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B. E.

Seventh Semester Examination, May-2008

REFRIGERATION AND AIR-CONDITIONING

Note : Attempt any five questions.

Section-A

Q. 1. (a) Define the following terms :

Tons of Refrigeration, COP, Nomenclature of refrigerants, Secondary refrigerants.

Ans. Tons of Refrigeration : The practical wit of refrigeration is expressed in terms of 'tonne of refrigeration.' A tonne of refrigeration is defined as the amount of refrigeration effect produced by the uniform melting of one tonne (1000 kg) of ice from and 0°C in 24 hours.

C.O.P. : The coefficient of performance (briefly written as C.O.P.) is the ratio of heat extracted in the refrigerator to the work done on the refrigerant. It is also known as theoretical coefficient of performance.

$$\text{C.O.P.} = \frac{Q}{W}.$$

Nomenclature of Refrigerants : The refrigerant is a heat carrying medium which during their cycle in the refrigeration system absorb heat from a low temperature system and discard the heat so absorbed to a higher temperature system. The natural ice and a mixture of ice and slat were the first refrigerants.

Secondary Refrigerants : Brines are secondary refrigerants and are generally used where temperatures are required to be maintained below the freezing point of water i.e. 0°C. In case the temperature involved is above the freezing point of water (0°C), then water is commonly used as a secondary refrigerants.

Q. 1. (b) With the help of layout and T-S diagram derive the expression for the COP and power required for Boot strap cycle used in aircrafts.

Ans. If θ tonnes of refrigeration is the cooling load in the cabin, then he quantity of air required for the refrigeration purpose will be

$$m_a = \frac{210\theta}{C_P\left(T_7 - T_6'\right)} \text{kg/min}$$

m_1 = Total mass of air bled from the main compressor.

m_2 = Mass of cold air bled from the cooling turbine for regenerative heat exchanges

For the energy balance of regenerative heat exchanger,

$$m_2 c_p\left(T_8 - T_6'\right) = m_c c_p\left(T_4 - T_5\right)$$

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$$m_2 = \frac{m_1(T_4 - T_5)}{(T_8 - T_6')}$$

T_8 = Temperature of air leaving to atmosphere from the regenerative heat exchanger. Power required for the refrigeration system.

$$P = \frac{m_1 c_p (T_3' - T_2')}{60} \text{kw}$$

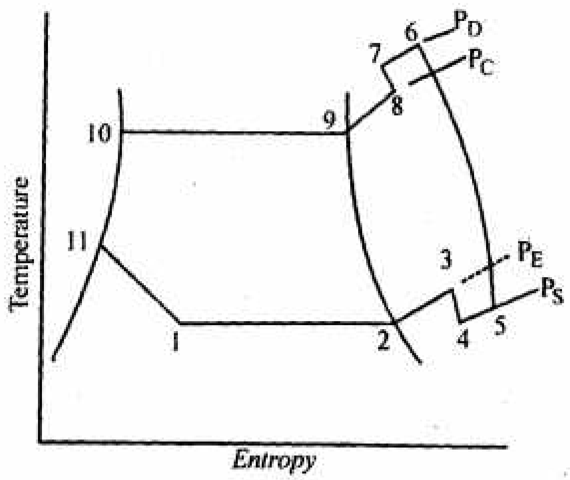
C.O.P. of the refrigerating system,

$$= \frac{210\theta}{m_1 c_p (T_3' - T_2')}$$

$$= \frac{210\theta}{P \times 60}$$

Q. 2. (a) With the help of T-S or P-H diagram explain the actual vapour compression cycle considering all the losses.

Ans. The actual vapour compression cycle on T-S diagram is shown in fig. The various processes are discussed below :



