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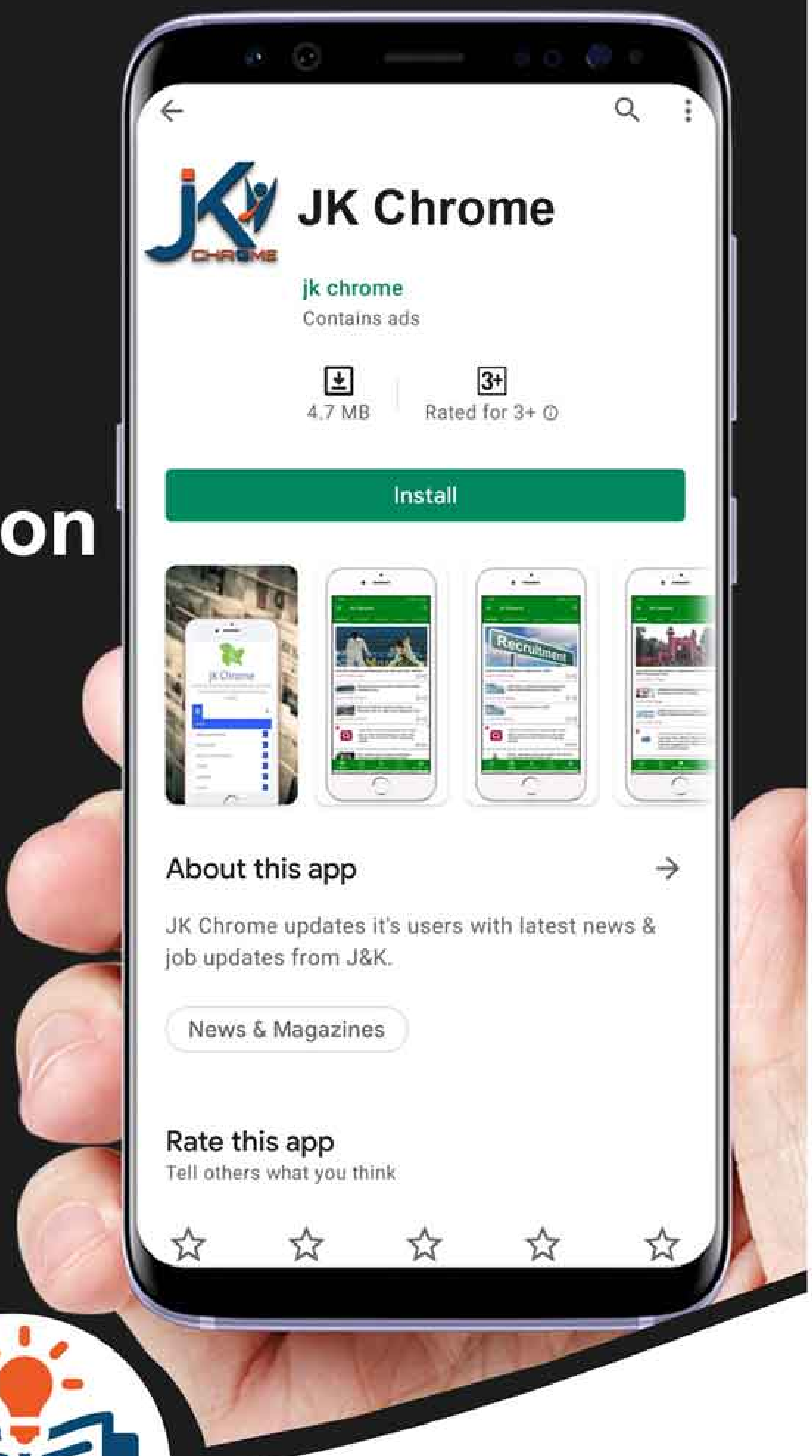

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Heat-Transfer

Index

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Conduction

FUNDAMENTALS OF CONDUCTION:

DIFFERENCE BETWEEN HEAT TRANSFER & THERMODYNAMICS

The main difference between thermodynamic analysis & heat transfer analysis of a problem is that in thermodynamics we deal with system in equilibrium i.e. to bring a system from one equilibrium state to another, how much heat is required (in Joules) is the main criteria in thermodynamics analysis.

But in heat transfer analysis, we evaluate at what rate that change of state occurs by calculating rate of heat transfer (in joule/sec or watt).

Modes of Heat Transfer:

(1) Conduction

(2) Convection

(3) Radiation

The thermal conductivity of gases increases with increase of their temperature. (The reason being the increase in molecular activity)

Liquids are better conductor than gases.

Example, k_{water} = 0.61 W/mk

Mercury (Liquid metal) is best-known conductor Liquid, k_{Hg} = 8.1 W/mk

Mercury has high thermal conductivity, Low vapour pressure & good expansion ability due to heat. Hence, it is used in a thermometer.

CONVECTION:

In case of forced convection heat transfer this macroscopic bulk motion of the fluid is provided by an external agency like far or a blower or a pump.

In case of free convection heat transfer, this motion is provided by buoyancy forces arising out of density changes of fluid due to its temperature change.

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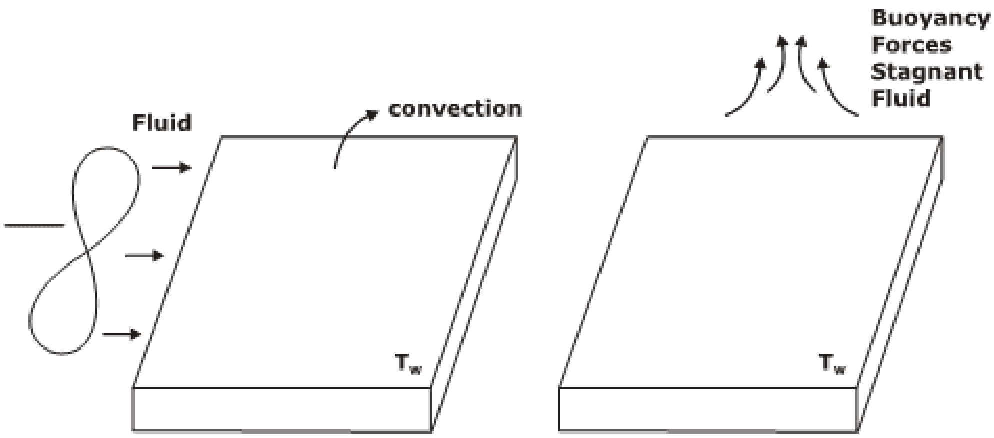
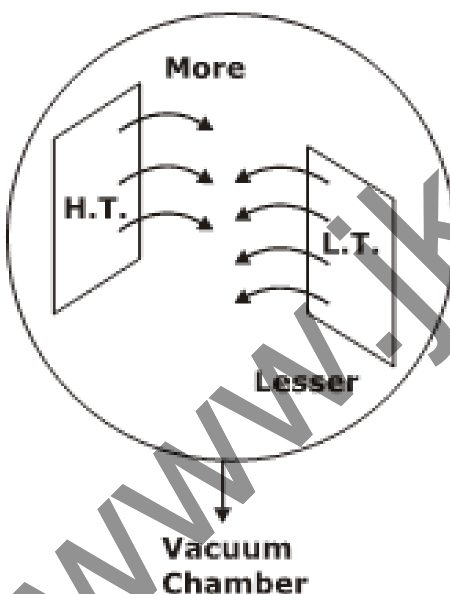


RADIATION:

Radiation mode of heat transfer predominates over conduction & convection particularly when the temperature difference is sufficiently high.

