HEAT TRANSFER

(CONDUCTION + RADIATION ONLY)
+ HEAT EXCHANGER

SILVER ($k = 410\ W/mK$)

COPPER ($k = 385\ W/mK$)

$K_{PURE} > K_{ALLOY}$

ALUMUNIUM ($k = 200\ W/mK$)

STEELS ($k = 17$ to $45\ W/mK$)

GOLD ($k = 319\ W/mK$)

DIAMOND ($k = 2300\ W/mK$)

ASBESTOS ($k = 0.2\ W/mK$)

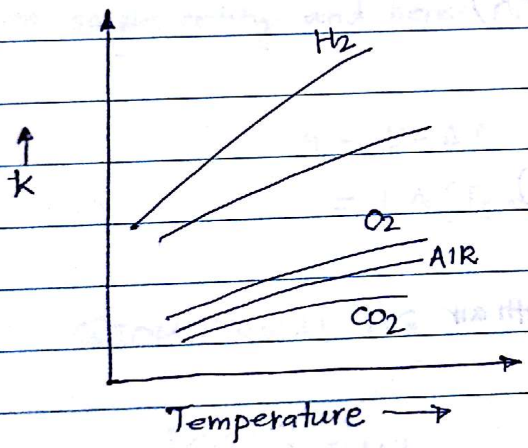
GLASSWOOL ($k = 0.075\ W/mK$)

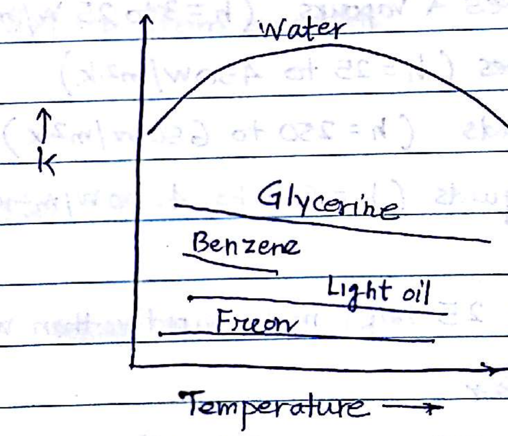
REFRACTORY BRICK ($k = 0.9\ W/mK$)

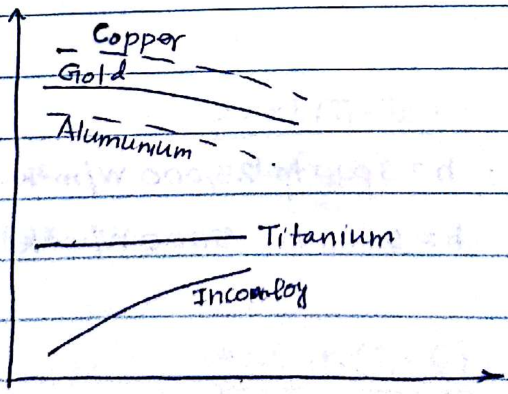
POLYURETHENE FOAM (PUF) ($k = 0.02\ W/mK$)

WATER ($k = 0.63\ W/mK$)

MERCURY ($k = 8.4\ W/mK$) ; highest K among liq.







Optical pyrometer :- sense radiation, wavelength converted to temperature

In forced convection H·T ;

$$h = f(\vec{V}, D, \underbrace{\rho, \mu, C_p, k}_{\Downarrow \text{ properties of fluid}})$$

In free convection

$$h = f(g, \beta, \Delta T, L, \underbrace{\mu, \rho, C_p, k}_{\Downarrow \text{ properties of fluid}})$$

β = Isobaric volume expansion coefficient of fluid

$$= \frac{1}{V}\left(\frac{\partial V}{\partial T}\right)_P$$

RANGES OF h :-

1) Free convection in Gases & vapours ($h = 3$ to $25\ W/m^2K$)
- Forced convection in gases ($h = 25$ to $450\ W/m^2K$)
- Free convection is liquids ($h = 250$ to $650\ W/m^2K$)
- Forced convection in liquids ($h = 600$ to $4000\ W/m^2K$)

Water takes away heat 25 times more rapidly than with air

Reasons:- $\rho_w \approx 1000\, \rho_{air}$

$C_{pw} \approx 4.18\, C_{p\,air}$

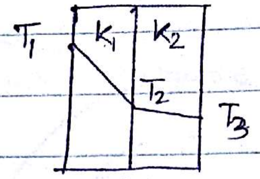
$K_{water} > K_{air}$

$\mu_{water} > \mu_{air}$

- Condensation heat transfer : $h = 3000$ to $25{,}000\ W/m^2K$
- Boiling heat transfer (Liq. to vapour) :- $h = 5000$ to $5000\ W/m^2K$

FOURIER LAW OF CONDUCTION

$$q_x = -KA\left(\frac{dT}{dx}\right) \text{ watt}$$



$K_2 > K_1$

CONVECTION THERMAL RESISTANCE

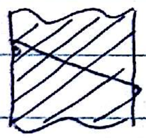
$$R_{TH} = \frac{\Delta T}{q}$$

$$q_{conv.} = hA\Delta T \Rightarrow (R_{TH})_{conv.} = \frac{\Delta T}{q} = \frac{1}{hA}\left(\frac{K}{watt}\right)$$

