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CMR INSTITUTE OF TECHNOLOGY, BENGALURU

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Internal Assessment Test III – MAYl 2018

Sub:	Heat Transfer							Code:	15ME63
Date:	22/05/2018	Duration:	90 mins	Max Marks:	50	Sem:	VI	Branch:	MECH

Note: Answer **all five** questions.

		Marks	OBE CO	RBT
1	a) State and prove Kirchoff's law of radiation. b) Explain the following terms: (i) Black body and Gray body (ii)Radiosity and Irradiation. **OR** Calculate the net radiant heat exchange per m^2 area for two large parallel plates at temperatures 427°C and 27°C respectively. Take emissivity of hot plate and cold plate as 0.9 and 0.6 respectively. If a polished aluminium shield with emissivity 0.4 is placed between them, find percentage reduction in heat transfer.	10	5	3,4
2	Derive an expression for LMTD for counter flow heat exchanger and state the assumptions made. **OR** 8000kg/h of air at 105°C is cooled by passing it through a counter flow heat exchanger. Find the exit temperature of air if water enters at 15^0C and flows at a rate of 7500kg/h. The heat transfer area is equal to $20m^2$ and overall heat transfer coefficient corresponding to this area is $145W/m^2K$. Take Cp of air=1kJ/kg-K.	10	6	3

3	Using dimensional analysis derive an expression relating Nusselt, Prandtl and Grashoff numbers for natural convection.	10	4	3
	OR			
	A vertical plate 4m high and 6m wide is maintained at 60°C and exposed to air at 10°C. Calculate the heat transfer from both sides of the plate. Use data book for air properties.			
4	a) Explain with sketches: (i) Boundary layer thickness (ii) Thermal boundary layer thickness. b) Explain the significance of Grashoff No.	10	4	1
	OR			
	Explain the significance of (i) Reynold's No (ii)Prandtl No (iii)Nusselt No (iv)Stanton No			
5	Air at 20°C flows past a 800mm long plate at velocity of 45m/s. If the surface of the plate is maintained at 300°C, determine: (i) The heat transferred from entire plate length to air taking into consideration both laminar and turbulent portion of boundary layer. (ii) The percentage error if the boundary layer is assumed to be of turbulent nature from the very leading edge of the plate. Assume unit width of the plate and critical Reynold's no to be $5X10^5$.	10	5	3
	OR			
	Hot air at atmospheric pressure and 80°C enters an 8m long uninsulated square duct of cross section 0.2mX0.2m that passes through the attic of a house at a rate of $0.15m^3/s$. The duct is observed to be nearly isothermal at 60°C. Determine the exit temperature of the air and the rate of heat loss from the duct to the attic space.			

IAT-III Solution.

Heat Transfer

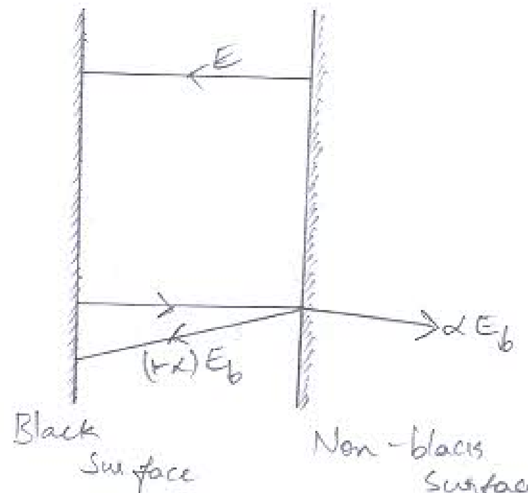
15ME63.

1)
a) State and Prove Kirchoff's law of radiation.

A:- It states that the emissivity of the surface of a body is equal to its absorptivity when the body is in thermal equilibrium with its surroundings. ie, $\varepsilon = \alpha$.

Explanation:-

Consider two surfaces one absolutely black at T_b, the other non-black at temperature T. The surfaces are arranged parallel to each other as shown in figure.



Let E = Radiant energy emitted by non block surface.

E_b = Radiant energy emitted by black surface.

Radiant energy emitted by non-black surface being fully absorbed by black surface. But non-Black surface absorbs only αE_b of radiant energy of black surface and reflects $(1-\alpha)E_b$ of radiant energy. Radiant interchange for the black surface equal to $E - \alpha E_b$.

If both the surfaces are at the same temperature.

ie, $T = T_b$, then

$$E - \alpha E_b = 0$$

$$(\text{or})\ \frac{E}{E_b} = \alpha \quad (\text{or})\ \frac{E}{\alpha} = E_b$$

For different surfaces, it can be written as

$$\frac{E_1}{\alpha_1} = \frac{E_2}{\alpha_2} = \frac{E_3}{\alpha_3} \text{------} = \frac{E_b}{\alpha_b} = E_b\ (\alpha_b = 1)$$

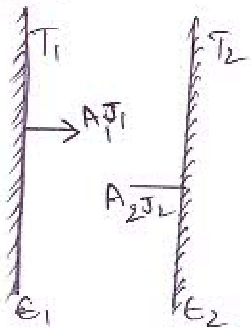
i.e. the ratio of emissive power to absorptivity is same for all bodies and is equal to emissive power of a black body. This relationship is known as Kirchoff's law.

