PIE Tech
POLLACHI INSTITUTE OF ENGINEERING AND TECHNOLOGY
(Approved by AICTE and Affiliated to Anna University)
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Department of Mechanical Engineering

Regulation 2021
III Year – VI Semester
ME3691 HEAT AND MASS TRANSFER

UNIT - I

Heat Transfer

It's can be defined as the transmission of energy from one region to another due to temp difference.

Modes of heat transfer

* Conduction
* Convection
* Radiation

Conduction:

Heat conduction is a mechanism of heat transfer from a region of high temp to region of low temp. within medium (solid, Liquid, gas)

OR

Different medium in direct physical contact

Convection:

It's a process of heat transfer that will occur between a solid surface & a Fluid medium, when they are diff temp.

Convection is possible only in the presence of fluid medium.

Radiation:

The heat transfer from 1 body to another body without any transmitting medium is.

It's a electro magnetic wave phenomenon.

Fourier's law of conduction.

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

A - Area in m^2

$\frac{dT}{dx}$ - Temp gradiant, k/m

k - Thermal conductivity W/mk

Q - Rate of heat conduction

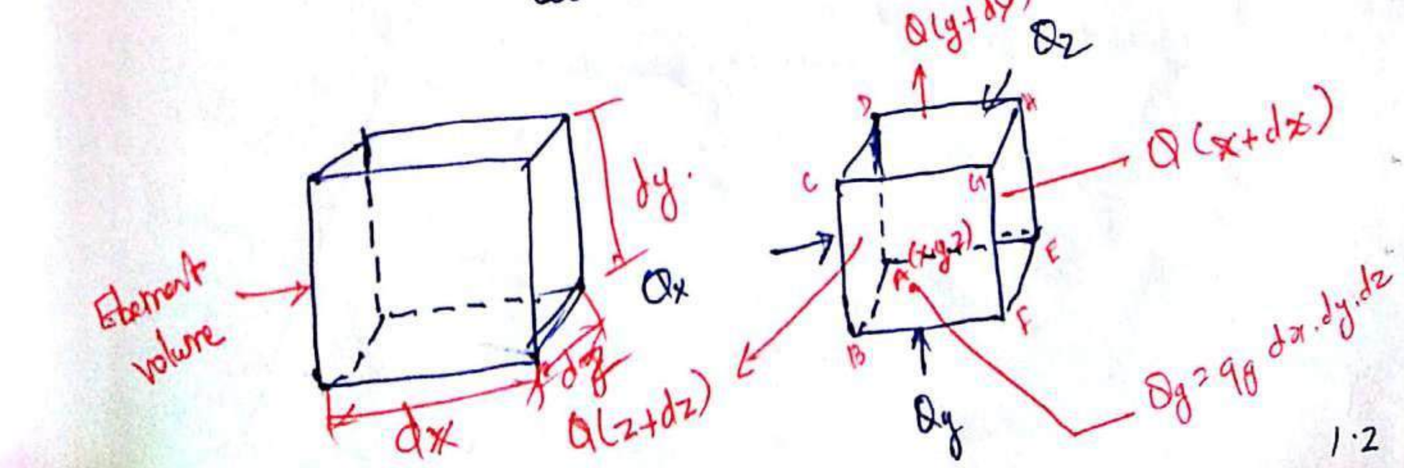
The rate of heat conduction is proportional to the area measured normal to the direction of heat flow and to the temp gradiant in that direction.

Thermal conductivity:-

'Ability of a substance to conduct heat.

General heat conduction equation in

Cartesian coordinates:



* Consider small rectangular element of sides $dx, dy, \& dz$

* The energy balance of the ~~regu~~ rectangular element is obtained from 1st law of Thermodynamics.

$\Delta U = Q - W$

~~$\Delta U + Q = W$~~ $\Delta U + W = Q$

$$\begin{bmatrix} \text{Net heat} \\ \text{conducted into} \\ \text{element from} \\ \text{all coordinate} \\ \text{directions} \end{bmatrix} + \begin{bmatrix} \text{Heat} \\ \text{generated} \\ \text{with the} \\ \text{element} \end{bmatrix} = \begin{bmatrix} \text{Heat} \\ \text{stored} \\ \text{the element} \end{bmatrix} \quad \text{------>} \; 1.1$$

Net heat conducted into element from all the coordinate directions:

- Let q_x be the heat flux in a direction of face ABCD
and
- $q_x + dx$ be the heat flux in " " " EFGH

The rate of heat flow into the elements in x direction through the face ABCD is:

$$Q_x = q_x \, dy \, dz$$

$$\boxed{Q_x = -k_x \frac{\partial T}{\partial x} dy \, dz} \quad \text{------>} (1.2)$$

where k – T/c

$\frac{\partial T}{\partial x}$ = T/m

The rate of heat flow out of the elements in x direction through the face EFGH is

$$Q_{x+dx} = Q_x + \frac{\partial}{\partial x}(Q_x)\, dx$$

$$-k_x \frac{\partial T}{\partial x} dy\, dz + \frac{\partial}{\partial x}\left[-k_x \frac{\partial T}{\partial x} dy\, dz\right] dx$$

$$\boxed{Q_{x+dx} = -k_x \frac{\partial T}{\partial x} dy\, dz - \frac{\partial}{\partial x}\left[k_x \frac{\partial T}{\partial x}\right] dx\, dy\, dz} \rightarrow 1.3$$

Subtracting (1.2 − 1.3)

~~$Q_x - Q_{x+d}$~~

$$Q_x - Q_{(x+dx)} = -k_x \frac{\partial T}{\partial x} dy\, dz - \left[-k_x \frac{\partial T}{\partial x} dy\, dz - \frac{\partial}{\partial x}\left[k_x \frac{\partial T}{\partial x}\right] dx\, dy\, dz\right]$$

$$= \cancel{-k_x \frac{\partial T}{\partial x} dy\, dz} + \cancel{k_x \frac{\partial T}{\partial x} dy\, dz} + \frac{\partial}{\partial x}\left[k_x \frac{\partial T}{\partial x}\right] dx\, dy\, dz$$

$$\boxed{Q_x - Q_{x+dx} = \frac{\partial}{\partial x}\left[k_x \frac{\partial T}{\partial x}\right] dx\, dy\, dz} \rightarrow (1.4)$$

Similarly:-

$$Q_y - Q_{(y+dy)} = \frac{\partial}{\partial y}\left[k_y \frac{\partial T}{\partial x}\right] dx\, dy\, dz$$

$$Q_z - Q_{(z+dz)} = \frac{\partial}{\partial z}\left[k_z \cdot \frac{\partial T}{\partial x}\right] dx\, dy\, dz.$$

Adding (1.4) + (1.5) + (1.6)

$$\boxed{= \left[\frac{\partial}{\partial x}\left[k_x \frac{\partial T}{\partial x}\right] + \frac{\partial}{\partial y}\left[k_y \frac{\partial T}{\partial y}\right] + \frac{\partial}{\partial z}\left[k_z \frac{\partial T}{\partial z}\right]\right] dx\, dy\, dz}$$

Heat stored in the element

WKT,

Heat stored in the element = Mass of the element × Sp. heat of the element × Rise in temp of element

$$= m \times C_p \times \frac{\partial T}{\partial t}$$

$$= \rho \times dx\,dy\,dz \times C_p \times \frac{\partial T}{\partial t}$$

// Density = mass / volume
mass = De. × volu

$$\boxed{\text{Heat stored in element} = \rho C_p \frac{\partial T}{\partial t}\,dx\,dy\,dz} \longrightarrow (1.8)$$

Heat Generated with in the element:

" " is given by

$$\boxed{Q = \dot{q}\,dx\,dy\,dz} \longrightarrow (1.9).$$

Sub equation 1.7 & 1.8 & 1.9 in equ (1.1)

$$1.1 = \left[\frac{\partial}{\partial x}\left[k_x \frac{\partial T}{\partial x}\right] + \frac{\partial}{\partial y}\left[k_y \frac{\partial T}{\partial y}\right] + \frac{\partial}{\partial z}\left[k_z \frac{\partial T}{\partial z}\right]\right] dx\,dy\,dz +$$

$$\dot{q}\,dx\,dy\,dz = \rho C_p \frac{\partial T}{\partial t}\,dx\,dy\,dz$$

{net heat conducted into ele from all the coordinate direction} + {Heat generated in elem} = {heat stored in ele}

Considering the material is isotropic so Isotropic

$$\boxed{k_x = k_y = k_z = k = \text{constant}}$$

$$\left[\frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2} + \frac{\partial^2 T}{\partial z^2}\right] k + \dot{q} = \rho C_p \frac{\partial T}{\partial t}$$

÷ k

$$'' \qquad \frac{\dot{q}}{k} = \frac{1}{\alpha}\frac{\partial T}{\partial t}$$

α = Themal diffusivity $= \frac{k}{\rho c_p}$ - m^2/s.

Case (i) No heat Source:

equ 1·10 · $\frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2} + \frac{\partial^2 T}{\partial z^2} = \frac{1}{\alpha}\frac{\partial T}{\partial t}$ ...... 1·11

diffusion equ on Fourier's equ

Case (ii) steady state conditions:-

" the temp does not change with time.

So, $\frac{\partial T}{\partial t} = 0$ so, (1.10) redus to equ

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

(or)

$$\boxed{\nabla^2 T + \frac{\dot{q}}{k} = 0}$$ Poisson's equ.

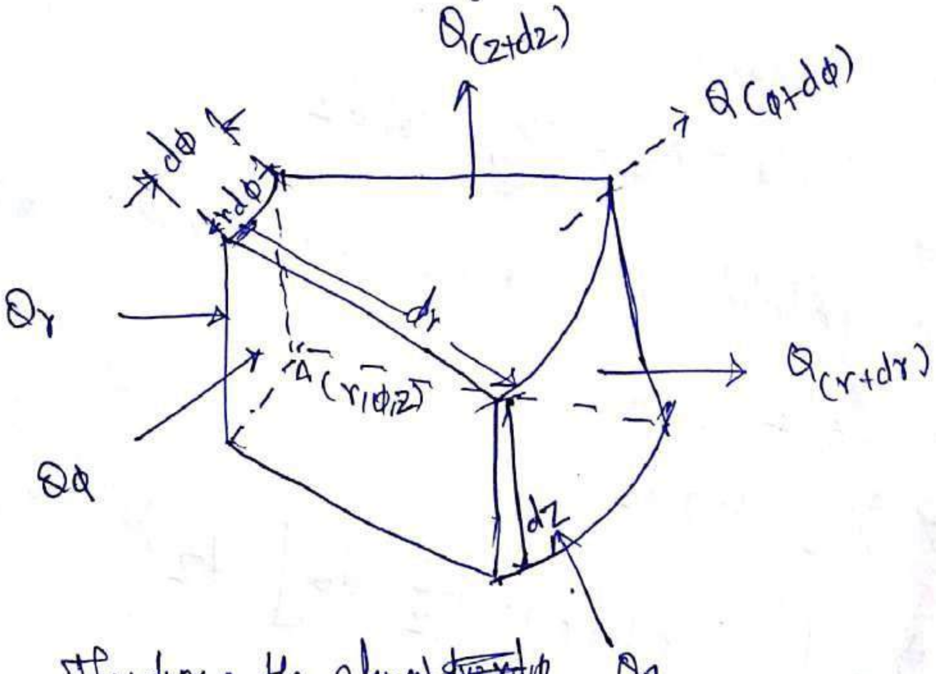
absence of internal heats generation

$\nabla^2 T = 0$ Laplace equation.

General heat conduction Equ in cylindrical co-ordinates.

The cartesian coordinate system is not applicable for the solids like cylinder, cones, sphere etc.

Consider a small cylinder element of sides dr, $d\phi$ & dz



The volume of the element ~~dr·rdφ~~

$$dv = r\,d\phi \cdot dr\,dz$$

Let us assume that thermal conductivity k, sp.h c_p & ρ density are constant

The energy balance to 1st law of thermodynamics

Net heat conduct into element from all the co-or directions	+	Heat generated within the element	=	Heat stored in the element	$\longrightarrow$ (1)

Net heat conducted into element from all the co-ordinate direction.

I. Heat entering into the element through (r, φ) plane in time $d\theta$

$$Q_z = -k(r \cdot d\phi\, dr) \frac{\partial T}{\partial z} d\theta$$

" leaving " " " , " " $d\theta$

$$Q_{z+dz} = Q_z + \frac{\partial}{\partial z}(Q_z)\, dz$$

Net heat conducted into the element through (r, φ) plane in time

$$= Q_z - Q_{z+dz}$$

$$= -\frac{\partial}{\partial z}(Q_z)\, dz$$

$$= \cdot \frac{\partial}{\partial z}\left[k(r d\phi \cdot dr) \cdot \left(\frac{\partial T}{\partial z}\right) d\theta\right] dz$$

$$= k\left[\frac{\partial^2 T}{\partial z^2}\right] [dr \cdot r d\phi \cdot dz)\, d\theta$$

$$\left.\begin{matrix}\text{Net heat conducted} \\ \text{through } (r, \phi) \text{ Plane}\end{matrix}\right\} = k\left[\frac{\partial^2 T}{\partial z^2}\right] [dr, r d\phi, dz) d\theta$$

II) Heat " " " (φ, z) plane in time $d\theta$.

$$Q_r = -k(r\, d\phi\, dz) \frac{\partial T}{\partial r} d\theta$$

" " leaving " (φ, z) " " $d\theta$

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

Net heat conducted into the element through (z, r) plane in time $d\theta$

$$Q_\phi - Q_{\phi+d\phi} = k\left[\frac{1}{r^2}\frac{\partial^2 T}{\partial \theta^2}\right](dr\ r d\phi\ dz)\ d\theta \rightarrow (4)$$

Add 2 + 3 + 4

$$k\,(dr\ r d\phi\ dz)\ d\theta\left[\frac{\partial^2 T}{\partial z^2} + \frac{\partial^2 T}{\partial r^2} + \frac{1}{r}\frac{\partial T}{\partial r} + \frac{1}{r^2}\frac{\partial^2 T}{\partial \theta^2}\right]$$

All coordinate directions

$$= k\,(dr\ r d\phi \cdot dz)\ d\theta\left[\frac{\partial^2 T}{\partial r^2} + \frac{1}{r}\frac{\partial T}{\partial r} + \frac{1}{r^2}\frac{\partial^2 T}{\partial \phi^2} + \frac{\partial^2 T}{\partial z^2}\right].$$

Heat generated within the element:-

Total .. .. .. is given by

$$Q = \dot{q}\,(dr\ r d\phi\ dz)\ d\theta \longrightarrow (5)$$

Heat stored in the element

The increase in internal energy of the element is equal to the net heat stored in the element.

Increase in internal energy = Net heat stored in the element

Net heat conducted into the element through (ϕ, z) Plane in

$$= Q_r - Q_{r+dr}$$

$$= -\partial/\partial r\,(Q_r)\,dr$$

$$= -\partial/\partial r\left[-k\,(r\,d\phi\,dz)\cdot\left(\frac{\partial T}{\partial r}\right)d\theta\right]dr$$

$$= k\,(dr\,d\phi\,dz)\ \partial/\partial r\left[r\cdot\partial T/\partial r\right]d\theta$$

$$= k\,(dr\cdot r\,d\phi\cdot dz)\left[\frac{\partial^2 T}{\partial r^2} + 1/r\ \partial T/\partial r\right]d\theta$$

Net heat conducted through (ϕ, z) Plane $= k\cdot(dr\cdot r\,d\phi\cdot dz)\ \frac{\partial^2 T}{\partial r^2} + \left[1/r\ \partial T/\partial r\right]d\theta$

III Heat ,, entering ,, (z, r) ,, in time $d\theta$

$$Q_\phi = -k(dr\cdot dz)\,\frac{\partial T}{r\,\partial\theta}\,d\theta$$

,, leaving ,, (z r) .. time $d\theta$

$$Q_{\phi+d\phi} = Q_\phi + \frac{\partial}{r\,\partial\phi}\left(Q_\phi\right)r\,d\phi$$

$$= \rho\,(dr\ r d\phi \cdot dz)\ C_p\ \frac{\partial T}{\partial \theta} \times d\theta \longrightarrow (5)$$

Substituting eqn 5, 4, 3 in eqn (1).