Example, Mode of heat transfer between hot flue gases & refractory brick walls in a large pulverized fuel fired power boiler is predominantly by Radiation.



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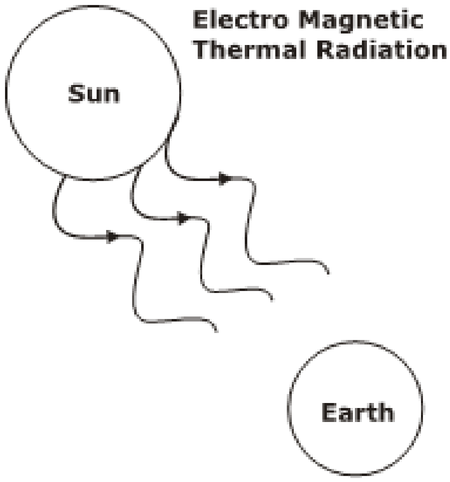
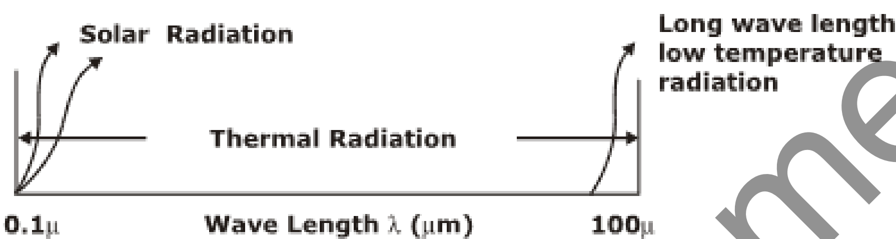
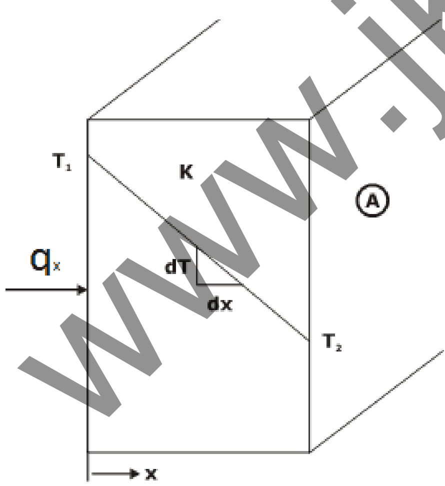




Fourier's Law of Conduction:

The law states that "the rate of heat transfer by conduction in a given direction is directly proportional to the temperature gradient along that direction & is also directly proportional to the area of heat transfer lying perpendicular to the direction of heat transfer".



Heat always flows in the downhill of temperature (Clausius IInd statement of thermodynamics)

Thermal conductivity 'k' is defined as the ability of the material to allow the heat energy to get conducted through the material (W/mk)

Therefore,

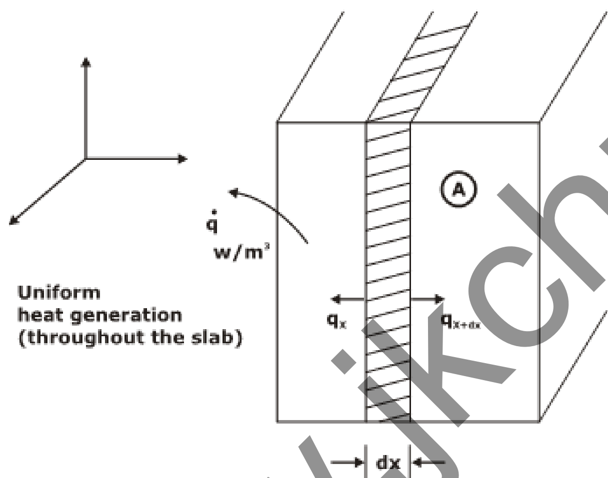
$$q_x = -kA\left(\frac{dT}{dx}\right) \text{ watts}$$

'k' is thermal conductivity (thermo physical property) of material of slab.

GENERALISED CONDUCTION EQUATION:

Rectangular Unsteady, 3D heat conduction with heat generation:

Heat can be generated in a slab or in a material by passing electric or by exothermic chemical reaction or by thermonuclear fission reaction,



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

If conditions are steady, then

$$\frac{\partial T}{\partial \tau} = 0$$

If there is no heat generation, then q = 0

Then this equation results into Laplace Equation:

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$$\frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2} + \frac{\partial^2 T}{\partial z^2} = 0$$

Poisson’s equation:

$$\frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2} + \frac{\partial^2 T}{\partial z^2} + \frac{q}{K} = 0$$

THERMAL DIFFUSIVITY (α):

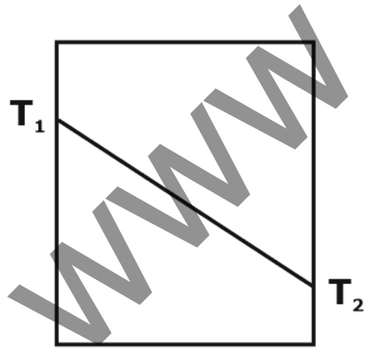
Defining thermal diffusivity (α), a thermophysical property of a material i.e., the ratio between thermal conductivity of material & its thermal capacity/heat capacity (heat storage capacity).

$$\alpha = \frac{k}{\rho C_p}$$

Thermal diffusivity (α) of a medium signifies the rate at which heat energy can diffuse through the medium. Higher the thermal conductivity (k) of material or lesser its heat capacity, it mean & more the thermal diffusivity of material.