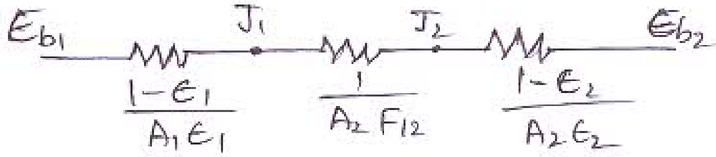
$$\text{or } \frac{E}{E_b} = \alpha \Rightarrow \boxed{E = \alpha}$$

1) b) Calculate the net radiant heat exchange per m² area for two infinite parallel plates held at temperature of 427°C and 27°C respectively. Take emissivity of hot plate and cold plate as 0.9 and 0.6 respectively. If a polished aluminium shield with emissivity 0.4 is placed between them. Find percentage reduction in heat transfer.

Sol: Data:- $T_1 = 427 + 273 = 700K$, $T_2 = 27 + 273 = 300K$, $\epsilon_1 = 0.9$, $\epsilon_2 = 0.6$, $\epsilon_{shield} = 0.4$

Case (i)



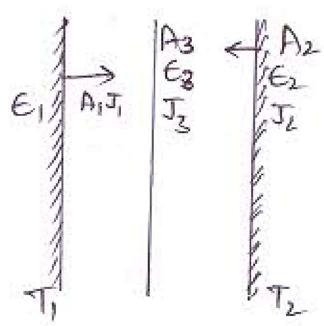


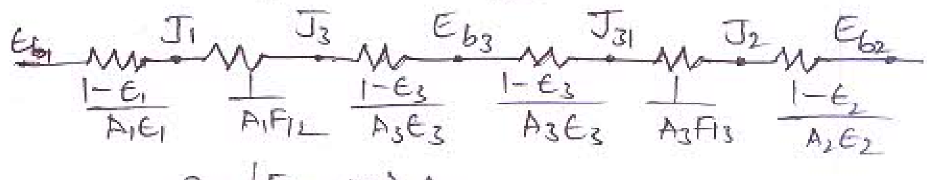
$$q = \frac{(E_{b1} - E_{b2})A_1}{\frac{1}{\epsilon_1} - 1 + 1 + \frac{1}{\epsilon_2} - 1} = \frac{\sigma(T_1^4 - T_2^4)A_1}{\frac{1}{\epsilon_1} + \frac{1}{\epsilon_2} - 1}$$

$$\frac{q}{A} = \frac{5.67\times10^{-8}(700^4 - 300^4)}{\frac{1}{0.9} + \frac{1}{0.6} - 1}$$

$$\frac{q}{A} = 7400 \text{ W/m}^2$$

Case (ii)





$$q = \frac{(E_{b1} - E_{b2})A_1}{\frac{1}{\epsilon_1} \cancel{-1} \cancel{+1} + \frac{1}{\epsilon_3} - 1 + \frac{1}{\epsilon_3} \cancel{-1} \cancel{+1} + \frac{1}{\epsilon_2} - 1}$$

$$q = \frac{\sigma(T_1^4 - T_2^4)A_1}{\frac{1}{\epsilon_1} + \frac{1}{\epsilon_2} + \frac{2}{\epsilon_3} - 2}$$

$$\frac{q}{A_1} = \frac{5.67\times10^{-8}[(700^4) - (300^4)]}{\frac{1}{0.9} + \frac{1}{0.6} + \frac{2}{0.4} - 2} = 2276.72 \text{ W/m}^2$$

$$\therefore \text{ \% reduction in heat transfer} = \frac{7400 - 2276.72}{7400} = 0.6923 = 69.23\%$$

2)a) Derive an expression for LMTD for Counter flow direction and state the assumption made.

A:- Assumptions:- For the analysis of heat exchanger under LMTD, the following assumptions are considered.

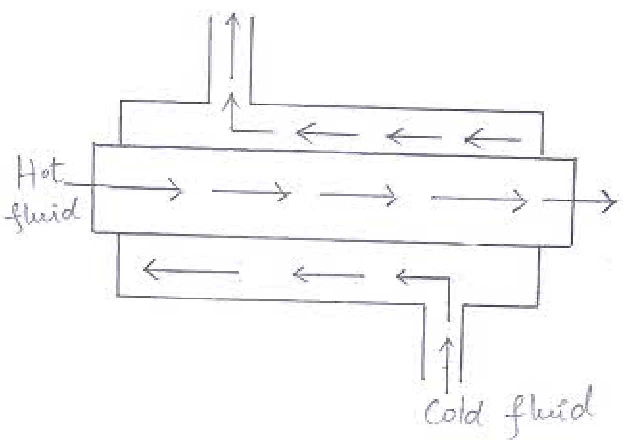
(a) The overall heat transfer coefficient is uniform through out the heat exchanger.

