



Madarju.com



Heat Transfer

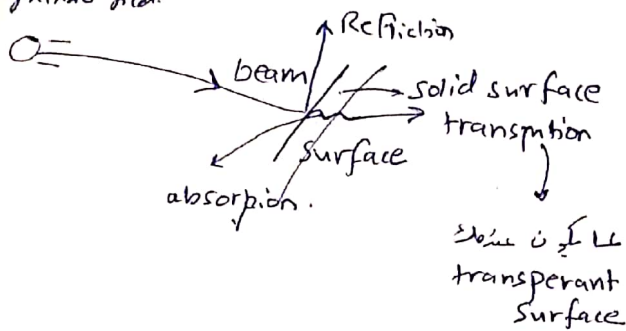
Dr Yahia Khreisha



Heat

Radiation:-

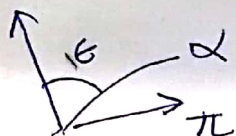
magnitude of



glass sheet: 3-
absorption
transmission

α = fraction absorptivity.

$$\epsilon + \alpha + \tau = 1.0$$



* Surface properties

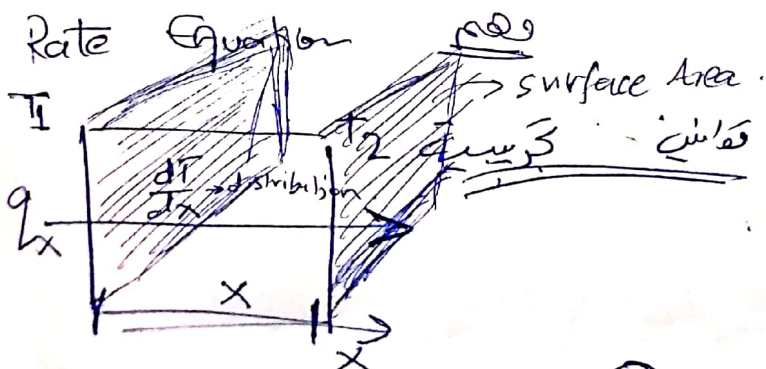
Color $\Rightarrow \epsilon, \alpha, \tau \rightarrow$ black body = max. as Ref

roughness.

Radiation. = ...

desu ad k-wal 2. ...

Rate Equation



①

$\frac{dT}{dx}$ heat gradient

$$q_x = -k A \frac{dT}{dx}$$

negativ
jeel 5.11

S.S $Q_x = \text{constant}$

$$q_x = -k A \frac{dT}{dx}$$

constant if st-st $k = k_w/m \cdot k$

valid

$q'' = \text{flux}$

= heat flow per unit area

$$= \frac{\text{heat flow}}{\text{unit Area}} = \frac{kJ/h}{m^2}$$

$$= \frac{kW/s}{m^2}$$

$$Q \propto A \Delta T$$

$$Q = h A \Delta T$$

driving force
($T_s - T_\infty$)

①

Problem!!

K, h = experimentally

analytically = simple shape

h mean for boundary layer

②

heat

1/1

سید

Measuring convection heat transfer coefficient st st
 $T = 15^\circ\text{C}$

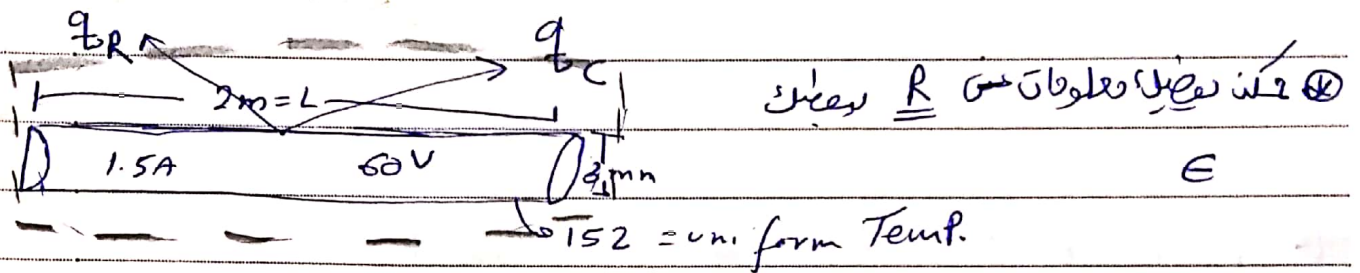
1.5A
 $0 \rightarrow 0 \leftarrow 152^\circ\text{C}$

$L = 2\text{m}$

60V

$D = 3\text{mm}$

Determine the con. heat transfer coefficient (C.H.T)?



$$Q = h A \Delta T$$

Assumption

- ① No Radiation
- ② st. st condition just convection
- ③ uniform Temp.
- ④ properties constant

$$A_1 = \pi D L \rightarrow$$

Area = $\frac{\pi D^2}{2} \times 2$
 نصف دایره

⑤ Assume A_2 very small

نکته ۱ و ۲ در این مسئله

⑥ E.B.:

$$\dot{E}_{IN} - \dot{E}_{OUT} + \dot{E}_g = 0$$

$\downarrow$ $\downarrow$ $\downarrow$
 $\dot{E}_{IN}$ $\dot{E}_{OUT}$ $\dot{E}_{current}$

$$\dot{E}_{out} = \frac{q}{C} = E_g = \underset{?}{h} A \Delta T$$

$$P - E_g = \underline{V} \cdot \underline{A} \rightarrow$$

$$\underline{q} = \boxed{h} \underline{A} \underline{\Delta T} \quad \text{to find } \dots$$

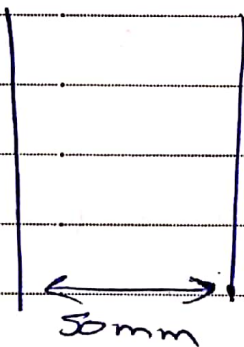
⇒ sphere & cylinder → find $G \cdot A$

⇒ surface area for sphere $4\pi r^2$

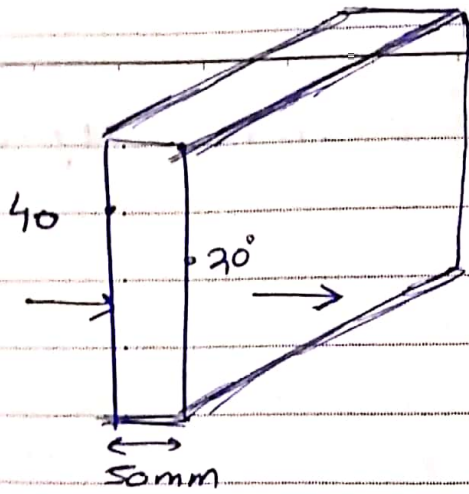
⇒ surface area for cylinder πDL
 $\pi 2rL$

Thermal Conductivity

Slab } wallpaper = thin layer
 wall



The heat flux through a wood slab (50mm) thick, whose inner & outer surface Temp are (40 & 20°C) respectively, has been determined to be (40 W/m²) what is the thermal conductivity



$$q = k A \frac{\Delta T}{\Delta x}$$

Assumption

- ① St. St
- ② ONE Direction
- ③ uniform temp.

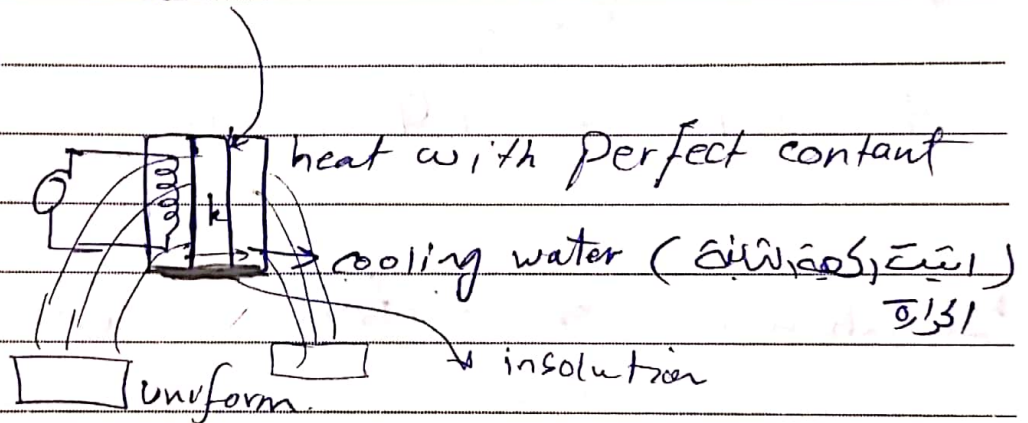
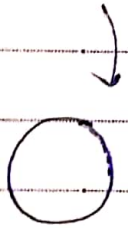
$$q'' = \frac{q}{A} = (k) \frac{\Delta T}{\Delta x}$$

⇒ experimentally:

① $D = 5\text{cm}$
thickness = 3mm

② k

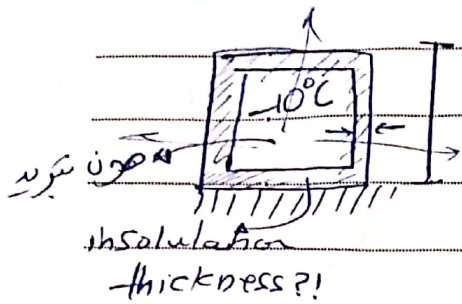
how to heat !!



* Thickness of insulation :-

A freezer compartment consist of a cubic cavity that is (2m) ^{dimension} a side. Assume the bottom to be perfectly insulated. What is the min. thickness of styrofoam insulation ($k = 0.03 \text{ W/m.k}$) that must be applied to the top & side walls to ensure a heat load of

less than 500 w, then the inner & outer surfaces are -10 and 35°C !



35°C

Assumption

- ① One Direction
- ② st-st
- ③ constant Properties

Cubic cavity = $5, 1, 7$

④ bottom insulated = $5, 1, 5, 6$

$$\frac{q}{L} = \frac{k}{L} \frac{\Delta T}{\Delta x} ??$$

$$A = 5 \times 2^2 = 8 \times 4 = 20 \text{ m}^2$$

Radiation

فقد على

Q: A spherical interplanetary probe, of (0.5m) diameter contains electronics that dissipate (150W). If the probe surface has an emissivity of 0.8 and the probe does receive radiation from other surfaces, what is its surface Temp?

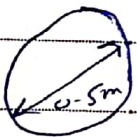
Rad = shoo

Assume {Vacuum = Rad}

→ $q = \text{dissipate} = 150\text{w}$

→ $q = \text{of side}$

→ Receiving energy from outside = ignore



$$q = \epsilon_0 \epsilon A \frac{V}{s}$$

$\downarrow$
 sphere
 Area
 $\{ \pi D^2 \}$

Ex 100

$$\text{sphere} = \text{S.A} = \pi D^2$$

$$\text{sphere} = \text{volum} = \frac{1}{6} \pi D^3$$



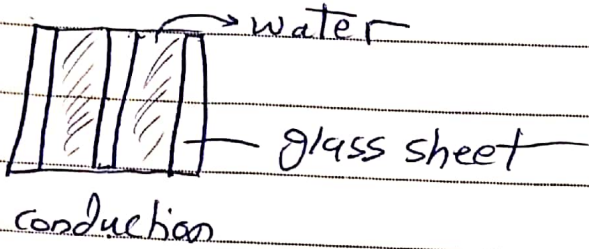
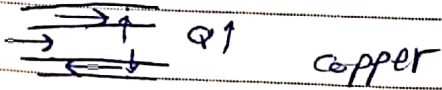
4.1.16

heat
w/21

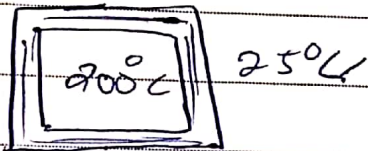
Thermal conductivity "k"

$$k = f(T)$$

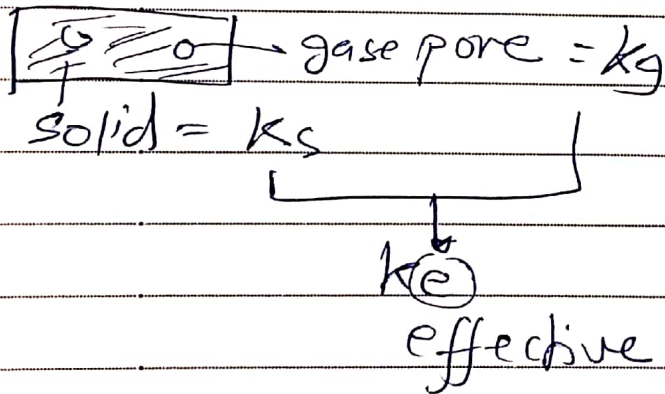
k = insulation Tojers



Insulation = $k \downarrow \downarrow$
to reduce the heat loss



flake $\rightarrow \uparrow$ radiation = reflective



$$\alpha = k / \rho c_p \rightarrow \text{Thermal diffusivity}$$

as a definition

$$\text{stored} = \rho c_p$$

$$\alpha = 20 \quad \alpha = 10$$

ability to
conduct
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stored energy $\propto \sqrt{t}$

(Figure 2.4)
 $k_{\text{metal}} \uparrow$

$k_{\text{gas}} \downarrow$

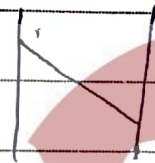
Heat diffusion eq.

① * solid }
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* energy balance eq.

② differential eq.
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③ $T = f(x)$
 $T = f(y)$ } distribution through solid medium



Fourier law $\rightarrow$ heat flow

ask ; believe & recieve

Heat transfer = Vector

is $\ominus$ to x-direction

convention sign

$\phi =$

Iso thermal = constant T

C.V = $x \rightarrow dx$

$y \rightarrow dy$

$z \rightarrow dz$

(S.A)

E generation

E stored

Energy balance eq

E_g = per unit volume

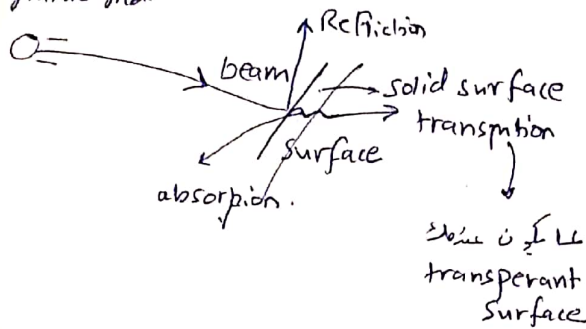
செலி சேமிக்கப்படும்



Heat

Radiation:-

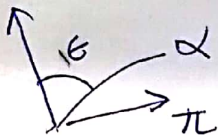
magnitude field.



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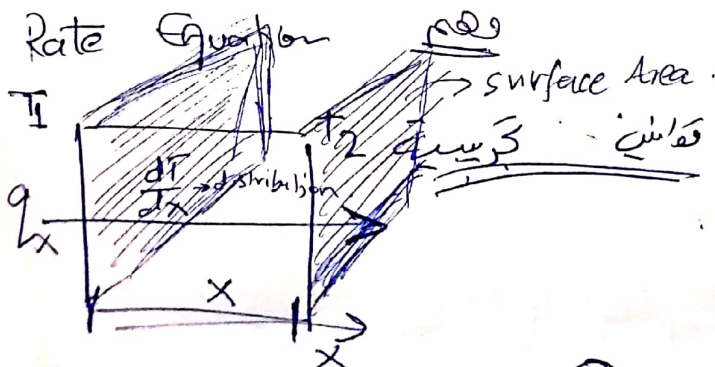
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desu as ...



