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PART - A

2 Marks Questions

1.
Ans. -Transfer of heat from one place to another place, Which occurs as a result of a temperature difference
- Conduction
 - Convection
 - Radiation.

2.
Ans. Conduction
- Heat transfer in bodies due to fixed molecules.
 - This take place in solid bodies.
- Convection**
- Heat transfer in bodies due to moving molecules.
 - This take place in fluid liquid or gas.

3.
Ans. $Q_{\text{conduction}} = -kA \times (\Delta T / \Delta x)$
 where,
 A is the cross-sectional area
 ΔT is temperature difference ($T_1 - T_2$)
 Δx is distance between the two surfaces

4.
Ans. Natural convection heat transfer process is the heat transfer process where the heat transfer takes place due to the natural temperature difference or density difference and no external forces are employed for the fluid movement.

5.
Ans. Planck's distribution law:
 The relationship between the monochromatic emissive power of a black body and wave length of a radiation at a particular temperature is given by the following expression, by Planck.

$$E_{b\lambda} = \frac{C_1 \lambda^{-5}}{\left(\frac{C_2}{\lambda T} \right)_{-1}}$$

Where $E_{b\lambda}$ = Monochromatic emissive power W/m^2
 λ = Wave length - m
 $c_1 = 0.374 \times 10^{-15} W/m^2$
 $c_2 = 14.4 \times 10^{-3} mK$

6.
Ans. The Wien's law gives the relationship between temperature and wave length corresponding to the maximum spectral emissive power of the black body at that temperature.

$$\lambda_{\text{max}} \times T = C_3$$

$$c_3 = 2.9 \times 10^{-3} \text{ [Radiation constant]}$$

$$\Rightarrow \lambda_{\text{max}} \times T = 2.9 \times 10^{-3} mK$$

7.
Ans. The emissive power of a black body is proportional to the fourth power of absolute temperature.

$E_b \propto T^4$
 $E_b = \sigma T^4$
 Where E_b = Emissive power, w/m^2
 σ = Stefan. Boltzmann constant
 $= 5.67 \times 10^{-8} W/m^2 K^4$
 Where T = Temperature, K.

8.
Ans.
- Plate exchangers are limited when high pressures, high temperatures, or aggressive fluids are present.
 - Because of this problem these type of heat exchangers have only been used in small, low pressure applications such as on oil coolers for engines.
9.
Ans.
- (i) Shell
 - (ii) Tubes
 - (iii) Rear header
 - (iv) Front header
 - (v) Baffles
 - (vi) Tubes sheet
10.
Ans. A heat exchanger is a device that transfers heat from one fluid to the others. A condenser is simply a heat exchanger. It compresses refrigerants into a hot gas to then condense them into a liquid. Majorly condenser is used to cool the required vapor and heat exchanger is used to heat the required liquid
11.
Ans. The earth can be considered as isochoric system that is volume of air is constant around our globe. As at constant volume the pressure is directly proportional to temp. as a result temp will also decrease as pressure decreases as we move higher and higher.
12.
Ans. If the fluid motion is artificially created by means of an external force like a blower or fan, that type of heat transfer is known as forced convection.
13.
Ans.
1. Reynolds number [Re].
 2. Nusselt number [Nu].
 3. Prandtl number [Pr].
14.
Ans. Fins are extended surfaces used primarily to enhance the heat transfer rate between the solid fins and an adjoining fluid.
15.
Ans.
1. Straight fin of uniform cross section
 2. Straight fin of non-uniform cross section
 3. Annular fin
 4. Pin fin
16.
Ans. Stefan Boltzman constant (σ) = $5.6697 \times 10^{-8} \text{ W / (m}^2 \text{ K}^4)$
17.
Ans. Heat conduction is a process in which heat is transferred from the hotter part to the colder part of a body without involving any actual movement of the body's molecules.
Example: When frying vegetables in a pan. Heat transfer occurs from the flame to the pan and the vegetables.
18.
Ans. Heat convection is a process in which heat is transferred in the liquid and gases from a higher temperature region to a lower temperature region. Convection heat transfer occurs partly due to the actual movement of molecules or due to the mass transfer.
Example: Heating of milk in a pan.
19.
Ans. Heat radiation is a process in which heat is transferred from one body to another without involving the medium's molecules. Radiation heat transfer does not depend on the medium.
Example: In a microwave, the substances are heated directly without any heating medium.

20.

Ans. The difference in temperature is the primary cause of heat transfer.

21.

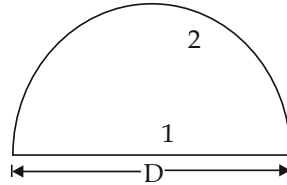
Ans. $F_{11} + F_{12} = 1$ (Principal of conservation)

$$F_{21} + F_{22} = 1$$

$$\Rightarrow F_{11} = 0$$

$$\Rightarrow F_{12} = 1$$

$$A_1 F_{12} = A_2 F_{21} \text{ (Reciprocity theorem)}$$



$$\left(\frac{\pi D^2}{4} \right) \times 1 = \left(\frac{\pi D^2}{2} \right) \times F_{21}$$

$$F_{21} = 1/2$$

22.

Ans. A heat exchanger (HX) is used for exchanging heat between two mediums i.e fluids.

Its consists of following two groups;

1. Inirect Contact HX

2. Direct Contact HX

In other words, heat exchanger is a device used for heat dissipation from a certain part where excess heat may can prove to be harmful to the system.

It increases the heat dissipation by simply exposing more area to the atmosphere as the heat transfer directly depends on the availability of area for heat transfer. For instance, fins on the engine cylinder

23.

- Ans.**
- After a period of operation the heat-transfer surfaces for a heat exchanger may become coated with various deposits present in the flow systems, or the surfaces may become corroded as a result of the interaction between the fluids and the material used for construction of the heat exchanger.
 - This coating represents an additional resistance to the heat flow, and thus results in decreased performance.
 - The overall effect is usually represented by a fouling factor, or fouling resistance (R)

$$R_f = \frac{1}{U_{\text{dirty}}} - \frac{1}{U_{\text{clean}}}$$

24.

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

where

k is the materials conductivity [W.m⁻¹.K⁻¹]

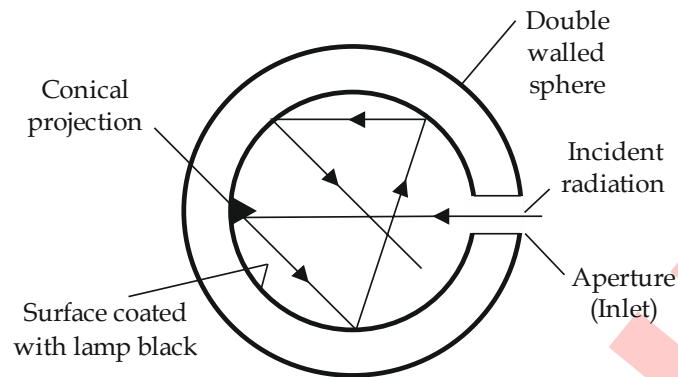
q_v is the rate at which energy is generated per unit volume of the medium [W.m⁻³]

ρ is the density [kg.m³]

c_p is the specific heat capacity [J.kg⁻¹.K⁻¹]

25.

Ans.

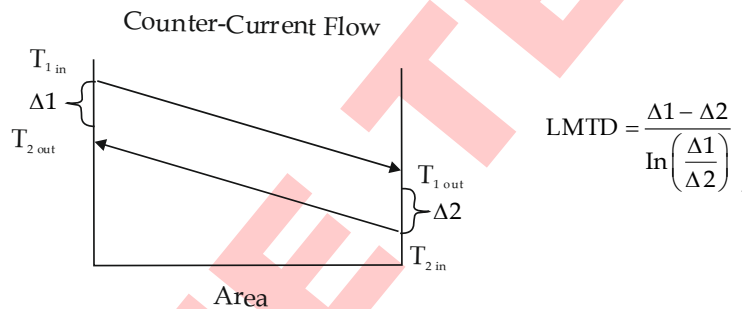


26.

Ans. In dropwise condensation the condensate liquid collects in the form of countless droplets of varying diameters on the condensing surface, instead of forming a continuous film, and does not wet the solid cooling surface.

27.

Ans.



28.

Ans. Convective heat transfer depends on following factors

- Flow velocity,
- Availability of Turbulence,
- Thermo-physical properties of Flowing fluid,
- Shape and size of Geometry
- Temperature gradient

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

Where

q is the local heat flux density [W.m^{-2}]

h is the heat transfer coefficient [$\text{W.m}^{-2}.\text{K}$]

ΔT is the temperature difference [K]

29.

Ans. This law states that the ratio of total emissive power to the absorptivity is constant for all surfaces which are in thermal equilibrium with the surroundings. This can be written as

$$\frac{E_1}{\alpha_1} = \frac{E_2}{\alpha_2} = \frac{E_3}{\alpha_3}$$

It also states that the emissivity of the body is always equal to its absorptivity when the body remains in thermal equilibrium with its surroundings.

$\alpha_1 = E_1$; $\alpha_2 = E_2$ and so on.



30.

Ans. It states that the total emissive power E_b from a radiating plane surface in any direction proportional to the angle of emission

$$E_b \propto \cos\theta$$

31.

Ans. **Effectiveness of fin (ϵ_{fin}):** It is defined as the ratio of the actual heat transfer that takes place from the fin to the heat that would be dissipated from the same surface area without fin.

Efficiency of fin (η_{eff}):

$$\eta_{fin} = \frac{\dot{Q}_{fin}}{\dot{Q}_{fin,max}} = \frac{\text{Actual heat transfer rate from the fin}}{\text{Ideal heat transfer rate from the fin}}$$

$$\dot{Q}_{fin} = \eta_{fin} \dot{Q}_{fin,max} = \eta_{fin} h A_{fin} (T_b - T_\infty)$$

32.

Ans. When steady state is reached, the power supplied to the wire equals the convective heat loss which is given by

$$Q = h A \Delta t = h (\pi d l) \times (t_s - t_w)$$

where,

l = Length of wire

d = Diameter of wire

t_s = Surface temperature of wire

t_w = Temperature of water

33.

Ans. The heat flux at the surface is the rate at which radiant energy leaves the surface per unit area.

$$q = \frac{Q}{A}$$

$$= \sigma_b T^4 = 5.67 \times 10^{-8} (273 + 60)^4 = 697.2 \text{ W/m}^2$$

34.

Ans. From an energy balance,

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

or

$$\frac{Q_a}{Q_0} + \rho + \tau = 1$$

or

$$\frac{Q_a}{Q_0} = 1 - \rho - \tau$$

$\therefore$

$$\text{Incident flux } (Q_0) = \frac{Q_a}{1 - \rho - \tau}$$

35.

Ans. From an energy balance,

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

$\therefore$ Absorption

$$Q_a = \alpha Q_0$$

Transmission

$$Q_t = \tau Q_0$$

Reflection

$$Q_r = \rho Q_0$$



36.

Ans. The radiation heat loss from the glass window is given by

$$Q = \sigma_b A T^4 \times \tau$$

Where,

τ = Transmissivity of glass

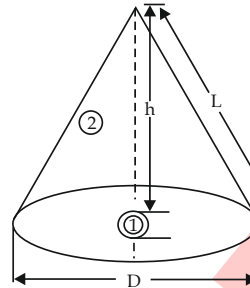
σ_b = Stefan boltzman constant = 5.67×10^{-8}

A = Area of surface

T = Temperature

37.

Ans.



$$F_{12} = 1; A_1 F_{12} = A_2 F_{21}$$

$$F_{21} = \frac{A_1}{A_2} \times 1 = \frac{\frac{\pi d^2}{4}}{\pi \times \frac{D}{2} \times L} = \frac{d^2}{2 \times D \sqrt{h^2 + \frac{D^2}{4}}} = \frac{d^2}{D \sqrt{4h^2 + D^2}}$$

38.

Ans. Dühring's Rule states that a linear relationship exists between the boiling point of a given solution and the boiling point of the pure solvent (usually water) at the same pressure.

39.

Ans. Extended surfaces, commonly known as fins, are significant in heat transfer primarily because they enhance the rate of heat transfer between a solid surface and a surrounding fluid.

40.

Ans. **Gray Body** : A grey body is a theoretical object that has an emissivity less than 1 but remains constant over all wavelengths of radiation. This means that while it does not emit radiation as perfectly as a black body (which has an emissivity of 1), its emissivity does not vary with the wavelength of the radiation it emits.

41.

Ans. The Colburn analogy is a theoretical relationship used in heat transfer and fluid mechanics to relate convective heat transfer to mass transfer.

It provides a way to express the Nusselt number (Nu), which is a dimensionless number representing the ratio of convective to conductive heat transfer, in terms of the Sherwood number (Sh), which represents the ratio of convective to diffusive mass transfer.

42.

Ans. Heat transfer $Q = \frac{\Delta T_{\text{overall}}}{R}$ Where $\Delta T = T_1 - T_2$

$R = \frac{L}{KA}$ - Thermal resistance of slab

L = Thickness of slab,

K = Thermal conductivity of slab,

A = Area

43.

Ans. Heat transfer $Q = \frac{\Delta T_{\text{overall}}}{R}$ Where, $\Delta T = T_1 - T_2$

$$R = \frac{1}{2\pi LK} \ln \left[\frac{r_2}{r_1} \right] \text{ thermal resistance of slab}$$

where,

L = Length of cylinder,

K = Thermal conductivity,

r_2 = Outer radius,

r_1 = inner radius

44.

Ans. Heat transfer by convection is given by Newton's law of cooling

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

Where,

A = Area exposed to heat transfer in m^2 ,

h = heat transfer coefficient in $\text{W}/\text{m}^2\text{K}$

T_s = Temperature of the surface in K,

T_∞ = Temperature of the fluid in K.