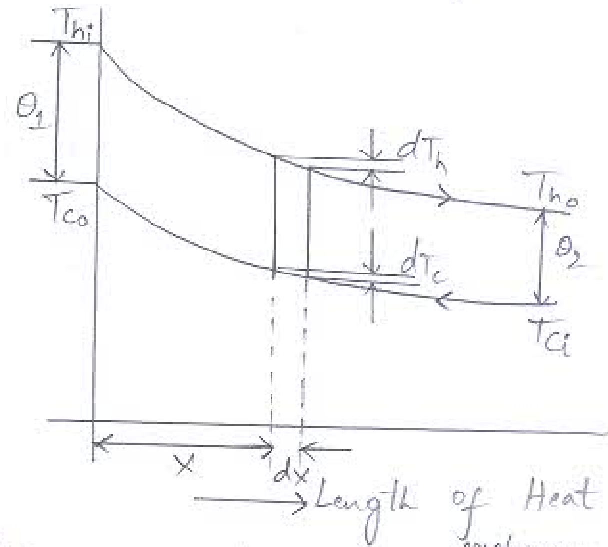
(b) The Specific heats and mass flow rates of both the fluids are constant.

(c) The potential and Kinetic energy changes are negligible.

(d) The heat exchange takes place only between the two fluids.

(e) The temperature of both the fluids are constant over a given cross section.





consider a counter flow heat exchanger as shown in fig.

Let T_{hi} and T_{ho} are temperature of hot fluid at inlet and outlet.

T_{ci} and T_{co} are temperature of cold fluid at inlet and outlet.

m_h and m_c are mass flow rate of hot and cold fluids.

C_{ph} and C_{pc} are specific heats of hot and cold fluids.

A_s = Surface area of heat exchanger

Let heat transfer between two fluids are

$$Q = m_h C_{ph}(T_{hi} - T_{ho}) = m_c C_{pc}(T_{co} - T_{ci})$$

$$Q = C_h(T_{hi} - T_{ho}) = C_c(T_{co} - T_{ci})$$

Also $\theta_1 = T_{hi} - T_{co}$

and $\theta_2 = T_{ho} - T_{ci}$

for a small elemental area dA, heat transfer is

$$dQ = -m_h C_{ph} dT_h = -C_h dT_h$$

and $dQ = -m_c C_{pc}\, dT = -C_c dT_c$

$$dT_h = -\frac{dQ}{C_h} \quad \text{and} \quad dT_c = -\frac{dQ}{C_c}$$

From the diagram at a distance of x

$$\theta = T_h - T_c$$

(or) $d\theta = dT_h - dT_c$

$$= -\frac{dQ}{C_h} - \left[-\frac{dQ}{C_c}\right] = -dQ\left[\frac{1}{C_h} - \frac{1}{C_c}\right]$$

$$d\theta = -U\,dA\,\theta\left[\frac{1}{C_h} - \frac{1}{C_c}\right]$$

$$\frac{d\theta}{\theta} = -U\left[\frac{1}{C_h} - \frac{1}{C_c}\right] dA$$

Integrating between inlet and outlet condition

$$\int_{\theta_1}^{\theta_2} \frac{d\theta}{\theta} = -U\left[\frac{1}{C_h} - \frac{1}{C_c}\right]\int_0^A dA$$

$$\log_e \frac{\theta_2}{\theta_1} = -UA_s\left[\frac{1}{C_h} - \frac{1}{C_c}\right]$$

$$= -UA_s\left[\frac{1}{\frac{Q}{T_{hi}-T_{ho}}} - \frac{1}{\frac{Q}{T_{co}-T_{ci}}}\right]$$

$$= -\frac{UA_s}{Q}\left[(T_{hi}-T_{ho}) - (T_{co}-T_{ci})\right]$$

$$= -\frac{UA_s}{Q}\left[(T_{hi}-T_{co}) - (T_{ho}-T_{ci})\right]$$

$$\log_e \frac{\theta_2}{\theta_1} = -\frac{UA_s}{Q}\left[\theta_1 - \theta_2\right] \qquad \left[\because \theta_1 = T_{hi} - T_{co},\ \theta_2 = T_{ho} - T_{ci}\right.$$

$$\log_e \frac{\theta_1}{\theta_2} = \frac{UA_s}{Q}\left[\theta_1 - \theta_2\right]$$

$$Q = UA_s \frac{[\theta_1 - \theta_2]}{\log_e \frac{\theta_1}{\theta_2}} = UA_s \theta_m$$

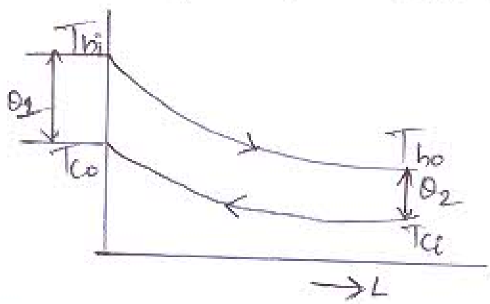
where, $\theta_m = \dfrac{\theta_1 - \theta_2}{\ln \frac{\theta_1}{\theta_2}}$ is the Logarithmic Mean Temperature Difference (LMTD)

2)b) 8000 kg/hr of air at 105°C is cooled by passing it through a counter flow heat exchanger. Find the exit temperature of air if water enters at 15°C and flows at a rate of 7500 kg/hr. The heat exchanger has heat transfer area equal to 20 m² and the overall heat transfer co-efficient corresponding to this area is 145 W/m²K. Take C_p of air = 1 kJ/kgK.

