Question 1 Hints

The tower coefficient is presented by $\frac{KaV}{\dot{m}_w}$

Tower coefficient could also found by an integration of the following equation from water temperature t_{w1} to t_{w2}.

$$\frac{KaV}{\dot{m}_w} = \int_{tw1}^{tw2} \frac{dt_w}{h' - h_a}$$

Numerical integration has to be adopted.

The heat balance of the air side and water side could be represented by the following equation: -

$$\dot{m}_a dh_a = \dot{m}_w c_{pw} dt$$

With ratio of mass flow rate given and C_{pw} = 4.18, you should get:

$$dh_a = 1.2 \text{ x } 4.18\, dt_w = 5.0\, dt_w$$

With the wet-bulb temperature of air at 29°C, the enthalpy of saturated air is 92.3 kJ/kg.

As the enthalpy of saturated film h' is 111.0 kJ/kg, hence the enthalpy difference (h'-ha) =(111.0 – 92.3) = 18.7 kJ/kg and 1/(h'-ha) = 0.05348 kg/kJ.

Question 1 Hints (cont'd)

For a temperature difference of 1 deg C and using the below equation

$$dh_a = 1.2 \text{ x } 4.18\, dt_w = 5.0\, x\, dt_w \qquad dh_a = 5.0 \text{ kJ/kg}$$

At a water temperature of 33 deg C, the enthalpy of air is 92.3 + 5.0 = 97.3 kJ/kg hence the enthalpy difference (h'-ha) = (116.8 – 97.3) = 19.5 kJ/kg and 1/(h'-ha) = 0.05128 kg/kJ.

The average value of 1/(h'-ha) at temperature of 32°C and 33°deg C is

$$\frac{1}{2}(0.05348 + 0.05128) = 0.05238$$

You could construct the following table (incomplete, just show two calculations)

Water temperature	Enthalpy of saturated firm	Enthalpy of air	h'-ha	1 / (h'-ha)	Mean of 1 / (h'-ha)
32	111.0	92.3	18.7	0.05348	
					0.05238
33	116.8	97.3	19.5	0.05128	

Summation of all the mean at last column got the tower coefficient

Question 1 Hints (cont'd)

Water temperature	Enthalpy of saturated firm	Enthalpy of air	h' -ha	1 / (h' -ha)	Mean of 1 / (h' -ha)
32	111	92.3	18.7	0.05348	
					0.05238
33	116.8	97.3	19.5	0.05128	
					0.0498
34	123	102.3	20.7	0.04831	
					0.04678
35	129.4	107.3	22.1	0.04525	
					0.04354
36	136.2	112.3	23.9	0.04184	
					0.04023
37	143.2	117.3	25.9	0.03861	
				Tower Coefficient	0.23272

$$0.23272 = 0.0167 + 0.033N(1.2)^{-0.6} \qquad N = 7.3 \text{ say } 8$$

(The above equation would be given.)

$$H = 0.6 \times 8 = 4.8 \text{ m}$$

Question 2 (Hints)

C max = 100 ppm

Since the car park is in continual use, c_o also equals 100 ppm.

$$Q' = V'\left(\frac{n}{3600}\right)$$

Since t is one hour

$$V' = 5400m^3$$

$$c_a = 0$$

$$V_c = 0.0024m^3s^{-1}$$

From the equation of Lecture 3,

$$100 = \left(\frac{10^6 x0.0024x3600}{5400n} + 0\right)(1 - e^{-n}) + 100e^{-n}$$

The number of air change n = 16.

Question 3

Hints

1) Supply conditions

SHR = Sensible heat/ Total heat gain

Psychrometric chart: Draw a line from point 25 deg C and 50%RH with SHR

The supply condition s lies on the space condition line and cooling coil cc lines on 95% RH. As supply fan & duct heat gain is 2 deg C, width of segment cc and s should be just 2 deg C.

The supply condition (temperature = 14 deg C, enthalpy = 35.2 kJ/kg and moisture content = 0.0082 kg/kg) could be read out from the psychrometric chart.