45.

Ans. Heat transfer $Q = \frac{\Delta T_{\text{overall}}}{R}$

Where,

$$\Delta T = T_a - T_b,$$

$$R = \frac{1}{2\pi L} \frac{1}{h_a r_1} + \frac{\ln \left[\frac{r_2}{r_1} \right]}{K_1} + \frac{\ln \left[\frac{r_1}{r_2} \right] L_2}{K_2} + \frac{1}{h_b r_3}$$

46.

Ans. It is defined as the ratio of product of inertia force and buoyancy force to the square of viscous force.

$$Gr = \frac{\text{Inertia force} \times \text{Buoyancy force}}{(\text{Viscous force})^2}$$

Stanton number is the ratio of nusselt number to the product of Reynolds number and prandtl number.

$$St = \frac{Nu}{Re \times Pr}$$

47.

Ans. If the fluid motion is produced due to change in density resulting from temperature gradients, the mode of heat transfer is said to be free or natural convection.

If the fluid motion is artificially created by means of an external force like a blower or fan, that type of heat transfer is known as forced convection.

48.

Ans. The thickness of the boundary layer has been defined as the distance from the surface at which the local velocity or temperature reaches 99% of the external velocity or temperature.



49.

Ans. The general heat transfer equation for internal heat flow in cylinder tube uses Newton's law of cooling.

$$Q = h.A.(T_w - T_B)$$

where,

h = Convecting heat transfer coefficient

A = Surface area of heat transfer

T_w = Wall temperature

T_B = Fluid bulk mean temperature

For detailed analysis we use dimensionless number like

$$Nu = 0.023 (Re)^{0.8} (Pr)^n$$

$n = 0.4$ for heating of fluids

$n = 0.3$ for cooling of fluids

50.

Ans. The types of heat exchangers are as follows

1. Direct contact heat exchangers
2. Indirect contact heat exchangers
3. Surface heat exchangers
4. Parallel flow heat exchangers
5. Counter flow heat exchangers
6. Cross flow heat exchangers
7. Shell and tube heat exchangers
8. Compact heat exchangers.

51.

Ans. In direct contact heat exchanger, the heat exchange takes place by direct mixing of hot and cold fluids.

52.

Ans. In this type of heat exchangers, the transfer of heat between two fluids could be carried out by transmission through a wall which separates the two fluids.

53.

Ans. In this type of heat exchangers, hot and cold fluids flow alternately through the same space. Examples: IC engines, gas turbines.

54.

Ans. This is the most common type of heat exchangers in which the hot and cold fluid do not come into direct contact with each other but are separated by a tube wall or a surface.

55.

Ans. In this type of heat exchanger, hot and cold fluids move in the same direction.
In this type of heat exchanger hot and cold fluids move in parallel but opposite directions.

56.

Ans. In this type of heat exchanger, one of the fluids move through a bundle of tubes enclosed by a shell. The other fluid is forced through the shell and it moves over the outside surface of the tubes.

57.

Ans. There are many special purpose heat exchangers called compact heat exchangers. They are generally employed when convective heat transfer coefficient associated with one of the fluids is much smaller than that associated with the other fluid.

58.

Ans. We know that the temperature difference between the hot and cold fluids in the heat exchanger varies from point in addition various modes of heat transfer are involved. Therefore based on concept of appropriate mean temperature difference, also called logarithmic mean temperature difference, also called logarithmic mean temperature difference, the total heat transfer rate in the heat exchanger is expressed as

$$Q = U A (\Delta T)_m$$

Where,

U = Overall heat transfer coefficient W/m^2K

A = Area m^2

$(\Delta T)_m$ = Logarithmic mean temperature difference.

59.

Ans. The heat exchanger effectiveness is defined as the ratio of actual heat transfer to the maximum possible heat transfer.

$$\text{Effectiveness } \varepsilon = \frac{\text{Actual heat transfer}}{\text{Maximum possible heat transfer}} = \frac{Q}{Q_{\max}}$$

60.

Ans. The emissive power is defined as the total amount of radiation emitted by a body per unit time and unit area. It is expressed in W/m^2 .

The energy emitted by the surface at a given length per unit time per unit area in all directions is known as monochromatic emissive power.

61.

Ans. Absorptivity is defined as the ratio between radiation absorbed and incident radiation.

Reflectivity is defined as the ratio of radiation reflected to the incident radiation.

Transmissivity is defined as the ratio of radiation transmitted to the incident radiation.

62.

Ans. Black body is an ideal surface having the following properties.

A black body absorbs all incident radiation, regardless of wave length and direction. For a prescribed temperature and wave length, no surface can emit more energy than black body.

If a body absorbs a definite percentage of incident radiation irrespective of their wave length, the body is known as gray body. The emissive power of a gray body is always less than that of the black body.

63.

Ans. Radiation shields constructed from low emissivity (high reflective) materials. It is used to reduce the net radiation transfer between two surfaces.

64.

Ans. It is defined as the total radiation incident upon a surface per unit time per unit area. It is expressed in W/m^2 .

It is used to indicate the total radiation leaving a surface per unit time per unit area. It is expressed in W/m^2 .

65.

Ans. The Dittus-Boelter equation is an empirical correlation widely significant in the field of heat transfer, specifically for calculating the convective heat transfer coefficient (and thus the Nusselt number) for fluids flowing in turbulent, fully developed flow inside smooth circular tubes.

66.

Ans. Counter current heat exchangers have several advantages worth knowing about. Undoubtedly, their significant advantages include high efficiency, wide application, low production costs, and reduced thermal stresses in the structure compared to concurrent exchangers.

67.

Ans. **According to Prevost theorem:-** All bodies at all temperatures emit thermal radiation except the body at $0\text{ K } (-273.15^\circ\text{C})$ in form of electromagnetic waves.

68.

Ans. It is a thermophysical property of the material which tells about the ability of the material that how fast a material can diffuse heat energy from the material as compared to storing it.

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

ρC_p = Heat capacity heat storage ability

Higher the thermal conductivity of material and lesser its heat capacity (storage ability) more will be the value of thermal diffusivity.



69.

Ans. Biot Number is the ratio of internal conduction resistance in the body to the external convection resistance from the surface.

$$\text{Biot Number} = \frac{\text{Internal Conduction Resistance}}{\text{External Convection Resistance}} = \frac{hL}{K}$$

where,

h = Convecting heat transfer coefficient

L = Characteristic length

K = Thermal conductivity

The significance of the Biot number lies in the following

- (i) Determines the suitability of applying lumped system analysis.
- (ii) Indicates the relative importance of conduction within a body compare to convection at its surface.
- (iii) Guides the selection of appropriate heat transfer analysis method.

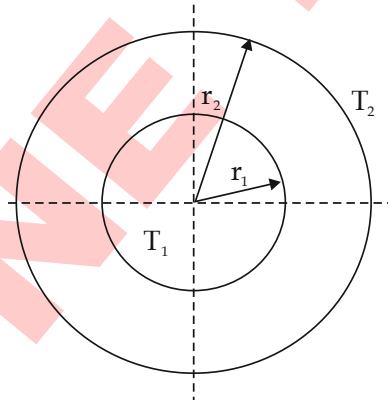
70.

Ans. The heat transfer by conduction through a thick sphere (Q) is given by

$$Q = \frac{4\pi k r_1 r_2 (T_1 - T_2)}{r_2 - r_1}$$

Note: In case of a composite sphere, heat transfer,

$$Q = \frac{4\pi(T_1 - T_2)}{\sum \frac{r_2 - r_1}{k r_1 r_2}}$$

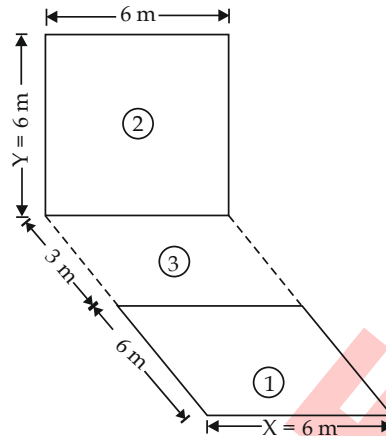


PART - B

5 Marks Questions

71.

Ans.



As per associative rule of shape factor,

$$F_{2(3,1)} = F_{23} + F_{21}$$

$$F_{23} = 0.15$$

... (i) (A_3 is hypothetical surface)

With $Y = 6$, $X = 6$, $Z = 3$, $Y/X = 1$, $Z/X = 0.5$

(From view factor figure for perpendicular rectangles with a common edge)

Similarly,

$$F_{2(3,1)} = 0.225 \text{ with } \frac{Y}{X} = 1, \frac{Z}{X} = \frac{3}{6} = 0.5$$

So from eq. (i)

$$F_{21} F_{2(3,1)} - F_{23} = 0.225 - 0.15 = 0.075$$

$\therefore$

$$A_1 F_{12} = A_2 F_{21}$$

or

$$F_{12} = \frac{A_2}{A_1} F_{21} = \frac{6 \times 6}{6 \times 6} \times 0.075 = 0.075$$

72.

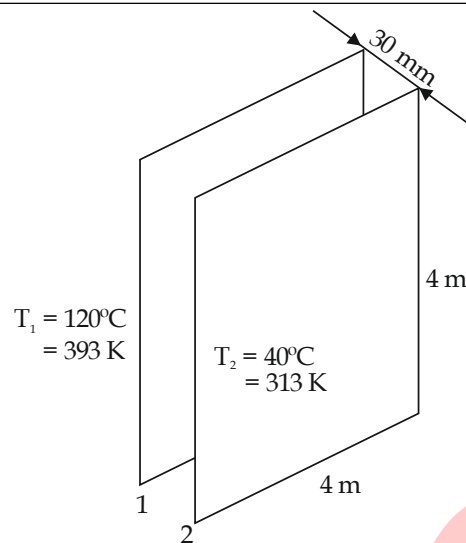
Ans. The gap between the plates is very small as compared to the size of plates.

$$\therefore \text{Effective resistance} = \frac{1}{\left(\frac{1}{\epsilon_1} + \frac{1}{\epsilon_2} - 1 \right)}$$

$\therefore$ Heat exchanged

$$\therefore Q_{12} = \frac{A\sigma(T_1^4 - T_2^4)}{\left(\frac{1}{\epsilon_1} + \frac{1}{\epsilon_2} - 1 \right)}$$





$$= \frac{4 \times 4 \times 5.67 \times 10^{-8} [393^4 - 313^4]}{\left(\frac{1}{0.736} + \frac{1}{0.736} - 1 \right)} = 5.2824 \times 10^{-7} [393^4 - 313^4]$$

$$= 7530.933 \text{ W} = 7.531 \text{ kW}$$

73.

Ans. Let the base be surface (1) and semicircle be surface (2)

Now as

$\Rightarrow$

Also

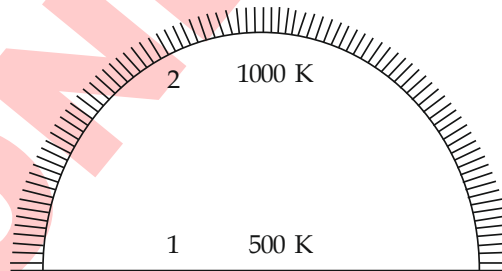
$$F_{11} + F_{12} = 1; \quad F_{21} + F_{22} = 1$$

$$F_{11} = 0$$

$$F_{12} = 1$$

$$A_1 F_{12} = A_2 F_{21} \text{ [Reciprocity rule]}$$

$$F_{21} = \frac{A_1}{A_2} \times F_{12} = \frac{D \times L}{\pi D L / 2} \times 1 = \frac{2}{\pi} = 0.6366$$



$$F_{22} = 1 - 0.6366 = 0.3634$$

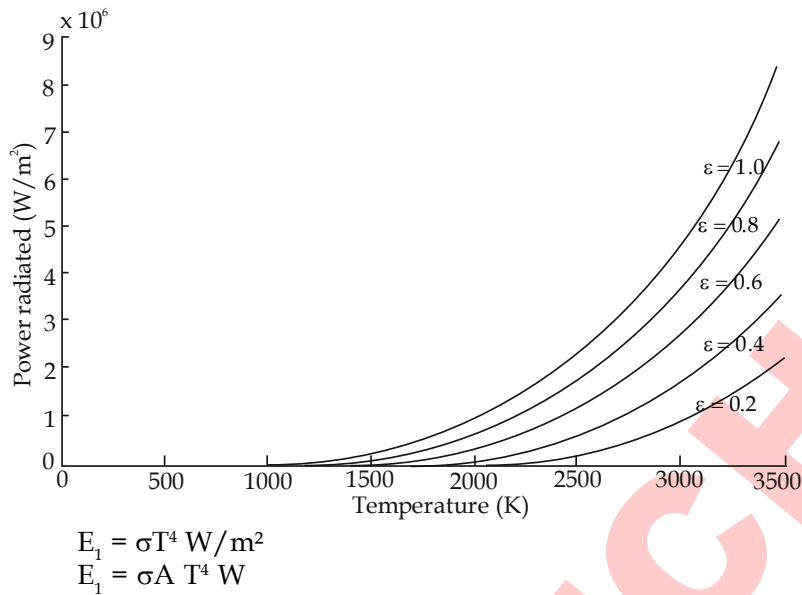
Now heat transfer rate

$$Q = A_1 F_{12} (\sigma_b) (T_1^4 - T_2^4) = D \times L \times 1 \times 5.67 \times 10^{-8} [1000^4 - 500^4]$$

$$\frac{Q}{L} = 53.16 \times 10^3 \text{ W/m.} \Rightarrow 53.16 \text{ D kW/m}$$

74.