$\div$ by $(dr\ r d\phi \cdot dz)\ d\theta$

$$k \left[\frac{\partial^2 T}{\partial r^2} + \frac{1}{r}\frac{\partial T}{\partial r} + \frac{1}{r^2}\frac{\partial^2 T}{\partial \phi^2} + \frac{\partial^2 T}{\partial z^2}\right] + \dot{q} = \rho \cdot C_p \frac{\partial T}{\partial \theta}$$

$$+ \frac{\dot{q}}{k} = \frac{\rho C_p}{k} \cdot \frac{\partial T}{\partial \theta}$$

"

$$\therefore \alpha = k/\rho C_p$$

" $$= \frac{1}{\alpha}\frac{\partial T}{\partial \theta}$$

$$\boxed{\frac{\partial^2 T}{\partial r^2} + \frac{1}{r}\frac{\partial T}{\partial r} = 0}$$

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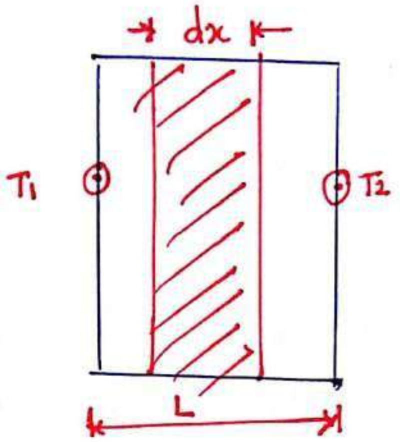
Conduction of heat through a slab or Plane wall



Considering a slab of uniform k, L thickness, with inner Temp T_1 & outer temp T_2

Let us consider a small element one of thickness 'dx'

From Fourier law of conduction, WKT

$$Q = -kA \frac{dT}{dx}$$

$$Q \cdot dx = -kA\, dT$$

$\int$ the above equ below the limits of 0 to L & T_1 to T_2

$$\Rightarrow \int_0^L Q\,dx = -\int_{T_1}^{T_2} kA\, dT$$

$$\Rightarrow Q \int_0^L dx = -kA \int_{T_1}^{T_2} dT$$

$$\Rightarrow Q[x]_0^L = -kA [T]_{T_1}^{T_2}$$

$$\Rightarrow Q(L-0) = -kA(T_2 - T_1)$$

$$\Rightarrow Q \times L = kA(T_1 - T_2)$$

$$\Rightarrow Q = \frac{kA}{L}[T_1 - T_2]$$

$$\boxed{Q \Rightarrow \frac{T_1 - T_2}{L/kA}} = Q = \frac{\Delta T_{overall}}{R} \rightarrow ①.$$

where

$\Delta T = T_1 - T_2$

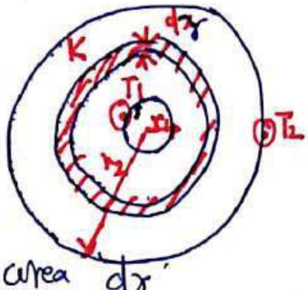
$R = L/kA$ - Thermal resistance of slab.

1.14

Conduction of Heat through Hollow cylinder

Consider a hollow cylinder of inner radius r_1 & outer radius r_2, inner temp & outer temp T_1 & T_2

k



From Fourier law of conduction

$Q = -KA \frac{dT}{dr}$

Let us consider small element area dr

Area of cylinder is $2\pi r L$

So,

$$Q = -k\, 2\pi r L\, dT/dr$$

$$Q \times \frac{dr}{r} = -k 2\pi L\, dT$$

Integrating the above equ from r_1 to r_2 & T_1 to T_2

$$\Rightarrow Q \int_{r_1}^{r_2} \frac{dr}{r} = -k 2\pi L \int_{T_1}^{T_2} dT$$

$$\Rightarrow Q [\ln r]_{r_1}^{r_2} = -k 2\pi L [T]_{T_1}^{T_2}$$

$$\Rightarrow Q[\ln r_2 - \ln r_1] = -k 2\pi L [T_2 - T_1]$$

$$\Rightarrow Q \ln\left[\frac{r_2}{r_1}\right] = 2\pi L k [T_1 - T_2].$$

$$\Rightarrow Q = \frac{2\pi L k [T_1 - T_2]}{\ln\left(\frac{r_2}{r_1}\right)}$$

$$\Rightarrow Q = \frac{T_1 - T_2}{\frac{1}{2\pi L k} \ln\left(\frac{r_2}{r_1}\right)}$$

$$\Rightarrow Q = \frac{\Delta T_{overall}}{R}$$

Where,

$T_1 - T_2 = \Delta T$

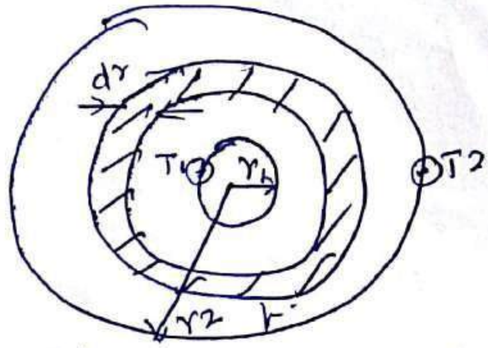
$R = \frac{1}{2\pi L k} \ln\left(\frac{r_2}{r_1}\right)$

Thermal Resistance of the hollow cylinder.

Conduction of Heat through hollow Sphere.

Consider a hollow sphere of inner radius r_1, outer radius r_2,

Inner & Outer temp : T_1 & T_2



Let us consider a small element area of thickness 'dr' from Fourier law of heat conduction

$$Q = -kA\, dT/dr$$

Area of sphere is $4\pi r^2 = A$

$$Q = -k 4\pi r^2 \frac{dT}{dr}$$

$$Q \times \frac{dr}{r^2} - k \times 4\pi dT$$

$\int$ on both sides

$$Q \int_{r_1}^{r_2} \frac{dr}{r^2} = -4\pi k \int_{T_1}^{T_2} dT$$

$\int \frac{dr}{r^2} = -1/r$

$\frac{dx}{x^2}, -1/x$

$$Q \left[\frac{-1}{r}\right]_{r_1}^{r_2} = -4\pi k [T]_{T_1}^{T_2}$$

$$Q \left[\frac{1}{r_1} - \frac{1}{r_2}\right] = -4\pi k [T_2 - T_1]$$

$$Q \left[\frac{r_2 - r_1}{r_1 r_2}\right] = 4\pi k [T_1 - T_2]$$

$$Q = \frac{4\pi k [T_1 - T_2]}{\frac{r_2 - r_1}{r_1 r_2}}$$

$$Q = \frac{T_1 - T_2}{\frac{r_2 - r_1}{4\pi k [r_1 r_2]}}$$

$$Q = \frac{\Delta T_{overall}}{R}$$

$$Q = \frac{\Delta T_{overall}}{R}$$

where $\Delta T = T_1 - T_2$

$$R = \frac{r_2 - r_1}{4\pi k (r_1 r_2)}$$: Thermal Resistance of hollow sphere.

1.16

Newtons Law of Cooling

Heat transfer by Convection is given by Newton law of cooling

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

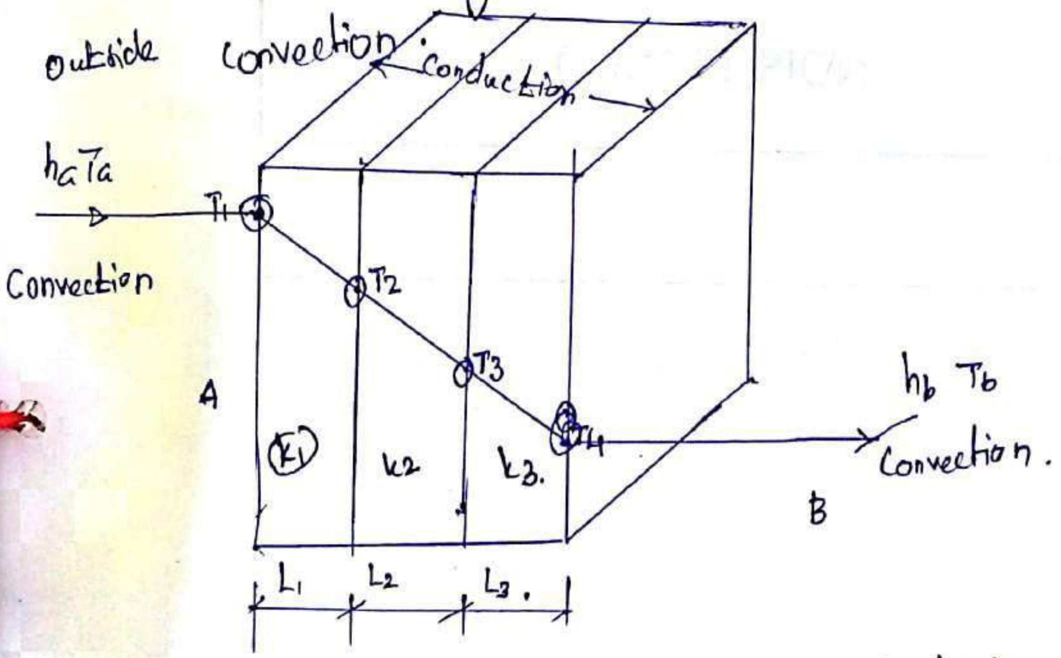
A - Area exposed to heat transfer in m^2

h - heat transfer co-efficient in W/m^2K

T_s - Temp of the surface in K

T_a = Temp of the Fluid in K.

Heat transfer Through a composite plane wall with Inside & outside convection



Composite wall thickness L_1, L_2, L_3

Thermal conductivity k_1, k_2, k_3

Interior & Exterior ~~bay~~ surface of the system are subjected to convection at mean temp T_a & T_b.

heat transfer co-efficient h_a & h_b

From Newton law of cooling ~~WKT~~ WKT

Heat transfer by convection at side A is

$$Q = h_a A [T_a - T_1] \rightarrow ①$$

Heat " " " at slab (1) is

$Q = \frac{kA}{L}(T_1 - T_2) \Rightarrow$ || $$Q = \frac{k_1 A \cdot [T_1 - T_2]}{L_1} \rightarrow ②$$

Heat " " " at slab(2) is

$$Q = \frac{k_2 A \, [T_2 - T_3]}{L_2} \rightarrow ③$$

Heat " " " at slab (3) is

$$Q = \frac{k_3 A \, (T_3 - T_4)}{L_3} \rightarrow ④$$

Heat transfer by convection at side B is

$$Q = h_b A (T_4 - T_b). \rightarrow ⑤$$

WKT :-

$$T_a - T_1 = Q \times \frac{1}{h_a A}$$

$$T_1 - T_2 = Q \times \frac{L_1}{k_1 A}$$

$$T_2 - T_3 = Q \times \frac{L_2}{k_2 A}$$

$$T_3 - T_4 = Q \times \frac{L_3}{k_3 A}$$

$$T_4 - T_b = Q \times \frac{1}{h_b A}$$

1.18

Adding both sides of the above equ

$$T_a - T_b = Q \left[\frac{1}{h_a A} + \frac{L_1}{k_1 A} + \frac{L_2}{k_2 A} + \frac{L_3}{k_3 A} + \frac{1}{h_b A} \right]$$

$$Q = \frac{T_a - T_b}{\left[1/h_a A + L_1/k_1 A + L_2/k_2 A + L_3/k_3 A + \frac{1}{h_b A} \right]}$$

$$Q = \frac{\Delta T_{overall}}{R}$$

|| T_a & T_b = heat transfer co-efficients

Where

$\Delta T = T_a - T_b$

Thermal Resistance $R = R_a + R_1 + R_2 + R_3 + R_b$

$\therefore R_a = \frac{1}{h_a A}$

$R_1 = L_1 / k_1 A$

WKT

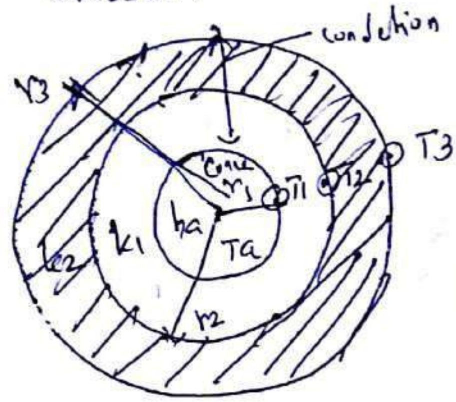
$$R = \frac{1}{UA}$$

$$Q = \frac{T_a - T_b}{1/UA}$$

$$\dot{Q} = UA \left[T_a - T_b \right].$$

U = over all heat transfer co-efficient $W/m^2 k$.

Heat transfer Through composite pipe or cylinder with Inside & outside convection.



Ta - hot Fluid temp

ha - heat transfer coefficient

k_1 & k_2 - Thermal conductivity.

Tb - cold fluid temp

hb - heat transfer co-efficient

Heat transfer by convection at side A is

$Q = h_a A (T_a - T_1)$

$\boxed{\because \text{Area} = 2\pi r_1 L}$

So,

$Q = 2\pi r_1 L \cdot h_a (T_a - T_1)$

Heat transfer by conduction at section 1 is

$$Q = \frac{2\pi L k_1 (T_1 - T_2)}{\ln(r_2/r_1)}$$

Similar At section 2

$$Q = \frac{2\pi L k_2 (T_2 - T_3)}{\ln r_3/r_2}$$

Heat transfer by convection at side B is

At Section 3

$Q = h_b A (T_3 - T_b)$

$Q = 2\pi r_3 L \; h_b (T_3 - T_b)$

W.k.t.

$$T_a - T_1 = \frac{Q}{2\pi L r_1 h_a}$$

$$T_1 - T_2 = \frac{Q}{2\pi L k_1} \ln\left(\frac{r_2}{r_1}\right)$$

$$T_2 - T_3 = \frac{Q}{2\pi L k_2} \ln\left(\frac{r_3}{r_2}\right)$$

$$T_3 - T_b = \frac{Q}{2\pi r_3 L h_b}$$

Add both side above equ

$$T_a - T_b = \frac{Q}{2\pi L}\left[\frac{1}{h_a r_1} + \frac{\ln r_2/r_1}{k_1} + \frac{\ln (r_3/r_2)}{k_2} + 1/h_b r_3\right].$$

$$Q = \frac{2\pi L (T_a - T_b)}{\frac{1}{h_a r_1} + \frac{\ln(r_2/r_1)}{k_1} + \frac{\ln(r_3/r_2)}{k_2} + \frac{1}{h_b r_3}}$$

1.20

$$Q = \frac{T_a - T_b}{\frac{1}{2\pi L}\left[\frac{1}{h_a r_1} + \frac{\ln r_2/r_1}{k_1} + \frac{\ln (r_3/r_2)}{k_2} + \frac{1}{h_b r_3}\right]}$$

$$Q = \Delta T_{overall} / R$$

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

$$\Delta T_{overall} = T_a - T_b$$

WKT

$$R = \frac{1}{UA}$$

$$Q = \frac{T_a - T_b}{1/UA}$$

$$Q = UA[T_a - T_b].$$

where

U - Overall heat transfer co-efficent w/m^2k

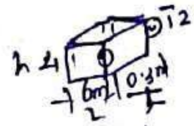
A = Area $= 2\pi r_3 L$

Date :

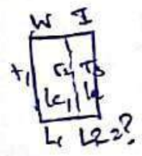
Ex. No. :

Problem's for slabs

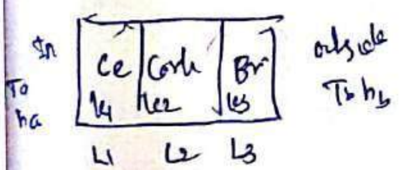
① Determine the heat transfer through the plane of length 6m, height 4m & thickness 0.30m. The temp of inner & outer surface are $100^\circ c$ & $40^\circ c$. Thermal conductivity of wall is $0.55 W/mK$ Q = watts
(HMT Pg No: 44 Seventh edition)



② A wall of 0.5m thickness having (k) of $1.2 W/mK$. The wall is to be insulated with a material having an average ~~ther~~ k of $0.3 W/mK$ $\Delta T = T_a - T_b$ (or) $T_1 - T_3$
Inner & outer temp are $1000^\circ C$ & $10^\circ C$ respectively. If heat transfer rate is Q/A $1400 W/m^2$ calculate the thickness of insulation. HMT PG NO = 44/45
$L_2 = ?$ m



③ The wall of a cold room is composed of three layers. The outer layer is brick 30cm thick. The middle layer is cork 20cm thick. The inner layer is cement 15cm thick. The temp of outside air is $25^\circ C$ (T_b) & on the inner air is $-20^\circ C$ (T_a). The film co-efficient for outside air & brick is $55.4 W/m^2K$ (h_a). Film co-efficient for inner air & cement is $17 W/m^2K$ (h_b). Take k for brick = $2.5 W/mK$
cork = $0.05 W/mK$
cem = $0.28 W/mK$
Find heat flow rate = Q/A HMT PG: 44/45



4) A wall of cold room is composed of three layers. The outer layer is ~~brick~~ Firewall 20cm thick, the middle layer is asbestos 10cm thick, the inside layer is cement 5 cm thick.
The temp of outside layer ~~of~~ air is 25°C and that on the inside air is -20°C.

the film co-efficients for outside air & ~~brick~~ Firewall h_1 is 45.4 W/m²K

The film co-efficients for Inside air Cement is 17 W/m²K.

