# HMT

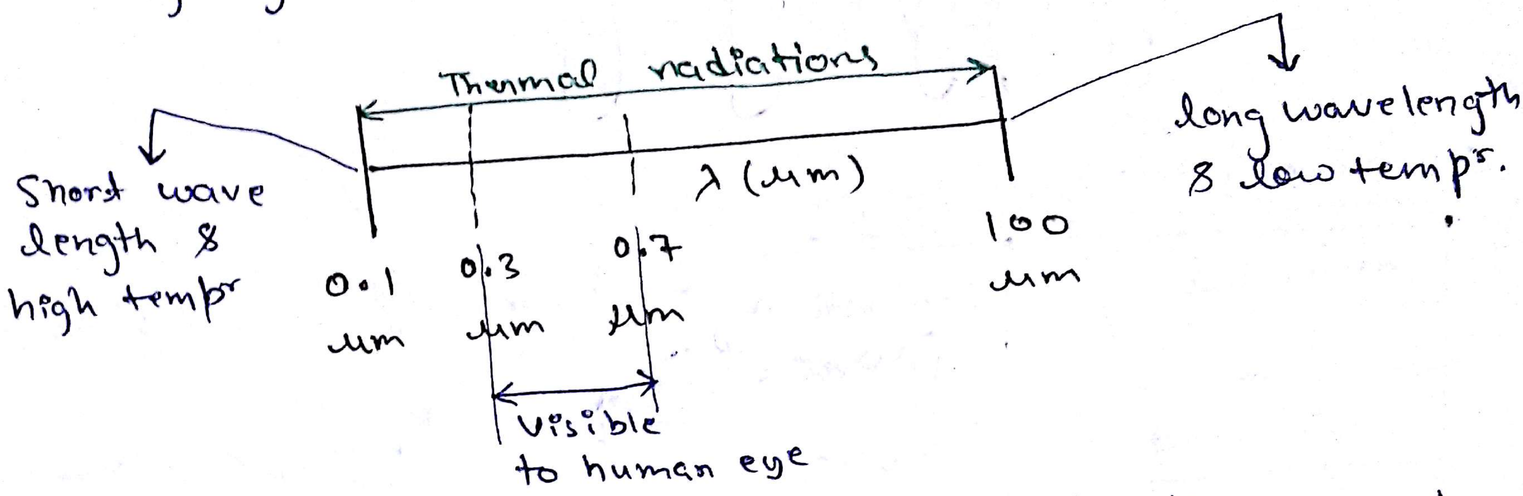
## CHAPTER-1 [Conduction]

| material | Conductivity (K) (w/mK) |
|---|---|
| Diamond | 2300 |
| Silver | 410 |
| Copper | 385 |
| Al | 200 |
| lead | 35 |
| Steel | 17 to 45 |
| Hg | 8.3 |
| glass | 1.5 |
| Refractory bricks | 0.9 |
| Asbestos | 0.2 |
| wood | 0.11 |
| glass wool | 0.075 |
| Steam (200°C) | 0.0334 |
| Air | 0.026 |
| Polyurethene Foam (PUF) | 0.02 |

* $K_{H_2O} = 0.63$ w/mK
* $K_{\text{pure metal}} > K_{\text{metal alloy}}$
* $K_{\text{liquid}} > K_{\text{gas}}$
* $K_{ice} > K_{H_2O} > K_{steam}$
*


| Conditions | h (w/m²K) |
|---|---|
| Free convection in Gases | 3 to 25 |
| Forced Convection in Gases | 25 to 400 |
| free Convection in liquid | 250 to 600 |
| forced Convection in liquid | 600 to 4000 |
| Condensation H.T. (vapour to liquid) | 3000 to 25,000 |
| Boiling Heat transfer | 5000 to 50,000 |

* Range of thermal radiations




⇒ optical pyrometer uses this wavelength concept to measure high temprs. [ $\lambda \propto \frac{1}{T}$ ]

- Fourier's Law of conduction

$$q_{cond} = -KA\left(\frac{dT}{dx}\right)$$

∴ $\frac{dT}{dx}$ is calculated at an 'x' where q is needed to be known.

- Newton's Law of cooling

$q \propto A$

$q \propto \Delta T$

$$q_{conv.} = hA(T_w - T_\infty)$$

- Stefan Boltzman's Law of Radiation

For a black body, Total hemispherical Emissive power per unit area per unit time = $E_b$

$$E_b \propto T^4 \Rightarrow E_b = \sigma T^4$$

$\sigma$ = Stefan boltzman const. = $5.67 \times 10^{-8}$ $W/m^2K^4$

- In General

$$Q = \frac{\Delta T}{\Sigma R_{th}}$$

| Resistance ⇓ | Slab | Cylinder | Sphere |
|---|---|---|---|
| Conduction | $\frac{l}{KA}$ | $\frac{\ln\left(\frac{r_2}{r_1}\right)}{2\pi K l}$ | $\frac{r_2 - r_1}{4\pi K r_1 r_2}$ |
| Convection | $\frac{1}{hA}$ | $\frac{1}{hA}$ | $\frac{1}{hA}$ |

- Overall heat transfer coefficient (U)

$$\frac{1}{U} = \Sigma R_{th} \times A \quad ; \quad \frac{1}{U_i} = \Sigma R_{th} \times A_i$$

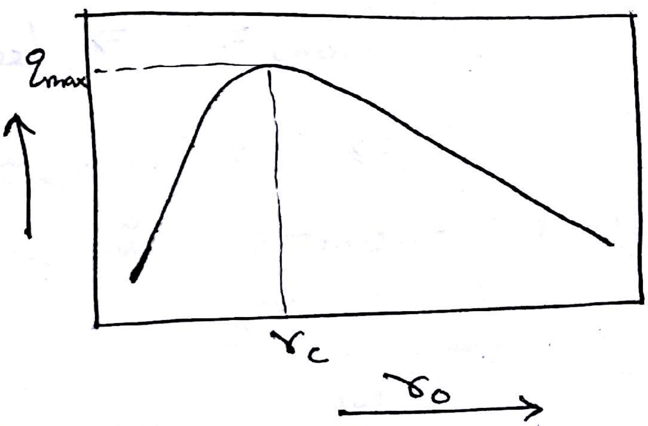
$$\frac{1}{U_o} = \Sigma R_{th} \times A_o$$

- Critical radius of insulation

cylinder ⇒ $r_c = \frac{K}{h}$

sphere ⇒ $r_c = \frac{2K}{h}$

here, K ⇒ insulating mat.