- Ans.
- The total radiant heat power emitted from a surface is proportional to the fourth power of its absolute temperature.
 - The emissive power of a black body, according to Stefan-Boltzmann, is proportional to absolute temperature to the fourth power



where,

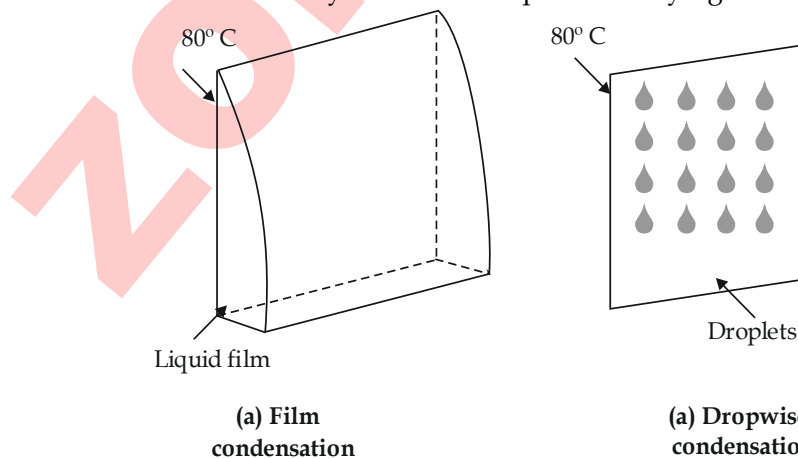
σ = Stefan-Boltzmann constant = $5.67 \times 10^{-8} \text{ W/m}^2 \text{ K}^4$

Key significance of Stefan Boltzmann law in engineering:

- (i) Radiative heat transfer calculation
- (ii) Furnace and Boiler design
- (iii) Insulation and Thermal management
- (iv) Thermal imaging and Pyrometry
- (v) Solar energy system
- (vi) Aerospace and Defence
- (vii) Material science

75.

Ans. In film condensation, the condensate wets the surface and forms a liquid film on the surface that slides down under the influence of gravity. The thickness of the liquid film increases in the flow direction as more vapor condenses on the film. This is how condensation normally occurs in practice. In drop wise condensation, the condensed vapor forms droplets on the surface instead of continuous film, and the surface is covered by countless droplets of varying diameters.

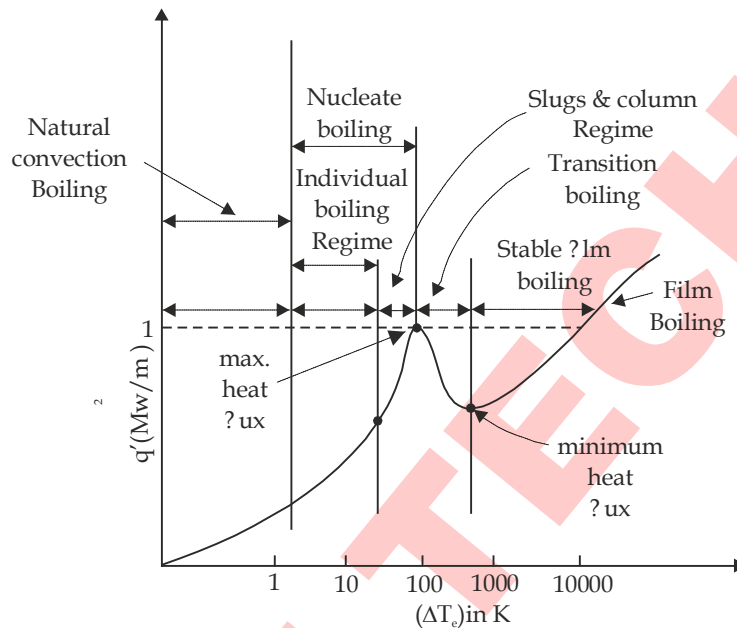


In film condensation, the surface is blanketed by a liquid film of increasing thickness, and this "liquid wall" between solid surface and the vapor serves as a resistance to heat transfer. The heat of vaporization h_{fg} released as the vapor condenses must pass through this resistance before it can reach the solid surface and be transferred to the medium on the other side. In dropwise condensation, however, the droplets slide down when they reach a certain size, clearing the surface and exposing it to vapor.

There is no liquid film in this case to resist heat transfer. As a result, heat transfer rates that are more than 10 times larger than those associated with film condensation can be achieved with dropwise condensation. Therefore, dropwise condensation is the preferred mode of condensation in heat transfer applications.

76.

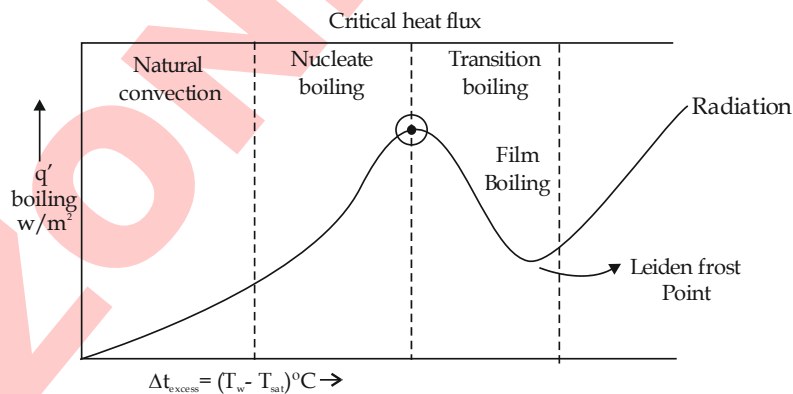
Ans.



Typical boiling curve for pool boiling of water at saturation temperature and atmospheric pressure.

77.

Ans. Typical boiling curve for water



78.

Ans. The Colburn analogy is a theoretical relationship used in heat transfer and fluid mechanics to relate convective heat transfer to mass transfer. It provides a way to express the Nusselt number (Nu), which is a dimensionless number representing the ratio of convective to conductive heat transfer, in terms of the Sherwood number (Sh), which represents the ratio of convective to diffusive mass transfer.

Key Concepts:

1. **Nusselt Number (Nu):**

Defined as:

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

where,

h is the convective heat transfer coefficient,

L is a characteristic length, and

K is the thermal conductivity of the fluid.

Represents the enhancement of heat transfer through convection compared to conduction.

2. **Sherwood Number (Sh):**

Defined as:

$$Sh = \frac{K_L L}{D}$$

where,

K_L is the mass transfer coefficient,

D is the diffusivity of the species being transferred, and

L is again a characteristic length.

Represents the enhancement of mass transfer through convection compared to diffusion.

Colburn Analogy:

The Colburn analogy states that the Nusselt number and the Sherwood number are proportional to each other, typically expressed as:

$$\frac{Nu}{Pr^{1/3}} = \frac{Sh}{Sc^{1/3}}$$

where:

Pr is the Prandtl number, defined as $Pr = \frac{v}{\alpha} = \frac{c_p \mu}{K}$,

where,

v is the kinematic viscosity,

α is the thermal diffusivity,

C_p is the specific heat, and

μ is the dynamic viscosity.

Sc is the Schmidt number, defined as $Sc = \frac{v}{D}$.

79.

Ans. Given data:

$$T_m = 315 \text{ K};$$

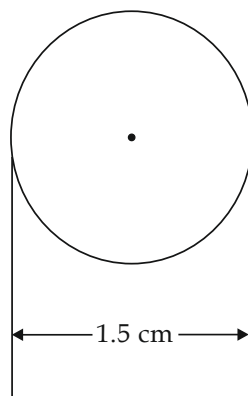
$$T_s = 385 \text{ K};$$

$$T_\infty = 293 \text{ K}$$

Grashoffs number,

$$\begin{aligned} Gr &= \frac{L^3 \beta g \Delta T}{v^2} \frac{D^3 \beta g \Delta T}{v^2} \\ &= \frac{(1.5 \times 10^{-2})^3 \times 2.86 \times 10^{-3} \times 9.81 \times (385 - 315)}{(2.076 \times 10^{-5})^2} \\ &= 15379.8 \end{aligned}$$





$$\overline{Nu} = 2 + 0.43(Gr \times Pr)^{1/4} = 2 + 0.4(15379.8 \times 0.697)^{1/4} = 6.375$$

$$\frac{hd}{k} = 6.375$$

$$h = \frac{6.375 \times 0.03}{1.5 \times 10^{-2}} = 12.75 \text{ W/m}^2\text{K}$$

$$Q = (4\pi r^2) \times h \times (\Delta T) = (4\pi \times (0.75 \times 10^{-2})^2) \times 12.75 \times (385 - 315) = 0.631 \text{ W}$$

80.

Ans.

- The lumped thermal capacity model is the simplest transient (unsteady) heat conduction approach.
 - In this model, the temperature of the solid body is a function of the time only, which means that the temperature is assumed to be spatially independent (uniform).
 - Lumped system analysis is exact when $Bi = 0$ and approximate when $Bi > 0$.
 - It is generally accepted that lumped system analysis is applicable if $Bi \leq 0.1$
- (Heat transfer into the body during dt) = (The increase in the energy of the body during dt)

$$hA_s (T_\infty - T) dt = mc_p dT$$

81.

Ans.

Nusselt Number : The ratio of convection heat transfer to conduction heat transfer under the same conditions.	$\frac{q_{\text{conv}}}{q_{\text{cond}}} = \frac{h\Delta T}{k \frac{\Delta T}{L}} = \frac{hL}{k} = Nu_L$
Prandtl Number : The ratio of momentum diffusivity (kinematic viscosity) to thermal diffusivity.	$Pr = \frac{\nu}{\alpha}$ Pr = Prandtl Number ν = momentum of diffusivity α = thermal diffusivity
Grashof Number : Ratio of buoyant forces to viscous forces; used to determine the flow regime in natural convection.	$Gr = \frac{\text{Buoyancy force}}{\text{Viscous force}} \quad Gr = \frac{g\beta(T_w - T_\infty)X^3}{\nu^2}$

Significance of Nusselt Number: Measure the enhancement of heat transfer by convection relative to pure conduction across the same fluid layer.

Significance of Prandtl Number: Relates the relative thickness of the momentum boundary layer to the thermal boundary layer.

Significance of Grashof Number: Quantifies the ratio of buoyancy forces to viscous forces in natural (free) convection.

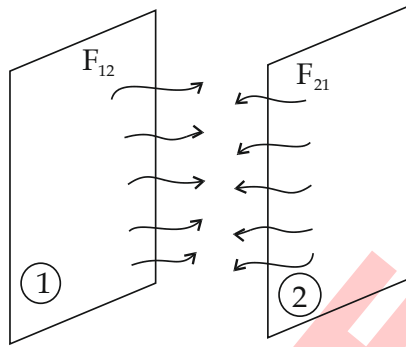


82.

- Ans.**
- A shape factor is a measurement of the direct exchange of heat energy between two surfaces.
 - The surfaces can be a rectangle, a circle, a cylinder, a cone, or some other basic geometric configurations.
 - The surfaces can be placed in parallel, in perpendicular, or inclining to each other at a specified angle.
 - The shape factor is denoted by the symbol F_{ij} , where i indicates radiating surface and j indicates receiving surface.

F_{12} = Fraction of radiation energy leaving surface 1 that reaches surface 2

F_{21} = Fraction of radiation energy leaving surface 2 that reaches surface 1



83.

- Ans.** Convective heat transfer coefficient,

$$h = 2.5 (\Delta T)^{0.25} = 2.5 (60-20)^{0.25} = 6.287 \text{ W/m}^2 \text{ K}$$

Heat lost by convection = $h A \Delta t$

$$= 6.287 \times 0.25 \times (60 - 20) = 62.87 \text{ W}$$

Heat lost by convection as fraction of heat supplied,

$$= \frac{62.87}{120} = 0.5239 \text{ or } 52.39\%$$

The remaining 47.61% would be lost to the surroundings by radiation.

84.

- Ans.** For steady state conditions, the electric power supplied to the rod equals the radiant heat loss from it. Further, since the walls of the furnace completely enclose the heating rod, all the radiant energy emitted by the surface of the rod is intercepted by the furnace walls. Thus

$$\begin{aligned} Q &= \sigma_b A \varepsilon (T_1^4 - T_2^4) \\ &= \sigma_b (\pi dl) \varepsilon (T_1^4 - T_2^4) \\ &= 5.67 \times 10^{-8} (\pi \times 0.02 \times 1.5) (0.9) (1000^4 - 800^4) \\ &= 5.67 (\pi \times 0.02 \times 1.5) \left[\left(\frac{1000}{100} \right)^4 - \left(\frac{800}{100} \right)^4 \right] \\ &= 5.67 (\pi \times 0.02 \times 1.5) (0.9) (10000 - 4096) = 2838 \text{ W} \end{aligned}$$

Thus the rate of electrical input to the rod must equal 2838 W.

85.

- Ans.**
- The Biot number is the ratio of the internal resistance of a body to heat conduction to its external resistance to heat convection.
 - **Physical Significance:** The Biot number is more likely to be larger for poorly conducting solids since such bodies have larger resistances against heat conduction.

$$Bi = \frac{\text{Internal conductive resistance}}{\text{External convective resistance}}$$

$$Bi = \frac{hL}{K}$$

h = heat transfer co-efficient ($\text{W}/\text{m}^2\text{K}$)

L = characteristic length

K = Thermal conductivity

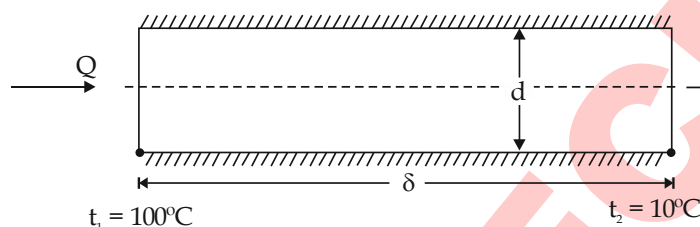
86.