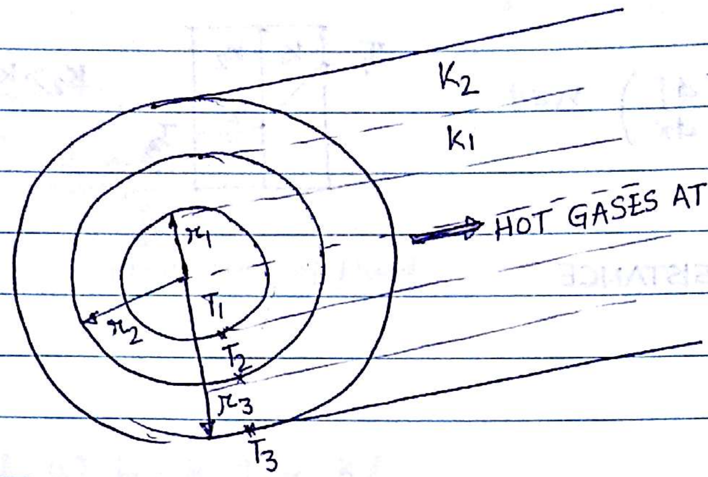
OVERALL HEAT TRANSFER COEFFICIENT (U)

It is a parameter which takes into account all the modes of heat transfer into single entity and hence is defined from equation

$$q = UA\Delta T$$
$$= UA(T_G - T_\infty)$$

GEOMETRICAL FIG.	HEAT TRANSFER (q)	THERMAL RESISTANCE	PROFILE
 (PLANE WALL)	$\frac{KA\Delta T}{L}$	$\frac{L}{KA}$	linear
 CYLINDRICAL	$\frac{2\pi KL(T_1-T_2)}{\ln\left(\frac{r_2}{r_1}\right)}$	$\frac{\ln(r_2/r_1)}{2\pi KL}$	Logarithmic
 SPHERICAL	$\frac{4\pi k r_1 r_2 (T_1-T_2)}{(r_2-r_1)}$	$\frac{(r_2-r_1)}{4\pi k r_1 r_2}$	Rect. hyperbola

CONDUCTION CONVECTION HEAT TRANSFER THROUGH COMPOSITE CYLINDER



$$T_G \quad \frac{1}{h_1(2\pi r_1 L)} \quad T_1 \quad \frac{\ln(r_2/r_1)}{2\pi k_1 L} \quad T_2 \quad \frac{\ln(r_3/r_1)}{2\pi k_2 L} \quad T_3 \quad \frac{1}{h_2(2\pi r_3 L)} \quad T_\infty$$

FOR HEAT EXCHANGERS:-

$$\frac{1}{U_{\text{Without fouling}}} = \frac{1}{h_1} + \frac{1}{h_2}$$

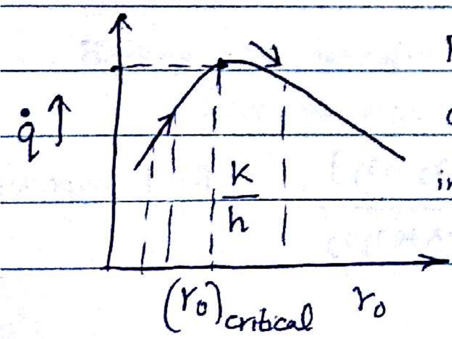
h_1 on hot side

h_2 on cold side

CRITICAL RADIUS OF INSULATION

Cylinder; $r_o = \frac{k}{h}$ Thickness $= r_o - r = \left(\frac{k}{h} - r\right)$

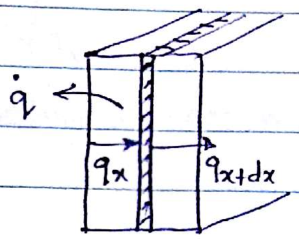
Sphere ; $r_o = \frac{2k}{h}$



For sufficiently thin wires whose radius is lesser than critical radius of insulation, any insulation wrapped around it shall result in increase in H.T. In case if radius of the wire already more than critical radius of insulation, any insulation wrapped around it shall directly decrease the H.T rate.

Cooling of cement for few days $\rightarrow$ Turing process

GENERALISED (3D, Steady or unsteady, with or without heat generation) CONDUCTION EQ^N.



Energy balance

heat conducted into the element + heat generated within element = Heat conduction out of element + Rate of change of I·E of element

i.e $\quad q_x + \dot{q}(A\,dx) = q_{x+dx} + \frac{\partial}{\partial \tau}(mC_p dT)$

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

Thermal diffusitivity $(\alpha) = \dfrac{k}{\rho C_p}$

α of a medium signifies how rapidly the heat energy can diffuse or pass through the medium

$$\alpha_{gases} > \alpha_{LIQUID}$$

Prandtl number of fluid $= Pr = \dfrac{\nu}{\alpha}$

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

If steady state $\dfrac{dT}{d\tau} = 0$

if no heat generation, $\dot{q} = 0$

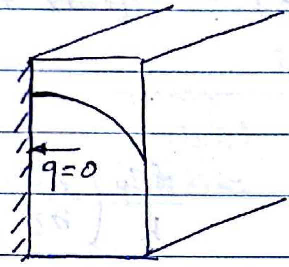
$$\frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2} + \frac{\partial^2 T}{\partial z^2} = 0 \Rightarrow \nabla^2 T = 0 \text{ (Laplace equation)}$$

CONDUCTION WITH HEAT GENERATION IN A SLAB :-

Temperature distribution within the slab is $T_0 - T = \frac{\dot{q}x^2}{2K}$; parabolic temp. distribution

$$\frac{T_0 - T}{T_0 - T_W} = \left(\frac{x}{L}\right)^2$$

- In case when one side of slab is perfectly insulated



$$-KA\frac{dT}{dx} = 0$$

$\Rightarrow \frac{dT}{dx} = 0$ at insulated surface.