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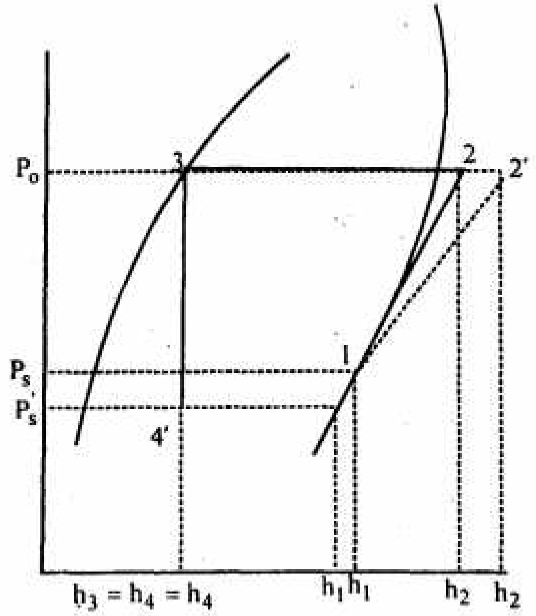

(a) Process 1-2-3 :

This process shows the flow of refrigerant in he evaporator. The point 1 represents the entry of refrigerant into the evaporator and the point 3 represents the exit of refrigerant from evaporator in a superheated state. The point 3 also represents the entry of refrigerant into the compressor in a superheated condition.

(b) Process 3-4-5-6-7-8 :

This process represents the flow of refrigerant through the compressor. When the refrigerant enters the compressor through the suction at point 3.

(c) Process 8-9-10-11 :

This process represents the flow of refrigerant though the condenser. The process 8-9 represents the cooling of superheated vapour refrigerant to the dry saturated state.

(d) Process 11-1 :

This process represents the expansion of subcooled liquid refrigerant by throttling from the condenser pressure to the evaporator pressure

Q. 2. (b) An NH_3 refrigerating machine fitted with an expansion value works between the temperature limits of –10°C and 30°C. The vapour is 95% dry at the end of isentropic compression and the fluid leaving the condenser is at 30°C. Assume the actual COP is 60% of theoretical. Calculate the Kg of

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ice produced per Kial. her at 0°C from water at 10°C. Latent heat of ice is 335 KJ/Kg.

Ans. Given

$$T_1 = -10°C$$

$$= -10 + 273 = 263K$$

$$T_2 = 30°C$$

$$= 30 + 273 = 303K$$

$$T_w = 30°C$$

$$h_{fg(ice)} = 335 KJ/kg$$

Now mass of the ice produced per day we know that heat extraction capacity of the refrigerator.

$$= 200 \times 210 \quad [\theta = 220TR]$$

$$= 42000 kJ/min.$$

and heat removed from 1 kg of water at 30°C to form ice at 0°C.

$$= \text{Mass} \times \text{Sp. heat} \times \text{Rise in temperature } + h_{fg}(ice)$$