STEADY STATE, 1-D, NO INTERNAL HEAT GENERATION:

Plane Wall:



$$\dot{Q} = \frac{KA(T_1 - T_2)}{L}$$

Cylinder:

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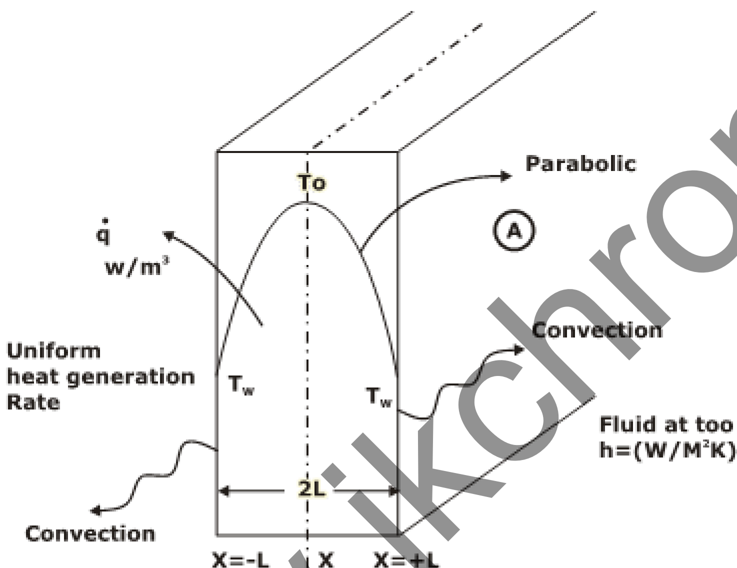
$$\dot{Q} = \frac{2K\pi L(T_1 - T_2)}{\ln(r_2 / r_1)}$$

Sphere:

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

CONDUCTION WITH HEAT GENERATION:

Conduction with Heat Generation in a Slab:



Parabolic Temperature Distribution,

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

Maximum temperature,

$$T_0 = T_w + \frac{q}{k}\frac{L^2}{2}$$

Conduction with Heat Generation in a Cylinder:

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$$T_0 - T = \frac{\dot{q} r^2}{4K}$$

Maximum temperature,

$$T_{max} = T_w + \frac{qR^2}{4k}$$

Conduction with Heat Generation in a Sphere:

Maximum temperature,

$$T_{max} = T_w + \frac{qR^2}{6k}$$

where center temperature T_0 ,Surface temperature Tw,

Critical Radius of Insulation for Cylinder:

$$r_o = \frac{k_{ins}}{h}$$

Critical Radius of Insulation for Sphere:

$$(r_o)_{sphere} = \frac{2K_{ins}}{h}$$

Thermal Resistance Concept:

ELECTRICAL ANALOGY OF HEAT TRANSFER:

Electrical	Thermal
I amperes	Q watts
Emf or ΔV	ΔT° C
R elec.	Rth.

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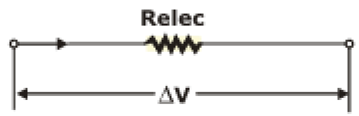
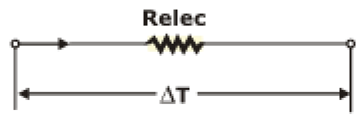


$$\text{Relec} = \frac{\Delta V}{i}\ \Omega$$



$$R_{Th} = \left(\frac{\Delta T}{q}\right) \text{K/watt}$$

For slab,

$$R_{Th} = \frac{T_1 - T_2}{q} = \frac{b}{KA} \text{ k / watt}$$

For Cylinder,

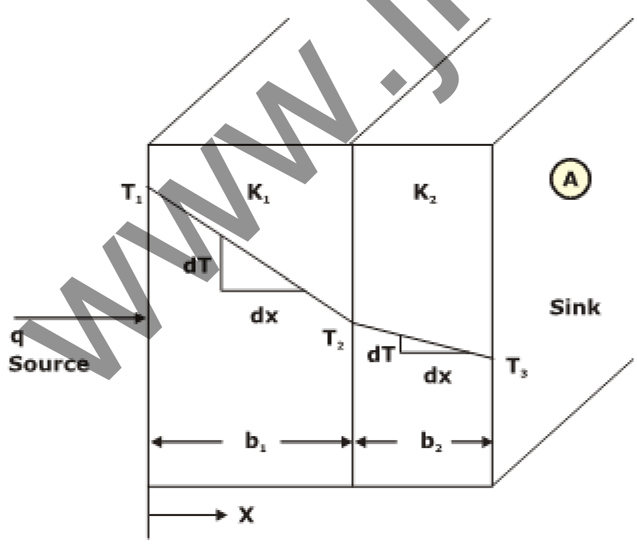
$$(R_{Th})_{cylinder} = \frac{(T_1 - T_2)}{q} = \frac{\ln\frac{r_2}{r_1}}{2\pi KL} \text{ K / watts}$$

For Sphere,

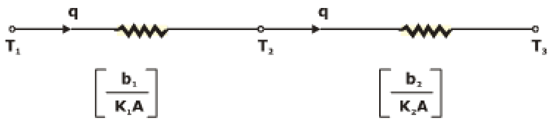
$$Rth = \frac{T_1 - T_2}{\dot{Q}} = \frac{r_2 - r_1}{4\pi K r_1 r_2}$$

Combinations:

Case 1: Conduction Heat Transfer Through A Composite Slab

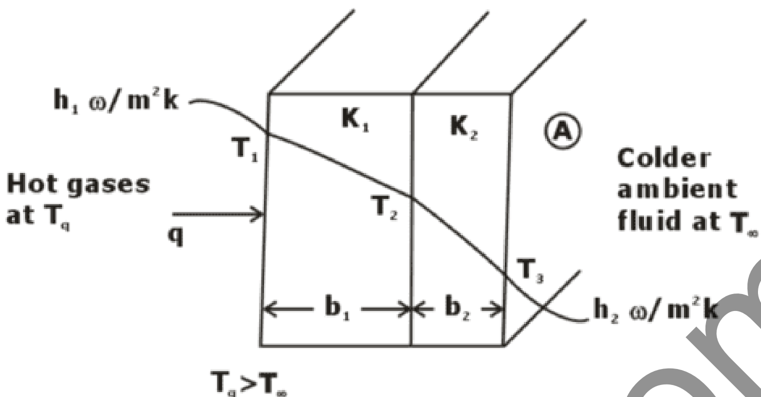


Drawing the equivalent thermal circuit

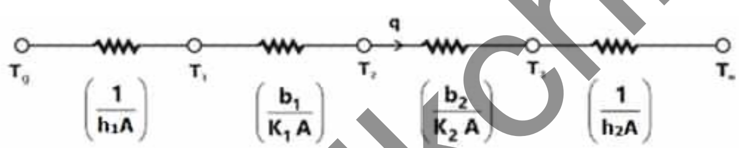


$$\sum R_{Th} = \left(\frac{b_1}{K_1A} + \frac{b_2}{K_2A}\right)$$

Case 2: Conduction – Convection Heat Transfer Through A Composite Slab:



Drawing the equivalent thermal circuit



$$\sum R_{Th} = \left(\frac{1}{h_1A}\right) + \left(\frac{b_1}{K_1A}\right) + \left(\frac{b_2}{K_2A}\right) + \left(\frac{1}{h_2A}\right)$$