Ans. Invoking Fourier law of heat conduction, the heat flow through the rod is given by

$$Q = \frac{k A_c (t_1 - t_2)}{\delta}$$

$$6 = \frac{k \left(\frac{\pi}{4} \times 0.03^2 \right) (100 - 10)}{0.2}$$

$$= 0.318 k$$

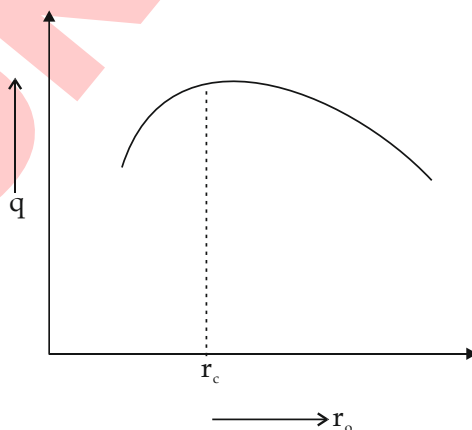


$\therefore$ Thermal conductivity of the rod material,

$$k = \frac{6}{0.318} = 18.87 \text{ W/M-deg}$$

87.

- Ans.**
- In a plane wall thicker the insulation, the lower the heat transfer rate. This is due to the fact the outer surface have always the same area.
 - But in cylindrical and spherical coordinates, the addition of insulation also increases the outer surface area which decreases the convection resistance at the outer surface.
 - Moreover, in some cases, a decrease in the convection resistance due to the increase in surface area can be more significant than an increase in conduction resistance due to thicker insulation.
 - As a result, the total resistance may actually decrease resulting in increased heat flow.



Thickness up to which heat flow increases and beyond which heat flow decreases is termed as critical thickness. For cylinders and spheres, it is the critical radius (r_c).

It can be derived that the critical radius of insulation depends on the thermal conductivity of the insulation k and the external convection heat transfer coefficient h .

For cylinder

$$r_c = \frac{k}{h}$$

For sphere

$$r_c = \frac{2k}{h}$$

88.

Ans. Conduction is the transfer of thermal energy between objects or within an object through direct contact, without any bulk movement of the material itself. This occurs due to the vibration and collision of particles (atoms, molecules, or free electrons) within the material, transferring energy from hotter regions to colder regions.

Fourier's law of heat conduction states that the rate of heat transfer through a material is proportional to the negative of the temperature gradient and the area through which the heat is transferred. Mathematically, it is expressed as:

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

Where:

- Q is the rate of heat transfer (in Watts or J/s)
- k is the thermal conductivity of the material (in $W/(m \cdot K)$)
- A is the cross-sectional area through which heat is flowing (in m^2)
- $\frac{dT}{dx}$ is the temperature gradient (change in temperature with dx respect to distance) (in K/m)

The negative sign in Fourier's law is used to indicate that heat flows in the direction of decreasing temperature. In other words, heat always flows from a region of higher temperature to a region of lower temperature, which is the opposite direction of the positive temperature gradient.

89.

Ans. The rate of heat transfer by convection, $Q_c = h_c A (\Delta T)$,

where $h_c = 2.512 (59-20)^{1/4} = 6.277 \text{ W/m}^2\text{K}$

$\therefore Q_c = 6.277 \times 0.2 \times 39 = 48.96 \text{ W}$

Fraction of the supplied heat lost by convection is $48.96/100$ or 0.4896 or 48.96 or 49% .

The remaining 51% heat transfer occurs by radiation.

90.

Ans. The rate of heat transfer through the gas in the cylinder is given by

$$Q_k = \frac{2\pi kL(T_2 - T_1)}{\ln(r_2/r_1)}$$

$$= \frac{2\pi \times 0.0251 \times 0.25(750 - 250)}{\ln(25/0.25)}$$

$$= 4.28 \text{ W Ans.}$$

Heat lost by radiation

$$Q = \sigma A_1 (T_1^4 - T_2^4)$$

$$= 5.67 \times 10^{-8} \times 2\pi \times 0.25 \times 10^{-3} \times 0.25 (750^4 - 250^4)$$

$$= 6.958 \text{ or } 7 \text{ W. Ans.}$$

91.

Ans. Total radiation from the sun

$$Q_r = 1.4 \times 4\pi R^2$$

where R is the radius of the earth's orbit.

Total radiation emitted by the sun

$$Q = \sigma 4\pi^2 T^4$$

where r is the radius of the sun and T is the surface temperature of the sun. Therefore,



$$\sigma 4\pi r^2 T^4 = 1.4 \times 4\pi R^2$$

$$\therefore T^4 = \frac{1.4 \times 10^3}{5.67 \times 10^{-8}} \times (216)^2 = 0.1152 \times 10^{16} \text{ K}^4$$

$$\therefore T = 5826 \text{ K Ans.}$$

92.

Ans. Convective heat flux to the surface

$$q = \frac{Q}{A} = h(T_{s,0} - T_{\infty})$$

$$= 10(16 - 20) = -40 \text{ W/m}^2 \text{ Ans.}$$

Since this must be equal to be heat conducted through the sides,

$$q = -k \frac{dT}{dx} = -k \frac{T_{s,0} - T_{s,i}}{L}$$

$$\therefore T_{s,i} = \frac{qL}{k} + T_{s,0} = -\frac{40 \times 0.03}{0.1} + 16$$

$$= 4^\circ\text{C Ans.}$$

93.

Ans. Solid

In solids, conduction mechanism works in two ways.

- Lattice vibration
- Transport of free electron

Liquid and Gas

In liquid and gases, conduction mechanism works in two different ways.

- Collision
- Diffusion

Gas

For gases, conduction mechanism is simple. The kinetic energy of molecules is the function of temperature. The kinetic energy is the result of random translation motion as well as vibrational and rotational motion. Molecules of gases are in continuous random motion which exchanges energy and momentum. When one molecule from high-temperature region collides with another molecule from the low-temperature region, it loses energy by collision.

Liquid

In liquids, conduction mechanism is similar to that for gases. Compared to gases, molecules in liquids are closed space. In liquids, due to closely spaced molecules, the intermolecular force comes into play.

94.

Ans. Heat flow has an analogy in the flow of electricity.

Ohm's law states that the current 'I' flowing through a wire is equal to the voltage potential ($E_1 - E_2$) divided by the electrical resistance R_e .

$$I = \frac{E_1 - E_2}{R_e}$$

Since the temperature difference and heat flux in conduction are similar to the potential difference and electrical current respectively, the rate of heat conduction through the wall can be written as

$$Q = \frac{T_1 - T_2}{L/kA} = \frac{T_1 - T_2}{R_{th}}$$

Where $R_{th} = \frac{L}{kA}$ is the conductive thermal resistance to heat flow offered by the wall.

Fourier's law of heat conduction is analogous to Ohm's law for electrical circuits. In the analogy:

- the heat flow (Q) corresponds to the electrical current (I)
- the thermal resistance to the electrical resistance

- temperature (T) to the electrical voltage (V)
- thermal conductivity to the electrical conductivity
- heat capacity to the capacitance

95.

Ans. Thermal Contact Resistance (TCR) is the resistance to heat flow across the interface between two materials in contact. It is a critical factor in thermal management applications, affecting the efficiency of heat transfer in systems such as electronics, heat exchangers, and thermal insulators.

Causes of Thermal Contact Resistance

1. **Surface Roughness:** The microscopic surface texture of materials can create gaps at the interface, reducing the contact area and impeding heat transfer.
2. **Material Properties:** Different materials have varying thermal conductivities. Lower thermal conductivity at the interface can increase resistance.
3. **Interfacial Material:** The presence of contaminants (like dust, oils, or oxidation) can form a layer that further increases resistance.
4. **Contact Pressure:** Insufficient pressure at the interface can lead to larger gaps and increased resistance. Higher pressure can improve contact but may also lead to deformation.
5. **Temperature:** TCR can vary with temperature due to changes in material properties and thermal expansion.

Effects of Thermal Contact Resistance

1. **Heat Transfer Efficiency:** Increased TCR reduces the efficiency of heat transfer, leading to overheating in electronic components or other systems.
2. **Temperature Gradient:** A higher TCR can create significant temperature differences across the interface, potentially affecting the performance and reliability of devices.
3. **Thermal Runaway:** In critical applications, insufficient heat dissipation due to high TCR can lead to thermal runaway, damaging components.
4. **System Performance:** Overall performance of thermal management systems can be compromised, leading to increased energy consumption and reduced lifespan of components.

96.

Ans. Thermal Conductivity

- **Definition :** Thermal conductivity (k) is a material property that measures a material's ability to conduct heat. It is defined as the amount of heat (Q) that passes through a unit area (A) of a material with a temperature gradient (dT/dx) over a specific time (t).
- **Units :** The SI unit of thermal conductivity is watts per meter-kelvin (W/m·K).
- **Formula :** The relationship can be expressed as:

$$Q = k \cdot A \cdot \frac{dT}{dx} \cdot t$$

- **Context :** It is an intrinsic property of the material, meaning it does not change with the size or shape of the material.

Thermal Conductance

- **Definition :** Thermal conductance (C) measures how well a specific geometry (like a wall, a layer, or a component) conducts heat. It takes into account the material's thermal conductivity as well as its thickness and area.
- **Units :** The SI unit of thermal conductance is watts per meter squared-kelvin (W/m²K).
- **Formula :** It can be expressed as:

$$C = \frac{kA}{d}$$

where, d is the thickness of the material.

- **Context :** It is an extrinsic property, meaning it depends on the geometry and dimensions of the material in addition to its thermal conductivity.

Summary

- **Thermal Conductivity:** Material property (W/m·K).
- **Thermal Conductance:** Geometry-dependent property (W/m²K).



97.

Ans. The table below shows the difference between natural and forced convection.

Characteristic	Natural Convection	Forced Convection
Driving Mechanism	Natural density and temperature gradients drive fluid motion.	External sources (e.g., fans or pumps) induce fluid motion.
Example	Boiling water, rising smoke, or hot air rising.	Cooling fans, exhaust fans, or pumps in HVAC systems.
Rate of Heat Transfer	Typically lower due to natural, unassisted movement.	Generally higher due to external equipment assistance.
Control and Regulation	Limited control, primarily dependent on ambient conditions.	Highly controllable and adjustable with external devices.
Overall Heat Transfer Coefficient	Lower compared to forced convection.	Higher compared to free convection.

98.

Ans. **Natural Convection:**

- **Velocity Profile:** In natural convection, fluid motion is driven by buoyancy forces arising from density differences caused by temperature gradients. Near a heated surface, the fluid becomes less dense and rises, while cooler, denser fluid descends. This creates a boundary layer where the velocity starts from zero at the surface, increases to a maximum away from the surface, and then gradually decreases to zero in the free stream. The profile is typically bell-shaped or parabolic-like.
- **Temperature Profile:** The temperature profile in natural convection shows a steep gradient near the heated surface, where heat is being transferred to the fluid. As you move away from the surface, the temperature gradually approaches the ambient temperature of the surrounding fluid.

Forced Convection:

- **Velocity Profile:** In forced convection, fluid motion is externally imposed (e.g., by a pump or fan). The velocity profile depends on the flow regime (laminar or turbulent) and the geometry of the flow path. For laminar flow over a flat plate, the velocity starts from zero at the surface and increases to a maximum in the free stream. For flow in a pipe, the velocity is zero at the walls and parabolic in the center for laminar flow, or relatively flat in the core with steep gradients near the walls for turbulent flow.
- **Temperature Profile:** The temperature profile in forced convection is also influenced by the imposed flow. Heat is transferred from the surface to the moving fluid, and the temperature gradient is typically concentrated within a thermal boundary layer near the surface. The temperature of the bulk fluid changes as it flows, and the profile reflects this change, often approaching the surface temperature as the fluid absorbs heat.

99.

Ans. Under the stipulations of one-dimensional steady state heat conduction, the electrical energy dissipation rate within the oven must equal the conduction heat transfer rate across the wall. That is:

$$Q = \frac{kA(t_1 - t_2)}{\delta}$$

$$36.5 = \frac{0.04A(420 - 310)}{0.75} = 5.87A$$

Hence, the required wall surface area,

$$A = \frac{36.5}{5.87} = 6.218 \text{ m}^2$$



100.

Ans. The temperature gradient in the direction of heat flow is

$$\frac{dt}{dx} = \frac{t_2 - t_1}{\delta} = \frac{30 - 100}{0.1} = -700^\circ\text{C} / \text{m}$$

Heat flow across the wall is given by Fourier's heat conduction equation

$$Q = -kA \frac{dt}{dx} = -8.4 \times 3 \times (-700) \\ = 17850 \text{ W or } 17.85 \text{ kW}$$

101.

Ans. (i) From Stefan-Boltzman law, the rate of energy emission E_b is :

$$E_b = \sigma_b AT^4 \\ = (5.67 \times 10^{-8}) \times 0.2 \times 800^4 = 4644.86 \text{ W}$$

(ii) The intensity of normal radiation is

$$I_n = \frac{\sigma_b T^4}{\pi} \\ = \frac{5.67 \times 10^{-8} \times 800^4}{\pi} = 7396.28 \text{ W/m}^2$$

(iii) From Lambert's cosine law :

$$I_\phi = I_n \cos \phi \\ = 7396.28 \times \cos 60^\circ = 3698.14 \text{ W/m}^2 \text{ steradian}$$

(iv) From Wien's displacement law, the wavelength $\lambda_{\max}$ for maximum monochromatic emissive power is :

$$\lambda_{\max} = \frac{2.898 \times 10^{-3}}{T} = \frac{2.898 \times 10^{-3}}{800} = 3625 \times 10^{-6} \text{ m}$$

102.