- 3D Generalised conduction eqn

1) Cartesian co-ordinates

$$\frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2} + \frac{\partial^2 T}{\partial z^2} + \frac{\dot{q}}{K} = \frac{1}{\alpha}\left(\frac{\partial T}{\partial \tau}\right)$$

$\alpha$ = Thermal diffusivity

$\alpha = \frac{K}{\rho C_p}$ $m^2/sec$

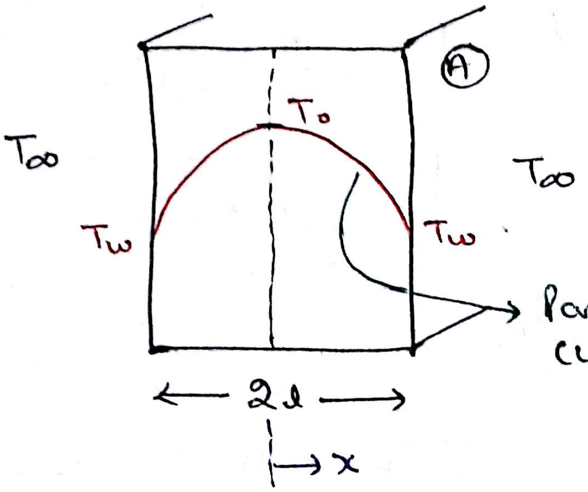
2) Cylindrical co-ordinates

$$\frac{\partial^2 T}{\partial r^2} + \frac{1}{r}\frac{\partial T}{\partial r} + \frac{1}{r^2}\frac{\partial^2 T}{\partial \phi^2} + \frac{\partial^2 T}{\partial z^2} + \frac{\dot{q}}{K} = \frac{1}{\alpha}\left(\frac{\partial T}{\partial \tau}\right)$$

3) 1D unsteady state eqn for sphere

$$\frac{1}{r^2} \cdot \frac{d}{dr}\left(r^2 \frac{dT}{dr}\right) + \frac{\dot{q}}{K} = \frac{1}{\alpha}\left(\frac{\partial T}{\partial \tau}\right)$$

- Heat Generation in slab




$$\Rightarrow T_0 - T = \frac{\dot{q} x^2}{2K}$$

$$T_0 - T_w = \frac{\dot{q} l^2}{2K}$$

$$\frac{T_0 - T}{T_0 - T_w} = \frac{x^2}{l^2}$$

$$T_w - T_\infty = \frac{\dot{q} l}{h}$$

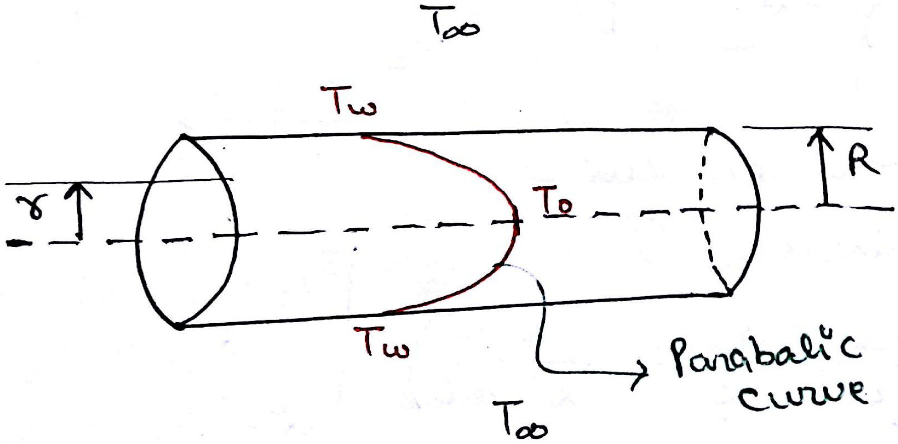
$$\Rightarrow \boxed{T_0 = \frac{\dot{q} l^2}{2K} + \frac{\dot{q} l}{h} + T_\infty}$$

$\Rightarrow$ when one side is insulated, then at that side

$q_{conv.} = 0 \;\&\; q_{cond} = 0 \Rightarrow \left(\frac{dT}{dx}\right)_{side} = 0$

$\Rightarrow T = T_{max}$ at insulated surface.

- Heat generation in cylinder




$$\Rightarrow T_0 - T = \frac{\dot{q} r^2}{4K}$$

$$T_0 - T_w = \frac{\dot{q} R^2}{4K}$$

$$\frac{T_0 - T}{T_0 - T_w} = \frac{r^2}{R^2}$$

$$T_w - T_\infty = \frac{\dot{q} R}{2h}$$

$$\Rightarrow \boxed{T_0 = \frac{\dot{q} R^2}{4K} + \frac{\dot{q} R}{2h} + T_\infty}$$

- Heat generation in sphere

$$\Rightarrow T_0 - T_w = \frac{\dot{q} R^2}{6K}$$

$$T_w - T_\infty = \frac{\dot{q} R}{3h}$$

$$\Rightarrow \boxed{T_0 = \frac{\dot{q} R^2}{6K} + \frac{\dot{q} R}{3h} + T_\infty}$$

- **Fins**

$$\theta = T - T_\infty = C_1 e^{-mx} + C_2 e^{mx}$$

$$m = \sqrt{\frac{hP}{KA}} \quad ; \quad P = \text{Perimeter of fin}, \quad A = \text{X-S/C Area}$$