T is max. at insulated surface.

Generalised conduction equation in cylindrical coordinate system is :-

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

Spherical Coordinate :-

$$\frac{1}{r^2}\frac{d}{dr}\left(r^2\frac{dT}{dr}\right) + \frac{q_G}{K} = \frac{1}{\alpha}\left(\frac{dT}{d\tau}\right)$$

$\nabla^2 T + \frac{q_G}{K} = 0$; POISSON'S EQ^N

$\nabla^2 T = 0$; LAPLACE EQ^N.

$\nabla^2 T = \frac{1}{\alpha}\frac{dT}{dt}$; DIFFUSION EQ^N

Heat generation in cylindrical surface (body):

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

$\downarrow T_{max}.$

Melting of wires will begin at the axis of wire.

FINS

Generalised solution of differential equation of fins

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

$$m = \sqrt{\frac{hP}{KA}}$$

CASE-I : FIN IS INFINITELY LONG

$$\frac{T - T_\infty}{T_0 - T_\infty} = e^{-mx} \quad \text{and} \quad Q = \sqrt{hPKA}\,(T_0 - T_\infty)$$

$$= M\theta_0 \quad \text{where} \quad M = \sqrt{hPKA}$$

CASE-II FIN IS FINITE IN LENGTH AND TIP IS INSULATED

$$\frac{\theta}{\theta_0} = \frac{T - T_\infty}{T_0 - T_\infty} = \frac{\cosh m(L-x)}{\cosh mL} \quad ; \quad Q = \sqrt{hPKA}\,(T_0 - T_\infty)\tanh ml$$

$$= M\theta_0 \tanh mL$$

If in any problem no case is mentioned, by default assume adiabatic tip.

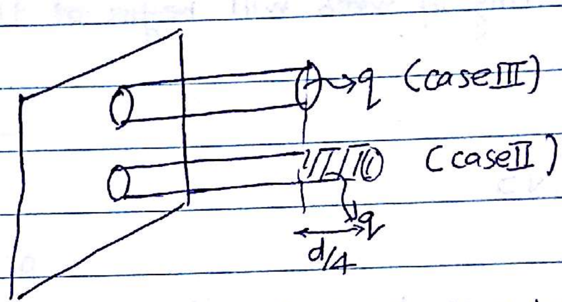
Case III : FINS FINITE IN LENGTH AND ALSO LOSES HEAT BY CONVECTION FROM ITS TIP (UNINSULATED TIP)

$$\frac{T-T_\infty}{T_0-T_\infty} = \frac{\theta}{\theta_0} = \frac{\cosh m(L_c-x)}{\cosh mL_c} \quad ; \quad Q = \sqrt{hPKA}\,(T_0-T_\infty)\tanh mL_c$$
$$= M\theta_0 \tanh mL_c$$

L_c = corrected length

$= L + \frac{t}{2}$ (for Rectangular fin)

$= L + \frac{d}{4}$ (for Pin fin)



q loss in case III is compensated by increasing length of rod and treating equivalent to case II.

$\tanh mL \geq 0.99$; condition for fin to be infinite length.

FIN REQUIREMENTS :
- Thin & closely spaced
- Large perimeter
- Good k value
- Moderatly short

FIN EFFICIENCY : $(\eta_{fin}) = \frac{q_{ACTUAL}}{q_{MAX.\ POSSIBLE}}$

$$\eta_{fin} = \frac{\tanh mL}{mL} \text{ or } \frac{\tanh mL_c}{mL_c}$$

(case II) (case III)

Fin effectiveness is defined as ratio b/w H.T with fin and H.T rate without fin.

$$\epsilon_{fin} = \sqrt{\frac{kp}{hA}}\tanh mL \quad ; \quad \epsilon_{fin} \propto \frac{1}{\sqrt{h}} \quad ; \quad \epsilon_{fin} \propto \sqrt{\frac{P}{A}}$$

- Alumunium is used as fin material not copper as it is less corrosive.