Ans. The Graetz number (Gz) is a dimensionless number used in fluid dynamics and heat transfer, primarily characterizing laminar flow and heat transfer in a conduit, such as a pipe. It is named after the German physicist Leo Graetz.

Key Aspects of the Graetz Number:

- **Definition:** It's defined as a ratio of the thermal energy transported by the bulk fluid flow to the thermal energy conducted radially within the fluid.
- **Formula:** For flow in a circular pipe, it can be expressed as:

$$Gz = \frac{(\text{Diameter}) \cdot \text{Reynolds Number} \cdot \text{Prandtl Number}}{\text{Length}} \text{ or } Gz = \frac{D}{L} \text{RePr.}$$

It can also be defined as $Gz = \frac{UD^2}{\kappa x}$ where U is fluid velocity, D is pipe diameter, κ is thermal diffusivity, and x is axial distance.

- **Significance:** It helps determine the thermally developing flow entrance length in ducts.

Interpretation:

- **Small Gz ($Gz < 20$):** Indicates that radial temperature profiles are fully developed, meaning the temperature distribution across the pipe's cross-section has reached a stable profile.
- **Large Gz:** Thermal boundary layer development needs to be considered, meaning the temperature profile is still evolving along the length of the pipe.
- **Applications:** It is critically important in analyzing heat transfer in various applications like chemical reactors, heat exchangers, and blood flow.



103.

Ans. Definition of Condensation:

The transformation of water vapour into liquid is known as condensation. It's the polar opposite of vapourisation, in which liquid water turns into a gas.

Forms of Condensation:

The temperature at which the dew point is reached can be used to classify the different types of condensation. Condensation can occur when the dew point is below freezing or 0°C , and it can also occur when the dew point is above freezing.

The types of condensation can be divided into two groups in this way:

- (1) When the dew point falls below freezing, frost, snow, and certain clouds occur.
- (2) When the dew point rises above freezing, dew, mist, fog, smog, and some clouds form.

Condensation can also be characterised by where it occurs, such as at or near the earth's surface or in free air. The first category includes dew, white frost, fog, and mist, whereas the second category includes clouds.

1. **Dew:**

- Forms when water vapor condenses into liquid droplets on cool surfaces near the ground, like grass or leaves, when the surface temperature drops below the dew point, which is above freezing.

2. **Frost (White Frost):**

- Forms on cold surfaces when the dew point is at or below freezing (0°C), causing water vapor to directly convert into minute ice crystals instead of liquid water droplets.

3. **Fog:**

- A cloud that forms near the Earth's surface when water vapor condenses around fine dust particles or other condensation nuclei in the air, significantly reducing visibility (typically less than 1 km).

4. **Mist:**

- Similar to fog but less dense, with visibility generally ranging between 1 and 2 km.

5. **Clouds:**

- Masses of minute water droplets or tiny ice crystals formed by the condensation of water vapor in free air at considerable elevations.
- Classified into various types (e.g., cirrus, cumulus, stratus, nimbus) based on their height, expanse, density, and appearance.

104.

Ans. The Number of Transfer Units (NTU) Method is used to calculate the rate of heat transfer in heat exchangers (especially counter current exchangers) when there is insufficient information to calculate the LogMean Temperature Difference (LMTD).

In heat exchanger analysis, if the fluid inlet and outlet temperatures are specified or can be determined by simple energy balance, the LMTD method can be used; but when these temperatures are not available The NTU or The Effectiveness method is used.

Difficulties with the LMTD method -

- From the design equation:

$$Q = UA\Delta T_{lm}$$
- From energy balance:

$$Q = m_c C_{pc} (T_{co} - T_{ci}) = m_h C_{ph} (T_{hi} - T_{ho})$$
- The LMTD method of heat exchanger design is difficult to use if we want to predict the performance of a heat exchanger.
- Here we would know: m_c , m_h , T_{ci} , T_{hi} , U , and A , however, we would not know:
 T_{co} or T_{ho}
- Hence, we cannot find: Q or ΔT_{lm}

To solve the above problem with the usual LMTD method:

1. We could guess a value for T_{ho} or T_{co} , find Q from a heat balance, and then from $UA\Delta T_{lm}$.
2. Using this Q value, find new values of T_{ho} and T_{co} .
3. We would need to progressively alter our guesses until the first and second-step values of T were equal.

This iterative method can readily be done by computer, but a direct method can also be used. This direct method is known as the Effectiveness-NTU method.



PART - C

20 Marks Questions

105.

Ans. Overall heat transfer coefficient (OHTC):

- OHTC is defined as the quantity of rate of heat transfer takes place due to any mode per unit area for unit temperature difference.
- It doesn't have any physical meaning, it's an experimentally determined coefficient value.
- The concept of OHTC is valid for a steady-state.
- In the case of a heat exchanger, there is a combined mode of heat transfer (conduction and convection) or multiple resistances present therefore we use,

$$Q = UA\Delta T \Rightarrow UA = \frac{1}{R_{\text{Total}}},$$

where R_{Total} is total resistance and U = Overall heat transfer coefficient

- Unit of 'U' is $\frac{W}{m^2K}$
- U indicates the quantity of rate of heat transfer takes place due to any mode per unit area for the unit temperature difference.

Individual Heat Transfer Coefficient (h):

- The individual heat transfer coefficient, denoted as 'h', represents the rate of heat transfer per unit area per unit temperature difference for a specific heat transfer mechanism at a particular surface or interface.
- It quantifies the efficiency of heat transfer for a single process, like heat transfer between a solid surface and a fluid (convection), or within a specific layer of material (conduction).
- For example, in a heat exchanger, there would be individual heat transfer coefficients for the hot fluid film (convection), the tube wall (conduction), and the cold fluid film (convection).
- The units of 'h' are typically $W/(m^2K)$ or $BTU/(h \cdot ft^2 \cdot ^\circ F)$.

Relationship:

The relationship between the overall heat transfer coefficient (U) and individual heat transfer coefficients (h) depends on the geometry and configuration of the system. For example, for heat transfer across a plane wall with two fluid films on either side, and potentially a fouling layer, the relationship is expressed as:

$$1/U = 1/h_i + R_w + 1/h_o + R_f$$

Where:

U = Overall heat transfer coefficient.

h_i = Individual heat transfer coefficient for the inside fluid film.

R_w = Thermal resistance of the wall (calculated as thickness divided by thermal conductivity).

h_o = Individual heat transfer coefficient for the outside fluid film.

R_f = Fouling resistance.

In essence, individual heat transfer coefficients (h) characterize heat transfer at specific points or layers, while the overall heat transfer coefficient (U) accounts for the combined effect of all resistances encountered in the entire heat transfer process.

106.

Ans. The correction factor, F , accounts for the less efficient heat exchange caused by these more complex flow arrangements. The actual mean temperature difference is calculated by multiplying the ideal LMTD by this correction factor:

$$\Delta T_{\text{mean}} = F \times (\text{LMTD})_{\text{counter-current}}$$

Reasons for adopting the LMTD correction factor

- **Adjusts for complex flow paths:** Most industrial heat exchangers use multiple passes (e.g., one shell pass, two tube passes) or cross-flow configurations to increase the heat transfer area in a compact



space. These configurations are a mixture of counter-current and co-current flows, and the standard LMTD formula does not apply directly.

- **Maintains accuracy:** Without the correction factor, calculating the heat transfer rate for complex flow patterns would be difficult and inaccurate. The correction factor allows engineers to use the straightforward LMTD method and adjust it for the specific, non-ideal flow arrangement.
- **Accounts for reduced effectiveness:** For a given inlet temperature and flow rate, a complex, multi-pass design is less effective than a pure counter-current heat exchanger of the same size. The correction factor is always less than 1 (except for pure counter-flow or when a phase change occurs) and quantifies this reduced performance.
- **Simplifies design and analysis:** Calculating the precise, point-by-point average temperature difference in a complex heat exchanger would be mathematically cumbersome. Correction factor charts, standardized by bodies like TEMA, simplify the process by providing pre-calculated values based on the number of shell and tube passes.
- **Helps avoid temperature crossover:** An LMTD correction factor (F) below 0.75 indicates a particularly inefficient design that is sensitive to temperature changes. It suggests the design may lead to a "temperature crossover," where the outlet temperature of the cold fluid is higher than the outlet temperature of the hot fluid, which is only possible in counter-flow and indicates the design should be avoided.

107.

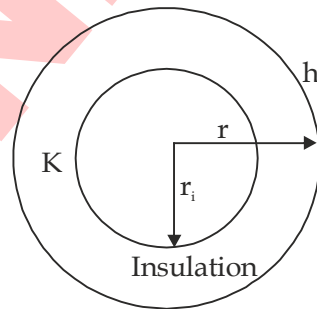
Ans. When insulating material is employed on the heat dissipating surface to reduce the dissipation, it will increase heat conductance resistance but at the same time reduce the convective resistance. So there is a conflicting situation and there exist a situation in which heat dissipation is maximum and the thickness of insulation in this condition is called critical thickness.

Assume a cylindrical surface

Total resistance offered by the insulating material

$$R_{th} = \frac{\ln(r/r_i)}{2\pi K \times 1} + \frac{1}{h \times 2\pi r \times 1} \quad \dots(i)$$

For critical dissipation $\frac{dR_{th}}{dr} = \frac{1}{2\pi K} \times \frac{r_i}{r} \times \frac{1}{r_i} - \frac{1}{2\pi hr^2} = 0$



$$\frac{1}{2\pi r} \left[\frac{1}{K} - \frac{1}{h \times r} \right] = 0$$

$$r_{critical} = \left(\frac{K}{h} \right) \text{ This is the radius of critical insulation.}$$

$$\therefore \frac{d^2 R_{th}}{dr^2} = \text{positive. So } R_{th} \text{ is min. and Heat dissipation is max.}$$

Note: For spherical surface Critical $r_{critical} = \frac{2K}{h}$

For application purposes, two cases arise one where the aim is to increase heat dissipation and other

where aim is to reduce heat dissipation. So

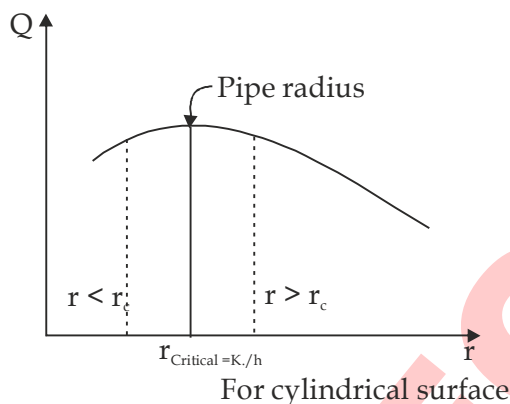
1. Maximum heat dissipation

- (a) Current carrying wire
- (b) Fuel rod in nuclear reactor
- (c) Heating coil in heater of A/c systems

2. Minimum heat dissipation

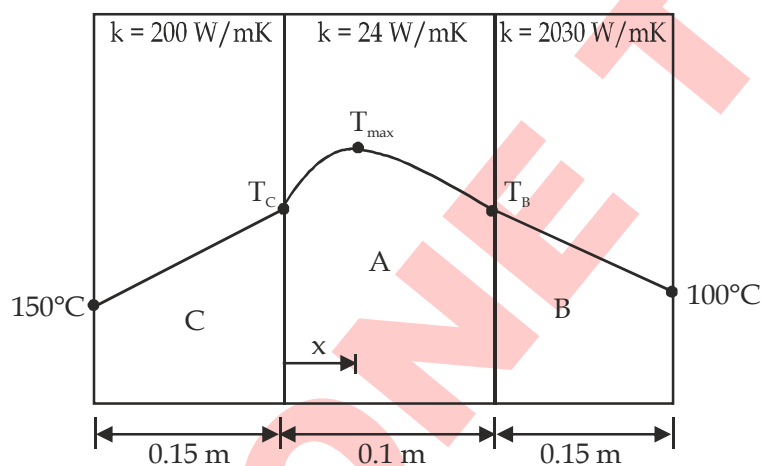
- (a) Cold fluid or gas storage
- (b) Refrigerant carrying line in air conditioning system

Graphical representation :



108.

Ans.



Metal wall C has more heat resistance than metal wall B (since $K_C < K_B$), therefore maximum temperature will shift toward wall C.

$\dot{Q}$ = Heat generated in rectangular block/per unit volume

q_c'' = Heat flux to metal wall C

q_b'' = Heat flux to metal wall B

$\dot{Q} \times l = q_c'' + q_b''$ (for unit cross sectional area)

$$2.5 \times 10^3 \times 0.1 = \frac{T_c - 150}{\left(\frac{0.15}{200}\right)} + \frac{T_b - 100}{\left(\frac{0.15}{2030}\right)} \quad \dots(i)$$

equation of steady state internal heat generation is

$$\frac{d^2 T}{dx^2} + \frac{\dot{Q}}{K} = 0$$

$$\frac{dT}{dx} = -\frac{\dot{Q}}{K}x + C_1$$

Boundary condition $x = 0; T = T_B$
 $x = L; T = T_C$

After putting boundary condition in equation (ii) we get three unknown i.e. T, T_B, T_C we have two equation i.e., equation (i) and (ii) therefore it can not be solve. Some data is missing.

109.