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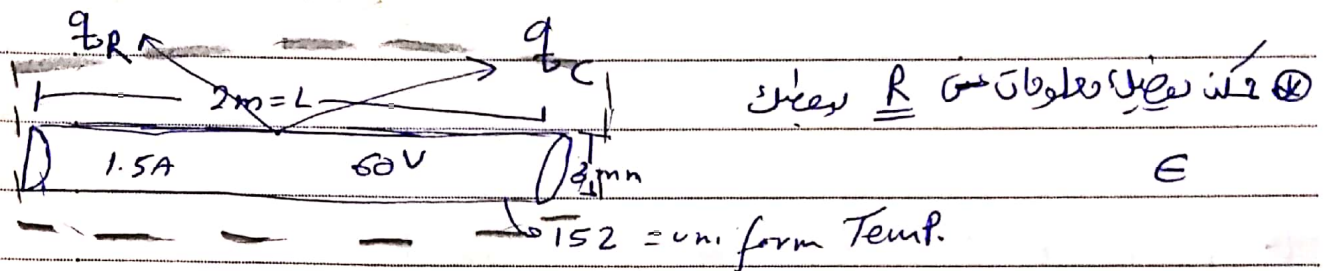
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$$A_1 = \pi D L \rightarrow$$

$$\text{Area} = \frac{\pi D^2}{4} \times 2$$

نقطه ۱ و ۲

⑤ Assume A_2 very small

در این حالت A بسیار کوچک است

and
 ⑥ E.B.:

$$\dot{E}_{IN} - \dot{E}_{OUT} + \dot{E}_g = 0$$

$\checkmark$ $\checkmark$ $\checkmark$
 current

$$\dot{E}_{out} = \frac{q}{c} = E_g = \underbrace{h}_{?} A \Delta T$$

$$P = E_g = \underline{V} \cdot A \rightarrow$$

$$q = \boxed{h} \underbrace{A} \underbrace{\Delta T}$$

to find ...

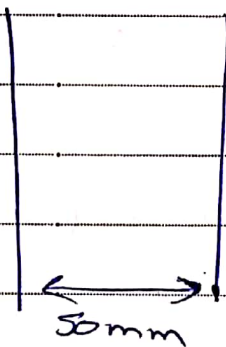
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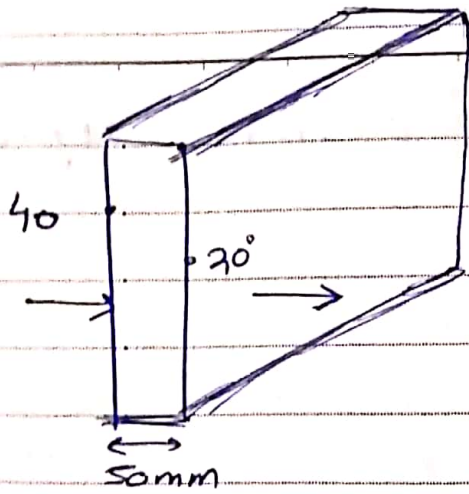
⇒ surface area for cylinder $\pi D L$
 $\pi 2r L$

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Slab } ~~sublayer~~ = thin layer
 wall



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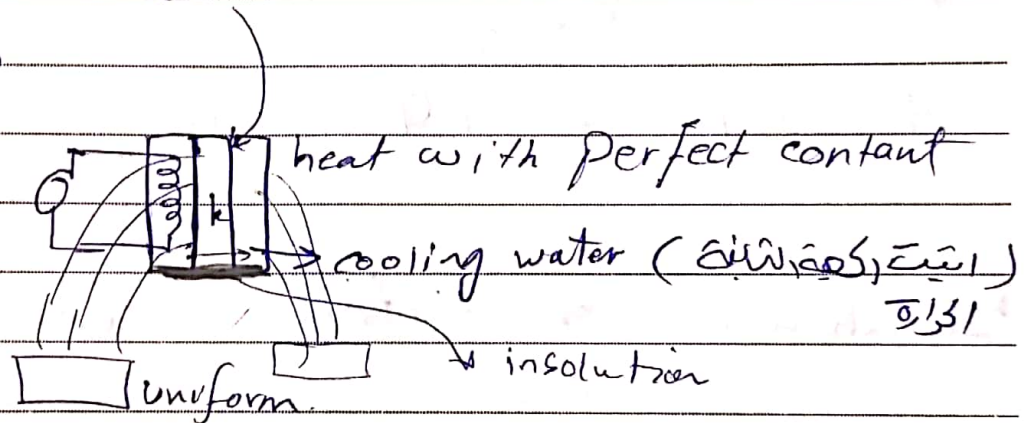
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⇒ experimentally:

① $k ??$
 $D = 50\text{mm}$
 thickness = 3mm



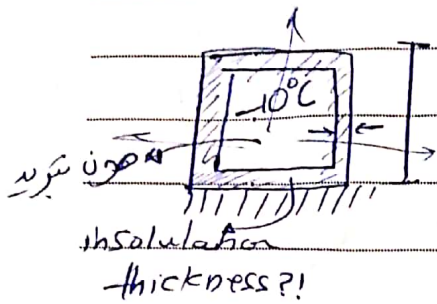
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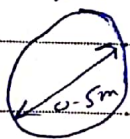
$$\frac{q}{L} = \frac{k A \Delta T}{L (\Delta x) ??}$$

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Radiation في حال على

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$q_{\text{rad}} = \text{shab}$



Assume $\{V_{\text{vacum}} = \text{Rad}\}$

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$\downarrow$
 sphere
 Area
 $\left\{ \frac{4}{3} \pi D^2 \right\}$

Ex 100

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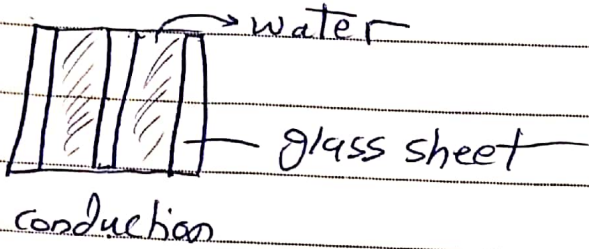
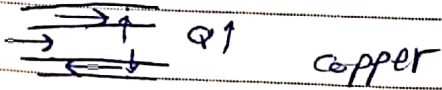
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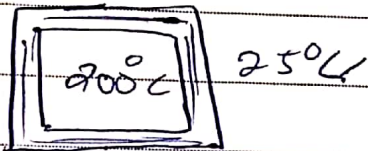
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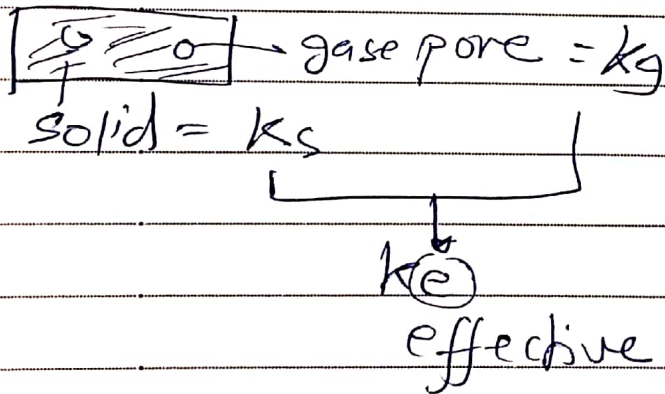
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(S.A)

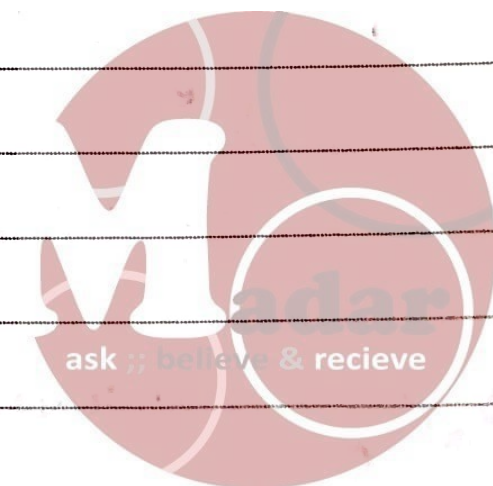
E_{generation}

E_{stored}

Energy balance eq

E_g = per unit volume

செலி சேலி செலி செலி



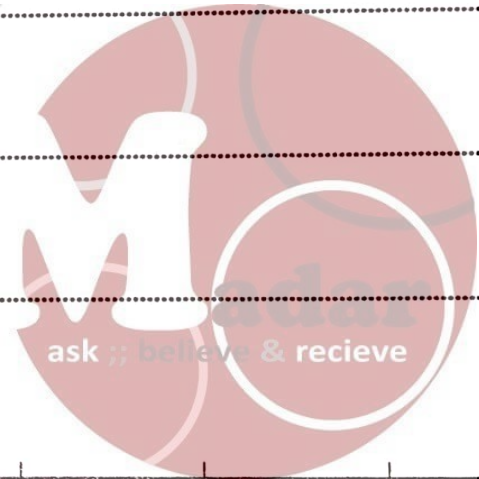
Fourier's law

∇T multidimension

$$q_u = -k \nabla T$$

$$= -k \frac{\partial T}{\partial x} \vec{i} - k \frac{\partial T}{\partial y} \vec{j}$$

$$-k \frac{\partial T}{\partial z} \vec{k}$$



6th Oct. 201

$$\nabla = \left(\frac{\partial}{\partial x} + \frac{\partial}{\partial y} + \frac{\partial}{\partial z} \right)$$

$$q_g \rightarrow \left(\frac{\text{kw}}{\text{m}^2} \right)$$

$$\dot{E} = q \times \text{volume}$$

① Energy balance EB

⇒ G. D. F

(General differential form)

⇒ B.C^s

(Boundary equation)

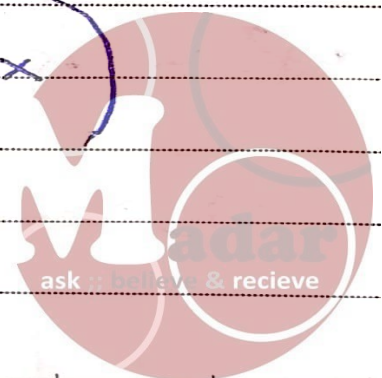
$$\textcircled{m} \rho c_p \frac{\partial T}{\partial t}$$

$$\rho \cdot V \cdot c_p \frac{\partial T}{\partial t}$$

$$\int \rho c_p \frac{\partial T}{\partial t} dx dy dz$$

$$q_x dx \rightarrow \left(q_x + \frac{\partial q_x}{\partial x} dx \right)$$

Taylor series



6th Oct - 2019

Fourier's law

$$q_x = -k A \frac{\partial T}{\partial x} \rightarrow \text{size} \sim \text{width}$$

$dz dy = \text{Area}$

$$\rightarrow \frac{\partial q_x}{\partial x} dx = \frac{\partial}{\partial x} \left(\int k dy dz \frac{\partial T}{\partial x} \right) dx \quad \text{volume}$$

$$= \frac{\partial}{\partial x} \left(k \frac{\partial T}{\partial x} \right) \rightarrow \text{Energy in}$$

$$\left[\rho c_p \frac{\partial T}{\partial t} \right] = \text{accumulation term of stored energy}$$

Ⓢt. st $\rightarrow \rho c_p \frac{\partial T}{\partial t} = 0$ but transit; keep it
 $\rho c_p \frac{\partial T}{\partial t} (z, y) \text{ of } (x) \text{ of } (x) \text{ of } (x) \rightarrow \text{st}$

planer Medium = Cartesian

constant Properties = $k, c_p = \text{const}$

one dimensional = just x-dimension

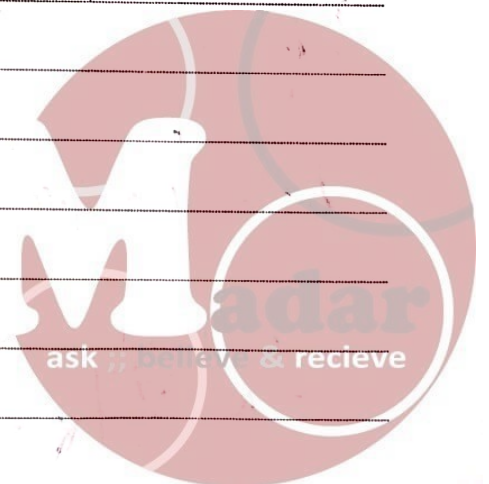
$$\frac{\partial}{\partial x} \left(k \frac{\partial T}{\partial x} \right) = \rho c_p \frac{\partial T}{\partial t} \rightarrow (\text{Transit}) (\text{un study stat})$$

[function of temp.]

const $\rightarrow \text{const}$

$$\frac{\partial}{\partial x} \left(\frac{\partial T}{\partial x} \right) = \frac{\rho c_p}{k} \frac{\partial T}{\partial t}$$

$$\frac{\partial^2 T}{\partial x^2} = \frac{1}{\alpha} \frac{\partial T}{\partial t}$$



$$\frac{\partial^2 T}{\partial x^2} = 0 \Rightarrow \text{1-D, no-g, properties constant} \Rightarrow \text{st. st}$$

Boundary condition.

$$\frac{\partial T}{\partial x} \rightarrow \text{ONE B.C.}$$

$$\frac{\partial^2 T}{\partial x^2} \rightarrow \text{two B.C.}$$

$$\text{Ex: BC @ } x=0 \quad T=100^\circ\text{C}$$

$$T(0, t) = T_s \quad \text{at any time} \quad \text{when st. st it doesn't matter}$$

$$\text{Surface balance} \Rightarrow \left(-k \frac{\partial T}{\partial x} \Big|_{x=0} = q_s'' \right)$$

$$\text{Adiabatic / insulated} \Rightarrow \frac{\partial T}{\partial x} \Big|_{x=0} = 0$$

$T(x, t)$

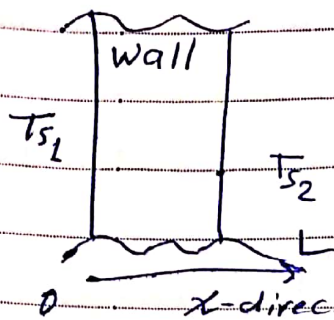
Convection

conv gas 

heat transfer of conv. = heat transfer of conf.

