INDIAN INSTITUTE OF TECHNOLOGY KHARAGPUR

Date : Time : 2 hours Full Marks : 30 No. of students : 20

Mid-Semester, Spring 2008 Mechanical Engineering ME2 and DD

Subject No. ME60096 Subject Name: Air conditioning & Ventilation

Note:

1. Use perfect gas model for calculating the required psychrometric properties.
2. Unless otherwise specified, assume the barometric pressure to be 101.3 kPa.
3. Make suitable assumptions, but state them clearly.

Given data:

Molecular weights: Dry air = 28.966 kg/kmol; Water = 18.02 kg/kmol

Universal gas constant = 8.314 kJ/kg.K; C_p of liquid water = 4.18 kJ/kg.K

Specific heats, C_p: Dry air=1.005 kJ/kga.K, Water vapor=1.88 kJ/kgw.K

Stefan_Boltzmann Constant, σ = 5.669 X 10^{-8} W/m^2.K^4

Unless otherwise specified, Specific heat of moist air, c_{pm} (kJ/kga.K):

c_{pm} = 1.005+1.88w, where w is humidity ratio of air (kg water/kg air)

Antoine's equation for saturation pressure (p_{sat}) of water:

$$\ln(p_{sat}) = 16.54 - \left(\frac{3985}{T - 39.00}\right); \quad p_{sat} \text{ in kPa \& T in K}$$

Latent heat of vaporization of water,h_{fg} in terms of temperature ($0 \le t \le 40^oC$):

$$h_{fg} = 2500.95 - 2.37267\,t \quad kJ/kgw$$

Humidity ratio(w_s) and enthalpy (h_s) of saturated air ($5 \le t \le 30^oC$):

$$w_s = 4.914 + t(0.0102 + 0.0243714\,t) \qquad \text{grams of water/kga}$$

$$h_s = 12.4312 + t(1.00455 + 0.0633071\,t) \quad kJ/kga$$

Carrier Equation:

$$p_v = p'_v - \frac{1.8(p_t - p'_v)(DBT - WBT)}{2800 - 1.3(1.8DBT + 32)}$$

where p_v, p'_v and p_t are vapour pressure, saturated vapour pressure at WBT and barometric pressure, respectively. DBT and WBT are in ^oC and units for pressures should be consistent.

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1. Atmospheric air at 42°C (DBT) and 28°C (WBT) is in contact with a water pond of 100 m^2 surface area. The surface of the pond is at 30°C and the convective heat transfer coefficient between the pond surface and surrounding air is 23 W/m^2.K. Assuming a value of 1.0 for the

Lewis number, find the direction and magnitude of heat transfer between the pond surface and the atmospheric air. Also find the evaporation rate per hour. (5+1=6)

2. An unshielded sling type psychrometer shows a dry bulb temperature of 32°C and a wet bulb temperature of 21°C. The surrounding surfaces are at an average temperature of 36°C. The sensing bulb surfaces of both dry bulb and wet bulb thermometers have an emissivity of 0.9. The convective heat transfer coefficient between surrounding air and the sensing bulbs of the dry bulb and wet bulb thermometers are 33 $W/m^2.K$ and 23 $W/m^2.K$, respectively. Assuming the humid specific heat of moist air to be 1.03 kJ/kga.K and the Lewis number for air-water mixture to be 0.9, find the actual air dry bulb temperature and humidity ratio. Neglect radiation between dry and wet bulbs. (3+5 = 8)

3a. Prove that when moist air flows over a wetted surface (maintained at a constant temperature), the exit condition lies on the straight line joining the inlet state of air and saturated air at the wetted surface temperature on a psychrometric chart. Assume the specific heat and the convective heat transfer coefficient to be constant, and a value of 1.0 for Lewis number. (4)

3b. Air at 25°C (DBT) and 60% relative humidity flows over a cooling and dehumidifying coil that has an apparatus dew point temperature (ADP) of 7°C. If the coil has a bypass factor 0.1, find the exit air temperature and relative humidity. (4)

4. A large auditorium is designed for a seating capacity of 800 people. The auditorium has a sensible cooling load of 400 kW and a room sensible heat factor (RSHF) of 0.8. The design inside and outside conditions are 27°C (DBT) and 50% relative humidity, and 42°C (DBT) and 28°C (WBT), respectively. Fresh outdoor air at a rate of 9 litres/s/person (measured at outdoor conditions) should be supplied to the building for ventilation purposes. For proper distribution of air inside the auditorium, the supply air flow rate to the auditorium should be 30 kg/s (dry air basis). The air conditioning system uses a cooling and dehumidifying coil that has an apparatus dew point temperature (ADP) of 7°C. The dry bulb temperature of air at the exit of the cooling and dehumidifying coil is 11°C. a) Draw the schematic of the proposed system that can satisfy the given requirements, b) Show the cycle on psychrometric chart, c) Find the bypass factor, cooling capacity and sensible heat factor of the cooling and dehumidifying coil. (2+2+4 = 8)

End of the paper

INDIAN INSTITUTE OF TECHNOLOGY KHARAGPUR

Date : Time : 3 hours Full Marks : 50 No. of students : 20

End-Semester, Spring 2008 Mechanical Engineering ME2 and DD

Subject No. ME60096 Subject Name: Air conditioning & Ventilation

Given data:

Use perfect gas model for calculating the required psychrometric properties.

Unless otherwise specified, assume the barometric pressure to be 101 kPa.

Molecular weights: Dry air = 28.966 kg/kmol; Water = 18.02 kg/kmol

Universal gas constant = 8.314 kJ/kg.K; C_p of liquid water = 4.18 kJ/kg.K

Specific heats, C_p: Dry air=1.005 kJ/kga.K, Water vapor=1.88 kJ/kgw.K

Stefan-Boltzmann Constant, σ = 5.669 X 10^{-8} W/m^2.K^4

1 met = 58.2 W/m^2

Solar angles: (l: latitude angle, h: hour angle, δ: declination, Σ: Tilt angle)

$$Altitude\ angle, \beta = \sin^{-1}(\cos l.\cos h.\cos \delta + \sin l.\sin \delta)$$

$$solar\ azimuth\ angle,\ \gamma = \cos^{-1}\left(\frac{\cos l.\sin \delta - \cos \delta.\cos h.\sin l}{\cos \beta}\right)$$

$$angle\ of\ incidence, \theta = \cos^{-1}(\sin \beta.\cos \Sigma + \cos \beta.\cos \alpha.\sin \Sigma)$$

$$wall\ solar\ azimuth\ angle,\ \alpha = 180 - \gamma \pm \xi$$

$$surface\ azimuth\ angle,\ \xi\ is\ '+'\ if\ it\ is\ to\ the\ east\ of\ south$$

Solar Radiation data for 40°N, June 21st:

$$direct\ normal\ radiation, I_{DN} = 1085.\exp\left(-\frac{0.205}{\sin \beta}\right)$$

$$diffuse\ radiation, I_d = 0.134 I_{DN}\left(\frac{1+\cos\Sigma}{2}\right)$$

$$reflected\ radiation, I_r = \rho_g I_{DN}(0.134 + \sin \beta)\left(\frac{1-\cos\Sigma}{2}\right)$$

In the above equations, ρ_g is the ground reflectivity. I_{DN}, I_d and I_r are in W/m^2.

Antoine's equation for saturation pressure (p_{sat}) of water:

$$\ln(p_{sat}) = 16.54 - \left(\frac{3985}{T - 39.00}\right); \quad p_{sat}\ in\ kPa\ \&\ T\ in\ K$$

Carrier Equation:

$$p_v = p_v^{'} - \frac{1.8(p_t - p_v^{'})(DBT - WBT)}{2800 - 1.3(1.8DBT + 32)}$$

where p_v, p'_v and p_t are vapour pressure, saturated vapour pressure at WBT and barometric pressure, respectively. DBT and WBT are in ^oC and units for pressures should be consistent.

Surface area of a human body (m^2): $A_{Du} = 0.202\, m^{0.425} h^{0.725}$, m in kg & h in m.

1a) Derive an expression for sol-air temperature and explain its physical significance. Find the sol-air temperature of a horizontal roof that is subjected to an incident solar radiation of 800 W/m^2. The absorptivity of the surface for solar radiation is equal to 0.9, outside air temperature is 35°C, long wave radiation from the surface to the surroundings is 72 W/m^2, and the convective heat transfer coefficient between the surface and surrounding air is 23.3 $W/m^2.K$. (2+1=3)

1b) The horizontal roof of an air conditioned building is made of 15 cm thick concrete (k=1.73 W/m.K). The roof has an effective surface area of 100 m^2, a decrement factor of 0.48 and a time lag of 5 hours. The inside and outside surface heat transfer coefficients of the roof are 6.3 $W/m^2.K$ and 23.3 $W/m^2.K$, respectively. The air conditioned space is maintained at 25°C. The following table shows the values of sol-air temperatures for a particular day between 8 A.M. to 7 P.M.

Time	$T_{sol\text{-}air}$, °C	Time	$T_{sol\text{-}air}$, °C	Time	$T_{sol\text{-}air}$, °C
8 A.M.	35.1	12 Noon	53.2	4 P.M.	45.5
9 A.M.	41.2	1 P.M.	54.3	5 P.M.	40.0
10 A.M.	46.2	2 P.M.	52.9	6 P.M.	33.6
11 A.M.	50.7	3 P.M.	50.3	7 P.M.	26.8

If the mean sol-air temperature for the day is 33.3°C, find a) Peak cooling load on the building due to the roof and time of occurrence of this peak load, b) Peak CLTD value, and c) Inner surface temperature of the roof at peak load. **(3+1+2=6)**

2a) Explain briefly the concepts of Solar Heat Gain Factor (SHGF) and Cooling Load Factor (CLF). **(2+1=3)**

2b) Find the heat transfer rate through a south facing, unshaded, vertical window of height 1.2 m and width 1.5 m at solar noon on June 21st (declination = 23.5°) of a building located at 40^oN. The window made of standard DSA glass has a transmittivity of 0.8 and an absorptivity of 0.12 for solar radiation and the surrounding ground has a reflectivity of 0.2. The transmittivity and absorptivity may be considered to be same for direct and diffuse radiations. Both the inside and outside surface heat transfer coefficients of the window are 3.66 $W/m^2.K$ and inside and outside temperatures are same and are equal to 24°C. **(5)**

2c) Neglecting the conduction resistance of the glass, find the window glass temperature at solar noon. **(2)**

2d) Find the heat transfer rate if the window is converted into an inset window by providing an external overhang of depth 0.3 m. For the shaded portion of the window, consider only the diffuse and reflected solar radiations. **(3)**

3a) Discuss briefly the concept of *adaptive thermal comfort* and how the concept is useful in tropical countries such as India? **(4)**

3b) Neglecting heat transfer due to respiration, and clothing effects, state whether a 1.7 m tall person weighing 60 kg and doing a light activity (M = 1.8 met) can be at neutral equilibrium or not when his skin temperature is 33°C. The space conditions are: air temperature = surrounding temperature = 29°C, air velocity,V = 0.4 m/s, relative humidity of air = 40%. The convective heat transfer coefficient between the skin and air can be estimated by the equation: h_c (W/m^2.K) = 13.5 $V^{0.6}$, where V is in m/s. Assume the skin to behave as a blackbody and make any other suitable assumptions, but state them clearly. It is given that for sustained activity, 50% of the total skin area can remain wet, but not more than that. **(4)**