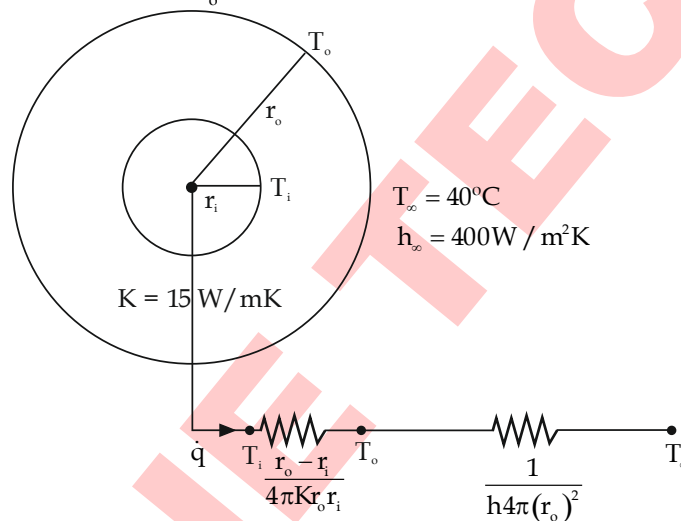
Ans. Assume:

- (i) Steady state condition,
- (ii) One dimensional heat transfer
- (iii) Uniform heat flux at inner surface

As per given information

Hollow sphere,

Inner radius, $r_i = 3 \text{ cm} = 0.03 \text{ m}$
 Outer radius, $r_o = 5 \text{ cm} = 0.05 \text{ m}$



Inner surface heat flux, $\dot{q} = 10^5 \text{ W/m}^2$

So, heat transfer from inner surface,

$$Q = \dot{q} \times (\text{Inner surface area})$$

$$Q = \dot{q} \times 4\pi(r_i)^2$$

$$= 10^5 \times 4\pi \times (0.03)^2$$

From convection, $Q = h_\infty A_o (T_o - T_\infty)$

$$1130.975 = 400 \times 4\pi (r_o)^2 (T_o - 40)$$

$$1130.975 = 400 \times 4\pi (0.05)^2 (T_o - 40)$$

Outer surface temperature, $T_o = 130.00^\circ\text{C}$

From conduction, $Q = \frac{T_i - T_o}{\left(\frac{r_o - r_i}{4\pi K r_o r_i} \right)}$

$$\Rightarrow 1130.975 = \frac{T_i - 130}{\left(\frac{0.05 - 0.03}{4\pi \times 15 \times 0.05 \times 0.03} \right)}$$

$$T_i - 130 = 80$$

Inner surface temperature, $T_i = 80 + 130 = 210^\circ\text{C}$

110.

Ans. **Design Factors:** The design of fins for heat dissipation considers high heat dissipation rate and low cost.

High dissipation rate is maintained by:

- (a) Optimum space between the two fins based upon boundary layer theory.
- (b) High value of thermal conductivity and low value of convection coefficient ensures good effectiveness.

$$\varepsilon = \sqrt{\frac{kP}{hA_c}}$$

Low cost is maintained by:

- (a) Manufacturing process and shape of fins.
- (b) Material used, etc.

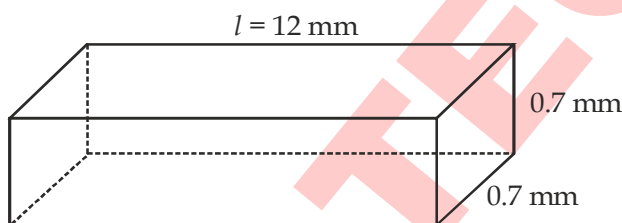
The given problem dissipate 480×10^{-3} W heat from square fins

$$A_c = 0.49 \text{ mm}^2$$

$$K = 170 \text{ W/mK}$$

$$h = 12 \text{ W/m}^2/\text{K}$$

$$L = 12 \text{ mm}$$



Heat dissipated by fins

$$Q = n\theta_b \sqrt{PKhA_c} \tanh(ml)$$

where,

$$n = \text{number of fins}$$

$$\theta_b = 70 - 30 = 40^\circ\text{C}$$

$$P = 2[0.7 + 0.7] = 2.8 \text{ mm}$$

$$0.48 = n \times 40 \sqrt{(2.8 \times 10^{-3} \times 12 \times 170 \times 0.49 \times 10^{-6})} \times \tanh(ml)$$

$$\therefore m = \sqrt{\frac{hP}{KA_c}} = \sqrt{\frac{12 \times 2.8 \times 10^{-3}}{170 \times 0.49 \times 10^{-6}}} = 20.084 \text{ m}^{-1}$$

$$\therefore 0.48 = n \sqrt{(2.8 \times 12 \times 170 \times 0.49 \times 10^{-9})} \times 40 \tanh(20.08 \times 12 \times 10^{-3})$$

$$n = \frac{0.05075}{\sqrt{2.8 \times 10^{-6}}} = 30.335 = 31$$

111.

Ans. Given:

Diameter, $d = 30 \text{ mm}$,

Base temperature, $T_0 = 500^\circ\text{C}$,

Atmospheric temperature, $T_\infty = 25^\circ\text{C}$

Convective heat transfer coefficient, $h = 25 \text{ W/m}^2\text{K}$

Conditions:

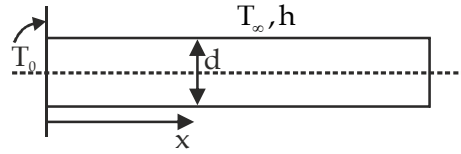
(i) at $x = x_1$, $T = T_1 = 400^\circ\text{C}$

(ii) at $x = x_1 + 10 \text{ mm}$, $T = T_2 = 390^\circ\text{C}$

Given relation

$$\frac{T - T_{\infty}}{T_0 - T_{\infty}} = e^{-mx}$$

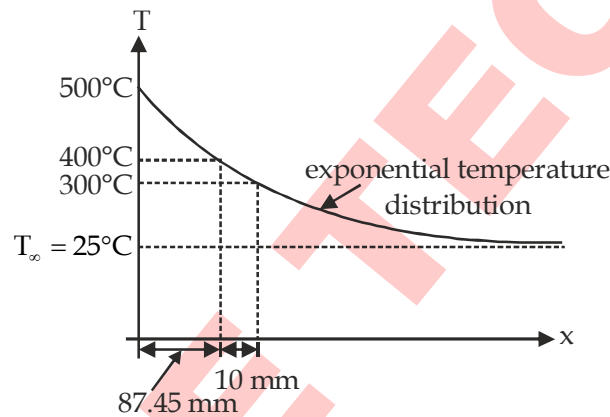
$$m = \sqrt{\frac{hP}{kA}}$$



for cylindrical rod;

$$\frac{A}{P} = \frac{(\pi/4)d^2}{\pi d} = \frac{d}{4}$$

So,
$$m = \sqrt{\frac{4h}{kd}}$$



Applying boundary conditions in given relation,

(i) at $x = x_1$ meter, $T_1 = 400^\circ\text{C}$

$$\frac{400 - 25}{500 - 25} = e^{-mx_1} \quad \dots(i)$$

(ii) at $x = x_1 + 0.01$ m, $T_2 = 390^\circ\text{C}$

$$\frac{390 - 25}{500 - 25} = e^{-m(x_1 + 0.01)} \quad \dots(ii)$$

Divide equation (i) by equation (ii),

$$\frac{400 - 25}{390 - 25} = e^{-m(x_1 - x_1 - 0.01)}$$

$$m = \frac{0.02703}{0.01} = 2.703 \text{ per meter} = \sqrt{\frac{4h}{Kd}}$$

or

$$(2.703)^2 = \frac{4 \times 25}{K \times 0.03}$$

$$K = 456.28 \text{ W/mK}$$

From equation (i)
$$\frac{400 - 25}{500 - 25} = e^{-2.703x_1}$$

$$x_1 = 0.08745 \text{ meter}$$

Now,
 $\Rightarrow$

$$\begin{aligned}x_1 &= 87.45 \text{ mm} \\x_2 &= x_1 + 0.01 \text{ meter} = 0.08745 + 0.01 = 0.09745 \text{ meter} \\x_2 &= 97.45 \text{ mm}\end{aligned}$$

112.

Ans. The heat transfer rate may be increased by increasing the surface area across which convection occurs. This may be done by using fins that extend from the wall into the surrounding fluid. The thermal conductivity of the fin material has a strong effect on the temperature distribution along the fin and thus the degree to which the heat transfer rate is enhanced.

Fin efficiency: In the limiting case of zero thermal resistance or infinite conductivity, the temperature of the fin will be uniform at the base value of T_b . The heat transfer from the fin will be maximum in this case and can be expressed as

$$Q_{\text{fin,max}} = hA_{\text{fin}}(T_b - T_{\infty})$$

In reality, however, the temperature of the fin will drop along the fin, and thus the heat transfer from the fin will be less because of the decreasing temperature difference toward the fin tip as shown in figure below. To account for the effect of this decrease in temperature on heat transfer, we define a fin efficiency as the ratio of actual heat dissipated by the fin to the maximum heat transferred by the pin if the whole fin has temperature equal to the base temperature of the fin.

$$\eta_{\text{fin}} = \frac{Q_{\text{fin}}}{Q_{\text{fin,max}}} = \frac{\text{Actual heat transfer rate from the fin}}{\text{Ideal heat transfer rate from the fin, if the entire fin were at base temperature}}$$

For the cases of constant cross section of very long fins and fins with adiabatic tips, the efficiency can be expressed as

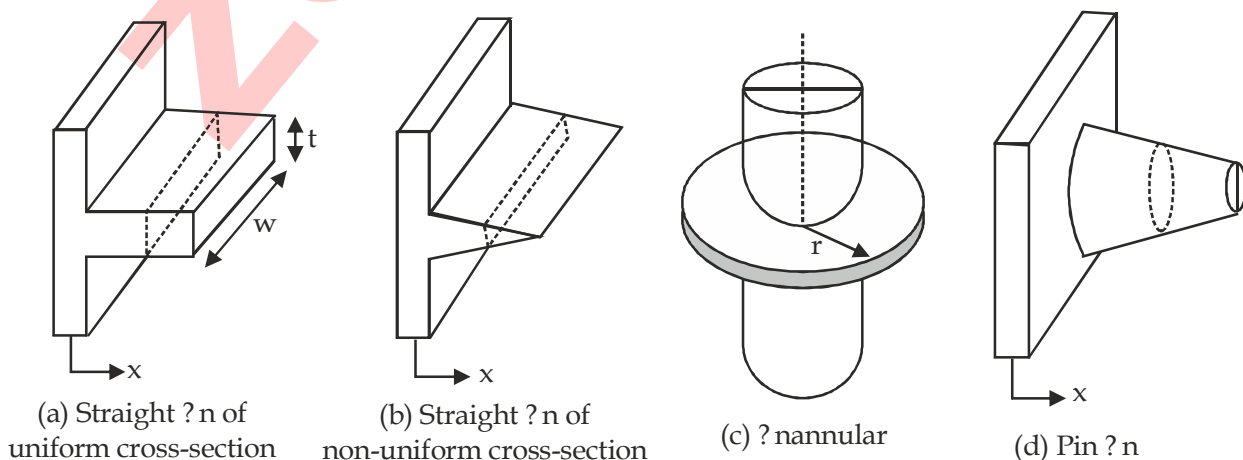
$$\eta_{\text{long fin}} = \frac{Q_{\text{fin}}}{Q_{\text{fin,max}}} = \frac{\sqrt{hPkA_c}(T_b - T_{\infty})}{hA_{\text{fin}}(T_b - T_{\infty})} = \frac{1}{L} \sqrt{\frac{kA_c}{hP}} = \frac{1}{mL}$$

and

$$\eta_{\text{adiabatic tip}} = \frac{Q_{\text{fin}}}{Q_{\text{fin,max}}} = \frac{\sqrt{hPkA_c}(T_b - T_{\infty}) \tanh mL}{hA_{\text{fin}}(T_b - T_{\infty})} = \frac{\tanh mL}{mL}$$

since $A_{\text{fin}} = PL$ for fins with constant cross section.

Figure shows below different fin configurations. A straight fin is any extended surface that is attached to a plane wall. It may be of uniform cross-sectional area, or its cross-sectional may vary which with the distance x from the wall. An annular fin is one that is circumferentially attached to a cylinder. A pin fin or spine is an extended surface of circular cross-sections. Pin fins may also be of uniform or non-uniform cross-section.



113.

Ans. **Lumped Capacity:** In heat transfer analysis, some bodies are observed to behave like a "lump" whose interior temperature remains essentially uniform at any times during a heat transfer process. The temperature of such bodies can be taken to be function of time only. Heat transfer analysis of this idealization is done by lumped capacity of the system.

Assumptions:

- (i) Lumped system analysis assumes a uniform temperature distribution throughout the body.
- (ii) Thermal conductive resistance of the body is zero.
- (iii) Convective heat transfer coefficient is constant and uniform.
- (iv) Lumped system is applicable if $Bi < 0.1$.

