

Department of Mechanical Engineering

Regulation: 2017

Year and Semester: III and V

SUBJECT CODE: ME 8693

SUBJECT NAME: HEAT AND MASS TRANSFER

STUCOR APP

UNIT: I - CONDUCTION

PART A - 2 Marks (Questions and Answers)

1. State Fourier's Law of conduction. (April/May 2011, Nov/Dec 14, Nov/Dec 16)

The rate of heat conduction is proportional to the area measured - normal to the direction of heat flow and to the temperature gradient in that direction.

$$Q \propto -A \frac{dt}{dx}$$

$$Q = -KA \frac{dt}{dx}$$

Where, A are in m²

$\frac{dt}{dx}$ is temperature gradient in K/m

K is Thermal Conductivity W/mk

2. State Newton's law of cooling or convection law. (May/June 2009)

Heat transfer by convection is given by Newton's law of cooling

$$Q = hA (T_s - T_\infty)$$

Where

A - Area exposed to heat transfer in m², h - heat transfer coefficient in W/m²K

T_s - Temperature of the surface in K, T_∞ - Temperature of the fluid in K.

3. Define overall heat transfer co-efficient. (May/June 2007)

The overall heat transfer by combined modes is usually expressed in terms of an overall conductance or overall heat transfer co-efficient 'U'.

$$\text{Heat transfer } Q = UA \Delta T.$$

4. Write down the equation for heat transfer through composite pipes or cylinder. (April/May 2008)

$$\text{Heat transfer } Q = \frac{\Delta T_{\text{overall}}}{R} \text{ where } \Delta T = T_a - T_b$$

$$R = \frac{1}{2\pi L} \left[\frac{1}{h_a r_1} + \frac{\ln\left[\frac{r_2}{r_1}\right]}{K_1} + \frac{\ln\left[\frac{r_1}{r_2}\right]}{K_1} L_2 + \frac{1}{h_a r_{12}} \right]$$

5. What is critical radius of insulation (or) critical thickness? (May/June 2014) (Nov/Dec 2008)

Critical radius = r_c Critical thickness = $r_c - r_1$

Addition of insulating material on a surface does not reduce the amount of heat transfer rate always. In fact under certain circumstances it actually increases the heat loss up to certain thickness of insulation. The radius of insulation for which the heat transfer is maximum is called critical radius of insulation, and the corresponding thickness is called critical thickness.

6. Define Fin efficiency and Fin effectiveness. (Nov/Dec 2015& Nov/Dec 2010)

The efficiency of a fin is defined as the ratio of actual heat transfer by the fin to the maximum possible heat transferred by the fin.

$$\eta = \frac{Q_{fin}}{Q_{max}}$$

Fin effectiveness is the ratio of heat transfer with fin to that without fin

$$fin\ effectiveness = \frac{Q_{withfin}}{Q_{withoutfin}}$$

7. Define critical thickness of insulation with its significance. [MAY-JUN 14]

Addition of insulating material on a surface does not reduce the amount of heat transfer rate always. In fact under certain circumstances it actually increases the heat loss up to certain thickness of insulation. The radius of insulation for which the heat transfer is maximum is called critical radius of insulation, and the corresponding thickness is called critical thickness. For cylinder, Critical radius = $r_c = k/h$, Where k- Thermal conductivity of insulating material, h- heat transfer coefficient of surrounding fluid. Significance: electric wire insulation may be smaller than critical radius. Therefore the plastic insulation may actually enhance the heat transfer from wires and thus keep their steady operating temperature at safer levels.

8. What is lumped system analysis? When is it applicable? [Nov/Dec 14 & April/May 2010]

In heat transfer analysis, some bodies are observed to behave like a "lump" whose entire body temperature remains essentially uniform at all times during a heat transfer process. The temperature of such bodies can be taken to be a function of time only. Heat transfer analysis which utilizes this idealization is known as the lumped system analysis. It is applicable when the Biot number (the ratio of conduction resistance within the body to convection resistance at the surface of the body) is less than or equal to 0.1.

9. Write the three dimensional heat transfer poisson and laplace equation in Cartesian co-ordinates(May/June 2012)(April/May 2010)

Poisson equation:

$$\frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2} + \frac{\partial^2 T}{\partial z^2} + \frac{g}{k} = 0$$

Laplace equation:

$$\frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2} + \frac{\partial^2 T}{\partial z^2} = 0$$

10. A 3 mm wire of thermal conductivity 19 W/mK at a steady heat generation of 500 MW/m³. Determine the center temperature if the outside temperature is maintained at 25°C (May 2012)

$$\begin{aligned} \text{Critical temperature} \quad T_c &= T_\infty + \frac{qr^2}{4K} \\ &= 298 + \left[\frac{500 \times 10^6 \times 0.0015^2}{4 \times 19} \right] \\ T_c &= 312.8K \end{aligned}$$

11. List down the three types of boundary conditions. (Nov/Dec 2005)

1. Prescribed temperature
2. Prescribed heat flux
3. Convection Boundary Conditions.

12. Define fins (or) extended surfaces.

It is possible to increase the heat transfer rate by increasing the surface of heat transfer. The surfaces used for increasing heat transfer are called extended surfaces or sometimes known as fins.

13. How thermodynamics differ from heat transfer?

- Thermodynamics doesn't deals with rate of heat transfer
- Thermodynamics doesn't tell how long it will occur
- Thermodynamics doesn't tell about the method of heat transfer

PART B - 13 Marks (Questions and Answers)

1. Derive the General Differential Equation of Heat Conduction in Cartesian coordianates.(NOV/DEC 2014)

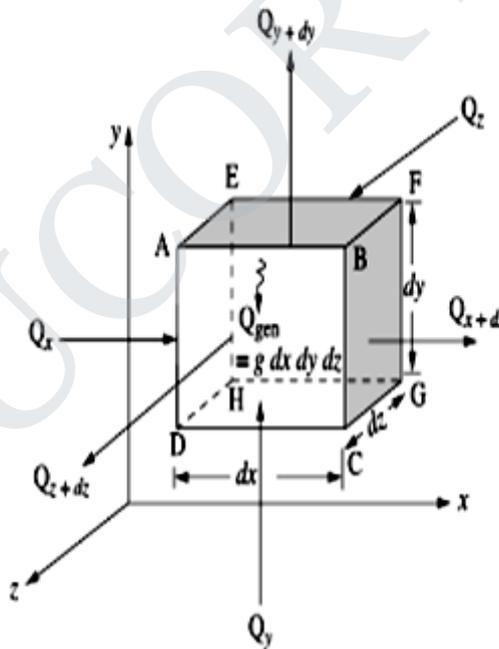


Fig 2.1

Consider a small volume element in Cartesian coordinates having sides dx, dy and dz as shown in Fig. 2.1 the energy balance for this little element is obtained from the first law of thermodynamics as:

$$\left\{ \begin{array}{l} \text{Net heat conducted into element} \\ dx dy dz \text{ per unit time} \\ \text{(I)} \end{array} \right\} + \left\{ \begin{array}{l} \text{Internal heat generated} \\ \text{per unit time} \\ \text{(II)} \end{array} \right\}$$

$$= \left\{ \begin{array}{l} \text{Increase in internal} \\ \text{per unit time} \\ \text{(III)} \end{array} \right\} + \left\{ \begin{array}{l} \text{Work done by element} \\ \text{per unit time} \\ \text{(IV)} \end{array} \right\} \quad (2.2)$$

The last term of Eqn. (2.2) is very small because the flow work done by solids due to temperature changes is negligible.

The three terms, I, II and III of this equation are evaluated as follows:

Let q_x be the heat flux in x-direction at x, face ABCD and q_{x+dx} the heat flux at $x + dx$, face A'B'C'D'. Then rate of heat flow into the element in x-direction through face ABCD is:

$$Q_x = q_x dydz = -k_x \frac{\partial T}{\partial x} dydz \quad (2.3)$$

Where k_x is the thermal conductivity of material in x-direction and $\frac{\partial T}{\partial x}$ is the temperature gradient in

x-direction. The rate of heat flow out of the element in x-direction through the face at $x+dx$. A'B'C'D' is:

$$Q_x = -k_x \frac{\partial T}{\partial x} dydz - \frac{\partial}{\partial x} \left(k_x \frac{\partial T}{\partial x} \right) dx dydz \quad (2.4)$$

Then, the net rate of heat entering the element in x-direction is the difference between the entering and leaving heat flow rates, and is given by:

$$Q_x - Q_{x+dx} = \frac{\partial}{\partial x} \left(k_x \frac{\partial T}{\partial x} \right) dx dydz \quad (2.5)$$

$$Q_y - Q_{y+dy} = \frac{\partial}{\partial y} \left(k_y \frac{\partial T}{\partial y} \right) dx dydz$$

$$Q_z - Q_{z+dz} = \frac{\partial}{\partial z} \left(k_z \frac{\partial T}{\partial z} \right) dx dydz$$

The net heat conducted into the element $dx dydz$ per unit time, term I in Eqn. (2.2) is:

$$I = \left[\frac{\partial}{\partial x} \left(k_x \frac{\partial T}{\partial x} \right) + \frac{\partial}{\partial y} \left(k_y \frac{\partial T}{\partial y} \right) + \frac{\partial}{\partial z} \left(k_z \frac{\partial T}{\partial z} \right) \right] dx dydz \quad (2.6)$$

Let q be the internal heat generation per unit time and per unit volume (W/m^3), the rate of energy generation in the element, term II in Eqn. (2.2), is

$$II = q dx dydz \quad (2.7)$$

The change in internal energy for the element over a period of time dt is: (mass of element) (specific heat) (change in temperature of the element in time dt)

$$(\rho dx dydz) (c_p) dT = (\rho c_p dT) dx dydz \quad (2.8)$$

Where ρ and c_p are the density and specific heat of the material of the element.

Then, the change in internal energy per unit time, term III of Eqn. (2.2) is:

$$\text{III} = \rho c_p \frac{\partial T}{\partial t} dx dy dz \quad (2.9)$$

Substitution of Eqns. (2.6),(2.7) and (2.9) into Eqn. (2.2) leads to the general three-dimensional equation for heat conduction:

$$\frac{\partial}{\partial x} \left(k_x \frac{\partial T}{\partial x} \right) + \frac{\partial}{\partial y} \left(k_y \frac{\partial T}{\partial y} \right) + \frac{\partial}{\partial z} \left(k_z \frac{\partial T}{\partial z} \right) + q = \rho c_p \frac{\partial T}{\partial t} \quad (2.10)$$

Since for most engineering problems the materials can be considered isotropic for which $K_x = K_y = K_z = k = \text{Constant}$, the general three-dimensional heat conduction equation becomes:

$$\frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2} + \frac{\partial^2 T}{\partial z^2} + \frac{q}{k} = \frac{\rho c_p}{k} \frac{\partial T}{\partial t} = \frac{1}{\alpha} \frac{\partial T}{\partial t}$$

The quantity $\frac{k}{\rho c_p}$ is known as the thermal diffusivity, α of the material. It has got the units m^2/s .

2. Derive the Heat conduction equation in cylindrical coordinates.

The heat conduction equation derived in the previous section can be used for solids with rectangular boundaries like slabs, cubes, etc. but then there are bodies like cylinders, tubes, cones, spheres to which Cartesian coordinates system is not applicable.

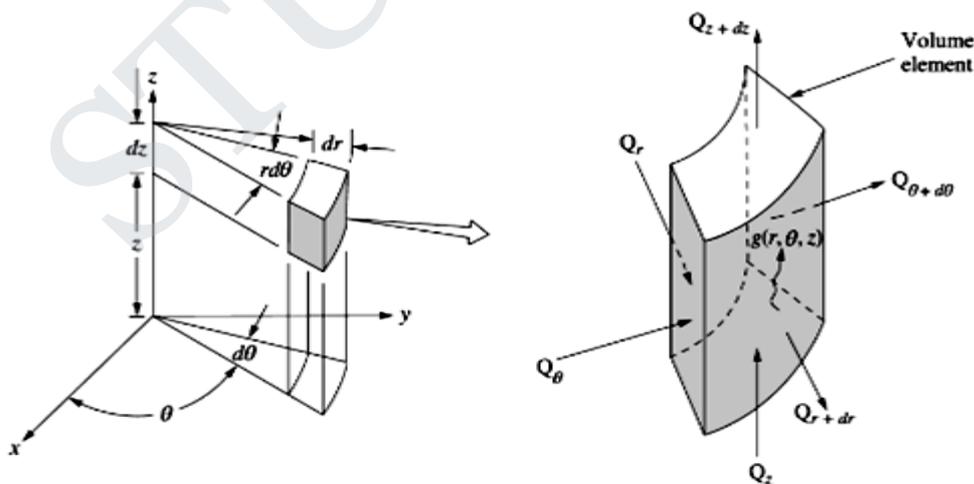


Fig 2.2

A more suitable system will be one in which the coordinate surfaces coincide with the boundary surfaces of the region. For cylindrical bodies, a cylindrical

coordinate system should be used. The heat conduction equation in cylindrical coordinates can be obtained by an energy balance over a differential element, a procedure similar to that described previously. The equation could also be obtained by doing a coordinate transformation from Fig. 2.2.

Consider a small volume element having sides dr , dz and $r d\theta$ as shown in Fig. 2.2. Assuming the material to be isotropic, the rate of heat flow into the element in r -direction is:

$$Q_r = -k \frac{\partial T}{\partial r} r d\theta dz$$

The rate of heat flow out of the element in r -direction at $r+dr$ is:

$$Q_{r+dr} = Q_r + \frac{\partial Q_r}{\partial r} dr$$

Then, the net rate of heat entering the element in r -direction is given by

$$\begin{aligned} Q_r - Q_{r+dr} &= k \frac{\partial}{\partial r} \left(r \frac{\partial T}{\partial r} \right) dr d\theta dz \\ &= k \left(\frac{\partial^2 T}{\partial r^2} + \frac{1}{r} \frac{\partial T}{\partial r} \right) dr d\theta dz \end{aligned}$$

Similarly,

$$\begin{aligned} Q_\theta - Q_{\theta+d\theta} &= -k \frac{\partial T}{r d\theta} dr dz - \left[-k \frac{\partial T}{r d\theta} dr dz - \frac{k \partial}{r d\theta} \left(\frac{\partial T}{r \partial \theta} \right) \cdot r d\theta dr dz \right] \\ &= k \left(\frac{1}{r^2} \frac{\partial^2 T}{\partial \theta^2} \right) r dr d\theta dz \\ Q_z - Q_{z+dz} &= -k \frac{\partial T}{\partial z} \cdot r d\theta dz - \left[-k \frac{\partial T}{\partial z} r d\theta dr - k \frac{\partial}{\partial z} \left(\frac{\partial T}{\partial z} \right) \cdot r d\theta dr dz \right] \\ &= -k \left(\frac{\partial^2 T}{\partial z^2} \right) r dr d\theta dz \end{aligned}$$

The net heat conducted into the element $dr \cdot r d\theta \cdot dz$ per unit time, term I of Eqn. (2.2)

$$I = k \left(\frac{\partial^2}{\partial r^2} + \frac{1}{r} \frac{\partial}{\partial r} \right) r dr d\theta dz$$

Taking q as the internal heat generation per unit time and per unit volume, term II of Eqn (2.2) is

$$II = q r dr d\theta dz$$

The change in internal energy per unit time, term III of Eqn. (2.2) is:

$$III = \rho c_p \frac{\partial T}{\partial t} r dr d\theta dz$$

Substitution of terms I, II and III into the energy balance Eqn. (2.2) leads to three-dimensional equation for an isentropic material in cylindrical coordinate system as

$$\frac{\partial^2 T}{\partial r^2} + \frac{1}{r} \left(\frac{\partial T}{\partial r} \right) + \frac{1}{r^2} \frac{\partial^2}{\partial \theta^2} + \frac{\partial^2}{\partial z^2} + \frac{q}{k} = \frac{1}{\alpha} \frac{\partial T}{\partial t}$$

3. A furnace wall is made up of three layer of thickness 25 cm, 10 cm and 15 cm with thermal conductivities of 1.65 W/mK and 9.2 W/mK respectively. The inside is exposed to gases at 1250⁰c with a convection coefficient of 25 W/m² K and the inside surface is at 1100⁰c , the outside surface is exposed to air at 25⁰C with convection coefficient of 12 W/m²K .Determine (i) the unknown thermal conductivity (ii)the overall heat transfer coefficient (iii) All the surface temperature.(May/June 2012)

Given:

Thickness $L_1 = 25 \text{ cm} = 0.25 \text{ m}$

$L_2 = 10 \text{ cm} = 0.1 \text{ m}$

$L_3 = 15 \text{ cm} = 0.15 \text{ m}$

Thermal conductivity , $k_1 = 1.65 \text{ W/mK}$,

$k_2 = 9.2 \text{ W/mK}$

Inside Gas Temperature , $T_a = 1250^\circ \text{C} = 1523 \text{ K}$

$T_b = 25^\circ \text{C} = 298 \text{ K}$

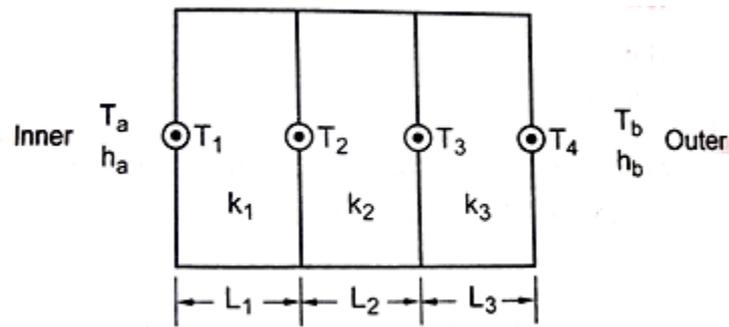
Inner surface temperature , $T_1 = 1100^\circ \text{C} = 1373 \text{ K}$

Inside heat transfer coefficient , $h_a = 25 \text{ W/m}^2 \text{ K}$

Outside Heat Transfer Coefficient , $h_b = 12 \text{ W/m}^2 \text{ K}$

To find:

- i) The Unknown Thermal Conductivity ,
- ii) The Overall Heat Transfer Coefficient
- iii) All The Surface Temperature



Solution:

STEP-1

$$Q = h_a A (T_a - T_1)$$

$$= 25(1523 - 1373) = 3750 \text{ W/m}^2$$

From HMT data book P.No 45

Heat Flow, $Q = \Delta T_{\text{overall}} / R$

$$R = \frac{1}{H_a A} + \frac{L_1}{k_1 A} + \frac{L_2}{k_2 A} + \frac{L_3}{k_3 A} + \frac{1}{H_b A}$$

$$Q = \frac{T_a - T_b}{\frac{1}{H_a A} + \frac{L_1}{k_1 A} + \frac{L_2}{k_2 A} + \frac{L_3}{k_3 A} + \frac{1}{H_b A}}$$

$$\frac{Q}{A} = \frac{1523 - 298}{\frac{1}{25} + \frac{0.25}{1.65} + \frac{0.10}{k_2} + \frac{0.15}{9.2} + \frac{1}{12}}$$

$k_2 = 2.816 \text{ W/mk}$

STEP-2

From HMT data book P.No 45

Overall Thermal resistance (R)

$$R = \frac{1}{H_a A} + \frac{L_1}{k_1 A} + \frac{L_2}{k_2 A} + \frac{L_3}{k_3 A} + \frac{1}{H_b A}$$

[Take $A = 1 \text{ m}^2$]

$R_{\text{total}} = 0.3267 \text{ W/m}^2$

$$U = 1/R_{\text{total}} = 1/0.3267 = 3.06 \text{ W/m}^2\text{K}$$

STEP-3

$$Q = \frac{T_a - T_1}{R_a} = \frac{T_1 - T_2}{R_1} = \frac{T_2 - T_3}{R_2} = \frac{T_3 - T_4}{R_3} = \frac{T_4 - T_b}{R_b}$$

$$Q = \frac{T_a - T_1}{R_a},$$

$$Q = \frac{T_1 - T_2}{R_1},$$

$$R_1 = \frac{L_1}{K_1} = 0.1515$$

$$3750 = \frac{1373 - T_2}{0.1515}$$

$T_2 = 804.8\text{K}$

$$Q = \frac{T_2 - T_3}{R_2} \left[\because R_2 = \frac{L_2}{K_2} \right]$$

$$3750 = \frac{804.8 - T_3}{\frac{0.10}{2.816}}$$

$T_3 = 671.45\text{K}$

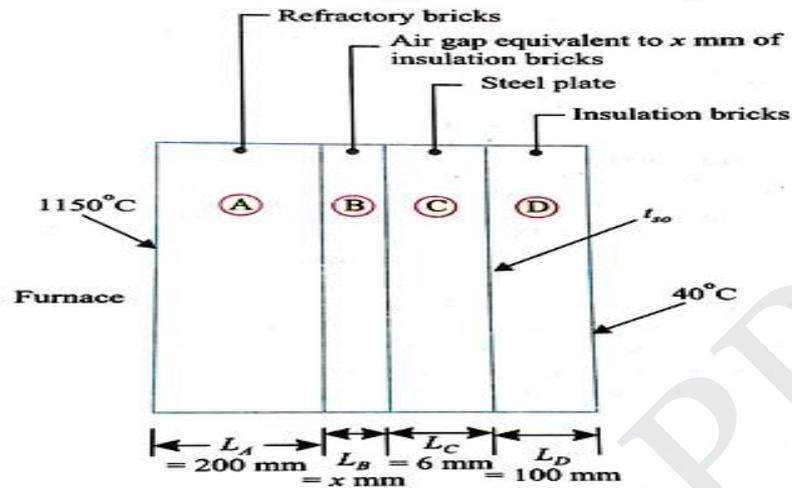
$$Q = \frac{T_3 - T_4}{R_3} \left[\because R_3 = \frac{L_3}{K_3} \right]$$

$$3750 = \frac{671.45 - T_4}{\frac{0.15}{9.2}}$$

$T_4 = 610.30\text{K}$

4. A furnace wall consists of 200mm layer of refractory bricks, 6 mm layer of steel plate and a 100mm layer of insulation bricks. The maximum temperature of the wall is 1150°C on the furnace side and the minimum temperature is 40°C on the outermost side of the wall. An accurate energy balance over the furnace shows that the heat loss from the wall is 400W/m². It is known that there is a thin layer of air between the layers of refractory bricks and steel plate. Thermal conductivities for the three layers are 1.52, 45 and 0.138 W/m°C respectively. Find

- i) To how many millimeters of insulation bricks is the air layer equivalent?
ii) What is the temperature of the outer surface of the steel plate? (Nov/Dec 2014)



Given

Thickness of refractory bricks, $L_A = 200 \text{ mm} = 0.2 \text{ m}$

Thickness of steel plate, $L_C = 6 \text{ mm} = 0.006 \text{ m}$

Thickness of insulation bricks, $L_D = 100 \text{ mm} = 0.1 \text{ m}$

Difference of temperature between the innermost and outermost sides of the wall,

$$\Delta t = 1150 - 40 = 1110^\circ\text{C}$$

$$K_A = 1.52 \text{ W/m}^\circ\text{C}$$

$$K_B = K_D = 0.138 \text{ W/m}^\circ\text{C}$$

$$K_C = 45 \text{ W/m}^\circ\text{C}$$

Heat loss from the wall, $q = 400 \text{ W/m}^2$

- i) The value of $x (=L_B)$

From HMT data book P.No 45

Heat Flow, $Q = \Delta T_{\text{overall}}/R$

$$R = \frac{1}{H_a A} + \frac{L_1}{k_1 A} + \frac{L_2}{k_2 A} + \frac{L_3}{k_3 A} + \frac{1}{H_b A}$$

$$400 = \frac{1110}{\frac{L_A}{K_A} + \frac{L_B}{K_B} + \frac{L_C}{K_C} + \frac{L_D}{K_D}}$$

$$400 = \frac{1110}{\frac{0.2}{1.52} + \frac{(x/1000)}{0.138} + \frac{0.006}{45} + \frac{0.1}{0.138}}$$

$$= \frac{1110}{0.1316+0.0072x+0.00013+0.7246}$$

$$= \frac{1110}{0.8563+0.0072x}$$

$$0.8563 + 0.0072x = \frac{1110}{400} = 2.775$$

$$x = \frac{2.775-0.8563}{0.0072} = 266.5 \text{ mm}$$

$x = 266.5 \text{ mm}$

ii) Temperature of the outer surface of the steel plate t_{so} :

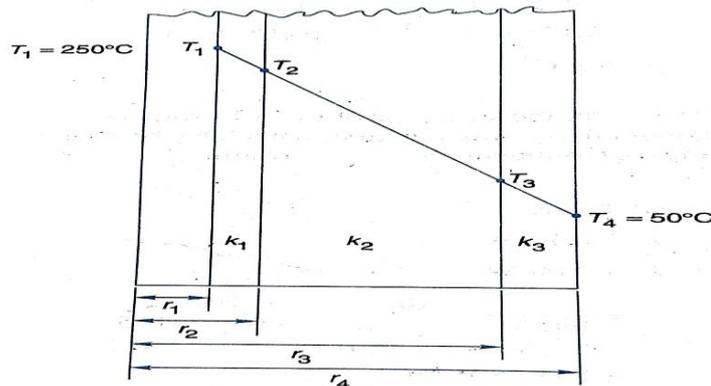
$$q = 400 = \frac{(t_{so} - 40)}{L_D/K_D}$$

$$400 = \frac{(t_{so} - 40)}{0.1/0.138}$$

$$t_{so} = \frac{400}{1.38} + 40 = 329.8^\circ\text{C}$$

$t_{so} = 329.8^\circ\text{C}$

5. A steel pipe line ($K=50\text{W/mk}$) of I.D 110mm is to be covered with two layers of insulation each having a thickness of 50mm. The thermal conductivity of the first insulation material is 0.06W/mk and that of the second is 0.12W/mk . Calculate the loss of heat per metre length of pipe and the interface temperature between the two layers of insulation when the temperature of the inside tube surface is 250°C and that of the outside surface of the insulation is 50°C . (April/ may 2015)



Given :

$$r_1 = 50mm$$

$$r_2 = 55mm$$

$$r_3 = 105mm$$

$$r_4 = 155mm$$

$$K_1 = 50 \frac{W}{mk}$$

$$K_2 = 0.06 \frac{W}{mk}$$

$$K_3 = 0.12 \frac{W}{mk}$$

$$T_1 = 250^{\circ}C$$

$$T_4 = 50^{\circ}C$$

To find

$$T_3 = ?$$

Solution:

step-1

From HMT data book P.No 46

Heat Flow, $Q = \Delta T_{\text{overall}}/R$

$$R = \frac{1}{2\pi L} \left[\frac{1}{h_a r_1} + \frac{1}{k_1} \ln \left(\frac{r_2}{r_1} \right) + \frac{1}{k_2} \ln \left(\frac{r_3}{r_2} \right) + \frac{1}{k_3} \ln \left(\frac{r_4}{r_3} \right) + \frac{1}{h_b r_4} \right]$$

$$\frac{Q}{L} = \frac{2\pi (T_1 - T_4)}{\frac{\ln \left(\frac{r_2}{r_1} \right)}{K_1} + \frac{\ln \left(\frac{r_3}{r_2} \right)}{K_2} + \frac{\ln \left(\frac{r_4}{r_3} \right)}{K_3}}$$

$$\frac{Q}{L} = \frac{2 \times 3.14 (250 - 50)}{\frac{\ln \left(\frac{55}{50} \right)}{50} + \frac{\ln \left(\frac{105}{55} \right)}{0.06} + \frac{\ln \left(\frac{155}{105} \right)}{0.12}}$$

$\frac{Q}{L} = 89.6 W/m$

step-2

The interface temperature, T_3 is obtained from the equation

$$\frac{Q}{L} = \frac{2\pi(T_3 - T_4)}{\frac{\ln\left(\frac{r_4}{r_3}\right)}{K_3}}$$

$$T_3 = \frac{\frac{Q}{L} \times \ln\left(\frac{r_4}{r_3}\right)}{2\pi K_3} + T_4$$

$$= \frac{89.6 \times \ln\left(\frac{155}{105}\right)}{0.12 \times 6.28} + 50$$

$$T_3 = 96.3^\circ\text{C}$$

6. A plane wall 10cm thick generates heat at a rate of $4 \times 10^4 \text{ W/m}^3$ when an electric current is passed through it. The convective heat transfer coefficient between each face of the wall and the ambient air is $50 \text{ W/m}^2\text{K}$. Determine a) the surface temperature b) the maximum air temperature on the wall, Assume the ambient air temperature to be 20°C and the thermal conductivity of the wall material to be 15 W/mK . (May/June 2016)

Given:

Thickness $L = 10\text{cm} = 0.10\text{m}$

Heat generation $\dot{q} = 4 \times 10^4 \text{ W/m}^3$

Convective heat transfer co-efficient $= 50 \text{ W/m}^2\text{K}$.

Ambient air temperature $T_\infty = 20^\circ\text{C} + 273 = 293\text{K}$

Thermal conductivity $k = 15 \text{ W/mK}$.

Solution:

Step 1

From HMT data book P.No 48

Surface temperature

$$T_w = T_\infty + \frac{\dot{q}L}{2h}$$

$$= 293 + \frac{4 \times 10^4 \times 0.10}{2 \times 50}$$

$$T_w = 333\text{K}$$

Step 2

Maximum temperature

$$T_{max} = T_w + \frac{\dot{q}L^2}{8k}$$

$$= 333 + \frac{4 \times 10^4 \times (0.10)^2}{8 \times 15}$$

$$T_{\max} = 336.3 \text{ K}$$

7. A cylinder 1m long and 5 cm in diameter is placed in an atmosphere at 45°C. It is provided with 10 longitudinal straight fins of material having $k=120 \text{ W/mK}$. The height of 0.76mm thick fins is 1.27cm from the cylinder surface. The heat transfer coefficient between cylinder and the atmospheric air is $17 \text{ W/m}^2\text{K}$. Calculate the rate of heat transfer and the temperature at the end of fins if the surface temperature of cylinder is 150°C. (Nov/Dec 2015)

Given:

Length of cylinder $W = 1 \text{ m}$

Length of the fin $L = 1.27 \text{ cm} = 1.27 \times 10^{-2} \text{ m}$.

Thickness of the fin $t = 0.76 \text{ mm} = 0.76 \times 10^{-3} \text{ m}$.

Thermal conductivity $k = 120 \text{ W/mK}$

heat transfer coefficient $h = 17 \text{ W/m}^2\text{K}$

Base temperature of the cylinder $T_b = 150^\circ\text{C} + 273 = 423 \text{ K}$

Ambient temperature $T_\infty = 45^\circ\text{C} + 273 = 318 \text{ K}$

Diameter of the cylinder $d = 5 \text{ cm} = 5 \times 10^{-2} \text{ m}$.

To find

- i) Heat transfer rate, Q_{total}
- ii) Temperature at the end of the fin, T

Solution:

Step-1

Perimeter = $2W = 2 \times 1 = 2 \text{ m}$

Area = $Wt = 1 \times 0.76 \times 10^{-3} = 0.76 \times 10^{-3} \text{ m}^2$

From HMT data book P.No 50

$$m = \sqrt{\frac{hp}{kA}}$$

$$= \sqrt{\frac{17 \times 2}{120 \times 0.76 \times 10^{-3}}}$$

$$m = 19.31$$

Step-2

$$\tan h (mL) = \tanh(19.81 \times 1.27 \times 10^{-2}) = 0.241$$

$$\frac{h}{mk} = \frac{17}{19.31 \times 120} = 0.00734$$

From HMT data book P.No 50

$$Q_{fin} = \sqrt{hpkA} (T_b - T_{\infty}) \left[\frac{\tanh(ml) + \left(\frac{h}{mk}\right)}{1 + \left(\frac{h}{mk}\right) \tanh(ml)} \right]$$

$$= \sqrt{17 \times 2 \times 120 \times 0.76 \times 10^{-3}} (423 - 318) \left[\frac{0.241 + (0.00734)}{1 + (0.00734)0.241} \right]$$

$$Q_{fin} = 45.65 \text{ KWperfin}$$

From HMT data book P.No 44

$$Q_b = h[\pi D - [10 \times 0.76 \times 10^{-3}]L(T_b - T_{\infty})]$$

$$= 17[\pi \times 0.05 - [10 \times 0.76 \times 10^{-3}]1(423 - 318)]$$

$$Q_b = 266.82W$$

Step-3

$$Q_{total} = 10Q_{fin} + Q_b$$

$$= (10 \times 45.7) + 266.82$$

$$Q_{total} = 723.82W$$

Step-4

From HMT data book P.No 50

The temperature at the end of the fin

$$T - T_{\infty} = \frac{T_b - T_{\infty}}{\text{Cosh}(ml) + \left(\frac{h}{mk}\right) \sinh(ml)}$$

$$T - 318 = \frac{423 - 318}{\text{Cosh}(19.81 \times 1.27 \times 10^{-2}) + (0.00734) \sinh(19.81 \times 1.27 \times 10^{-2})}$$

$$T = 419.74K$$

8. A circumferential rectangular fins of 140mm wide and 5mm thick are fitted on a 200mm diameter tube. The fin base temperature is 170°C and the ambient temperature is 25°C. Estimate fin Efficiency and heat loss per fin. Take Thermal conductivity $K = 220\text{W/mk}$ Heat transfer co-efficient $h = 140\text{W/m}^2\text{k}$.

Given:

Wide $L = 140\text{mm} = 0.140\text{m}$

Thickness $t = 5\text{mm} = 0.005\text{m}$

Diameter $d = 200\text{mm} \Rightarrow r = 100\text{mm} = 0.100\text{m}$

Fin base temperature $T_b = 170^\circ\text{C} + 273 = 443\text{K}$

Ambient temperature $T_\infty = 25^\circ\text{C} + 273 = 298\text{K}$

Thermal conductivity $k = 220\text{W/mk}$

Heat transfer co-efficient $h = 140\text{W/m}^2\text{k}$

To find:

Fin Efficiency, η

Heat loss Q

Solution:

A rectangular fin is long and wide. So heat loss is calculated by fin efficiency curves

From HMT data book P.No 52

Step1

$$\begin{aligned} \text{Corrected length } L_c &= L + \frac{t}{2} \\ &= 0.140 + \frac{0.005}{2} \end{aligned}$$

$$L_c = 0.1425 \text{ m}$$

Step2

$$\begin{aligned} r_{2c} &= r_1 + L_c \\ &= 0.100 + 0.1425 \end{aligned}$$

$$r_{2c} = 0.2425\text{m}$$

Step 3

$$A_s = 2\pi [r_{2c}^2 - r_1^2]$$

$$= 2\pi[(0.2425)^2 - (0.100)^2]$$

$$A_s = 0.30650m^2$$

Step4

$$A_m = t[r_{2c} - r_1]$$

$$A_m = 0.005[0.2425 - 0.100]$$

$$A_m = 7.125 \times 10^{-4}m^2$$

From the graph, we know that, [HMT data book page no.51]

$$X_{axis} = (L_c)^{1.5} \left[\frac{h}{KA_m} \right]^{0.5}$$

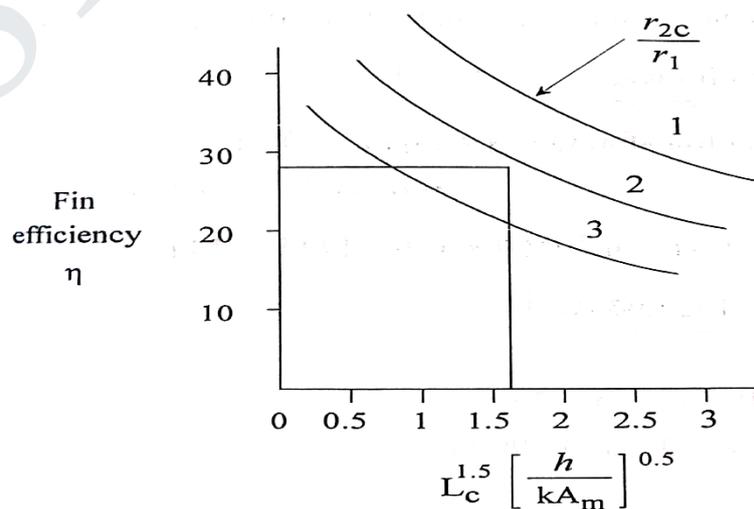
$$= (0.1425)^{1.5} \left[\frac{140}{220 \times 7.125 \times 10^{-4}} \right]^{0.5}$$

$$X_{axis} = 1.60$$

$$\text{Curve} \rightarrow \frac{r_{2c}}{r_1} = \frac{0.2425}{0.1} = 2.425$$

X_{axis} value is 1.60

Curve value is 2.425



By using these values we can find fin efficiency, η from graph

$$\text{Fin Efficiency } \eta = 28 \%$$

Heat transfer = $\eta A_s h (T_b - T_\infty)$ from HMT data book P.No 50

$$= 0.28 \times 0.30650 \times 140 \times [443 - 298]$$

$$Q = 1742.99W$$

9. A metallic sphere of radius 10mm is initially at a uniform temperature of 400°C. It is heat treated by first cooling it in air ($h=10 \text{ W/m}^2\text{k}$) at 20°C until its central temperature reaches 335°C . It is then quenched in a water bath at 20°C with $h=6000 \text{ W/m}^2\text{K}$ until the centre of the sphere cools from 335°C to 50°C. compute the time required for cooling in air and water for the following physical properties of the sphere.

$$\text{Density, } \rho = 3000 \text{ kg/m}^3$$

$$c = 1000 \text{ J/kgK}$$

$$K = 20 \text{ W/mK}$$

$$\alpha = 6.66 \times 10^{-6} \text{ m}^2/\text{s}$$

Given

$$\text{Density, } \rho = 3000 \text{ kg/m}^3$$

$$c = 1000 \text{ J/kgK} \quad K = 20 \text{ W/mK}$$

$$\alpha = 6.66 \times 10^{-6} \text{ m}^2/\text{s}$$

To find

Surface temperature at end of cooling in water.

Solution

Step-1

i) Cooling in air .