CASE 4: Conduction – Convection Heat Transfer Through Composite Cylinder

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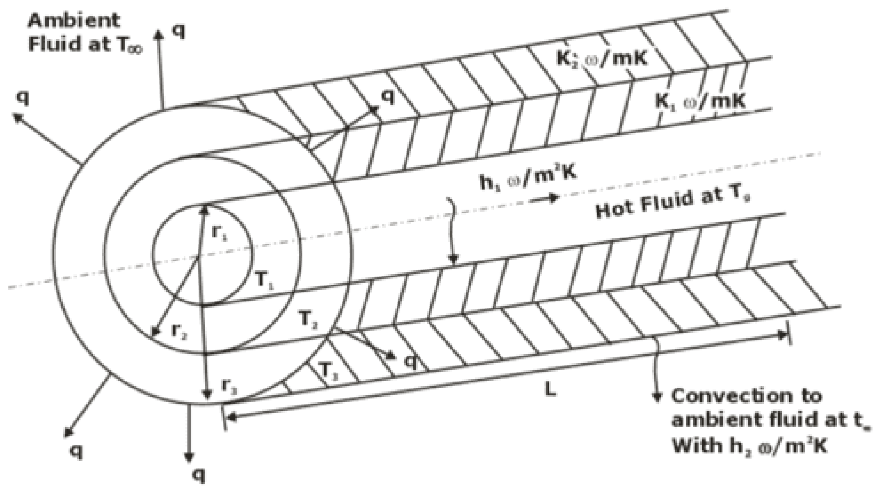
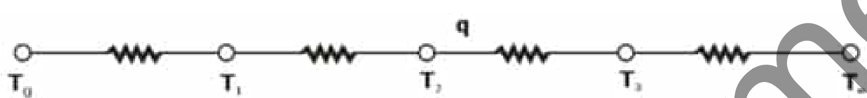
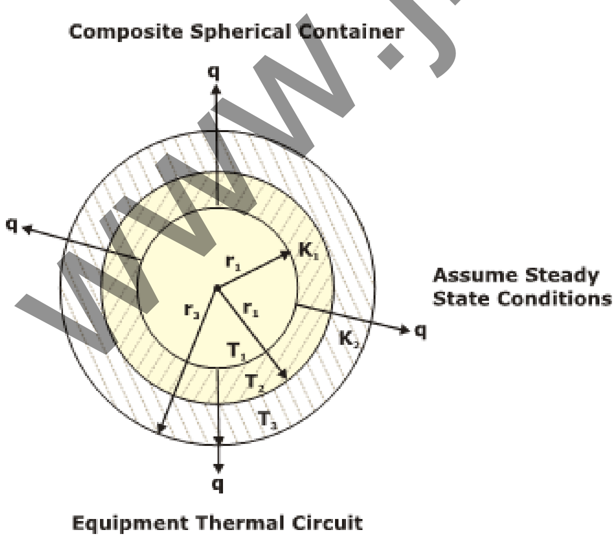


Drawing the equivalent thermal circuit



$$q = \frac{(T_G - T_\infty)}{\frac{1}{h_1 2\pi r_1 L} + \frac{\ln\frac{r_2}{r_1}}{2\pi k_1 L} + \frac{\ln\frac{r_3}{r_2}}{2\pi K_2 L} + \frac{1}{h_2 2\pi r_3 L}}$$

Case 5: Composite Spherical Container



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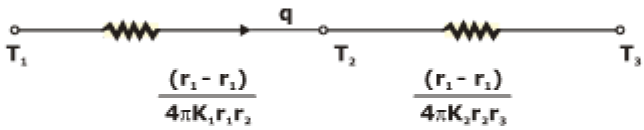
Drawing the equivalent thermal circuit,

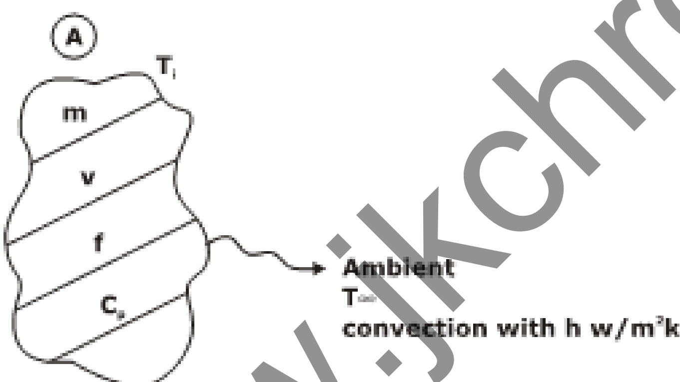


Unsteady State Heat Conduction & Fins

Lumped System Analysis

- Interior temperatures of some bodies remain essentially uniform at all times during a heat transfer process.
- The temperature of such bodies is only a function of time, T = T(t).
- The heat transfer analysis based on this idealization is called **lumped system analysis**.

Consider a body of the arbitrary shape of mass m, volume V, surface area A, density ρ and specific heat C_p initially at a uniform temperature T_i.



- At time t = 0, the body is placed into a medium at temperature T_∞ ($T_\infty > T_i$) with a heat transfer coefficient h.
- An energy balance of the solid for a time interval dt can be expressed as:
  heat transfer into the body during
  dt = the increase in the energy of the body during dt

$$h A (T_\infty - T) dt = m C_p dT$$

With m = ρV and change of variable dT = d(T − T_∞):

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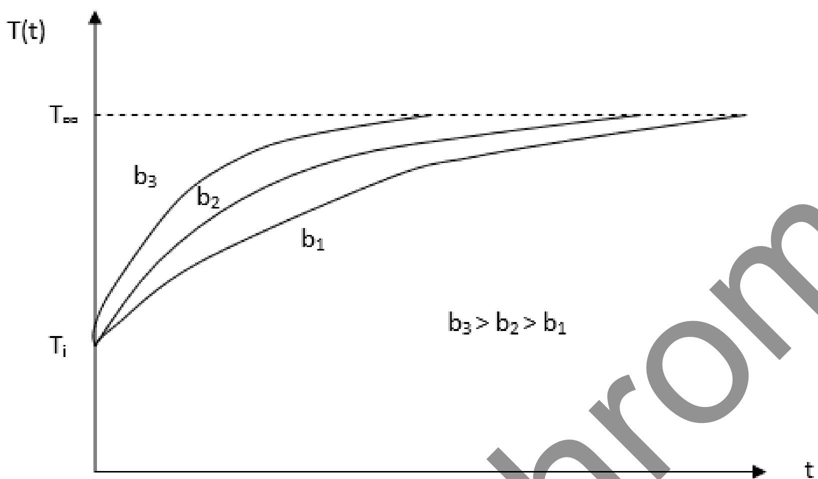
$$\frac{d(T-T_\infty)}{T-T_\infty} = -\frac{hA}{\rho V C_p} dt$$

Integrating from t = 0 to T = T_i

$$\frac{T(t)-T_\infty}{T_i-T_\infty} = e^{-bt}$$

$$b = \frac{hA}{\rho V C_p} \quad (1/s)$$



Criterion for Lumped System Analysis

- Lumped
  system approximation provides a great convenience in heat transfer an alysis.

We
want to establish a criterion for the applicability of the lumped system analysi s.

A characteristic length scale is defined as: **L_c= V/A** (where ,V = Volumn, A = Surface area)

A non-dimensional parameter, the Biot number, is defined:

$$\text{Biot Number} = \frac{\text{Internal conductive resistance offered by body}}{\text{External convective resistance}}$$

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- The Biot number is the ratio of the internal resistance (conduction) to the external resistance to heat convection.
- Lumped system analysis assumes a uniform temperature distribution throughout the body, which implies that the conduction heat resistance is zero. Thus, the lumped system analysis is exact when Bi = 0.
- It is generally accepted that the lumped system analysis is applicable if: **Bi ≤ 0.1**
- Therefore, small bodies with high thermal conductivity are good candidates for lumped system analysis. Note that assuming h to be constant and uniform is an approximation.