UNSTEADY STATE OR TRANSIENT HEAT CONDUCTION :-

$$\frac{T - T_\infty}{T_i - T_\infty} = e^{-\frac{hA}{\rho c V}t} \quad ; \quad \frac{\theta}{\theta_i} = e^{-Bi \cdot Fo}$$

$Bi = \frac{hL}{k}$ and $Fo = \frac{\alpha t}{L^2}$

Biot no: = $\frac{\text{Internal conductive resistance}}{\text{External convective resistance}}$

For sphere, $L = \frac{4/3\,\pi r^3}{4\pi r^2} = \frac{r}{3}$ $\left[\frac{\text{Volume}}{\text{Area}}\right]$

For cylinder, $L = \frac{\pi r^2 l}{2\pi r l} = \frac{r}{2}$

For cuble of side l, $L = \frac{l^3}{6l^2} = \frac{l}{6}$

$$Q = hA\theta_i \exp(-Bi \cdot Fo) \text{ kW}$$

$$\Delta U = \rho c V \theta_i (1 - e^{-Bi \cdot Fo}) \text{ kJ}$$

$\frac{\rho c V}{hA}$ is called time constant of system

small value preferable.

At the end of time constant period, ΔT b/w body and ambient is 37% of initial temperature difference.

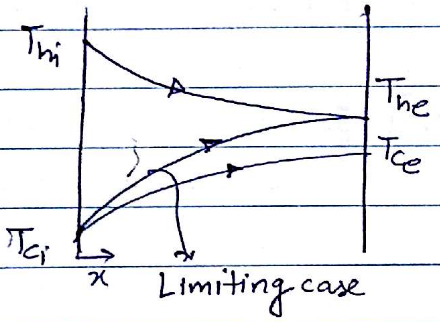
CRITERIA FOR LUMPED HEAT CAPACITY ANALYSIS ; $Bi < 0.1$

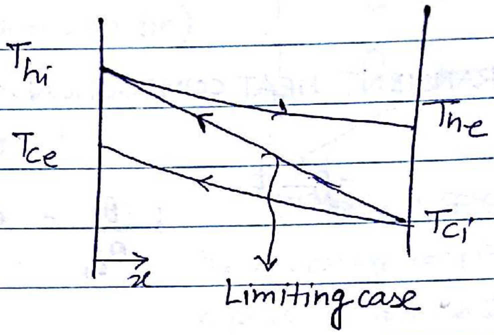
HEAT EXCHANGER

↓

open system + Adiabatic + constant pressure.

- Parallel flow more irreversible than counter flow





x direction is the direction of hot fluid flow

- Mean Temperature difference (MTD) :- takes into account the variation of ΔT w.r.t x by averaging it from inlet to exit of H.E

$$Q = UA(\Delta T)_m$$

Q = Total heat transfer rate between hot fluid and cold fluid (in entire H.E)

U = Overall H.T coefficient

ΔT_m = Total area of H.E

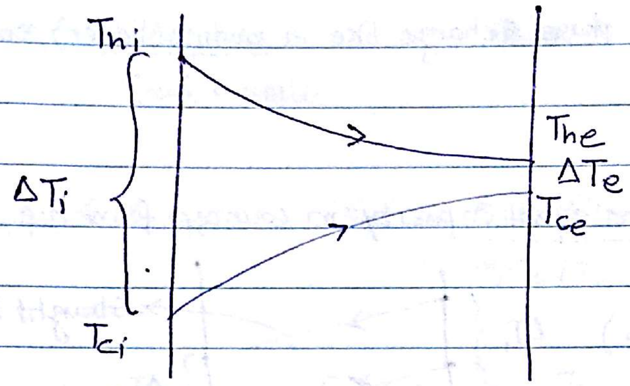
ΔT_m = M.T.D

$$\Delta T_m = \frac{1}{A}\int_{\text{INLET}}^{\text{EXIT}} \Delta T \, dA$$

→ helps in designing H.E for given inlet exit condition and Q

↓ can find requred area.

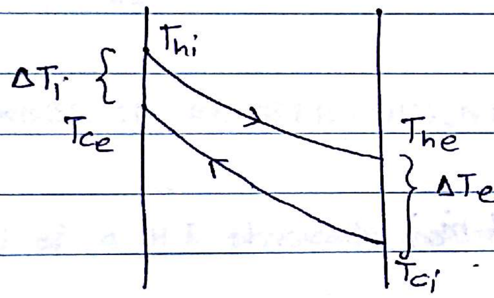
PARALLEL FLOW H·E



$$Q = UA\left[\frac{\Delta T_i - \Delta T_e}{\ln\left(\frac{\Delta T_i}{\Delta T_e}\right)}\right]$$

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

COUNTER FLOW



$$(\Delta T_m)_{C \cdot F} = \frac{\Delta T_i - \Delta T_e}{\ln\left(\frac{\Delta T_i}{\Delta T_e}\right)}$$

- For same inlet and exit temperature of both hot & cold fluid used in parallel flow and counter flow H·E the value of L·M·T·D for counter flow is greater than that of parallel flow.