3c) With neat sketches explain how the locations of external openings have to be decided if the aim is to provide as much natural ventilation as possible by utilizing the wind and stack effects. Assume the building to be non-air conditioned. **(3)**

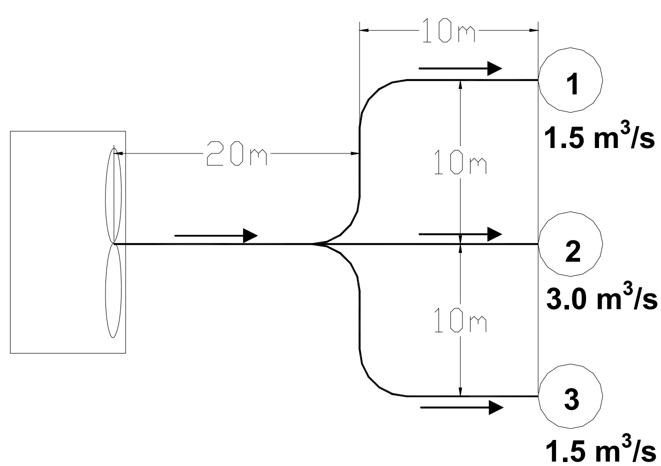
4) With neat sketches, discuss briefly the working principle, advantages, disadvantages and applications of any 2 of the following systems: **(2 X 4 = 8)**

i. Variable Air Volume Systems

ii. Air-water systems

iii. Room air conditioners

5a) Shown below is the duct layout of an air conditioning system. The fan used in the system develops a Fan Total Pressure of 120 Pa at design conditions. Using Equal Friction Method find the required diameters of the duct runs and the amount of dampering required at each of the outlets, 1,2 and 3.



Use the following equation for estimating the friction pressure drop: **(4)**

$$\left(\frac{\Delta p_f}{L}\right) = \frac{0.022243\,\dot{Q}_{air}^{\,1.852}}{D^{4.973}}$$; (Δp_f/L) in Pa/m; L and D are in m and Q_{air} is in m^3/s

For estimating momentum pressure drops, take the equivalent lengths as: upstream-to-downstream = 3m, upstream-to-branch = 12m, elbow = 3m, fan outlet = 2m and air outlet in conditioned space = 3m.

5b) Assuming the duct characteristics to remain same, what is the required FTP, if the air flow rate to each zone is reduced by 50%? **(2)**

5c) If the fan operates at 900 RPM and consumes 1200 W at design conditions, what will be the required fan speed and power consumption when the airflow rate is reduced by 50%? What are the assumptions made? **(1+1+1=3)**

End of the paper

INDIAN INSTITUTE OF TECHNOLOGY KHARAGPUR

Date : Time : 2 hours Full Marks : 30 No. of students : 15

Mid-Semester, Spring 2009 Mechanical Engineering ME2,DD & UG

Subject No. ME60096 Subject Name: Air conditioning & Ventilation

Note:

4. Use perfect gas model for calculating the required psychrometric properties.
5. Unless otherwise specified, assume the barometric pressure to be 101.3 kPa.
6. Make suitable assumptions, but state them clearly.

Given data:

Molecular weights: Dry air = 28.966 kg/kmol; Water = 18.02 kg/kmol

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Specific heats, C_p: Dry air=1.005 kJ/kga.K, Water vapor=1.88 kJ/kgw.K

Stefan_Boltzmann Constant, σ = 5.669 X 10^{-8} W/m^2.K^4

1 met = 58.15 W/m^2, 1 clo = 0.155 m^2.K/W

1 Ton of Refrigeration (TR) = 3.517 kW

Antoine's equation for saturation pressure (p_{sat}) of water:

$$\ln(p_{sat}) = 16.54 - \left(\frac{3985}{T - 39.00}\right); \quad p_{sat}\ in\ kPa\ \&\ T\ in\ K$$

Latent heat of vaporization of water, h_{fg} in terms of temperature ($0 \le t \le 40^oC$):

$$h_{fg} = 2500.95 - 2.37267\,t \quad kJ/kgw$$

Carrier Equation:

$$p_v = p_v' - \frac{1.8(p_t - p_v')(DBT - WBT)}{2800 - 1.3(1.8DBT + 32)}$$

where p_v, p_v' and p_t are vapour pressure, saturated vapour pressure at WBT and barometric pressure, respectively. DBT and WBT are in ^{o}C and units for pressures should be consistent.

Equations for Predicted Mean Vote (PMV) & Percent People Dissatisfied (PPD):

$$PMV = \left[0.303\exp(-0.036M) + 0.028\right]L$$

where M is the metabolic rate and L is the thermal load on the body(difference between internal heat generation and heat loss to the actual environment)

$$PPD = 100 - 95\exp\left[-(0.03353\ PMV^4 + 0.2179\ PMV^2)\right]$$

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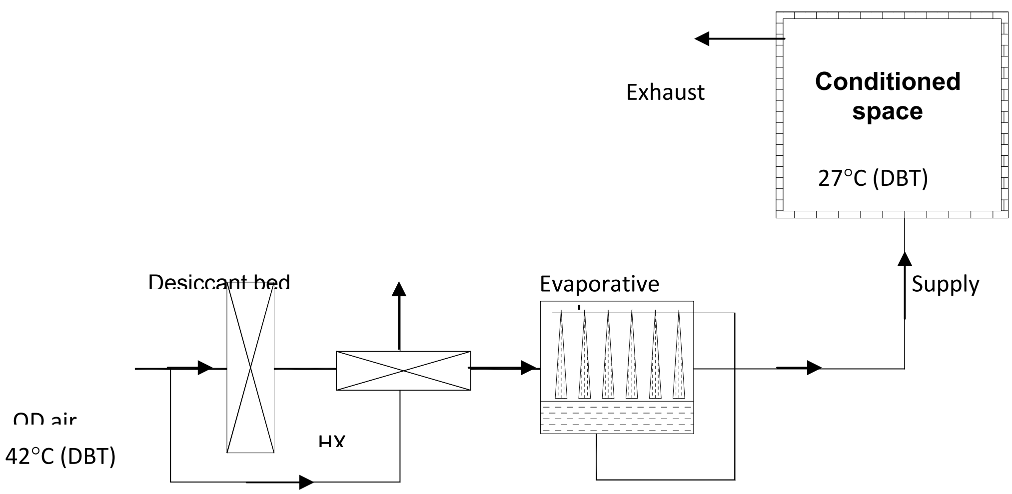
1. Find the direction and magnitude of total heat transfer (sensible + latent) when air at 40^{o}C and 27% RH flows over a wetted surface that is maintained at a constant temperature of 24^{o}C. The

wetted surface has a surface area of 0.25 m^2 and the convective heat transfer coefficient between the wet surface and the air flowing over it is 60 W/m^2.K. Show the process undergone by air on psychrometric chart. Make suitable assumptions and state them clearly. (3+1 = 4 marks)

2. Shown below is a hybrid air conditioning system that uses a desiccant bed for dehumidification of outdoor (OD) air followed by sensible cooling in a heat exchanger (HX) and cooling and humidification in an evaporative cooler. The cool air from the evaporative cooler is supplied to the conditioned space at a flow rate of 0.4 kg/s. The outdoor air is at 42^{o}C (DBT) and 28^{o}C (WBT). The dry and wet bulb temperatures of air at the exit of the heat exchanger are 45^{o}C and 20.8^{o}C, respectively. The evaporative cooler has an efficiency of 0.9, while the desiccant bed has an efficiency (defined as the ratio of reduction in humidity ratio of air as it flows through the desiccant bed to the humidity ratio at the inlet of the bed) of 0.7. Find a) the sensible and latent cooling capacities of the above system (in kW) if it maintains a conditioned space at 27^{o}C (DBT) and 65% RH, and b) heat transfer rate in the heat exchanger (HX). Is it possible to maintain the required conditions in the conditioned space by using only evaporative cooler? (4+1+1= 6 marks)



3. A condensation resistance factor $\tau = \left(\dfrac{t_s - t_c}{t_a - t_c}\right)$ is a parameter that is used in connection with condensation of water vapour on building walls. In the above expression t_s and t_c are the temperatures of the hot and cold surfaces of the wall and t_a is the dry bulb temperature of air on the hot side of the wall. The inside surface temperature of a cold storage wall is to be maintained at -15^oC when the outside ambient air is at 35^{o}C (DBT) and 40% RH. For this cold storage wall, find the minimum value of condensation resistance factor and the minimum wall thickness (in mm) required to prevent condensation. The effective thermal conductivity of the wall is 0.6 W/m.K and the design heat transfer coefficient between the outer surface of the wall and surroundings is 23 W/m^2.K. Supposing the cold storage is constructed using the minimum insulation thickness

calculated from the above data, do you expect condensation to take place when, other conditions remaining same, a) Ambient RH increases to 45%, and b) Surface heat transfer coefficient drops to 15 $W/m^2.K$ due to change in wind velocity. (4 + 2 = 6 marks)

4. Netaji auditorium has a seating capacity of 900. The design inside conditions are: 27^{o}C (DBT), humidity ratio = 0.015 kgw/kga, mean radiant temperature = 30^{o}C. The convective heat transfer coefficient and the average radiative heat transfer coefficient between the occupants and surroundings are both equal to 4.7 $W/m^2.K$. To calculate the evaporative heat transfer rate from the human body, a Lewis number of 0.85 may be assumed with a skin wettedness factor of 0.1. Assume that the clothes worn do not offer any resistance to evaporative heat transfer from skin and all the sweat generated from the body evaporates from the skin. For the occupants assume an average body area of 1.7 m^2, activity level of 1.2 met, clothing resistance of 0.5 clo and clothing area factor of 1.15. The average skin temperature is 33.8^{o}C. The average respiration rate of the occupants may be taken as 0.17 grams of air/s, and the temperature and humidity ratio of air that is breathed out are 35^{o}C and 0.032 kgw/kga, respectively. From the above data, a) find how many people inside the auditorium are likely to be dissatisfied with the prevailing thermal environment, and b) find the required cooling capacity of the air conditioning plant in TR when the total external cooling load (excluding ventilation) on the conditioned space is 30 kW. For the purpose of ventilation, outside air at a flow rate of 8 litres/person/s (measured at outside conditions) is supplied to the auditorium. The outside air is at 35^{o}C (DBT) and 50% RH. The internal load on the auditorium is due to the occupants only. (6+2 = 8 marks)

5. Answer briefly any 2 questions: (2 X 3 = 6 marks)

a) Explain the concept of adaptive thermal comfort and state how it is different from the normal thermal comfort standards developed based on heat balance.

b) Explain why people may experience sick building syndrome in modern air conditioned buildings, even though the buildings are maintained at conditions that provide thermal comfort to the occupants.

c) With suitable equations show the process followed by air on a psychrometric chart when air is humidified by: a) spraying water, and b) by adding saturated steam.

d) Discuss the concept of effective sensible heat factor (ESHF). How is it useful in air conditioning calculations?