Let us check whether lumped capacity method can be used here

$$Bi = \frac{hr_0}{3k} = \frac{10 \times 0.01}{3 \times 20} = 16.66 \times 10^{-4} \ll 0.1$$

From HMT data book P.No 58

$$\therefore \frac{T - T_\infty}{T_0 - T_\infty} = \exp \left[- \left\{ \frac{hA}{\rho c V} \right\} \cdot t \right]$$

$$t = \frac{\rho c V}{hA} \ln \frac{T_0 - T_\infty}{T - T_\infty} = \frac{\rho r_0 c}{3h} \ln \frac{T_0 - T_\infty}{T - T_\infty}$$

$$t = 188s$$

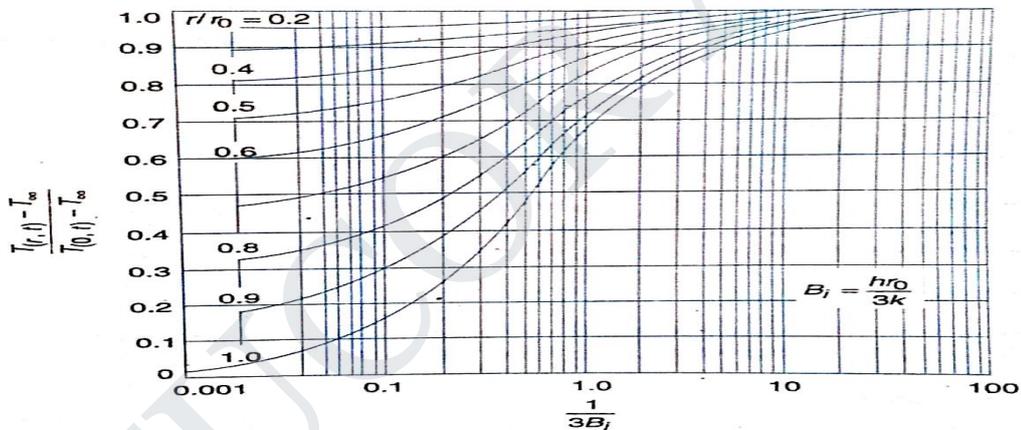
$$= \frac{3000 \times 0.01 \times 1000}{3 \times 10} \ln \frac{400 - 20}{335 - 20}$$

Step-2

ii) Cooling in water

$$B_i(\text{for lumped capacity method}) = \frac{hr_0}{3k} = \frac{6000 \times 0.01}{3 \times 20} = 1.0 > 0.1$$

So the lumped capacity method cannot be employed, but heisler charts can be used



$$\frac{1}{B_i} = \frac{k}{hr_0} = \frac{20}{6000 \times 0.01} = 0.33$$

$$\frac{T_{(0,t)} - T_\infty}{T_0 - T_\infty} = \frac{50 - 20}{335 - 20} = 0.095$$

$$F_o = \frac{\alpha t}{r_0^2} = 0.5$$

$$t = \frac{F_o r_0^2}{\alpha} = \frac{0.5 \times 0.01^2}{6.66 \times 10^{-6}} = 7.5s$$

The surface temperature at the end of quenching in water may be obtained from fig with

$$\frac{1}{3B_i} = 0.33$$

$$\frac{r}{r_0} = 1$$

$$\frac{T(r_0) - T_\infty}{T(0,t) - T_\infty} = 0.33$$

$$T(r_0) = [0.33 \times (50 - 20)] + 20 = 30^\circ\text{C}$$

$$T(r_0) = 30^\circ\text{C}$$

10. A thermocouple junction is in the form of 8 mm diameter sphere. Properties of material are $c=420 \text{ J/kg}^\circ\text{C}$, $\rho=8000 \text{ kg/m}^3$, $k=40 \text{ W/m}^\circ\text{C}$ and $h=40 \text{ W/m}^2\text{C}$. The junction is initially at 40°C and inserted in a stream of hot air at 300°C . Find

i) Time constant of the thermocouple

ii) The thermocouple is taken out from the hot air after 10 seconds and kept in still air at 30°C . Assuming the heat transfer coefficient in air $10\text{W/m}^2\text{C}$, find the temperature attained by the junction 20 seconds after removing from hot air.(Nov/Dec 2008)

Given

$$R=4 \text{ mm}= 0.004\text{m}$$

$$C= 420 \text{ J/kg}^\circ\text{C}$$

$$\rho=8000 \text{ kg/m}^3$$

$$k=40 \text{ W/m}^\circ\text{C}$$

$$h=40 \text{ W/m}^2\text{C} \text{ (gas stream)}$$

$$h=10 \text{ W/m}^2\text{C} \text{ (gas air)}$$

To Find

i) Time constant of the thermocouple τ^*

ii) The temperature attained by the junction (t)

Solution

Step-1

$$\tau^* = \frac{\rho VC}{hA_s} = \frac{\rho \times \left[\frac{4}{3}\pi R^3\right] \times c}{h \times 4\pi R^2} = \frac{\rho Rc}{3h}$$

$$\tau^* = \frac{8000 \times 0.004 \times 420}{3 \times 40} = 112 \text{ s}$$

$$\tau^* = 112 \text{ s}$$

Step-2

$$t_i = 40^\circ\text{C}, t_a = 300^\circ\text{C}, \tau = 10\text{s}$$

The temperature variation with respect to time during heating (when dipped in gas stream) is given by

From HMT data book P.No 58

$$\frac{t - t_a}{t_i - T_a} = \exp \left[- \left\{ \frac{hA}{\rho c V} \right\} \cdot t \right]$$

$$\frac{t - 300}{40 - 300} = \exp \left[- \left\{ \frac{\tau}{\tau^*} \right\} \right] = e^{(10/112)}$$

$$\frac{1}{e^{(10/112)}} = 0.9146$$

$$t = 300 + 0.9146(40 - 300) = 62.2^\circ\text{C}$$

$$t = 62.2^\circ\text{C}$$

The temperature variation with respect to time during cooling (when exposed to air) is given by

$$\frac{t - t_a}{t_i - T_a} = e^{-\frac{\tau}{\tau^*}}$$

Where

$$\tau^* = \frac{\rho R c}{3h} = \frac{8000 \times 0.004 \times 420}{3 \times 10} = 448\text{s}$$

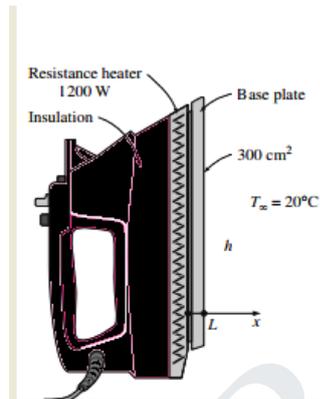
$$\frac{t - 30}{62.2 - 30} = e^{-\left(\frac{20}{448}\right)}$$

$$t = 30 + 0.9563(62.2 - 30) = 60.79^\circ\text{C}$$

$$t = 60.79^\circ\text{C}$$

PART C - 15 Marks (Questions and Answers)

1. Heat Conduction in the Base Plate of an Iron Consider the base plate of a 1200-W household iron that has a thickness of $L = 0.5$ cm, base area of $A = 300$ cm², and thermal conductivity of $k = 15$ W/m · °C. The inner surface of the base plate is subjected to uniform heat flux generated by the resistance heaters inside, and the outer surface loses heat to the surroundings at $T_{\infty} = 20^{\circ}\text{C}$ by convection, as shown in Figure



Taking the convection heat transfer coefficient to be $h = 80$ W/m² · °C and disregarding heat loss by radiation, obtain an expression for the variation of temperature in the base plate, and evaluate the temperatures at the inner and the outer surfaces.

SOLUTION

The base plate of an iron is considered. The variation of temperature in the plate and the surface temperatures are to be determined.

Assumptions

- 1 Heat transfer is steady since there is no change with time.
- 2 Heat transfer is one-dimensional since the surface area of the base plate is large relative to its thickness, and the thermal conditions on both sides are uniform.
- 3 Thermal conductivity is constant.
- 4 There is no heat generation in the medium.
- 5 Heat transfer by radiation is negligible.
- 6 The upper part of the iron is well insulated so that the entire heat generated in the resistance wires is transferred to the base plate through its inner surface.

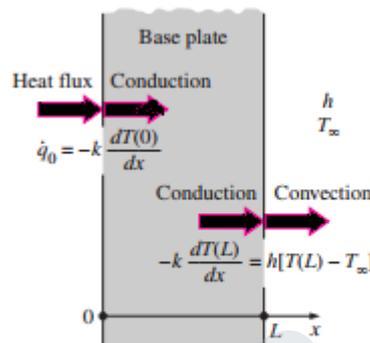
Properties

The thermal conductivity is given to be $k = 15 \text{ W/m} \cdot ^\circ\text{C}$.

Analysis The inner surface of the base plate is subjected to uniform heat flux at a rate of

$$q_0 = \frac{Q_0}{A_{\text{base}}} = \frac{1200}{0.03} = 40,000 \text{ W/m}^2$$

The outer side of the plate is subjected to the convection condition. Taking the direction normal to the surface of the wall as the x-direction with its origin on the inner surface, the differential equation for this problem can be expressed as fig



$$\frac{d^2 T}{dx^2} = 0$$

With the boundary conditions

$$-k \frac{dT(0)}{dx} = q_0 = 40000 \text{ W/m}^2$$

$$-k \frac{dT(L)}{dx} = h[T(L) - T_\infty]$$

The general solution of the differential equation is again obtained by two successive integrations to be

$$\frac{dT}{dx} = C_1$$

And

$$T(x) = C_1 x + C_2 \text{ -----(1)}$$

Where C_1 and C_2 are arbitrary constants. Applying the first boundary condition,

$$-k \frac{dT(0)}{dx} = q_0 \longrightarrow -k C_1 = q_0 \longrightarrow C_1 = -\frac{q_0}{k}$$

$$-k \frac{dT(L)}{dx} = h[T(L) - T_\infty] \longrightarrow -k C_1 = h[(C_1 L + C_2) - T_\infty]$$

Substituting $C_1 = -\frac{q_0}{k}$ and solving for C_2 We obtain

$$C_2 = T_\infty + \frac{q_0}{h} + \frac{q_0}{k} L$$

Now substituting C_1 and C_2 into the general solution (1) gives

$$T(x) = T_\infty + q_0 \left(\frac{L-x}{k} + \frac{1}{h} \right) \text{-----(2)}$$

Which is the solution for the variation of the temperature in the plate. The temperatures at the inner and outer surfaces of the plate are determined by substituting $x=0$ and $x=L$, respectively, into the relation (2)

$$\begin{aligned} T(0) &= T_\infty + q_0 \left(\frac{L}{k} + \frac{1}{h} \right) \\ &= 20^\circ \text{C} + (40000 \text{ W/m}^2) \left(\frac{0.005 \text{ m}}{15} + \frac{1}{80} \right) = 533^\circ \text{C} \end{aligned}$$

And

$$T(L) = T_\infty + q_0 \left(0 + \frac{1}{h} \right) = 20^\circ \text{C} + \frac{40000}{80} = 520^\circ \text{C}$$

Discussion Note that the temperature of the inner surface of the base plate will be 13°C higher than the temperature of the outer surface when steady operating conditions are reached. Also note that this heat transfer analysis enables us to calculate the temperatures of surfaces that we cannot even reach. This example demonstrates how the heat flux and convection boundary conditions are applied to heat transfer problems.

2. A person is found dead at 5 PM in a room whose temperature is 20°C . The temperature of the body is measured to be 25°C when found, and the heat transfer coefficient is estimated to be $h = 8 \text{ W/m}^2 \cdot ^\circ \text{C}$. Modeling the body as a 30-cm-diameter, 1.70-m-long cylinder, estimate the time of death of that person

SOLUTION A body is found while still warm. The time of death is to be estimated.

Assumptions **1** The body can be modeled as a 30-cm-diameter, 1.70-m-long cylinder. **2** The thermal properties of the body and the heat transfer coefficient are constant. **3** The radiation effects are negligible. **4** The person was healthy(!) when he or she died with a body temperature of 37°C .

Properties The average human body is 72 percent water by mass, and thus we can assume the body to have the properties of water at the average temperature of $(37 + 25)/2 = 31^\circ \text{C}$; $k = 0.617 \text{ W/m} \cdot ^\circ \text{C}$, $\rho = 996 \text{ kg/m}^3$, and $C_p = 4178 \text{ J/kg} \cdot ^\circ \text{C}$

Analysis The characteristic length of the body is

$$L_c = \frac{V}{A_s} = \frac{\pi r_0^2 L}{2\pi r_0 L + 2\pi r_0^2} = \frac{\pi (0.15)^2 (1.7)}{2\pi(0.15)(1.7) + 2\pi(0.15)^2} = 0.0689$$

Then the biot number becomes

$$B_i = \frac{hL_c}{k} = \frac{8 \times 0.0689}{0.617} = 0.89 > 0.1$$

Therefore lumped system analysis is not applicable. However, we can still use it to get a rough estimate of the time of death.

$$\frac{T(t) - T_\infty}{T_i - T_\infty} = e^{-bt} \text{-----(1)}$$

The exponent b in this case is

$$b = \frac{hA_s}{\rho C_p V} = \frac{h}{\rho C_p L_c} = \frac{8}{996 \times 4178 \times 0.0689} = 2.79 \times 10^{-5}$$

now substitute these values into equation (1)

$$\frac{25 - 20}{37 - 20} = e^{-2.79 \times 10^{-5} t}$$

$t = 43860 \text{ s} = 12.2 \text{ h}$
--

The person died about 12 h before the body was found and thus the time of death is 5 AM.

UNIT: II - CONVECTION
PART A - 2 Marks (Questions and Answers)

1. Define critical Reynolds number. What is its typical value for flow over a flat plate and flow through a pipe? (May 2013, Nov/Dec 16)

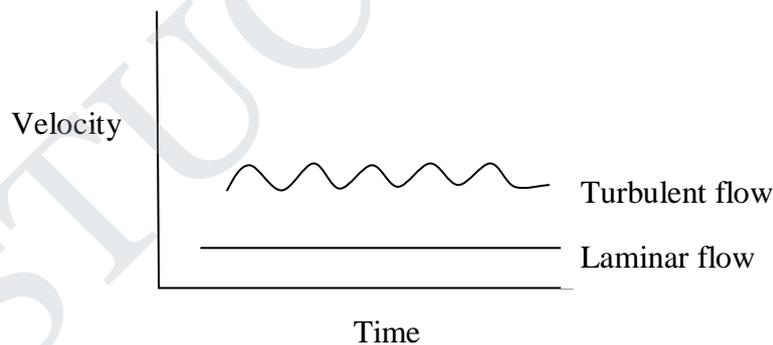
The critical Reynolds number refers to the transition from laminar to turbulent flow.

The critical Reynolds number for flow over a flat plate is 5×10^5 ; the critical Reynolds number for flow through a pipe is 4000.

2. How does or Distinguish laminar flow differ from turbulent flow? (May 2013 & May 2015)

Laminar flow: Laminar flow is sometimes called stream line flow. In this type of flow, the fluid moves in layers and each fluid particle follows a smooth continuous path. The fluid particles in each layer remain in an orderly sequence without mixing with each other.

Turbulent flow: In addition to the laminar type of flow, a distinct irregular flow is frequently observed in nature. This type of flow is called turbulent flow. The path of any individual particle is zig-zag and irregular.



3. Differentiate viscous sub layer and buffer layer. (May 2014)

In the turbulent boundary layer, a very thin layer next to the wall where viscous effect is dominant called the viscous sub layer. The velocity profile in this layer is very nearly linear and the flow is streamlined.

In the turbulent boundary layer, next to viscous sub layer, a layer called **buffer layer** in which turbulent effects are becoming significant, but the flow is still dominated by viscous effects.

4. Define grashoff number and prandtl number. Write its significance. (May 2014 & Nov 2014 & Nov 2015-Reg 2008)(Nov 2015) (APR/MAY 2017)

Grashoff number is defined as the ratio of product of inertia force and buoyancy force to the square of viscous force.

$$Gr = \frac{\text{Inertia Force} * \text{Buoyancy Force}}{(\text{Viscous Force})^2} \quad [\text{HMT Data Book, P.No 112}]$$

Significance: Grashoff number has a role in free convection similar to that played by Reynolds number in forced convection.

Prandtl number is the ratio of the momentum diffusivity of the thermal diffusivity.

$$Pr = \frac{\text{Momentum Diffusivity}}{\text{Thermal Diffusivity}} \quad [\text{HMT Data Book, P.No. 112}]$$

Significance: Prandtl number provides a measure of the relative effectiveness of the momentum and energy transport by diffusion.

5. Define velocity boundary layer thickness. (May 2015)

The region of the flow in which the effects of the viscous shearing forces caused by fluid viscosity are felt is called velocity boundary layer. The velocity boundary layer thickness, δ , is defined as the distance from the surface at which velocity, $u = 0.99V$

6. Air at 27°C and 1 atmospheric flow over a flat plate at a speed of 2m/s. Calculate boundary layer thickness at a distance 40 cm from leading edge of plate. At 27°C viscosity (air) = $1.85 * 10^{-5}$ kg/ms. (Nov 2012)

Given Data:

$$T = 27^\circ\text{C} = 27 + 273 = 300\text{K}$$

$$P = 1 \text{ atm} = 1 \text{ bar} = 1.01325 * 10^5 \text{ N/m}^2$$

$$U = 2 \text{ m/s}$$

$$\mu = 1.85 * 10^{-5} \text{ kg/ms. (At } 27^\circ\text{C)}$$

$$R = 287 \text{ (Gas constant)}$$

To Find: δ at $X = 40 \text{ cm} = 0.4 \text{ m}$

Solution:

$$\begin{aligned} \text{Step: 1 Density } \rho &= P/RT \\ &= \frac{1.01325 * 10^5}{(287 * 300)} \end{aligned}$$

$$= 1.177 \text{ Kg/m}^3$$

(Note: If Surface temperature (T_w) is given, then properties to be taken for T_f Value.)

Step: 2 Reynolds Number $Re = \rho UX / \mu$ [HMT Data Book, P.No. 112]

$$= \frac{1.177 * 2 * 0.4}{1.85 * 10^{-5}}$$

$$= 55160. \text{ (Re} < 5 * 10^5, \text{ flow is laminar)}$$

Step: 3 Boundary layer thickness $\delta = 5 * X * (Re)^{-0.5}$

[HMT Data Book, P.No.113]

$$= 5 * 0.4 * (55160)^{-0.5}$$

$$= 0.0085 \text{ m}$$

Boundary layer thickness δ at X (0.4m) = 0.0085 m

7. A square plate 40*40 cm maintained at 400K is suspended vertically in atmospheric air at 300 K. Determine the boundary layer thickness at trailing edge of the plate. (Nov 2012)

Given Data:

Length of horizontal plate $X = 40 \text{ cm} = 0.4 \text{ m}$

Wide $W = 40 \text{ cm} = 0.40 \text{ m}$

Plate temperature $T_w = 400 \text{ K} = 127^\circ \text{C}$

Fluid temperature $T_\alpha = 300 \text{ K} = 27^\circ \text{C}$

$$\Delta T = (T_w - T_\alpha) = 400 - 300 = 100$$

To Find: δ at $X = 40 \text{ cm} = 0.4 \text{ m}$

Solution:

$$\text{Step: 1 Film Temperature } (T_f) = \frac{T_w + T_\alpha}{2}$$

$$= \frac{127 + 27}{2} = 77^\circ \text{C} = 350 \text{ K}$$

2

Step: 2 Properties of air at 77°C (apprx 75°C)

[HMT Data Book, P.No.34]

$$v = 20.56 * 10^{-6} \text{ m}^2/\text{s}$$

$$\text{Pr} = 0.693$$

Step: 3 Find $\beta = 1 / T_f$ in K

$$= 1 / 350$$

$$= 2.857 * 10^{-3} \text{ K}^{-1}$$

Step: 4 For free Convection (Note: As Velocity not given)

$$\text{Gr} = \frac{g * \beta * X^3 * \Delta T}{v^2} \quad [\text{HMT Data Book, P.No.135}]$$

$$= \frac{9.81 * 2.857 * 10^{-3} * (0.4)^3 * (400-300)}{(20.56 * 10^{-6})^2}$$

$$= 4.24 * 10^8$$

Step: 5 Boundary layer thickness $\delta = 3.93 * X * (\text{Pr})^{-0.5} * (0.952 + \text{Pr})^{0.25} * \text{Gr}^{-0.25}$

[HMT Data Book, P.No.135]

$$= 3.93 * 0.4 * (0.693)^{-0.5} * (0.952 + 0.693)^{0.25} * (4.24 * 10^8)^{-0.25}$$

$$= 0.0155 \text{ m}$$

Boundary layer thickness δ at X (0.4m) = 0.0155 m
--

8. Define the term thermal boundary layer thickness. (Nov 2013)

The thickness of the thermal boundary layer δ_t at any location along the surface is defined as the distance from the surface at which the temperature difference equals to $0.99(T_\alpha - T_s)$, in general $T = 0.99T_\alpha$

9. Why heat transfer coefficient for natural convection is much lesser than that for forced convection? (Nov 2013 & May 2016)

Heat transfer coefficient depends on the fluid velocity.

In natural convection, the fluid motion occurs by natural means such as buoyancy. Since the fluid velocity associated with natural convection is relatively low, the heat transfer coefficient encountered in natural convection is low.

The reason for higher heat transfer rates in forced convection is because the hot air surrounding the hot body is immediately removed by the flow of air around it. This is why forced convection heat transfer coefficient is greater than natural convection heat transfer coefficient.

10. Name four dimensions used for dimensional analysis. (Nov 2014)

1. Velocity
2. Density
3. Heat transfer coefficient
4. Thermal conductivity

11. Mention the significance of boundary layer. (Nov 2015)

Boundary layer is the layer of fluid in the immediate vicinity of a bounding surface where the effects of viscosity are significant.

12. What is Dittus Boelter equation? When does it apply? (Nov 2015)

Dittus-Boelter equation (for fully developed internal flow - turbulent flow) is an explicit function for calculating the Nusselt number. It is easy to solve but is less accurate when there is a large temperature difference across the fluid. It is tailored to smooth tubes, so use for rough tubes (most commercial applications) is cautioned.

The Dittus-Boelter equation is:

$$Nu_D = 0.023 Re_D^{0.8} Pr^n \quad [\text{HMT Data Book, P.No.126}]$$

13. What is the difference between friction factor and friction coefficient? (May 2016)

Friction factor, a dimensionless quantity used in the Darcy-Weisbach equation, for the description of friction losses in pipe flow as well as open-channel flow. Friction coefficient applied at the value of x ($x=x$ -Local friction coefficient, $x=L$ - Average friction coefficient)

14. Differentiate free and forced convection. (May 2016) (Nov/Dec 16)

Natural convection, or free convection, occurs due to temperature differences which affect the density, and thus relative buoyancy, of the fluid. Free convection is governed by Grashoff number and Prandtl number.

Example: Rise of smoke from a fire.

In forced convection, fluid movement results from external forces such as a fan or pump. Forced convection is typically used to increase the rate of heat exchange. It is governed by the value of the Reynolds number.

Example: Cooling of IC engines with fan in a radiator.

15. Differentiate hydrodynamic and thermal boundary layer. (May 2016)

The hydrodynamic boundary layer is a region of a fluid flow, near a solid surface, where the flow patterns (velocity) are directly influenced by viscous drag from the surface wall. The velocity of the fluid is less than 99% of free stream velocity.

The thermal boundary layer is a region of a fluid flow, near a solid surface, where the fluid temperatures are directly influenced by heating or cooling from the surface wall. The temperature of the fluid is less than 99% of free stream temperature.

16. What are the difference between natural convection and forced convection? (Nov/Dec 16)

Natural convection is a mechanism of heat transportation in which the fluid motion is not generated by an external source.

Forced convection is a mechanism, or type of heat transport in which fluid motion is generated by an external source (like a pump, fan, suction device, etc.)

PART B - 13 Marks (Questions and Answers)

1. Air at 25°C at the atmospheric pressure is flowing over a flat plate at 3m/s. If the plate is 1m wide and the temperature $T_w = 75^\circ\text{C}$. Calculate the following at a location of 1m from leading edge.

- a) Hydrodynamic boundary layer thickness,
- b) Local friction coefficient,
- c) Thermal heat transfer coefficient,
- d) Local heat transfer coefficient.

Given Data:

Fluid temperature, $T_\alpha = 25^\circ\text{C}$

Velocity, $U = 3\text{m/s}$

Wide, $W = 1\text{m}$

Plate surface temperature, $T_w = 75^\circ\text{C}$

Distance, $x = 1\text{m}$

To Find: δ_{hx} , C_{fx} , δ_{Tx} , h_x ,

Solution:

[From HMT Data Book, P.No.113]

Film temperature, $T_f = \frac{T_w + T_\alpha}{2}$

$$= \frac{75 + 25}{2} = 323\text{K}$$

$$T_f = 50^\circ\text{C}$$

Properties of air at 50°C:

[From HMT Data Book, P.No.34]

Density, $\rho = 1.093\text{kg/m}^3$

Kinematic viscosity, $\nu = 17.95 \times 10^{-6} \text{m}^2/\text{s}$

Prandtl number $Pr = 0.698$

Thermal conductivity, $k = 0.02826 \text{W/mk}$

Reynolds number, $Re = UL/\nu$

[From HMT Data Book, P.No.112]

[∵ $x = L = 1\text{m}$]

$$\frac{3 \times 1}{17.95 \times 10^{-6}} = 1.67 \times 10^5$$

$$Re = 1.67 \times 10^5 < 5 \times 10^5$$

Since $Re < 5 \times 10^5$ flow is laminar.

For the plate, laminar flow.

[From HMT Data Book, P.No.113]

1. Hydrodynamic boundary layer thickness,

$$\delta_{hx} = 5 \cdot x \cdot Re^{-0.5}$$

$$= 5 \cdot x \cdot (1.67 \times 10^5)^{-0.5}$$

$$\delta_{hx} = 0.0122\text{m}$$

2. Local friction coefficient,

[From HMT Data Book, P.No.113]

$$C_{fx} = 0.664 Re^{-0.5}$$

$$= 0.664 \cdot (1.67 \times 10^5)^{-0.5}$$

$$C_{fx} = 1.62 \times 10^{-3}$$

3. Thermal heat transfer coefficient,

[From HMT Data Book, P.No.113]

$$\begin{aligned}\delta_{Tx} &= \delta_{hx} * (Pr)^{-0.333} \\ &= 0.0122 * (0.698)^{-0.333}\end{aligned}$$

$\delta_{Tx} = 0.01375$

4. Local heat transfer coefficient, h_x

[From HMT Data Book, P.No.113]

$$\begin{aligned}\text{Local nusselt number } Nu_x &= 0.332 Re^{0.5} (Pr)^{0.333} \\ &= 0.332 (1.67 * 10^5)^{0.5} (0.698)^{0.333} \\ Nu_x &= 120.415\end{aligned}$$

[From HMT Data Book, P.No.112]

$$Nu_x = \frac{h_x * L}{k}$$

$$120.415 = \frac{h_x * 1}{0.02826}$$

[∵ $x=L=1m$]

Local heat transfer coefficient, $h_x = 3.4W/m^2K$
--

Result:

- a) $\delta_{hx} = 0.0122m$
- b) $C_{fx} = 1.62 * 10^{-3}$
- c) $\delta_{Tx} = 0.01375$
- d) $h_x = 3.4W/m^2K$

2. Air at 290°C flows over a flat plate at a velocity of 6 m/s. The plate is 1m long and 0.5 m wide. The pressure of the air is 6 KN/m². If the plate is maintained at a temperature of 70°C, estimate the rate of heat removed from the plate.

Given:

Fluid temperature $T_{\infty} = 290^{\circ}C$

Velocity $U = 6 m/s$.

Length $L = 1 m$

Wide $W = 0.5 m$

Pressure of air $P = 6 KN/m^2 = 6 \times 10^3 N/m^2$

Plate surface temperature $T_w = 70^\circ\text{C}$

To find:

Heat removed from the plate

Solution:

[From HMT Data Book, P.No.113]

$$\text{Film temperature } T_f = \frac{T_w + T_\infty}{2}$$

$$T_f = \frac{70 + 290}{2}$$

$$T_f = 180^\circ\text{C}$$

Properties of air at 180°C (At atmospheric pressure)

[From HMT Data Book, P.No.34]

$$\rho = 0.799 \text{ Kg/m}^3$$

$$\nu = 32.49 \times 10^{-6} \text{ m}^2/\text{s}$$

$$\text{Pr} = 0.681$$

$$K = 37.80 \times 10^{-3} \text{ W/mK}$$

Note: Pressure other than atmospheric pressure is given, so kinematic viscosity will vary with pressure. Pr, K, C_p are same for all pressures.

$$\text{Kinematic viscosity } \nu = \nu_{\text{atm}} \frac{P_{\text{atm}}}{P_{\text{given}}}$$

[$\because 1 \text{ bar} = 1 \times 10^5 \text{ N/m}^2$]

$$\nu = 32.49 \times 10^{-6} \times \frac{1 \times 10^5}{6 \times 10^3}$$

$$\text{Kinematic viscosity } \nu = 5.145 \times 10^{-4} \text{ m}^2/\text{s}$$

[From HMT Data Book, P.No.112]

$$\text{Reynolds number } \text{Re} = \frac{UL}{\nu}$$

$$= \frac{6 \times 1}{5.145 \times 10^{-4}}$$

$$\text{Re} = 1.10 \times 10^4 - 5 \times 10^5$$

Since $\text{Re} < 5 \times 10^5$, flow is laminar

For plate, laminar flow, $UL \nu$

[From HMT Data Book, P.No.113]

$$\text{Local nusselt number } \text{Nu}_x = 0.332 \text{ Re}^{0.5} (\text{Pr})^{0.333}$$

$$= 0.332 (1.10 \times 10^4)^{0.5} (0.681)^{0.333}$$

$$Nu_x = 30.63$$

[From HMT Data Book, P.No.112]

$$Nu_x = \frac{h_x L}{K}$$

$$30.63 = \frac{h_x \times 1}{37.80 \times 10^{-3}} \quad [\because L = 1 \text{ m}]$$

Local heat transfer coefficient $h_x = 1.15 \text{ W/m}^2\text{K}$

Average heat transfer coefficient $h = 2 \times h_x$

$$h = 2 \times 1.15$$

$$h = 2.31 \text{ W/m}^2\text{K}$$

Heat transferred $Q = h A (T_\alpha - T_w)$

$$= 2.13 \times (1 \times 0.5) \times (563 - 343)$$

$$Q = 254.1 \text{ W}$$

Heat transfer from both side of the plate = 2×254.1

$$= 508.2 \text{ W.}$$

Result: Heat transfer from both side of the plate = 508.2 W

3. A large vertical plate 4 m height is maintained at 606°C and exposed to atmospheric air at 106°C. Calculate the heat transfer is the plate is 10 m wide.

Given :

Vertical plate length (or) Height, $L = 4 \text{ m}$

Wall temperature, $T_w = 606^\circ\text{C}$

Air temperature, $T_\infty = 106^\circ\text{C}$

Wide, $W = 10 \text{ m}$

To find:

a) Heat transfer, (Q)

Solution:

[From HMT Data Book, P.No.113]

$$\begin{aligned} \text{Film temperature } T_f &= \frac{T_w + T_\alpha}{2} \\ &= \frac{606 + 106}{2} \end{aligned}$$

$$T_f = 356^\circ\text{C}$$

[From HMT Data Book, P.No.34]

Properties of air at $356^{\circ}\text{C} = 350^{\circ}\text{C}$

Density, $\rho = 0.566\text{kg/m}^3$

Kinematic viscosity, $\nu = 55.46 \times 10^{-6} \text{m}^2/\text{s}$

Prandtl number $\text{Pr} = 0.698$

Thermal conductivity, $k = 49.08 \times 10^{-3} \text{W/mk}$

Coefficient of thermal expansion $\beta = \frac{1}{T_f \text{ in K}}$

$$= \frac{1}{356 + 273} = \frac{1}{629}$$

$$\beta = 1.58 \times 10^{-3} \text{K}^{-1}$$

$$\text{Grashof number } Gr = \frac{g \times \beta \times L^3 \times \Delta T}{\nu^2}$$

$$\Rightarrow Gr = \frac{9.81 \times 2.4 \times 10^{-3} \times (4)^3 \times (606 - 106)}{(55.46 \times 10^{-6})^2}$$

$$\Rightarrow Gr = 1.61 \times 10^{11}$$

$$Gr \text{ Pr} = 1.61 \times 10^{11} \times 0.676$$

$$Gr \text{ Pr} = 1.08 \times 10^{11}$$

Since $Gr \text{ Pr} > 10^9$, flow is turbulent

For turbulent flow,

$$\text{Nusselt number } Nu = 0.10 [Gr \text{ Pr}]^{0.333}$$

$$\Rightarrow Nu = 0.10 [1.08 \times 10^{11}]^{0.333}$$

$$Nu = 471.20$$

[From HMT Data Book, P.No.112]

$$\text{Nusselt number } Nu = \frac{hL}{K}$$

$$\Rightarrow 472.20 = \frac{h \times 4}{49.08 \times 10^{-3}}$$

Heat transfer coefficient $h = 5.78 \text{ W/m}^2\text{K}$

Heat transfer $Q = h A \Delta T$

$$= h \times W \times L \times (T_w - T_{\infty})$$

$$= 5.78 \times 10 \times 4 \times (606 - 106) \quad Q = 115600 \text{ W}$$

$$Q = 115.6 \times 10^3 \text{ W}$$

Result:

Heat transfer $Q = 115.6 \times 10^3 \text{ W}$

4. A thin 100 cm long and 10 cm wide horizontal plate is maintained at a uniform temperature of 150°C in a large tank full of water at 75°C. Estimate the rate of heat to be supplied to the plate to maintain constant plate temperature as heat is dissipated from either side of plate.

Given :

Length of horizontal plate, $L = 100 \text{ cm} = 1 \text{ m}$

Wide, $W = 10 \text{ cm} = 0.10 \text{ m}$

Plate temperature, $T_w = 150^\circ\text{C}$

Fluid temperature, $T_\infty = 75^\circ\text{C}$

To find:

a) Heat loss (Q) from either side of plate

Solution:

$$\text{Film temperature, } T_f = \frac{T_w + T_\infty}{2} \quad [\text{From HMT Data Book, P.No.113}]$$

$$= \frac{150 + 75}{2} = 323 \text{ K}$$

$$T_f = 112.5^\circ\text{C}$$

Properties of water at 112.5°C

$\rho = 951 \text{ Kg/m}^3$

$\nu = 0.264 \times 10^{-6} \text{ m}^2/\text{s}$

$Pr = 1.55$

$k = 683 \times 10^{-3} \text{ W/mK}$

$$\text{Coefficient of thermal expansion } \beta = \frac{1}{T_f \text{ in K}} = \frac{1}{112.5 + 273} = 2.59 \times 10^{-3} \text{ K}^{-1}$$

$$\text{Grashof Number } Gr = \frac{g \times \beta \times L^3 \times \Delta T}{\nu^2}$$

For horizontal plate,

$$\text{Characteristic length } L_c = \frac{W}{2} = \frac{0.10}{2}$$

$$L_c = 0.05 \text{ m}$$

$$Gr = \frac{9.81 \times 2.59 \times 10^{-3} \times (0.05)^3 \times (150 - 75)}{(0.264 \times 10^{-6})^2}$$

$$Gr = 3.41 \times 10^9$$

$$GrPr = 3.14 \times 10^9 \times 1.55$$

$$GrPr = 5.29 \times 10^9$$

Gr Pr value is in between 8×10^6 and 10^{11}

i.e., $8 \times 10^6 < GrPr < 10^{11}$

For horizontal plate, upper surface heated:

$$\text{Nusselt number } Nu = 0.15 (GrPr)^{0.333}$$

[From HMT Data Book, P.No.114]

$$Nu = 0.15 (5.29 \times 10^9)^{0.333}$$

$$Nu = 259.41$$

$$\text{Nusselt number } Nu = \frac{h_u L_c}{K}$$

$$259.41 = \frac{h_u \times 0.05}{683 \times 10^{-3}}$$

$$h_u = 3543.6 \text{ W/m}^2\text{K}$$

Upper surface heated, heat transfer coefficient $h_u = 3543.6 \text{ W/m}^2\text{K}$

For horizontal plate, lower surface heated:

$$\text{Nusselt number } Nu = 0.27 [GrPr]^{0.25}$$

$$Nu = 0.27 [5.29 \times 10^9]^{0.25}$$

$$Nu = 72.8$$

[From HMT Data Book, P.No.113]

$$\text{Nusselt number } Nu = \frac{h_l L_c}{K}$$

$$72.8 = \frac{h_l L_c}{K}$$

$$72.8 = \frac{h_l \times 0.05}{683 \times 10^{-3}}$$

$$h_1 = 994.6 \text{ W/m}^2\text{K}$$

Lower surface heated, heat transfer coefficient $h_1 = 994.6 \text{ W/m}^2\text{K}$

Total heat transfer $Q = (h_u + h_1) \times A \times \Delta T$

$$= (h_u + h_1) \times W \times L \times (T_w - T_\infty)$$

$$= (3543.6 + 994.6) \times 0.10 \times (150 - 75)$$

$Q = 34036.5 \text{ W}$

Result:

Total heat transfer $Q = 34036.5 \text{ W}$

5. Explain in detail about the boundary layer concept.

The concept of a boundary layer as proposed by Prandtl forms the starting point for the simplification of the equation of motion and energy.

When a real i.e., viscous fluid, flows along a stationary solid boundary, a layer of fluid which comes in contact with the boundary surface and undergoes retardation. This retarded layer further causes retardation for the adjacent layer of the fluid. So a small region is developed in the immediate vicinity of the boundary surface in which the velocity of the flowing fluid increases rapidly from zero at the boundary surface and approaches the velocity of the main stream.

Types of boundary layer

1. Velocity boundary layer (or) hydrodynamic boundary layer
2. Thermal boundary layer

Velocity boundary layer (or) hydrodynamic boundary layer

In the velocity boundary layer, the velocity of the fluid is less than 99% of the free stream velocity.

The fluid approaches the plate in the x-direction with uniform velocity u_∞ . The fluid particles in the fluid layer adjacent to the surface get zero velocity. This motionless layer acts to retard the motion of particles in the adjoining fluid layer as a result of friction between the particles of these two adjoining fluid layers at two different velocities. This fluid layer then acts to retard the motion of particles of the next fluid layer and so on, until a distance $y = \delta$ from the surface is reached, where

these effects become negligible and the fluid velocity u reaches the free stream velocity u_∞ as a result of frictional effects between the fluid layers.

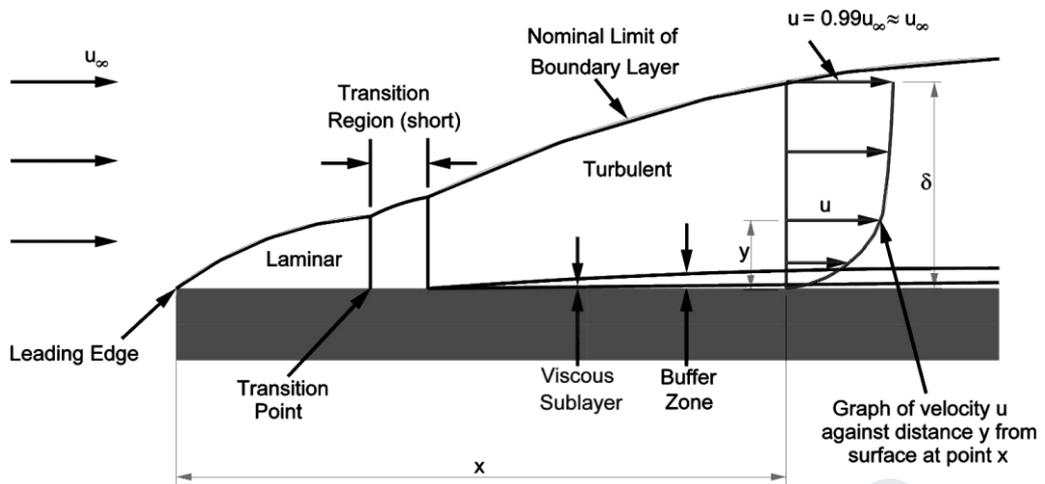
Thermal boundary Layer:

In the Thermal boundary layer, temperature of the fluid is less than 99% of free stream temperature.

If the fluid flowing on a surface has a different temperature than the surface, the thermal boundary layer developed is similar to the velocity boundary layer. Consider a fluid at a temperature T_∞ flows over a surface at a constant temperature T_s . The fluid particles in adjacent layer to the plate get the same temperature that of surface. The particles exchange heat energy with particles in adjoining fluid layers and so on. As a result, the temperature gradients are developed in the fluid layers and a temperature profile is developed in the fluid flow, which ranges from T_s at the surface to fluid temperature T_∞ sufficiently far from the surface in y direction.

