CBCS SCHEME

USN

15ME63

Sixth Semester B.E. Degree Examination, June/July 2019
Heat Transfer

Time: 3 hrs. Max. Marks: 80

Note: ***1. Answer any FIVE full questions, choosing ONE full question from each module.***
2. Use of heat transfer data hand book and steam tables are permitted.

Important Note : 1. On completing your answers, compulsorily draw diagonal cross lines on the remaining blank pages.
2. Any revealing of identification, appeal to evaluator and /or equations written eg, 42+8 = 50, will be treated as malpractice.

Module-1

1 a. State the laws governing three basic modes of heat transfer. (06 Marks)

b. Derive the general three-dimensional conduction equation in Cartesian coordinates and state the assumptions made. (10 Marks)

OR

2 a. Derive an expression for the temperature distribution through the plane wall with uniform thermal conductivity. (06 Marks)

b. A metal [K = 45 W/m°C] steam pipe of 5 cm inside diameter and 6.5 cm outside diameter is lagged with 2.75 cm thickness of high temperature high insulation having thermal conductivity 1.1 W/m°C. convective heat transfer coefficients on the inside and outside surfaces are 4650 W/m^2K and 11.5 W/m^2K respectively. If the steam temperature is 200°c and the ambient temperature is 25°C. Calculate:

i) Heat loss per metre length of pipe
ii) Temperature at the interfaces
iii) Overall heat transfer coefficient to inside and outside surfaces. (10 Marks)

Module-2

3 a. Derive an expression for critical thickness of insulation for a cylinder. (06 Marks)

b. The handle of a ladle used for pouring molten metal at 327°C is 30 cm long and is made of 2.5 cm × 1.5 cm mild steel bar stock [K = 43 W/mK]. In order to reduce grip temperature, it is proposed to make a hallow handle of mild steel plate 0.15 cm thick to the same rectangular shape. If the surface heat transfer coefficient is 14.5 W/m^2K and the ambient temperature is 27°C, estimate the reduction in the temperature of grip. Neglect the heat transfer from inner surface of the hallow shape. (10 Marks)

OR

4 a. What is lumped system analysis? Derive the temperature variation using lumped parameter analysis. (06 Marks)

b. An iron sphere of diameter 5 cm is initially at a uniform temperature of 225°C. It is suddenly exposed to an ambient at 25°C with convection coefficient of 500 W/m^2K.

i) Calculate the centre temperature 2 minute after the start of exposure.
ii) Calculate the temperature at a depth of 1 cm from the surface after 2 minute of exposure.
iii) Calculate the energy removed from the sphere during this period.

Take thermo physical properties of iron sphere K = 60 W/mK, ρ = 7850 kg/m^3, C = 460 J/kg, $\alpha = 1.6 \times 10^{-5}$ m^2/s. (10 Marks)

Module-3

5 a. Explain the three types of boundary conditions are applied in finite difference representations. (06 Marks)

b. Derive the relation between normal intensity and emissive power. (10 Marks)

OR

6 a. Explain:

i) Stefan Boltzman law

ii) Kirchoff's law

iii) Planks law (06 Marks)

b. Two large parallel plates with $\in = 0.5$ each, are maintained at different temperatures and are exchanging heat only by radiation. Two equally large radiations shields with surface emissivity 0.05 are introduced in parallel to the plates. Find the percentage reduction in net radiative heat transfer. (10 Marks)

Module-4

7 a. Explain the physical significance of:

(i) Prandtl number (ii) Reynolds number (iii) Nusselt number (06 Marks)

b. Air at 1 atm pressure and temperature 25°C flowing with a velocity 50 m/s crosses an industrial heater made of long solid rod of diameter 20 mm. The surface temperature of the heater is 457°C. Determine the allowable electrical power density (W/m^3) within the heater per meter length. (10 Marks)

OR

8 a. A circular plate of 25 cm diameter with both surfaces maintained at a uniform temperature of 100°C is suspended horizontally in atmospheric air at 20°C. Determine the heat transfer from the plate. (10 Marks)

b. Obtain the fundamental relationship between Nusselt, Prandtle and Reynolds number using Buckingham's π - theorem for forced convection heat transfer. (06 Marks)

Module-5

9 a. Derive an expression for LMTD for a parallel flow heat exchanger. (06 Marks)

b. A refrigerator is designed to cool 250 kg/hr of hot fluid of specific heat 3350 J/kg°C at 120°C using a parallel arrangement 1000 kg/hr of cooling water is available for cooling purposes at a temperature of 10°C. If the overall heat transfer coefficient is 1160 $W/m^2{}°C$ and the surface area of the heat exchanger is $0.25\ m^2$. Calculate the outlet temperature of the cooled liquid and water and also the effectiveness of the heat exchanger and rate of heat transfer. (10 Marks)

OR

10 a. Sketch and explain boiling curve. (06 Marks)

b. The outer surface of a vertical tube 80 mm in outer diameter and 1m long is exposed to saturated steam at atmospheric pressure. The tube surface is maintained at 50°C by flow of water through the tube. What is the rate of heat transfer to coolant and what is the rate of condensation of steam? (10 Marks)

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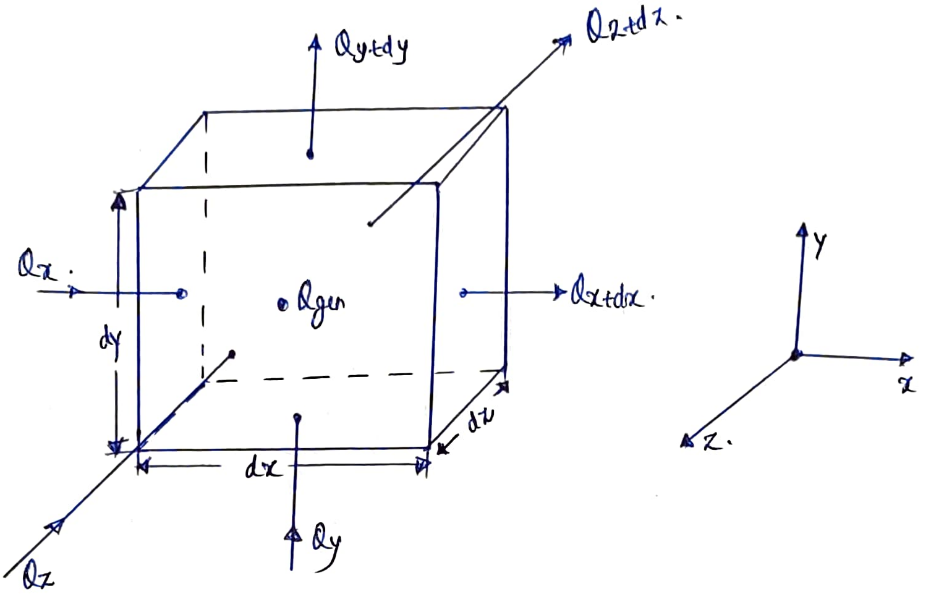
Department of Mechanical Engineering

Subject with code: Heat transfer (15ME63)   June/July – 2019.

Q.No.	Solution and Scheme	Marks
1 a)	The three modes of heat transfer are 1) Conduction: Fourier Law of heat conduction. It states that the rate of heat flow by conduction in any medium in any direction is proportional to the area normal to the direction of heat flow and temperature gradient in that direction. $Q_x = -KA\, dT/dx$	02
	2) Convection: Newton's law of cooling: It states that the rate of convection heat transfer is proportional to the surface area and temperature difference. ie $Q = h A_s (T_s - T_\infty)$	02
	3) Radiation: Stefan Boltzmann's law of radiation. It states that the emissive power of a blackbody is proportional to the fourth power of absolute temperature of the body. if E_b is the emissive power of a blackbody at temperature T °K, then $E_b = \sigma T^4$.	02

Q.No.	Solution and Scheme	Marks
1b.	Three dimensional conduction equation in the cartesian coordinate system.	



(02)

Assumptions:

1) Heat transfer takes place in x, y and z direction.
2) Unsteady state heat transfer.
3) Anisotropic and homogenous material. (02)
4) With heat generation term (Q_{gen})
5) Neglecting the convection and radiation heat losses.

Let us consider a volume element of dimensions dx, dy and dz in x, y and z directions. The conduction heat transfer across the six surfaces of the element is shown in above figure.