Given data:

$d = 50 \text{ mm} = 0.05 \text{ m};$
 $\rho = 7800 \text{ kg/m}^3,$
 $c = 0.46 \text{ kJ/kgK};$
 $K = 35 \text{ W/mK},$
 $h = 10 \text{ W/m}^2\text{K};$
 $T_i = 600^\circ\text{C},$
 $T = 100^\circ\text{C};$
 $T_\infty = 30^\circ\text{C}$

$$Bi = \frac{hs}{K} = \frac{10 \times \left(\frac{0.05}{6}\right)}{35} = 0.0024 < 0.1$$

i.e. lumped analysis is valid.

$$\frac{T - T_\infty}{T_i - T_\infty} = \exp\left[-\frac{hA}{\rho vc} t\right] \quad \dots(i)$$

$$\frac{T - T_\infty}{T_i - T_\infty} = \frac{100 - 30}{600 - 30} = \frac{70}{570} = 0.1228$$

$$\frac{hA}{\rho vc} = \frac{10 \times \pi \times 0.05^2 \times 3}{7800 \times 4 \times \pi (0.025)^3 \times 460} = 3.34 \times 10^{-4} \text{ s}^{-1}$$

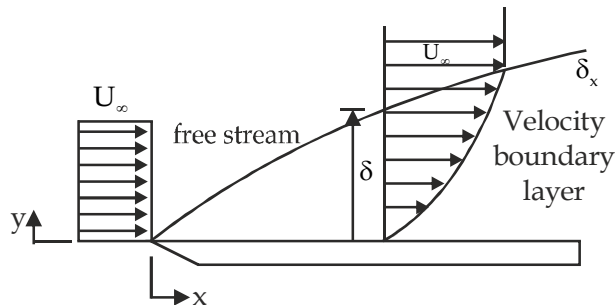
from equation (i), $\frac{hA}{\rho vc} t = 2.0972$

$$t = \frac{2.0972}{3.34 \times 10^{-4}} = 6270 \text{ sec}$$

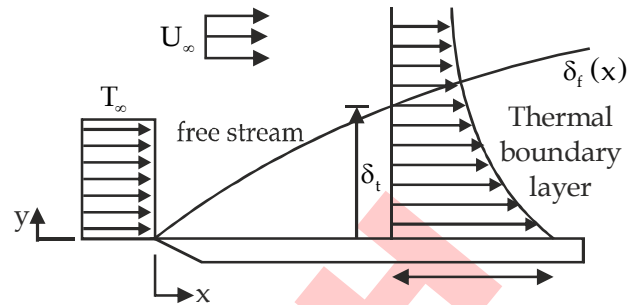
or, $t = \frac{6270}{60} = 104.51 \approx 104 \text{ minute } 30 \text{ seconds}$

114.

Ans. Consider a flow of fluid along the surface of a thin plate with its sharp leading edge set parallel to flow direction.



Velocity boundary layer development on a flat plate.



Thermal boundary layer development on an isothermal flat plate

Thermal boundary layer development on an isothermal flat plate.

The free stream undisturbed flow has a uniform velocity U_∞ in the x direction. Particles of fluid adhere to the plate surface as they approach it and the fluid is slowed down considerably. The fluid becomes stagnant or virtually so in the immediate vicinity of the plate surfaces. Thus there exists a region where the flow velocity changes from that of solid boundary to that of main stream fluid, and in this region the velocity gradients exist in fluid. This thin layer of changing velocity has been called the hydrodynamic boundary layer.

Thermal Boundary Layer:

When a fluid flows past a heated or cold surface, a temperature field is set up in the fluid next to the surface. If the plate surface is hotter than the fluid, the temperature distribution will be as indicated in the figure below. This zone or thin layer where the temperature field exists is called thermal boundary layer.

Now if velocity profile is

$$U_{x,y} = U_\infty \left\{ \frac{3}{2} \left(\frac{y}{\delta_x} \right) - \frac{1}{2} \left(\frac{y}{\delta_x} \right)^3 \right\}$$

and boundary layer

$$\delta_x = \sqrt{\frac{280}{13} \times \frac{\nu x}{U_\infty}}, \quad \nu = (\text{Kinematic viscosity})$$

In question ν is given as ' u '

Now shear stress,
$$\tau = \mu \left(\frac{dU}{dy} \right)_{y=0} = C_{fx} \cdot \frac{1}{2} \rho U_\infty^2$$

$$\Rightarrow \mu \left(\frac{3}{2\delta_x} \times U_\infty \right) = C_{fx} \cdot \frac{1}{2} \rho U_\infty^2$$

$$C_{fx} = \frac{\mu U_\infty \times 3 \sqrt{\frac{13}{280} \times \frac{U_\infty}{\nu x}}}{\rho U_\infty^2} = \frac{3 \times \sqrt{\frac{13}{280}}}{\sqrt{\frac{\rho^2 U_\infty^2}{\mu^2} \times \frac{\nu x}{U_\infty}}} = \frac{3 \times \sqrt{\frac{13}{280}}}{\sqrt{\frac{U_\infty \nu x}{\mu^2}}} = \frac{0.6464}{\sqrt{Re_x}}$$

115.

Ans. (a) Nusselt Number (Nu): Nusselt number can be defined in several ways:

(i) It is the 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 of L meters. Thus,

$$Nu = \frac{Q_{\text{conv.}}}{Q_{\text{cond.}}} = \frac{h}{K/L} = \frac{hL}{K} \quad \dots(i)$$

(ii) It is the ratio of heat transfer rate, Q to the rate at which heat would be conducted within the fluid under a temperature gradient of $\Delta\theta/L$. Thus,

$$Nu = \frac{Q}{(\Delta\theta K)L} = \frac{Q}{\Delta\theta} \times \frac{L}{K} = \frac{hL}{K} \quad \dots(ii)$$

(iii) It is the ratio of characteristic length L to the thickness Δx of a stationary fluid layer conducting the heat at the same rate under the same temperature difference as in the case of convection process. Thus,

$$Q = K \frac{\Delta t}{\Delta x} h \Delta t$$

or
$$\Delta x = \frac{K}{h}$$

$$\therefore Nu = \frac{L}{\Delta x} = \frac{L}{K/h} = \frac{hL}{K}$$

The Nusselt number is a convenient 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 the fluid and inversely proportional to the significant length parameter.

(b) Prandtl Number (Pr): It is the ratio of kinematic viscosity (ν) to thermal diffusivity (α)

$$Pr = \frac{\mu c_p}{K} = \frac{\rho \nu c_p}{K} = \frac{\nu}{(K/\rho c_p)} = \frac{\nu}{\alpha} \quad \dots(i)$$

Kinematic viscosity indicates the impulse transport through molecular friction whereas thermal diffusivity indicates the heat energy transport by conduction process.

- Prandtl number provides a measure of the relative effectiveness of the momentum and energy transport by diffusion.
- 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.

(c) Biot Number (Bi): The non-dimensional factor $\frac{hL_c}{K}$ is called the Biot Number.

i.e
$$Bi = \frac{hL_c}{K} = \text{Biot Number}$$

It gives an indication of the ratio of internal (conduction) resistance to surface (convection) resistance.

When the value of Bi is small, it indicates that the system has a small internal (conduction) resistance, i.e., relatively small temperature gradient or the existence of practically uniform temperature within the system. The convective resistance then predominates and the transient phenomenon is controlled by the convective heat exchange.

If $Bi < 0.1$, the lumped heat capacity approach can be used to advantage with simple shape such as plates, cylinder, spheres and cubes.

(d) Thermal Diffusivity: In case of homogeneous (in which properties e.g., specific heat, density, thermal conductivity etc, are same everywhere in the material) and isotropic (in which properties are independent of surface orientation) material, $K_x = K_y = K_z = K$.

$$\frac{\partial^2 t}{\partial x^2} + \frac{\partial^2 t}{\partial y^2} + \frac{\partial^2 t}{\partial z^2} + \frac{q_g}{K} = \frac{\rho c}{K} \cdot \frac{\partial t}{\partial \tau} = \frac{1}{\alpha} \cdot \frac{\partial t}{\partial \tau}$$

Where,

$$\alpha = \frac{K}{\rho \cdot c} = \frac{\text{Thermal conductivity}}{\text{Thermal capacity}}$$

The quantity,

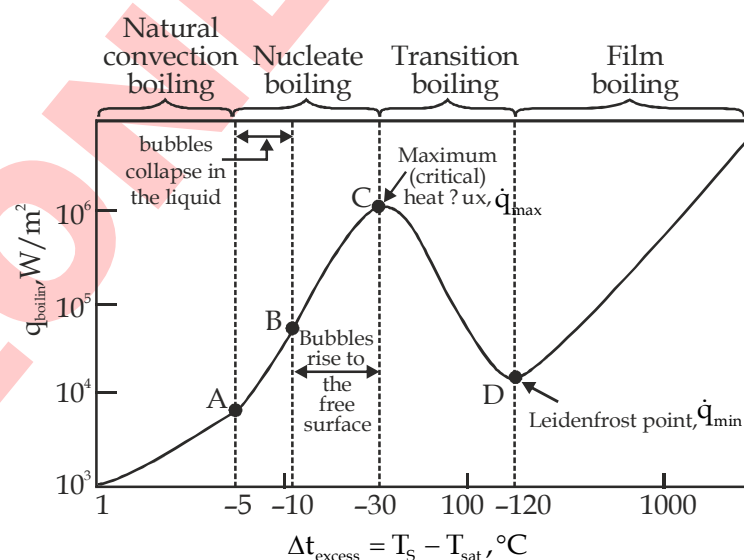
$$\alpha = \frac{K}{\rho \cdot c} \text{ is known as thermal diffusivity}$$

- The larger value of α , the faster will the heat diffuse through the material and its temperature will change with time. This will result either due to a high value of thermal conductivity k or a low value of heat capacity ($\rho \cdot c$). A low value of heat capacity means the less amount of heat entering the element, would be absorbed and used to raise its temperature and more would be available for onward transmission. Metals and gasses have relatively high value of α and their response to temperature changes is quite rapid. The non-metallic solids and liquids respond slowly to temperature changes because of their relatively small value of thermal diffusivity.
- Thermal diffusivity is an important characteristic quantity for unsteady conduction situation.

116.

- Ans. (i) There are two distinct forms of condensation which are observed: film condensation and dropwise condensation. In film condensation, the condensate wets the surface and forms a liquid film on the surface that slides down under the influence of gravity. In dropwise condensation, the condensed vapor forms droplets on the surface instead of a continuous film and surface is covered by countless drops of varying diameters. In dropwise condensation, the small droplets that form at the nucleation sites on the surface grow as a result of continued, coalesce into large droplets, and slide down when reach a certain size, clearing the surface and exposing it to the vapor. There is no liquid film in this case to resist heat transfer. As a result, with dropwise condensation, heat transfer coefficients can be achieved that are more than 10 times larger than those associated with film condensation. Large heat transfer coefficients enable designers to achieve a specified heat transfer rate with a smaller surface area.

(ii)



Typical boiling curve for water at 1 atm

Nucleate Boiling: The nucleate boiling can be represented between points A and C. Point A is referred to as the onset of nucleate boiling (ONB). The bubbles form at an increasing rate at an increasing number of nucleation sites as we move along the boiling curve towards point C. The nucleate boiling regime can be separated into two distinct regions. In region A-B,



($5^{\circ}\text{C} \leq \Delta T_{\text{excess}} \leq 10^{\circ}\text{C}$), isolated bubbles are formed at various preferential nucleation sites on the heated surface. In region B-C ($10 \leq \Delta T_{\text{excess}} \leq 30^{\circ}\text{C}$), the heater temperature is further increased, and bubbles form at such great rates at such a large number of nucleation sites that they form numerous continuous columns of vapor in the liquid. These bubbles move all the way up to the free surface, where they break up and release their vapor content. The large heat fluxes obtainable in this region are caused by the combined effect of entrainment and evaporation.

Film boiling: In this, region, the heater surface is completely covered by a continuous stable vapor film. Point D, where the heat flux reaches a minimum is called Leidenfrost point. The presence of a vapor film between the heater surface and the liquid is responsible for the low heat transfer rates in the film boiling region. The heat transfer rate increases with increasing excess temperature as a result of heat transfer from the heated surface to the liquid through the vapor film by radiation, which becomes significant at high temperatures.

117.

Ans. **Condensation:** Condensation is the process by which a vapor turns into a liquid. This phase change occurs when the vapor is cooled to its dew point temperature or comes into contact with a surface at a temperature below the dew point. During condensation, the latent heat of vaporization is released.

Filmwise Condensation: Filmwise condensation occurs when the liquid condensate and forms a continuous film over the surface. The characteristics of filmwise condensation are:

- The liquid film grows in thickness as more vapor condenses.
- The heat transfer through the liquid film occurs by conduction, which is generally less efficient due to the thermal resistance of the film.
- The film can create a barrier, reducing the overall heat transfer coefficient.

Dropwise Condensation: Dropwise condensation occurs when the vapor condenses into discrete droplets on the surface rather than forming a continuous film. The characteristics of dropwise condensation are:

- The droplets form and grow in size until they are large enough to be pulled away by gravity or other forces.
- The surface remains partially exposed, allowing for higher heat transfer rates.
- The heat transfer is more efficient due to the direct contact between the vapor and the cooled surface.

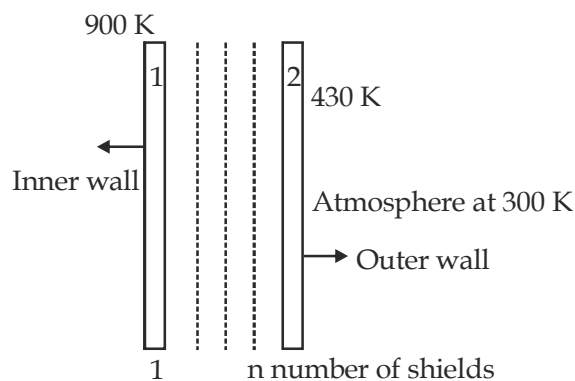
Dropwise condensation is generally more preferred over filmwise condensation for the following reasons:

1. **Higher Heat Transfer Coefficient:** Dropwise condensation has a significantly higher heat transfer coefficient compared to filmwise condensation. This is because the direct contact between the vapor and the cooled surface reduces thermal resistance.
2. **Efficiency:** The absence of a continuous liquid film allows for more effective heat transfer, making dropwise condensation more efficient for practical applications.
3. **Surface Exposure:** In dropwise condensation, the surface remains exposed to the vapor as droplets form and fall away. This continuous renewal of the surface helps maintain high heat transfer rates.

118.