$$= 1 \times 4.187(30 - 0) + 335$$

$$= 439.7 kJ/kg$$

$\therefore$ Mass of ice produced per min,

$$= \frac{42000}{439.7}$$

$$= 95.52 \text{ kg/min.}$$

And mass of ice produced per day,

$$= 95.52 \times 60 \times 24$$

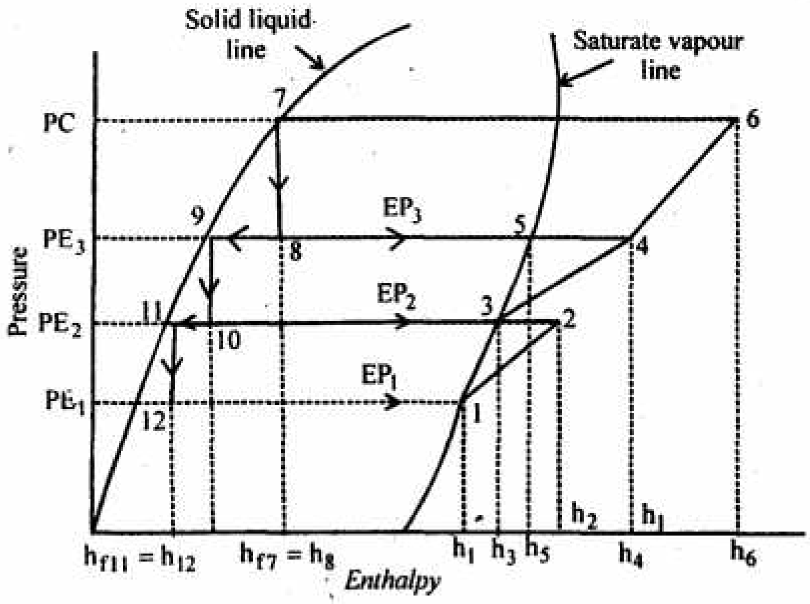
$$= 137550 kg$$

$$= 137.55 \text{ tonnes}.$$

Q. 3. (a) Draw the layout and P-H diagram for a multiload (TR-1 & TR-2) with compound compression single expansion value and flash intercooler. Derive the expression for COP and power required.

Ans. If θ tonnes of refrigeration is the load on the evaporator, then the mass of refrigerant passing through the evaporator or low pressure compressor,

$$m_1 = \frac{210\theta}{h_1 - h_2} = \frac{210\theta}{h_1 - h_{f11}} \text{ kg/min.}$$



Thus, the mass of vapour refrigerant passing through the intermediate compressor.

$$m_2 = m_1 + m_1\left(\frac{x_{10}}{1-x_{10}}\right) + m_1\left(\frac{h_2 - h_3}{h_3 - h_{10}}\right)$$

$$= m_1\left[1 + \frac{x_{10}}{1-x_{10}} + \frac{h_2 - h_3}{h_3 - h_{10}}\right]$$

Mass of vapour refrigerant,

$$m_3 = m_2 + \frac{m_1 x_8}{(1-x_{10})(1-x_8)} + m_2\left(\frac{h_4 - h_5}{h_5 - h_8}\right)$$

$$= m_2\left[1 + \frac{h_4 - h_5}{h_5 - h_8}\right] + \frac{m_1 x_8}{(1-x_{10})(1-x_8)}$$

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We know that work done in L.P. compressor

$$W_L = m_1(h_2 - h_1)$$

Work done in I.P. compressor,

$$W_I = m_2(h_4 - h_3)$$

Work done in H.P. compressor,

$$W_H = m_3(h_6 - h_5)$$

and total work done in three compressors

$$W = W_L + W_I + W_H$$

$$= m_1(h_2 - h_1) + m_3(h_4 - h_3) + m_3(h_6 - h_5)$$

∴ Power required to drive the co..pressors,

$$P = \frac{W}{60} KW$$

We know that refrigerant effect,

$$R_E = m_1(h_1 - h_{f11})$$

$$= 2100 \, KJ/min$$

∴ $$\text{COP of the} = \frac{R_E}{W} = \frac{2100}{P \times 60}.$$

Q. 3. (b) Calculate the power needed to compress 20 Kg/min of NH_3 from saturated vapour at 1.4 bar to a condenser pressure of 10 bar by two stage pressure with intercooling by liq., at 4 bar assume saturated liq., leaves the condenser and dry vapour leaves the evaporator.

Ans. $m_1 = 20 \, kg/min$, $P_E = 1.4 \, bar$, $P_2 = P_3 = 4 \, bar$. The p-h diagram for a two-stage compression with intercooling by liquid refrigerant is shown in fig. The various values for R-12 as read from the p-h diagram are as follows : Enthalpy of saturated vapour refrigerant entering the low pressure compressor at point 1.

$$h_1 = 178 \, KJ/kg$$

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

$$S_1 = 0.71 \, kJ/kg \, K$$

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

$$h_2 = 195 \, kJ/kg$$

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Enthalpy of saturated vapour refrigerant leaving the interscalar or entering the high pressure compressor at point 3.

$$h_3 = 191 \, KJ/kg$$

Entropy of saturated vapour refrigerant entering the high pressure compressor at point 3.