Applying I law of thermodynamics for the volume element we have

or.

Q.No.	Solution and Scheme	Marks

or

Energy balance

Net heat accumulated in the element due to conduction of heat from all the coordinate directions (A) + heat generated within the element (B) = Energy stored in the element (C).

ie (A) + (B) = (C)

(A) ie Net heat accumulated in the element due to conduction of heat from all the coordinate system = $Q_{in} - Q_{out}$

Considering, in only x direction

$$= Q_x - Q_{x+dx}$$

$$= \cancel{Q_x} - \left[\cancel{Q_x} + \frac{\partial Q_x}{\partial x} dx\right]$$

$$= -\frac{\partial Q_x}{\partial x} dx$$

$\left[Q_x = -KA\frac{\partial T}{\partial x}\right.$

$\left.A = dy\,dz\right]$

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

$$= \frac{d}{dx}\left(K\frac{dT}{dx}\right)dx\,dy\,dz$$

Similarly in

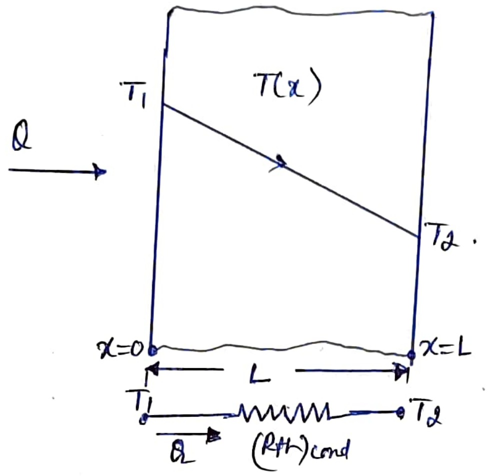
y direction $= \frac{d}{dy}\left(K\frac{dT}{dy}\right)dx\,dy\,dz$ — (04)

z direction $= \frac{d}{dz}\left(K\frac{dT}{dz}\right)dx\,dy\,dz$

Q.No.	Solution and Scheme	Marks
	(B) ie Q_g = Energy generated per unit volume $= Q_g\,dx\,dy\,dz$.	
	(C) ie Energy stored in the element $= \frac{dE}{dt}$ $\left[E = M C_p\, dT = \rho C_p dT\, dx\,dy\,dz \right)$ $= \rho C_p \frac{dT}{dt}\,dx\,dy\,dz$.	
	Hence (A) + (B) = (C)	
	$\frac{d}{dx}\left(K\frac{dT}{dx}\right)dx\,dy\,dz + \frac{d}{dy}\left(K\frac{dT}{dy}\right)dx\,dy\,dz + \frac{d}{dz}\left(K\frac{dT}{dz}\right)dx\,dy\,dz + Q_g\,dx\,dy\,dz = \rho C_p \frac{dT}{dt}\,dx\,dy\,dz$.	
	Dividing throughout by $dx\,dy\,dz$.	02
	$\boxed{\frac{d}{dx}\left(K\frac{dT}{dx}\right) + \frac{d}{dy}\left(K\frac{dT}{dy}\right) + \frac{d}{dz}\left(K\frac{dT}{dz}\right) + Q_g = \rho C_p \frac{dT}{dt}}$	
	OR.	
2 a)	Temperature distribution through the plane wall with uniform thermal conductivity.	

Q.No.	Solution and Scheme	Marks

Consider a plane wall of homogenous material of width 'L' through which heat is flowing only in x direction.



(02)

The general conduction equation for the cartesian coordinates is

$$\frac{d^2T}{dx^2} + \frac{d^2T}{dy^2} + \frac{d^2T}{dz^2} + \frac{Q_g}{K} = \frac{1}{\alpha}\frac{dT}{dt} \quad \text{—(1)}$$

Assumptions:

1) 1D ie $T = T(x)$, $y=0$, $z=0$.
2) Without heat generation, $(Q_g = 0)$
3) Isotropic material. $(K = K_x = K_y = K_z)$ (02)
4) Steady heat transfer $\left(\frac{dT}{dt} = 0\right)$

After simplifying wrt above assumptions, eqn(1) reduces to

$$\frac{d^2T}{dx^2} = 0$$

Integrating above eqn. wrt x

$$\frac{dT}{dx} = C_1, \qquad T(x) = C_1 x + C_2 \quad \text{—(2)}$$

Q.No.	Solution and Scheme	Marks

Boundary conditions:

At $x=0$, $T(x)=T_1$, $x=L$, $T(x)=T_2$.

At $x=0$, $T(x)=T_1$

$T_1 = C_1 \times 0 + C_2$, $\boxed{C_2 = T_1}$

At $x=L$, $T(x)=T_2$,

$T_2 = C_1 L + T_1$, $\boxed{C_1 = \frac{T_2 - T_1}{L}}$

From (2) we can write.

$$\boxed{T(x) = \left(\frac{T_2 - T_1}{L}\right)x + T_1} \quad -(3)$$ 02

Eqn (3) gives the temperature distribution in the plane wall.

2(b) Given data:

$r_1 = 0.025m$.

$r_2 = 0.0325m$.

$r_3 = r_2 + 2.75cm$.

$= 0.06m$

$K_1 = 45 W/m°C$

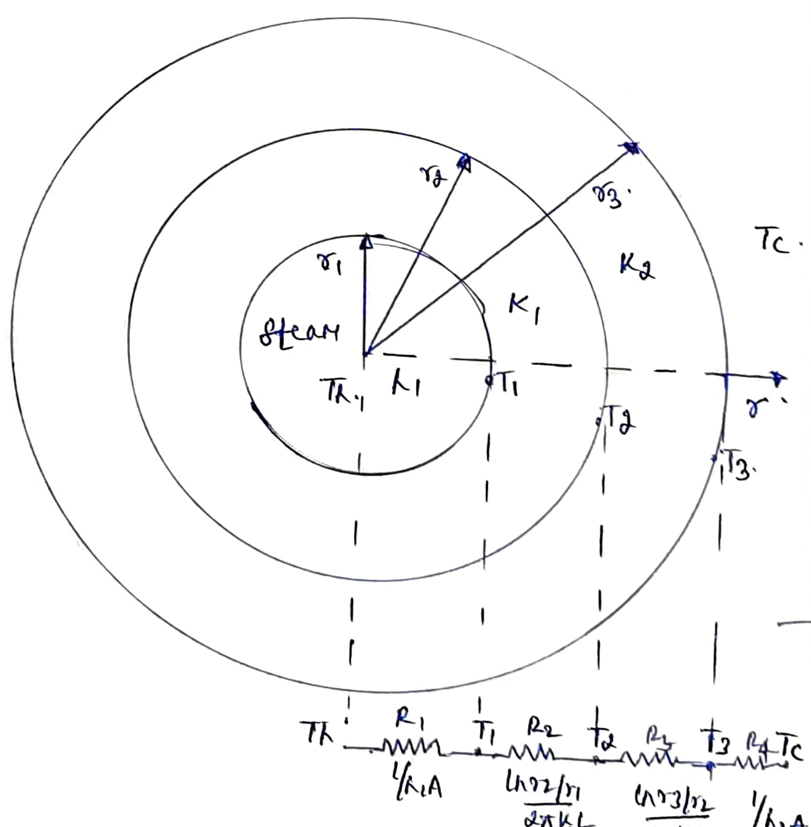
$K_2 = 11 W/m°C$.

$h_1 = 4650 W/m^2K$

$h_2 = 11.5 W/m^2K$.

$T_h = 200°C$

$T_c = 25°C$.



02

Q.No.	Solution and Scheme	Marks

i) Heat loss per metre length of pipe.

$$Q = \frac{\Delta T}{R_{th}} = \frac{T_h - T_c}{R_{th}}.$$

$$R_{th} = R_1 + R_2 + R_3 + R_4$$

$$= \frac{1}{h_1 A} + \frac{\ln r_2/r_1}{2\pi K_1 L} + \frac{\ln r_3/r_2}{2\pi K_2 L} + \frac{1}{h_2 A}$$

$$R_{th} = \frac{1}{h_1 \cdot 2\pi r_1} + \frac{\ln r_2/r_1}{2\pi K_1} + \frac{\ln r_3/r_2}{2\pi K_2} + \frac{1}{h_2\, 2\pi r_3}.$$

$$= 1.369\times10^{-3} + 9.28\times10^{-4} + 8.86\times10^{-3} + 0.230$$

$$R_{th} = 0.241157 \ {}^\circ C/W m$$

$$Q = \frac{T_h - T_c}{R_{th}} = \frac{200-25}{0.241157}$$

$$\boxed{Q = 725.668 \ W/m.}$$ (04)

ii) Temperature at the interface (T_1, T_2, T_3).