Fourier's law = Newton law @ $(x=0)$

$$-k \frac{\partial T}{\partial x} \Big|_{x=0} = h [T_\infty - T(0, t)]$$



no. 9 / Conc. Properties / 56.5t

Find general rate Heat diffusion eq
for the Temp distribution, then find
the heat rate eq

$$E_{in} - E_{out} + \cancel{E_g} = \cancel{E_{st}} \\ = \cancel{\int \rho c_p \frac{dT}{dt}}$$

$$\frac{\partial}{\partial x} \left(k \frac{\partial T}{\partial x} \right) + \frac{\partial}{\partial y} \left(k \frac{\partial T}{\partial y} \right) + \frac{\partial}{\partial z} \left(k \frac{\partial T}{\partial z} \right) + \cancel{\dot{q}} = \cancel{\rho c_p \frac{dT}{dt}}$$

$$\frac{\partial}{\partial x} \left(k \frac{\partial T}{\partial x} \right) = 0$$

$$\frac{\partial}{\partial x} \left(\frac{\partial T}{\partial x} \right) = 0$$

$$@ x=0 \quad T=T_{s1}$$

$$@ x=L \quad T=T_{s2}$$

↓ integration 1

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

↓ integration 2

$$[T = C_1 x + C_2] \text{ Temperature distribution}$$

$$T_{s1} = C_2$$

$$T_{s2} = C_1 L + C_2$$

$$T_{s2} = C_1 L + T_{s1}$$

$$C_1 = \frac{T_{s2} - T_{s1}}{L}$$

$$T = \frac{T_{s2} - T_{s1}}{L} x + T_{s1} \quad \#$$

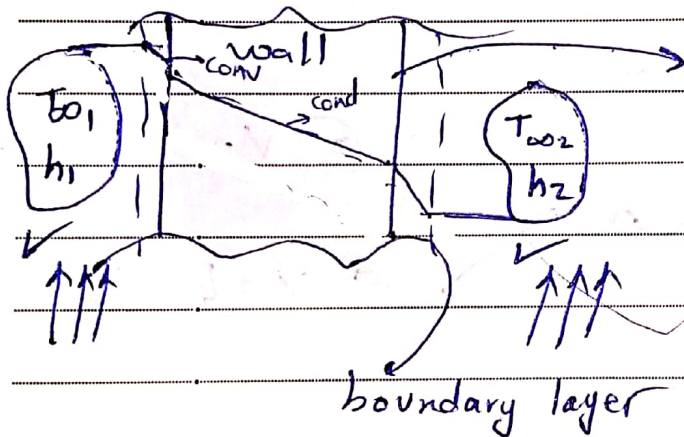
Heat Rate !!

$$q'' = -k \left[\frac{\partial T}{\partial x} \right] \rightarrow \text{heat distribution}$$

$$T = \frac{T_{s2} - T_{s1}}{L} x + T_{s1} \quad (\text{derivation})$$

$$\frac{\partial T}{\partial x} = \frac{T_{s2} - T_{s1}}{L}$$

$$q'' = -k \frac{T_{s2} - T_{s1}}{L} = \left[k \frac{T_{s1} - T_{s2}}{L} \right]$$



IN COMMON, NOT Given!!

no. gen
cons. properties

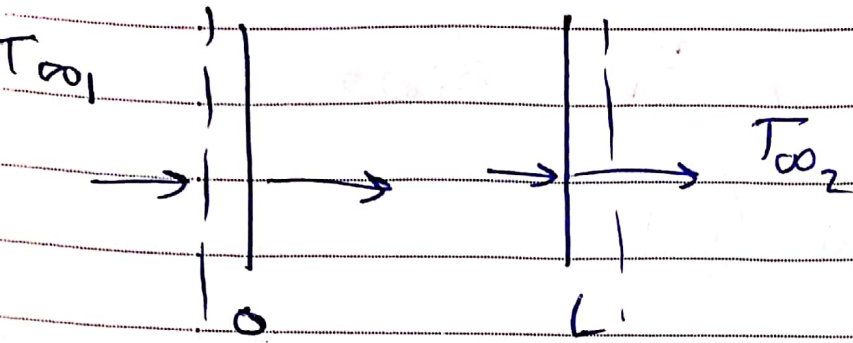
St-St

Find general rate heat diffusion eq.

The Temp. distribution & the heat Rate eq

$$\frac{\partial}{\partial x} \left(k \frac{\partial T}{\partial x} \right) + \frac{\partial}{\partial y} \left(k \frac{\partial T}{\partial y} \right) + \frac{\partial}{\partial z} \left(k \frac{\partial T}{\partial z} \right) = \rho c_p \frac{\partial T}{\partial t}$$

① boundary Condition @ surface balance



$$-k \frac{dT}{dx} \Big|_{x=0} = h(T_{\infty 1} - T(0)) \quad \text{--- ①}$$

$$-k \frac{dT}{dx} \Big|_{x=L} = h(T(L) - T_{\infty 2}) \quad \text{--- ②}$$

Back to eq $\frac{\partial}{\partial x} \left(k \frac{\partial T}{\partial x} \right) = 0$

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

eq 23

$$T = C_1 x + C_2$$

$$\left. \begin{array}{l} T(0) = 0 + C_2 \\ T(L) = C_1 L + C_2 \end{array} \right\} \rightarrow \text{back to eq ① and eq ②}$$

$$-k C_1 = h(T_{\infty 1} - C_2) \quad \text{--- ①}$$

$$-k C_1 = h(C_1 L + C_2 - T_{\infty 2}) \quad \text{--- ②}$$

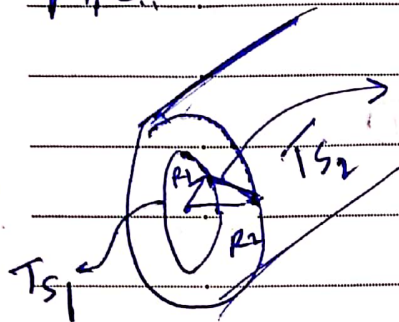
Cylinder → solid Rod/wire

hollow pipe

sphere → solid

hollow

pipe



This is distribution

boundary $r=r_1$ $T=T_1$
 $r=r_2$ $T=T_2$

one wall $q = \frac{T_1 - T_2}{L} K$ wall, cylinder, sphere

Remember:
cylindrical

$$\frac{1}{r} \frac{\partial}{\partial r} (kr \frac{\partial T}{\partial r}) + \frac{1}{r^2} \frac{\partial}{\partial \phi} (kr \frac{\partial T}{\partial \phi}) + \frac{\partial}{\partial z} (k \frac{\partial T}{\partial z}) + \dot{q} = \rho c_p \frac{\partial T}{\partial t}$$

Spherical

$$\frac{1}{r^2} \frac{\partial}{\partial r} (kr^2 \frac{\partial T}{\partial r}) + \frac{1}{r^2 \sin^2 \theta} \frac{\partial}{\partial \phi} (k \frac{\partial T}{\partial \phi}) + \frac{1}{r^2 \sin^2 \theta} \frac{\partial}{\partial \theta} (k \sin \theta \frac{\partial T}{\partial \theta}) + \dot{q} = \rho c_p \frac{\partial T}{\partial t}$$

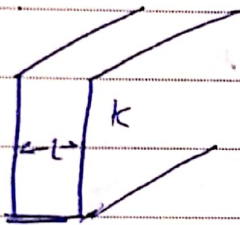
heat
wall, cylindrical, spherical

10th Oct - 2019

	plane wall	cylindrical wall	spherical wall
H.D.E	✓	✓	
heat (diffusion) equation			$\frac{d}{dr}(r^2 \frac{dT}{dr}) = 0$
Temp. distribution	$\frac{T(x) - T_{s1}}{T_{s2} - T_{s1}} = \frac{x}{L}$	✓	✓
q or q''	$q = kA \frac{\Delta T}{L}$	$q = \frac{2\pi L k \Delta T}{\ln(r_2/r_1)}$	$q = 4\pi k r_1 r_2 \frac{\Delta T}{r_2 - r_1}$ $q = 4\pi k \frac{\Delta T}{\frac{1}{r_1} - \frac{1}{r_2}}$

Cond

$$q = kA \frac{\Delta T}{L} = \frac{\Delta T}{(L/kA)} = \frac{\Delta T}{R_{t,cond}} \rightarrow \text{driving force} \text{ Resis}$$



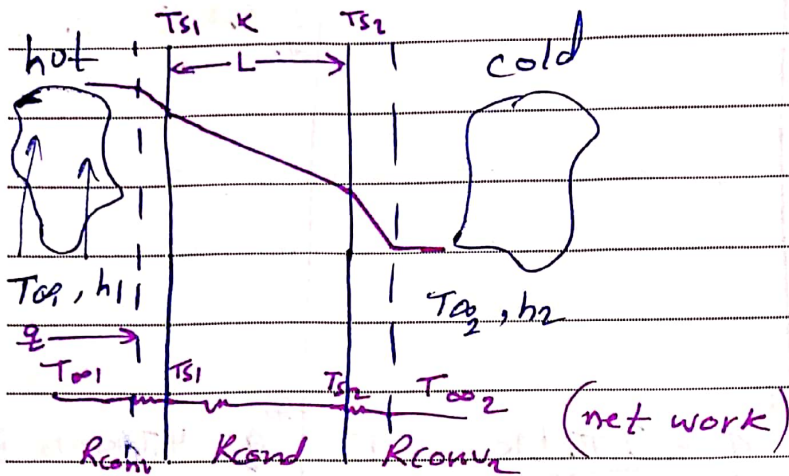
Conv

$$q = hA \Delta T \rightarrow (T_{\infty 1} - T_{s1}) = \frac{\Delta T}{(1/hA)} = \frac{\Delta T}{R_{t(conv)}}$$

$$q = \frac{2\pi L k \Delta T}{\ln(r_2/r_1)} \rightarrow q = \frac{\Delta T}{\frac{\ln(r_2/r_1)}{2\pi L k}} \text{ so } R_{resis} = \frac{\ln(r_2/r_1)}{2\pi L k}$$

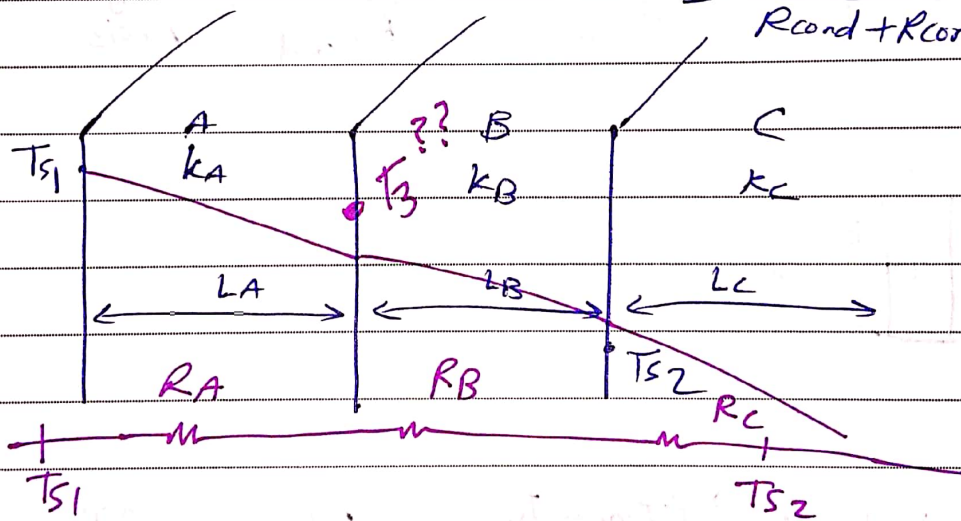
$$R \text{ for sphere} = \frac{(1/r_1 - 1/r_2)}{4\pi k}$$

$$q = \frac{\Delta T}{R}$$



$$q = \frac{T_{\infty 1} - T_{s1}}{R_{conv1}} = \frac{T_{s1} - T_{s2}}{R_{cond}} = \frac{T_{s2} - T_{\infty 2}}{R_{conv2}} = \frac{T_{\infty 1} - T_{\infty 2}}{\sum R}$$

$$= \frac{T_{s1} - T_{\infty 2}}{R_{cond} + R_{conv2}}$$



$$\frac{L_A}{k_A} \quad \frac{L_B}{k_B} \quad \frac{L_C}{k_C}$$

$$q = \frac{\Delta T}{\sum R} \rightarrow (T_{s1} - T_{s2})$$

$$q = \frac{T_{s1} - T_3}{R_A}$$

ask, believe & recieve

O.H.T.C [U]

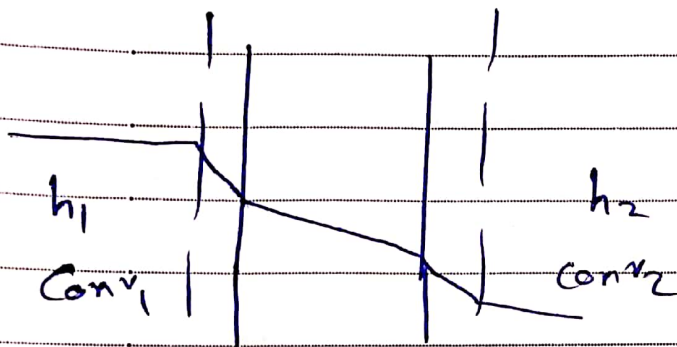
$$Q = UA \Delta T \rightarrow \text{overall temp. difference}$$

$$Q = \frac{\Delta T}{R_{\text{total}}} \dots \text{so;}$$

$$\underline{UA = \frac{1}{R_{\text{total}}}}$$

you can write $U = \frac{1}{A R_{\text{total}}}$

⊗ ex

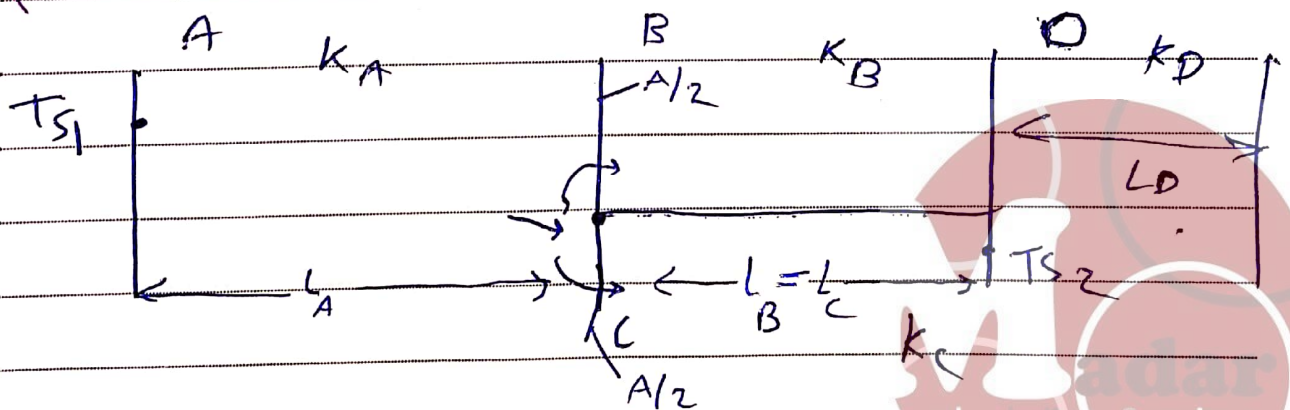


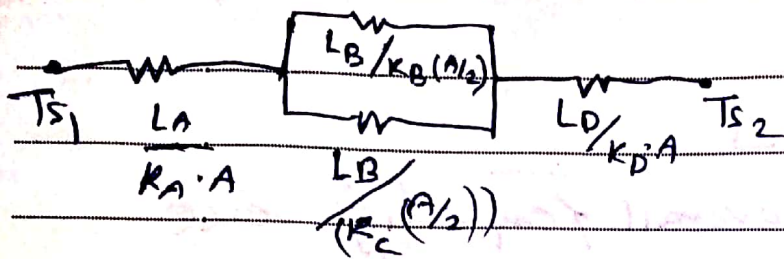
$$U = \frac{1}{A R_{\text{total}}} = \frac{1}{\left(\frac{1}{h_1} + \frac{L}{RA} + \frac{1}{h_2} \right) A}$$

$$= \frac{1}{\left(\frac{1}{h_1} + \frac{L}{R} + \frac{1}{h_2} \right)}$$

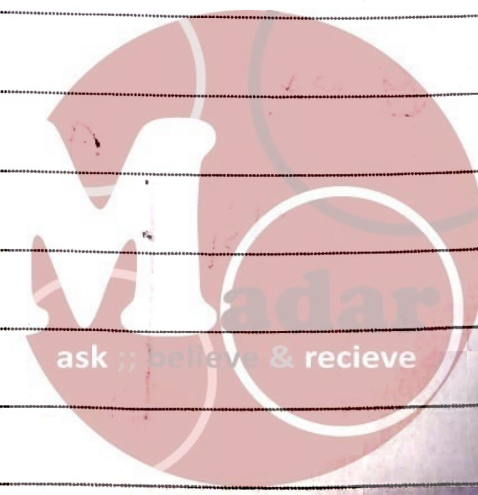
$$R_{\text{total}} = \frac{1}{UA} \rightarrow \text{مقاومة كلية}$$

