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Heat Transfer

2022

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Madar
ask :: believe & recieve

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* modes of transfer heat :

1 Conduction

$$q = -kA \frac{dT}{dx}$$ Fourier's law

A ⇒ surface area (perpendecular on heat transfer direction)

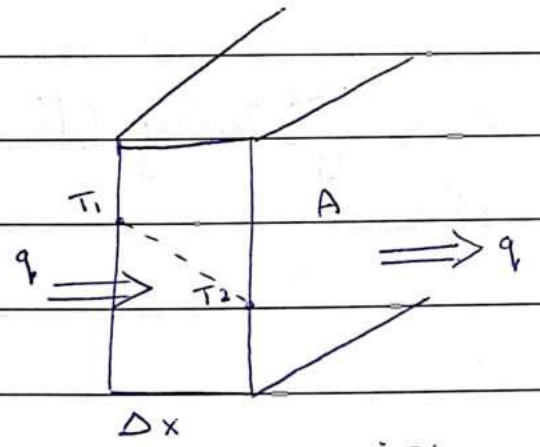
$\frac{dT}{dx}$ ⇒ temperature gradient

dT ⇒ driving force

dx ⇒ distance (thickness)

* ملاحظة : لا يطبق القانون إلا في حالة steady state condition

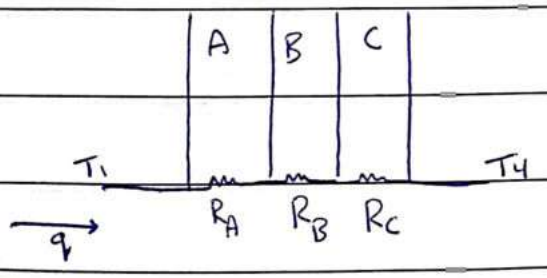
* الوسط الأكثر شيوعاً لانتقال الحرارة بهذا الأسلوب هو الـ solid ، وقد تنتقل أيضاً في الغاز والسائل ولكن بشرط أن يكون الـ fluid في حالة سكون (stagnant) .



heat flux ⇒ $q'' = \frac{q}{A} = -k \frac{dT}{dx}$

$$q = \frac{\Delta T}{dx / -kA} = \frac{\Delta T}{R}$$

* هذا الشكل مهم في حال وجود حائط تحتوي على أكثر من طبقة (composite wall)



$$q = \frac{\Delta T_{overall}}{\Sigma R}$$

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2 Convection

* الوسط المناسب لانتقال الحرارة هو ⇐ fluid
ولكن الشرط أن يكون المائع في حالة motion

يحدث بين جزيئات ال fluid من خلال عمليات ال mixing بين الساخن والبارد ويحدث بين سطح صلب والـ fluid

أنواع حركة ال fluid

bulk movement	diffusion
حركة كلية	انتشار

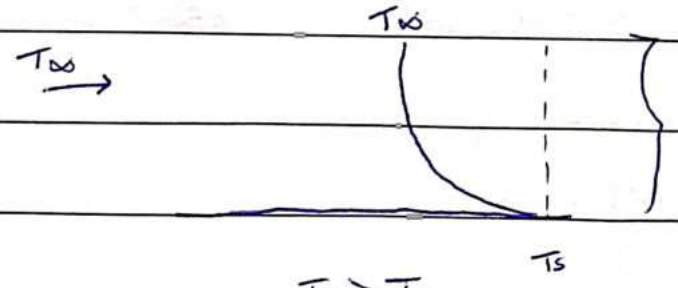
convection

forcing	natural (free convection)
حركة التيارات تكون غير طبيعية باستخدام مراوح (fan)	من ضمن التيارات الصاعدة والهابطة من غير تدخل لإنسان
driven by forced over a surface	Drive by density different

$q = hA\Delta T$ ⇒ Newton's law of cooling

$= \frac{\Delta T}{1/hA} = \frac{\Delta T}{R}$

h ⇒ heat transfer coefficient



Thermal boundary layer in convection heat transfer

U ... of a system.

* **Heat**: a type of energy 'thermal energy'.

* **Heat transfer**: energy in transit due to temperature difference.

$q = m C_p \Delta T$ } type of heat

$q \Rightarrow$ latent heat

↓ change T

↳ phase change & T not change

& phase not change

* **process heat transfer**: A study which focuses on the heat transfer during the physical or chemical process.

For examples:

heating of crude oil in heat exchangers & pipe still heater.

فمن كمونات البترول تتم سخونة البترول في heat exchanger

ثم يتم فصلها بواسطة distillation column.

* Relation between thermodynamics & heat transfer:

- first law: ما تخسره جسم من طاقة يكتسبه جسم آخر

وهذا يعني انتقال حرارة

A hot block	B cold block

الساخن سوف يفقد طاقة وبالتالي (يبرد)

البارد = يكتسب = (يسخن)

2nd - mar - 2022

* Temperature gradiant: change of temperature per distance ⇒ $\frac{dT}{dx}$

* Temperature distribution: It is very important to be known in order to compute the heat flow.

* Modes of heat transfer:
- Conduction
- Convection
- Radiation

q ⇒ latent heat

change T & phase not change

phase change & change

* process heat transfer: A study which focuses on the heat transfer during the physical or chemical process.

For examples:

heating of crude oil in heat exchangers & pipe still heater.

heat exchanger في مصفى كونات البترول تتم عملية تسخين البترول

ثم يتم فصلها بواسطة distillation column.

* Relation between thermodynamics & heat transfer:

- first law : ما تخسره جسم من طاقة يكتسبه جسم آخر

وهذا يعني انتقال حرارة

A hot block	B cold block

الساخن سوف يفقد طاقة وبالتالي (يبرد)

البارد = يكتسب = (يسخن)

- Second law: (اتجاه انتقال الحرارة بزيادة الـ entropy)

$S_{final} > S_{initial}$

* Heat transfer on radiation: (vaccume)

$q = \sigma A T_{ab}^4$ ⇒ black surface $\epsilon = 1$

$q = \epsilon \sigma A T_{ab}^4$ ⇒ actual surface

T_{ab} ⇒ above zero (absolute temperature)

الـ radiation يختلف بالقالب convection

ϵ emissivity (0 - 1)

color of surface — nature of surface

another color — black

0 - 1 — $\epsilon = 1$

$q'' = \epsilon \sigma (T_1^4 - T_2^4)$ ⇒ difference of temperature

* application of heat transfer:

1- Chemical Rxn: exothermic or endothermic, such as combustion.

2- Biological Rxn: like, cooling & freezing of food.

3- physical change: like, evaporation & Condensation e.g: distillation & Melting &

4- Power generation

5- Air conditioning & space heating.

6- Waste Heat recovery

7- insulation

8- Control of temperature

3rd - mar - 2020

* General heat transfer and applications :

- Heat rate, pipe lagging & wall insulations

process industry examples of process using heat:

1- Steel making
2- Chemicals
3- non metalic minerals
4- food & drink
5- paper & printing
6- Textiles

* أعلى نسبة لاستهلاك الطاقة الحرارية تقصد بـ :

- steel - chemicals

* Comparison between different heat transfer modes:

Conduction	Convection	Radiation
Material continuity	Material continuity	Possible in the absence of material continuity => (vaccume)
تحتاج لوسط	تحتاج لوسط	لا تحتاج لوسط
Motion not necessary	Depend on fluid motion (fluid dynamic)	Motion not necessary
In solid, liquid & gas	In liq & gas & multiphase mixture	In transparent media, mainly gases

* Thermal conductivity:

تختلف الموصلية باختلاف المادة ،

for metals ⇒ high conductivity

Insulation ⇒ low conductivity

gas ⇒ low conductivity

- Thermal conductivity function of temperature.

بعض المواد بزيادة الـ T تزداد conductivity ، وبعض المواد تقل الموصلية بزيادة الحرارة ، وبعضها يبقى ثابت بتغير درجة الحرارة .

* Conservation of energy : (first law)

surrounding , S

Control volume (CV) (system)

Boundary , B

(control surface , CS)

Addition through inlet $\dot{E}_{in}$ → -Accumulation (storage) $\dot{E}_{st}$ -generation $\dot{E}_g$ → loss through outlet $\dot{E}_{out}$

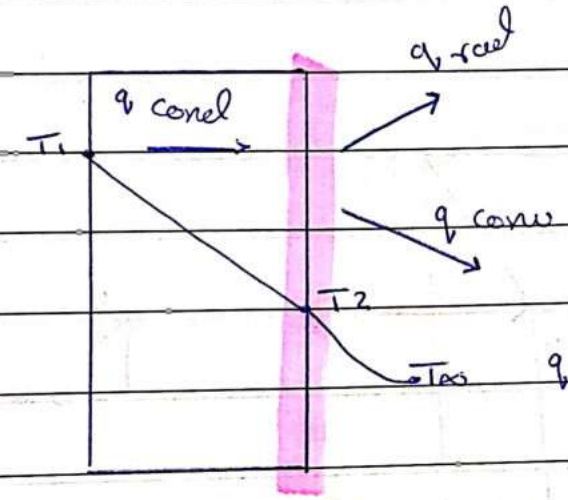
* energy conservation on a rate basis:

$$\dot{E}_{in} + \dot{E}_g - \dot{E}_{out} = \frac{dE_{st}}{dt} = \dot{E}_{st}$$

units $W = J/s$ storage

* Inflow & outflow are surface phenomena. ($\dot{E}_{in}$ & $\dot{E}_{out}$)
* generation & accumulation are volumetric phenomena. ($\dot{E}_g$ & $\dot{E}_{acc}$)

* Surface energy balance:
for a control surface



$\dot{E}_{in} - \dot{E}_{out} = 0$

or

$q_{cond} - q_{rad} - q_{conv} = 0$

* In general, the determination of h is a very complex problem because h is affected by:

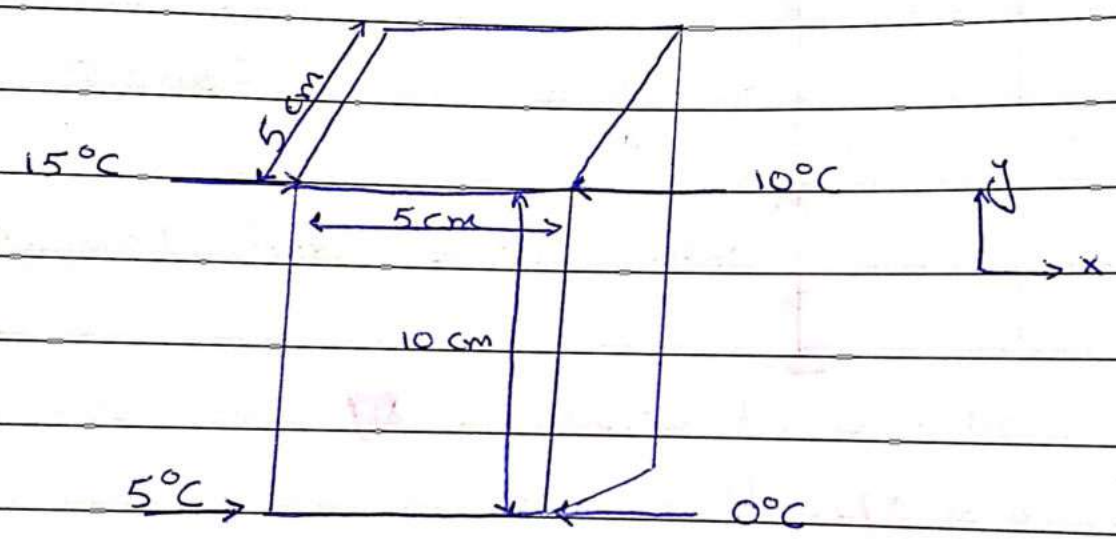
1. The type of flow, laminar turbulent or transition.
2. The geometery of the body
3. The physical properties of fluid.
4. The temperature difference.
5. The position along the surface of the body.
   تختلف قيمة h باختلاف الموقع على السطح ، لذلك يتم أخذ ال
   mean value over the surface
6. Mechanism is forced or free.

* Example 1 :

The block of 304 stainless steel ~~[illegible]~~ shown below is well insulated on the front & back surface, & the temperature in the block varies linearly in both x & y direction, find:

a- The heat fluxes & heat flows in the x & y- direction.

b- " magnitude & direction of the heat flux vector.



* مصطلحات :

quantities

vector
↓
magnitude & direction
↓
heat

scalar
magnitude only
↓
Temperature

solⁿ

heat ⊥ Area

$A_x = yz = 10\times10^{-2} \times 5\times10^{-2} = .005\ m^2$

$A_y = xz = 5\times10^{-2} \times 5\times10^{-2} = .0025\ m^2$

from table A.1 thermal conductivity of 304 stainless steel is 14.4 W/m.k

a) $q_x = -k \frac{dT}{dx}$

$= -14.4 \left(\frac{-5}{5\times10^{-2}}\right) = 1440\ W/m^2$

$\Delta T_x = \left(\frac{10+0}{2}\right) - \left(\frac{15+5}{2}\right) = -5$

average of T_2 - average of T_1

$q_y = -k \frac{dT}{dx}$

$= -14.4 \left(\frac{10}{10\times10^{-2}}\right) = -1440\ W/m^2$

$\Delta T_y = \left(\frac{10+15}{2}\right) - \left(\frac{5+0}{2}\right) = 10$

يدل على أن اتجاه انتقال الحرارة كان ~~خطأ~~ ~~ولهذا~~ ~~الحرارة~~

بالاتجاه المعاكس

تنتقل من الشمال للشرق

$q_x = q_x A = 1440 \frac{W}{m^2} \times .005\ m^2 = 7.2\ W$

$q_y = q_y A = -1440 \frac{W}{m^2} \times .0025\ m^2 = -3.6\ W$

b)

$\hat{q} = \hat{q}_x + \hat{q}_y$

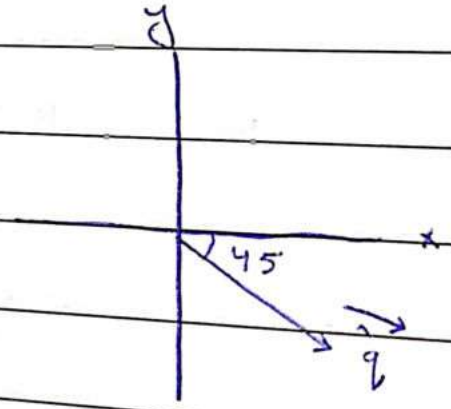
$\hat{q} = 1440i - 1440j$

$\hat{q} = \sqrt{(1440)^2 + (1440^2)} = 2036.5 \ W/m^2$

magnitude

* direction :

$\theta = \tan^{-1}\left(\frac{\hat{q}_y}{\hat{q}_x}\right) = \left(\frac{-1440}{1440}\right) = -45$



* ex : 2

A 5-cm (2 in) schedule 40 steel pipe carries a heat-transfer fluid and is covered with a 2-cm layer of calcium silicate insulation ($k = 0.06\ W/m \cdot K$) to reduce the heat loss. The inside and outside pipe diameters are 5.25 cm & 6.03 cm, respectively. If the inner pipe surface is at 150°C & the exterior surface of the insulation is at 25°C, calculate:

a) The rate of heat loss per unit length of pipe.

b) The temperature of the outer pipe surface.

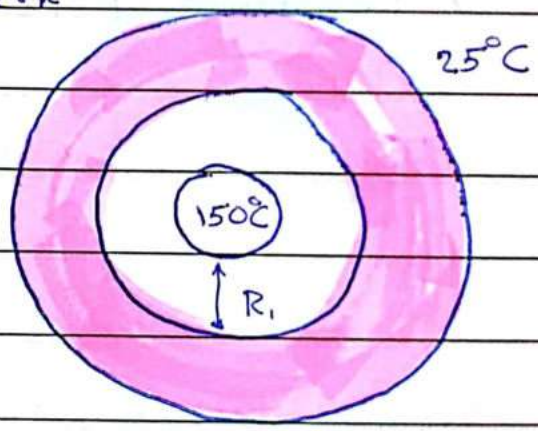
soln k for insulation = 0.06 W/m·K

$r_1 = .02625\ m$

$r_2 = .03015\ m$

$r_3 = .05015\ m$

k steel = 43 W/m·K (Table A.1)



$$R_1 = \frac{\ln(r_2/r_1)}{2\pi k L}$$

$$= \frac{\ln(.03015/.02625)}{2\pi(43)(1)}$$

$R_1 = .000513$

* critical radius

هو قطر المادة العازلة التي تبدأ عندها فقدان الحرارة يقل، ذلك يعني أن هذا القطر يوجد ضياع في الحرارة حتى مع وجود المادة العازلة ولكن عندما نصل إلى هذا القطر يقل معدل فقدان الحرارة

$$R_2 = \frac{\ln(r_3/r_2)}{2\pi k}$$

$$= \frac{\ln(.05015/.03015)}{2\pi(.06)} = 1.344723$$

$$q = \frac{\Delta T}{\Sigma R} = \frac{150-25}{1.344723 + .000513} = 92.576 \ W/m$$

b) $q = \frac{\Delta T}{R_1}$

$$92.576 = \frac{150 - T}{.000513} \Rightarrow T = 149.95°C$$

* Resistance of insulation higher than resistance of metal.

لابد الجزء من المادة العازلة هو منع فقدان الحرارة .

* ملاحظة أخرى :

قيمة الـ q المحسوبة سابقاً بناءً على وجود مقاومتين فقط ، ولكن يوجد نوع ثالث من المقاومات وهو مقاومة وجود فراغ بين المادة العازلة والـ pipe لا بد من غير الممكن أن تكون المادة العازلة محاطة بـ pipe بشكل مثالي لابد من وجود فراغات (يسمى هذا النوع من المقاومات (thermal contact

يؤثر على هذه المقاومة الـ (thermal contact) ، الـ pressure الموضوع
بداخل الـ system وخصائص الـ materials ، وتحسب هذه المقاومة
experimental .

فتصبح المقاومة الكلية :

$$R = R_{pipe} + R_{insulation} + R_{contact}$$

* Comments:

- It should be pointed out that the calculation in the previous examples tends to overestimate the rate of heat transfer because it assumes that the insulation is in perfect thermal contact with the pipe wall.

- Since solid surfaces are not perfectly smooth, there will generally be air gaps between two adjacent solid materials. Since air is a very poor conductor of heat, even a thin layer of air can result in a substantial thermal resistance.

* All modes of heat transfer require the existence of a temperature difference.

* All modes are from the heigh-temp medium to a lower-temp one.

* In gases & liquids conduction is due to the collision & diffusion of the molecules during their random motion.

* In solids conduction is due to the combination of vibrations of the molecules in a lattic & the energy transport by free electrons.

* fourier's law:

- Assumptions:

1- Homogeneous medium.

2- One-dimensional heat flow

$Q = -kA \frac{dT}{dx}$ ⇒ fourier's law of heat conduction

* when $k \uparrow$ ⇒ conduct material

* when $k \downarrow$ ⇒ insolator material

$Q = -kA \frac{\Delta T}{\Delta x}$ ⇒ special case from fourier's law.

when T ⇒ linear like:

$T = a + bx$

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

* negative sign ⇒ because heat is transferred in the direction of decreasing temperature.

* $k \propto \sqrt{T}$ ⇒ $k = f(T)$

* for most gases, at low P

$k = f(T)$ only

* At high P, close to critical P:

$k = f(T, P)$

* for liquids & Solids:

- بعض المواد تزداد قيمة k بزيادة درجة الحرارة ،

- بعض المواد تقل k بزيادة درجة الحرارة.

- بعض المواد قيمة k ثابتة بتغير درجة الحرارة.

* Thermal diffusivity:

$$\alpha = \frac{\text{Heat conduct}}{\text{Heat stored}} = \frac{k}{\rho C_p}$$

نسبة الطاقة المنتقلة على نسبة الطاقة المخزنة

* Conduction ⇒ fluid في سطح stagnant تحتاج إلى

* convection ⇒ motion fluid تحتاج إلى

* Radiation ⇒ تنتقل في الفراغ على شكل electromagnetic waves

* Newtons law (Convection):

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

T_s ⇒ T of surface

T_∞ ⇒ T of fluid

* Convection

- Nature → difference in density
- forced → external force applied on fluid

* Radiation:

- Is transport of electromagnetic light waves, similar to the transport of light.
- No physical medium is required, It can take place in a vaccum.

* المواد الصلبة والسائلة تميل إلى امتصاص الإشعاعات التي تنتقل من خلالها لذلك يحدث انتقال الإشعاعات عبر الغازات والفضاء.

* examples: Transfer of heat from the sun to the earth.

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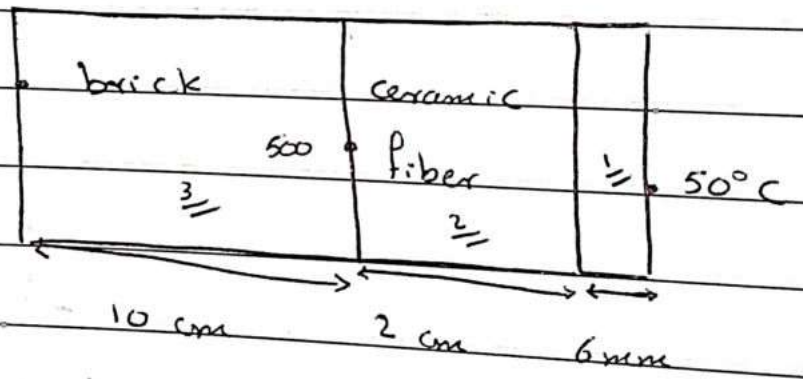
* ex 3:

A one square meter, 6 mm-thick steel furnace door (conductivity 30 W/m·k) is insulated on the inside by a 2-cm thick layer of ceramic fiber metting (conductivity 0.05 W/m·k) and a 10-cm thick layer of refractory brick (conductivity 1.0 W/m·k). If the temperature of the brick surface in the furnace is 700°C & the out side steel surface of the door is at 50°C, what is the heat loss by conduction through the door? If the maximum temperature of ceramic fiber were limited to 500°C, what would its thickness have to be and what effect would this have on heat loss?



Soln:

$k = 1.0$ $\quad k = .05$ $\quad k = 30\ W/m\cdot k$

$$R_1 = \frac{\Delta x_1}{kA} = \frac{6\times10^{-3}}{30\,(1)} = 2\times10^{-4}\ k/W$$

$$R_2 = \frac{\Delta x_2}{kA} = \frac{2\times10^{-2}}{.05\,(1)} = 0.4\ k/W$$

$$R_3 = \frac{\Delta x_3}{kA} = \frac{10\times10^{-2}}{1\,(1)} = 0.1\ k/W$$

$$q_{loss} = \frac{\Delta T}{\Sigma R} = \frac{700 - 50}{(2\times10^{-4} + .4 + .1)} \simeq 1300W$$

نهمل مقاومة ال steel لأنها صغيرة جداً مقارنةً بمقاومات ال insulators

* إذا لم يذكر السؤال قيمة ال area نفرضها $1m^2$
∴ assume Area = $1m^2$
* قيم ال resistance المواد ال insulation أكبر من قيم ال resistance ال metal.

* المطلوب رقم 2 : إيجاد ال thickness لطبقة ال ceramic في حال فرض ال max temperatur = 500 ، هذا يعني أن ال thickness المُعطى لدرجة حرارة غير ال 500°C ، بداية نجد درجة حرارة ال max عند ال thickness المُعطى

$$q = \frac{\Delta T}{R_2}$$

$$1300 = \frac{T_{max} - 50}{0.4}$$

* T_{max} of ceramic at thickness (.02m) = 570

نلاحظ أنها أكبر من T_{max} المذكورة في السؤال (500 °C) ، يعني بدي أشوف العلاقة بين درجة حرارة وال thickness

* إيجاد ال thickness | $T_{max} = 500$

* تغير ال thick يعني تغير ال resistance وهذا يعني تغير قيمة ال q لذلك بداية نجد قيمة ال q بين طبقتين brick & ceramic جزء من =500 ثم نجد ال thick

$$q = \frac{\Delta T \text{ between brick \& ceramic}}{R_3}$$

$$q = \frac{700 - 500}{.1}$$

$$q = 2000\ W$$

* فيه ↓ fric

$T_{max} = 500°C$

$$q = \frac{\Delta T}{\Delta x / kA}$$

$$2000 = \frac{500 - 50}{\Delta x} (.05 \times 1)$$

$$\Delta x = 1.125\ cm$$

$T_{max} = 500°C$ & q loss $= 2000W$

* Note of ceramic layer:

Max temp ↓ thick ↓ R ↓ Q ↑

* Extended Surface: (fins) أسطح زائدة

* الهدف منها زيادة مساحة السطح وبالتالي زيادة heat rate transfer

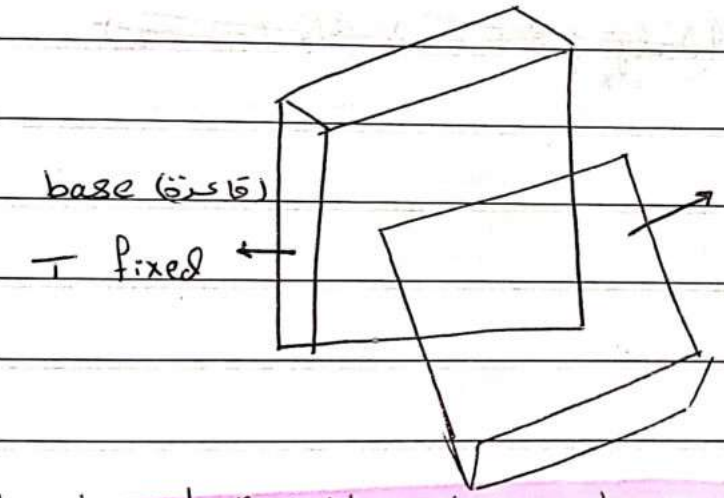
$q = hA\Delta T$

الهدف زيادة q ، يمكن زيادة q بإحدى الطرق:

- $h\uparrow$ $q\uparrow$ - ولكن قد تكون قيمة h ثابتة للمادة الواحدة أي لا يمكن زيادتها

- $\Delta T\uparrow$ $q\uparrow$ قد تكون ΔT ثابتة

- $A\uparrow$ $q\uparrow$ العامل المؤثر الأقوى ، حيث يتم عمل سطوح على سطح ثابت لزيادة الـ A وبالتالي زيادة الـ q.



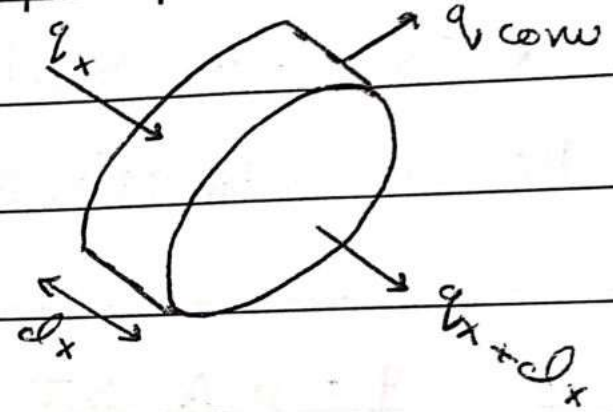
* معظم الـ extend مصنوعة من الـ metal لزيادة الـ heat

* تنتقل الحرارة من سطح الـ base لداخل الـ fine عن طريق conduction

ثم من سطح الـ fin للهواء radiation & convection

energy conservation:

$q_x = q_{x+dx} + q_{conv}$ --- 1



* q_x ⇒ conduction heat transfer

← انتقال الحرارة تم بين سطح ال base وسطح ال fin

* q_{x+dx} ⇒ دائماً الكمية الخارجة تساوي q_{in} + التغيير ال q

* The convection heat transfer rate may be expressed as:

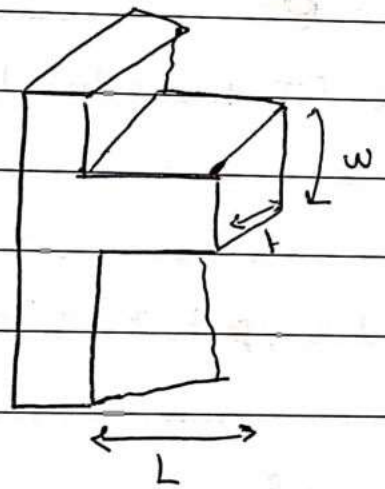
$$dq_{conv} = h\, dA_c\, (T - T_\infty)$$

* The temperature distribution:

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

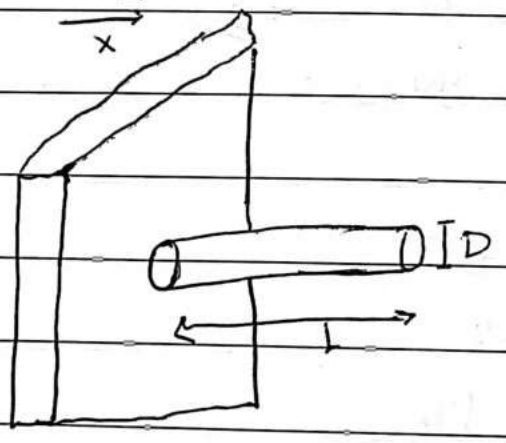
surface area

if cross section uniforms ⇒ Delete $\frac{dA_c}{dx}$ because $\frac{dA_c}{dx} = 0$



$P = 2w + 2t$ الحيط

$A_c = wt$



$P = \pi D$ الحيط

$A_c = \pi \frac{D^2}{4}$

$A_s = \pi D L$

* Surface area of cylinder $= \pi D L$

$= \pi D x$ ⇒ at any point in cylinder

$$A_s = P x$$

$$\frac{dA_s}{dx} = P$$

يتم تعويض قيمة $\frac{dA_s}{dx} = P$ في المعادلة رقم 1 ويتم حذف term dA_c لأن uniform cross تصبح المعادلة بالشكل التالي:

$$\frac{d^2T}{dx^2} - \frac{h}{A_c}\frac{1}{k} P (T - T_\infty) = 0 \quad -- 2$$

excess temperature $= T - T_\infty = \theta$

reference (T atmosphere)

$$m^2 = \frac{h}{A_c}\frac{P}{k}$$

يتم تعويض قيمهم في المعادلة رقم 2

$$\frac{d^2\theta}{dx^2} - m^2\theta = 0 \quad -- 3$$

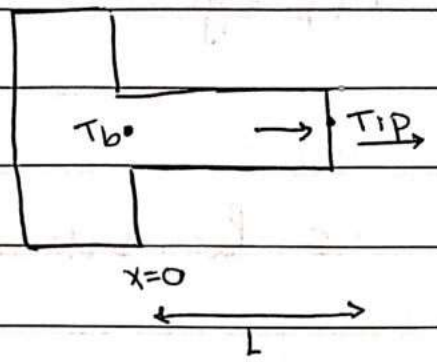
• general solution معادلة من الدرجة الثانية لها

$$\theta(x) = C_1 e^{mx} + C_2 e^{-mx}$$

⇓

Temperature profile

* لإيجاد قيمة C_1 & C_2 بالـ boundary conditions



$T_b = T_{base} = given$

بقى T عند الـ tip لها ٤ احتمالات :

1- Convection heat transfer:

~~T_b = fixed~~ ~~T_{tip} = given~~

بعمل Surface balance on the tip لإيجاد الـ boundary cond:

heat transfer by conduction = heat transfer by convection

$$E_{in} = E_{out}$$

$$-k A_c \frac{dT}{dx} = h A_c (T - T_\infty)$$

$$-k \frac{d\theta}{dx} = h \underset{(L)}{\theta} \Rightarrow$$ تسمى boundary condition for convection heat transfer

$$\frac{d\theta}{\theta} = \frac{h}{k} dx$$

بعمل تكامل وبعرف كم الـ T عند أي نقطة على الـ fins ولكن يوجد tables فيها

T profile ⇒ table 3.4

2 prescribied temperature

تكون قيمة T tip معطى

T_b = fixed T tip = given

3 adiabatic

بعمل : Surface balance

$E_{in} = E_{out}$

$-k A_c \frac{dT}{dx} = 0$

هذا يعني أن $\frac{dT}{dx} = 0$

4 Infinite length ($L \to \infty$)

at $L = \infty$ $T = T_\infty$ $\theta = T - T_\infty = 0$

تصبح درجة حرارة ال tip تساوي T_∞

* مثال : تطبيق ال boundary condition في فرض ال infinite length على ال general solution .

$\theta(x) = C_1 e^{mx} + C_2 e^{-mx}$

sol^n at $x=0$ $T = T_b$ $\theta(0) = T_b - T_\infty = \theta_b$

$\theta(b) = C_1 e^0 + C_2 e^0$

$\theta(b) = C_1 + C_2$ --- 1

B.C 2

at $x = L \to \infty$ $\quad T = T_\infty \quad \theta = 0 = T_\infty - T_\infty$

$$0 = C_1 e^{m\infty} + C_2 \cancel{e^{-m\infty}}$$

صغيرة جداً 0

$\therefore C_1 = 0$

نعوضها في معادلة

$$\theta_b = C_1 + C_2$$

$$\boxed{\theta_b = C_2}$$

نعوضهم في الـ general solution

$$\theta(x) = \theta_b e^{-mx}$$

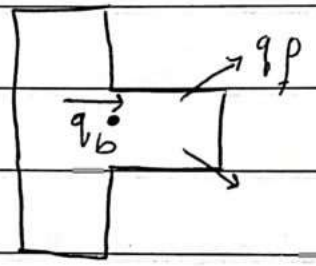
$$\frac{\theta(x)}{\theta_b} = e^{-mx}$$

هيا جدول جاهز لدايمي للاشتقاق

$$\Rightarrow \frac{T - T_\infty}{T_b - T_\infty} = e^{-mx} \Rightarrow \text{temperature distribution}$$

table 3.4 $\Leftarrow$: fine لـ q الخارجة من الـ

بناءً على قانون حفظ الطاقة $q_f = q_b$



$q_{in} = q_{out}$

q_b = heat transfer by conduction between base & fine

q_f = heat transfer from fine

$\Rightarrow$

هنا شرح الاشتقاق $\Leftarrow$ بوجد table 3.4

فيه q_f & T distribution

$$q_b = q_f = -kA\frac{dT}{dx}$$

$$q_b = q_f = -kA\frac{d\theta}{dx}\Big|_{x=0}$$

$$q_f = -kA\frac{d\,\theta_b e^{-mx}}{dx}$$

$$\cancel{\theta_x = \theta_b m}$$

$$\underbrace{\theta_x = \theta_b e^{-mx}}_{\text{at base}}$$

$$q_f = kAm\,\theta_b \cancel{e^{-mx}}^{\,1}\Big|_{x=0}$$

$$q_f = kA\,\theta_b\left(\frac{kP}{kA_c}\right)^{\frac{1}{2}}$$

$$q_f = \sqrt{kA_c kP}\;\theta_b$$

* fine performance: يريد كفاءة ال fines في نقل الحرارة

- fine efficiveness ⇐ تم وضع معيار وهو

ε : It is defined as the ratio of the fine heat transfer rate to the heat transfer rate that would exist without the fine.

$$\varepsilon = \frac{q_f}{kA_{c,b}\,\theta_b}$$

$A_{c,b}$ ⇒ Is the fine cross-sectional area at the base

النسبة بين q_f أي بوجود ال fine و q من غير وجود fine

A_c ⇒ مساحة قاعدة ال fine الموجودة على ال base

* مثال لإيجاد ε :

* Assume infinite length :

$L \to \infty$

$q_f = M = \sqrt{hPkA_c}\ \theta_b$

$\varepsilon = \dfrac{\sqrt{hPkA_c}\ \cancel{\theta_b}}{h A_{c,b}\ \cancel{\theta_b}} = \left(\dfrac{kP}{hA_c}\right)^{\frac{1}{2}}$

- بزيادة قيمة k تزداد ε ، ويمكن زيادة قيمة k بـ metal
- كلما قلت h تزداد الـ ε
- كلما زاد نسبة $\frac{P}{A_c}$ تزداد قيمة ε ، نزيد $\frac{P}{A_c}$ بزيادة طول الـ fin وتقليل الـ A_c .

* Notes :

1- k should be as high as possibole . (copper , aluminum & iron) . (metals)

2- P/A_c should be as high as possibole . (Thin plate fins & slender pin fins) .

3- Most effective in applications where h is low . (Use of fins justified if when the medium is gas & heat transfer is by natural convection).

4- max heat rate could be achieved by using very long fins . However , it is not reasonable to use very long fins to achieve near max . heat transfer .

* من طرق زيادة قيمة ε زيادة النسبة $\frac{P}{A_c}$ التي تتحقق بزيادة length of fins

ولكن زيادة الطول بشكل كبير غير عملي ، وسوف نصل إلى طول معين بعد هذا الطول لن تتغير قيمة الحرارة (تثبت) ⇐ تثبت قيمة الـ ε .

* How to obtain a reasonable length ?

بافتراض أن at length → ∞ ⇐ تصبح درجة الحرارة at tip
مساوية لدرجة حرارة الجو المحيط ، هذا يعني أنه لم تنتقل حرارة
من ال fin ⇐ tip of adiabatic tip fin

ومن الجدول نجد قيمة q_f at adiabatic

$$q_f = \sqrt{h P k A_c}\ \theta_b \tanh mL$$

Assume 98% of the max possible heat transfer

$$q_{f,max} = M = (h P k A_c)^{\frac{1}{2}}\ \theta_b$$

$$.98\ q_{f,max} = q_{f,adiabatic}$$

$$.98\ \cancel{(h P k A_c)^{\frac{1}{2}}\ \theta_b} = \cancel{(h P k A_c)^{\frac{1}{2}}\ \theta_b} \tanh mL$$

$$\Rightarrow mL = 2.3$$

* conclusion :

It is more suitable to use fin with $L = 2.3/m$ which yield 98% max possible heat transfer rather than to use $L > 2.3/m$ or infinite length.

Conclusion

If $\varepsilon > 1$ adding fins enhance heat transfer

$\varepsilon = 1$ adding fins has no effect

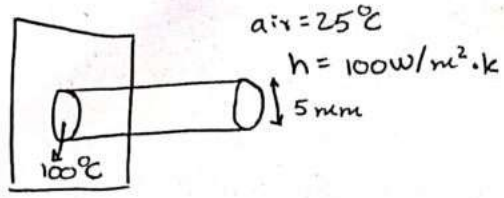
$\varepsilon < 1$ adding fins decrease heat transfer (as insulator

Problem 1 _ Fins

infinte length

A very long rod 5 mm in diameter has one end maintained at 100 °C. The surface of the rod is exposed to ambient air at 25 °C with a convection heat transfer coefficient of 100 W/m² K.

1. **Determine the temperature distributions along rods constructed from pure copper,** 2024 aluminum alloy, and type AISI 316 stainless steel. What are the corresponding heat losses from the rods?
2. **Estimate how long the rods must be for the assumption of *infinite length to yield an*** accurate estimate of the heat loss.



very long = infinite length

sol^n : مكونات ال rod

* pure copper & 2024 aluminum alloy & AISI 316 stainless steel

1. Determine the Temperature distribution & heat losses from the rods.

- Temperature distribution from table 3.4 for infinite length:

$$\frac{\theta}{\theta_b} = e^{-mx}$$

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

$$T = e^{-mx}(T_b - T_\infty) + T_\infty$$

- heat losses from table 3.4 for infinite length:

$$q = M = \sqrt{hPkA_c}\ \theta_b$$

جميع ال properties ثابتة لجميع مكونات ال rods ما عدا قيمة k تختلف باختلاف ال metals ، لذلك يتم حساب q لكل metal :

* k (pure copper) ⇒

interpolation A.1

$$\frac{393 - 413}{400 - 200} = \frac{x - 413}{335 - 200}$$

$k|_{335°k}$ copper = 398 W/m.k

نحسب قيمة k عند درجة حرارة 335 k

$$\frac{100 + 25}{2} = 62.5\,°C \simeq 335\,k$$

$$P = \pi D = .0157\ m$$

$$A_c = \pi \frac{D^2}{4} = .00001963\ m^2$$

q_f copper heat transfer $= \sqrt{100 \times .0157 \times 398 \times .00001963}\ (100 - 25)$

① q_f copper = 8.3 W

② q_f 2024 aluminum $= \sqrt{100 \times .0157 \times 180 \times .00001963}\ (100 - 25) = 5.6\ W$

③ q_f 316 stainless steel $= \sqrt{100 \times .0157 \times 14 \times .00001963}\ (100 - 25) = 1.6\ W$

2 estimate how long the rods must be for the assumption of infinte length?

$$.99\ q_{f\,max} = q_{f\,adiabatic}$$

$$.99\ (hPk\cancel{A_c})^{\frac{1}{2}}\ \cancel{\theta_b} = (hPk\cancel{A_c})^{\frac{1}{2}}\ \cancel{\theta_b}\ \tanh mL$$

$$.99 = \tanh mL$$

$$.99 = \frac{e^{mL} - e^{-mL}}{e^{mL} + e^{-mL}}$$

$$\vdots$$

$$1 + .99 = (e^{mL})^2 (1 - .99)$$

$$1.99 = e^{2mL} (.01)$$

$$199 = e^{2mL}$$

$$\ln 199 = 2mL \ln e$$

$$5.2933 = 2mL$$

$$\boxed{mL = 2.65}$$

$$L \text{ for copper} = 2.65 \left(\frac{kA_c}{hP}\right)^{\frac{1}{2}} = 2.65 \left(\frac{398 \times .00001963}{100 \times .0157}\right)^{\frac{1}{2}} = .19\ m$$

$$L \text{ for aluminum} = 2.65 \left(\frac{180 \times .00001963}{100 \times .0157}\right)^{\frac{1}{2}} = .13\ m$$

$$L \text{ for stainless steel} = 2.65 \left(\frac{14 \times .00001963}{100 \times .0157}\right)^{\frac{1}{2}} = .04\ m$$

* كيف يمكن التخلص من ال tanh !؟

$$x = \tanh y$$

$$x = \frac{e^{y} - e^{-y}}{e^{y} + e^{-y}}$$

$$\vdots$$

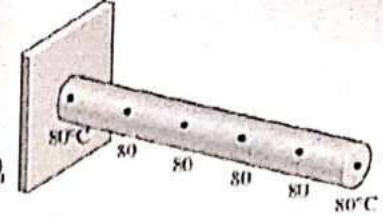
$$1 + x = (e^{y})^2 (1 - x)$$

Efficiency of Fins, η_f

- Definition:

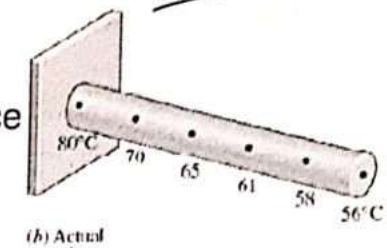
(0 - 1) $$\eta_f = \frac{q_f}{q_{max}} = \frac{q_f}{hA_f\theta_b}$$

يتم تقديرها من الـ tables



(a) Ideal

Ideal (Isothermal)



(b) Actual

where A_f is the surface area of the fin.

Look! Max heat transfer takes place when the surface temp. of the fin equals the base temperature.

Assume adiabatic tip fin, the previous eq. becomes

$$\eta_f = \frac{M\tanh mL}{hPL\theta_b} = \frac{\sqrt{hPkA_c}\,\theta_b\tanh mL}{hPL\theta_b} = \frac{\tanh mL}{mL}$$

$\eta_f \rightarrow$ max as $L \rightarrow 0$; $\eta_f \rightarrow$ min as $L \rightarrow \infty$

$\eta \uparrow$ when $L \downarrow$ والعكس صحيح

$q_{max} \Rightarrow$ at uniform T يتم حسابها وبالتالي $T - T_\infty$ أكبر من

الـ actual على الـ actual يكون في تقريب بدرجة الحرارة مع x

Approximation for heat transfer from a convection tip fin

- The heat transfer, q_f, of a convection tip fin; eq. 3.72, can be calculated via using adiabatic tip eq.3.76 by making a correction for the length; L_c=L+(t/2) for a rectangular fin and L_c=L+(D/4) for a pin fin.
- Therefore, with tip convection, the fin heat transfer rate may be approximated as

$$q_f = M\tanh mL_c$$

من الـ table هي نفسها معادلة الـ q adiabatic ولكن بسبب استبدال L_c بدل L نحصل على الـ convection

where $M = \sqrt{hPkA_c}\,\theta_b$

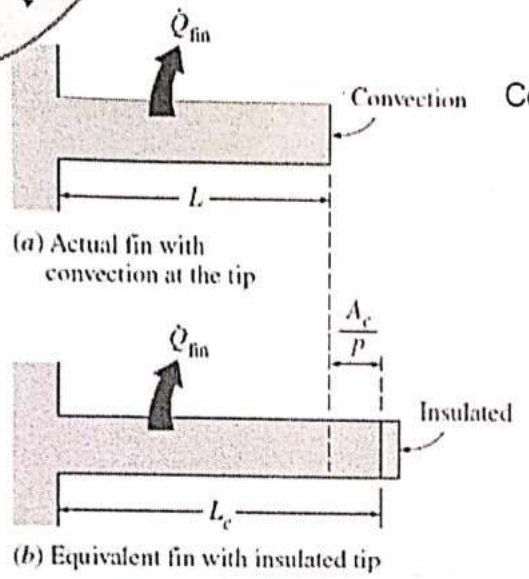
and $$\eta_f = \frac{\tanh mL_c}{mL_c}$$

* أخذنا من عمل correction for the length لنحصل على قيمة q_f في حالة الـ convection tip من تطبيق معادلة الـ adiabatic.

* عندما نعوض L_c في معادلة $q_{adiabatic}$ نحصل على $q_{convection}$

SPECIAL NOTE

Corrected fin length



(a) Actual fin with convection at the tip

(b) Equivalent fin with insulated tip

Corrected fin length: $L_c = L + \frac{A_c}{P} = L + \frac{\frac{\pi D^2}{4}}{\pi D}$

$\boxed{L_c = L + \frac{D}{4}}$

Multiplying the relation above by the perimeter gives

$A_{corrected} = A_{fin\ (lateral)} + A_{tip}$

المساحة الجانبية

$L_{c,rectangular fin} = L + \frac{t}{2}$

$L_{c,cylindrical fin} = L + \frac{D}{4}$

Corrected fin length L_c is defined such that heat transfer from a fin of length L_c with insulated tip is equal to heat transfer from the actual fin of length L with convection at the fin tip.