(ΔT_m) or LMTD OF CROSS-FLOW HE $= (\Delta T_m)_{counterflow} \times F$

where F = correction factor < 1
(from data book)

- When all 4 temp. are same (in all types)

$$(\Delta T_m)_{COUNTER} > (\Delta T_m)_{CF} > (\Delta T_m)_{PF}$$

SPECIAL CASES

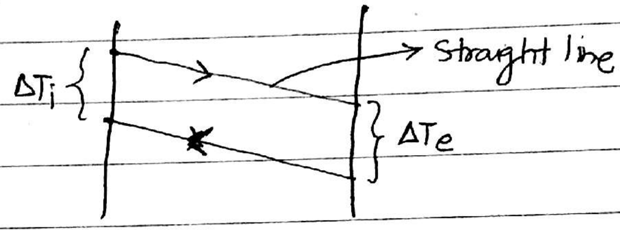
CASE I :- When one of fluid is undergoing phase change like in boiling (boiler), condensor

Then, $(\Delta T_m)_{PF} = (\Delta T_m)_{CF}$

Case II :- When both hot and cold fluids have equal capacity in counter flow H.E

Then, $\Delta T_i = \Delta T_e$

$(LMTD)_{C.F}$ = either (ΔT_i) or (ΔT_e)



$$\frac{dT_h}{dx} = \frac{dT_c}{dx} = \text{constant.}$$

DESIGN OF HEAT EXCHANGER

LMTD METHOD

Given data :-

1) Both the mass flow rate of hot & cold fluid ($\dot{m}_h$ & $\dot{m}_c$)
2) Both the specific heat of fluids (C_{ph}, C_{pc})
3) U or $\frac{1}{U} = \frac{1}{h_1} + \frac{1}{h_2}$
4) only 3 temp. among 4

STEP 1 :- Calculate 4th unknown temperature $[\dot{m}_h C_{ph}(T_{hi} - T_{he}) = \dot{m}_c C_{pc}(T_{ce} - T_{ci})]$

STEP 2 :- Find LMTD

STEP 3 :- Find Q $\dot{m}_h C_{ph}(T_{hi} - T_{he})$ or other way

STEP 4 :- $Q = UA\Delta T_m$; find Area.

Area $= (\pi d L) \times n \times p$; where n = no. of tubes per pass

$\dot{m}_{c.w}$ = mass flow rate of C.W $= \rho \times \frac{\pi}{4} \times d^2 \times V \times n$ (kg/s) [Don't multiply no. of passes].

Effectiveness of Heat exchanger :-

$$\epsilon = \frac{q_{ACTUAL}}{q_{MAX.\ POSSIBLE}} \qquad q_{max} = \dot{m}c_p \times (T_{hi} - T_{ci})$$

If $\dot{m}_h c_{ph} < \dot{m}_c c_{pc}$; $\epsilon_{HE} = \frac{\dot{m}_h c_{ph}(T_{hi} - T_{he})}{\dot{m}_h c_{ph}(T_{hi} - T_{ci})} = \frac{T_{hi} - T_{he}}{T_{hi} - T_{ci}}$

If $\dot{m}_c c_{pc} < \dot{m}_h c_{ph}$; $\epsilon_{HE} = \frac{T_{ce} - T_{ci}}{T_{hi} - T_{ci}}$

only one of the formula can be used at a time

Usually $\epsilon_{HE} = 0.5$ to 0.8

- NUMBER OF TRANSFER UNITS (NTU) $= \frac{UA}{(\dot{m}c_p)_{small}}$

NTU of a H.E physically indicates overall size of H.E

$$\frac{1}{U_{with\ fouling}} = \frac{1}{h_1} + F_1 + \frac{1}{h_2} + F_2 \quad ; \ NTU \ngtr 18$$

CAPACITY RATIO (C) : $= \frac{(\dot{m}c_p)_{small}}{(\dot{m}c_p)_{big}} \Rightarrow 0 \le C \le 1$

C=0 when one of the fluids is undergoing phase change.

For any Heat exchanger, $\epsilon = f(NTU, C)$

For parallel flow H.E

$$\epsilon = \frac{1 - e^{-(1+C)NTU}}{1+C}$$

For counter-flow

$$\epsilon = \frac{1 - e^{-(1-C)NTU}}{1 - Ce^{-(1-C)NTU}}$$

SPECIAL CASE REGARDING EFFECTIVENESS

Case I :- When one of the fluids is undergoing phase change ($C=0$)

$\epsilon_{\parallel el} = \epsilon_{counter} = 1-e^{-NTU}$; LMTD same

Area same

NTU same

Case II :- When both hot and cold fluids have equal capacity rates ($\dot{m}_h C_{ph} = \dot{m}_c C_{pc}$) [$C=1$]

$$\epsilon_{parallel} = \frac{1-e^{-2NTU}}{2}$$

$$\epsilon_{COUNTER} = \frac{NTU}{1+NTU}$$

NTU METHOD :-

The main purpose of this method is to obtain both exit temperature of hot & cold fluids for a given H·E area A.

Given, Data — To find $T_{he} = ?$ and $T_{ce} = ?$

- Both the mass flow rate ($\dot{m}_h$ & $\dot{m}_c$)
- Both the specific heats of fluid (C_{ph} or C_{pc})
- U
- Only two inlet temperature (T_{hi} and T_{ci})
- Area of HE (A)

SOL^N

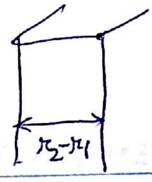
STEP 1 :- Calculate capacity ratio

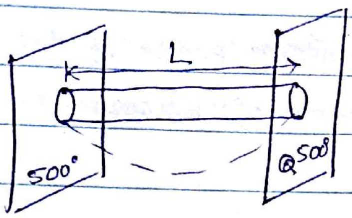
STEP 2 :- Calculate $NTU = \frac{UA}{(\dot{m}C_p)_{small}}$

STEP 3 :- OBTAIN $\epsilon = f(NTU, C)$

STEP 4 :- Select appropriate formula for ϵ ($\dot{m}_h C_h \lesseqgtr \dot{m}_c C_{pc}$)

Thermally equivalent means
1. Total thickness must be same.
2. R_{TH} must be same





?

$$\frac{r_2 - r_1}{4\pi r_1 r_2 k} = \frac{r_2 - r_1}{A_m k} \Rightarrow A_m = 4\pi r_1 r_2$$

At $\left(\frac{L}{3}, \frac{L}{4}\right)$, Q non zero.