⊗ ex



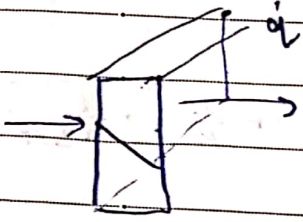


$$R_{total} = \left[\frac{1}{R_2} + \frac{1}{R_3} \right]^{-1} \rightarrow \text{سلسلة}$$



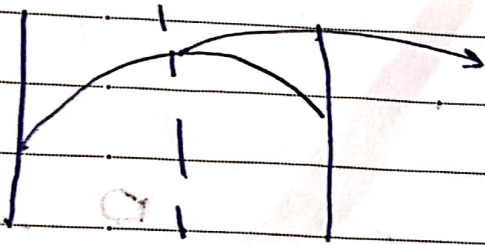
problem 2 on slides:-

$$\dot{E}_{in} = \dot{E}_{out} + \dot{E}_g \rightarrow 0 \rightarrow \frac{k_w}{m^2} \cdot A - q'' \times A + \underbrace{\dot{q} \times A \times L}_{\text{volum}} = 0$$



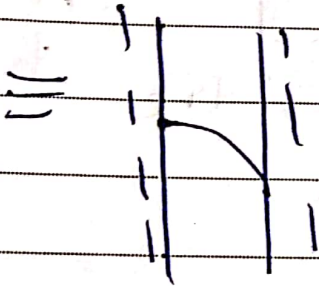
The plane wall:-

If there is generation term as the method or approach if there is not $\dot{E}_g$



insulated B.C
symmetrical

symmetrical



ask :: believe & recieve

$$5Lx\dot{w}_1$$

15th Oct

$$\frac{d^2 I}{dx^2} + \frac{\dot{q}}{K} = 0 \Rightarrow 10$$

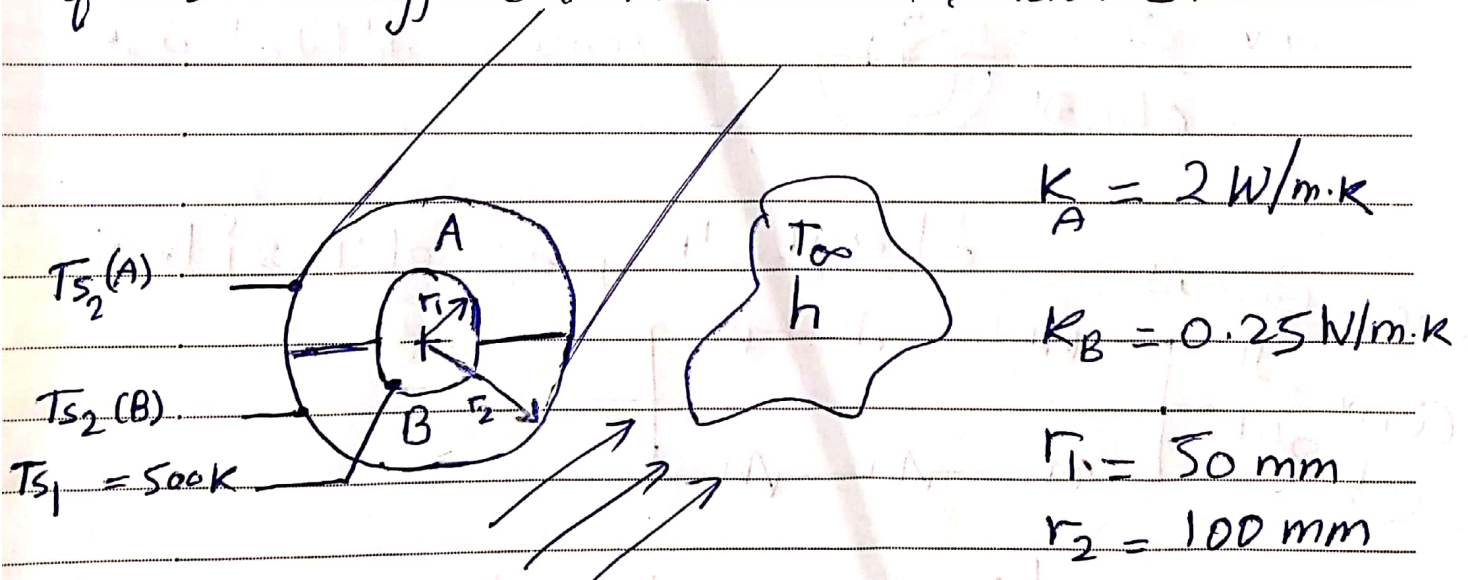
R constant

There is a heat generation

surface balance $\Delta \sigma = \gamma_1 - \gamma_2$ T_s $\Delta \sigma$ $\Delta \sigma$ (*)

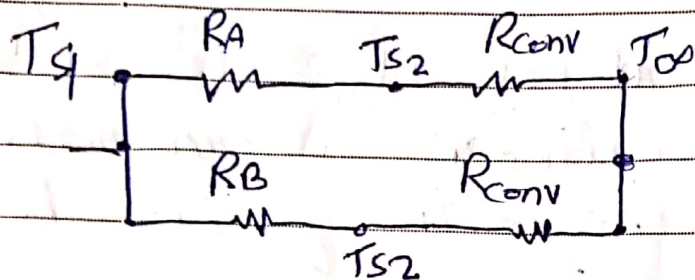
if we want T_s for a cylinder; we can make $\odot$ energy balance at the centre of it because at $r=0$ its adiabatic.

Q: (3.52) Steam flowing through a long thin-walled pipe maintains the pipe wall at a uniform temperature of 500K . The pipe is covered with an insulation blanket comprised of two different materials, A and B.

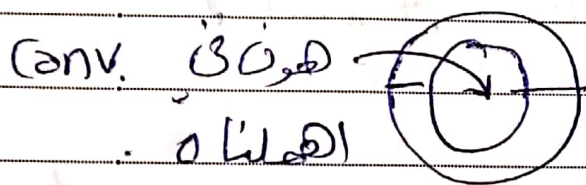
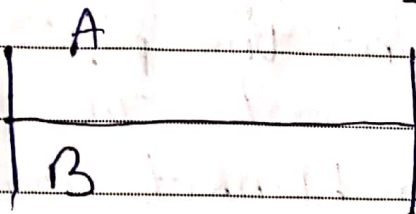


The interface between the two materials may be assumed to have an infinite contact resistance and the entire outer surface is exposed to air for which $T_{\infty} = 300 \text{ K}$ & $h = 25 \text{ W/m}^2\text{K}$

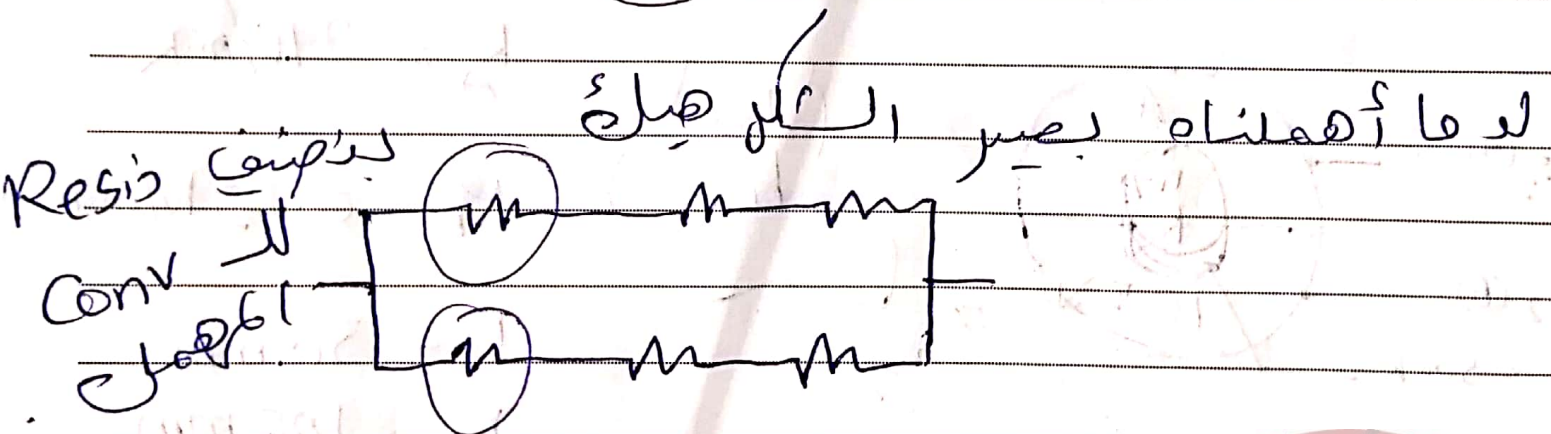
@ sketch the thermal circuit of the system.
label (using the above symbols) all pertinent nodes and resistances.



كيف عرفنا أنه توازي؟ التحليل إلى فاتحة الـ Pipe
يسير Wall $\leftarrow$ فصول بالتوازي



هو هذا الـ Conv



لما فهمنا يسير الـ في الخارج

Resis الـ في الخارج

Conv الـ في الخارج

Resis الـ في الخارج

$(A/2)$

$$R_{cond} = \frac{\ln(r_2/r_1)}{2\pi k_A L}$$

R_{conv}

R_{cond}

$$R_{conv} = \frac{1}{h \pi r_2 L}$$

b) for the prescribed conditions, what is the total heat loss from the pipe? what are the outer surface temperatures $T_{s(A)}$ & $T_{s(B)}$?

$$q_{\text{total loss}} = q_A + q_B$$

(Conv h) $\Rightarrow$

T_{s1} constant = 500K

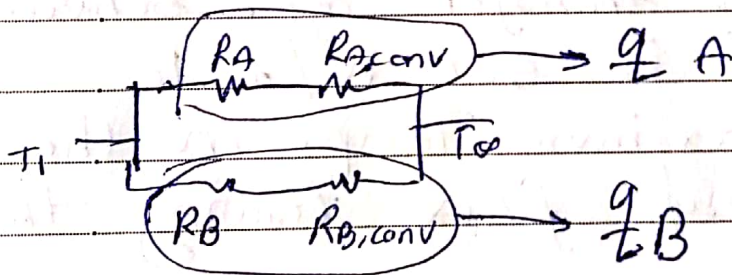
2 assumption $\rightarrow T_{s1}$ constant

$\rightarrow$ neglected the conv. here

$$q_A = \frac{T_{s1} - T_{\infty}}{\sum R}$$

$$q_B = \frac{T_{s1} - T_{\infty}}{\sum R}$$

$$(\sum R) \rightarrow (R_B + R_{conv})$$



but R'_A or R'_B (per unit length $L=1$)

(1) $\text{m}^2/\text{K} \rightarrow (L)$ $\leftarrow$ K^{-1}

$$R'_A = \frac{\ln(r_2/r_1)}{2\pi k (L)=1}$$

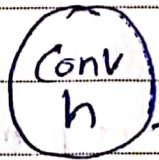
$$R'_{conv} = \frac{1}{hA} = \frac{1}{h 2\pi r_2 (L)=1}$$

ask ; believe & recieve

$$R_{conv} = \frac{1}{h \pi r_2 L}$$

b) for the prescribed conditions, what is the total heat loss from the pipe? what are the outer surface temperatures $T_{s(A)}$ & $T_{s(B)}$?

$$q_{total loss} = q_A + q_B$$



T_{s1} constant = 500K

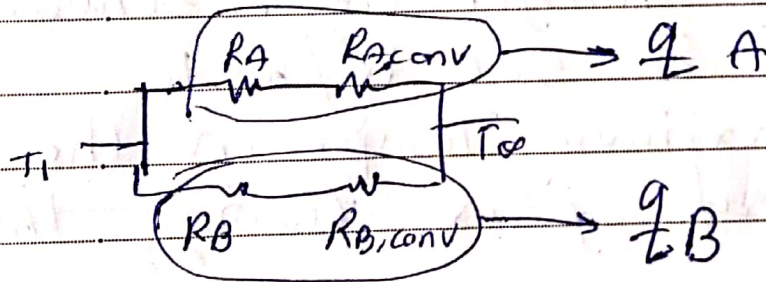
2 assumption $\rightarrow T_{s1}$ constant

$\rightarrow$ neglected the conv. here

$$q_A = \frac{T_{s1} - T_{\infty}}{\sum R}$$

$$q_B = \frac{T_{s1} - T_{\infty}}{\sum R}$$

$$\sum R \rightarrow (R_B + R_{conv})$$



but R'_A or R'_B (per unit length $L=1$)

(1) air jal $\rightarrow (L)$ $\underline{L=1}$

$$R'_A = \frac{\ln(r_2/r_1)}{2\pi k (L)=1}$$

$$R'_{conv} = \frac{1}{hA} = \frac{1}{h 2\pi r_2 (L)=1}$$

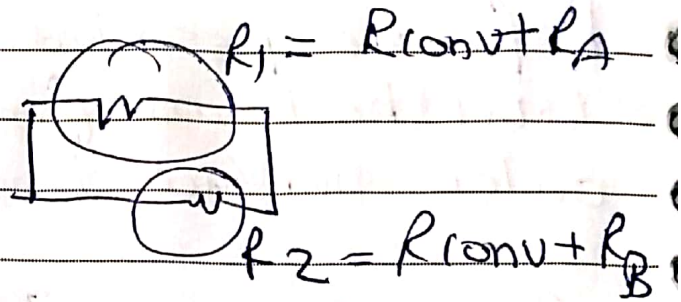
ask, believe & receive

R_{total} de $\dot{Q} = \frac{\Delta T}{R}$

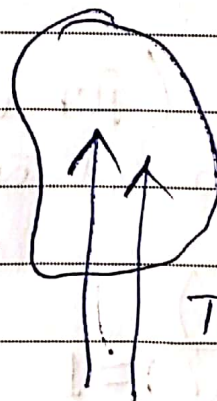
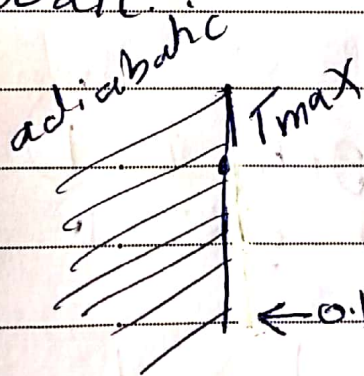
R_{total}

$$\dot{Q} = \frac{T_{S1} - T_{\infty}}{\sum R \rightarrow total}$$

$$R = \left[\frac{1}{R_1} + \frac{1}{R_2} \right]^{-1}$$



Q2: A plane wall of thickness 0.1 m and Thermal conductivity 25 W/m.K having uniform volumetric heat generation of 0.314 W/m^3 is insulated on one side, while the other side is exposed to a fluid @ 92°C . The convection heat transfer coeff. between wall and the fluid is $500 \text{ W/m}^2\cdot\text{K}$. Determine the maximum Temp. in the wall and the heat flux leaving the wall.

