B. E.

Seventh Semester Examination, Dec-2006

REFRIGERATION AND AIR-CONDITIONING

Note : Attempt any five questions.

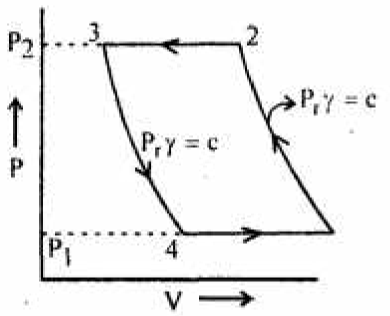
Part-A

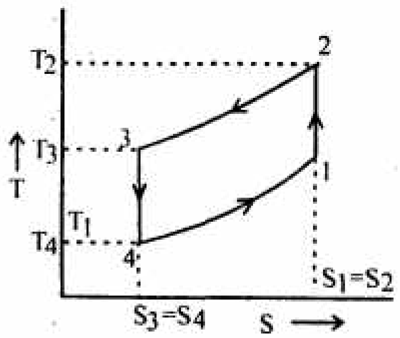
Q. 1. (a) What are the advantages of air refrigeration? Derive an expression for COP of an air refrigerator working on Bell Coleman cycle.

Ans. Advantages of Air-Refrigeration : The following advantage can obtain as :

* Air-available always.
* No-cost.
* Not required extra container or cylinder.
* Not pollutant.
* Ozone friendly.
* No operating cost.

Reverse brayton cycle or bell-coleman cycle :





Work done during the cycle = Heat rejected-heat absorbed

$$c_p(T_2 - T_3) - c_p(T_1 - T_4)$$

Heat absorbed $= c_p(T_1 - T_4)$

$$COP = \frac{\text{Heat absurbed}}{\text{Workdone}} = \frac{c_p(T_1 - T_4)}{c_p(T_2 - T_3) - c_p(T_1 - T_4)}$$

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

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

We know that for isentropic process 1–2.

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

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

Since $P_2 = P_3$ and $P_1 = P_4$

$$\frac{T_2}{T_1} = \frac{T_3}{T_4} \text{ or, } \frac{T_2}{T_3} = \frac{T_1}{T_4}$$

So, $$COP = \frac{T_4}{T_3 - T_4} = \frac{1}{\frac{T_3}{T_4} - 1}$$

$$= \frac{1}{\left(\frac{P_3}{P_4}\right)^{\frac{r-1}{r}} - 1} = \frac{1}{\left(\frac{P_2}{P_1}\right)^{\frac{r-1}{r}} - 1} = \frac{1}{(rp)^{\frac{r-1}{r}} - 1}$$

$$w_1 = \frac{n}{n-1}(p_2v_2 - p_1v_1) = \frac{n}{n-1}(RT_2 - RT_1)$$

$$w_2 = \frac{n}{n-1}(p_2v_2 - p_1v_1) = \frac{n}{n-1}(RT_3 - RT_4)$$

$$\text{Network} = \frac{n}{n-1}\{R[T_2 - T_1] - (T_3 - T_4)\}$$

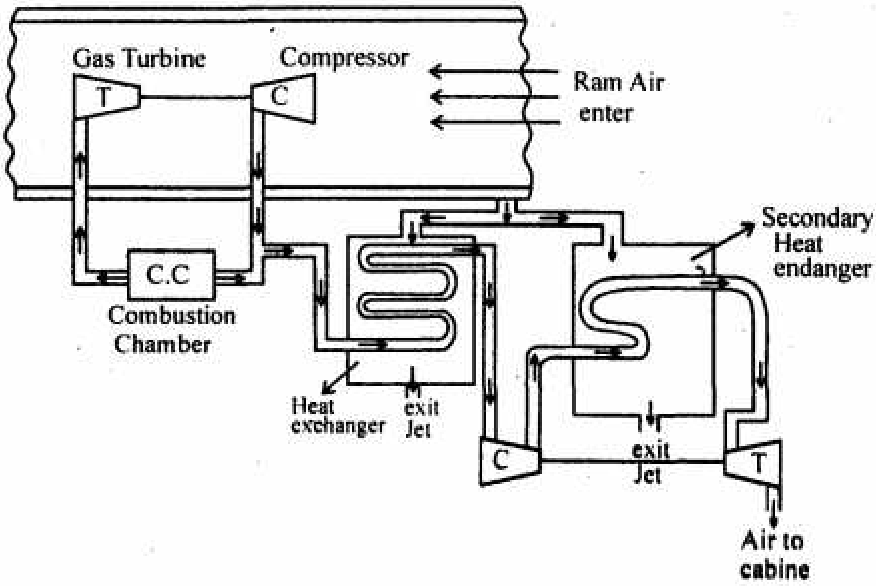
$$COP = \frac{c_p(T_1 - T_4)}{\frac{n}{n-1} \times R[(T_2 - T_1) - (T_3 - T_4)]}$$

$$R = c_p - c_v = c_v(r-1)$$

$$\boxed{COP = \frac{c_p(T_1 - T_4)}{\frac{n}{n-1} \times c_v(r-1)[(T_2 - T_1) - (T_3 - T_4)]}}$$

Q. 1. (b) Describe with a sketch a bootstrap cycle of air refrigeration system.

Ans. Boot-strap air cooling system :



This type of air-cooling system used in transport vehicles. Where two compression, two heat exchanges and two turbines installed.