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$$Q = \frac{T_h - T_1}{R_1}$$

$$T_1 = T_h - QR_1$$

$$T_1 = 199\ {}^\circ C$$

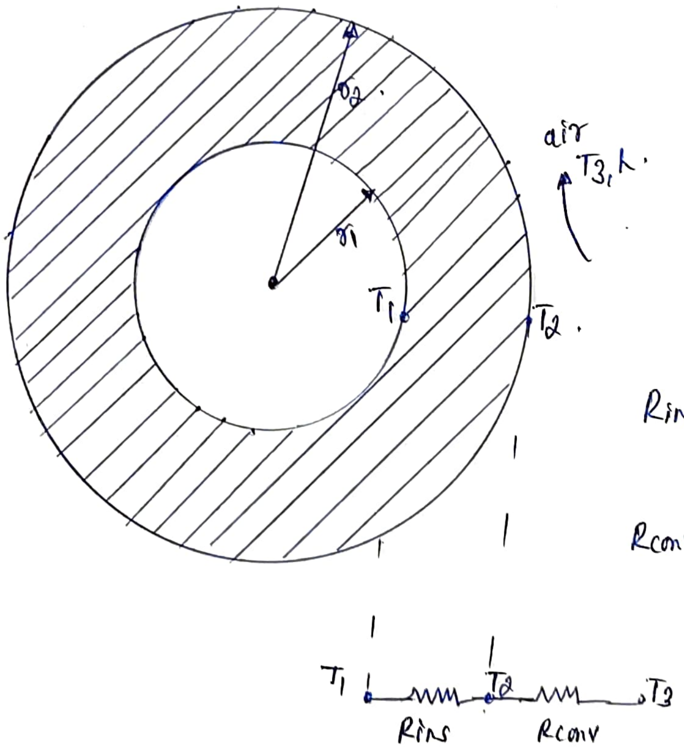
$$Q = \frac{T_1 - T_2}{R_2}.$$

$$T_2 = T_1 - QR_2.$$

$$= 198\ {}^\circ C$$ (02)

$$T_3 = QR_4 + T_c.$$

$$= 191\ {}^\circ C$$

Q.No.	Solution and Scheme	Marks
	iii) Overall heat transfer coefficient to inside and outside surfaces (U)	
	W.K.T $U = \frac{1}{R_{th}}$	
	$= \frac{1}{0.241157}$	
	$U = 4.146\ W/m^2\ ^\circ C$	02
	Module-02.	
3 a)	Critical thickness of insulation for cylinder	
		02
	$R_{ins} = \frac{\ln r_2/r_1}{2\pi KL}$	
	$R_{conv} = \frac{1}{h 2\pi r_2 L}$	
	Consider a cylinder of radius r_1 maintained at uniform temperature 'T_1' covered with a layer of insulation of radius r_2. $(r_2 - r_1)$ is the thickness of insulation. Let T_3 be the temperature.	

Q.No.	Solution and Scheme	Marks

of ambient air with heat transfer coefficient 'h'

The rate of heat transfer is

$$Q = \frac{T_1 - T_3}{R_{in} + R_{con}} = \frac{T_1 - T_3}{\frac{\ln r_2/r_1}{2\pi K L} + \frac{1}{h 2\pi r_2 L}}$$ — (02)

For the critical radius, the R_{total} is minimum.

Mathematically ie for R_{total} is to be minimum.

$$\frac{dR_{total}}{dr_2} = 0, \quad = \frac{d}{dr_2}\left[\frac{\ln r_2/r_1}{2\pi K L} + \frac{1}{h 2\pi r_2 L}\right] = 0$$

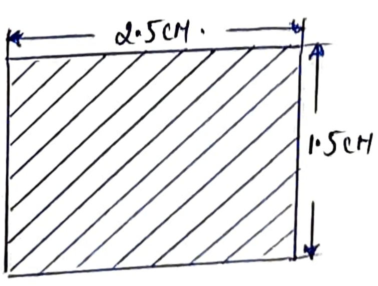
$$\frac{1}{K}\,\frac{1}{r_2\, 2\pi L} + \frac{1}{h 2\pi L}\left(\frac{-1}{r_2^2}\right) = 0$$

$$\frac{1}{2\pi K L r_2} = \frac{1}{2\pi K h}\left(\frac{1}{r_2^2}\right)$$

$$\boxed{r_2 = \frac{K}{h} = r_c.}$$ — (3) — (02)

Eqn(3) represents the expression for critical thickness of insulation for cylinder.

3 b) a) When the handle is made of solid steel bar.



$h = 14.5\ W/m^2K$

$K = 43\ W/mK$

$T_b = 327 - 27$

$= 300°C.$

Q.No.	Solution and Scheme	Marks

Area of cross section of the bar.

$$A = 2.5 \times 1.5 \times 10^{-4} = 3.75 \times 10^{-4} m^2$$

Perimeter of the bar

$$P = 2\{2.5 + 1.5) \times 10^{-2} = 8 \times 10^{-2} m.$$

Therefore $m = \sqrt{\frac{hP}{KA}} = \sqrt{\frac{14.5 \times 8 \times 10^{-2}}{43 \times 3.75 \times 10^{-4}}} = 8.48\ m^{-1}$ — (02)

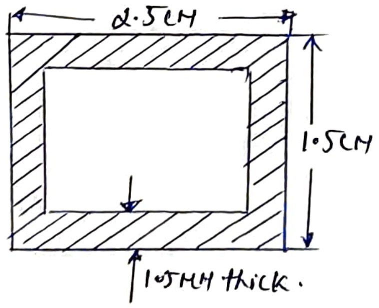
When the heat loss from the tip of the handle is neglected the temperature at any point along the length of the handle is given by

$$\frac{T(x) - Ta}{Tb - Ta} = \frac{\cosh m(L-x)}{\cosh mL}$$

$$\frac{T(x) - 27}{327 - 27} = \frac{1}{\cosh(8.48 \times 0.3)}$$

$$T(x)_{x=L} = 74^\circ C.$$ — (02)

b) When the handle is hollow made out of a sheet.



Area of the cross section of the fin is

$$A = [(2.5 \times 1.5) - (2.5 \times 0.3) \times (1.5 - 0.3)]$$

$$= 1.11 \times 10^{-4} m^2$$ — (02)

$$P = 2 \times \{2.5 + 1.5) = 8 \times 10^{-2} m.$$

Q.No.	Solution and Scheme	Marks

$$m = \sqrt{\frac{hP}{KA}} = \sqrt{\frac{14.5 \times 8 \times 10^{-2}}{43 \times 1.11 \times 10^{-4}}}$$

$$m = 15.59\ m^{-1}$$ (02)

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$$\frac{T(x) - T_a}{T_b - T_a} = \frac{1}{\cosh mL}$$

$$\frac{T(x) - 27}{327 - 27} = \frac{1}{\cosh(15.59 \times 0.3)}$$

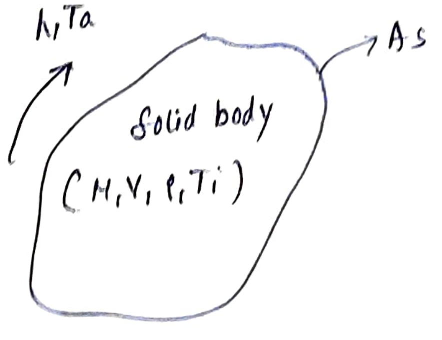
$$T(x)_{x=L} = 32.57^\circ C.$$

Reduction in grip temperature $= 74 - 30.57$
$= 41.43^\circ C.$ (02)

OR.