Ans. Heat lost by outer surface per unit area

$$\begin{aligned}
 &= \varepsilon \sigma (T_2^4 - T_0^4) + h(T_2 - T_0) \\
 &= 0.85 \times 5.67 \times 10^{-8} [430^4 - 300^4] + 1.5(T_2 - T_0)^{0.33} \times (T_2 - T_0) \\
 &= 1257.3116 + 971.932 \\
 &= 2229.24 \text{ W} \qquad \dots(i)
 \end{aligned}$$



This heat must transfer through walls and shields

∴ Heat transfer through composite system of walls and n shields of equal emissivity is

$$\frac{\sigma(T_1^4 - T_2^4)}{(n+1)\left[\frac{2}{\epsilon} - 1\right]} = 2229.24$$

$$\frac{5.67 \times 10^{-8} [900^4 - 430^4]}{(n+1)\left[\frac{2}{0.85} - 1\right]} = 2229.24$$

$$\frac{4.19087}{n+1} [900^4 - 430^4] \times 10^{-8} = 2229.24$$

$$(n+1) = \frac{26063.52}{2229.24} = 11.7$$

$$n = 10.7$$

$$n \approx 11$$

Hence eleven shields are inserted.

119.

Ans. (a) Lambert's Cosine Law: The law states that the total emissive power E_θ from a radiating plane surface in any direction is directly proportional to the cosine of the angle of emission. The angle of emission is the angle subtended by the normal to the radiating surface and the direction vector of emission of the receiving surface. If E_n be the total emissive power of the radiating surface in the direction of its normal, then,

$$E_\theta = E_n \cos \theta$$

$$E = \epsilon \sigma T^4 = 0.7 \times 5.67 \times 10^{-8} \times (1073)^4$$

$$E = I_{bn} \pi = 52611.4 \text{ W / m}^2$$

Normal intensity

$$I_{bn} = \frac{E}{\pi} = 16746.73 \text{ W / m}^2$$

Intensity of radiation at an angle of 30° with respect to normal direction

$$I_\theta = I_n \cos \theta = 16746.73 \times \cos 30^\circ$$

$$= 14503.09 \text{ W / m}^2 \text{ steradian}$$

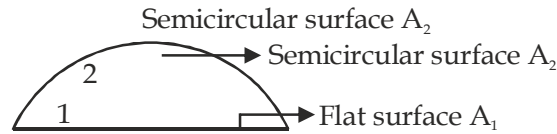
(b)

$$F_{11} = 0$$



$$\therefore F_{11} + F_{12} = 1$$

$$F_{12} = 1$$



By using reciprocity theorem we have

$$A_1 F_{12} = A_2 F_{21}$$

$$F_{21} = \frac{A_1}{A_2} = \frac{\pi r^2}{2\pi r^2} = 0.5$$

Also

$$F_{21} = F_{22} = 1$$

$$0.5 + F_{22} = 1$$

$$F_{22} = 0.5$$

120.

Ans. Thermal capacity rates of the hot and cold fluids are :

$$C_h = m_h c_h = \frac{4}{60} \times 4200 = 280 \text{ W / K}$$

$$C_c = m_c c_c = \frac{8}{60} \times 2400 = 320 \text{ W / K}$$

Obyiously, $C_{\min} = 280 \text{ W / K}$ and $C_{\max} = 320 \text{ W / K}$

$$\text{Capacity ratio, } C = \frac{C_{\min}}{C_{\max}} = \frac{280}{320} = 0.875$$

(a) The effectiveness for a parallel flow heat exchanger is

$$\varepsilon = \frac{1 - \exp[-NTU(1+C)]}{1+C}$$

Effectiveness will be maximum when $NTU = \infty$ for the given value of capacity ratio C .

$$\therefore \text{Maximum possible effectiveness} = \frac{1 - \exp[-\infty]}{1+C}$$

$$= \frac{1}{1+C} = \frac{1}{1+0.875} = 0.533$$

(b) The effectiveness for a counter flow heat exchanger is

$$\varepsilon = \frac{1 - \exp[-NTU(1-C)]}{1 - C \exp[-NTU(1-C)]}$$

Effectiveness will be maximum when $NTU = \infty$ for the given value of capacity ratio C

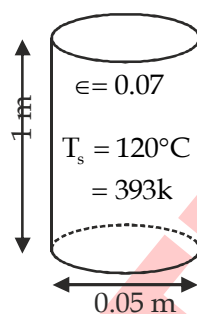
$$\therefore \text{Maximum possible effectiveness} = \frac{1 - \exp(-\infty)}{1 - C \exp(-\infty)}$$

$$= \frac{1-0}{1-0} = 1$$

121.

Sol.

Given:

Length of rod, $L = 1$ m;Diameter of rod, $D = 5$ cm = 0.05 m;Surface temperature of rod, $T_s = 120^\circ\text{C}$;Surrounding temperature, $T_\infty = 34^\circ\text{C}$;Surface emissivity of rod, $\epsilon = 0.7$;Density of air, $\rho = 0.998$ kg / m³;Kinematic viscosity of air, $\nu = 2.076 \times 10^{-5}$ m²/s;Thermal conductivity of air, $K = 0.03$ W/mK;Prandtl number, $Pr = 0.697$ **Assumptions:**

(i) Steady state heat transfer.

(ii) Convection heat transfer is taking place naturally.

(iii) Surrounding air and wall temperature of room remains at same temperature.

(iv) All properties of air are taken at mean film temperature.

(v) For radiation heat transfer the cylindrical body is assumed to be a small body enclosed in a large room.

(vi) Radiation through the ends of the rod is negligible.

Characteristic length, $L_c = L = 1$ mCoefficient of thermal expansion, $\beta = \frac{1}{T_{mf}}$ Mean film temperature, $T_{mf} = \frac{T_s + T_\infty}{2} = \frac{120 + 34}{2}$ $T_{mf} = 77^\circ\text{C} = 350$ K $\therefore \beta = \frac{1}{350} = 2.857 \times 10^{-3} \text{ K}^{-1}$ Grashoff number, $Gr = \frac{g \beta \Delta T L_c^3}{\nu^2}$

$$Gr = \frac{9.81 \times 2.857 \times 10^{-3} \times (120 - 34) \times 1^3}{(2.076 \times 10^{-5})^2}$$

$$Gr = 5.593 \times 10^9$$

Nusselt number correlation as given in question,

$$\overline{Nu}_L = 0.1 (Gr \cdot Pr)^{0.33}$$

$$\overline{Nu}_L = 0.1 \times (5.593 \times 10^9 \times 0.697)^{0.33}$$



$$\overline{Nu}_L = 146.214$$

$$\Rightarrow \frac{\bar{h}L_c}{K} = 146.214$$

$$\Rightarrow \frac{\bar{h} \times 1}{0.03} = 146.214$$

$$\Rightarrow \bar{h} = 4.3864 \text{ W/m}^2\text{K}$$

(a) Heat lost by the rod by convection

$$\begin{aligned} q_{\text{conv}} &= \bar{h}.A.\Delta T \\ &= \bar{h}.(\pi DL) \times (T_s - T_\infty) \\ &= 4.3864 \times \pi \times 0.05 \times 1 \times (120 - 34) \end{aligned}$$

$$q_{\text{conv}} = 59.255 \text{ W}$$

(b) Heat lost by the rod by radiation,

$$\begin{aligned} q_{\text{rad}} &= \epsilon \cdot \sigma \cdot A_s (T_s^4 - T_\infty^4) \\ q_{\text{rad}} &= 0.7 \times 5.67 \times 10^{-8} \times \pi \times 0.05 \times 1 \times (393^4 - 307^4) \\ &= 93.34 \text{ W} \end{aligned}$$

(c) Total heat lost by the rod

= Heat lost by convection + Heat lost by radiation

$$\begin{aligned} q_{\text{total}} &= q_{\text{conv}} + q_{\text{rad}} \\ &= 59.255 + 93.34 \end{aligned}$$

$$q_{\text{total}} = 152.59 \text{ W}$$

(d) Percentage of convection and radiation heat loss

$$\text{Percentage of convection heat loss} = \frac{q_{\text{conv}}}{q_{\text{total}}} \times 100 = \frac{59.255}{152.59} \times 100 = 38.83\%$$

$$\text{Percentage of radiation heat loss} = \frac{q_{\text{rad}}}{q_{\text{total}}} \times 100 = \frac{93.34}{152.59} \times 100 = 61.17\%$$

(e) Since radiation heat loss has high percentage contribution (61.17%) in the total heat loss, it is not justifiable to neglect radiation heat loss for this situation.

122.

Ans. (a) In a heat exchanger, during normal operation the tube surface gets covered by deposits of ash, soot, dirt and scale etc. This phenomenon of rust formation and deposition of fluid impurities is called fouling. Due to these surface deposits the thermal resistance is increased and eventually the performance of the ascertain the thickness and thermal conductivity of the scale deposits, the effect of scale on heat flow is considered by specifying an equivalent scale heat transfer coefficient h_s . If h_{si} and h_{so} be the heat transfer coefficients for the scale deposited on the inside and outside surfaces respectively, then the thermal resistances to scale formation on the inside surface (R_{si}) and outside surface (R_{so}) are given by

$$R_{si} = \frac{1}{A_i h_{si}}; \quad R_{so} = \frac{1}{A_o h_{so}}$$

The reciprocal of scale heat transfer coefficient, h_s is called the fouling factor, R_f . Thus

$$R_f = \frac{1}{h_s} \text{ m}^2\text{°C/W}$$

Fouling factors are determined experimentally by testing the heat exchanger is both the clean and dirty conditions. The fouling factor, R_f is thus defined as

$$R_f = \left(\frac{1}{h_s} \right) = \frac{1}{U_{\text{dirty}}} - \frac{1}{U_{\text{clean}}}$$

(b) Nature of heat exchanger process:

Heat exchangers, on the basis of nature of heat exchange process, are classified as follows:

- (i) Direct contact (or open) heat exchanges.
- (ii) Indirect contact heat exchangers.
- (i) **Direct contact heat exchangers:** In a direct contact or open heat exchanger the exchange of heat takes place by direct mixing of hot and cold fluids and transfer of heat and mass takes place simultaneously. The use of such units is made under conditions where mixing of two fluids is either harmless or desirable. Example: (i) Cooling towers; (ii) Jet condensers; (iii) Direct contact feed heaters.
- (ii) **Indirect contact heat exchangers:** In this type of heat exchanger, the heat transfer between two fluid could be carried out by transmission through wall which separates the two fluids. According to this heat exchanger are further classified into two categories:
 - (a) Transfer type heat exchanger.
 - (b) Storage type heat exchanger.
- (a) **Direct transfer type heat exchanger (Recuperator):** In this type heat exchanger, the two fluids are kept separate and do not mix as they flow through heat exchanger. Heat is transferred through the separating wall between hot and cold fluids by convection and conduction. Ex: Boilers, superheaters, air-preheater, economiser, condenser, evaporator. etc.
- (b) **Storage type heat exchanger (Regenerator):** In this type of heat exchanger, hot and cold fluids flow alternatively on the same surface. The hot fluid gives heat to the surface and cold fluid gains heat from the same surface. Ex: Regenerator of reciprocating compressors, I.C. engines etc.

Advantage:

1. Higher heat transfer coefficient;
2. Less weight per kW of the plant;
3. Minimum pressure loss;
4. Quick response to load variation;
5. Small bulk weight;
6. Efficiency quit high.

Disadvantages:

1. Costlier compared to recuperative heat exchangers.
2. Leakage is the main trouble, therefore, perfect sealing is required.

123.

Ans. Given,

Surface temperature $T = 3000\text{K}$

1. **Monochromatic Emissive Power :**

From Planck's distribution law, we know

$$E_{b\lambda} = \frac{C_1 \lambda^{-5}}{\exp\left(\frac{C_2}{\lambda T}\right) - 1}$$

Where,

c_1 = First radiation constant $= 2\pi hc^2 = 0.374 \times 10^{-15} \text{ Wm}^2$

c_2 = First radiation constant $= \frac{hc}{K} = 14.4 \times 10^{-3} \text{ mK}$

$\lambda = 1 \times 10^{-6} \text{ m}$

⇒

$$E_{b\lambda} = \frac{0.374 \times 10^{-15} [1 \times 10^{-6}]^{-5}}{\left[\frac{144 \times 10^{-3}}{1 \times 10^{-6} \times 3000} \right]_{-1}}$$

$$E_{b\lambda} = 3.10 \times 10^{12} \text{ W/m}^2$$

2. Maximum wave length ($\lambda_{\max}$)

From Wien's law, we know

$$\lambda_{\max} T = 2.9 \times 10^{-3} \text{ mK}$$

⇒

$$\lambda_{\max} = \frac{2.9 \times 10^{-3}}{3000}$$

$$\lambda_{\max} = 0.966 \times 10^{-6} \text{ m}$$

3. Maximum emissive power ($E_{b\lambda}$) max:

Maximum emissive power

$$\begin{aligned} (E_{b\lambda})_{\max} &= 1.307 \times 10^{-5} T^5 \\ &= 1.307 \times 10^{-5} \times (3000)^5 \\ (E_{b\lambda})_{\max} &= 3.17 \times 10^{12} \text{ W/m}^2 \end{aligned}$$

4. Total emissive power (E_b):

From Stefan - Boltzmann law, we know that

$$E_b = \sigma T^4$$

Where,

$$\begin{aligned} \sigma &= \text{Stefan - Boltzmann constant} \\ &= 5.67 \times 10^{-8} \text{ W/m}^2 \text{K}^4 \\ &= 5.67 \times 10^{-8} \text{ W/m}^2 \text{K}^4 \end{aligned}$$

$$\Rightarrow E_b = (5.67 \times 10^{-8})(3000)^4$$

$$E_b = 4.59 \times 10^6 \text{ W/m}^2$$

5. Total emissive power of a real surface:

$$(E_b)_{\text{real}} = \varepsilon \sigma T^4$$

Where,

$$\varepsilon = \text{Emissivity} = 0.85$$

$$(E_b)_{\text{real}} = 0.85 \times 5.67 \times 10^{-8} \times (3000)^4$$

$$(E_b)_{\text{real}} = 3.90 \times 10^6 \text{ W/m}^2$$