Notes

1. Errors associated with the approximation are negligible if (ht/k) or (hD/2k) ≤0.0625

معيار لتقدير تسبب الخطأ في approximation

إذا كانت النسبة أقل من 0.625 تقريباً لا يوجد error

أما إذا كان أكبر من 0.625 يوجد error ولازم أخذه بعين الاعتبار

2. If w >> t for rectangular fin

Pin fin

Rectangular fin

∴ P ≈ 2w

المفروض P=2w+2t ولكن t >> w لذلك نهمل t

$$mL_c = \left(\frac{hp}{kA_c}\right)^{1/2} L_c = \left(\frac{h2w}{kwt}\right)^{1/2} L_c = \left(\frac{2h}{kt}\right)^{1/2} L_c$$

Introducing a corrected fin profile area, $A_p = L_c t$

$$\therefore\ mL_c = \left(\frac{2h}{kt}\right)^{1/2}\left(\frac{L_c}{L_c}\right)^{1/2} L_c = \left(\frac{2h}{ktL_c}\right)^{1/2} L_c^{3/2} = \left(\frac{2h}{kA_p}\right)^{1/2} L_c^{3/2}$$

عند ضربهم × بـ $\frac{L_c^{1/2}}{L_c^{1/2}}$

See Figure 3.18 and figure 3.19 {η vis $\left(\frac{h}{kA_p}\right)^{1/2} L_c^{3/2}$ }

Fig. is developed from this eq. $\eta_f = \frac{\tanh mL_c}{mL_c}$

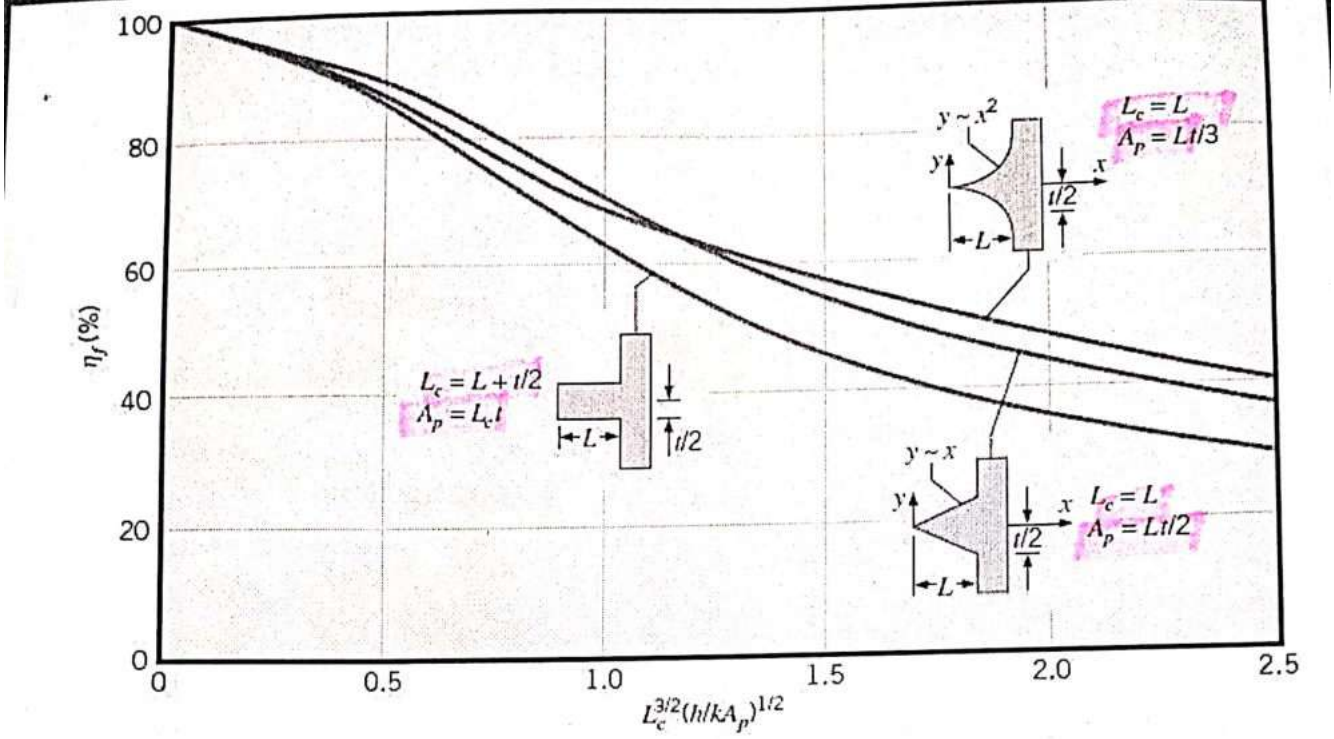


FIGURE 3.18 Efficiency of straight fins (rectangular, triangular, and parabolic profiles).

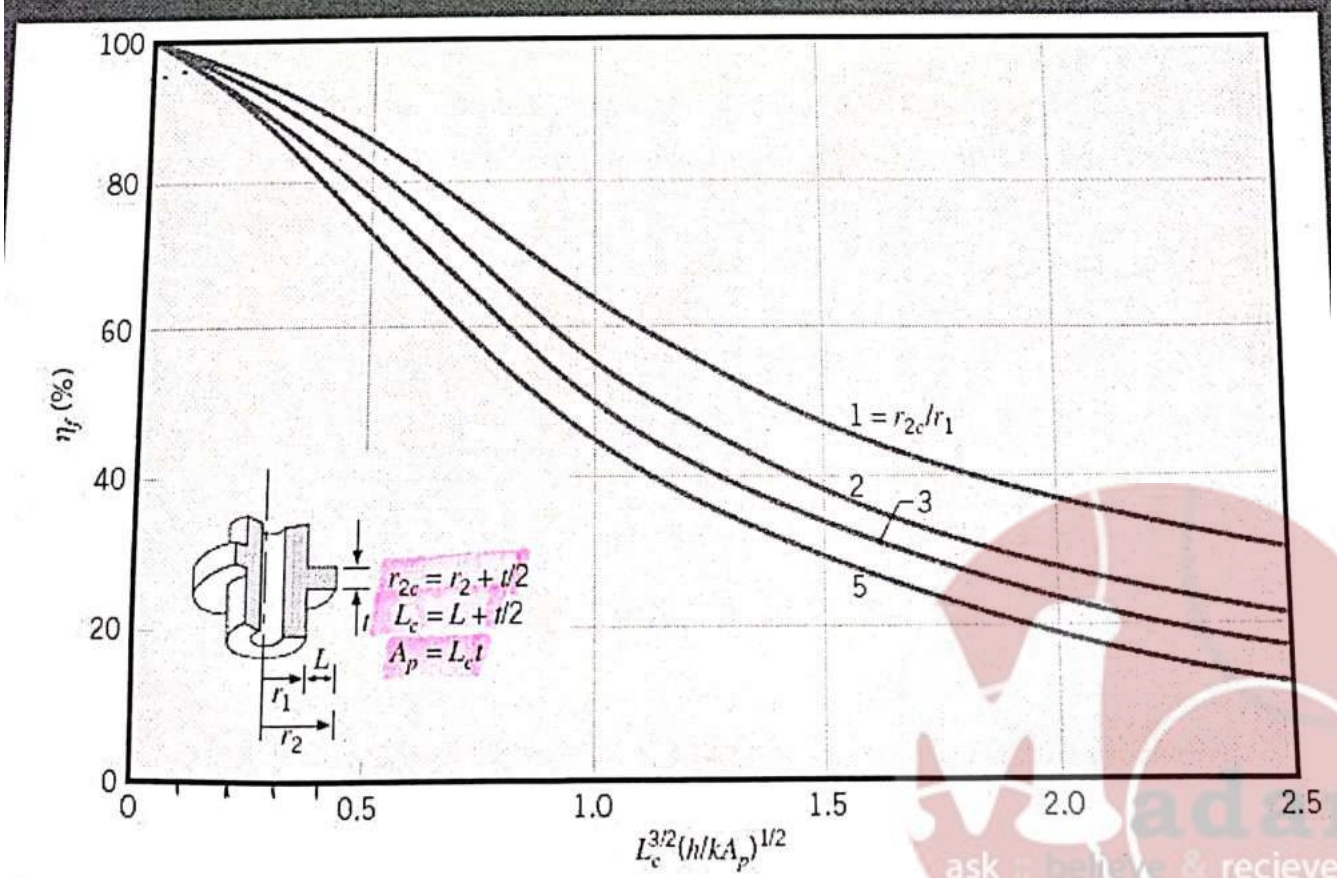


FIGURE 3.19 Efficiency of annular fins of rectangular profile.

TABLE 3.5 E[illegible]

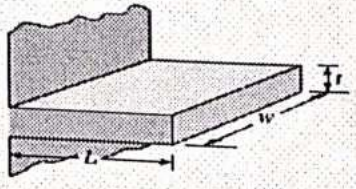
Straight Fins

Rectangular[a]

$A_f = 2wL_c$

$L_c = L + (t/2)$

$A_p = tL$

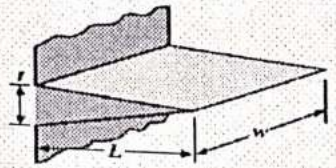


$$\eta_f = \frac{\tanh mL_c}{mL_c} \tag{3.89}$$

Triangular[a]

$A_f = 2w[L^2 + (t/2)^2]^{1/2}$

$A_p = (t/2)L$



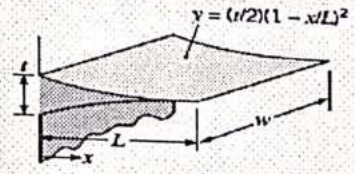
$$\eta_f = \frac{1}{mL}\frac{I_1(2mL)}{I_0(2mL)} \tag{3.93}$$

Parabolic[a]

$A_f = w[C_1L + (L^2/t)\ln(t/L + C_1)]$

$C_1 = [1 + (t/L)^2]^{1/2}$

$A_p = (t/3)L$



$$\eta_f = \frac{2}{[4(mL)^2 + 1]^{1/2} + 1} \tag{3.94}$$

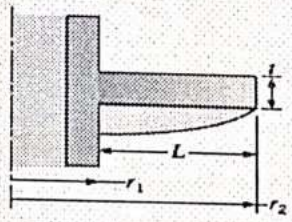
Circular Fin

Rectangular[a]

$A_f = 2\pi(r_{2c}^2 - r_1^2)$

$r_{2c} = r_2 + (t/2)$

$V = \pi(r_2^2 - r_1^2)t$



$$\eta_f = C_2\frac{K_1(mr_1)I_1(mr_{2c}) - I_1(mr_1)K_1(mr_{2c})}{I_0(mr_1)K_1(mr_{2c}) + K_0(mr_1)I_1(mr_{2c})} \tag{3.91}$$

$$C_2 = \frac{(2r_1/m)}{(r_{2c}^2 - r_1^2)}$$

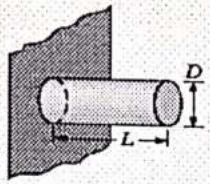
Pin Fins

Rectangular[b]

$A_f = \pi DL_c$

$L_c = L + (D/4)$

$V = (\pi D^2/4)L$

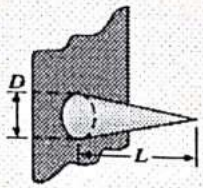


$$\eta_f = \frac{\tanh mL_c}{mL_c} \tag{3.95}$$

Triangular[b]

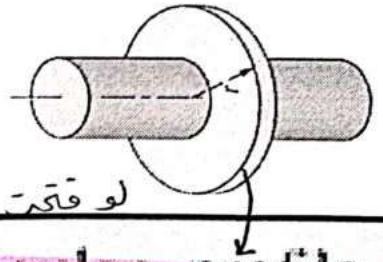
$A_f = \frac{\pi D}{2}[L^2 + (D/2)^2]^{1/2}$

$V = (\pi/12)D^2L$



$$\eta_f = \frac{2}{mL}\frac{I_2(2mL)}{I_1(2mL)} \tag{3.96}$$

Problem_2 Fin



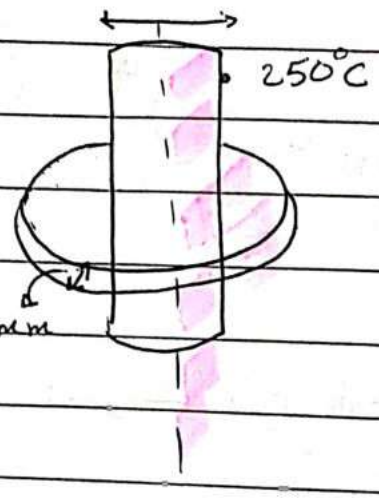
لو فتحت ال annular هي بيطلعلي شكل مستطيل

An annular aluminum fin of rectangular profile is attached to a circular tube having an outside diameter of 25 mm and a surface temperature of 250°C. The fin is 1 mm thick and 10 mm long, and the temperature and the convection coefficient associated with the adjoining fluid are 25°C and 25 $W/m^2 \cdot K$, respectively.

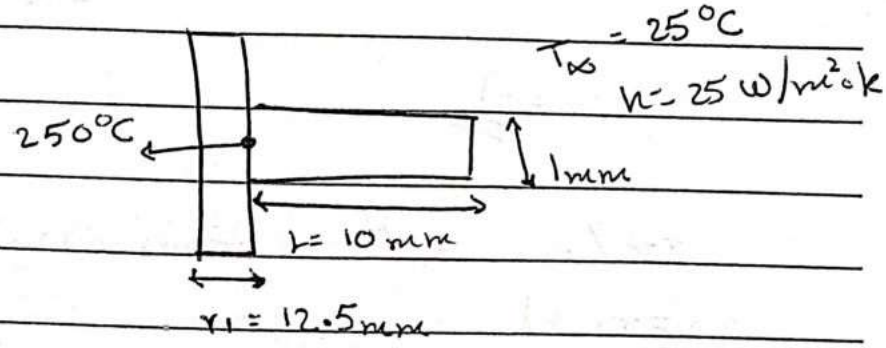
(a) What is the heat loss per fin?

Subject :

D = 25 mm



* بدي آخذ مقطع من ال circular tube



إذا فتحت ال annular الشكل بيطلع مستطيل

سطح ال tube هو نفس ال base

∴ Tb = 250°C

∴ ال thickness = width of rectangular

length of annular = length of rectangular

من ال chart (3.19) بدي أحسب المقدار الموجود على x-axis ثم بجيب ال η وبعدها بضرب ال $\eta \times q_{max}$ وبحصل على مقدار q المطلوب.

$$x\text{-axis} \Rightarrow L_c^{3/2} \left(\frac{h}{k A_p}\right)^{1/2}$$

$$L_c = L + \frac{t}{2} = 10\times10^{-3} + \frac{1\times10^{-3}}{2} = 10.5\times10^{-3}\ m$$

$$A_p = L_c\, t = (10.5\times10^{-3})(1\times10^{-3}) = 10.5\times10^{-6}\ m^2$$

$k = 240\ W/m\cdot k$

$h = 25\ W/m^2\cdot k$

Subject :

$$x\text{-axis} \Rightarrow L_c^{\frac{3}{2}} \left(\frac{h}{kA_p}\right)^{\frac{1}{2}}$$

$$= (10.5\times10^{-3})^{\frac{3}{2}} \left(\frac{25}{240\times10.5\times10^{-6}}\right)^{\frac{1}{2}}$$

$$= 0.1$$

لمعرفة أي المنحنيات نستخدم نظهر r_{2c}/r_1

$$r_2 = r_1 + L$$

$$r_2 = (12.5+10)\times10^{-3} = 22.5\times10^{-3}\,m$$

$$r_{2c} = r_2 + \frac{t}{2}$$

$$r_{2c} = 22.5\times10^{-3} + \frac{1\times10^{-3}}{2} = 23\times10^{-3}\,m$$

$$\frac{r_{2c}}{r_1} = \frac{23\times10^{-3}}{12.5\times10^{-3}} = 1.84$$

ومن الـ chart نجد قيمة $\eta = .97$

$$q_f = \eta\, q_{max}$$

$$\cancel{= \eta\, h A_f\, \theta_b}$$

$$\cancel{= .97(25)}$$

$$= \eta\, h A_f\, \theta_b$$

$$= \eta\, h\, 2\pi (r_{2c}^2 - r_1^2)\, \theta_b$$

$$= .97(25)(2\pi)(23^2 - 12.5^2)\times10^{-6}\,(250-25)$$

$$q_f = 12.8\,W$$

ضربت بـ 2 لأن 2 surfaced annular وقانون الـ A_f يعني مساحة السطح (πr^2) لذلك ضربت بـ 2 ليشمل السطحين.

13th - mar - 2022

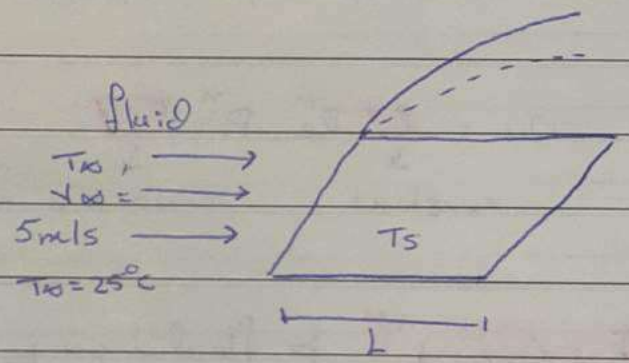
* Forced convection:

- In external flow
- In pipes & Ducts

* أهم قانون في الـ forced convection هو ⇐ Newton's law

* forced convection in external flow:

أسهل تطبيق في حركة الموائع على سطح صلب هو plate surface



الخطوط تدل على أن الـ fluid uniform يمر على السطح بـ T & V

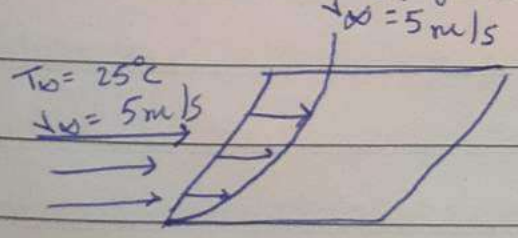
* ماذا يحدث عند مرور الـ stream فوق الـ flat surface ؟

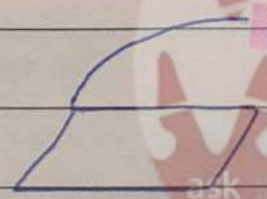
1. heat transfer (hydrodynamic)
2. momentum transfer (velocity)
3. mass

هي فقط تلتزم في هذا الـ coarse

* لما يمر على الـ plate ⇐ 2 boundary layer :

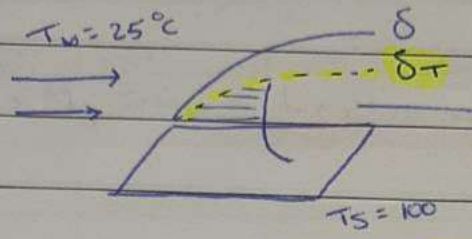
نظرياً الـ fluid مع plate بسرعة = 0 ، ثم تزداد سرعة الـ fluid بالاتجاه y لتصل إلى V_∞ (أي السرعة التي دفع بها الـ fluid)





$\delta \rightarrow$ boundary layer thickness

* boundary layer : هي المسافة التي يحدث فيها تغير (T or v)



هذه المسافة فقط التي يحدث فيها تغير لـ T ، فوق δ_T تثبت درجة الحرارة (T_∞)

* يستخدم ٣ معادلات بشكلهم وكلهم للوصول إلى الـ Correlation التالي ⇐

$$Nu = C\, Re^{m}\, Pr^{n}\, (F)$$

C ↓ constant

(F) ↓ correction factor

$F = \left(\frac{\mu}{\mu_s}\right)^{0.14}$ ⇐ يتعلق بالـ viscosity ويعتمد على نوع الـ fluid وقد تصل قيمة F إلى 1

* الطريقة الثانية للحصول على الـ Correlation هو عن طريق التجربة ⇐ (empirical)

* المعادلات المستخدمة في الحصول على Nu :

- momentum balance
- Temperature balance
- mass balance

* الهدف من دراسة ال Nu :

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

∴ الهدف هو معرفة قيمة h من قيمة Nu :

convection heat transfer coefficient

* factors effect on h value:

1) property of fluid
2) Type of flow

laminar Turbulent

3) geometry

*Notes:

- The heat transfer coefficient is obtained from the following correlations:

where:

$$Nu = \frac{hL}{k} \quad \& \quad Re = \frac{\rho L V_\infty}{\mu}$$

قد تكون length وقد تكون diameter حسب ال phase

- Nusselt number (Nu): It the ratio of convective to conductive heat transfer at a boundary in a fluid.
- Reynolds number: It the ratio of inertial forces to viscous forces within a fluid.

* أهم شي في ال design هو اختيار ال properties بشكل صحيح.

* In these equations all fluid properties are evaluated at the film temperature, T_f, defined by:

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

لازم أميز بين المعادلات اللي تحتوي على T_∞ أو T_f، لأنه لازم اختار ال properties المستخدمة في المعادلات بناءً على قيمة T الموجودة في المعادلة، إذا كانت T_∞ باخذ جميع ال properties at T_∞

أما إذا كانت T_f باخذ جميع ال properties at T_f

* لما أشوف معادلة تحتوي على T_∞ لازم أعمل correction

- Hence, the rate of heat transfer between the plate & the fluid is given by:

$$q = hA(T - T_\infty)$$

* الخلاصة:

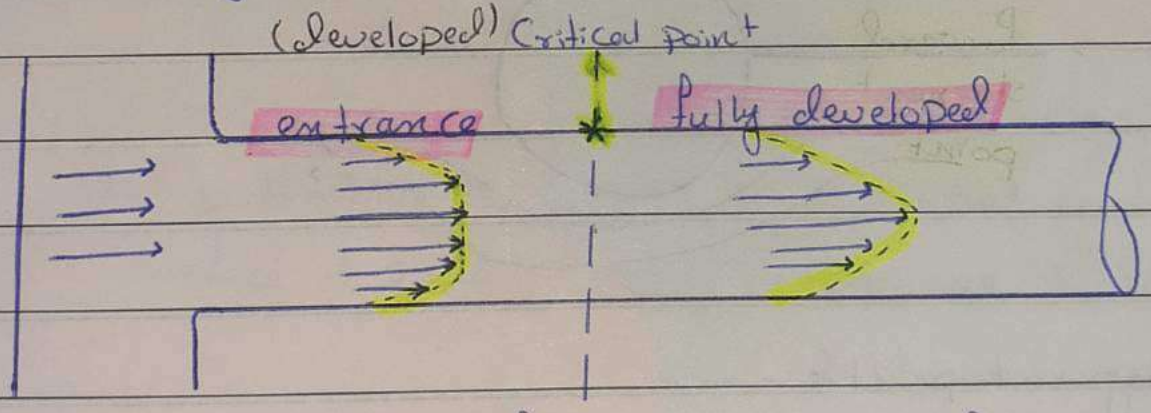
بدي أحسب قيمة ال q:

أول شي بعمل check على reynolds number. حدد نوع التدفق، بعدين حسب قيمة h من ال Nu وركز انتبه لدرجة الحرارة المستخدمة في المعادلات قد تكون (T_∞ or T_f) لأن الجزيئات تنتقل من T إلى الأخرى ثم بطبق بقانون Newton's law ← $q = hA(T - T_\infty)$

* معادلات Nu موجودة بـ tables ، تختلف من بعضها باختلاف الـ regeme ، يعني نحدد الـ regeme من Re ثم نحدد معادلة Nu من الجداول ، ثم نحسب قيمة h من Nu ثم نطبق Newton's law ونحسب قيمة q

* أثناء تدفق fluid داخل pipe ، بيكون منطقتين :

1. Entrance (developed)
2. fully developed



velocity profile is changing

velocity profile doesn't change

↓

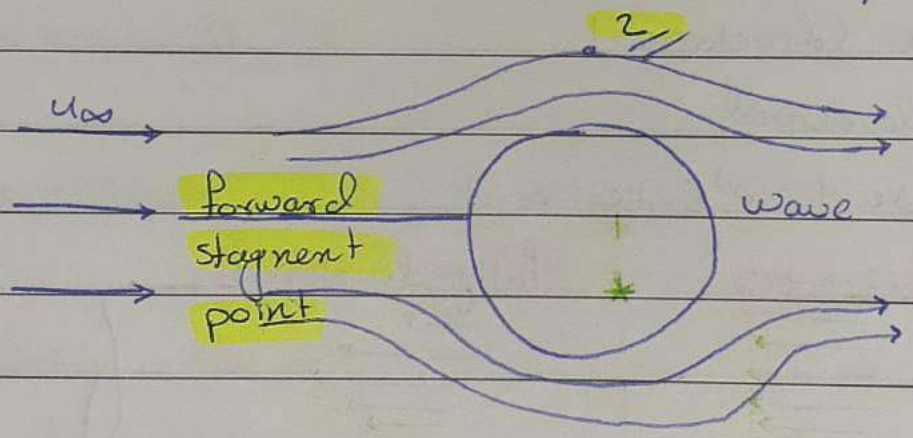
تصير في الـ laminar & Turbulent

* معادلات الـ derevation (Theory) لـ تطبيق على Laminar في حساب Nu .

* بينما معادلات الـ Nu (empirical) لـ تجريبية تطبيق على الـ turbulent

* flow past cylinder:

المعادلات اللي كانت مستخدمة قبل في حساب الـ Nu كانت مخصصة في حالة flat plate ، هون غيّرنا الشكل إلى cylinder وتم الحصول على معادلات Nu جديدة .



* at stagnent point :

$V = 0 \Rightarrow k = 0$ & P max

* at point 2 :

$P = 0$ & V max $\Rightarrow$ k max

* ماذا يفرض في عمليات ال design Turbulent or laminar

لنقل الحرارة ؟

Turbulent ⇒ لما Re أكبر بالتالي Nu أكبر

$\therefore h \uparrow$

$Nu = C\, Re^{m}\, Pr^{n}\, (F)$

$Re \uparrow \quad Nu \uparrow$

$Re_D \Rightarrow \frac{\rho u_\infty D}{\mu}$

$\therefore$ cylinder

$Re_L \Rightarrow \frac{\rho u_\infty L}{\mu}$

$\therefore$ plate

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

$Nu \uparrow \quad h \uparrow$

$q = hA\,\Delta T$

$h \uparrow \quad q \uparrow$

$Pr_s \Rightarrow$ at T_s

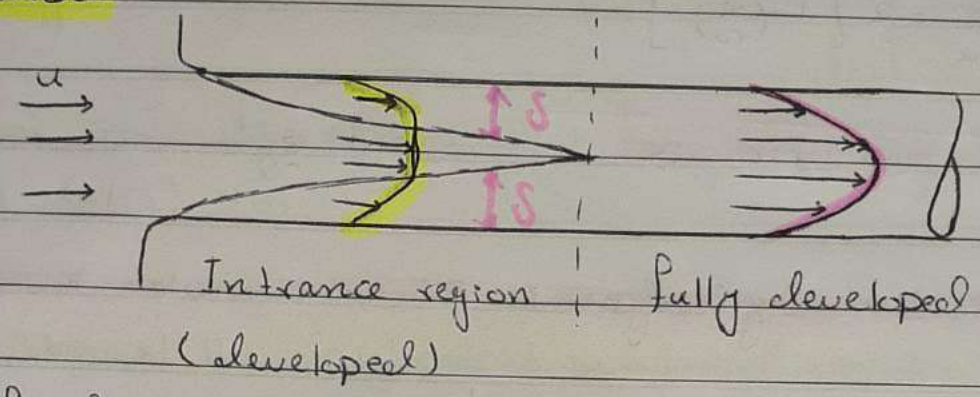
يعني بطلع P_r على درجة حرارة ال surface

* جميع المعادلات بتعطي قيم مختلفة وهذا بسبب أخطاء في :

- properties & experimental & human error.

* Internal flow : Laminar & Turbulent regimes

- Laminar hydrodynamic B.L development in a circular tube.



السرعة لا تتغير على طول الـ pipe (parabolic velocity)

حدث تغير بسرعة الـ fluid

∴ خارج شكل البوروكيه فيا تغير بسرعة

الـ fluid يعني يوجد δ ⇐

thickness boundary layer

إذا رأى البوروكيه السرعة منتظمة

(لا يحدث تغير)

* Notes: In general, the entrance length, $x_h = f(Re_D)$

& $(x/D)_{lam} = .05\, Re_D$ ⇒ entrance كسابي طول الـ

معادلة خاصة بـ laminar

- A force balance on a differential fluid element gives a parabolic velocity for fully developed laminar flow.

$$u(r) = \frac{-1}{4\mu}\left(\frac{dP}{dx}\right) r_0^2 \left[1-\left(\frac{r}{r_0}\right)^2\right]$$

drop pressure

$$u_m = -r_o^2 \frac{dp}{dx}$$

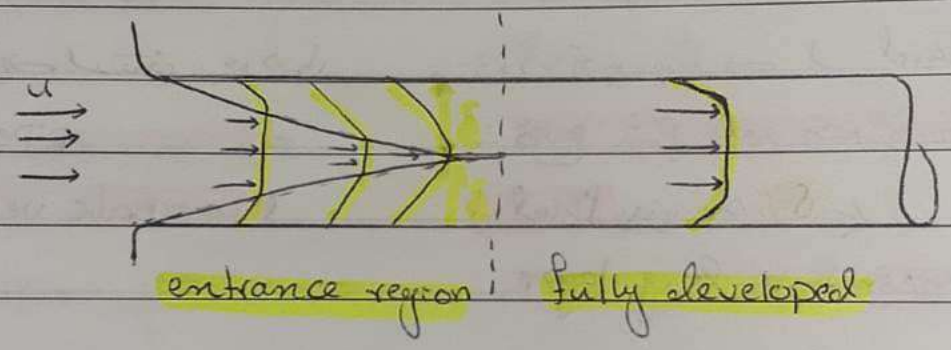
where:

$u_m \Rightarrow$ mean velocity

$$\frac{u_r}{u_m} = 2\left[1-\left(\frac{r}{r_o}\right)^2\right]$$

u_{max} at $r=0 \Rightarrow$ في المنتصف

- Turbulent hydrodynamic B.L in a circular tube:



* Notes

- Unlike laminar flow, the entrance length, x_n is not function in Re_D.
- Experiments have shown that:

$$10 < \left(\frac{x_n}{D}\right) < 60$$

يعني بفرض قيمة $\frac{x_n}{D}$ بين (10-60) مثل 30 و 20 وحد لفرض

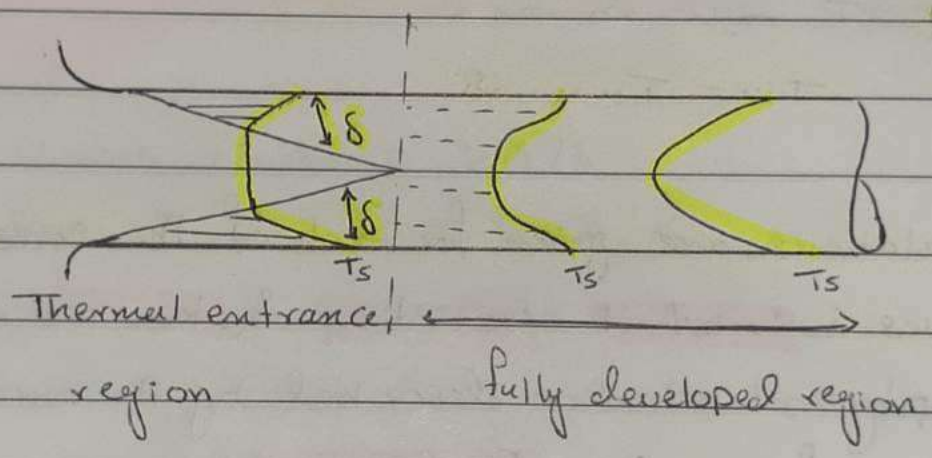
نحسب قيمة x_n

- velocity profile for fully developed turbulent flow is not parabolic & is flatter due to turbulent mixing in the radial direction.

بسبب ال mixing اللي بصير في ال turbulent تكون ال velocity profile أكثر flat

- for simplicity, assume that fully developed occurs at $(x/D) > 10$

* Thermal B.L development in a heated circular tube:



هذا الملف للتسخين لأنه المنحنى بيتناقص من T_s لـ T_s

Notes:

- for laminar flow, the thermal entrance length, x_t:

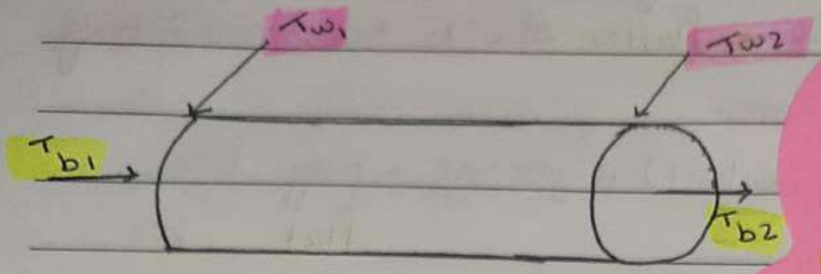
$$x_t = f(Re, Pr)$$

$$\frac{x_t}{D} = .05\, Re_D\, Pr$$

- for turbulent, $x_t \neq f(Re, Pr) \sim (x_t/D) > 10$ or assume $(x_t/D) = 10$

17th - mar

* Correlations for forced convection in pipes and Ducts:



figure

* هنا ال pipe - ثابتة على T3 في حال كان ال isothermal surface

يعني $T_{w1} = T_{w2}$

* وقد تكون عليه T4 إذا لم تكن $T_{w1} = T_{w2}$

* صعب دع نقوم بانتقال الحرارة داخل ال pipes يعني بيهمني T_{in} & T_{out}

* T_b بتسمى T bulk or T average

دلالة على أن ال T تتغير مع ال r

$T(r)$

- Correlations are given for each of the three flow regimes; turbulent, transition & laminar.

The physical situation described by the correlations is given in figure ↑. Fluid enters the pipe at an average Temperature T_{b1} & leaves at an average temperature T_{b2} (T_b stands for bulk temperature. It is the fluid temperature averaged over the flow area).

The pipe wall temperature is T_{w1} at the entrance & T_{w2} at the exit.

* ربع نبلش بال correlations تختلف باختلاف ال regime

* Turbulent Region :

with respect to heat transfer in circular pipes, fully developed turbulent flow is achived at a Reynolds number of approximate 10^4. for this flow regime $Re \geq 10^4$, the following correlation is widely used:

$$Nu = .027\, Re^{0.8}\, Pr^{1/3} \left(\mu/\mu_w\right)^{0.14}$$ => seider - Tate correlation

T ال تغير ع ال fraction ال ناتج

where:

- $Nu \equiv$ Nusselt Number $\equiv hD/k$
- $Re \equiv$ Reynolds Number $\equiv D v \rho / \mu$
- $Pr \equiv$ Panelt number $\equiv C_p \mu / k$
- $D \equiv$ inside pipe diameter
- $v \equiv$ average fluid velocity
- C_p, μ, ρ, k = fluid properties evaluated at the average bulk fluid temperature
- $\mu_w \equiv$ fluid viscosity evaluated at average wall temperature.

average bulk fluid temperature & average wall temperature. ← تعريف ال

physical properties ال
average bulk fluid temperature. او نحددها عند
at average wall temperature او عند μ_w

* The average bulk fluid temperature & average wall temperature are given by:

$$T_{b,avg} = \frac{T_{b1} + T_{b2}}{2}$$

$$T_{w,avg} = \frac{T_{w1} + T_{w2}}{2}$$

* Note:

- The equation is frequently written with the coefficient 0.027 replaced by .023. The latter value gives a somewhat more conservation estimate for the heat-transfer coefficient, which is often desirable for design purpose.

* .14 قد تكتب معادلة Nu

$$Nu = .027 Re^{0.8} Pr^{1/3} (\mu/\mu_w)$$

باستبدال .027 بـ .023

لشغلها في design

- The previous correlation is valied for fluids with Pranelt numbers between 0.5 and 17000 & for pipes with $L/D > 10$.

- For short pipes with $10 < L/D < 60$, the right-hand side of the equation is often multiplied by the factor $[1 + (D/L)^{2/3}]$ to correct for entrance & exit effects

$$Nu = .027 Re^{.8} Pr^{1/3} (\mu/\mu_w)^{.14} [1 + (D/L)^{2/3}]$$

في حالة ال short pipes إذا كان نسبة $10 < L/D < 60$
نظهر تأثير ال (developed) entrance ، فيجب ضرب ال Nu
بـ correction factor ⇐ $[1 + (D/L)^{2/3}]$

*Notes:

- The correlation is generally accurate to within ±20% to ±40%. It is more accurate for fluids with low to moderate Pranelt numbers ($0.5 \leq Pr \leq 100$), which includes all gases & low viscosity process liquids such as water, organic solvents, light hydrocarbons.

* عندما يكون ال fluid فيه ال Pr ال قليل تعطي ال correlations ⇐ accurate data & عندما تكون فيه accurate data
ال μ قليل للسوائل ⇐ accurate data

- It is less accurate for highly viscous liquids, which have correspondingly large prandtl numbers.

$\mu \downarrow$ $P_r \downarrow$

accurate data from correlations $\uparrow$

$P_r = C_p \mu / k$

$\Rightarrow$ $\mu \uparrow$ $P_r \uparrow$

accurate data from correlations $\downarrow$

* for laminar flow in circular pipes ($Re < 2100$), the Seider-Tate correlation takes the form:

$$Nu = 1.86 \left[Re\, Pr\, D/L\right]^{1/3} (\mu/\mu_w)^{0.14}$$

- This equation is valied for $0.5 < Pr < 17000$ and $\left[Re\, Pr\, D/L\right]^{1/3} (\mu/\mu_w)^{0.14} > 2$, and is generally accurate to within $\pm 25\%$.
  - إذا كان correction factor > 2 ، يطبق المعادلة

- For $\left[Re\, Pr\, D/L\right]^{1/3} (\mu/\mu_w)^{0.14} < 2$, the nusselt number should be set to 3.66, which is the theoretical value for laminar flow in an infinitely long pipe with constant wall temperature.

  إذا كان correction factor < 2 ، يعتبر فيزيد
  $Nu = 3.66$ دائماً.

- Fluid properties are evaluated at $T_{b,ave}$, except for μ_w, which is evaluated at $T_{w,ave}$

- Also, at low Reynolds numbers heat transfer by natural convection can be significant (see Natural convection), and this effect is not accounted for in the Seider-Tate correlation.

convection

Natural → $q = \frac{\Delta T}{1/Ah}$

forced → تحتوي على قيم Nu, Pr ---

when Re is low :

نطبق معادلات الـ Natural

إذا طبقت الـ correlations السابقة وكان الـ Re قليل

طالع تعطيني نتائج مش مظبوطة

المسيطر أكثر هو الـ Natural

* for transition Region:

معادلات ال transition
تعطي نتائج عالية الدقة
لأنها عبارة عن
combined regime
laminar & turbulent

• for flow in the transition region ($2100 < Re < 10^4$), the Hausen equation correlation

$$Nu = 0.116 \left[Re^{2/3} - 125 \right] Pr^{1/3} (\ldots$$

• Heat transfer calculations in the transition region are subject to a higher degree of uncertainty than those in the laminar & fully developed turbulent regimes.

• Although industrial equipment is sometimes designed to operate in the transition region, it is generally recommended to avoid working in this flow regime if possible.

يفضل عدم الشغل عن هذا ال regime

• An alternative equation for the transition & turbulent regimes has been proposed by Gnielinski:

$$Nu_D = \frac{(f/8)(Re_D - 1000)\, Pr}{1 + 12.7 (f/8)^{1/2} (Pr^{2/3} - 1)} \left[1 + \left(\frac{D}{L}\right)^{2/3} \right]$$

$.6 < Pr < 2000$
$Re > 2300$

f ⇒ Darcy friction factor:

$f = (0.782 \ln Re - 1.51)^{-2}$

approximation of Colebrook equation

* f ⇒ friction factor

* المشكلة في هذه المعادلة
تتطلب حساب قيمة f

* ducts and conduits with non-circular cross-sections.

The pipe non circular

* في هذه الحالة بطبق نفس المعادلات السابقة لـ circular
ولكن الـ D المستخدم في المعادلات نستبدله بـ D equivalent

$$D_{eq} = 4 \times \text{hydraulic radius} = 4 \times \frac{\text{flow area}}{\text{wetted parameter}}$$

المحيط المبلل

- This approximation generally gives reliable results for turbulent flow. However, it is not recommended for laminar flow.

* التطبيق الأكثر شيوعاً هو:

laminar flow in non-circular ducts is flow in the annulus of a double pipe heat exchanger.

- for laminar annular flow, the following equation given by Gnielinski is recommended:

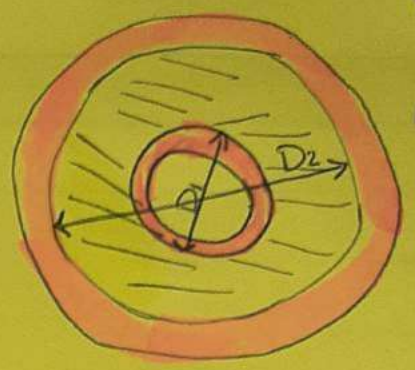
$$Nu = 3.66 + 1.2\,(D_2/D_1)^{.8} + \frac{.19\,[1 + .14\,(D_2/D_1)^{.5}]\,[Re\,Pr\,D_e/L]^{.8}}{1 + .117\,[Re\,Pr\,D_e/L]^{.467}}$$

where:

D_1 : out side diameter of inner

D_2 : inside " " outer

D_e : equivalent diameter $= D_2 - D_1$



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

$$D_h = \frac{\cancel{4}\,\frac{\cancel{\pi}}{\cancel{4}}\,(D_2^2 - D_1^2)}{\cancel{\pi}\,(D_1 + D_2)} = \frac{(D_2 - D_1)\cancel{(D_2 + D_1)}}{\cancel{(D_1 + D_2)}}$$

$$D_h = D_2 - D_1$$

- Nu number in aprevious equation based on De.

* لما هو عرفت قيمة Nu ومن قيمة لـ Nu نحسب قيمة h ثم نعوض بمعادلة $q = hA\Delta T$

ولكن هو ΔT مختلف بسبب وجود T_w & T_b.

$$\Delta T_{ln} = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1 / \Delta T_2)}$$

where:

$\Delta T_1 = |T_{w1} - T_{b1}|$

$\Delta T_2 = |T_{w2} - T_{b2}|$

يصبح شكل معادلة لـ q بعد استبدال ΔT

$$q = hA\Delta T_{ln}$$

where:

ΔT_{ln} = logarithmic mean temperature

* في حالات تكون فيها $\Delta T_{avg} \simeq \Delta T_{ln}$:

- عندما تكون ΔT_1 & ΔT_2 قريبات من بعض

$\Delta T_1 = 50$

$\Delta T_2 = 60$

$\Delta T_{avg} = \frac{50+60}{2} = 55$

$\Delta T_m = \frac{60-50}{\ln(60/50)} = 54.85$

$\Delta T_{avg} \simeq \Delta T_m$

- عندما يكون التدفق laminar & $Re\ Pr\ D/L > 0$ ، يصبح الفرق بين ΔT_{avg} & ΔT_m قليل .

15th - mar

* Example 1 :

Air at 20°C is blown over a 6 cm OD pipe that has a surface temperature of 140°C . The free stream air velocity is 10 m/s . what is the rate of heat transfer per meter of pipe ?

Soln :

في خطوات معينة كل السؤال :

١- برسم schematic (رسمة توضح السؤال)

٢- بحط ال assumptions اللي بيلزموا بالسؤال

* بعض ال assumptions المستخدمة :

- assume steady state
- assume the length = 1m
- assume uniform surface temperature

٣- رح نكتب جميع ال physical properties اللازمة للسؤال .

بعض ال properties المستخدمة في الحل :

- ρ , μ , k , Pr

* بعض في هذه المسائل تطلع من ال tables قيمة ν وليس μ ، لماذا ؟

معادلة Re عبارة عن :

$$Re = \frac{\rho v D}{\mu} = \frac{v D}{\nu}$$

بدل ما أطلع ال ρ & μ ، بطلع قيمة واحدة فقط وهي ν وهكذا أسهل في حالة بدي أعمل interpolation

*Note:

type of viscosity

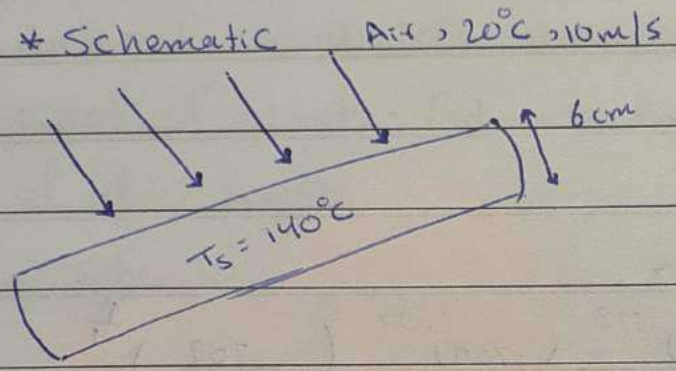
Kinematic	Dynamic
$\nu = \frac{\mu}{\rho}$	μ

٤- نحدد الـ regime عن طريق قيمة Re

٥- نختار المعادلة المناسبة لـ Nu من الـ tables

٦- $q = A h \Delta T$

* Schematic



$\nu|_{293.15k} \Rightarrow \frac{15.89 - 11.44}{300 - 250} = \frac{x - 11.44}{293.15 - 250}$

$\therefore \nu = 15.28 \times 10^{-6} \frac{m^2}{s}$

$Pr|_{293.15k} \Rightarrow \frac{.707 - .720}{300 - 250} = \frac{x - .720}{293.15 - 250}$

$Pr = .709$

$k|_{293.15} = \frac{26.3 - 22.3}{300 - 250} = \frac{x - 22.3}{293.15 - 250} \Rightarrow k = 25.75 \times 10^{-3}$

$Re = \frac{V D}{\nu} = \frac{10 \frac{m}{s} \times 6 \times 10^{-2} m}{15.28 \times 10^{-6} \frac{m^2}{s}} =$

$Re = 39267$

$\therefore$ flow is turbulent

نختار من الـ tables معادلة Nu في حالة الـ turbulent

- Re in pipe < 2300
  $\therefore$ flow is laminar

- $Re > 2300$
  $\therefore$ flow is turbulent

Eq: 2

$$Nu = C Re^{m} Pr^{n} (Pr/Pr_s)^{1/4}$$

$$Pr_s\Big|_{413.15K} = \frac{.686 - .690}{450 - 400} = \frac{x - .690}{413.15 - 400}$$

$$Pr_s\Big|_{413.15} = .689$$

if $Pr \leq 10 \Rightarrow n = .37$

$$Nu = .193 (39267)^{.618} (.709)^{.37} \left(\frac{.709}{.689}\right)^{1/4}$$

$$Nu = 118.17$$

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

$$118.17 = h \frac{(.06)}{25.75 \times 10^{-3}}$$

$$h = 50.7 \ W/K.m^2$$

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

$$= 50.7 (\pi D L)(T_s - T_\infty)$$

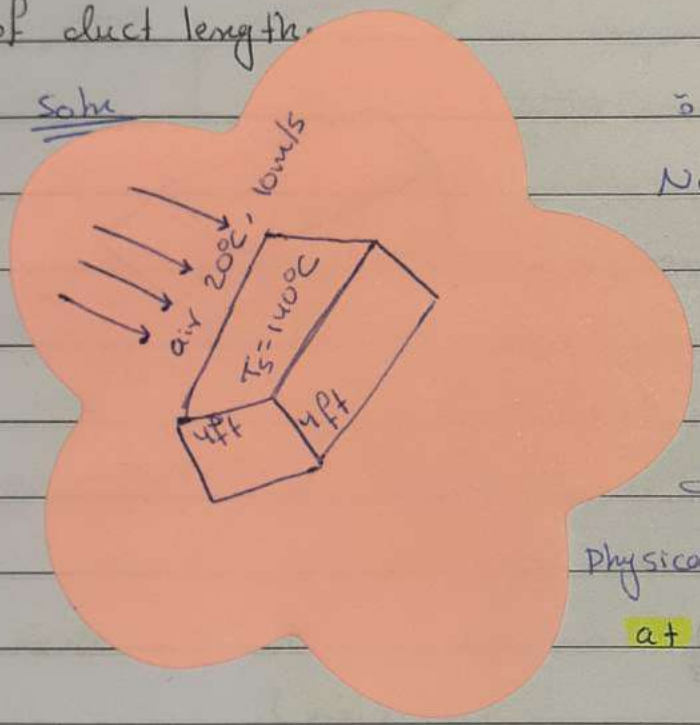
$$\frac{q}{L} = 50.7 (\pi .06)(140 - 20)$$

$$\frac{q}{L} = 1146.8 \ \frac{W}{m}$$

* Example 2:

A duct carries hot waste gas from a process unit to a pollution control device. The duct cross section is 4 ft x 4 ft & it has a surface temperature of 140°C. Ambient air at 20°C flows across the duct with a wind speed of 10 m/s. Estimate the rate of heat loss per meter of duct length.