The process 1–2 → Isentropic ramming process.

Process 2–2' → Isothermal process.

Process 1–2' → Actual ramming.

Process 2'–3 → Ideal compression.

Process 3'–2' → Actual compression process.

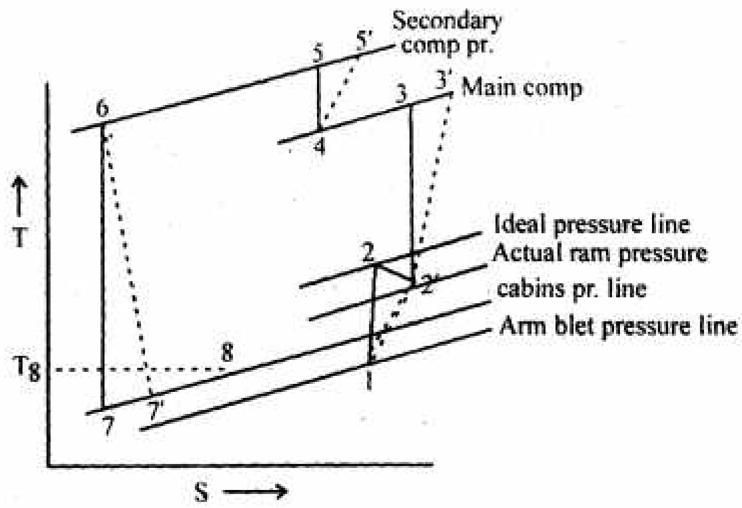
Process 3'–3°–4 → Heat exchange.

Process 4–5–5' → Ideal/Actual secondary comp.

Process 5–5'–6 → Secondary heat exchanging.

Process 6–7 → Expanding isontropic.

Process 7–8 → Cabine supply



Quantity of air requires m_a.

$$m_a = \frac{210Q}{c_p(T_8 - T_7)} \text{ kg / min}$$

Where Q = Tonnes of refrigeration.

Power required for refrigerating system.

$$p = \frac{m_a c_p (T_3' - T_2')}{60} \text{ k}\omega$$

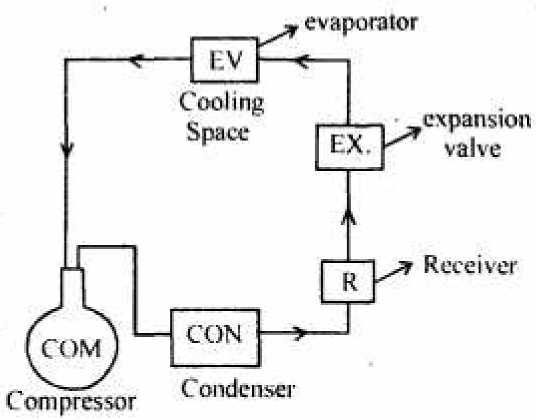
$$COP = \frac{210Q}{m_a c_p (T_3' - T_2')}$$

Q. 2. (a) Discuss the effect of operating conditions on the performance of Vapor compression refrigeration system.

Ans. Vapour compression refrigeration system : It is the close type system which contain the circulation of refrigerants (very low boiling temp), does not leave the system and it circulated in entire cycle regularly.

In this refrigeration system, the refrigerant. Take the latent heat of cooling space and delivered it in the condenses. Where the water regularly easing this above heat or through air.

Mechanism :



The component used.

Compressor : It take refrigerant from low pressure from output of evaporation and after comp it out the high pressure and temperature.

Condenser : Its cool the refrigerant and deliver to capillary.

Capillary : It expand the refrigerant to, temp. and pressure decrease at very low.

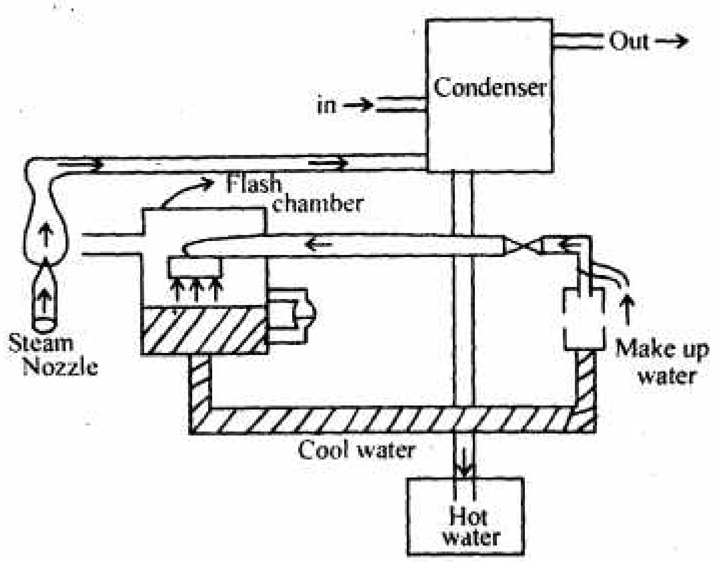
Evaporator : This space is belongs to where requires maintain temperature.