If Biot number > 1 ; Lumped heat analysis not valid ; HIESLER CHART USED.

- TOTAL HEMISPHERICAL EMMISIVE POWER (E)

It is defined as the radiation energy emitted from surface of a body (by virtue of its absolute temperature) per unit time per unit area in all possible hemispherical direction integrated over all wavelength.

- TOTAL EMMISIVITY (ϵ)

Emissivity for a surface is defined as the ratio between total hemispherical emissive power of a non-black body and total hemispherical power of a black surface, both being at same temperature. $\epsilon = \frac{E}{E_b}$

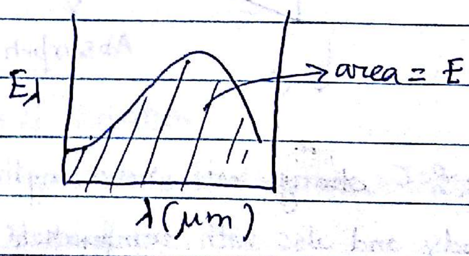
BLACKBODY → IDEAL EMITTER
BLACKBODY → PERFECT ABSORBER
BLACKBODY → DIFFUSIVE

- A tiny hole in furnace wall is a example of black body.
- Sun is treated a blackbody

- MONOCHROMATIC OR SPECTRAL EMMISIVE POWER

E_λ at particular wavelength λ is defined as the quantity which when multiplied by $d\lambda$ shall give radiation energy emitted from surface of body per unit time per unit area

E = Total emmisive power = $\int_0^\infty E_\lambda d\lambda$ W/m^2



At same temperature E_λ is $f(\lambda)$

• MONOCHROMATIC EMISSIVITY OR SPECTARAL EMISSIVITY :-

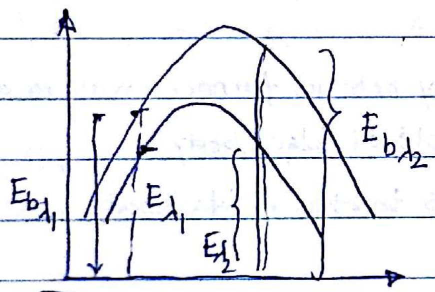
It is defined as the ratio between monochromatic hemispherical emissive power of a non-blackbody and monochromatic hemispherical emissive power of black body being at same temperature and wavelength.

$$\epsilon_\lambda = \frac{E_\lambda}{E_{b\lambda}}$$

$$E_\lambda = \epsilon_\lambda E_{b\lambda}$$

$$\epsilon = \frac{E}{E_b} = \frac{\int_0^\infty E_\lambda \, d\lambda}{\int_0^\infty E_{b\lambda} \, d\lambda} = \frac{\int_0^\infty \epsilon_\lambda E_{b\lambda} \, d\lambda}{\int_0^\infty E_{b\lambda} \, d\lambda}$$

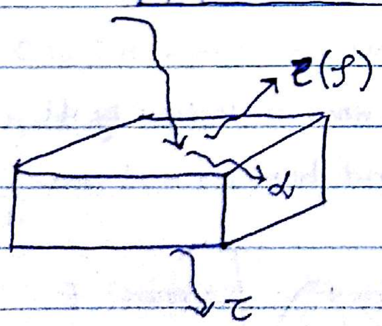
If $E_\lambda \neq f(\lambda)$ i.e for gray body i.e $\epsilon = \epsilon_\lambda \dfrac{\int_0^\infty E_{b\lambda} \, d\lambda}{\int_0^\infty E_{b\lambda} \, d\lambda} \Rightarrow \boxed{\epsilon = \epsilon_\lambda}$



For gray body,

$$\frac{E_{\lambda_1}}{E_{b\lambda_1}} = \frac{E_{\lambda_2}}{E_{b\lambda_2}}$$

* The ratio between the area under bottom curve on x-axis and area under top curve on axis is equal to total emissivity of graybody.



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

Absorpbitivity Reflectivity Transmissivity

α, ρ, τ change with wavelength of incident thermal radiation, surface roughness of the body and also with temperature.