Soln



هذا الشكل له معادلة واحدة

$Nu = C\,Re_D^{m}\,Pr^{\frac{1}{3}}$

∵ non-circular cylinder

* ملاحظة مهمة :

من شروط تطبيق هذه المعادلة

ان يكون جميع الـ physical properties

at T_f

$$T_f = \frac{T_\infty + T_s}{2} = \frac{20 + 140}{2} = 80°C$$

$\nu|_{353.15} \Rightarrow \frac{26.41 - 20.92}{400 - 350} = \frac{x - 20.92}{353.15 - 350}$

$\nu = 21.266 \times 10^{-6}\ m^2/s$

$k \Rightarrow \frac{33.8 - 30}{400 - 350} = \frac{x - 30}{353.15 - 350}$

$k = 30.24 \times 10^{-3}$

$Pr|_{353.15} \Rightarrow \frac{.69 - .7}{400 - 350} = \frac{x - .7}{353.15 - 350}$

$Pr = .699$

width ⇒ square ان

$$Re = \frac{D\,v}{\nu} = \frac{w\,v}{\nu} =$$

$$\frac{4\,ft \;\; .3048\,m}{1\,ft} \times 10\,\frac{m}{s} \;\; \frac{s}{21.266\times10^{-6}\,m^2}$$

$Re = 573309.5$

∴ turbulent regime

$$Nu = C\,Re^{m}\,Pr^{\frac{1}{3}}$$

$$= .102\,(573309.5)^{.675}\,(.699)^{\frac{1}{3}}$$

$Nu = 697.7$

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

$$697.7 = \frac{h\,(1.22)}{30.24\times10^{-3}}$$

$h = 17.29\ w/m^2.k$

$q = hA\,(T_s - T_\infty)$

$$q = 17.29\,\left(4 \times \left(4\,ft\ \frac{.3048\,m}{1\,ft}\right) L\right)\,(140-20)$$

عدد المزدوج التي حدث ⇐ 4
فيها انتقال للطاقة ، القاعدية
تهمل لان مقدار الطاقة صغير
جداً مقارنة مع السطح الأخرى

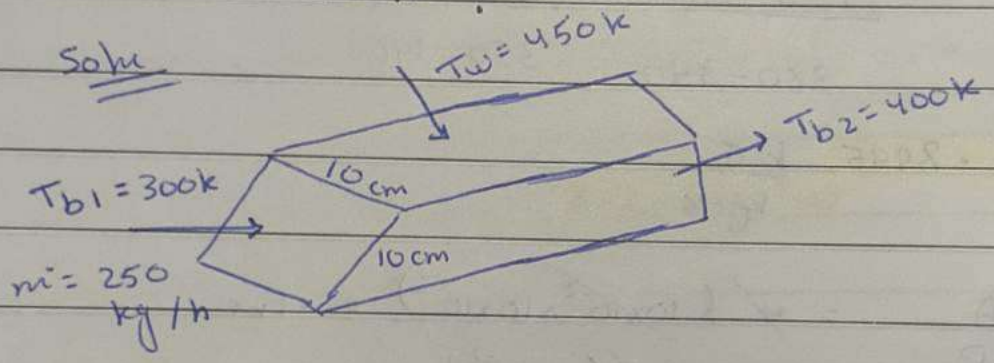
$$\frac{q}{L} = 17.29\,(4\times1.22)\,(120)$$

$$\frac{q}{L} = 10\,118.38\ \frac{w}{m}$$

* Ex 4:

Carbon dioxide at 300 k & 1atm is to be pumped through a duct with a 10 cm * 10 cm square cross-section at a rate of 250 kg/h. The walls of ducts will be at temperature of 450 k. what distance will the CO_2 travel through the duct before its temperature reaches a 400 k ?

Soln



* physical properties at $T_{avg} = \frac{300+400}{2} = 350\,k$

$$\rho\Big|_{350} \Rightarrow \frac{1.4743 - 1.5618}{360 - 340} = \frac{x - 1.5618}{350 - 340}$$

$$\rho\Big|_{350} = 1.518\ kg/m^3$$

$$\mu\Big|_{350} \Rightarrow \frac{173 - 165}{360 - 340} = \frac{x - 165}{350 - 340}$$

$$\mu\Big|_{350} = 169 \times 10^{-7}\ N \cdot s/m^2$$

$$k\Big|_{350} \Rightarrow \frac{21.2 - 19.7}{360 - 340} = \frac{x - 19.7}{350 - 340}$$

$$k\Big|_{350} = 20.45 \times 10^{-3}\ w/m \cdot k$$

$$Pr\Big|_{350} \Rightarrow \frac{.741 - .746}{360 - 340} = \frac{x - .746}{350 - 340}$$

$$Pr\Big|_{350} = .7435$$

$$\mu_w\Big|_{450} = 210 \times 10^{-7}\ N \cdot s/m^2$$

$$Cp\Big|_{350} \Rightarrow \frac{.908 - .891}{360 - 340} = \frac{x - .891}{350 - 340}$$

$$Cp\Big|_{350} = .8995\ \frac{kJ}{kg \cdot k}$$

$$D_{eq} = \frac{4A}{wet\ P} = \frac{\cancel{4}(\cancel{10\times10^{-2}} \times 10\times10^{-2})}{\cancel{4}(\cancel{10\times10^{-2}})} = .1\,m$$

$$Re = \frac{\rho\, v\, D_{eq}}{\mu}$$

$$Re = 1.518\ \frac{\cancel{kg}}{\cancel{m^3}} \times 4.57\ \frac{\cancel{m}}{\cancel{s}} \times \frac{.1\,\cancel{m}\ \cancel{m^2}}{169\times10^{-7}\ \cancel{N \cdot s}}$$

$$Re = 41049$$

$\therefore$ turbulent regime

$$Nu = .027\, Re^{0.8}\, Pr^{1/3}\, (\mu/\mu_w)^{0.14}$$

$$Nu = .027\,(41049)^{.8}\,(.7435)^{\frac{1}{3}}\left(\frac{169\times10^{-7}}{210\times10^{-7}}\right)$$

$$Nu = 116.38$$

$$\dot{V}olume = \frac{mass\ rate}{\rho}$$

$$\dot{V}olume = \frac{250\ \cancel{kg}\ m^3\ \cancel{h}}{1.518\ \cancel{h}\ \cancel{kg}\ 3600\,s}$$

$$\dot{V} = .0457\ m^3/s$$

$$velocity = \frac{\dot{V}olume}{area}$$

$$velocity = \frac{.0457\ \cancel{m^3}}{.01\ \cancel{m^2}\ s}$$

$$v = 4.57\ \frac{m}{s}$$

$$Nu = \frac{h\, D_{eq}}{k}$$

$$116.38 = \frac{h \times .1}{20.45 \times 10^{-3}}$$

$$h = 23.8 \ W/m^2 \cdot k$$

$$q = C_p m \Delta T$$

$$q = .8995\,(250)\,(400-300)\ \frac{kJ}{kg \cdot k} \times \frac{kg}{h} \times k \ \frac{h}{3600\,s}$$

$$q = 6.246\ kW = 6.246 \times 10^3\ W$$

$$q = hA\,\Delta T_m$$

$$6.246 \times 10^3\ W = 23.8\ \frac{W}{m^2 \cdot k}\,(4 \times 10 \times 10^{-2}\ m\ L)\ 91\ k$$

$$L = 7.2\ m$$

$$\Delta T_1 = 450 - 300 = 150$$

$$\Delta T_2 = 450 - 400 = 50$$

$$\Delta T_m = \frac{150-50}{\ln(150/50)}$$

$$\boxed{\Delta T_m = 91}$$

* كيف يمكن قياس قيمة k or h عملياً بالمختبر ؟!

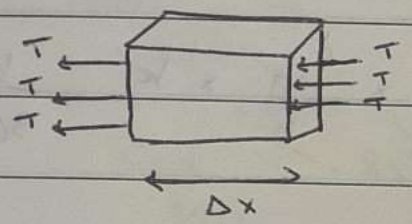
- بجيب heater power وبعرض سطح المادة لمقدار معين من الطاقة .

- بقيس أبعاد الجسم وبحسب ال A

- بحسب ΔT على السطح الأمامي والسطح الخلفي من خلال thermometer

وفي أكثر من جهاز لقياس درجة الحرارة

على أكثر من نقطة .



$$q = k A \frac{dT}{dx}$$

أصبحت كل القيم معلومة ما عدا ال k بحسبها من معادلة

Fourier's

وبنفس الفكرة بحسب قيمة h ولكن من قانون Newton's

$$q = hA \Delta T$$

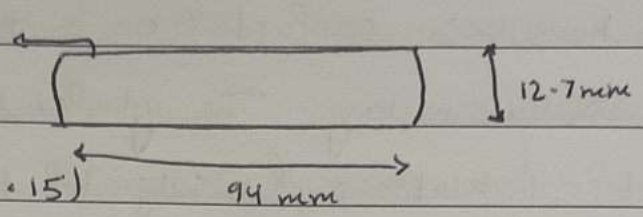
* Ex 5: experiment

Experiments have been conducted on a metalic cylinder 12.7 mm in diameter & 94 mm long. The cylinder is heated internally by an electrical heater and is subjected to across flow of air in a low-speed wind tunnel. Under a specific set of operating conditions for which the upstream air velocity & temperature were maintained at $v = 10$ m/s & 26.2°C, respectively, the heater power dissipation was measured to be $P = 46$ W, while the average cylinder surface temperature was determined to be $T_s = 128.4$°C. It is estimated that 15% of the power dissipation is lost through the commulative effect of surface radiation & conduction through the end pieces.

1- Determine the convection heat transfer coefficient from the experimental observations.

2- Compare the experimental result with the convection coefficient computed from an appropriate correlation.

air 10m/s $T_\infty = 26.2°C$



$q = q \times (1 - .15)$

قدرة بعد الـ loss $= 46\ (.85) = 39.1\,W$

1/ $q = hA\,\Delta T$

$39.1W = h\ \pi D L\ (T_s - T_\infty)$

$39.1\,W = h\ \pi\ (12.7\times10^{-3})m\ (94\times10^{-3})m\ (128.4 - 26.2)k$

$h = 102.4\ W/m^2.k$

experiments هي القيمة الناتجة من الـ

correlation لنفس الـ بدي أحسب قيمة h من الـ

2/ Physical properties at T_∞

$\nu\big|_{299.35k} = 15.89\times10^{-6}\ \frac{m^2}{s}$

$Pr = .707$

$k = 26.3\times10^{-3}\ W/m.k$

$Pr_s = 401.55 = .690$

$Re = \frac{VD}{\nu} = \frac{10\,m/s \times 12.7\times10^{-3}\,m}{15.89\times10^{-6}\,m^2/s} = 7992.448$

$\therefore$ Turbulent flow

$Nu = C\ Re^m\ Pr^n\ (Pr/Pr_s)^{1/4}$

$C = .193$ $\quad m = .618$ $\quad n = .37 \Rightarrow Pr < 10$

$$Nu = .193\,(7992.448)^{.618}\,(.707)^{.37}\left(\frac{.707}{.690}\right)^{\frac{1}{4}}$$

$Nu = 44.09$

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

$$44.09 = h\,\frac{(12.7\times10^{-3})}{26.3\times10^{-3}}$$

$h = 91.3\ W/m^2.K$

* ملاحظة : الدكتور استخدم أكثر من correlation وكانت قيمة h تقريباً بتتراوح بين (80 - 110) ± رقم

24th - mar - 2022

* flow across Bank of tubes:

Bank of tubes ⇐ heat exchanger من أنواع ال

ده تركز على ال external flow

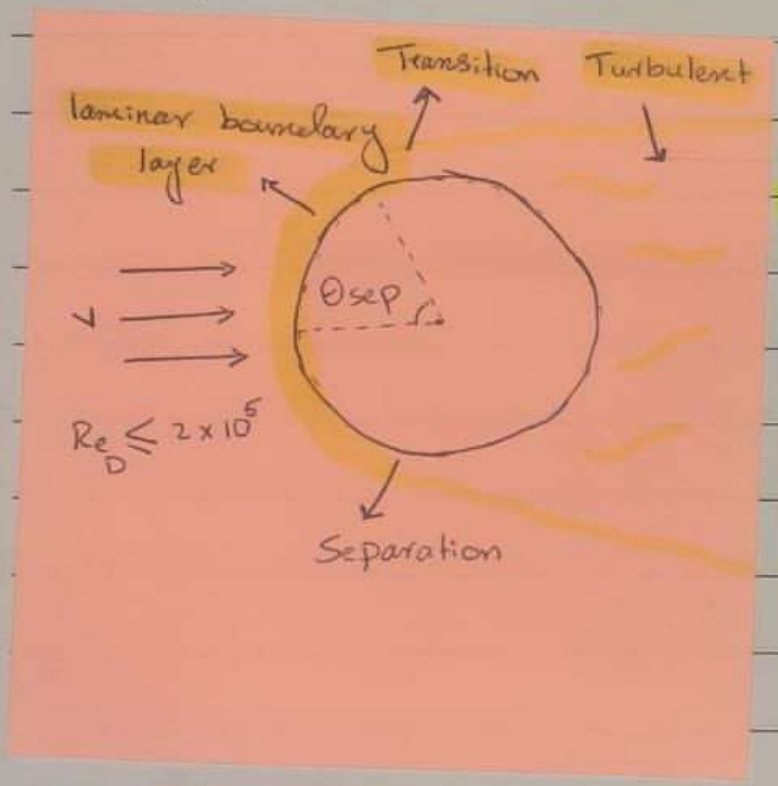
* مناقشة ل one tube :

بيكون boundary layer

في نقطة بيتحول ال regime

من transition ال laminar

ثم turbulent



* Bank of tubes ⇒ عبارة عن مجموعة من ال tubes

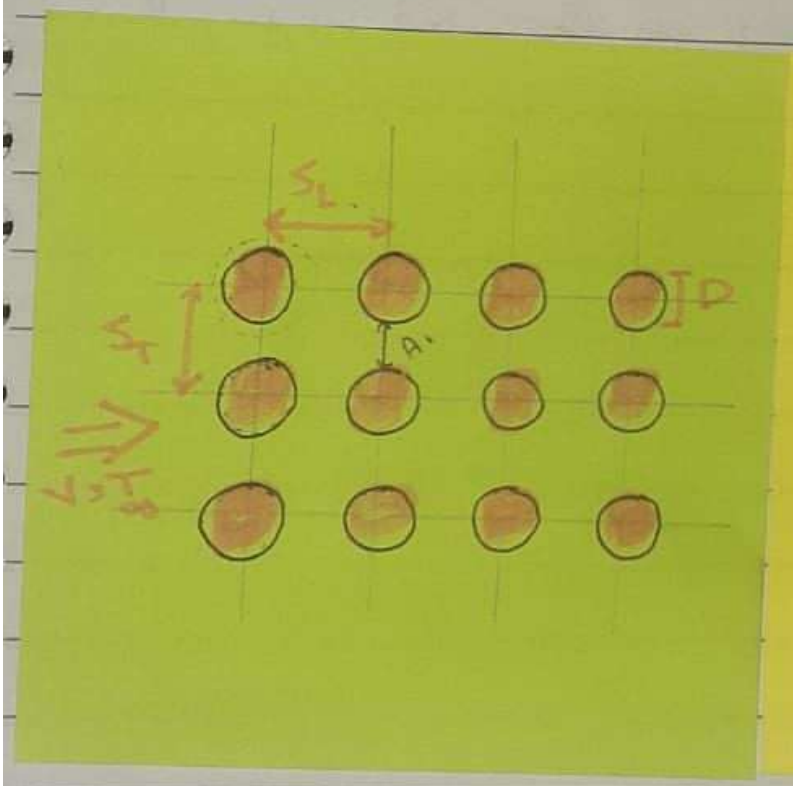
* 2 rows ده بيتأثر بال turbulent flow بالتالي ده بيتحسن ال heat transfer.

* في عمليات ال design أول شي بهتم في هو ال arrangement
(ترتيب ال tubes والمسافة بينهم)

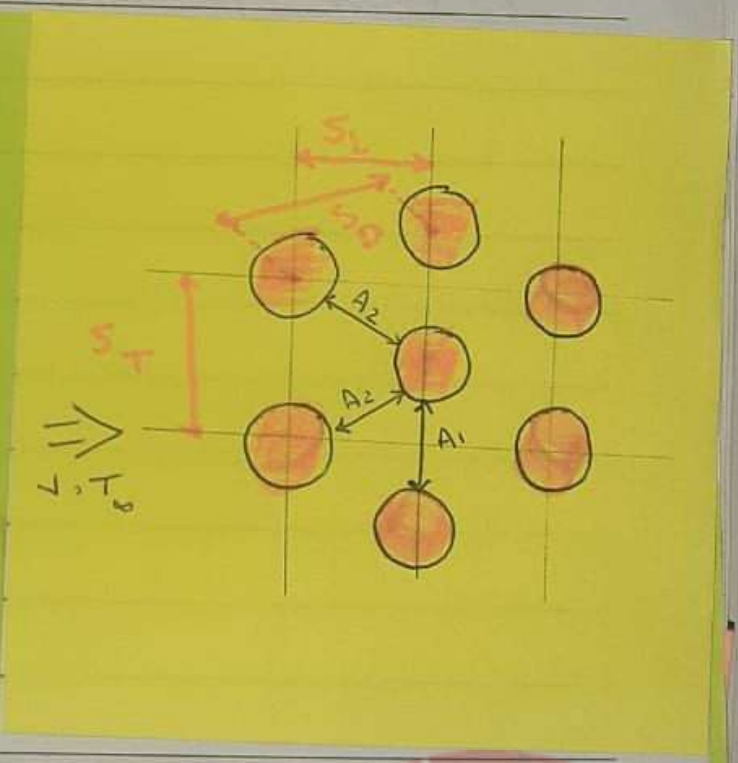
* Type of arrangement

Aligned
بيج شكل مربع
بين ال tubes

Staggered
بيج شكل مثلث بين
ال tubes



(Aligned)



(Staggered)

where:

S_T => Transverse Pitch

S_L => Longitudinal Pitch

* أهم شي المسافات بين ال Tubes بتلعب دور كبير في ال heat transfer

* Notes:

- The configuration (ترتيب) is characterized by the tube diameter D and by the transverse pitch S_T and longitudinal pitch S_L measured between tube centers.

- Conditions within the bank are dominated by boundary layer separation effects and by wake interactions, which in turn influence convection heat transfer.

- Flow around the tubes in the first Row of a tube bank is similar to that for a single (isolated) cylinder in cross flow.

أول row الي بنتبع ال correlation لـ single tube

* Correlations of heat transfer:

$$\overline{Nu}_D = C_1 Re^{m}_{D,max} \, Pr^{0.36} \left(\frac{Pr}{Pr_s}\right)^{\frac{1}{4}}$$

تطبق هذه المعادلة إذا :

- عدد الصفوف ≥ 20

- $N_L \geq 20$

- $0.7 \leq Pr \leq 500$

$10 \leq Re_{D,max} \leq 2 \times 10^6$

where:

$N_L \Rightarrow$ Number of tube rows

* المقصود بـ Remax يعني نحسب Re عند velocity max

* جميع الخصائص ما عدا Prs بأخذ قيمهم عند Taug

$$T_{aug} = \frac{T_i + T_o}{2} = \frac{T_{in} + T_{out}}{2}$$

* إذا لم يكن عدد الصفوف ≥ 20 يعني تصحيح لـ قيمة Nu عن طريق ضرب الـ Nu بالمعادلة السابقة بعامل تصحيح

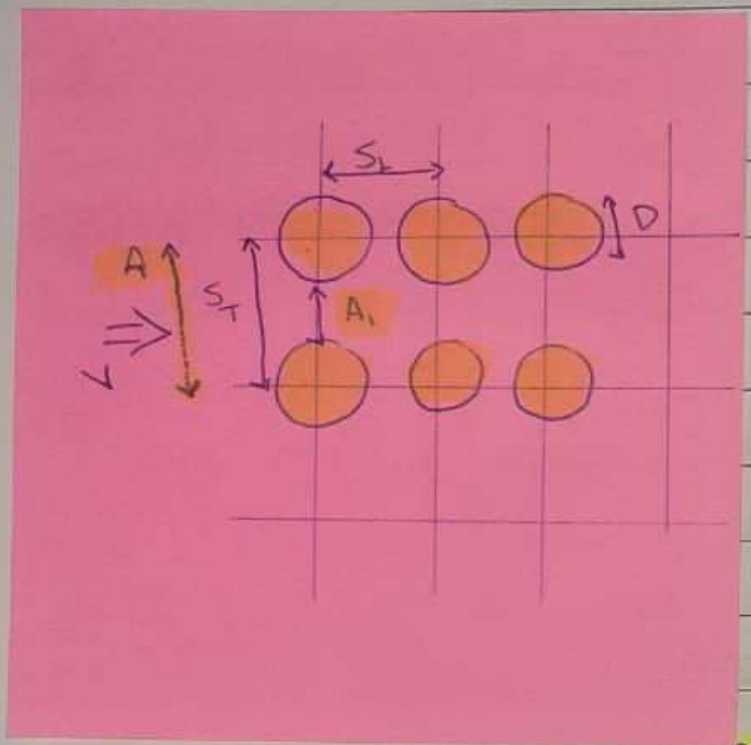
تصبح المعادلة بعد التصحيح :

$$\overline{Nu}_D \Big|_{N_L \leq 20} = C_2 \, \overline{Nu}_D \Big|_{N_L \geq 20}$$

where: C_2 is a correction factor قيمة تعطى في table

* هذا السؤال كيف بدي أحسب v_{max} حتى أقدر أحسب Re_{max} ؟!

الحالة 1 : Aligned



في هذه الحالة ال fluid بيمر من ال bank of tubes من خلال A ، ثم بيمر ال fluid بين ال tubes من خلال A_1 .

والمعروف أنه ما في إلا A_1 بين ال tubes في حالة ترتيب Aligned والمراد حساب قيمة v عند هذه ال A_1 area

ومن خلال معادلة ال Continuity equation نحسب v_{max} عند A_1 .

ثبات ال density

$$Q_{in} = Q \text{ between tubes}$$

$$A\,v = A_1\,v_{max}$$

$$(S_T \cancel{L})\,v = (S_T - D)\cancel{L}\,v_{max}$$

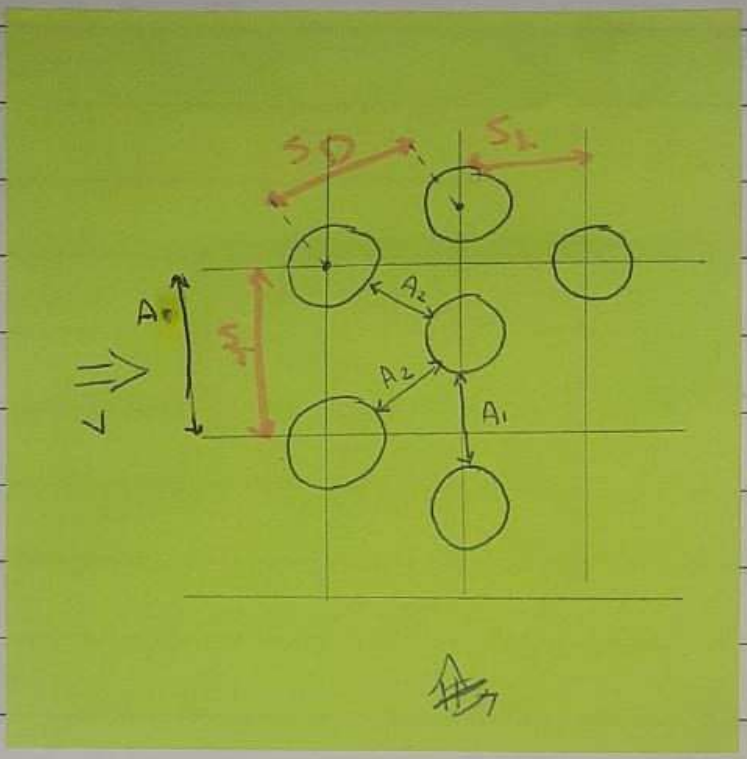
$$S_T\,v = (S_T - D)\,v_{max}$$

$$v_{max} = \frac{S_T\,v}{S_T - D} = \frac{S_T}{S_T - D}\,v$$

at A_1

السرعة اللي رايح فيها ال fluid

* كلما قلت ال area تزداد ال velocity



الحالة 2 : Staggered

في هذه الحالة ال fluid بيمر عبر ال bank of tubes من خلال A ، ثم بيتدفق ال fluid بين ال tubes من خلال مساحتين A_2 & A_1 ، بدي أعمل check أي area أقل بكون عندها V_{max} .

أول شي بعمل assumption أن :

$$2A_2 < A_1$$

ضربت بـ 2 لأن التدفق up & down

إذا كان ال assumption صحيح .. كم ال V_{max} عند A_2

بدي أعوض بدل ال A قيمها $2A_2 < A_1$

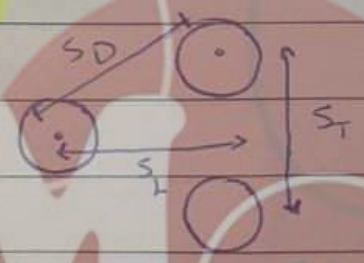
$$2(S_D - D)\cancel{L} < (S_T - D)\cancel{L}$$

كيف بدي أحسب قيمة S_D لأنها لا تكون معطى ؟!

من نظرية فيثاغورس

$$S_D = \left(\left(\frac{S_T}{2}\right)^2 + (S_L)^2 \right)^{\frac{1}{2}}$$

بعوضها في المتباينة



$$2\left[\left(\left(\frac{S_T}{2}\right)^2+(S_L)^2\right)^{\frac{1}{2}}-D\right] \leq S_T - D \quad \div 2$$

$$\left(\left(\frac{S_T}{2}\right)^2+(S_L)^2\right)^{\frac{1}{2}}-D \leq \frac{S_T}{2}-\frac{D}{2}$$

$$\left[\left(\frac{S_T}{2}\right)^2+(S_L)^2\right]^{\frac{1}{2}} \leq \frac{S_T}{2}+\frac{D}{2}$$

$$S_D \leq \frac{S_T+D}{2}$$

this means $2A_2 < A_1$

يعني v_{max} عند A_2

$$v_{max}(2A_2) = vA$$

السرعة التي دخل بها ال fluid

$$v_{max} = \frac{AV}{2A_2}$$

$$v_{max} = \frac{S_T \cancel{L} v}{2(S_D-D)\cancel{L}} = \frac{S_T}{2(S_D-D)} v$$

أما إذا كانت المتباينة خطأ ركز v_{max} عند A'

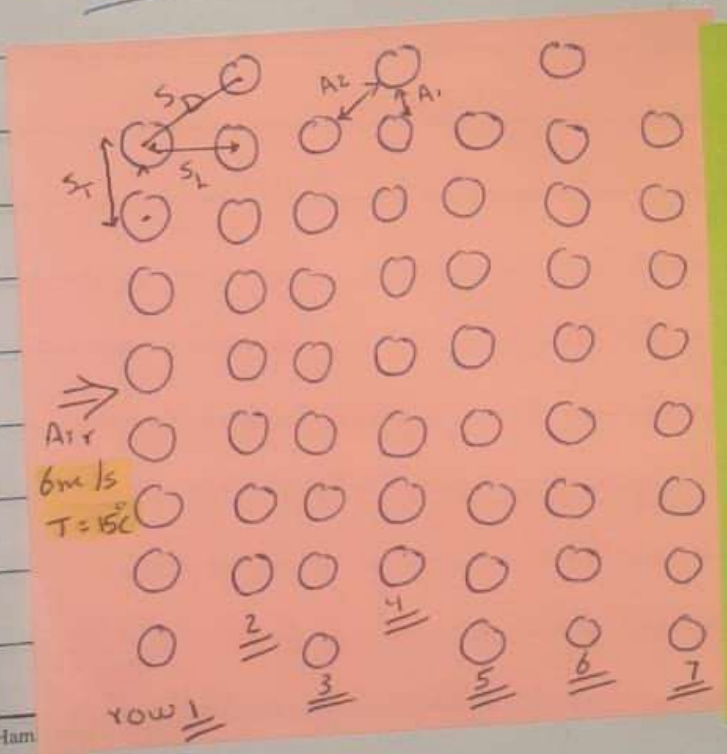
$$v_{max}\Big|_{A_1} = \frac{S_T}{(S_T-D)} v$$

* أهم خطوة في ال design هو ال assumption

* ex: Pressurized water is often available at elevated temperatures and may be used for space heating or industrial process applications. In such cases it is customary to use a tube bundle in which the water is passed through the tubes, while air is passed in cross flow over the tubes. Consider a staggered arrangement for which the tube outside diameter is 16.4 mm and the longitudinal and transverse pitches are $S_L = 34.3$ mm & $S_T = 31.3$ mm. There are seven rows of tubes in the airflow direction and eight tubes per row. Under typical operating conditions the cylinder surface temperature is at 70°C, while the air upstream temperature and velocity are 15°C and 6 m/s, respectively. Determine the air-side convection coefficient.

Sol:

أول شي برسم رسمة توضح السؤال



* بعمل assumption ع درجة حرارة ال tubes.

T_s Assume uniform wall temperature → T_s

T_o ← external flow temperature — T_i

$\Delta T_2 = T_s - T_o$

$\Delta T_1 = T_s - T_i$

$$\Delta T_{lm} = \frac{\Delta T_1 - \Delta T_2}{\ln\left(\frac{\Delta T_1}{\Delta T_2}\right)}$$

* physical properties of Air at 15°C (288.15 k)

$\nu = 14.835 \times 10^{-6}\ m^2/s$

$$\frac{11.44 - 15.89}{250 - 300} = \frac{x - 15.89}{288.15 - 300}$$

$Pr = .71$

$$\frac{.707 - .720}{300 - 250} = \frac{x - .720}{288.15 - 250}$$

$k = 25.352 \times 10^{-3}\ W/m.k$

$$\frac{26.3 - 22.3}{300 - 250} = \frac{x - 22.3}{288.15 - 250}$$

* physical properties of air at surface 70°C (343.15) k

$Pr = .701$

$$\frac{.7 - .707}{350 - 300} = \frac{x - .707}{343.15 - 300}$$

* cheack of area ($2A_2 < A_1$)

$$S_D = \left(\left(\frac{1}{2} S_T\right)^2 + (S_L)^2\right)^{\frac{1}{2}}$$

$$S_D = \left(\left(\frac{31.3 \times 10^{-3}}{2}\right)^2 + (34.3 \times 10^{-3})^2\right)^{\frac{1}{2}}$$

$S_D = 37.7 \times 10^{-3}\ m$

$$S_D \overset{?}{<} \frac{S_T + D}{2}$$

$$\frac{(31.3 \times 10^{-3}) + (16.4 \times 10^{-3})}{2}$$

$S_D > 23.85 \times 10^{-3}$ ∴ ~~Max at Air~~

$$S_D > \frac{S_T + D}{2}$$

$\therefore V_{max}(A_1) = V A$

$$V_{max} = \frac{S_T}{(S_T - D)} V$$

$$V_{max} = \frac{31.3\times10^{-3}}{2(31.3\times10^{-3} - 16.4\times10^{-3})} \times 6 \frac{m}{s}$$

$V_{max} = 12.604\ m/s$

$$Re_{D_{max}} = \frac{D V_{max}}{\nu} = \frac{16.4\times10^{-3} \times 12.604}{14.835\times10^{-6}} =$$

$Re_{D_{max}} = 13933$

$C_1 \Rightarrow$

$$\frac{S_T}{S_L} = \frac{31.3\times10^{-3}}{34.3\times10^{-3}} = .9 < 2$$

$$\therefore C_1 = .35 \left(\frac{S_T}{S_L}\right)^{1/5} = .3427$$

$m = .6$

$$Nu_D\Big|_{N_L \geq 20} = .3427 (13933)^{.6} .71^{.36} \left(\frac{.71}{.701}\right)^{1/4} = 93.15$$

$$Nu_{D_{N_L<20}} = Nu\Big|_{N_L \geq 20} * C_2$$

$$Nu_D\Big|_{N_L<20} = 93.15 \times .95 = 88.5$$

$$h = Nu \frac{k}{D} = 88.5 \times 25.352\times10^{-3} \frac{W}{m\cdot K} \frac{}{16.4\times10^{-3} m} = 136 \frac{W}{m^2\cdot K}$$

* energy balance :

Newton's law of cooling

1 --- $q = hA\,\Delta T_{lm} = \bar{h}\,(N \pi D L)\,\Delta T_{lm} = (\rho V S_T N_T L)\, c_p (T_o - T_i)$

عبارة عن $\dot{m}$

where:

N => Total number of tubes

N_T => number of tubes per row => عدد الانابيب في الصف

$N_T =$ عدد الممرات بين + 1

tubes لـ

2 --- $\Delta T_{lm} = \dfrac{\Delta T_1 - \Delta T_2}{\ln\left(\dfrac{\Delta T_1}{\Delta T_2}\right)}$

Combine Eqs (1) & (2), we obtain

$$\frac{T_s - T_o}{T_s - T_i} = \exp\left[-\frac{\pi D N \bar{h}}{\rho V N_T S_T c_p}\right]$$

* تستخدم هذه المعادلة في الغالب لإيجاد قيمة T_o ، في أغلب الأسئلة تكون T_o غير معروفة

وقد تستخدم أيضاً لإيجاد h وتكون غير دقيقة

29th - may - 2022

* Heat exchanger :

هو جهاز يتكون من tubes يتم فيه تبادل حرارة بين cold & hot fluid

H.E ⇒ heat exchanger

H.E Calculations

Thermal & hydraulic calculations

Mechanical calculations

* hydraulic ⇒
هو العلم الذي يدرس الخواص الميكانيكية للسوائل
ويدرس القوى المؤثرة على السوائل.

* Mechanical ⇒ mainly stress on pipes ...

رح نركز على ال
Thermal & hydraulic

Heat exchanger problems

Rating

من المعروف أن الـ H.E
تقل كفاءته مع الزمن ، فيتم عمل
فحوصات معينة لاتخاذ قرار
استبدال الـ H.E بآخر أو
يتم عمل تعديلات بتصميمه .
وذكر كيف يتم عمل الفحوصات ؟
بيشوف الـ performance (الأداء)
لـ H.E حسب الـ (المواصفات)
الموجودة specification .
ويجب data بقارنها مع الـ
performance إذا لم تطابق التقارير
يكون في مشكلة بـ heat exchanger

* يعني من هذه المشاكل بقرر أغلق الـ H.E أو استبداله .

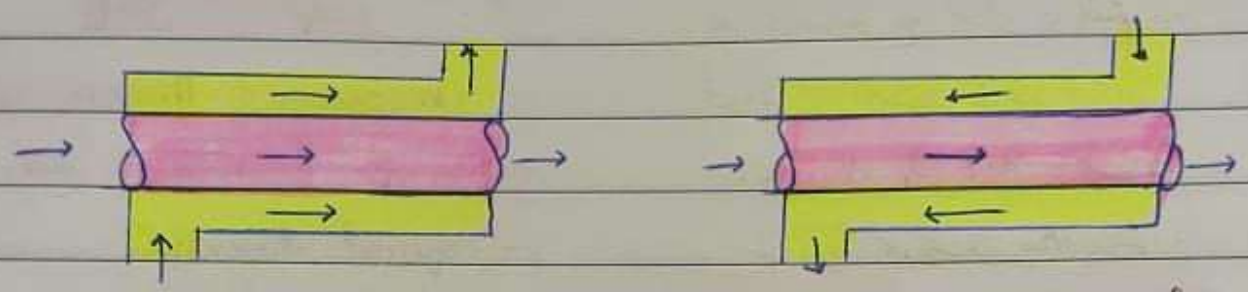
Design

بجيب 2 stream
hot cold
بجيب temperature flowrate
وبطلع الـ specification بالشكل :
- required dimension
- number of tubes
- total surface area
- h overall مهمة جداً

* أبسط أنواع الـ heat exchanger :

- Double - pipe equipment

يتكون من : 2 tubes



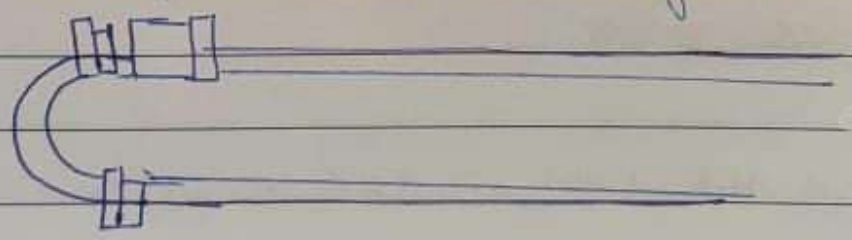
parallel or (co-current) flow counter - current flow

— Inner fluid

— Annulus fluid

Concentric tube exchanger

* Simple double-pipe exchanger (one hairpin)



هذا وهو فقط hair pin والمطلوب تجيب من الـ hairpin

* characteristic of hairpin exchanger :

- Batteries of hairpins connected in series or in series-parallel arrangements are commonly employed to provide adequate surface area for heat transfer.

- The outer pipe may be insulated to minimize heat transfer to or from the environment.

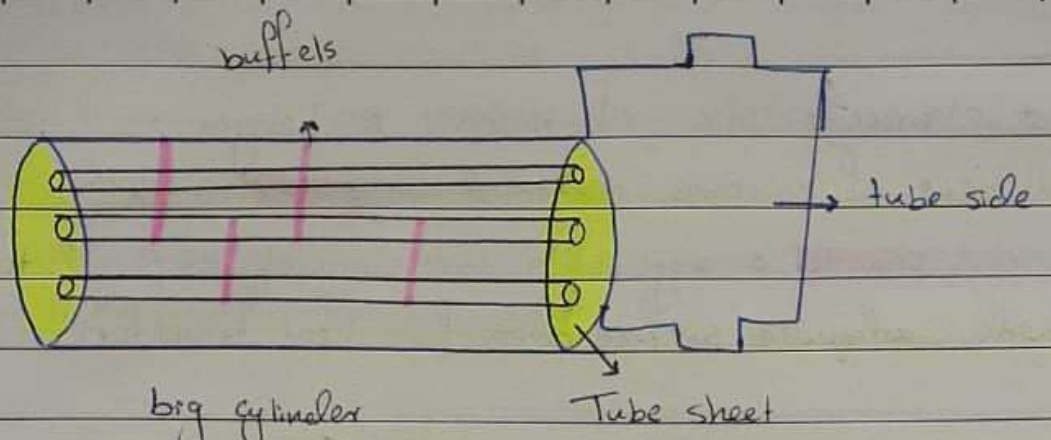
مهم جداً

- Nozzles may be provided on the inner pipe (tube) side as well as on the annulus side to facilitate connection to process piping.

يمكن توفير فوهات على جانب الأنبوب الداخلي وكذلك على جانب الحلقة لتسهيل التوصيل بأنابيب المعالجة.

- Multi-tube exchanger are also available in which the inner pipe is replaced by a bunelle of U-tubes.

تتوفر أيضاً مبادلات متعددة الأنابيب حيث يتم استبدال الأنبوب الداخلي بحزمة من أنابيب U.



* أهميه ال tube sheet :

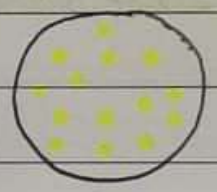
- لتثبيت ال tubes

- حتى ما يختلط ال fluid ال tube side & shell

Calculations

shell side	tube side
الشل الخارجي	التنابيب الداخلية

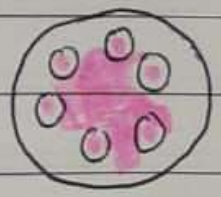
* Tube sheet : عبارة عن disk تحتوي على فتحات بعدد ال tubes



Tube sheet

⇒ تحتوي على فتحات لتثبيت ال tubes وركب عدد الفتحات يجب أن يكون محدود (standard) حتى ما تضعف أو تكسر ال sheet

* Multi - tube hairpin exchanger:



⇒ Cross section area of fin tubes inside shell أهميه زيادة ال surface area

- Simple double-pipe exchangers are commerically available with outer pipe sizes ranging from 2 to 8 in and inner pipes from 3/4 to 6 in.

يتميز باستعمال هذا الـ size

- Multi-tube units with outer-pipe sizes as large as 36 in are commerically available. أكبر قطر ممكن 36 in

- Double-pipe exchangers are commonly used in applications involving relatively low flow rates & high temperatures or pressure, for which they are well suited.

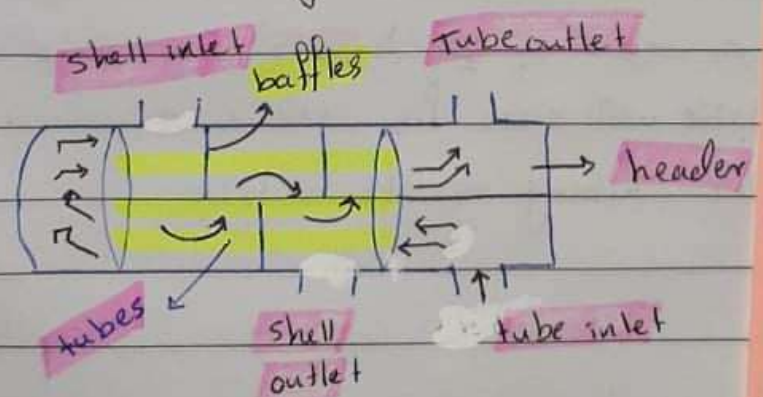
يستخدم في المصانع الصغيرة (low capacity)

- Other advantages include low installation cost, ease of maintenance, and flexibility.

أسهل الصيانة — لا يحتاج مساحة كبيرة

* Shell and Tube Equipment

A shell and tube exchanger consists of a bunelle of tubes contained in a cylinelrical shell.



- Tubes could be fixed or removable.
- Tube sheets are used to fix the tubes & to prevent fluids mixing
- A number of different head & shell designs are commerically available

TEMA designations for shell & tube exchanger: عبارة عن شركة
فخصصوا لتحديد ال H.E

Front end Head types	Shell types	Rear end types
تحتوي على أشكال	تحتوي على أشكال	تحتوي على أشكال
برموز مختلفة	برموز مختلفة	برموز مختلفة

* يتم تركيب H.E من 3 رموز.

*Notes:

- The tubular exchanger Manufacturers Association (TEMA) employes a three-latter code to specify the front-end & shell & rear end types.

من المفترض على كودات ال H.E ← AES

• Shells ⇒ are avilable with inside diameters in discrete size up to 120 in. Shells up to 24 in. in diameter are generally made from steel pipes, while larger sizes are made from rolled steel plate.

shells ال يجيب 24in أقل من 120in إلى size of inside diameter ↑ قد يصل
جاهزين، أما إذا كان ال diameter أكبر من 24in يصنع ال shell من
rolled steel plate (يعني بلق شيت بالـ Diameter المطلوب)

• Tubes ⇒ are available with sizes from 1/4 to 2 in. outside-diameter. Finned tubing is also avilable.

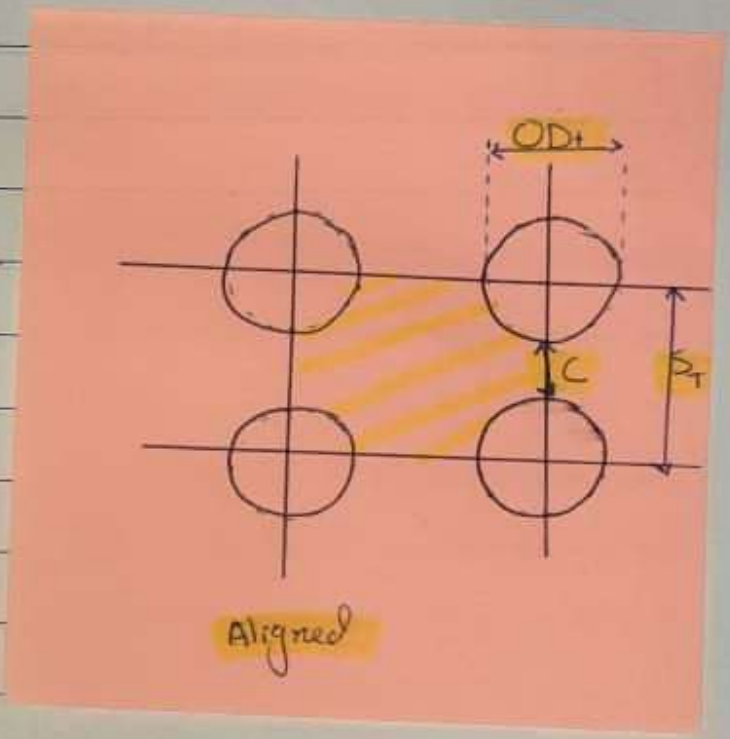
31th mar - 2022

* حسابات ل heat transfer coefficent في ل shell side :

- Equivalent diameter for shell side :

$$De = \frac{4\,Area}{wetted\ perimeter}$$

$$De = \frac{4\left(S_T^2 - \pi (OD_t)^2/4\right)}{\pi (OD_t)}$$



كيف اشتقيت القانون ؟!

1 المساحة عبارة عن مربع مطروح منه ٤ أرباع دوائر

يعني عندي ٤ دوائر كل دائرة ¼ منها ضمن مساحة المربع .

مساحة المربع عبارة عن ^{2}(طول الضلع)

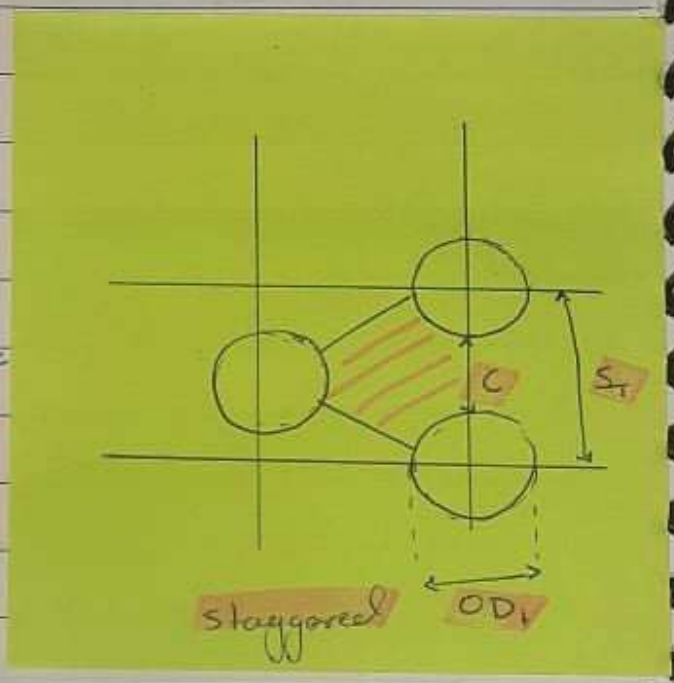
$= S_T^2$

2 المحيط ← الأجزاء المبينة هي ٤ أرباع يعني دائرة كاملة لذلك المحيط عبارة عن محيط دائرة

ولكن المربع راح منه ٤ أجزاء عبارة عن ربع دائرة ، وإذا جمعت ال ٤ أجزاء رح يصير دائرة كاملة لذلك المساحة عبارة عن مساحة مربع - مساحة دائرة .

كيف اشتقيت القوانين ؟!

$$D_e = \frac{4\left((S_T/2)(.86 S_T) - \pi (OD_1)^2/8\right)}{\pi (OD_1)/2}$$



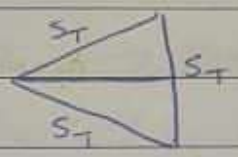
التشكل الناتج من التوصيل بين الدوائر هو مثلث متساوي الأضلاع ويعني زواياه $= 60°$

المساحة عبارة عن مثلث مطروح منه أجزاء من أرباع الكرات.