$l_c = l + \frac{t}{2} \Rightarrow$ rectangular fin

$l_c = \frac{d}{4} + l \Rightarrow$ Pin fin

| Characteristic ⇓ | Infinite long fin | Fin → finite length, ↳ insulated tip [use this case if not given in Q] | Fin → finite length, ↳ non-insulated tip |
|---|---|---|---|
| $\frac{\theta}{\theta_0} = \frac{T - T_\infty}{T_0 - T_\infty} =$ | $e^{-mx}$ | $\frac{\cosh[m(l-x)]}{\cosh(ml)}$ | $\frac{\cosh[m(l_c-x)]}{\cosh(ml_c)}$ |
| $q_{fin} \Rightarrow$ | $\sqrt{hPKA} \cdot (T_0 - T_\infty)$ | $\sqrt{hPKA} \times (T_0 - T_\infty) \tanh(ml)$ | $\sqrt{hPKA} \times (T_0 - T_\infty) \times \tanh(ml_c)$ |
| Efficiency $\eta_{fin} = \frac{q_{actual}}{q_{max}} \Rightarrow$ | $\frac{1}{ml}$ | $\frac{\tanh(ml)}{(ml)}$ | $\frac{\tan(ml_c)}{(ml_c)}$ |
| Effectiveness $\epsilon = \frac{q_{with\ fin}}{q_{without\ fin}} \Rightarrow$ | $\sqrt{\frac{PK}{hA}}$ | $\sqrt{\frac{PK}{hA}} \times \tanh(ml)$ | $\sqrt{\frac{PK}{hA}} \times \tanh(ml_c)$ |

- Unsteady or Transient conduction Heat transfer

$\Rightarrow hA(T-T_\infty) = -mC_p \frac{dT}{d\tau}$ ⇒ to be used when rate of cooling is asked.

$\Rightarrow e^{\left(\frac{hA}{mC_p}\right)\tau} = \frac{T_i - T_\infty}{T - T_\infty}$ ⇒ To be used when $T$ or $\tau$ is asked

here, $\frac{mC_p}{hA} = \frac{\rho V C_p}{hA}$ = Time const (TC) sec

- Reading on thermocouple is taken at 3×(TC)

- Lumped Heat Analysis

This is applied when Biot no. < 0.1

Biot no. $= \frac{hS}{K} = \frac{S/KA}{1/hA}$ = $\frac{\text{Internal conduction Resistance}}{\text{External convection Resistance}}$

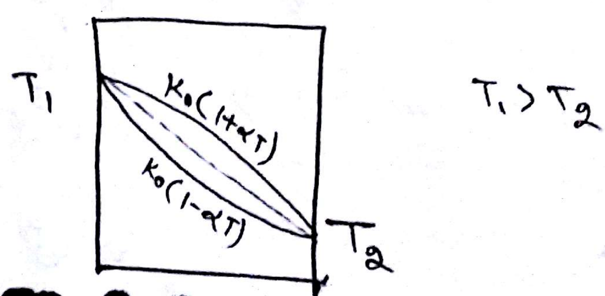
K = conductivity of object ; $S = \frac{\text{Volume}}{\text{Surface Area}}$

| material shape | $S = \frac{V}{A}$ |
|---|---|
| Slab | $\frac{V}{A} = \frac{At}{2A} = \frac{t}{2}$ |
| Cylinder | $\frac{V}{A} = \frac{\pi d^2 l}{4 \times \pi d l} = \frac{d}{4}$ |
| Sphere | $\frac{V}{A} = \frac{\text{Vol.}}{\text{SA.}} = \frac{d}{6}$ |
| Cube | $\frac{V}{SA} = \frac{a}{6}$ |

- If Biot no. > 0.1 then Heisler charts must be used.
- Variable thermal conductivity

$K = K_0(1 \pm \alpha T)$

Solve question by taken K at average value



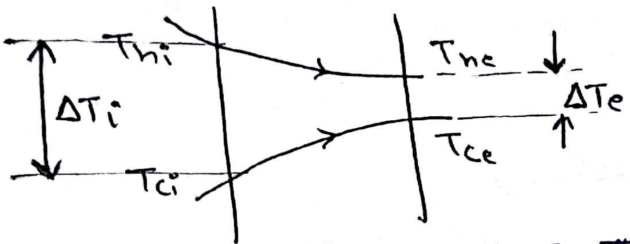
- Two objects are said to be thermally equivalent only if ⇒ Total thickness is same
  ⇒ $R_{th}$ is same

## CHAPTER-2 [Heat exchangers]

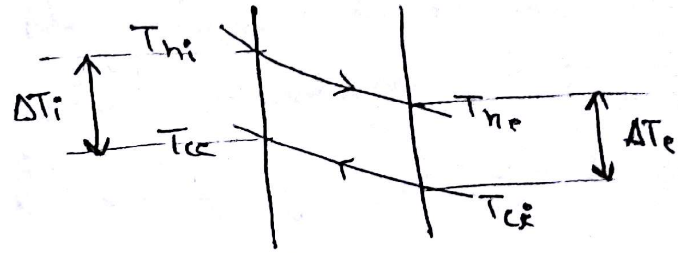
- Energy balance equation

$$\dot{m}_h C_{ph} (T_{hi} - T_{he}) = \dot{m}_c C_{pc} (T_{ce} - T_{ci})$$

Parallel flow HE




Counter flow HE




$$\Rightarrow LMTD = \Delta T_m = \frac{\Delta T_i - \Delta T_e}{\ln\left(\frac{\Delta T_i}{\Delta T_e}\right)}$$

$$\Rightarrow LMTD = \Delta T_m = \frac{\Delta T_i - \Delta T_e}{\ln\left(\frac{\Delta T_i}{\Delta T_e}\right)}$$

$$(\Delta S)_{\substack{uni. \\ parallel}} > (\Delta S)_{\substack{uni. \\ counter}}$$

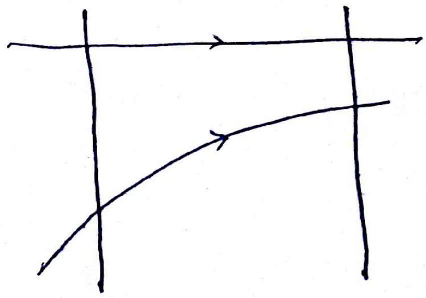
- Counter flow HE $(\Delta T)_m$ is higher so smaller in size.
- $(LMTD)_{cross\ flow} = F \times (LMTD)_{counterflow}$

  F = correction factor

- $(\Delta T_m)_{counter} > (\Delta T_m)_{cross} > (\Delta T_m)_{parallel}$

Special case

① when one of the fluid is changing its phase



- LMTD will be calculated as earlier
- $(LMTD)_{parallel} = (LMTD)_{counter}$

② when both hot and cold fluids have equal heat capacity ( $\dot{m}_h C_{p_h} = \dot{m}_c C_{p_c}$ )

then, $LMTD_{counter} = \Delta T_i = \Delta T_e$

- This type of HE is called balanced counterflow HE
- Tempr profile will be parallel straight line.