End of the paper

INDIAN INSTITUTE OF TECHNOLOGY KHARAGPUR

Date : Time : **3 hours** Full Marks : **50** No. of students : **15**

End-Semester, Spring 2009 Mechanical Engineering **ME2,DD & UG**

Subject No. **ME60096** Subject Name: **Air conditioning & Ventilation**

Given data

Use perfect gas model for calculating the required psychrometric properties.

Unless otherwise specified, assume the barometric pressure to be **101 kPa**.

Molecular weights: Dry air = 28.966 kg/kmol; Water = 18.02 kg/kmol

Universal gas constant = 8.314 kJ/kg.K; **C_p of liquid water =** 4.18 kJ/kg.K

Specific heats, C_p: Dry air=1.005 kJ/kga.K, Water vapor=1.88 kJ/kgw.K

Antoine's equation for saturation pressure of water (p_{sat}):

$$\ln(p_{sat}) = 16.54 - \left(\frac{3985}{T - 39.00}\right)$$, p_{sat} in kPa and T in K

Solar angles:(**l**: latitude, **h**: hour angle, δ: declination, Σ: Tilt angle, ξ: wall azimuth angle)

$$Altitude\ angle, \beta = \sin^{-1}(\cos l.\cos h.\cos\delta + \sin l.\sin\delta)$$

$$solar\ azimuth\ angle(from\ north),\ \gamma = \cos^{-1}\left(\frac{\cos l.\sin\delta - \cos\delta.\cos h.\sin l}{\cos\beta}\right)$$

$$angle\ of\ incidence, \theta = \cos^{-1}(\sin\beta.\cos\Sigma + \cos\beta.\cos\alpha.\sin\Sigma)$$

$$wall\ solar\ azimuth\ angle,\ \alpha = 180 - \gamma \pm \xi$$

Solar Radiation data for June 21st:

$$I_{DN} = 1085.\exp\left(-\frac{0.205}{\sin\beta}\right),\ I_d = 0.134 I_{DN}\left(\frac{1+\cos\Sigma}{2}\right),\ I_r = \rho_g I_{DN}(0.134 + \sin\beta)\left(\frac{1-\cos\Sigma}{2}\right)$$

I_{DN}, I_d and I_r are direct normal, diffuse & reflected radiation in W/m^2, ρ_g is ground reflectance.

Velocity distribution for an isothermal, free jet through a circular opening:

$$V(x,r) = \frac{7.41\, V_o \sqrt{A_o}}{x\left[1 + 57.5\left(r^2/x^2\right)\right]^2}$$, V_o and A_o are velocity and area of opening at supply air outlet.

Answer Question No.1 and any 5 questions from the remaining

1a) From basic definitions and using ideal gas equations, show that:

$i)\quad v = \left(\frac{R_a T}{0.622 P_t}\right)(0.622 + W);$ $\qquad ii)\quad \phi = \left(\frac{P_t}{0.622 P_{sat}}\right)\left(\frac{0.622W}{0.622 + W}\right)$

where v is specific volume of moist air, W is humidity ratio, T is DBT, P_{sat} and P_t are saturated vapour pressure and total pressure, respectively, R_a is gas constant of dry air and ϕ is relative humidity. (**2+2 = 4 marks**)

1b) Using basic principles of psychrometry, explain why moist air is dehumidified when it is compressed and then stored in the storage tank of air compressor? How does this method of dehumidification compare with other methods of dehumidification? (**3 marks**)

1c) An air conditioned room is maintained at **26°C** (DBT) and **50%** relative humidity. The room has a room sensible heat factor (RSHF) of **0.65**. If air is supplied to the room at **12°C** (DBT), what should be the relative humidity of supply air so that it can take care of the room sensible and latent cooling loads? (**3 marks**)

2a) Find the total solar radiation flux incident on **June 21st**, on a west facing, vertical wall of a building at a time when the horizontal projection of sun's rays is normal to the west facing vertical wall. The building is located at **42°N** and the declination for June 21st is **23.47°**. Assume a ground reflectance of **0.2**. (**5 marks)**

2b) Find the clock time at which the sun rises and sets at Kharagpur (**23°N** latitude & **80°E** longitude) on 1st of May (declination = **14.51°**, equation of time = 2 minutes and 50 seconds). The clocks in India are set for a standard meridian of 82.5°E. (**3 marks**)

3a) Derive an expression for sol-air temperature considering both shortwave (solar) and longwave radiations. Explain how the wind speed affects sol-air temperature? (**3 marks**)

3b) The sol-air temperature for the horizontal roof of an air conditioned building is given by the equation: $t_{sol-air}(^oC) = 42 + 23.3 * \cos(15\theta - 192)$, where θ is the solar time in hours measured from the midnight (i.e., θ = 0 hours at 12'0 clock, midnight). The roof has an area of **84 m²** and is made up of **150 mm thick** concrete (**k=1.73 W/m.K**) with **6mm thick** plaster (**k=8.65 W/m.K**) on both sides of the roof. The internal and external surface conductance values for the roof are **8.3 W/m².K and 23.3 W/m².K**, respectively. The roof has a time lag of **5.2 hours** and a decrement factor of **0.6**. If the air conditioned space is maintained at a dry bulb temperature of **26°C**, find the minimum and maximum cooling loads on the building due to the roof and the corresponding solar times.

(5 marks)

4a) What are the factors that cause infiltration in buildings? Explain a practical method that can be used for measuring infiltration rate in buildings. (**1+2 = 3 marks**)

4b) In a naturally ventilated building, the air flow rates due to wind and stack effects are given by the equations: $Q_{wind}(m^3/s) = 0.55 * A * V_{wind}$ and $Q_{stack}(m^3/s) = 0.6 * A * \sqrt{2g * \Delta h_{NPL}(T_i - T_o)/T_i}$, where A is the area of opening in m², V_{wind} is wind speed in m/s, Δh_{NPL} is the difference in heights between the opening and the neutral pressure level in m, T_i and T_o are the inside and outside air temperatures in K and g is acceleration due to gravity in m^2/s. From the following input data find the area of the opening required so that the air flow rate due to natural ventilation takes care of the entire internal heat generation rate (Q_{int}) of the building. Wind speed = **25 kmp**h, Δh_{NPL} = **1.5 m**, Q_{int} =**3 kW**, T_i = **33^{o}C** and T_o= **29^{o}C**. Assume an average air density of **1.2 kg/m^3** and a c_p value of **1.02 kJ/kg.K**.

(**5 marks**)

5a) Derive Borda-Carnot equation, and then using this equation, obtain an expression for pressure loss due to sudden contraction. (**2+2 = 4 marks**)

5b) In a sudden enlargement in an air conditioning duct, the cross-sectional area increases from **1 m^2 to 2.2 m^2**. If the available static pressures in the upstream and downstream of the fitting (i.e., the sudden enlargement) are **0.5 inches of H_2O column** and **0.7 inches of H_2O column**, respectively, what is the maximum possible air flow rate through the fitting? The densities of water and air may be taken as **1000 kg/m^3** and **1.2 kg/m^3**, respectively. (**4 marks**)

6a) Define the terms Effective Draft Temperature (EDT), Air Distribution Performance Index (ADPI) and Space Diffusion Effectiveness Factor (SDEF). (**3 marks**)

6b) An isothermal, free jet enters the room through a circular outlet. What should be the velocity of air at the supply outlet, if it is required to provide a supply air flow rate of **90 litres/s** with a throw of **12 m**? Also find the entrainment ratio at throw. (**3+2 =5 marks**)

7a) Find the minimum **life cycle cost** of a ducting system by optimizing the duct diameter. Use the data given below:

a) Thickness of the duct material	: **1.5 mm**
b) Density of the duct material	: **8000 kg/m^3**
c) Cost of the ducting material	: **Rs. 12/- per kg**
d) Volumetric flow rate of air	: **2.0 m^3/s**
e) Density of air	: **1.2 kg/m^3**
f) Friction coefficient, f	: **0.02**
g) Number of operating hours	: **12000 hours**
h) Cost of electricity	: **Rs. 5 per kWh**
i) Efficiency of fan	: **60 %**
j) Total length of duct	: **150 m**

Assume the cost of fan, fan efficiency, density of air and friction coefficient to remain constant independent of the duct diameter. **(5 marks)**

7b) A fan-duct system is designed such that when the air temperature is **20^{o}C**, the mass flow rate is **3 kg/s** when the fan speed is **25 rps** and the fan motor requires **1.2 kW**. Now the operating air temperature is changed to **5^{o}C**, and the fan speed is changed so that the same mass flow rate of air prevails. What are the revised fan speed and power requirement? **(3 marks)**

End of the paper

Indian Institute of Technology Kharagpur, Department of Mechanical Engineering

ME60096 (Air conditioning & Ventilation) – Mid Semester – 2010

Duration: 2 hours **Total marks : 45**

Given data: **Barometric Pressure, p_t** = 101 kPa, **Universal gas constant**: 8.314 kJ/kmol.K

Molecular weights (kg/kmol): Dry air = 28.966; Water = 18.02

C_p(kJ/kg.K): Dry air=1.005, moist air (avg.): 1.0216; Water vapor=1.88; Liquid water = 4.18

Latent heat of vaporization of water, $h_{fg}(t)$ = 2501 –2.368*t; h_{fg} in (kJ/kg) and t in ^oC

1 met = 58.2 W/m^2, **1 clo** = 0.155 m^2.K/W; **f_{cl}** = $(A_{cl}/A_D) = 1+0.3*I_{cl}$, I_{cl} in clo

Du-Bois area, A_D = $0.202m^{0.425}h^{0.725}$; m in kg, h in m, A_D in m^2, $\sigma = 5.678 \times 10^{-8}$ W/m^2.K^4

a) Antoine's equation for saturation pressure of water (p_{sat})

$$\ln(p_{sat}) = 16.54 - \left(\frac{3985}{T - 39.00}\right)$$, p_{sat} in kPa and T in Kelvin

b) **Apjohn Equation**: $p_v = p'_v - \frac{1.8 \quad p_t(DBT - WBT)}{2700}$, DBT & WBT in ^oC

p_v is the vapour pressure, p'_v is the saturated vapour pressure at WBT and p_t is the barometric pressure (pressure units should be consistent)

c) **Saturation humidity ratio (kgw/kga)**, $W_{sat}(t)$ = 0.00364+0.000375*t, (Validity: $0.1 \le t \le 10^oC$)
$W_{sat}(t)$ = –0.000042+0.00073*t, (Validity: $11 \le t \le 21^oC$)

Part-A is compulsory. Answer any 2 questions from Part-B

Part-A

A1. Using suitable equations and the assumption of ideal gas behavior, explain the procedure for generating the psychrometric chart for a given barometric pressure. **(5 marks)**