مساحة المثلث $= \frac{1}{2} \times base \times high$

القاعدة $= S_T$

الارتفاع $= .86 S_T$



نظرية فيثاغورس:

$$(S_T)^2 = \left(\frac{1}{2} S_T\right)^2 + (high)^2$$

$$.75 S_T^2 = (high)^2$$

$$high = .86 S_T$$

* الأجزاء المحذوفة من المثلث هي نص دائرة والسبب أن مجموع الزوايا حول نقطة $= 360°$ وهي الدائرة، والمثلث الناتج مجموع زواياه $180°$ وهو $\frac{1}{2}$ مجموع زوايا الدائرة

$\therefore$ Area $=$

$$\frac{1}{2} \times base \times high - \underbrace{\frac{1}{2}}_{\text{نص دائرة}} \left(\pi \frac{OD^2}{4}\right)$$

$$= \left(\frac{1}{2} S_T \times .86 S_T\right) - \frac{1}{8}(\pi OD^2)$$

* wetted perimeter =

محيط $\frac{1}{2}$ الدائرة

$$= \frac{1}{2}(\pi (OD_1))$$

* Baffles:

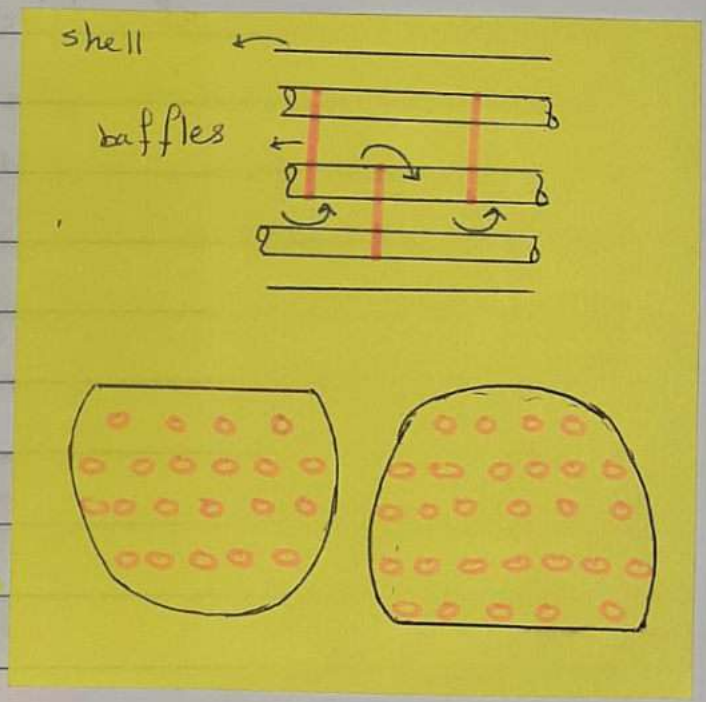
- Baffles are a number of discs installed inside the shell side of the H.E.

- Used to support the tube bundle and to withstand vibrations and bending

- Typical types are:

Segmental baffles

عبارة عن disk جزئي بها فتحات
يمر فيها ال tubes ، وبتشيل 25%
تقريباً من مساحة ال disk لتصبح
بشكل ، الشكل ▭ أهميته هذا الشكل
حتى يمر من خلالها ال fluid
وتثبت ال tubes وتغير اتجاه حركة
ال fluid من الشرق إلى الغرب وهكذا
ويتعرض ال fluid لأكبر مساحة
ممكنة من ال tubes ، ويحدث
اضطراب (turbulent) يزيد من
ال heat exchanger



* عيوب ال Baffles :

- Increase pressure drop

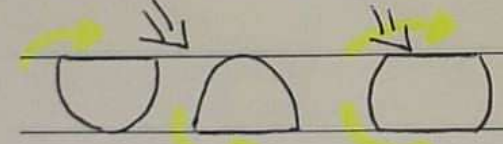
لازم أحدد عدد معين من ال baffels لأن زيادته تزيد من ال pressure drop

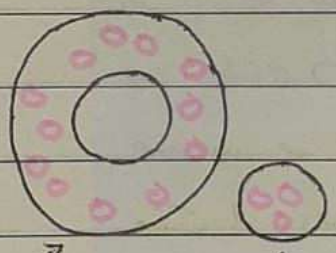
وأيضاً من العوامل المؤثرة

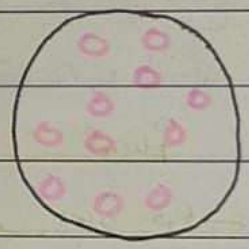
- المسافات بين ال baffles وطول ال disk

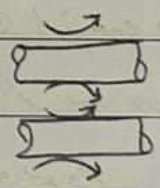
Types of baffles

- Cut - Segmental baffle
  - Single Segmented baffle
  - Double Segmented baffle
- Disk and doughnut baffle
- Orific baffle









* تشبه الدونات حركة ال fluid من خلال الفتحة بالمنتصف

* تكون حركة ال fluid متزن أو المتسقي حسب القطع

يتم زيادة قطر الفتحة عن قطر ال tube بحيث يصبح هو اللي ال tube شاغل (gap) يسمح بمرور ال fluid من خلاله

* أهمية الـ Baffle :

- Change direction of fluid
- Increase the mixing ⇒ Increase heat transfer

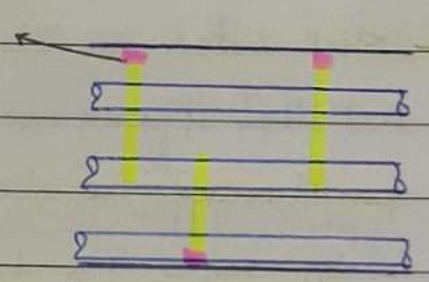
* Notes:

The baffles serve the function of directing the flow of the shell side fluid across the tube bundle, thereby enhancing the rate of heat transfer

* Notes:

Sealing strips (metal strips attached to the baffles) are used to minimize the channeling of fluid between the outer row of tubes and the shell.

شرائح معدنية متصلة بالـ baffles لتقليل تدفق السوائل بين الصف الخارجي للأنابيب والـ shell



* Tube and shell passes:

- Tube side:

▷ multiple passes are achieved by means of U-tubes or by partitioning the headers.

▷ The number of tube-side passes is usually one, two, four or six, but may be as high as 16.

المقصود بال passes ⇐ الممرات التي يسلكها ال fluid داخل ال tube من ال entrance لنهاية (مسار).

- Shell side:

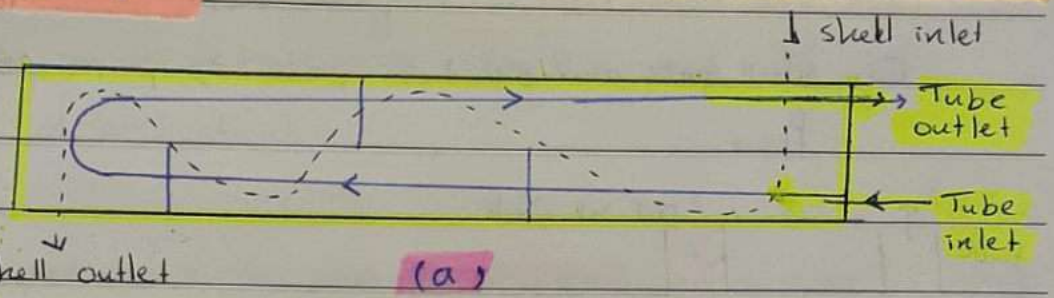
▷ Multiple passes on the shell side are achieved by partitioning the shell with a longitudinal baffle (type F-shell) or by connecting two or more single-pass shells together.

◁ Number of shell-side passes is usually between one and six.

* أهمية المسارات المتعددة ⇐ التقليل من ال space area وبالتالي التقليل من length of heat exchanger.

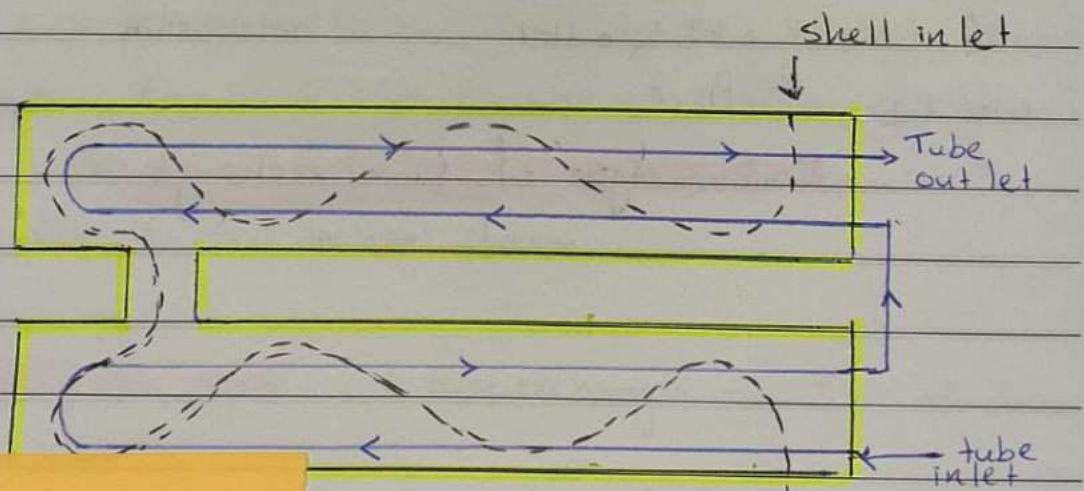
* يمكن تحقيق multipasses
لل shell :
- عن طريق ربط ال shell
ببعضها البعض (series)

a ⇒
1-2 heat exchanger



(a)

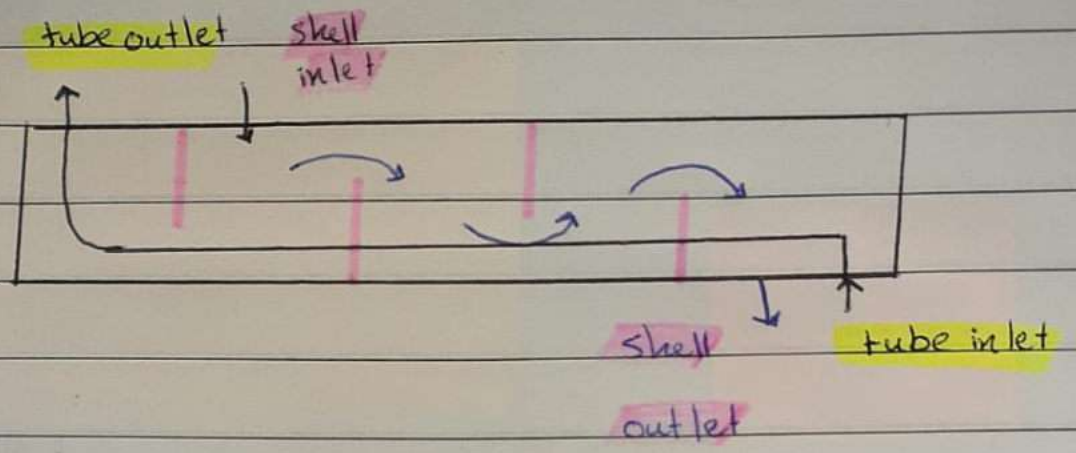
* تحتوي على مسارين داخل ال tube
* بمجرد تغيير اتجاه الحركة يظهر مسار جديد
* تحتوي على مسار واحد لل shell



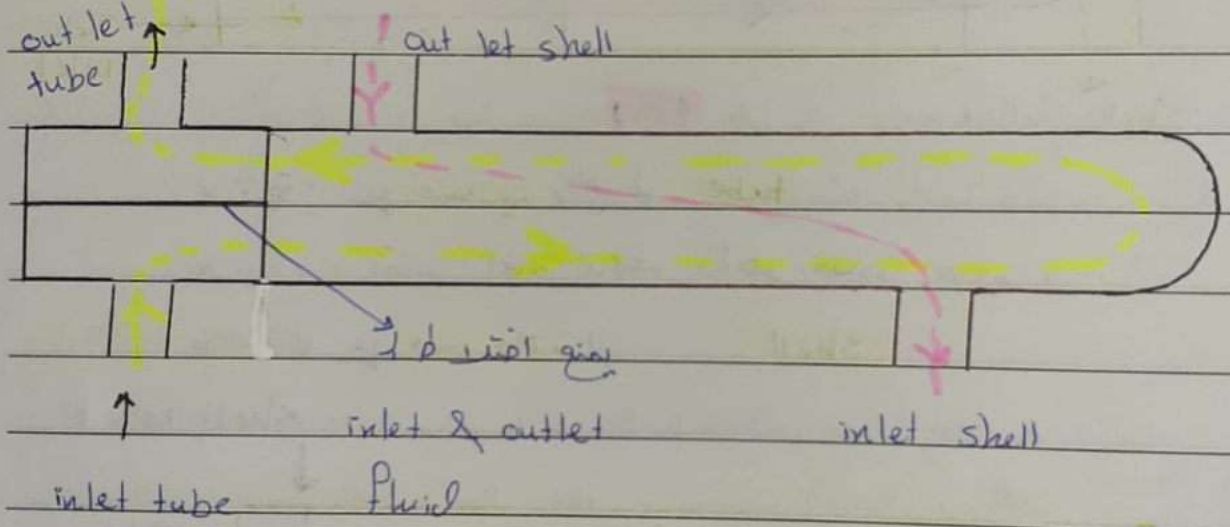
(b)

* تحتوي على ٤ مسارات داخل ال tube
* " " مسارين داخل ال shell

b ⇒
2-4 heat exchanger
where :
2 ⇒ number of shell pass
4 ⇒ number of tube pass



One shell pass and one tube pass ⇒ pure counter flow



Common type of (1-2) exchanger

* Notes:

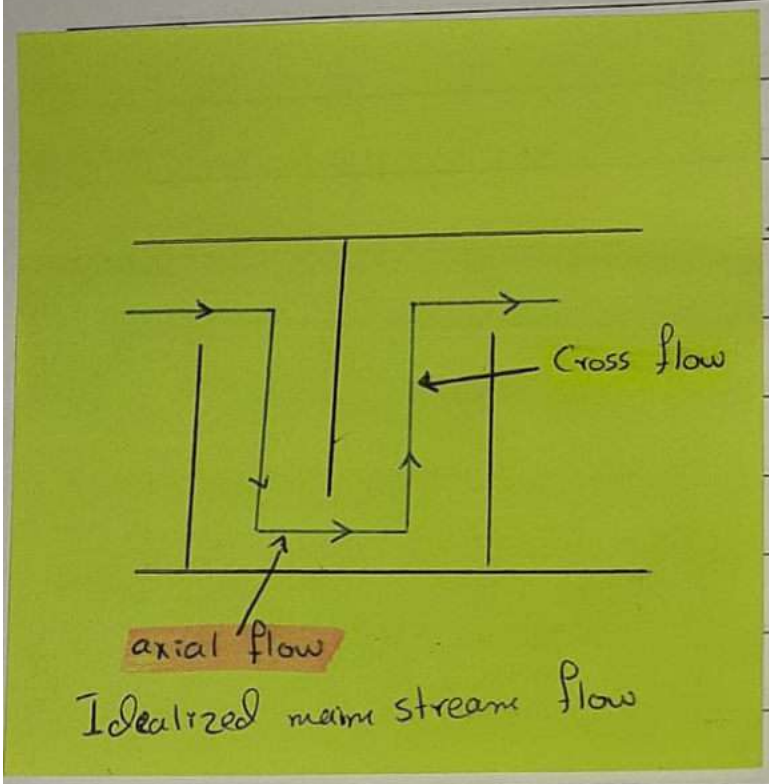
- The main advantage of shell and tube exchanger is that they provide a relatively large amount of heat transfer surface per unit of volume and weight, and require a minimum number of connections. They are extremely versatile, and can be designed to meet almost any heat transfer service. As a result, they are used in a wide variety of applications.

- Flow pattern in the shell is a sinuous motion both transverse and parallel to the tubes.

--- up-down-up ← baffle حول ال fluid حركة ال

- Streams bypass: part of the fluid bypasses the main heat transfer surface as a result of various leakages. The bypass streams include the tube-to-baffle leakage stream, the tube bundle to shell bypass stream, and the baffle to shell leakage stream

(تسرب) leakages ← heat exchanger ال مشاكل



* Shell side leakage and bypass:

- Tube to baffle leakage stream
تسرب هدايي بين ال tube & baffle
- actual cross-flow stream
- bundle to shell bypass stream
- baffle to shell leakage stream.
- partitioning longitudinal

* Shell - Side Bypass flows:

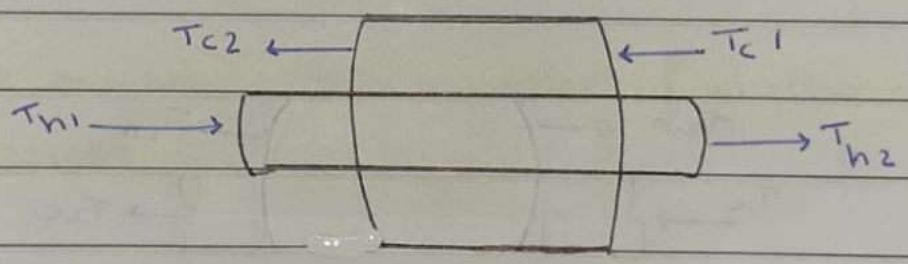
Stream	Percentage in turbulent flow	Percentage in laminar flow
Cross flow	30 - 65	10 - 50
Bundle bypass	15 - 35	30 - 80
Baffle - shell leakage	6 - 21	6 - 48
Tube - shell leakage	9 - 23	0 - 10

Look:

- The main stream flowing across the tube bundle may comprise less than 50% of the total shell-side flow due to bypass flow.
- The presence of these bypass streams complicates the analysis of shell-side heat transfer and pressure drop.

* The overall Heat-Transfer Coefficient:

- Consider the double-pipe section as shown below.
- Let the hot fluid (arbitrarily) to flow through the inner pipe.



* Explain the heat transfer from hot to cold stream.
* To describe this overall process, an **overall heat transfer coefficient, U,** is defined by:

$$q = U A \Delta T_m \rightarrow \text{driving force}$$

بتعني ال area للسطح الخارجي من الحرارة

في المثال السابق بتعني ال area لل hot tube

وهي ال area اللي بتوصلها في الخارج

بالنسبة لل area = $\pi D L$ ولكن الأنبوب له

inside diameter & out side diameter وإذا استخدمت كل منهم صحيح ،

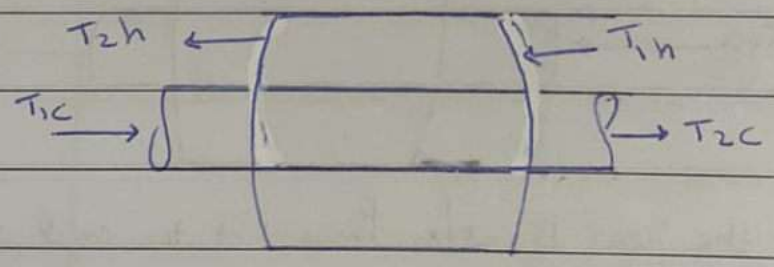
ولكن عملياً تم تقديم استخدام D_o (out side)

$\therefore Area = \pi D_o L$

* Heat transfer area:

The heat transfer area, A, is the surface area of the inner pipe, and may be based on either the inside and outside diameter. In practice, however, the out side diameter is commonly used, so that:

$$A = A_o = \pi D_o L$$



في هذه الحالة بما أن ال heat تنتقل من hot → cold

$A = \pi D_i L$

لل tube الخارجي

وفي هذه الحالة يجب عمل insulation على سطح

الأنبوب الخارجي للتقليل من ضياع الحرارة

* Temperature difference:

- The temperature difference, ΔT_m, is the mean temperature difference between the two fluid streams.

- It can be shown that when U is independent of position along the exchanger, ΔT_m is the logarithmic mean temperature difference.

$$\Delta T_m = \Delta T_{ln} = \frac{\Delta T_2 - \Delta T_1}{\ln(\Delta T_2 / \Delta T_1)}$$

- where ΔT_1 & ΔT_2 are the temperature difference at the two ends of the exchanger.

- The above equation is valid regardless of whether counter flow or parallel flow is employed.

* Thermal resistance & OHTC "U"

OHTC ⇒ over all heat transfer coefficient

- Thermal resistance:

$$R_{th} = \frac{1}{UA}$$

$$\frac{1}{UA_o} = \frac{1}{h_i A_i} + \frac{\ln(D_o/D_{in})}{2\pi k L} + \frac{1}{h_o A_o} \quad \text{--- } 1$$

المقاومة بين inner fluid & inner tube (inner diameter)

المقاومة بين قطر الانبوب الداخلي

المقاومة بين outer fluid & outer surface (inner tube)

where: h_i & h_o are the heat transfer coefficients for flow in the inner & outer pipes

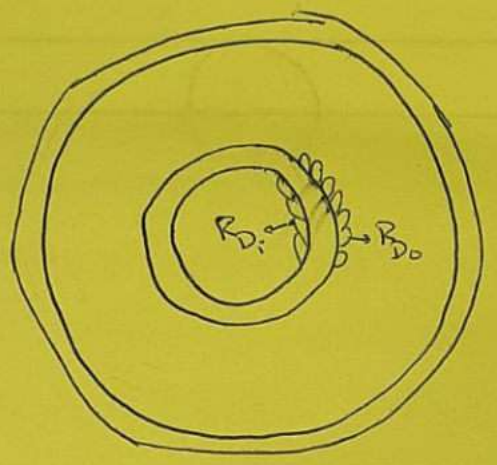
A_i ⇒ surface area of the inner pipe based on the inside diameter. $A_i = \pi D_i L$

$$U = \left[\frac{D_o}{h_i D_i} + \frac{D_o \ln(D_o/D_i)}{2k} + \frac{1}{h_o}\right]^{-1} \quad \text{--- } 2$$

- Equation 2 is correct when the heat exchanger is new & the heat transfer surfaces are clean.

لهيك اشتغل b هون هذه المعادلة إذا كان ال heat exchanger new & surface clean

لدنه مع كثرة استخدام ال H.E يتكون على سطحه تكلسات تزيد من مقاومة انتقال ال heat .



Scales build up
(تكلسات)

* Adding these two additional resistances to the main OHTC equation :

$$U_D = \left[\frac{D_o}{h_i D_i} + \frac{D_o \ln(D_o/D_i)}{2k} + \frac{1}{h_o} + \frac{R_{D_i} D_o}{D_i} + R_{D_o}\right]^{-1}$$

where :

U_D is the overall coefficient after fouling has occured.

- It is necessary to provide more heat transfer area than is actually required when the exchanger is clean.

Subject :

* مع مرور الزمن يظهر داخل الـ H.E تكلسات ، تعيق انتقال الحرارة ،
يتم حساب هذه المقاومات experimental ، وتوجد في tables حسب نوع
الـ fluid المسبب للتكلس .

fouling mechanism

- Corrosion
- Sedimentation
- Crystallization
- Biological activity
- Decomposition
- polymerization

*ex: A double pipe heat exchanger will be used to cool a hot stream from 350°F to 250°F by heating a cold stream from 80°F to 120°F. The hot stream will flow in the inner pipe, which is 2-in. schedule 40 carbon steel with a thermal conductivity of 26 Btu/h.ft².°F fouling factors of .001 h.ft².°F should be provided for each stream. The heat transfer coefficients are estimated to be $h_i = 200$ & $h_o = 350$ Btu/h.ft².°F and the heat load is 3.5×10^6 Btu/h.

a) For counter-current operation, what surface area is required?

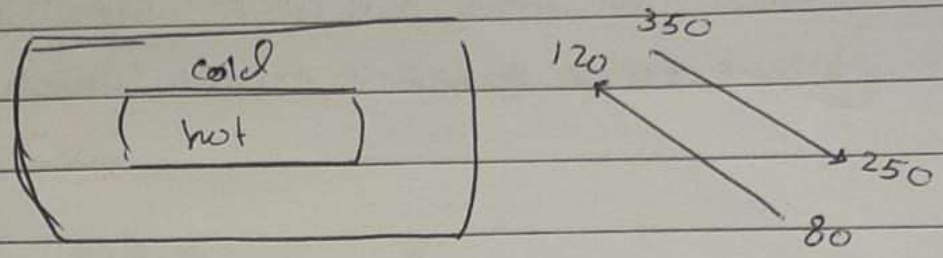
b) for co-current operation, what surface area is required?

Subject :

بسم الله

sol^n

a)



from table:

$D_i = 2.067$ in $\quad D_o = 2.375$ in

$$\Delta T_{lm} = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1/\Delta T_2)}$$

$\Delta T_1 = 350 - 120 = 230$

$\Delta T_2 = 250 - 80 = 170$

$$= \frac{230 - 170}{\ln(230/170)} =$$

$\Delta T_{lm} = 198.5\ ^\circ F$

$$U_D = \left(\frac{D_o}{h_i D_i} + \frac{D_o \ln(D_o/D_i)}{2k} + \frac{1}{h_o} + R_{D_i}\frac{D_o}{D_i} + R_{D_o}\right)^{-1}$$

$$= \frac{2.375}{2.067(200)} + \frac{2.375 \ln(2.375/2.067)}{2(26)}$$

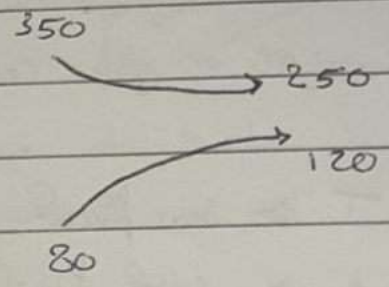
$$+ \frac{1}{350} + .001\left(\frac{2.375}{2.067}\right) + .001\Big)^{-1}$$

$= 88.65\ Btu/h \cdot ft^2 \cdot ^\circ F$

$$A = \frac{q}{U \Delta T_{lm}} = \frac{3.5 \times 10^6\ Btu/h}{88.65\ Btu/h \cdot ft^2 \cdot ^\circ F\ (198.5)^\circ F} = 198.9\ ft^2$$

Subject :

b) co-current



$\Delta T_1 = 350 - 80 = 270$

$\Delta T_2 = 130$

$$\Delta T_{lm} = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1 / \Delta T_2)}$$

$$\Delta T_{lm} = \frac{270 - 130}{\ln(270/130)} = 191.5^\circ F$$

$$A = \frac{3.5 \times 10^6}{88.65(191.5)} = 206 \ ft^2$$

* Comments:

A co-current > A counter

* كلما حصلت على area أقل في حال ثبات الـ q يكون أخفض من ناحية cost .

$q = U A \Delta T_{lm}$

ثوابت

$A\uparrow \ \Delta T_{lm}\downarrow$ & $\Delta T_{lm}\uparrow \ A\downarrow$

الزيادة في واحدة يقابلها نقص بالثانية

& $\Delta T_{lm} \Rightarrow$ driving force for heat transfer

فبزيادة الـ Driving force يكون الوضع أفضل

* في حالة الـ co-current بقدر أرفع درجة حرارة الـ exit بتكون أخفض من counter

Subject : _____ / /

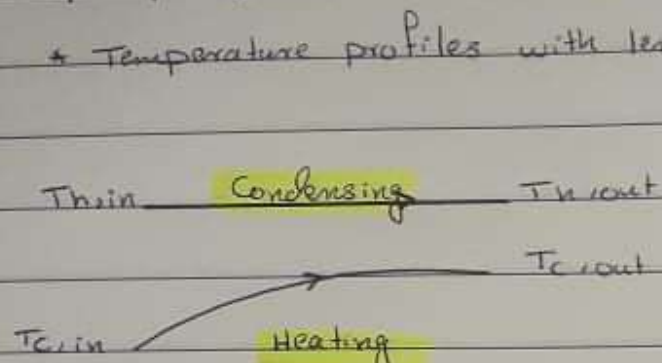
* في مرحلة ال design ← ال counter ; فرض من ناحية area

T ⇒ hot fluid ال تدفق ال

t ⇒ cold fluid " "

5th-Apr-2022

* Temperature profiles with length:



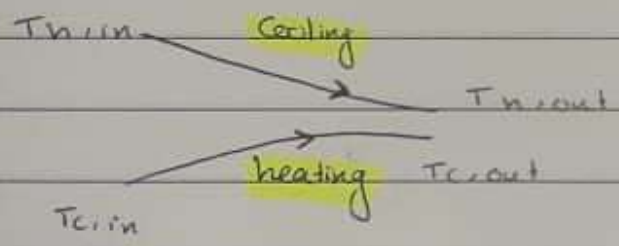
1

Condensor , change phase

hot stream ⇒ condensation

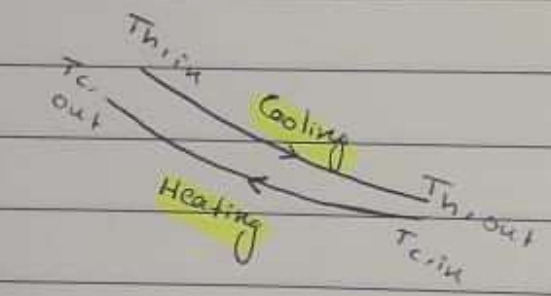
الطرف الذي يتقدم يكثف ال

cold fluid



2

Single phase , Co-current

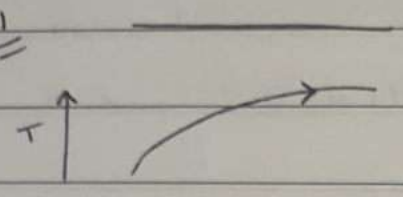


3

Counter , single phase

* Special operating Conditions:

1



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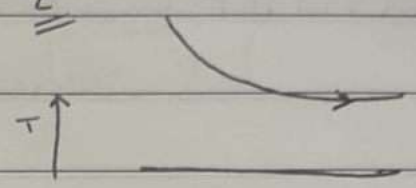
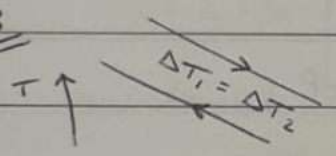


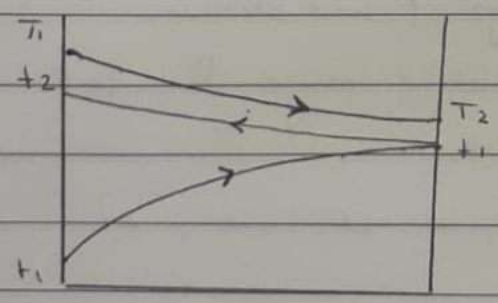
لم يحدد لل hot stream اتجاه ، طبعاً انه ما بيفرق إذا كان Counter or co-current لهم نفس ال performance

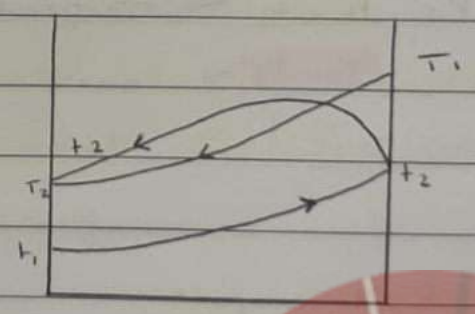
نفس الشي ما حددت اتجاه لل cold stream لنفس السبب

3



* The LMTD Correction factor





Combination between Counter & co-current هي عبارة عن

* كيف يكون في التركيب (combination) ؟

بنفرض (counter or co-current) pure ، يعني H.E (11) ،

ثم نحسب (LMTD) وبعدين correction .

$\Delta T_m = F\,(\Delta T_{lm})$ → pure

→ combination

where: F ⇒ Correction factor

* كيف نحسب ال F :-

N ⇒ # of shell-side passes

$R = \frac{T_a - T_b}{t_b - t_a}$ → Temperature range of hot stream

$t_b - t_a$ → Temperature range of cold stream

$P = \frac{t_b - t_a}{T_a - t_a}$ → Temperature range of cold stream

$T_a - t_a$ → maximum temperature range for H.E

بعد ما أحسب هذه النسب بطبق على قوانين :

for $R \neq 1$, compute

$$\alpha = \left(\frac{1 - RP}{1 - P}\right)^{1/N}$$

$$S = \frac{\alpha - 1}{\alpha - R}$$

$$\therefore F = \frac{\sqrt{R^2+1}\ \ln\left(\frac{1-S}{1-RS}\right)}{(R-1)\ \ln\left[\frac{2-S(R+1-\sqrt{R^2+1})}{2-S(R+1+\sqrt{R^2+1})}\right]}$$

* for $R=1$, compute

$$S = \frac{P}{N-(N-1)P}$$

$$F = \frac{S\sqrt{2}}{(1-S)\ \ln\left[\frac{2-S(2-\sqrt{2})}{2-S(2+\sqrt{2})}\right]}$$

معادلات F لا تستخدم في الحل وفائدتها في كتابة البرامج فقط

أما في مجال الهندسة بستخدم الـ chart method في إيجاد قيمة F

* يوجد 3 charts تمثل قيمة F أو الـ chart المستخدم باختلاف

# of shell passes والتعقيد في # of tube passes

* ملاحظة:

H.E (1-2) بستخدم الـ chart 1

H.E (1-4) " " " "

يعني مش مهم # of tube passes أما إذا تغير # of shell passes

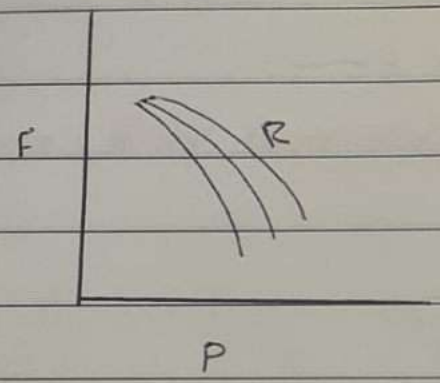
بشوف chart تانية بطلع منه قيمة F

* الرسومات لـ 3 وحدة لـ One shell pass

الثانية لـ two shell pass

الثالثة لـ three shell pass

بغض النظر عن # of tube passes



بطلع من P خط عامودي يتقاطع مع R ثم بمد خط أفقي يتقاطع من F وبأخذ قيمة F

* ما تأثير قيمة F على الـ H.E ؟!

$q = U A F (\Delta T_m)$

افرض قيمة أن تكون قيمة F = 1 ، يكون الـ H.E (pure)

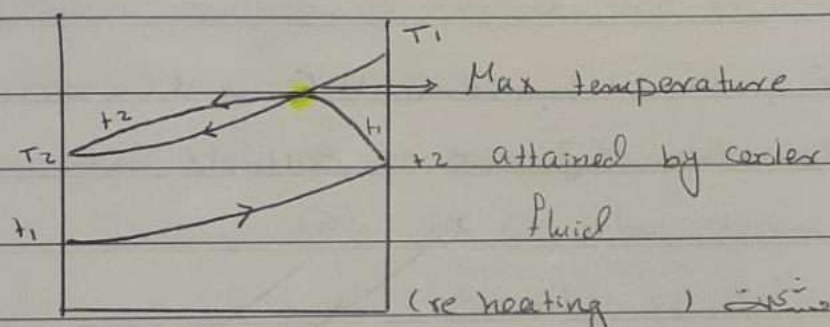
وكلما قلت قيمة F قل مقدار الـ heat transfer

∴ Note:

F must be not less than 0.8

* في عمليات الـ design بيبدأ دائماً من H.E (1-2) ، وإذا كنت بدي قيمة F أعلى بنتقل إلى 2 shell لزيادة قيمة F ، وإذا لم يفي بالغرض بنتقل إلى 3 shell إلى أن أحصل على قيمة F المرغوبة.

# of shell ↑ ⇒ F↑



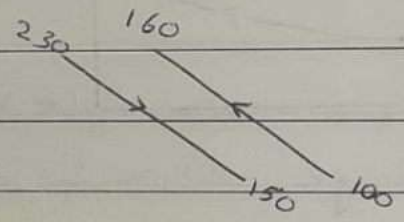
بعد هذه النقطة زاوية مشكلة (re heating)

لذلك مع زيادة # of shell بحول هذه النقطة بحيث أحصرها بـ range ضيق.

* Ex 1: A fluid is to be heated from 100°F to 160°F by heat exchange with a hot fluid that will be cooled from 230°F to 150°F, The heat transfer rate will be 540000 Btu/h and the hot fluid will flow in the tubes. Will a (1-2) exchanger (an exchanger with one shell pass and a multiple of two tube passes) be suitable for this servic?
Find the mean temperature difference in the exchanger.

لما يسأل عن 1-2 انه مناسب او لا ، نحسب قيمة F واذا كانت أعلى
بكثير suitable واذا كانت أقل not suitable وبده تغيير.

N = 1



soln:

$$R = \frac{T_a - T_b}{t_b - t_a} = \frac{160 - 100}{230 - 150} = .75$$

$$P = \frac{t_b - t_a}{T_a - t_a} = \frac{150 - 230}{100 - 230} = .615$$

نجد قيمة F من chart (1-2) exchanger :

F = .72 < .8 , not suitable & should be not used

Two shell passes ⇒ F = .94 will be suitable

* **Mean temperature difference**

$\Delta T_1 = 230 - 160 = 70°F$

$\Delta T_2 = 150 - 100 = 50°F$

$\Delta T_m = F\ (\Delta T_{ln})$

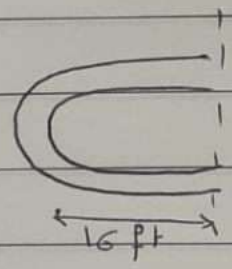
$= .94 \left(\frac{70-50}{\ln(70/50)} \right) \simeq 56°F \Rightarrow$ driving force for H.E

* ex 2: **Double pipe thermal analysis:**

Thermal analysis of a double-pipe exchanger is illustrated in the following example:

10000 lb/h of benzene is to be heated from 60°F to 120°F by heat exchange with an online stream that will be cooled from 150°F to 100°F. A number of 16 ft hairpins consisting of 2 in. by 1.25 in. schedule 40 stainless steel pipe (type 316, k = 9.4 Btu/h.ft.°F are available and will be used for this service. how many hairpins will be required?

Soln



إذا بدي طول ال pipe كامل

بضربه 16x2

طول ال H.E بأسوء الحالات بصل إلى 10m ويفضل أقل

* reference of length for hairpins = 16, 18, 20

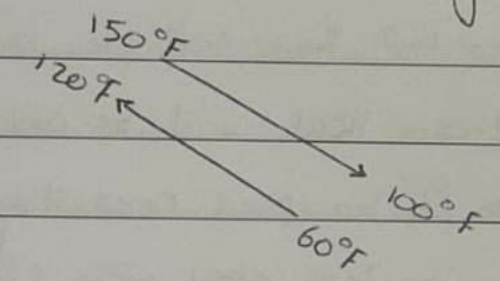
* Basic Assumptions:

- Assume heat from hot stream to cold stream
- Assume benzene flows in the inner pipe

عكس وفرض تدفق البنزين في ال outer pipe

- properties at average properties or according to the correlation conditions.

* Sketch T-x diagram



Mean temperature difference calculations

$$\Delta T = 50°\text{F} \left\{ \begin{array}{l} 100°\text{F} \longrightarrow 160°\text{F} \\ 150°\text{F} \longleftarrow 230°\text{F} \end{array} \right\} \quad \Delta T = 70°\text{F}$$

$$(\Delta T_{\ln})_{cf} = \frac{\Delta T_2 - \Delta T_1}{\ln(\Delta T_2/\Delta T_1)} = \frac{70 - 50}{\ln(70/50)} = 59.6°\text{F}$$

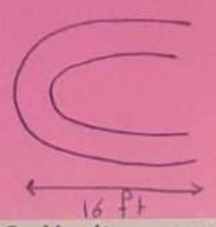
$$\Delta T_m = F(\Delta T_{\ln})_{cf} = 0.94 \times 59.6 = 56°\text{F}$$

* hairpins consisting of 2 in by 1.25 in.

⇒ by ⇒ (2in x 1.25in) تعني ضربه

2 in schedule ⇒ outer pipe

1.25 schedule ⇒ inner pipe



* what is the mass flowrate of aniline ?

- $q_{loss} = q_{accept}$

- $q_{aniline} = q_{Benzene}$

- $m\, c_p\, \Delta T = m\, c_p\, \Delta T$

?1 ✓ ✓ ✓ ✓ ✓

from table from table

herma

ble-pipe e

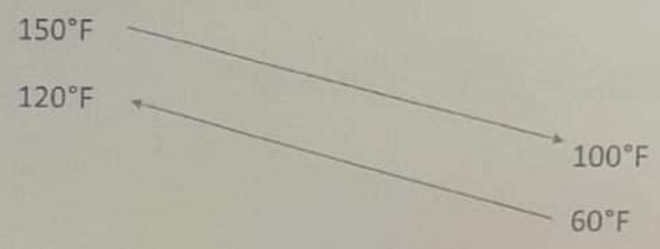
g example:

10,000 lb/h of benzene is to be heated from 60°F to 120°F by heat exchange with an aniline stream that will be cooled from 150°F to 100°F A number of 16 ft hairpins consisting of 2 in. by 1.25 in. schedule 40 stainless steel pipe (type 316, k = 9.4 Btu/h. ft. °F are available and will be used for this service. how many hairpins will be required?

Basic Assumptions

- Assume heat from hot stream = cold stream. (adiabatic)
- Assume benzene flows in the inner pipe.
- Properties at average properties or according to the correlation conditions.
- Assume counter current

Sketch T-x diagram



Solution _ Procesdure

Find properties for both streams

↓

find the heat load of the exchanger & unknown flow rate

↓

Calculate the LMTD

↓

Calculate h_i assuming $\phi_i = 1.0$. firstly find D_i from tables, then Re, then Nu. ⇒ ϕ_i = correction $\left(\frac{\mu_{avg}}{\mu_{sur}}\right)$ & prandtle Number

↓

check of developed & fully developed

Check entrance effect L/Di >60. In this example the length is not given, therefore assume one hair pin. Hence take L = 32 ft (one hairpin). If the ratio is greater than 60, you don't need to use entrance effects correction.

L = 32 = total length = 16×2

inside diameter for (2in)

Calculate ho assuming $\phi_o = 1.0$. Firstly find D_1, D_2 from tables. Then obtain D_e and A_f. Then find Re, Nu. → out side diameter for (1.25in)

↓

Check entrance effect L/De > 60. **Note** in the annulus the flow is disrupted at the return bends, so it is appropriate to use the length of pipe in one leg of a hairpin to estimate entrance effects.

↓

Calculate wall pipe temperature T_w See next slide. ⇒ energy balance

↓

Find ϕ_i , ϕ_o .

↓

Correct the values of h_i and h_o

↓

Obtain fouling Factors

↓

Compute the O.H.T.C

↓

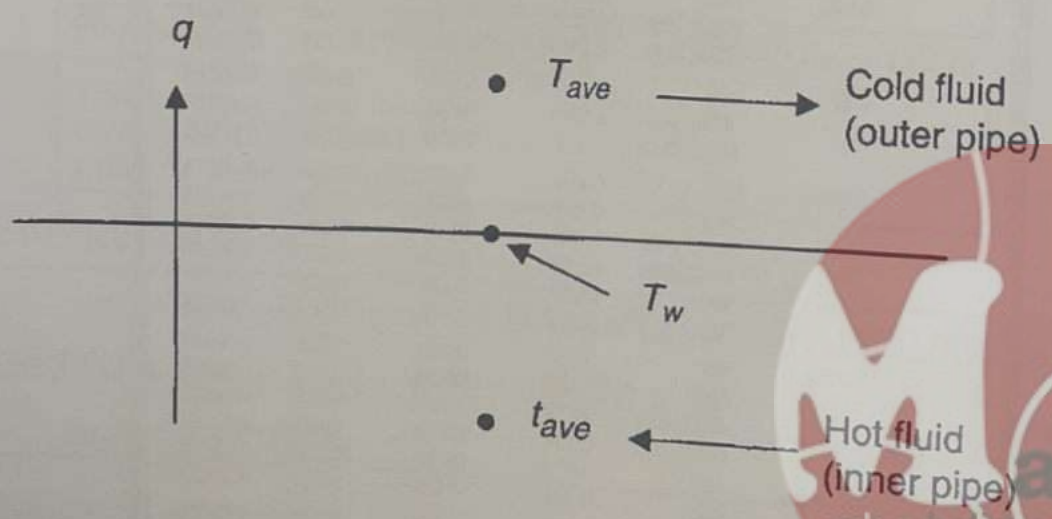
Calculate the required surface area. Then find L by using the table show in the next slides. Finally find the no. of hairpins.

* ليش فرضت قيمة ϕ_o & ϕ_i = 1.0 حسبها ؟!
لسبب لسبب T_{sur} ما لسبب

$$\phi = \frac{\mu_{avg}}{\mu_{sur}}$$

$$N = \frac{L \text{ total calculated}}{L \text{ of one hairpin}}$$ ⇒ $2\pi r L$ ← A من حسبها.

How to calculate T_w



An energy balance gives:

$$q = h_i A_i (t_{ave} - T_w) = h_o A_o (T_w - T_{ave})$$

q loss = q accept

Solving for T_w yields:

* فرضنا انه ما في thickness لـ metal
بين السائل (Benzene & anline

$$T_w = \frac{h_i A_i t_{ave} + h_o A_o T_{ave}}{h_i A_i + h_o A_o}$$

Substituting $A = \pi DL$ for the areas and rearranging, we obtain:

$$T_w = \frac{h_i t_{ave} + h_o (D_o/D_i) T_{ave}}{h_i + h_o (D_o/D_i)}$$

Criteria for Fluid Placement, in Order of Priority

Tube-side fluid	Shell-side fluid
Corrosive fluid	Condensing vapor (unless corrosive)
Cooling water	Fluid with large ΔT (>100°F)
Fouling fluid	
Less viscous fluid	
Higher-pressure stream	
Hotter fluid	

لاني بدي أعرض surface area لأقل

* القواعد العامة لاختيار موقع ال fluids

* يجب الأخذ بعين الاعتبار ال metal المستخدم

10th - Apr - 2022

* Heat exchanger 3 :

* Preliminary Design of Shell - and - Tube Exchangers :

تستخدم هذه الطريقة لـ :

Obtain the heat transfer area for the purpose of making a preliminary cost estimate of the exchanger.

* تؤخذ من الجداول قيمة U و A tables

* Ex 1 :

In a petroleum refinery, it is required to cool 30,000 lb/h of kerosene from 400°F to 250°F by heat exchange with 75,000 lb/h of gas oil, which is at 110°F. A shell-and-tube exchanger will be used, and the following data are available :

fluid property	kerosene	Gas oil
C_p (Btu/lbm·°F)	0.6	0.5
μ (cp)	0.45	3.5
k (Btu/h·ft·°F)	0.077	0.08

for the purpose of making a preliminary cost estimate, determine the required heat-transfer area of the exchanger.

Sol :

المطلوب

$q = U A (\Delta T_{lm})$

نحسبها من قانون — من ال — غير جاهزة بالسؤال

$\dot{m} c_p \Delta T$ — tables — قيم ال T المعطى

فقط 3 من أصل 4 درجات حرارة

a) Calculate the heat load & out let temperature by energy balances on the two streams.

$q = \dot{m} c_p \Delta T$

$q = 30{,}000 \frac{lb}{h} \times .6 \frac{Btu}{lbm \cdot °F} (400 - 250)°F$

$q = 2.7 \times 10^6$ Btu/h

$q_{loss} = q_{accept}$

$2.7 \times 10^6 \, Btu/h = 75{,}000 \frac{lb}{h} \times 0.5 \frac{Btu}{lbm \cdot °F} (T - 110)$

$T = 182°F$

b) Calculate the LMTD (assume pure counter current)

$\Delta T_1 = 400 - 182 = 218°F$

$\Delta T_2 = 250 - 110 = 140°F$

$(\Delta T_{lm})_{cf} = \frac{\Delta T_2 - \Delta T_1}{\ln(\Delta T_2 / \Delta T_1)}$

$= 176.13°F$

400 → 250°F

182°F ← 110°F

c) Calculate the LMTD correction factor

for the purpose of this calculation, assume that the kerosene will flow in the shell. This assumption will not affect the result since the value of F is the same, regardless of which fluid is in the shell.

$$R = \frac{T_a - T_b}{t_b - t_a} = \frac{400 - 250}{182 - 110} = 2.083$$

$$P = \frac{t_b - t_a}{T_a - t_a} = \frac{182 - 110}{400 - 110} = .248$$

*assume 1-2 exchanger

$F \simeq .94$

d) estimate U_D from table ⇒ U_D in range 20 - 35 Btu/h.ft².°F

Therefor take $U_D = 27.5$ Btu/h.ft².°F

range بأخذ أي قيمة ضمن ال

e) Calculate the required area

$$q = U A F (\Delta T_{lm})_{cf}$$

$$2.7 \times 10^6 \ \cancel{Btu/h} = 27.5 \frac{\cancel{Btu}}{\cancel{h} \cdot ft^2 \cdot \cancel{°F}} \ A \ (.94)(176.13) \ \cancel{°F}$$

$$A = 593 \ ft^2$$

* Rating a Shell-and-Tube Exchanger:

- Check of performance

بعدها بقدر أعرف H.E مناسب أو جيب مجهات ثاني

* تشبيه حسابات ال double-pipe exchanger وكمان اختلاف في ال shell & tube وجود عدد كبير من ال tubes أما ال double كان عندي ال one pipe فقط.