Velocity boundary layer on a flat plate:

It is most essential to distinguish between laminar and turbulent boundary layers. Initially, the boundary layer development is laminar as shown in figure for the flow over a flat plate. Depending upon the flow field and fluid properties, at some critical distance from the leading edge small disturbances in the flow begin to get amplified, a transition process takes place and the flow becomes turbulent. In laminar boundary layer, the fluid motion is highly ordered whereas the motion in the turbulent boundary layer is highly irregular with the fluid moving to and from in all directions. Due to fluid mixing resulting from these macroscopic motions, the turbulent boundary layer is thicker and the velocity profile in turbulent boundary layer is flatter than that in laminar flow.



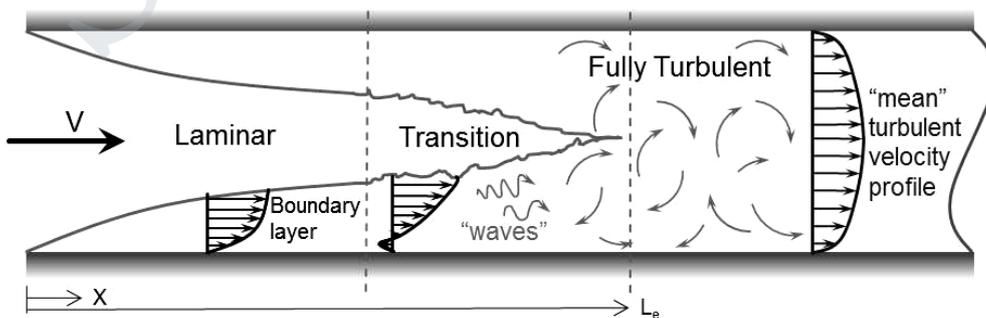
Velocity boundary layer on a tube:

Laminar Boundary Layer Flow

The laminar boundary is a very smooth flow, while the turbulent boundary layer contains swirls or "eddies." The laminar flow creates less skin friction drag than the turbulent flow, but is less stable. Boundary layer flow over a wing surface begins as a smooth laminar flow. As the flow continues back from the leading edge, the laminar boundary layer increases in thickness.

Turbulent Boundary Layer Flow

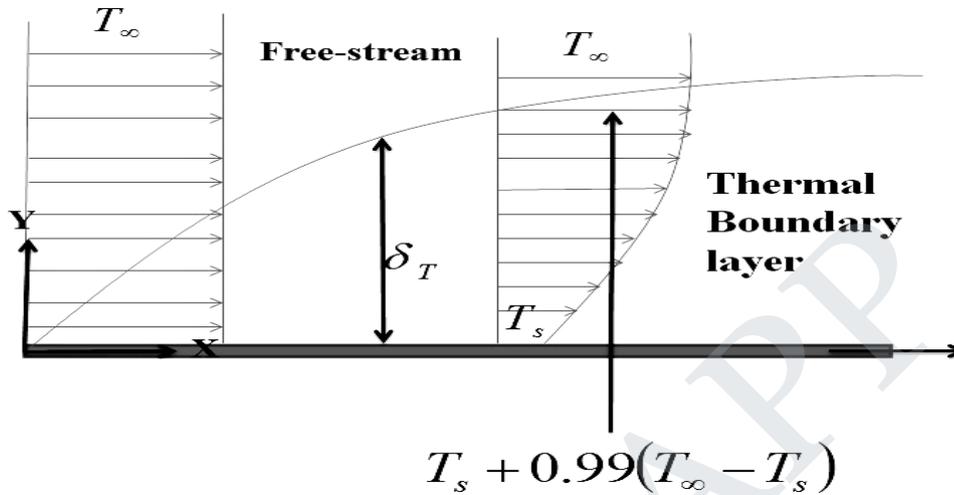
At some distance back from the leading edge, the smooth laminar flow breaks down and transitions to a turbulent flow. From a drag standpoint, it is advisable to have the transition from laminar to turbulent flow as far aft on the wing as possible, or have a large amount of the wing surface within the laminar portion of the boundary layer. The low energy laminar flow, however, tends to break down more suddenly than the turbulent layer.



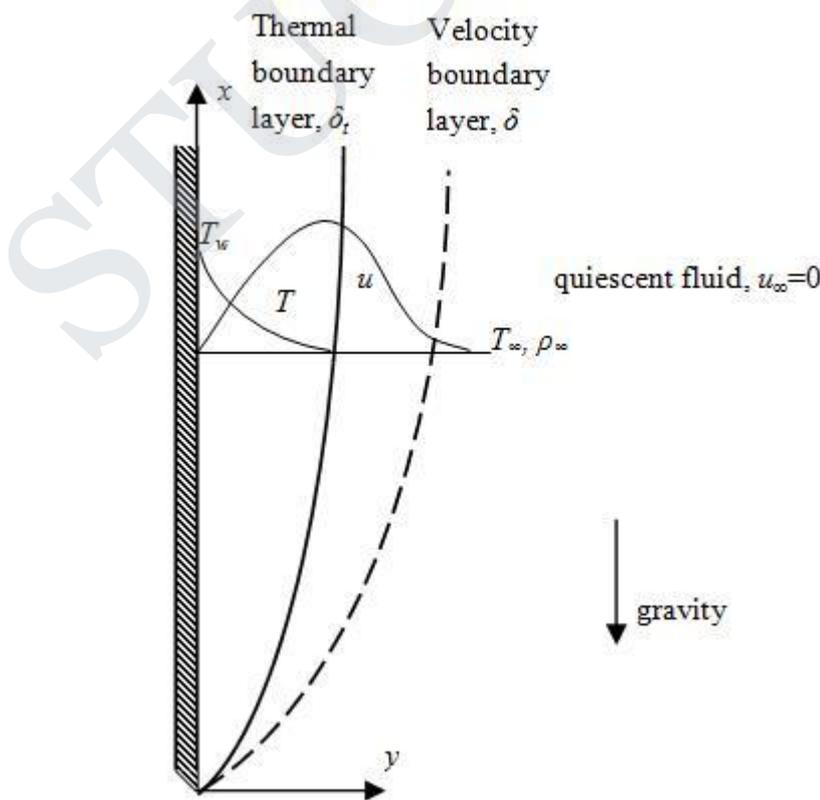
Thermal boundary Layer on a flat plate:

Consider a fluid of uniform temperature T_α approaching a flat plate of constant temperature T_s in the direction parallel to the plate. At the solid/liquid interface

the fluid temperature is T_s since the local fluid particles achieve thermal equilibrium at the interface. The fluid temperature T in the region near the plate is affected by the plate, varying from T_s at the surface to T_∞ in the main stream. This region is called the thermal boundary layer.



Velocity and Temperature boundary layer (Profile) for a vertical plate



6. In a long annulus (3.125 cm ID and 5 cm OD) the air is heated by maintaining the temperature of the outer surface of inner tube at 50°C. The air enters at 16°C and leaves at 32°C. Its flow rate is 30 m/s. Estimate the heat transfer coefficient between air and the inner tube.

Given : Inner diameter $D_i = 3.125 \text{ cm} = 0.03125 \text{ m}$

Outer diameter $D_o = 5 \text{ cm} = 0.05 \text{ m}$

Tube wall temperature $T_w = 50^\circ\text{C}$

Inner temperature of air $T_{m1} = 16^\circ\text{C}$

Outer temperature of air $t_{m2} = 32^\circ\text{C}$

Flow rate $U = 30 \text{ m/s}$

To find: Heat transfer coefficient (h)

Solution:

$$\begin{aligned} \text{Step 1. Mean temperature } T_m &= \frac{T_{m1} + T_{m2}}{2} \\ &= \frac{16 + 32}{2} \\ T_m &= 24^\circ\text{C} \end{aligned}$$

Properties of air at 24°C

[From HMT Data book page no. 34]

$$\rho = 1.185 \text{ Kg/m}^3$$

$$\nu = 15.53 \times 10^{-6} \text{ m}^2/\text{s}$$

$$\text{Pr} = 0.702$$

$$k = 0.02634 \text{ W/mK}$$

Step 2. Hydraulic or Equivalent diameter

$$\begin{aligned} D_h &= \frac{4A}{P} = \frac{4 \times \frac{\pi}{4} [D_o^2 - D_i^2]}{\pi [D_o + D_i]} \\ &= \frac{[D_o + D_i][D_o - D_i]}{[D_o + D_i]} \\ &= D_o - D_i \\ &= 0.05 - 0.03125 \end{aligned}$$

$$D_h = 0.01875 \text{ m}$$

$$\begin{aligned} \text{Step 3. Reynolds number, } Re &= \frac{UD_h}{\nu} \\ &= \frac{30 \times 0.01875}{15.53 \times 10^{-6}} \end{aligned}$$

$$Re = 36.2 \times 10^3$$

Since $Re > 2300$, flow is turbulent.

For turbulent flow, general equation is ($Re > 10000$).

$$Nu = 0.023 (Re)^{0.8} (Pr)^n$$

[From HMT Data book, Page No. 126]

This is heating process. So $n = 0.4$.

$$[T_{mo} > T_{mi}]$$

Step 4. $Nu = 0.023 \times (36.2 \times 10^3)^{0.8} (0.702)^{0.4}$

$$Nu = 88.59$$

Step 5. $Nu = \frac{hD_h}{k}$

$$88.59 = \frac{h \times 0.01875}{26.34 \times 10^{-3}}$$

$$h = 124.4 \text{ W/m}^2\text{K}$$

Heat transfer coefficient, $h = 124.4 \text{ W/m}^2\text{K}$

7. In a surface condenser, water flows through staggered tubes while the air is passed in cross flow over the tubes. The temperature and velocity of air are 30°C and 8 m/s respectively. The longitudinal and transverse pitches are 22 mm and 20 mm respectively. The tube outside diameter is 18 mm and tube surface temperature is 90°C. Calculate the heat transfer coefficient.

Given:

Fluid temperature, $T_\infty = 30^\circ\text{C}$

Velocity, $U = 8 \text{ m/s}$

Longitudinal pitch, $S_l = 22 \text{ mm} = 0.022 \text{ m}$

Transverse pitch, $S_t = 20 \text{ mm} = 0.020 \text{ m}$

Diameter, $D = 18 \text{ mm} = 0.018 \text{ m}$

Tube surface temperature, $T_w = 90^\circ\text{C}$

To find:

Step 1. Heat transfer coefficient.

Solution:

We know that,

$$\text{Film temperature, } T_f = \frac{T_w + T_\infty}{2}$$

$$= \frac{90 + 30}{2}$$

$$T_f = 60^\circ\text{C}$$

Properties of air at 60°C

[From HMT data book, Page No. 34]

$$\nu = 18.97 \times 10^{-6} \text{ m}^2/\text{s}$$

$$\text{Pr} = 0.696$$

$$K = 0.02896 \text{ W/mK}$$

$$\text{Step 2. Maximum velocity, } U_{\max} = U \times \frac{S_t}{S_t - D}$$

$$U_{\max} = 8 \times \frac{0.020}{0.020 - 0.018}$$

$$U_{\max} = 80 \text{ m/s}$$

$$\text{Step 3. Reynolds Number, } \text{Re} = \frac{U_{\max} \times D}{\nu}$$

$$= \frac{80 \times 0.018}{18.97 \times 10^{-6}}$$

$$\text{Re} = 7.5 \times 10^4$$

$$\frac{S_t}{D} = \frac{0.020}{0.018} = 1.11$$

$$\boxed{\frac{S_t}{D} = 1.11}$$

$$\frac{S_l}{D} = \frac{0.022}{0.018} = 1.22$$

$$\boxed{\frac{S_l}{D} = 1.22}$$

$\frac{S_t}{D} = 1.11$, $\frac{S_l}{D} = 1.22$, corresponding C, n values are 0.518 and 0.556 respectively.

[From HMT data book, page No. 123]

$$C = 0.518$$

$$n = 0.556$$

Step 4. Nusselt Number, $Nu = 1.13 (Pr)^{0.333} [C (Re)^n]$

[From HMT data book, Page No. 123]

$$Nu = 1.13 \times (0.696)^{0.333} \times [0.518 \times (7.5 \times 10^4)^{0.556}]$$

$$Nu = 266.3$$

Step 5. Nusselt Number, $Nu = \frac{hD}{k}$

$$266.3 = \frac{h \times 0.018}{28.96 \times 10^{-3}}$$

$$\text{Heat transfer coefficient, } h = 428.6 \text{ W/m}^2\text{K}$$

8. A thin 100 cm long and 10 cm wide horizontal plate is maintained at a uniform temperature of 150°C in a large tank full of water at 75°C. Estimate the rate of heat to be supplied to the plate to maintain constant plate temperature as heat is dissipated from either side of plate.

Given:

Length of horizontal plate $L = 100 \text{ cm} = 1 \text{ m}$

Wide $W = 10 \text{ cm} = 0.10 \text{ m}$

Plate temperature $T_w = 150^\circ\text{C}$

Fluid temperature $T_\infty = 75^\circ\text{C}$

To find: Heat loss (Q) from either side of plate:

Solution:

$$\begin{aligned} \text{Step 1. Film temperature, } T_f &= \frac{T_w + T_\infty}{2} \\ &= \frac{150 + 75}{2} \end{aligned}$$

$$T_f = 112.5^\circ\text{C}$$

Properties of water at 112.5°C:

[From HMT data book, Page No. 22]

$$\rho = 951 \text{ Kg/m}^3$$

$$\nu = 0.264 \times 10^{-6} \text{ m}^2/\text{s}$$

$$Pr = 1.55$$

$$k = 0.683 \text{ W/mK}$$

$$\beta_{(\text{for water})} = 0.8225 \times 10^{-3} \text{ K}^{-1}$$

[From HMT data book, Page No. 30]

Step 2. Grashof Number, $Gr = \frac{g \times \beta \times L_c^3 \times \Delta T}{\nu^2}$

For horizontal plate,

Characteristic length, $L_c = \frac{W}{2} = \frac{0.10}{2}$

$$L_c = 0.05 \text{ m}$$

$$Gr = \frac{9.81 \times 0.8225 \times 10^{-3} \times (0.05)^3 \times (150 - 75)}{(0.264 \times 10^{-6})^2}$$

$$Gr = 1.0853 \times 10^9$$

$$GrPr = 1.0853 \times 10^9 \times 1.55$$

$$GrPr = 1.682 \times 10^9$$

GrPr value is in between 8×10^6 and 10^{11}

i.e., $8 \times 10^6 < GrPr < 10^{11}$

For horizontal plate, upper surface heated:

Step 3. Nusselt Number, $Nu = 0.15 (GrPr)^{0.333}$

[From HMT data book, Page No. 136]

$$Nu = 0.15 [1.682 \times 10^9]^{0.333}$$

$$Nu = 177.13$$

Step 4. Nusselt Number, $Nu = \frac{h_u L_c}{k}$

$$177.13 = \frac{h_u \times 0.05}{0.683}$$

$$h_u = 2419.7 \text{ W/m}^2\text{K}$$

Upper surface heated, heat transfer coefficient

$$h_u = 2419.7 \text{ W/m}^2\text{K}$$

For horizontal plate, lower surface heated:

Step 5. Nusselt Number $Nu = 0.27 [GrPr]^{0.25}$

[From HMT data book, Page No. 136]

$$Nu = 0.27 [1.682 \times 10^9]^{0.25}$$

$$Nu = 54.68$$

Step 6. Nusselt Number, $Nu = \frac{h_l L_c}{k}$

$$54.68 = \frac{h_u \times 0.05}{0.683}$$

$$h_l = 746.94 \text{ W/m}^2\text{K}$$

Lower surface heated, heat transfer coefficient, $h_l = 746.94 \text{ W/m}^2\text{K}$

Step 7. Total heat transfer, $Q = (h_u + h_l) \times A \times \Delta T$

$$= (h_u + h_l) \times W \times L \times (T_w - T_\infty)$$

$$= (2419.7 + 746.94) \times 0.10 \times (150 - 75)$$

Heat transfer, $Q = 23749.8 \text{ W}$

9. Atmospheric air at 275 K and a free stream velocity of 20 m/s flows over a flat plate 1.5 m long that is maintained at a uniform temperature of 325 K. Calculate the average heat transfer coefficient over the region where the boundary layer is laminar, the average heat transfer coefficient over the entire length of the plate and the total heat transfer rate from the plate to the air over the length 1.5 m and width 1 m. Assume transition occurs at $Re_c = 2 \times 10^5$.

Given: Fluid temperature, $T_\infty = 275 \text{ K} = 2^\circ\text{C}$

Velocity, $U = 20 \text{ m/s}$

Length, $L = 1.5 \text{ m}$

Plate surface temperature, $T_w = 325 \text{ K} = 52^\circ\text{C}$

Width, $W = 1 \text{ m}$

Critical Reynolds number, $Re_c = 2 \times 10^5$

- To find:**
1. Average heat transfer coefficient, h_l [Boundary layer is laminar]
 2. Average heat transfer coefficient, h_t [Entire length of the plate]
 3. Total heat transfer rate, Q .

Solution:

Step 1. Film temperature, $T_f = \frac{T_w + T_\infty}{2}$

$$= \frac{52 + 2}{2}$$

$$T_f = 27^\circ\text{C}$$

Properties of air at $27^\circ\text{C} \approx 25^\circ\text{C}$

[From HMT data book, Page No. 34]

$$\rho = 1.185 \text{ Kg/m}^3$$

$$\nu = 15.53 \times 10^{-6} \text{ m}^2/\text{s}$$

$$\text{Pr} = 0.702$$

$$k = 0.02634 \text{ W/mK}$$

Case (i): Reynolds number, $\text{Re} = \frac{UL}{\nu}$

Transition occurs at $\text{Re}_c = 2 \times 10^5$

ie., Flow is laminar upto Reynolds number value is 2×10^5 , after that flow is turbulent.

$$2 \times 10^5 = \frac{20 \times L}{15.53 \times 10^{-6}}$$

$$L = 0.155 \text{ m}$$

For flat plate, laminar flow,

Step 2. Local Nusselt number, $\text{Nu}_x = 0.332 (\text{Re})^{0.5} (\text{Pr})^{0.333}$

[From HMT data book, Page No. 113]

$$\text{Nu}_x = 0.332 (2 \times 10^5)^{0.5} (0.702)^{0.333}$$

$$\text{Nu}_x = 131.97$$

Step 3. Local Nusselt Number, $\text{Nu}_x = \frac{h_x L}{k}$

$$131.97 = \frac{h_x \times 0.155}{0.02634}$$

$$h_x = 22.42 \text{ W/m}^2\text{K}$$

Local heat transfer coefficient, $h_x = 22.42 \text{ W/m}^2\text{K}$
--

Step 4. Average heat transfer coefficient, $h = 2 \times h_x$

$$= 2 \times 22.42$$

$$= 44.84 \text{ W/m}^2\text{K}$$

Case (ii):

Step 5. Reynolds number, Re_L [For entire length] = $\frac{UL}{\nu}$

$$= \frac{20 \times 1.5}{15.53 \times 10^{-6}}$$

$$= 1.93 \times 10^6 > 5 \times 10^5$$

Since $Re_L > 5 \times 10^5$, flow is turbulent.

For flat plate, laminar-turbulent combined flow,

Step 6. Average Nusselt number, $Nu = (Pr)^{0.333} [0.037 (Re_L)^{0.8} - 871]$

$$Nu = (0.702)^{0.333} [0.037 (1.93 \times 10^6)^{0.8} - 871]$$

$$Nu = 2737.18$$

Step 7. Nusselt number, $Nu = \frac{hL}{k}$

$$2737.18 = \frac{h \times 1.5}{0.02634}$$

$$h = 48.06 \text{ W/m}^2\text{K}$$

Average heat transfer coefficient for turbulent flow, $h_t = 48.06 \text{ W/m}^2\text{K}$

Step 8. Total heat transfer rate, $Q = h_t \times A \times \Delta T$

$$= h_t \times W \times L \times (T_w - T_\infty)$$

$$= 48.06 \times 1 \times 1.5 \times (52 - 2)$$

$$Q = 3604.5 \text{ W}$$

10. A steam pipe 10 cm outside diameter runs horizontally in a room at 23°C. Take the outside surface temperature of pipe as 165°C. Determine the heat loss per metre length of the pipe. [Dec 2004]

Given: Diameter of the pipe, $D = 10 \text{ cm} = 0.10 \text{ m}$

Ambient air temperature, $T_\infty = 23^\circ\text{C}$

Wall temperature, $T_w = 165^\circ\text{C}$

To find: Heat loss per metre length.

Solution:

Step 1. Film temperature, $T_f = \frac{T_w + T_\infty}{2}$

$$= \frac{165 + 23}{2}$$

$$T_f = 94^\circ\text{C}$$

Properties of air at $94^\circ\text{C} \approx 95^\circ\text{C}$

[From HMT data book, Page No. 34]

$$\rho = 0.959 \text{ Kg/m}^3$$

$$\nu = 22.615 \times 10^{-6} \text{ m}^2/\text{s}$$

$$Pr = 0.689$$

$$k = 0.03169 \text{ W/mK}$$

Step 2. Coefficient of thermal expansion, $\beta = \frac{1}{T_f \text{ in K}}$

$$= \frac{1}{94+273}$$

$$= 2.72 \times 10^{-3} \text{ K}^{-1}$$

$$\beta = 2.72 \times 10^{-3} \text{ K}^{-1}$$

Step 3. Grashof Number, $Gr = \frac{g \times \beta \times D^3 \times \Delta T}{\nu^2}$

[From HMT data book, Page No. 135]

$$Gr = \frac{9.81 \times .72 \times 10^{-3} \times (0.10)^3 \times (165-23)}{(22.615 \times 10^{-6})^2}$$

$$Gr = 7.40 \times 10^6$$

$$GrPr = 7.40 \times 10^6 \times 0.689$$

$$GrPr = 5.09 \times 10^6$$

For horizontal cylinder, Nusselt number, $Nu = C [GrPr]^m$

[From HMT data book, Page No. 138]

$GrPr = 5.09 \times 10^6$, corresponding $C = 0.48$, and $m = 0.25$

$$Nu = 0.48 [5.09 \times 10^6]^{0.25}$$

$$Nu = 22.79$$

Step 4. Nusselt number, $Nu = \frac{hD}{k}$

$$22.79 = \frac{h \times 0.10}{0.03169}$$

$$h = 7.22 \text{ W/m}^2\text{K}$$

Step 5. Heat loss, $Q = hA\Delta T$

$$= h \times \pi DL(T_w - T_\infty)$$

$$\frac{Q}{L} = h \times \pi \times D \times (T_w - T_\infty)$$

$$= 7.22 \times \pi \times 0.10 \times (165 - 23)$$

$$\frac{Q}{L} = 322.08 \text{ W/m}$$

$$\text{Heat loss per metre length, } \frac{Q}{L} = 322.08 \text{ W/m}$$

PART C – 15 Marks (Questions and Answers)

1. Consider the flow of oil at 20° C in a 30cm diameter pipeline at an average velocity of 2 m/s. a 200m long section of the pipeline passes through icy waters of a lake at 0° C. Measurements indicate that the surface temperature of the pipe is very nearly 0° C. Disregarding the thermal resistance of the pipe material determine (a) the temperature of the oil when the pipe leaves the lake, (b) the rate of heat transfer from the oil, and (c) the pumping power required to overcome the pressure losses and to maintain the flow of the oil in the pipe.

Solution

Oil flows in a pipeline that passes through icy waters of a lake at 0° C. The exit temperature of the oil, the rate of heat loss, and the pumping power needed to overcome pressure losses are to be determined.

Assumptions

1. Steady operating conditions exist.
2. The surface temperature of the pipe is very nearly 0° C.
3. The thermal resistance of the pipe is negligible.
4. The inner surfaces of the pipeline are smooth.
5. The flow is hydrodynamically developed when the pipeline reaches the lake.

Properties

We do not know the exit temperature of the oil, and thus we cannot determine the bulk mean temperature, which is the temperature at which the properties of oil are to be evaluated. The mean temperature of the oil at the inlet is 20°C, and we expect this temperature to drop somewhat as a result of heat loss to the icy waters of the lake. We evaluate the properties of the oil at the inlet temperature, but we will repeat the calculations, if necessary, using properties at the evaluated bulk mean temperature. At 20° C from HMT data book

$$\rho = 888 \text{ kg/m}^3 \quad U = 901 \times 10^{-6} \text{ m}^2/\text{s}$$

$$k = 0.145 \text{ W/m } ^\circ\text{C} \quad C_p = 1880 \text{ J/kg } ^\circ\text{C}$$

$$\text{Pr} = 10,400$$

$$R_e = \frac{UD}{\nu} = \frac{2 \times 0.3}{901 \times 10^{-6}} = 666$$

which is less than the critical Reynolds number of 2300. Therefore, the flow is

laminar, and we assume thermally developing flow and determine the nusselt number from

$$\begin{aligned} Nu &= \frac{hD}{k} = 3.66 + \frac{0.065 (D/L) Re Pr}{1+0.04[(D/L) Re Pr]^{2/3}} \\ &= 3.66 + \frac{0.065 (0.3/200) \times 666 \times 10400}{1+0.04[(0.3/200) 666 \times 10400]^{2/3}} = 37.3 \end{aligned}$$

This nusselt number is considerably higher than the fully developed value of 3.66 then

$$h = \frac{k}{D} Nu = \frac{0.0145}{0.3} (37.3) = 18.0 \frac{W}{m^2} \text{ } ^\circ C$$

also we determine the exit temperature of air from

$$T_e = T_s - (T_s - T_i) \exp(-h A_s / m C_p)$$

here

$$A_s = PL = \pi DL = \pi (0.3 \text{ m})(200 \text{ m}) = 188.5 \text{ m}^2$$

$$m = \rho V = (1.009 \text{ kg/m}^3)(0.15 \text{ m}^3/\text{s}) = 0.151 \text{ kg/s}$$

Substitute A_s and m in T_e

$$T_e = 60 - (60 - 80) \exp(-13.5 \times 6.4 / 0.151 \times 1008) = 71.3 \text{ } ^\circ C$$

Then the logarithmic mean temperature difference and the rate of heat loss from the air become

$$\Delta T_{\ln} = \frac{T_i - T_e}{\ln \frac{T_s - T_e}{T_s - T_i}} = -15.2 \text{ } ^\circ C$$

$$Q = h A_s \Delta T_{\ln} = (13.5 \text{ W/m}^2 \text{ } ^\circ C)(6.4 \text{ m}^2)(-15.2 \text{ } ^\circ C) = -1313 \text{ W}$$

Therefore, air will lose heat at a rate of 1313 W as it flows through the duct in the attic.

- 2. In condenser water flows through two hundred thin walled circular tubes having inner diameter 20mm and length 6 m. the mass flow rate of water is 160 kg/s. the water enters at 30° C and leaves at 50 ° C. Calculate the average heat transfer coefficient.**

Given :

Inner diameter $D = 20\text{mm}$

Length $L = 6 \text{ m}$

Mass flow rate $m = 160 \text{ kg/s}$

Inlet water temperature $T_{mi} = 30^\circ \text{C}$

Outlet water temperature, $T_{mo} = 50^\circ \text{C}$

To find: Heat transfer coefficient (h)

Solution:

$$\text{Bulk mean temperature } T_m = \frac{T_{mi} + T_{mo}}{2} = \frac{30 + 50}{2} = 40^\circ \text{C}$$

Properties of water at 40°C [from HMT data boo page no 21]

$$\rho = 995 \text{ kg/m}^3$$

$$\nu = 0.657 \times 10^{-6} \text{ m}^2/\text{s}$$

$$\text{Pr} = 4.340$$

$$k = 0.628 \text{ W/mK}$$

$$C_p = 4178 \text{ J/kg K}$$

$$\text{Reynolds Number } R_e = UD/\nu$$

$$m = \rho AU$$

$$\text{Velocity } U = m/\rho A$$

$$= \frac{\left(\frac{160}{200}\right)}{995 \times \frac{\pi}{4} \times 0.020^2} = 2.55 \text{ m/s} \quad (\text{no of tubes} = 200)$$

$$R_e = UD/\nu$$

$$= \frac{2.55 \times 0.020}{0.657 \times 10^{-6}} = 77625.57$$

Since $R_e > 2300$, flow is turbulent

For turbulent flow, general equation is ($R_e > 10000$)

$$\text{Nu} = 0.023 \times R_e^{0.8} \text{Pr}^n \quad [\text{from HMT data boo page no 125}]$$

This is heating process so $n = 0.4$ ($T_{mo} > T_{mi}$)

$$\text{Nu} = 0.023 \times 77625.57^{0.8} \times 4.340^{0.4}$$

$$\text{Nu} = 337.8$$

$$\text{Nu} = \frac{hD}{k}$$

$$337.8 = \frac{h \times 0.020}{0.628}$$

$$\text{Heat transfer coefficient } h = 10606.9 \text{ w/m}^2\text{K}$$

UNIT: III PHASE CHAGNE HEAT TRANSFER AND HEAT EXCHANGERS

PART A - 2 Marks (Questions and Answers)

1. What is burnout point in boiling neat transfer? Why is it called so? (May /June 2013)

In the Nucleate boiling region, a point at which heat flow is maximum is known as burnout point. Once we cross this point, large temperature difference is required to get the same heat flux and most material may burn at this temperature. Most of the boiling heat transfer heaters are operated below the burnout heat flux to avoid that disastrous effect.

2. Define NTU and LMTD of a heat exchanger. (May/June 2013 & May/June 2016)

LMTD (Logarithmic Mean Temperature Difference)

The temperature difference between the hot and cold fluids in the heat exchanger varies from point in addition various modes of heat transfer are involved. Therefore based on concept of appropriate mean temperature difference, also called logarithmic mean temperature difference, the total heat transfer rate in the heat exchanger is expressed as

$$Q = U A (\Delta T)_m$$

Where U - Overall heat transfer coefficient W/m²K

A - Area m²

(ΔT)_m - Logarithmic mean temperature difference.

NTU (No. of Transfer Units)

It is used to calculate the rate of heat transfer in heat exchangers, when there is insufficient information to calculate the Log-Mean Temperature Difference (LMTD). In heat exchanger analysis, if the fluid inlet and outlet temperatures are specified or can be determined, the LMTD method can be used; but when these temperatures are not available The NTU or The Effectiveness method is used.

3. What are the different regimes involved in pool boiling? (May/June 2014)

The different boiling regimes observed in pool boiling are

1. Interface evaporation
2. Nucleate boiling
3. Film boiling.

4. Write down the relation for overall heat transfer coefficient in heat exchanger with fouling factor. (May/June 2014)

Overall heat transfer coefficient in heat exchanger

$$\frac{1}{U_o} = \frac{1}{h_o} + R_{fo} + \frac{r_o}{k} \ln \frac{r_o}{r_i} + \frac{r_o}{r_i} R_{fi} + \frac{r_o}{r_i} \frac{1}{h_i}$$

Where R_{fi} and R_{fo} are the fouling factors at inner and outer surfaces.

[HMT Data Book, P.No.157]

5. How heat exchangers are classified? (May/June 2015)

The heat exchangers are classified as follows

1. Direct contact heat exchangers
2. Indirect contact heat exchangers
3. Surface heat exchangers
4. Parallel flow heat exchangers
5. Counter flow heat exchangers
6. Cross flow heat exchangers
7. Shell and tube heat exchangers
8. Compact heat exchangers.

6. What are the limitations of LMTD method? Discuss the advantage of NTU over the LMTD method. (May/June 2015 & Nov/Dec 2012 & Nov/Dec 2013)

The LMTD method cannot be used for the determination of heat transfer rate and outlet temperature of the hot and cold fluids for prescribed fluid mass flow rates and inlet temperatures when the type and size of heat exchanger are specified.

Effectiveness NTU is superior for the above case because LMTD requires tedious iterations for the same.

7. Differentiate between pool and forced convection boiling. (Nov/Dec 2012 & Nov/Dec 2013 & Nov/Dec 2015) (NOV/DEC 2016)

Boiling is called pool boiling in the absence of bulk fluid flow, and flow boiling (or forced convection boiling) in the presence of it.

In pool boiling, the fluid is stationary, and any motion of the fluid is due to natural convection currents and the motion of the bubbles due to the influence of buoyancy. Example: Boiling of water in a pan on top of a stove.

8. What is pool boiling? Give an example for it. (Nov/Dec 2014)

If heat is added to a liquid from a submerged solid surface, the boiling process referred to as pool boiling. In this case the liquid above the hot surface is essentially stagnant and its motion near the surface is due to free convection and mixing induced by bubble growth and detachment.

Example: Boiling of water in a pan on top of a stove.

9. What do you understand by fouling and effectiveness? (Nov/Dec 2014 & Nov/Dec 2015)

The surfaces of heat exchangers do not remain clean after it has been in use for some time. The surfaces become fouled with scaling or deposits. The effect of these deposits affecting the value of overall heat transfer coefficient. This effect is taken care of by introducing an additional thermal resistance called the fouling resistance or fouling factor.

10. Define effectiveness. (May/June 2016)

The heat exchanger effectiveness is defined as the ratio of actual heat transfer to the maximum possible heat transfer.

$$\text{Effectiveness } \varepsilon = \frac{\text{Actual heat transfer}}{\text{Maximum possible heat transfer}}$$

11. What is meant by sub-cooled and saturated boiling? (Nov/Dec 2015)

The sub-cooled boiling or saturated boiling, depending on the bulk liquid temperature.

Sub-cooled boiling:

There is sharp increase in temperature near to the surface but through most of the liquid, temperature remains close to saturation temperature. ($T_{\alpha} < T_{sat}$)

Saturated boiling:

When the temperature of the liquid equals to the saturation temperature. ($T_{\alpha} = T_{sat}$)

12. What is a compact heat exchanger? Give applications. (May/June 2016)

Special purpose heat exchangers called compact heat exchangers. They are generally employed when convective heat transfer coefficient associated with one of the fluids is much smaller than that associated with the other fluid.

In variety of applications including,

- Compressed Gas / Water coolers
- Condensers and evaporators for chemical and technical processes of all kinds.
- Oil and water coolers for power machines
- Refrigeration and air-conditioning units

13. What are the assumptions made in Nusselt theory of condensation? (May/June 2016)

1. The plate is maintained at a uniform temperature which is less than the saturation temperature of vapour. ($T_w < T_{sat}$)
2. Fluid properties are constant.
3. The shear stress at the liquid vapour interface is negligible.
4. The heat transfer across the condensate layer is by pure conduction and the temperature distribution is linear.

14. How fouling affect the rate of heat transfer? (May/June 2016)

"Fouling" is any kind of deposit of extraneous material that appears upon the heat transfer surface during the life time of the heat exchanger.

This fouling will cause an additional resistance to heat transfer is introduced and the operational capability of the heat exchanger is correspondingly reduced. In many cases, the deposit is heavy enough to significantly interfere with fluid flow and increase the pressure drop required to maintain the flow rate through the exchanger.

PART B - 13 Marks (Questions and Answers)

1. Discuss briefly the pool boiling regimes of water at atmospheric pressure (May/June 2013, May/June 2014, Nov/Dec 2013)

Boiling is classified as pool boiling or flow boiling, depending on the presence of bulk fluid motion. Boiling is called pool boiling in the absence of bulk fluid flow and flow boiling in the presence of bulk fluid motion.

Boiling takes different forms, depending on the value of the excess temperature ΔT_{excess} . Four different boiling regimes are observed: natural convection boiling, nucleate boiling, transition boiling, and film boiling. These regimes are illustrated on the boiling curve in fig, which is a plot of boiling heat flux versus the excess temperature.

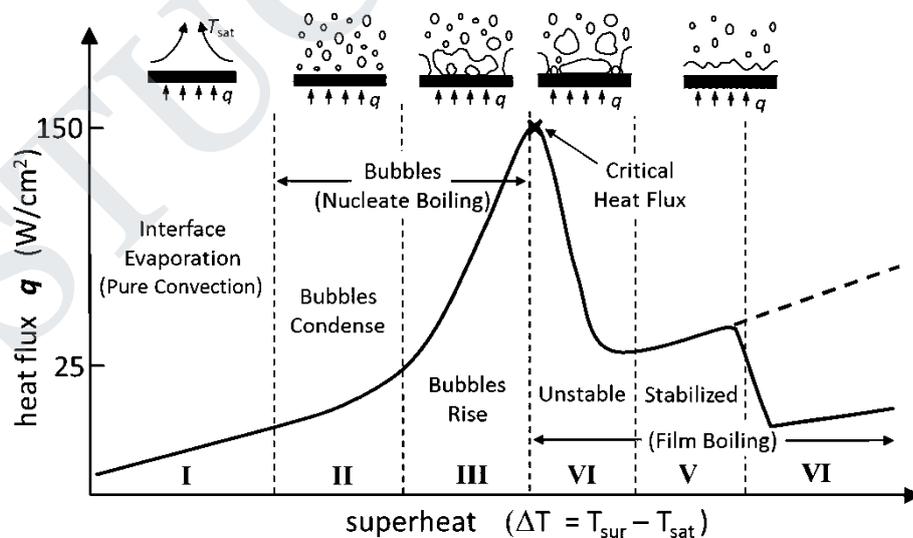


Fig: Typical boiling curve for water at 1 atmospheric pressure

NATURAL CONVECTION BOILING (to point A on the Boiling curve)

We know from thermodynamics that a pure substance at a specified pressure starts boiling when it reaches the saturation temperature at that

pressure. But in practice we do not see any bubbles forming on the heating surface until the liquid is heated a few degrees above the saturation temperature (about 2 to 6° C for water). Therefore, the liquid is slightly superheated in this case and evaporates when it rises to the free surface. The fluid motion in this mode of boiling is governed by natural convection currents, and heat transfer from the heating surface to the fluid is by natural convection. For the conditions of fig, natural convection boiling ends at excess temperature of about 5° C.

NUCLEATE BOILING (between points A and C)

The first bubbles start forming at point A of the boiling curve at various preferential sites on the heating surface. Point A is referred to as the onset of nucleate boiling (ONB). The bubbles form at an increasing rate at an increasing number of nucleation sites as we move along the boiling curve toward point C. From fig nucleate boiling exists in the range from about 5° C to about 30°C.

The nucleate boiling regime can be separated into two distinct regions. In regions A-B ($5^{\circ}\text{C} \leq \Delta T_{\text{excess}} \leq 10^{\circ}\text{C}$), isolated bubbles are formed at various preferential nucleation sites on the heated surface. But these bubbles are dissipated in the liquid shortly after they separate from the surface. The space vacated by the rising bubbles is filled by the liquid in the vicinity of the heater surface, and the process is repeated. The stirring and agitation caused by the entrainment of the liquid to the heater surface is primarily responsible for the increased heat transfer coefficient and heat flux in this region of nucleate boiling.

In region B-C ($10^{\circ}\text{C} \leq \Delta T_{\text{excess}} \leq 30^{\circ}\text{C}$), the heater temperature is further increased, and bubbles form at such great rates at such a large number of nucleation sites that they form numerous continuous columns of vapour in the liquid. These bubbles move all the way up to the free surface, where they break up and release their vapor content. The large heat fluxes obtainable in this region.