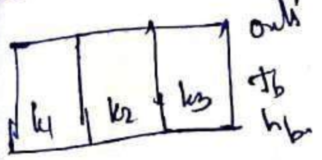
Find (i) Thermal resistance = R = ?

(ii) The heat flow ~~rat~~ rate Q/A = ?

k for Firewall = 3.45 W/mK

k for aspetus = 0.043 W/mK

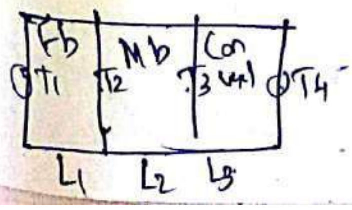
k for cement = 0.294 W/mK



For unit Area A = 1

5). A Furnace wall consists of 3 layer. The inner layer of 10cm thickness is made of ~~Glass~~ F.b Glasswhole (k = 1.04 W/mk). The intermediate layer of 25 cm thickness is made of masonry brick k = 6.69 W/mk) followed by 5cm thick Concreat wall (k = 1.37 W/mk). When the Furnace is continuous operation the inner surface of the Furnace is at 800°C while the outer Concreat surface is at 50°C. Calculate the rate of heat transfer loss per unit area wall, the temp at the Interface of F.b Glasswhol & masonry brick & ..
The temp at the interface of the masonry brick & concert.

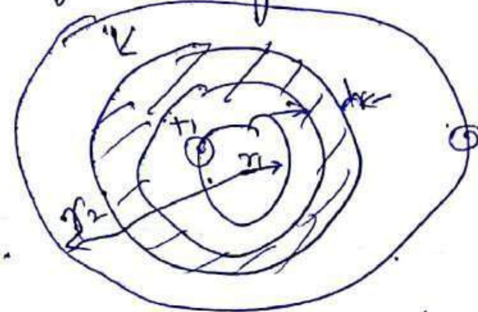
To find 1) Q/A
2) T2
3) T3
} ?



1.23

①

Conduction of heat through Hollow cylinder



inner radius r_1, r_2
outer
Tem T_1, T_2.
k.

Small element area 'dr'

Area of cylinder is $2\pi rL$

$$Q = -kA \frac{dT}{dr}$$

$$Q = -k\,2\pi rL \frac{dT}{dr}$$

$$Q \times \frac{dr}{r} = -k 2\pi L\, dT$$

∫ the eq r_1 to r_2 T_1 to T_2.

$$Q\int_{r_1}^{r_2} \frac{dr}{r} = -k2\pi L\int_{T_1}^{T_2} dT$$

$$Q[\ln]_{r_1}^{r_2} = -k2\pi L[T]_{T_1}^{T_2}$$

$$Q(\ln r_2 - \ln r_1) = -k2\pi L\,[T_2 - T_1]$$

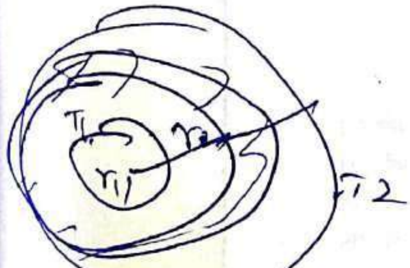
$$Q\ln\left(\frac{r_2}{r_1}\right) = 2\pi Lk\,(T_1 - T_2)$$

$$Q = \frac{2\pi Lk(T_1 - T_2)}{\ln r_2/r_1}$$

$$= \frac{T_1 - T_2}{\frac{\ln \frac{r_2}{r_1}}{2\pi Lk}} \quad T_1 - T_2 = \Delta T \quad \ln\frac{r_2}{r_1} = R$$

$$Q = \frac{\Delta T_{overall}}{R}$$

T_b
h_b



Area of sphere is $4\pi r^2$

$$Q = K\,4\pi r^2 \frac{dT}{dr}$$

$$Q \times \frac{dr}{r^2} = -k \times 4\pi\, dT$$

∫ both

$$Q\int_{r_1}^{r_2} \frac{dr}{r^2} = -4\pi k\int_{T_1}^{T_2} dT$$

$$Q\left[\frac{-1}{r}\right]_{r_1}^{r_2} = -4\pi k\,[T]_{T_1}^{T_2}$$

$$Q\left(\frac{1}{r_1} - \frac{1}{r_2}\right) = -4\pi k\, T_2 - T_1$$

$$Q\,\frac{r_2 - r_1}{r_1 r_2} = 4\pi k\, T_1 - T_2$$

$$Q = \frac{4\pi k\, T_1 - T_2}{\left(\frac{r_2 - r_1}{x\, r_1 r_2}\right)}$$

$$Q = \frac{''}{4\pi k \frac{r_2}{r_1 r_2}}$$

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Solved Problems on cylinder.

compare $R = \frac{1}{2\pi L}\left[\frac{1}{h_a r_1} + \frac{\ln(r_2/r_1)}{k_1} + \frac{\ln(r_3/r_2)}{k_2} + \frac{1}{h_b r_3}\right]$. | cylinder $R = \frac{1}{2\pi L k}\ln\left(\frac{r_2}{r_1}\right)$.

$Q = UA(T_a - T_b)$ | $U = W/m^2K$

$A = 2\pi r_3 L$

A hollow cylinder 5cm inner radius & 10cm outer radius has inner surface temp of 200°C & outer surface temp of 100°C. If the thermal conductivity is 70W/mK. Find heat transfer per unit length. $Q/L = \frac{2\pi k(T_1 - T_2)}{\ln(r_2/r_1)}$

Determine thermal conductivity of asbestos powder packed in b/w concentric copper pipes 25mm & 36mm diameter length. The inner pipe housing has a heating coil to which 120W power is supplied. The average temp of inner & outer pipes are 42.4°C & 27.9°C respectively.

$Q = \frac{T_1 - T_2}{\frac{1}{2\pi L(k)}\ln(r_2/r_1)}$

1.24

Problem for hollow sphere.

$$\dot{Q} = \frac{\Delta T_{overall}}{R} \qquad \Delta T = T_a - T_b$$

$$R = \frac{1}{4\pi}\left[\frac{1}{h_a r_1^2} + \frac{1}{k_1}\left[\frac{1}{r_1} - \frac{1}{r_2}\right] + \frac{1}{k_2}\left[\frac{1}{r_2} - \frac{1}{r_3}\right] + \frac{1}{h_b r_3^2}\right]$$

) A hollow sphere ($k - 65\ W/mK$) of 120mm inner dia & 350 mm outer dia is covered 10mm layer of insulation ($k = 10\ W/mK$). The inside & outside temp are $500°C$ & $50°C$ respectively. Calculate the rate of heat flow through this sphere.

CRITICAL RADIUS OF INSULATION:

Addition of insulation material on a surface does not reduce the amount of heat transfer rate always. In fact under certain circumstances it actually increases the heat low up to certain thickness of insulation. The radius of insulation for which the heat transfer is max is called CR of In.

Critical radius $= r_c = m$

Critical thickness $: r_c - r_1$

Critical Radius of Insulation for A cylinder

$$Q = \frac{T_i - T_\infty}{\ln\left(\frac{r_o}{r_1}\right) / 2\pi k L}$$

$$r_o = k/h = r_c\ (m)$$

Problem for Critical thickness:

① An electrical wire of 10m length & 1mm dia dissipates 200W in air at $25°C$. The convection heat transfer co-efficient b/w the wire surface & air is $15\ W/m^2K$. The k for wire is $0.582\ W/mK$. Calculate the critical radius of insulation (r_c) & also determine the critical thickness of insulation. Also determine the temp of the wire (T_a) if it is insulated to the critical thickness of insulation.

T_a / T_{wire} / T_b

Also find critical thickness

$$Q = \frac{(T_a) - T_b}{\frac{1}{2\pi L}\left[\frac{\ln r_c/r_1}{k_1} + \frac{1}{h_b r_c}\right]}$$

Problem for Planewall with $\underline{IHG}$ (Internal heat generation)

① An electric current is passed through a plane wall of thickness 150mm which generated heat at the rate of 50,000 W/m^3 ($\dot{q}$). The convective heat transfer co-efficient between wall & ambient air is 65 W/m^2k (h). ambient air temp is $28^\circ C$ (T_∞) & the K of the wall materials 22 W/mk. Calculate (i) Surface temp (ii) Max temp.

② An electric current is passed through a plane wall of thickness 25mm & 120mm wide, which used to heat a Fluid at $30^\circ C$ (T_∞). The heat generation rate is $65 \times 10^5\ W/m^3$ ($\dot{q}_g$), k of the plate is 25 W/mk. calculate the heat transfer co-efficient to maintained the temp of plate is below $\underline{150^\circ C}$ (Tmax).

③ An ele current is ,, ,, composite wall made up of 2 layer. 1st layer is steel of 10cm thick & 2nd layer is brass of 8cm thick. The outer surface temp of steel & brass one maintained at $120^\circ C$ (T_1) & 65° (T_3) respectively. Assume that the contacts b/w 2 slab is perfect & the heat generation is 1.65 000 W/m^3. Determine.

$q_1 = \frac{Q_1}{A} = \frac{T_1 - T_2}{L_1/k_1 A}$ $\quad Q_2 = \frac{T_2 - T_3}{L_2/k_2 A}$

$\boxed{\dot{q}_{g} = \dot{q}_1 + \dot{q}_2}$

① Heat Flux through the outer surface of brass slab q_2.

② Interface temp. T_2.

Problem for cylinder with $\underline{IHG}$

A copper wire of 40mm dia ($\phi/2 r$) carries 250A (I) & has a resistance of $0.25 \times 10^{-4}\ \Omega$ cm/length (R) surface temp of copper wire of is $25^\circ C$ (T_s) & the ambient air temp is $10^\circ C$ (T_∞). If the k of copper wire is 175 w/mk. Calculate

① Heat transfer co-efficient b/w with surface & ambient air (h)

② max temp of ä in the wire (Tmax)

$Q = \cancel{I \times R}\ I^2 \times R^2$ w/cm

$\dot{q} = Q/V = \frac{Q}{\pi r^2 L}$

Problem for sphere with IHG

① A sphere of 100mm dia having k of 0.18 W/mk. The outer surface temp is $8^\circ C$ & 250 W/m^2 of energy is released due to heat source. Calculate ① heat generated $\dot{q} = Q/V = \frac{4}{3}\pi r^3$

② Temp at the centre of the sphere.

$\dot{q}/A = \frac{Q/A}{V}$

$\frac{\boxed{\dot{q}}}{4\pi r^2} = \frac{Q/A}{4/3 \pi r^3}$

Heat conduction with Heat Generation :

In many practical cases, there is a heat generation within the system.

Ex: 1) Electric coils, 2) Resistance heater.

3) Nuclear Reactor 4) Combustion of fuel in the fuel of boiler Furnaces

Plane wall with internal heat generation:

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

(ii) $T_w = T_\infty + \frac{\dot{q}L}{2h}$ (Surface temp)

where

T_∞ = Fluid temp k

q = Heat generation

L = Thickness m

h - heat transfer co

k - Thermal conduc

Cylinder with internal heat generation:

(i) $T_{max} = T_w + \frac{\dot{q}r_0^2}{4k}$ | heat generation $\dot{q} = \frac{Q}{V}$

(ii) $T_w = T_\infty + \frac{r_0\dot{q}}{2h}$

cylinder: V - volume - πr^2

r - radius - r_0

Sphere with internal heat Generation:

$T_c = T_w + \frac{\dot{q}r_0^2}{6k}$

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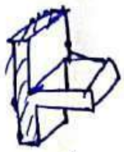
FINS

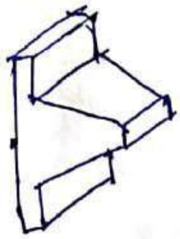
It's possible to increase the heat transfer rate by increasing the surface of heat transfer. The surface used for increasing heat transfer are called extended surfaces (or) fins.

TYPES:-

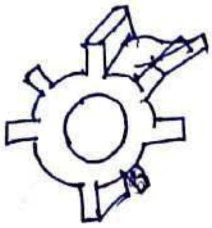
(1) Uniform straight fin

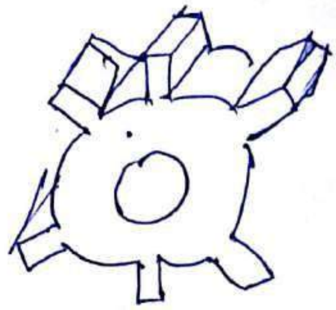


(2) Tapered straight fin

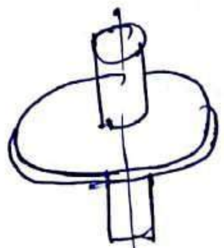


(3) Splines

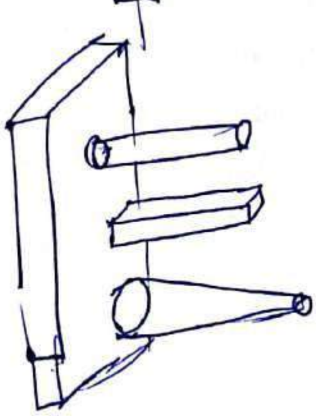




(4) Annular fin



(5) Pin fins



: types

(1) Infinitely long fine

(2) short fin (end is insulated)

(3) short fin (end is not insulated).

Applications:-

* Cooling of electrical components, motor cycle engines, small capacity compressor, transformers, radiators, refrigerators etc.

Fin efficiency

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

The ratio of actual heat transferred fin to the maximum possible heat transferred by the fin.

For insulated end

$$\eta_{fin} = \frac{\tanh(mL)}{mL}$$

Fin effectiveness:

It is defined as the ratio of heat transfer with fin to heat transfer without fin

$$\text{Fin Effectiveness } E = \frac{Q_{with\ fin}}{Q_{without\ fin}}$$

For insulated end

$$\text{Fin effectiveness } E = \frac{\tanh(mL)}{\sqrt{hA/kP}}$$

Problem for long fin or infinitely long fin

(1) Find the heat loss from a rod of 3 mm in diameter & infinitely long when its base is maintained at 140°C. The conductivity of the material is 150 W/mk and the heat transfer coefficient on the surface of the rod is 300 W/m²k. The temp of the surrounding the rod is 15°C

(2) A long rod 5 cm dia its base is connected to a furnace wall at 150°C, while the end is projecting into the room at 20°C. The temp of the rod at distance of 20 cm apart from its base is 60°C. The conductivity of the material is 200 W/mk. Determine convective heat transfer coefficient. $\boxed{e = \infty}$

furnace 150 60 20°C, 20 cm

(3) One end of the long solid rod of 50 mm dia is inserted in to a furnace with the other end is projected the atmosphere at 25°C. Once the steady state is reached, the temp of the rod is measured at 2 points 20 cm apart are found to be 150° & 100°C. The convective heat transfer coefficients b/w the rod & the surrounding air is 30 W/m²k. Calculate the thermal conductivity of the rod material.

$mx = \sqrt{hP/kA}$

(4) A carbon steel (k = 55 W/mk) 90 mm long rod with cross sectional area 5×10^{-3} m² and perimeter 0.69 m is attached to a plane wall which is maintained at a temp of 400°C. The surrounding environment is at 50°C & h = 90 W/m²k. Calculate the heat dissipated by the rod (Q)

(5) A turbine blades are made of stainless steel. Each blade carries 85 W heat (Q). The cross sectional area of each blade is 4.5 cm² (4.5×10^{-4} m²), Perimeter of each blade is 7 cm. The gas temp flowing over the blades is 800°C (T_∞). The temp of the roots of the blade is 1250°C. k = 22 W/mk & h = 110 W/m²k. Determine the height of the blade neglecting the heat flow from the gas to the end of the blade

1·35

INFINITELY LONG FIN (or) LONG FIN.