إذن بعمل assumption في حساب hi :

assumed that all tubes in the exchanger are exposed to the same thermal & hydraulic conditions. The value of hi is then the same for all tubes & the calculation can be made for a single tube.

- In computing the Reynolds number, the mass flowrate per tube must be used:

$$\dot{m}_{per\ tube} = \frac{\dot{m}_t}{n_t} n_p$$

where:

$\dot{m}_t$ = total mass flowrate of tube side fluid

n_p = # of tube side passes

n_t = # of tubes

بعدين نحسب ال hi متل ال double pipe

* المشكلة بـ ho ما بيربط : one correlation

* Method of shell-side coefficient, ho calculations:

1- Heat transfer Research, Inc method

غير مطابق بالـ coarse

2- Delaware method:

The method utilizes the graph of modified Colburn factor, j_H

هنا الـ factor (j_H) بطلعه من chart

- Read j_H from the graph & then compute ho from

$$h_o = j_H \left(\frac{k}{D_e}\right) P_r^{1/3} \left(\frac{\mu}{\mu_w}\right)^{.14}$$

- Appropriate fit for the curve is:

$$j_H = 0.5\,(1 + B/d_s)\,(0.08 Re^{.6821} + .7\, Re^{.1772})$$

- flow area across the shell:

$$a_s = d_s\, C\, B / 144\, P_T$$

where:

d_s ⇒ shell inside diameter in

C ⇒ clearance in

B ⇒ baffle spacing in

P_T ⇒ Pitch, in

as ⇒ shell flow area, ft

$N+1 = \frac{L}{B}$

هذا الشكل مهم في ال

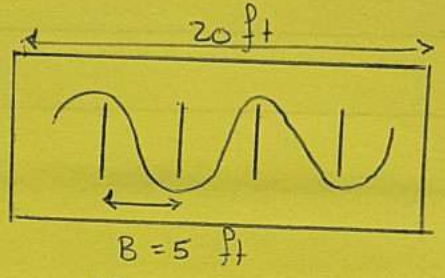
drop pressure

* عندما تقل B تزيد عدد ال

baffles بالتالي تزيد عدد ال

cross ← تزيد ال

drop pressure



B ⇒ baffle space

$\#$ of baffles $= \frac{L}{B} = \frac{20}{5} = 4$

N = number of cross :

$N = \frac{L}{B} - 1 \Rightarrow N+1 = \frac{L}{B}$

Shell mass flux

$G = m / a_s \Rightarrow lb/ft^2 \cdot h$

Reynolds number for shell side fluid

$Re = G D_e / \mu$

* Thermal rating procedure for a shell-and-tube heat exchanger:

1- Calculate the required overall coefficient

$$U_{req} = \frac{q}{A F (\Delta T_{lm})_{cf}}$$

2- Calculate the clean overall coefficient

$$U_c = \left[\frac{D_o}{h_i D_i} + \frac{D_o \ln (D_o / D_i)}{2k} + \frac{1}{h_o} \right]^{-1}$$

U_c ⇒ clean heat exchanger

أول م يتم صناعة ال H.E

3 Is $U_c > U_{req}$?!

yes ⇓ Continue

No ⇓ exchanger is not suitable

4 Obtain required fouling factors, R_{Di} & R_{do} then calculate R_D

from tables

$$R_D = R_{Di} (D_o / D_i) + R_{Do}$$

5 Calculate the design overall coefficient

$$U_D = (1/U_c + R_D)^{-1}$$

6 Is $U_D > U_{req}$?

yes ⇓ exchanger is thermally suitable

No ⇓ exchanger is not suitable

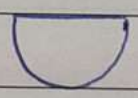
13th - Apr - 2022

* Ex 2:

30 000 lb/h of kerosene are to be cooled from 400°F to 250°F by heat exchange with 75 000 lb/h of gas oil which is at 110°F Available for this duty is a shell-and-tube exchanger having 156 tubes in a 21 1/4 in. ID shell. The tubes are 1-in OD, 14 BWG, 16 ft long on a $1\frac{1}{4}$ in. square pitch. There is one pass on the shill side and six passes on the tube side. The baffles are 20% cut segmental type and are spaced at 5 in. intervals. Both the shell & tubes are carbon steel having k = 26 Btu/h.ft².°F. fluid properties are given in the previous example. Will the exchanger be thermally suitable for this service ?!

fluid properties	kerosene	Gas oil
Cp (Btu/lbm.°F)	0.6	0.5
μ (cp)	0.45	.5
k (Btu/h.ft.°F)	0.077	0.08

BWG ⇒ Birmingham wire gauge

20% segmental baffles ⇒ 

يعني ال cross مقطوع من 20% من ال area

Soln

step 1

أول شي بحسب قيمة q من ال fluid المعطى لي بجميع القيم ، وكمان
الحرارة المفقودة = الحرارة المكتسبة بحسب T لل fluid اللي مو معطى
لها قيمة T ، ثم بحسب ΔT_m ، وبطبق قانون U_{req}

$q = C_p \dot{m} \Delta T$

$q = .6 \frac{Btu}{lbm \cdot °F} \times 30000 \frac{lb}{h} \times (400 - 250)°F$

$q = 2.7 \times 10^6$ Btu/h ⇒ from kerosene

q to gas oil = $C_p \dot{m} \Delta T$

$\frac{Btu}{h} \; 2.7 \times 10^6 = .5 \frac{Btu}{lbm \cdot °F} \times 75000 \frac{lb}{h} (T - 110)$

$T_2 = 182 °F$

400 ⟶ 250

182 ⟵ 110

$\Delta T_1 = 218$

$\Delta T_2 = 140$

$(\Delta T_m)_{cf} = \frac{218 - 140}{\ln(218/140)} = 176 °F$

ومن ال chart one pass of shell بطلع قيمة F

$F = .93$

- The surface area is obtained from the dimensions of the exchanger

$$A = 156 \text{ tubes} \times 16 \text{ ft} \times \underbrace{0.2618 \text{ ft}^2}_{\text{ft of tube}} = 653 \text{ ft}^2$$

table B.1 من

area / unit length الى

$$U_{req} = \frac{q}{A\,F(\Delta T_{ln})_{cf}} = \frac{2.7 \times 10^6}{653 \times .93 \times 176} = 25.3 \text{ Btu/h-ft}^2\cdot{}^\circ F$$

step 2 Calculate , U_c

- calculate the tube side Re

$$\dot{m} \text{ per tube} = \frac{\dot{m} \times \#\text{ of passes of tubes}}{\#\text{ of tubes}}$$

$$= \frac{75\,000}{156} \times 6 = 2885 \text{ lb/h}$$

$$Re = \frac{4\dot{m}}{\pi D_i \mu} = \frac{4 \times 2885}{\pi\left(\frac{.834}{12}\right) \times 35 \times 2.419} = 6243 \Rightarrow \text{transition regime}$$

assumbtion حسبت الـ Re في tube واحد لأنني عملت

⇒ all tubes are exposed to the same thermal

Calculate hi

$$Nu = \frac{h_i D_i}{k} = .116\left(Re^{2/3} - 125\right) P_r^{1/3} \left(\frac{\mu}{\mu_w}\right)^{.14} \left(1 + (D_i/L)^{2/3}\right)$$

$$h_i = .08 \times .116 \left(6243^{2/3} - 125\right) \left(\frac{.5 \times 3.5 \times 2.419}{.08}\right)^{1/3} (1) \left(1 + \left(\frac{.834}{12 \times 16}\right)^{2/3}\right)$$

$h_i = 110$ Btu/h·ft²·°F

* Note: we assume the viscosity correction factor = 1, since no data were given for the temperature dependence of the oil viscosity

Calculate the shell side Re

from figure 1 $d_e = .99$ in & $C' = .250$ in

$D_e = d_e/12 = .99/12 = .0825$ ft

$$a_s = \frac{d_s C' B}{144 P_T} = \frac{21.25 \times .25 \times 5}{144 \times 1.25} = .1476 \text{ ft}^2$$

$$G = \frac{\dot{m}}{a_s} = \frac{30000}{.1476} = 203294 \text{ lbm/h·ft}^2$$

$$Re = \frac{D_e G}{\mu} = \frac{.0825 \times 203294}{.45 \times 2.419} = 15407$$

Calculate h_o

$B/d_s = 5/21.25 = .235$

from figure 1 . $j_H = 40$

$$h_o = j_H \frac{k}{D_e} P_r^{1/3} \left(\frac{\mu}{\mu_w}\right)^{.14}$$

$$= \frac{40 \times .077}{.0825} \left(\frac{.6 \times .45 \times 2.414}{.077}\right)^{\frac{1}{3}}$$

$h_o = 76$ Btu/h.ft².°F

Calculate U_c

$$U_c = \left(\frac{D_o}{h_i D_i} + \frac{D_o \ln(D_o/D_i)}{2k} + \frac{1}{h_o}\right)^{-1}$$

$$= \left(\frac{1}{110 \times .834} + \frac{(1/12) \ln(1/.834)}{2 \times 26} + \frac{1}{76}\right)^{-1}$$

$U_c = 41.1$ Btu/h.ft².°F

$\therefore$ $U_c > U_{req}$, proceed to step 3

step 3 Obtain the required fouling factors. Taking .0025 for kerosene & .0035 for gas oil

$$R_D = \frac{R_{Di} D_o}{D_i} + R_{Do}$$

$$= \frac{.0035 \times 1}{.834} + .0025$$

$$R_D = .0067 \ h \cdot ft^2 \cdot {}^\circ F / Btu$$

step 4 Calculate U_D

$$U_D = \left(\frac{1}{U_c} + R_D\right)^{-1}$$

$$= \left(\frac{1}{41.1} + .0067\right)^{-1}$$

$$U_D = 32 \ Btu / h \cdot ft^2 \cdot {}^\circ F$$

$\Rightarrow U_D > U_{req}$

$\therefore$ The exchanger is suitable

Heat Exchanger 4

DESIGN OF DOUBLE-PIPE HEAT EXCHANGERS

Complete design * حتى يصير عندي
thermal analysis ⇐ ركزنا بالسابق عليه
وكيف حسبنا ال area
hydraulic analysis ⇐ هسه بدنا نركز عليه
function on ΔP ⇐ (operating cost)
يعني بدي أجيب معادلات ΔP وأدخلها بالحسابات

* أهم نقطة في عمليات ال design هو ال cost
* Total cost have 2 parameters:
- Capital cost - Operating cost
لازم يكونوا low

Two parameters conflicting of H-E

Operating cost ⇐ function on energy or power

Capital cost ⇐ يعتمد على area
كلما كانت ال area أقل (small area) تكون ال cost أقل

Introduction

- Design of heat-transfer equipment off between the two conflicting goals of low capital cost (high overall heat-transfer coefficient, small heat-transfer area) and low operating cost (small stream pressure drops).
- Optimal design thus involves capital and energy costs, which are constantly changing.
- A simpler procedure is to specify a reasonable pressure drop allowance for each stream and design the exchanger within these constraints. The pressure drop allowances determine (approximately) the trade-off between capital and operating costs.

Introduction _ continue

- For low-viscosity liquids such as water, organic solvents, light hydrocarbons, etc., an allowance in the range of 7-20 psi is commonly used. For gases, a value in the range of 1-5 psi is often specified.
- The basic calculations involved in the thermal analysis of double-pipe heat exchangers (which given in the previous slides) are still required in this section of analysis. Moreover, Series-parallel configurations of hairpins are considered.

* أثناء الشغل على الـ design لازم يكون عندي data معينة ، بخلي الشغل يتبع فيها هذه الـ data
* من هذه الـ data :
- Range of pressure
يختلف هذا الـ range باختلاف حالة المادة والـ viscosity

* كلما ارتفعت الـ viscosity لـ liquids حالة
تقترب من 20 psi .
* كلما قل الـ range يكون أخفض ← يكون على low operating cost
* وإذا زاد الضغط من هذا الـ range إنت في دراسة
* بالنسبة للغازات الـ range كما أتى .

Heat-Transfer Coefficients for Exchangers without Fins

- Heat-transfer correlations for pipes and annuli were presented in the previous lectures and their application to the analysis of double-pipe exchangers was also illustrated. <u>In short</u> (summary):
  - ✓ For turbulent flow ($Re \geq 10^4$), the Seider-Tate equation is used in the form:

$$Nu = 0.023\, Re^{0.8} Pr^{1/3} (\mu/\mu_w)^{0.14} \quad \text{.....(1)}$$

<u>Note</u>: A coefficient of 0.023 is used here rather than 0.027. This coefficient (0.023) is often preferred for design work.

✓ For the transition region ($2100 < Re < 10^4$), the Hausen equation is used:

10^4 هيه في تعمل

$$Nu = 0.116\,[Re^{2/3} - 125]\,Pr^{1/3}(\mu/\mu_w)^{0.14}[1 + (D_i/L)^{2/3}] \quad \text{.....(2)}$$

✓ Equations (1) and (2) are used for both pipes and annuli, with the equivalent diameter, De ,replacing D_i , the pipe ID, in the case of an annulus.

✓ For laminar flow ($Re \leq 2100$) in pipes, the Seider-Tate equation is used:

$$Nu = 1.86\,[Re\,Pr\,D_i/L]^{1/3}(\mu/\mu_w)^{0.14} \quad \text{.....(3)}$$

But for laminar flow in annuli, the following equation should be used.

$$Nu = 3.66 + 1.2(D_2/D_1)^{0.8} + \frac{0.19[1 + 0.14(D_2/D_1)^{0.5}][Re\,Pr\,D_e/L]^{0.8}}{1 + 0.117[Re\,Pr\,D_e/L]^{0.467}} \quad \text{.....(4)}$$

* تكرير ال Double pipes عبارة عن
عبارة عن ال hair pins بوصلوهم ب
- Series
- Series parallel

hair pins

ations for Exchangers without Fins

The main contribution to pressure drop in double-pipe exchangers is from fluid friction in the straight sections of pipe. For isothermal flow, this pressure drop can be expressed in terms of the friction factor, *f*, as:

American unit ⇐

$$\Delta P_f = \frac{f\,L\rho V^2}{2g_c D_i} \quad \text{.....(5)}$$

* أنواع ال pressure drop :
- في طول ال hair pin (total length)
(straight section of pipe)
- صدفة بسبب الانحناء ⇐
change of direction in hair pin
- بسبب ال nozzels

where

* تستخدم هذه المعادلة في حالة isothermal
شو يعني ؟!
يعني درجة average temperature of fluid ≃ temperature of wall

L = pipe length

D_i = pipe ID

ρ = fluid density

V = average fluid velocity

This equation can be written in terms of the mass flux, G, and the specific gravity, s, of the fluid by making the substitutions $V = G/\rho$ and $\rho = S * \rho_{water}$ to give:

$$\Delta P_f = \frac{f\,LG^2}{2g_c\rho_{water}D_i s} \quad \text{.....(6)}$$

* specific gravity (s) ⇒

$$s = \frac{\rho}{\rho_{water}}$$

* يتم احسب ال ΔP بقارنها مع ال range اللي بعده

* إذا كان ال system ← فيه درجة حرارة ال wall عن ال fluid لازم أحسب ال correction factor وأقسمها على ΔP اللي حسبتها في ال isothermal.

⇒ correction factor = 144 ϕ

فيها حسب نوع ال flow.

Notes

is basic hydraulic equation by dividing the right side by 144ϕ.

$(\mu/\mu_w)^{0.14}$ for turbulent flow

$(\mu/\mu_w)^{0.25}$ for laminar flow

The viscosity correction factor accounts for the effect of variable fluid properties on the friction factor in non-isothermal flow, while the factor of 144 converts the pressure drop from lbf/ft^2 to psi.

converts factor

➢ The friction factor can be computed as follows: (f)

معادلة ΔP ← تتعامل مع 144 لأنه الرقم American uni

يحولي من lbf/ft² إلى psi

For laminar flow in the inner pipe,

$$f = 64/Re$$

For laminar flow in the annulus

$$f = \left(\frac{64}{Re}\right)\left[\frac{(1-\kappa)^2}{1+\kappa^2+(1-\kappa^2)/\ln\kappa}\right]$$

where
$\kappa = D_1/D_2$
D_1 = OD of inner pipe
D_2 = ID of outer pipe

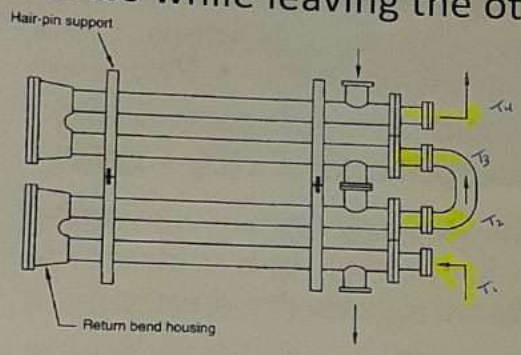
- For commercial pipe and turbulent flow with $Re > 3000$, the following equation can be used for both the pipe and the annulus:

$$f = 0.3673\, Re^{-0.2314}$$

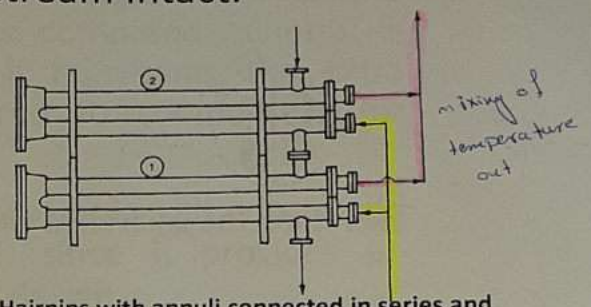
* في عمليات الـ design دائماً بيبقى ب series إذا عايزة بيجي لـ parallel

Series/Parallel Configurations of Hairpins

In order to meet pressure drop constraints, it is sometimes convenient to divide one stream into two or more parallel branches while leaving the other stream intact.



- *Two hairpins connected in series*
- *Gives pure counter current flow (true counter flow)*



- **Hairpins with annuli connected in series and inner pipes connected in parallel.**
- **Although the flow is countercurrent in each hairpin, the overall flow pattern is not true counter flow.**

Note: in particular that the temperature of the fluid enter
hairpin is different in the two cases.

* اعرف بين الـ parallel & series :
⇐ (series) : بيدخل الـ inner fluid من وحدة وبيخرج داخل الـ hair pin على شكل سلسلة متعرجة .
⇐ (parallel) : بقسم الـ inner fluid لجزئين : جزء بيدخل في الـ hair pin الأول والثاني

To account for the departure from true co
parallel configurations, the counter-flow
temperature difference (LMTD) is multip
factor, F, given by the following equations:

$$F = \left[\frac{(R-x)}{x(R-1)}\right] \frac{\ln[(1-P)/(1-P}{\ln\left[\frac{(R-x)}{R(1-PR)^{1/x}} + \right.}$$

$$F = \frac{P(1-x)}{x(1-P)\ln\left[\frac{(1-x)}{(1-P)^{1/x}} + x\right]}$$

$$P = (t_b - t_a)/(T_a - t_a)$$

$$R = (T_a - T_b)/(t_b - t_a)$$

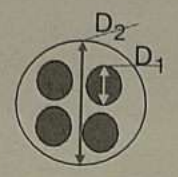
⇐ (series) : بكون في ٤ درجات حرارة مابينها بالسلسلة بيعض
⇐ (parallel) : بتكون Tin مشتركة & Texit مابينها mixing بعد طلع

* في الـ parallel not true counter flow
شو يعني ؟!
- بتفرق true counter ويعمل (F) correction
وبضرب الـ F بـ (LMTD) .

where
x = number of parallel branches
T_a, T_b = inlet and outlet temperatures of
t_a, t_b = inlet and outlet temperatures of

Multi-tube Exchangers

Un-finned multi-tube hairpin exchangers can be handled using the methods presented above with an appropriate value of the equivalent diameter. For an outer pipe with an ID of D_2 containing n_t tubes, each with OD of D_1, the flow area and wetted perimeter are:



$$A_f = (\pi/4)(D_2^2 - n_t D_1^2)$$

$$\text{Wetted perimeter} = \pi(D_2 + n_t D_1)$$

Therefore, the equivalent diameter is given by:

$$D_e = 4A_f/\text{wetted perimeter}$$

$$D_e = (D_2^2 - n_t D_1^2)(D_2 + n_t D_1)$$

Note that for $n_t = 1$, this reduces to $D_e = D_2 - D_1$.

double pipe يعني

Over-Surface and Over-Design

U clean U design

- Over-surface is a measure of the safety factor incorporated in the design of a heat exchanger through fouling factors and the use of standard equipment sizes. Since it deals directly with exchanger surface area, it is easier to visualize than fouling factors and calculated versus required heat-transfer coefficients.
- The percentage over-surface is defined as follows:

$$\%\ \text{over-surface} = \frac{A - A_C}{A_C} \times 100$$

where
A = actual heat-transfer surface area in the exchanger
A_C = calculated heat-transfer surface area based on U_C

- Over-surface depends on the relative magnitudes of the total fouling allowance and the film and wall resistances. While values of 20-40% may be considered typical, higher values are not unusual.
- The previous equation is often applied with the surface area calculated using the design coefficient, U_D, rather than the clean coefficient. In this case the computed quantity is referred to as the over-design; it represents the extra surface area beyond that required to compensate for fouling.
- Some over-design (typically about 10% or less) is considered acceptable and (often) desirable, since it provides an additional safety margin in the final design.
- The design procedure is illustrated via the following example.

* في أسئلة ال design بالبداية بعمل thermal calculation بعدين بعمل hydraulic calculation

Example

10,000 lb/h of benzene will be heated from 60°F to 120°F by heat exchange with an aniline stream that will be cooled from 150°F to 100°F. A number of 16-ft hairpins consisting of 2-in. by 1.25-in. schedule 40 stainless steel pipe (type 316, $k = 9.4$ Btu/h·ft·°F) are available and will be used for this service. A maximum pressure drop of 20 psi is specified for each stream. The specific gravity of benzene is 0.879 and that of aniline is 1.022. Determine the number and configuration of hairpins that are required.

Solution

Assumptions;

1) We begin by assuming that the hairpins are connected in series on both sides, since this is the simplest configuration, and that the flow pattern is counter-current.
2) We also place the benzene in the inner pipe (however, either fluid could be placed in the inner pipe).

* أول شي كنا ال assumptions :
- بنفرض بالأول أن ال connection عبارة عن series وإذا طلعت كون ال parallel series
- بنفرض ال inner & outer fluid حسب قواعد سابقة .

* من السؤال يتم عمل حل Try & error

First trial

a) Fluid properties at the average stream temperatures are obtained from Figures A.1 and A.2, and Table A.15.

Fluid property	Benzene ($t_{ave} = 90°F$)	Aniline ($T_{ave} = 125°F$)
μ (cp)	0.55	2.0
C_P (Btu/lbm · °F)	0.42	0.52
k (Btu/h · ft · °F)	0.092	0.100

b) Determine the heat load and an
energy balances on the two stre

$$q = (\dot{m}C_P\Delta T)_B = 10{,}000 \times 0.42 \times$$
$$252{,}000 = (\dot{m}C_P\Delta T)_A = \dot{m}_A \times 0.5$$
$$\dot{m}_A = 9692\,\text{lb/h}$$

* لسنا أعمل energy balance وأستفيد من المعطيات المعطى ، حتى أطلع باقي المجاهيل مثل : mass flowrate of other fluid

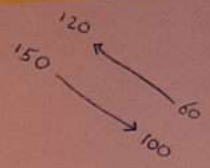
(c) Calculate the LMTD.

$$\Delta T_{ln} = \frac{40 - 30}{\ln(40/30)} =$$

* التوصيل عن طريق ال series
ينتج عنه true pure counter

$\Delta T_1 = 40°F$

$\Delta T_2 = 30°F$



(d) Calculate h_i assuming $\phi_i = 1.0$. ⇒ لسه Tw غير معلومة

*properties of inner pipe:

1.25 in Schedule 40

$D_i = 1.38$ in

$D_o = 1.66$ in

$$D_i = 1.38/12 = 0.115\,\text{ft} \quad \text{(from Table B.2)}$$

$$Re = \frac{4\dot{m}}{\pi D_i \mu} = \frac{4 \times 10{,}000}{\pi \times 0.115 \times 0.55 \times 2.419} = 83{,}217 \Rightarrow \text{turbulent flow}$$

يفضل انه يكون turbulent في حسابات ال design
ولازم أبعد عن ال laminar & transition

$$h_i = \frac{k}{D_i} \times 0.023\, Re^{0.8} Pr^{1/3}$$

$$h_i = \frac{0.092}{0.115} \times 0.023(83{,}217)^{0.8} \left(\frac{0.42 \times 0.55 \times 2.419}{0.092}\right)^{1/3}$$

$$h_i = 290\,\text{Btu/h} \cdot \text{ft}^2 \cdot {}^\circ\text{F}$$

$D_{eq} = D_{in}$ of outer tube $-$ D_{out} of inner tube

$D_{eq} = 2.067 - 1.66 = .407$ in

(e) Calculate h_o assuming $\phi_o = 1.0$. ⇒ فرضناها لسه Tw غير معروفة

*properties of outer pipe:

2 in schedule 40

$D_i = 2.067$ in

$D_o = 2.375$ in

$$\left.\begin{aligned} D_2 &= 2.067\,\text{in.} \\ D_1 &= 1.660\,\text{in.} \end{aligned}\right\} \quad \text{(from Table B.2)}$$

$$D_e = D_2 - D_1 = \frac{2.067 - 1.660}{12} = 0.0339\,\text{ft}$$

$$\text{flow area} \equiv A_f = \frac{\pi}{4}(D_2^2 - D_1^2) = 0.00827\,\text{ft}^2$$

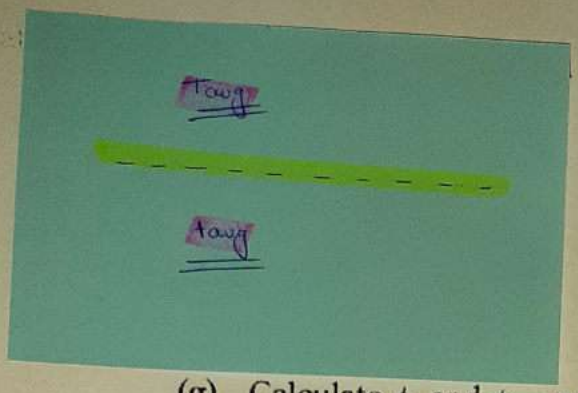
$$Re = \frac{D_e(\dot{m}/A_f)}{\mu} = \frac{0.0339 \times (9692/0.00827)}{2.0 \times 2.419} = 8212 \Rightarrow \text{transition flow}$$

Using the Hausen equation with $[1 + (D_e/L)^{2/3}] = 1.0$ gives:

$$h_o = \frac{k}{D_e} \times 0.116[Re^{2/3} - 125]Pr^{1/3}$$

$$= \frac{0.1}{0.0339} \times 0.116 \times [(8212)^{2/3} - 125]\left(\frac{0.52 \times 2.0 \times 2.419}{0.1}\right)^{1/3}$$

$$h_o = 283\,\text{Btu/h} \cdot \text{ft}^2 \cdot {}^\circ\text{F}$$



temperature.

$$T_w = \frac{h_i t_{ave} + h_o(D_o/D_i)T_{ave}}{h_i + h_o(D_o/D_i)}$$

$$T_w = \frac{290 \times 90 + 283(1.66/1.38) \times 125}{290 + 283 \times (1.66/1.38)}$$

$$T_w = 108.9°\text{F}$$

(g) Calculate ϕ_i and ϕ_o, and corrected values of h_i and h_o.
From Figure A.1, at 108.9°F, $\mu_B = 0.47$ cp and $\mu_A = 2.4$ cp. Therefore,

$q_{loss} = q_{accept}$

$h_o A (T_{avg} - T_w) = h_i A (T_w - t_{avg})$

$h_o A T_{avg} - h_o A T_w = h_i A T_w - h_i A t_{avg}$

$h_i A t_{avg} + h_o A T_{avg} = h_i A T_w + h_o A T_w$

$T_w = \dfrac{h_i t_{avg} + h_o T_{avg}(D_o/D_i)}{h_i + h_o(D_o/D_i)}$

$$\phi_i = (0.55/0.47)^{0.14} = 1.0222$$

$$\phi_o = (20/2.4)^{0.14} = 0.9748$$

$$h_i = 290(1.0222) = 296\,\text{Btu/h} \cdot \text{ft}^2 \cdot °\text{F}$$

$$h_o = 283(0.9748) = 276\,\text{Btu/h} \cdot \text{ft}^2 \cdot °\text{F}$$

بعد حساب Tw
بعمل correction لـ
h_i & h_o
$h_i = \phi_i h_i$
$h_o = \phi_o h_o$

*ملاحظة:
D_o & D_i ⇒ for inner tube
لـ surface of inner wall يعتبر
tube

(h) Obtain fouling factors. For liquid organic process chemicals such as benzene and aniline, a value of 0.001 h·ft²·°F/Btu is appropriate.

(i) Compute the overall heat-transfer coefficient.

$$U_D = \left[\frac{D_o}{h_i D_i} + \frac{D_o \ln(D_o/D_i)}{2k} + \frac{1}{h_o} + \frac{R_{Di} D_o}{D_i} + R_{Do}\right]^{-1}$$

$$U_D = \left[\frac{1.66}{296 \times 1.38} + \frac{(1.66/12)\ln(1.66/1.38)}{2 \times 9.4} + \frac{1}{276} + \frac{0.001 \times 1.66}{1.38} + 0.001\right]^{-1}$$

$$U_D = 89\,\text{Btu/h} \cdot \text{ft}^2 \cdot °\text{F}$$

(j) Calculate the required surface area and number of hairpins.

$$q = U_D A \Delta T_{ln}$$

$$A = \frac{q}{U_D \Delta T_{ln}}$$

$$A = \frac{252{,}000}{89 \times 34.76} = 81.5\,\text{ft}^2$$

From Table B.2, the external surface area per foot of 1.25-in. schedule 40 pipe is 0.435 ft^2.

$$L = \frac{81.5}{0.435} = 187.4\,\text{ft}$$

2 ft of pipe,

er of hairpins $= \dfrac{187.4}{32} = 5.9 \Rightarrow 6$

* Number of hairpins = $\dfrac{\text{Total length}}{\text{length of one hair pins}} = \dfrac{187.4}{(16 \times 2)}$

$5.9 \simeq 6$

إذا كان 0.x > 0.5 بقرب الرقم للي بعده

$\underline{\underline{6}} \simeq 5.6$ or 5.7 or 5.8 مثلاً

$\Rightarrow A = \pi D_o L$ of inner tube

$A = \pi\left(\frac{1.66}{12}\right) L$

$L = \dfrac{A\ 12}{\pi(1.66)}$

$\underline{\underline{L = 187.4\ ft}}$

* السطح الذي يتم من خلاله انتقال الحرارة هو السطح الخارجي للأنبوب الداخلي

(k) Calculate the pressure drop for the benzene stream (inner pipe).
The friction factor is calculated from Equation (4.8):

* هسه بدي أطبق بارامترات hydraulic calculations بعد ما حسبت الـ thermal

$$f = 0.3673\,Re^{-0.2314} = 0.3673(83{,}217)^{-0.2314}$$
$$f = 0.0267$$
$$A_f = 0.0104\,\text{ft}^2 \quad \text{(Table B.2)}$$
$$G = \dot{m}/A_f = 10{,}000/0.0104 = 961{,}538\,\text{lbm/h}\cdot\text{ft}^2$$

The pressure drop in the straight sections of pipe is calculated using Equation (4.5):

$$\Delta P_f = \frac{f L G^2}{7.50 \times 10^{12} D_i s \phi}$$
$$= \frac{0.0267(6 \times 32)(961{,}538)^2}{7.50 \times 10^{12}(1.38/12) \times 0.879 \times 1.022}$$
$$\Delta P_f = 6.1\,\text{psi}$$

The pressure drop in the return bends is obtained from Equation (4.12):

$$\Delta P_r = 1.6 \times 10^{-13}(2N_{HP} - 1)G^2/s$$
$$= 1.6 \times 10^{-13}(2 \times 6 - 1)(961{,}538)^2/0.879$$
$$\Delta P_r = 1.85\,\text{psi}$$

Since the nozzle losses associated with the inner pipes are negligible, the total pressure drop, ΔP_i, is:

$$\Delta P_i = \Delta P_f + \Delta P_r = 6.1 + 1.85 = 7.95 \cong 8.0\,\text{psi}$$

< 20 psi

ما في مشكلة

هسه بدنا نشوف ΔP للـ inline

(l) Calculate the pressure drop for the aniline stream (annulus).
The friction factor is calculated from Equation (4.8):

$$f = 0.3673\,Re^{-0.2314} = 0.3673(8212)^{-0.2314}$$
$$f = 0.0456$$
$$G = \dot{m}/A_f = 9692/0.00827 = 1{,}171{,}947\ \text{lbm/h}\cdot\text{ft}^2$$

The pressure drop in the straight sections of pipe is again calculated using Equation (4.5) with the pipe diameter replaced by the equivalent diameter:

$$\Delta P_f = \frac{f\,LG^2}{7.50\times10^{12}D_e s\phi}$$
$$= \frac{0.0456(6\times32)(1,171,947)^2}{7.50\times10^{12}\times0.0339\times1.022\times0.9748}$$
$$\Delta P_f = 47.5\ \text{psi}$$

> 20 psi

يعني في مشكلة
كبيرة ومعكم اصل هي المشكلة

Since this value greatly exceeds the allowed pressure drop, the minor losses will not be calculated. This completes the first trial.

* اول حاجة بعمل switch بالكي يعني؟
كنا فرضنا ان ال Benzene inner & aniline outer
صار بعمل العكس يعني ال Benzen inner & outer
* او المحاولة الثانية إذا عاوزين نحنا اول ما
بعمل splitting لل aniline لسه $\Delta P \propto G$
يعني بدي ال aniline ع دفعات باخذ منها $\frac{1}{2}$ وغيره

Note

In summary, there are two problems with the initial configuration of the heat exchanger:

I. The pressure drop on the annulus side is too large.
II. The Reynolds number in the annulus is less than 10,000.

Since the dimensions of the hairpins are fixed in this problem, there are relatively few options for modifying the design. Two possibilities are:

- Switch the fluids, i.e., put the aniline in the inner pipe and the benzene in the annulus.
- Connect the annuli in parallel.

- The effects of these changes on the Reynolds numbers and pressure drops can be estimated as follows:

Try Method 1

- Switch the fluids. Since the flow rates of the two streams are approximately the same, the Reynolds numbers are essentially inversely proportional to the viscosity. Thus,

correction for Re عملت عن طريق ضرب الـ Re بـ (ϕ_w/ϕ)

$$\begin{cases} Re_i \rightarrow 83{,}217(0.55/2.0) \cong 23{,}000 \\ Re_o \rightarrow 8212(2.0/0.55) \cong 30{,}000 \end{cases}$$

Hence, switching the fluids will result in fully turbulent flow on both sides of the exchanger.

To estimate the effect on pressure drops, assume that the number of hairpins does not change. Then the main factors affecting ΔP are f and s. Hence,

$$\Delta P \sim f/s \sim Re^{-0.2314} s^{-1}$$

f ⇒ function on Re

$$\Delta P_{f,i} \rightarrow 6.1(23{,}000/83{,}000)^{-0.2314}(1.022/0.879)^{-1} \cong 7 \text{ psi}$$

$$\Delta P_{f,o} \rightarrow 47.5(30{,}000/8200)^{-0.2314}(0.879/1.022)^{-1} \cong 41 \text{ psi}$$ ⇒ still high > 20

لسا في مشكلة بدي أجرب هي ثاني

Clearly, switching the fluids does not reduc

* أحل الثاني أعمل spliting للـ aniline
كيف عكس أجزاء الكمية ؟!
عن طريق أعمل connection للـ aniline
parallel ، هكيتا في حالة الـ parallel
ينقسم الـ inlet إلى جزئين .
$\therefore m' = \frac{1}{2} m$

Try method 2

- Connect the annuli in two parallel banks. This change will have no effect on the fluid flowing in the inner pipe. For the fluid in the annulus, however, both the flow rate and the length of the flow path will be halved. Therefore,

$$Re_o \rightarrow 8212 \times 1/2 \cong 4100$$

$Re \propto m$

Assuming that the number of hairpins does not change,

$$\Delta P_{f,o} \sim fG^2L$$

$$\Delta P_{f,o} \rightarrow 47.5(4100/8200)^{-0.2314}(1/2)^2(1/2) \cong 7 \text{ psi}$$

$\frac{1}{2}$

تيم إذا عملنا spliting رح تنحل مشكلة ΔP وتكون 20psi > 7psi

- Apparently this modification will take care of the pressure-drop problem, but will push the Reynolds number further into the transition region.
- Although neither modification by itself will correct the problems with the initial design, in combination they might. Hence, we consider a third alternative.

Try method 3

- Switch the fluids and connect the annuli in two
  Reynolds numbers will become:

$$Re_i \cong 23,000$$
$$Re_o \cong 15,000$$

30000/2

* في عندي مشكلة وحدة وهي الـ transition
كيف بدي اتخلص منها ؟!
بحافظ ع الـ spliting اللي عملت بين يدي
اعمل switch. حيث يصير الـ Benzen هو
الـ annuli و الـ aniline بالداخل

The pressure drops will become (assuming no change in the

$$\Delta P_{f,i} \cong 7 \text{ psi}$$

$$\Delta P_{f,o} \rightarrow 47.5(15{,}000/8200)^{-0.2314}(1/2)^2(1/2)(0.879/1.022) \cong 6 \text{ psi}$$

It appears that this alternative will meet all design requirements. However, it is necessary to perform the detailed calculations because h_i, h_o and the mean temperature difference will all change, and hence the number of hairpins can be expected to change as well.

Second trial

* بما أنه فضينا من مشكلة الـ DP بـ transition نقدر أن التصميم
سيكون كالتالي:
- الـ Benzen بالخارج (annula) & الـ aniline بالداخل
- التوصيل بين الـ fluid الخارجي يكون parallel
ويرجع نعمل كل الحسابات كما في السابق ع هذا التصميم

Second trial

a) Calculate the LMTD correction factor for the series/parallel configuration. Aniline in the inner pipe is the series stream and benzene in the annulus is the parallel stream. "See the previous figure for connection".

هذا الـ correction بسبب الـ parallel
بضرب الـ F في (ΔT_m)

$$\begin{cases} T_a = 150°\text{F} \quad t_a = 60°\text{F} \\ T_b = 100°\text{F} \quad t_b = 120°\text{F} \\ P = (t_b - t_a)/(T_a - t_a) = (120-60)/(150-60) = 0.6667 \\ R = (T_a - T_b)/(t_b - t_a) = (150-100)/(120-60) = 0.8333 \\ x = 2 \text{ (number of parallel branches)} \end{cases}$$

Substituting into Equation (4.16) gives:

$$F = \left[\frac{(0.8333-2)}{2(0.8333-1)}\right] \frac{\ln\left[(1-0.6667)/(1-0.6667\times 0.8333)\right]}{\ln\left[\dfrac{(0.8333-2)}{0.8333(1-0.6667\times 0.8333)^{1/2}} + \left(\dfrac{2}{0.8333}\right)\right]}$$

$$F = 0.836$$

Shell-and-tube Exchanger 5

Complete Design Procedure

Hydraulic Calculations

* Note:
n_p ⇒ number of tube passes
L ⇒ tube length
$n_p L$ ⇒ total Length of tube

• Tube-side pressure drop

(main النوع الرئيسي) ⇐

$$\Delta P_f = \frac{f\, n_p L G^2}{7.50 \times 10^{12} D_i s \phi}$$

where
f = Darcy friction factor (dimensionless)
L = tube length (ft)
G = mass flux (lbm/h · ft²)
D_i = tube ID (ft)
s = fluid specific gravity (dimensionless)
ϕ = viscosity correction factor (dimensionless)
$= (\mu/\mu_w)^{0.14}$ for turbulent flow
$= (\mu/\mu_w)^{0.25}$ for laminar flow

$f = \frac{64}{Re}$ Laminar

$f = 0.4137\, Re^{-0.2585}$ turbulent, Re ≥3000

Note the effect of number of tube passes, n_p

return النتيجة ⇐

$$\Delta P_r = 1.334 \times 10^{-13} \alpha_r G^2/s$$

number of velocity heads, See table to get it.

* أنواع الـ ΔP :
- ΔP الرئيسي الذي يحدث في الـ tube
- ΔP نتيجة الـ return في الـ headers
- ΔP نتيجة الـ Nozzle

Number of Velocity Heads, α_r, Allocated for Minor Losses on Tube Side

Flow regime	Regular tubes	U-tubes
Turbulent	$2n_p - 1.5$	$1.6\,n_p - 1.5$
Laminar, $Re \geq 500$	$3.25\,n_p - 1.5$	$2.38\,n_p - 1.5$

• Nozzle pressure loss

No. of shells

$10^{-13} N_s G_n^2/s$ (turbulent flow)

$10^{-13} N_s G_n^2/s$ (laminar flow, $Re_n \geq 100$)

Re_n are the mass flux and Reynolds number

$\Delta P_{tot} = \Delta P_f + \Delta P_r + \Delta P_n$

بعدين بقارن الـ ΔP_{tot} مع الـ range المسموح (allowance)

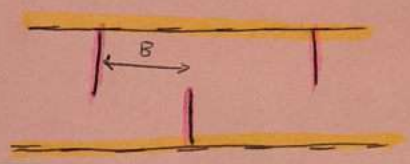
- # Shell-side pressure drop

$$\Delta P_f = \frac{f\, G^2\, d_s (n_b + 1)}{7.50 \times 10^{12}\, d_e s \phi}$$

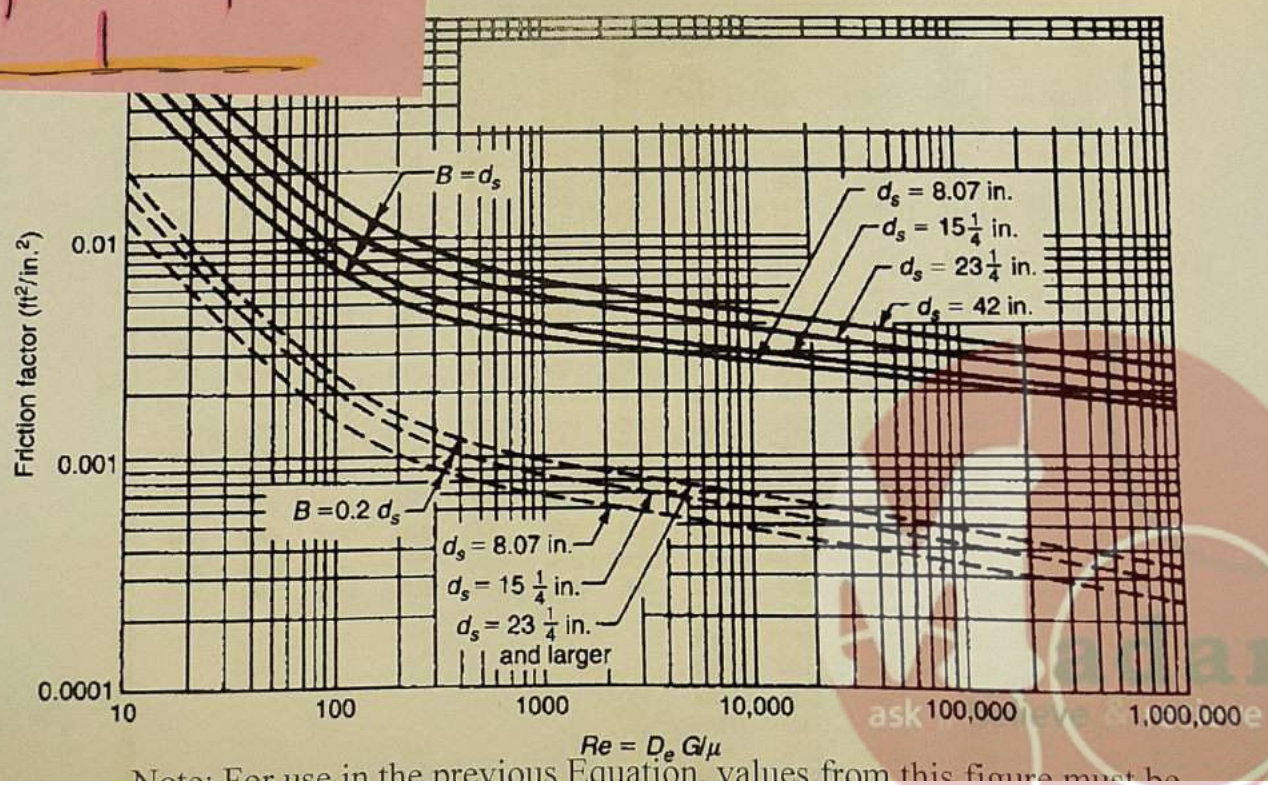
number of crosses

where
f = friction factor (dimensionless)
G = mass flux = $\dot{m}/a_s$ (lbm/h · ft^2)
a_s = flow area across tube bundle (ft^2)
= $d_s C' B/(144 P_T)$
d_s = shell ID (in.)
C' = clearance (in.)
B = baffle spacing (in.)
P_T = tube pitch (in.); replaced by $P_T/\sqrt{2}$ for 45° tube layout
n_b = number of baffles
d_e = equivalent diameter from Figure 3.12 (in.)
s = fluid specific gravity
ϕ = viscosity correction factor = $(\mu/\mu_w)^{0.14}$
ΔP_f = pressure drop (psi)

* The friction factor function on:
- Re
- B => baffle space



the following figure



Note: For use in the previous Equation, values from this figure must be

Tube-Count Tables

⇒ standard based on tube size & pitch & Codes (TEMA)

Table C.1 *Tube Counts for 5/8-in. OD Tubes on 13/16-in. Square Pitch*

Shell ID (in.)	TEMA P or S				TEMA U		
	Number of passes						
	1	2	4	6			
8	55	48	34	24			
10	88	78	62	56			
12	140	138	112	100	1		
13.25	178	172	146	136	1		
15.25	245	232	208	192	2		
17.25	320	308	274	260	3		
19.25	405	392	352	336	4		
21.25	502	484	442	424	5		
23.25	610	584	536	5[illegible]	6		
25	700	676	618	600	7		
27	843	812	742	716	8		
29	970	942	868	840	998	956	920
31	1127	1096	1014	984	11		
33	1288	1250	11[illegible]	1148	13		
35	1479	1438	[illegible]	1308	14		
37	1647	1604	1520	1480	16		
39	1840	1794	1700	1664	18		
42	2157	2112	2004	1968	21		
45	2511	2458	2326	2288	25		
48	2865	2808	2686	2656	29		
54	3656	3600	3462	3404	37		
60	4538	4472	4310	4256	46		

* أول شي بحدده ال designer هو ال OD & pitch.