At large values of ΔT_{excess} , the rate of evaporation at the heater surface reaches such high values that a large fraction of the heater surface

is covered by bubbles, making it difficult for the liquid to reach the heater surface and wet it. Consequently, the heat flux increases at a lower rate with increasing ΔT_{excess} , and reaches a maximum at point C. the heat flux at this point is called the critical heat flux.

TRANSITION BOILING (between points C and D)

As the heater temperature and thus the ΔT_{excess} , is increased past point C, the heat flux decreases, as shown in fig. this is because a large fraction of the heater surface is covered by a vapour film, which acts as an insulation due to the low thermal conductivity of the vapour relative to that of the liquid. In the transition boiling regime, both nucleate and film boiling partially occur. Nucleate boiling at point C is completely replaced by film boiling at point D. for water, transition boiling occurs over the excess temperature range from about 30°C to about 120°C.

FILM BOILING (beyond point D)

In this region the heater surface is completely covered by a continuous stable vapour film. Point D, where the heat flux reaches a minimum, is called the Leidenforst point. The liquid droplets on a very hot surface jump around and slowly boil away. The presence of a vapour film between the heater surface and the liquid is responsible for the low heat transfer rates in the film boiling region. The heat transfer rate increases with increasing excess temperature as a result of heat transfer from the heated surface to the liquid through the vapour film by radiation, which becomes significant at high temperatures.

- 2. Water is to be boiled at atmospheric pressure in a polished copper pan by means of an electric heater. The diameter of the pan is 0.38 m and is kept at 115° C. calculate the following**
- 1. Power required boiling the water**
 - 2. Rate of evaporation**
 - 3. Critical heat flux.** (Nov/Dec 2012, Nov/Dec 2015)

Given:

Diameter, $d = 0.38 \text{ m}$;

Surface temperature, $T_w = 115^\circ \text{ C}$.

To find:

1. Power required, (p)
2. Rate of evaporation, (ṁ)
3. Critical heat flux, (Q/A)

Solution:

Step 1:

Need to find the nucleate pool boiling or film pool boiling process.

$\Delta T = \text{Excess Temperature} = T_w - T_{\text{sat}} = \text{Answer}$, which is less than 50°C then it is Nucleate pool boiling or greater than 50°C then it is film pool boiling.

$$\Delta T = T_w - T_{\text{sat}}$$

We know that saturation temperature of water is 100°C . i.e. $T_{\text{sat}} = 100^\circ \text{C}$

$\Delta T = 115 - 100 = 15^\circ \text{C}$ so this is nucleate pool boiling process.

Step 2:

Need to find the properties of water at 100°C .

(From HMT data book page No. 21)

Density, $\rho_l = 961 \text{ kg/m}^3$

Kinematic viscosity, $\nu = 0.293 \times 10^{-6} \text{ m}^2/\text{s}$

Prandtl Number, $P_r = 1.740$

Specific heat, $C_{pl} = 4216 \text{ J/kg K}$

Dynamic viscosity, $\mu_l = \rho_l \times \nu = 961 \times 0.293 \times 10^{-6} = 281.57 \times 10^{-6} \text{ Ns/m}^2$

Enthalpy of evaporation, $h_{fg} = 2256.9 \text{ KJ/kg}$ (from steam table)

Specific volume of vapour, $\nu_g = 1.673 \text{ m}^3/\text{kg}$

Density of vapour, $\rho_v = (1/\nu_g) = 0.597 \text{ kg/m}^3$

Step 3:

Need to find the heat flux, power

$$\text{Heat flux, } \frac{Q}{A} = \mu_l \times h_{fg} \left[\frac{g \times (\rho_l - \rho_v)}{\sigma} \right]^{0.5} \times \left[\frac{C_{pl} \times \Delta T}{C_{sf} \times h_{fg} P_r^n} \right]^3 \dots\dots 1 \text{ (from HMT data}$$

book page no. 142)

Where $\sigma =$ surface tension for liquid vapour interface at 100°C .

$\sigma = 0.0588 \text{ N/m}$

(from HMT data book page no. 144)

For water - copper $\rightarrow C_{sf}$ = surface fluid constant = 0.013 and $n=1$ for water (from HMT data book page no.143)

Substitute the $\mu_l, h_{fg}, \rho_l, \rho_v, C_{pl}, \Delta T, C_{sf}, n$ and Pr values in equation 1

$$\frac{Q}{A} = 4.83 \times 10^5 \text{ W/m}^2$$

Heat transfer $Q = 4.83 \times 10^5 \times A$

$$\text{Area } A = \left(\frac{\pi}{4}\right)d^2 = 0.113 \text{ m}^2$$

Power = 54.7 kW

Step 4:

Need to find Rate of evaporation, (\dot{m})

$$\text{Heat transferred } Q = \dot{m} \times h_{fg}$$

Substitute Q and h_{fg}

$\dot{m} = 0.024$ kg/s

Step 5:

Need to find the critical flux

For nucleate pool boiling, critical heat flux,

$$\frac{Q}{A} = 0.18 \times h_{fg} \times \rho_v \left[\frac{\sigma \times g \times (\rho_l - \rho_v)}{\rho_v^2} \right]^{0.25}$$

(from HMT data book page no. 142)

Critical heat flux, $q = \frac{Q}{A} = 1.52 \times 10^6 \text{ W/m}^2$
--

3. A wire of 1 mm diameter and 150mm length is submerged horizontally in water at 7 bar. The wire carries a current of 131.5 ampere with an applied voltage of 2.15 Volt. If the surface of the wire is maintained at 180° C, calculate the heat flux and the boiling heat transfer coefficient.(May/June 2014 Reg 2008)

Given:

Diameter, $D = 1 \text{ mm} = 1 \times 10^{-3} \text{ m}$;

Length, $L = 150 \text{ mm} = 150 \times 10^{-3} \text{ m}$;

Pressure, $P = 7 \text{ bar}$

Voltage, $V = 2.15 \text{ V}$

Current, $I = 131.5 \text{ amps}$

$T_w = 180^\circ \text{ C}$

To find:

1. Heat flux, $\frac{Q}{A}$
2. Heat transfer coefficient, h

Solution:

Step 1:

Need to find heat flux

$$Q = V \times I = 2.15 \times 131.5 = 282.72 \text{ W}$$

$$A = \pi DL = \pi \times 1 \times 10^{-3} \times 150 \times 10^{-3} = 471.23 \times 10^{-6} \text{ m}^2$$

$$\text{Heat flux} = \frac{Q}{A} = 282.72 / 471.23 \times 10^{-6} = 599.950 \times 10^3 \text{ W/m}^2$$

$$\frac{Q}{A} = 599.950 \times 10^3 \text{ W/m}^2$$

Step 2:

Need to find the heat transfer coefficient h

At pressure $P = 7 \text{ bar}$: $\Delta T = 180 - 100 = 80^\circ \text{ C}$

Heat transfer coefficient, $h = 5.56 (\Delta T)^3$

(From HMT data book page no: 143)

$$h = 2846720 \text{ W/m}^2 \text{ K}$$

Heat transfer coefficient other than atmospheric pressure

$$h_p = h P^{0.4} = 2846720 \times 7^{0.4} = 6.19 \times 10^6 \text{ W/m}^2 \text{ K}$$

$$h_p = 6.19 \times 10^6 \text{ W/m}^2 \text{ K}$$

4. A vertical cooling fin approximating a flat plate 40 cm in height is exposed to saturated steam at atmospheric pressure. The fin is maintained at a temperature of 90° C . estimate the thickness of the film at the bottom of the fin, overall heat transfer coefficient and heat transfer rate after incorporating McAdam's correction, the rate of condensation of steam. (Nov/Dec 2015 Reg 2008)

Given:

Height (or) Length, $L = 40 \text{ cm} = 0.4 \text{ m}$

Surface temperature, $T_w = 90^\circ \text{ C}$

To find:

1. The film thickness δ_x
2. Overall heat transfer coefficient h (McAdam's correction)
3. Heat transfer rate Q
4. Rate of condensation of steam \dot{m}

Solution:

Step 1:

We know that, saturation temperature of water is 100° C , i.e. $T_{\text{sat}} = 100^\circ \text{ C}$

$h_{fg} = 2256.9 \text{ kJ/kg}$ (from steam table)

We know that

Film temperature, $T_f = \frac{T_w + T_{\text{sat}}}{2} = 95^\circ \text{ C}$

Properties of saturated water at 95° C (from HMT data book page no: 21)

Density, $\rho_l = 967.5 \text{ kg/m}^3$

Kinematic viscosity, $\nu = 0.328 \times 10^{-6} \text{ m}^2/\text{s}$

Specific heat, $C_{pl} = 4205.5 \text{ J/kg K}$

Thermal conductivity $K = 0.674 \text{ W/mk}$

Dynamic viscosity, $\mu_l = \rho_l \times \nu = 967.5 \times 0.328 \times 10^{-6} = 3.173 \times 10^{-4} \text{ Ns/m}^2$

Step 2:

We need to find the film thickness

$$\delta_x = \left[\frac{4 \mu K x (T_{\text{sat}} - T_w)}{g h_{fg} \rho_l^2} \right]^{0.25} \quad \text{(from HMT data book page no: 148)}$$

substitute all appropriate property value in above formula

$\delta_x = 1.13 \times 10^{-4} \text{ m}$

Step 3:

We need to find the heat transfer coefficient h

For vertical surface laminar flow (assume) or find by Re-Reynolds number

$Re = \frac{4 \dot{m}}{P \mu}$ here $P = \text{perimeter}$; $Re > 1800$ then that flow is turbulent flow,

$Re < 1800$ then that flow is laminar flow,

$$h = 0.943 \left[\frac{k^3 \times \rho^2 \times g h_{fg}}{\mu \times L \times (T_{sat} - T_w)} \right]^{0.25} \quad (\text{from HMT data book page no: 148})$$

The factor 0.943 may be replaced by 1.13 for more accurate result as suggested by Mc Adams

$$h = 1.13 \left[\frac{k^3 \times \rho^2 \times g h_{fg}}{\mu \times L \times (T_{sat} - T_w)} \right]^{0.25}$$

Substitute all the properties in above formula

$$h = 1495.3 \text{ W/m}^2 \text{ K}$$

Step 4:

We need to find the heat transfer rate Q

$$Q = h A (T_{sat} - T_w) = h L W (T_{sat} - T_w)$$

$$Q = 1495.3 \times 0.4 \times 1 \times 10 = 5981.26 \text{ W}$$

$$Q = 5981.26 \text{ W}$$

Step 5:

We need to find the rate of condensation of steam \dot{m}

$$Q = \dot{m} h_{fg}$$

$$\dot{m} = Q/h_{fg}$$

$$\dot{m} = 0.00265 \text{ kg/s}$$

5. A condenser is to be designed to condense 600 kg/h of dry saturated steam at a pressure of 0.12 bar. A square array of 400 tubes, each of 8 mm diameters is to be used. The tube surface is maintained at 30° C. Calculate the heat transfer coefficient and the length of each tube. (April/May 2015) (NOV/DEC 2013)

Given:

$$\dot{m} = 600 \text{ kg/h} = 0.166 \text{ kg/s}$$

$$\text{Pressure } P = 0.12 \text{ bar}$$

$$\text{No. of tubes} = 400$$

$$\text{Diameter, } D = 8 \text{ mm} = 8 \times 10^{-3} \text{ m}$$

$$\text{Surface temperature, } T_w = 30^\circ \text{ C.}$$

To find:

1. Heat transfer coefficient h
2. Length

Solution:

Step 1:

We need find the properties of steam at 0.12 bar (from steam table)

$$T_{sat} = 49.45^\circ \text{C}.$$

$$h_{fg} = 2384.3 \times 10^3 \text{ J/kg}$$

$$\text{Film temperature, } T_f = \frac{T_w + T_{sat}}{2} = 39.72^\circ \text{C} = 40^\circ \text{C}$$

Properties of saturated water at 40°C (from HMT data book page no: 21)

$$\text{Density, } \rho_l = 995 \text{ kg/m}^3$$

$$\text{Kinematic viscosity, } \nu = 0.657 \times 10^{-6} \text{ m}^2/\text{s}$$

$$\text{Thermal conductivity } K = 0.628 \text{ W/mk}$$

$$\text{Dynamic viscosity, } \mu_l = \rho_l \times \nu = 995 \times 0.657 \times 10^{-6} = 653.7 \times 10^{-6} \text{ Ns/m}^2$$

With 400 tubes, a 20 × 20 tube of square array could be formed

$$N = \sqrt{400} = 20$$

Step 2:

We need to find the heat transfer coefficient h

$$h = 0.728 \left[\frac{k^3 \times \rho^2 \times g \times h_{fg}}{\mu \times N \times D \times (T_{sat} - T_w)} \right]^{0.25} \quad (\text{from HMT data book page no: 148})$$

$$h = 5304.75 \text{ W/m}^2 \text{ K}$$

Step 3:

$$Q = h A (T_{sat} - T_w) = h D L (T_{sat} - T_w) = 1.05 \times 10^6 L \text{ -----1}$$

We know that

$$Q = \dot{m} h_{fg} = 0.3957 \times 10^6 \text{ W} \text{ -----2}$$

Equating (1) and (2) We get, L = 0.37

6. In a double pipe counter flow heat exchanger, 10000 kg/hr of an oil having a specific heat of 2095 J/kg-k is cooled from 80°C to 50°C by 800kg/hr of water entering at 25°C. Determine the heat exchanger area

for an overall heat transfer co-efficient of $300 \text{ W/m}^2\text{k}$. Take C_p for water as 4180 J/kg-k .

Given:

Hot fluid – oil (T_1-T_2) Cold fluid - water (t_1-t_2)

The mass flow rate of oil (Hot fluid), $m_h = 10000 \text{ kg/hr}$

$$= \frac{10000 \text{ kg}}{3600 \text{ s}}$$

$$m_h = 2.277 \text{ kg/s}$$

Specific heat of oil, $C_{ph} = 2095 \text{ J/kg-k}$

Entry temperature of oil, $T_1 = 80^\circ\text{C}$

Exit temperature of oil, $T_2 = 50^\circ\text{C}$

Mass flow rate of water (Cold fluid), $m_c = 8000 \text{ kg/hr}$

$$= \frac{8000 \text{ kg}}{3600 \text{ s}}$$

$$m_c = 2.22 \text{ kg/s}$$

Entry temperature of water, $t_1 = 25^\circ\text{C}$

Overall heat transfer co-efficient, $U = 300 \text{ W/m}^2\text{k}$

Specific heat of water, $C_{pc} = 4180 \text{ J/kg-k}$

To find:

Heat exchanger area, A

Solution:

Heat lost by oil (Hot fluid) = Heat gained by water (Cold fluid)

$$Q_h = Q_c$$

$$m_h C_{ph} (T_1 - T_2) = m_c C_{pc} (t_1 - t_2)$$

$$2.277 \times 2095 (80 - 50) = 2.22 \times 4180 \times (t_2 - 25)$$

$$174.53 \times 10^3 = 9.27 \times 10^3 t_2 - 231.99 \times 10^3$$

$$t_2 = 43.85^\circ\text{C}$$

$$\text{Exit temperature of water, } t_2 = 43.85^\circ\text{C}$$

$$\text{Heat transfer, } Q = m_h C_{ph} (T_1 - T_2) \text{ or } m_c C_{pc} (t_1 - t_2)$$

$$Q = 2.22 \times 4180 \times (43.85 - 25)$$

$$Q = 174.92 \times 10^3 \text{ W}$$

We know that,

$$\text{Heat transfer, } Q = UA (\Delta T)_m \quad \dots(1)$$

Where,

$(\Delta T)_m$ – Logarithmic Mean Temperature Difference. (LMTD)

$$\text{For counter flow, } (\Delta T)_m = \frac{[(T_1 - t_2) - (T_2 - t_1)]}{\ln \left[\frac{T_1 - t_2}{T_2 - t_1} \right]}$$

$$= \frac{[(80 - 43.85) - (50 - 25)]}{\ln \left[\frac{80 - 43.85}{50 - 25} \right]}$$

$$(\Delta T)_m = 30.23^\circ\text{C}$$

Substitute $(\Delta T)_m$, U and Q value in eqn (1)

$$Q = UA (\Delta T)_m$$

$$174.92 \times 10^3 = 300 \times A \times 30.23$$

$$\text{Heat exchanger area } A = 19.287 \text{ m}^2$$

7. In a cross flow heat exchangers, both fluids are mixed, hot fluid with a specific heat of 2300 J/kg K, enters at 380°C and leaves at 300°C. Cold fluids enter at 25°C and leaves 210°C. Calculate the required surface area of heat exchanger. Take overall heat transfer co-efficient is 750 W/m²K. Mass flow rate of hot fluid is 1 kg/s.

Given:

Specific heat of hot fluid $C_{ph} = 2300 \text{ J/Kg K}$

Entry temperature of hot fluid $T_1 = 380^\circ \text{C}$

Exit temperature of hot fluid $T_2 = 300^\circ \text{C}$

Entry temperature of Cold fluid $t_1 = 25^\circ \text{C}$

Exit temperature of Cold fluid $t_2 = 210^\circ \text{C}$

Overall heat transfer co-efficient, $U = 750 \text{ W/m}^2\text{K}$

The mass flow rate of hot fluid, $m_h = 1 \text{ kg/s}$

To find:

Heat exchanger area (A)

Solution:

This is Cross flow, both fluids unmixed type heat exchanger.

For cross flow heat exchanger,

$$Q = F UA (\Delta T)_m \text{ (counter flow) } \dots\dots(1)$$

[From HMT Data book page No.152]

Where,

$(\Delta T)_m$ – Logarithmic Mean Temperature Difference for counter flow.

$$\begin{aligned} \text{For counter flow, } (\Delta T)_m &= \frac{[(T_1 - t_2) - (T_2 - t_1)]}{\ln \left[\frac{T_1 - t_2}{T_2 - t_1} \right]} \\ &= \frac{[(380 - 210) - (300 - 25)]}{\ln \left[\frac{380 - 210}{300 - 25} \right]} \end{aligned}$$

$$\boxed{(\Delta T)_m = 218.3^\circ\text{C}}$$

$$\text{Heat transfer, } Q = m_h C_{ph} (T_1 - T_2)$$

$$Q = 1 \times 2300 \times (380 - 300)$$

$$\boxed{Q = 184 \times 10^3 \text{ W}}$$

To find correction factor F, refer HMT data book page No 162

[Single pass cross flow heat exchanger – Both fluids unmixed]

From graph,

$$\begin{aligned} X_{\text{axis value}} P &= \left[\frac{t_2 - t_1}{T_1 - t_1} \right] \\ &= \left[\frac{210 - 25}{380 - 25} \right] \end{aligned}$$

X axis Value is 0.52, Curve Value is 0.432,

corresponding Y_{axis} Value is 0.97 i.e

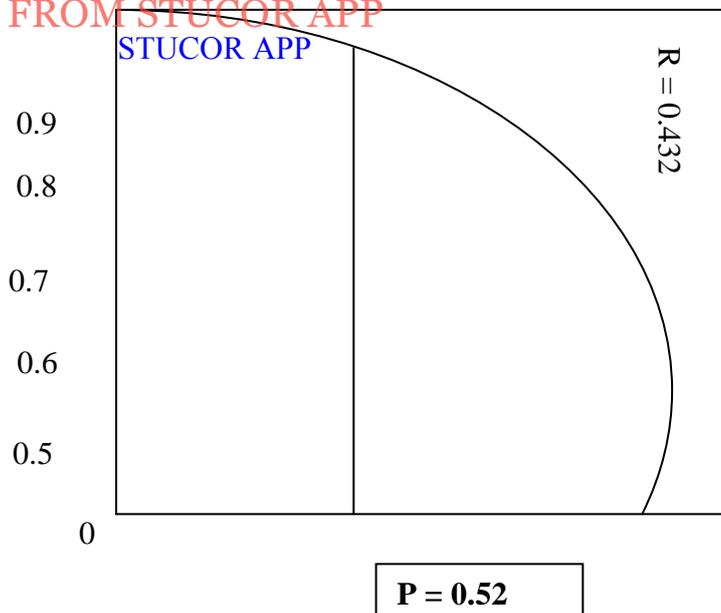
$$\boxed{F = 0.97}$$

$$P = 0.52$$

$$\text{Curve Value } R = \left[\frac{T_1 - T_2}{t_2 - t_1} \right]$$

$$= \left[\frac{380 - 300}{210 - 25} \right]$$

$$R = 0.432$$



Substitute, Q , $F (\Delta T)_m$, and U value in eqn (1)

$$Q = F UA (\Delta T)_m$$

$$184 \times 10^3 = 0.97 \times 750 \times A \times 218.3$$

$$\text{Surface Area } A = 1.15 \text{m}^2$$

8. Classify the heat exchangers, draw the temperature distribution in a condenser and evaporator.

There are several types heat exchangers which may be classified on the basis of

- I. Nature of heat exchange process
- II. Relative direction of fluid motion
- III. Design and constructional features
- IV. Physical state of fluids.

I. Nature of heat exchange process

On the basis of the nature of heat exchange processes, heat exchangers are classified as

Direct contact heat exchangers or open heat exchangers

a) Indirect contact heat exchangers

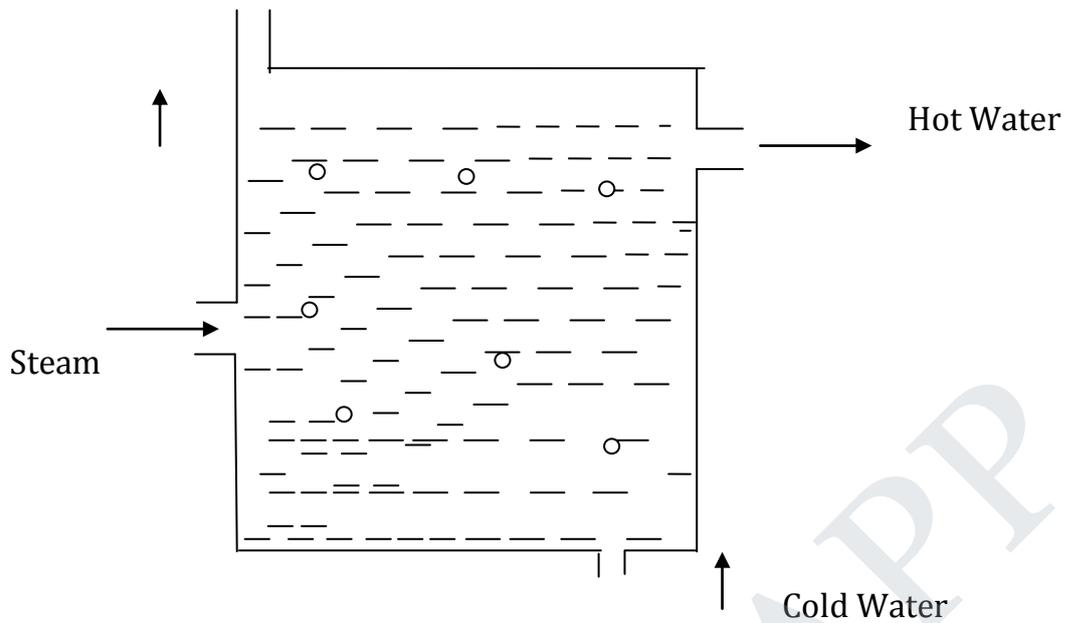
a. Direct contact heat exchangers

The heat exchange takes place by direct mixing of hot and cold fluids.

This heat transfer is usually accompanied by mass transfer.

Ex: cooling towers, direct contact feed heaters

Gas



b. Indirect contact heat exchangers could be carried out by transmission through a wall which separates the two fluids

It may be classified as

- i) Regenerators
- ii) Recuperators

Regenerators

Hot and cold fluids flow alternately through the same space

Ex: IC engines, gas turbines

Recuperators

This is most common type of heat exchanger in which the hot and cold fluid do not come into direct contact with each other but are separated by a tube wall or a surface.

Ex: Automobile radiators, Air pre heaters, Economisers

Advantages

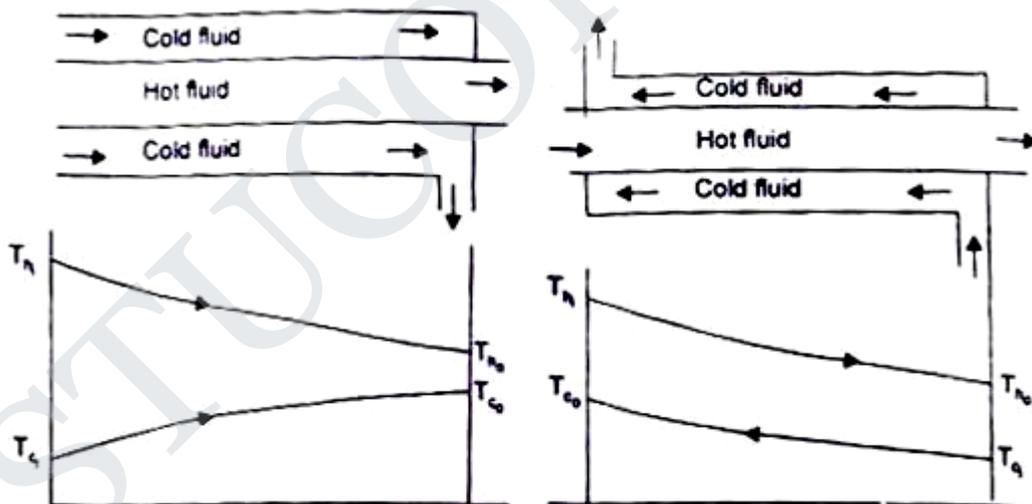
- 1. Easy construction
- 2. More economical
- 3. More surface area for heat transfer

Disadvantages

1. Less heat transfer co-efficient
 2. Less generating capacity
- II. Relative direction of fluid motion
- a. Parallel flow heat exchanger
 - b. Counter flow heat exchanger
 - c. Cross flow heat exchanger

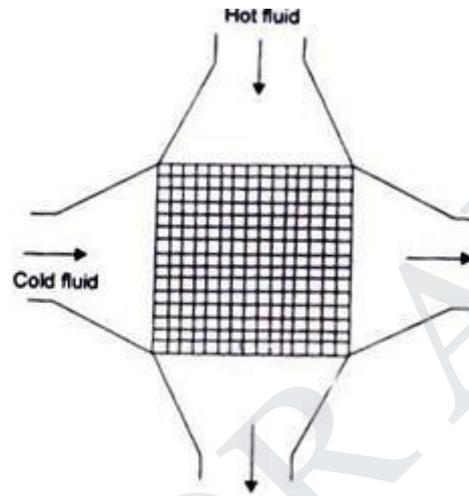
a) Parallel Flow – the hot and cold fluids flow in the same direction. Depicts such a heat exchanger where one fluid (say hot) flows through the pipe and the other fluid (cold) flows through the annulus.

(b) Counter Flow – the two fluids flow through the pipe but in opposite directions. A common type of such a heat exchanger. By comparing the temperature distribution of the two types of heat exchanger



We find that the temperature difference between the two fluids is more uniform in counter flow than in the parallel flow. Counter flow exchangers give the maximum heat transfer rate and are the most favoured devices for heating or cooling of fluids. When the two fluids flow through the heat exchanger only once, it is called one-shell-pass and one-tube-pass

(c) Cross-flow - A cross-flow heat exchanger has the two fluid streams flowing at right angles to each other. illustrates such an arrangement An automobile radiator is a good example of cross-flow exchanger. These exchangers are 'mixed' or 'unmixed' depending upon the mixing or not mixing of either fluid in the direction transverse to the direction of the flow stream and the analysis of this type of heat exchanger is extremely complex because of the variation in the temperature of the fluid in and normal to the direction of flow



III.Design and constructional features

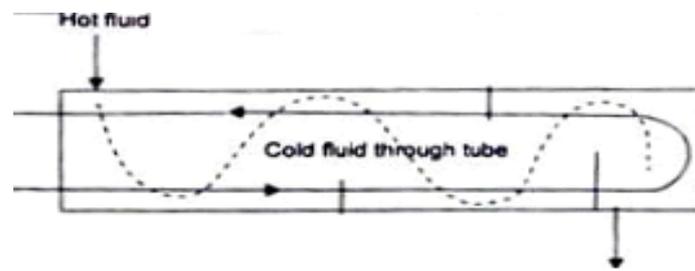
- a. Concentric tubes
- b. Shell and tube
- c. Multiple shell and tube passes
- d. Compact heat exchangers

a Concentric tubes

Two concentric pipes ,each carrying one of the fluids are used as a heat exchanger.The direction of flow may be parallel or counter.

b. Shell and tube

One of the fluids move through a bundle of tubes enclosed by a shell.The other fluid is forced through the shell and it moves over the outside surface of the tubes.

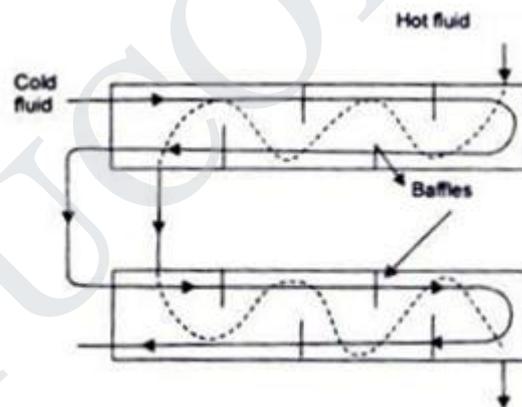


c. Multiple shell and tube passes

If the fluid flowing through the tube makes one pass through half of the tube, reverses its direction of flow, and makes a second pass through the remaining half of the tube, it is called 'one-shell-pass, two-tube-pass' heat exchanger, Many other possible flow arrangements exist and are being used. depicts a 'two-shell-pass, four-tube-pass' exchanger.

d. Compact heat exchangers

There are many special purpose heat exchangers called compact heat exchangers. They are generally employed when convective heat transfer coefficient associated with one of the fluids is much smaller than that associated with the other fluid.



IV. Physical state of fluids

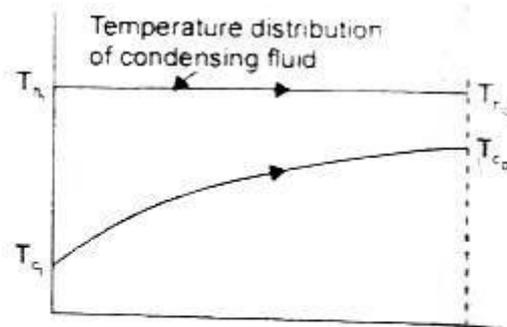
- a. Condensers
- b. Evaporators

a) Condenser

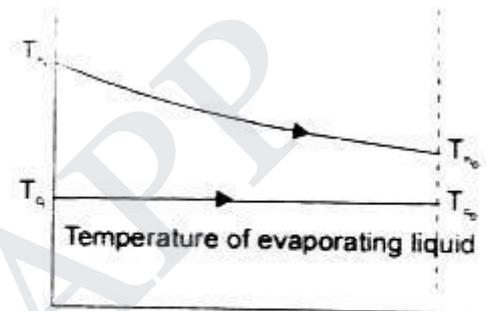
In a condenser, the condensing fluid temperature remains almost constant throughout the exchanger and temperature of the colder fluid gradually increases from the inlet to the exit.

b) Evaporator

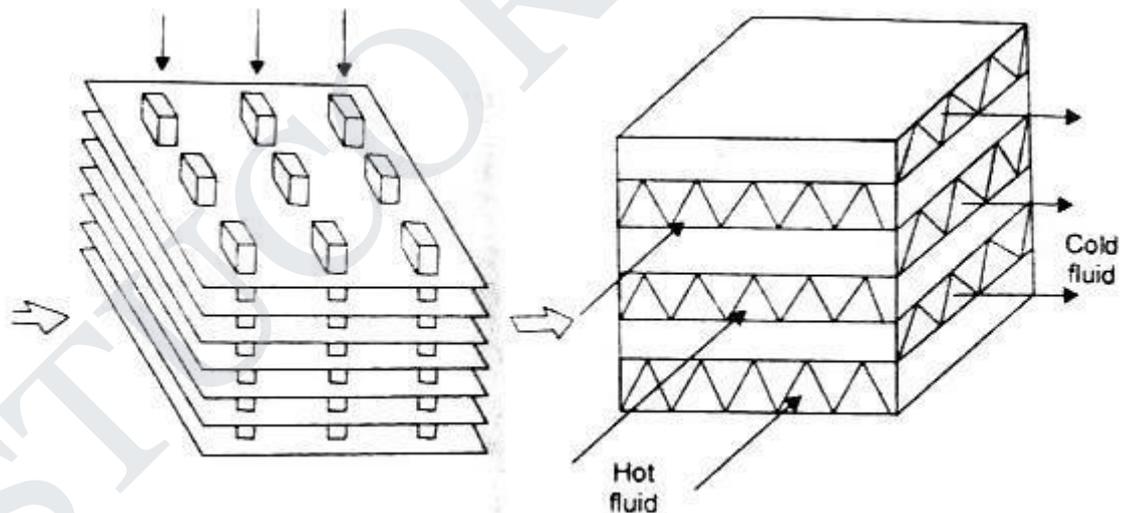
Temperature of the hot fluid gradually decreases from the inlet to the outlet whereas the temperature of the colder fluid remains the same during the evaporation process. Since the temperature of one of the fluids can be treated as constant, it is immaterial whether the exchanger is parallel flow or counter flow.



(a) A condenser



(b) An evaporator



9. Water at the rate of 4 kg/s is heated from 38°C to 55°C in a shell-and-tube heat exchanger. The water is flow inside tube of 2 cm diameter with an average velocity 35 cm/s. How water available at 95°C and at the rate of 2.0 kg/s is used as the heating medium on the shell side. If the length of tubes must not be more than 2m calculate the number of tube passes, the number of tubes per pass and the length of the tubes for one pass shell, assuming $U_0 = 1500 \text{ W/m}^2\text{K}$.

Given:

$$M_c = 4 \text{ kg/s}$$

$$T_{CI} = 38^\circ \text{C}$$

$$T_{CO} = 55^\circ \text{C}$$

$$U = 35 \text{ m/s}$$

$$T_{hi} = 95^\circ \text{C}$$

$$C_h = 2 \text{ kg/s}$$

$$U_o = 1500 \text{ w/m}^2\text{k}$$

To find:

- 1) Number of tubes per pass
- 2) Number of passes
- 3) Length of tube per pass

Solution:

The heat transfer rate for the cold fluid is

$$\begin{aligned} Q &= m_c c_c \Delta T_c \\ &= 4 \times 4186 (55-38) \\ Q &= 284.65 \text{ KW} \end{aligned}$$

The exit temperature of hot fluid can be calculated

$$\begin{aligned} Q &= m_h C_h \Delta T_h \\ &= 284.65 \text{ kw} \\ \Delta T_h &= \frac{284650}{4186 \times 2} \\ &= 34^\circ \text{C} \end{aligned}$$

$$T_{ho} = 95 - 34 = 61^\circ \text{C}$$

Counter flow heat exchanger

$$\Delta T_{ln} = \frac{(\Delta T_1 - \Delta T_2)}{\ln (\Delta T_1 / \Delta T_2)}$$

$$\begin{aligned} \Delta T_1 &= T_{h,i} - T_{c,o} \\ &= 95 - 55 = 40^\circ \text{C} \end{aligned}$$

$$\begin{aligned} \Delta T_2 &= T_{h,o} - T_{c,i} \\ &= 61 - 38 = 23^\circ \text{C} \end{aligned}$$

$$\Delta T_{ln} = \frac{(40 - 23)}{\ln (40/23)} = 30.72^\circ \text{C}$$

$$A = \frac{Q}{U \Delta T_{lm}} = 284.65 \times 1000 / ((1500) \times 30.72)$$

$$= 6.177 \text{m}^2$$

Using average velocity of water in the tubes and its flow rates

$$m_c = \rho A U$$

$$A = 4 / [(1000)(0.35)]$$

$$A = 0.011429 \text{m}^2$$

This area is can also be put as the number of tubes

$$0.011429 = n \pi \frac{d^2}{4}$$

$$n = 36.38$$

Taking $n = 36$, the total surface area of tubes for one shell pass exchanger in terms of L ,

$$A = 6.177 = n \pi d L$$

$$L = 6.177 / [(36) \pi (0.02)]$$

$$L = 2.731 \text{m}$$

Since this length is grater than the permitted length of 2m,

$$P = \frac{t_o - t_i}{T_1 - t_i}$$

$$= 0.3$$

$$R = \frac{T_1 - T}{t_o - t_i}$$

$$R = 2$$

Thus the total area required for one shall pass, 2 tube pass exchanger is

$$A' = Q / [U F \Delta T_{lm}]$$

$$A' = 6.863 \text{m}^2$$

Due to velocity requirement let the number of tubes pr pass still be 36

$$A' = 2 n \pi d l$$

$$L = 6.863 / [2 \times 36 \times \pi \times 0.02]$$

$$L = 1.517 \text{m}$$

PART C - 15 Marks (Questions and Answers)

1. A nickel wire carrying electric current of 1.5 mm diameter and 50 cm long, is submerged in a water bath which is open to atmospheric pressure. calculate the voltage at the burn out point, if at this point the wire carries a current of 200A.

Given:

$$D = 1.5 \text{ mm} = 1.5 \times 10^{-3} \text{ m}$$

$$L = 50 \text{ cm} = 0.50 \text{ m}$$

$$\text{Current, } I = 200 \text{ A.}$$

To find:

Voltage (v)

Solution:

We know that, saturation temperature of water is 100° C .

$$\text{i.e., } T_{\text{sat}} = 100^\circ \text{ C.}$$

PROPERTIES OF WATER AT 100° C .

From HMT Data book page no 21

$$\rho_l = 961 \text{ Kg / m}^3$$

$$v = 0.293 \times 10^{-6} \text{ m}^2/\text{s}$$

$$P_r = 1.740$$

$$C_{pl} = 4216 \text{ J /Kg k}$$

$$\mu_l = \rho_l \times v$$

$$= 961 \times 0.293 \times 10^{-6}$$

$$= 281.57 \times 10^{-6} \text{ N s/ m}^2$$

From steam table at 100° C .

$$h_{fg} = 2256.9 \text{ KJ/Kg}$$

$$h_{fg} = 2256.9 \times 10^3 \text{ J/Kg}$$

$$v_g = 1.673 \text{ m}^3/\text{Kg}$$

$$\rho_v = 1/ v_g = 1/ 1.673$$

$$= 0.597 \text{ Kg / m}^3$$

σ = surface tension for liquid - vapour interface

At 100° C (From HMT databook page no 144)

$$\sigma = 0.0588 \text{ N/m}$$

For Nucleate pool boiling critical heat flux (at burn out)

$$Q/A = 0.18 * h_{fg} * \rho_v [((\sigma * g * (\rho_l - \rho_v)) / (\rho_v^2))]^{0.25}$$

From HMT databook page no 142

Substitute h_{fg} , ρ_l , σ , ρ_v

$$Q/A = 0.18 * 2256.9 * 10^3 * 0.5978 [((0.0588 * 9.81 * (961 - 0.597)) / (0.597)^2)]$$

$$Q/A = 1.52 * 10^6 \text{ W/m}^2.$$

Heat transferred, $Q = V * 1$

$$Q/A = (V * 1) / A$$

$$1.52 * 10^6 = (V * 200) / (\pi d L)$$

$$1.52 * 10^6 = ((V * 200) / (\pi * 1.5 * 10^{-3} * 0.50))$$

$$V = 17.9 \text{ Volts}$$

2. An oil cooler of the form of tubular heat exchanger cools oil from a temperature of 90° C to 35° C by a large pool of stagnant water assumed at constant temperature of 28° C. The tube length is 32 m and diameter is 28 mm. The specific heat and specific gravity of the oil are 2.45 KJ / Kg K and 0.8 respectively. The velocity of the oil is 62 cm / s. Calculate the overall heat transfer co-efficient.