A2. A car's interiors are kept at **23^{o}C (DBT) and 40 %** (relative humidity), while the outside conditions are: **4^{o}C (DBT) and 90 %** (relative humidity). The windshield of the car is made up of **6 mm thick**, Pyrex glass (**k = 1.01 W/m.K**). If the internal and external heat transfer coefficients between the inner and outer surfaces of the windshield and surrounding air are **8 W/m^2.K** and **60 W/m^2.K**, respectively, find whether any condensation of water vapour takes place on the windshield. If condensation takes place, state on which side (inner or outer) of windshield it occurs. Does the speed of car influence the formation of condensation in any manner? **(5 marks)**

A3. In a mixing chamber **1 kg/s** of air stream 1 at –**30^{o}C** (DBT) and **100%** (relative humidity) mixes with **1 kg/s** of air stream 2 at **25^{o}C** (DBT) and **60%** (relative humidity). Assuming the mixing process to be adiabatic, find the condition of air at the exit of mixing chamber. Show the process on psychrometric chart. **(5 marks)**

A4. In a humidifier air is humidified by bringing it in contact with a spray of hot water at **80^{o}C**. If the air enters the humidifier at **40^{o}C** (DBT) and **30%** (relative humidity), and its humidity ratio increases by **5 grams/kg of dry air**, what is the dry bulb temperature of air at the exit of the humidifier? Show the process on psychrometric chart. **(5 marks)**

A5. Based on the requirement of a neutral condition, explain briefly whether the air dry bulb temperature in a room should be increased, decreased or kept constant when:

a) Activity level increases; **b)** Air velocity increases; **c)** Surrounding temperature increases; **d)** Clothing resistance decreases; and **e)** Moisture content in room increases **(5 marks)**

Part-B

B1. A psychrometer measures dry bulb and wet bulb temperatures of air in a room as **27°C** and **18°C**, respectively. The mean radiant temperature of the room is **30°C**. The convective heat transfer coefficients between the dry bulb and air, and between the wet bulb and air are both equal to **8 W/m^2.K**. Both the dry and wet bulb sensors have emissivity values of **0.95**. The Lewis number for air may be taken as **0.90**. The dry bulb sensor is unshielded, while a perfect radiation shield is provided for the sensor of the wet bulb only. From this data, find a) The true dry bulb temperature of air, and b) Humidity ratio of air, c) What will be the temperature indicated by the wet bulb thermometer, if the wet bulb is also unshielded? Make reasonable assumptions and state and justify them clearly. **(10 marks)**

B2. A winter air conditioning system is to be designed for a sensible heat loss from the building of **40 kW** and a latent heat loss from the building of **5 kW**. The required inside conditions are **22°C** (dry bulb) and **40 %** (relative humidity). The outside conditions are **–8°C** (dry bulb) and **100 %** (relative humidity). Air is to be supplied to the room at a dry bulb temperature of **37°C**. The supply air should consist of **10 %** of outdoor air (by mass) for ventilation. The system consists of a pre-heater, steam humidifier and a re-heater. The dry bulb temperature of air increases by **10°C** in the pre-heater. In the humidifier, dry saturated steam is added at a temperature of **120°C** (**h_g=2706 kJ/kg**). Draw a schematic of the system and show the processes on psychrometric chart. Find a) Heat input to the pre-heater; b) Steam consumption in kilograms per hour; and c) Heat input to the re-heater. Verify the overall energy balance for the system **(10 marks)**

B3. A room is maintained at **25°C** (dry bulb) and **60 %** (relative humidity). The average height and weight of the occupants inside the room are **1.7 m** and **60 kg**, respectively. The average activity level of the occupants is **1.5 met** and clothing resistance (I_{cl}) is **0.5 clo**. The convective and radiative heat transfer coefficients between the occupants and the air and surroundings are **8.1 W/m^2.K and 4.7 W/m^2.K**, respectively. a) What should be the mean radiant temperature of the room so that the skin temperature of the occupants is **33°C**, and the total evaporative heat loss from the skin is **40 %** of the total heat loss from skin? b) What is the mean clothing temperature? Use the following equations for estimating heat losses due to respiration. **(10 marks)**

$$Q_{resp,sensible}\text{ (in W)} = 0.0014 * M * A_D(34-ta); \quad Q_{resp,latent}\text{ (in W)} = 0.0173 * M * A_D(5.87-p_v)$$

where M is metabolic rate in W/m^2; A_D is Du-Bois area in m^2; t_a is dry bulb temperature of air in °C, and p_v is water vapour pressure in air in kPa.

INDIAN INSTITUTE OF TECHNOLOGY KHARAGPUR

Date : Time : **3 hours** `Full Marks : **50** No. of students : **21**
End-Semester, Spring 2010 Mechanical Engineering **ME2 & 5DD**
Subject No. **ME60096** Subject Name: **Air conditioning & Ventilation**

Answer all questions

Given data

saturation pressure of water: $\ln(p_{sat}) = 16.54 - \left(\dfrac{3985}{T-39.00}\right)$; p_{sat} in kPa and T in K

Solar Radiation data for June 21st:

$$I_{DN} = 1085.\exp\left(-\frac{0.205}{\sin\beta}\right),\ I_d = 0.134 I_{DN}\left(\frac{1+\cos\Sigma}{2}\right),\ I_r = \rho_g I_{DN}(0.134+\sin\beta)\left(\frac{1-\cos\Sigma}{2}\right)$$

I_{DN}, I_d and I_r are direct normal, diffuse & reflected radiation in W/m^2, ρ_g is ground reflectance. Σ is the tilt angle and β is the altitude angle.

altitude angle, $\beta = \cos(l).\cos(h).\cos(\delta) + \sin(l).\sin(\delta)$; where l, h and δ are the latitude, hour angle and declination, respectively.

Velocity distribution for an isothermal, free jet through a circular opening:

$V(x,r) = \dfrac{7.41\ V_o\ \sqrt{A_o}}{x\left[1+57.5\left(r^2/x^2\right)\right]^2}$; V_o and A_o are velocity and area of opening at supply air outlet

Frictional pressure drop and equivalent diameters of ducts:

Frictional pressure drop through ducts: $\left(\dfrac{\Delta p_f}{L}\right) = \dfrac{0.022243\ \dot{Q}_{air}^{1.852}}{D^{4.973}}$; Δp_f in Pa; D & L in m; $\dot{Q}_{air}$ in m^3/s

Equivalent diameter of a rectangular duct (for same $\dot{Q}_{air}$ *and* $\Delta p_f/L$): $D_{eq} = \dfrac{1.3(a.b)^{0.625}}{(a+b)^{0.25}}$

==

1) An air conditioned classroom measuring **20m X 12m X 8m** has a seating capacity of **200** students. On an average each student generates **16 cm^3 of CO_2 per second**. The CO_2 concentration of outdoor air is **35 ppm** (parts per million). The air conditioned system re-circulates **75 percent** of supply air, while the remaining **25 percent** is the outdoor air. The space air distribution system is such that **20 percent** of the air supplied to the classroom by passes the occupied zone. If the maximum permissible concentration of CO_2 inside the classroom is **1000 ppm**, find the amount of outdoor air required in terms of a) litres/second/student and b) air changes per hour (ACH). **(5)**

2) To measure the natural ventilation rate in a building of internal volume **1200 m^3**, a tracer gas is injected into the building at a constant rate of **18 cm^3/s**. Find the ventilation rate, if at steady state the concentration measured in the building shows a value of **0.6 cm^3/m^3** of air. What is the time required for the tracer gas concentration inside the building to reach a value of **99 percent** of the steady-state value? Assume that tracer gas concentration before injection is 0 and there are no tracer gas sources or sinks inside the building. **(2+3 = 5)**

3) Find the heat flux through a thin, opaque wall (overall heat transfer coefficient of the wall, **U=1.2 W/m^2.K**) when the total solar radiation incident on the surface of the wall is **800 W/m^2.** The wall has an absorptivity of **0.8.** The external heat transfer coefficient is **23 W/m^2.K**, the outer and inner dry bulb temperatures are **37^{o}C and 25^{o}C**, respectively. b) Under what assumption, the answer obtained is correct? c) Account for total energy balance for the wall. **(3+1+1=5)**

4) Find the maximum solar radiation incident on the flat roof of a building on June 21st. The roof measures **21 m by 12 m**, and the building is located at 22°20'N and 87°25'E. Also find the total number of sunshine hours for this location on June 21st . Declination for June 21st is 23.5°.

(4+1 = 5)

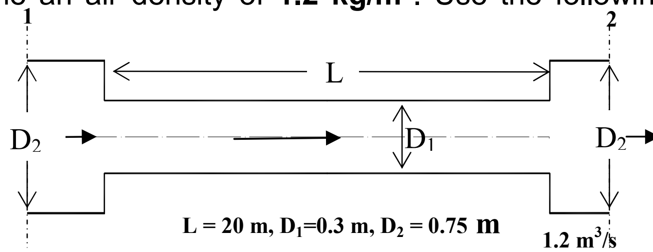
5) A duct run made of a circular duct consists of a sudden contraction, followed by **20 m** of straight duct and then a sudden expansion. If the flow rate of air through the duct run is **0.9 m³/s**, find the total pressure drop from section 1 to 2. Assume an air density of **1.2 kg/m³**. Use the following equation for contraction coefficient:

Contraction coefficient (C_c):

$C_c = 0.61 + 0.17A_r - 0.34A_r^{\ 2} + 0.56A_r^{\ 3}$

Where $A_r = A_2/A_1$.

(5)



6) A two-branch duct system of rectangular cross section is shown below. The fittings have the following equivalent length of the straight duct; upstream to branch: **8 m**, elbow: **4 m** and outlets: **8m**. There is negligible pressure loss in the straight through section (upstream to downstream) of the branch. The designer selects a fan that provides 40 Pa fan total pressure at a flow rate of **3.2 m³/s**. Taking an aspect ratio of **4** everywhere, select the dimensions of the duct system. **(5)**

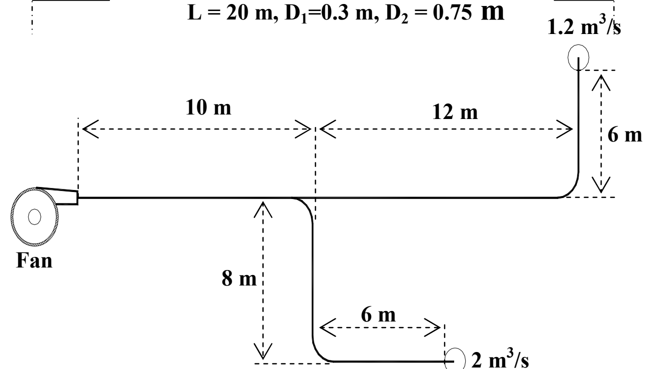


Figure for Problem 3b

7) A centrifugal fan, provides **2.4 m³/s** at a fan total pressure of **50 Pa** and, consumes **160 W** when the speed is **3000 RPM** and air temperature is **21°C**. What will be the flow rate, fan total pressure and power consumption when the changes to **4500 RPM** and the temperature is **27°C**? What is the fan efficiency? **(1+1+2+1 = 5)**

8) Write short notes on **ANY 1** of the following with neatly drawn diagrams, wherever necessary: **(5)**

a) Evaporative cooling systems for hot and humid climates
b) Multi-zone, single duct, constant volume, all-air systems
c) Radiative cooling systems