2) Supply air volume flow rate

Mass flow rate = Sensible heat/ (specific heat capacity x difference in room supply air temperature and room return air temperature) = 3.03 kg/s

Supply air flow rate= mass flow rate x moist volume of supply air = 3.03 x 0.825 = 2.5 m3/s)

Moist volume could be read out from psychrometric chart at supply point

Question 3

3) Enthalpy of the mixture of outdoor and return air

The outdoor air conditions (moist volume, enthalpy, humidity ratio) should be read from psychrometric chart at first

Mass flow rate of outdoor air = volume flow / moist volume = 0.44/0.89 = 0.49 kg/s

Enthalpy of the mixed air could be found by using

$$h_m = \frac{m_o h_o + (m_a - m_o) h_{ru}}{m_a} \quad = 58.65 \text{ kJ/kg}$$

h_m = enthalpy of mixed air

m_o = mass flow rate of outdoor air

h_o = enthalphy of outdoor air

m_a = mass flow rate of supply air

h_{ru} = enthalpy of return air before mixing (duct heat gain included)

Question 4 Hints

<u>Ratio of re-circulated to fresh air</u>

Plot the process on the psychrometric chart.

With the same max. sensible heat gain and turndown ratio, each zone will have the same maximum and minimum mass flow rates

The max. mass flow rate = 49 kW/ (1.02kW/KJ.K x 8K) = 6 kg/s

Minimum mass flow rate = 6 kW/ 2.8 = 2.14 kg/s

Since both zones have the same min. mass flow rate, the zone requiring the most fresh air will control the re-circulation ratio.

Re-circulation ratio = (2.14-0.8)/0.8= 1.675

<u>Temperature and percentage saturation in each zone during maximum simultaneous gains</u>

Supply air temperature = 22 – 8 = 14 °C

Zone 1 , Room Ratio Line = 14/(14+7) = 0. 5

Zone 2, Room Ratio Line = 49/(49+7) = 0.875

The minimum load to maintain zone A at 22 °C is 49/2.8 = 17.5 kW with a load of only 14kW the temperature must fall below this.

Zone 1: room temperature = 14 + 14/(2.14 x 1.02) = 20.4 °C

Zone 2: room temperature is at design temperature, i.e. 22 °C

Question 4

The two room ratio lines can be drawn from the supply state to establish the room states R1 and R2. From the chart, zone 1 is 67 % and zone 2 is at 50 % saturation.

The maximum cooling coil load

The mass flow rate thru the central plant is (6.13 + 2.45) = 8.58 kg/s.

Cooling Coil load = 8.58 (50.5 – 31.4) = 164 kW

The return air temperature (after mixing)

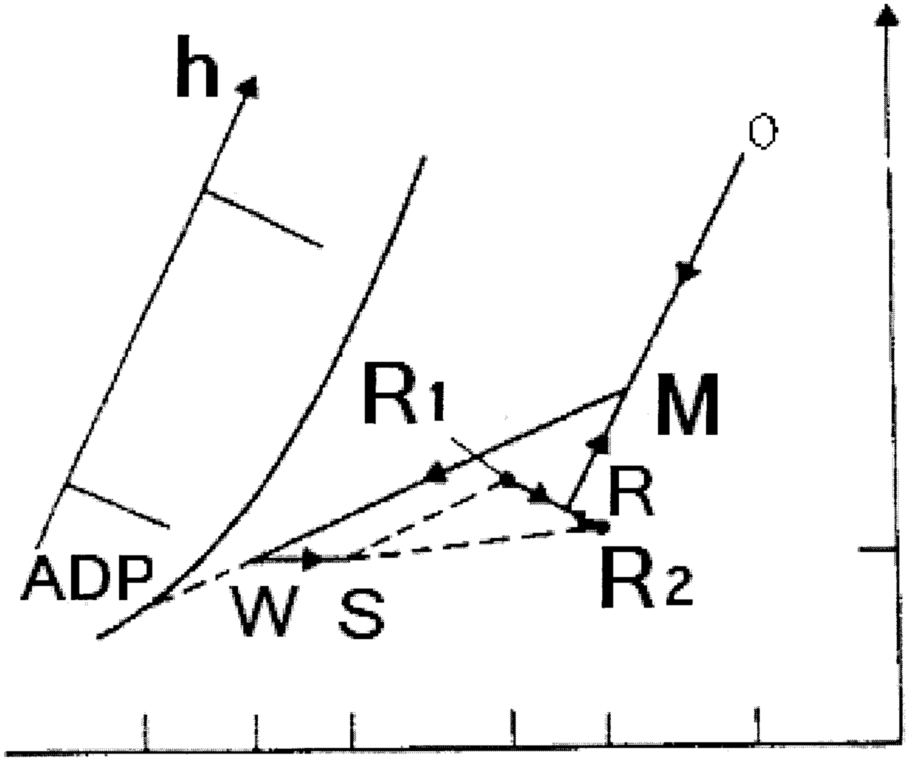
The return air state R will lie along a line joining R1 and R2. It can be fixed by proportioning the lines or calculating the mixed temperature, i.e.