Important formula for Fin

(a) Temp distribution:

$$\frac{T - T_\infty}{T_b - T_\infty} = e^{-mx}$$

where

T_b = Base Temp, K

T_∞ = Surrounding Temp K

T = Intermediate temp K

x : Distance, m

Where, $m = \sqrt{\frac{hp}{KA}}$

where h = heat transfer co-eff W/m²K

p = Perimeter $\pi \times D$

k - Thermal cond W/mK

A - Area $\pi/4 d^2$

(b) Heat transfered

$$Q = (T_b - T_\infty)\sqrt{hpkA}$$

SHORT FIN

(a) Temprature distribution

$$\frac{T - T_\infty}{T_b - T_\infty} = \frac{\cosh m[L-x]}{\cosh[mL]}$$

(b) Heat transferred:

$$Q = (hpkA)^{1/2} (T_b - T_\infty)\tanh(mL)$$

(c) Number of fin = $\dfrac{\text{Heat generated}}{\text{Heat transfer per fin}}$

Perimeter : $2 \times L$ (Approximately)

Area : $t \times L$.

L.I heat Analysis [Negligible internal resistance]

(6) An aluminium alloy Fin of 7mm thick & 50mm long protrudes from a wall, which is maintained at 120°C. The ambient air temp is 22°C the h & k = 140 W/m²K & 55 W/mK respectively. Determine.

1. Temp at the end of the fin x = L
2. Temp at the middle of the fin x = L/2
3. Total heat dissipated by the fin Q

1/cosh(mL)

$P = 2 \times L$

$A = L \times T$

(7) An rectangular aluminium fins of 0.5mm square and 12mm long are attached on a plane plate which is maintained at 80°C. $T_\infty = 22°C$. Calculate the number of fins required to generate 35×10^{-3} W of heat. k = 165 W/mK
h = 10 W/m²K. Assume no ~~heat~~ heat loss from the tip of the fin insulated.

p = 2 (b + L)

Number of fin = H.G / Heat transfer (Q)

Lumped heat Analysis [Negligible internal resistance]

$-hA(T-T_\infty) = \rho c_v \, dT/dt$ Convective heat loss from the body = Rate of change of internal Energy.

T_∞

$T = T_0$ at $t = 0$

$$\frac{T - T_\infty}{T_0 - T_\infty} = e^{\left[\frac{-hAt}{\rho C_p V}\right]}$$ Pg.no:57

Where,

T_0 = Initial temp
T = Intermediate temp } T_∞ = Surface
T_∞ = Surface/surr final temp.

The lumped parameter system, Biot number value is less than 0.1.

i.e $B_i < 0.1 = \frac{hL_c}{k} < 0.1$

$L_c = V/A$

Problem for Lumped heat Analysis

(1) A 50 x 50 cm^2 aluminium slab of 6 mm thick is at 400°C (T_0) initially & its suddenly immersed in water, so its surface temp is lowered to 50°C (T_∞). Determine the time required for slab to reach 120°C (T). Take h = 100 W/m²k. D(Properties of Aluminium HMT book ρ, cp, k

Slab L/2, $B_i = \frac{hL_c}{k}$ $1.24 \times 10^{-3} < 0.1$

(2) A copper rod of outer dia 20mm initially at a temp of 380°C (T_0) is suddenly immersed in water at 100°C. Determine the time req for reach 210°C. h = 95 W/m²k cylin $L_c = r/2$

(3) ~~A solid copper cylinder of 7cm dia is~~ A 5cm thic copper slab is at 260°C initially & its suddenly immersed in water. T_∞ = 90°C (surface temp), In test run, initial temp is decrease by 40°C. & time taken is 6 minutes. Determine h lumped heat capacity method analysis.

(4) 1000 Thousand spheres made of copper of dia 6mm, T_0 = 20°C (Initial temp), These sphere are annealed in a quenching furnace. The temp of the inner furnace is 450 (T_∞). Calculate time required for the sphere to reach to temp of 320°C (T), h = 30 W/m²k

(5) A cylindrical stainless steel ingot 170mm dia & 50cm long passes through a heat treatment furnace which is 5000mm length. The temp of the furnace gas is 1300°C (T_∞), the initial ingot temp is 120°C (T_0). The combined radiation & convection h = 105 W/m²k. Calculate the maximum speed with which the ingot moves through the furnace in it attain 800°C (T), Take k = 45 W/mk, α = 0.46 x 10^{-5} m²/s

cyl $L_c = r/2$

alternator.

Problem for Semi-Infinite solids.

① A large concrete high way initially at a temp of 70°C (T_i) & stream water is directed on the high way so that surface temp is suddenly lowered to 40°C (T_0). Determine the time required to reach 55°C (T_x) at a depth of 4cm from the surface (α is not given).

② A large block of steel is initially at 35°C (T_i). The surface temp is suddenly raised & maintained at 250°C (T_0). Calculate the temp at a depth of 2.5cm after a time of 30s. The $\alpha = 1.4\times10^{-5}$ m/s, k = 45 W/mK.

③ A large wall 2cm thick has uniform temp 30°C (T_i) initially and the wall temp is suddenly raised & maintained at 400°C (T_0). Find

① The temp at a depth of 0.8cm from the surface of the wall after 10s

② Instantaneous heat flow rate through that surface per m^2 per hour

Take $\alpha = 0.008\ m^2/h^2$ · k = 6 W/m°C

(2) $q_x = \dfrac{k(T_0-T_i)}{\sqrt{\alpha\pi t}}\, e^{\left[\frac{-x^2}{4\alpha t}\right]}$

③ Total heat energy $Q_T = 2k[T_0 - T_i]\sqrt{t/\pi\alpha}$

iv

④ A semi infinite slab of aluminium is exposed to a constant heat flux (q_0) at the surface of $0.25\ MW/m^2$. Initial temp of slab is 25°C (T_i), calculate the surface temp after 10 minutes & also find the temp at a distance of 30cm from the surface after 10 minutes.

$q_0 = \dfrac{k(T_0-T_i)}{\sqrt{\alpha\pi t}}$

(1.4)

Spr = L/3

(1) Alloy steel ball of 12mm dia heated to 808C (To) is quenched in a bath at 100C (T∞). The properties of the ball are k = 205kJ/m hr k, ρ = 7860 kg/m³, Cp = 0.45 kJ/kg k J/kg, h = 150 W/mK

Determine (i) Temp of ball after 10sec (T) & (ii) Time for ball to cool to 400C (t)

205kJ/m hr k

$\frac{205 \times 10^3}{3600 \times mk}$

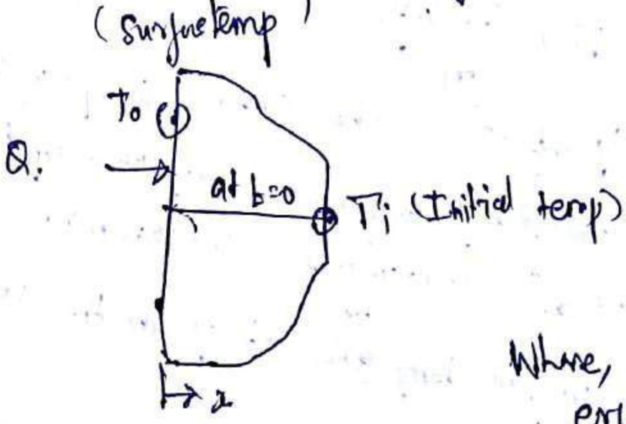
$\boxed{J/s = W}$

INFINITE SOLIDS:

A solid which extends itself infinitely in all directions of space is known as infinite solids.

SEMI INFINITE SOLIDS:

If an infinite solids is split in the middle by a plane, each half is known as semi infinite solids.



Where,
erf = error function of
α = Thermal diffusivity m²/s
t = Time S
Ti = Init temp k
To = Surface temp k
Tx = Intermediate temp k

$$\frac{T_x - T_o}{T_i - T_o} = erf\left[\frac{x}{2\sqrt{\alpha t}}\right]$$

Note:
1) In semi infinite solid heat transfer co-efficient (h) biot ~~number~~ number value is ∞ i.e. $\boxed{h \to \infty \text{ (or) } Bi \to \infty}$

Initial T_i, T_o (Surface), T_x (Intermediate)

Transients Heat Flow in an Infinite plate (16)

$$\frac{T_0 - T_\infty}{T_i - T_\infty} = f\left[\frac{x}{L}, \frac{hL}{k}, \frac{\alpha t}{L^2}\right].$$

Note :-
In infinite solids biot number value is in between 0.1 & 100

ie $0.1 < Bi < 100$.

1) A aluminium slab of 5cm thic initially at a temp of 400 (T_i) . its suddenly immersed in a (T_∞) water at 90°c . Calculate the mid plane (T_0) temp after 1 minute & also calculate the temp inside the plate at a distance of 10mm from the mid Plane. h = 1500 W/m²k . // HMT : 66 2) T_x

(i) Lc, (ii) Bi = 0.1 to 100 Case(ii) Fourier num = $\alpha t / L_c^2$ curve : $\frac{hLc}{k}$
L/2 raw

2) case (ii) x axis : $Bi = hLc/k$
curve : k/Lc

1) $\frac{T_0 - T_\infty}{T_i - T_\infty}$, 2) $\frac{T_x - T_\infty}{T_0 - T_\infty}$

2. A slab of aluminium 10cm thick is originally at a temp of 500°c (T_i) . its suddenly Immerged in a liquid at 100°c (T_∞) resulting in a h = 1200 W/m²k . Determine the temp at the centre line & the surface 1 minute after the immersion . Also calculate the total thermal energy removed per unit area of the slab during this period.
Take $\alpha = 8.4 \times 10^{-5}$ m²/s k = 215 W/mk $\rho = 2700$ kg/m³ C = 0.9 kJ/kg k.
1) To , 2) x = Lc = 0.05 (Temp at the surface.) 3) x axis = $\frac{h^2 \alpha t}{k^2}$
Curve : hLc/k.

$Q/Q_0 = 0.34$

$Q_0 = \rho C_p L [T_i - T_\infty]$

$Q/$ ... = J/m² .

T_i - Initial Temp

T_∞ = Final Temp
midplane T_0 = centre line Temp } k.
T_x = Intermediate temp

1.42

(3) A 10 cm dia apple approximately spherical in shape is taken from a 20°C (T_i) environmental & placed in a refrigerator where temp is 5°c (T_∞) & average heat $h = 6 W/m^2 K$. Calculate the temp at the centre of the apple after a period of 1 hour. Properties $\rho = 998 kg/m^3$ $c = 4180 J/kgK$, $k = 0.6 W/m K$.

$L_c = R/3$

$\alpha = k/\rho c_p$ x axis $\frac{\alpha t}{R^2}$

y axis, hR/k

cylinder

At centre

curve : hR/k

$x = \alpha t / R^2$

UNIT: II

CONVECTION

CONVECTIVE HEAT TRANSFER

Dimensional analysis

is a mathematical method which makes use of study of dimensions for solving several engineering problems.

Dimensions:-

Mass - M Temp - θ } are commonly used quantities in heat transfer
Length - L Time - T

Buckingham π Theorem

If there are n variable in a dimensionally homogeneous equation & if these contain m fundamental dimensions. then the variable are arranged into (n-m) dimensionless terms.

Advantages (dimensional analysis)

1) It expresses the Functional relationship b/w the variable in dimensional terms

2) It enable getting up a theoritical solution in a simplified dimensionless form.

Limitation

1) No information is given about the internal mechanism of physical Phenomenon.

2) It does not give any clue regarding the selection of variables

Renolds number (Re)

$$Re = \frac{UL}{\nu}$$

u - Velocity m/s
L - Length m
ν - kinematic viscosity m^2/s

$$Re = \frac{\text{Inertia force}}{\text{Viscous force}}$$

Prandtl Number (Pr)

$$Pr = \frac{\mu C_p}{k} = \frac{\nu}{\alpha}$$

ν - kin viscosity m^2/s
α - Thermal diff m^2/s

Nusselt Number (Nu)

$$Nu = \frac{hL}{k}$$

h - W/m^2K
L - m
k - W/mK

$$\frac{q_{conv}}{q_{cond}}$$

Grashof Number (Gr)

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

β = coeffeint of expansion k^{-1}
L = m
ν - m^2/s
ΔT - Temp difference k.

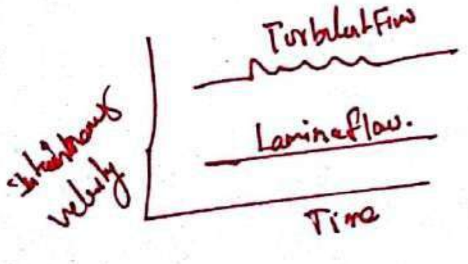
Stanton Number (St)

$$St = \frac{h}{\rho U C_p}$$

Newtonian & Non-Newtonian Fluids:

They fluid which obey the Newton's law of viscosity - Newtonian
- not obey - Non-Newtonian

Laminar & Turbulent flow



Irregular flow. Zig Zag.

Smooth & continuous path.
(Stream line flow)

Types of Boundary layer

1) Hydrodynamic (or) Velocity - velocity of fluid less than 99% of stream velocity
2) Thermal - Temp of the fluid less than 99% of stream velocity.

Convection - solid to liquid

Newton's law of convection:

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

$h = W/m^2 k$

$A = m^2$

T_w = Surface or wall temp k

T_∞ - Temp of fluid k

Free convection / Natural convection:

If the fluid motion is produced due to change in density resulting from temp gradients, the mode of heat transfer is said to be free (or) natural convection.

Forced convection:-

If the fluid motion is artificially created by means of an external force like a blower or fan, that type of heat transfer is known as FC.

Heat transfer from flat surface - Formulae

1) Velocity is given then forced convection problems.

2) Film temp $= T_f = \frac{T_w + T_\infty}{2}$ | T_w = Plate temp or surface, T_∞ = fluid temp

3) $Re = \frac{UL}{\nu} < 5\times10^5 \Rightarrow$ Laminar Flow | U - velocity m/s, L = m, $\nu = m^2/s$ kinematic viscosity

$Re = \frac{UL}{\nu} > 5\times10^5 \Rightarrow$ Turbulent Flow

For Flat Plate: Laminar Flow HMT 115

① Local Nusselt Number

$$Nu_x = 0.332 (Re)^{0.5} (Pr)^{0.333}$$ / $0.0296 (Re)^{0.8} (Pr)^{0.333}$

② Local Nusselt Numb $Nu_x = \frac{h_x L}{K}$ / $h_x L / k$ | h_x: Local heat transfer coefficient $W/m^2 k$

③ Average heat transfer coeff $h = 2 \times h_x$ / $1.25 h_x$

Q. Air at $20°C$ (T_∞) at a pressure of 1 bar is flowing over a flat plate at a velocity of 3 m/s. If the plate is maintained at $60°C$ (T_w). Calculate the ~~heat~~ heat transfer per unit width of the plate. Assuming the length of the plate along the flow of air is 2m.

1) T_f 2) Properties of air at T_f 3) Re 4) Nu, $\frac{h_x \times L}{k}$ 4) $h = 2 \times h_x$

Area $= W \times L = 1 \times 2 = 2$

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

2. Air at $25°C$ (T_∞) flows over a flat plate at a speed of 5 m/s & heated to $135°C$. The plate is 3 m long & 1.5 m wide. Calculate the local h_x at $x = 0.5$ & the Q from the first 0.5 m of the plate.

3) Air at $20°C$ (T_∞) at atmospheric pressure flows over a flat plate at a velocity of 3 m/s. If the plate is 1m wide & $80°C$ (T_∞). the following at $x = 300mm$. // Area $= 1 \times 0.3$

① Hydro-dy boundary Thick in m ② Thermal ... (m) ③ Local friction coefficients C_{fx} C_{fx}

④ Average Friction coefficients $\overline{C_{fL}}$ ⑤ Local heat transfer coeffs h_x ⑥ Average h ⑦ Q

4) Air at $20°C$ (T_∞) flows over a flat plate at a velocity of 3.5 m/s. The plate is 5m long & 2m wide.

Calculate the following

① Length of the plate over which the boundary layer is laminar 5×10^5

② Thickness of boundary layer δ (At laminar flow)

③ Shear stress at the location where boundary layer is laminar τ_x - Shear stress

$C_{fx} = \frac{\tau_x}{\frac{1}{2}\rho U^2}$

④ Total drag force on both sides of the plate where boundary layer is laminar

Q $F_D =$ Area $\times$ Average shear stress (τ)

$W \times L \times \tau$

Take $\rho = 1.205\ kg/m^3$

$\nu = 15.06 \times 10^{-6}\ m^2/s$.

$\overline{C_{fL}} = \tau / \frac{\rho U^2}{2}$

5) Air at $40°C$ (T_∞) flows over a flat plate at a velocity of 2 m/s. The plate is maintained at $100°C$. The length of the plate is 2.5m. Calculate the heat transfer per unit width. Using i) Exact method Q

ii) Approximate method Q_2

HOD

PRINCIPAL

In a cross flow h/c ...