Advantage :

* Its capacity/performance batter.
* It is compact size.
* It has less running cost.
* It can be implode large range of temperature.
* COP is quit high.

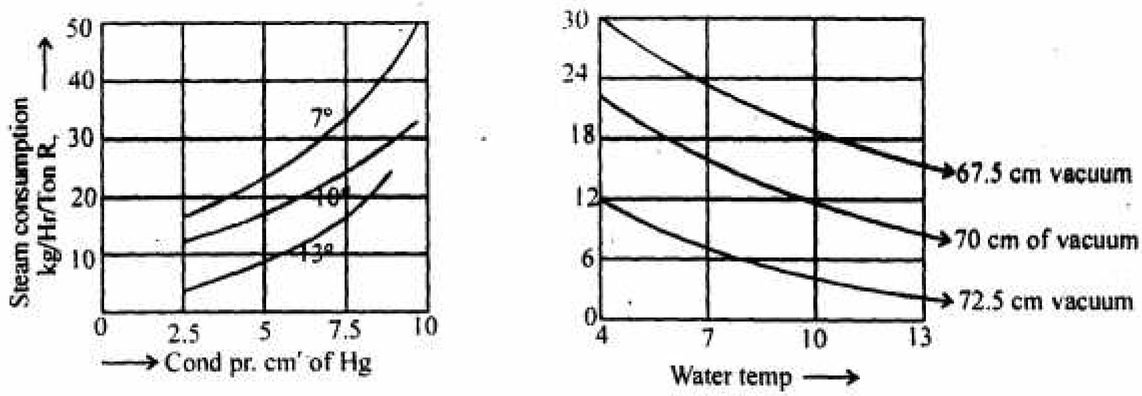
Q. 2. (b) Explain with a neat sketch the working of a steam jet refrigeration system. What are the advantages and disadvantages of this system over other types of refrigeration system?

Ans.



(Steam jet refrigeration system)

Performance : The system of steam jet as the "mass of steam required per tonnes of refrigerate on with the decrease in temperature of chilled water (evaporating temperature) or increase in pressure of condensing vapour.



Where Fig. (a) For comparing condenser Pr vs. steam consumption

Fig. (b) Chilled water temp v.s steam consumption

If the refrigeration load drop; the ref capacity of the steam jet unit should be reduced. This can be done by throttling the steam before entering the nozzle but this method is not practical because if make the operation of ejector unstable at low pressure of motive steam. At low loads the flow of steam through some nozzle is cut-off.

Q. 3. (a) A cascade refrigeration system, using R-22 and R-13 is required to produce 20 tonnes of refrigeration at −70°C. Heat is rejected in cascade condenser by R-13 at −5°C to R-22 at −15°C. The condensation in the water-cooled condenser is at 40°C. Assume simple saturated cycles for both the circuits. Determine :

(a) Pressure ratio and mass flow rate of each cascade.

(b) COP and piston displacement of each cascade

(c) COP of combined system.

Ans. Cascade refrigeration system used

R_{-22}, R_{-13}.

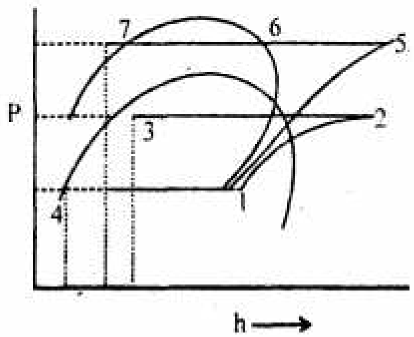
Capacity $C_1 = 20TR$

$T_1 = -70 + 273 = 203k$

Heat rejection

R_{13} at $273 - 5 = 268k = T_2$

R_{22} at $275 - 15 = 258k = T_3$



Condensing process at $40^C = T_4 = 273 + 46 = 313^k$

So,

$P_1 = 0.3745$ bar by steam table.

$h_1 = 223.7$

$s_1 = 1.054 \, kg / kgk^\circ$

$h_2 = 275 \, kJ / kg$

$$p_2 = 2.458 \text{ bar}$$

$$h_{f_7} = h_4 = 22.2 \text{ kJ/kg}$$

So, $$\frac{p_2}{p_1} = \frac{2.458}{0.3745} = 6.56$$

$$\frac{p_6}{p_5} = \frac{6.518}{1.044} = 6.24$$

$$m_1 = \frac{210Q}{h_1 - h_4} = \frac{210 \times 20}{223.7 - 22.2}$$

$$= 20.8 \text{ kg/min for } R_{22}$$

$$m_2 = \frac{m_1(h_2 - h_4)}{h_5 - h_8} = \frac{20.8(275 - 22.2)}{174.2 - 54.9}$$