The return air temperature = (2.14/(2.14 + 6) x 20.4) + (6./(2.14+6) x 22) = 21.57 °C

The psychrometric cycle can now be completed using the re-circulation ratio of 2.8:1 to establish point M:

Mixed temperature = (27 x 1/3.8) + (21.57 x 2.8/3.8) = 23.0 °C

Question 4 (Hints)



X-axis = Temperature

Y-axis = Specific Humidity of air

Question 5 Hints

Since the minimum allowable stack temperature is 145°C

The reduction in temperature possible for the flue gas is (185-145°C) = 40°C.

The amount of fuel used = 1501itre/hour = 135 kg/h (assuming the density to be 900 kg/m^3).

The air to-fuel ratio is 1:14.

Therefore, the amount of combustion air is approximately 14 times the weight of the fuel used. i.e. 14 X 135 = 1890 kg/h.

Total mass flow rate of flue gas = 1890 kg/h + 135 kg/h = 1995 kg/h = 0.55 kg/s

As the specific heat capacity of flue gas is 1.1 kJ/kg.K, the amount of heat recovered can be estimated as follows:

Heat recovered = mass flow rate x specific heat capacity x temperature drop for flue gas

=0.55 x 1.1 x 40 = 24.2 kW (kJ/s) = 0.0242 X 3,600 = 87.1 MJ/hr.

Question 5

Loading A	Operation hours per day B	Fan motor power (with damper) (kW) C	Fan motor power with VSD drive (kW)* D	Power Saving (kW) E	Power Saving (kWh) F
100%	3	15	15	0	0
80%	4	14	7.68	6.32	25.28
60%	10	12	3.24	8.76	87.6
40%	7	11	0.96	10.04	70.28
					183.16

*D = A^3 x 15

Assume \$1.3/kWH and no interest rate, the money saving per year is \$183.16 * 365 = \$66,853.

With the interest rate at 3%, the money saving for the first year would be

$$\frac{66{,}853}{(1+0.03)^1} = \$64{,}906$$

With the interest rate at 3%, the money saving for the second year would be $\frac{66{,}853}{(1+0.03)^2} = \$63{,}015.5$

Question 6 Hints

Refrigeration effect = difference in enthalpies before and after evaporator = 111.9 kJ/kg

Work input = the differences in enthalpies before and after the compressor = 20.5 kJ/kg

Coefficient of performance = refrigeration effect/ work input = 5.46

Mass flow rate = Cooling capacity/ refrigeration effect = 1500/111.9 = 13.4 kg/s

Power input per kW refrigeration produced = 1kW/ Coefficient of Performance = 0.183kW/kW

With the sub-cooling, new enthalpy to be found for refrigerant leaving condenser

New refrigeration effect (= 114 kJ/kg), new coefficient of performance (=5.56) and new power input per kW refrigeration (= 0.1799kW/kW) produced should be got.