* بعدين بحسب ال total number of tubes طريقة :

$$\frac{\text{Surface area total}}{\text{surface of one tube}} = \text{total number of tubes}$$

* كمان بحسب ال total surface area :

$$q = h A \Delta T_m$$

$$A = \text{total surface area} = \frac{q}{h \Delta T_m}$$

* بعد ما أطلع ال total number of tubes بحدد ال code (P or S & U) ، بعدين بحدد ال number of passes.

* بعد ما أحدد ال np بيني خط عامودي لحد ما أوصل ل number of tubes بقرأ كم قيمة ال shell (ID)

Table C.2 *[illegible] OD Tubes on 15/16-in. Triangular Pitch*

Shell ID (in.)	TEMA P or S								TEMA U		
	Number of passes				Number of passes				Number of passes		
	1	2	4	6	1	2	4	6	2	4	6
[illegible]	[illegible]	[illegible]	34	24	34	32	16	18	32	24	24
[illegible]	[illegible]	[illegible]	52	50	60	62	52	44	64	52	52
12	122	114	94	96	109	98	78	68	98	88	78
13.25	151	142	124	112	126	120	106	100	126	116	108
15.25	204	192	166	168	183	168	146	136	180	160	148
17.25	264	254	228	220	237	228	202	192	238	224	204
19.25	332	326	290	280	297	286	258	248	298	280	262
21.25	417	396	364	348	372	356	324	316	370	352	334
23.25	495	478	430	420	450	430	392	376	456	428	[illegible]
25	579	554	512	488	518	498	456	444	534	500	[illegible]
27	676	648	602	584	618	602	548	532	628	[illegible]	570
29	785	762	704	688	729	708	650	624	736	[illegible]	668
31	909	878	814	792	843	812	744	732	846	812	780
33	1035	1002	944	920	962	934	868	840	97[illegible]	928	904
35	1164	1132	1062	1036	1090	1064	990	972	[illegible]	1060	1008
37	1304	1270	1200	1168	1233	1196	1132	1100	1238	1200	1152
39	1460	1422	1338	1320	1365	1346	1266	12[illegible]	1390	1336	1290
42	1703	1664	1578	1552	1611	1580	1498	[illegible]	1632	1568	1524
45	1960	1918	1830	1800	1875	1834	1736	1708	1882	1820	1770
48	2242	2196	2106	2060	2132	2100	19[illegible]	1964	2152	2092	2044
54	2861	2804	2682	2660	2730	2684	[illegible]	2536	2748	2680	2628
60	3527	3476	3360	3300	3395	3346	3228	3196	3420	3340	3286
66	4292	4228	4088	4044							
72	5116	5044	4902	4868							
78	6034	5964	5786	5740							
84	7005	6934	6766	6680							
90	8093	7998	7832	7708							
96	9203	9114	8896	8844							
108	11696	11618	11336	11268							
120	14459	14378	14080	13984							

Design Guidelines

*لازم قبل الـ design ادرس طبيعة الـ fluid & operating condition (T&P)

*قواعد عامة في الـ design

- Fluid placement
  - ❖as given before.
  - ❖corrosion and fouling very important factors that affect the fluid placement. Corrosive fluids should be placed on the tube side.

    تآكل في الـ metal — ترسبات

    *المادة اللي بتعمل fouling & corrosion أكثر بمرورها في الـ tube side
  - ❖Fluids that are heavy fouler should be placed on the tube side because it is (usually) easier to clean deposits from the interior surfaces of the tubes than from the exterior surfaces.
  - ❖Cooling water is usually placed in the tubes due to its tendency to corrode carbon steel and to form scale, which is difficult to remove from the exterior tube surfaces.
  - ❖See text for details.

- Tubing selection
  - ➢The most frequently used tube sizes are 3/4 and 1 in.

    common
  - ➢ For water service, 3/4 in., 16 BWG tubes are recommended.
  - ➢For oil (liquid hydrocarbon) service, 3/4 in., 14 BWG tubes are recommended if the fluid is non-fouling, while I in., 14 BWG tubes should be used for fouling fluids.
  - ➢Tube lengths typically range from 8 to 30 ft, and sometimes longer depending on the type of construction and the tubing material.
  - ➢A good value to start with is 16 or 20 ft.

ترتيب ال tubes

- **Tube layout** (square or triangular)
  - ✓ the most commonly used tube layouts are either triangular or square, with a pitch of 1.0 in. (for 3/4-in. tubes) or 1.25 in. (for 1-in. tubes).

* بعض المعلومات لازم تكون معطى لي :
mass flowrate of fluids &
Tin & Tout & P
في عمليات ال design

* العمليات التي تحدث باستخدام ال H.E :
- cooling & reheating &
heat recovery

* Note :
- cooling water ⇒ cooling ال
- steam ⇒ reheating " " "

* Initial configuration :
- OD tubes
- Pitch
- length
مجموعة من ال assumption بنفرضهم أول شي .

esign Strategy

* كيف بدي أبلش بال design :

Shell-and-tube design is an inherently iterative process, the main steps of which can be summarized as follows:

a) Obtain an initial configuration for the heat exchanger. This can be accomplished by using the preliminary design procedure as given in previous sections to estimate the required heat-transfer surface area, along with the design guidelines and tube-count tables discussed above to completely specify the configuration.

b) Rate the design to determine if it is thermally and hydraulically suitable.

c) Modify the design, if necessary, based on the results of the rating calculations.

d) Go to step (b) and iterate until an acceptable design is obtained.

بدي أقارن النتائج مع القيم المسموح الشغل فيها ، فمثلاً
بدي أشوف DP من ال calculations هل يقع ضمن ال range المسموح أو لا
وبعدين أعمل تعديلات في ال design حتى أوصل لل design المطلوب

The design procedure is illustrated in the following examples

* مثال

See text Ch. 5 and Read these examples

اجباري

* آخر شي مطلوب من المهندس يقدم summery sheet ، تحتوي على جميع مواصفات الـ tubes & fluid

مثل :

hot fluid	cold fluid
OD	
pitch	
length	
U_c	
U_D	
ID shell	
number of baffles	
⋮	

lec 11 — 24th - Apr - 2022

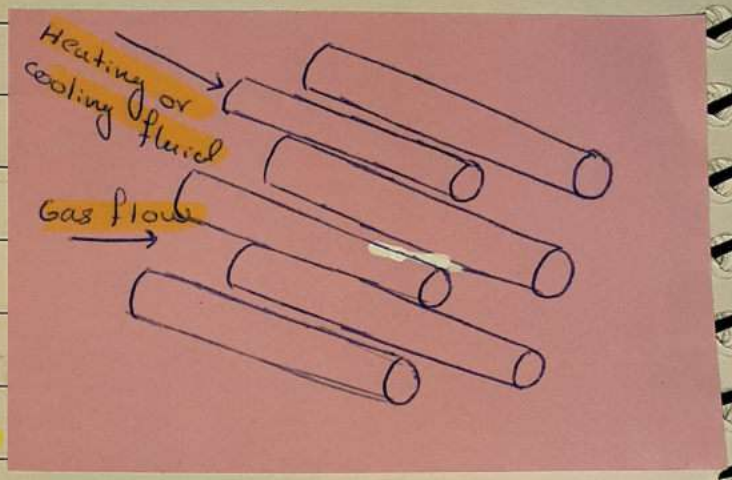
* Cross flow Exchanger and Exchanger effectiveness:

1 - Cross flow mixed - unmixed flow:

Cross ⇒

Heating or cooling fluid ⊥ Gas flow

بشكلوا زاوية = 90



* ماذا تعني mixed & unmixed؟

-mixed ⇒ أكثر من fluid - خلطوا معاً

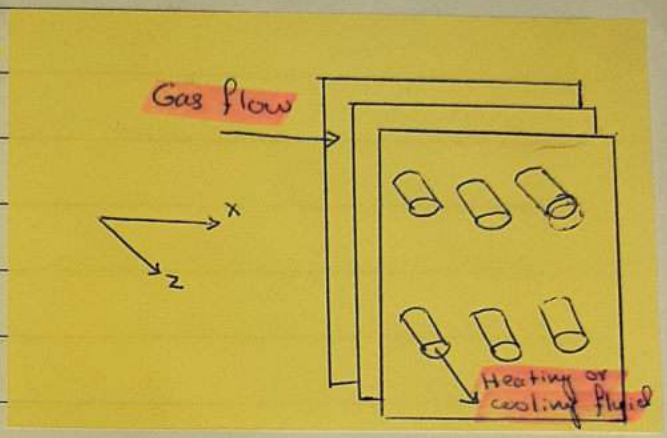
-unmixed ⇒ ما بصير اختلاط يعني بس لـ fluid داخل special channel

* Gas flow ⇒ over tubes ⇒ special channel يعني ما في mix يعني بصير

* Heating or cooling fluid ⇒ بمر داخل لـ pipes يعني unmixed

2 Cross flow exchanger both fluids unmixed :

* Cross ⇒ Gas flow ⊥
Heating or cooling fluid



* both fluids unmixed :
channel or Gas flow ⊥ بين
Heating or cooling ⊥ وكذلك
pipes ⊥ داخل fluid بين

* بنشوف هذا النوع من H.E في Air conditioning system

* ميزته :

$\frac{\text{Surface area}}{\text{unit volume}}$ ⇒ very large

يطلق عليه اسم Combat exchanger !!
very large لسبب ratio ⊥

* Note :

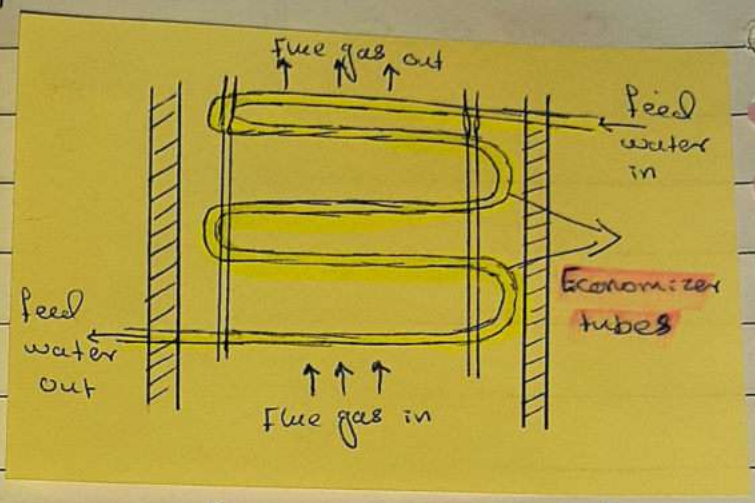
- Cross flow exchanger mixed-unmixed flow ⇒
Unfinned with one fluid mixed and the other unmixed

- Cross flow exchanger both fluids unmixed ⇒
Finned with both fluid unmixed

* Applications - Economizer :

يستخدم هذا التطبيق في مجال ال energy recovery

يوضع هذا الجهاز على hot steam
ناتج من احتراقه، فيتم استخدام
حرارة هذا ال steam لتسخين
ال water ، لذلك يستخدم في
recovery energy



Cross flow mixed - unmixed

* chart 1 (P, F) : (cross flow with both unmixed)

يلزم قيمة F ما تقل عن 0.8 كما في السابق

هي الرسمة بتبين أن كل stream
ال channel خاص به يعني ←
unmixed

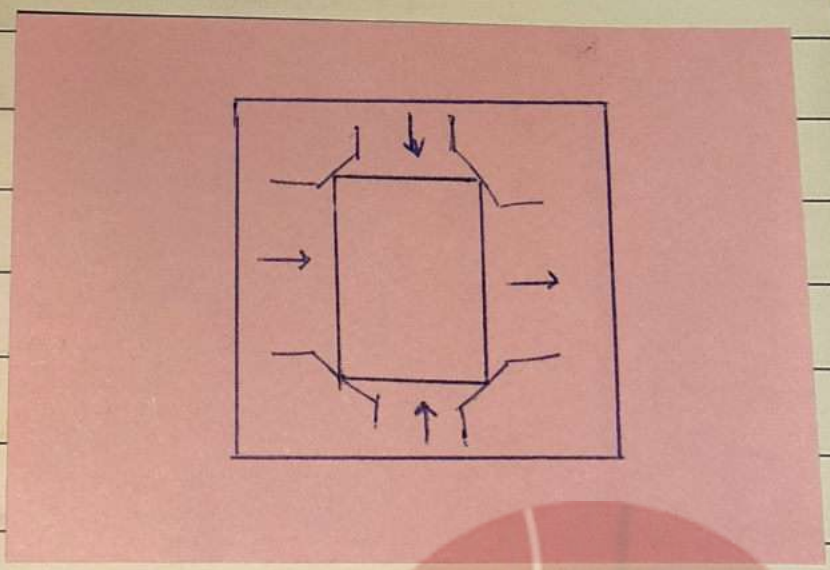


chart 2 Cross flow mixed - unmixed :

نفس الحسابات السابقة بفرض pure counter

نحسب قيمة (ΔT_{lm}) ثم نضربها بـ F ، بعدين نحسب الـ

q overall

* ملاحظة :

في هذه النوع الـ area متشابكة مع بعض ، لهذا السبب بتعامل مع

UA كـ one term

* Heat Exchanger analysis : The effectiveness - NTU Method:

- This method is preferable when only inlet temperatures are known.

$$\text{efficiency} = \frac{\text{actual heat transfer}}{\text{maximum heat transfer}}$$

* هي الطريقة valied لجميع أنواع الـ H.E

سواء كان :

- double pipe
- shell and tube
- Cross flow

$q_{max} \Rightarrow$ In Counter Current

* Note:

بوجود ال passes في ال shell & tube تظهر F
وهاي القيمة بتقلل من q ، لذلك عكس أعتبر ال
F مقياس لل performance .
لسلام نكون قيمة F عالية

$q = mcp\,\Delta T$

هذا المقدار بسميه C

وينقسم إلى نوعين

C_c — for cold stream

C_h — for hot stream

* نسمي C ⇐ heat capacitance

• Max possible heat transfer rate, q_{max} can be achieved in case of counter current flow exchanger of infinite length. But infinite length is not practical

* هسه كيف بدي أحسب q_{max} ؟!

$q_{hot} = q_{cold}$

$C_h \Delta T_h = C_c \Delta T_c$

∴ If $C_c < C_h$

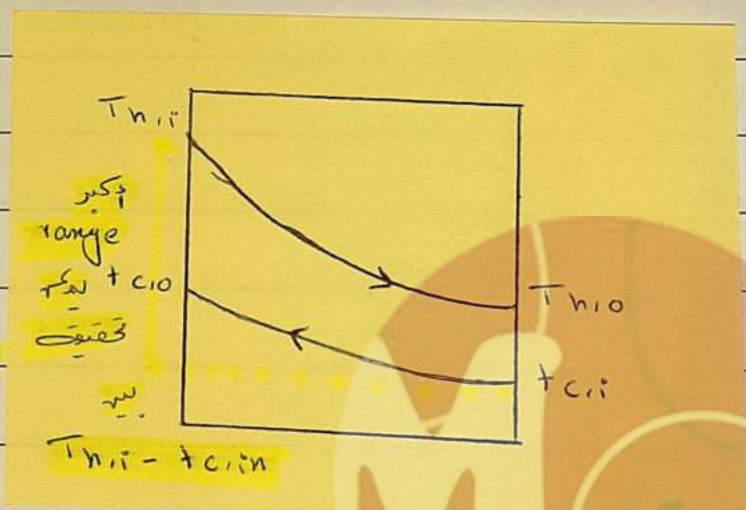
$\therefore \Delta T_c > \Delta T_h$

range of temperature أكبر لل

$T_{hi} - t_{ci}$ يكون تحقيق بين

فهذا ال range هو الذي يتم تعويض

q_{max} بـ



* The cold fluid would be heated to inlet temperature of the hot fluid

* هنا بعض C أي فهم أعوّض بـ q_{max} ؟!

بعوّض C_{min} سواء كانت C_{min}

C_c C_h

$\therefore q_{max} = C_{min} (T_{h,i} - t_{c,i})$

الحالة رقم 1 ← $C_{min} = C_c$

$\therefore q_{max} = C_c (T_{h,i} - t_{c,i})$

لهذا السبب هذه الطريقة يشترط معرفة inlet temperature

2) If $C_h < C_c$

$q_{cold} = q_{hot}$

$C_c \Delta T_c = C_h \Delta T_h$

$\therefore \Delta T_h > \Delta T_c$

* The hot fluid would be cooled to the inlet temperature of the cold fluid.

$q_{max} = C_{min} (T_{h,i} - t_{c,i})$

هنا range ثابت

اللي بيختلف فقط C_{min}

$\therefore q_{max} = C_h (T_{h,i} - t_{c,i})$

* In general, $q_{max} = C_{min} (T_{h,i} - t_{c,i})$

where C_{min} is equal to C_h or C_c whichever is smaller.

* Effectiveness, ε, is defined as the ratio of the actual heat transfer rate:

$$\varepsilon = \frac{q_{actual}}{q_{max}}$$

∴ Note:

$$q_{actual} = q_c = q_h = C_c (T_{c,o} - T_{c,i})$$

$$= C_h (T_{h,i} - T_{h,out})$$

مبدأ حفظ الطاقة

$$\therefore \varepsilon = C_h (T_{h,i} - T_{h,o}) / C_{min} (T_{h,i} - t_{c,i})$$

or

$$\varepsilon = C_c (t_{c,o} - t_{c,i}) / C_{min} (T_{h,i} - t_{c,i})$$

يعني بقدر أحسب ال ε باستخدام ال:

hot or cold stream

* Assume parallel - flow for which $C_{min} = C_h$, hence:

$$\varepsilon = \cancel{C_h}(T_{h,i} - T_{h,o}) / \cancel{C_h}(T_{h,i} - t_{c,i})$$

$$\varepsilon = (T_{h,i} - T_{h,o}) / (T_{h,i} - t_{c,i})$$

• If ε, $T_{h,i}$ & $t_{c,i}$ are known, the actual heat transfer:

$$q_{act} = \varepsilon\, q_{max}$$

$$q_{act} = \varepsilon\, C_{min}(T_{h,i} - t_{c,i})$$

إذا عرفت الـ q_{act} بستخدمها في إيجاد أي unkown من:

Texit or surface area

وكمان هنا لسا تحتاج إلى try & error

* for any exchanger it can be shown that

$$\varepsilon = f(NTU, C_{min}/C_{max})$$

this mean the ε function on $(NTU, C_{min}/C_{max})$

where:

NTU => Number of transfer unit

$$NTU = UA / C_{min}$$

where:

U => Over all heat transfer coefficient

A => Surface area

* where :

C_{min} / C_{max} is equal to C_c / C_h or C_h / C_c , depending on the relative values of heat capacity rates.

- ε and NTU can be obtained from equations or charts.

* إذا طلعت chart يرجع على المعادلات.

* Example 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 of 1 kg/s from 35° to 125°C. The overall heat transfer coefficient based on the gas side surface area is $U_h = 100\ W/m^2 \cdot K$.

Determine the required gas-side surface area A_h using the NTU method.

* exhaust gas ⇒

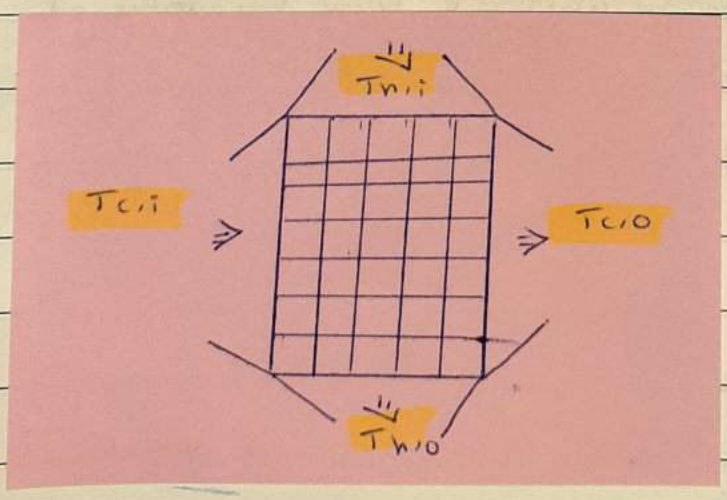
هو الغاز الخارج من المركبات ويتكون من :

CO_2 , CO , O_2 , N_2

أكبر نسبة هو N_2

* لو يكون معطيني composition من غير mass or mole fraction ثم يحدد انها percent weight يفترض

pressurized water ⇒ boiler من system ال تبل ع ا

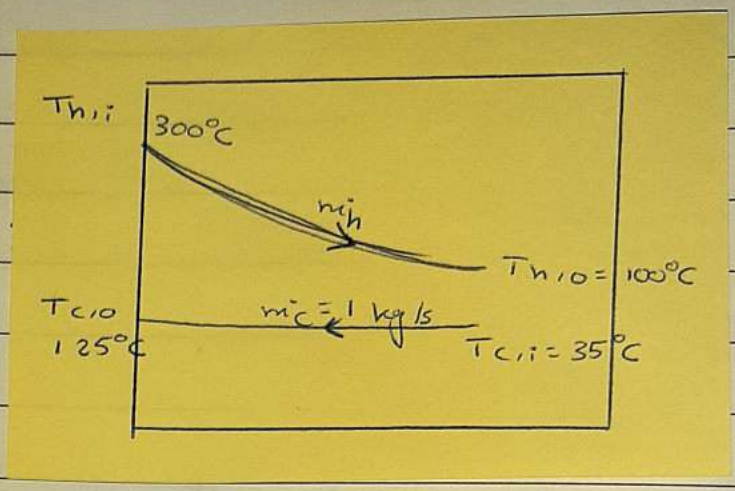


finned tube, cross flow heat exchanger

$U_h = 100\ W/m^2 \cdot K$

Both fluids unmixed

* Assume counter



Sol :

Assumptions:

1. Negligible heat loss to the surroundings and kinetic and potential energy changes.
2. Constant properties

properties:

$T_{c,avg} = 80°C$ $\quad c_p = 4197\ J/kg\cdot K$

$\Rightarrow$ for water

* Analysis:

The required surface area may be obtained from knowledge of the number of transfer units, which, in turn, may be obtained from knowledge of the ratio of heat capacity rates and the effectiveness.

* حساب بدي أدور على C_{min} :

To determine the minimum heat capacity rate, we begin by computing:

$$C_c = \dot{m}\, c_{p,c} = 1\,\frac{kg}{s} \times 4197\,\frac{kJ}{kg \cdot K} = 4197\ W/K$$

Since $\dot{m}_h$ is not specified, C_h is obtained by combining the overall energy balances:

$$q_{cooled} = q_{hot} \Rightarrow \text{Assume no loss heat}$$

$$C_c\,\Delta T_c = C_h\,\Delta T_h$$

$$C_h = \frac{C_c\,\Delta T_c}{\Delta T_h} = \frac{C_c\,(T_{c,o} - T_{c,i})}{(T_{h,i} - T_{h,o})}$$

$$C_h = 4197\,\frac{W}{K}\,\frac{(125-35)\,K}{(300-100)\,K} = 1889\ W/K$$

$$C_h < C_c$$

$$C_{min} = C_h$$

$$q_{max} = C_{min}\,(T_{h,i} - T_{c,i}) = 1889\,\frac{W}{K}\,(300-35)\,K$$

$$q_{max} = 5\times10^5\ W$$

$$q_{act} = C_c\,\Delta T_c$$

$$= 4197\,\frac{W}{K}\,(125-35)\,K = 3.78\times10^5\ W$$

$$\varepsilon = \frac{q_{act}}{q_{max}} = \frac{3.78 \times 10^5 \, \cancel{w}}{5 \times 10^5 \, \cancel{w}} = .755$$

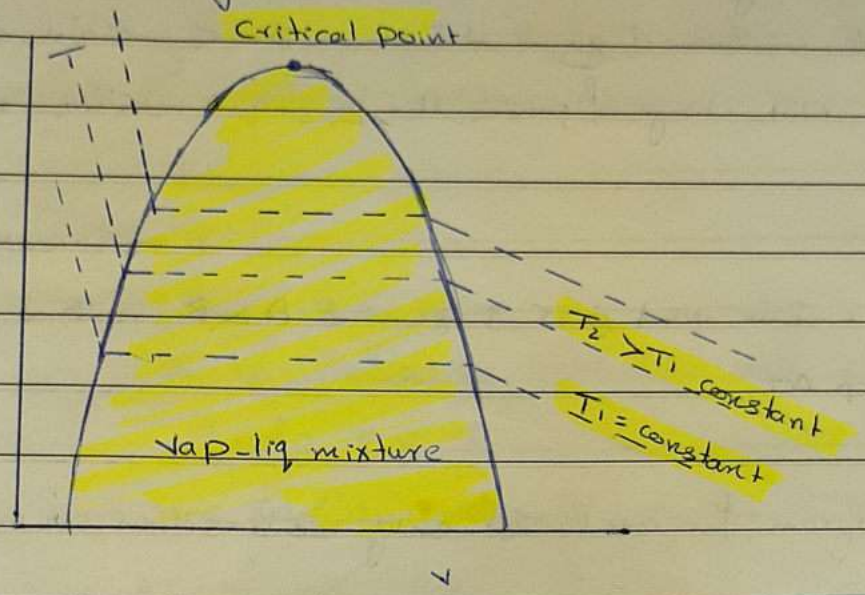
$$\frac{C_{min}}{C_{max}} = \frac{1889}{4197} = .45$$

from figure both fluids unmixed :

$$NTU = \frac{U_n A_n}{C_{min}} = 2$$

$$A_n = \frac{2 \, (1889 \, w/k)}{100 \, w/m^2 \cdot k} = 37.8 \, m^2$$

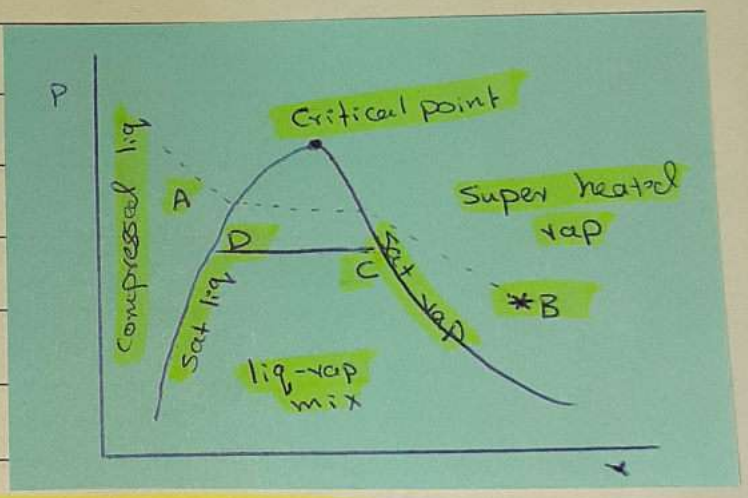
* Thermodynamics :



النقطة B ← تقع في منطقة ال super heated

مع زيادة الضغط مع ادص نقط ال Saturated vapor

تسمى درجة الحرارة عند هذا ال line dew point



* وتسمى درجة الحرارة عند نقط ال Saturated liquid bubble point

في المنطقة من C ← D يتم تحويل جميع ال vap إلى liq بثبات درجة الحرارة ، هذا يعني يتم إزالة حرارة (q) ، تسمى هذه الحرارة latent heat وهذا عمل ال condenser

* المنطقة B تحتوي على H.E ← Not change of phase

* المنطقة C ← D تحتوي H.E من نوع ال condenser ← change of phase

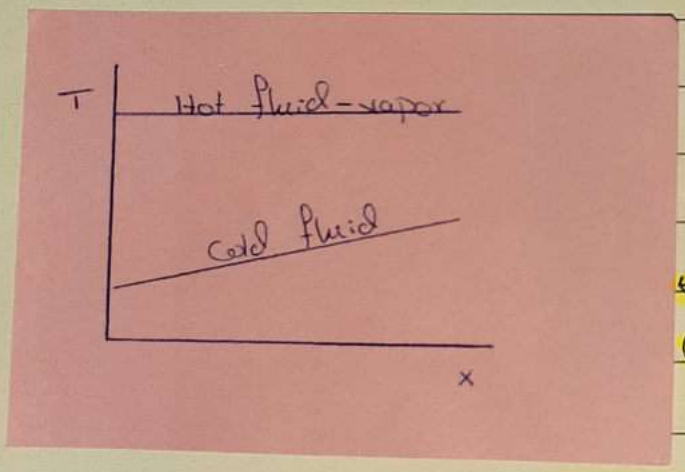
* المنطقة A تحتوي على H.E ← Not change of phase

* ملاحظة :

المنطقة A ← B نحسبها q كما في السابق مثلاً :

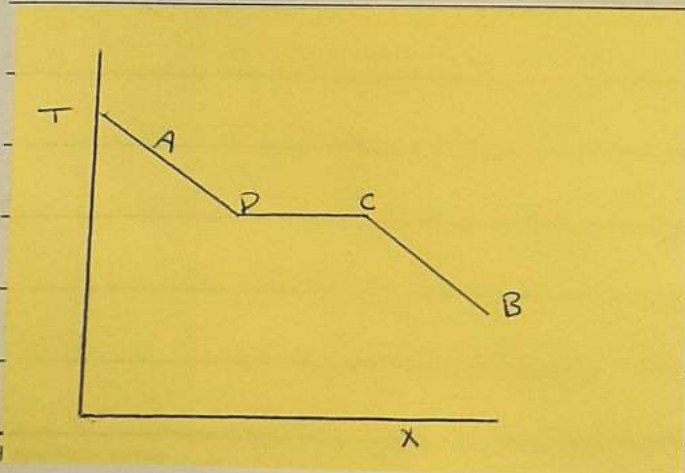
$q = m c_p \Delta T$

* T-X diagram for condenser only condensation ; No sub-cooling :



هاد ال section يعني منطقة ال mixture constant of temperature

ما بيفرق إذا صفيت بخار → أو ← (condenser) cost temp لأن



هاد يعني H.E تحتوي على 3 sections

* غالباً لما يصير انتقال للحرارة بتنقص بـ 3 modes وصعب انو أغلب
واحد من ال mode من الثلاثة .

* modes of transfer heat :

- Conduction
- Convection
- Radiation

* كيف عكس ألقى انتقال الحرارة by radiation ؟!

$$q_{rad} = \varepsilon \sigma T_s^4$$

إذا كانت T_s تؤول إلى الصفر ، بلغى انتقال الحرارة by radiation
أمّا إذا زادت ال T_s بتزيد انتقال الحرارة by radiation وبالتالي بصير
transfer by radiation هو ال dominante (المسيطر)

* ما يميز ال modes of convection هو وجود moving fluid

* Modes of transfer heat in the condensor ⇒ Convection
بسبب وجود moving fluid & سطح صلب يحدث على ال condensation

* شروط حدوث ال Condensation :

- وجود solid surface
- Temperature of solid surface must be below the saturation temperature

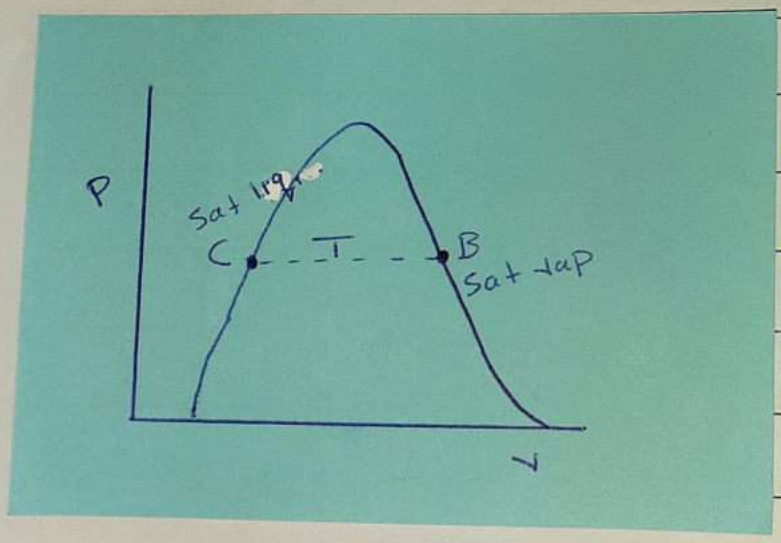
$$T_s < T_{sat\ vap}$$

* Condensation - Types and Mechanism:

- Condensation occurs when the temperature of the vapor is reduced below its saturation temperature.

- The solid surface whose temperature is below the saturation temperature of the vapor.

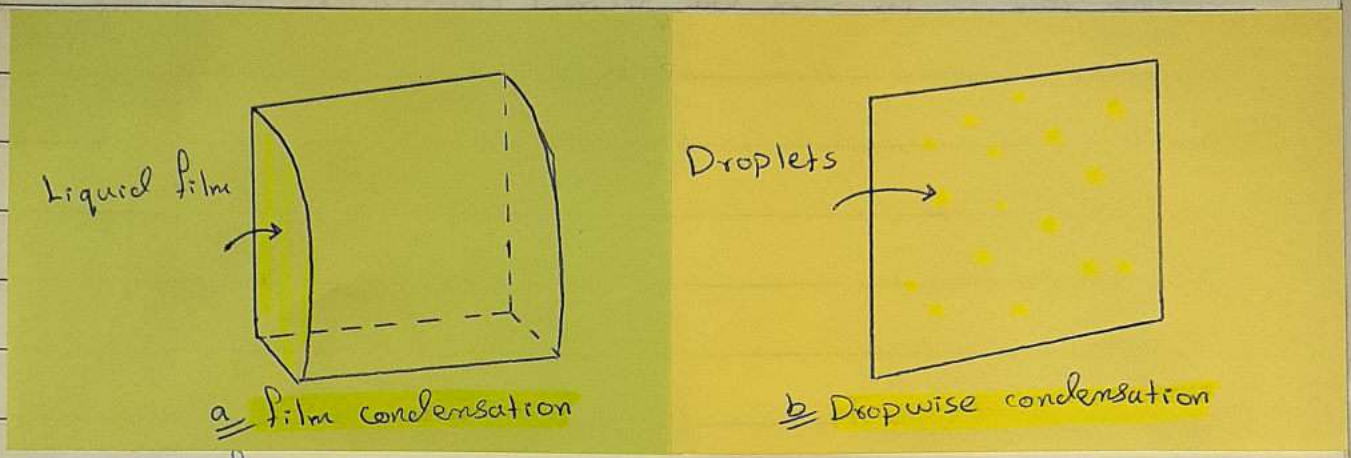
* ليش اشترطت أنّ درجة حرارة ال surface أقل من saturation temperature of vapor تحديداً ؟!



ليه ال condensor يجب أن يكون ال feed له عبارة عن vapor sat vapor فقط أو vapor بشكل عام ويكون لا يحتوي على liquid بالمرة ، لأنه يخرب الجهاز.

وعشان ال condensation تحدث من B ← C .

• Two distinct forms of condensation :



a Film condensation b Dropwise condensation

Cover film (shell model)

• gravity بيزداد مع ال thickness ال

Complete covery (contact) partially cover of solid surface

* Film wise Condensation :

- when liquid formed by condensation wets the surface. It is more common type of condensation to occur.

* أكثر شيوعاً في الصناعة

- In filmwise condensation liquid condensate forms a continuous film over the surface, this film flows down the surface under the action of gravity, shear force due to vapor flow, other forces.

* ملاحظة :

كلمة condensate تشير إلى ال liquid

أي flow بصير over surface

بينشئ عندي shear force .

* أنواع ال force المؤثرة على ال film :

- gravity - shear

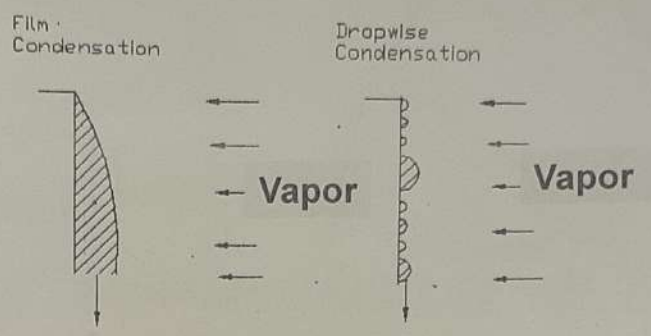
- buoyant

- The layer of liquid condensate acts as a barrier to heat flow due to its very low thermal conductivity and hence low heat transfer rate.

* layer of liquid acts as thermal resistance for heat transfer.

* انتقال الحرارة بهذا النوع أقل من النوع الثاني

Filmwise and dropwise Condensation



Heat transfer coefficient

$h_{film} << h_{drop}$

لأن film ب

★ شروط حدوث Condensation :

- وجود سطح صلب
- $T_s < T_{sat}$ (أي تكون درجة حرارة الـ surface أقل من T_{sat})

Filmwise Condensation: Nu

Idealizations:

- Laminar flow
- Constant properties
- Negligible subcooling of liquid
- Inertia effects negligible in momentum balance
- Vapor stationary, no drag
- Smooth liq-vap interface

s film by

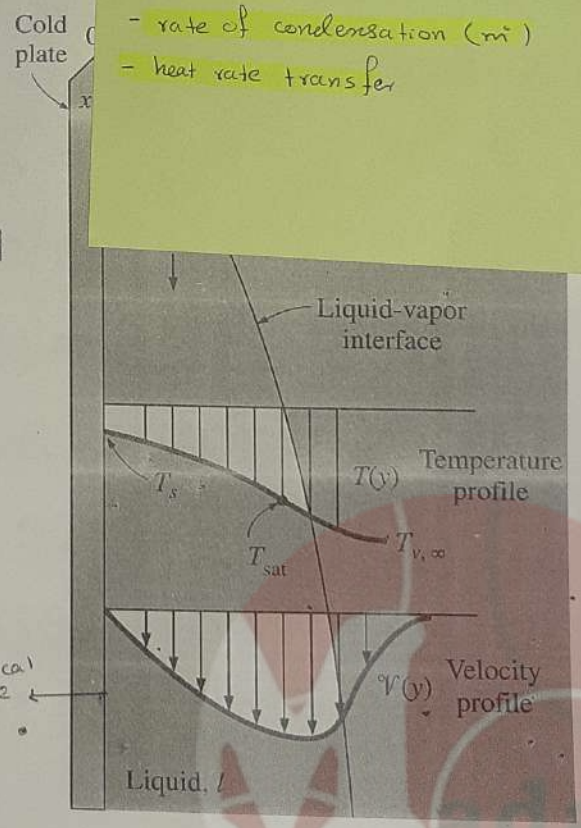
velocity of liquid at surface = 0 & increase with δ

★ بالابتعاد عن السطح الصلب تزداد الـ v

رح يكون (thickness δ) طبقة التكثف يزداد الـ thick باتجاه الـ downward.

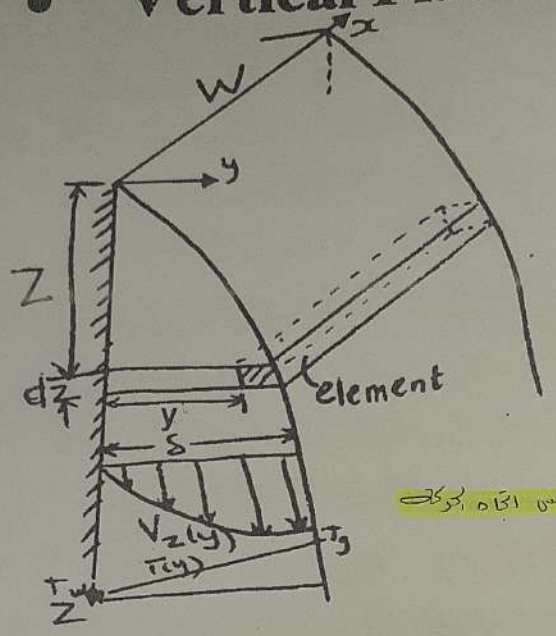
★ لإيجاد اللي بدي احسبها بجيب المسائل :

- thickness (δ)
- rate of condensation ($\dot{m}$)
- heat rate transfer



Laminar Film Condensation

- **Vertical Flat surface**



2-dimension

* نحلل الـ force على الجسم :

$\rho_v \frac{g}{g_c}(\delta - y) W dz$

buoyancy force

$\tau_{yz}\, W dz$

$\tau_{yz} W dz$

shear force

(friction)

دائماً دائماً بعكس اتجاه الحركة

$\rho_f \frac{g}{g_c}(\delta - y) W dz$

gravity force

gravity force & buoyancy force

متشابهات باختلاف بين الاختلاف

ρ_f for gravity

ρ_v for buoyancy

Definitions

Film thickness = δ at any z location

Width of the plate = W

Vol of element = (δ-y) W dz

Types of forces acting on element

1. **gravity force "+ve downward"**
2. **buoyancy force "-ve"**
3. **friction due to viscosity "-ve"**

Force Balance on Element

- **Gravity Force =Buoyancy + Friction**

$$\rho_f \frac{g}{g_c}(\delta - y)Wdz = \rho_v \frac{g}{g_c}(\delta - y)Wdz + \tau_{yz}(Wdz) \qquad (1)$$

T_g is the sat. temp. may be < ambient temp "T_∞" or = ambient temp "T_∞"

Since $\tau_{yz} = \mu \frac{dV_z}{dy}$ **. Substitute in eq. 1**

→ momentum transfer

هذا القانون مشابه لـ :

heat transfer لـ fourier's law - Newton's law for viscosity

mass transfer لـ fick's law -

$$\rho_f \frac{g}{g_c}(\delta - y)\omega dz = \rho_v \frac{g}{g_c}(\delta - y)\omega dz + \tau_{yz}(\omega dz)$$

$$\frac{g}{g_c}(\delta - y)\omega dz(\rho_f - \rho_v) = \mu \frac{dv_z}{dy}$$

$$\Rightarrow \frac{dv_z}{dy} = \frac{g}{\mu g_c}(\delta - y)\omega dz(\rho_f - \rho_v)$$

and rearranging

$$\frac{dV_z}{dy} = \frac{g}{\mu g_c}(\delta - y)(\rho_f - \rho_v)$$

Integrating gives

$$V_z = \frac{g}{\mu g_c}(\rho_f - \rho_v)(\delta y - y^2/2) + C_1$$

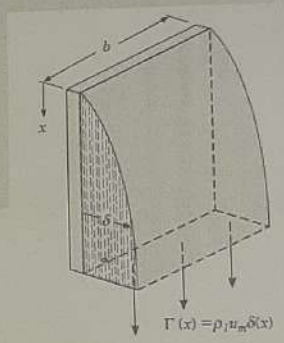
$$B.C \quad y = 0 \;,\; V_z = 0 \quad \therefore C_1 = 0$$

$\Rightarrow \frac{dv_z}{dy} \Rightarrow$ velocity gradient

$$\therefore velocity\,Profile \; V_z = \frac{\delta^2 g}{\nu_f}(1 - \rho_v/\rho_f)\left(y/\delta - y^2/2\delta^2\right) \qquad (2)$$

Note

Eq. 2 can be integrated over the cross section ' normal to the direction of the Z-directed velocity V_z' to get the mass flow rate of condensate, $\dot{m}_f$



$$\therefore \dot{m}_f = \iint \rho_f V_z \, dA$$

$$= \int_0^w \int_0^\delta \frac{\rho_f g}{\nu_f}\left(1-\frac{\rho_v}{\rho_f}\right)\left(\delta y - \frac{y^2}{2}\right) dy \, dx$$

kinematic viscosity

$$= \frac{w \delta^3 g \rho_f}{3 \nu_f}\left(1-\frac{\rho_v}{\rho_f}\right) \quad \text{.....(3)}$$

Heat transfer equation

The heat flow at the wall within the area $W\,dz$ is given by

$$q_y = -k_f\,(W\,dz)\,\frac{dT}{dy}\Big|_{y=0}$$ ⇒ fourier's law

surface temperature for film

Assume a linear temp. profile within the film varies from T_w at the wall to T_g, the saturation temp. at the liq-vap interface.

$$q_y = -k_f\,W\,dz\,\frac{T_w - T_g}{\delta}$$

* mechanism of heat transfer in boiler & condensor ⇒ Convection moving fluid

The amount of condensate between z and $z+dz$ is given by

$$\text{Added Condensate in } dz = \dot{m}_f\Big|_{z+dz} - \dot{m}_f\Big|_{z}$$

$$\text{condensat in } dz = \frac{d\dot{m}_f}{dz}\,dz = \frac{d\dot{m}_f}{d\delta}\cdot\frac{d\delta}{dz}\,dz$$

$$= \frac{d\dot{m}_f}{d\delta}\cdot d\delta$$

from eq. 3

$$\frac{d\dot{m}_f}{d\delta}\cdot d\delta = \frac{W\rho_f\,\delta^2 g}{\nu_f}\left(1-\rho_v/\rho_f\right)d\delta$$ ⇒ mass flowrate within differential element

..... (4)

يضرب بـ mass flowrate within differential element ← h_{fg} حيث يصبح هذا المقدار عبارة عن حرارة (q) ،
بساوي هذه الحرارة (q) مع fourier's law

heat vaporization

Multiplying eq.4 by h_{fg} gives q_y

$$\frac{W \rho_f \delta^2 g}{\nu_f}\left(1-\frac{\rho_v}{\rho_f}\right) h_{fg}\, d\delta = k_f W\, dz\, \frac{T_g - T_w}{\delta}$$

OR

$$\delta^3\, d\delta = \frac{k_f \nu_f (T_g - T_w)}{\rho_f g h_{fg}(1-\rho_v/\rho_f)}\, dz$$

الهدف نطلع على δ لـ film

Integrating

$$\delta^4 = \frac{4 k_f \nu_f Z (T_g - T_w)}{\rho_f g h_{fg}(1-\rho_v/\rho_f)} + C_1$$

B.C at $Z=0$, $\delta=0$ $\therefore C_1 = 0$

$$\therefore \delta = \left[\frac{4 k_f \nu_f Z (T_g - T_w)}{\rho_f g h_{fg}(1-\rho_v/\rho_f)}\right]^{1/4} \quad \text{---- (5)}$$

⇒ thickness at any high

Express H. transfer in terms of local conv. coeff. "h_z"

local at any point ← h_z $w\,dz\,(T_w - T_g) = -k_f\, w\,dz\, \frac{T_g - T_w}{\delta}$

OR

$$h_z = k_f/\delta$$

Newton's law & fourier's law

Substituting in eq. (5)

$$h_z = k_f \left[\frac{\rho_f g\, h_{fg} (1 - \rho_v/\rho_f)}{4 k_f \nu_f Z (T_g - T_w)} \right]^{1/4} \quad \text{.....(6)}$$

The Local Nusselt No is

$$Nu_z = \frac{h_z Z}{k_f} = \left[\frac{\rho_f g Z^3 h_{fg} (1 - \rho_v/\rho_f)}{4 k_f \nu_f (T_g - T_w)} \right]^{1/4} \quad \text{.....(7)}$$

Average Coeff. over the entire surface

$$\bar{h} = \frac{1}{LW} \int_0^W \int_0^L h_z \, dz \, dx$$

total length of plate — area

$w \Rightarrow$ constant

يعني بقدر ما أكامل

$\frac{1}{L}\int_0^L h_z dz$ يعني التكامل

$$= \frac{1}{L} \int_0^L k_f \left[\frac{\rho_f g\, h_{fg} (1 - \rho_v/\rho_f)}{4 k_f \nu_f Z (T_g - T_w)} \right]^{1/4} dz$$

Solving, we get

$$\bar{h} = \frac{4}{3} k_f \left[\frac{\rho_f g\, h_{fg} (1 - \rho_v/\rho_f)}{4 k_f \nu_f L (T_g - T_w)} \right]^{1/4} = \frac{4}{3} h_L \quad \text{.....(8)}$$

$$= 0.943 \left[\frac{\rho_f k_f^3 h_{fg} (1 - \rho_v/\rho_f)}{\nu_f L (T_g - T_w)} \right]^{1/4} \quad \text{.....(8)}$$

Nu based on the average coeff.