For, R_{-13}

$$m_2 = 44.08 \text{ kg/min}$$

(b) COP for each unit of displacement,

$$R_{E_1} = m_1(h_1 - h_2) = 210 \times 20 = 4200 \text{ kJ/min for } R_{-22}.$$

For R_{-13} :

$$RE_2 = m_2(h_5 - h_8)$$

$$= 44.08(174.2 - 54.9)$$

$$= 5258 \text{ kJ/min}$$

$$W_1 = m_1(h_2 - h_1) = 20.8(275 - 223.7)$$

$$= 1067 \text{ kJ/min}$$

$$W_2 = 44.08(207 - 174.2) = 1446 \text{ kJ/min}$$

∴ COP for $$R_{22} = \frac{RE_1}{W_1} = \frac{4200}{1067}$$

$$\boxed{(COP)_{R-22} = 3.936}.$$

in/

$$(COP)_{R-13} = \frac{5258}{1446} = \boxed{3.64}.$$

(c) $$(COP)_{Whole} = \frac{210 \times 20}{1067 + 1446} = \boxed{1.6713\,TR}.$$

Q. 4. A single compressor using R-12 as refrigerant has three evaporators of capacity 30 TR, 20 TR and 10 TR. The temperature in the three evaporators is to be maintained at –10°C, 5°C and 10°C respectively. The condenser pressure is 9.609 bar. The liquid refrigerant leaving the condenser is sub-cooled to 30°C. The vapors leaving the evaporators and dry and saturated. Assuming isentropic compression, find (a). The mass of refrigerant flowing through each evaporator. (b) The power required to drive the compressor, and (c) COP of the system.

Ans. $$Q_1 = 30TR,\ t_1 = t_{12} = 0$$

$$Q_2 = 20TR,\ t_3 = t_{10} = t_{11} - 5°C$$

$$Q_3 = 10TR,\ t_5 = t_8 = t_9 = 10°C$$

$$t_c = 40°C,\ t_7 = 30°C$$

By the chart,

$$h_1 = 183.19\,kJ/kg$$

$$s_1 = 0.702\,kJ/kg\,k°$$

$$h_3 = 189.65\,kJ/kg$$

$$s_3 = 0.6943\,kJ/kg\,k°$$

Vapour refrigerant leaving at 10°C.

$$h_5 = 191.74\,kJ/kg$$

$$s_5 = 0.6921\,kJ/kg\,k°$$

In order to obtain the condition of refrigent leaving the first compressor at 2.

$$h_2 = 194.1\,kJ/kg$$

$$h_4 = 195.3\,kJ/kg$$

$$h_6 = 207.18\,kJ/kg$$

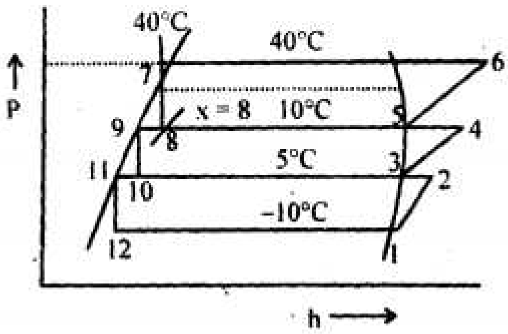
$$h_{f_7} = h_g = 64.59\,kJ/kg$$

$$x_8 = 0.13$$

$$h_{f_g} = h_{10} = 45.37\,kJ/kg$$

$$x_{10} = 0.04$$

$$h_{f_{11}} = h_{12} = 40.69 \text{ kJ / kg}$$



(a) Mass of refrigerant through compressor.

$$m_{c_1} = m_1 = \frac{210Q_1}{h_1 - h_{12}} = \frac{210 \times 30}{183.3 - 40.69}$$

$$= 44.2 \text{ kg / min}.$$

Mass of refrigerant requires to circulates in Second evaporator.

$$m_2 = \frac{210Q_2}{h_3 - h_{10}} = \frac{210 \times 20}{189.65 - 45.37} = 29.1 \text{ kg / min.}$$

$$m_2^1 = \frac{m_{c_2}(h_4 - h_8)}{h_5 - h_8} = \frac{76.5(195.3 - 191.74)}{191.74 - 64.59}$$

$$= 2.14 \text{ kg / min.}$$

$\therefore$ Mass of vapour formed $\quad m_{c_2} = 76.5 \text{ kg / min.}$

$$m_3'' = m_{c_2}\left(\frac{x_8}{1 - x_8}\right) = 76.5\left(\frac{.13}{1 - .13}\right)$$

$$= 11.47 \text{ kg / min}$$

Total refrigerant passing through Third compression,

$$m_{c_3} = m_{c_2} + m_3 + m_3' + m_3''$$

$$= 76.5 + 16.5 + 2.14 + 11.47 = 106.61 \text{ kg / min}$$

(b) Power required to drive,

$$p_1 = \frac{m_{c_1}(h_2 - h_1)}{60} = \frac{44.2(194.1 - 183.19)}{60}$$

$= 8.04$ kw

$$p_2 = \frac{m_{c_2}(h_4 - h_3)}{60} = \frac{76.5(195.3 - 189.65)}{60}$$

$= 7.2$ kg

$$p_3 = \frac{m_{c_3}(h_6 - h_5)}{60} = \frac{106.61(201.18 - 191.74)}{60}$$

$= 27.4$ kw

Total power $= p_1 + p_2 + p_3$

$= 8.04 + 7.2 + 27.4 = 42.64$ kw.