4 a) In the lumped system analysis, the internal conduction resistance of the body to heat flow is negligible compared to the convective resistance at the surface. (02)

Derivations:



Q.No.	Solution and Scheme	Marks
	Consider a solid body of arbitrary shape, volume V, mass M, density ρ, surface area A_s and specific heat C_p. To start with at $t=0$ let the temperature throughout the body be uniform at $T=T_i$. At the instant $t=0$, let the body be suddenly placed in a medium at a temperature of T_a. Let us assume that $T_a > T_i$. Then heat will be transferred from the medium to the body and the temperature of the body will increase with time. Let the temperature of the body rise by differential amount dT in a differential time interval dt, thus increasing the internal energy of the solid. Making the energy balance ie Amount of heat transferred into the body in time interval 'dt' (ie convection) = Increase in internal energy of the body in time interval 'dt' $h A_s (T_a - T(t)) = M C_p \frac{dT}{dt}$ — (1) $\rho = \frac{M}{V}$, $M = \rho V$ $dT = d(T(t) - T_a)$ $h A_s (T_a - T(t)) = \rho V C_p \, d(T(t) - T_a)$ $\frac{d(T(t) - T_a)}{T(t) - T_a} = \frac{-h A_s}{\rho V C_p} dt$ Integrating between $t=0$ (ie $T=T_i$) and any t (ie $T=T(t)$)	02

Q.No.	Solution and Scheme	Marks
	$\ln\left[\frac{T(t)-T_a}{T_i-T_a}\right] = -\frac{hA_st}{\rho C_p V}$	
	$\frac{T(t)-T_a}{T_i-T_a} = e^{-\left(\frac{hA_st}{\rho C_p V}\right)}$ —— (2)	02
	Let, $\frac{\rho C_p V}{h \cdot A_s} = \tau$	
	Where 'τ' is known as time constant &	
	Eqn (2) represents the temperature distribution for the lumped system.	
4 b)	Given data:	
	$T_i = 225°C$, $T_\infty = 25°C$, $h = 500\, W/m^2K$,	
	$k = 60\, W/mK$.	
	Biot number (Bi)	
	$Bi = \frac{hR_0}{K} = \frac{500 \times 2.5 \times 10^{-2}}{60}$	
	$= 0.208 > 0.1$	02
	Lumped analysis is not applicable.	
	Solution is obtained from Heisler Chart	
	$Fo = \frac{\alpha t}{R_0^2} = \frac{1.6 \times 10^{-5} \times 120}{(2.5 \times 10^{-2})^2} = 3.072$	
	i) Centre temperature (T_0):	
	Data. handbook, pg No: 72, for sphere., chart corresponding to $Bi = 0.208$, $Fo = 3.072$	

Q.No.	Solution and Scheme	Marks
	We have	
	$$\frac{T_0 - T_\infty}{T_i - T_\infty} = 0.18$$	
	$$\frac{T_0 - 25}{225 - 25} = 0.18$$	02
	$$T_0 = 61^\circ C$$	
	ii) The temperature at the depth 1cm from surface	
	$$\frac{r}{R_0} = 0.6.$$	
	Data handbook, pg.no: 73 for sphere, temp. at any position, corresponding to $Bi = 0.208$, $\frac{r}{R_0} = 0.6.$	
	We have	
	$$\frac{T(r/R) - 25}{61 - 25} = 0.95$$	02
	$$T(r/R) = 59.2^\circ C$$	
	iii) Energy removed (Q)	
	$$Bi^2 Fo = (0.208)^2 \times (3.072) = 0.133$$	
	Data handbook, pg. no: 74, for sphere, heat flow corresponding to $Bi^2 Fo = 0.133$ and $Bi = 0.208$	
	We have $$\frac{Q}{Q_0} = 0.75$$	02
	But Q_0 for sphere, pg. no: 65	
	$$Q_0 = 4/3\, \pi R_0^3 \times \rho \times C \times (T_i - T_\infty)$$	
	$$= 4/3 \times \pi \times (2.5\times10^{-2})^3 \times 2850 \times 460 \times (225 - 25)$$	
	$$Q_0 = 47.26\times10^3 \text{ J}$$	
	$\frac{Q}{Q_0} = 0.75$, $Q = 0.75 \times 47.26\times10^3$	02
	$= 35.45\times10^3$ Joules	

Q.No.	Solution and Scheme	Marks
	Module-03.	

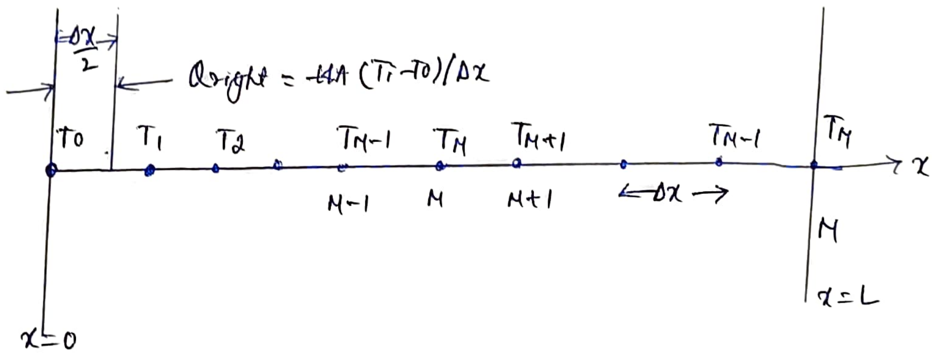
5a. **Types of Boundary conditions:**

i) Prescribed temperatures at the boundaries:

Let the temperatures at $x=0$ and $x=L$ be given as T_0 and T_M respectively. Then $T(0)=T_0$ and $T(L)=T_M$, give the two additional equations required to solve for $M+1$ unknown node temperatures. ~~In this case, there~~.

Apply energy balance to volume elements of nodes at the boundaries ie nodes 0 and M.

$$Q_{left} + KA\left(\frac{T_1-T_0}{\Delta x}\right) + \dot{q}_0\left(\frac{A\cdot\Delta x}{2}\right) = 0 \quad \text{---(1)}$$



(02)

ii) Prescribed heat flux at the boundaries.

Let q_{left} and q_{right} be the heat flux at nodes 0 and M respectively.

W.K.T

$$Q_{left} + KA\left(\frac{T_1-T_0}{\Delta x}\right) + \dot{q}_0\left(\frac{A\cdot\Delta x}{2}\right) = 0 \quad \text{---(1)}$$

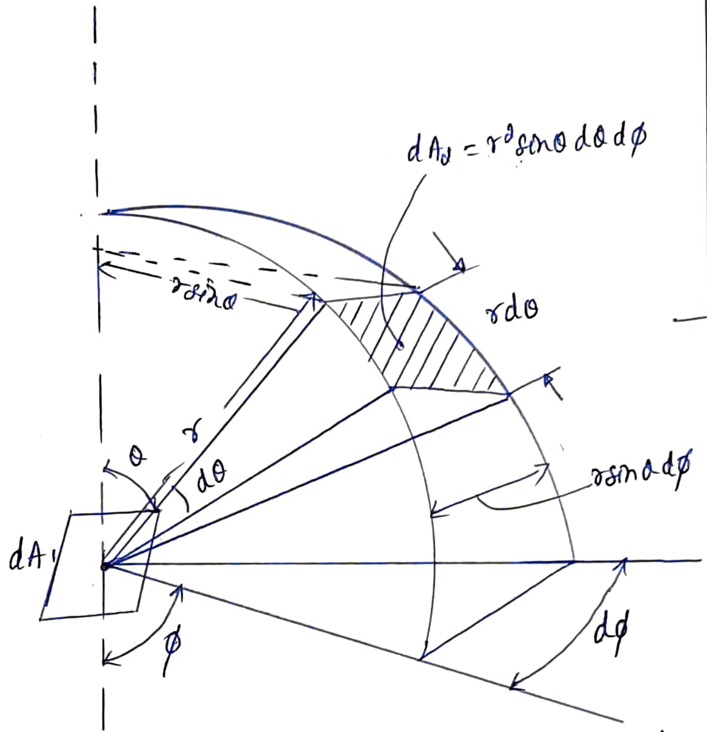
For node 0

$$q_{left}\,A + KA\left(\frac{T_1-T_0}{\Delta x}\right) + \dot{q}_0\left(\frac{A\Delta x}{2}\right) = 0 \quad \text{---(2)}$$

$$2T_1 - 2T_0 + \frac{(\Delta x)^2\cdot \dot{q}_0}{K} + \frac{2\Delta x\, q_{left}}{K} = 0 \quad \text{---(3)}$$