Fourier number:

Fourier number is given by:

$$Fo = \frac{\alpha t}{L_c^2}$$

Characteristic length for sphere	$l_c = \frac{\frac{4}{3}\pi R^3}{4\pi R^2} = \frac{R}{3}$
Characteristic length for solid cylinder	$l_c = \frac{\pi R^2 L}{2\pi R(L+R)}$ $if\ l >> R, l_c = \frac{R}{2}$
Characteristic length for cube	$l_c = \frac{L^3}{6L^2} = \frac{L}{6}$
Characteristic length for rectangular plate	$l_c = \frac{lbt}{2lb} = \frac{t}{2}$
Characteristic length for hollow cylinder	$l_c = \frac{\pi(r_0^2 - r_i^2)l}{2\pi r_0 l + 2\pi r_i l + 2\pi(r_0^2 - r_i^2)}$

Heat Transfer from Extended Surface (Fin):

- A fin is a surface that extends from an object to increase the rate of heat transfer to or from the environment by increase convection.
- Adding a fin to an object increases the surface area and can sometimes be an economical solution to heat transfer problems.

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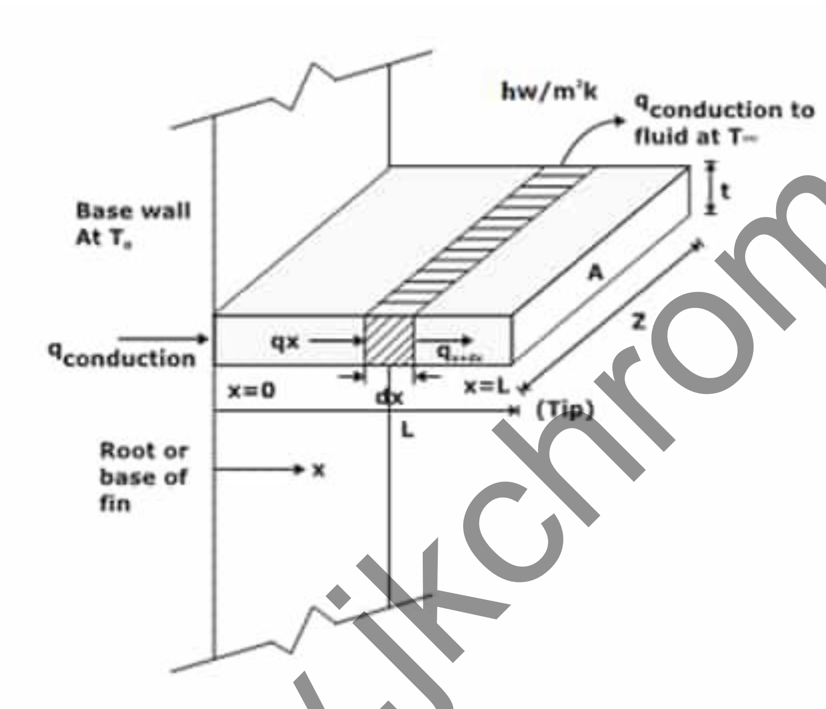
- Finned surfaces are commonly used in practice to enhance heat transfer. In the analysis of the fins, we consider steady operation with no heat generation in the fin.
- We also assume that the convection heat transfer coefficient h to be constant and uniform over the entire surface of the fin.
- The rate of heat transfer from a solid surface to atmosphere is given by $Q = hA\Delta T$ where, h and ΔT are not controllable.
- So, to increase the value of Q surface area should be increased. The extended surface which increases the rate of heat transfer is known as fin.



$$\frac{d^2\theta}{dx^2} - \frac{hP}{KA}(T - T_\infty) = 0$$

- General equation of 2nd order: $\boldsymbol{\theta = c_1e^{mx} + c_2e^{-mx}}$
- Heat dissipation can take place on the basis of three cases.

Case 1: Heat Dissipation from an Infinitely Long Fin (L → ∞):

- In such a case, the temperature at the end of Fin approaches to surrounding fluid temperature ta as shown in figure. The boundary conditions are given below

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- At x=0, t=t_0 : $\theta = t_0 - t_a = \theta_0$
- At x =L→ ∞: t = t_a, θ = 0

  $$\frac{t - t_a}{t_0 - t_a} = e^{-mx}$$

  $\theta = \theta_0\, e^{-mx}$
- Heat transfer by conduction at base:

$$Q_{fin} = \sqrt{KA_cPh}\,(t_0 - t_a)$$

Case 2: Heat Dissipation from a Fin Insulated at the End Tip:

- Practically, the heat loss from the long and thin film tip is negligible, thus the end of the tip can be, considered as insulated.
- At x=0, t=t_0 and $\theta = t_0 - t_a = \theta_0$

$$\frac{\theta}{\theta_0} = \frac{T - T_\infty}{T_0 - T_\infty} = \frac{\cosh m(L - x)}{\cosh mL}$$

$$q_{thro\ fin} = \sqrt{hPkA}\,\theta_0 \tanh mL \text{ watt}$$

$$q_{thro\ fin} = \sqrt{hPkA}\,(T_0 - T_\infty)\tanh mL \text{ watt}$$

Fin Efficiency:

Fin efficiency is given by:

$$\eta = \frac{\text{Actual heat rate from fin Q}}{\text{Maximum heat transfer rate } Q_{max}}$$

Note: The following must be noted for a **proper fin selection**:

- The **longer the fin**, the **larger the heat transfer area** and thus the higher t he rate of heat transfer from the fin.
- The larger the fin, the bigger the mass, the higher the price, and larger th e fluid friction.
- The fin efficiency decreases with increasing fin length because of the d ecrease in fin temperature with length.

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Fin Effectiveness:

- The performance of fins is judged on the basis of the enhancement in heat transfer relative to the no-fin case, and expressed in terms of the fin effectiveness:

$$\varepsilon_{fin} = \frac{\dot{Q}_{fin}}{\dot{Q}_{no\ fin}} = \frac{\dot{Q}_{fin}}{hA_b(T_b - T_\infty)} = \frac{\text{Heat transfer rate from the fin}}{\text{heat transfer rate from the surface area of } A_b}$$

$$\varepsilon_{fin} = \begin{cases} <1 & \text{fin acts as insulation} \\ =1 & \text{fin does not affect heat transfer} \\ >1 & \text{fin enhances heat transfer} \end{cases}$$

To increase **fins effectiveness**, one can conclude:

- The thermal conductivity of the fin material must be as high as possible
- The ratio of perimeter to the cross-sectional area p/Ac should be as high as possible
- The use of fin is most effective in applications that involve low convection heat transfer coefficient, i.e. natural convection.

Radiation

RADIATION HEAT TRANSFER:

(i) Thermal energy emitted by matter as a result of vibrational and rotational movements of molecules, atoms and electrons.