Sol^n:- Data:- $m_h = 8000\,kg/hr = 2.22\,kg/s.$

$T_{hi} = 105°C,\ T_{ci} = 15°C,\ m_c = \frac{7500}{3600} = 2.083\,kg/s,\ A = 20\,m^2,\ U = 145\,W/m^2K$

$C_{ph} = 1\times10^3\,J/kgK,\ C_{pc} = 4.18\times10^3\,J/kgK.$



$C_h = m_h C_{ph} = 2.22\times1\times10^3 = 2.22\times10^3 \longrightarrow C_{min}$

$C_c = m_c C_{pc} = 2.083\times4.18\times10^3 = 8.71\times10^3 \longrightarrow C_{max}$

$C = \frac{C_{min}}{C_{max}} = \frac{2.22\times10^3}{8.71\times10^3} = 0.26$ and $NTU = \frac{UA_s}{C_{min}} = \frac{145\times20}{2.22\times10^3} = 1.306$

From NTU v/s C chart for counter flow heat exchanger at NTU = 1.306 and C = 0.26

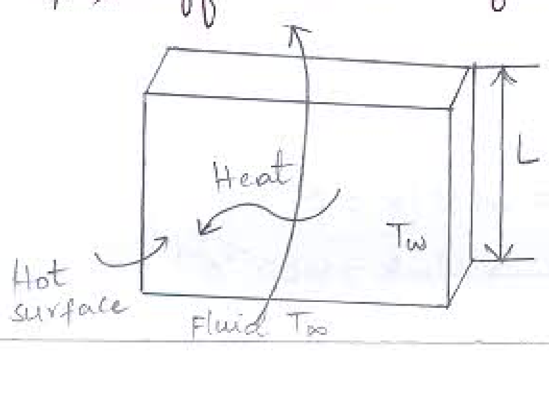
we get $\epsilon = 0.69$

$$\epsilon = \frac{C_h(T_{hi} - T_{ho})}{C_{min}(T_{hi} - T_{ci})}$$

$$0.69 = \frac{105 - T_{ho}}{105 - 15}$$

$$T_{ho} = 42.9°C$$

3)a) Using dimensional analysis derive an expression relating Nusselt, Prandtl and Grashoff numbers for natural convection.



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The different variables affecting on the behaviour of natural convection heat transfer are

a) Density of fluid, ρ.

b) Fluid viscosity, μ.

c) Specific heat of fluid, c

d) Fluid thermal conductivity, K.

e) Acceleration due to gravity, g.

f) Volume thermal expansion coefficient, β.

g) Temperature difference between fluid and plate, ΔT.

h) Significant length, L

i) Heat transfer coefficient, h.

ie, $f(\rho, \mu, C_p, k, g, \beta, \Delta T, L, h) = 0$

Coefficient of thermal expansion of fluid 'β' is defined as

$$\rho = \rho_0(1+\beta\Delta T)$$

Where, ρ = Bulk fluid density which depends on g

ρ_0 = Fluid density inside the heated layer.

ΔT = Temperature difference between the heated fluid and the bulk value.

Therefore, β and g are taken together.

Now number of variables $= n = 8$,

Number of fundamental dimension $m = 4$ (M, L, T, θ)

Number of π terms $= n - m = 8 - 4$, $f(\pi_1, \pi_2, \pi_3, \pi_4) = 0$.

Repeating variables L, ρ, μ, k.

Geometric property L

Flow property $= \rho$

Fluid property $= \mu$

Thermal property $= k$

Dimension of variables

$$\rho = kg/m^3 = ML^{-3}$$

$$\mu = Ns/m^2 = ML^{-1}T^{-1}$$

$$c = J/kgk = L^2T^{-2}\theta^{-1}$$

$$\beta g = m/s^2/k = \theta^{-1}LT^{-2}$$

$$k \longrightarrow w/mk = MLT^{-2}\theta^{-1}$$

$$\Delta T = c = \theta$$

$L \longrightarrow m = L$

$h \longrightarrow W/m^2k = MLT^{-3}\theta^{-1}$

Now $\pi_1 = L^{a_1}\rho^{b_1}\mu^{c_1}k^{d_1}\Delta T$

$\pi_2 = L^{a_2}\rho^{b_2}\mu^{c_2}k^{d_2}\beta g$

$\pi_3 = L^{a_3}\rho^{b_3}\mu^{c_3}k^{d_3}c_p$

$\pi_4 = L^{a_4}\rho^{b_4}\mu^{c_4}k^{d_4}h$

$\pi_1 = L^{a_1}\rho^{b_1}\mu^{c_1}k^{d_1}\Delta T$

$M^0L^0T^0\theta^0 = L^{a_1}(ML^{-3})^{b_1}(ML^{-1}T^{-1})^{c_1}(MLT^{-3}\theta^{-1})^{d_1}\theta.$