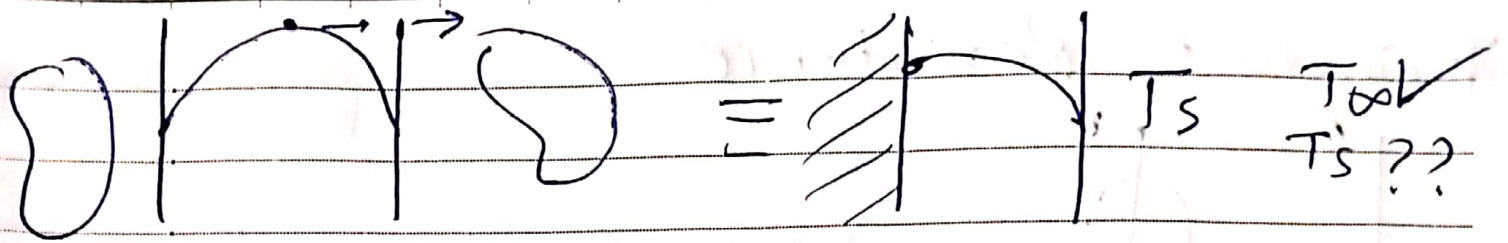


$$h = 500 \text{ W/m}^2\cdot\text{K}$$

$$K = 25$$

$$\dot{q} = 0.3 \text{ MW/m}^3$$

$\text{mega} = 10^6$



from surface balance:

$$T_s = T_\infty + \frac{q \cdot L}{h} \quad \leftarrow \text{from this.}$$

$$T(0) = \frac{q \cdot L^2}{2k} + \bar{T}_s$$

العوائق في السطح إلى نفس الموضع
symmetric wall.

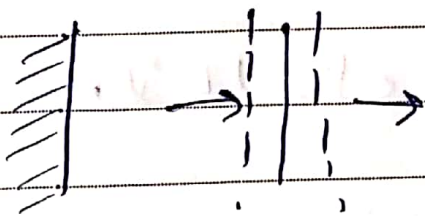
heat loss $q_{out} = \frac{h A (T_s - T_\infty)}{L}$ (1) 1

من مساحة عادي اعتبره flux

$$q'' = h (T_s - T_\infty)$$

Energy balance

(2) 2



$$-E_{out} + E_g = 0$$

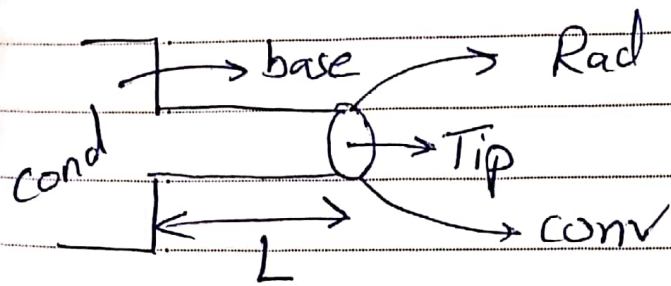
Conv $E_{out} = E_g = q \times \text{volume}$
 $A \cdot L$

assumption

$A=1$ 3

$$q_{conv} = E_{out} = E_g = q \cdot A \cdot L$$

Fins $\uparrow A$



$$\frac{d}{dx} \left(A_c \frac{dT}{dx} \right) - \frac{h}{k} \frac{dA_s}{dx} (T - T_\infty) = 0$$

A_c area
gradient

$\Downarrow$ simplify

$$A_c \frac{d^2 T}{dx^2} + \frac{dT}{dx} \frac{dA_c}{dx} - \frac{h}{k} \frac{dA_s}{dx} (T - T_\infty) = 0$$

$\div A_c$

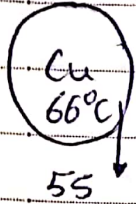
$$\frac{d^2 T}{dx^2} + \left(\frac{1}{A_c} \frac{dA_c}{dx} \right) \frac{dT}{dx} - \left(\frac{1}{A_c} \frac{h}{k} \frac{dA_s}{dx} \right) (T - T_\infty) = 0$$

Rectangular fin

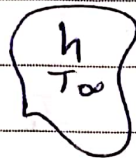
مستطیل
شکل



Prob ① The heat transfer coeff. for air flowing over a sphere is to be determined by observing the Temp. - time history of a sphere fabricated from pure copper. The sphere, which is 12.7 mm in diameter, is at 66°C before it is inserted into an airstream having a Temp. of 27°C. A thermocouple on the outer surface of the sphere



$t = 69\text{ s}$



Indicates 55°C 69s after the sphere is inserted in the airstream.

Assume isothermal copper sphere and calculate the heat transfer coeff.?

$$\frac{\theta}{\theta_i} = \exp\left[-\frac{t}{\tau}\right] \rightarrow \frac{55 - 27}{66 - 27} = \exp\left[-\frac{69}{\tau}\right]$$

$$\tau = \frac{\rho V c_p}{h A_s}$$

$$L_c = \frac{V}{A_s} = \frac{r_0}{3}$$

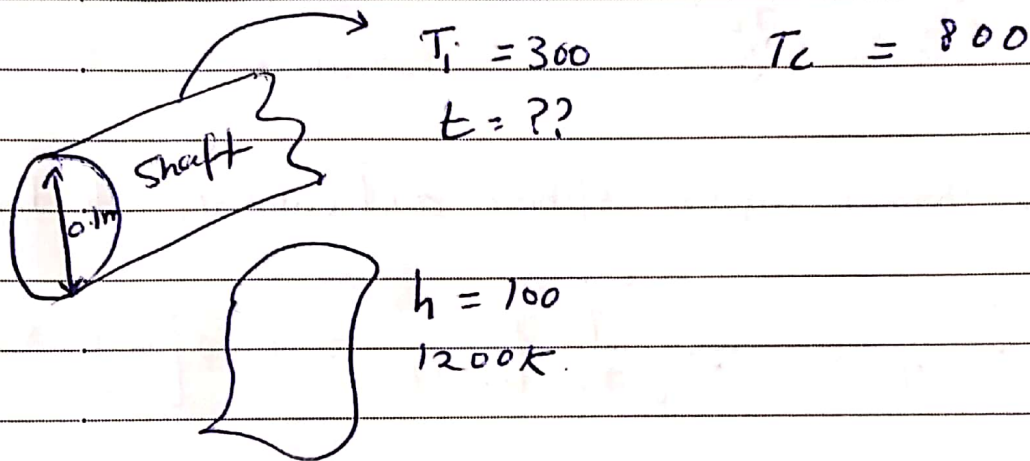
$$A_s = 4\pi r^2$$

$$V = \frac{4}{3}\pi r^3$$

⇒ Justify ⇒ Bi < 0.1

$$Bi = \frac{h r/3}{k} \rightarrow \text{Appendix} \rightarrow \text{Avg}$$

Prob 2: Carbon steel 1010 shaft of 0.1 m diam. are heat treated in a gas-fired furnace whose gases are at 1200 K and provide a convection heat transfer coeff. of 100 W/m².K. If the shafts enter the furnace @ 300 K, How long must they remain in the furnace to achieve a centerline Temp of 800 K.



$$Bi = \frac{h L_c}{k} \rightarrow \left(\frac{r_o}{2}\right) \text{ outer radius}$$

$k \rightarrow T_{avg} = T_c = T_i = T_o = T_{\infty}$ center = outer, The same Temp. 1

$$= 0.1$$

second sol: $\frac{T - T_{\infty}}{T_i - T_{\infty}} = \text{Exp} \left[\frac{-t}{\tau} \right]$ to find

General lumped Capacitance Analysis.

$$Bi < 0.1 \Rightarrow \text{neg Rad} \Rightarrow \underline{5.19}$$

$$\theta = T - T_{\infty}$$

$$\frac{dT}{dt} = \frac{d\theta}{dt} \Rightarrow \text{put in the eq (5.15)}$$

$$\frac{h}{\rho V_c} = a \quad \frac{q'' A_s h + E_g}{\rho V_c} = b$$

$$\Rightarrow \frac{d\theta}{dt} + a\theta - b = 0$$

$$\theta' = \theta - \frac{b}{a} \text{ to be homogenous.}$$

Pulverized coal $\therefore$ (5.21)

prob: before being injected into a furnace, pulverized coal is preheated by passing it through a cylindrical tube whose surface is maintained at $T_{sur} = 1000^\circ\text{C}$. The coal pellets are suspended in an airflow and are known to move with a speed of 3 m/s. If the pellets may be approximated as spheres of 1 mm diameter and it may be assumed heated by radiation transfer from the tube surface. $\rightarrow$ black surface = assumption

$$T_i = 25^\circ\text{C}$$

$$T_{sur} = 1000^\circ\text{C}$$

$$\epsilon = 0.6$$

$$L = ??$$

$\downarrow$
the needed

$$v = 3 \text{ m/s}$$

$$L = v \times t$$

$\Rightarrow$ how long must the tube be to heat coal entering @ 25°C at a Temp of 600°C ?

NO gen

NO flux

(5.17)

$t =$

$$Bi = \frac{h \times (V_3 r_0)}{K}$$

Chapter 1

$$q = hA(T_{\text{sur}} - T)$$

$$h = (\quad) (\quad)$$



31st Oct - 2019

u-31

Spatial $\Rightarrow T(x)$ exist

$$\frac{\partial^2 T}{\partial x^2} = \frac{1}{\alpha} \frac{\partial T}{\partial t}$$

① $t=0 \quad T=T_i$

② $x=0 \quad \frac{\partial T}{\partial x} = 0$ adiabatic

③ Convection = conduction

$$\theta^* = \frac{\theta}{\theta_i} = \frac{T - T_\infty}{T_i - T_\infty}$$

T, x, t

$$x^* = \frac{x}{L} \text{ dimensionless}$$

$$t^* = \tau_0 = \frac{\alpha t}{L^2} \Rightarrow \text{dimensionless time}$$

$$\frac{\theta}{\theta_i} = \exp \left[-t/\tau \right] \rightarrow \exp \left[- \left(\frac{t h A_s}{P V_c} \right) \right]$$

$$\tau = \frac{P V_c}{h A_s}$$

$$\frac{t h A_s}{P V_c} \rightarrow (5.11)$$

$$\frac{\theta}{\theta_i} = \exp \left[-B_i / \tau_0 \right]$$

$$\frac{d^2 T}{dx^2} = \frac{1}{\alpha} \frac{dT}{dt} \quad \text{we put } x^* \quad t^* \quad T^*$$

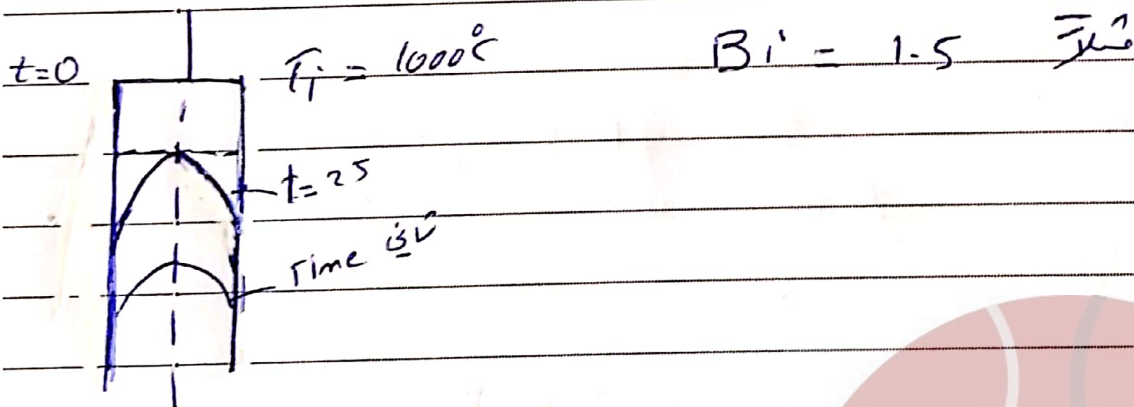
$$\frac{\partial}{\partial x^*} \frac{\partial^2 \theta^*}{\partial x^{*2}} = \frac{1}{\alpha} \frac{\partial}{\partial t^*} \frac{\partial \theta}{\partial f_0} \Rightarrow \frac{\partial^2 \theta^*}{\partial x^{*2}} = \frac{\partial \theta}{\partial f_0}$$

$$\theta^* (x^*, 0) = 1 \quad (T = T_i)$$

$$\frac{\partial \theta^*}{\partial x^*} \bigg|_{x^*=0} = 0$$

$$-k \frac{\partial T}{\partial x} \bigg|_{x=L} = h [T(L, t) - T_\infty] \quad (8-29)$$

$\Rightarrow f_0$ include time $\frac{\alpha t}{L^2}$



* $\frac{h r_0}{k} \Rightarrow$ cylinder $\Rightarrow \frac{h r_0}{k} = 2 \Rightarrow \overline{f_0}$

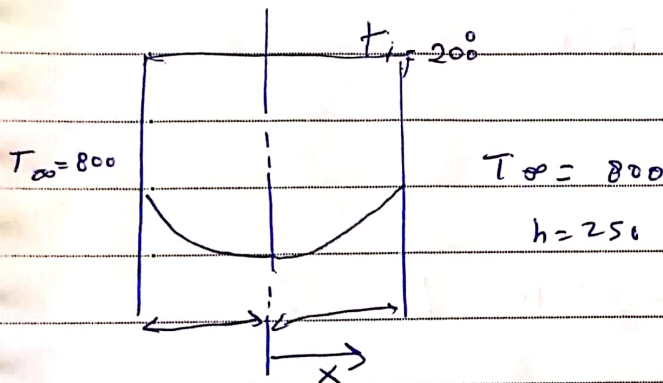
Bi lumped $\Rightarrow$ cylinder $\Rightarrow Bi = \frac{h r_0/2}{k} = 0.2$

lumped $\Rightarrow$ special

$$Bi = \frac{h r_o}{K}, \text{ Rad.}$$

Example: Annealing is a process by which steel is reheated and then cooled to make it less brittle. Consider the reheat stage for a 100 mm thick steel plate ($\rho = 7830 \text{ kg/m}^3$, $c = 550 \text{ J/kg}\cdot\text{K}$, $K = 48 \text{ W/m}\cdot\text{K}$), which is initially at a uniform Temp of $T_i = 200^\circ\text{C}$ and is to be heated to a minimum Temp of 550°C . Heating is effected in a gas-fired furnace.