Q.No.	Solution and Scheme	Marks
	For node M, replace the subscript 0 by M and subscript 1 by M−1	
	$q_{right} A + K.A \frac{(T_{M-1} - T_M)}{\Delta x} + q_M \left(\frac{A \cdot \Delta x}{2}\right) = 0$ — (4)	
	$2T_{M-1} - 2T_M + \frac{(\Delta x)^2 q_M}{K} + \frac{2\Delta x \cdot q_{right}}{K} = 0$ — (5)	
	Eqn (3) and (5) are finite difference representation of the prescribed heat flux conditions at nodes 0 and M respectively.	02
	iii) Convection boundary condition:	
	Let the boundaries at $x=0$ and $x=L$ be subjected to convection to a fluid at a temperature of T_a with a heat transfer coefficient of h.	
	We have	
	$h.A(T_a - T_0) + K.A\frac{(T_1 - T_0)}{\Delta x} + q_0\left(\frac{A \cdot \Delta x}{2}\right) = 0$ — (1)	
	$2T_1 - 2T_0\left(1 + \frac{h \cdot \Delta x}{K}\right) + \frac{(\Delta x)^2 q_0}{K} + \frac{2h \cdot \Delta x}{K} T_a$ — (2)	
	For node M, replace the subscript 0 by M and subscript 1 by M−1, we get	
	$h.A(T_a - T_M) + K.A\left(\frac{T_{M-1} - T_M}{\Delta x}\right) + q_M\left(\frac{A \Delta x}{2}\right) = 0$ — (3)	
	$2T_{M-1} - 2T_M\left(1 + \frac{h \cdot \Delta x}{K}\right) + \frac{(\Delta x)^2 q_M}{K} + \frac{2h\Delta x}{K} T_a = 0$ — (4)	
	Eqn (3) and (4) are finite difference representations for convective boundary conditions at nodes '0' and M.	02

Q.No.	Solution and Scheme	Marks
5b)	Relation between normal intensity and emissive power.	03



Consider a small black surface of area dA_1 emitting radiation in different directions. A black body radiation collector through which the radiation pass is located at an angular position 'θ' towards the surface normal and angle ϕ of a spherical coordinate system. The collector subtends a solid angle $d\omega$ when viewed from a point on the emitter.

Let us consider radiation from the elementary area dA_1 at the centre of a sphere. Suppose this radiation is absorbed by a second elemental area dA_2, a portion of the hemispherical surface.

The projected area of dA_1 on a plane perpendicular to the line joining dA_1 and

$dA_2 = dA_1 \cos\theta$.

Q.No.	Solution and Scheme	Marks

The Solid angle subtended by

$$d\omega = \frac{dA_2}{r^2}.$$

The intensity of radiation, $I = \dfrac{dQ_{1-2}}{dA_1 \cos\theta \times \dfrac{dA_2}{r^2}}.$

Where dQ_{1-2} is the rate of radiation heat transfer from dA_1 to dA_2

$$dQ_{1-2} = I \cdot dA_1 \sin\theta \cos\theta \, d\theta \, d\phi$$ 05

The total radiation through the hemisphere is

$$Q = I \, dA_1 \int_{\theta=0}^{\pi/2} \int_{\phi=0}^{2\pi} \sin\theta \cos\theta \, d\theta \, d\phi.$$

$$= 2\pi I \, dA_1 \int_{\theta=0}^{\pi/2} \sin\theta \cos\theta \, d\theta$$

$$= \pi I \, dA_1 \int_{\theta=0}^{\pi/2} \sin 2\theta \, d\theta.$$

$$Q = \pi \cdot I \, dA_1$$

or $Q = E \, dA_1$ 02

$$E \, dA_1 = \pi I \, dA_1$$

$$E = \pi I$$

The total emissive power of a diffuse surface is equal to π times its intensity of radiation.

Q.No.	Solution and Scheme	Marks
	OR.	
6a)	Explain	
	i) Stefan Boltzman law: It states that the emissive power of a black body is directly proportional to the fourth power of the absolute temperature of the body. ie $E_b \propto T^4$; $E_b = \sigma T^4$. Where, E_b – Emissive power of black body. σ – Stefan Boltzmann constant. T – Absolute temp. in K.	02
	ii) Kirchoff's law: It states that at any temperature, the ratio of total emissive power (E) to the total absorptivity (α) is a constant for all substances which are in thermal equilibrium with their environment. ie $\varepsilon = \alpha$.	02
	iii) Planks law: This law states that the monochromatic power of a blackbody is given by $$E_{b\lambda} = \frac{C_1}{\lambda^5 \left(e^{C_2/\lambda T} - 1\right)}$$	02

Q.No.	Solution and Scheme	Marks

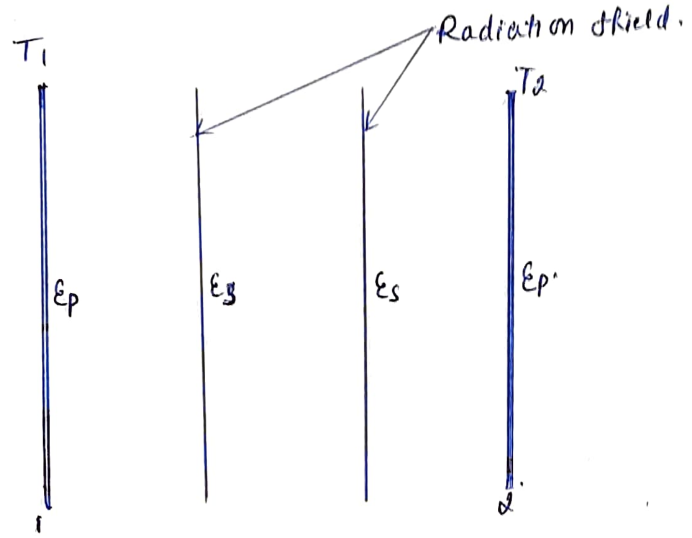
Where, C_1 and C_2 are constants whose values are found from experimental data.

$C_1 = 3.7415 \times 10^{-16}\ W/m^2$ and $C_2 = 1.4388 \times 10^{-2}\ mK$.

λ – is the wavelength

T – is the absolute temperature in K.

6 b)



(02)

i) The rate of heat transfer without shield.

$$(Q)_{\text{without shield}} = \frac{\sigma (T_1^4 - T_2^4)}{\left(\frac{1-\varepsilon_p}{\varepsilon_p}\right) + 1 + \left(\frac{1-\varepsilon_p}{\varepsilon_p}\right)}$$

$$= \frac{\sigma (T_1^4 - T_2^4)}{\frac{1}{\varepsilon_p} + \frac{1}{\varepsilon_p} - 1}$$

(02)

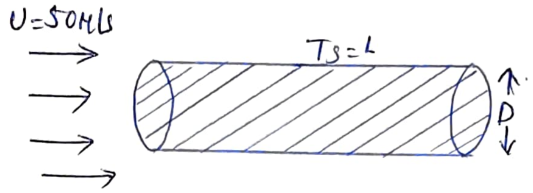
$$= \frac{\sigma (T_1^4 - T_2^4)}{\frac{1}{0.5} + \frac{1}{0.5} - 1}$$

$$= \sigma (T_1^4 - T_2^4) \times 0.33 \quad \text{——} (1)$$

Q.No.	Solution and Scheme	Marks
	ii) The rate of heat transfer with shield.	
	$$(Q)_{\text{with shield}} = \frac{\sigma(T_1^4 - T_2^4)}{\frac{1}{\varepsilon_p} + \frac{1}{\varepsilon_p} + 2\left[\frac{1}{\varepsilon_s} + \frac{1}{\varepsilon_s}\right] - (2+1)}$$	
	$$= \frac{\sigma(T_1^4 - T_2^4)}{\frac{2}{\varepsilon_p} + \frac{4}{\varepsilon_s} - 3}$$	03
	$$= \frac{\sigma(T_1^4 - T_2^4)}{\frac{2}{0.5} + \frac{4}{0.05} - 3}$$	
	$$= \sigma(T_1^4 - T_2^4) \times 0.012345 \quad \text{—(2)}$$	
	Percentage reduction in heat flow	
	$$= \left(\frac{Q_{\text{without shield}} - Q_{\text{with shield}}}{Q_{\text{without shield}}}\right) \times 100$$	
	$$= \frac{\sigma(T_1^4 - T_2^4) \times 0.33 - \sigma(T_1^4 - T_2^4) \times 0.012345}{\sigma(T_1^4 - T_2^4) \times 0.33}$$	02
	$= 96.26\%$	
	Module – 04	
7 a.	Physical significance.	
	i) Prandtl number (Pr): It is the ratio of kinematic viscosity to thermal diffisivity.	02
	$$Pr = \frac{\mu C_p}{K}$$	