⇒ Heat transfered will be minimum in Parallel flow HE when heat capacity is same for both fluids. (i.e. $\dot{m}_h C_{p_h} = \dot{m}_c C_{p_c}$ )

⇒ Heat transfered ⇒ $Q = UA(\Delta T_m)$

U = overall heat transfer coefficient

$$\frac{1}{U} = \frac{1}{h_1} + F_1 + \frac{1}{h_2} + F_2$$

$F_1$, $F_2$ → Fouling factor

A = Total surface area of tubes

$A = (\pi d l) \times n \times p$

n = no. of tubes per pass

p = no. of passes

- Effectiveness of HE ( $\epsilon$ )

$$\epsilon = \frac{q_{actual}}{q_{max.\,Possible}}$$

$\therefore q_{max} = (\dot{m} C_p)_{smaller} \times (\Delta T)_{max}$

$\Delta T_{max} = T_{hi} - T_{ci}$

$$\epsilon = \frac{T_{hi} - T_{he}}{T_{hi} - T_{ci}}$$ ⇒ when $\dot{m}_h C_{p_h} < \dot{m}_c C_{p_c}$

$$\epsilon = \frac{T_{ce} - T_{ci}}{T_{hi} - T_{ci}}$$ ⇒ when $\dot{m}_c C_{p_c} < \dot{m}_h C_{p_h}$

- Number of transfer units (NTU)

$$NTU = \frac{UA}{(\dot{m} C_p)_{smaller}}$$

U → W/m²K, A → m², $C_p$ → J/KgK, $\dot{m}$ → Kg/s

- Capacity Ratio (C)

$$C = \frac{(\dot{m} c_p)_{smaller}}{(\dot{m} c_p)_{larger}}$$

$\therefore$ $\dot{m}c_p$ = capacity rate

$0 \leq C \leq 1$

$C = 0$, when one of fluid is changing its phase

- $\epsilon = f(NTU, C)$

$$\epsilon = \frac{1 - e^{-(1+C)NTU}}{1+C} \Rightarrow \text{for parallel flow}$$

$$\epsilon = \frac{1 - e^{-(1-C)NTU}}{1 - Ce^{-(1-C)NTU}} \Rightarrow \text{for counter flow.}$$

Case-1 when $C = 0$ [when one fluid is changing phase]

$$\epsilon = 1 - e^{-NTU} \Rightarrow \text{for both parallel \& counter flow}$$

Case-2 when $C = 1$ [$\dot{m}_h c_{ph} = \dot{m}_c c_{pc}$]

$$\epsilon = \frac{1 - e^{-2NTU}}{2} \Rightarrow \text{for parallel flow}$$

$$\epsilon = \frac{NTU}{NTU + 1} \Rightarrow \text{for counter flow}$$

## CHAPTER -3 (Radiation)

E = Total hemispherical emissive Power

$$E = \int_0^\infty E_\lambda \, d\lambda \quad \frac{watt}{m^2}$$ = Area under $E_\lambda$ & $d\lambda$ graph

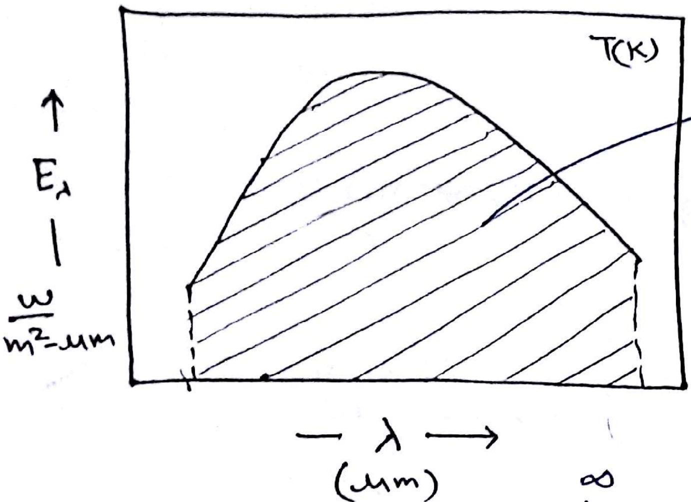
$E_\lambda$ = monochromatic/spectral hemispherical emissive Power

$$E_\lambda \Rightarrow \frac{watt}{m^2 - \mu m}$$

$\epsilon$ = Total emissivity

$\epsilon_\lambda$ = monochromatic / spectral emissivity

$$\epsilon = \frac{E}{E_b} \quad ; \quad \epsilon_\lambda = \frac{E_\lambda}{E_{b\lambda}}$$




→ Area under $E_\lambda$ & $\lambda$ graph

$$\Rightarrow E = \int_0^\infty E_\lambda \, d\lambda$$

$$\boxed{\alpha_{avg} = \frac{\int_0^\infty \alpha_\lambda G_\lambda \, d\lambda}{\int_0^\infty G_\lambda \cdot d\lambda}}$$