Equating the powers of M, L, T,

For $M;\ 0 = b_1 + c_1 + d_1 \longrightarrow (1)$

$L:\ 0 = a_1 - 3b_1 - c_1 + d_1 \longrightarrow (2)$

$T:\ 0 = -c_1 - 3d_1 \longrightarrow (3)$

$\theta:\ 0 = -d + 1 \longrightarrow (4)$

From (4) $d = -1$

From (3) $0 = -c_1 - 3d_1 \quad \therefore 0 = -c_1 - 3(-1) \Rightarrow c_1 = -3$

From (1) $0 = b_1 - 3 + 1 \Rightarrow b_1 = 2$

From (2) $0 = a_1 - (3\times2) + (3) + 1 \Rightarrow a = 2$

$\therefore \pi_1 = L^2\rho^2\mu^{-3}k^{1}\Delta T$

(or) $\boxed{\pi_1 = \dfrac{L^2\rho^2 \cancel{k_s} \Delta T}{\mu^3}}$

$\pi_2 = L^{a_2}\rho^{b_2}\mu^{c_2}k^{d_2}\beta g$

$M^0L^0T^0\theta^0 = L^{a_2}(ML^{-3})^{b_2}(ML^{-1}T^{-1})^{c_2}(MLT^{-3}\theta^{-1})^{d_2}LT^{-2}\theta^{-1}$

Equating the powers of M, L, T, θ

$M: 0 = b_2 + c_2 + d_2 \longrightarrow (1)$

$L: 0 = a_2 - 3b_2 - c_2 + d_2 + 1 \longrightarrow (2)$

$T: 0 = -c_2 - 3d_2 - 2 \longrightarrow (3)$

$\theta: 0 = -d_2 - 1 \longrightarrow (4)$

From (1) $d_2 = -1$

From (2) $0 = -c_2 - 3(-1) - 2 \Rightarrow c_2 = 1$

From (1) $0 = b_2 + 1 + (-1) \Rightarrow b_2 = 0$

From (2) $0 = a_2 - 3(0) - 1 + 1 \Rightarrow a_2 = 1$

$\therefore \pi_2 = L^1\rho^0\mu^1k^{-1}\beta g$

$$\boxed{\Pi_2 = \frac{L\mu\beta g}{k}}$$

$$\Pi_3 = L^{a_3}\rho^{b_3}\mu^{c_3}k^{d_3}c_p$$

$$M^0L^0T^0\theta^0 = L^{a_3}(ML^{-3})^{b_3}(ML^{-1}T^{-1})^{c_3}(MLT^{-3}\theta^{-1})^{d_3}L^2T^{-2}\theta^{-3}$$

Equating the powers of M, L, T, θ

$M: 0 = b_3 + c_3 + d_3 \longrightarrow (1)$

$L: 0 = a_3 - 3b_3 - c_3 + d_3 + 2 \longrightarrow (2)$

$T: 0 = -c_3 - 3d_3 - 2 \longrightarrow (3)$

$\theta: 0 = -d_3 - 1 \longrightarrow (4)$

From (4) $\Rightarrow d_3 = -1$

From (3) $0 = -c_3 - 3(-1) - 2 \Rightarrow c_3 = 1$

From (1) $0 = b_3 + 1 + (-1) \Rightarrow b_3 = 0$

From (2) $0 = a_3 - 3(0) - 1 - 1 + 2 \Rightarrow a_3 = 0$

$\therefore \Pi_3 = L^0\rho^0\mu^1k^{-1}c_p$

$$\therefore \boxed{\Pi_3 = \frac{\mu c_p}{k}}$$

$$\Pi_4 = L^{a_4}\rho^{b_4}\mu^{c_4}k^{d_4}h$$

$$M^0L^0T^0\theta^0 = L^{a_4}(ML^{-3})^{b_4}(ML^{-1}T^{-1})^{c_4}(MLT^{-3}\theta^{-1})^{d_4}MT^{-3}\theta^{-1}$$

Equating the powers of M, L, T, θ.

$M: 0 = b_4 + c_4 + d_4 + 1 \longrightarrow (1)$

$L: 0 = a_4 - 3b_4 - c_4 + d_4 \longrightarrow (2)$

$T: 0 = -c_4 - 3d_4 - 3 \longrightarrow (3)$

$\theta: 0 = -d_4 - 1 \longrightarrow (4)$

From (4) $d_4 = -1$

From (3) $0 = -c_4 - 3(-1) - 3 \Rightarrow c_4 = 0$

From (1) $0 = b_4 + 0 + (-1) + 1 \Rightarrow b_4 = 0$

From (2) $0 = a_4 - 3(0) - 0 - 1 \Rightarrow a_4 = 1$

$\therefore \Pi_4 = L^1\rho^0\mu^0k^{-1}h$

$$\therefore \boxed{\Pi_4 = \frac{hL}{k}}$$

Several experimental and analytical studies have shown that the first two π terms (Π_1, Π_2) always appear together as a single dimensionless group. The parameter so found is called Grashoff Number.

i.e. $G = \pi_1 \pi_2 = \frac{L^2 \rho^2 K \Delta T}{\mu^3} \times \frac{L \mu \beta g}{k} = \frac{L^3 \rho^2}{\mu^2} \beta g \Delta T$