LAWS OF THERMAL RADIATION :-

• KIRCHOFF'S LAW OF THERMAL RADIATION :-

Whenever a body is in thermal equilibrium with its surrounding its emissivity is equal to its absorptivity

$$\boxed{\alpha = \epsilon}$$

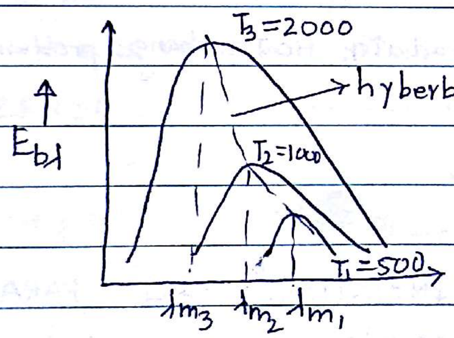
i.e a good absorber is good emitter

• PLANK'S LAW OF THERMAL RADIATION :-

The law states that the monochromatic or spectral emissive power of a blackbody is dependent on both absolute temperature of blackbody & wavelength of energy emitted from body.

$$E_{b\lambda} = f(T, \lambda)$$

$$E_{b\lambda} = \frac{2\pi C_1}{\lambda^5 \left(e^{C_2/\lambda T} - 1\right)} \quad \frac{watt}{m^2\text{-}\mu m}$$; C_1 and C_2 are experimental constant



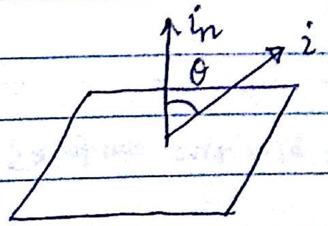
$\lambda_m T$ = constant = 2900 μm-K.

• WEIN'S DISPLACEMENT LAW

• STEFAN - BOLTZMAN'S LAW :-

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

• LAMBERT'S COSINE LAW :-



i_n = Normal intensity of Radiation

i = Intensity of radiation along a direction making an angle 'θ' w.r.t Normal

$$i = i_n \cos\theta$$

Intensity of Radiation (i) along a specified direction is defined as the radiation energy emitted from the surface of body per unit time per unit area and per unit solid angle about that direction.

$\therefore E_b = i\pi$ (for black body only)

SHAPE FACTOR OR CONFIGRATION FACTOR OR VIEW FACTOR

F_{mn} = Fraction of Radiation energy leaving surface (m) that reaches surface (n)

$$0 \leq F_{mn} \leq 1$$

- Shape factor only depends on how two surfaces are geometrically oriented w.r.t each other.

- RECIPROCITY RELATION : $A_1 F_{12} = A_2 F_{21} = \frac{1}{\pi} \int_{A_1} \int_{A_2} \frac{\cos\phi_1 \cos\phi_2 \, dA_1 \, dA_2}{r^2}$

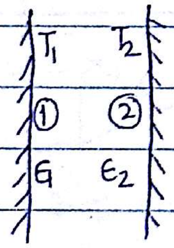
- SUMMATION RULE AMONG SHAPE FACTOR

If there are n number of surfaces involved in any radiation Heat exchange problem, then

$$F_{11} + F_{12} + F_{13} + \dots F_{1n} = 1$$

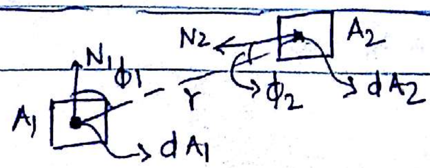
also $F_{21} + F_{22} + F_{23} + \dots F_{2n} = 1$

RADIATION HEAT EXCHANGE BETWEEN TWO INFINITELY LARGE PARALLEL PLATE SURFACES/ PLANES



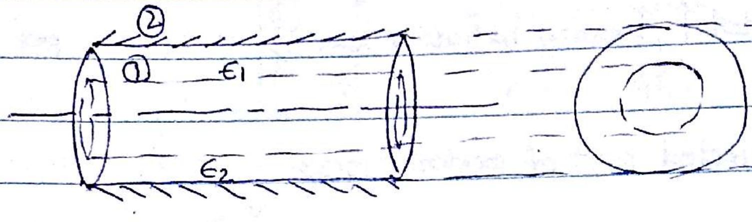
$$\left(\frac{q}{A}\right)_{1-2} = \frac{\sigma(T_1^4 - T_2^4)}{\frac{1}{\epsilon_1} + \frac{1}{\epsilon_2} - 1} \text{ Watt/m}^2$$

$dQ_{1-2} = \frac{I_1 \, dA_1 \cos\phi_1 \, dA_2 \cos\phi_2}{r^2}$; heat exchange b/w two surfaces



RADIATION HEAT EXCHANGE BETWEEN TWO INFINITELY LONG SURFACE.

(May be ecenteric)



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

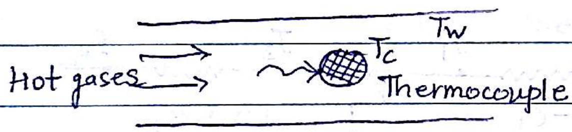
- When a small body is kept in large room/enclosure

$\frac{A_1}{A_2} \to 0$; $(q_{net})_{1-2} = \sigma(T_1^4 - T_2^4) A_1 \epsilon_1$

- For steady state condition of filament,

Electric power $= \sigma(T_{fil}^4 - T_{amb}^4)\epsilon_{fil} \times A_{fil}$

- TO GET ERROR IN TEMPERATURE MEASUREMENT (by using Thermocouple)



The rate of convection H·T b/w gas & thermocouple = Net Radiation H·E b/w T·C & Ductwall

$$\underbrace{h A_{T/C}(T_G - T_C)}_{\text{error}} = \sigma(T_C^4 - T_W^4)\epsilon_{T/C} A_{T/C} \text{ watt}$$