Given:

Hot fluid - oil

Cold fluid - water

(T1, T2)

(t1, t2)

Entry temperature of oil T1 = 90° C

Exit temperature of oil T2 = 35° C

Entry and Exit temperature of water, t1 = t2 = 28° C

Tube length L = 32 m

Diameter D = 28 mm = 0.028 m

Specific heat of oil, $C_{ph} = 2.45 \text{ KJ/Kg k} = 2.45 * 10^3 \text{ J/Kg k}$

Specific gravity of oil = 0.8

Velocity of oil, $C = 62 \text{ cm / s} = 0.62 \text{ m/s}$.

To Find:

Overall heat transfer co- efficient U

Solution:

Specific gravity of oil = Density of oil / density of water

$$= \rho_0 / \rho_w$$

$$0.8 = \rho_0 / 1000$$

$$\rho_0 = 800 \text{ Kg / m}^3.$$

Mass flow rate of oil, $m_h = \rho_0 * A * C$

$$= 800 * ((\pi/4)*(D^2)*0.62$$

$$= 800 * ((\pi/4)*(0.028^2)*0.62$$

$$m_h = 0.305 \text{ Kg / s.}$$

Heat transfer , $Q = m_h * C_{ph} * (T_1 - T_2)$

$$= 0.305 * 2.45 * 10^3 * (90 - 35)$$

$$Q = 41 * 10^3 \text{ W.}$$

We know that

$$\text{Heat transfer , } Q = UA (\Delta T)_m$$

From HMT databook page no 151

$(\Delta T)_m$ = logarithmic mean temperature difference (LMTD)

For parallel flow

$$(\Delta T)_m = [((T_1 - t_1) - (T_2 - t_2))] / \ln [((T_1 - t_1) / (T_2 - t_2))]$$

$$= [((90 - 28) - (35 - 28))] / \ln [((90 - 28) / (35 - 28))]$$

$$(\Delta T)_m = 25.2^\circ \text{ C.}$$

Substitute $(\Delta T)_m$ value in Q Equation

$$Q = UA (\Delta T)_m$$

$$41 * 10^3 = U * \pi * D * L * (\Delta T)_m$$

$$41 * 10^3 = U * \pi * 0.028 * 32 * 25.2$$

$$U = 577.9$$

Overall heat transfer co - efficient , $U = 577.9 \text{ W / m}^2 \text{ K}$

UNIT: IV RADIATION

PART A - 2 Marks (Questions and Answers)

1. State Planck's distribution law. (Nov/Dec 2013)

The relationship between the monochromatic emissive power of a black body and wave length of a radiation at a particular temperature is given by the following expression, by Planck.

$$E_{b\lambda} = \frac{C_1 \lambda^{-5}}{e^{\left(\frac{C_2}{\lambda T}\right) - 1}}$$

Where

$$c_1 = 0.374 \times 10^{-15} \text{ W m}^2$$

$$c_2 = 14.4 \times 10^{-3} \text{ mK}$$

2. State Wien's displacement law & Stefan - Boltzmann law. (Nov/Dec 2010)

The Wien's law gives the relationship between temperature and wave length corresponding to the maximum spectral emissive power of the black body at that temperature.

$$\lambda_{\max} T = 2.9 \times 10^{-3} \text{ mK}$$

The emissive power of a black body is proportional to the fourth power of absolute temperature.

$$E_b = \sigma T^4$$

Where σ = Stefan - Boltzmann constant

$$= 5.67 \times 10^{-8} \text{ W/m}^2\text{K}^4$$

$$\Rightarrow E_b = (5.67 \times 10^{-8}) (2773)^4$$

$$E_b = 3.35 \times 10^6 \text{ W/m}^2$$

3. State Kirchoff's law of radiation. (April/May 2015)

This law states that the ratio of total emissive power to the absorptivity is constant for all surfaces which are in thermal equilibrium with the surroundings. This can be written as

$$\frac{E_1}{\alpha_1} = \frac{E_2}{\alpha_2} = \frac{E_3}{\alpha_3} \dots\dots\dots$$

It also states that the emissivity of the body is always equal to its absorptivity when the body remains in thermal equilibrium with its surroundings.

$$\alpha_1 = E_1; \alpha_2 = E_2 \text{ and soon.}$$

4. What is the purpose of radiation shield? (Nov/Dec 2014)

Radiation shields constructed from low emissivity (high reflective) materials. It is used to reduce the net radiation transfer between two surfaces.

5. Define irradiation (G) and radiosity (J) (Nov/Dec 2015)

It is defined as the total radiation incident upon a surface per unit time per unit area. It is expressed in W/m^2 .

It is used to indicate the total radiation leaving a surface per unit time per unit area. It is expressed in W/m^2 .

6. What are the factors involved in radiation by a body. (Nov /Dec 2014)

- Wave length or frequency of radiation
- The temperature of surface
- The nature of the surface

7. What is meant by shape factor?

The shape factor is defined as the fraction of the radiative energy that is diffused from on surface element and strikes the other surface directly with no intervening reflections. It is represented by Fig. Other names for radiation shape factor are view factor, angle factor and configuration factor.

8. How radiation from gases differs from solids? (Nov/Dec 2013)

A participating medium emits and absorbs radiation throughout its entire volume thus gaseous radiation is a volumetric phenomenon, solid radiation is a surface phenomena Gases emit and absorb radiation at a number of narrow wavelength bands. This is in contrast to solids, which emit and absorb radiation over the entire spectrum.

9. What is black body and gray body?

Black body is an ideal surface having the following properties. A black body absorbs all incident radiation, regardless of wave length and direction. For a prescribed temperature and wave length, no surface can emit more energy than black body. If a body absorbs a definite percentage of incident radiation irrespective of their wave length, the body is known as gray body. The emissive power of a gray body is always less than that of the black body.

10. Define emissive power [E] and monochromatic emissive power. [Ebλ]

The emissive power is defined as the total amount of radiation emitted by a body per unit time and unit area. It is expressed in W/m².

The energy emitted by the surface at a given length per unit time per unit area in all directions is known as monochromatic emissive power.

11. Two parallel radiating Planes 10 x 50 cm are separated by a distance Of 50 cm .what is the radiation shape factor between the planes?(May/June 2012)

L=100 cm B= 50 cm D= 50 cm [From HMT data book ,Page no.92]

X=L/D=100/50=2 Y=B/D = 50/50=1

From table,for X=2 and Y=1

$$F_{12}=F_{21}=0.28588$$

12. What does the view factor represent? When is the view factor from a surface to itself not zero?

The view factor F_{i-j} represents the fraction of the radiation leaving surface i that strikes surface j directly. The view factor from a surface to itself is non-zero for concave surfaces.

13. State Lambert’s cosine law.

It states that the total emissive power E_b from a radiating plane surface in any direction proportional to the cosine of the angle of emission

$$E_b \propto \cos \theta$$

14. Find the temperature of the sun assuming as a Block Body, if the intensity of radiation is maximum at the wavelength of 0.5μ

According to Wien's displacement law:

$$\lambda_{\max} T = 2.9 \times 10^{-3} \text{ mK}$$

$$0.5 \times 10^{-6} T = 2.9 \times 10^{-3}$$

$$\mathbf{T = 5800 \text{ K}}$$

15. What is a radiation shield? Why is it used?

Radiation heat transfer between two surfaces can be reduced greatly by inserting a thin, high reflectivity (low emissivity) sheet of material between the two surfaces. Such highly reflective thin plates or shells are known as radiation shields. Multilayer radiation shields constructed of about 20 shields per cm. thickness separated by evacuated space are commonly used in cryogenic and space applications to minimize heat transfer. Radiation shields are also used in temperature measurements of fluids to reduce the error caused by the radiation effect.

16. State Lamberts cosine law for radiation (April/May 2017)

It states that the total emissive power E_b from a radiating plane surface in any direction proportional to the cosine of the angle of emission. $E_b \propto \cos \theta$

17. Define monochromatic emissive power (Nov/Dec 2016)

The monochromatic emissive power E_λ , is defined as the rate, per unit area, at which the surface emits thermal radiation at a particular wavelength λ . Thus the total and monochromatic hemispherical emissive power are related by

$$E = \int_0^\infty E_\lambda d\lambda$$

18. What is meant by infrared and ultra violet radiation (Nov/Dec 2016)

Infrared radiation, or simply infrared or IR, is electromagnetic radiation (EMR) with longer wavelengths than those of visible light, and is therefore invisible. Ultraviolet (UV) radiation is a type of radiation that is produced by the sun and some artificial sources, such as solariums

PART B - 13 Marks (Questions and Answers)

1. Calculate the following for an industrial furnace in the form of a black body and emitting radiation at 2500°C

Monochromatic emissive power at 1.2 μm wave length.

- i) Wave length at which emission is maximum.
- ii) Maximum emissive power.
- iii) Total emissive power,
- iv) The total emissive of the furnace if it is assumed as a real surface having emissivity equal to 0.9. (Nov / Dec 2014) (Nov / Dec 2015)

Given: Surface temperature $T = 2500^{\circ}\text{C} = 2773\text{K}$

Monochromatic emissive power $\lambda = 1.2 \times 10^{-6} \text{ m}$

Emissivity = 0.9

Solution:

Step 1. Monochromatic Emissive Power:

From Planck's distribution law, we know

$$E_{b\lambda} = \frac{C_1 \lambda^{-5}}{e \left(\frac{C_2}{\lambda T} \right)^{-1}} \quad [\text{From HMT data book, Page No.82}]$$

Where

$$c_1 = 0.374 \times 10^{-15} \text{ W m}^2$$

$$c_2 = 14.4 \times 10^{-3} \text{ mK}$$

$$\lambda = 1.2 \times 10^{-6} \text{ m} \quad [\text{Given}]$$

$$E_{b\lambda} = 5.39 \times 10^{12}$$

Step 2. Maximum wave length (λ_{max})

From Wien's law, we know

$$\lambda_{\text{max}} T = 2.9 \times 10^{-3} \text{ mK}$$

$$\lambda_{\text{max}} \times 2773 = 2.9 \times 10^{-3} \text{ mK}$$

$$\lambda_{\text{max}} = 5.37 \times 10^{-6}$$

Step 3. Maximum emissive power ($E_{b\lambda}$) max:

Maximum emissive power

$$(E_{b\lambda})_{\max} = 1.307 \times 10^{-5} T^5$$

$$= 1.307 \times 10^{-5} \times (2773)^5$$

$$(E_{b\lambda})_{\max} = 2.14 \times 10^{12} \text{ W/m}^2$$

Step 4. Total emissive power (E_b):

From Stefan – Boltzmann law, we know that

$$E_b = \sigma T^4 \quad [\text{From HMT data book Page No.72}]$$

Where σ = Stefan – Boltzmann constant

$$= 5.67 \times 10^{-8} \text{ W/m}^2\text{K}^4$$

$$\Rightarrow E_b = (5.67 \times 10^{-8}) (2773)^4$$

$$E_b = 3.35 \times 10^6 \text{ W/m}^2$$

Step 5. Total emissive power of a real surface:

$$(E_b)_{\text{real}} = \varepsilon \sigma T^4$$

Where ε = Emissivity = 0.9

$$(E_b)_{\text{real}} = 0.9 \times 5.67 \times 10^{-8} (2773)^4$$

$$(E_b)_{\text{real}} = 3.011 \times 10^6 \text{ W/m}^2$$

2. Two parallel plates of size 1.0 m x 1.0 m spaced 0.5 m apart are located in a very large room, the walls are maintained at a temperature of 27°C. One plate is maintained at a temperature of 900°C and the other at 400°C. Their emissivities are 0.2 and 0.5 respectively. If the plates exchange heat with themselves and surroundings, find the heat transfer to each plate and to the room. Consider only the plate surface facing each other. (May/June 2012 & Nov/Dec 2014)

Given:

Size of the Plate = 1.0 m x 1.0 m

Distance between plates = 0.5 m

Room Temperature, $T_3 = 27^\circ\text{C} + 273 = 300 \text{ K}$

First plate temperature, $T_1 = 900^\circ\text{C} + 273 = 1173 \text{ K}$

Second plate temperature , $T_2=400^{\circ}\text{C} + 273 =673 \text{ K}$

Emissivity of first plate, $\epsilon_1 = 0.2$

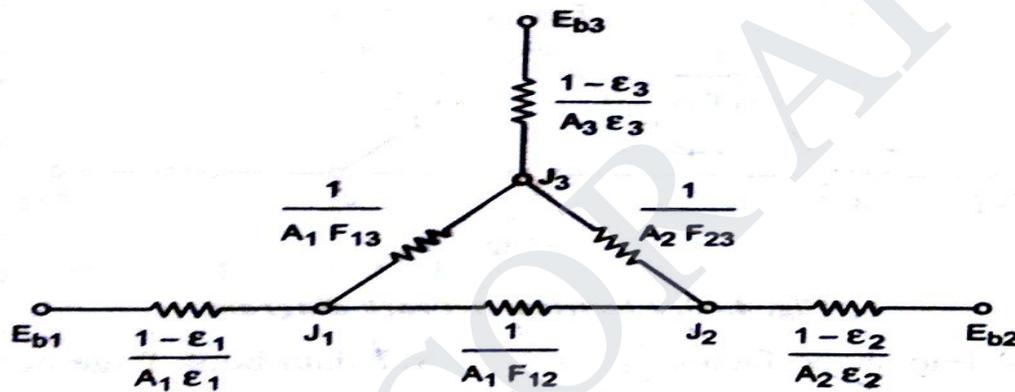
Emissivity of second plate, $\epsilon_2 = 0.5$

To Find:

1. Net Heat Transfer to each
2. Net heat transfer to room

Solution:

In this problem heat exchange take place between two plates and the room .so, this is three surface problem and the corresponding radiation network is given below.



Electrical network diagram

Area , $A_1 = 1 \times 1 = 1 \text{ m}^2$

$$A_1 = A_2 = 1 \text{ m}^2$$

Since the room is large , $A_3 = \infty$

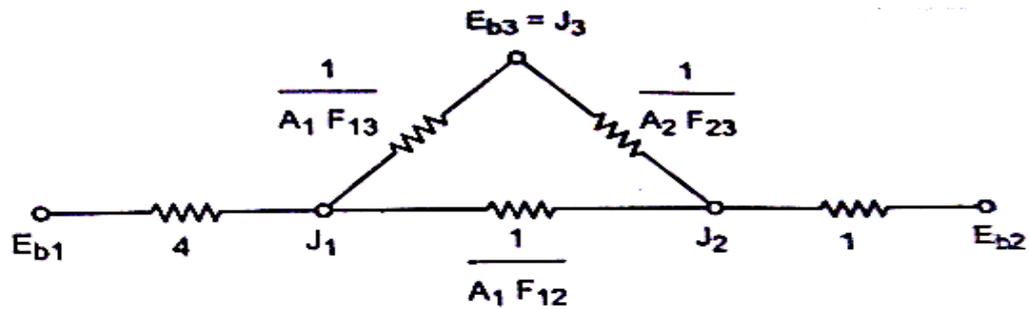
Step:1 From electrical network diagram,

$$\frac{1 - \epsilon_1}{A_1 \epsilon_1} = \frac{1 - 0.2}{1 \times 0.2} = 4$$

$$\frac{1 - \epsilon_2}{A_2 \epsilon_2} = \frac{1 - 0.5}{1 \times 0.5} = 1$$

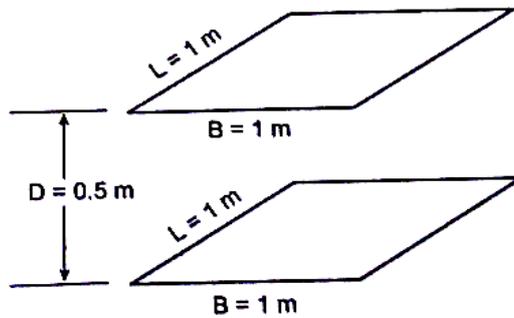
$$\frac{1 - \epsilon_3}{A_3 \epsilon_3} = 0 \quad [A_3 = \infty]$$

Apply $\frac{1 - \epsilon_1}{A_1 \epsilon_1} = 4, \frac{1 - \epsilon_2}{A_2 \epsilon_2} = 1, \frac{1 - \epsilon_3}{A_3 \epsilon_3} = 0$ values in electrical network diagram.



Electrical network diagram

Step:2 To find shape factor F_{12} , refer HMT data book page no.92 and 93



$$X = \frac{L}{D} = \frac{1}{0.5} = 2$$

$$Y = \frac{B}{D} = \frac{1}{0.5} = 2$$

X value is 2, Y value is 2. From that, we can find corresponding shape factor value is 0.41525 [From the table]

i.e $F_{12} = 0.41525$

we know that,

$$F_{11} + F_{12} + F_{13} = 1, \text{ we know that } F_{11} = 0$$

$$F_{13} = 1 - 0.41525$$

$$F_{13} = 0.5847$$

Similarly, $F_{21} + F_{22} + F_{23} = 1$ We know that, $F_{22} = 0$

$$\begin{aligned} F_{23} &= 1 - F_{21} \\ &= 1 - F_{12} = 1 - 0.41525 \\ &= 0.5847 \end{aligned}$$

From electrical network diagram,

$$\frac{1}{A_1 F_{13}} = \frac{1}{1 \times 0.5847} = 1.7102$$

$$\frac{1}{A_2 F_{23}} = \frac{1}{1 \times 0.5847} = 1.7102$$

$$\frac{1}{A_1 F_{12}} = \frac{1}{1 \times 0.41525} = 2.408$$

Step: 3 From stefan-Boltzmann Law,

$$E_b = \sigma T^4$$

$$E_{b1} = \sigma T_1^4$$

$$= 5.67 \times 10^{-8} [1173]^4$$

$$E_{b1} = 107.34 \times 10^3 \text{ W/m}^2$$

$$E_{b2} = \sigma T_2^4$$

$$= 5.67 \times 10^{-8} [673]^4$$

$$E_{b2} = 11.63 \times 10^3 \text{ W/m}^2$$

$$E_{b3} = \sigma T_3^4$$

$$= 5.67 \times 10^{-8} [300]^4$$

$$E_{b3} = 459.27 \text{ W/m}^2$$

From the electrical network diagram , we know that

$$E_{b3} = J_3 = 459.27 \text{ W/m}^2$$

Step: 4

The radiosities J_1 and J_2 can be calculated by using Krichoff's

The sum of current entering the node J_1 is zero.

At Node J_1 :

$$\frac{E_{b1} - J_1}{4} + \frac{J_2 - J_1}{A_1 F_{12}} + \frac{E_{b3} - J_1}{A_1 F_{13}} = 0 \quad [\text{From electrical network diagram}]$$

$$\frac{107.34 \times 10^3 - J_1}{4} + \frac{J_2 - J_1}{2.408} + \frac{459.27 - J_1}{1.7102} = 0$$

$$26835 - 0.25J_1 + 0.415J_2 - 0.415J_1 + 268.54 - 0.5847J_1 = 0$$

$$-1.2497 J_1 + 0.415 J_2 = -27.10 \times 10^3 \text{ -----(1)}$$

At Node J_2 :

$$\frac{J_1 - J_2}{A_1 F_{12}} + \frac{E_{b3} - J_1}{A_2 F_{23}} + \frac{E_{b2} - J_2}{1} = 0$$

$$\frac{J_2 - J_1}{2.408} + \frac{459.27 - J_1}{1.7102} + \frac{11.63 \times 10^3}{1} = 0$$

$$0.415 J_1 - 1.4997 J_2 = -11.898 \times 10^3 \text{ -----(2)}$$

Solving the equation (1) and (2)

$$-1.2497 J_1 + 0.415 J_2 = -27.10 \times 10^3$$

$$-0.415 J_1 - 1.4997 J_2 = -11.898 \times 10^3$$

$$J_1 = 26.780 \times 10^3 \text{ W/m}^2$$

$$J_2 = 15.34 \times 10^3 \text{ W/m}^2$$

Step: 5

Heat lost by plate (1) $Q_1 = \frac{E_{b1} - J_1}{\frac{1 - \epsilon_1}{A_1 \epsilon_1}}$ [From electrical network diagram]

$$= \frac{107.34 \times 10^3 - 26.780 \times 10^3}{\frac{1 - 0.2}{1 \times 0.2}}$$

$$Q_1 = 20.140 \times 10^3 \text{ W}$$

Heat lost by plate (2) $Q_2 = \frac{J_2 - E_{b2}}{\frac{1 - \epsilon_2}{A_2 \epsilon_2}}$

$$= \frac{15.34 \times 10^3 - 11.63 \times 10^3}{\frac{1 - 0.5}{1 \times 0.5}}$$

$$Q_2 = 3710 \text{ W}$$

Total heat lost by the plates(1) and(2)

$$Q = Q_1 + Q_2$$

$$Q = 20.140 \times 10^3 + 3710$$

$$Q = 23.850 \times 10^3 \text{ W}$$

Total heat received or absorbed by the room

$$Q = \frac{J_1 - J_3}{\frac{1}{A_1 F_{13}}} + \frac{J_2 - J_3}{\frac{1}{A_2 F_{23}}}$$

$$Q = \frac{26.780 \times 10^3 - 459.27}{1.7102} + \frac{11.06 \times 10^3 - 459.27}{1.7102}$$

$$Q = 24.09 \times 10^3 \text{ W}$$

Result:

1. Net heat lost by each plates

$$Q_1 = 20.140 \times 10^3 \text{ W}$$

$$Q_2 = 3710 \text{ W}$$

2. Net heat transfer to the room

$$Q = 24.09 \times 10^3 \text{ W}$$

3. Emissivities of two large parallel planes maintained at 800°C and 300°C are 0.3 and 0.5 respectively. Find the net radiant heat exchange per square meter of the plates. Find the percentage of reduction in heat transfer when a polished aluminium shield ($\epsilon = 0.05$) is placed between them. Also find the temperature of the shield (April/May 2015)(Nov/Dec 2015).(NOV/DEC 2013)

Given:

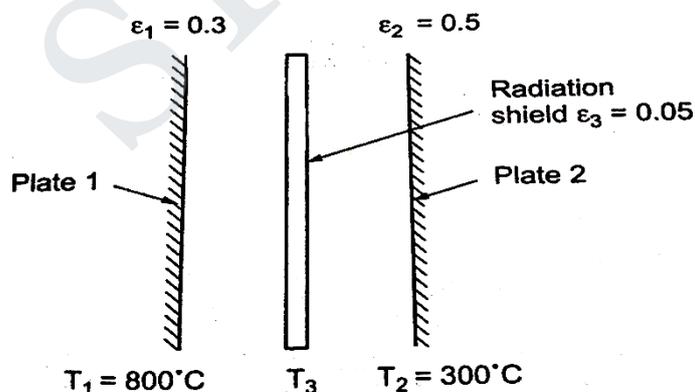
$$T_1 = 800^\circ\text{C} + 273 = 1073 \text{ K}$$

$$T_2 = 300^\circ\text{C} + 273 = 573 \text{ K}$$

$$\epsilon_1 = 0.3$$

$$\epsilon_2 = 0.5$$

$$\text{Radiation shield emissivity } \epsilon_3 = 0.05$$



To find:

- (i) Percentage of reduction in heat transfer due to radiation shield.
- (ii) Temperature of the shield (T_3)

Solution:

Case: 1 Heat transfer without radiation shield:

Heat exchange between two large parallel plates without radiation shield is given by

Step: 1 $Q_{12} = \epsilon^- \sigma A [T_1^4 - T_2^4]$

Where $\epsilon^- = \frac{1}{\frac{1}{\epsilon_1} + \frac{1}{\epsilon_2} - 1}$
 $= \frac{1}{\frac{1}{0.3} + \frac{1}{0.5} - 1}$

$\epsilon^- = 0.2307$

$Q_{12} = 0.2307 \times 5.67 \times 10^{-8} \times A \times [(1073)^4 - (573)^4]$

Step: 2

$\frac{Q_{12}}{A} = 15.9 \times 10^3 \text{ W/m}^2$

Heat transfer without radiation shield $\frac{Q_{12}}{A} = 15.9 \times 10^3 \text{ W/m}^2$ ----- (1)

Case : 2 Heat transfer with radiation shield:

Heat exchange between radiation plate 1 and radiation shield 3 is given

Step: 3

$Q_{13} = \epsilon^- \sigma A [T_1^4 - T_3^4]$

Where $\epsilon^- = \frac{1}{\frac{1}{\epsilon_1} + \frac{1}{\epsilon_3} - 1}$

$Q_{13} = \frac{\sigma A [T_1^4 - T_3^4]}{\frac{1}{\epsilon_1} + \frac{1}{\epsilon_3} - 1}$ ----- (2)

Heat exchange between radiation shield 3 and plate 2 is given

Step: 4

$Q_{32} = \epsilon^- \sigma A [T_3^4 - T_2^4]$

Where $\varepsilon^- = \frac{1}{\frac{1}{\varepsilon_3} + \frac{1}{\varepsilon_2} - 1}$

$$Q_{32} = \frac{\sigma A [T_3^4 - T_2^4]}{\frac{1}{\varepsilon_3} + \frac{1}{\varepsilon_2} - 1} \text{----- (3)}$$

Step: 5

We know that,

$$Q_{13} = Q_{32}$$

$$\frac{\sigma A [T_1^4 - T_3^4]}{\frac{1}{\varepsilon_1} + \frac{1}{\varepsilon_3} - 1} = \frac{\sigma A [T_3^4 - T_2^4]}{\frac{1}{\varepsilon_3} + \frac{1}{\varepsilon_2} - 1}$$

$$\frac{\sigma A [1073^4 - T_3^4]}{\frac{1}{0.3} + \frac{1}{0.05} - 1} = \frac{\sigma A [T_3^4 - 573^4]}{\frac{1}{0.05} + \frac{1}{0.5} - 1}$$

$$3.02 \times 10^{13} = 43.3 T_3^4$$

$$T_3 = 913.8 \text{ K}$$

Temperature of the shield T₃ = 913.8 K

Substitute T₃ value in equation (2) or (3),

Heat transfer with radiation shield $Q_{13} = \frac{\sigma A [1073^4 - 913.8^4]}{\frac{1}{0.3} + \frac{1}{0.05} - 1}$

$$\frac{Q_{13}}{A} = 159.46 \text{ W/m}^2$$

Step: 6

Percentage of reduction in heat transfer due to radiation shield

$$= \frac{Q_{withoutshield} - Q_{withshield}}{Q_{withshield}}$$

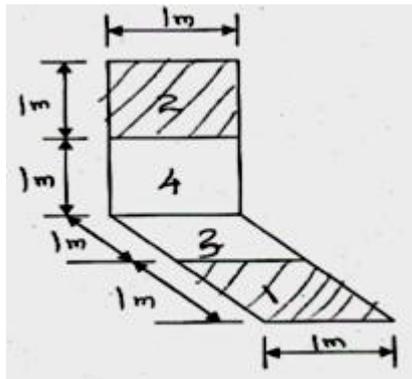
$$= \frac{Q_{12} - Q_{13}}{Q_{12}}$$

$$= \frac{15.8 \times 10^3 - 1594.6}{15.8 \times 10^3} \times 100$$

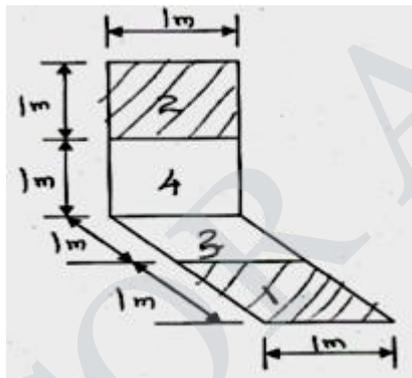
$$= 0.899 \times 100 \% = 89.9\%$$

Percentage of reduction in heat transfer due to radiation shield= 89.9%

4. The area A_1 and A_2 are perpendicular but do not share the common edge .find the shape factor F_{1-2} for the arrangement. (Nov/Dec 2015).

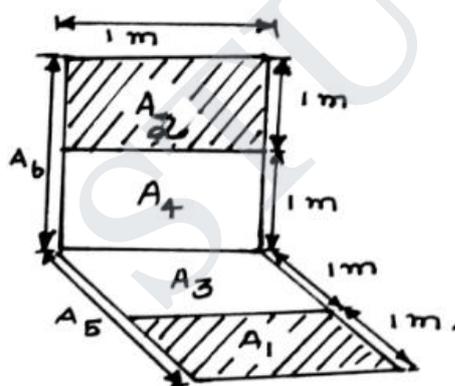


Given:



To find : Shape Factor of F_{1-2}

Solution:



From the figure we know that

Step: 1

$$A_5 = A_1 + A_3$$

$$A_6 = A_2 + A_4$$

Further Step: 2

$$A_5 F_{5-6} = A_1 F_{1-6} + A_3 F_{3-6}$$

$$[A_5 = A_1 + A_3, F_{5-6} = F_{1-6} + F_{3-6}]$$

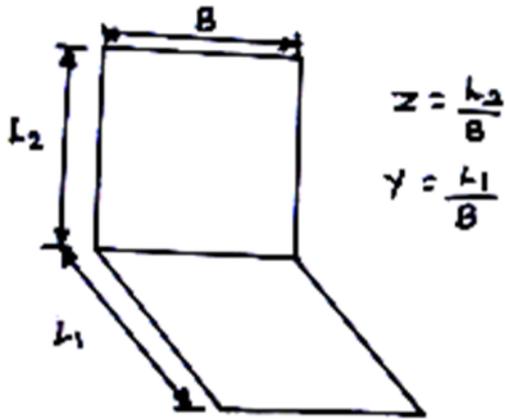
$$= A_1 F_{1-2} + A_1 F_{1-4} + A_3 F_{3-6} \quad [F_{1-6} = F_{1-2} + F_{1-4}]$$

$$A_5 F_{5-6} = A_1 F_{1-2} + A_5 F_{5-4} - A_3 F_{3-4} + A_3 F_{3-6} \quad [A_1 = A_5 - A_3, F_{1-4} = F_{5-4} - F_{3-4}]$$

$$A_1 F_{1-2} = A_5 F_{5-6} - A_5 F_{5-4} + A_3 F_{3-4} - A_3 F_{3-6}$$

$$F_{1-2} = \frac{A_5}{A_1} [F_{5-6} - F_{5-4}] + \frac{A_3}{A_1} [F_{3-4} - F_{3-6}] \text{-----(1)}$$

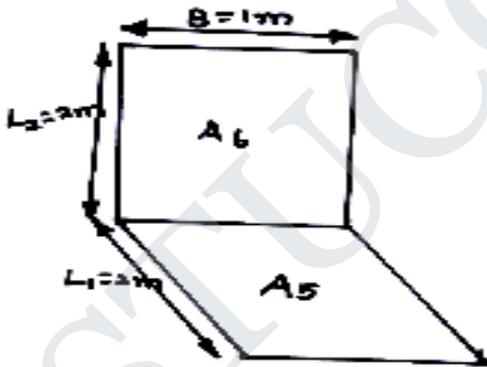
[Refer HMT data book, Page no.95]



$$Z = \frac{L_2}{B}, Y = \frac{L_1}{B}$$

Step: 3

Shape Factor for the area A5 and A6



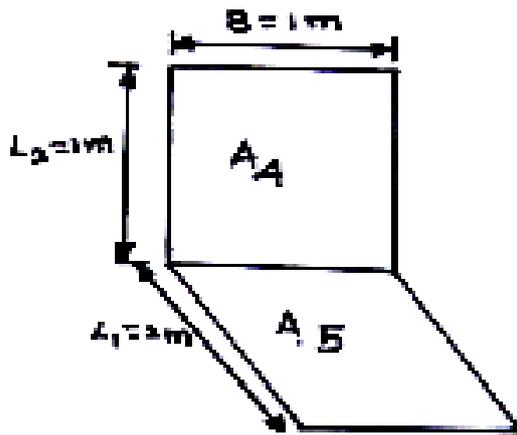
$$Z = \frac{L_2}{B} = \frac{2}{1} = 2$$

$$Y = \frac{L_1}{B} = \frac{2}{1} = 2$$

Z value is 2, Y value is 2. From that, we can find Corresponding shape factor value is 0.14930

$$F_{5-6} = 0.14930$$

Shape Factor for the area A_5 and A_4



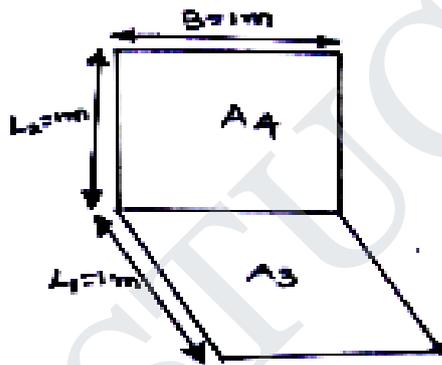
$$Z = \frac{L_2}{B} = \frac{1}{1} = 1$$

$$Y = \frac{L_1}{B} = \frac{2}{1} = 2$$

Z value is 1, Y value is 2. From that, we can find Corresponding shape factor value is 0.11643

$$F_{5-4} = 0.11643$$

Shape Factor for the area A_3 and A_4



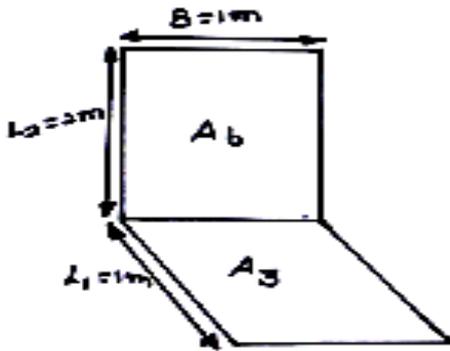
$$Z = \frac{L_2}{B} = \frac{1}{1} = 1$$

$$Y = \frac{L_1}{B} = \frac{1}{1} = 1$$

Z value is 1, Y value is 1. From that, we can find Corresponding shape factor value is 0.2004

$$F_{3-4} = 0.2004$$

Shape Factor for the area A_3 and A_6 :



$$Z = \frac{L_2}{B} = \frac{2}{1} = 2$$

$$Y = \frac{L_1}{B} = \frac{1}{1} = 1$$

Z value is 2, Y value is 1. From that, we can find Corresponding shape factor value is 0.23285

$$F_{3-6} = 0.23285$$

Step: 4

Substitute F_{3-6} , F_{3-4} , F_{5-4} and F_{5-6} in equation (1)

$$F_{1-2} = \frac{A_5}{A_1} [F_{5-6} - F_{5-4}] + \frac{A_3}{A_1} [F_{3-4} - F_{3-6}]$$

$$A_5 = 2 ; A_3 = A_1 = 1$$

$$F_{1-2} = \frac{2}{1} [0.14930 - 0.11643] + \frac{1}{1} [0.2004 - 0.23285]$$

$$F_{1-2} = 0.03293$$

$$F_{1-2} = 0.03293$$

5. (a) State and Prove Kirchhoff's law of thermal radiation.

This law states that the ratio of total emissive power to the absorptivity is constant for all surfaces which are in thermal equilibrium with the surroundings.

$$\frac{E_1}{\alpha_1} = \frac{E_2}{\alpha_2} = \frac{E_3}{\alpha_3} \dots\dots\dots$$

It also states that the emissivity of the body is always equal to its absorptivity when the body remains in thermal equilibrium with its surroundings.

$$\alpha_1 = E_1 ; \alpha_2 = E_2 \text{ and soon.}$$

(b) What is a black body? A 20 cm diameter spherical ball at 527°C is suspended in the air. The ball closely approximates a black body. Determine the total black body emissive power, and spectral black body emissive power at a wavelength of 3 μm.

A black body absorbs all incident radiation, regardless of wave length and direction. For a prescribed temperature and wave length, no surface can emit more energy than black body.

Given:

In sphere, (Black body)

Diameter of sphere, $d = 20 \text{ cm} = 0.2 \text{ m}$

Temperature of spherical ball, $T = 527^\circ\text{C} + 273 = 800 \text{ K}$

To Find:

- (i) Total black body emissive power, E_b
- (ii) Spectral black body emissive power at wavelength of 3 μm.