Q.No.	Solution and Scheme	Marks
	Prandtl number is a connecting link between the velocity field and temperature field and its value strongly influences relative growth of velocity and thermal boundary layers.	
	ii) Reynolds number: It is defined as the ratio of the inertia force to the viscous force. $Re = \frac{\text{Inertia force}}{\text{viscous force}} = \frac{\rho UL}{\mu}$.	02
	Higher the value of Re, the greater will be the relative contribution of inertia effect. The smaller the value of Re, the greater will be the relative magnitude of the viscous stresses.	
	iii) Nusselt number: It is ratio of heat flow rate by convection process under a unit temperature gradient to the heat flow rate by conduction process under a unit temperature gradient through a stationary thickness. $Nu = \frac{hL}{k}$ or $\frac{hD}{k}$.	02
	The Nusselt number is a measure of the convective heat transfer coefficient. For a given value of the Nusselt number, the convective heat transfer coefficient is directly proportional to thermal conductivity of fluid and inversely proportional to the significant length.	

Q.No.	Solution and Scheme	Marks
7b)	$T_\infty = 25°C$, $U = 50 m/s$, $D = 20 mm = 20 \times 10^{-3} m.$ $T_s = 457°c$	



Reynolds number (Re)

$$Re = \frac{UD}{\nu}.$$

Mean bulk fluid temp

or

film temp $(T_f) = \frac{T_s + T_\infty}{2} = \frac{25+457}{2}.$ — 02

$= 241°c.$

At $T_f = 241°c$,

The corresponding properties are, refer HMT databook book.

Refer pg. 33

$\rho = 0.674 \; kg/m^3$

$\nu = 40.61 \times 10^{-6} \; m^2/s$

$Pr = 0.677$

$K = 0.04268 \; W/m°C$

$$Re = \frac{UD}{\nu} = \frac{50 \times 20 \times 10^{-3}}{40.61 \times 10^{-6}}$$

$Re = 24,624.47$ — 03

From HMT handbook, pg No: 115

We have $Nu = C \, Re^m \, Pr^{0.333}$

Q.No.	Solution and Scheme	Marks

$C = 0.193$, $m = 0.618$

$\therefore$

$Nu = 0.193\, Re^{0.618}\, Pr^{0.333}$

$= 0.193\,(24624.47)^{0.618}\,(0.677)^{0.333}$

$Nu = 87.70$

WKT

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

$87.70 = \frac{h \times 20\times10^{-3}}{0.04268}$

$h = 187.1518\ W/m^2\,^\circ C$ — (03)

The electrical power density

$\therefore\quad Q = hA\,(T_s - T_\infty)$

$= 187.1518 \times \pi \times 20\times10^{-3}\cdot(457-25)$

$Q = 5080.58\ W/m$ — (02)

$= 16,169891\ W/m^3$

OR.

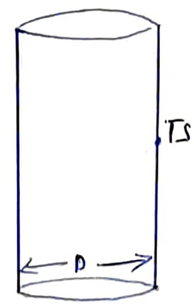
8 a) Given data:

$D = 25\,cm = 25\times10^{-2}\,m$

$T_s = 100^\circ C$

$T_\infty = 20^\circ C$

$Q = ?$



Mean film temperature (T_f)

$T_f = \frac{100+20}{2} = 60^\circ C$,

Q.No.	Solution and Scheme	Marks

Properties of air at 60°C are $\beta = \frac{1}{(60+273)} = 3.003 \times 10^{-3}\ 1/K$.

$Pr = 0.696$, $\nu = 18.97 \times 10^{-6}\ m^2/s$ $K = 0.02896\ W/mK$.

$\nu = 18.97 \times 10^{-6}\ m^2/s$, $K = 0.02896\ W/mK$.

$$Ra_L = \frac{g\beta\Delta T L^3}{\nu^2} Pr = \frac{9.81 \times (3.003 \times 10^{-3}) \times (100-20) \times (0.0625)^3}{(18.97 \times 10^{-6})^2} \times 0.696$$

(03)

$$Ra_L = 1.112 \times 10^{6}$$

From HMT data handbook,

We have, Nusselt number at top. (Nu_{top})

i) $$Nu_{top} = h_{top} L/K = 0.54 \times (Ra_L)^{0.25}$$

Characteristic length (L) $= A/P = \pi D^2/4 / \pi D = 0.0625 m$.

$$\therefore h_{top} \times \frac{0.0625}{0.02896} = 0.54 \times (1.112 \times 10^{6})^{0.25}$$

(03)

$$h_{top} = 8.12\ W/m^2K.$$

ii) Nusselt number at bottom (Nu_{bottom})

$$Nu_{bottom} = h_{bottom} L/K = 0.27 \times (1.112 \times 10^{6})^{0.25}$$

$$\therefore h_{bottom} \times \frac{0.0625}{0.02896} = 0.27 \times (1.112 \times 10^{6})^{0.25}$$

(02)

$$h_{bottom} = 4.059\ W/m^2K.$$

Total heat loss to air (Q_{total})

$$Q_{total} = Q_{top} + Q_{bottom}$$

$$= (\pi D L) h_{top} \Delta T + (\pi D L) h_{bottom} \Delta T$$

$$= (\pi \times 0.25 \times 1) \times (100-20) \times (8.76 + 4.059)$$

(02)

$$Q_{total} = 805.44\ W$$

Q.No.	Solution and Scheme	Marks

8 b) Let us assume that the heat transfer coefficient in a fully developed forced convection in a tube is a function of the following variables.

$$h = f(\rho, D, V, \mu, C_p, K)$$

or $$f(h, \rho, D, V, \mu, C_p, K)$$

h - heat transfer coefficient - $MT^{-3}\theta^{-1}$
ρ - Fluid density - ML^{-3}
D - Tube diameter - L
V - Fluid velocity - LT^{-1}
μ - Fluid viscosity - $ML^{-1}T^{-1}$
C_p - Specific heat - $L^2T^{-2}\theta^{-1}$
K - Thermal conductivity - $MLT^{-3}\theta^{-1}$

The total number of variables, $n = 7$

Fundamental dimensions in the problem are M, L, T, θ and hence $m = 4$

Number of dimensionless π-terms $= n - m = 7 - 4 = 3$.

The repeating variables are μ, K, V, D.

π_1-Term.

$$\pi_1 = \mu^{a_1} K^{b_1} V^{c_1} D^{d_1} \cdot h.$$

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

Equating the powers

For M, $0 = a_1 + b_1 + 1$

For L, $0 = -a_1 + b_1 + c_1 + d_1$ — (02)