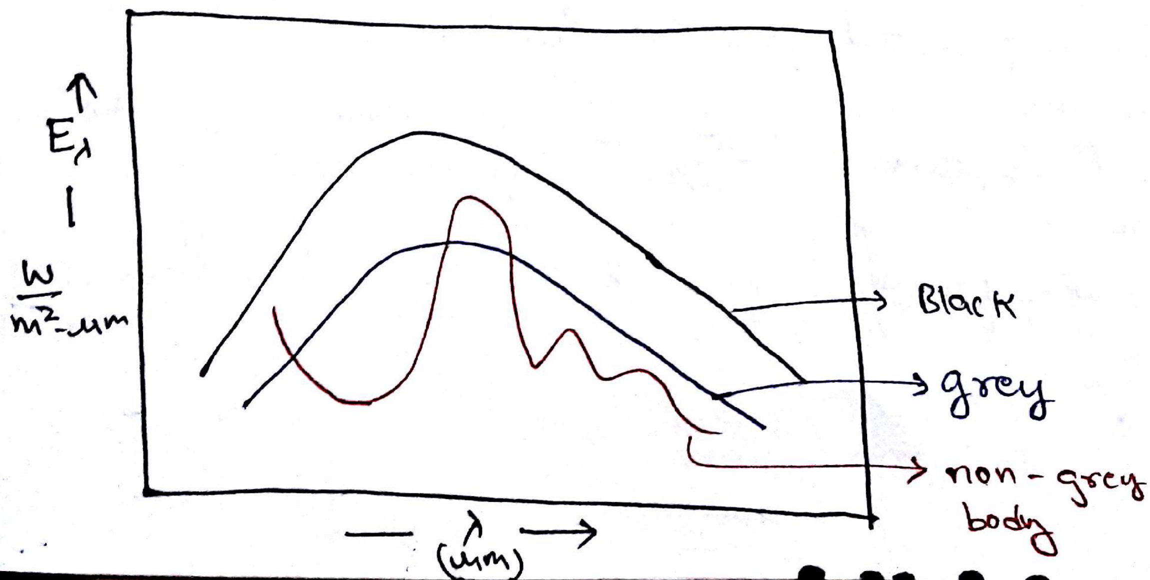
$$* \quad \epsilon = \frac{E}{E_b} = \frac{\int_0^\infty (E_\lambda) d\lambda}{\int_0^\infty E_{b\lambda} \, d\lambda} = \frac{\int_0^\infty (\epsilon_\lambda \cdot E_{b\lambda}) \, d\lambda}{\int_0^\infty E_{b\lambda} \, d\lambda}$$

above formula can be used to calculate $\epsilon_{avg}$. Break the limits on Numerator if different "$\epsilon$" is given for different range of "$\lambda$".

- Grey body

$$\epsilon_\lambda \neq f(\lambda)$$

hence from above formula $\epsilon = \epsilon_\lambda$ = const. for a grey body.



- $\alpha$ = Absorptivity :→ fraction of incident radiation absorbed

$\rho$ = Reflectivity :→ fraction of incident radiation reflected

$\tau$ = Transmissivity :→ fraction of incident radiation transmitted

$\alpha + \rho + \tau = 1$ [ for opaque body $\tau = 0$ ]

- For a black boby ⇒ $\alpha_b = 1$ ; $\epsilon_b = 1$

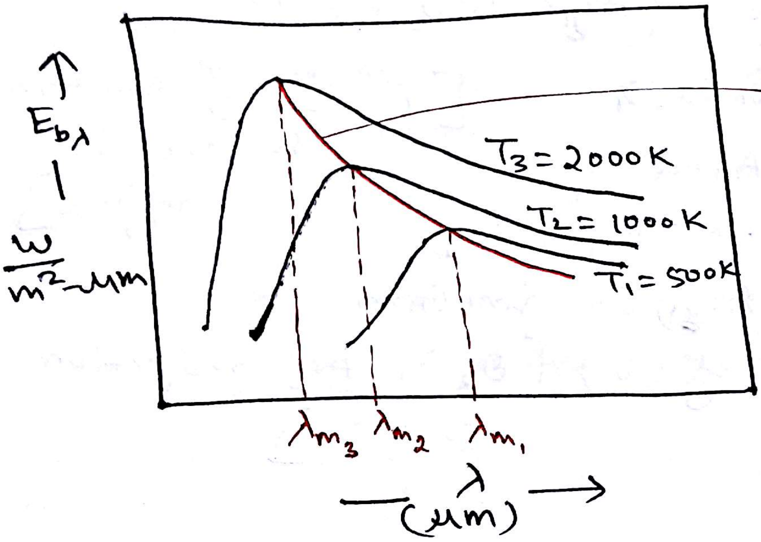
Laws of radiation

1. Kirchoff's Law   At thermal equilibrium

$$\alpha = \epsilon$$

2. Plank's Law

$E_{b\lambda} = f(\lambda, T)$ ; $E_{b\lambda} = \dfrac{2\pi C_1}{\lambda^5 \left[e^{C_2/\lambda T} - 1\right]}$




$\lambda_m$ = wavelength at which $E_{b\lambda}$ is maximum for a given tempr

3. Wein's Displacement Law

$$\lambda_m T = \text{constant} = 2898\ \mu m\text{-}K$$

4. Stefan - Boltzman's Law

$E_b \propto T^4$

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

$\sigma$ = Stefan boltzmann const.

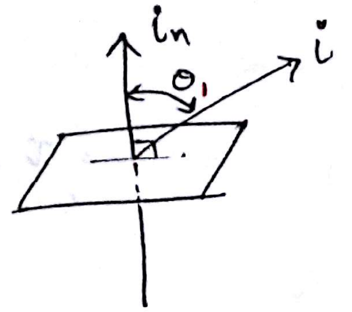
$= 5.67 \times 10^{-8}$ $W/m^2K^4$

For a non-black body

$E = \epsilon E_b$

$E = \epsilon \sigma T^4$ $\frac{W}{m^2}$

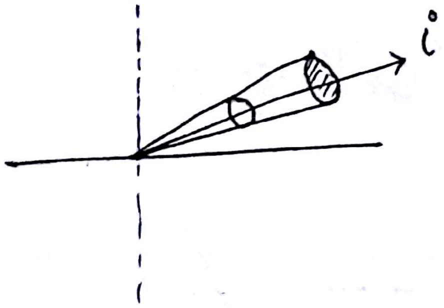
## 5. Lambert's Cosine Law




$i = i_n \cos\theta_1$

$\theta_1$ = angle with normal of plane

$i \Rightarrow \dfrac{w}{m^2 - \text{steradian}}$




Steradian is solid angle ($\omega$)

$$\omega = \frac{A_{r_2}}{r^2} = \frac{A_{r_2}}{r^2}$$

here, $A_{r_2}$ = Area of receiving body

$r$ = distance b/w 2 bodies

Total Emission $\Rightarrow E_{1-2} = i \times (A_{r_1} \times \cos\theta_2) \times \omega$

here $i = i_n \cos\theta_1$ [$i = i_n$ if emitting surface is diffuse]