(6) Air at $\overset{T_\infty}{25^\circ C}$ flows over 1m×3m (3m long) horizontal plate maintained at $\overset{T_w}{200^\circ C}$. at 10m/s. Calculate the average heat transfer co-efficient (h) for both laminar & turbulent regions. Take $Re = 3.5\times10^5$ (critical)

A particular engine the underside of the crankcase can be idealised as a flat plate measuring 80cm × 20cm ($A = 0.16 m^2$). The engine runs at $\overset{U}{80 km/hr}$ and the crankcase is cooled by air flowing past it at the same speed. Calculate the loss of heat from the crankcase surface of temp $\overset{T_w}{75^\circ C}$ to the ambient $\overset{air}{}$ temp $\overset{T_\infty}{25C}$

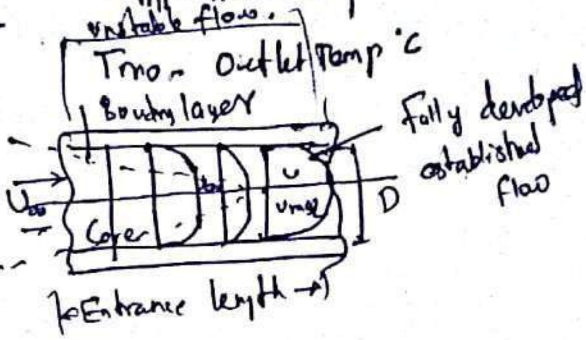
Q=?

Formulae used for Flow through cylinders (internal flow).

① Bulk mean temp $T_m = \frac{T_{mi} + T_{mo}}{2}$

T_{mi} - Inlet Temp °C

T_{mo} - Outlet Temp °C



② Reynolds Number $Re = UD/\nu$

$Re < 2300$ flow is laminar.

$Re > 2300$ Turbulent

③ Laminar flow: $Nu = 3.66$

④ Turbulent (general eqn): $Nu = 0.023 (Re)^{0.8} (Pr)^n$

$0.6 < Pr < 160$
$Re > 10000$
$L/D > 60$
$n = 0.4$ } Heating process.
$n = 0.3$ } cooling process

$$Nu = 0.036 (Re)^{0.8} Pr^{0.33} (D/L)^{0.055}.$$

This eqn valid for $10 < L/D < 400$,

$Re < 10,000$

⑤ equivalent dia for rectangular section

$$D_h \text{ or } D_e = \frac{4A}{P} = \frac{4(L \times W)}{2(L+W)}$$

A - Area m^2
P - Perimeter m
L - Length m
W - width m.

⑥ " " for hollow cylinder

$$D_h \text{ or } D_e = \frac{4A}{P}$$

$$= \frac{4 \times \pi/4 (D_o^2 - D_i^2)}{\pi (D_o + D_i)}$$

D_o - Outer dia

D_i - Inner dia

mass fluid ... 4.

Problem for internal flow.

Pg.no: 22

1) Water flows inside a tube of 20mm dia & 3m long at U of 0.03 m/s. The water gets heated from 40°C to 120°C while passing through the tube. The tube wall is maintained at T:C of 160°C. Find Q?

$Re = UD/\nu < 2300$ | $Nu = 3.66 = hD/k$ | $Q = hA(T_w - T_m)$. $A = \pi \times D \times L$.

2) When 0.4 kg of water per minute is passed through a tube of 2cm dia, it is found to be heated from 20°C (T_i) to 60°C (T_o). The heating is achieved by condensing steam on the surface of tube and subsequently the surface temp. of the tube is maintained at 90°C (T_w). Determine the length of the tube req for fully developed flow. ① $\dot{m} = \rho A$ ... $A = \pi/4\, d^2$ $Q = m C_p \Delta T$ ($\Delta T = T_{mo} - T_{mi}$)

$A = \pi \times D \times L$

Re > 10000, L/D > 60, Pr 0.6 ...

Water at 50°C (T_{mi}) enters 50mm dia and 4m long tube with a velocity of 0.8 m/s. The tube wall is maintained at a const temp of 90°C (T_w). Determine h & Q if out water temp is 70°C (T_{mo}). Re 8.36×10⁴ (Turbulent) $Nu = 0.023 \times (Re)^{0.8} \times (Pr)^n$

Water flows through of 8 cm dia, 3m long tube at an average temp of 40°C. Calculate (h). The flow velocity is 0.15 m/s & tube wall temp is 140°C (T_w).

$Nu = 0.036 (Re)^{0.8} (Pr)^{0.33} (D/L)^{0.055}$

for Turbulent L/D = 375

Air at 15°C (T_{mi}), 35 m/s, flows through a hollow cylinder of 4cm inner dia & 6 cm outer diameter & leaves at 45°C (T_{mo}). Tube wall is maintained at 60°C (T_w). Calculate the heat transfer co-efficient b/w the air & the inner tube. Hollow cylinder = $Re = \frac{UD_e}{\nu}$

Engine oil flows through a 50mm dia tube at an average temp. of 147°C. The flow velocity is 80cm/s (0.8 m/s). Calculate the average h, if the tube wall is maintained at a temp of 200°C (T_w) & it is 2m long.

Free convection:

If the fluid motion is produced due to change in density resulting from temp gradients. the mode of heat transfer is said to be Free or natural convection.

Formulae used for free convection:

① Film temp $T_f = \frac{T_w + T_\infty}{2}$ T_w - Surface °C, T_∞ - Fluid °C

② coefficient of thermal expansion:

$$\beta = \frac{1}{T_f \text{ in K.}}$$

③ Nusselt Number $Nu = hL/k$

④ Grashof Number for vertical Plate } $Gr = \frac{g \times \beta \times L^3 \times \Delta T}{\nu^2}$

$\Delta T = T_w - T_\infty$
$GrPr$ value $< 10^9$ - L
$GrPr > 10^9$ T

⑤ Nu for laminar Flow (Vertical Plate) $10^4 < GrPr < 10^9$ PG:134

$$Nu = 0.59 (GrPr)^{0.25}$$

⑥ Nu For Turbulent Flow (vertical plate)

$$Nu = 0.10 (GrPr)^{0.333}$$

⑦ Q for vertical plate) $A = \pi DL$ (cylinder), A - Area (Plate)

$$Q = hA(T_w - T_\infty).$$

① Grashof Number For horizontal Plate: $Gr = \frac{g \times \beta \times L_c^3 \times \Delta T}{\nu^2}$ $L_c = w/2$

② For horizontal Plate, upper surface heated,

$$Nu = 0.54 (GrPr)^{0.25}$$

This expression is valid for $2 \times 10^4 < GrPr < 8 \times 10^6$ Pg: 136

③ $Nu = 0.15 [GrPr]^{0.333}$ $8 \times 10^6 < GrPr < 10^{11}$

④ For horizontal Plate Lower surface heated $10^5 < GrPr < 10^{11}$

$$Nu = 0.27 (GrPr)^{0.25}$$

⑤ Q for horizontal Plate $= (h_u + h_L) \times A \times (T_w - T_\infty)$

2.10

PART - B.

8) (a) ~~Explain~~ generalized

For horizontal cylinder :

$Nu = C (GrPr)^m$ (pg 138).

$Q = hA (T_w - T_\infty)$ $A = \pi DL$

For sphere

$Nu = 2 + 0.43 [GrPr]^{0.25}$ $\boxed{\text{Page } 138}$

$Q = h \times A \times (T_w - T_\infty)$ $A = 4\pi r^2$

Boundary layer thickness $\delta_x = \left[3.93 \times (Pr)^{-0.5} (0.952 + Pr)^{0.25} \times (Gr)^{-0.25}\right] \times x$ $\boxed{135}$

Maximum velocity

$u_{max} = 0.766 \times v \times (0.952 + Pr)^{-1/2} \times \left[\frac{g\beta (T_w - T_\infty)}{v^2}\right]^{1/2} \times x^{1/2}$

Mass flow rate

$\dot{m} = 1.7 \times \rho \times v \left[\frac{Gr}{(Pr)^2 (Pr + 0.952)}\right]^{0.25}$.

Date .... Expt. No. .... Page No. ....

UNIT: III

PHASE CHANGE HEAT TRANSFER & HEAT EXCHANGERS

3.1 Boiling & condensation:.

Boiling & condensation are such convective heat transfer process that are associated with change in phase of liquid.

Boiling:

The change of Phase from liquid to vapour state.

Condensation

The change of Phase from vapour to Liquid.

Application:-

* Thermal & Nuclear power plant
* Refrigerating system
* Heating metal in furnaces
* Air conditioning system

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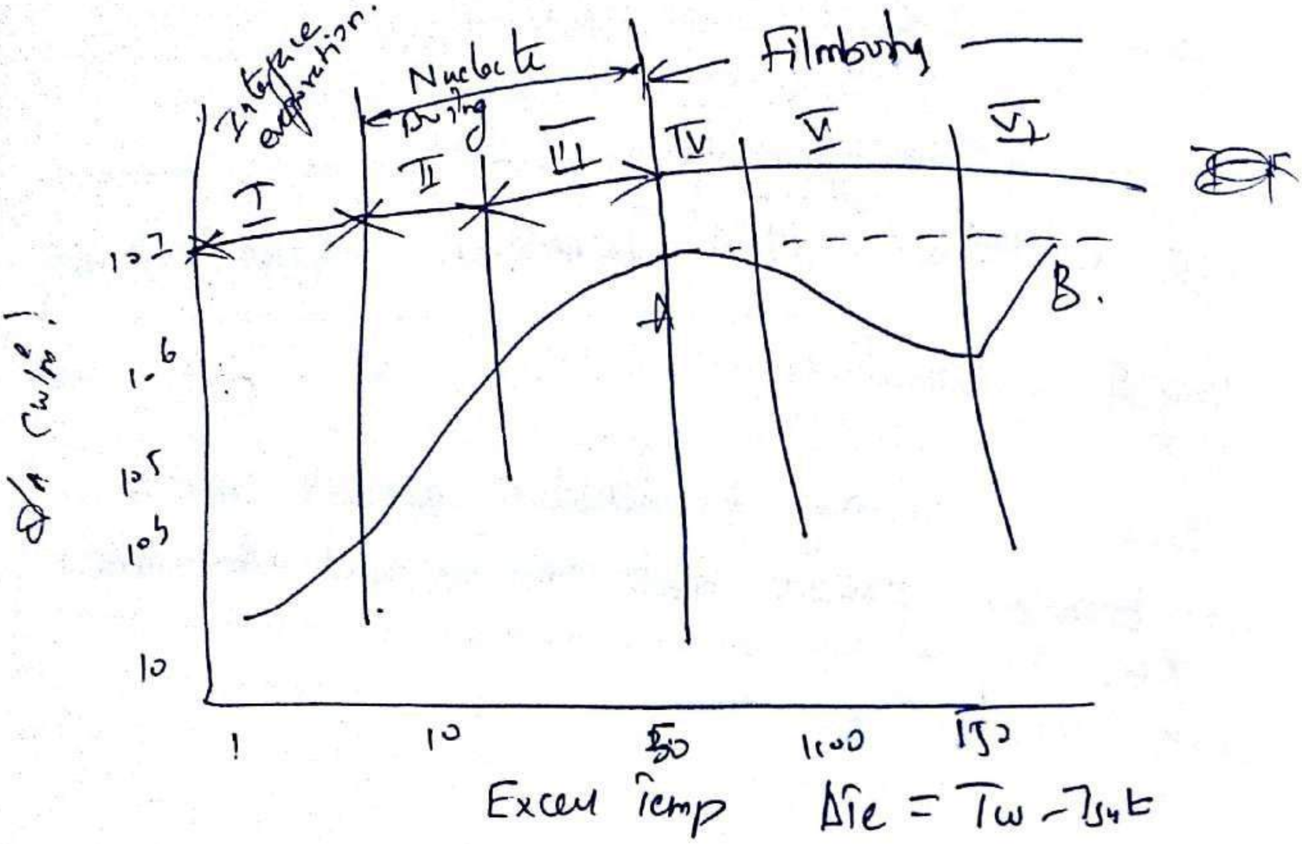

I - Free convection

II - Bubble condense in super heated liquid

III - Bubble raise ~~the~~ to surface

IV - Unstable film

V Stable film

VI Radiation coming into play.

(1) Nucleate Pool Boiling [HMT DB : 143]

(a) Heat flux $Q/A = \mu_l h_{fg} \left[\frac{g \times (\rho_l - \rho_v)}{\sigma}\right]^{0.5} \times \left[\frac{C_{pl} \times \Delta T}{C_{sf} \times h_{fg} \cdot Pr^n}\right]^3$

$Q/A = q$ = heat flux W/m^2

μ_l - Dynamic viscosity of liquid Ns/m^2

h_{fg} - enthalpy of evaporation J/kg

g - Acceleration due to gravity ~~m/s~~

ρ_l - Density of liquid kg/m^3 ρ_v: density vapour

C_{pl} - Sp. heat liquid J/kgK

C_{sf} - Surface fluid constant

Pr - Prandtl Num ΔT - Excess temp

T_w - Surface temp T_{sat} - Sat temp.

$\boxed{n = 1 \text{ for water}}$ $\boxed{n = 1.7 \text{ for other Fluid}}$

(b) Critical head flux

$$Q/A = 0.18\, h_{fg}\, \rho_v \left[\frac{\sigma \times g (\rho_l - \rho_v)}{\rho_v^2} \right]^{0.25}$$

(c) Excess temp:

$\Delta T = T_w - T_{sat} < 50°C$ for Nucleate pool boiling

(d) Heat transfer $Q = m \times h_{fg}$.

Date .... Expt. No. .... Page No. ....

(2) Flim Pool Boiling:

(a) Heat transfer coefficient

$$h = h_{conv} + 0.75\, h_{rad}.$$

$$h_{conv} = 0.62 \left[\frac{k_v^3 \times \rho_v \times (\rho_l - \rho_v) \times g \times (h_{fg} + 0.4\, C_{pv} \Delta T)}{\mu_v D \Delta T} \right]^{0.25}$$

k_v - of vapor w/mk

C_{pv} - sp heat of vapour at constant presure.

μ_v - Dynamic viscosity of vapour.

$$h_{rad} = \sigma \epsilon \frac{T_w^4 - T_{sat}^4}{T_w - T_{sat}}$$

$\sigma = 5.67 \times 10^{-8}\ w/m^2 k^4$

ϵ - Emissivity

σ - Stefan Boltzmann constant.

T_{sur} - Surface Temp °C

T_{sat} - Saturated Temp.

(b) Excess Temp

$\Delta T = T_w - T_{sat} > 50°$ for film boiling

①24

Problem for boiling

① Water is to be boiled at atm pressure in a polished copper pan by means of an ele heater. The dia of the pan is 0.38 m & is kept at 115°C (T_s). Calculate following

① Power req. to boil the water (P) (ii) ③ critical heat flux

FB ② Rate of evaporation ($\dot{m}$) $Q = \dot{m} \times h_{fg}$ W/m²

① Properties of water at 100°C (Sat) ② from ST at 100°C $h_{fg} = 2257.9 \times 10^3$ $v_g = 1.673$ ① $\rho_v = 1/v_g$

③ σ - from DDB 1145 at 100°C ⑤ $Q/A = $ $Q = \_ \times A(\pi/4 d^2)$ b) $\rho_v + \rho_l = v$

C_{sf} - 124

② It is desired to boil water at atmospheric pressure on a copper surface which is electrically heated. Estimate the heat

FB from the surface to the water, if the surface is maintained

and also the peak heat flux

Q/A

$T_w = 260 + 273 = 4$

$\sigma = \cancel{5.67} \times 10^{-8}$

FB ③ A heating element cladded with metal is 8 mm dia & of 0.92. The element is horizontally immersed in a water bath. The temp of the metal is 260°C (T_w) under steady state boiling conditions. Calculate power dissipation (P) per unit length of the heater

1) ΔT = FB / 2) T_f = / 3) ρ_v / ④ h_{fg}, $\sigma = 5.67 \times 10^{-8}$,

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... fluid unmixed, but fluid with a

boiling

① It's desired to boil water at atmospheric pressure on a Copper surface which is electrically heated. Estimate the heat flux from the surface to the water, if the surface maintained at 110°C and also the peak heat flux.

Condensation :-

The change of phase from vapour to liquid state.

Modes of condensation.

Two types
① Filmwise condensation
② Dropwise condensation.

FWC :-

The liquid condensate wets the solid surface, spreads out & forms a continuous film over the entire surface.

DWC

The vapour condenses into small liquid droplets of various size which fall down the surface in a random fashion.