(ii) The energy is transported by electromagnetic waves (or photons).

(iii) Radiation requires no medium for its propagation, therefore, can take place also in vacuum.

BASIC DEFINITIONS:

TOTAL HEMISPHERICAL EMISSIVE POWER (E):

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(i) It is defined as the radiation energy emitted from the surface of a body per unit time & per unit area in all possible hemispherical directions integrated over all the wavelengths (in J/s m^2 or W/m^2).

BLACK BODY:

(i) Different bodies emits different amount of radiation even if they are at same temperature.

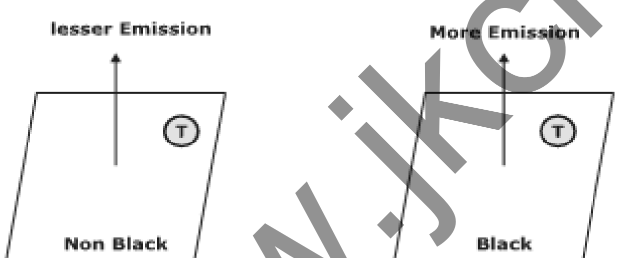
(ii) A body which emits maximum amount of radiation at a given temperature is known as **BLACK BODY**.

(iii) A black body is defined as perfect emitter and absorber of radiation.

TOTAL EMISSIVITY (Є):

Total emissivity of a body is defined as the ratio between total hemispherical emissive power of a non-black body & total hemispherical emissive power of a black body both being at the same temperature.

$$\epsilon = \frac{E}{E_b}$$



ABSORPTIVITY (α), REFLECTIVITY (ρ) & TRANSMISSIVITY (τ)

- Absorptivity(α) is the fraction of incident radiation absorbed.
- Reflectivity(ρ) is the fraction of incident radiation reflected.
- Transmissivity (τ) is the fraction of incident radiation transmitted.

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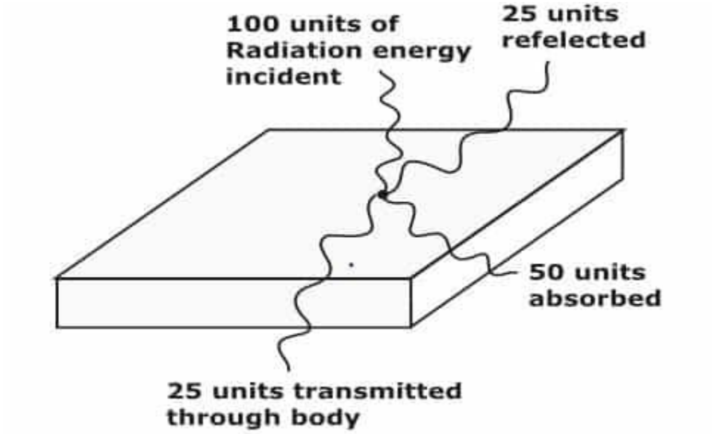

ABSORPTIVITY (α) = 50/100 = 0.50 = Fraction of radiation energy incident upon a surface which is absorbed by it.

REFLECTIVITY (ρ) = 25/100 = 0.25 = Fraction of radiation energy incident upon a surface which is reflected by it.

TRNASMISSIVITY (τ) = 25/100 = 0.25 = Fraction of radiation energy incident upon a surface which is transmitted by it.

- For any Body:

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

- For opaque body, which do not transmit any energy,

$\tau = 0, \ \alpha + \rho = 1$

- For black body, α = 1 [as it absorbs all energy incident]
- For white body, which reflects all energy incident, τ = 1

LAWS OF THERMAL RADIATION:

KIRCHOFF'S LAW OF RADIATION:

The law states that whenever a body is in thermal equilibrium with its surrounding's its emissivity is equal to its absorptivity.

$\alpha = \epsilon$

STEFAN BOLTZMANN'S Law:

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The law states that the total hemispherical emissive power of a black body is directly proportional to fourth power of the absolute temperature of black body.

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

Stefan Boltzmann Constant= 5.67×10^{-8} $W/m^2 k^4$

PLANCK'S LAW OF THERMAL RADIATION:

The planck law describes the theoretical spectral distribution for the emissive power of a black body which is the amount of radiation energy emitted by a blackbody at a thermodynamic temperature T per unit time per unit surface area and per unit wavelength.

$$(E_\lambda)_h = \frac{C_1 \lambda^{-5}}{\exp[C_2 / \lambda T] - 1}$$

WEIN'S DISPLACEMENT LAW:

The wavelength at which $E_{b\lambda}$ will be maximum at a specified temperature can be found out by differentiating $E_{b\lambda}$ w.r.t λ and then putting equal to zero.

We get, $\lambda_m T = 2897.8$ μmK

LAMBERT'S COSINE LAW:

The intensity of radiation in a direction θ from the normal to a diffuse emitter is proportional to cosine of the angle θ.

$I = I_n \cos\theta$, I_n = normal intensity of radiation

RADIATION INTENSITY:

The ideal intensity I_b is defined as the energy emitted from an ideal body(black body which is diffuse emitter) ,per unit projected area, per unit time ,per unit solid angle.

SHAPE FACTOR OR VIEW FACTOR OR CONFIGURATION FACTOR:

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Radiation heat transfer between surfaces depends upon the orientation of the surfaces relative to each other as well as their radiation properties and temperatures.

To account for the effects of orientation on radiation heat transfer between the surfaces,a new parameter **VIEW FACTOR** is defined which is purely **geometric quantity** and is independent of radiation properties and temperatures.

Assumptions:

- The surfaces are diffuse emitter.
- The radiation that strikes a surface need not to be absorbed by that surface.
- Radiation that strikes a surface after being reflected by other is not considered in the evaluation of view factor.

The radiation shape factor is represented by the symbol F_{ij} ,

which means the shape factor form a surface, A_i to another surface A_j.

Thus the radiation shape factor F_{12} of surface A_1 to surface A_2 is

$$F_{12} = \frac{\text{direct radiation from surface 1 incident upon surface 2}}{\text{total radiation from emitting surface 1}}$$

SHAPE FACTOR RELATIONS:

(i) RECIPROCITY RELATION:

$A_1F_{12} = A_2 F_{21}$

Reciprocity Relation is valid between any two surfaces even when there are more than two number of surfaces involved in Radiation Heat Exchange.

(ii) SUMMATION RULE:

If there are n number of surfaces involved in any radiation heat exchange then.

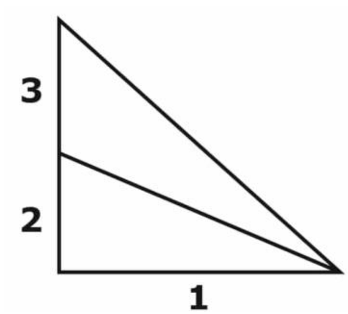
$F_{11} + F_{12} + F_{13} + \ldots\ldots\ldots\ldots\ldots F_{1n} = 1$

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

.

