(h_3' - h_4) + \text{heat added by motor}$$

$$= 2639.32(58 - 37) + 10$$

$$= \frac{25.39}{3.5} \text{kw} = 7.25\text{TR}$$

(iv) ADP : We already calculated the apparatus dew point temperature of the cooling coil.

$$t_{d_4} = 15°C \ \& \ t_{d_6} = 10.9°C$$

By-pass factor (BPF) = 0.2

Q. 8. Discuss the following :

(a) Duct system design,

(b) Design of summer air conditioning.

Ans. (a) Duct system design : The object of duct design is to determine the dimensions of all ducts in the given system. The ducts should carry the necessary volume of conditional air from the fan outlet to the conditioned space within minimum frictional and dynamic losses. The duct layout must be made so as to reach the outlet without least number of bends, obstruction and area changes. The area changes must be gradual where possible and limited to not more than 20° for diverging area and 60° for converging area. For rectangular ducts, the aspect ratio of 4 and less is desirable but it should not be greater than in any case. The minimum sheet metal is required with square cross-section for given cross-sectional area.

The velocities in the ducts must be high enough to reduce +ve size of the duds but it should be low enough to reduce the noise and pressure losses to economic power requirement.

After the layout of the duct is decided and the requirements of air quantities at varies outlets are known, then the size of the ducts may be obtained.

(b) Design of summer air conditioning : It is the most important type of air conditioning, in which the air is cooled and generally humified.

The outside air flows through the damper and mixes up with recirculated air. The mixed air passes through a filter to remove dirt, dust and outer impurities. The air now passes through a cooling coil. The coil has a temperature much below the required dry bulb temperature of the air in the conditioned space. The cooled air passes through a perporated membrane and leases its moisture in the condensed from which is collected in a sump. After that, the air is made to also through a heating coil which heats up air slightly. This done to being the air to the designed dry bulb temperature and relative humidity.

Now the conditioned air is supplied to the conditioned space by a fan. From the conditioned space, a part of the used air is exhausted to the atmosphere by the exhaust fans or ventilations.