IRRADIATION (G) :- It is defined as total thermal radiation incident upon a surface for unit time and per unit area

RADIOSITY (J) is defined as the total thermal radiation leaving a surface per unit time and per unit area

J = Emitted + Reflected part of incident energy

EQUIVALENT RADIATION CIRCUIT

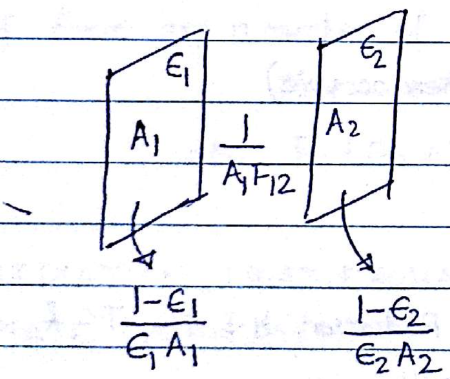
$$E_b \quad (q_{net})_{surf-surr.} \quad J$$

$$\left(\frac{1-\epsilon}{\epsilon A}\right) \rightsquigarrow \text{surface resistance.}$$

This resistance shall exist at every surface which is exchanging HEAT BY RADIATION

$$J_1 \qquad J_2$$

$$\left(\frac{1}{A_1 F_{12}}\right) \rightarrow \text{Space resistance}$$



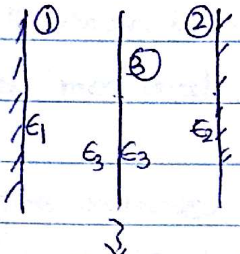
RADIATION NETWORK:-

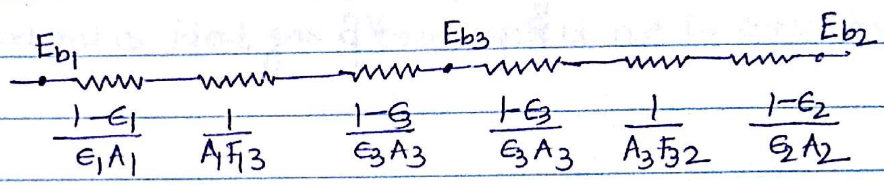
$$E_{b_1} \quad \underset{\frac{1-\epsilon_1}{\epsilon_1 A_1}}{\text{ww}} \quad J_1 \quad \underset{\frac{1}{A_1 F_{12}}}{\text{ww}} \quad J_2 \quad \underset{\frac{1-\epsilon_2}{\epsilon_2 A_2}}{\text{ww}} \quad E_{b_2}$$

No multiplication of area.

$$(q_{1-2})_{net} = \frac{(E_{b_1} - E_{b_2})}{\Sigma R_{TH}} = \frac{\sigma(T_1^4 - T_2^4)}{\frac{1-\epsilon_1}{\epsilon_1 A_1} + \frac{1}{A_1F_{12}} + \frac{1-\epsilon_2}{\epsilon_2 A_2}} \text{ watt}$$

RADIATION SHIELDS :-





Radiation shield

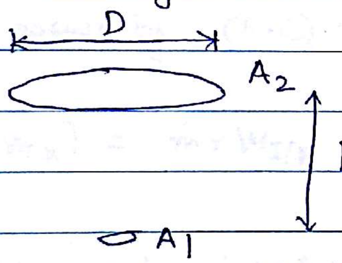
$$\left(\frac{q}{A}\right)_{\text{WITH SHIELD}} = \frac{\sigma(T_1^4 - T_2^4)}{\frac{1}{\epsilon_1} + \frac{2}{\epsilon_3} + \frac{1}{\epsilon_2} - 2}$$

IF ALL SURFACES INCLUDING SHIELD HAVE SAME EMISSIVITY (i.e $\epsilon_1 = \epsilon_2 = \epsilon_3 = \cdots$)

$$\left(\frac{q}{A}\right)_{\text{WITH SHIELD}} = \left(\frac{1}{n+1}\right) \times \left(\frac{q}{A}\right)_{\text{WITHOUT SHIELD}}$$

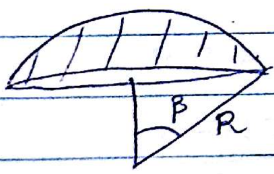
where n = no. of shields.

Net radiation exchange between ② and ① $= A_2 F_{21} \sigma (T_2^4 - T_1^4)$

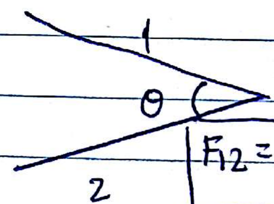


If $A_2 >> A_1$

then, $F_{12} = \frac{D^2}{D^2 + 4L^2}$



$A = 2\pi R^2 (1 - \cos\beta)$



$$\boxed{F_{12} = F_{21} = \frac{1 - \sin\theta}{2}}$$

$\Omega = \frac{dA_n}{r^2}$; dA_n = projection of surface to direction of radiation

$$\text{Total absorpitivity} = \frac{\int_0^\infty \alpha G_\lambda \, d\lambda}{\int_0^\infty G_\lambda \, d\lambda} = \frac{\int_0^\infty (1-\rho) G_\lambda \, d\lambda}{\int_0^\infty G_\lambda \, d\lambda}$$