$$Bi = \frac{h L_o}{K} = \frac{250 \times}{48} = 5.2 \quad \text{spatial}$$



$$\theta = \frac{T_{\text{min}} - T_\infty}{T_i - T_\infty} = \theta_o^* = C_1 \exp\left(-\xi_1^2 Fo\right)$$

$$Bi = \infty \rightarrow \left(\begin{array}{l} 200 \text{ is like} \\ \infty \text{ is } \infty \end{array} \right)$$

ask :: believe & recieve
 1. $\text{Resistance very very small} \Rightarrow T_{\text{wall}} \equiv T_\infty$

Problem (2):

Spheres of 40 mm diameter heated to a uniform Temp of 400°C are suddenly removed from the oven and placed in a forced-air bath operating at 25°C with conv. coeff. of $300 \text{ W/m}^2\cdot\text{K}$ on the sphere surfaces. The thermophysical prop. of the sphere material

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

$$c = 850 \text{ J/kg}\cdot\text{K}$$

$$k = 15 \text{ W/m}\cdot\text{K}$$

$$\frac{Q}{Q_0} = 1 - \frac{3\theta_0^*}{\xi_1^3} [\sin(\xi_1) - \xi_1 \cos(\xi_1)]$$

$$\theta_0^* = \eta \exp(-\xi_1^2 Fo)$$

$$Q = 0.80 Q_0$$

$$\frac{Q}{Q_0} = 0.8 = 1 - 3 \frac{\eta \exp(-\xi_1^2 Fo)}{\xi_1^3} [\dots]$$

$$Bi = \frac{h r_0}{k} \Rightarrow \text{lumped.}$$

$$Bi = \frac{h r_0}{k} \Rightarrow Bi < 1 \quad \xi_1$$

ask :: believe & recieve

$$S_+ \uparrow \quad x \uparrow \quad (T_s - T_\infty) \text{ (on)}$$

$$\frac{d}{dx} \downarrow \quad h \downarrow$$

$$\bar{h} = \frac{1}{L} \int_0^L h \, dx \rightarrow \text{average}$$

shear stress $\uparrow$ mixing turbulent

$$\nu = \frac{\mu}{\rho} \rightarrow \frac{\nu}{\alpha} = \frac{\mu \rho}{k}$$

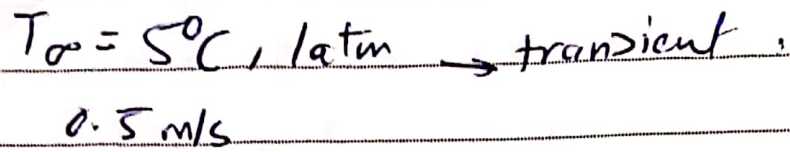
problem: a fluid @ 20°C flows across a flat surface with a constant velocity. find the distance from the leading edge @ which the flow becomes critical for the following fluids and free stream-velocity

$$Re = \frac{\rho U_\infty x}{\mu} = 15 \times 10^5$$

example: assume an organic particles @ Temp 15°C are required to be cooled by an air @ 5°C, 1 atm and a velocity of 0.5 m/s. The heat transfer coeff by convection is determined experimentally and is found to be $h = 5.05 \text{ K} \cdot \text{m}^{-1} \cdot \text{s}^{-1} Re^{1/3}$ for air velocities $0.1 \text{ K} \leq 0.33 \text{ m/s}$

where D is the diameter of oranges. Determine

① The initial rate of heat transfer from 7 cm diam
a $k = 0.5 \text{ W/m} \cdot \text{°C}$



$$\textcircled{2} \quad Nu = \frac{q_{conv}}{q_{cond}}$$

$$Re = \frac{v_{\infty} D}{\nu} \rightarrow \frac{D}{x}$$

$$\dot{Q} = A_s h (T_s - T_\infty)$$

إذا كان K لا يتغير مع K_{water} assumption.

$$Nu = \frac{hD}{k} = 68.11$$

14th Oct - 2019

$$Nu = \frac{hL}{k_f} = \frac{\partial T^+}{\partial y^+} \bigg|_{y=0} = F(\vec{x^+}, Re, Pr)$$

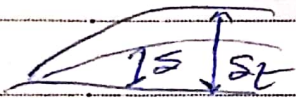
$$\overline{Nu} = F(Re, Pr) \rightarrow \text{Average of } F \text{ no effect of } x^+$$

Similarity parameter

$$Sh = f(Re, Sc)$$

$\frac{h_m L}{D_{AB}}$

D_{AB}



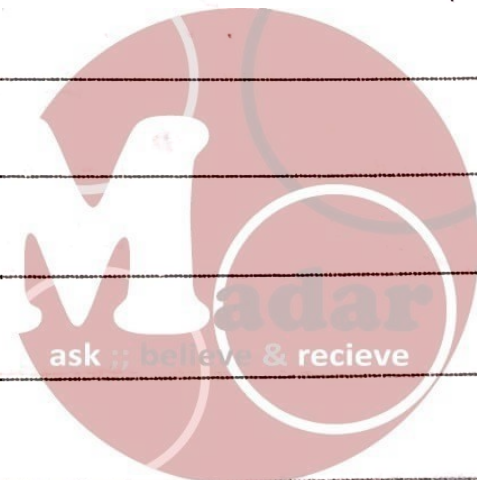
$$Pr \gg 1$$

$$Pr = 1$$

$$Pr \ll 1$$

$$\frac{S}{S_k} = Pr^n$$

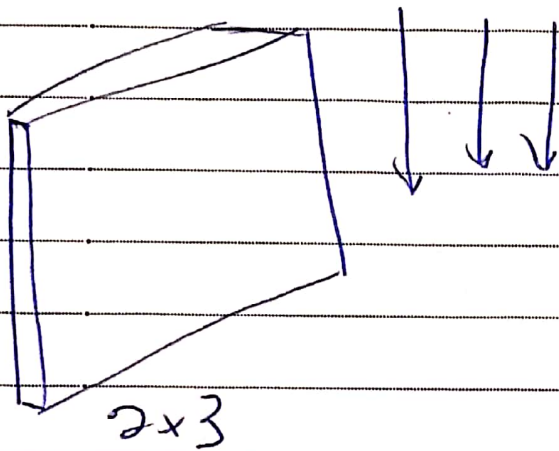
E-66



Sun.
17th - Nov. 2019

(ex)

A 2m. x 2m plate is suspended in a room, and is subjected to air flow parallel to its surfaces along its 3m long side. The free stream Temp. and velocity of air are 20°C & 7 m/s. The total drag force acting on the plate is measured to be 0.86 N. Determine the avg convection heat transfer coefficient for



$$T_{\infty} = 20^{\circ}\text{C}$$

$$u = 7 \text{ m/s.}$$

$$F_D = 0.86 \text{ N}$$

$$\frac{C_f}{2} = s_L = 5 \text{ m}$$

$$j_H = \frac{C_f}{2} = s_L \uparrow P_r^{2/3}$$

$\frac{h}{\rho V C_p}$

$$C_f = \frac{\tau_s}{\frac{1}{2} \rho V^2} \rightarrow \left[\frac{F_D}{A} \right] \rightarrow 0.86$$

2x3



The heat transfer rate per unit width (normal to the page) from a longitudinal section $(x_2 - x_1)$, can be expressed as $q'_{12} = \bar{h}_{12} (x_2 - x_1) (T_s - T_\infty)$ where $\bar{h}_{12}$ is the avg coefficient for the section of length $(x_2 - x_1)$, consider laminar flow over a flat plate with a uniform temp T_s . The spatial variation of the local convection coeff. is of the form $h_x = C x^{-1/2}$, where C is a constant

$$dq' = h_x dx (T_s - T_\infty)$$

$$\bar{h}_{12} = f(C, T_s, x_2)$$

$$\bar{h} = \frac{1}{L} \int_0^L h dx$$

$$q'_{12} = \bar{h}_{12} (x_2 - x_1) (T_s - T_\infty)$$

$$q'_{12} = (T_s - T_\infty) \int_{x_1}^{x_2} h_x dx$$

$$\bar{h}_{12} = \frac{1}{x_2 - x_1} \int_{x_1}^{x_2} h_x dx \rightarrow C_1 x^{-1/2}$$

(P-3)

An object of irregular shape has a characteristic length of $L = 1\text{m}$ and is maintained at a uniform surface temp. of $T_s = 400\text{K}$. when placed in atmospheric air at a temp of $T_\infty = 300\text{K}$ and moving with a velocity of $V = 100\text{m/s}$, the avg. heat flux from the surface to the air is 20000W/m^2 . If a second object of the same shape, but with a characteristic length of $L = 5\text{m}$, is maintained at a surface Temp of $T_s = 400\text{K}$ and is placed in atmospheric air at $T_\infty = 300\text{K}$, what will the value of the average convection coefficient be, if the air velocity is $V = 20\text{m/s}$?

$$\overline{Nu} = f(Re, Pr) \rightarrow \text{avg air velocity}$$

to approve there is analogy between the both objects

Case 1

Case 2

Then $\overline{Nu}_{\text{case(1)}} = \overline{Nu}_{\text{case(2)}}$

$\downarrow$ $\downarrow$
 h_1 h_2

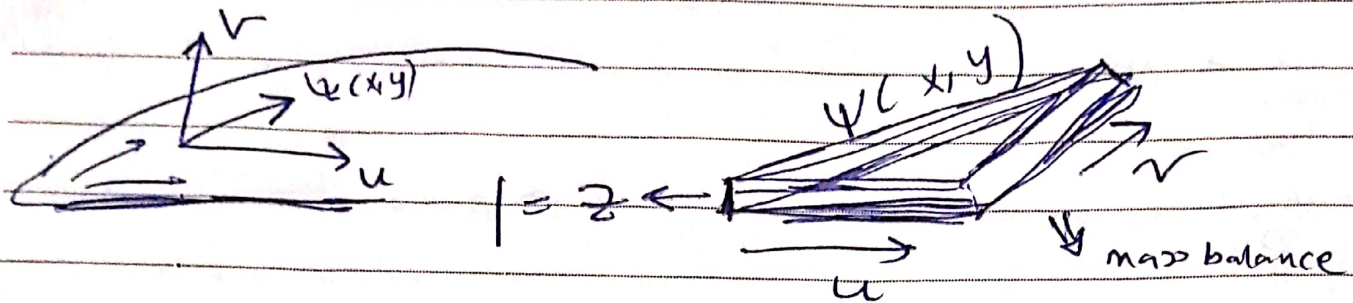
External flow

T_{∞} $\rightarrow$ factor $\rightarrow \frac{V(\text{viscosity})}{V(\text{surface})} \rightarrow$ properties visc

$$\left(\frac{\overline{Nu}}{Pr^n} \right) = \underset{\substack{\text{Intersect} \\ \downarrow}}{\text{C}} Re_L^{\overset{m}{\text{Slope}}} \Rightarrow \text{from chart}$$



laminar flow over an isothermal 19th Nov - 2019
 external flow $\equiv$ internal flow!!



$$\Rightarrow u dy * \rho = v dx (1) \rho$$

$$\psi(x, y) = \text{constant} \rightarrow \partial \psi = \frac{\partial \psi}{\partial x} dx + \frac{\partial \psi}{\partial y} dy$$

$$-v dx + u dy = 0$$

$$\psi = u_{\infty} \left(y \sqrt{\frac{u_{\infty}}{v_x}} \right) \cdot \left(\sqrt{\frac{v_x}{y_{\infty}}} \right)$$

$$\psi = u_{\infty} \sqrt{\frac{v_x}{u_{\infty}}} f(\eta)$$

$$\eta = y \sqrt{\frac{u_{\infty}}{v_x}}$$

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

$$u = \frac{\partial \psi}{\partial y} = \frac{\partial \psi}{\partial \eta} \frac{\partial \eta}{\partial y} \quad (\text{chain rule})$$

$$= \frac{\partial \psi}{\partial f} \frac{\partial f}{\partial \eta} \frac{\partial \eta}{\partial y} \rightarrow \frac{f \eta}{\partial \eta} \sqrt{\frac{u_{\infty}}{\nu x}} \cdot \frac{1}{\sqrt{\frac{u_{\infty}}{\nu x}}}$$

$$u = \frac{\partial \psi}{\partial y} = f$$

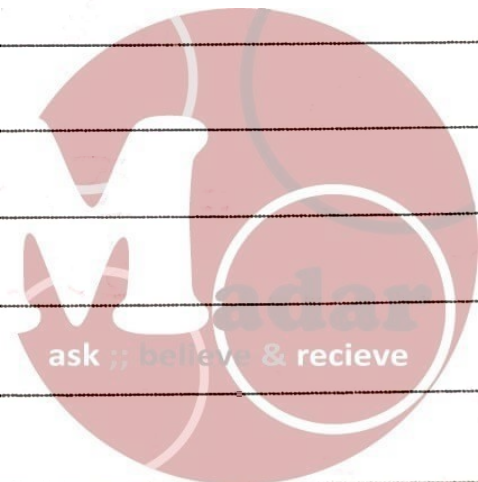
$$\frac{\partial u}{\partial y} = \frac{\partial f}{\partial y} = \frac{\partial f}{\partial \eta} \frac{\partial \eta}{\partial y} = \sqrt{\frac{u_{\infty}}{\nu x}} \frac{\partial f}{\partial \eta} \frac{1}{\sqrt{\frac{u_{\infty}}{\nu x}}} = \frac{\partial f}{\partial \eta}$$

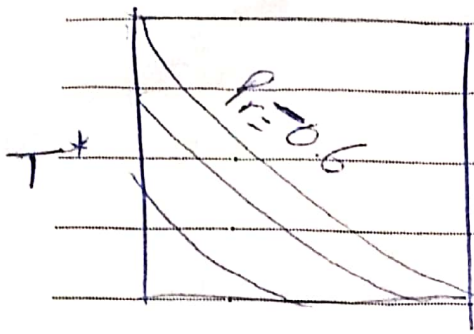
$$\left. \frac{\partial f}{\partial \eta} \right|_{\eta=0} = 0 \quad u(x,0) = 0 \quad \left. \frac{\partial f}{\partial \eta} \right|_{\eta=0} = 1$$

$$u(x,\infty) = u_{\infty}$$

$$S = \text{thickness} \quad \begin{array}{c} y \\ \text{---} \\ S \end{array} \quad y = 0.99$$

$$\frac{\delta}{2} = St Pr^{1/3}$$





$$T^* = \frac{T - T_s}{T_\infty - T_s}$$

$$y =$$

$$\left. \frac{dT}{dy} \right|_{y=0} \rightarrow \text{at the wall}$$

$$\left. \frac{\partial T^*}{\partial y} \right|_{y=0}$$

approximate analysis = integral method.