Solution:

(i) **Step:1** Total black body emissive power , E_b

$$E_b = \sigma AT^4 = 5.67 \times 10^{-8} \times \pi \times (0.2)^2 \times (800)^4$$

$$E_b = 2920 \text{ W}$$

(ii) **Step:2** Spectral black body emissive power: at $\lambda = 3 \mu\text{m}$

$$E_{b\lambda} = \frac{C_1}{\lambda^5 [\exp(\frac{C_2}{\lambda T}) - 1]}$$

$$= \frac{0.374 \times 10^{-15}}{(3 \times 10^{-6})^5 [\exp(\frac{14.14 \times 10^{-3}}{3 \times 10^{-6} \times 800}) - 1]}$$

$$E_{b\lambda} = 3824.3 \times 10^6 \text{ W/m}^2$$

$$E_{b\lambda} = 3824.3 \times 10^6 \text{ W/m}^2$$

6. Consider a cylinder furnace with outer radius = 1m and height=1 m. The top (surface 1) and the base (surface2) of the furnace have emissivities 0.8 and 0.4 and are maintained at uniform temperature of 700 K and 500 K

respectively. The side surface closely approximates a black body and is maintained at a temperature of 400 K. Find the net rate of radiation heat transfer at each surface during steady state operation. (May/June 2015)

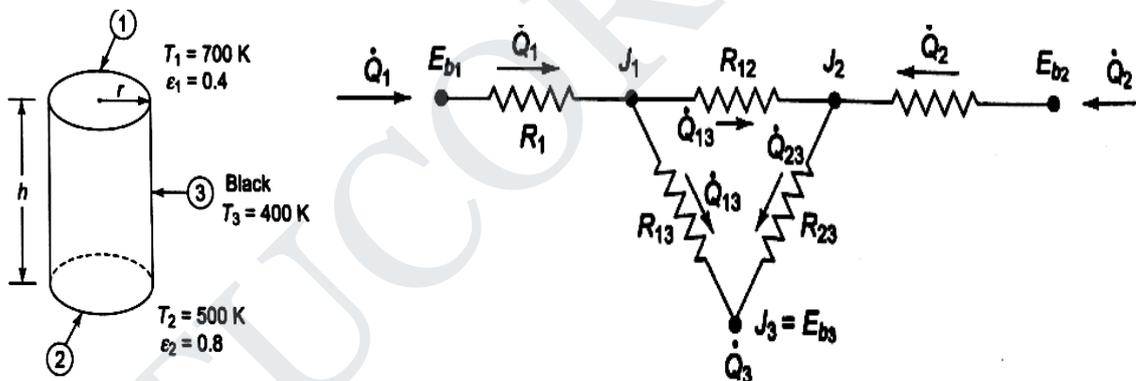
Given:

- Radius of the cylinder = 1m
- Height of the cylinder = 1m
- Top surface temperature $T_1 = 700$ K
- Base surface temperature $T_2 = 500$ K
- side surface temperature $T_3 = 400$ K
- Top surface emissivities $\epsilon_1 = 0.8$
- Base surface emissivities $\epsilon_2 = 0.4$

To Find:

- Net rate of radiation heat transfer at each surface

Solution:



The furnace and the radiation network are shown in above figure .writing the energy balance for the node 1 and 2,

Step: 1

$$\frac{E_{b1} - J_1}{R_1} = \frac{J_1 - J_2}{R_{12}} + \frac{J_1 - J_3}{R_{13}} \text{----- (1)}$$

$$\frac{E_{b2} - J_2}{R_2} = \frac{J_2 - J_1}{R_{12}} + \frac{J_2 - J_3}{R_{23}} \text{----- (2)}$$

$$E_{b1} = \sigma T_1^4 = 5.67 \times 10^{-8} (700)^4 = 13614 \text{ W/m}^2$$

$$E_{b2} = \sigma T_2^4 = 5.67 \times 10^{-8} (500)^4 = 3544 \text{ W/m}^2$$

$$E_{b3} = \sigma T_3^4 = 5.67 \times 10^{-8} (400)^4 = 1452 \text{ W/m}^2$$

$$A_1 = A_2 = \Pi r^2 = \Pi(1)^2 = 3.14\text{m}^2$$

Step: 2

From the HMT data Book [page no. 91]

The view factor from the base to top is found to be $F_{12} = 0.38$

Now, $F_{11} + F_{12} + F_{13} = 1$, we know that $F_{11} = 0$

$$F_{13} = 1 - F_{12} = 1 - 0.38 = 0.62$$

$$R_1 = \frac{1 - \varepsilon_1}{A_1 \varepsilon_1} = \frac{1 - 0.8}{3.14 \times 0.8} = 0.0796 \text{ m}^2$$

$$R_2 = \frac{1 - \varepsilon_2}{A_2 \varepsilon_2} = \frac{1 - 0.4}{3.14 \times 0.4} = 0.4777 \text{ m}^2$$

$$R_{12} = \frac{1}{A_1 F_{12}} = \frac{1}{3.14 \times 0.38} = 0.8381 \text{ m}^2$$

$$R_{23} = \frac{1}{A_2 F_{23}} = \frac{1}{3.14 \times 0.62} = 0.5137 \text{ m}^2 = R_{13}$$

Step: 3

On substitution, of this value in above equation (1) and (2)

$$\frac{13614 - J_1}{0.0796} = \frac{J_1 - J_2}{0.8381} + \frac{J_1 - 1452}{0.5137}$$

$$\frac{3544 - J_2}{0.4777} = \frac{J_2 - J_1}{0.8381} + \frac{J_2 - 1452}{0.5137}$$

By solving the above equations,

$$J_1 = 11418 \text{ W/m}^2 \text{ and } J_2 = 4562 \text{ W/m}^2$$

$$Q_1 = \frac{E_{b1} - J_1}{R_1} = \frac{13614 - 11418}{0.0796} = 27,588 \text{ W}$$

$$Q_2 = \frac{E_{b2} - J_2}{R_2} = \frac{3544 - 4562}{0.4777} = 2132 \text{ W}$$

$$Q_3 + \frac{J_1 - J_3}{R_{13}} + \frac{J_2 - J_3}{R_{23}} = 0$$

$$Q_3 = \frac{1452 - 11418}{0.5137} + \frac{1452 - 4562}{0.5137} = 25455 \text{ W}$$

Net rate of radiation heat transfer at each surface

$$\begin{aligned} Q_1 &= 27,588 \text{ W} \\ Q_2 &= 2132 \text{ W} \\ Q_3 &= 25455 \text{ W} \end{aligned}$$

7. The spectral emissivity function of an opaque surface at 1000 K is approximated as

$$\epsilon_{\lambda 1} = 0.4, 0 \leq \lambda < 2 \mu\text{m};$$

$$\epsilon_{\lambda 2} = 0.7, 2 \mu\text{m} \leq \lambda < 6 \mu\text{m};$$

$$\epsilon_{\lambda 3} = 0.3, 6 \mu\text{m} \leq \lambda < \infty$$

Determine the average emissivity of the surface and the rate of radiation emission from the surface, in W/m² (Nov / Dec 2015)

Given:

Surface temperature = 1000 K

$$\epsilon_{\lambda 1} = 0.4, 0 \leq \lambda < 2 \mu\text{m};$$

$$\epsilon_{\lambda 2} = 0.7, 2 \mu\text{m} \leq \lambda < 6 \mu\text{m};$$

$$\epsilon_{\lambda 3} = 0.3, 6 \mu\text{m} \leq \lambda < \infty$$

To Find: Rate of radiation emission from the surface, in W/m²

Solution:

The average emissivity can be determined by breaking the integral

Step:1

$$\begin{aligned} \epsilon(T) &= \frac{\epsilon_1 \int_0^{\lambda_1} E_{b\lambda}(T) d\lambda}{\sigma T^4} + \frac{\epsilon_2 \int_{\lambda_1}^{\lambda_2} E_{b\lambda}(T) d\lambda}{\sigma T^4} + \frac{\epsilon_3 \int_{\lambda_2}^{\infty} E_{b\lambda}(T) d\lambda}{\sigma T^4} \\ &= \epsilon_1 f_{0-\lambda_1}(T) + \epsilon_2 f_{\lambda_1-\lambda_2}(T) + \epsilon_3 f_{\lambda_2-\infty}(T) \\ &= \epsilon_2 f_{\lambda_1} + \epsilon_2 (f_{\lambda_2} - f_{\lambda_1}) + \epsilon_3 (1 - f_{\lambda_2}) \end{aligned}$$

Where f_{λ_1} and f_{λ_2} are black body radiation function corresponding to $\lambda_1 T$ to $\lambda_2 T$

Step:2

$$\lambda_1 T = 2 \times 1000 = 2000 \mu\text{mK}, f_{\lambda_1} = 0.066728$$

$$\lambda_2 T = 6 \times 1000 = 6000 \mu\text{mK}, f_{\lambda_2} = 0.737818 \text{ [From HMT data Book Page No: 83]}$$

$$\epsilon = 0.4 \times 0.066728 + 0.7(0.737818 - 0.066728) + 0.3(1 - 0.737818)$$

$$\epsilon = 0.5751$$

Step:3

$$E = \epsilon \sigma T^4 = 0.5715 \times 5.67 \times 10^{-8} \times (1000)^4$$

$$E = 32608 \text{ W/m}^2$$

$$E = 32608 \text{ W/m}^2$$

8. The inner sphere of a liquid oxygen container is 400 mm dia., outer sphere is 500 mm dia., both have emissivity 0.05. Determine the rate of liquid oxygen evaporation at -183°C , when the outer sphere temperature is 20°C . The latent heat of evaporation 210 kJ/kg . Neglect losses due to other modes of heat transfer. (May/ June 2016)

Given:

Inner wall temperature $T_1 = -183^{\circ}\text{C} + 273 = 90\text{K}$

Outer wall Temperature $T_2 = 20^{\circ}\text{C} + 273 = 293\text{K}$

Inner diameter $D_1 = 400\text{ mm} = 0.4\text{ m} = r_1 = 0.2\text{ m}$

Outer diameter $D_2 = 500\text{ mm} = 0.5\text{ m} = r_2 = 0.25\text{ m}$

Emissivity, $\varepsilon_1 = \varepsilon_2 = 0.05$

Latent heat of evaporation = $210\text{ kJ/kg} = 210 \times 10^3\text{ J/kg}$

To Find:

Rate of liquid oxygen evaporation

Solution:

$$\text{Heat transfer } Q_{12} = \bar{\varepsilon} \sigma A_1 [T_1^4 - T_2^4]$$

$$\bar{\varepsilon} = \frac{1}{\frac{1}{\varepsilon_1} + \frac{A_1}{A_2} \left(\frac{1}{\varepsilon_2} - 1 \right)}$$

$$A_1 = 4\pi r_1^2 = 4 \times 3.14 \times (0.2)^2 = 0.5026$$

$$A_2 = 4\pi r_2^2 = 4 \times 3.14 \times (0.25)^2 = 0.7853$$

$$= \frac{1}{\frac{1}{0.05} + \frac{0.5026}{0.7853} \left(\frac{1}{0.05} - 1 \right)}$$

$$\bar{\varepsilon} = 0.0310$$

$$Q_{12} = \bar{\varepsilon} \sigma A_1 [T_1^4 - T_2^4]$$

$$= 0.0310 \times 5.67 \times 10^{-8} \times 0.5026 [90^4 - 293^4]$$

$$Q = -6.4529\text{ W}$$

$$\text{Rate of Evaporation} = \frac{\text{heatTransfer}}{\text{LatentHeat}}$$

$$= \frac{6.4529}{210 \times 10^3}$$

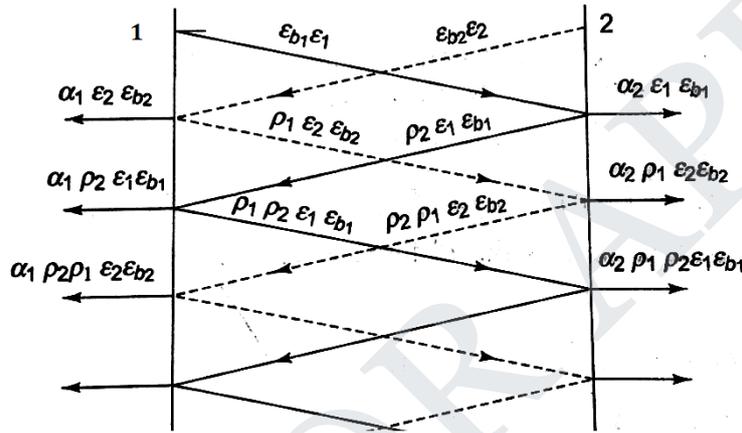
$$= 3.07 \times 10^{-5}$$

Rate of liquid oxygen evaporation = 3.07×10^{-5}

Rate of liquid oxygen evaporation = 3.07×10^{-5}

9. Derive relation for heat exchange between infinite parallel planes. (May/June 2014).

The radiant interchange between two infinite parallel gray planes involves no geometry factor, since $F_{12} = F_{21} = 1.0$. let us consider two gray planes,



For gray surface $\alpha = \epsilon$ and $\rho = 1 - \epsilon$. Surface 1 emits $\epsilon_1 E_{b1}$ per unit time and area. surface 2 absorbs $\alpha_2 \epsilon_2 E_{b2}$ or $\alpha_2 \epsilon_1 E_{b1}$ and reflects $\rho_2 \epsilon_1 E_{b1}$ or $(1 - \epsilon_2) \epsilon_1 E_{b1}$ back towards A_1 . the net heat transferred per unit of surface 1 to 2 is the emission $\epsilon_1 E_{b1}$ minus the fraction of $\epsilon_1 E_{b1}$ and $\epsilon_2 E_{b2}$ which is ultimately absorbed by surface 1 after successive reflections. Therefore.

$$\begin{aligned} (Q_{1-2})_{net} &= \{A_1 \epsilon_1 E_{b1} [1 - \epsilon_1(1 - \epsilon_2) - \epsilon_1(1 - \epsilon_1)(1 - \epsilon_2)^2 - \epsilon_1(1 - \epsilon_1)^2(1 - \epsilon_2)^3 - \dots] - \\ &\quad \{A_2 \epsilon_2 E_{b2} [\epsilon_1 + \epsilon_1(1 - \epsilon_1)(1 - \epsilon_2) + \epsilon_1(1 - \epsilon_1)^2(1 - \epsilon_2)^2 + \dots]\} \\ &= A_1 \epsilon_1 E_{b1} \left[1 - \frac{\epsilon_1(1 - \epsilon_2)}{1 - (1 - \epsilon_1)(1 - \epsilon_2)} \right] - A_2 \epsilon_2 E_{b2} \left[1 - \frac{\epsilon_1}{1 - (1 - \epsilon_1)(1 - \epsilon_2)} \right] \\ &= A \frac{\epsilon_1 \epsilon_2}{\epsilon_1 + \epsilon_2 - \epsilon_1 \epsilon_2} [E_{b1} - E_{b2}] \quad \text{since } [A_1 = A_2 = A] \end{aligned}$$

$$(Q_{1-2})_{net} = A \sigma \frac{1}{\frac{1}{\epsilon_1} + \frac{1}{\epsilon_2} - 1} (T_1^4 - T_2^4)$$

$$(Q_{1-2})_{net} = A \sigma F_{1-2} (T_1^4 - T_2^4)$$

$$F_{1-2} = \frac{1}{\frac{1}{\epsilon_1} + \frac{1}{\epsilon_2} - 1}$$

$$(Q_{1-2})_{\text{net}} = A \sigma F_{1-2} (T_1^4 - T_2^4)$$

10.A gas mixture contains 20% CO₂ and 10% H₂O by volume. The total pressure is 2 atm. The temperature of the gas is 927°C. The mean beam length is 0.3 m. Calculate the emissivity of the mixture.

Given : Partial pressure of CO₂, P_{CO₂} = 20% = 0.20 atm

Partial pressure of H₂O, P_{H₂O} = 10% = 0.10 atm.

Total pressure P = 2 atm

Temperature T = 927°C + 273
= 1200 K

Mean beam length L_m = 0.3 m

To find: Emissivity of mixture (ε_{mix}).

Solution: Step: 1

To find emissivity of CO₂

$$P_{\text{CO}_2} \times L_m = 0.2 \times 0.3$$

$$P_{\text{CO}_2} \times L_m = 0.06 \text{ m - atm}$$

From HMT data book, Page No.106, we can find emissivity of CO₂.

From graph, Emissivity of CO₂ = 0.09

$$\epsilon_{\text{CO}_2} = 0.09$$

Step: 2

To find correction factor for CO₂

Total pressure, P = 2 atm

$$P_{\text{CO}_2} L_m = 0.06 \text{ m - atm.}$$

From HMT data book, Page No.107, we can find correction factor for CO₂

From graph, correction factor for CO₂ is 1.25

$$C_{\text{CO}_2} = 1.25$$

$$\varepsilon_{\text{CO}_2} \times C_{\text{CO}_2} = 0.09 \times 1.25$$

$$\varepsilon_{\text{CO}_2} \times C_{\text{CO}_2} = 0.1125$$

Step: 3

To find emissivity of H₂O:

$$P_{\text{H}_2\text{O}} \times L_m = 0.1 \times 0.3$$

$$P_{\text{H}_2\text{O}} L_m = 0.03 \text{ m - atm}$$

From HMT data book, Page No.108, we can find emissivity of H₂O.

From graph Emissivity of H₂O = 0.048

$$\varepsilon_{\text{H}_2\text{O}} = 0.048$$

Step: 4

To find correction factor for H₂O:

$$\frac{P_{\text{H}_2\text{O}} + P}{2} = \frac{0.1 + 2}{2} = 1.05$$

$$\frac{P_{\text{H}_2\text{O}} + P}{2} = 1.05,$$

$$P_{\text{H}_2\text{O}} L_m = 0.03 \text{ m - atm}$$

From HMT data book, Page No.108 we can find emission of H₂O

PART C - 15 Marks (Questions and Answers)

1. Two large parallel plates with $\varepsilon = 0.5$ each, are maintained at different temperatures and are exchanging heat only by radiation. Two equally large radiation shields with surface emissivity 0.05 are introduced in parallel to the plates. find the percentage of reduction in net radiative heat transfer.

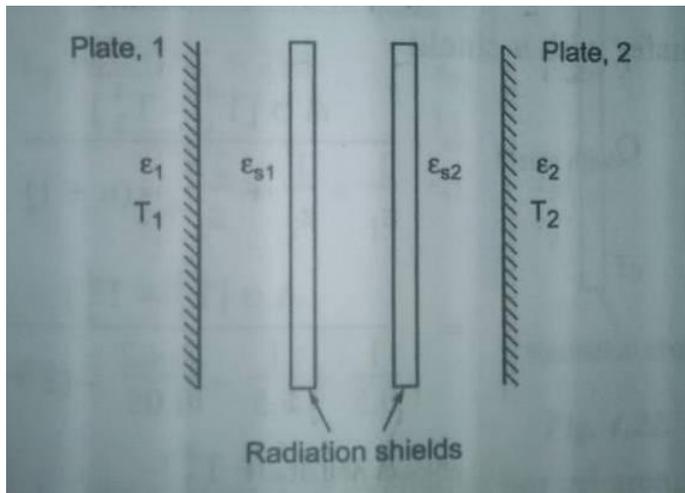
Given:

Emissivity of plate 1, $\varepsilon_1 = 0.5$

Emissivity of plate 2, $\varepsilon_2 = 0.5$

Emissivity of shield, $\varepsilon_s = \varepsilon_{s1} = \varepsilon_{s2} = 0.05$

Number of shields, $n = 2$



To find:

Percentage of reduction in net radiative heat transfer

Solution:

Case 1:

Heat transfer without radiation shield

$$Q_{12} = \epsilon * \sigma * A * [T_1^4 - T_2^4]$$

$$\epsilon = 1 / ((1/\epsilon_1) + (1/\epsilon_2) - 1)$$

$$\epsilon = 1 / ((1/0.5) + (1/0.5) - 1)$$

$$\epsilon = 0.333.$$

$$Q_{12} = \epsilon * \sigma * A * [T_1^4 - T_2^4]$$

$$Q_{12} = 0.333 * \sigma * A * [T_1^4 - T_2^4]$$

CASE 2: Heat transfer with radiation shield

$$\begin{aligned} Q_{\text{with shield}} &= (\sigma * A * [T_1^4 - T_2^4]) / ((1/\epsilon_1) + (1/\epsilon_2) + (2n/\epsilon_s) - (n+1)) \\ &= (\sigma * A * [T_1^4 - T_2^4]) / ((1/0.5) + (1/0.5) + ((2*2)/0.05) - (2+1)) \\ &= (\sigma * A * [T_1^4 - T_2^4]) / 81 \end{aligned}$$

$$Q_{\text{with shield}} = 0.0123 * (\sigma * A * [T_1^4 - T_2^4])$$

We know that

Radiation in heat transfer due to radiation shield

$$\begin{aligned} &= (Q_{\text{WITHOUT SHIELD}} - Q_{\text{WITH SHIELD}}) / Q_{\text{WITHOUT SHIELD}} \\ &= \frac{ ((0.333 * \sigma * A * [T_1^4 - T_2^4]) - (0.0123 * (\sigma * A * [T_1^4 - T_2^4]))) }{ (0.333 * \sigma * A * [T_1^4 - T_2^4]) } \\ &= 0.963 \end{aligned}$$

= 96.3 %

Percentage of reduction in net radiative heat transfer = 96.3 .

2. A black body at 3000 K emits radiation Calculate the following

1. Monochromatic emissive power at 1 μm wave length

2. Wave length at which emission is maximum

3. Maximum emissive power

4. Total emissive power

5. Calculate the total emissive of the furnace if it is assumed as a real surface having emissivity equal to 0.85

Given

Surface temperature T = 3000K

To find

1. Monochromatic emissive power $E_{b\lambda}$ at $\lambda=1 \mu = 1 \times 10^{-6}m$

2. Maximum wave length, (λ_{max})

3. Maximum emissive power $(E_{b\lambda})_{max}$

4. Total emissive power, E_b

5. Emissive power of real surface at $\epsilon=0.85$

Solution

1. Monochromatic emissive power

From Planck's distribution law, we know that

$$E_{b\lambda} = \frac{c_1 \lambda^{-5}}{c_2 e^{\lambda T - 1}}$$

$$C_1 = 0.374 \times 10^{-15} \text{ Wm}^2$$

$$C_2 = 14.4 \times 10^{-3} \text{ mK}$$

$$\lambda = 1 \times 10^{-6} \text{ m}$$

$$E_{b\lambda} = \frac{0374 \times 10^{-15} [1 \times 10^{-6}]^{-5}}{\left[\frac{14.4 \times 10^{-3}}{1 \times 10^{-6} \times 3000} \right] - 1}$$

$$E_{b\lambda} = 3.10 \times 10^{12} \text{ W/m}^2$$

2. Mamimum wave length (λ_{max})

$$\lambda_{max} T = 2.9 \times 10^{-3} \text{ mK}$$

$$\lambda_{\max} = \frac{2.9 \times 10^{-3}}{3000}$$

$$\lambda_{\max} = 0.966 \times 10^{-6} \text{ m}$$

3. Maximum emissive power $(E_{b\lambda})_{\max}$

$$\begin{aligned} (E_{b\lambda})_{\max} &= 1.307 \times 10^{-5} T^5 \\ &= 1.307 \times 10^{-5} \times (3000)^5 \end{aligned}$$

$$(E_{b\lambda})_{\max} = 3.17 \times 10^{12} \text{ W/m}^2$$

4. Total emissive power E_b

$$E_b = \sigma \times T^4 \text{ (From HMT data book P.No 8)}$$

$$\sigma = \text{Stefen Boltzman Constant}$$

$$= 5.67 \times 10^{-8} \text{ W/m}^2 \text{ K}^4$$

$$E_b = (5.67 \times 10^{-8}) \times (3000)^4$$

$$E_b = 4.59 \times 10^6 \text{ W/m}^2$$

5. Total emissive power of real surface

$$(E_b)_{\text{real}} = \epsilon \sigma T^4$$

$$\epsilon - \text{Emissivity} = 0.85$$

$$(E_b)_{\text{real}} = 0.85 \times 5.67 \times 10^{-8} \times (3000)^4$$

$$(E_b)_{\text{real}} = 3.90 \times 10^6 \text{ W/m}^2$$

Result

1. $E_{b\lambda} = 3.10 \times 10^{12} \text{ W/m}^2$ 2. $\lambda_{\max} = 0.966 \times 10^{-6} \mu\text{m}$

3. $(E_{b\lambda})_{\max} = 3.17 \times 10^{12} \text{ W/m}^2$ 4. $(E_b)_{\text{real}} = 3.90 \times 10^6 \text{ W/m}^2$

UNIT: V MASS TRANSFER

PART A - 2 Marks (Questions and Answers)

1. What is mass transfer?

The process of transfer of mass as a result of the species concentration difference in a mixture is known as mass transfer.

2. Give the examples of mass transfer.

Some examples of mass transfer.

1. Humidification of air in cooling tower
2. Evaporation of petrol in the carburettor of an IC engine.
3. The transfer of water vapour into dry air.

3. What are the modes of mass transfer? (Nov/Dec 2010)(Nov/Dec 2104)

There are basically two modes of mass transfer,

1. Diffusion mass transfer
2. Convective mass transfer

4. What is molecular diffusion?

The transport of water on a microscopic level as a result of diffusion from a region of higher concentration to a region of lower concentration in a mixture of liquids or gases is known as molecular diffusion.

5. What is Eddy diffusion?

When one of the diffusion fluids is in turbulent motion, eddy diffusion takes place.

6. What is convective mass transfer? (May/June 2006)

Convective mass transfer is a process of mass transfer that will occur between surface and a fluid medium when they are at different concentration.

7. State Fick's law of diffusion. (April/May 2012) (NOV-DEC 14)(Nov/Dec 16)

The diffusion rate is given by the Fick's law, which states that molar flux of an element per unit area is directly proportional to concentration gradient.

$$\frac{ma}{A} = -D_{ab} \frac{dC_a}{dx}$$

Where,

$$\frac{ma}{A} - \text{Molar flux, } \frac{\text{kg} - \text{mole}}{\text{s} - \text{m}^2}$$

Dab- Diffusion coefficient of species a and b, m²/s

$$\frac{dCa}{dx} - \text{Concentration gradient, kg/m}^3$$

8. What is free convective mass transfer?

If the fluid motion is produced due to change in density resulting from concentration gradients, the mode of mass transfer is said to be free or natural convective mass transfer.

Example: Evaporation of alcohol.

9. Define forced convective mass transfer.

If the fluid motion is artificially created by means of an external force like a blower or fan, that type of mass transfer is known as convective mass transfer.

Example: The evaporation of water from an ocean when air blows over it.

10. Define Schmidt and Lewis number. What is the physical significance of each? (NOV/DEC 13)

The dimensionless Schmidt number is defined as the ratio of momentum diffusivity to mass diffusivity $Sc = \nu/DAB$, and it represents the relative magnitudes of momentum and mass diffusion at molecular level in the velocity and concentration boundary layers, respectively. The Schmidt number diffusivity corresponds to the Prandtl number in heat transfer. A Schmidt number of unity indicates that momentum and mass transfer by diffusion are comparable, and velocity and concentration boundary layers almost coincide with each other.

The dimensionless Lewis number is defined as the ratio of thermal diffusivity to mass diffusivity $Le = \alpha / DAB$ and it represents the relative magnitudes of heat and mass diffusion at molecular level in the thermal and concentration boundary layers, respectively. A Lewis number of unity indicates that heat and mass diffuse at the same rate, and the thermal and concentration boundary layers coincide.

11. Define Sherwood Number. (April/May 2012)

It is defined as the ratio of concentration gradients at the boundary.

$$Sc = \frac{hmX}{D_{ab}}$$

hm- Mass transfer coefficient, m/s

D_{ab}-Diffusion coefficient, m²/s

X- length, m

12. What is mass average velocity?(May/June 2010)

The bulk velocity of mixture, in which different components may have different mobilities, is compared either on mass average. If a fluid mixture of two components A and B if u_A and u_B are the mean velocities then the average velocity is

$$u_{\text{mass}} = (\rho_A u_A + \rho_B u_B) / (\rho_A + \rho_B)$$

13. Distinguish between mass concentration and molar concentration (April/May 2017)

Mass Concentration

Mass of a component per unit volume of the mixture. It is expressed in kg/m³

$$\text{Mass concentration} = \frac{\text{Mass of a component}}{\text{Unit volume of mixture}}$$

Molar concentration

Number of molecules of a component per unit volume of the mixture. It is expressed in Kg - mole /m³

$$\text{Molar concentration} = \frac{\text{Number of moles of component}}{\text{Unit volume of mixture}}$$

14. Define Schmidt number and state its physical significance.) (Nov/Dec 16)

Schmidt number (Sc) is a dimensionless number defined as the ratio of momentum diffusivity (viscosity) and mass diffusivity, and is used to characterize fluid flows in which there are simultaneous momentum and mass diffusion convection processes.

Significance:

Analogous of Prandtl number in Heat Transfer. Used in fluid flows in which there is simultaneous momentum & mass diffusion. It is also ratio of fluid boundary layer to mass transfer boundary layer thickness.

PART B - 13 Marks (Questions and Answers)

1. A vessel contains binary mixture of O₂ and N₂ with partial pressure in the ratio 0.21 and 0.79 at 15°C =. The total pressure of the mixture is 1.1 bar. Calculate the following.

- I. Molar concentrations
- II. Mass densities
- III. Mass fractions
- IV. Molar fraction of each species.

[APRIL/MAY 2014; NOV/DEC 2015]

Given:

Partial pressure of O₂ = 0.21 x total pressure

$$(P_{O_2}) = 0.21 \times 1.1$$

$$P_{O_2} = 0.231 \times 10^5 \text{ N/m}^2$$

So partial pressure of N₂ = P_{N₂} = 86.9 x 10³ N/m²

Temperature T = 15° C = 288 K

To find

- I. Molar concentrations, C_{O₂}, C_{N₂}
- II. Mass densities, ρ_{O₂}, ρ_{N₂}
- III. Mass fractions, ṁ_{O₂}, ṁ_{N₂}
- IV. Molar fraction of each species. x_{O₂}, x_{N₂}

Solution:

STEP-1

$$\text{Molar concentration, } c_{O_2} = \frac{p_{O_2}}{GT}$$

Universal Gas Constant, G = 8314 J/kg mole K

$$C_{O_2} = \frac{0.231 \times 10^5}{8314 \times 288}$$

$$C_{O_2} = 9.64 \times 10^{-3} \text{ kg - mole / m}^3$$

$$C_{N_2} = \frac{P_{N_2}}{GT}$$

$$C_{N_2} = \frac{86.9 \times 10^3}{8314 \times 288}$$

$$C_{N_2} = 0.036 \text{ kg-mole/m}^3$$

Total concentration,

$$C = C_{O_2} + C_{N_2} = 0.045 \text{ kg mole/m}^3$$

STEP-2

Molar concentration

$$C = \frac{\rho}{\mu}$$

$$\Rightarrow \rho_{O_2} = C_{O_2} \times \mu_{O_2}$$

$$= 9.64 \times 10^{-3} \times 32$$

$$\rho_{O_2} = 0.308 \text{ kg/m}^3$$

$$= 0.0362 \times 28$$

$$\rho_{N_2} = 1.013 \text{ kg/m}^3$$

Overall density, $\rho = \rho_{O_2} + \rho_{N_2}$

$$= 0.308 + 1.0136$$

$$\rho = 1.3216 \text{ kg/m}^3$$

STEP-3

Mass fractions $\dot{m}_{O_2} = \frac{\rho_{O_2}}{\rho} = \frac{0.308}{1.3216}$

$$\dot{m}_{O_2} = 0.233$$

$$\dot{m}_{N_2} = \frac{\rho_{N_2}}{\rho} = \frac{1.0136}{1.3216}$$

$$\dot{m}_{N_2} = 0.766$$

STEP-4

Mole fractions, $X_{O_2} = \frac{C_{O_2}}{C} = \frac{9.64 \times 10^{-3}}{0.045}$

$$X_{O_2} = 0.210$$

$$X_{N_2} = \frac{C_{N_2}}{C} = \frac{0.0362}{0.045}$$

$$X_{N_2} = 0.804$$

RESULT:

I. Molar concentrations, $C_{O_2} = 9.64 \times 10^{-3} \text{ kg-mole/m}^3$

$$C_{N_2} = C_{N_2} = 0.036 \text{ kg-mole/m}^3$$

II. Mass densities, $\rho_{O_2} = 0.308 \text{ kg/m}^3$

$$\rho_{N_2} = 1.013 \text{ kg/m}^3$$

III. Mass fractions, $\dot{m}_{O_2} = 0.233$

$$\dot{m}_{N_2} = 0.766$$

IV. Molar fraction of each species. $X_{O_2} = 0.210$

$$X_{N_2} = 0.804$$

2. Air at 20°C ($\rho = 1.205 \text{ kg/m}^3$; $\nu = 15.06 \times 10^{-6} \text{ m}^2/\text{s}$; $D = 4.16 \times 10^{-6} \text{ m}^2/\text{s}$) flows over a tray (length = 32 cm, width = 42 cm) full of water with a velocity of 2.8 m/s. The total pressure of moving air is 1 atm and the partial pressure of water present in the air is 0.00658 bar. If the temperature on the water surface is 15°C calculate the evaporation rate of water.

(MAY/JUNE 2012; NOV/DEC 2014; NOV/DEC 2015; APRIL/MAY 2016)

Given:

Fluid temperature, $T_\infty = 20^\circ\text{C}$

Speed, $U = 2.8 \text{ m/s}$

Flow direction is 32 cm side. So, $x = 32 \text{ cm} = 0.32 \text{ m}$

Area, $A = 32 \text{ cm} \times 42 \text{ cm} = 0.32 \times 0.42 \text{ m}^2$

Partial pressure of water, $P_{w_2} = 0.0068 \text{ bar}$

$$P_{w_2} = 0.0068 \times 10^5 \text{ N/m}^2$$

Water surface temperature, $T_w = 15^\circ\text{C}$

To find:

Evaporation rate of water (M_w)

Solution:

Properties of air is given

$\rho = 1.205 \text{ kg/m}^3$;

$\nu = 15.06 \times 10^{-6} \text{ m}^2/\text{s}$;

Diffusion coefficient $D = 4.16 \times 10^{-6} \text{ m}^2/\text{s}$

STEP-1

$$\text{Re} = \frac{UL}{\nu} = \frac{2.8 \times 0.32}{15.06 \times 10^{-6}}$$

$$= 0.594 \times 10^5 < 5 \times 10^5$$

Since $\text{Re} < 5 \times 10^5$, flow is laminar

Flat plate laminar flow:

Sherwood number (Sh) = $[0.664 (\text{Re})^{0.5} (\text{Sc})^{0.333}] \dots\dots\dots \{1\}$

[From HMT data book, P.no-175]

STEP-2

$\text{Sc} \rightarrow$ Schmidt number = $\frac{\nu}{D_{ab}} = \frac{15.06 \times 10^{-6}}{4.16 \times 10^{-5}}$

Sc = 0.3620

Sub Sc, Re in {1}

(Sh) = $[0.664 (0.594 \times 10^5)^{0.5} (0.3620)^{0.333}]$

Sh = 115.37

STEP-3

Sherwood number $\text{Sh} = \frac{h_m L}{D_{ab}}$

$115.37 = \frac{h_m \cdot 0.32}{4.16 \times 10^{-5}}$

$h_m = 0.0149 \text{ m/s}$

STEP-4

Mass transfer coefficient based on pressure difference is given

$h_{mp} = \frac{h_m}{RT_w} = \frac{0.0149}{287 \times 288} \quad [T_w = 15^\circ\text{C} + 273 = 288 \text{ K, So } R = 287 \text{ J/kg K}]$

$h_{mp} = 1.80 \times 10^{-7} \text{ m/s}$

Saturation pressure of water at 15°C

$P_{w1} = 0.017 \text{ bar}$

$P_{w1} = 0.017 \times 10^5 \text{ N/m}^2 \quad [\text{From steam table (R.S khurmi) P.no-1}]$

STEP-5

The evaporation of water

$$M_w = h_{mp} \times A (P_{w1} - P_{w2})$$

$$M_w = 2.66 \times 10^{-5} \text{ kg/s}$$

Result:

The evaporation rate of water $M_w = 2.66 \times 10^{-5} \text{ kg/s}$

3. Dry air at 27°C and 1 atm flows over a wet flat plate 50 cm long at a velocity of 50 m/s. Calculate the mass transfer coefficient of water vapour in air at the end of the plate.

(NOV/DEC 2014; APRIL/MAY 2015) (NOV/DEC 2013)

Given:

Fluid temperature $T_\infty = 27^\circ\text{C}$

Velocity $u = 50 \text{ m/s}$

Length $x = 35 \text{ mm} = 0.035 \text{ m}$

To find:

Mass transfer co-efficient, (h_m)

Solution:

STEP-1

Properties of air at 27°C:

$$\nu = 16 \times 10^{-6} \text{ m}^2/\text{s}$$

$$\text{Re} = \frac{UL}{\nu} = \frac{50 \times 0.035}{16 \times 10^{-6}}$$

$$= 1.09375 \times 10^5 < 5 \times 10^5$$

Since $\text{Re} < 5 \times 10^5$, flow is laminar

Flat plate laminar flow:

$$\text{Sherwood number (Sh)} = [0.664 (\text{Re})^{0.5} (\text{Sc})^{0.333}] \dots \dots \dots \{1\}$$

[From HMT data book, P.no-175]

STEP-2

$$[D_{ab} - \text{Diffusion coefficient (water+ air) @ } 27^\circ\text{C} = 25.38 \times 10^{-6} \text{ m}^2/\text{s}]$$

$$Sc \rightarrow \text{Schmidt number} = \frac{\nu}{D_{ab}} = \frac{16 \times 10^{-6}}{25.38 \times 10^{-6}}$$

$$Sc = 0.6304$$

STEP-3

Sub Sc, Re in {1}

$$(Sh) = [0.664 (1.09375 \times 10^5)^{0.5} (0.6304)^{0.333}]$$

$$Sh = 188.32$$

STEP-4

$$\text{Sherwood number } Sh = \frac{h_m L}{D_{ab}}$$

$$188.32 = \frac{h_m \cdot 0.35}{25.38 \times 10^{-6}}$$

$$h_m = 0.1365 \text{ m/s}$$

Result:

Mass transfer coefficient of water vapour $h_m = 0.1365 \text{ m/s}$.

4. CO_2 and air experience equimolar counter diffusion in a circular tube whose length and diameter are 1 m and 50 mm respectively. The system of total pressure of 1 atm and a temperature of 25°C . The ends of the tube are connected to large chambers in which the species concentrations are maintained at fixed values. the partial pressure of CO_2 at one end is 190 mm of Hg while at the other end is 95 mm Hg. Estimate the mass transfer rate of CO_2 and air through the tube.

[MAY/JUNE 2012; APRIL/MAY 2016]

Given:

Diameter, $d = 50 \text{ mm} = 0.05 \text{ m}$

Length = 1 m $[x_2 - x_1]$

Total pressure, $p = 1 \text{ atm} = 1 \text{ bar}$

Temperature, $T = 25^\circ\text{C} = 298 \text{ K}$

Partial pressure of CO_2 at one end

$$P_{a1} = 190 \text{ mm of Hg} = \frac{190}{760} \text{ bar}$$

$$P_{a1} = 0.25 \text{ bar} \quad [1 \text{ bar} = 760 \text{ mm of Hg}]$$

$$P_{a1} = 0.25 \times 10^5 \text{ N/m}^2 \quad [1 \text{ bar} = 10^5 \text{ N/mm}^2]$$

Partial pressure of CO₂ at other end

$$P_{a2} = 95 \text{ mm of Hg} = \frac{95}{760} \text{ bar}$$

$$P_{a2} = 0.0312 \text{ bar} \quad [1 \text{ bar} = 760 \text{ mm of Hg}]$$

$$P_{a2} = 0.0312 \times 10^5 \text{ N/m}^2 \quad [1 \text{ bar} = 10^5 \text{ N/mm}^2]$$

To find:

1. Mass transfer rate of Co₂
2. Mass transfer rate of air

Solution:

STEP-1

$$\frac{m_a}{A} = \frac{D_{ab} [C_{a1} - C_{a2}]}{GT [X_2 - X_1]}$$

Diffusion coefficient (D_{ab}) for CO₂-Air combination is 11.89 × 10⁻⁶ m²/s

[HMT data book page no.180]

$$G = \text{Universal gas constant} = 8314 \frac{\text{J}}{\text{kg-mole-K}} \quad (\text{J/kg-mole-K})$$

$$A = \text{Area} = \frac{\pi}{4} (d)^2$$

$$A = 1.9634 \times 10^{-3} \text{ m}^2$$

$$\frac{m_a}{A} = \frac{D_{ab} [C_{a1} - C_{a2}]}{GT [X_2 - X_1]}$$

$$\frac{m_a}{A} = \frac{11.89 \times 10^{-6} [0.25 \times 10^5 - 0.031 \times 10^5]}{8314 \times 298} \quad [1]$$

$$\text{Molar transfer rate of Co}_2, m_a = 1.050 \times 10^{-7} \frac{\text{kg-mole}}{\text{s}}$$

STEP-2

We know,

Mass Transfer Rate Co₂ = Molar Transfer x Molecular Weight

$$= 1.050 \times 10^{-7} \times 44.01$$

[Molecular weight of Co₂ Refer HMT D.B Page 182]

$$\text{Mass Transfer Rate Co}_2 = 4.625 \times 10^{-6} \text{ kg/s}$$

$$\text{Mass Transfer Rate of Air} = m_b = -1.050 \times 10^{-7} \frac{\text{kg} - \text{mole}}{\text{s}}$$

STEP-3

$$[m_a = -m_b]$$

$$\begin{aligned} \text{Mass Transfer Rate Air} &= \text{Molar Transfer} \times \text{Molecular Weight of air} \\ &= 1.050 \times 10^{-7} \times 29 \end{aligned}$$

$$\text{Mass Transfer Rate Air} = -3.045 \times 10^{-6} \text{ kg/s}$$

Result:

1. Mass transfer rate of Co₂ = **4.625 x 10⁻⁶ kg/s**

2. Mass transfer rate of air = **-3.045 x 10⁻⁶ kg/s**

5. Discuss briefly the Analogy between heat and mass transfer.

[MAY/JUNE 2013; NOV/DEC 2015; APRIL/MAY 2016]

There is similarity among heat and mass transfer. The three basic equations dealing with these are

- I. Newtonian equation of momentum
- II. Fourier law of heat transfer
- III. Fick law of mass transfer

The momentum, heat and mass transfer equation can be written as

$$\text{Continuity equation, } \frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} = 0$$

$$\text{Momentum transfer, } u \frac{\partial u}{\partial x} + v \frac{\partial v}{\partial y} = \nu \frac{\partial^2 u}{\partial y^2}$$

$$\text{Heat transfer, } u \frac{\partial T}{\partial x} + v \frac{\partial T}{\partial y} = \alpha \frac{\partial^2 T}{\partial y^2}$$

$$\text{Mass transfer, } u \frac{\partial C_a}{\partial x} + v \frac{\partial C_n}{\partial y} = D \frac{\partial^2 C_a}{\partial y^2}$$

Heat and mass transfer takes place due to temperature difference. As per Fourier's law of conduction

$$Q = -kA \frac{dt}{dx}$$

Where Q= rate of heat transfer

K= thermal conductivity of material

A= Heat transfer area

$$\frac{dt}{dx} = \text{Temperature gradient}$$

As per Newton's law of cooling

$$Q = hA\Delta T$$

Where h= heat transfer coefficient

Mass transfer takes place due to concentration difference.