(c) COP,

$$R_E = 210(Q_1 + Q_2 + Q_3)$$

$= 210(30 + 20 + 10) = 12600$ kJ / min

Total work done $= w = 60 \times p$

$= 42.64 \times 60 = 2558.4$ kg / min

$$COP = \frac{R_E}{W} = \frac{12600}{2558.4} = 4.9 \text{ TR}.$$

Part-B

Q. 5. (a) Prove the following relations :

(i) $w = 0.622 \dfrac{p_v}{p - p_s}$

(ii) $\mu = \dfrac{w}{w_s} = \dfrac{p_v}{p_s}\left[\dfrac{1 - p_s / p}{1 - p_v / p}\right]$

(iii) $\phi = \dfrac{\mu}{1 - (1 - \mu)p_s / p}$.

Ans. (i) In air-conditioning system from dalton low :

$$p_b = p_a + p_v$$

Specific humidity : It is mass of water vapour present in 1 kg of dry air.

For dry air

p_a, v_a, T_a, m_a and R_a ; P_s pressure, volume, temp. r s, gas constant of dry air similarly, water vapour,

$$p_v, v_v, T_v, m_v, R_v$$

$$p_a v_a = m_a R_a T_a$$

$$p_v v_v = m_a R_a T_a$$

Also,
$$v_a = v_v$$

$$\frac{p_v}{p_a} = \frac{m_v R_v}{m_a R_a}$$

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

Where, $R = 0.287$ kJ/kg k for dry air.

$R_v = 0.461$ kJ/kg k for water vapour.

$$w = \frac{0.287 \times p_v}{0.461 \times p_a} = 0.622 \times \frac{p_v}{p_a}$$

$\because$
$$p_b = p_a + p_v$$

$$\boxed{w = 0.622 \times \frac{p_v}{p_b - p_v}}$$

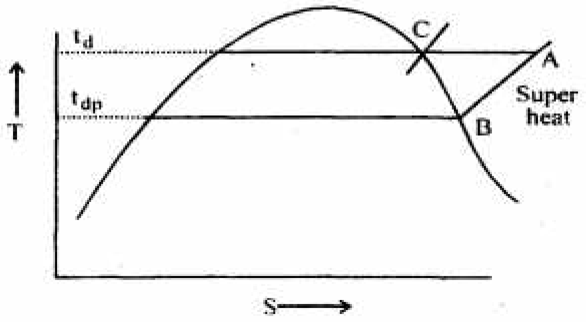
(ii)
$$w_s = w_{max}$$

$$w_s = 0.622 \times \frac{p_s}{p_b - p_s}$$

p_s = Partial pressure.

So, degree of saturation.

$$\mu = \frac{w}{w_s} = \frac{\dfrac{0.622 p_v}{p_b - p_v}}{\dfrac{0.622 p_s}{p_b - p_s}} = \frac{p_v}{p_s}\left(\frac{p_b - p_s}{p_b - p_v}\right).$$



(iii) Relative humidity

$$\phi = \frac{m_v}{m_s}$$

$$p_v v_v = m_v R_v T_v \qquad ...(i)$$

$$p_s v_s = m_s R_s T_s \qquad ...(ii)$$

$$v_v = v_s$$

$$T_v = T_s$$

$$R_v = R_s = 0.461 \text{ kJ/kg k}^o.$$

So, $$\phi = \frac{m_v}{m_s} = \frac{p_v}{p_s}$$

$$\mu = \frac{p_v}{p_s}\left[\frac{1 - \dfrac{p_s}{p_b}}{1 - \dfrac{p_v}{p_b}}\right] = \phi\left[\frac{1 - \dfrac{p_s}{p_b}}{1 - \phi + \dfrac{p_s}{p_b}}\right] \qquad \therefore \ \phi = \frac{p_v}{p_s}$$

$$\phi = \frac{\mu}{1(-1\mu)\frac{p_s}{p_b}}.$$

Q. 5. (b) Calculate :

(i) Relative humidity

(ii) Humidity ratio

(iii) Dew point temperature

(iv) Density

(v) Enthalpy

of atmospheric air, when DBT is 35°C, WBT is 23°C and the barometer reads 750 mm Hg.

Ans. Given, $t_d = 35°$, $t\omega = 23°$; $p_b = 750$ mm of fig.

$$p_b = 750 \times 133.3 = 0.99975 \text{ bar.}$$

From steam table,

$p_w = 0.02808$ correspond to wet bulb temp.

$p_s = 0.05622$ correspond to dry bulb temp.

So,

$$p_v = p_w - \frac{(p_b - p_w)(t_d - t_w)}{1547 - 1.44t_w}$$

$$p_v = 0.02808 - \frac{(.99975 - .02808)(35 - 23)}{1547 - 1.44 \times 23}$$

$$p_v = 0.02808 - \frac{11.66}{1513.88}$$

$$p_v = 0.02037.$$

(i) Relative humidity

$$\phi = \frac{p_v}{p_s} = \frac{0.02037}{.05622} = \boxed{36.23\%}$$

(ii) Humidity ratio,

$$w = \frac{0.622 p_v}{p_b - p_v} = \frac{0.622 \times .02037}{.99975 - .02037}$$

$$w = \frac{.01267}{.97938} = 0.01293$$

$$w = 0.0293$$

$$\boxed{w = 12.93 g/kg}.$$

(iii) Dew point temperature $\boxed{17°C}$ from chart.