$$Nu_x = \frac{h_x x}{K} = 0.332 Re_x^{1/2} Pr^{1/3} \quad Pr \geq 0.6$$

table $\rightarrow$ $\frac{u}{u_\infty}$ $\frac{v}{u_\infty}$ $\frac{T - T_s}{T_\infty - T_s}$ Geometry

$$\frac{\delta}{\delta_t} = Pr^{1/3}$$

$$\overline{C}_{fx} = \frac{\overline{q}_{sA}}{\frac{1}{2} \rho u_\infty^2 x}$$

$$\overline{C}_{fx} = \frac{1}{x} \int_0^x T_{sx} dx \quad 0.332 u_\infty \sqrt{\rho \mu u_\infty / x}$$

$$\bar{h} = \frac{1}{x} \int_0^x h_x dx \quad \leftarrow \text{معدل}$$

$$= \frac{0.332 u_{\infty} (\rho \mu u_{\infty})^{1/2} x^{1/2}}{(x)^{1/2}}$$

$$= \frac{2 \times 0.332}{\rho u_{\infty}} (\rho \mu u_{\infty} / x)^{1/2}$$

$$= 4 \times 0.332 u_{\infty} \left(\frac{\rho \mu u_{\infty}}{x} \right)^{1/2}$$

$$= 4 \times 0.332 Re_x^{1/2}$$

$$\bar{C}_{fx} = 2 C_{fx}$$

$$\frac{\bar{C}_{fx}}{C_{fx}}$$

$$\bar{Nu} = \frac{\bar{h} x}{k}$$

Isothermal Plate

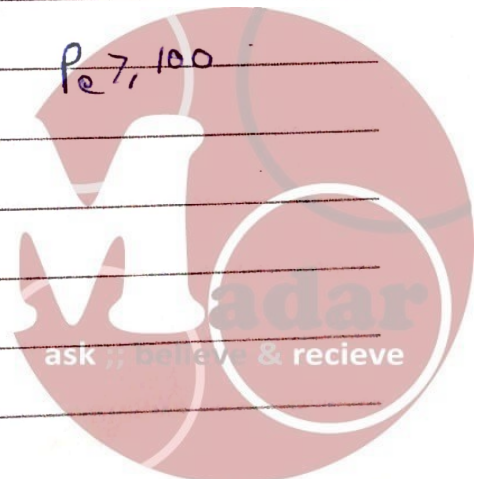
$$Nu = \frac{0.332 Re_x^{1/2} Pr^{1/3}}{[1 + (0.0468/Pr)^{1/3}]^{1/2}}$$

$Pr > 100$

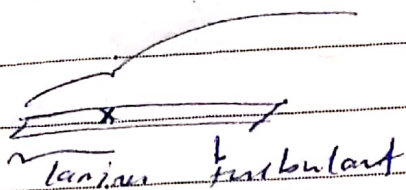
$$\bar{Nu} = 2 Nu$$

$$\delta = \delta_t = \delta_c \rightarrow \text{Thickness}$$

$$Re_c \rightarrow \text{turbulent}$$

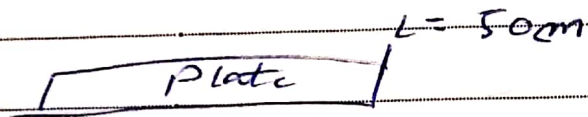


= Mixed.

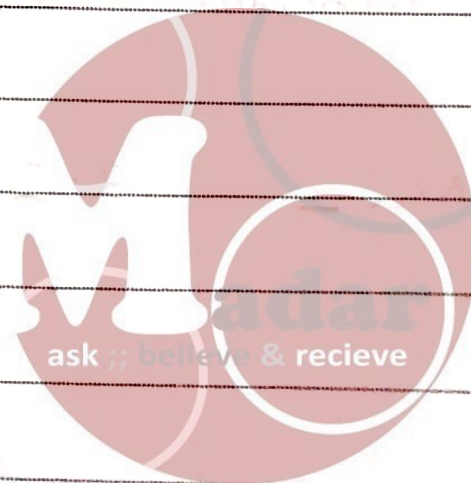


$$\bar{h}_x = \frac{1}{L} \left[\int_0^{x_c} h_x dx + \int_{x_c}^L h_x dx \right]$$

Labels: x_c → critical → النقطة الحرجة



$x_c = 30 \text{ cm} \rightarrow \text{Mixed.}$



The cylinder

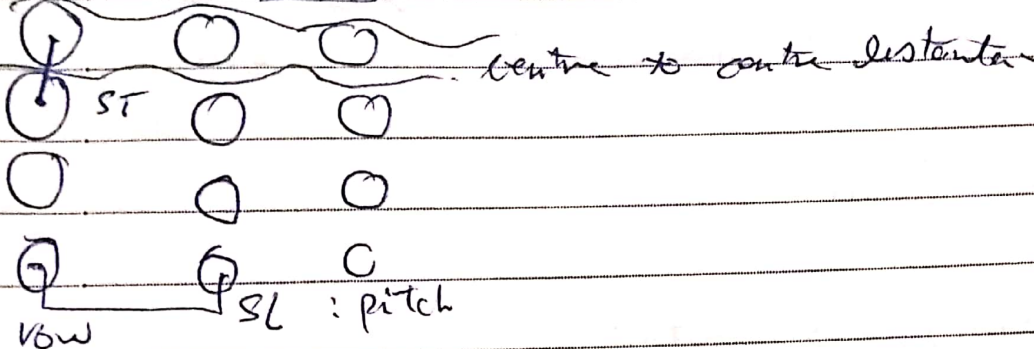
24th Nov - 2019.

$$Nu = x Re Pr * \boxed{\text{correlation}}$$

$$\text{average } T_f = \frac{T_s + T_\infty}{2}$$

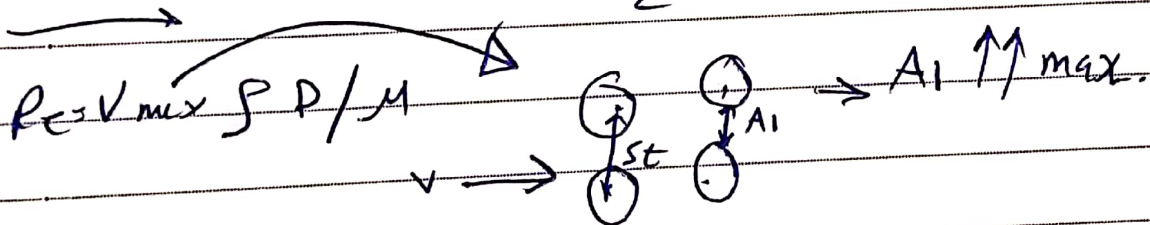
$\downarrow$ = falling drop

$h \rightarrow$ cylinder



h first Row $\rightarrow$ single cylinder

$\rightarrow$ properties @ $\frac{(T_s - T_\infty)}{2} \rightarrow$ expection Pr .

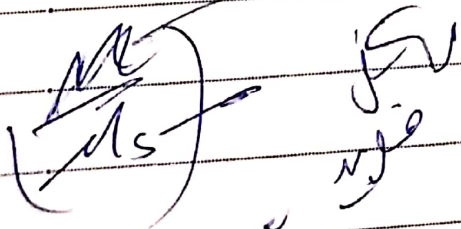


$$V(st) = V_{max} (ST - D) \times L$$

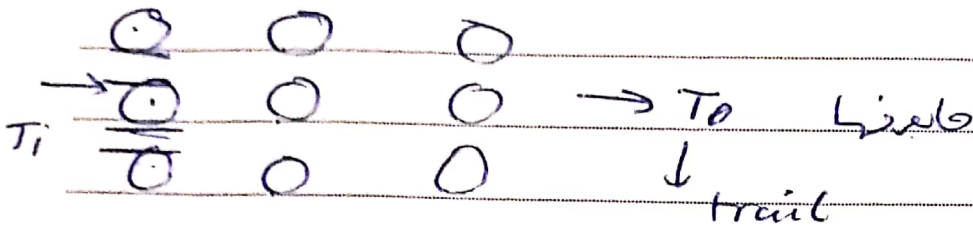
$$V_{max} = V \frac{ST}{ST - D}$$

$\rightarrow$ viscosity of the

the fluid



$$\dot{m} = \dot{V} \rho = v s_T \underline{N_T} L \rho$$

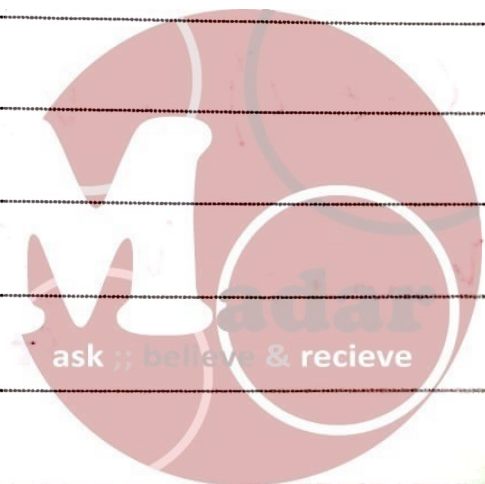


$$h \rightarrow \text{properties} \rightarrow T_{avg} \rightarrow \frac{T_i + T_o}{2} \rightarrow L \text{ and } l$$

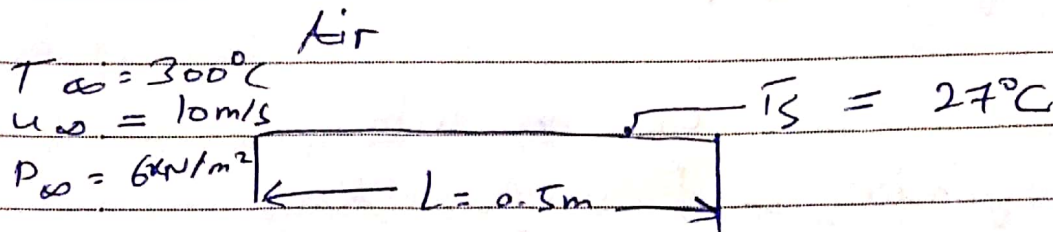
pressure drop $\rightarrow$ power

T_s (isothermal) $\rightarrow$ constant

constant flux $\rightarrow$



(example): Air at P of 6 kN/m^2 and a T of 300°C flows with a velocity of 10 m/s over a flat plate 0.5 m long. Estimate the cooling rate per unit width of the plate needed to maintain it at a surface temp. 27°C



$$Q = h A (T_\infty - T_s)$$

↳ (5 m) OR (Q/W)

$$Re = \frac{u L}{\nu} \Rightarrow \text{how to calculate the } \underline{h}$$

↳ Turbulent
 ↳ Laminar @ The last Point → outlet

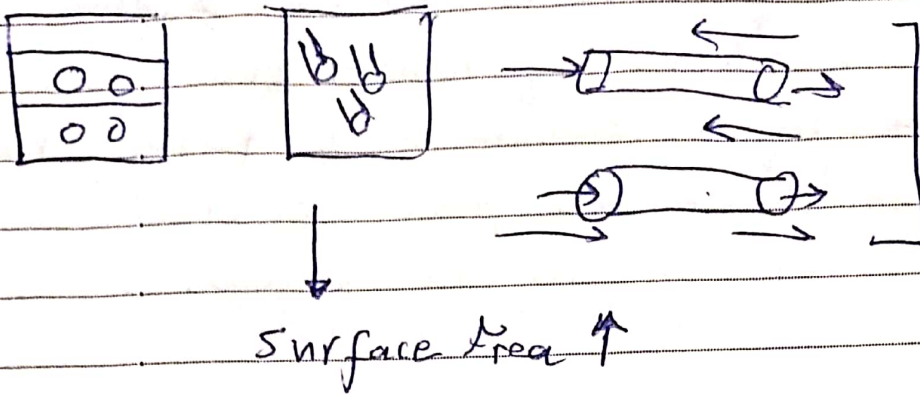
Turbulent ✓✓ ∴ → mixed +

← $[Re_c]$ →

Laminar → Turbulent

Ignore ⇒ [laminar] $\frac{L}{\nu}$

Pressure = correction = because table based on
 1 atm



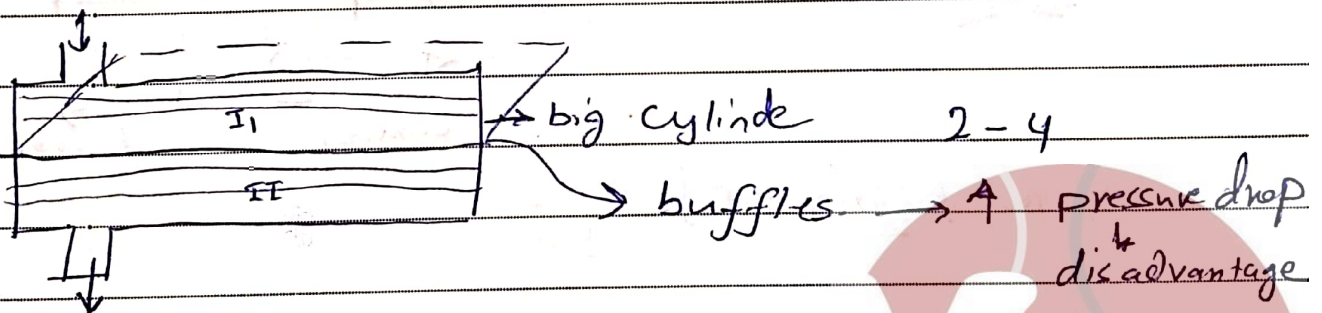
Surface Area ↑

Tube → may be more than one tube
Shell = big cylinder → ↑ surface Area

→ Single pass → No change in the direction of flow direction

1-1] shell and tube heat exchange
1-2]
 length $\propto$ 1/2

Passes → to control length

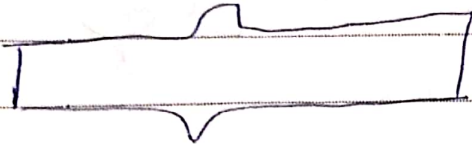


The most common heat exchanger
(1-2) → series

* Straight tube (fixed tube sheet) →
If there is not rust or deposition

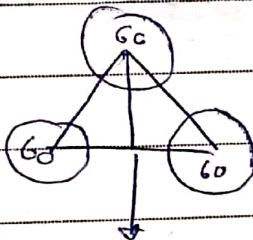
3rd Dec-2019

maintenance!! → remove dirt
removable → substance.

 → Temperature ↑↑↑

thickness ↑ pressure drop ↑ → ISF > $\frac{C_{\text{exp}} P_0}{\text{nominal sizes}}$

flow area / wetted Parameter



$\Delta - \frac{1}{2} \odot$

equivalent length of triangle



$$P_h = \frac{4A}{D}$$

$$Q_h = \boxed{\approx 100} \text{ friction} \rightarrow \text{pressure drop}$$

Inner $\rightarrow$ chapter 8.

Assumption = adiabatic system

=
No change in phase = sensible heat

basis = outside surface

$$Q = UA \Delta T \rightarrow U = \text{resistance}$$

based on inside $\frac{1}{h_o} \rightarrow$ correction.