$G_r = \frac{L^3 \beta g \Delta T}{\nu^2}$ $\quad \left[\because \nu = \frac{\mu}{\rho}\right.$

The terms π_3 and π_4 are called Prandtl and Nusselt numbers

$\pi_3 = $ Prandtl number $P_r = \frac{\mu C_p}{k}$

$\pi_4 = $ Nusselt number $N_u = \frac{hL}{k}$

$\therefore f\left(\frac{L^3 \beta g \Delta T}{\nu^2}, \frac{\mu C_p}{k}, \frac{hL}{k}\right) = 0$

$f(G_r, P_r, N_u) = 0$

(Or) $N_u = \phi(G_r P_r)$

$\boxed{N_u = C G_r^{n_1} P_r^{n_2}}$ where, C, n_1 and n_2 are constants.

~~(Or) $N_u = \phi(G_r P_r)$~~

~~$N_u = C G$~~

3)b) A vertical plate 4m high and 6m wide is maintained at 60°c and exposed to air at 10°c. calculate the heat transfer from both sides of the plate. Use data book for air properties.

Soln: Data :- plate of (4×6) m, $T_p = 60°c$

$T_a = 10°c$

$q = ?$

characteristic length $L = \frac{A}{P} = \frac{4 \times 6}{2(4+6)} = \frac{24}{20} = 1.2m$.

Mean film Temperature

$T_m = \frac{T_p + T_a}{2} = \frac{60+10}{2} = 35°c$.

From data hand book at $T_m = 35°c$

$\nu = 16.48 \times 10^{-6}$ $\quad P_r = 0.7$ $\quad k = 0.0271$

$q = hA\Delta T$

$A = 4 \times 6 = 24 m^2$ $\quad \beta = \frac{1}{35+273}$

$\Delta T = 60 - 10 = 50°c$ $\quad \beta = 3.247 \times 10^{-3}$

$h = h_{top} + h_{bottom}$

$$h_{top} = \frac{Nu \times k}{L}$$

$$Nu = 0.54 (Gr\,Pr)^{0.25}$$

$$Gr = \frac{g \beta L^3 \Delta T}{\nu^2} = \frac{9.81 \times 3.847 \times 10^{-3} \times 4^3 \times 50}{16.48 \times 10^{-6}}$$