$$Nu = \frac{\bar{h} L}{k_f} = \frac{4}{3} \left[\frac{\rho_f g\, h_{fg} (1 - \rho_v/\rho_f) L^3}{4 k_f \nu_f (T_g - T_w)} \right]^{1/4} \quad \text{.....(9)}$$

condensation, as $Re < 1800 \Rightarrow$ laminar

$Re > 1800 \Rightarrow$ turbulent

$$\therefore Re = \frac{V D_h}{\nu_f}$$

V : avg. film velocity

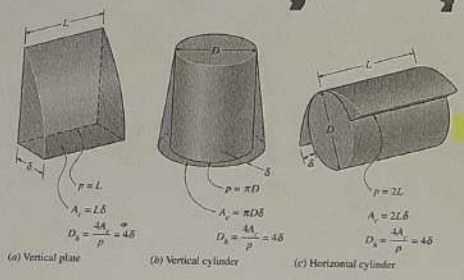
D_h : Hydraulic diam

$$D_h = \frac{4A}{P} = \frac{4\delta W}{W} = 4\delta$$

$$\therefore Re = \frac{V(4\delta)}{\nu_f}$$

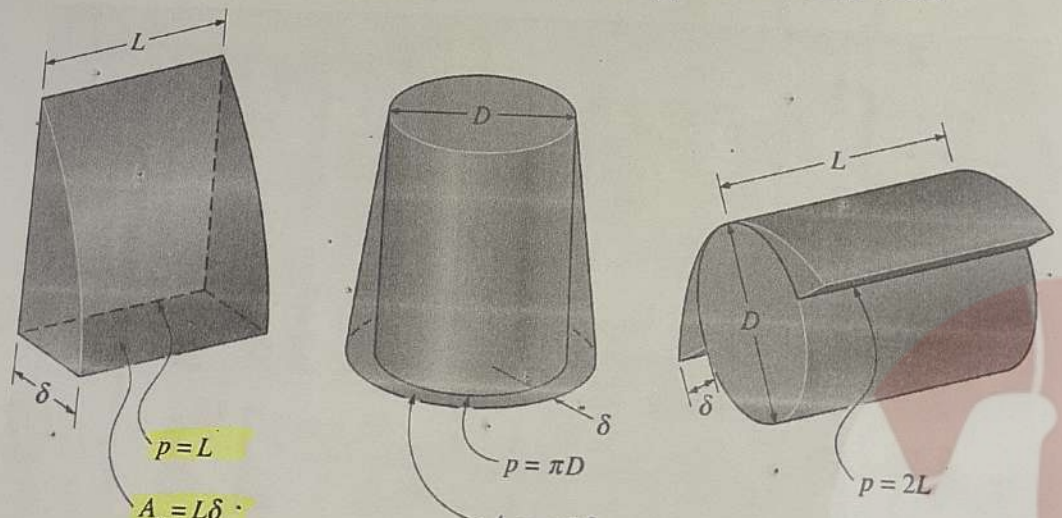
$$= \frac{\dot{m}_f}{\rho_f A} \cdot \frac{4\delta}{\nu_f} = \frac{4\dot{m}}{\rho_f(\delta W)} \frac{\delta}{\nu_f} = \frac{4\dot{m}_f}{W \rho_f \nu_f} \quad \text{.....(19)}$$

for laminar flow $Re < 1800$



(a) Vertical plate (b) Vertical cylinder (c) Horizontal cylinder

The wetted perimeter p, the condensate cross-sectional area A_c, and the hydraulic diamter D_h for some geometries



(a) Vertical plate (b) Vertical cylinder (c) Horizontal cylinder

Note 1

Re can be written in terms of conv. coef

$$\therefore q = h_L A_s (T_g - T_w) \quad \ldots\ldots\ldots (11)$$

where

A_s : surface area of plate in contact with the film

but

$$q = \dot{m}_f h_{fg}$$

$$\therefore \dot{m}_f = \frac{h_L A_s (T_g - T_w)}{h_{fg}} \quad \ldots\ldots\ldots (12)$$

Substituting into eq. (10)

$$\therefore Re = \frac{4 h_L A_s (T_g - T_w)}{W h_{fg} \mu_f \nu_f} \quad \ldots\ldots\ldots (13)$$

Note 2

- Experimental values of the convective coefficient can be as much as 20 % higher than those predicted by eq. (8) for laminar flow;

هي المعادلة أخذ من المعادلة 8

$$\bar{h} = 1.13\left[\frac{\rho_f k_f^3 h_{fg} g (1 - \rho_v / \\ \rho_f)}{\upsilon_f L (T_g - T_w)}\right]^{1/4}$$

20% بـ h derived أكبر من h experemint

Notes

1. Tube loading, $\Gamma = \frac{\dot{m}_f}{\pi D}$ Condensate (liq) per unit tube perimeter

wetted perimeter

2. Plate loading, $\Gamma = \frac{\dot{m}_f}{W}$

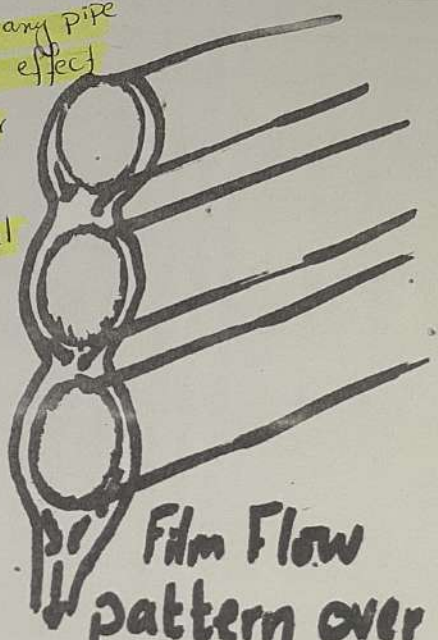
$\therefore Re_\delta \text{ (tube)} = \frac{4\Gamma}{\mu_f}$

3. for a tube bundel, $\Gamma = \frac{\dot{m}_f}{N_t \pi D}$

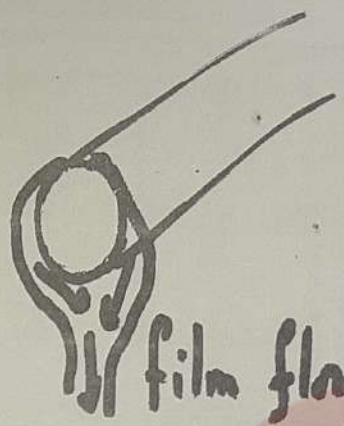
number of tube

Film Condensation on a horizontal tube and a horizontal tube bank

- assume any pipe doesn't effect on other pipe
- Isothermal



Film Flow pattern over tube bank



film flow pattern for Single tube 'horizontal'

for one tube

$$\bar{h} = 0.728 \left[\frac{g \rho_f (1 - \rho_v/\rho_f) k_f^3 h_{fg}}{\nu_f (T_g - T_w) D} \right]^{1/4}$$ ..... (15)

أقل من ال vertical

laminar filmwise

where D: outside diam

number of rows

For j tubes vertically above each other

$$\bar{h} = 0.728 \left[\frac{g \rho_f (1 - \rho_v/\rho_f) k_f^3 h_{fg}}{\nu_f (T_g - T_w) j D} \right]^{1/4}$$

نفس المعادلة السابقة بس مقسومة على j

OR

$$\bar{h}_j = \bar{h} \cdot j^{-1/4}$$ ..... (16)

Note

In practice, it is better to use index 1/6 instead of 1/4

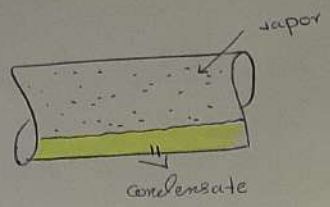
$$h_{|\,tube\ bundle} = \bar{h} \cdot j^{-1/6}$$ ..... (17)

من ناحية عملية بقدر استخدم 1/6 بتكون أدق من 1/4

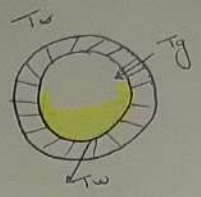
* لهون التركيز على ال condensation خارج ال pipes

* film condensation within Horizontal tubes :

- for low vapor velocities, the following expression for condensation of refrigerants can be used.



condensation in a tube conveying a two phase mixture at a low flow rate



$T_{gas} = T_{vap}$ at saturated temperature

* يتكون داخل الـ 2 phase in the pipe
يتجمع الـ liq في الـ bottom والـ vap في الأعلى

* في الأنابيب : Air condition

$$Re_D = \frac{V_v \, ID_t}{\nu_v} < 35000$$

$\nu_v \Rightarrow$ kinematic viscosity

$ID_t \Rightarrow$ inside diameter

* the Chato Equation has been derived:

$$\bar{h}_D = .555 \left[\frac{g \rho_f (1 - \rho_v/\rho_f) k_f^3 h'_{fg}}{\nu_f (T_g - T_w) ID_t} \right]^{1/4}$$

where: V_v is the average velocity of the vapor

ID is the inside diameter of the tube

$$h'_{fg} = h_{fg} + \frac{3}{8} C_{pf} (T_g - T_w)$$

Reynolds number evaluated at the inlet temperature

* Liquid properties at $(T_g + T_w)/2$

k_{fg} at T_{vapor}

* for higher flow rates, the flow becomes quite complicated. Correlations are available in the literature, however.

* ملاحظة الكتشر :

- يفترض laminar

- بشيك على Re إذا كان أقل من 1800 يكمل الحل على الـ laminar ، وإذا كان أكبر يروح على معادلات الـ turbulent

- نحسب q (heat of condensation)

- $\dot{m} = q / h_{fg}$

* في حالات يكون multi component
كيف بدي أطلع الـ properties ؟!
بدي أحسب الـ dew Temperature
وعندها نحسب الـ properties

12th - May - 2022

* Introduction

Phase change + Convective fluid motion

* Modes of heat transfer in condensation & boiling =>

Convection heat transfer due to moving fluid

* Important features:

- Large heat transfer rates with small temperature differences (nearly isothermal)

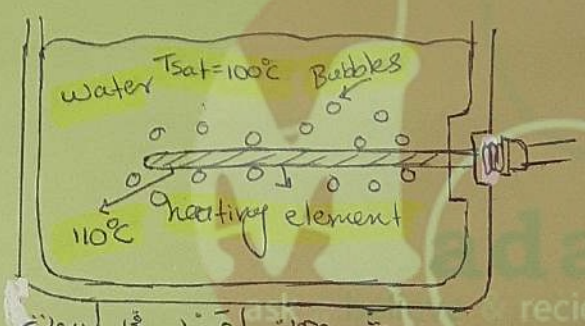
* في حالة بكون عندي change of phase العملية في درجة الحرارة بتساوي 0.

- High heat transfer coefficient
- Excellent for high heat fluxes (compact).

* Some of applications:

- Refrigeration, boilers, heat exchangers.
- Petroleum refining, chemical processing, cryogenics physical separation of gases (N_2, etc).

Boiling occurs when a liquid is brought into contact with a surface at a temperature above the sat temperature of the liq.



مثل جهاز الغيزر في البيوت

* Boiling:

Is evaporation at a solid-liquid interface, and occurs when $T_s > T_{sat}$, where T_{sat} is the temperature for liquid to gas phase change, and is a function of pressure.

e.g. for water at 1 atm, $T_{sat} = 100°C$ & $h_{fg} = 2256.4$ kJ/kg

- In boiling, the rate equation (Newton's law of cooling) is:

$$q'' = h(T_s - T_{sat}) = h\,\Delta T_e$$

where: ΔT_e is the excess temperature

* Modes of Boiling:

Boiling can be classified as:

1- Pool Boiling

quiescent liquid, motion near the surface is due to free convection and mixing due to bubble growth and detachment.

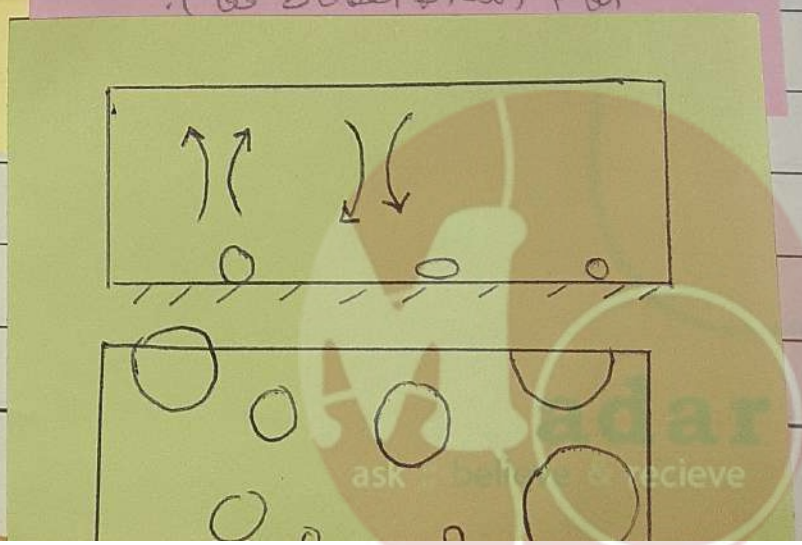
* liquid motion near surface.

تكون current (تيارات) نتيجة لـ density difference (natural convection)

وتكون فقاعات بالقرب من السطح بعدين تنفصل السائل إلى سطح السائل إلى أن تفلت من السطح وتتحول لـ vapor.

* سائل هادئ الحركة، الحركة بالقرب من السطح ناتجة عن الحمل الحراري والاضطراب بسبب نمو الفقاعات وانفصالها.

* الفقاعات عند بعضها (coalescence) القاع (تندمج الفقاعات معاً).



2- forced convection:

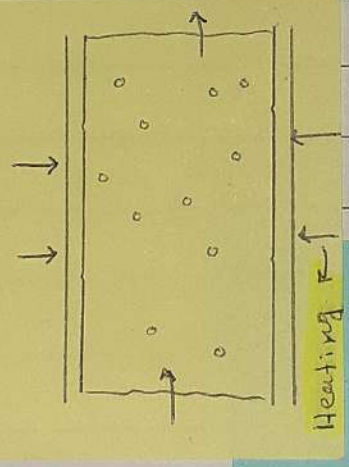
Boiling (flow boiling)

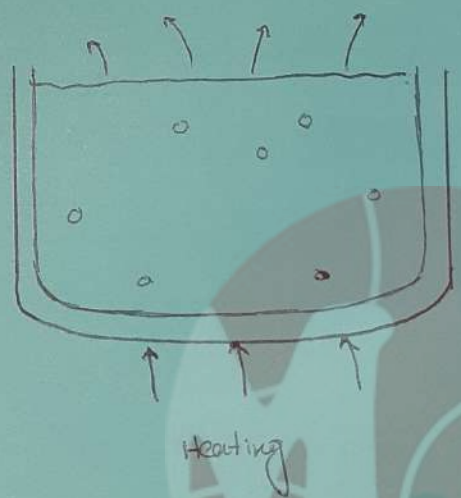
• external means drive fluid motion

* تحدث through pipes في
under pressure

في boiler system

flow boiling



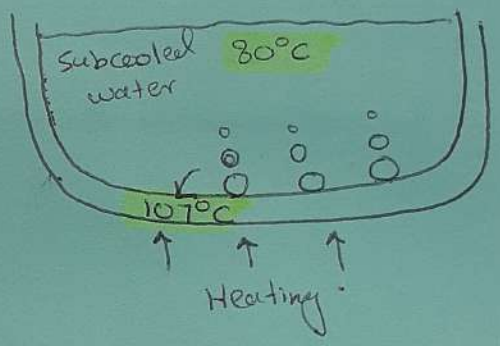


Pool boiling

* Modes of boiling:

Boiling can also be classified, alternatively, as:

- Subcooled (local) boiling
  - T liq is below Tsat
  - bubbles formed at the solid surface condense in the liquid.



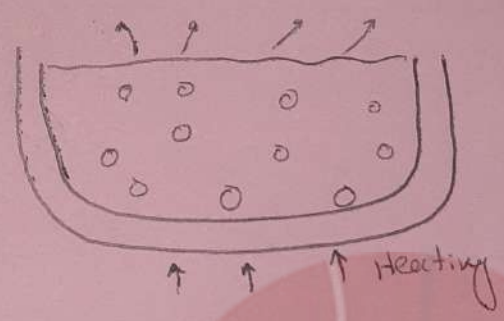
* الفقاعات ما بتوصل لسطح ال liq
لانها تتكثف شوي خلال انتقالها بال liq

* درجة حرارة ال liq < درجة حرارة ال surface
* يتكون فقاعات بالقرب من السطح الصلب، درجة حرارة الفقاعات عالية، ترتفع خلال ال liq
يحدث لها تكاثف لانها تبرد خلال ال liq درجة حرارته أقل منها، يحدث نقصان بحجم الفقاعة ثم تتلاشى

- Saturated boiling

T liq is slightly > Tsat

- bubbles can rise and escape



* الفقاعة تحافظ على حجمها، وترتفع إلى الأعلى (على سطح ال liq)، بعدين تنفجر وتتحول إلى vap.

* Dimensionless Parameter:

- Grashof like number : $[\rho g (\rho_l - \rho_v) L^3] / \mu^2$

بتبين ال buoyancy force

* Pool boiling:

Pool boiling is boiling at the surface of a body in an extensive pool of a motionless liquid.

• Variables:

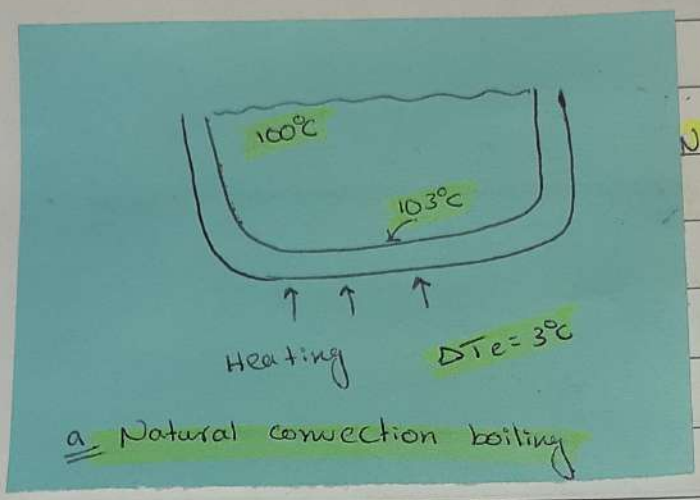
- heat flux
- thermophysical properties (liquid & vapor)
- surface material and finish
- size of the heated surface.

• Two possibilites (control ال مش ناقص)

- Temperature control برفعها أو بنزلها
- Heat flux control

* Different boiling regimes in pool boiling:

* Control of temperature: ΔT_e ⇒ بتتغير



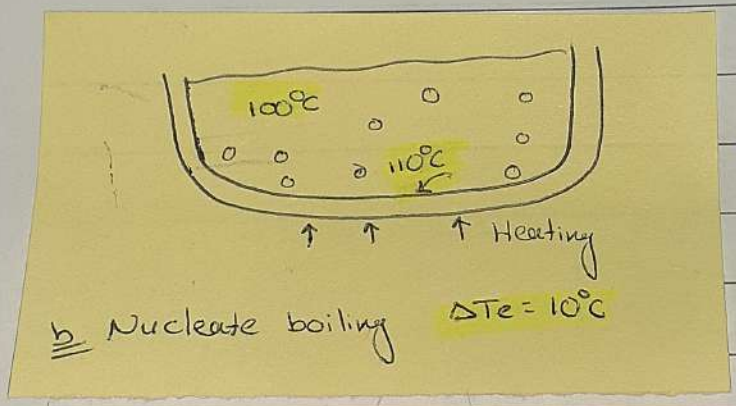
a. Natural convection boiling

ΔT_e ⇒ قليلة

نسمي ال system Natural

يكون current نتيجة

ال Density difference



b. Nucleate boiling

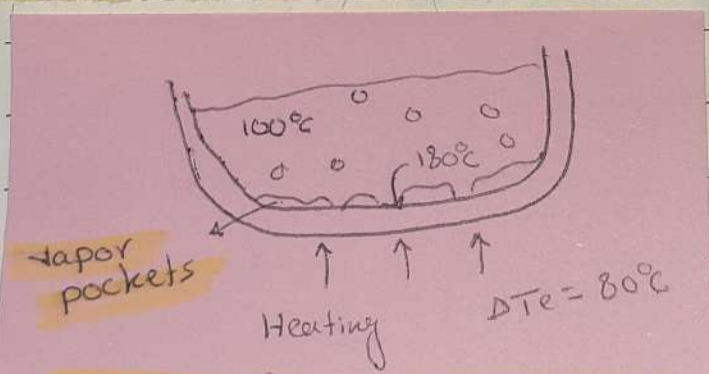
مع رفع ΔT_e بصير

generation of bubbles

الفقاعات عم

تكبر تنشق

حسب ΔT_e



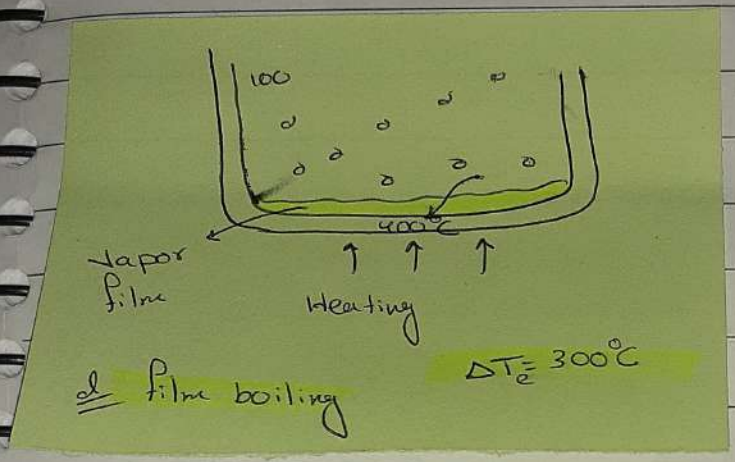
c. Transition boiling

Solid surface يتكون على ال

vapor pockets

تعمل كـ (insulation)

⇒ Resistance of heat transfer



ال film vapor بيعمل ك
(insulation).
Resistance of heat transfer.

* هي الشغل ال 4 ملخصين في chart (q'' vs ΔT_e).

- In Natural convection & Nucleate boiling
  q'' (w/m^2) ↑ with ΔT_e ↑

(Maximum critical heat flux q''_{max}) بيستمر بالارتفاع إلى نقطة تسمى (C)

- In transition boiling
  q'' ↓ with ΔT_e ↑

بسبب تكون pockets تعزل الحرارة عن ال liq
لحد ما أوصل ال minimum تسمى هذه النقطة (D)
(Leidenfrost point)

- In film boiling

$q'' \uparrow$ with $\Delta T_e \uparrow$

بترجع الحرارة ترتفع مع زيادة ΔT_e وبسبب تحول طريقة انتقال الحرارة by radiation

* heat transfer by radiation from hot solid surface.

تستمر الحرارة بالارتفاع مع ΔT إلى أن نصل إلى نقطة مميزة (E) تسمى (burn out) لأنها تقابل النقطة C

* Pool Boiling correlations:

1- Nucleat boiling

$$q = \mu_L h_{fg} \left[\frac{g(\rho_L - \rho_v)}{\sigma} \right]^{1/2} \left[\frac{C_{p,L}(T_s - T_{sat})}{C_{s,f}\, h_{fg}\, P_{r_L}^{n}} \right]^3$$

where:

$\mu_L \Rightarrow$ viscosity of liquid

$h_{fg} \Rightarrow$ heat of vaporization

$\rho_L \Rightarrow$ density of liquid

$\rho_v \Rightarrow$ " " vapor

$\sigma \Rightarrow$ surface tension

$C_{p,L} \Rightarrow$ heat capacity of liq

$C_{s,f} \Rightarrow$ surface fluid combination

n & $C_{s,f}$ يعتمد على نوع الـ solid وهي معدل يطلع منه قيمة

2. Critical heat flux:

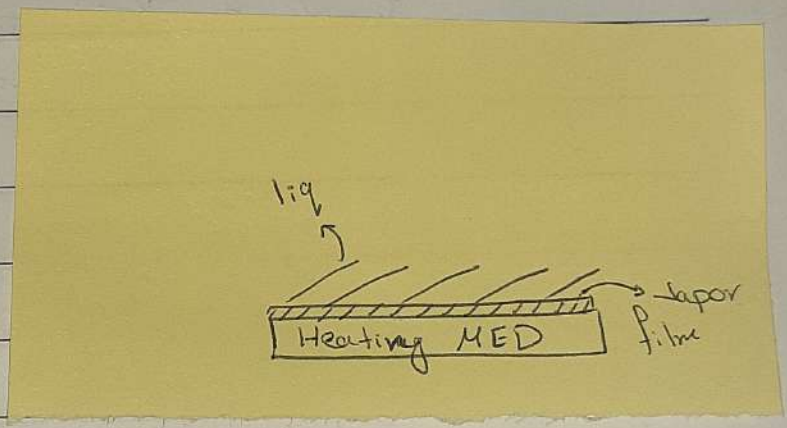
$$q_{max} = \frac{\pi}{24} h_{fg} \rho_v \left[\frac{\sigma g (\rho_L - \rho_v)}{\rho_v^2}\right]^{1/4} \left(\frac{\rho_L + \rho_v}{\rho_L}\right)^{1/2}$$

لا تعتمد على ΔT

3. Minimum heat flux

$$q_{min} = .09 \rho_v h_{fg} \left[\frac{g \sigma (\rho_L - \rho_v)}{(\rho_L + \rho_v)^2}\right]^{1/4}$$

4. film boiling:



$$Nu = \frac{\bar{h}_{con} \cdot D}{k_v} = C \left[\frac{g (\rho_f - \rho_v) h'_{fg} D^3}{\nu_v k_v (T_s - T_{sat})}\right]^{1/4}$$

where:

$C = .62$ for horz cylinder

$C = .67$ " sphere

$$h'_{fg} = h_{fg} + .8 C_{p,v} (T_s - T_{sat})$$

where:

$\bar{h}_{conv}$: average boiling H.T.C in absence of radiation.

في غياب radiation

At high temperature. $T_s \geq 300°C$; radiation mode affect the process : Total H.T.C is

$$(\bar{h})^{4/3} = (\bar{h}_{conv})^{4/3} + (\bar{h}_{rad})^{1/3}$$

If $\bar{h}_{rad} < \bar{h}_{conv}$

$$\bar{h} = \bar{h}_{conv} + \frac{3}{4}\bar{h}_{rad}$$

The effictive rad. coeff. $\bar{h}_{rad}$ is obtained from

$$\bar{h}_{rad} = \frac{\varepsilon \sigma (T_s^4 - T_{sat}^4)}{(T_s - T_{sat})}$$

where:

σ : Stefan - Boltzman cons.

ε : emissivity of the solid.

Note:

It is recommended to operate near or below the temp. excess that corresponds to critical flux "peak point".

* Radiation

All objects with a temperature above absolute zero radiate.

* Radiation propagates in vaccum. (No medium)

* heat transfer by radiation at high temperature

* يتوقف ال radiation عندما تصبح درجة حرارة الجسم = abs zero

* تأثير انتقال الحرارة بالإشعاع عند درجات حرارة عالية جداً (هو يبين أثر ال radiation)

* يعتمد الانتقال بالإشعاع على T of surface

* جميع أنواع انتقال الحرارة لا يمكن فصلها (radiation & conduction & convection)

ولكن ممكن يكون mode مسيطر أكثر من غيره.

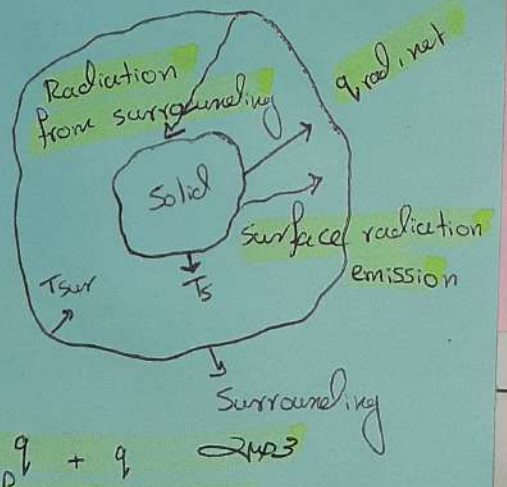
* Physical concepts:

- Mechanism of radiation:

1- Electromagnetic waves ~ Solar radiation, X-ray radio waves

2- Photons a discrete packets of energy.

* Radiation propagates in vaccum



$q_{rad} \Rightarrow q + q$ مهم

from sur to sur

* تنتقل الحرارة بالإشعاع عن طريق: تحول الحرارة إلى موجات كهرومغناطيسية تنطلق بالأجسام وتتحول إلى حرارة.

أو عن طريق الفوتونات.

- The wave nature of thermal radiation implies that the wave length - λ should be associated with the frequency of radiation, ν. The relation between λ & ν is given by:

$$\lambda = c / \nu$$

where:

$c \Rightarrow$ سرعة الضوء في الفراغ $= 3 \times 10^8$ m/s $\Rightarrow$ speed of propagation

- Thermal radiation is the radiation due to the temperature of body.

* Spectrum of electromagnetic radiation:

يهتم بالأطوال الموجية (λ) بين $(10^{-1} - 10^{2})\ \mu m$ ← يسمى

Thermal radiation

$(.4 - .7)\ \mu m$ ← (visible) ← الأطوال الطيفية

* Characteristics of blackbody (ideal):

- for a given surface temperature & wavelength, no surface can emit more radiation than a black body, a black body is a perfect emitter. (radiation هذا الجسم 100% يبعث)

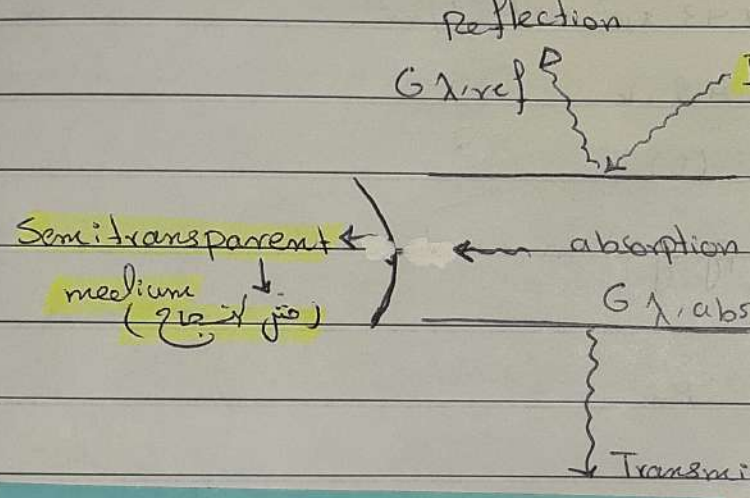
- A black body is absorb all incident radiation regardless of its direction and wavelength, a black body is a perfect absorber.

* A black body emits radiation in all directions it is a diffuse emitter.

* Cavity model of a black body:

يمتص جميع الإشعاع الساقط عليه.

* Absorption, reflection & Transmission by Real surface:



* $G_{\lambda} = G_{\lambda ab} + G_{\lambda ref} + G_{\lambda tr}$

$\frac{G_{\lambda abs}}{G_{\lambda}}$ = absorptivity

$\frac{G_{\lambda ref}}{G_{\lambda}}$ = reflectivity

$\frac{G_{\lambda tr}}{G}$ = transmetivity

مجموعهم كلهم = 1

for black surface

⇒ absorptivity = 1

* The black body gives max emissive or max absorption

* Black body emissive power:

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

where:

$C_1 = 2\pi h c^2 = 3.743 \times 10^8 \ W \mu m^4 / m^2$

$C_2 = 1.4387 \times 10^4 \ \mu m \cdot k$

T = absolute temp (k)

C = Speed of propagation, m/s

h = Plank's cons

λ = wave length, μm

* Black body Radiation function:

$$E_{b,0\to\lambda}(T) = \int_0^{\lambda} E_{b\lambda}(T)\, d\lambda$$

band

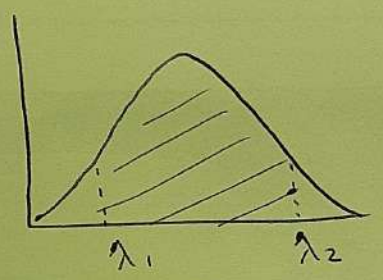
$$f_{0\lambda}(T) = \frac{\int_0^{\lambda} E_{b\lambda}(T)\, d\lambda}{\int_0^{\infty} E_{b\lambda}(T)\, d\lambda} = \frac{\int_0^{\lambda} E_{b\lambda}(T)\, d\lambda}{\sigma T^4}$$

table في جدول

* $f_{0\lambda}(T) \Rightarrow$ black body Radiation function given in table

* over a finite -wave length band $\lambda_1 - \lambda_2$

$$f_{\lambda_1-\lambda_2}(T) = f_{0-\lambda_2}(T) - f_{0-\lambda_1}(T)$$



$f_{0-\lambda_2}(T) - f_{0-\lambda_1}(T)$

* Example (3):

A light bulb radiates like a black body at 2800k. find the percentage of total emitted energy that lies in the visible range.

visible rang $\Rightarrow$ $\lambda = (.4 - .7)\ \mu m$

$$f_{\lambda_1-\lambda_2}(T) = f_{0-.7} - f_{0-.4}$$

$\lambda T = .7(2800) = 1960$ بروح على table بطلع

$\lambda T = .4(2800) = 1120$ قيمة F ثم بطرحهم

من بعض

* القيمة اللي طلعتها عبارة نسبة إشعاع الجسم للطاقة إذا ضربتها

بالنسبة الكلية (total) ← ج رح تعطيني quantity emitted

$$\int_0^\infty E_{b\lambda}(T) = \sigma T^4$$

* Radiation properties of surfaces

- Absorption and emission of radiation depend on :
  - Type of material
  - Temp or wavelength

* Type of materials

metals, wood, stone	glass, water
- Opaque to thermal radiation.	- Semitransparent to radiation
معتم للإشعاع الحراري	شبه شفاف للإشعاع
- Emission or absorption within a very short distance.	- penetrate into the depth
	تخترق العمق
يقطع الإشعاع مسافة قصيرة داخله	

* Note:

Most engineering materials are opaque to thermal radiation (gray)

معظم المواد تكون معتمة وتسمى (gray)

* Emissivity: ϵ :

$$\epsilon = \frac{\text{Energy emitted by ~~areal surface~~}}{\text{Energy emitted by a black body}} \Big\} \text{ at the same temp}$$

ϵ for black body = 1

* $\epsilon = f$ (T, wavelength λ , Direction)

* Hemi spherical emissivity $\epsilon(T)$ => reference shape

Definition:

$$\epsilon(T) = \frac{\text{Rad. flux by real body overall wavelength}}{\text{energy emitted by black body at same T}}$$

" into a hemi spherical space "

$$\epsilon(T) = \frac{q(T)}{E_b(T)} \quad \text{OR}$$

$$q(T) = \epsilon(T)\, E_b(T) = \epsilon(T)\, \sigma T^4 \ \text{w/m}^2$$

* Note 1 :

Oxidation ↑ $\epsilon(T)$ ↑

كل ما ال metal ال is Oxidation بتزيد ال ϵ

* Note (2) :

Spectral hemispherical emissivity ϵ_λ

at a given λ

يتم حسابها experemental

$\equiv$ emissivity at a specific wavelength λ at a certain temp

* determine experimentally

* Can be used to obtain the average value of ϵ like this

$$\epsilon = \frac{\int_0^\infty \epsilon_\lambda E_{b\lambda}(T)\, d\lambda}{\int_0^\infty E_{b\lambda}(T)\, d\lambda} = \frac{\int_0^\infty \epsilon_\lambda E_{b\lambda}\, d\lambda}{\sigma T^4} \quad \text{--- 1}$$

look $\int_0^\infty \epsilon_\lambda E_{b\lambda}(T)\, d\lambda$ can be treated as band of cons ϵ

يتم تقسيم التكامل إلى فترات بافتراض أن ϵ ثابتة

from 0 to λ_1 $\quad \epsilon_\lambda = \epsilon_1 = \text{cons}$

from λ_1 to λ_2 $\quad \epsilon_\lambda = \epsilon_2 = \text{cons}$

from λ_2 to ∞ $\quad \epsilon_\lambda = \epsilon_3 = \text{cons}$

ويعوضهم في المعادلة 1

$$\epsilon = \frac{\int_0^{\lambda_1} \epsilon_1 E_{b\lambda}(T)\, d\lambda}{\sigma T^4} + \frac{\int_{\lambda_1}^{\lambda_2} \epsilon_2 E_{b\lambda}\, d\lambda}{\sigma T^4} + \frac{\int_{\lambda_2}^{\infty} \epsilon_3 E_{b\lambda}\, d\lambda}{\sigma T^4}$$

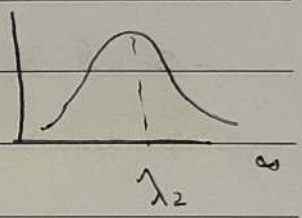
ϵ_1 & ϵ_2 & ϵ_3 ⇒ constant

$$\epsilon = \epsilon_1 \frac{\int_0^{\lambda_1} E_{b\lambda}(T)\, d\lambda}{\sigma T^4} + \epsilon_2 \frac{\int_{\lambda_1}^{\lambda_2} E_{b\lambda}\, d\lambda}{\sigma T^4} + \epsilon_3 \frac{\int_{\lambda_2}^{\infty} E_{b\lambda}\, d\lambda}{\sigma T^4}$$

$$\epsilon = \epsilon_1 f_{0\to\lambda_1} + \epsilon_2 [f_{0\to\lambda_2} - f_{0\to\lambda_1}] + \epsilon_3 [1 - f_{0\to\lambda_2}]$$

$0 \to \lambda_2 + \lambda_2 \to \infty = 1$ ⇐

$\lambda_2 \to \infty = 1 - 0 \to \lambda_2$



⇒ knowing ϵ_1, ϵ_2 & ϵ_3, we can obtain ϵ from the previous equation by using the table of black body radiation functions "$f_{0\lambda}(T)$"

* Emission of Radiation:

- Monochromatic radiation ≡ radiation of a single wavelength.
- A beam of thermal radiation is not monochromatic but a wide or a band of wavelengths.

(ban ⇒ wavelength ، مدى ⇒)

- At temp above 500°C ⇒ heat radiation in the visible spectrum becomes significant "Red heat" and "white heat" are commonly used.

تجمع الجزيئات

* State of aggregation and Molecular structure of the substance affect radiations

ذرة واحدة ← - Mono atomic and diatomic (جزيئة) gases 'O_2, N_2, Ar' radiate weakly even at high temp.

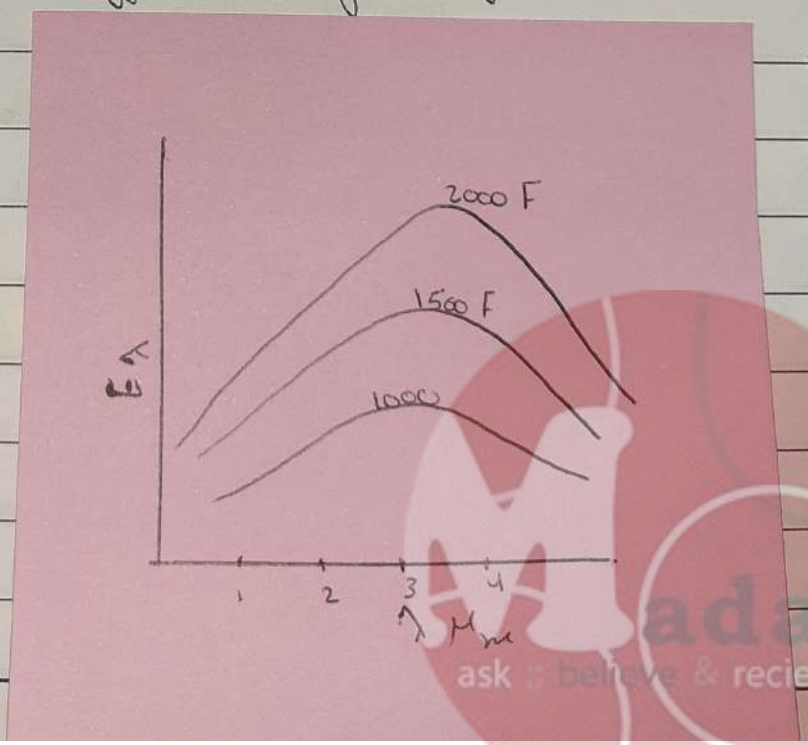
- Poly atomic gases (H_2O, CO_2) emit and absorb radiation at several wavelength.

- Solid & liquids absorbs and emit radiation over the entire spectrum in thin layers

* Radiation travels in astraight line "beam". like ligh
* Net energy transfer from heter body A to cold body B.

* Mono Chromatic energy emitted by an object depends on (Temp & λ)

تزداد E مع λ لنقطة معينة
(max) يعبر بمعنى تم
تزداد E مع T



* Types of Bodies :

1- Opaque body (معتم) => No transmission of electromagnetic radiation is possible

$t = 0$ $\alpha + r = 1$

2- White body : any body whose surface reflects incident rays of radiation

$\alpha = 1$ $r = 1$

الاشعة في الصيف بتصطدم بالاجسام البيضاء السطح ، لذلك الامتصاصية

$= 0$

3- Black body :

$\alpha = 1$ $r = 0$

4- Gray body

$0 < \alpha < 1$

* Absorptivity similar to emissivity

- Hemispherical absorptivity α
- Spectral hemispherical absorptivity α_λ

$\alpha = f (T_{source} , Material \lambda)$

حسب طبيعة المادة عكس α تتوقف مع الحرارة

وعكس تم

* Gray body Approximation :

في الماده ϵ & α تعتمد على λ وهذا معنى عملي في حساب الحرارة عند كل λ .

لحل هذي المشكلة اعتبرنا ان ϵ ثابتة (uniform) وهيك صار

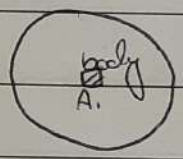
Gray body $\neq f(\lambda)$

* Relation between ϵ & α

after equilibrium :

T body = T enclosure

sphere & body تتساوى درجات الحرارة بين ال



hollow sphere at conc. T

I ⇒ The intensy of radiation falling upon the body A_1 .

شدة الإشعاع الساقط على الجسم

E_1 ⇒ The total emessive power by the body.

الطاقة المنبعثة من الجسم

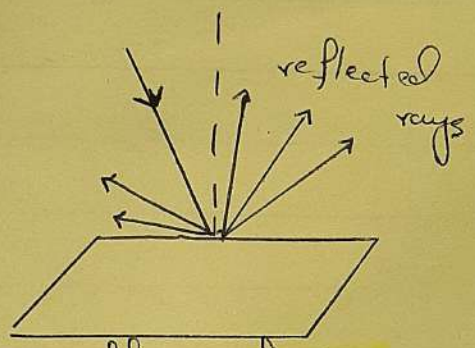
The energy emitted by the body A_1 = The energy recived.

$E_1 \cancel{A_1} = I \alpha_1 \cancel{A_1}$

$E_1 = I \alpha_1$

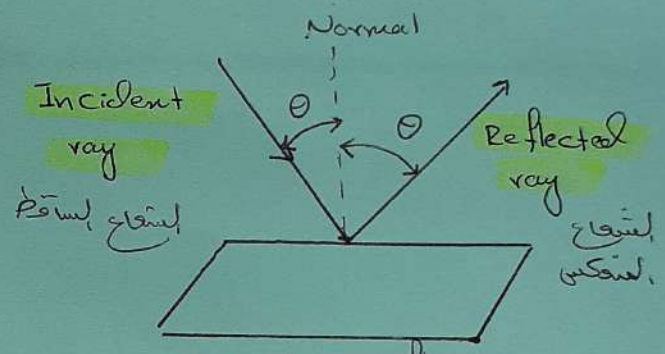
for black body $\alpha = 1$

* Radiation Intensity: كثافة الإشعاع



Diffuse reflection

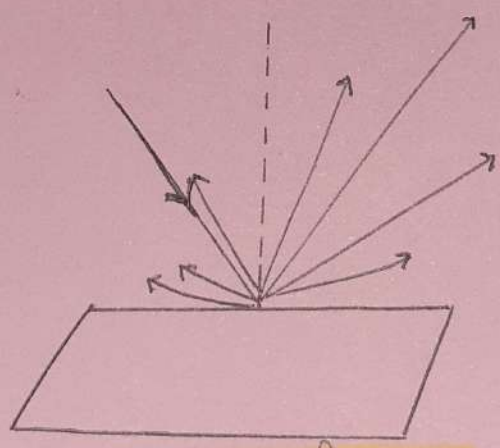
* هاد الحقيقي



Specular reflection

The surface is very smooth

θ of incidence = θ of Reflection

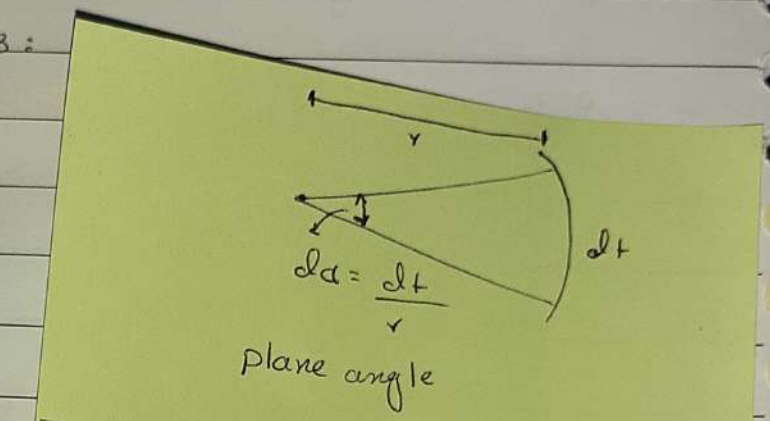


Irregular reflection

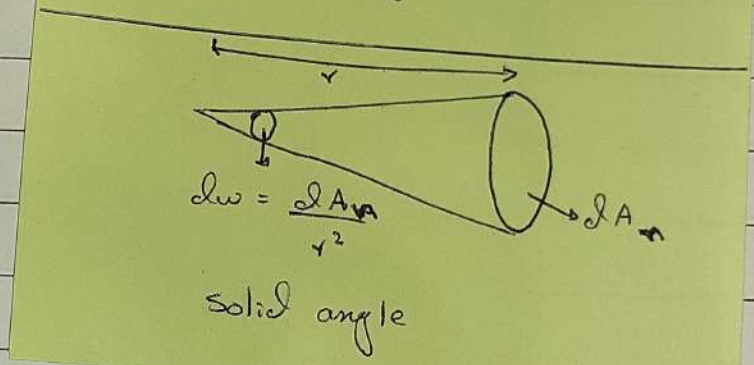
* أفرض coordinate system لتمثيل الاتجاهات (اتجاه بسيكاس) ؟

hemi spherical coordinate system

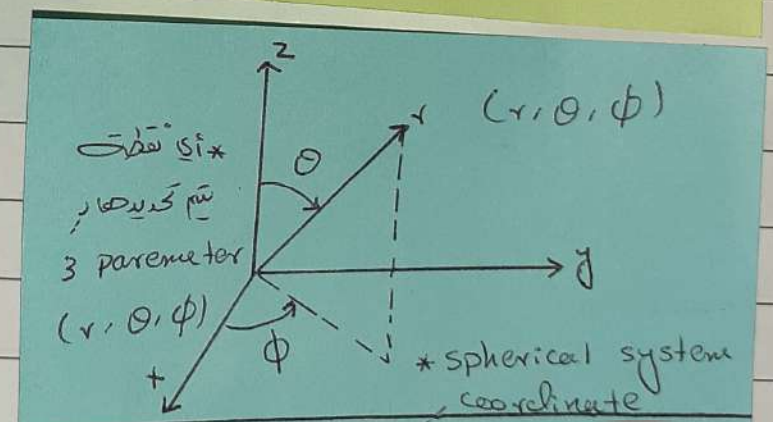
* Mathemitical Definitions :



plane angle

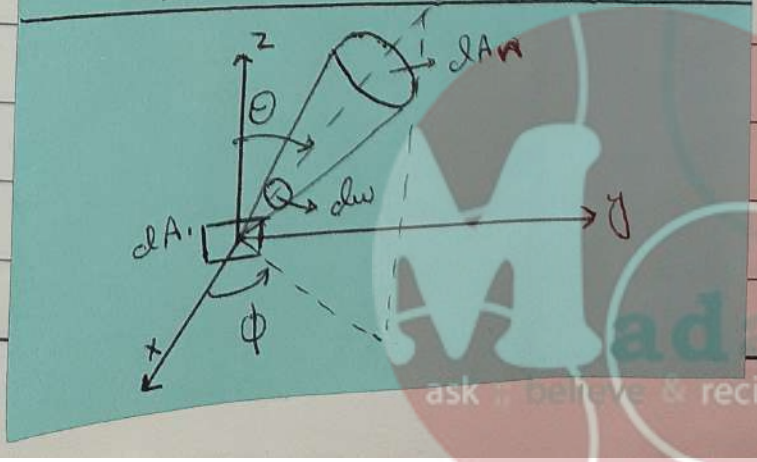


solid angle



* أي نقطة يتم تحديدها بـ 3 parameter (r, θ, ϕ)

* spherical system coordinate



* معطيات ال solid angle داخل ال spherical coordinate system.