$A_{r_1}$ = Area of emitting body

$\theta_2$ = ($A_{r_1}$) in direction of ($i$) [$\theta_2$ is $A_{r_2}$ projection angle]

~~Generally~~ or always

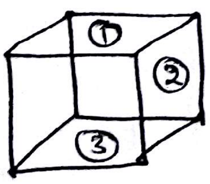
$\theta_1 = \theta_2$

- For black body

$$E_b = \pi \times i_n \quad w/m^2$$

- Reciprocity relation $\Rightarrow A_1 F_{12} = A_2 F_{21}$
- $F_{11} + F_{12} + F_{13} \; --- \; F_{1n} = 1$

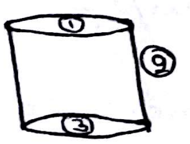
**For cube**




$F_{12} = F_{23} = 0.2$

$\Rightarrow \boxed{F_{13} = 0.2}$

**For cylinder [$l = d$]**




$F_{12} = 0.83$

$\boxed{F_{13} = 0.17}$

$F_{21} = 0.2075$

$\Rightarrow F_{22} = 0.585$

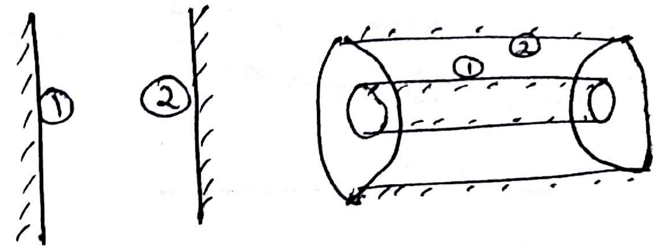
- Surface resistance $\Rightarrow \dfrac{1-\epsilon}{A\epsilon}$
- Space resistance $\Rightarrow \dfrac{1}{AF}$
- General radiation Equation

$$(q_{1-2})_{net} = \frac{E_{b_1} - E_{b_2}}{\Sigma R_{th}} = \frac{\sigma(T_1^4 - T_2^4)}{\Sigma R_{th}}$$

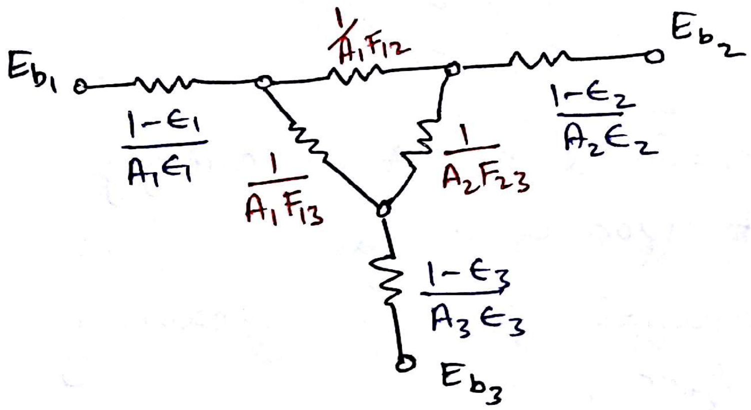
- Calculation of $\Sigma R_{th}$

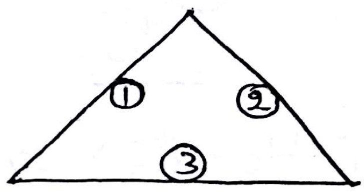
① For 2 bodies


$$R_{th} = \frac{1-\epsilon_1}{A_1\epsilon_1} + \frac{1}{A_1F_{12}} + \frac{1-\epsilon_2}{A_2\epsilon_2}$$




② For 3 bodies




Case-1 Infinitely long Parallel plates

$$\frac{q_{1-2}}{A} = \frac{\sigma(T_1^4 - T_2^4)}{\frac{1}{\epsilon_1} + \frac{1}{\epsilon_2} - 1} \quad W/m^2$$

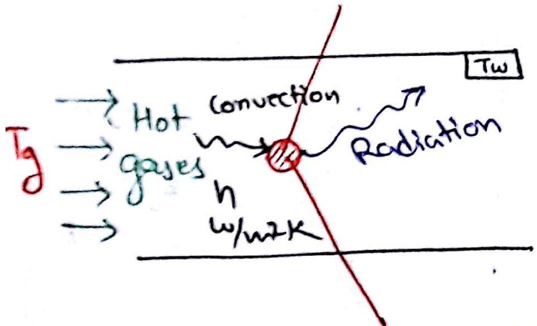
Case-2 Infinitely long Parallel Plates with shield in b/w

$$\frac{q_{1-2}}{A} = \frac{\sigma(T_1^4 - T_2^4)}{\frac{1}{\epsilon_1} + \frac{2}{\epsilon_s} + \frac{1}{\epsilon_2} - 2} \quad W/m^2$$

Case-3 Infinitely long concentric cylinder

$$q_{1-2} = \frac{\sigma(T_1^4 - T_2^4)}{\frac{1-\epsilon_1}{A_1\epsilon_1} + \frac{1}{A_1F_{12}} + \frac{1-\epsilon_2}{A_2\epsilon_2}} = \frac{\sigma(T_1^4 - T_2^4)A_1}{\frac{1}{\epsilon_1} + \frac{A_1}{A_2}\left(\frac{1}{\epsilon_2} - 1\right)}$$

* Error in tempr measurement of gases by using thermocouple




$T_g$ = True gas tempr

$T_c$ = reading on thermocouple

$T_w$ = Duct wall tempr

Convection HT = Radiation HT

$$h\cancel{A}\underbrace{(T_g - T_c)}_{\text{error}} = \sigma(T_c^4 - T_w^4)\cancel{A}\,\epsilon_{\text{th. coup.}}$$

- Irradiation (G) ⇒ Total thermal radiation energy incident upon a surface
- Radiosity (J) ⇒ Total thermal radiation leaving the surface.