As per Fick's law of diffusion

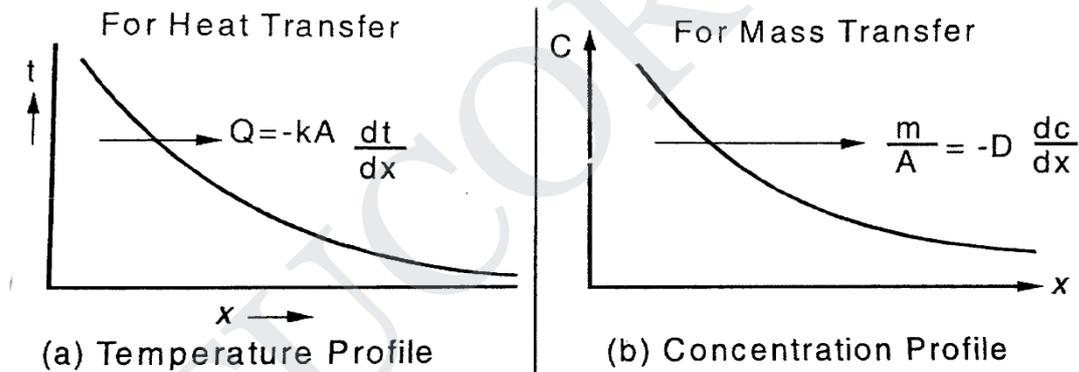
$$N_A = \frac{m_A}{A} = -D_{AB} \frac{dC_A}{dx}$$

m_A = Mass flow rate of species A by diffusion.

A = Area through which mass is flowing

D_{AB} = Diffusion coefficient.

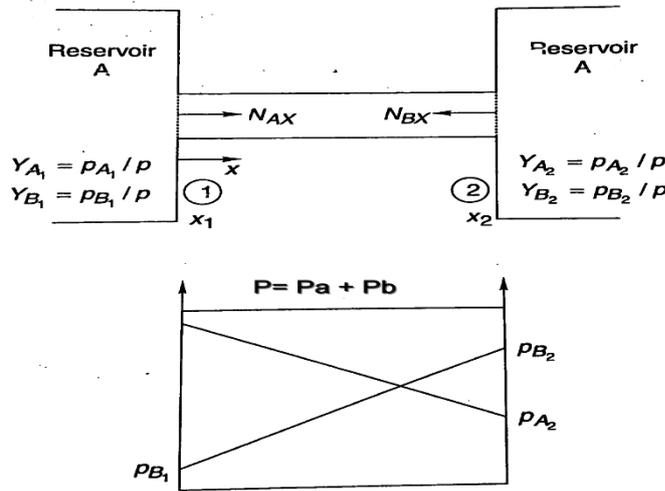
$\frac{dC_A}{dx}$ = concentration gradient.



6. Explain Equimolar Counter diffusion in gases.

[APRIL/MAY 2013; NOV/DEC 2014]

Two large chambers 'a' and 'b' connected by a passage as shown below.



Equimolar Counter Diffusion in a Binary Mixture

N_a and N_b are the steady state molar diffusion rates of component a and b respectively.

Equimolar diffusion is defined as each molecule of 'a' is replaced by each molecule of 'b' and vice versa. The total pressure $p = p_a + p_b$ is uniform throughout the system.

$$P = P_a + P_b$$

Differentiating with respect to x

$$\frac{dP}{dx} = \frac{dP_a}{dx} + \frac{dP_b}{dx}$$

Since the total pressure of the system remains constant under steady state conditions

$$\frac{dP}{dx} = \frac{dP_a}{dx} + \frac{dP_b}{dx} = 0$$

$$\frac{dP_a}{dx} = -\frac{dP_b}{dx}$$

Let the total molar flux is zero, $N_a + N_b = 0$

$$\rightarrow N_a = -N_b$$

From flick's law,

$$-D_{BA} \frac{A}{GT} \frac{dP_a}{dx} = D_{BA} \frac{A}{GT} \frac{dP_b}{dx}$$

$$D_{AB} = D_{BA} = D$$

$$N_a = \frac{ma}{A} = -D \frac{A}{GT} \int_1^2 \frac{dP_a}{dx}$$

Molar flux,
$$N_a = \frac{ma}{A} = -D \frac{A}{GT} \left[\frac{Pa_1 - Pa_2}{x_2 - x_1} \right]$$

Similarly,

$$N_b = \frac{mb}{A} = -D \frac{A}{GT} \left[\frac{Pb_1 - Pb_2}{x_2 - x_1} \right]$$

Where,

$$\frac{ma}{A} - \text{Molar flux} \frac{\text{kg} - \text{mole}}{\text{s} - \text{m}^2}$$

D- Diffusion coefficient

$$G - \text{Universal constant} - 8314 \frac{\text{J}}{\text{kg} - \text{mole} - \text{K}}$$

A- Area - m²

Pa₁- Partial pressure of constituent at 1 in N/m²

Pa₂- Partial pressure of constituent at 2 in N/m²

T - Temperature - K

- 7. An open pan of 150 mm diameter and 75 mm deep contains water at 25°C and is exposed to atmosphere air at 25°C and 50°C R.H. calculate the evaporation rate of water in grams per hour.**

[APRIL/MAY 2002]

Given:

Diameter, d = 150 mm = 0.150 m

Deep, (x₂-x₁) = 75 mm = 0.075 m

Temperature, T = 25°C+273 = 298 K

Relative Humidity = 50%

To Find:

Evaporation rate of water in grams per hour.

Solution:

Diffusion co-efficient (D_{ab}) [water + air] at 25°C

[From HMT data book, page no, 180]

$$D_{ab} = 25.83 \times 10^{-6} \text{ m}^2/\text{s}$$

STEP-1

We know that, for isothermal evaporation,

$$\text{Molar flux, } \frac{ma}{A} = \frac{D_{ab}}{GT} \frac{p}{(x_2 - x_1)} \ln \left[\frac{P - P_{w2}}{P - P_{w1}} \right]$$

$$\text{Area, } A = 0.0176 \text{ m}^2$$

$$P = \text{Total pressure} = 1 \text{ atm} = 1.013 \times 10^5 \text{ N/mm}^2$$

P_{w1} = Partial pressure at the bottom of the test tube corresponding to saturation temperature 25°C.

At 25°C

$$\rightarrow P_{w1} = 0.03166 \times 10^5 \text{ N/mm}^2 \quad [\text{From steam table p.no 2}]$$

P_{w2} = Partial pressure at the top of the pan corresponding to 25°C and 50% relative humidity.

At 25°C

$$\rightarrow P_{w2} = 0.03166 \times 10^5$$

$$\text{R.H} = 50\% = 0.50$$

$$P_{w2} = 0.03166 \times 10^5 \times 0.50$$

$$\rightarrow P_{w2} = 1583 \text{ N/m}^2$$

STEP-2	$\frac{ma}{0.076} = \frac{25.83 \times 10^{-6}}{8314 \times 298} \times \frac{1.013 \times 10^5}{0.075} \ln \left[\frac{1.013 \times 10^5 - 1583}{1.013 \times 10^5 - 0.03166 \times 10^5} \right]$
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$$\text{Molar rate of water vapour, } ma = 3.96 \times 10^{-9} \frac{\text{kg-mole}}{\text{s}}$$

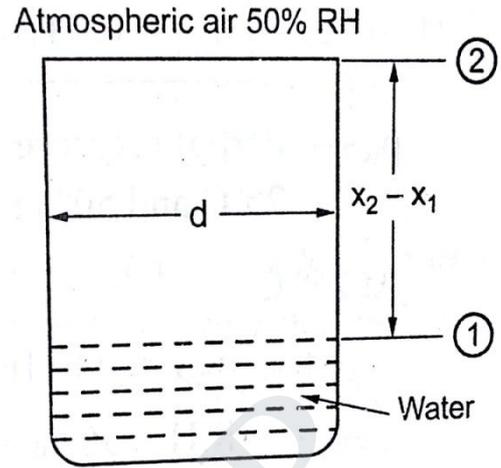
STEP-3

Mass rate of water vapour = molar rate of water vapour X molecular weight of steam

$$= 3.96 \times 10^{-9} \times 18.016 \text{ kg/s}$$

$$\text{Mass rate of water vapour} = 0.256 \text{ g/h}$$

Result:



Evaporation rate of water = 0.256 g/h.

Evaporation rate of water = 0.256 g/h.

PART C - 15 Marks (Questions and Answers)

1. Two large tanks, maintained at the same temperature and pressure are connected by a circular 0.15m diameter duct, which is 3 m length. One tank contains a uniform mixture of 60 mole % ammonia and 40 mole % air and other tank contains a uniform mixture of 20 mole % ammonia and 80 mole % air. The system is at 273 K and 1.013×10^5 Pa. Determine the rate of ammonia transfer between the two tanks. Assuming a steady state mass transfer.

Given:

Diameter $d = 0.15$ m

Length $(x_2 - x_1) = 3$ m

$P_{a1} = \frac{60}{40} = 0.6$ bar = 0.6×10^5 N/m²

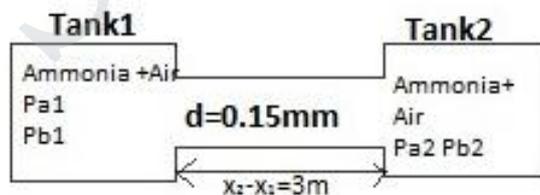
$P_{b1} = \frac{40}{40} = 0.4$ bar = 0.4×10^5 N/m²

$P_{a2} = \frac{20}{40} = 0.2$ bar = 0.2×10^5 N/m²

$P_{b2} = \frac{80}{40} = 0.8$ bar = 0.8×10^5 N/m²

$T = 273$ K

$P = 1.013 \times 10^5$ N/m²



a-Ammonia

b-Air

To find

Rate of ammonia transfer

Solution:

Equimolar counter diffusion

Molar flux,

$$\frac{ma}{A} = \frac{Dab}{GT} \left[\frac{Pa1 - Pa2}{X2 - X1} \right]$$

Where G -universal constant =8314 J/Kg-mole-K

$$A = \text{area} = \frac{\pi}{4} d^2$$

$$A = \frac{\pi}{4} (0.15)^2$$

$$A = 0.017 \text{ m}^2$$

Dab-Diffusion co efficient of ammonia with air = $21.6 \times 10^{-6} \text{ m}^2/\text{s}$
(From HMT data book P.No 180 (sixth edition))

$$Dab = 21.6 \times 10^{-6} \text{ m}^2/\text{s}$$

$$(1) = \frac{ma}{0.017} = \frac{21.6 \times 10^{-6}}{8314 \times 273} \times \frac{0.6 \times 10^5 - 0.2 \times 10^5}{3}$$

$$\text{Molar transfer rate of ammonia, } m_a = 2.15 \times 10^{-9} \text{ Kg-mole/s}$$

Mass transfer rate of ammonia = Molar transfer rate of ammonia x Molecular weight of ammonia

$$= 2.15 \times 10^{-9} \times 17.03 \text{ (Refer HMT data book P.No 182)}$$

$$\text{Mass transfer rate of ammonia} = 3.66 \times 10^{-8} \text{ Kg/s}$$

Result

$$\text{Mass transfer rate of ammonia} = 3.66 \times 10^{-8} \text{ Kg/s}$$

2. An open pan 20cm in diameter and 8 cm deep contains water at 25°C and is exposed to dry atmospheric air. If the rate of diffusion of water vapour is $8.5 \times 10^{-4} \text{ kg/h}$, estimate the diffusion co efficient of water in air.

Given :

$$\text{Diameter } d = 20 \text{ cm} = 0.20 \text{ m}$$

$$\text{Length } (x_2 - x_1) = 8 \text{ cm} = 0.08 \text{ m}$$

$$\text{Temperature } T = 25^\circ\text{C} + 273 = 298 \text{ K}$$

Diffusion rate (or)

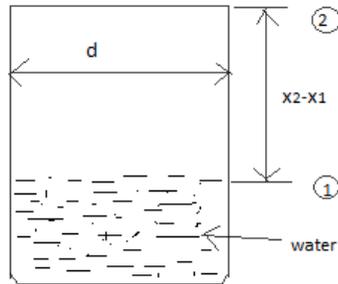
$$\text{Mass rate of water vapour} = 8.54 \times 10^{-4} \text{ kg/h}$$

$$= \frac{8.54 \times 10^{-4} \text{ kg}}{3600 \text{ s}}$$

$$= 2.37 \times 10^{-7} \text{ kg/s}$$

To find

Diffusion coefficient D_{ab}



Solution

Molar rate of water vapour

$$\frac{m_a}{A} = \frac{D_{ab}}{GT} \frac{p}{x_2 - x_1} \times \ln \left[\frac{p - p_{w2}}{p - p_{w1}} \right]$$

$$m_a = \frac{D_{ab} \times A}{GT} \frac{p}{x_2 - x_1} \times \ln \left[\frac{p - p_{w2}}{p - p_{w1}} \right]$$

We know that

Mass transfer rate of steam = Molar transfer rate of steam x Molecular weight of steam

$$2.37 \times 10^{-7} = \frac{D_{ab} \times A}{GT} \frac{p}{x_2 - x_1} \times \ln \left[\frac{p - p_{w2}}{p - p_{w1}} \right] \times 18.016$$

Where

$$\text{Area } A = \frac{\pi}{4} d^2$$

$$= \frac{\pi}{4} (0.20)^2$$

$$A = 0.0314 \text{ m}^2$$

G - universal constant = 8314 J/Kg-mole-K

P - Total Pressure = 1 atm = 1.013 bar = 1.013 x 10⁵ N/m²

P_{w1}- Partial pressure at the bottom of the test tube corresponding to saturation temperature 25°C

At 25°C (From Rs Khurmi Steam table P.No 2)

$$P_{w1} = 0.03166 \times 10^5 \text{ N/m}^2$$

P_{w2}- Partial pressure at the top of the pan Hence air is dry and there is no water vapour So, P_{w2} = 0

$$P_{w2} = 0$$

(1) =

$$= \frac{2.37 \times 10^{-7}}{8314 \times 298} \times \frac{1.013 \times 10^5}{0.08} \times \ln \left[\frac{1.013 \times 10^5 - 0}{1.013 \times 10^5 - 0.03166 \times 10^5} \right] \times 18.016$$

$$D_{ab} = 2.58 \times 10^{-5} \text{ m}^2/\text{s}$$

Result

Diffusion coefficient, D_{ab} = 2.58 x 10⁻⁵ m²/s

STUCOR APP

B.E/B.Tech DEGREE EXAMINATION, APR/MAY 2017.

Fifth Semester

Mechanical Engineering

ME 6502 – HEAT AND MASS TRANSFER

(Regulation 2013)

Time: Three hours

Maximum: 100 marks

Answer ALL questions

PART A – (10 x 2= 20 marks)

1. Distinguish between conduction and convection heat transfer
2. State some practical applications of transient heat transfer analysis.
3. How Reynolds and Colburn analogies differ with each other.
4. Define Grashof number and explain its significance in free convection heat transfer. Refer page No.36, Question No.4
5. Draw the pool boiling curve for water
6. Sketch the temperature variation of condenser and evaporator
7. State Lamberts cosine law for radiation Refer page No.93, Question No.13
8. What are the application of radiation shields? Refer page No.94, Question No.16
9. Distinguish between mass concentration and molar concentration. Refer page No.121, Question No.13
10. Give examples for free and forced convection mass transfers.

PART B- (5 x 16= 80 marks)

- 11.A) i) Steam at 320°C flows in a cast pipe ($k=80 \text{ W/mK}$) whose inner and outer diameter are $D_1=5\text{cm}$ and $D_2 = 5.5\text{cm}$, respectively. The pipe is covered with 3-cm thick glass wool insulation with $k=0.05 \text{ W/mK}$. Heat is lost to the surroundings at 5°C by natural convection and radiation, with a combined heat transfer coefficient of $h_2= 18 \text{ W/m}^2\text{K}$. Taking the heat transfer coefficient inside the pipe to be $h_1= 60 \text{ W/m}^2\text{K}$. Determine the rate of heat loss from the steam per unit length of the pipe. Also determine the temperature drops across the pipe shell and the insulation. (10)

Refer page No.20, Question No.5 (similarity)

- (ii) Write short notes on types of extended surfaces or fins (6)

(Or)

- B) i) Circumferential aluminium fins ($k= 200 \text{ W/mK}$) of rectangular profile (1.5cm x 1cm thick) are fitted onto a 2.5cm diameter tube. The fin base temperature is 170°C and the ambient air temperature is 25°C . Estimate the heat loss per fin. The heat transfer coefficient "h" may be taken as $130 \text{ W/m}^2\text{K}$. Refer page No.25, Question No.8 (similarity) (8)

- (ii) The ground at a particular location is covered with snow pack at -10°C for a continuous period of three months and the average soil properties at that location are $k=0.4 \text{ W/mK}$ and $\alpha=0.15 \times 10^{-6} \text{ m}^2/\text{s}$. Assuming an initial uniform temperature of 15°C for the ground, determine the minimum burial depth to prevent the water pipes from freezing. (8)

12 a) i) Air flows a flat plate at a velocity of 10 m/s . Air and surface temperature of the plate are 20°C and 580°C respectively. Calculate the amount of heat transferred per meter width from both sides of the plate over a distance of 40 cm from the leading edge. Refer page No.42, Question No.2 (similar) (8)

(ii) Engine oil flows through a 50 mm diameter tube at an average temperature of 147°C . the flow velocity is 80 cm/s . Calculate the average heat transfer coefficient if the tube wall is maintained at a temperature of 200°C and it is 2 m long? (8)

(Or)

b) Consider a $0.6\text{ m} \times 0.6\text{ m}$ thin square plate in a room at 30°C . One side of the plate is maintained at a temperature of 90°C , while the other side is insulated. Determine the rate of heat transfer from the plate by natural convection if the plate is (i) vertical (ii) horizontal with hot surface facing up, and (iii) horizontal with hot surface facing down.

13 (a) Water is to be boiled at atmospheric pressure in a mechanically polished stainless steel pan placed on top of a heating unit the inner surface of the bottom of the pan is maintained at 108°C . If the diameter of the bottom of the pan is 30 cm , determine (i) the rate of heat transfer to the water and (ii) the rate of evaporation of water. Refer page No.70, Question No.2 (similar) (16)

(Or)

(b) A shell passes and 4 tube passes heat exchanger is used to heat glycerin from 20°C to 50°C by hot water which enters the thin-walled 2 cm diameter tubes at 80°C and leaves at 40°C . the total length of the tubes in the heat exchanger is 60 cm . the convection heat transfer coefficient is $25\text{ W/m}^2\text{K}$ on the glycerin (shell) side and $160\text{ W/m}^2\text{K}$ on the water (tube) side. determine the rate of heat transfer in the heat exchanger (i) before any fouling and (ii) after fouling with a fouling factor of $0.0006\text{ m}^2\text{ k/w}$ occurs on the outer surfaces of the tubes.. (16)

14. a) (i) The filament of a round bulb is maintained at a temperature of 2000 K and it is assumed to be black body. The transmissivity of the bulb glass is 0.92 in the visible range ($0.65 < \lambda < 0.75\text{ }\mu\text{m}$) of the radiation. Calculate the amount of energy transmitted. (8)

(ii) The surfaces of a double walled spherical vessel used for storing liquid oxygen are covered with a layer of silver having an emissivity of 0.03 . The temperature of the outer surface of the inner wall is -153°C and the temperature of the inner surface of the outer wall is 27°C . The spheres are 21 cm and 30 cm in diameter, with the space between them evacuated. Calculate the radiation heat transfer through the walls into the vessel and the rate of evaporation of liquid oxygen if its rate of vaporization is 220 kJ/Kg . (8)

(Or)

b) Two parallel plates $2\text{ m} \times 1\text{ m}$ are spaced 1 m apart. The plates are at a temperature of 727°C and 227°C and their emissivities are 0.3 and 0.5

respectively. The plates are located in a large room, the walls of which are at 27°C . determine the rate of heat loss from each plate and heat gain by the walls. Refer page No.101, Question No.3 (similarity) (16)

15. a) Two large vessel contain uniform mixture of air and sulphur dioxide at 1 atm and 273k but at difference concentration. Vessel 1 contains 80 % air and 20 % SO_2 by volume or mole percentage whereas vessel 2 contains 30% air and 70 % SO_2 by mole percentage. .The vessels are connected by a 10 cm inner diameter 1.8m long pipe .Determine the rate of transfer of air between these two vessels by assuming that a steady state transfer takes place .the mass diffusivity of air SO_2 mixture at 1 atm 273 k is $0.122 \times 10^{-4} \text{ m}^2/\text{s}$ (16)
Refer page No.134, Question No.1 (similarity)

(Or)

- b) The water in a 5 cm 15 m outdoor swimming pool is maintained at a temperature of 27°C . The average temperature and relative humidity are 37°C and 40 % respectively. Assuming a wind speed of 2 m/s in the direction of the long side of the pool. Estimate the mass transfer coefficient for the evaporation of water from the pool surface and the rate of evaporation in Kg/day. Refer page No.132, Question No.7 (similarity) (16)

STUCOR APP

B.E/B.Tech DEGREE EXAMINATION, NOV/DEC 2016.
Fifth Semester
Mechanical Engineering
ME 6502 – HEAT AND MASS TRANSFER
(Regulation 2013)

Time: Three hours

Maximum: 100 marks

Answer ALL questions

PART A – (10 x 2= 20 marks)

1. State the Fourier's law of heat conduction. Why is the negative sign used?
Refer page No.9, Question No.1
2. Under what circumstances from the heat transfer point of view, will the use of finned walls be better?
3. What are the differences between natural and forced convection? Refer page No.39, Question No.14
4. What is critical Reynolds number for the flow over the flat plate? Refer page No.35, Question No.1
5. Give examples for pool boiling and flow boiling.
Refer page No.66, Question No.7
6. What are the fouling factors?
7. Define monochromatic emissive power. Refer page No.94, Question No.17
8. What do you mean by infrared and ultraviolet radiation
Refer page No.94, Question No.18
9. State Fick's law of diffusion.
Refer page No.109, Question No.7
10. Define Schmidt number and state its physical significance. Refer page No.120, Question No.10

PART B- (5 x 16= 80 marks)

1. A) i) A body of an electric motor is 360 mm in diameter and 240mm long. It dissipates 360W of heat and its surface temperature should not exceed 55°C. longitudinal fins of 15 mm thickness and 40 mm height are proposed. The heat transfer coefficient is 40 W/ m² K when the ambient temperature 30°C. Determine the number of fins required, if K of the fine material is 40 W/m K.
(10)
(ii) Derive an expression for critical radius of insulation for a cylindrical system.
(6)

(Or)

B) (i) Determine the minimum depth at which one must place a water main below the soil surface to avoid freezing. The soil is initially at a uniform temperature of 20°C. in severe winter condition it is subjected to a temperature of -15°C for a period of 60 days. Use the following properties of the soil: $\rho = 2050 \text{ kg/m}^3$, $C = 1840 \text{ j/kg K}$ and $k = 0.52 \text{ W/m K}$.
(8)

(ii) A steel pipe with 50mm OD is covered with two layers of insulation. The inner layer is 7.5mm thick and has a $k = 0.3 \text{ W/m K}$ and the top layer is

20mm thick and $k = 0.12 \text{ W/m K}$. The pipe wall is 315°C and the outside air temperature is 25°C . determine the surface temperature and heat loss per metre length for 10 minutes. Take the convective heat transfer coefficient between air the surface as $16 \text{ W/ m}^2 \text{ K}$. (8)

12. a) (i) Define velocity boundary layer and thermal boundary layer. (4)
Refer page No.48, Question No.5

(ii) Air at 200 kPa and 200°C is heated as it flows through a tube with a diameter of 25 mm at a unit length of the tube. If a constant heat flux condition is maintained at the wall and the wall temperature is 20°C above the air temperature, all along the length of the tube. How much would the bulk temperature increases over 3 m length of the tube? (12)

(Or)

b) (i) A 0.5 m high flat plate of glass at 93°C is removed from an annealing furnace and hung vertically in the air at 28°C , 1 atm . Calculate the initial rate of heat transfer to the air. The plate is 1 m wide. (10)

Refer page No.44, Question No.3 (similarity)

(ii) A fine wire having a diameter of 0.02 mm is maintained at a constant temperature of 54°C by an electric current. The wire is exposed to air at 1 atm . and 0°C . Calculate the electric power necessary to maintain the wire temperature if the length is 50 cm . (6)

13. (a) The bottom of copper pan, 300 mm in diameter is maintained at 120°C by an electric heater. Calculate the power required to boil water in this pan. What is the evaporation rate? Estimate the critical heat flux. (16)

(Or)

(b) Water at the rate of 4 kg/s is heated from 40°C to 55°C in a shell and tube heat exchanger. On shell side one pass is used with water as heating fluid ($\dot{m} = 2 \text{ kg/s}$), entering the exchanger at 95°C . The overall heat transfer coefficient is $1500 \text{ W/m}^2\text{C}$ and the average water velocity in the 2 cm diameter tubes is 0.5 m/s . Because of space limitations the tube length must not exceed 3 m . calculate the number of tube passes, keeping in mind the design constraint. (16)

Refer page No.85, Question No.10

14. a) (i) A gray, diffuse opaque surface ($\alpha = 0.8$) is at 100°C and receives an irradiation 1000 W/m^2 . If the surface area is 0.1 m^2 . Calculate

1) Radiosity of the surface

2) Net radiative heat transfer rate from the surface

3) Calculate above quantities, if surface is black. (10)

(ii) Emissivities of two large parallel plate maintained at 800°C and 300°C and 0.3 and 0.5 respectively. Find the net radiant heat exchange per square metre for these plates. (6)

Refer page No.101, Question No.3

(Or)

- b) Two rectangles 50 x 50 cm are placed perpendicular with common edge. One surface has $T_1 = 1000$ K, $\epsilon = 0.6$, while the other surface is insulated and in radiant balance with a large surrounding room at 300 K. Determine the temperature of the insulated surface and heat lost by the surface at 1000 K.
15. a) Air is contained in a tyre tube of surface area 0.5 m^2 and wall thickness 10 mm. The pressure of air drops from 2.2 bar to 2.18 bar in a period of 6 days. The solubility of air in the rubber is 0.072 m^3 of air per m^3 of rubber at 1 bar. Determine the diffusivity of air in rubber at operating temperature of 300 K if the volume of air in the tube is 0.028 m^3 .

(Or)

- b) Along a horizontal water surface an air stream with velocity $u_c = 3 \text{ m/s}$ is flowing. The temperature of the water on the surface is 15°C , the air temperature is 20°C the total pressure is 1 atm (10^6 N/m^2), and the saturation pressure of the water vapour in the air at 20°C is 2337 N/m^2 . The relative humidity of the air is 33%. The water surface along the wind direction has a length of 10 cm. calculate the amount of water evaporated per hour per meter from the water surface. The binary diffusivity of water vapour in the air may be taken as $3.3 \times 10^{-5} \text{ m}^2/\text{s}$. The saturation vapour pressure of water at 15°C 1705 N/m^2 and kinematic viscosity of the air is $1.5 \times 10^{-5} \text{ m}^2/\text{s}$.

STUCOR APP

B.E/B.Tech DEGREE EXAMINATION, MAY/JUNE 2016.

Fifth Semester

Mechanical Engineering

ME 6502 – HEAT AND MASS TRANSFER

(Regulation 2013)

Time: Three hours

Maximum: 100 marks

Answer ALL questions

PART A – (10 x 2= 20 marks)

1. What are the modes of heat transfer? Give examples.
2. What is lumped capacitance analysis?
3. Differentiate forced and free convection
Refer page No.39, Question No.14
4. Differentiate hydrodynamic boundary layer and thermal boundary layer.
Refer page No.40, Question No.15
5. What is black body radiation?
6. Define emissive power and monochromatic emissivity.
Refer page No.94, Question No.17
7. What are the assumptions made in Nusselt theory of condensation?
8. What is fouling and how does it affect the rate of heat transfer?
9. Define Fick's law.
Refer page No.119, Question No.7
10. Diffusivity of ammonia in air at temperature 30°C and pressure 1 atm, is 0.228 cm²/sec. find the diffusivity of ammonia in cm²/sec in air at temperature 50°C and pressure 1.1 atm.

PART B- (5 x 16= 80 marks)

11. (a) Write short notes on :
 - (i) Heat transfer with extended surfaces (3)
 - (ii) Critical radius of insulation (3)
 - (iii) A flat furnace wall is constructed of 114 mm layer of sil-o-gel brick with a thermal conductivity of 0.318 W/m°C backed by a 229mm layer of common brick of conductivity 1.38 W/m°C. The temperature of inner face of the wall is 760°C and that of the outer face is 76.6°C. (a) What is the heat loss through composite wall? (b) What is the temperature of interface between refractory brick and common brick? (10)

(OR)

- (b) (i) A plane wall 10 cm thick generates heat at the rate of 4×10^4 W/m³, when electric current is passed through it. The conductive heat transfer coefficient between each phase of wall and ambient air is 50

$W/m^2\text{C}$. The thermal conductivity of wall is $15 W/m^\circ C$ (a) derive the temperature profile for given slab; (b) determine the surface temperature, (c) the maximum temperature in the wall. (10)

Refer page No.22, Question No.6

(ii) What is the significance of Heisler charts in transient heat conduction? (6)

12.(a) (i) Explain about three layer model for internal convection n tubular flow. (8)

(ii) A fluid of kinematic viscosity equal to $15 \times 10^{-6} m^2/s$ flows with an average velocity of $10 m/s$ in a square duct of $0.08 \times 0.08 m$ cross section. What is the Reynolds number based on the hydraulic diameter? Is the flow laminar or turbulent? What is the Nusselt number if the flow is fully developed and the Prandtl number is 0.7 ? (8)

(OR)

(b) (i) Explain about thermal boundary layer flow past a flat plate. (6)

(ii) Consider flow of air at atmospheric pressure and $300 K$ parallel to a flat plate $2 m$ long. The viscosity of air far away from the plate is $10 m/ sec$. the plate surface is held at a constant temperature of $400 K$. Determine the heat transfer coefficient at the trailing edge of the plate using the colburn analogy ? Data: Properties at film temperature are: Density $0.995 kg/ m^3$, kinematic viscosity $20.92 \times 10^{-6} m^2/s$, thermal conductivity $0.03 W m^{-1}k^{-1}$. Prandtl number 0.7 . (10)

13. (a) (i) Hot water enters a counter flow exchanger at $95^\circ C$. This hot water is used to heat a cool stream of water from 8 to $40^\circ C$. The flow rate of the cool water is $1.2 kg/s$, and that of the hot water is $2.7 kg/s$. the overall heat transfer coefficient is $850 W/m^2\text{C}$. What is the area of the heat exchanger and its effectiveness? (10)

(ii) Name and brief the different types of heat exchangers (6)

Refer page No.80, Question No.8

(OR)

(b) (i) A hot stream is cooled from $120^\circ C$ to $30^\circ C$ while the cold stream temperature changes from 20 to $60^\circ C$. Find out the LMTD for both counter and co-current phenomenon. Justify how counter current is effective than co-current? (10)

(ii) What is flow boiling and pool boiling? Describe how heat transfer coefficient varies in regimes of pool boiling. (6)

Refer page No.61, Question No.1

- 14.(a) (i) Determine an expression for heat transfer rate by using electrical analogy (i) Without any shield between 2 parallel plates (ii) with shield in between 2 parallel plates. (16)

(OR)

- (b) (i) What is view factor and shape factor ? (5)
(ii) State laws of blackbody radiation? (5)
(iii) Two large parallel plates are at temperature $T_1 = 500\text{K}$ and $T_2 = 300\text{K}$. Their emissivities are $\epsilon_1 = 0.85$ and $\epsilon_2 = 0.90$. What is the radiant flux between the plates? (6)
- 15.(a) (i) derive an expression for mass flux in steady state molecular diffusion
(a) A through non diffusing B. (8)
(b) Equimolar Counter Diffusion. (8)
- (ii) NH_3 gas (A) diffuses through N_2 (B) under steady state condition with non-diffusing N_2 . The total pressure is 101.325 K pa and temperature is 298 K. the diffusion thickness is 0.15×10^4 pa and at the other point is 5×10^3 pa. the D_{AB} for mixture at 1 atm and 298 K is $2.3 \times 10^{-5} \text{ m}^2 / \text{sec}$. (a) Calculate flux of NH_3 (A through non diffusing B). Calculate flux for equimolar counter diffusion. (8)
- (OR)**
- (b) (i) Write a note on convective mass transfer coefficients for liquids and gases. (8)
(ii) Give a brief description on heat, momentum and transfer analogies. (8)

B.E/B.Tech DEGREE EXAMINATION, MAY/JUNE 2016.
Fourth Semester
Mechanical Engineering
ME 2251 – HEAT AND MASS TRANSFER
(Regulation 2008)

Time: Three hours

Maximum: 100 marks

Answer ALL questions

PART A – (10 x 2= 20 marks)

1. List the differences between thermodynamics and heat transfer
Refer page no.12, Question no.13
2. State the assumptions on which the fourier's law of conduction is based.
3. What is the difference between friction factor and friction co-efficient?
Refer page no.39, Question no.13
4. Why heat transfer co-efficient of natural convection is much less than those forced convection? Refer page no.38, Question no.9
5. What is a compact heat exchanger? Give applications.
Refer page no.67, Question no.12
6. Define effectiveness and NTU of a heat exchanger.
Refer page no.64, Question no.10 & 2.
7. Define radiation intensity.
8. Differentiate black body and grey body. Refer page no.86, Question no.9
9. Enumerate important aspects of Fick's law of diffusion.
Refer page no.119, Question no.7
10. What is mass transfer?
Refer page no.119, Question no.1

PART B- (5 x 16= 80 marks)

11. (a) A copper wire of 10 mm dia is covered with 10 mm thick of plastic insulation. The plastic insulation is exposed to air at 35°C with $h = 8 \text{ W/m}^2\text{K}$. the K for Cu and plastic are $400 \text{ W/m}^\circ\text{C}$ and $0.5 \text{ W/m}^\circ\text{C}$. the resistivity is $3 \times 10^{-3} \text{ ohms mm}$. the plastic insulation temperature should not exceed 180°C. Determine (i) heat transfer rate and current carrying capacity (ii) q_{\max} , maximum current carrying capacity.

(OR)

- (b) A motor body has 500 mm O.D and 400 mm long. It is maintained at 60°C. 40 longitudinal fins with the height of 20 mm and thickness of 8 mm are attached with the body. Thermal conductivity of fin material is 55

$W/m^{\circ}C$. Heat transfer co-efficient is $23 W/m^2K$. Find (i) Area weighed fin efficiency (ii) % increase in heat transfer due to addition of fins.

- 12.(a) Air at $20^{\circ}C$, $1 m/sec$ flows over a flat plate of $2 m \times 1 m$ maintained at $40^{\circ}C$. Determine (i) boundary layer thickness at $40 cm$ from leading edge, (ii) boundary layer thickness at $2m$ from leading edge (iii) localised heat transfer co-efficient at $2m$, (iv) average heat transfer co-efficient from leading to $2m$ length.

(OR)

(b) $1000 kg/hr$ of cheese at $150^{\circ}C$ is pumped through a tube of $7.5 cm$ dia. After passing through an unheated length of about 50 diameters, it passes through a $1.2 m$ length of tube maintained at $90^{\circ}C$. Calculate the heat transfer co-efficient and mean temperature of cheese leaving the heated section. For cheese $k=1.55 W/mK$, $C_p = 2.85 kJ/kgK$, $\rho = 1100 kg / m^3$ and $\mu = 56400 kg/ hr-m$.

- 13.(a) In cross flow heat exchanger, both fluid unmixed, hot fluid with a specific heat of $2300 J/kg K$ enters at $380^{\circ}C$ and leaves at $300^{\circ}C$. cold fluid enters at $25^{\circ}C$ and leaves at $210^{\circ}C$. Calculate the required surface area of heat exchanger. Take over all heat exchanger co-efficient as $750 W/m^2K$. Mass flow rate of hot fluid is $1 kg/s$.

Refer page no.78, Question no.7

(OR)

(b) A parallel flow heat exchanger is used to cool $4.2 kg/min$ of hot liquid as specific heat $3.5 kJ/kg K$ at $130^{\circ}C$. A cooling water of sp.heat $4.18 kJ/kg K$ is used for cooling purpose at a temperature of $15^{\circ}C$. The mass flow rate of cooling water is $17 kg/min$. Calculate the following (i) Outlet temperature of the liquid, (ii) Outlet temperature of water (iii) Effectiveness of heat exchanger.

- 14.(a) The inner sphere of a liquid oxygen container is $400 mm$ dia., outer sphere is $500 mm$ dia., and both have emissivity 0.05 . Determine the rate of liquid oxygen evaporation at $-183^{\circ}C$, when the outer sphere temperature is $20^{\circ}C$. The latent heat of evaporation is $210 kJ/kg$. Neglect losses due to other modes of heat transfer. Refer page no.112, Question no.8

(OR)

(b) A large isothermal enclosure is maintained at $2500 K$. Determine (i) emissive power of radiation that emerge from a small aperture on the enclosed surface on the enclosed surface (ii) wavelength, below which 10% of emission is constructed (iii) wavelength, above which 10% emission is constructed (iv) max spectral intensity and corresponding wavelength.