9) Using the assumption of similarity in velocity profiles and conservation of x-momentum, find how the mid-plane velocity of air for an isothermal, plane, rectangular, free-jet varies with the distance measured from the outlet. Compare this with a circular free-jet and comment on the applicability of rectangular and circular jets. **(3+1+1=5)**

10) What should be the diameter of a circular outlet, which provides an entrainment ratio of **60** at throw, when the flow rate at the outlet is **0.12 m³/s**? **(5)**

End of the paper

Indian Institute of Technology Kharagpur

Department of Mechanical Engineering

ME60096 (Air conditioning & Ventilation) – Mid Semester – 2011

Duration: 2 hours **Total marks : 60**

Given data

a) Barometric Pressure, p_t = 101 kPa, **Universal gas constant**: 8.314 kJ/kmol.K

b) Molecular weights (kg/kmol): Dry air = 28.966; Water = 18.02

c) $C_{p,avg}$ (kJ/kg.K): Dry air = 1.005, moist air : 1.0216; Water vapor = 1.88;

d) **$C_{p,avg}$ (kJ/kg.K):** Liquid water = 4.18

e) **Latent heat of vaporization of water at temperature t,**

$h_{fg}(t) = 2501 - 2.368t$; h_{fg} in (kJ/kg) and t in ^oC

f**) Antoine's equation for saturation pressure of water (p_{sat})**

$$\ln(p_{sat}) = 16.54 - \left(\frac{3985}{T - 39.00}\right)$$, p_{sat} in kPa and T in Kelvin

g) **Apjohn Equation**:

$$p_v = p'_v - \frac{1.8 \quad p_t(DBT - WBT)}{2700}$$, DBT & WBT in ^oC

p_v is the vapour pressure, p'_v is the saturated vapour pressure at WBT and p_t is the barometric pressure (pressure units should be consistent)

Make suitable assumptions, wherever necessary, and state the assumptions clearly

1. In a spray type cooling tower, the temperature of water is reduced by **4^{o}C** by exchanging heat with an air stream flowing through the cooling tower. The flow rate of water is **20 kg/s** and its average temperature is **32^{o}C**. The average dry bulb and wet bulb temperatures of air are **42^{o}C** and **28^{o}C,** respectively. The water is sprayed in the cooling tower using spray nozzles and energy exchange takes place between air and water droplets (assumed to be spherical and of uniform diameter) only. The convective heat transfer coefficient between air and the water droplets is **240 W/m^2.K**. Neglecting heat transfer between the surface of the cooling tower body and surroundings, find **a)** the required **diameter of the water droplets** to accomplish the given task. **b)** Amount of **make-up water required** to account for loss of water due to evaporation (in kg/s). Assume an average density of liquid water to be **995 kg/m^3**.

(12+4 = 16 marks)

2. An air conditioning system has to be selected for a building that has a maximum occupancy of **1000** people. The building has a sensible cooling load of **300 kW** and a latent cooling load of **100 kW**. The design inside conditions are **26^{o}C (DBT) and 50% (relative humidity)**, while the design outside conditions are **42^{o}C (DBT) and 28^{o}C (WBT).** For ventilation outside air at the rate of **8 liters per second per person** (measured at outside conditions) is required. A cooling and dehumidification coil with a bypass factor of **0.15** is to be chosen. From the data given find: **a)** Required supply air flow rate in kg/s, and **b)** Sensible and latent cooling capacities of the cooling & dehumidification coil in kW. The humidity ratio (**w_{sat}**) and enthalpy (**h_{sat}**) of saturated moist air in terms of temperature **t** (**4 < t < 14^{o}C**) can be obtained using the following equations (t is in ^{o}C for both the equations):

$$\mathbf{w_{sat} = 2.97 \times 10^{-3} + 4.78 \times 10^{-4}\, t} \text{ (kgw/kgda)}$$

$$\mathbf{h_{sat} = 7.37 + 2.22\, t} \text{ (kJ/kgda)}$$

(12+4 = 16 marks)

3. Find the outdoor air required (in liters per second) for an air conditioned building such that the concentration of CO_2 inside the building does not exceed **1000 parts per million** (ppm). The building has occupancy of **1000 persons** with an average CO_2 generation rate of **10 cm^3/second/person**. The air conditioning system is designed for a recirculation of **80%** of the room air. The air distribution system is such that **20 %** of the air supplied to the building bypasses the occupied zone. The CO_2 concentration of outdoor air is **350 ppm**. The recirculation and bypass are in terms of volumetric flow rates of air. **(10 marks)**

4. An experiment is conducted in a building to measure the outside air flow rate into the building due to natural ventilation. In the experiment Sulfur Hexafluoride (SF_6) is injected into the building for a certain amount of time so that the concentration of SF_6 at the end of injection is **20 ppm**. Now the injection is stopped and the decrease in concentration is measured as a function of time. Measurements show that after **1 hour** the concentration of CO_2 drops to **2 ppm**. Assuming well mixed conditions inside the building and concentration of SF_6 in the OD air to be 0 and no other sources of SF_6 inside the building, find the outdoor air flow rate into the building in litres per second if the internal volume of the building is **4800 m^3**.

(8 marks)

5. Write a short note on **any one** of the following:

(10 marks)

a) Adaptive thermal comfort; its meaning and relevance
b) Evaporative cooling systems for hot and humid conditions
c) Practical measurement of psychrometric properties; instruments required and precautions to be taken

End of the paper

INDIAN INSTITUTE OF TECHNOLOGY KHARAGPUR

Date : Time : **3 hours** Full Marks : **50** No. of students : **10**

End-Semester, Spring 2011 Mechanical Engineering **ME2 & 5DD**

Subject No. **ME60096** Subject Name: **Air conditioning & Ventilation**

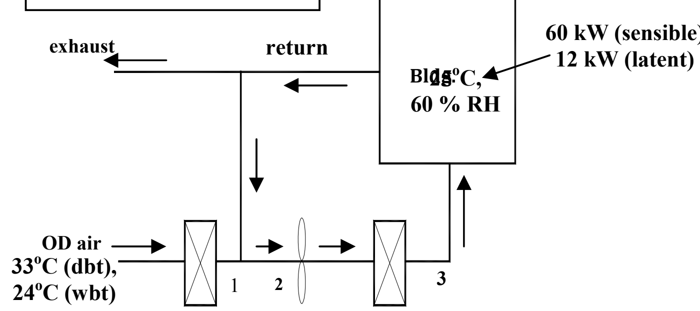
Answer all questions

1a. An evaporative cooler can cool incoming air at **27°C** (dry bulb) and **15°C** (wet bulb) to a final dry bulb temperature of **18°C**. Assuming the air and water flow rates to remain constant, to what final temperature the evaporative cooler can cool air if the dry bulb and wet bulb temperatures of air at the inlet to the cooler change to **35°C** (dry bulb) and **21°C** (wet bulb)? Justify your answer with suitable equations and assumptions. **(4)**

1b. Air at **40°C** (dry bulb) and **50 %** relative humidity is in contact with water which is at a temperature of **32°C**. Find the magnitude and direction of total heat transfer rate between air and water per unit area of the water surface. Take the convective heat transfer coefficient between air and water surface as **23 W/m^2.K**, and the Lewis number as **0.85**. **(4)**

2. An air conditioned building has design sensible and latent loads of **60 kW** and **12 kW**, respectively. The design inside and outside conditions are: **25°C and 60 %** relative humidity, and **33°C** (dry bulb) and **24°C** (wet bulb), respectively. For ventilation outside air at a mass flow rate of **1.2 kg/s** is mixed with the return air as shown in the figure. The air conditioning system consists of two cooling coils, Coil 1 and Coil 2, respectively. Coil 1 with a sensible heat factor of **0.6** is used to cool and dehumidify the outside air only, while Coil 2 handles only sensible load (sensible heat factor = 1.0). If the temperature of air supplied to the conditioned space is **12°C**, find the latent and sensible cooling capacities of Coil 1 and 2. Show the complete process on psychrometric chart. **(6+2 = 8)**

Figure for Problem 2



3a. To reduce cooling load, the exposed roof of a building is evaporatively cooled by spraying water. Write the relevant governing equation(s), boundary and initial conditions required for estimating the cooling load due to this evaporatively cooled roof. **(3)**

3b. Derive Borda-Carnot equation and, using this equation derive an expression for dynamic loss for air flow through a sudden contraction. **(6)**

3c. Explain the basis of fan laws and state how they are useful in practice. **(3)**

4. An air-water system shown in the figure given below is used to maintain the conditioned space of a building at **25°C** (dry bulb) and **60 %** (relative humidity).The building has a sensible load of **300 kW** and a latent load of **80 kW**. For ventilation purpose, **20 %** of the supply air (by mass) is outside air, which is at **42°C** (dry bulb) and **28°C** (wet bulb). The cooling coil used in the primary air system (Coil 1) handles the entire latent load on the building and has a coil ADP of **7°C** and a bypass factor of **0.15**. In the chilled water system, chilled water at **7°C** is supplied to the fan coil unit (FCU) kept inside the building. After extracting the required amount of sensible heat, the chilled water leaves the FCU at **16°C**. Find a) Required sensible and latent cooling capacities of coil 1, b) Chilled water flow rate through FCU, and c) Effective surface temperature of the fan coil unit such that no condensation takes place in FCU. **(5+2+1 = 8)**

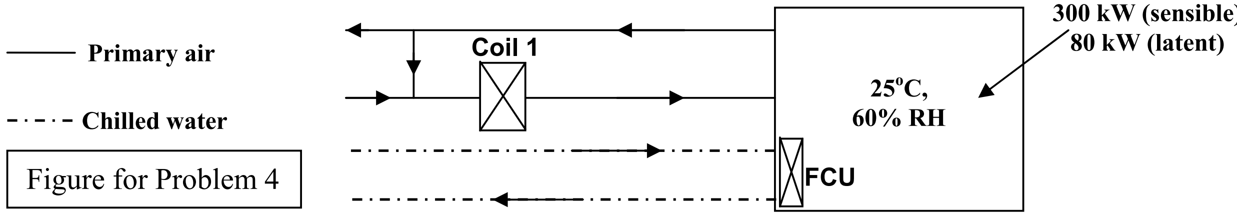


Figure for Problem 4

5a. It is required to design suitable supply air outlets for a movie theatre so that a uniform distribution is obtained in the conditioned space. The air outlets, circular in shape have to be located on the opposite walls separated by a distance of **16 m**. The total supply air flow rate is **12 m^3/s**. If the maximum allowable velocity at the supply air outlet is **6 m/s**, find the size and number of outlets required. Also find the entrainment ratio at the end of throw (V_{max} = 0.25 m/s). State the assumptions made while arriving at the above answers? **(5+2 = 7)**

5b. A dual duct, all air system is used to air condition a building which has a peak sensible cooling load of **150 kW** and a peak sensible heating load of **60 kW**. The room-to-supply air temperature difference is **12 K** for cooling and **28 K** for heating. The maximum allowable velocity of air in both cold and hot air ducts is **9 m/s**. The index run of the cold air duct has an equivalent length of **180 m**, while the index run of hot air duct has an equivalent length of **75 m**. Assuming uniform diameter throughout the index runs, find a) Minimum diameters of cold and hot air ducts, b) Using Equal Friction Method, find the maximum power rating of the electric motor that drives the fan in kW, assuming an overall fan-motor efficiency of **0.6**. Take the density of air as **1.15 kg/m^3** and the specific heat to be **1.02 kJ/kg.K**. **(5+2 = 7)**