$$Gr = 6184591.85$$

$$Gr\,Pr = 4329214$$

$$(Gr\,Pr)^{0.25} = 45.61$$

$$h_{top} = \frac{0.54 \times 45.69 \times 0.0271}{4}$$

$$h_{top} = 0.1668 \ W/m^2K$$

$$h_{bottom} = \frac{0.27 \times 45.69 \times 0.0271}{4}$$

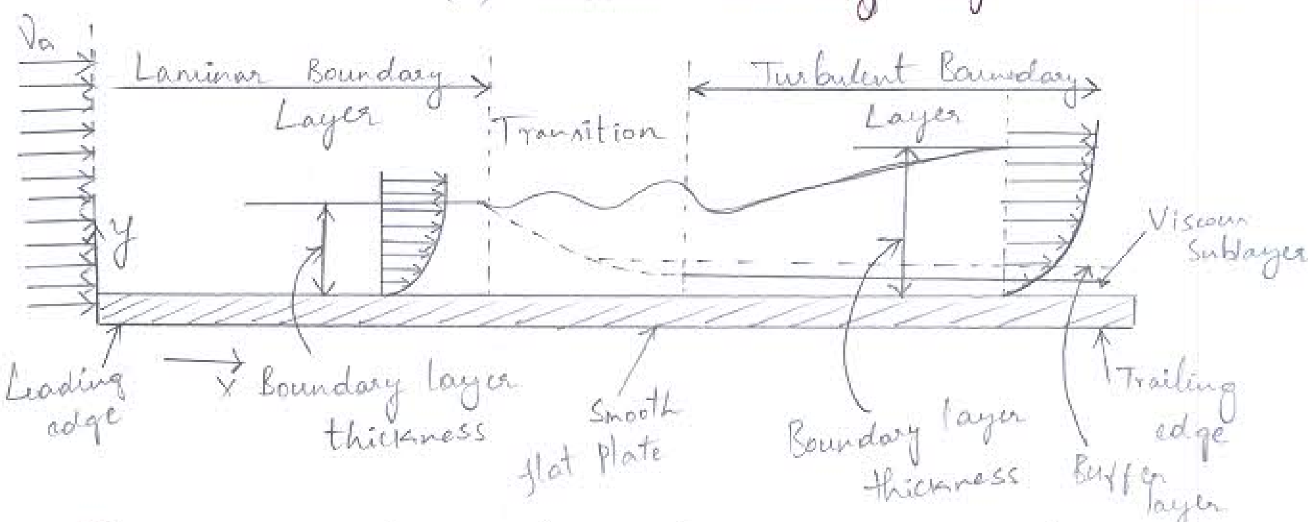
$$h_{bottom} = 0.0834 \ W/m^2K$$

$$q = hA\Delta T$$

$$q = (0.1668 + 0.0834)(24)(60 - 10)$$

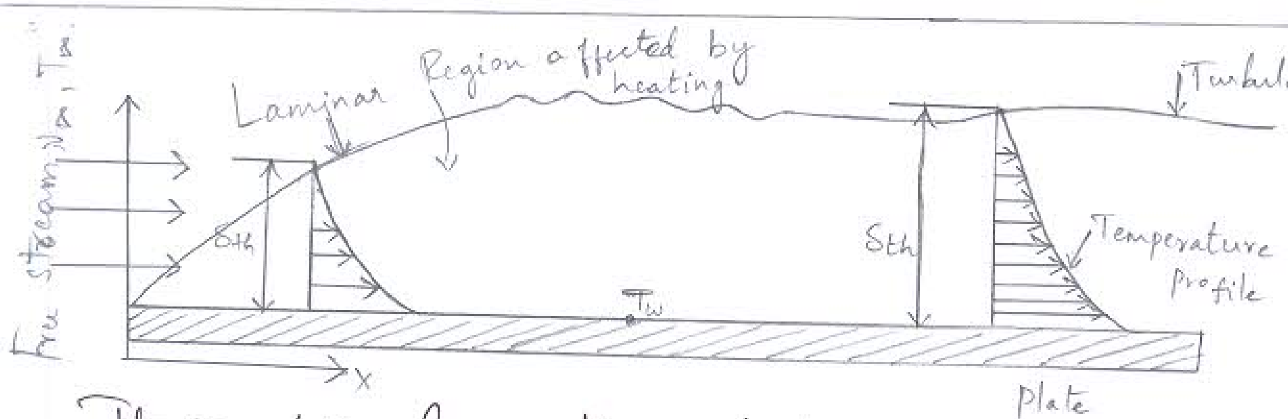
$$q = 300.24 \ W$$

4) a) Explain with Sketches (i) Boundary layer thickness.
(ii) Thermal boundary layer thickness.



It is defined as the distance measured perpendicularly from the boundary in which the velocity reaches 99% of the free Stream.

(1) Displacement thickness
(2) Momentum thickness.
(3) Energy thickness.



It is defined as the distance from the boundary in which the temperature difference reaches 99% of the initial temperature.

4) b) Explain the Significance of

(i) Reynolds No. :- It is defined as the ratio of inertia to viscous force.

$$Re = \frac{\rho V D}{\mu}$$

It signifies the relative pre dominance of inertia effect to the viscous effect in the flow system.

(ii) Prandtl no :- It is the ratio of kinematic viscosity to thermal diffusivity.

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

It is the connecting link between the velocity field and temperature field. Its value strongly influences the relative growth of velocity and thermal boundary layers.

(iii) Nusselt no :- It is the ratio of heat flow by convection under a unit temperature difference to the heat flow by conduction through a stationary thickness in L meters.

$$Nu = \frac{q_{conv}}{q_{cond}} = \frac{hA}{\frac{KA}{L}}$$

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

(iv) Stanton number :- It is the ratio of heat transfer coefficient to the flow of heat per unit temp rise due to the velocity of the fluid.

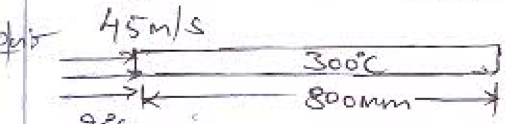
$$St = \frac{h}{\rho v C_p} \times \frac{k \times L \times \mu}{k \times L \times \mu}$$

$$St = \frac{hL}{k} \times \frac{k}{\mu C_p} \times \frac{\mu}{\rho VL}$$

$$St = \frac{N}{Pr \times Re}.$$

Used only for correlating forced convection data.

5)
a) Air at 20°c flows past a 800mm long plate at velocity of 45 m/s. If the surface of the plate maintained at 300°c, determine (i) The heat transferred from entire plate length to air taking into consideration both laminar and turbulent portion of boundary layer. (ii) The percentage error if the boundary layer is assumed to be of turbulent nature from the very leading edge of the plate. Assume unit width of the plate and Critical Reynold's no to be 5×10^5.



$Re_c = 5 \times 10^5$

$$T_m = \frac{T_s + T_\infty}{2} = \frac{300 + 20}{2} = 160°c$$

At 160°c,

$\nu = 30.09 \times 10^{-6}$

$Pr = 0.682$

$k = 0.03640.$

(i) $Re = \frac{VL}{\nu}$

$$5 \times 10^5 = \frac{45 \times L}{30.09 \times 10^{-6}}$$

$L_L = 0.334$ m

'L_L' is the length of Laminar flow

Nusselt no for the forced convection, Laminar flow

$$Nu = 0.332\, Re^{0.5}\, Pr^{0.333}$$

$$Nu = 0.332 \times (5 \times 10^5)^{0.5} \times (0.682)^{0.333}$$

$$Nu = 206.66.$$

$$h_L = \frac{Nu \times k}{L} = \frac{206.66 \times 0.03640}{0.334} = 22.52 \text{ W/m}^2\text{K}$$