$$J = E + (\rho + \tau)G \qquad [\rho + \tau = 1 - \alpha]$$

$$J = \epsilon E_b + (1-\alpha)G$$

$$J = \epsilon E_b + (1-\epsilon)G \qquad \text{[during thermal Equilibrium]}$$

- When "n" no. of shields are inserted b/w two infinite parallel plates [given ⇒ $\epsilon_1 = \epsilon_2 = \epsilon_{s_1} = \epsilon_{s_2} = \epsilon$] then,

$$(q_{1-2})_{\text{with n shield}} = \left(\frac{1}{n+1}\right)(q_{1-2})_{\text{without shield}}$$

# CHAPTER - 4 [Convection]

- Reynold's no. ⇒ $Re = \frac{\rho \vec{V} D}{\mu} = \frac{\text{inertia force}}{\text{viscous force}}$

$$Re = \frac{VD}{\nu} \; ; \; \nu = \frac{\mu}{\rho} \quad \text{or} \quad \mu = \rho \nu$$

- Nusselt no. ⇒ $Nu = \frac{hD}{K}$ [Dimensionless H.T. coefficient]

- Prandtl no. ⇒ $Pr = \frac{\mu C_p}{K} = \frac{\nu}{\alpha}$ $\left[\because \nu = \frac{\mu}{\rho}, \; \alpha = \frac{K}{\rho C_p}\right]$

- Stanton no. ⇒ $St = \frac{h_x}{\rho V_\infty C_p}$

- Newton's Law of viscosity ⇒ $\tau = \mu\left(\frac{\partial u}{\partial y}\right)$

## A) Forced convection [Flow over flat plate]

- D = characteristic Dimension of body which is used to calculate dimensionless number.

  For flat plate ⇒ D = length of plate [in dir$^n$ of flow]

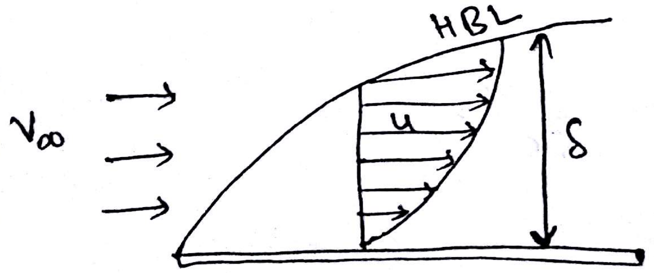
- if $Re < 5 \times 10^5$ ⇒ Laminar flow

  $Re > 6.5$ to $7 \times 10^5$ ⇒ turbulent flow

- **HBL [Laminar flow]**

1) Velocity Distribution

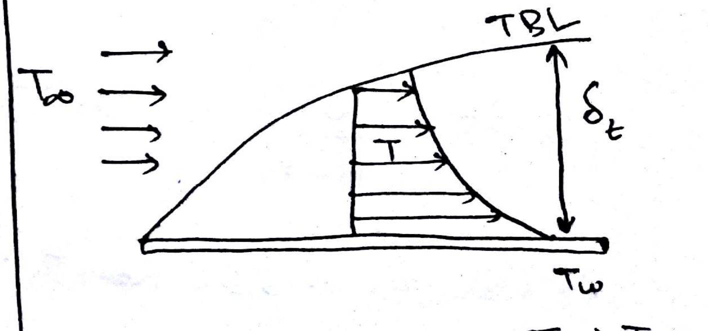
$$\frac{u}{V_\infty} = \frac{3}{2}\left(\frac{y}{\delta}\right) - \frac{1}{2}\left(\frac{y}{\delta}\right)^3$$




**TBL [Laminar flow]**

1) Temp$^r$ Distribution

$$\frac{T - T_w}{T_\infty - T_w} = \frac{3}{2}\left(\frac{y}{\delta_t}\right) - \frac{1}{2}\left(\frac{y}{\delta_t}\right)^3$$




$T_w > T_\infty$

2) momentum equation

$$u\frac{\partial u}{\partial x} + v\frac{\partial u}{\partial y} = \nu\frac{\partial^2 u}{\partial y^2}$$

3) $\tau_{w_x} = \mu\left(\frac{\partial u}{\partial y}\right)_{y=0}$

$$\tau_w = \mu \times V_\infty \times \frac{3}{2\delta}$$

$\Rightarrow \delta \propto x^{1/2}$

$\Rightarrow \tau_w \propto x^{-1/2}$

$\Rightarrow h_x \propto x^{-1/2}$

2) Energy equation

$$u\frac{\partial T}{\partial x} + v\frac{\partial T}{\partial y} = \alpha\frac{\partial^2 T}{\partial y^2}$$

3) $-K_f \cancel{A}\left(\frac{\partial T}{\partial y}\right) = h_x \cancel{A}(T_w - T_\infty)$

$$\Rightarrow h_x = \frac{-K_f\left(\frac{\partial T}{\partial y}\right)_{y=0}}{T_w - T_\infty}$$

$$\Rightarrow h_x = \frac{3K_f}{2\delta_{tx}}$$

$$\Rightarrow Nu_x = \frac{3}{2}\frac{x}{\delta_{tx}}$$

- $\delta = 1.026 \times \delta_t \times Pr^{1/3}$
- $\delta = \dfrac{4.64x}{\sqrt{Re_x}}$ or $\delta = \dfrac{5x}{\sqrt{Re_x}}$ [Blausius eqn] more accurate
- $\delta_t \propto x^{1/2}$
- For fully developed laminar flow over plate

$$Nu_x = \frac{h_x D}{K} = 0.332\, Re_x^{1/2}\, Pr^{1/3} \qquad \left[\therefore \bar{h} = 2h_e\right]$$

$$\bar{Nu} = \frac{\bar{h}D}{K} = 0.664\, Re_e^{1/2}\, Pr^{1/3}$$

$$\bar{h} = \frac{1}{l}\int_0^l h_x\,dx \quad \Rightarrow \quad \bar{h} = 2h_e$$

$$St = \frac{Nu}{Re\,Pr} = 0.332\, Re^{-1/2}\, Pr^{-2/3}$$

$$Pr^{2/3} \times St = 0.332\, Re^{-1/2} = \frac{Cf_x}{2}$$

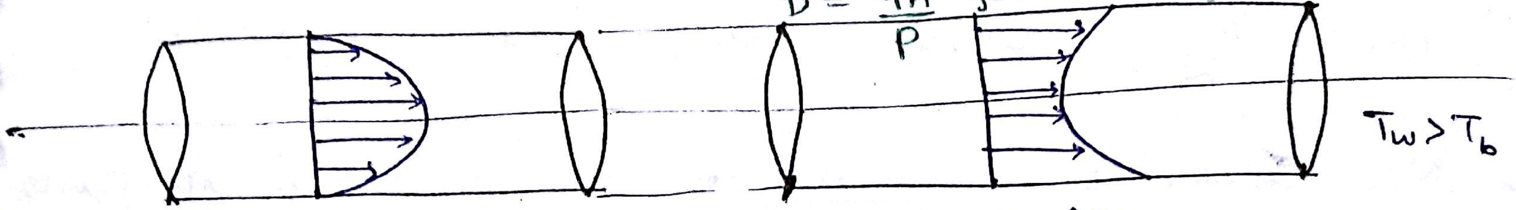
[$Cf_x$ = skin friction coeficient]