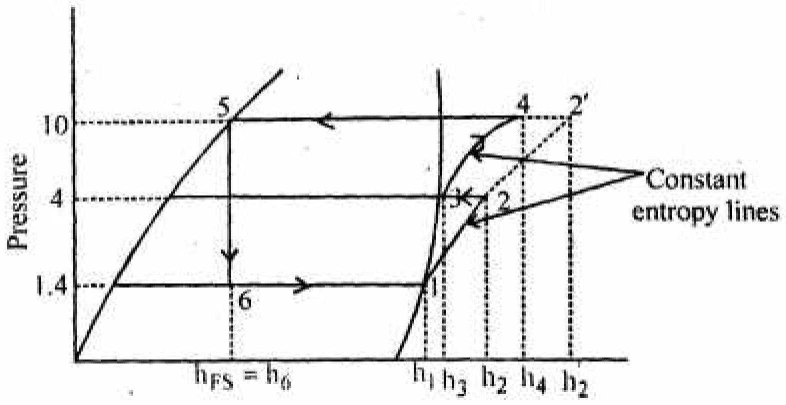
$$S_3 = 0.695 \, kJ/kg\,k$$

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

$$h_4 = 210 \, kj/kgk$$

Enthalpy of saturated liquid refrigerant leaving the condenser at point 5.

$$h_{f5} = h_6 = 77 \, kJ/kg$$



We know that mass of refrigerant passing through the condenser (or high pressure compressor).

$$m_2 = \frac{m_1(h_2 - h_{f5})}{h_3 - h_{f5}}$$

$$= \frac{20(195 - 77)}{181 - 77} = 207 \, kg/min$$

Work done in low pressure compressor,

$$w_L = m_1(h_2 - h_1) = 20(195 - 178) = 340 kJ/min$$

Work done in high pressure compressor,

$$w_H = m_2(h_4 - h_3) = 20.7(210 - 191) = 393 \text{ kJ / min.}$$

and total work done in both the compressors,

$$w = w_L + w_H = 230 + 393 = 733 \text{ kJ / min}$$

$$= 733/60 = 12.2 \text{ kw}.$$

Q. 4. (a) Derive an expression for the COP of a vapour absorption refrigeration system.

Ans. Neglecting the heat due to pump work (θ_P). We have according to first law of thermodynamics,

$$\theta_C = \theta_G + \theta_E$$

T_G = Temperature at which heat (θ_G) is given to the generator.

T_C = Temperature at which heat (θ_C) is discharged to atmosphere or cooling water from the condenser and absorber and

T_E = Temperature at which heat (θ_E) is absorbed in the evaporator.

Since the vapour absorption system can be considered as a perfectly reversible system therefore the initial entropy of the system must be equal to the entropy of the system after the change in its condition.

$$\frac{\theta_G}{T_G} + \frac{\theta_E}{T_E} = \frac{\theta_C}{T_C} = \frac{\theta_C + \theta_E}{T_C}$$

$$\frac{\theta_G}{T_G} - \frac{\theta_G}{T_C} = \frac{\theta_E}{T_C} - \frac{\theta_E}{T_E}$$

$$\theta_G\left(\frac{T_C - T_G}{T_G \times T_C}\right) = \theta_E\left(\frac{T_E - T_C}{T_C \times T_E}\right)$$

$$\theta_G = \theta_E\left[\frac{T_E - T_C}{T_C \times T_E}\right]\left[\frac{T_G \times T_C}{T_C - T_G}\right]$$

$$= \theta_E\left[\frac{T_C - T_E}{T_C \times T_E}\right]\left[\frac{T_G \times T_C}{T_G - T_C}\right]$$