- 15.(a) Air at $20^{\circ}C$ ($\rho = 1205 kg/m^3$; $\nu = 15.06 \times 10^{-6} m^2/s$; $D = 0.15 m^2/hr$) flows over a tray (length = $25 cm$, width = $40 cm$) full of water with a velocity of $2.5 m/s$. The total pressure of moving air is $1.01 bar$ and the partial pressure of water present in the air is $0.0075 bar$. If the

temperature on the water surface is 15°C calculate the evaporation rate of water. Refer page no.124, Question no.2 (16)

(OR)

(b) CO_2 and air experience equimolar counter diffusion in a circular tube whose length and diameter are 1 m and 50 mm respectively. The system of total pressure of 1 atm and a temperature of 25°C . The ends of the tube are connected to large chambers in which the species concentrations are maintained at fixed values. the partial pressure of CO_2 at one end is 190 mm of Hg while at the other end is 95 mm Hg. estimate the mass transfer rate of CO_2 and air through the tube. Refer page no.127, Question no.4 (16)

STUCOR APP

B.E/B.Tech DEGREE EXAMINATION, NOVEMBER/DECEMBER 2015.
Fourth Semester
Mechanical Engineering
ME 2251 - HEAT AND MASS TRANSFER
(Regulation 2008)

Time: Three hours

Maximum: 100 marks

Answer ALL questions
PART A - (10 x 2= 20 marks)

1. What do you understand by critical thickness of insulation? Give its expression. Refer page no.10, Question no.5
2. What is lumped capacity analysis? Refer page no.11, Question no.8
3. Mention the significance of boundary layer.
4. Define Prandtl number and Grashoff number.
Refer page no.36, question no.4
5. Differentiate between pool and flow boiling.
Refer page no.66, question no.7
6. What do you understand by fouling and heat exchanger effectiveness?
Refer page no.66, question no.9
7. Assuming the sun to be Black body emitting radiation with maximum intensity at $\lambda = 0.49 \mu\text{m}$, calculate the surface temperature of the sun.
Refer page no.94, question no.14
8. What is irradiation and radiosity?
Refer page no.92, question no.5
9. How mass transfer takes through Diffusion and convection?
10. What do you mean by equimolar counter diffusion?

PART B- (5 x 16= 80 marks)

11. (a) (i) A reactor's wall 320 mm thick is made up of an inner layer of fire brick ($k=0.84 \text{ W/m}^0\text{c}$) covered with a layer of insulation ($k=0.16 \text{ W/m}^0\text{c}$).the reactor operates at a temperature of 1325^0C and the ambient temperature is 25^0C .Determine the thickness of the fire brick and insulation which gives minimum heat loss. Calculate the heat loss presuming that the insulating material has a maximum temperature of 1200^0C . (8)

Similar problem Refer page no.16, question no.3

(ii) Derive an expression for the heat conduction through a hollow cylinder from the general heat conduction equation. Assume steady state unidirectional heat flow in radial direction and no internal heat generation. (8)

Refer page no.14, question no.2

(OR)

(b)(i) A 25 mm diameter rod of 360 mm length connects two heat sources maintained at 127°C and 227°C respectively. The curved surface of the rod is losing heat to the surrounding air at 27°C . The heat transfer coefficient is $10\text{W/m}^2\text{C}$. Calculate the loss of heat from the rod if it is made of copper ($k=335\text{W/m}^0\text{c}$) and steel ($k=40\text{W/m}^0\text{c}$) (8)

(ii) A thermocouple junction is in the form of 8 mm diameter sphere. The properties of the material are $c = 420\text{J/kg}^0\text{C}$, $\rho=8000\text{kg/m}^3$; $k=40^0\text{C}$ and inserted in a stream of hot air at 300^0C . Find the time constant of the thermocouple. The thermocouple is taken out from the hot air after 10 seconds and kept in still air of $10\text{W/m}^0\text{c}$, find the temperature attained by the junction 20 seconds after removing from hot air. (8)

Refer page no.29, question no.10

12.(a) Air at 20^0C and at a pressure of 1 bar is flowing over a flat plate at a velocity of 3 m/s. If the plate is 280 mm wide and at 56^0C calculate the following at $x=280\text{mm}$:

- (i) Boundary layer thickness
- (ii) Local friction coefficient
- (iii) Average friction coefficient
- (iv) Thickness of the thermal boundary layer
- (v) Local convective heat transfer coefficient
- (vi) Average convective heat transfer coefficient
- (vii) Rate of heat transfer by convection
- (viii) Total drag force on the plate. (16)

Refer page no.40, question no.1

(OR)

(b)(i) A cylindrical body of 300 mm diameter and 1.6 m height is maintained at a constant temperature of 36.5^0C . The surrounding temperature is 13.5^0C . Find the amount of heat generated by the body per hour if $c_p=0.96\text{kJ/kg}^0\text{C}$; $\rho=1.025\text{kg/m}^3$; $k=0.0892\text{W/m}^0\text{C}$, $\nu=15.06 \times 10^{-6}\text{m}^2/\text{s}$ and $\beta=1/298\text{K}^{-1}$. Assume $\text{Nu}=0.12(\text{Gr.Pr})^{1/3}$ (8)

(ii) A nuclear reactor with its core constructed of parallel vertical plates 2.2 m high and 1.4 m wide has been designed on free convection heating

of liquid bismuth .the maximum temperature of the plate surfaces is limited to 960°C while the lowest allowable temperature of bismuth is 340°C .calculate the maximum possible heat dissipation from both sides of each plate. The properties of bismuth at film temperature are $c_p=150.7\text{KJ/kg}^{\circ}\text{C}$; $\rho=10000\text{ kg/m}^3$; $k=13.02\text{W/m}^{\circ}\text{C}$, $\mu=3.12 \times 10^{-6}\text{kg/m h}$. Assume $\text{Nu}=0.12(\text{Gr.Pr})^{1/3}$. (8)

- 13.(a) (i) water at atmospheric pressure is to be boiled in a polished copper pan. The diameter of the pan is 350 mm and is kept at 115°C .calculate the power of the burner, rate of evaporation in kg/h and the critical heat flux. (8)

Refer page no.70, question no.2

(ii) A vertical cooling fin approximating a flat plate 40 cm in height is exposed to saturated steam at atmospheric pressure. The fin is maintained at a temperature of 90°C .Estimate the thickness of the film at the bottom of the fin, overall heat transfer coefficient and heat transfer rate after incorporating McAdam's correction. (8)

Refer page no.73, question no.4

(OR)

- (b) (i) Explain how heat exchangers are classified? (8)

Refer page no.80, question no.8

(ii) A counter flow double pipe heat exchanger using superheated steam is used to heat water at the rate of 10500 kg/h. The steam enters the heat exchanger at 180°C and leaves at 130°C .The inlet and exit temperature of water are 30°C and 80°C respectively.if $U=814\text{ w/m}^2\text{C}$, Calculate the heat transfer area. What would be increase in area if the fluid flows were parallel? (8)

Refer page no.76, question no.6

14. (a)(i) Calculate the following for an industrial furnace in the form of a black body and emitting radiation at 2500°C
Monochromatic emissive power at $1.2\ \mu\text{m}$ wave length.
v) Wave length at which emission is maximum.
vi) Maximum emissive power.
vii) Total emissive power,
iv) The total emissive of the furnace if it is assumed as a real surface having emissivity equal to 0.9. (10)

Refer page no.95, question no.1

- (ii) Define the following

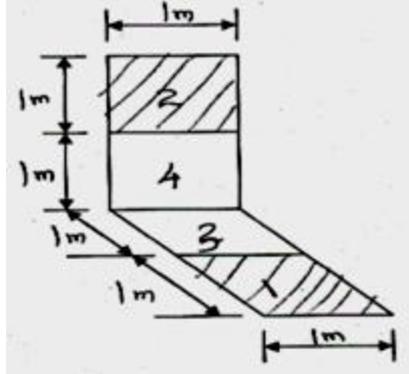
(i) Black body, (ii) Gray body, (iii) Opaque body, (iv) white body

(v) Specular reflection and (vi) Diffuse Defection (6)

Refer page no.93&94

(OR)

(b) (i) The area A_1 and A_2 are perpendicular but do not share the common edge. Find the shape factor F_{1-2} for the arrangement. (12)



Refer page no.104, question no.4

(ii) Determine the radiant heat exchange in W/m^2 between two large parallel steel plates of emissivities 0.8 and 0.5 held at temperatures of 1000 K and 500 K respectively, if a thin copper plate of emissivity 0.1 is introduced as a radiation shield between the two plates. (4)

Refer page no.101, question no.3

15. (a)(i) State Fick's Law of diffusion and give its expression. Obtain an expression for the same in terms of partial pressures. (4)

(ii) Derive the general mass transfer equation in Cartesian coordinates. (12)

(OR)

(b) (i) A vessel contains binary mixture of O_2 and N_2 with partial pressures in the ratio 0.21 and 0.79 at $15^\circ C$. The total pressure of the mixture is 1.1 bar. Calculate the following:

- (1) Molar concentrations
- (2) Mass densities
- (3) Mass fractions
- (4) Molar fractions of each species. (8)

Refer page no.122, question no.1

(ii) Air at $20^\circ C$ with $D = 4.166 \times 10^{-5} m^2/s$ flows over a tray (length = 320 mm, width = 420 mm) full of water with a velocity of 2.8 m/s. The total pressure of moving air is 1 atm and the partial pressure of water present in the air is 0.0068 bar. If the temperature on the water surface is $15^\circ C$. Calculate the evaporation rate of water. (8)

Refer page no.132, question no.7 (similarity)

B.E/B.Tech DEGREE EXAMINATION, NOVEMBER/DECEMBER 2015.

**Fifth Semester
Mechanical Engineering
ME 6502 - HEAT AND MASS TRANSFER
(Regulation 2013)**

Time: Three hours

Maximum: 100 marks

Answer ALL questions

PART A - (10 x 2= 20 marks)

1. Will the thermal contact resistance be greater for smoother or rough plain surfaces? Why?
2. Distinguish between fin efficiency and its effectiveness.
Refer page no.10, question no.6
3. What is Dittus-Boetler equation? When does it apply?
Refer page no.39, question no.12
4. Define Grashof number and explain its significance in free convection heat transfer. Refer page no.36, question no.4
5. What is meant by sub-cooled and saturated boiling?
Refer page no.66, question no.11
6. What advantages does the NTU method have over the LMTD method?
Refer page no.65, question no.6
7. Define irradiation and radiosity. Refer page no.92, question no.5
8. What is the greenhouse effect? Why is it a matter of great concern among atmospheric scientists?
9. What is the driving force for
 - a) Heat transfer and
 - b) Mass transfer
10. Define Lewis number and state its physical significance

PART B- (5 x 16= 80 marks)

11. (a) (i) Consider a 1.2 m high and 2 m wide double-pane window consisting of two 3mm thick layer of glass ($k=0.78\text{W/mK}$) separated by a 12mm wide stagnant air space ($k=0.026\text{ W/mK}$). Determine the steady rate of heat transfer through this double -pane window and the temperature of its inner surface when the room is maintained at 24°C while the temperature of the outdoors is -5°C . Take the convective heat transfer coefficients on the inner and outer surfaces the window to be $10\text{ W/m}^2\text{K}$ and $25\text{ W/m}^2\text{K}$ respectively. (8)

Similar problem Refer page no.16, question no.3

(ii) Derive the general 3-Dimensional heat conduction equation in Cartesian coordinates. Refer page no.12, question no.1 (8)

(OR)

(b) A cylinder 1m long and 5 cm in diameter is placed in an atmosphere at 45°C. It is provided with 10 longitudinal straight fins of materials having $k=120 \text{ W/mK}$. The height of 0.76 mm thick fins is 1.27 cm from the cylinder surface. The heat transfer coefficient between cylinder and atmospheric at the end of fins if surface temperature of cylinders is 150°C. (16)

12. (a) (i) A long 10 cm diameter steam pipe whose external surface temperature is 110°C passes through some open area that is not protected against the winds. Determine the rate of heat loss from the pipe per unit length when the air is at 1 atm and 10°C and the wind is blowing across the pipe at a velocity of 8 m/s. (8)

Similar problem Refer page no.52, question no.6

(ii) An air stream at 0°C is flowing along a heated plate 90°C at a speed of 75 m/s. The plate is 45 cm long and 60 cm wide. Calculate the average values of friction coefficient for the full length of the plate. Also calculate the rate of energy dissipation from the plate. (8)

Similar problem Refer page no.38, question no.2

(OR)

(b) (i) Explain the concept of Hydrodynamic and thermal boundary layers. (6)

Refer page no.48, question no.5

(ii) A 6 m long section of an 8 cm diameter horizontal hot water pipe passes through a large room whose temperature is 20°C. If the outer surface temperature and emissivity of the pipe are 70°C and 0.8 respectively, Determine the rate of heat loss from the pipe by

(1) Natural convection

(2) Radiation. (10)

13. (a) Water is boiled at atmospheric Pressure by horizontal polished copper heating element of diameter $D= 5 \text{ mm}$ and emissivity 0.05 immersed in water. If the surface temperature of the heating wire is 350°C, determine the rate of heat transfer from the wire to the water per unit length of the wire. (16)

(OR)

(b) Hot oil ($C_p = 2200 \text{ J/kg K}$) is to be cooled by water ($C_p = 4180 \text{ J/kg K}$) in a 2 - shell - pass and 12 - tube - pass heat exchanger. The tubes are thin - walled and are made of copper with diameter of 1.8 cm. The length of each

tube pass in the heat exchanger is 3 m, and the overall heat transfer coefficient is $340 \text{ W/m}^2\text{K}$. Water flows through the tubes at a total rate of 0.1 kg/s , and the oil through the shell at a rate of 0.2 kg/s . The water and the oil enter at temperature 18°C and 160°C , respectively. Determine the rate of heat transfer in the heat exchanger and the outlet temperatures of the water and the oil. (16)

Similar problem Refer page no.76, question no.6

14.(a)(i) Two very large parallel plates are maintained at uniform temperatures of $T_1=1000 \text{ K}$ and $T_2 = 800 \text{ K}$ and have emissivities of $\epsilon_1=\epsilon_2=0.2$, respectively. It is desired to reduce the net radiation heat transfer between the two plates to one-fifth by placing thin aluminium sheets with an emissivity of 0.15 on both sides between the plates. Determine the number of sheets that need to be inserted (10)

Similar problem Refer page no.101, question no.3

(ii) Define the following terms:

(1) Monochromatic emissivity

(2) Gray body

(3) Shape factor. (6)

Refer page no. 93,94

(OR)

(b)(i) The spectral emissivity function of an opaque surface at 1000 K is approximated as

$$\epsilon_{\lambda 1} = 0.4, 0 \leq \lambda < 2 \mu\text{m};$$

$$\epsilon_{\lambda 2} = 0.7, 2 \mu\text{m} \leq \lambda < 6 \mu\text{m}$$

$$\epsilon_{\lambda 3} = 0.3, 6 \mu\text{m} \leq \lambda < \infty$$

Determine the average emissivity of the surface and the rate of radiation emission from the surface, in W/m^2 (8)

Refer page no.104, question no.7

(ii) Emissivities of two large parallel plates maintained at 800°C and 300°C are 0.3 and 0.5 respectively. Find the net radiant heat exchange per square meter for these plates? (8)

Similar problem Refer page no.101, question no.3

15. (a) (i) A 3-cm-diameter Stefan-tube is used to measure the binary diffusion coefficient of water vapour in air at 20°C at an elevation of 1600 m where the atmospheric is 8.5 kPa. The tube is partially filled with water, and the distance from the water surface to the open end of the tube is 40 cm. Dry air is blown over the open end of the tube so that water vapour rising to the top is removed immediately and the concentration of vapour at the top of the tube is zero. In 15 days of continuous operation at constant pressure and temperature, the amount

of water that has evaporated is measured to be 1.23 g. Determine the diffusion coefficient of water vapour in air at 20°C and 83.5 kPa. (10)

(ii) State some analogies between heat and mass transfer. (6)
Refer page no.129, question no.5

(OR)

(b)(i) A thin plastic membrane separates hydrogen from air. the molar concentrations of hydrogen in the membrane at the inner and outer surface are determined to be 0.045 and 0.002 k mole/m³, respectively .the binary diffusion coefficient of hydrogen in plastic at the operation temperature is 5.3×10^{-10} m²/s.Determine the mass flow rate of hydrogen by diffusion through the membrane under steady conditions if the thickness of the membrane is

(1) 2 mm and

(2) 0.5 mm. (8)

Similar problem Refer page no.127, question no.4

(ii) Dry air at 15°C and 92 kPa flows over a 2 m long wet surface with a free stream velocity of 4 m/s. Determine the average mass transfer coefficient. (8)

STUCOR APP

B.E/B.Tech DEGREE EXAMINATION, APRIL/MAY 2015.

Fifth Semester

Mechanical Engineering

ME 2251 - HEAT AND MASS TRANSFER

(Regulation 2008)

Time: Three hours

Maximum: 100 marks

Answer ALL questions

PART A - (10 x 2= 20 marks)

1. What is meant by transient heat conduction?
2. What is thermal diffusivity?
3. Define velocity boundary layer thickness
Refer page no. 36, Question no.5
4. Distinguish between laminar and turbulent flow
Refer page no. 35, Question no.2
5. How heat exchangers are classified?
Refer page no. 65, Question no.5
6. Discuss the advantage of NTU method over the LMTD method.
Refer page no. 64, Question no.2
7. State Kirchhoff's law?
Refer page no. 91, Question no.3
8. Define Radiosity
Refer page no.92, Question no.5
9. State Fick's law of diffusion.
Refer page no. 119, Question no.7
10. Define molar concentration.
Refer page no. 121, Question no.13

Part -B (5X16=80 Marks)

- 11.(a) (i) Differentiate between conductivity and conductance. (4)
- (ii) A steel pipe line (thermal conductivity $K = 50 \text{ W/mK}$) of inner diameter 100 mm and outer diameter 110mm is to be covered with two layers of insulation each having a thickness of 50mm. the thermal conductivity of the first insulation material is 0.06 W/mK and that of the second is 0.12 W/mK . Calculate the loss of heat per meter length of pipe and the interface temperature between the two layers of insulation when the temperature of the inside tube surface is 250° C and that of the outside surface of the insulation is 50° C . (12)

Refer page no. 20, Question no.5

(OR)

(b)(i) With a neat sketch, explain different types of fin profile. (4)

(ii) A metallic sphere of radius 10mm is initially at a uniform temperature of 400° C. It is heat treated by first cooling it in air (heat transfer coefficient $h = 10 \text{ W/m}^2\text{K}$ at 20° C until its central temperature reaches 335° C to 50° C. Compute the time required for cooling in air and water for the following physical properties of the sphere density = 3000 Kg/m^3 specific heat = 1000 J/Kg K : thermal conductivity = 20 W/mK thermal diffusivity = $6.66 \times 10^{-6} \text{ m}^2/\text{s}$. (12)

Refer page no. 27, Question no.9

12.(a) (i) Explain the velocity boundary layer profile on a flat plate and mention its significance (4)

Refer page no. 48, Question no.5

(ii) Engine oil at 20°C is forced over a 20 cm square plate at a velocity of 1.2 m/s. The plate is heated to a uniform temperature of 60°C. Calculate the heat loss of the plate. (12)

(OR)

(b) (i) Considering a heated vertical plate in quiescent fluid, draw the velocity and temperature profile. (4)

Refer page no. 48, Question no.5

(ii) Water at 60°C enters a tube of 2.54 mm diameter at a mean flow velocity of 2 cm/s. Calculate the exit water temperature if the tube is 3 m long and the wall temperature is constant at 80°C. (12)

13.(a) (i) with a neat sketch explain various regimes of pool boiling. (4)

Refer page no. 68, Question no.1

(ii) A 10 X 10 array of horizontal tubes of 1.27 cm diameter is exposed to pure steam at atmospheric pressure. If the tube wall temperature is 98° C, estimate the mass of steam condensed assuming a tube length of 1.5 m. (12)

(OR)

(b) (i) What are the different types of fouling in heat exchangers? (4)

(ii) Water enters a cross flow heat exchanger (both fluid unmixed) at 5° C and flows at the rate of 4600 kg/h to cool 4000 kg/h of air that is initially at 40° C. Assume the overall heat transfer coefficient value to be 150 $\text{W/m}^2\text{K}$. For an exchanger surface area of 25 m^2 . Calculate the exit temperature of air and water. (12)

14.(a) Consider a cylindrical furnace with outer radius = 1 m and height = 1 m. The top (surface 1) and the base (surface 2) of the furnace have emissivities 0.8 & 0.4 and are maintained at uniform temperatures of 700 K and 500K respectively. The side surface closely approximates a black body and is maintained at a temperature of 400 K. Find the net rate of radiation heat transfer at each surface during steady state operation. (16)

Refer page no.108, Question no.6

(OR)

(b) Emissivities of two large parallel plates maintained at 800°C and 300°C are 0.3 and 0.5 respectively. Find the net radiant heat exchange per square meter for these plates. Find the percentage reduction in heat transfer when a polished aluminium radiation shield ($\epsilon = 0.05$) is placed between them. Also find the temperature of shield. (16)

Refer page no.101, Question no.3

15. (a) Dry air at 27°C and 1 bar flows over a wet plate of 50 cm at 50 m/s. Calculate the mass transfer coefficient of water vapour in air at the end of the plate. Refer page no.126, Question no.3 (16)

(OR)

(b) (i) What are the assumptions made in the 1 - D Transient mass diffusion problems? (4)

(ii) The dry bulb and wet bulb temperatures recorded by a thermometer in moist air are 27°C and 17°C respectively. Determine the specific humidity of air assuming the following values: Prandtl number = 0.74 Schmidt number = 0.6 specific heat at constant pressure = 1.004 kJ/kg K, pressure = 1.0132×10^5 N/m². (12)

STUCOR APP

B.E/B.Tech. DEGREE EXAMINATION, NOVEMBER/DECEMBER 2014.

**Fourth Semester
Mechanical Engineering
ME 2251 – HEAT AND MASS TRANSFER
(Regulation 2008)**

Answer ALL questions.

PART A – (10 X 2 = 20 marks)

1. State Fourier's Law of conduction.
Refer Page No.09 Question No.01
2. What is meant by lumped heat capacity analysis?
Refer Page No.11 Question No.08
3. Name four dimensions used for dimensional analysis.
Refer Page No.39 Question No.10
4. Define Grashof Number.
Refer Page No.36 Question No.04
5. What is pool boiling? Give an example for it.
Refer Page No.66 Question No.08
6. What is meant by effectiveness?
Refer Page No.66 Question No.10
7. What are the factors involved in radiation by a body?
Refer Page No.92 Question No.06
8. What is the use of radiation shield?
Refer Page No.92 Question No.04
9. State Fick's law diffusion.
Refer Page No.119 Question No.07
10. List out the various modes of mass transfer.
Refer Page No.119 Question No.03

PART B-(5×16=80 marks)

11. (a) (i) Derive general heat conduction equation in Cartesian coordinates.
Refer Page No.12 Question No.01 (10)

(ii) Compute the heat loss per square meter surface area of a 40 cm thick furnace wall having surface temperatures of 300°C and 50°C if the thermal conductivity k of wall material is given by $k=0.005T-5\times 10^{-6}T^2$ where T =temperature in °C. (6)

(Or)

(b) (i) A furnace wall consists of 200mm layer of refractory bricks, 6 mm layer of steel plate and 100 mm layer of insulation bricks. The maximum temperature of the wall is 1150°C on the furnace side and the minimum temperature is 40°C on the outermost side of the wall. An accurate energy balance over the furnace shows that the heat loss from wall is 400 W/m^2 . It is known that there is a thin layer of air between the layers of refractory bricks and steel plate. Thermal conductivities for the three layers are 1.52, 45 and $0.138 \text{ W/m}^{\circ}\text{C}$ respectively. Find

(1) To how many millimetres of insulation brick is the air layer equivalent?

(2) What is the temperature of the outer surface of the steel plate? (8)

Similar Problem Refer Page No.16 Question No.03

(ii) Find out the amount of heat transferred through an iron fin of length 50 mm, width 100 mm and thickness 5 mm. Assume $k=210 \text{ KJ/mh}^{\circ}\text{C}$ and $h=42 \text{ kJ/mh}^{\circ}\text{C}$ for the material of the fin and the temperature at the base of the fin as 80°C . Also determine the temperature at tip of the fin, if atmosphere temperature is 20°C . (8)

Similar Problem Refer Page No.23 Question No.07

12. (a) (i) Explain about velocity boundary layer on a flat plate. (6)

Refer Page No.48 Question No.05

(ii) Assuming that a man can be represented by a cylinder 30 cm in diameter and 1.7 m high with a surface temperature of 30°C , calculate the heat he would lose while standing in a 36 km/h wind at 10°C . (10)

(Or)

(b) (i) A metal plate 0.609 m high forms the vertical wall of an oven and is at a temperature of 161°C . Within the oven air is at a temperature of 93.0°C and one atmosphere. Assuming that natural convection conditions hold near the plate, estimate the mean heat transfer coefficient and the rate of heat transfer per unit width of the plate. (8)

(ii) A 10 mm diameter spherical steel ball at 260°C is immersed in air at 90°C . Estimate the rate of convective heat loss. (8)

13. (a) A vertical tube of 50 mm outside diameter and 2 m long is exposed to steam at atmospheric pressure. The outer surface of the tube is maintained at a temperature of 84°C by circulating cold water through the tube. Determine the rate of heat transfer and also the condensate mass flow rate. (16)

Similar Problem

(Or)

(b) (i) Explain about Fouling Factors. (4)

Refer Page No.66 Question No.09

(ii) Hot oil with a capacity rate of 2500 W/K flows through a double pipe heat exchanger. It enters at 360°C and leaves at 300°C . Cold fluid enters at 30°C and leaves at 200°C . If the overall heat transfer coefficient is $800 \text{ W/m}^2\text{K}$. Determine the heat exchanger area required for (1) Parallel flow and (2) Counter flow. (12)

Refer Page No. & 76, Question No.06

14. (a) Calculate the following for an industrial furnace in the form of a black body and emitting radiation at 2500°C . (i) Monochromatic emissive power at 1.2

μm length. (ii) Wave length at which the emission is maximum. (iii) maximum emissive power (iv) Total emissive power and (v) Total emissive power of the furnace if it is assumed as a real surface with emissivity equal to 0.9. (16)
Refer Page No.95 Question No.01

(Or)

(b) Two parallel plates of size 1.0 m by 1.0 m spaced 0.5 m apart are located in a very large room, the walls of which are maintained at temperature of 27°C. One plate is maintained at a temperature of 900°C and the other at 400°C. Their emissivities are 0.2 and 0.5 respectively. If the plates exchange heat between themselves and surroundings, find the net heat transfer to each plate and to the room. Consider only the plate surfaces facing each other. (16)

Refer Page No.96 Question No.02

15. (a) (i) Discuss about steady state equimolar counter diffusion. (8)

Refer Page No.130 Question No.06

(ii) Hydrogen gas is maintained at pressures of 2.4 bar and 1 bar on opposite sides of a plastic membrane 0.3 mm thick. The binary diffusion coefficient of hydrogen in the plastic is $8.6 \times 10^{-8} \text{ m}^2/\text{s}$ and solubility of hydrogen in the membrane is $0.00145 \text{ kg mole}/\text{m}^3\text{-bar}$. Calculate, under uniform temperature conditions of 24°C the following (1) Molar concentrations of the membrane and (2) Molar and mass diffusion flux of hydrogen through the membrane. (8)

(Or)

(b) (i) Air at 20°C ($\rho = 1.205 \text{ kg}/\text{m}^3$, $\nu = 15.06 \times 10^{-6} \text{ m}^2/\text{s}$, $D = 4.166 \times 10^{-5} \text{ m}^2/\text{s}$), flows over a tray (length = 320 mm, width = 420 mm) full of water with velocity of 2.8m/s. The total pressure of moving air is 1 atm and the partial pressure of water present in the air is 0.0068 bar. If the temperature on the water surface is 15°C, calculate the evaporation rate of water. (8)

Similar problem Refer Page No.132 Question No.07

(ii) Dry air at 27°C and 1 atm flows over a wet flat plate 50cm long at a velocity of 50m/s. Calculate the mass transfer coefficient of water vapour in air at the end of the plate. (8)

Refer Page No.126 Question No.03

B.E/B.Tech DEGREE EXAMINATION, APRIL/MAY_2014.
Fifth Semester
Mechanical Engineering
ME 2251 – HEAT AND MASS TRANSFER
(Regulation 2008)

Time: Three hours

Maximum: 100 marks

Answer ALL questions

PART A – (10 x 2= 20 marks)

1. Write any two examples of heat conduction with heat generation.
2. Define critical thickness of insulation with its significance.
Refer page no: 10, Q.No. 7
3. Differentiate viscous sub layer and buffer layer.
Refer page no. 35, Q. No 3
4. Define grashoff number and prandtl number.
Refer page no. 36, Q. No 4
5. What are the different regimes involved in pool boiling.
Refer page no. 58, Q. No 3
6. Write down the relation for overall heat coefficient in heat exchanger with fouling factor.
Refer page no. 65, Q. No 4
7. Define irradiation and emissive power.
Refer page no. 92&93, Q. No 5 & 10
8. Write down any two shape factor algebra.
9. State fick's law of diffusion.
Refer page no.119, Q. No 7
10. Write down the analogous terms in heat and mass transfer.

Part –B (5X16=80 Marks)

11. (a) Drive the general heat conduction equation in cylindrical coordinate and solve the following. (8)

Refer page no. 14, Q. No 1

Hot air at a temperature of 65⁰C is flowing through steel pipe of 120 mm diameter. The pipe is covered with tow layer of different insulating

material of thickness 60mm and 40mm, and their corresponding thermal conductivities are 0.24 and 0.4W/m²K. The inside and outside heat transfer coefficients are 60 W/m²K and 12 W/m²K respectively. The atmosphere is at 20°C. Find the rate of heat loss from 60 m length of pipe.

Refer page no.20, Q. No 5 (similar) (8)

(OR)

(b) Drive the heat dissipation equation through pin fin with insulated end and solve the following. (8)

A temperature rise of 50°C in a circular shaft of 50 mm diameter is caused by the amount of heat generated due to friction in the bearing mounted on the crankshaft. The thermal conductivity of shaft material is 55 W/mK and heat transfer coefficient is 7 W/m²K. Determine the amount of heat transferred through shaft assume that the shaft is a rod of infinite length. (8)

12.(a) Using dimensional analysis find dimensionless groups involved in free convection any solved the following.

A horizontal heated plate measuring 1.5m x 1.1m and at 215°C, facing upwards is placed in still air at 25°C. Calculate the heat loss by natural convection. Use the relation

$H=3.05(T_f)^{1/4}$, T_f = Mean film temperature. (16)

(OR)

(b) Explain development of hydrodynamic and thermal boundary layers with suitable figure and solve the following Refer page no.45, Q. No 5

In a straight tube of 50mm diameter, water is flowing at 15m/s. The tube surface temperature is maintained at 60°C and the flowing water is heated from the inlet temperature 15°C to an outlet temperature of 45°C calculate the heat transfer coefficient from the tube surface to the water and length of the tube. (16)

13.(a) Explain nucleate boiling and solve the following.

Refer page no.68, Q. No 1

A wire of 1mm diameter and 150mm length is submerged horizontally in water at 1 bar. The wire carries a current of 131.5 ampere with an applied voltage of 2.15 volt. If the surface of the wire is maintained at 180°C, calculate the heat flux and the boiling heat transfer coefficient. (16)

Refer page no.72, Q. No 3

(OR)

(b) Classify the heat exchangers, draw temperature distribution in a condenser and evaporator and derive the expression for effectiveness of parallel flow heat exchanger by NTU method. (16)

Refer page no.80, Q. No 9

14.(a) State laws of radiation and solve the following

Assuming the sun to be black emitting radiation with maximum intensity at $\lambda=0.5\mu\text{m}$, calculate the surface temperature of the sun and heat flux at its surface. (16)

(OR)

(b) Derive the relation for heat exchange between infinite parallel planes and solve.

Refer page no.113, Q. No 9

Consider double wall as two infinite parallel planes. The emissivity of the walls is 0.3 and 0.8 respectively. The space between the walls is evacuated. Find the heat transfer/unit area when inner and outer surface temperatures are 300K and 260K. To reduce the heat flow, a shield of polished aluminium with $\epsilon=0.05$ is inserted between the walls. Find the reduction in heat transfer. (16)

Refer page no.96, Q. No 2 (similar problem)

15.(a) Explain different modes of mass transfer and derive the general mass diffusion equation in stationary media. (16)

(OR)

(a) Explain Reynold's number, Sherwood number, Schmidt number and solve the following.

A vessel contains a binary mixture of oxygen and nitrogen with partial pressures in the ratio 0.21 and 0.79 at 15°C. The total pressure of the mixture is 1.1 bar. Calculate the following

- i. Molar concentrations (4)
- ii. Mass densities(4)
- iii. Mass fractions(4)
- iv. Molar fractions of each species. (4)

Refer page no.122, Q. No 1

B.E/B.Tech. DEGREE EXAMINATION, NOVEMBER/DECEMBER 2013.

Fourth Semester

Mechanical Engineering

ME 2251 – HEAT AND MASS TRANSFER

(Regulation 2008)

Answer ALL questions.

PART A – (10 X 2 = 20 marks)

1. Define Fourier's law of conduction.
Refer Page No.9, Question No.1
2. A plane wall 10 cm thick generates heat at the rate of $4 \times 10^4 \text{ W/m}^3$ when an electric current passed through it. The convective heat transfer coefficient between each face of the wall and ambient air $50 \text{ W/m}^2\text{K}$. Determine the surface temperature. Assume the air temperature to be 20°C and $k(\text{for wall})=15 \text{ W/mK}$.
3. Define the term thermal boundary layer.
Refer Page No.40, Question No.15
4. Why heat transfer coefficient for natural convection is much lesser than that for forced convection?
Refer Page No.38, Question No.9
5. Distinguish the pool boiling from forced convection boiling.
Refer Page No.66, Question No.7
6. What are limitations of LMTD method? How is ϵ -NTU method superior to LMTD method?
Refer Page No.65, Question No.6
7. State Planck's law?
Refer Page No.91, Question No.1
8. How radiation from gases differs from solids?
Refer Page No.92, Question No.8
9. Write general mass diffusion equation.
10. Define Schmidt and Lewis numbers. What is the physical significance of each?
Refer Page No.120, Question No.10

Part -B (5X16=80 Marks)

11. a) The door of an industrial furnace is $2\text{m} \times 4\text{m}$ in surface area and is to be insulated to reduce the heat loss to not more than 1200 W/m^2 . The interior and exterior walls of the door are 10 mm and 7mm thick steels ($k= 25 \text{ W/m}^2\text{K}$). Between these two sheets, a suitable thickness of insulation material is to be placed. The effective gas temperature inside the furnace is 1200°C and the overall heat transfer coefficient between the gas and door is $20 \text{ W/m}^2\text{K}$. The heat transfer coefficient outside the door is $5 \text{ W/m}^2\text{K}$. The surrounding air temperature is 20°C . Select suitable insulation material and its size. (16)

(Or)

b) (i) A turbine blade 6 cm long and having a cross-sectional area 4.65 cm^2 and parameter 12 cm is made of stainless steel ($k= 23.3 \text{ w/m k}$).The temperature at the root is 500°c .The blade is exposed to a hot gas at 870°c .The heat transfer coefficient between blade surface and gas is $442 \text{ w/m}^2 \text{ k}$.Determine the temperature distribution and rate of heat flow at the root of the blade. Assume the tip of the blade to be insulated. (8)

ii) An ordinary egg can be approximated as a 5 cm diameter sphere. The egg is initially at a uniform temperature of 5°c and is dropped into boiling water at 95°c .Taking the convection heat transfer coefficient to be $h = 200 \text{ w/m}^2\text{c}$, determine how long it will take for the centre of the egg to reach 70°c . (8)

Refer similar problem Page No. 27, Question No. 9

12. a) i) Explain in detail about boundary layer concept (6)

Refer similar problem Page No. 48, Question No. 5

ii) An aeroplane flies with a speed of 450 km/h at a height where the surrounding air has a temperature of 1°c and pressure of 65 cm of Hg. The aeroplane wing idealized as a flat plate 6m long ,1.2 m wide is maintained at 19°c .If the flow is made parallel to the 1.2 m width calculate: (1) Heat loss from the wing; (2) Drag force on the wing. (10)

(Or)

b) A two stroke motor cycle petrol engine cylinder consists of 15 annular fins. If outside and inside diameters of each fin are 200 mm and 100mm respectively. The average fin surface temperature is 475°c and they are exposed in air at 25°c .Calculate the heat transfer rate from the fins for the following condition (i) When motor cycle is at rest (ii)when motor cycle is running at a speed of 60 km/h. The fin may be idealized as a single horizontal flat plate of same area. (16)

13. a) i) Explain the various regions of flow boiling in detail (6)

Refer similar Page No. 68, Question No. 1

ii) The outer surface of a vertical tube which is 6m long and has an outer diameter of 80 mm, is exposed to saturated steam at atmospheric pressure and is maintained at 50°c by the flow of cool water through the tube. What is the rate of heat transfer to coolant and what is the rate at which steam is condensed at the surface? (10)

(Or)

b) A Counter -Flow Concentric Tube Heat Exchanger Is Used To Cool the lubricating oil for a large industrial gas turbine engine. The flow rate of cooling water through the inner tube ($d_i=20\text{mm}$) is 0.18 kg/s while the flow rate of oil through the outer annular ($d_o=40\text{mm}$)is 0.12 kg/s . The inlet and outlet temperatures of an are 95°c and 65°c respectively. The water enters at 30°c to the exchanger Neglecting tube wall thermal resistance, fouling factors and heat loss to the surroundings, Calculate the length of the tube . Take the following properties at the bulk mean temperature:

Engine oil at 80°c , $C_p = 2131 \text{ J/Kg}^\circ\text{c}$, $\mu = 0.0325 \text{ N-sm}^2$; $k = 0.138 \text{ w/m}^\circ\text{c}$;

water at 35°C; $C_p = 4174 \text{ J/Kg}^\circ\text{C}$, $\mu = 725 \times 10^{-3} \text{ N}\cdot\text{sm}^2$; $k = 0.625 \text{ W/m}^\circ\text{C}$, $P_r = 4.85$ (16)
Refer similar problem Page No. 75, Question No. 5

14 a) i) A Truncated cone has top and bottom diameters of 10 and 20 cm and a height of 10 cm. Calculate the shape factor between the top surface and the side and also the shape factor between the side and itself. (10)

ii) Emissivities of two large parallel plates maintained at 800°C and 300°C and 0.3 and 0.5 respectively. Find the net radiant heat exchange per square meter for these plates. (6)

Refer similar problem Page No. 101, Question No. 3

(Or)

b) A 12 mm outside diameter pipe carries a cryogenic fluid at 90 K. Another pipe of 15 mm outside diameter and 290 K surrounds it coaxially and the space between the pipes is completely (i) determine the radiant heat flow for 3.5 m length of pipe if the surface emissivity for both surfaces is 0.25 (ii) Calculate the percentage reduction in heat flow if a shield of 13.5 mm diameter and 0.06 surface emissivity is placed between pipes. (16)

15 a) Air is contained in a tyre tube of surface area 0.5 m² and wall thickness 10 mm. The pressure of air drops from 2.2 bar and 2.18 bar in a period of 6 days. The solubility of air in the rubber is 0.072 m³ of air per m³ of rubber at 1 bar. Determine the diffusivity of air in rubber at the operating temperature of 300 K if the volume of air in the tube is 0.028 m³. (16)

(Or)

b) Air at 35°C and 1 atmospheric pressure flows at a velocity of 60 m/s over (i) a flat plate 0.5 m long (ii) a sphere 5 cm in diameter. Calculate the mass transfer coefficient of water in air. Neglect the concentration of vapour in air. (16)

Refer similar problem Page No. 126, Question No. 3 (similarity)

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