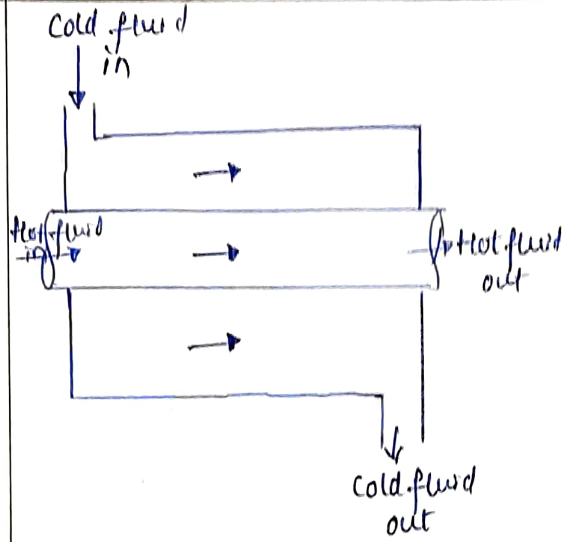
For T, $0 = -a_1 - 3b_1 - c_1 - 3$

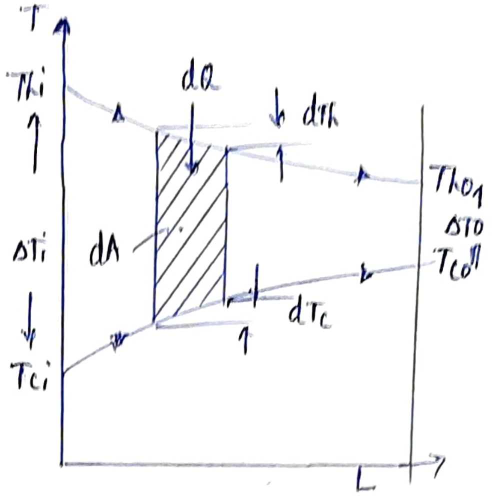
For θ, $0 = -b_1 - 1$

Q.No.	Solution and Scheme	Marks
	$\therefore\ b_1=-1,\ a_1=0,\ c_1=0,\ d_1=1$	
	$\therefore\ \pi_1 = \mu^0 k^{-1} V^0 D^1 h.$	
	$\boxed{\pi_1 = \frac{hD}{k}}$	
	π_2 - Term:	
	$\pi_2 = \mu^{a_2} k^{b_2} V^{c_2} D^{d_2} \cdot \rho.$	
	$M^0L^0T^0\theta^0 = (ML^{-1}T^{-1})^{a_1}(MLT^{-3}\theta^{-1})^{b_2}(LT^{-1})^{c_1}(L)^{d_1}ML^{-3}$	
	Equating the powers	
	For M: $0 = a_2+b_2+1$	
	For L: $0 = -a_2+b_2+c_2+d_2-3$	
	For T: $0 = -a_2-3b_2-c_2$	
	For θ: $0 = -b_2.$	
	$\therefore\ b_2=0,\ a_2=-1,\ c_2=1\ \ d_2=1$	02
	$\pi_2 = \mu^{-1} k^0 V^1 D^1 \rho.$	
	$\boxed{\pi_2 = \frac{\rho V D}{\mu}}$	
	π_3 - Term:	
	$\pi_3 = \mu^{a_3} k^{b_3} V^{c_3} D^{d_3} \cdot C_p.$	
	$M^0L^0T^0\theta^0 = (ML^{-1}T^{-1})^{a_3}(MLT^{-3}\theta^{-1})^{b_3}(LT^{-1})^{c_3}(L)^{d_3}L^2T^{-2}\theta^{-1}$	
	Equating the powers	
	For M: $0 = a_3+b_3$	
	For L: $0 = -a_3+b_3+c_3+d_3+2$	
	For T: $0 = -a_3-3b_3-c_3-2.$	

Q.No.	Solution and Scheme	Marks
	For θ : $0 = -b_3 - 1$ $b_3 = -1$ $\therefore \pi_3 = \mu^1 k^{-1} V^0 D^0 C_p$ $\pi_3 = \frac{\mu C_p}{k}$ $\therefore$ We can write $f(\pi_1, \pi_2, \pi_3) = 0$ $f\left(\frac{hD}{k}, \frac{\rho VD}{\mu}, \frac{\mu C_p}{k}\right) = 0$ $\frac{hD}{k} = f\left(\frac{\rho VD}{\mu}, \frac{\mu C_p}{k}\right)$ $\boxed{Nu = f(Re, Pr)}$ Where, Nu = Nusselt number $= \frac{hD}{k}$ Re = Reynolds number $= \frac{\rho VD}{\mu}$ Pr = Prandtl number $= \frac{\mu C_p}{k}$	03
	Module - 05	
9 a)	Consider a double pipe, parallel flow heat exchanger in which a hot fluid and a cold fluid flow parallel to each other separated by a solid wall. Hot fluid enters at a temperature of T_{hi} and leaves the heat exchanger at a temperature of T_{ho}, cold fluid enters the heat exchanger at a temperature of T_{ci} and leaves at a temperature of T_{co}	

Q.No.	Solution and Scheme	Marks





— (2)

Assumptions:

1) The overall heat transfer coefficient (U) is constant.
2) The flow conditions are steady.
3) The specific heats and mass flow rates of both fluids are constant.
4) There is no loss of heat to the surroundings due to the heat exchanger being perfectly insulated
5) There is no change of phase either of fluid during the heat transfer.
6) The changes in potential and kinetic energies are negligible.
7) Axial conduction along the tubes of the heat exchanger is negligible.

Let us consider an elementary area dA of the heat exchanger. The rate of flow of heat through the elementary area is given by

$$dQ = U\,dA\,(T_h - T_c)$$

$$= U\,dA \cdot \Delta T \quad \text{—— (1)}$$

Q.No.	Solution and Scheme	Marks

As a result of heat transfer dQ through the area dA, the hot fluid is cooled by dT_h whereas the cold fluid is heated up by dT_c. The energy balance over a differential area dA may be written as

$$dQ = -M_h C_{ph} dT_h = M_c C_{pc} dT_c \quad \text{———} \quad (2)$$

$$\therefore \quad dQ = -M_h C_{ph} dT_h .$$

$$dT_h = -\frac{dQ}{M_h C_{ph}} \quad \text{———} \quad (3)$$

and

$$dT_c = \frac{dQ}{M_c C_{pc}} \quad \text{———} \quad (4)$$

where, M_h and M_c are the mass flow rates, C_{ph} and C_{pc} are the specific heat of hot and cold fluids.

Therefore.

Eqn (3) − Eqn (4)

$$dT_h - dT_c = -dQ\left[\frac{1}{M_h C_{ph}} + \frac{1}{M_c C_{pc}}\right]$$

$$d(T_h - T_c) = -dQ\left[\frac{1}{M_h C_{ph}} + \frac{1}{M_c C_{pc}}\right] \quad \text{—} \quad 0.2$$

$$\left[dQ = U dA\,\Delta T = U dA\,(T_h - T_c)\right]$$

$$d(T_h - T_c) = -U dA\,(T_h - T_c)\left[\frac{1}{M_h C_{ph}} + \frac{1}{M_c C_{pc}}\right]$$

$$\frac{d(T_h - T_c)}{(T_h - T_c)} = -U dA\left[\frac{1}{M_h C_{ph}} + \frac{1}{M_c C_{pc}}\right]$$

Q.No.	Solution and Scheme	Marks

ie $\frac{d\Delta T}{\Delta T} = -U dA \left[\frac{1}{M_h C_{ph}} + \frac{1}{M_c C_{pc}}\right]$

Integrating above eqn. between the inlet and outlet of the heat exchanger.

$$\int_{\Delta T_i}^{\Delta T_o} \frac{d\Delta T}{\Delta T} = -U \left[\frac{1}{M_h C_{ph}} + \frac{1}{M_c C_{pc}}\right] \int_0^A dA$$

$$\ln \frac{\Delta T_o}{\Delta T_i} = -U \left[\frac{1}{M_h C_{ph}} + \frac{1}{M_c C_{pc}}\right] A \quad \text{— (5)}$$

Now, considering the total heat transfer rate for the entire heat exchanger, we have.

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

$$M_h C_{ph} = \frac{Q}{T_{hi} - T_{ho}} \quad \text{or} \quad \frac{1}{M_h C_{ph}} = \frac{T_{hi} - T_{ho}}{Q}.$$

$$\text{and} \quad \frac{1}{M_c C_{pc}} = \frac{T_{co} - T_{ci}}{Q}.$$

$$\ln \frac{\Delta T_o}{\Delta T_i} = -U \left[\frac{T_{hi} - T_{ho}}{Q} + \frac{T_{co} - T_{ci}}{Q}\right] A$$

$$= \frac{UA}{Q} \left[(T_{ho} - T_{co}) - (T_{hi} - T_{ci})\right]$$ (02)

$$Q = \frac{UA\,(\Delta T_o - \Delta T_i)}{\ln \frac{\Delta T_o}{\Delta T_i}} \quad \text{or} \quad Q = UA\,\text{LMTD}.$$

where, $\boxed{\text{LMTD} = \frac{\Delta T_o - \Delta T_i}{\ln \frac{\Delta T_o}{\Delta T_i}}}$

Q.No.	Solution and Scheme	Marks

9b)

Given data.