(iv) Density,

$$\delta_v = \frac{w(p_b - p_v)}{R_a T_d}$$

$$= \frac{.01293(.99975 - .02808)}{287(273 + 35)}$$

$$\boxed{\delta_v = .01537 kg/m^3}.$$

(v) Enthalpy,

$$h_{fgdp} = 2461.4$$

$$h = 1.022 t_d + w(h_{fgdp} + 2.3 t_{dp})$$

$$h = 1.022 \times 35 + .01293(2461.4 + 2.3 \times 17)$$

$$\boxed{h = 60.3 kJ/kg}.$$

Q. 6. A hall is to be maintained at 24° C DBT and 60% RH under the following conditions :

Outdoor conditions 38°C DBT and 28°C WBT

Sensible heat load in the room 46.4 kW

Latent heat load in the rom 11.6 kW

Total infiltrated air $1200 m^3/hr$

ADP 10°C.

If the quantity of recirculated air is mixed with the conditioned air after the cooling coil, find the following :

(a) The condition of air leaving the conditioner coil and before mixing with the recirculated air.

Ans. Hall maintain at,

$\left.\begin{array}{r} 24°C\ DBT \\ \phi = 60\%\ RH \end{array}\right\}$ in side condition.

$DBT = 38°C$

$WBT = 28°C$

Sensible heat load $= 46.4\ kw$

Latent heat load $= 11.6\ kw$

Out side condition.

$ADP = 10°C$

Total infiltration air $= 1200\ m^3 / Hr$

So, $td_2 = 24°C,\ \phi = 60\%$

$td_1 = 38°C$

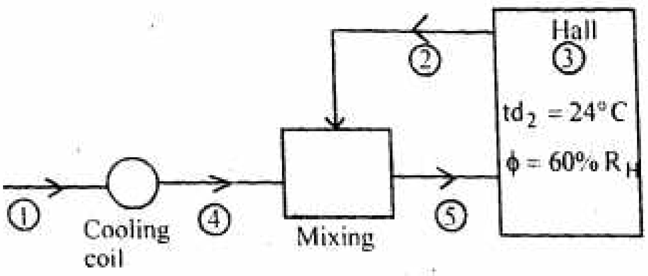
$tw_1 = 28°C$

$Q_S = 46.4\ kw$

$Q_L = 11.6\ kw$

$V_1 = 1200\ m^3 / h$

$ADP = 10°C$



First of all work the out door condition.

$VS1 = 0.907\ m^3 /kg$ of drug air.

Enthalpy $h_1 = 89.6\ kJ / kg$

$$h_2 = 53 kJ/kg$$

$$h_A = 68 kJ/kg$$

$$m_1 = \frac{V_1}{V_{s_1}} = \frac{1200}{.907}$$

$$= 1323 \text{ kg/Hr.}$$

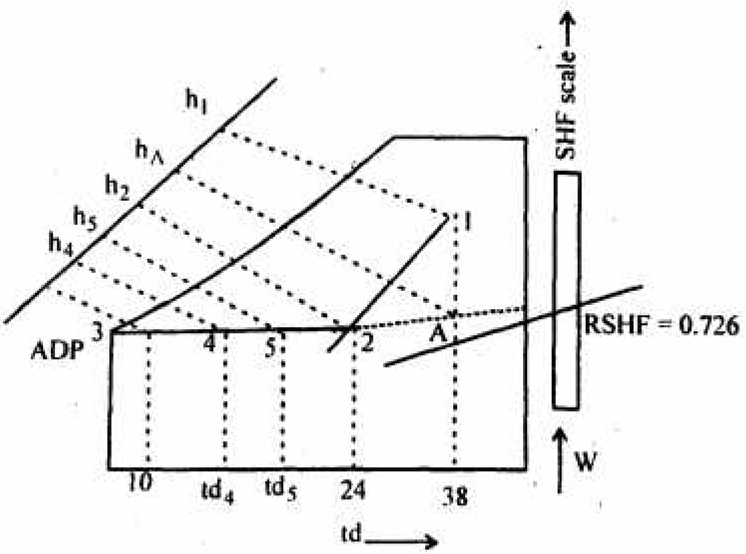
∴ Sensible heat gain due to filtration,

$$= m_1(h_A - h_2) = 1323(68-53) = 19845 kJ/kg$$

$$= \frac{19845}{3600} = 5.5 kw$$

And latent real gain to filtration.

$$= m_1(h_1 - h_A) = \frac{1323(89.6-68)}{3600}$$



$$= 7.94 kw$$

$$RSH = 46.4 + 5.5 = 51.9 kw$$

$$RLH = 11.6 + 7.94 = 19.54 \text{ kw}$$

$$RSHF = \frac{RSH}{RSH + RLH} = \frac{51.9}{51.9 + 19.54} = .726$$

Now, $td_4 = 16$ by chart.

$$\phi_4 = 90\% .$$

Q. 6. (b) The condition of air before entering the hall.

Ans. Condition of air before entering in Hall length 2–5 = length $2 - 4 \times 0.4$.