Given data: Unless otherwise specified, use the following data

Barometric pressure,p_t = **101 kPa**, Latent heat of vaporization of water at 0^{o}C = **2501 kJ/kg,**

Molecular Weights: Dry air: **28.97 kg/kmol**, water: **18.03 kg/kmol.**

Specific heats: Dry air: **1.005 kJ/kga.K**, moist air: **1.0216 kJ/kga.K**, water vapour: **1.88 kJ/kgw.K**

$$\text{Saturation pressure of water: } \ln(p_{sat}) = 16.54 - \left(\frac{3985}{T-39}\right);\ p_{sat}\ \text{in kPa};\ T\ \text{in K}$$

$$\text{Water vapour pressure of moist air: } p_v = p_{sat(wbt)} - \frac{1.8p_t(dbt - wbt)}{2700};$$

dbt and wbt in ^{o}C, pressure units to be consistent

Velocity distribution for isothermal, free jets through a circular opening:

$$V(x,r) = \frac{7.41\ \ V_o\ \ \sqrt{A_o}}{x\left[1+57.5\left(r^2/x^2\right)\right]^2};\ \text{V}_o \text{ and A}_o \text{ are velocity and area of supply air outlet}$$

Frictional pressure drop for air flow through a circular duct:

$$\text{Frictional pressure drop through ducts: } \left(\frac{\Delta p_f}{L}\right) = \frac{0.022243\ \ \dot{Q}_{air}^{\ 1.852}}{D^{4.973}};\ \ \Delta p_f \text{ in Pa; D \& L in m; } \dot{Q}_{air} \text{ in m}^3\text{/s}$$

End of the paper

INDIAN INSTITUTE OF TECHNOLOGY KHARAGPUR

Date : Time : **2 hours** Full Marks : **60** No. of students : **35**

Mid-Semester, Spring 2012 Mechanical Engineering **ME2 & 5DD**

Subject No. **ME60096** Subject Name: **Air conditioning & Ventilation**

Given data: **Molecular weights:** Dry air = 28.966 kg/kmol; Water = 18.02 kg/kmol

Specific heat, C_p: Dry air=1.005 kJ/kga.K, Water vapor=1.88 kJ/kg.K, moist air (avg)=1.0216 kJ/kga.K

Specific heat of Liquid water = 4.18 kJ/kg.K, **Universal gas constant** = 8.314 kJ/kmol.K

Latent heat of vaporization of water = 2501 kJ/kg, **Barometric Pressure, p_t** = 101 kPa, **1TR** = 3.517 kW

a) Antoine's equation for saturation pressure of water (p_{sat})

$$\ln(p_{sat}) = 16.54 - \left(\frac{3985}{T - 39.00}\right),\ p_{sat} \text{ in kPa and T in Kelvin}$$

b) Apjohn Equation: $p_v = p'_v - \dfrac{1.8 p_t (DBT - WBT)}{2700}$, DBT & WBT in ^oC

p_v is the vapour pressure, p'_v is the saturated vapour pressure at WBT and p_t is the barometric pressure (pressure units should be consistent)

c) Saturation humidity ratio (kgw/kga), $W_{sat}(t) =$ **$0.00359+0.00028*t$**, (Validity: $-10 \le t \le 0^oC$)

$W_{sat}(t) =$ **$0.00364+0.000375*t$**, (Validity: $0.1 \le t \le 10^oC$)

$W_{sat}(t) =$ **$-0.000042+0.00073*t$**, (Validity: $10 < t \le 21^oC$)

d) Enthalpy (kJ/kgw) of superheated steam at temperature t (^oC) = **$2501+1.88*t$**

e) Du-Bois area of human body, A_D = **$0.202m^{0.425}h^{0.725}$**; m in kg, h in m, A_D in m^2

Important Note: Either psychrometric equations or psychrometric chart can be used for performing calculations. If psychrometric chart is used then the state points and processes have to be clearly marked on the chart, and the chart should be submitted along with the answer paper

Answer all questions

1a) Prove that on the psychrometric chart the exit state of moist air in contact with a cooling and dehumidification coil lies on the straight line joining the inlet state of air to the saturation condition corresponding to the apparatus dew point (ADP) of the cooling coil. State the assumptions made while arriving at this conclusion. **(6)**

1b) An air conditioning system for a hospital uses **100%** outdoor air, which is at **42^{o}C** (DBT) and **28^{o}C** (WBT). The sensible and latent loads on the conditioned space maintained at **26^{o}C** (DBT) and **50%** relative humidity are **53 kW and 25 kW**, respectively. Air is supplied to the conditioned space at DBT of **12^{o}C**. If chilled water is used as coolant in the cooling and dehumidifying coil, find whether a conventional system or a system with reheating is required for this purpose. Justify your answer with suitable reasoning. **(9)**

2a) Air at **26^{o}C** (DBT) and **50%** relative humidity flows through a humidifier in which superheated steam at **150^{o}C** is added such that the humidity ratio of air increases by **10 grams/kg of dry air**. Find the dry bulb temperature of air at the exit of the humidifier. **(6)**

2b) In one element of a counterflow type cooling tower, the temperature of **12 kg/s** of water decreases from **35^{o}C** to **33^{o}C** as it exchanges energy with **16 kg/s of air**. The condition of air at the inlet to the element is **42^{o}C** (DBT) and **28^{o}C** (WBT). If the convective heat transfer coefficient between water and air is **480 W/m^2.K**, find the required heat transfer area of the element. What is the dry bulb temperature of air at the exit of the element? **(9)**

3) A human being with a body mass of **60 kg** and height of **1.71 m** is in a conditioned space that is at **26^{o}C** (DBT) and **50%** relative humidity with an average air velocity (V_{air}) of **0.25 m/s.** The activity level of the person is **1.5 met** (**1 met = 58.15 W/m^2**).The average skin temperature is **34^{o}C** while the surrounding surface temperature is **29^{o}C**. The radiative heat transfer coefficient between the human being and surrounding surfaces is **4.7 W/m^{2}K**. The convective heat transfer coefficient (h_c in W/m^{2}K) is given as a function of surrounding air velocity (V_{air} in m/s) as: $\mathbf{h_c = 8.3V_{air}^{0.6}}$. The heat transfer resistance offered by the clothes worn by the human being is **0.5 clo** (**1 clo = 0.155 m^{2}K/W**) and the surface area ratio of the person with and without clothes is **1.15**. The clothes do not offer any resistance to evaporation from skin. The respiration rate of the human being is **0.2 grams/s** and the temperature and humidity ratio of air that is breathed out are **35^{o}C and 0.032 kgw/kga**, respectively. Find a) Average clothing temperature, b) whether the human being finds the thermal environment of the conditioned space comfortable or not. Studies show that human beings may not be comfortable if the skin wettedness ratio is greater than **0.10**. c) If the person is not comfortable suggest a simple and practical method which can make the surroundings more comfortable. Justify your answer briefly. Assume a Lewis number of **0.9** and a Lewis Ratio (LR) of **16.5 K/kPa**. **(5+5+5 = 15)**

4) An air conditioned building has a total cooling load of **500 kW** with a room sensible heat factor of **0.75**. The inside and outside design conditions are: **26^{o}C** (DBT)/**50%** relative humidity and **42^{o}C** (DBT)/**28^{o}C** (WBT), respectively. The cooling and dehumidification coil has bypass factor of **0.1**. For ventilation purposes, outdoor air at a flow rate of **5 m^3/s** (measured at room conditions) is supplied to the building. Find a) The required supply air conditions (flow rate, DBT and humidity ratio) and b) Required sensible and latent cooling capacities of the air conditioning system in Tons of Refrigeration (TR). **(6+9 = 15)**

End of the paper

INDIAN INSTITUTE OF TECHNOLOGY KHARAGPUR

Date : Time : **3 hours** Full Marks : **60** No. of students : **35**

End-Semester, Spring 2012 Mechanical Engineering **M.Tech & DD**

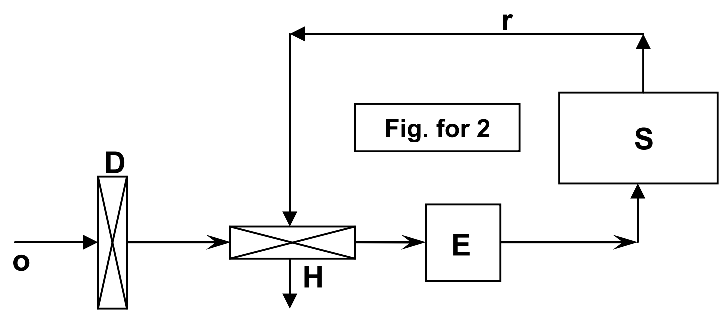
Subject No. **ME60096** Subject Name: **Air conditioning & Ventilation**

Answer all questions

The attached psychrometric chart may be used, if required

1. The dry and wet bulb thermometers of a psychrometer indicate **27°C** and **18°C**, respectively. The convective heat transfer coefficient between the thermometer bulbs and surrounding air is **4.7 W/m²K**. If the mean radiant temperature of the surroundings is **30°C**, and emissivity of both the bulbs is **0.9**, find a) the true dry bulb temperature of air, and b) humidity ratio of air. Assume that the sensing bulbs of the dry and wet bulb thermometers do not have radiation shields. However the sensing bulbs of dry and wet bulb thermometers do not see each other. (Stefan Boltzmann constant, σ **= 5.678x10^{-8} W/m^2.K^4**). **(2+4 = 6)**

2. Shown here is a hybrid air conditioning system that uses a desiccant wheel (**D**) for dehumidification of outdoor (**o**) air followed by sensible cooling in heat exchanger (**HX**) using room return air (**r**), and cooling and humidification in an evaporative cooler (**E**). The cool air from the evaporative cooler is supplied to the conditioned space (**S**) at a flow rate of **0.5 kg/s**. The outdoor air (**o**) is at **42°C (DBT) and 28°C (WBT)**. The heat exchanger effectiveness is such that the wet bulb temperature at the exit of heat exchanger is **18°C**. The evaporative cooler (E) has an efficiency of **0.9**, while the desiccant wheel (D) has an efficiency (defined as the ratio of reduction in humidity ratio of air as it flows through the desiccant bed to the humidity ratio at the inlet of the bed) of **0.60**. Find a) the sensible and latent cooling capacities of the above system (in kW) if it maintains a conditioned space at **27°C (DBT) and 60% RH**, and b) heat transfer rate in the heat exchanger, HX in kW.