$\boxed{\text{Q for DWC 10 time more than FWC}}$

Note

Laminar flow Re < 1800

Turbulent flow Re > 1800

$$Re = \frac{4\dot{m}}{P\mu}$$

HMT DB - pg no: 148

Vertical

(a) film thickness for laminar flow vertical surface. (δ_x) - Boundary layer thickness - (m)

$$\delta_x = \left[\frac{4\mu k x (T_{sat} - T_w)}{g \times h_{fg} \times \rho^2} \right]^{0.25}$$

μ - dynamic viscosity Ns/m²

$\mu - \rho \times \nu$

(b) Local heat transfer co-efficient (h_x) || for vertical surface Laminar flow

$$h_x = k/\delta_x$$

(c) Average heat transfer co-efficient (h) for vertical surface laminar flow.

$$h = 0.943 \left[\frac{k^3 \rho^2 g h_{fg}}{\mu L (T_{sat} - T_w)} \right]^{0.25}$$

The factor 0.943 may be replaced by 1.13 for more accurate result suggested by Mc adams.

horizontal

(1) Average heat transfer co-efficient for horizontal surface, laminar flow

$$h = 0.728 \left[\frac{k^3 \rho^2 g h_{fg}}{\mu N D (T_{sat} - T_w)} \right]^{0.25}$$

N - Next formula Avg h for bank of tubes

(2) Average heat transfer co-efficient for vertical surface / turbulent flow.

$$h = 0.0077 (Re)^{0.4} \left[\frac{k^3 \rho^2 g}{\mu^2} \right]^{0.333}$$

$Re < 1800$ L.F

$Re > 1800$ T.F

$Re = \frac{4m}{P\mu}$: P - Perimeter

Laminar flow vertical surface.

① Dry saturated steam at a pressure of 3 bar, condenses on the surfaces of a vertical tube of height of 1m. The tube surface temp is kept 110°C. Calculate the following

① Thickness of the condensate film 'δ_x'

② Local heat trans. $x = 0.25$m (h_x).

② A vertical tube of 65mm outside dia and 1.5m long is exposed to steam at atmospheric pressure. The outer surface of the tube is maintained at a temp of 60°C by circulating cold water through the tube. Calculate the following

① The rate of heat transfer to the coolant (Q)

② The rate of condensate of steam ($\dot{m}$)

① $T_{sat} = 100$ ② T_f ③ properties H_2O

④ h, ⑤ $Q = hA\Delta T$

$A = \pi \times D \times L$

③ A vertical flat plate in the form of fin is 500mm in height and is exposed to steam at atmospheric pressure. If surface of the plate is maintained at 60°C.

Calculate the following.

$x = L = 0.5$m

① Film thickness at the trailing edge (δ_x)

② Over all heat transfer co-efficient (h)

③ Heat transfer rate (Q)

④ The condensate mass flow rate. ($\dot{m}$)

④ A vertical plate 0.4m height and 0.3m wide at 40°C, is exposed to saturated steam at atmospheric pressure. Find the following

① Film thickness at the bottom of plate $\cdot \delta_x$

② Max velocity (U_{max}) $= \dfrac{\rho g (\delta_x)^2}{2\mu}$

Advanced: Easy construction, more economical, more surface area for Heat transfer

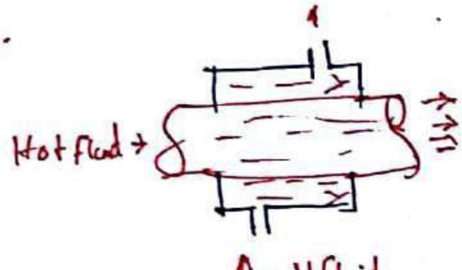
Disadvan: Less h, less generative capacity.

II Relative direction of Fluid motion:

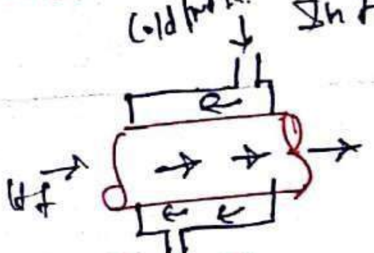
Types: (a) Parallel flow HE (b) counter flow HE (c) Cross flow Heat Exchanger.

→ same directions (Hot & cold fluid)

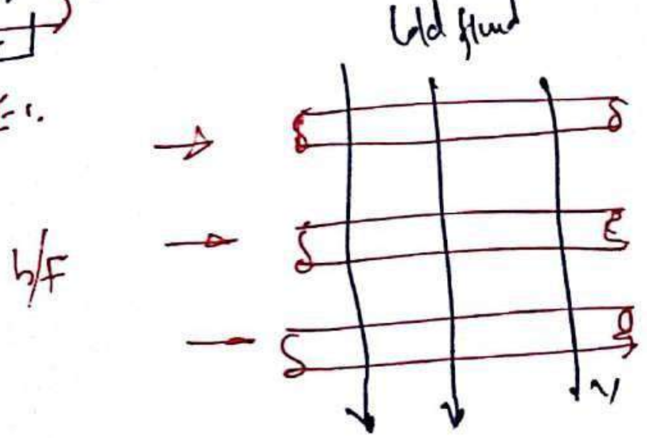
(a) Parallel Flow HE:



(b) Counter flow HE: In this type hot & cold fluids move in parallel but opposite directions



(c) Cross flow H/E:



[hot & cold fluid moves opposite directions].

Parallel flow & Counter flow heat exchanger

① Heat transfer $Q = UA(\Delta T)_m$ - $(\Delta T)_m$ Logarithmic mean Temp Diff (LMTD)

For Parallel flow

$$\Delta T_m = \frac{(T_1 - t_1) - (T_2 - t_2)}{\ln\left[\frac{T_1 - t_1}{T_2 - t_2}\right]}$$

T_1 & T_2 => Entry & Exit of hot fluid

t_1 & t_2 => Entry & Exit cold fluid

For counter flow:

$$(\Delta T_m) = \frac{(T_1 - t_2) - (T_2 - t_1)}{\ln\left[\frac{T_1 - t_2}{T_2 - t_1}\right]}$$

② Heat lost by hot fluid = Heat gained by cold fluid

$$Q_h = Q_c$$

$$m_h C_{ph} (T_1 - T_2) = m_c C_{pc} (t_2 - t_1)$$

m_h - massflowrate h/f

m_c - " " c/f kg/s

C_{ph} - sp. heat h/f J/kgK

③ Surface area of tube

$A = \pi D_1 L$ D_1 - Inner dia

$Q = m_h C_{ph} (T_1 - T_2)$

$Q = m_c C_{pc} (t_2 - t_1)$

④ $Q = \dot{m} \times h_{fg}$

⑤ Mass flow rate

$\dot{m} = \rho A C$

Problem for heat Exchanger

① In a counter Flow double pipe H/E, oil is cooled from 85°C to 55°C by water entering at 25°C. The mass flow rate of oil is 9800 kg/h & sp heat of oil is 2000 J/kgk. The mass flow rate of water is 8000 kg/h & sp heat of water is 4180 J/kgk. Determine the H/E area & heat transfer rate for an overall heat transfer co-efficient (U) = 280 W/m²k

hot fluid T_1, T_2
Cold Flow t_1, t_2

① $Q_h = Q_c$ ② Q ③ $Q = U(A)\Delta T_m$

(1.3)

② Water flows at the rate of 65 kg/m through a double pipe parallel flow H/E. Water is heated from 50°C to 75°C by an oil flowing through the tube. The sp oil is 4.2163 J/kg 1.780 kJ/kgk. The oil enters at 115°C & leaves 70°C. U = 340 W/m²k. Calculate the following ① Heat transfer Area ② Rate of heat transfer

③ In a Counter Flow Single pass H/E is used to cool the engine oil from 150°C to 55°C with water, available at 23°C as a cooling medium. Sp heat of oil is 2125 J/kgk. The flow rate of cooling water through the inner tube of 0.5 m dia is 2.4 kg/s. U = 240 W/m²k. how long must the H/E be to meet its cooling requirement?

Parallel H/E

④ Saturated steam at 120°C is condensing on the outer tube surface of a single pass H/E. The H/E heats 1050 kg/h of water from 20°C to 95°C. U = 1500 W/m²k. Calculate the following ① Area of H/E ② Rate of condensation of steam. Take h_{fg} = 2185 kJ/kg.

Parallel H/E

① Q = ? ② $Q = \dot{m}_h \times h_{fg}$, ③ $Q = U(A)\Delta T_m$

U
$T_1 T_2$
t_1
Cp_h
Cp_w
2.4 kJ

3.14

1) In a ~~counter~~ Counter flow double pipe heat Exchanger is used to cool the oil from 200°C to 120°C with water available at 25°C as the cooling medium. The out temp of water is 70°C. Sp. heat of oil is 1.5 kJ/kgK & m_h = 0.5 kg/s if U is 400 W/m²K. Find the following ① Rate of heat transfer (Q) 1) $Q = m_h C_{ph}(T_1 - T_2)$
2) Mass flow rate of water $\dot{m}_c$ ③ Area of heat exchanger (A) 2) $Q_h = Q_c$ 3) ΔT = LMTD

Problem on Cross Flow Heat Exchanger or Shell & tube Heat Exchanger.

Pg No: 151.

① $Q = FUA(\Delta T)_m$ (counter flow)

f – Correction factor HMTDB

$$(\Delta T)_m = \frac{(T_1 - t_2) - (T_2 - t_1)}{\ln\left[\frac{T_1 - t_2}{T_2 - t_1}\right]}$$

② $Q_h = Q_c$

$$m_h C_{ph}(T_1 - T_2) = m_c C_{pc}(t_2 - t_1)$$

③

$$\text{X axis value } P = \frac{t_2 - t_1}{T_1 - t_1}$$

$$\text{Curve value } R = \frac{T_1 - T_2}{t_2 - t_1}$$

formula used for NTU method

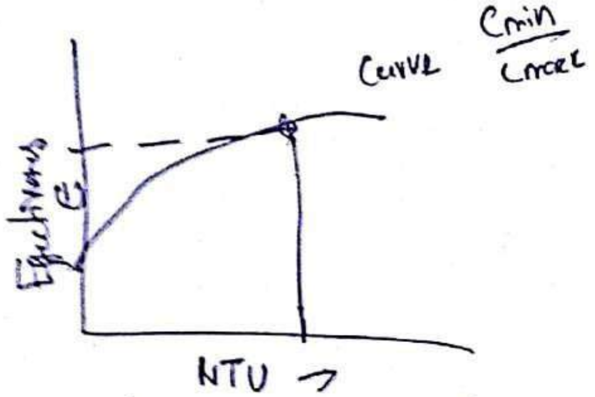
1) Capacity rate of hot fluid liquid : $C = \dot{m}_h \times C_{ph}$ W/K.

2) Capacity rate of cold fluid $= C = \dot{m}_c \times C_{pc}$

3) $\frac{C_{min}}{C_{max}}$

4) $NTU = \frac{UA}{C_{min}}$

5) Effective to find



5) $Q_{max} = C_{min}(T_1 - t_1)$ (max possible heat transfer)

6) Actual heat transfer rate $Q = \epsilon \times Q_{max}$

7) Heat transfer $Q = \dot{m}_c C_{pc}(t_2 - t_1)$ cold fluid)

Heat transfer $Q = \dot{m}_h C_{ph}(T_1 - T_2)$ (hot fluid)

It : NTU method is used to determine the inlet and exit temp of heat exchanger.

A parallel flow heat exchanger is used to cool 4.2 kg/min of hot liquid of sp. heat 3.5 kJ/kgk at $130°C$. A cooling water of sp. heat 4.18 kJ/kgk is used for cooling purpose at a temp of $15°C$. The mass flow rate of cooling water is 17 kg/min. Calculate the following

① Outlet temp of liquid (T_2) ② Outlet temp of water (t_2) ③ Effectiveness (ϵ)

Take $U = 1100$ W/m²k Area $= 0.30$ m²

) In a counter flow H/E water at $20°C$ flowing at the rate of 1200 kg/h. It is heated by oil of sp. heat 2100 J/kgk flowing at the rate of 520 kg/h at inlet temp of $95°C$. Determine the following

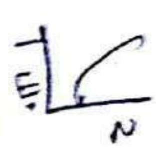
① Total heat transfer (Q) Take $U = 1000$ W/m²k

② Outlet temp of water $A = 1m^2$.

③ Outlet temp of oil

cold fluid water t_1 t_2 Hot fluid - oil T_1 T_2

① Capacity $m_h c_{ph}$, $m_c c_{pc}$ | ② $\frac{C_{min}}{C_{max}}$ | ③ $NTU = \frac{UA}{C_{min}}$ | ④



⑤ $Q_{max} = C_{min}(T_1 - t_1)$ | ⑥ $Q = \epsilon \times Q_{max}$ | $Q = m_c c_{pc}(t_2 - t_1)$ $Q = m_h c_{ph}(T_1 - T_2)$

UNIT: 4
RADIATION.

Introduction:

The heat is transferred from one body to another without any transmitting medium.

It is an electromagnetic wave phenomenon.

3×10^8 m/s

Emission properties

1) The wavelength
2) The temp of the surface
3) The nature of the surface.

Emissive power [Eb]

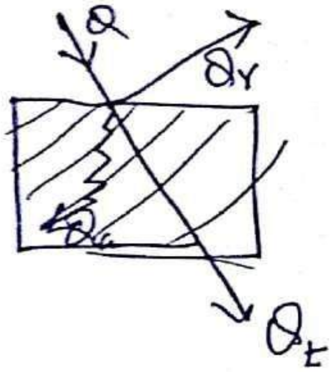
E_b = Total amounts of radiation emitted / body/unit time & body unit area

$$\boxed{E_b = W/m^2}$$

Monochromatic Emissive power ($E_{b\lambda}$)

The energy emitted by the surface at a given length per unit time per unit area in all directions.

Absorption, Reflection & Transmissions.



r - Reflection
T - transmission
a - Absorption.

$$Q = Q_a + Q_r + Q_t$$

$\div Q$ $\quad = \frac{Q}{Q} = \frac{Q_a}{Q} + \frac{Q_r}{Q} + \frac{Q_t}{Q}$

$$1 = \alpha + \rho + \tau$$

$$\text{Absorptivity } (\alpha) = \frac{\text{Radiation absorbed}}{\text{Incident radiation}}$$

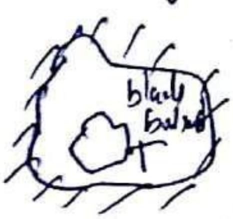
$$\text{Reflectivity } (\rho) = \frac{\text{Radiation reflected}}{\text{Incident radiation}}$$

$$\text{Transmissivity } (\tau) = \frac{\text{Radiation transmitted}}{\text{Incident radiation.}}$$

Concept of Blackbody :-

* A black body absorbs all incident radiation, regardless of wave length & direction.

* For a prescribed temp & wave length, no surface can emit more energy than black body



Enclosure at uniform temp

PLANCK'S DISTRIBUTION LAW

$$E_{b\lambda} = \frac{C_1 \lambda^{-5}}{e^{\left[\frac{C_2}{\lambda T}\right]} - 1}$$

$E_{b\lambda}$ = Monochromatic emissive power W/m^2

λ = Wavelength - m

$C_1 = 0.374 \times 10^{-15}\ W\text{-}m^2$

$C_2 = 14.4 \times 10^{-3}\ mk$

The Relationship b/w the MEP of a Black body & Wavelength of radiation at a particular temp.

WIEN'S DISPLACEMENT LAW :

The wien's Law gives the relationship b/w temp & wavelength corresponding to the max spectral emissive power of the black body at that temp.

$\lambda_{max} \cdot T = 2898\ \mu mk$ $\boxed{Pg-81}$

$$\boxed{\lambda_{max} T = 2.9 \times 10^{-3}\ mk}$$

STEFAN-BOLTZMANN LAW :-

$E_b \propto T^4$

$E_b = \sigma T^4$

Emissive power of a black body is proportional to the 4th power of absolute temp.

E_b - Emissive power W/m^2

σ = SB constant $5.67 \times 10^{-8}\ W/m^2k^4$

T = Temp - K

MAXIMUM EMISSIVE POWER $(E_{b\lambda})_{max}$

A combination of Planck & Wien's law yields the condition for the max. monochromatic emissive power for a black body.

$$(E_{b\lambda})_{max} = C_4 T^5$$

$C_4 = 1.307 \times 10^{-5}$ [Radiation constant]

$$\boxed{(E_{b\lambda})_{max} = 1.307 \times 10^{-5} T^5}$$

EMISSIVITY:-

$$\boxed{\varepsilon = \frac{E}{E_b}} = \frac{\text{Emissive power of any body}}{\text{Emissive power of a black body}}$$

GRAY BODY :-

* If a body absorbs a definite percentage of incident radiation irrespective of their wave length.
* The emissive power of a grey body is always less than black body

KIRCHOFF'S LAW of RADIATION:-

$$\frac{E_1}{\alpha_1} = \frac{E_2}{\alpha_2} = \frac{E_3}{\alpha_3}$$

INTENSITY OF RADIATION (I_n)

$$I_n = \frac{E_b}{\pi}$$

LAMBERT'S COSINE LAW

$$E_b \propto \cos\theta$$

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FORMULAE USED!.