Electric energy save = 1.7% (By using 0.183kW/kW & 0.1799kW/kW)

Question 7 (Hints)

1) The Coefficient of Performance

$$\text{The actual enthalphy of refrigerant leaving the compressor} = 960 + \frac{977 - 960}{0.8} = 981.3\ \text{kJ/kg}$$

$$\text{The Coefficient of Performance} = \frac{960 - 790}{981.3 - 960} = 8.0$$

2) The mass flow rate of the refrigerant at compressor inlet:

$$\text{Mass flow rate} = \frac{380\ \text{kW}}{(960 - 790)} = 2.24\ \text{kg/s}$$

3) The diameter of the impeller of the compressor

$$u_{b2} = \sqrt{\left(\frac{h_2 - h_1}{\sigma}\right)} = \sqrt{(981.3 - 960)}\sqrt{10^3} = 146\ \text{m/s}$$

$$D_0 = \frac{u_{b2}}{\pi N} = \frac{146}{\pi x 60} = 0.78\ \text{m}$$

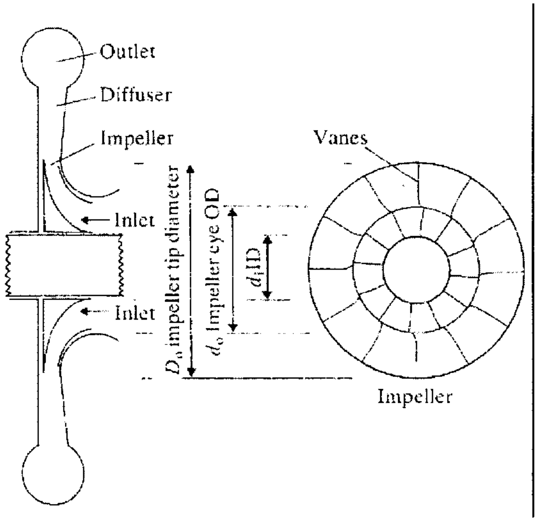
Question 7

4) The outside diameter of the eye of the impeller inlet

The mass flow rate at inlet = 2.24 x 0.4 = 0.896 m3/s

Mass flow rate = $\frac{\pi}{4}(d_o^2 - d_i^2)u_1 = \frac{\pi}{4}(d_o^2 - 0.065^2)85 = 0.896$

$d_o = 0.132$m



Question 8 Hints

Calculation of fin surface efficiency:

R-value of fin (Unit = m^2K/W)

Fin efficiency $\eta_f = \frac{1}{1 + h_o R_f}$ = **0.79**

Surface fin efficiency $\eta_s = 1 - \left(\frac{A_f}{A_o}\right)(1 - \eta_f) = 0.81$

Determination of Heat capacities of air and water

Air

Heat Capacity of air (unit= kW/K) =

Face area of coil x velocity of air x density of air x specific heat of air = 2.93kW/K

(C_{pa}= 1.02kJ/kg.K)

Water

Heat Capacity of water (unit= kW/K)=

Flow rate of water x density of water x specific heat of water = 8.4 kW/K

(specific heat of water C_{pw} = 4.2kJ/kg.K)

Question 8

Determination on heat capacity value

$$C = \frac{C_a}{C_w} = 0.35$$

Determine the overall heat transfer value: -

$$\frac{1}{U_o A_o} = \frac{1}{\eta_s h_o A_o} + \frac{1}{h_i A_i}$$

Hence, $U_o A_o = 3758$

Determine Number of Transfer Unit

Number of transfer unit, NTU= $\frac{U_o A_o}{C_a} = 1.28$

Determine Dry coil effectiveness

Dry coil effectiveness $\varepsilon = \frac{1 - e^{[-NTU(1-c)]}}{1 - ce^{[-NTU(1-c)]}} = 0.6663$

Calculate Sensible Cooling Coil's load

$$q_{cs} = \dot{m}_a C_{pa} (t_{ae} - t_{we}) \varepsilon = 35.1 \text{ kW}$$

(Equations in this question would be given)