- Peclet no ⇒ $Pe = Re \cdot Pr$
- $St\, Pr^{2/3} = \frac{C_{fx}}{2}$ ⇒ Reynold's analogy
- $Nu = (0.037 Re^{0.8} - 871) Pr^{1/3}$ ⇒ for flow upto turbulence on flat plate

(B) Forced convection [through pipes & Ducts]

D = characteristic dimension ⇒ D = dia for pipes

$D = \frac{4A}{P}$ for rectangular X-S/c




$u = f(r)$ at a given X-S/c

$T = f(r)$ at a given X-S/c

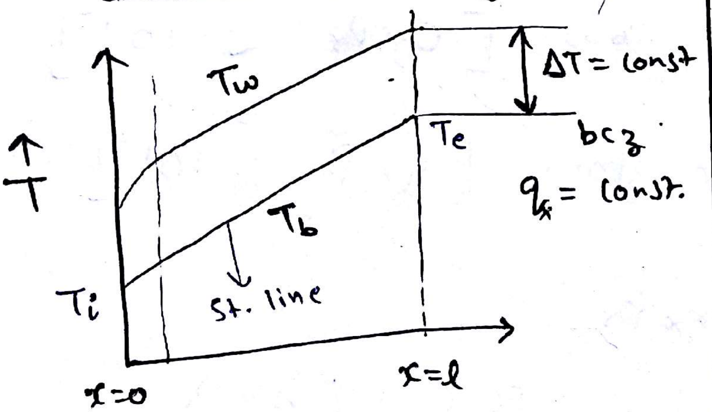
- mean velocity = $U_m$

$$U_m = \frac{2}{R^2}\int_0^R u\, r\, dr \qquad [\because u = f(r)]$$

- Bulk mean temp^r = $T_b$

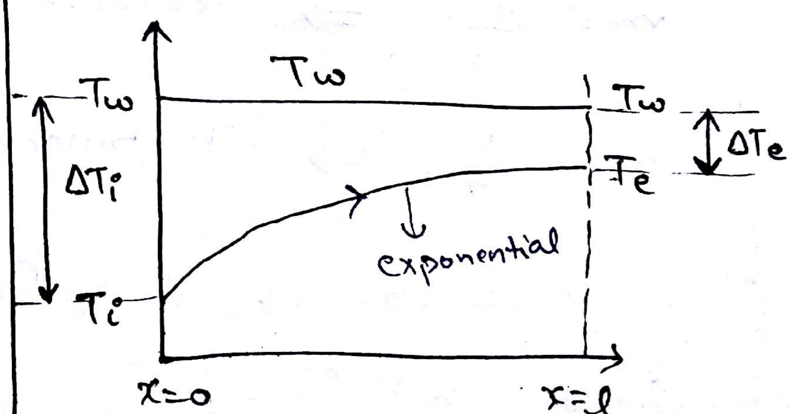
$$T_b = \frac{2}{U_m R^2}\int_0^R u T\, r\, dr \qquad \begin{bmatrix} u = f(r) \\ T = g(r) \end{bmatrix}$$

**Constant Heat flux**




- $h_x$ = const.
- $\dot m C_p (T_e - T_i) = q A_s = h A_s \Delta T$

**Const. wall temp^r**




- $$\frac{\Delta T_i}{\Delta T_e} = e^{\frac{h A_s}{\dot m C_p}}$$
- $Q = \dot m C_p (T_e - T_i) = h A_s (\Delta T_m)$
- $$(\Delta T_m) = \frac{\Delta T_i - \Delta T_e}{\ln\left(\frac{\Delta T_i}{\Delta T_e}\right)} = LMTD$$

- Nusselt no. for fully developed laminar flow through pipes / Ducts [$Re < 2000$]

$$Nu = \frac{hD}{K} = 4.36 \Rightarrow \text{for const. Heat flux}$$

$$Nu = \frac{hD}{K} = 3.66 \Rightarrow \text{for const. wall temp}^r$$

- Nusselt no. for turbulent flow through pipes/Ducts [$Re > 4000$]

$$Nu = \frac{hD}{K} = 0.023\, Re^{0.8}\, Pr^{n}$$

$n = 0.4 \Rightarrow$ for Heating of fluid

$n = 0.3 \Rightarrow$ for cooling of fluid

(C) Free convection or Natural convection

- $Nu = C\,(Gr.Pr)^m$ [C will be given in Question]

$m = \frac{1}{4} \Rightarrow$ Laminar flow [$Gr.Pr < 10^9$]

$m = \frac{1}{3} \Rightarrow$ Turbulent flow [$Gr.Pr > 10^9$]

- Rayleigh no. $\Rightarrow$ $Ra = Gr \times Pr$.
- $Gr = \dfrac{g \beta \Delta T l^3}{\nu^2}$ $\qquad \beta = \frac{1}{V}\left(\frac{\partial V}{\partial T}\right)_P$

$l$ = characteristic dimention $\qquad \beta = \dfrac{1}{T_{mean}(K)}$ for ideal gas

$l$ = height [for plates (verticle)] & verticle cylinder $\qquad \Delta T = T_w - T_\infty$

$l$ = dia for horizontal cylinder/sphere $\qquad T_{mea} \Rightarrow$ in Kelvin

$l = \dfrac{Area}{Perimeter}$ for horizontal plate