$\boxed{\text{HMT D'8: 82}}$

1) Emissive power (or) Total Emissive power!.

$$E_b = \sigma T^4 \quad w/m^2$$

$\sigma = 5.67 \times 10^{-8} w/m^2k^4$

2) Wien's Law

$$\lambda_{max} T = 2.9 \times 10^{-3} mk$$

3) Monochromatic Emissive power (or) Spectral Emissive power $(E_{b\lambda})$

$$E_{b\lambda} = \frac{C_1 \lambda^{-5}}{e^{\left[\frac{C_2}{\lambda T}\right]} - 1}$$

$C_1 = 0.374 \times 10^{-15} w\text{-}m^2$

$C_2 = 14.4 \times 10^{-3} mk$

4) Max Emissive power $(E_{b\lambda})_{max}$

$$(E_{b\lambda})_{max} = C_4 T^5$$

$C_4 = 1.307 \times 10^{-5}$

5) Intensity of Radiation (I_b)

$$I_n = E_b / \pi$$

6) Black body high Emissivity > Grey body low Emissivity . compare black body

7) Absorptivity $(\alpha) = \frac{\text{Radiation absorbed}}{\text{Incident radiation}}$ Transmissivity $\tau = \frac{RT}{IR}$

Reflectivity $(\rho) = \frac{RR}{IR}$

Problem for radiation:

) Assuming sun to be black body emitting radiation with maximum intensity at $\underset{\text{wavelength}}{\lambda = 0.5\mu}$, calculate its surface temperature & emissive power $E_b = \sigma T^4$ W/m^2

$\lambda_{max} T = 2.9 \times 10^{-3}$ mk

$\boxed{1\mu = 10^{-6} m}$

A black body at 3000k emits radiation.

Calculate the following:

$\boxed{1\mu = 10^{-6} m}$

① Monochromatic Emissive power at 1μm wave length.

② Wave length at which emission is maximum (λ_{max})

③ Maximum emissive power $(E_{b\lambda})_{max}$

④ Total emissive power $(E_b) = \sigma T^4$

⑤ Calculate the total emissive of the furnace if it is assumed as a real surface having emissivity equal to $\underset{\varepsilon}{0.85}$.

① $\lambda = 1\mu = 1 \times 10^{-6} m$ $E_{b\lambda} = \dfrac{C_1 \lambda^{-5}}{e^{[C_2/\lambda T]} - 1}$ w/m^2

② $\boxed{\lambda_{max} T = 2.98 \times 10^{-3} m k}$

③ $= 1.307 \times 10^{-5} T^5 \,(w/m^2)$

$$(E_b)_{real} = \varepsilon \sigma T^4$$

A grey surface is maintained at a temp of 90°c & max emissive power at that temp is $\underset{(E_{b\lambda})_{max}}{1.4 \times 10^{10} W/m^2}$. Calculate the emissivity ε of the body & wave length corresponding (λ_{max}) to the maximum intensity of radiation.

$C_4 = 1.307 \times 10^{-5} W/m^2 k^5$

① $(E_{b\lambda})_{max} = C_4 T^5$

$$\varepsilon = \frac{1.4 \times 10^{10}}{2.90 \times 10^{10}}$$

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$800 W/m^2$ of radiant energy is incident upon a surface, out of which $300 W/m^2$ is absorbed, $100 W/m^2$ is reflected & the remainder is transmitted through the surface. Calculate the α, ρ, τ = $800-(300+100)$

A black body of $\underset{1200 \times 10^{-4} m^2}{\underline{1200 cm^2}}$ emits radiation at 1000k.

calculate the

1.) Total rate of energy emission $E_b = \sigma T^4$ W/m^2

2) Intensity of normal radiation $(I_n) = \frac{E_b}{\pi}$ W/m^2

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smooth:

Radiation Exchage B/w Surface:

① All surfaces are considered to be either black or Gray.

② Radiation & reflection process are assumed to be diffuse.

Formulae used:

① Heat Exchange b/w two large parallel plate is given by

$$Q_{12} = \bar{\varepsilon}\, \sigma A (T_1^4 - T_2^4)$$

$$\text{emissivity} = \frac{1}{\frac{1}{\varepsilon_1} + \frac{1}{\varepsilon_2} - 1}$$

ε_1, ε_2 Surface Emissivity ①, ②

② Heat exchange b/w two large concentric cylinder (or) sphere is given by

$$Q_{12} = \bar{\varepsilon} A_1 \sigma (T_1^4 - T_2^4)$$

for cylinder $A = 2\pi r l$ or $2\pi L$

$$\bar{\varepsilon} = \frac{1}{\frac{1}{\varepsilon_1} + \frac{A_1}{A_2}\left(\frac{1}{\varepsilon_2} - 1\right)}$$

for sphere $4\pi r^2$

③ Heat transfer with n shield is given by

$$Q_{1n} = \frac{A \sigma (T_1^4 - T_2^4)}{1/\varepsilon_1 + 1/\varepsilon_2 + \frac{2n}{\varepsilon_s} - (n+1)}$$

n - number shields

ε_s - Emissivity of shields

Problems for Radiation 2 through Plate, cylinder, sphere

1) Calculate the net radiant interchange per sq.m. for two large planes at a temp of 900k & 400k respectively. Assume that the emissivity of hot plane is 0.9 & that of cold plane is 0.7   i) Q/A   w/m²

T1   T2

ε1   ε2

2) Estimate the net radiant heat exchange per square meter from a very large plate at a temp of 550°c & 320°c. Assume that emissivity of hot plate is 0.8 ε1 cold plate is 0.6.   i) Q/A

3) Two large parallel plates are maintained at a temp of 900k & 500k respectively. Each plate has an area of 6 m². Compare the net heat exchange b/w the plates for the following cases:

① Both planes are black = ε=1   Case i

② Plate have an emissivity of 0.5 $\varepsilon_1 = \varepsilon_2 = 0.5$ Case ii

4) Calculate the heat exchange by radiation b/w the surface of 2 long cylinders having radii 120mm & 60mm respectively. The axis of the cylinder are parallel to each other. The inner cylinder is maintained at a temp of 130°c & emissivity of 0.6. outer cylinder is maintained at a temp of 30°c & emissivity of 0.5.

T1 ε1   T2 ε2

$A = \pi D L$; $L = L_2$

5) A liquid oxygen is stored in double walled spherical vessel. Inner wall temp is -160°c & outer wall temp is 30°c. Inner dia of sphere is 20cm & outer dia is 32cm. Calculate the following: $4\pi r^2$

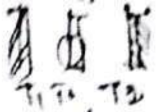
① Heat transfer if emissivity of spherical surface is 0.05.

② Rate of evaporation of liquid oxygen if its rate of vaporization of latent heat is 200 kJ/kg. $\varepsilon_1 = 0.05 = \varepsilon_2$ Latent heat 200kJ/kg 200×10^3 J.

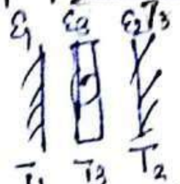
$A = 4\pi r^2$ [(-) sign indicates heat is transferred from outer surface to inner surface].

Rate of evaporation: $\frac{\text{Heat transfer}}{\text{Latent heat}}$

Problem for Radiation shield :-



① Emissivities of two large parallel plates maintained at $800°C$ & $300°C$ are 0.3 & 0.5 respectively. Find net radiant heat exchange per square metre for these plates. Find the percentage reduction in heat transfer when a polished aluminium radiation shield of emissivity 0.05 is placed b/w them. Also find the temp of the shield



Q_{13} & Q_{32}.

$$\frac{Q_{\text{without}} - Q_{\text{with sh}}}{Q_{\text{w/o sh}}} = \frac{Q_{12} - Q_{13}}{Q_{12}}$$

② A pipe of dia 30cm, carrying steam runs in a large room & is exposed to air at a temp of $25°C$. The surface temp of the pipe of $300°C$. Calculate the loss of heat to surrounding per metre length of pipe due to thermal radiation. The emissivity to pipe surface is 0.8.

What would be the loss of heat due to radiation of the pipe which is enclosed in a 55cm dia brick of emissivity 0.91? $A = \pi D L$

$Q/L = W/m$

③ Emissivity of two large parallel plates maintained at T_1 & T_2 are 0.6 & 0.6 respectively. Heat transfer is reduced 75 times when a polished aluminium radiation shields of emissivity 0.04 are placed in b/w them. Calculate the number of shields required. $\epsilon_S = \epsilon_3$

$$① \quad Q_{1n} = \frac{A\sigma (T_1^4 - T_2^4)}{\frac{1}{\epsilon_1} + \frac{1}{\epsilon_2} + \left(\frac{2n}{\epsilon_S}\right) - (n+1)}$$

② Heat transfer reduce 75 times

$$\frac{Q_{\text{without shield}}}{Q_{\text{with shield}}} = \frac{Q_{12}}{Q_{1n}} = 75 = \frac{\dfrac{A\sigma (T_1^4 - T_2^4)}{\frac{1}{\epsilon_1} + \frac{1}{\epsilon_2} - 1}}{\dfrac{A\sigma (T_1^4 - T_2^4)}{\frac{1}{\epsilon_1} + \frac{1}{\epsilon_2} + \frac{2n}{\epsilon_S} - (n+1)}}$$

Eg:- 11

③

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① Introduction:

The heat is transferred from one body to another without any transmitting medium.

It is an electromagnetic phenomenon. (3×10^{8} m/s)

② Emission Properties:

① The wavelength (or frequency of radiation

② the temp of surface

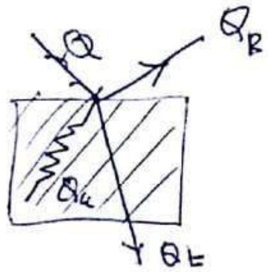
③ the nature of the surface.

③ Emissive power [E_b]

It is defined as the total amount of radiation emitted by a body per unit time & unit area. It's expressed in W/m^2

④ MONOCHROMATIC EMISSIVE POWER ($E_{b\lambda}$)

The energy emitted by the surface at a given length per unit time per unit area in all directions is known as MEP.



⑤ Absorption, Reflection & Transmission.

When the radiant energy falling on a body, 3 things happen.

A part is reflected back, a part is transmitted through the surface, and the remaining is absorbed.

$$Q = Q_a + Q_r + Q_t$$

÷ Q for Above equ:

$$\frac{Q}{Q} = \frac{Q_a}{Q} + \frac{Q_r}{Q} + \frac{Q_t}{Q}$$

where α = Absorptivity

ρ = Reflectivity

τ = Transmissivity

$$1 = \alpha + \rho + \tau$$

$$\alpha = \frac{\text{Radiation absorbed}}{\text{Incident Radiation}}, \quad \rho = \frac{\text{Radiation Reflected}}{\text{In / Rad}}, \quad \tau = \frac{\text{Radiat / Transmitte}}{\text{Inc / Radi}}$$

⑥ Black Body.

① A black body absorbs all incident radiation, regardless of wave length & direction

② For a ~~per~~ prescribed temp & wavelength, ~~regardless of~~ no surface can emitted more energy than black body.

(7) ~~B~~ Planck's Distribution Law:-

The relationship b/w the monochromatic emissive power of a black body and wave length of a radiation at a particular temp is given by the following expression

$E_{b\lambda}$ = monochromatic emissive power W/m²

λ = wavelength (m)

$$E_{b\lambda} = \frac{C_1 \lambda^{-5}}{e^{\left[\frac{C_2}{\lambda T}\right]} - 1}$$

$C_1 = 0.374 \times 10^{-15}\ W\text{-}m^2$

$C_2 = 14.4 \times 10^{-3}\ mk$

(8) WIEN'S displacement Law:-

Wine's law gives the relationship b/w temp & wavelength corresponding to the maximum spectral emissive power of the black body at the temp.

$\lambda_{max} T = 2898\ \mu mk$

$\boxed{\therefore \mu = 10^{-6} m}$

$\lambda_{max} T = 2.9 \times 10^{-3}\ mk$

(9) Stefan-~~boltz~~ boltzmann Law

The emissive power of a black body is proportional to the fourth power of absolute temp.

$E_b \propto T^4 \ / \ \therefore E_b = \sigma T^4$

Eb - Emissive power

σ - Stefan boltzmann constant $5.67 \times 10^{-8}\ W/m^2k^4$

(10) Maximum Emmisive power $(E_{b\lambda})_{max}$

A combination of Planck's law and wine's displacement law yields the condition for the max monochromatic emissive power for a black body.

$(E_{b\lambda})_{max} = C_4 T^5 \Rightarrow \underbrace{1.307 \times 10^{-5}}_{C_4} T^5$ (Radiation constant)

(11) Emissivity :- $\Rightarrow \boxed{\varepsilon = E/E_b}$

The ratio of emissive power of any body to the emissive power of a black body of equal temp

(12) GRAY Body: If a body absorbs a definite ~~%~~ % of incident radiation irrespective of their wave length, the body is known a grey body. emissive power of grey body < emissive power of black body.

(13) Intensity of Radiation (I_b)

$I_n = E_b/\pi$

(14) LAMBERT'S COSINE LAW

$E_b \propto \cos\theta$

(5) A Furnace wall emits radiation at 2000k. Treating it as black body radiation, calculate.

(1) Monochromatic radiant flux density at 1μm wave length. ($E_{b\lambda}$)

(2) Wave length at which emission is maximum & the corresponding (λ_{max}) emissive power. $\lambda_{max} T = 2.9 \times 10^{-3}$

(3) Total emissive power $E_b = \sigma T^4$

(6) The temp of a black body. surface 0.25 m^2 of area is 650°C

Calculate,

(1) The total rate of energy emission. $E_b = \sigma T^4$

(2) The intensity of normal radiation $= \frac{E_b}{\pi}$

(3) The wave length of max monochromatic emissive power.

$\lambda_{max} T = 2.9 \times 10^{-3}$,

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Problem for Radiation.

1) Assuming sun to be black body emitting radiation with maximum intensity at $\lambda = 0.5\mu$ (0.5×10^{-6} m) (wave length), calculate its surface temp & emissive power. ① $\lambda_{max} T = 2.9\times10^{-3}$ ② $E_b = \sigma T^4$

2) A black body at 3000k emits radiation. Calculate the following:

① Monochromatic emissive power at 1 μm wave length, $E_{b\lambda}$ at $1\mu m = 1\times10^{-6}$ m

② Wave length at which emission is maximum. (λ_{max}) on Wien's law

③ maximum emissive power $(E_{b\lambda})_{max}$

④ Total emissive power (E_b) (σT^4)

⑤ Calculate the total emissive of the furnace if it is assumed as a real surface having emissivity equal to 0.85 (ε)

① E_b $\quad (E_b)_{real} = \varepsilon \sigma T^4$

3) A gray surface is maintained at a temp of 900°C & the max. emissive power at that temp is 1.4×10^{0} W/m² ($(E_{b\lambda})_{max}$). Calculate the emissivity of the body (ε) C_4 the

$(E_b)_{max} \cdot C_4 T^5$

$\varepsilon = \frac{E}{E_b}$ gray

$\lambda_{max} = 0.52\times10^{-6}$ m.