$$= Q_E\left[\frac{T_C - T_E}{T_E}\right]\left[\frac{T_G}{T_G - T_C}\right]$$

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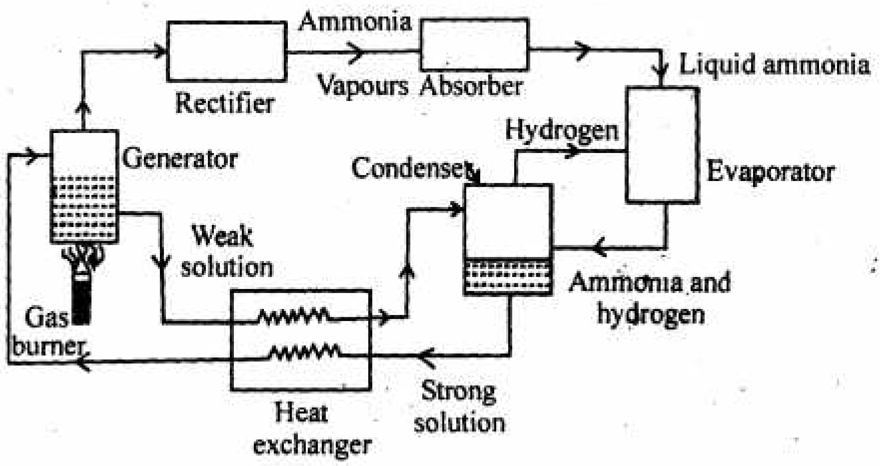
Maximum coefficient of performance of the system is given by

$$(\text{C.O.P.})_{\text{max.}} = \frac{\theta_E}{\theta_G} = \frac{\theta_G}{\theta_E\left[\frac{T_C - T_E}{T_E}\right]\left[\frac{T_G}{T_G - T_C}\right]}$$

$$= \left[\frac{T_E}{T_C - T_E}\right]\left[\frac{T_G - T_C}{T_G}\right].$$

Q. 4. (b) With the help of neat sketch explain the working of an Electrolux refrigerators.

Ans. Electrolux Refrigerator : The domestic absorption type refrigerator was invented by two Swedish engineers curl minters and Boltzer von latan in 1925 while they were studying for their under-graduate course of Royal Institute of technology in stockholm.



This type of refrigerator is also called three-fluids absorption system. The main purpose of this system is to eliminate the pump so that in the absence of moving parts, the machine become noise-less. The three fluids used in this system are ammonia, hydrogen and water. The ammonia is used as a refrigerant because it possesses most of the desirable properties.

Section-B

Q. 5. (a) Define the all psychrometric processes with the air washer.

Ans. Psychrometric Processes with the Help of Air Washer are :

1. Heating and Humidification :

The mean surface temperature of water is greater than the dry bulb temperature of air. The water is externally heated.

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2. Humidification :

The mean surface temperature of water is equal to the dry bulb temperature of air. The enthalpy of air increases. Hence the water is required to be externally heated.

3. Cooling and Humidification :

The mean surface temperature of water is less than the dry bulb temperature of air but greater than the wet bulb temperature of air.

4. Adiabatic Saturation : This is the case of pumped recirculation of water without any external heating or cooling. The recicrulated water reaches the equilibrium temperature which is equal to the thermodynamic wet bulb temperature of air.

5. Cooling and Humidification : The process is similar to 1-2C with the difference that the enthalpy of air decreases in this case.

6. Cooling and Dehumidification :

The mean water surface temperature is lower than the dew point temperature of air. Air is simultaneously cooled and dehumidified. The process is exactly similar to that of a cooling and dehumidifying coil.

Q. 5. (b) Prove the following relation :

(i) $\phi = \dfrac{\mu}{1-(1-\mu)\dfrac{Pws}{P}}$

(ii) $w = 0.622\dfrac{Pw}{Pa}$

Ans. (i) Let P_V, V_V, T_V, m_V and R_V = Pressure, volume, temperature, mass and gas constant respectively for water vapour is actual conditions and

P_S, V_S, T_S, M_S and R_S = Corresponding values for water vapour in saturated air.