$F_{n1} + F_{n2} + F_{n3} + \ldots\ldots\ldots.F_{nn} + 1$

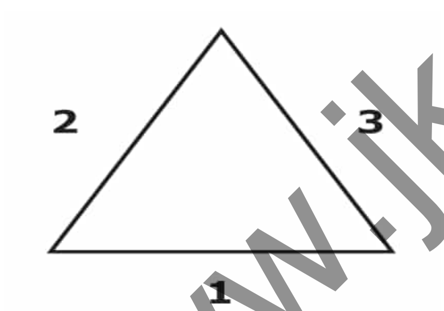
SUPERPOSITION RULE:



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

SYMMETRY RULE:

Two or more surfaces that possess symmetry about a third surface will have identical view factors from that surface.



If the surface 2 and 3 are symmetric about surface 1 then $F_{1-2}=F_{1-3}$

GENERAL CASES:

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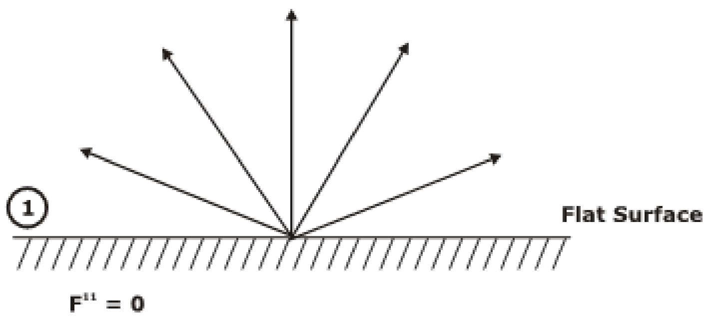
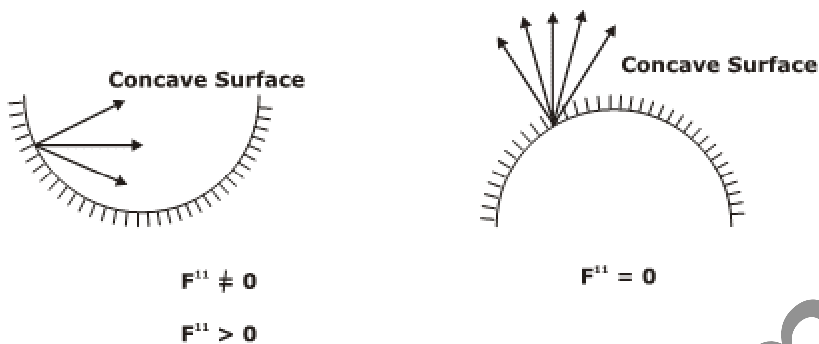
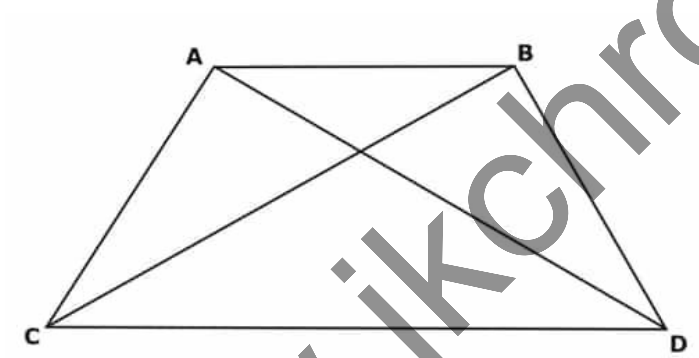




CROSS STRING METHOD:



The each length is assumed to be a strings connected between two points.

Cross Strings → The strings which crosses each other.

Uncrossed strings → The strings which do not cross each other.

$$F_{ij} = \frac{\sum \text{Crossed String} - \sum \text{Uncrossed string}}{2L_i}$$

L_i = length of string on surface i.

AB → 1 surface

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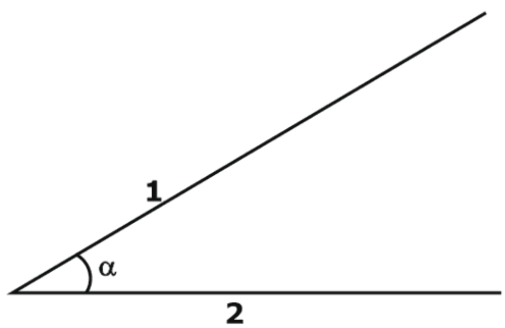
CD → 2 surface

$$F_{12} = \frac{(L_{AD} + L_{BC}) - (L_{AC} + L_{BD})}{2L_{AB}}$$

Special Case:



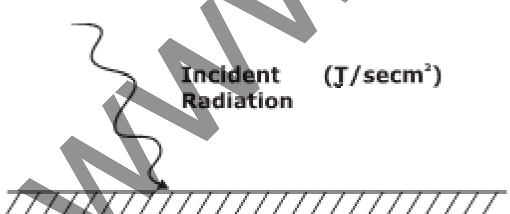
$(Length)_1 = (Length)_2$ Both surfaces are infinite in plane perpendicular to plane of paper.

$$F_{12} = 1 - \sin\frac{\alpha}{2}$$

RADIATION NETWORKS:

IRRADIATION (G):

Irradiation (G) is defined as the total thermal radiation incident upon a surface per unit time per unit area (in W/m^2)



RADIOSITY (J):

The total thermal radiation leaving a surface per unit time & per unit area is called radiosity of the surface.

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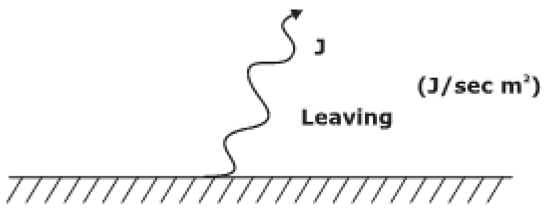
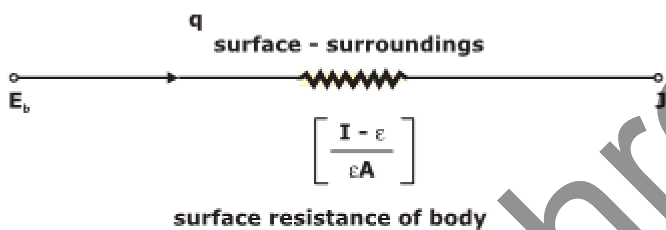


J = Emitted Energy + Reflected Part of Incident energy,

The net radiation heat exchange between the surface & all of its surroundings is given by,

$$q_{net} = \frac{E_b - J}{\left(\frac{1-\epsilon}{\epsilon A}\right)} \text{ watts}$$

The equivalent radiation circuit is,

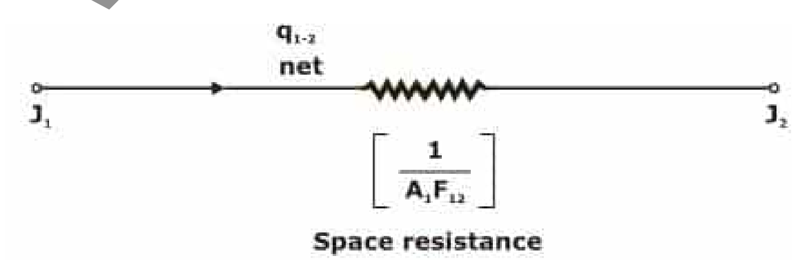


$$\left(\frac{1-\epsilon}{\epsilon A}\right) = \text{Surface resistance of body}$$