4) The sun emits max radiation at $\lambda = 0.52\mu$. Assuming the sun to be a black body. Calculate the surface temp of the sun (T). Also calculate the monochromatic emissive power of the sun's surface ($E_{b\lambda}$)

Tofind T, ($E_{b\lambda}$)

$\lambda_{m}T = 2.9\times10^{-3}$

Formulae Used

① Emissive power (or)

Total emissive power

$$E_b = \sigma T^4 \ \underline{W/m^2}$$

$\sigma = 5.67\times10^{-8}\ W/m^2K^4$
Stefan boltzmann constant

② Wien's Law

$$\lambda_{max} T = 2898\ \mu mK = 2.9\times10^{-3}\ mK$$

$\mu = 10^{-6}\ m$

③ Monochromatic Emissive power (or)
Spectral Emissive power

$$E_{b\lambda} = \frac{C_1 \lambda^{-5}}{e^{\left[C_2/\lambda T\right]} - 1}$$

$C_1 = 0.374\times10^{-15}\ Wm^2$
$C_2 = 14.4\times10^{-3}\ mK$

④ Maximum Emissive power $(E_{b\lambda})_{max}$

$$\left[E_{b\lambda}\right]_{max} = C_4 T^5$$

$$C_4 = 1.307\times10^{-5}$$

⑤ Intensity of Radiation (I_{bn})

$$I_n = \frac{E_b}{\pi}$$

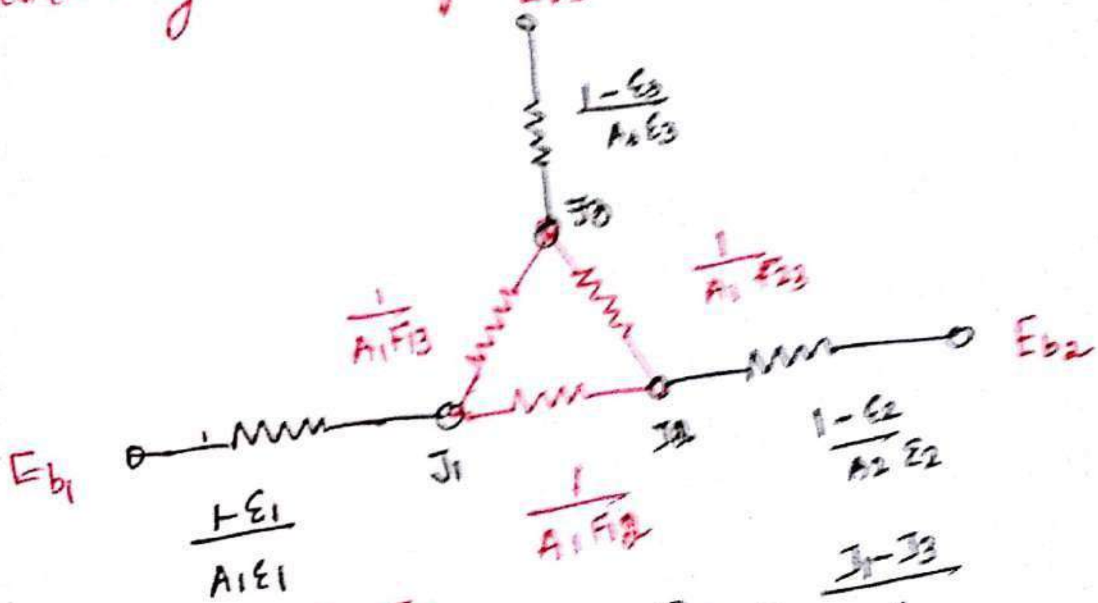
$$Q_{12} = \frac{\sigma T_1^4 - \sigma T_2^4}{\frac{1-\varepsilon_1}{\varepsilon_1 A_1} + \frac{1}{A_1 F_{12}} + \frac{1-\varepsilon_2}{\varepsilon_2 A_2}}$$

$\boxed{\varepsilon_1 = \varepsilon_2 = 1}$

Where f_{12} - shape factor

$$Q_{12} = \sigma [T_1^4 - T_2^4] \times A_1 F_{12}.$$

Radiation of Heat exchange for three gray surfaces.



$$Q_{12} = \frac{J_1 - J_2}{\frac{1}{A_1F_{12}}} \quad ; \quad Q_{13} = \frac{J_1 - J_3}{\frac{1}{A_1F_{13}}}$$

$$Q_{23} = \frac{J_2 - J_3}{\frac{1}{A_2F_{23}}}$$

$\frac{4}{4}$-13

UNIT: 5

MASS TRANSFER

Introduction:

In a system consisting of 2 or more components whose concentrations vary from point to point, there is a natural tendency for species (Particles) to be transported from a region of higher concentration side to a " " lower " " side.

Examples of mass transfer

* Humidification of air in cooling tower.
* Evaporation of petrol in the carburettor of an IC engine
* Transfer of water vapour into dry air
* Dissolution of sugar added to a cup of coffee

Modes of Mass transfer

① Diffusion mass transfer

② Convective mass transfer

Diffusion mass transfer:

* Molecular diffusion

* Eddy diffusion.

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 " in a mixture of liquid (or) Gases

Concentrations:

(i) Mass concentration = $\dfrac{\text{Mass of component}}{\text{Unit volume of mixture}}$ kg/m³

(ii) Molar concentration = $\dfrac{\text{Number of molecules of component}}{\text{Unit volume of mixture}}$

kg-mole/m³

$$C_A = \frac{\rho_A}{M_A} - \frac{\text{Density of component A}}{\text{Molecular weight of component A}}$$

(iii) Mass fraction = $\dfrac{\text{Mass concentration of a species}}{\text{Total mass density}}$

$$m_A = \frac{\rho_A}{\rho}$$

(iv) mole fraction = $\dfrac{\text{Mole concentration of a species}}{\text{Total molar concentration}}$

$$x_A = C_A / C.$$

Fick's law of diffusion:

$$\boxed{-D_{ab}\frac{dc_a}{dy}} = N_a = \frac{m_a}{A}$$ - molar flux units kg-mole/s-m²

(or)

Mass flux - kg/s-m²

D_{ab} - diffusion co-efficient of species a & b m²/s

$\frac{dc_a}{dx}$ - Concentration gradient.

- Used in industries for water pumping

Steady state diffusion through a Plane membrane:

Molar flux $\frac{m_a}{A} = \frac{D_{ab}}{L}[C_{a2} - C_{a1}]$.

C_{a1} - inner side } concentration kg.mole/m^3
C_{a2} - outer side }

For cylinders,

$L = r_2 - r_1$

$A = \frac{2\pi L(r_2 - r_1)}{\ln(r_2/r_1)}$

For sphere

$L = r_2 - r_1$

$A = 4\pi r_1 r_2$

r_1 - inner } radius (m)
r_2 - outer }
L - Length (m)

4

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using gears to reduce the [illegible]

water from the interior level make this project most successful in the future

Gaseous hydrogen is stored in a rectangular container. The walls of the container are of steel having 2.5mm thick. At the inner surface of the container the molar concentration of hydrogen in the steel is 1.2 kg mole/m^3 while at the outer surface ... molar concentration is zero. Take D_{ab} = 0.24×10^{-12} m^2/s.

Hydrogen gases at 3 bar & 1 bar are separated by a plastic membrane having thickness 0.25mm. The binary diffusion ~~coef~~ co-efficient of hydrogen in the plastic is 9.1×10^{-8} m^2/s. The solubility of H_2 in the membrane is 2.1×10^{-3} kg-mole/m^3-bar. An uniform temp condition of 20°C is assumed. Calculate follow.

(i) Molar concentration of H_2 on both side - C_{a1} & C_{a2} = Solu × Outer pr, Solubility × Inner pr

(2) Molar flux of hydrogen $m_a/A = D_{ab}/L\,[C_{a1} - C_{a2}]$

(3) Mass flux of hydrogen - Molar flux × molecular weight (H_2 = 2)

Solved Problem on concentration:

① A vessel contains a binary mixture of O_2 & N_2 with Partial Pressure in the ratio $\underset{P_{O2}}{\underline{0.21}}$ & $\underset{P_{N2}}{\underline{0.79}}$ at $\underset{T}{\underline{20°}}$ C. If the total pressure of mixture is 1.1 bar, calculate following:

(i) Molar concentration (C) $\left\{ P/GT = \frac{P_{O2}}{GT} \right.$ kg-mole/m^3

(ii) Mass densities $\rho_{O2} = C_{O2} \times M_{O2}$ (32)
$\rho_{N2} = C_{N2} \times M_{N2}$ (28)
$C = \rho/M$, $\rho = C \times M$ | overall density (ρ) = $\rho_{O2} + \rho_{N2}$

|| $C_{O2} + C_{N2} = C$

(iii) Mass fraction $m_{O2} = \rho_{O2}/\rho$ $\quad m_{N2} = \frac{\rho_{N2}}{\rho}$

(iv) Molar fraction of each species $x_{O2} = \frac{C_{O2}}{C}$ $\quad x_{N2} = \frac{C_{N2}}{C}$

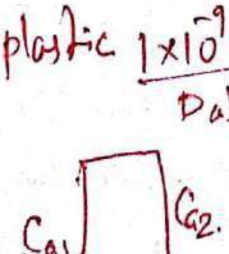
② A mixture of O_2 & N_2 with their partial. Pressure in the ratio 0.21 & 0.79 in a container at 25°c. Calculate the molar concentration (C_{O2}, C_{N2}), the mass density ($\rho_{O2}\ \rho_{N2}$), and the mass fraction ($m_{O2}\ m_{N2}$) of each species for a total pressure of 1 bar. What would be the average molecular weight of the mixture?

$M = \rho_{O2} M_{O2} + \rho_{N2} M_{N2}$

Solved Problems on members

① Helium diffuse through a plane membrane of $\underset{L}{\underline{2\text{ mm}}}$ thick. At the inner side the concentration of helium is $\underset{C_{a1}}{\underline{0.025}}$ kg mole/m^3. At the outer " " " " is $\underset{C_{a2}}{\underline{0.007}}$ kg mole/m^3. What is diffusion flux? of helium through the membrane. Assume diffusion co-efficient of helium with respect to plastic $\underset{D_{ab}}{\underline{1\times10^{-9}\,m^2/s}}$

diffusion flux of helium $\Big\}$ $m_a/A = \frac{D_{ab}}{L}[C_{a1} - C_{a2}] = 9\times10^{-9}\ \frac{\text{kg-mole}}{S\,m^2}$



Units

C_a & C_2 $\quad \frac{\text{kg-mole}}{m^3}$ | molar flux = $\frac{\text{kg-mole}}{S\text{-}m^2}$ | mass flux = $\frac{\text{kg}}{S\text{-}m^2}$

Oxygen at 25 °C & Pressure of 2 bar is flowing through a rubber pipe of inner dia 25 mm & wall thickness 2.5 mm. The diffusivity of O_2 through the rubber is $0.21 \times 10^{-9}\ m^2/s$ (D_{ab}) & the solubility of O_2 in rubber is $3.12 \times 10^{-3}\ \frac{kg \cdot mol}{m^3\ bar}$. Find the loss of O_2 by diffusion per metre length of pipe.

$d_i = r_1 = 12.5\ mm$

r_2 = Inner radius + thickness.

C_{a1} = Solubility ...
C_{a2} = 0 + ...

Molar flux: $\frac{m_a}{A} = \frac{D_{ab}\,[C_{a1} - C_{a2}]}{L}$

$L = r_2 - r_1$

$A = \frac{2\pi L\,(r_2 - r_1)}{\ln r_2/r_1}$

$L = 1\ m$

Unit $\frac{kg \cdot mol}{s}$

CO_2 and air experience equimolar counter diffusion in a circular tube whose length & diameter are 1.2 m & 60 mm respectively. The system is at a total pressure of 1 atm & a temp of 273 K. The ends of the tube are connected to large chambers. Partial pressure of CO_2 at one end is 200 mm of Hg while other end is 90 mm Hg. Calculate

$P_{a1} = \frac{200}{760} = 0.263$ bar $= 0.263 \times 10^5$ N/m²

$P_{a2} = 90/760$ bar $= 0.118 \times 10^5$ N/m²

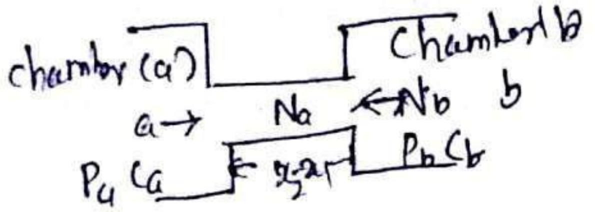
(i) Mass transfer of CO_2 kg/s

(ii) mass transfer air. kg/s

Data from HMTDB Pg 181 & 182 CO_2.

Steady state Equimolar counter diffusion

two chamber (large) a & b connected by a passage



Na } steady state molar diffusion
Nb }

Molar Flux $N_a = \frac{m_a}{A} = \frac{D}{GT}\left[\frac{P_{a1} - P_{a2}}{x_2 - x_1}\right]$

Molar Flux $N_b = \frac{m_b}{A} = \frac{D}{GT}\left[\frac{P_{b1} - P_{b2}}{x_2 - x_1}\right]$

$\frac{m_a}{A}$ - molar flux kg-mole/s-m^2

D - Diffusion co-efficient m^2/s

G - Universal gas constant 8314 J/kg-mole-k

P_{a1} } Partial pressure of constituent at 1 & 2 in N/m^2
P_{a2} }

Problem for equimolar counter diffusion.

1) Ammonia and air are in equimolar counter diffusion in a cylindrical tube of 2.5mm dia. & 15m length. The total pressure is 1 atm & the temp is 25°c. One end of the tube is connected to a large reservoir of ammonia & the other end of tube is open to atmosphere. If the mass diffusivity for the mixture is 0.28×10^{-4} m^2/s. Calculate the following

1) Mass rate of ammonia in kg/h

2) mass flow rate of air in kg/h.

1) Total pres $P = P_{a1} + P_{a2}$

2) $A = \pi/4 d^2$

1) find m_a - kg-mole/s

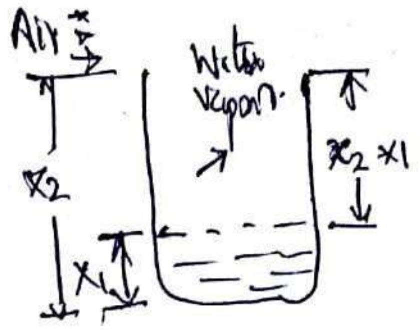
2) m_a x molecular weight of ammonia

P_{a1} = 1.013 bar

3) 6.36×10^{-2} kg/s ×3600

4) $m_a = -m_b$. mass of air

sothermal evaporation of water in to air.



Assumptions:

1) the system is isothermal & total pressure remains constant

2) system is steady state condition.

3) Air & water vapour behave as ideal gases.

From Fick's law diffusion

$$\text{molar flux } \frac{m_a}{A} = \frac{D_{ab}}{GT} \cdot \frac{P}{(x_2-x_1)} \ln\left(\frac{P_{a2}}{P_{a1}}\right)$$

or

$$\text{molar flux } \frac{m_a}{A} = \frac{D_{ab}}{GT} \frac{P}{(x_2-x_1)} \ln\left(\frac{P-P_{w2}}{P-P_{w1}}\right)$$

$\frac{m_a}{A}$ molar flux kg.mole/s-m^2

P - Total pressure

P_{w1} } Partial pressure of water vapour corresponding to saturation temp at 1 and 2

P_{w2} } Partial pressure of dry air at 2 in N/m^2.

Solved Problems Based on evaporation of water into air.

(1) Determine the diffusion rate of water from the bottom of a test tube of 25 mm diameter and 35 mm long into dry air at 25°C. Take diffusion co-efficient of water in air is $0.28 \times 10^{-4}\ m^2/s$.
254 = 5A Pg no: 183

(2). An open pan 20cm in diameter and 8cm deep contains water at 25°C and is exposed to dry atmospheric air. If the rate of diffusion of water vapour is $8.54 \times 10^{-4}\ kg/h$, estimate the diffusion co-efficient of water in air.

(3) Estimate the diffusion rate of water from the bottom of a test tube 10mm in diameter and 15cm long into dry atmosphere air at 25°C. Diffusion co-efficient of water into air is $0.255 \times 10^{-4}\ m^2/s$.

(Water - moisture
water vapour - steam

HEAT EXCHANGERS:-

Introduction:

A heat Exchanger is defined as an equipment which transfer the heat from a hot Fluid to a cold Fluid.

Types of Heat Exchanger:

① Nature of heat Exchange process
② Relative direction of Fluid motion
③ Design & constructional features
④ Physical state of fluids.

① Nature of heat exchange process:

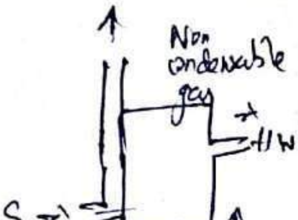
On the basis of the nature of heat exchange process heat exchange are classified as.

ⓐ Direct contact heat Exchanger or Open HE

ⓑ Indirect CHE

ⓐ OHE

The heat exchange takes place by direct mixing of hot & cold fluid. This H transfer is usually accompained by mass transfer. Ex: cooling tower, Direct contact Feed heaters.



ⓑ I contact HE

The transfer of heat b/w 2 fluids could be carried out by transmission through a wall which seperates the 2 Fluids.

① Regenerators ② Recuperators (or) Surface heat Exchangers

(i) Regenerators: In the type of HE, hot & cold Fluids flow alternately through the same space. Ex IC engine, gasturbine.

(ii) Recuperators (or) Surface heat Exchanger.
This most common type of HE in which the hot & cold Fluid do not come into direct contact with each other but are seperate by a tube wall or a surface
Automobile radiators, Air pre heater, Economisers