$$td_5 = 20.8°C$$

$$\phi_5 = 70\%$$

Q. 6. (c) The mass of air entering the cooler.

Ans.

$$m_c = \frac{\text{Total mass heat}}{\text{Total heat remove}} = \frac{RSH + RLH}{h_2 - h_4}$$

$$= \frac{51.9 + 19.54}{53 - 42} = 6.5 \text{ kg/s}$$

$$= 390 \text{ kg/min.}$$

Q. 6. (d) The mass of total air passing through the hall.

Ans.

$$m_H = 0.6 m_H + 390$$

$$m_H = \frac{390}{0.4} = 975 \text{kg/min}$$

Q. 6. (e) The bypass factor of the cooling coil.

Ans.

$$BPF = \frac{td_4 - ADP}{td_1 - ADP} = \frac{16 - 10}{38 - 10} = 0.214$$

Q. 6. (f) The refrigeration load on the cooling coil in TR.

Ans.

(f) Refrigeration load on cooling coil,

$$= m_c[h_1 - h_4] = 390(89.6 - 42)$$

$$= 18564 \text{ kJ / min}$$

$$= \frac{18564}{210} = 88 \text{ TR}$$

Q. 7. (a) Explain in detail the design of summer and winter air-conditioning systems.

Ans. Summer air condition : The summer air-condition may be design on the basis of the following condition.

* Depends on the type of installed condition.
* Depend on the amount of cooling required.
* Basis of the humidification required.
* Type of refrigerant used.
* Type of explanation devices used in the system.

The schematic figure may given as :

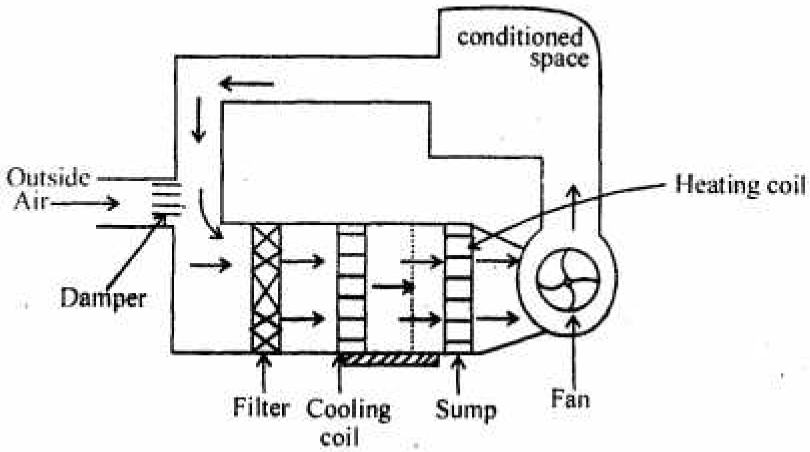
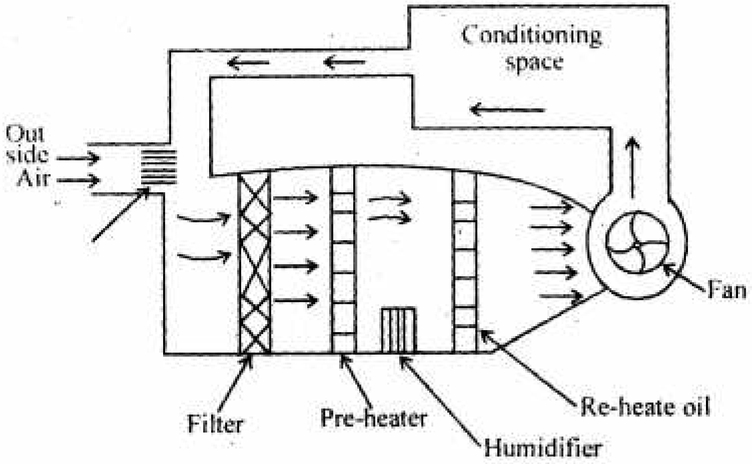


Fig. Summer A/c.

Winter air-condition : In winter air-conditioning system. The air is heated which is generally accompanied by humidification. The schematic arrangement of the system is shown.



The outside air flow through damper and mixed up with recirculated air and this mixture passed through filter for removing dust and impurities. The air now pass through a preheat coil in order to prevent the possible freezing of water and to control the humidifier.

This winter air conditioning system; is produced for used in winter.

Where required heating and humidifier.

Q. 7. (b) A restaurant with a capacity of 100 persons is to be air-conditioned with the following conditions :

Outside conditions 30°C DBT and 70% RH

Desired inside conditions 23°C DBT and 55% RH

Quantity of air supplied $0.5 m^3$ / min/ person.

Desired conditions are achieved by cooling, dehumidifying and then heating. Determine (a) Capacity of cooling coil in TR (b) Capacity of heating coil (c) Amount of water removed by dehumidifier (d) Bypass factor of the heating coil, if its surface temperature is 35°C.