Length of turbulent flow

$$L_t = 0.8 - 0.334 = 0.466.$$

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

$Re = \frac{45 \times 0.466}{30.09 \times 10^{-6}}$

$Re = 0.6969 \times 10^{6}$

$Nu = 0.0296\, Re^{0.8}\, Pr^{0.33}$

$Nu = 0.0296\, (0.6969 \times 10^{6})^{0.8}\, (0.682)^{0.33}$

$Nu = 1233.03$

$h = \frac{1233.03 \times 0.03640}{0.466} = 96.31\ W/m^2K$

$q = q_L + q_T \Rightarrow q = [(22.52 \times 0.334) + (96.31 \times 0.466)][280]$

$\boxed{q = 14672.6\ W}$

(ii) At the leading edge itself is turbulent flow occurs then

$Re = \frac{VL}{\nu} = \frac{45 \times 0.8}{30.09 \times 10^{-6}} = 1.196 \times 10^{6}$

$Nu = 0.0296\, Re^{0.8}\, Pr^{0.33}$

$Nu = 0.0296\, (1.196 \times 10^{6})^{0.8}\, (0.682)^{0.33}$

$Nu = 1899.43$

$h = \frac{Nu \times k}{L} = \frac{(899.43 \times 0.03640)}{0.8}$

$h = 86.42\ W/m^2K$

$q = 86.42 \times 0.8 \times 1\ (280)$

$q = 19359.08\ W$

$\therefore$ Percentage error $= \frac{19359.08 - 14672.6}{19359.08} = 0.242$

$\boxed{\%\ \text{error} = 24.2\%}$

5) b) Hot air at atmospheric pressure and 80°c enters an 8m long uninsulated square duct of cross section 0.2m × 0.2m that passes through the attic of a house at a rate of 0.15 m³/s. The duct is observed to be nearly isothermal at 60°c. Determine the exit temperature of the air and the rate of heat loss from the duct to the attic space.

Soln: Data :- $T_1 = 80°c$, $L = 8m$, $q = 0.15\ m^3/s$, $T_w = 60°c$, $T_2 = ?$

Properties of air at $T_1 = 80°c$

$\rho = 0.9994\ kg/m^3$, $C_p = 1008\ J/kgK$, $K = 0.02953\ W/mK$, $P_r = 0.7154$, $\nu = 20.9\times10^{-6}\ m^2/s$.

Hydraulic dia, $D_h = \frac{4A}{P} = \frac{4\times(0.2\times0.2)}{2(0.2+0.2)} = 0.2m.$

$q = AV = 0.2^2 \times V = 0.15$

$\Rightarrow V = 3.75\ m/s.$

$R_e = \frac{VD_h}{\nu} = \frac{3.75\times0.2}{20.9\times10^{-6}} = 35765.$

Since $R_e > 3000$, Flow is turbulent

$Nu = 0.023 R_e^{0.8} \times P_r^{0.3}$

$Nu = 0.023\times(35765)^{0.8}\times(0.7154)^{0.3}$

$Nu = 91.4$

$\frac{hD_h}{K} = 91.4 \Rightarrow \frac{h\times0.2}{0.02953} = 91.4$

$h = 13.5\ W/m^2K.$

$Q = mC_p(T_2-T_1) = hA_s\left[T_w - \left[\frac{T_1+T_2}{2}\right]\right]$

$\rho AVC_p(T_2-T_1) = hA_s\left[T_w - \frac{T_1+T_2}{2}\right]$

$0.9994\times0.2\times0.2\times3.75\times1008.0(T_2-80) = 13.5\times4\times0.2\times8\left(60 - \frac{80+T_2}{2}\right)$

$T_2 = 71.3°C$

$Q = mC_p(T_2-T_1) = 0.9994\times0.2\times3.75\times1008(71.3-80)$

$Q = -314.65\ W$

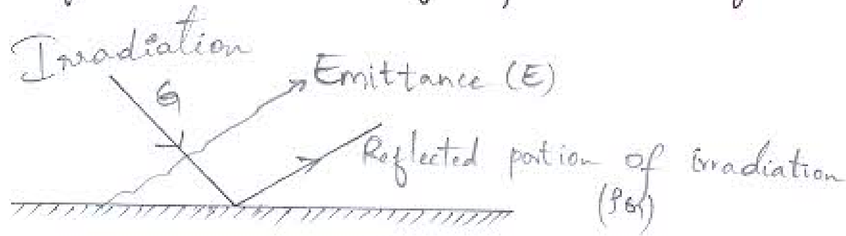
1) a) Explain the following terms:-

(a)(i) Black body:- It is defined as the body in which it is assigned to be a perfect absorber of radiation. Therefore for a black body, $\alpha = 1$, $\rho = \tau = 0$. Absorptivity can be increased by creating lamp black (or) a dark rough paint.

(ii) Grey body:- When a surface absorbs a certain fixed percentage of impringing radiation then the surface is called grey body ($\alpha < 1$).

(b) (i) Radiosity :- It is the total radiant energy leaving a surface per unit time per unit surface area. It comprises original emittance from the surface plus the reflected portion of any radiation incident upon it.

(ii) Irradiation :- It is the total incident energy inclined upon a surface per unit time per unit area. Some of it may be reflected to become part of the radiosity of the surface.



$$\underline{\underline{G = E + \rho G}}$$