* Radiation Intensity, I

$$I(\lambda, \theta, \phi) = \frac{dq}{dA_1 \cos\theta \cdot d\omega \cdot d\lambda}$$

I λ,e depend on:

- q
- Geometry (A)
- direction (θ)
- solid angle
- wavelength

$$\frac{dq}{d\lambda} = dq_\lambda = I_{\lambda e}(\lambda, \theta, \phi)\, dA_1 \cos\theta\, d\omega$$

where: $(dq/d\lambda)\, dq_\lambda$ is the rate at which radiation of wavelength λ leaves dA_1 & passes through dA_n

But solid angle is:

$d\omega = \sin\theta\, d\theta\, d\phi$,

Therefore, the heat flux from dA_1 becomes:

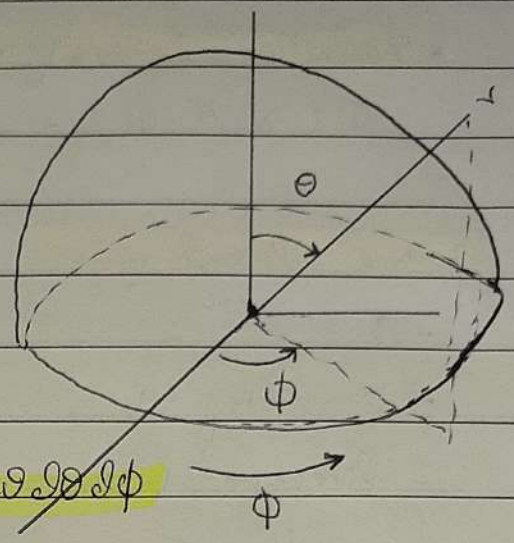
$$dq''_\lambda = I_{\lambda e}(\lambda, \theta, \phi) \cos\theta \sin\theta\, d\theta\, d\phi$$

* Hemispherical direction:

$\theta \Rightarrow$ من الرسمة تتراوح من

$0 \rightarrow \frac{\pi}{2}$

$\phi \Rightarrow$ $0 \rightarrow 2\pi$ تتراوح من



$$E_\lambda(\lambda) = q''_\lambda(\lambda) = \int_0^{2\pi}\int_0^{\pi/2} I_{\lambda e} \cos\theta \sin\theta \, d\theta \, d\phi$$

* The total, hemispherical emissive power, E (w/m²) is the rate at which radiation is emitted per unit area at all possible wavelengths and in all possible directions:

المعادلة رقم 1 كنت مقسم ال q على A & λ

هي الفترة بدها تفتح مجال ال q وتضرب المعادلة السابقة بـ λ

فبتصير ال E تحتوي على 3 تكامل

$$E = \int_0^{\infty}\int_0^{2\pi}\int_0^{\frac{\pi}{2}} I_{\lambda e}(\lambda, \theta, \phi) \cos\theta \sin\theta \, d\theta \, d\phi \, d\lambda$$

* Suppose a diffuse surface (emitted radiation is independent of direction), intensity, I, doesn't depend on direction θ and φ ⇒ $I_{\lambda,e}(\lambda, \theta, \phi) = I_{\lambda,e}(\lambda)$. Hence, the previous eq. becomes:

$$E = \pi I_e$$

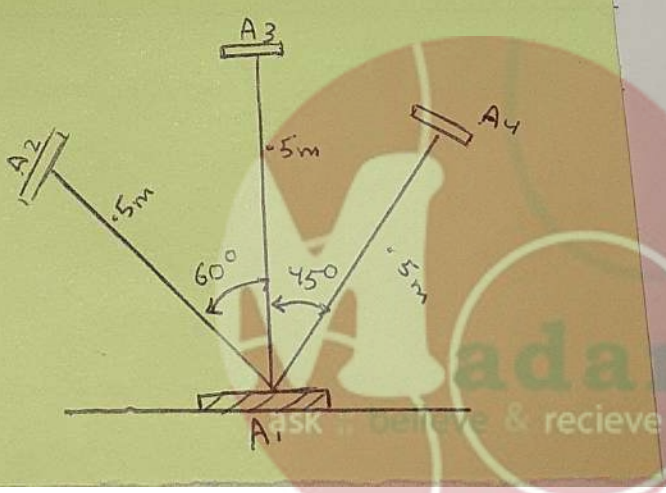
* بنفهم بدراسة إ Diffuse surface لـ لا يعتمد على إ direction
يعني قيمة E لا تعتمد على الزوايا

تصبح المعادلة السابقة إ E لـ $\cos\theta \sin\theta \, d\theta \, d\phi$ عبارة عن
ثابت بنطلع بين التكامل بتضل E تعتمد على λ & I فقط

* The constant in the above in the above equation is π and not 2π and has the unit steradians
وحدة إ solid angle ↓ زي إ radians & degree

* ex: A small surface of area $A_1 = 10^{-3} m^2$ is known to emit diffusely, and from measurements the total intensity associated with emission in the normal direction is $I_n = 7000 \, W/m^2 \cdot st$.

* Radiation emitted from the surface is intercepted by 3 other surfaces of Area $A_2 = A_3 = A_4 = 10^{-3} m^2$, which are 0.5 m from A_1 and are oriented as shown.



what is the intensity associated with emmision in each of the three directions ? what are the solid angles subtended by the three surfaces when viewed from A_1 ? what is the rate at which radiation emitted by A_1 is intercepted by the the three surfaces ?!

* الإشعاع خارج من A1 يتم اعتراضه بواسطة ٣ أسطح A2 & A3 & A4
- شدة الإشعاع في كل اتجاه من الاتجاهات الثلاثة ؟!
- solid angle عند النظر من A1 ؟!
- معدل الإشعاع المنبعث من A1 ؟!

a) Diffuse surface ⇒ لا يعتمد على الاتجاهات (ثابت)

$\therefore I = 7000 \ w/m^2.sr$

Continue ...

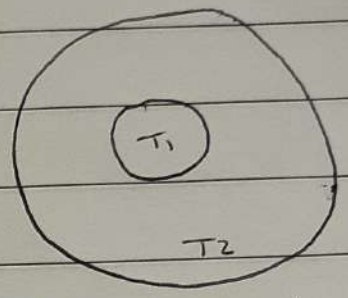
* General Comment :

The Net heat transfer between Two surfaces :

$$Q_{net} = A_1 \sigma (T_2^4 - T_1^4)$$

black body يحتوي على ε

- If T_1 is more than T_2, the black body cools down. because the net loss of heat until T_1 and T_2 are equal.



Black body inside another black body

لح يستمر انتقال الحرارة من 1 الى 2 لحد ما تصير درجات الحرارة متساوية

- If T_1 is less than T_2, net heat will added to the black body until T_1 & T_2 are again equal.

* Note:

for gray body inside another gray body

$$Q_{net} = A_1 \varepsilon_1 \sigma T_1^4 - A_1 \alpha_1 \sigma T_2^4$$

Example 1

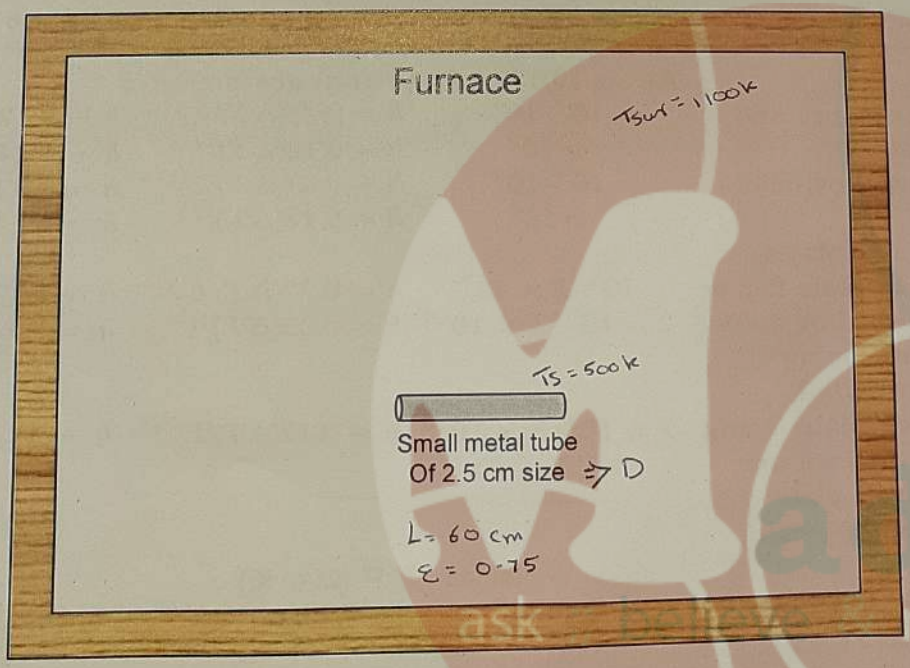
A small oxidized metal tube of diam = 2.5 cm and long = 60 cm, Ts = 500 K is in a very large furnace enclosure with fire-brick walls and surrounding air at 1100 K. Assume oxidized metal tube emissivity = 0.75. Calculate the heat transfer to the metal tube.

$q = h_r (T_{sur} - T_s)$

$= \varepsilon \sigma (T_{sur}^4 - T_s^4) \cdot A$

$= 0.75 \times 5.67 \times 10^{-8} \frac{W}{m^2 \cdot K^4} (1100^4 - 500^4) K^4 \times \pi \times .025\, m \times 0.6\, m = 28.087\, kW$

Schematic



Example 2

Recalculate Example 1 for combined radiation plus natural convection to the

* أول شي بدي أحسب h_{conv} عن طريق :
- بحدد ال regime هل laminar or turbulent
- بعدين بختار معادلة Nu المناسبة حسب ال geometry & regime وحسب قيمة h_{conv}
- بعدين بحسب q_{con} وبجمعها مع q_{rad} السابق
* أو بستخدم هاد ال table في حساب قيمة h_{conv}

$Pr \Big|_{1100} = 0.728$

Note: *Simplified Equations for Natural Convection from Various Surfaces*

Physical Geometry	$N_{Gr} N_{Pr}$	Equation: $h = btu/h \cdot ft^2 \cdot °F$, $L = ft$, $\Delta T = °F$, $D = ft$	Equation: $h = W/m^2 \cdot K$, $L = m$, $\Delta T = K$, $D = m$
	Air at 101.32 kPa (1 atm) abs pressure		
Vertical planes and cylinders	10^4–10^9	$h = 0.28(\Delta T/L)^{1/4}$	$h = 1.37(\Delta T/L)^{1/4}$
	$> 10^9$	$h = 0.18(\Delta T)^{1/3}$	$h = 1.24\ \Delta T^{1/3}$
Horizontal cylinders	10^3–10^9	$h = 0.27(\Delta T/D)^{1/4}$	$h = 1.32(\Delta T/D)^{1/4}$
	$> 10^9$	$h = 0.18(\Delta T)^{1/3}$	$h = 1.24\ \Delta T^{1/3}$
Horizontal plates			
Heated plate facing upward or cooled plate facing downward	10^5–2×10^7	$h = 0.27(\Delta T/L)^{1/4}$	$h = 1.32(\Delta T/L)^{1/4}$
	2×10^7–3×10^{10}	$h = 0.22(\Delta T)^{1/3}$	$h = 1.52\ \Delta T^{1/3}$
Heated plate facing downward or cooled plate facing upward	3×10^5–3×10^{10}	$h = 0.12(\Delta T/L)^{1/4}$	$h = 0.59(\Delta T/L)^{1/4}$
	Water at 70°F (294 K)		
Vertical planes and cylinders	10^4–10^9	$h = 26(\Delta T/L)^{1/4}$	$h = 127(\Delta T/L)^{1/4}$
	Organic liquids at 70°F (294 K)		
Vertical planes and cylinders	10^4–10^9	$h = 12(\Delta T/L)^{1/4}$	$h = 59(\Delta T/L)^{1/4}$

example 2

$N_{Gr} \Rightarrow$ Grashof number

$$Gr = g\,\beta\,\frac{(T_{wall} - T_\infty)\,L^3}{\nu^2}$$

$g \Rightarrow$ gravity acceleration

$\beta \Rightarrow \frac{1}{T_\infty}$ for T_∞ in k

$T_\infty \Rightarrow$ buk temperature

$\nu \Rightarrow$ kinematic viscosity

ν for air at 1100 k° = $141.8 \times 10^{-6}\ m^2/s$

$$Gr = 9.8\,\frac{m}{s^2} \times \frac{1}{1100\,k}\,(1100 - 500)\,k \times .6^3\,m^3\,\frac{s^2}{(141.8\times10^{-6})^2\,m^4}$$

$Gr = 5.7423 \times 10^7$

$Gr \times Pr = 4.18 \times 10^7$

$h_{conv} = 1.32\,(\Delta T / D)^{1/4} = 1.32\,(600/.025)^{1/4}$

$h_{conv} = 16.42\ W/m^2 \cdot k$

$q''_{conv} = h\,(T_\infty - T_w) = 16.42\,\frac{W}{m^2 \cdot k}\,(1100 - 500)\,k = 9852\,\frac{W}{m^2}$

$q_{con} = q''_{conv} \times A = 9852\,\frac{W}{m^2} \times \pi\,(.025)\,m \times .6\,m = 464.26\ W$

$q_{tot} = q_{rad} + q_{conv} = 28.087 + .46426\,kW = 28.55\ kW$

Example 3

A light bulb radiates like a black body at 2800 K. find the percentage of total emitted energy that lies in the visible range.

Example 3:

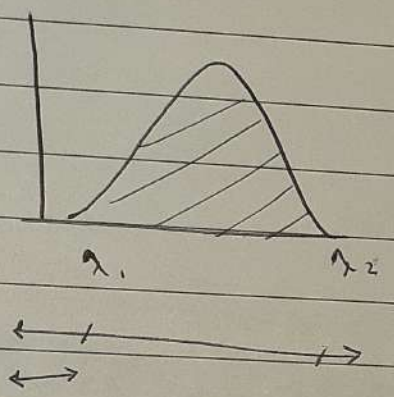
$T_s = 2800\,k$

Find the percentage of total emitted energy from $(.4 - .7)\,\mu m$

black body radiation هي دالة في استخدام function

visible range

$$f_{\lambda_1 - \lambda_2}(T) = f_{0 - \lambda_2}(T) - f_{0 \to \lambda_1}(T)$$



$$f_{.4 - .7}(T) = f_{0 \to .7}(T) - f_{0 - .4}(T)$$

$\lambda_1 T = .4 \times 2800 = 1120\ \mu m \cdot k$

$\lambda_2 T = .7 \times 2800 = 1960\ \mu m \cdot k$

من الـ table بطلع قيمة $f_{0 \to .7}$ & $f_{0 \to .4}$

$f_{.4 - .7} = .0612506 - .002134 = .0591166$

by interpolation — interpolation — percentage of total emitted

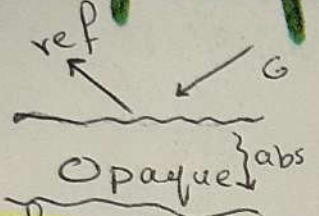
إذا طلب الـ quantity emitted بضربها بالكمية الكلية للإشعاع (σT_s^4)

$$\text{quantity emitted} = .0591166 \times 5.67 \times 10^{-8} \frac{w}{m^2 \cdot k^4} \times (2800)^4\, k^4$$

$$= 206.027\ \frac{kw}{m^2}$$

* Types of Bodies:

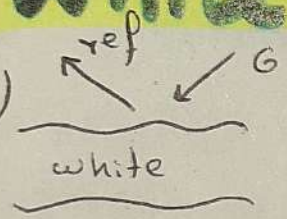
1. Opaque body:



no transmission of electromagnetic radiation is possible. $t=0$, $\alpha + r = 1$

$(G = ref + abs)$ ÷ G

2. White body:



÷ $(G = ref)$

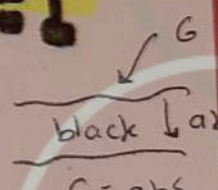
$\boxed{1 = r}$

any body whose surface relects incident rays of radiation. $\alpha = 0$, $r = 1$

ما بيمتص الإشعاع كله بيعكسه
لهيك بالعين
يعطي اللون الأبيض

3. Black body:

$\alpha = 1$, $r = 0$



$G = abs$

$\boxed{1 = a}$

4. Gray body:

$0 < \alpha < 1$

مضروبين لـ I في جزء الطاقة المتشعة

$$E_1 A_1 = I \alpha_1 A_1$$

لأنه يعبر عن شدة الإشعاع الساقط على الجسم

$$\therefore E_1 = I \alpha_1 \quad \ldots\ldots (1)$$

- If the body is replaced by another (of identical shape and equil. is again attained

الاختلاف في الجسم وليس شدة الإشعاع

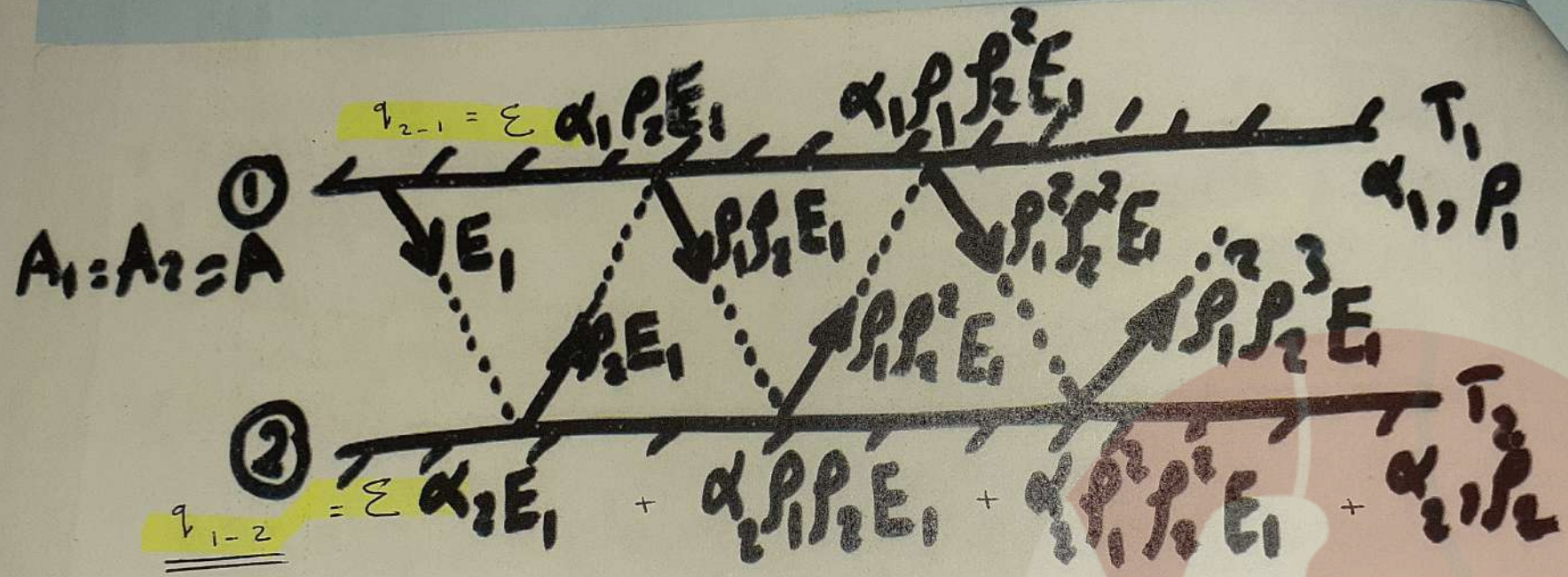
لأن I ثابتة

$$\therefore E_2 = I \alpha_2 \quad \ldots\ldots (2)$$

- Now, If a blackbody is placed instead of the later

$$\therefore E_b = I \quad \text{because } \alpha = 1 \quad \ldots\ldots (3)$$

Application



$\rho = 1 - \alpha$

- سقط على السطح 2 طاقة مقدارها E_1 ، امتص السطح 2 جزء من E_1 تساوي $(E_1 \alpha_2)$ ، وارتد ما تبقى من الطاقة بعد الامتصاص ، ما تبقى من الطاقة $= ((1-\alpha_2) E_1) = (\rho_2 E_1)$

- امتص السطح 1 ما ارتد من السطح 2 مقدار الطاقة الممتصة $(\alpha_1 \rho_2 E_1)$ ، بعدين ارتد ما تبقى من الطاقة $((1-\alpha_1) \rho_2 E_1) = (\rho_1 \rho_2 E_1)$ وهكذا ---

For Gray Surfaces with zero transmissivity ;

$$\rho = 1 - \alpha = 1 - \epsilon$$

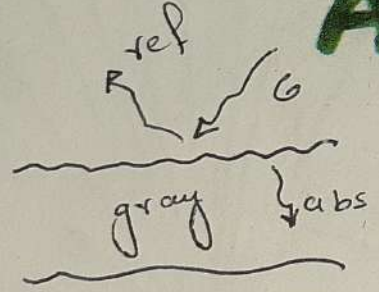
$\Rightarrow \cancel{1} - \alpha = \cancel{1} - \epsilon$

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

and

$$E = \epsilon \sigma T^4$$

$$\therefore \text{Net } \frac{q_{1-2}}{A} = \frac{\overset{\alpha_2}{\epsilon_2}\overset{E_1}{\epsilon_1 \sigma T_1^4} - \overset{\alpha_1}{\epsilon_1}\overset{E_2}{\epsilon_2 \sigma T_2^4}}{1 - (1-\epsilon_1)(1-\epsilon_2)}$$



$$= \frac{\epsilon_2 \epsilon_1 [\sigma (T_1^4 - T_2^4)]}{1 - 1 + \epsilon_2 + \epsilon_1 - \epsilon_1 \epsilon_2}$$

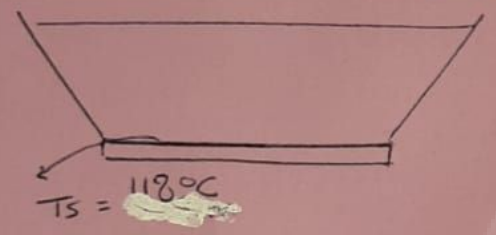
$$\text{Net } \frac{q_{1-2}}{A} = \frac{\sigma (T_1^4 - T_2^4)}{\frac{1}{\epsilon_1} + \frac{1}{\epsilon_2} - 1} \quad \text{III}$$

for parallel plate

* Example: The bottom of a copper pan, 0.3 m in diameter, is maintained at 118°C by an electric heater. Estimate the power required to boil water in this pan. what is the evaporation rate? Estimate the critical heat flux.

at 1atm

$T_{sat} = 100°C$ ⇒ لأنه مفتوح على atmosphere



من الشكل عرفت أنه Pool boil لأنه لا يحتوي على tubes or pipes

* لازم أحسب قيمة ΔT_e وأروح على الـ chart وأعرف المنطقة اللي رح أشتغل عليها

$$\Delta T_e = T_s - T_{sat} = 118 - 100 = 18°C$$

من الـ chart عرفت اني بمنطقة Nucleate boiling

$\Delta T_e = 5 \rightarrow 10$ ⇒ Bubbles collapse

$\Delta T_e = 10 \rightarrow 30$ ⇒ Bubbles rise to the free surface

هي المنطقة المذكورة في السؤال

* بختار الـ correlation المناسبة ومنها بحسب q ، ثم من $m = \frac{q}{h_{fg}}$

بعدين من معادلة q_{max} بحسب المطلوب بالأخير

1) $$q'' = \mu_l h_{fg} \left[\frac{g(\rho_l - \rho_v)}{\sigma}\right]^{1/2} \left[\frac{C_{p,l}(T_s - T_{sat})}{C_{s,f}\, h_{fg}\, Pr_l^n}\right]^3$$

heat flux ⇒ heat لأنه بعطيني بضربها بـ A عشان

$$A = \frac{\pi}{4} D^2$$

* All properties at T_{sat}

1// $\Delta T_e = 118 - 100 = 18\ ^\circ C$ so Nucleate boiling

all properties at 100°c (373 k)

$\mu_l = 279 \times 10^{-6}\ N \cdot s / m^2$

$h_{fg} = 2257\ kJ/kg$

$\rho_l = \frac{1}{v} = \frac{1}{1.044 \times 10^{-3}} = 957.85\ kg/m^3$

$\rho_v = \frac{1}{v} = \frac{1}{1.679} = 0.5956\ kg/m^3$

$\sigma = 58.9 \times 10^{-3}\ \frac{N}{m}$

$C_{p_l} = 4.217\ kJ/kg \cdot k$

$P_r = 1.76$

& from table find $C_{s,f} = .0128$ & $n = 1$

$$q = 279 \times 10^{-6} \times 2257 \left(\frac{9.8\,(957.85 - .5956)}{58.9 \times 10^{-3}} \right)^{1/2} \left(\frac{4.217\,(118-100)}{.0128 \times 2257 \times 1.76} \right)^3$$

$q'' = 836.12\ \frac{kW}{m^2}$

$\Rightarrow$ power $= q'' A = q'' \frac{\pi}{4} D^2 = 836.12\ \frac{kW}{m^2} \times \frac{\pi}{4} (.3)^2\ m^2$

$\Rightarrow$ power $= 59.1\ kW$

2// $q = m\, h_{fg}$

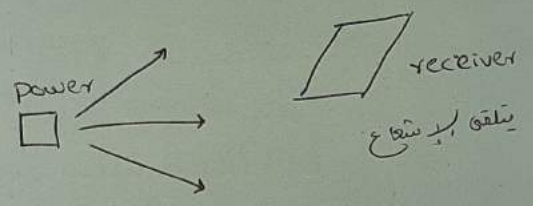
$\Rightarrow m = \frac{q}{h_{fg}} = 59.1\ \frac{kJ}{s} \cdot \frac{kg}{2257\ kJ} = .026\ \frac{kg}{s}$

3// Critical heat flux

$$q_{max} = \frac{\pi}{24} h_{fg} \rho_v \left(\frac{\sigma g (\rho_l - \rho_v)}{\rho_v^2} \right)^{1/4} \left(\frac{\rho_l + \rho_v}{\rho_l} \right)^{1/2}$$

كل ال prop محسوبة تعويض مباشر

View Factor Concept



view factor ⇒ كيف تنظر لمكان معين

size of surfaces Orientation يعتمد على

Introduction

- Radiation Between surfaces are strongly affected by :
  a. Orientation
  b. size of the surfaces relative to each other.
- In order to formulate the effects of orientation in the analysis of radiation heat exchange among surfaces, the concept of view factor must be considered.

- View Factor ~ Shape Factor ~ angle Factor ~ Configuration Factor have been used in literature.

View Factor

يعتمد على شكل السطح

رح نهتم بهاد الموضوع فقط

Diffuse View Factor
Surfaces are diffuse reflectors and diffuse emitters

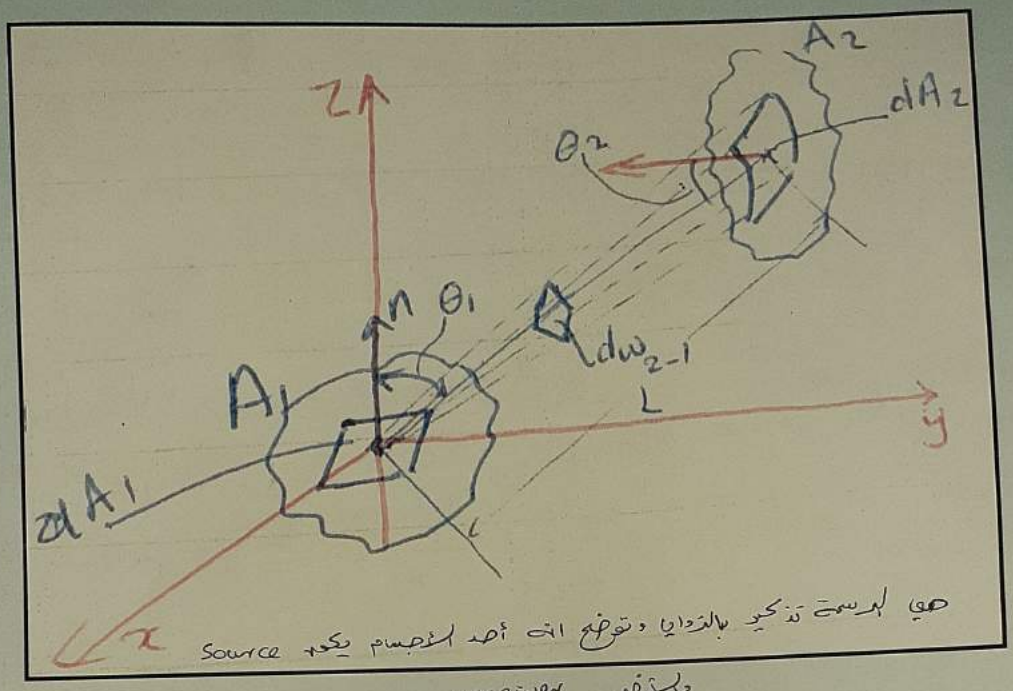
Specular View Factor
Surfaces are diffuse emitters and specular reflectors

view factor ⇒ fraction ⇒ مقدار الطاقة التي وصلت جسم معين / كل الطاقة المنبعثة من المصدر ⇒ حسب ال Orientation

- We are going to consider only the surfaces of diffuse emitters and diffuse reflectors. (Diffuse view factor) Thus, we simply use the term view factor, and it will imply the diffuse view factor.
- The physical significance of the view factor:

The view factor between two surfaces represents the fraction of the radiative energy leaving one surface that strikes the other surface directly.

View factor definition



هي الرسمة تذكير بالزوايا وتوضح انه أحد الأجسام يكون Source والثاني receiver

View Factor Derivation

$$dq_{1-2} = I \cos\theta_1 \, dA_1 \, d\omega_{2-1} \quad \text{.....(1)}$$

from one to two

solid angle $d\omega_{2-1} = \dfrac{\cos\theta_2 \, dA_2}{L^2}$ .....(2)

distance ⇒ عكس تكون عبارة عن radius & length حسب المسافة بين الجسمين

But for black surface

$$I = \frac{E}{\pi} = \frac{\sigma T_1^4}{\pi} \quad \text{.....(3)}$$

Substituting eq. 2 and 3 in 1, we obtain

$$dq_{1-2} = \frac{\sigma T_1^4}{\pi} \frac{\cos\theta_1 \cos\theta_2 \, dA_1 \, dA_2}{L^2} \quad \text{.....(4)}$$

- Following the same procedure, we can write

q_{1-2} ← لفرق بين T_2^4 موجبنا بالتعاقب

$$dq_{2-1} = \frac{\sigma T_2^4}{\pi} \frac{\cos\theta_1 \cos\theta_2 \, dA_1 dA_2}{L^2} \quad \text{.....(5)}$$

- The net of radiation between dA_1 and dA_2 is

$$dq_{1\to 2} = dq_{1-2} - dq_{2-1} \quad \text{.....(6)}$$

$$= \sigma(T_1^4 - T_2^4) \frac{\cos\theta_1 \cos\theta_2 \, dA_1 dA_2}{\pi L^2} \quad \text{.....(7)}$$

on a per unit area basis

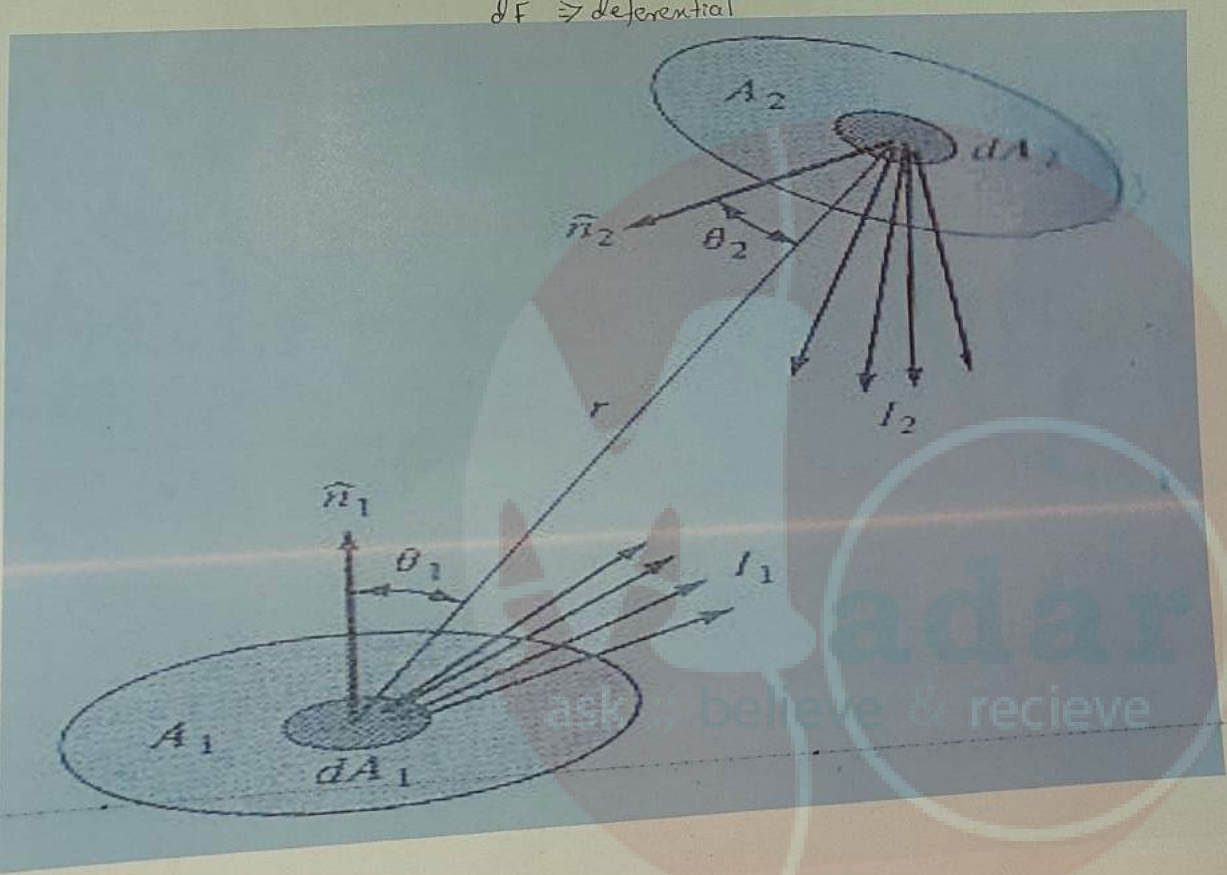
View factor, F_A

$$\frac{dq_{1\to 2}}{dA_1} = \sigma(T_1^4 - T_2^4) \frac{\cos\theta_1 \cos\theta_2 \, dA_2}{\pi L^2} \quad \text{.....(8)}$$

وكونه من زوايا و area و length

The elemental View Factor

dF ⇒ deferential



Definition: The elemental View Factor

The *elemental view factor* $dF_{dA_1-dA_2}$, by definition, is the ratio of the radiative energy leaving dA_1 that strikes dA_2 directly to the radiative energy leaving dA_1 in all directions into the hemispherical space. It is given by

$$dF_{dA_1-dA_2} = \frac{\cos\theta_1 \cos\theta_2 \, dA_2}{\pi r^2}$$

The elemental view factor $dF_{dA_2-dA_1}$ from dA_2 to dA_1 is now immediately obtained from previous eq. by interchanging subscripts 1 and 2. We find

$$dF_{dA_2-dA_1} = \frac{\cos\theta_1 \cos\theta_2 \, dA_1}{\pi r^2}$$

Reciprocity Relation

- The reciprocity relation between the view factors $dF_{dA1-dA2}$ and $dF_{dA2-dA1}$ of the previous two Eqs. Is :

$$dA_1 dF_{dA1-dA2} = dA_2 \, dF_{dA2-dA1}$$

area ل view عن طريق ل يعكس factor

- The view factors between surfaces A_i and A_j of the enclosure obey the following relation:

$$A_i F_{Ai-Aj} = A_j F_{Aj-Ai}$$

- Summation relation

$$\sum_{k=1}^{N} F_{A_i - A_k} = 1$$

مجموع $F_{a \to k}$ عبارة عن fraction

Note

$F_{A_i - A_i} = 0$ if A_i plane or convex (مستوي) → الجسم لا يرى إشعاع من نفسه

$F_{A_i - A_i} \neq 0$ if A_i concave ⇒ مقعر → الجسم يرى إشعاع من نفسه

*كيف بدي أحسب ال view - factor ؟ by correlation or chart

View-Factor Determination

- Equations were derived for simple shapes. These expressions given in tables. لا يتوقع بال correlation
- Graphs for simple configurations.
- Note: View factor Algebra

$F_{i-j} = F_{i-1} + F_{i-2} \ldots F_{i-n}$

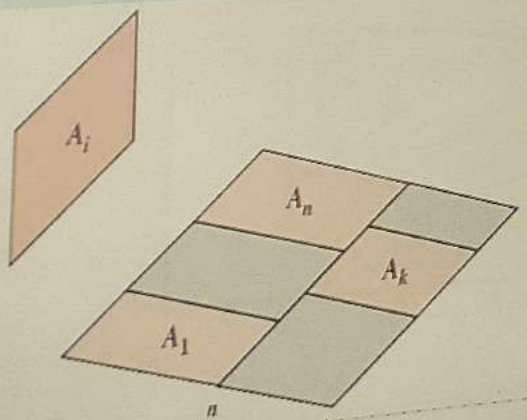
$A_i F_{i-j} = A_i F_{i-1} + A_i F_{i-2} \ldots A_i F_{i-n}$

Use reciprocity relation

$A_j F_{j-i} = A_1 F_{1-i} + A_2 F_{2-i} + \ldots + A_n F_{n-i}$

$Fj-i = [A1\ F1-i + A2\ F2-i + \ldots + An\ Fn-i]/A_j$

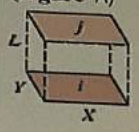
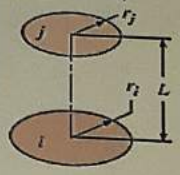
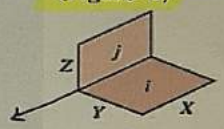
$Fj-i = [A1F1-i + A2\ F2-i + \ldots + An\ Fn-i]/[A1+A2+\ldots+An]$



$A = \sum^{n} A_k$

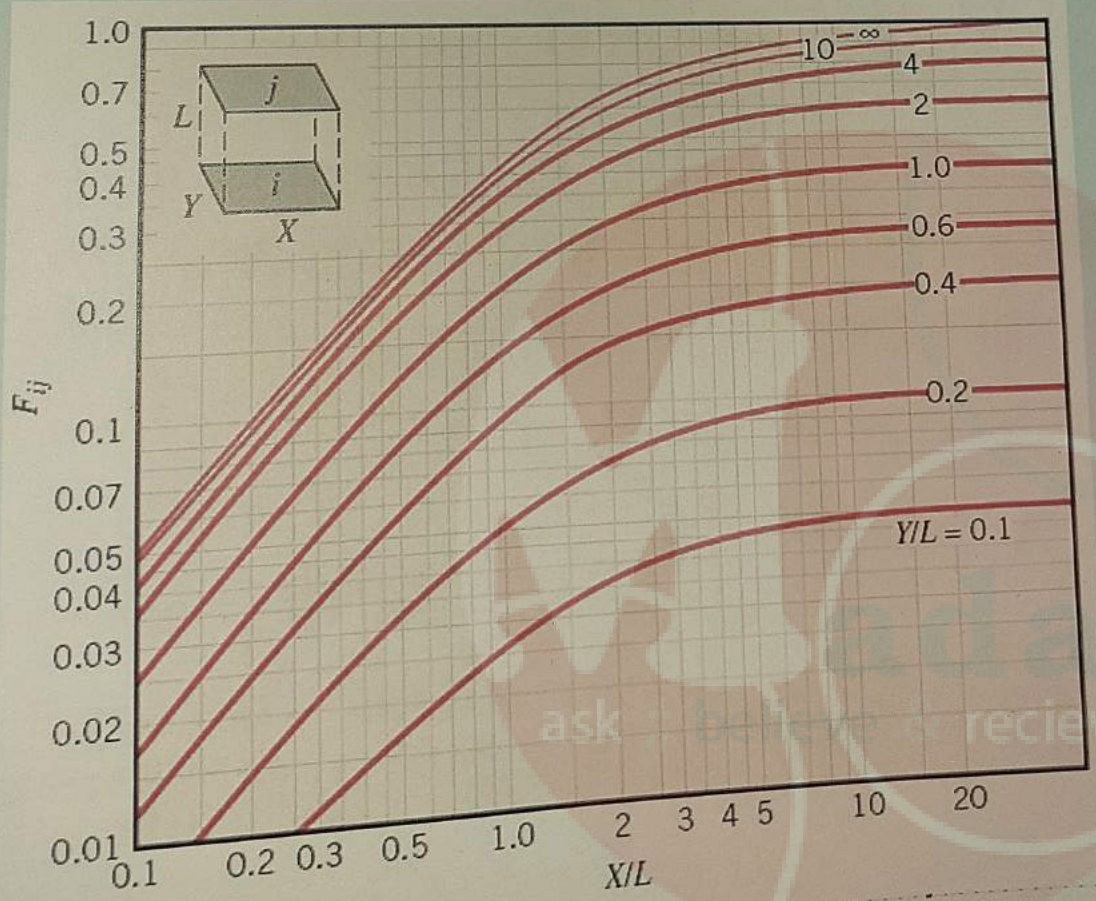
Areas used to illustrate view factor relations.

View Factors for Three-Dimensional Geometries

Geometry	Relation
Aligned Parallel Rectangles (Figure A) 	$\overline{X} = X/L, \overline{Y} = Y/L$ $F_{ij} = \frac{2}{\pi \overline{X}\,\overline{Y}}\left\{\ln\left[\frac{(1+\overline{X}^2)(1+\overline{Y}^2)}{1+\overline{X}^2+\overline{Y}^2}\right]^{1/2} + \overline{X}(1+\overline{Y}^2)^{1/2}\tan^{-1}\frac{\overline{X}}{(1+\overline{Y}^2)^{1/2}} + \overline{Y}(1+\overline{X}^2)^{1/2}\tan^{-1}\frac{\overline{Y}}{(1+\overline{X}^2)^{1/2}} - \overline{X}\tan^{-1}\overline{X} - \overline{Y}\tan^{-1}\overline{Y}\right\}$
Coaxial Parallel Disks (Figure B) 	$R_i = r_i/L, R_j = r_j/L$ $S = 1 + \frac{1+R_j^2}{R_i^2}$ $F_{ij} = \frac{1}{2}\{S - [S^2 - 4(r_j/r_i)^2]^{1/2}\}$
Perpendicular Rectangles with a Common Edge (Figure C) 	$H = Z/X, W = Y/X$ $F_{ij} = \frac{1}{\pi W}\left(W\tan^{-1}\frac{1}{W} + H\tan^{-1}\frac{1}{H} - (H^2+W^2)^{1/2}\tan^{-1}\frac{1}{(H^2+W^2)^{1/2}} + \frac{1}{4}\ln\left\{\frac{(1+W^2)(1+H^2)}{1+W^2+H^2}\left[\frac{W^2(1+W^2+H^2)}{(1+W^2)(W^2+H^2)}\right]^{W^2} \times \left[\frac{H^2(1+H^2+W^2)}{(1+H^2)(H^2+W^2)}\right]^{H^2}\right\}\right)$

Common edge
مهم جداً

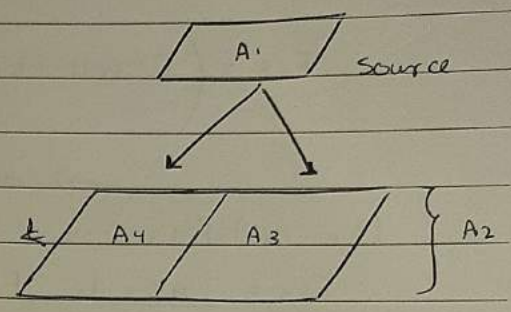
استخدم الـ chart أفضل



* view vactor :

if F_{12} & F_{14} given find F_{31} ؟!

من السطح 1 إلى 2 من السطح 1 إلى 4



$F_{12} = F_{13} + F_{14}$

$F_{13} = F_{12} - F_{14}$

$A_3 F_{31} = A_1 F_{13}$

$\Rightarrow F_{31} = \frac{A_1}{A_3}(F_{12} - F_{14})$ given #

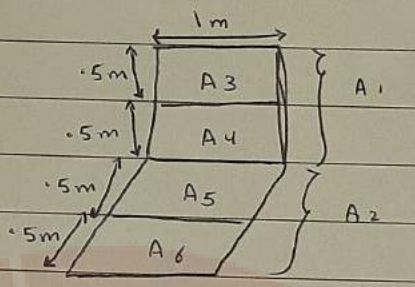
$(A_3 + A_4) F_{(3+4)-1} = A_3 F_{31} + A_4 F_{41}$ ⇒ إذا فرضت أن الـ source هو A2

* Example: find F_{63} ؟!

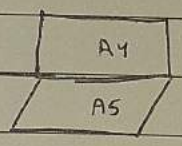
كتب كل قيم FAB اللي بقدر أجيبهم من الـ chart

* شرط استخدام الـ chart يكون بين سطح وسطح لا فرق

اللي بدي أحسبهم قيمة FAB & خط مشترك مش



هذا الخط المشترك. (common)



F_{15} or $F_{(3+4)-5}$

F_{45}

F_{42} or $F_{4-(5+6)}$

F_{12} or $F_{(3+4)-(5+6)}$

بقدر أطلعهم بسهولة من الـ chart & بينهم خط مشترك (common)

هاد وكمان أكتبه $F_{36} + F_{46}$ وبعدين بضرب بـ area حتى أحصل على F_{63}

$$F_{(3+4)-(5+6)} = F_{(3+4)-5} + F_{(3+4)-6}$$

$$= F_{(3+4)-5} + \frac{A_6}{A_3+A_4}\left(F_{63} + F_{64}\right)$$

هاد المطلوب

$$F_{63} = \left(F_{(3+4)-(5+6)} - F_{(3+4)-5}\right)\frac{A_3+A_4}{A_6} - F_{6-4} \quad \text{---} \; 1$$

* طريقة F_{6-4} نفسها F_{64} لكن غير معروفة

$$(A_5+A_6)\,F_{(5+6)-4} = A_5 F_{54} + A_6 F_{64}$$

$$F_{64} = \frac{A_5+A_6}{A_6} F_{(5+6)-4} - \frac{A_5}{A_6} F_{54}$$

بعوضها بالمعادلة 1

وحسب قيمة F_{63}

مثال ثاني:

$F_{(3+4)-5}$