Hot fluid

$M_h = 250\ kg/hr$

$= 0.069\ kg/s$

$C_{ph} = 3350\ J/kg^{\circ}C$

$T_{hi} = 120^{\circ}C$

Cold fluid

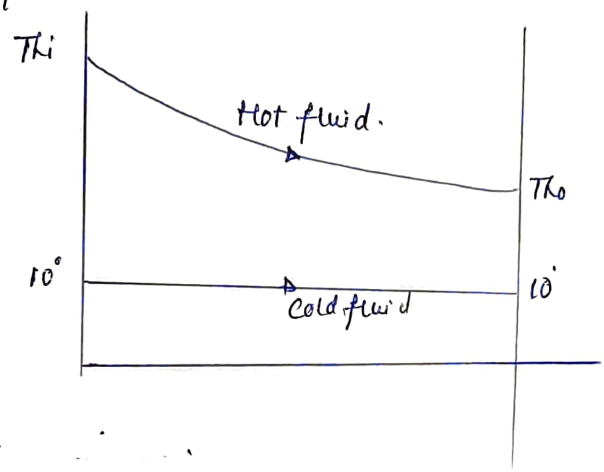
$M_c = 1000\ kg/hr$

$= 0.277\ kg/s$

$T_{ci} = 10^{\circ}C$

$A = 0.25\ m^2.$

$U = 1160\ W/m^2\ {}^{\circ}C$



02

In Evaporators, $C_h << C_c$, $C_{min} = M_h C_{ph}$.

The effectiveness is

$$\varepsilon = 1 - e^{-N}$$

(HMT. databook. Pg No: 153: sixth edition)

W.K.T

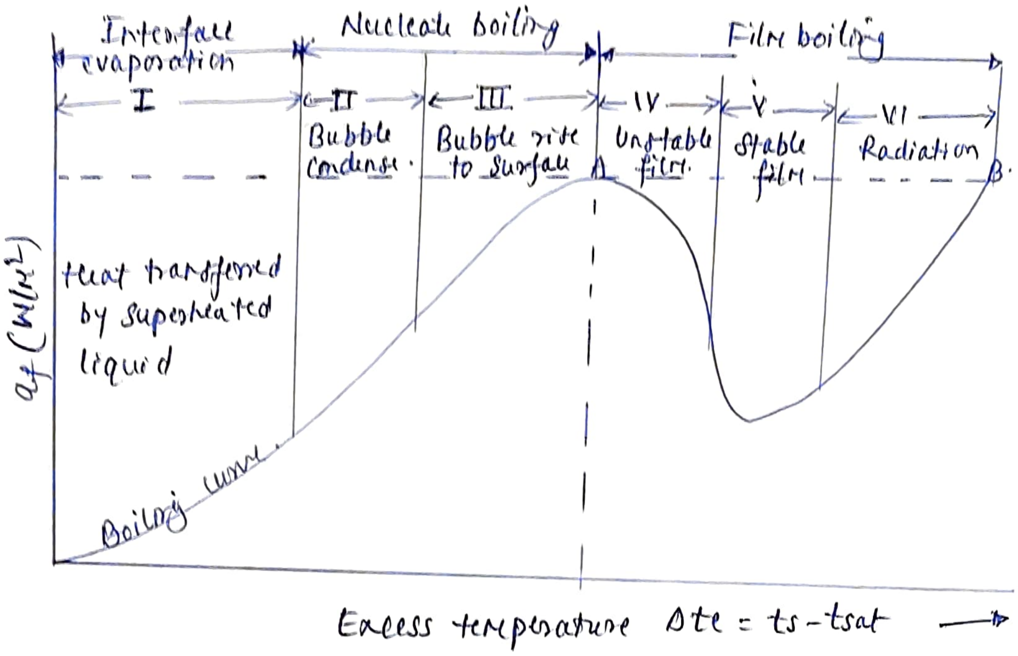
$$N = \frac{UA}{C_{min}} = \frac{1160 \times 0.25}{0.069 \times 3350}$$

$$N = 1.25$$

02

Q.No.	Solution and Scheme	Marks
	We have $\varepsilon = 1 - e^{-N}$ $= 1 - e^{-1.25}$ $\varepsilon = 0.71$	02
	From HMT data hand book (Pg.No:152) For $C_{min} = m_h c_h$, we have $\varepsilon = \frac{T_{hi} - T_{ho}}{T_{hi} - T_{ci}}$, $0.71 = \frac{120 - T_{ho}}{120 - 10}$ $T_{ho} = 41.9°C$	02
	Rate of heat transfer. $Q = \varepsilon \times Q_{max}$ $= \varepsilon \times C_{min} \times (\Delta T)_{max}$ $= 0.71 \times (0.069 \times 3350) \times (120 - 10)$ $Q = 18.052$ KW	02
	OR.	
10 a)	Boiling curve for water. Whether the boiling phenomenon corresponds to pool boiling or forced circulation boiling, there are three definite regimes of boiling ie 1) Interface evaporation 2) Nucleate boiling 3) Film boiling	

Q.No.	Solution and Scheme	Marks



(02)

1. Interface evaporation:

Here the excess temperature, Δte is very small and $\simeq 5^{\circ}C$. In this region the liquid near the surface is superheated slightly, the convection currents circulate the liquid and evaporation takes place at the liquid surface.

2. Nucleate boiling:

With the increase in Δte, the formation of bubbles on the surface of the wire at certain localised spots commences. The bubbles condense in the liquid without reaching the liquid surface. It is the region II where nucleate boiling starts. With further increase in Δte, the bubbles are formed more rapidly and rise to the surface of the liquid resulting in rapid evaporation as indicated

(02)

Q.No.	Solution and Scheme	Marks

in the region III. The ~~noble~~ nucleate boiling is thus characterised by formation of bubbles at the nucleation sites and the resulting liquid agitation. Nucleate boiling exists upto $\Delta T_e \simeq 50^\circ C$.

3. Film boiling:

Film boiling comprises of regions IV, V and VI. The trend of increase of heat flux with increase in excess temperature observed upto region III is reversed in region IV (called film boiling region). This is due to the fact that the bubble formation is very rapid and the ~~blankets~~ bubbles blanket the heating surface and prevent the incoming fresh liquid from taking their place. Eventually the bubbles coalesce and form a vapour film which covers the surface completely. Since the thermal conductivity of vapour film is much less than that of the liquid the heat flux drops with growth in ΔT_e. 02

Within the temperature range $50^\circ C < \Delta T_e < 150^\circ C$, conditions oscillate between nucleate and film boiling and the phase is referred to as transition boiling, unstable film boiling or partial film boiling (region IV). With further increase in ΔT_e the vapour film is stabilised and the heating surface is completely covered by a vapour blanket and the heat flux is the lowest as shown in region V.

Q.No.	Solution and Scheme	Marks
10 b	Given data:	
	$D = 80\,mm = 80 \times 10^{-3}\,m$.	
	$L = 1\,m$.	
	$T_s = 50^\circ C$	
	Mean film temperature $(T_f) = \frac{100+50}{2} = 75^\circ C$.	
	The properties of water at $T_f = 75^\circ C$	(03)
	$\rho_L = 975\,kg/m^3$, $\mu_L = 375 \times 10^{-6}\,Ns/m^2$, $k = 0.67\,W/m^\circ C$	
	At saturation temperature, $T_v = 100^\circ C$, $h_{fg} = 2257\,kJ/kg$	
	i) Rate of heat transfer 'Q'	
	Laminar condensation on a vertical surface.	
	$h = 0.943 \left[\frac{k^3 \rho^2 g h_{fg}}{\mu_L D (T_v - T_s)} \right]^{0.25}$ (HMT data hand book)	(03)
	$= 0.943 \left[\frac{0.67^3 \times 975^2 \times 9.81 \times 2257 \times 10^3}{375 \times 10^{-6} \times 80 \times 10^{-3} (100-50)} \right]^{0.25}$	
	$h = 8060.012\,W/m^2\,^\circ C$	(02)
	$Q = hA(T_v - T_s) = 8060 \times \pi \times 80 \times 10^{-3} \times 1 \times (100-50)$	
	$= 101.296\,kW$	
	ii) The rate of condensation of steam, m'	(02)
	$m^\circ = \frac{Q}{h_{fg}} = \frac{101.296 \times 10^3}{2257 \times 10^3} = 0.04488\,kg/s$	