Ans. Given :

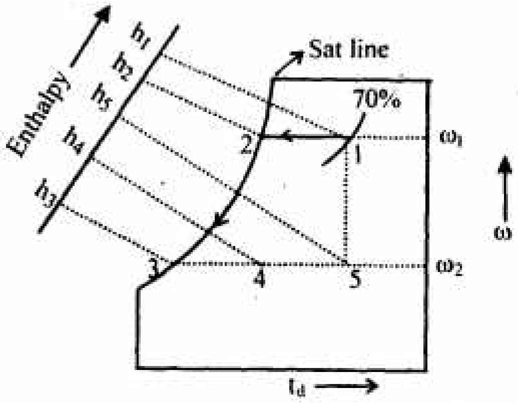
Number of person = 100

$$td_1 = 30°C,\ \phi_1 = 70\%,\ td_4 = 23°C.$$

$$\phi_4 = 55\%,\ v_1 = 0.5 m^3/\text{min/person}.$$

$$v_1 = 0.5 \times 100 = 50 m^2 / \text{min}$$

First make the outside condition at 30° t_d & 70% RH at chart.



From chart, $v_{s_1} = 0.885$

$$m_a = \frac{v_1}{v_{s_1}}$$

$$m_a = \frac{50}{0.885} = 56.5 \text{ kg/min.}$$

(i) Capacity of cooling coil in TR.

$h_1 = 78.5$ kJ/kg of dry air.

$h_3 = 37.8$ kJ/kg.

Capacity of cooling $= m_a(h_1 - h_3)$

$= 56.5(18.5 - 37.8) = 2300$ kJ/min.

Capacity $= \frac{2300}{210} = \boxed{10.95 \text{ TR}}$.

(ii) Capacity of heating coil,

$h_4 = 47.6$ kJ/kg

$= m_a(h_4 - h_3) = 56.5(47.6 - 37.8)$

$= 54$ kJ/min

$$= \frac{554}{60} = \boxed{9.23\,kw}.$$

(iii) Amount of water through dehumidifier : By psychrometric chart,

$w_2 = 0.0188$ kJ/kg of dry air

$w_3 = 0.0095$

Amount of water $= m_a(w_2 - w_3)$

$= 56.5(0.0188 - 0.0095)$

$= 0.525\,kg/min$

$= 0.525 \times 60 = \boxed{31.5\,kg/h}.$

(iv) By-pass factor :

$td_3 =$ Surface temp of heating coil $= 35°C$

$td_3 = 13.5°C$

$$BPF = \frac{td_5 - td_4}{td_5 - td_3} = \frac{35 - 23}{35 - 13.5} = 0.558$$

$$\boxed{BPF = 0.558}.$$

Q. 8. (a) Describe with a sketch a centrifugal compressor. Where are centrifugal compressors preferred over reciprocating compressors in refrigerating system?

Ans. Centrifugal Compressor : The centrifugal compressor for refrigeration system was worked and generate. Head due to functioning of centrifugal force and its generally used for refrigerant that require large displacement and low condensing pressure; such $R-11$, $R-113$ the single stage, number of curved vans fitted on impeller.

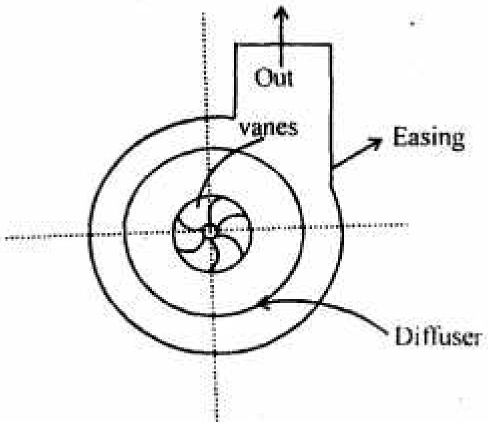
The some advantage of centrifugal compressor :

- Since the centrifugal compressor have no valve, piston, cylinder etc., so comparative working life more than reciprocating one.
- These compression operate with little or no-vibration. So, no, unbalance mass.
- The operation of centrifugal comp is quiet & calm.
- The centrifugal comp run at high speed 300 rpm.

- Efficiency also, quit high.

The main disadvantage is surging phenomenon.

So, from above discussed matter it is clear that the centrifugal compressor is the batter than the reciprocation compressor.



Centrifugal comp.

Q. 8. (b) What are the desirable properties of an ideal refrigerant? Discuss in detail secondary and eco friendly refrigerants.

Ans. Refrigerant : It is the heat carrying medium which take heat from cold medium and deliver it in to hot medium. Under the control condition.

Properties of ideal refrigerant :

- Law boiling point.
- High critical temperature.
- High latent heat of vaporisation.
- Low specific heat of liquid.
- Low specific volume of vapour.
- Non-corrosive to metal.
- Non-flammable, non explosive.
- Non-toxic.
- Low cost.
- Easy to highfied at moderate temperature & pressure.

- Mixed well with oil.
- Easy to locate leakage etc. by order etc.

Ozone friendly : Traditionally used refrigerant emitted cfcs. So, ozone layer goes decreased. But not a matter for discuss.

It is requires to used ozone friendly refrigerant which always exists in India. It not goes bad impact.

Secondary Refrigerant : The secondary refrigerant not take the parts for cooling the medium. But it required for the cooling in intermediate one.

The brine is the most important used in ice plant; so, some SO_3, $R-113$ etc. son etc.