(6+2 = 8)

3. Find the total solar radiation flux incident on **June 21st**, on a west facing, vertical wall of a building at a time when the horizontal projection of sun's rays is normal to the west facing vertical wall. The building is located at **32°N** and the declination for June 21st is **23.5°**. Assume a ground reflectance of **0.2**. **(8)**

4. A building has a U-value of **0.8 W/m².K** and a total exposed surface area of **400 m²**. The building is subjected to an external load (only sensible) of **3 kW** and an internal load of **1.8 kW** (only sensible). If the required internal temperature is **25°C**, using the concept of balanced outdoor temperature, state whether a cooling or a heating system is required when the ambient air temperature is **8°C**. How the results will change, if the U-value of the building is reduced to **0.6 W/m².K**? What conclusions can be drawn from this example regarding the effect of insulation on cooling and heating periods of buildings? **(3+1+2 = 6)**

5. An air conditioned building located in Kharagpur (latitude,l = **22.3° N**) has an opaque flat roof with an area of **72 m²**. The roof is made out of a material that has a U-value of **2.4 W/m².K** and negligible thermal capacity (negligible thermal lag and decrement). The internal and external surface heat transfer coefficients for the roof are **8.3 W/m².K** and **23.3 W/m².K**, respectively. The design inside and outside dry bulb temperatures are **25°C** and **37°C**, respectively. The external surface of the roof has a solar absorptivity of **0.9** (same for direct and diffused radiation). The effect of longwave radiation between the roof and sky is equivalent to a reduction of **3 K** in the sol-air temperature. Using the given data and ASHRAE clear sky model a) estimate the cooling load on the building due to the roof at solar noon on June 21st (declination,δ = **23.5°**), b) the internal and external surface temperatures of the roof. **(6+4 = 10)**

6. In a naturally ventilated building, the air flow rates due to wind and stack effects are given by the equations: $Q_{wind}(m^3/s) = 0.55 * A * V_{wind}$ *and* $Q_{stack}(m^3/s) = 0.6 * A * \sqrt{2g * \Delta h_{NPL}(T_i - T_o)/T_i}$, where A is the area of opening in m^2, V_{wind} is wind speed in m/s, Δh_{NPL} is the difference in heights between the opening and the neutral pressure level in m, T_i and T_o are the inside and outside air temperatures in K and g is acceleration due to gravity in m/s^2. From the following input data find the area of the opening required so that the air flow rate due to natural ventilation takes care of the entire internal sensible heat generation rate (Q_{int}) of the building. Wind speed = **12 km/h**, Δh_{NPL} = **1.5 m**, Q_{int} = **1.8 kW**, T_i = **305 K** and T_o= **300 K**. Assume an average air density of **1.2 kg/m³** and a c_p value of **1.02 kJ/kg.K**. **(6)**

7. In a straight, **30m** long, horizontal duct of uniform cross-section, air flows at a rate of **1.6 m³/s**. If the velocity of air through the duct is **8 m/s**, find the required fan power when, a) a circular duct is used, and b) a rectangular duct of aspect ratio **1:4** is used. Take the efficiency of the fan to be **0.7**. If a GI sheet of **0.5 mm thick** with a density of **8000 kg/m³** is used to construct the duct, how many kilograms of sheet metal is required for circular and rectangular cross sections? Use the following equations: **(6+2 = 8)**

$$\frac{\Delta P_f}{L} = \frac{0.022243 Q^{1.852}}{D_{eq}^{4.973}}; \; D_{eq} = \frac{1.3(a.b)^{0.625}}{(a+b)^{0.25}}$$

, where Q is the flow rate in m^3/s, $\Delta P_f/L$ is the frictional pressure drop per unit length (Pa/m), D_{eq} is the equivalent diameter of the rectangular duct of sides a and b.

8. In an air conditioning system, **1.8 m^3/s** of air (density = **1.2 kg/m^3**) flows through a straight circular duct that has a cross sectional area of **0.25 m^2** for a length of **8 m**, followed by a sudden expansion to an area of **0.6 m^2**. The total length of the straight duct is **18 m**. Find a) the total pressure drop (frictional + dynamic) for the duct system, and b) Static Regain factor for the expansion. Use the equation given in Problem 6, for estimating the frictional pressure drop and Borda-Carnot equation for estimating the dynamic loss due to sudden expansion. Assume dynamic loss coefficients of **0.03** and **1.0** for duct entry and exit. **(6+2 = 8)**

Given data: Barometric pressure,p_t **= 101 kPa**, Latent heat of vaporization of water at 0^oC = **2501 kJ/kg,** Molecular Weights**:** Dry air: **28.97 kg/kmol**, water: **18.03 kg/kmol**.

Specific heats: Dry air: **1.005 kJ/kga.K**, moist air: **1.0216 kJ/kga.K**, water vapour: **1.88 kJ/kgw.K**

$$saturation\ pressure\ of\ water\ vapour, p_{sat} = exp^{\left(16.54 - \frac{3985}{T-39}\right)}$$, p_{sat} in kPa and T is in K

$$water\ vapour\ pressure\ of\ moist\ air, p_v = p_{sat(wbt)} - \left(\frac{1.8 p_t (dbt - wbt)}{2700}\right)$$, dbt and wbt in ^oC

Solar angles:(**l**: latitude, **h**: hour angle, δ: declination, Σ: Tilt angle, ξ: wall azimuth angle)

$$Altitude\ angle, \beta = \sin^{-1}(\cos l.\cos h.\cos \delta + \sin l.\sin \delta)$$

$$solar\ azimuth\ angle(from\ north),\ \gamma = \cos^{-1}\left(\frac{\cos l.\sin \delta - \cos \delta.\cos h.\sin l}{\cos \beta}\right)$$

$$angle\ of\ incidence, \theta = \cos^{-1}(\sin \beta.\cos \Sigma + \cos \beta.\cos \alpha.\sin \Sigma)$$

$$wall\ solar\ azimuth\ angle,\ \alpha = 180 - \gamma \pm \xi$$

ASHRAE Clear sky model for Solar Radiation data for June 21^{st}:

$$I_{DN} = 1085.\exp\left(-\frac{0.205}{\sin\beta}\right),\ I_d = 0.134 I_{DN}\left(\frac{1+\cos\Sigma}{2}\right),\ I_r = \rho_g I_{DN}(0.134 + \sin\beta)\left(\frac{1-\cos\Sigma}{2}\right)$$

I_{DN}, I_d and I_r are direct normal, diffuse & reflected radiations in W/m^2, ρ_g is ground reflectance.

End of the paper

INDIAN INSTITUTE OF TECHNOLOGY KHARAGPUR

Date : Time : **2 hours** Full Marks : **60** No. of students : **20**

Mid-Semester, Spring 2013 Mechanical Engineering **UG/PG/DD**

Subject No. **ME60096** Subject Name: **Air conditioning & Ventilation**

Given data: **Molecular weights:** Dry air = 28.966 kg/kmol; Water = 18.02 kg/kmol

Specific heat, C_p: Dry air=1.005 kJ/kga.K, Water vapor=1.88 kJ/kg.K, moist air (avg)=1.0216 kJ/kga.K

Specific heat of Liquid water = 4.18 kJ/kg.K, **Universal gas constant** = 8.314 kJ/kmol.K

Latent heat of vaporization of water = 2501 kJ/kg, **Barometric Pressure, p_t** = 101 kPa, **1TR** = 3.517 kW

a) Antoine's equation for saturation pressure of water (p_{sat})

$$\ln(p_{sat}) = 16.54 - \left(\frac{3985}{T - 39.00}\right)$$, p_{sat} in kPa and T in Kelvin

b) Apjohn Equation: $p_v = p'_v - \frac{1.8 p_t (DBT - WBT)}{2700}$, DBT & WBT in ^oC

p_v is the vapour pressure, p'_v is the saturated vapour pressure at WBT and p_t is the barometric pressure (pressure units should be consistent)

c) Saturation humidity ratio (kgw/kga), $W_{sat}(t)$ = **0.00359+0.00028*t**, (Validity: $-10 \le t \le 0^oC$)

$W_{sat}(t)$ = **0.00364+0.000375*t**, (Validity: $0.1 \le t \le 10^oC$)

$W_{sat}(t)$ = **–0.000042+0.00073*t**, (Validity: $10 < t \le 21^oC$)

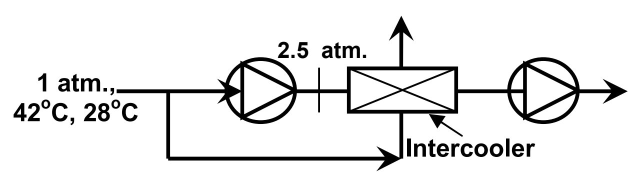
d) Enthalpy of superheated steam h_v (kJ/kgw) = ***2501+1.88*t,*** where t is temperature of water in ^oC

e) Latent heat of vaporization of water, h_{fg} (kJ/kgw) **=** ***2501–2.36*t***, where t is temperature of water in ^oC

Answer all questions

Make suitable assumptions/approximations, state and justify them clearly

1) Ambient air at **42^{o}C** (DBT) and **28^{o}C** (WBT) is compressed adiabatically to a pressure of **2.5 atm.** in the 1st stage of a 2-stage air compressor. The compressed air after 1st stage is cooled in an intercooler placed between the 1st and 2nd stage compressors by using ambient air as shown in the figure. Find whether there is any possibility of condensation of water vapour in the intercooler? **(6)**



2) Measurements on a room air conditioner show that the dry and wet bulb temperatures of the air supplied to the room are **15^{o}C** and **12^{o}C**, respectively, while the supply air flow rate is **0.24m^3/s**. If the air conditioner maintains the room at **26^{o}C** (dry bulb) and **50%** (relative humidity) and consumes **2.0 kW** of electrical power, find the sensible, latent, total cooling capacities and COP of the air conditioner. Assume **100 %** recirculation of room air without any outdoor air for ventilation. **(10)**

3) A winter air conditioning system is to be designed to provide for a building sensible and latent heat losses of **80 kW** and **10 kW**, respectively. The required inside conditions are **22°C** (dry bulb) and **40 %** (relative humidity). The outside conditions are **–10°C** (dry bulb) and **100 % (**relative humidity). Air is to be supplied to the room at a dry bulb temperature of **40°C**. The supply air should consist of **10 %** of outdoor air (by mass) for ventilation. The system consists of a pre-heater, a steam humidifier and a re-heater. Outdoor air is first preheated in the pre-heater before mixing it with the re-circulated room air. The dry bulb temperature of air increases by **16°C** in the pre-heater. In the humidifier, dry saturated steam is added at a temperature of **120°C**. Find a) Mass flow rate of air supplied to the building, b) Heat input to the pre-heater; c) Steam consumption in kilograms per hour; and d) Heat input to the re-heater. Verify the overall energy balance for the system. Represent the complete process on a psychrometric chart.

(12)

4) Air is cooled and dehumidified in a parallel flow type, chilled water spray washer by bringing it in contact with **40 kg/s** of chilled water sprayed at a temperature of **8°C**. The chilled water exchanges heat and mass with air which enters the spray washer at a mass flow rate of **50 kg/s**, inlet dry bulb temperature of **26°C** and a relative humidity of **50 %** (dew point ≈ **14.75°C**). Assuming the interfacial area for heat and mass transfer between water spray and air to be infinite, find the maximum possible cooling capacity of this spray washer.