We know that for water vapour in actual conditions,

$$P_V V_V = m_V R_V T_V \qquad ...(1)$$

For water vapour in saturated air,

$$P_S V_S = m_S R_S T_S \qquad ...(2)$$

According to the definitions,

$$V_V = V_S$$

$$T_V = T_S$$

$$R_V = R_S = 0.461 \text{ kJ/kgk}$$

From equation (1) and (2), relative humidity,

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

We know that the degree of saturation,

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

$$\phi = \frac{\mu}{1-(1-\mu)\frac{P_s}{P_b}}$$

(ii) Let P_a, V_a, T_a, M_a and R_a = Pressure, volume, absolute, temperature, mass and gas constant respectively for dry air and P_v, V_v, T_v, M_v and R_v = Corresponding values for the water vapour.

Assuming that the dry air and water vapour behave as perfect gases, we have for dry air,

$$P_a V_a = m_a R_a T_a \quad ...(1)$$

For water vapour,

$$P_v = P_v = m_v R_v T_v \quad ...(2)$$

$$V_a = V_v$$

$$T_a = T_v = T_d$$

From equation (1) and (2)

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

Humidity Ratio

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

Substituting $R_a = 0.287\,kJ/kg\,K$ for dry air and $R_v = 0.461\,kJ/kg\,K$ for water vapour in the above equation,

$$W = \frac{0.287\times P_v}{0.461\times P_a} = 0.622\times\frac{P_v}{P_a}$$

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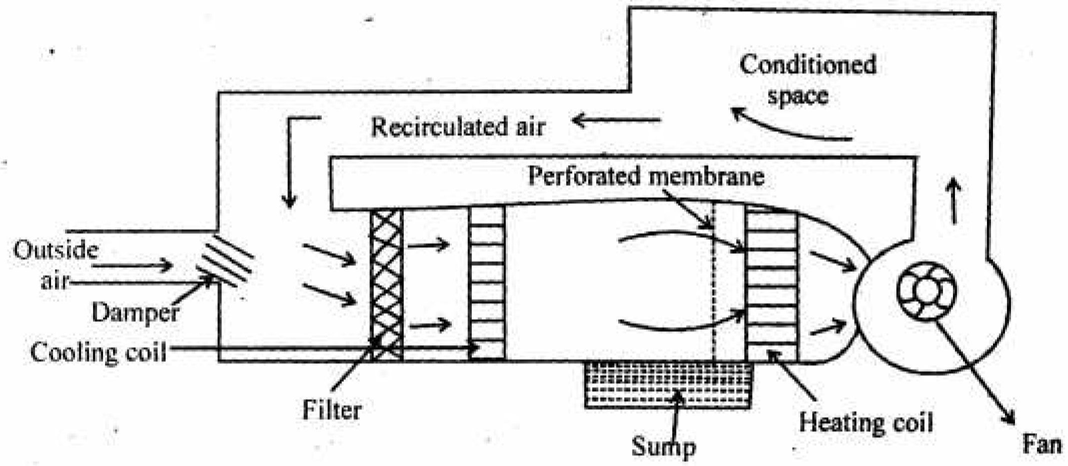
$$= 0.622 \times \frac{P_v}{P_b - P_v}.$$

Q. 6. (a) With the help of neat sketch and psychrometric diagram. Explain the summer air conditioning for hot and dry outdoor conditions.

Ans. It is the most important type of air conditioning, in which the air is cooled and generally dehumidified. The schematic arrangement of a typical summer air conditioning system is shown in fig.

The outside air flows through the damper, and mines up with recirculated air (which is obtained from the conditioned space). The mixed air passes through a filter a remove dirt, dust and other impurities. The air now passes through a cooling coil. The coil has a temperature much below the required dry build temperature of the air in the conditioned space. The cooled air passes through a perforated membrane and losses its moisture in the condensed from which is collected in a sump.