$\left(\frac{A_i}{A_o}\right)$ correction factor

$h_i \rightarrow$ chapter 8.

$$h_{io} = h_i \left(\frac{A_i}{A_o}\right) \rightarrow \text{corrected}$$

$$\frac{1}{A_i} R'_{f,i} + R'_{f,o} \rightarrow \text{compound dirt}$$

$$(U_o): \text{dirt or design } (U_e) = \left[\frac{1}{\frac{1}{h_i} + \frac{1}{h_o} + R'_{f,i} + R'_{f,o}} \right]$$

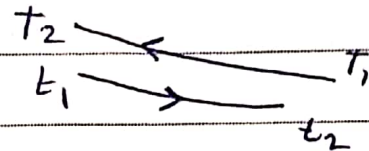
Pure Counter current =

$$F(\Delta T_{lm})$$

1-1 Shell and tube
log 1

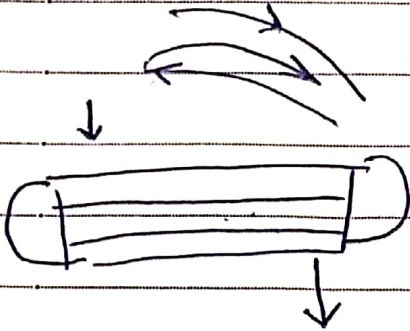
$$F=1$$

Pure counter current



1-2

↓ efficiency → multipasses



$F \rightarrow$ chart $\rightarrow$ 1 shell

$$R = \frac{(\dot{m} c_p)_c}{(\dot{m} c_p)_h} = \frac{\Delta T_h}{\Delta T_c} = \frac{T_1 - T_2}{T_2 - T_1}$$

$$\dot{m}_c c_{p,c} \Delta T = \dot{m}_h c_{p,h} \Delta T$$

$F > 0.9$ to avoid the drop

$F < 0.45$ bad design heat exchanger

$$\frac{A_T}{A_{one tube}} = n$$

Sizing $\rightarrow$ n -tube $\rightarrow$ flow in $\rightarrow$ given
surface area $\rightarrow$ flow out $\rightarrow$ given

Rating problem $\rightarrow$ efficiency $\rightarrow$ ΔT , f factors ...

$$C_c \equiv (m c_p)_c \quad C_c < C_h$$

Energy balance : $(m c_p \Delta T)_h = (m c_p \Delta T)_c$

$$C_h \Delta T_h = C_c \Delta T_c \quad \text{so } \Delta T_c > \Delta T_h$$

$$\Delta T_c = (T_o - t_i)$$

$$q_{\max} = C_c (T_{hi} - T_{ci}) \rightarrow \text{given} \rightarrow C_h < C_c$$

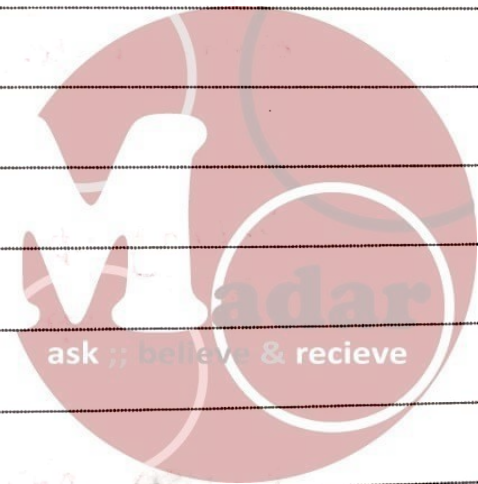
$$> C_h (T_{hi} - T_{ci})$$

$$q_{\max} = C_{\min} (T_{hi} - T_{ci})$$

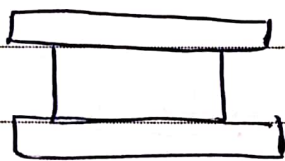
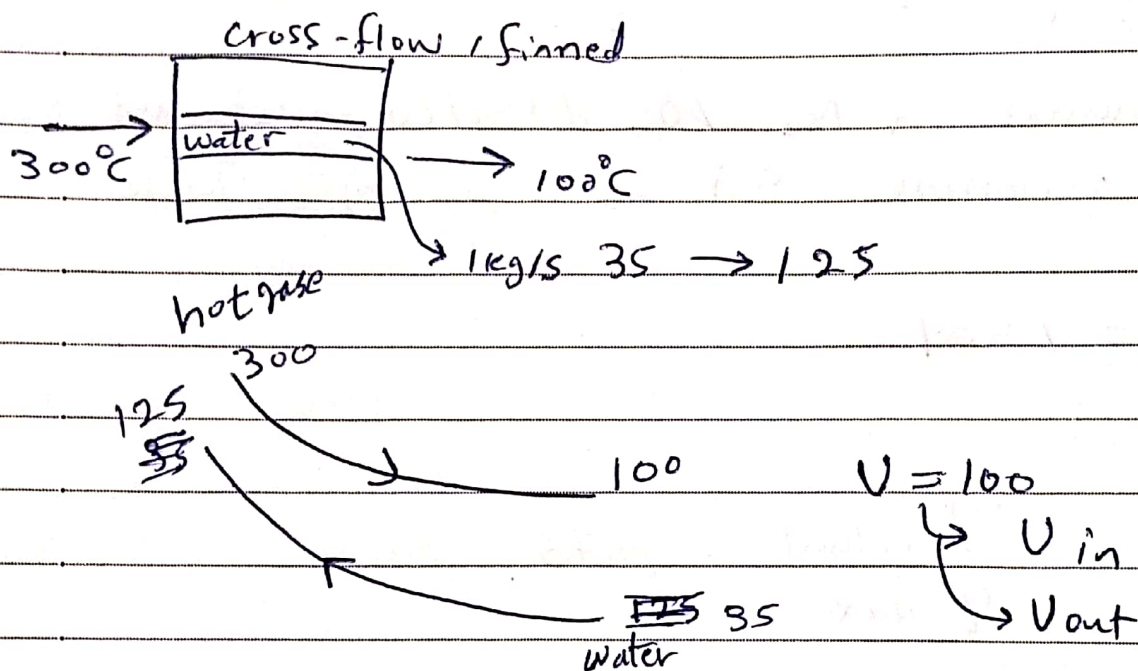
$$\epsilon = q_{\text{actual}} / q_{\max} = \frac{C_c (T_{co} - T_{ci})}{C_{\min} (T_{hi} - T_{ci})}$$

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

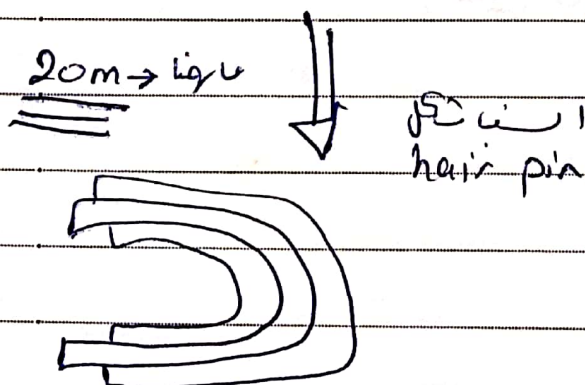
$\epsilon \Rightarrow$ chart / table



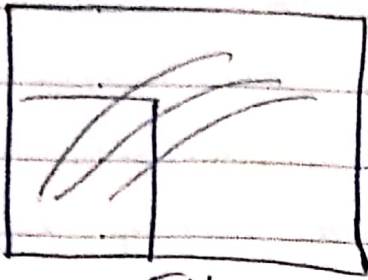
(Ex 1) Hot exhaust gases, which enter a finned-tube cross-flow heat exchanger at 300°C and leave at 100°C , are used to heat pressurized water at a flow rate 1 kg/s from 35 to 125°C . The overall heat transfer coeff. based on the gas-side surface area is $U_h = 100\text{ W/m}^2\cdot\text{K}$. Determine the required gas-side surface area A_h using the NTU method.



check the pressure drop



$\frac{20}{4} = 5$ hair pin.
 $\frac{20}{4} \rightarrow$ effective length.
 length of hair pin = 2m .



NTU

$$C_c = (m c_p)_c$$

$$C_h = (m c_p)_h$$

$$C_c \rightarrow \frac{125}{35} = 80 \xrightarrow{\text{Table}} 4197 \text{ J/kg}\cdot\text{K}$$

$$C_{c_{\text{max}}} = 1 \left(\frac{\text{kg}}{\text{s}} \right) \times 4197 \frac{\text{J}}{\text{kg}\cdot\text{K}} = 4197 \text{ W/K}$$

exhaust $\rightarrow \text{N}_2 / \text{O}_2 / \text{CO}_2 / \text{CO} \rightarrow \text{exhaust}$
 assume air $\rightarrow C_p(\text{air}) \rightarrow \text{Table}$

$$C_{h_{\text{min}}} = 1889$$

$$\epsilon = \frac{Q_{\text{actual}}}{Q_{\text{max}}} = 0.75$$

finned $\rightarrow$ unmixed $\rightarrow$ chart

سے سے، وہاں سے کیے کیے

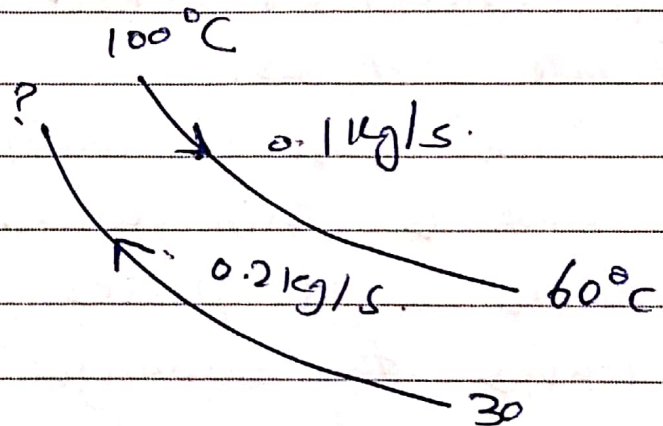
$$Q = UA \Delta T_{\log} \leftarrow \text{اے}$$

Concentric tube Heat exchanger

a Counterflow, concentric tube heat exchanger is used to cool the lubricating oil for a large industrial engine. The flow rate of cooling water through the inner tube ($D_i = 25\text{mm}$) is 0.2kg/s while the flow rate of oil through the outer annulus ($D_o = 45\text{mm}$) is 0.1kg/s . The oil and water enter at Temp. of 100 and 30°C . respectively. How long must the tube be made if the out-let Temp. of the oil is to be 60°C

1. energy balance

find L exist



2. $\Delta T = \dots$

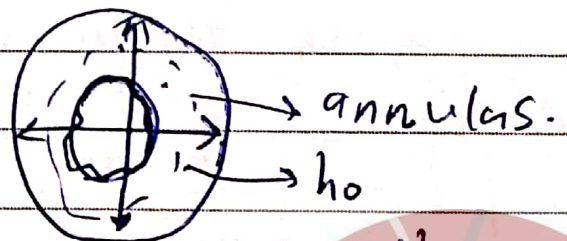
$$Q = U A \Delta T_{\text{LM}} = \pi D L \Delta T_{\text{LM}}$$

h_i h_o

chapter 8

Re $\rightarrow$ Turbulent
 $\rightarrow$ laminar

oil, water



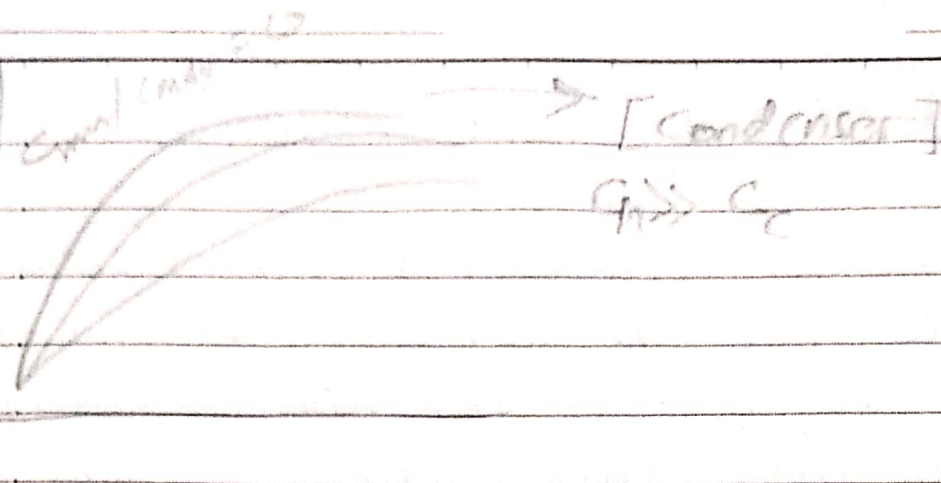
Correction!!

1) lubric equilibrium

oil, water

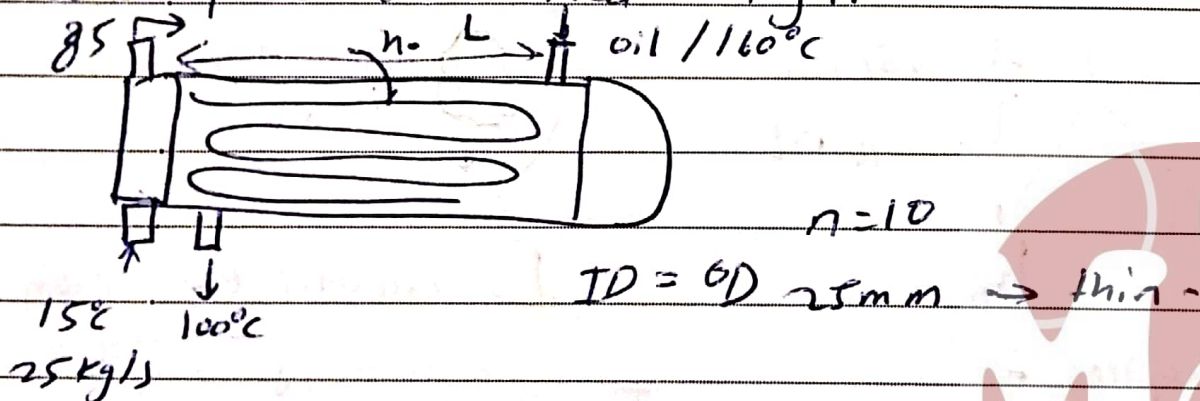
friction

heat



Q: A shell and tube heat exchanger must be designed to heat 2.5 kg/s of water from 15 to 35°C. The heating is to be accomplished by passing hot engine oil, which is available at 160°C, through the shell side of the exchanger. The oil is known to provide an avg. convection coeff. of $h_o = 400 \text{ W/m}^2\text{K}$ on the outside of tubes.

Ten tubes pass the water through the shell. Each tube is thin walled, of diam $D = 25 \text{ mm}$ and makes eight passes through the shell. If the oil leaves the exchanger at 100°C, what is the flow rate? How long must the tubes be to accomplish the desired heating?!

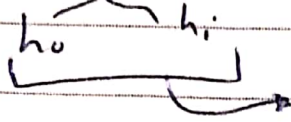


① $C_{p, \text{oil}}$ $C_{p, \text{water}}$ → from table

$$\dot{m}_o = \frac{2.5 \text{ kg/s} (35 - 15)}{C_{p, \text{oil}} (160 - 100)}$$

② ΔT_{lm} ③ $F \rightarrow [\text{Chart (1-2)}]$

④ $Q = U A F \Delta T_{lm} \rightarrow \pi D L \Delta T_{lm}$



ملا بينهم ما جلاه
كل الطورين هجاب
الوحدة 2.5 / 10

$A_i = A_o \rightarrow \frac{1}{U} = \frac{1}{h_i} + \frac{1}{h_o}$

Another Method!

[NTU]

$C_p > C_p \rightarrow$ check to know which min & Max

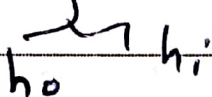
$\frac{Q}{C_{min}(100-15)}$



MFV

AV / C_{pmin}

AV



total $\rightarrow A_{flow} \times \text{كل}$