(10)

5) A building has sensible and latent cooling loads of **100 kW** and **25 kW**, respectively. The conditioned space of the building is to be maintained at **26°C (DBT)** and **18°C** (WBT) by using a cooling and dehumidification coil that has a bypass factor of **0.1**. Outdoor air at a flow rate of **1200 litres/s** (measured at outdoor conditions) is to be supplied to the building to meet the ventilation requirements. The design outdoor conditions are: **42°C** (DBT) and **28°C** (WBT). Find a) Supply air flow rate in kg/s, b) Sensible and latent cooling capacity of the cooling & dehumidification coil, and c) Supply air temperature and humidity ratio. **(12)**

6) Using suitable control volume equations explain how the size of a counter-flow type cooling tower can be calculated once the air and water flow rates, inlet conditions of air and inlet and outlet temperatures of water are specified. State clearly all the assumptions made while arriving at the equations. Explain the terms **range**, **approach** and **NTU** (Number of Transfer Units) with reference to the cooling tower. **(10)**

End of the paper

INDIAN INSTITUTE OF TECHNOLOGY KHARAGPUR
Department of Mechanical Engineering

Date : Time : **3 hours** Full Marks : **80** No. of students : **20**

End-Semester, Spring 2013 Mechanical Engineering

UG/PG/DD

Subject No. **ME60096** Subject Name: **Air conditioning & Ventilation**

Given data: **Molecular weights:** Dry air = 28.966 kg/kmol; Water = 18.02 kg/kmol

Specific heat, C_p: Dry air=1.0 kJ/kga.K, Water vapor=1.88 kJ/kg.K, moist air (avg)=1.02 kJ/kga.K

Specific heat of Liquid water = 4.18 kJ/kg.K, **Universal gas constant** = 8.314 kJ/kmol.K

Latent heat of vaporization of water = 2501 kJ/kg, **Barometric Pressure, p_t** = 101 kPa, **1TR** = 3.517 kW

Saturation pressure of water, p_{sat}: $\ln(p_{sat}) = 16.54 - \left(\frac{3985}{T-39.00}\right)$; p_{sat} *in kPa & T in K*

Humidity ratio of saturated air, w_{sat} = *–0.0191+0.00156*DBT*, where w_{sat} is in kgw/kga, **24 ≤ *DBT* ≤ 36°C**

Answer all questions. Make suitable assumptions wherever necessary and state them clearly

1a) Starting from the fundamental principles, derive the heat balance equation for a human being who is at **thermally neutral** condition. Include all the possible modes of energy transfer in the heat balance equation. State what other conditions have to be satisfied for this human being to be **thermally comfortable**. From the thermal comfort equation derived above, list all the relevant environmental, personal and physical parameters on which the thermal comfort of the human being depends? Of these, which parameter(s) can be controlled easily by the air conditioning engineer? **(6+3+1=10)**

1b) State and describe briefly the methods used in practice to provide acceptable **indoor air quality** in occupied spaces. **(6)**

2a) A condensation resistance factor $\tau = \left(\frac{t_s - t_c}{t_a - t_c}\right)$ is a parameter that is used in connection with condensation of water vapour on building walls. In the above expression t_s and t_c are the temperatures of the hot and cold surfaces of the wall and t_a is the dry bulb temperature of air on the hot side of the wall. The inside surface temperature of a cold storage wall is to be maintained at –**15°C** when the outside ambient air is at **35°C (DBT) and 40% RH**. For this cold storage wall, find the minimum value of condensation resistance factor and the minimum wall thickness (in mm) required to prevent condensation. The effective thermal conductivity of the wall is **0.9 W/m.K** and the design heat transfer coefficient between the outer surface of the wall and surroundings is **23 W/m².K**. **(6)**

2b) At a particular instance, the thin, flat roof of a building of area **120 m²** is exposed to a total solar radiation flux of **900 W/m²**, of which **75%** is direct and **25%** is diffuse radiation. The absorptance of the surface to both direct and diffuse radiation is **0.9**. The ambient dry and wet bulb temperatures and humidity ratio at this instance are **35°C, 23.8°C and 0.014 kgw/kga**, respectively, while the inside air temperature is **28°C** (dry bulb). The external and internal heat transfer coefficients are **23 W/m^2.K and 6 W/m^2.K,** respectively. The thermal capacity and heat transfer resistance of the roof and the long-wave radiation from the roof may be assumed to be negligible. From this data calculate the rate at which heat is transferred to the inside air, when:

a) The roof is completely exposed without any insulation
b) An insulation with an R-value of **0.5 m^2.K/W** and negligible thermal capacity is added
c) The roof is not insulated but is shaded such that it receives only diffuse radiation
d) The roof is not insulated but water is sprayed on the exposed roof such that evaporation takes place over the entire roof without any run-off of water **(2X3 + 4 = 10)**

3a) Explain briefly why for houses located in northern hemisphere it is preferable to locate the windows on south facing walls. A glass window of **3 m^2** area is subjected to a total solar radiation flux of **600 W/m^2** at a particular instance of time. The average transmittance and absorptance of glass for solar radiation are **0.8 and 0.12,** respectively. At steady state, the glass rejects **60%** of the absorbed radiation to the outside. The window has a U-value of **5.9 W/m^2.K**, a shading coefficient of **0.8** and a CLF value of **0.83**. Find the cooling load on the building due to this window, when the inside and outside dry bulb temperatures are **26°C and 42°C**, respectively. **(2+4 = 6)**
3b) Find the sol-air temperature of a horizontal roof that is subjected to an incident solar radiation of **900 W/m^2**. The absorptance of the surface for solar radiation is equal to **0.9**, outside air temperature is **33°C**, long wave radiation from the roof to the surroundings is **90 W/m^2**, and the convective heat transfer coefficient between the surface and surrounding air is **23 W/m^2.K**. **(4)**
3c) The horizontal roof of an air conditioned building is made of **15 cm** thick concrete (**k=1.73 W/m.K**). The roof has an effective surface area of **120 m**^{2}, a decrement factor of **0.48** and a time lag of **5 hours**. The inside and outside surface heat transfer coefficients of the roof are **8 W/m^2.K** and **23 W/m^2.K**, respectively. The air conditioned space is maintained at **25°C**. The following table shows the calculated values of sol-air temperatures for a particular day between 8 A.M. to 7 P.M.

Time	$T_{sol-air}$, °C	Time	$T_{sol-air}$, °C	Time	$T_{sol-air}$, °C
8 A.M.	35.1	12 Noon	53.2	4 P.M.	45.5
9 A.M.	41.2	1 P.M.	54.3	5 P.M.	40.0
10 A.M.	46.2	2 P.M.	52.9	6 P.M.	33.6
11 A.M.	50.7	3 P.M.	50.3	7 P.M.	26.8

If the mean sol-air temperature for the day is **33.3°C**, find the peak cooling load on the building due to the roof, time of occurrence of this peak load and the inner surface temperature of the roof at peak load. (**6**)
4a) A large function hall has to be maintained at **26°C** (DBT) and **0.01** kgw/kga (humidity ratio) when the outdoor conditions are **35°C (**DBT**)** and **0.016** kgw/kga (humidity ratio). The total external load on the function hall consists of **16 kW** (sensible load) and **8 kW** (latent load). The occupancy is **200** with sensible and latent heat gains of **60 W** and **40 W** per occupant, respectively. The total lighting and other sensible internal equipment load is **5 kW**. Ventilation is required at the rate of **8 liters/s/occupant** (air density = **1.16 kga/m^3**). Estimate the required cooling capacity of the air conditioning plant in TR and the coil sensible heat factor. Based on the value of coil sensible heat

factor, State what practical problems may be encountered with this design and a possible solution to the problem. **(7+1+2 = 10)**

4b) The required amount of air for the above system is to be supplied to the air conditioned space at a supply temperature of **12°C** and density of **1.2 kg/m³** from an air handling room that is located **15 m** away. A uniform duct of square cross-section is used for supplying this air to the conditioned space. The duct is to be designed using equal friction method with a frictional pressure drop of **1.2 Pa/m**. The total dynamic loss from the fan outlet to the supply air outlet is equal to an equivalent straight duct length of **60 m**. Find a) The required size of the duct and velocity of air through the duct, b) The required fan power input assuming a fan efficiency of **0.7**. The frictional pressure drop ($\Delta P_f/L$ in Pa/m) in terms of the volumetric air flow rate (Q in m^3/s) and diameter (d_{eq}), and equivalent diameter (d_{eq}) of a rectangular duct of sides *a and b* (in m) are given by: **(5+1 = 6)**

$$\frac{\Delta p_f}{L} = \left[\frac{0.022243Q^{1.852}}{{d_{eq}}^{4.973}}\right]; \qquad d_{eq} = \left[\frac{1.3(ab)^{0.625}}{(a+b)^{0.25}}\right]$$

5a) An air-water system shown in the figure given below is used to maintain the conditioned space of a building at **26°C** (dry bulb) and **0.01 kgw/kga** (humidity ratio).The building has a sensible load of **300 kW** and a latent load of **80 kW**. For ventilation purpose, **10 %** of the supply air (by mass) is outside air, which is at **35°C** (dry bulb) and **0.016 kgw/kga** (humidity ratio). The cooling coil used in the primary air system (Coil 1) handles the entire latent load on the building and has a coil ADP of **7°C** and a bypass factor of **0.1**. In the chilled water line, the chilled water temperature rises by **7K** due to heat transfer as it flows through the fan coil unit (FCU) kept inside the building. Find a) Required sensible and latent cooling capacities of Coil 1, and b) Required chilled water flow rate through FCU. **(6+2 = 8)**

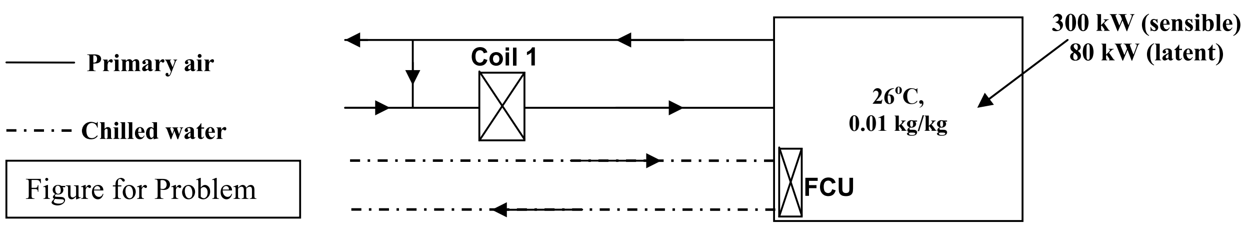


Figure for Problem

5b) What is the maximum allowable airflow rate through a circular opening of diameter **3 cm**, if the throw is not to exceed **6 m**? What is the total flow rate of air (supplied + entrained) at a distance of **3 m** from this opening? The velocity distribution for isothermal, free air jets through a circular opening is given by the equation:

$$V(x,r) = \frac{7.41V_o\sqrt{A_o}}{x\left[1+57.5\left({r^2}/{x^2}\right)\right]^2}$$

where V_o and A_o are the air velocity at the outlet and area of the outlet, respectively. **(3+5 = 8)**

End of the paper