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 ventilators. The remaining part of the used air (know as recirculated air) is again conditioned as shown in fig. The outside air is sucked and made to min the recirculated air in order to make up for the loss of conditioned air through exhaust- fans or ventilation from the conditioned space.

Q. 6. (b) In a cooling application moist air enters a refrigerations coil at a rate of 100 Kg of dry air per min at 35°C and 50% RH. The ADP of the coil is 5°C and by pass factor 0.15. Determine the outlet state of moist air and cooling capacity of coil in TR.

Ans. Given

$$m_a = 100\,\text{kg/min}$$

$$t_{d_1} = 35^\circ\text{C}$$

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$$\phi_1 = 50\%$$
$$ADP = 5°C$$
$$BPF = 0.15$$

Outlet State of Moist Air :

Let t_{d_2} and ϕ_2 = Temperature and relative humidity of air leaving the cooling coil.

First of all, mark the initial condition of air, i.e., 35°C dry bulb temperature and 50% relative humidity on the psychrometric chart at point 1. From the psychrometric chart, we find that the dew point temperature of the entering air at point 1.

$$t_{dp_1} = 23°C$$

Since the coil or apparatus dew point (ADP) is less than the dw point temperature of entering air, therefore it is a process of cooling and dehumidification.

We know that by pass factor,

$$BPF = \frac{t_{d_2} - t_{d_4}}{t_{d_1} - t_{d_4}} = \frac{t_{d_2} - ADP}{t_{d_1} - ADP}$$

$$0.15 = \frac{t_{d_2} - 5}{35 - 5} = \frac{t_{d_2} - 5}{30}$$

$$t_{d_2} = 0.15 \times 30 + 5 = 9.5°C.$$

From the psychrometric chart, we find that the relative humidity corresponding to a dry bulb temperature (t_{d_2}) of 9.5°C on the line 1-3 is

$$\phi_2 = 99\%.$$

Cooling Capacity of the Coil :

The resulting condition of the air coming out of the coil is shown by point 2. On the line joining the point 1 and 4. The line 1-2 represents the cooling and dehumidification process which may be assumed to have followed the path 1-A (i.e. dehumidification) and A-2 (i.e. cooling). Now from the psychrometric chart, we find that enthalpy of entering air at point 1.

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

And enthalpy of air at point 2.

$$h_2 = 28\,kJ/kg \text{ of dry air.}$$

We know that cooling of the coil

$$= m_a(h_1 - h_2)$$

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$$= 100(81-28)$$

$$= 5300\,kJ/min$$

$$= \frac{5300}{210}$$

$$= 25.24\,TR.$$

Q. 7. An air conditioning plant is to be designed for a small office for winter conditions outdoor conditions 10°C DBT & 8° WBT required condition 20°C DBT & 60% R.H. Amount of air circulation $= 0.3m^3/min/$ Person. Seating capacity of the office = 50 Persons.

The required condition is achieved first by heating and than by adiabatic humidification find.

(a) Heating capacity of coil in KW and the surface temperature, if the by pass factor of the coil is 0.32.

(b) Capacity of the humidifier.

Ans. Given : $t_{d_1} = 10°C$, $t_{w_1} = 8°C$, $t_{d_2} = 20°C$, $\phi_2 = 60\%$ seating capacity = 50 persons, $V_1 = 0.3\,m^3/min/person = 0.3 \times 50 = 15\,m^3/min$, BPF = 0.32. First of all, mark the initial condition of air at 10°C dry bulb temperature and 8°C wet bulb temperature on the psychrometric chart as point 1, as shown in fig. Now mark the final condition of air at 20°C dry bulb temperature and 60% relative humidity on the chart as point 2. Now locate point 3 on the chart by drawing horizontal line through point 1 and constant enthalpy line through and 2. Form the psychrometric chart, we find that the specific volume at point 1.

$$V_{S1} = 0.81\,m^3/kg \text{ of dry arise}$$

Mass of air supplied per minute,

$$m_a = \frac{V_1}{V_{S1}} = \frac{15}{0.81} = 18.52\,kg/min.$$

1. Heating capacity of the coil in kw and the surface temperature.

From the psychrometric chart, we find that enthalpy at point 1.

$$h_1 = 24.8\,kJ/kg \text{ of dry air.}$$

and enthalpy at point 2.

$$h_2 = 42.6\,kJ/kg \text{ of dry air}$$

We know that heating capacity of the coil,

$$= m_a(h_2 - h_1) = 18.52(423.6 - 24.8) = 329.66\,kJ/min$$

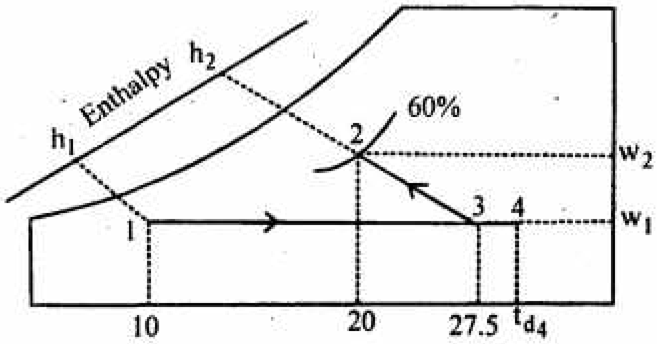
$$= \frac{329.66}{60} = 5.5\,kw$$

t_{d4} = Surface temperature of the coil.

We know that by pass factor (BPF).

$$0.32 = \frac{t_{d_4} - t_{d_3}}{t_{d_4} - t_{d_1}} = \frac{t_{d_4} - 27.5}{t_{d_4} - 10}$$

$$0.32\left(t_{d_4} - 10\right) = t_{d_4} - 27.5$$



2. Capacity of the humidifier. From the psychrometric chart, we find that specific humidity at point 1.

$$W_1 = 0.0058\,\text{kg / kg of dry air}$$

And specific humidity at point 2.

$$W_2 = 0.0088\,\text{kg / kg of dry air}$$

We know that capacity of the humidifier,

$$= m_a(w_2 - w_1) = 18.52(0.0088 - 0.0058) = 0.055\,\text{kg / min.}$$

$$= 0.055 \times 60\,\text{kg / h}$$

$$= 3.3\,\text{kg / h}.$$

Q. 8. Attempt any three of the following :

(a) Duct System Design,

(b) Type of Evaporations,

(c) Air-distribution Systems.

(d) Sources of Heat Load.

Ans. (a) Duct System Design :

The object of duct design is to determine the dimensions of all ducts in the given systems. The ducts

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should carry the necessary volume of conditioned air from the fan outlet to the conditioned space with minimum frictional and dynamic losses. The duct layout must be made so as to reach the outlet without least number of bends, obstructions 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.

(b) Type of Evaporations :

1. According to the type of construction.

(a) Bare tube coil evaporator.

(b) Finned tube evaporator.

(c) Plate evaporator.

(d) Shell and tube evaporator.

(e) Shell and coil evaporator and

(f) Tube in tube evaporator.

2. According to manner in which liquid refrigerant is fed :

(a) Flooded evaporator and

(b) Dry expansion evaporator.

3. According to mode of heat transfer :

(a) Natural convection evaporator and

(b) Forced convection evaporator.

4. According to operating conditions :

(a) Frosting evaporator.

(b) Non-frosting evaporator and

(c) Defrosting evaporator.

(c) Air-distribution systems :

When there is problem of air condition in a space essentially reduces to the calculation of the state and mass rate of air to be supplied to the space necessary to pick up its sensible heat and latent heat loads, than air distribution system is use. For the simplest air conditioning system, consider a space which is to be maintained at the room or inside conditions of dry bulb temperature.

(d) Sources of heat load :

Sources of heat load are :

1. Occupancy heat load.
2. Lighting load.
3. Appliances heat load.
4. Product heat load.
5. Process heat load.