Radiation heat exchange between two finite surfaces:

$$\left(q_{1-2}\right)_{net} = \frac{\left(J_1 - J_2\right)}{\left(\frac{1}{A_1 F_{12}}\right)} \text{ watts}$$

,The equivalent radiation circuit is



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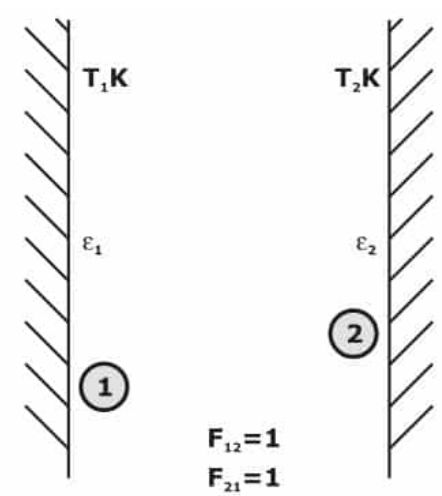
This space resistance shall exist in the space between two surfaces exchanging heat by radiation.

Radiation Exchange Between two Infinitely Large Plane Surfaces:



$$\text{Radiation Heat Flux} = \frac{E_{b1} - E_{b2}}{\left(\frac{1}{\epsilon_1} + \frac{1}{\epsilon_2} - 1\right)}$$

Radiation Shields:

$F_{12} = 1$, $F_{21} = 1$

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

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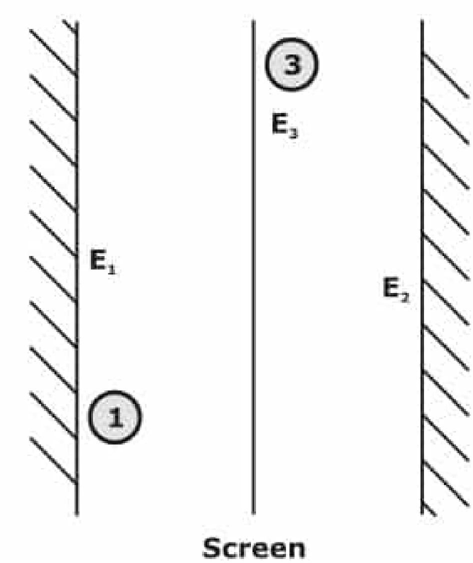

Since surfaces are very large

$A_1 = A_2 = A_3 = 1$ and $F_{13} = 1$, $F_{32} = 1$

Therefore,

$$\left(\frac{Q}{A}\right)_{(1-2)} = \frac{\sigma\left(T_1^4 - T_2^4\right)}{\frac{1}{\in_1} + \frac{1}{\in_2} + \frac{2}{\in_3} - 2}$$

Heat Exchangers

HEAT EXCHANGERS:

Introduction:

Heat exchanger is a steady flow adiabatic device (open system) in which two flowing fluids exchange or transfer heat between them without using or gaining any heat from the ambient.

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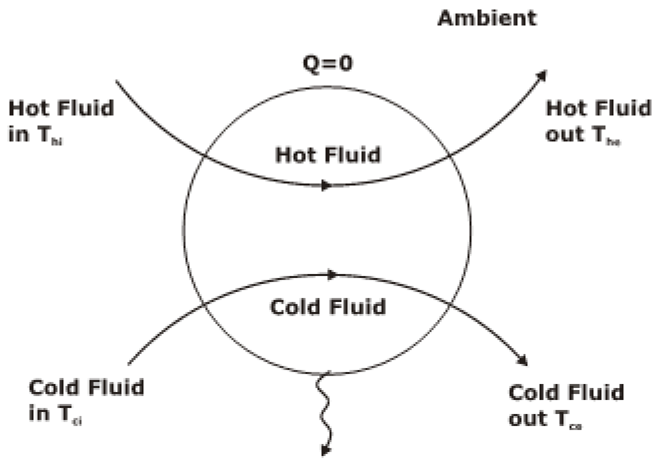

Examples:

(1) Economizer [Flue gases – Feed water]

(2) Air Preheater [Hot gases – Combustion air]

(3) Superheater [Flue gases – dry saturated Steam]

(4) Cooling Tower [Hot water –Atmosphere air]

(5) Jet Condenser [Steam – cold water]

(6) Oil cooler

(7) Steam Condenser (Vapour – Liquid): Steam condenser is a shell & tube Heat Exchanger.

From thermodynamics, we know that heat transferred in a constant pressure process is equal to change in enthalpy of fluid and as we commonly assume in any heat exchanger that both hot & cold fluids are flowing through the heat exchanger at constant pressure.

Rate of decrease of enthalpy of hot fluid = Rate of increase of enthalpy of cold fluid

$$\dot{m}_h C_{ph} (T_{hi} - T_{he}) = \dot{m}_c C_{pc} (T_{ce} - T_{ci})$$

CLASSIFICATION OF HEAT EXCHANGERS ON BASIS OF CONTACT:

(i) Indirect contact transfer type heat exchangers:

In this type of heat exchanger, both hot & cold fluids do not come into direct contact with each other but the transfer of heat occurs between them through a pipe wall of separation.

Examples:

(1) Surface condenser

(2) Economizer

(3) Superheater

(ii) Direct contact transfer type heat exchangers:

In this type of heat exchanger, both hot & cold fluids physically mix up with each other & exchange heat.

Example:

(1) Cooling Tower

(2) Jet condenser

CLASSIFICATION OF HEAT EXCHANGER ON BASIS OF RELATIVE DIRECTION:

(i) Parallel flow heat exchanger:

In this heat exchanger, both Hot & Cold Fluid travel in the same direction parallel to each other.

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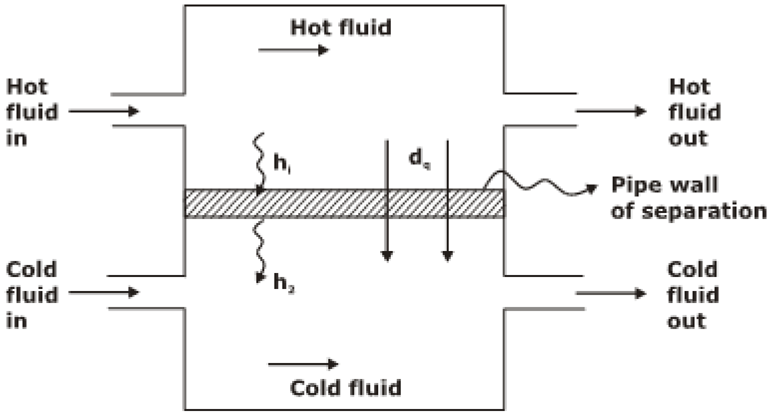
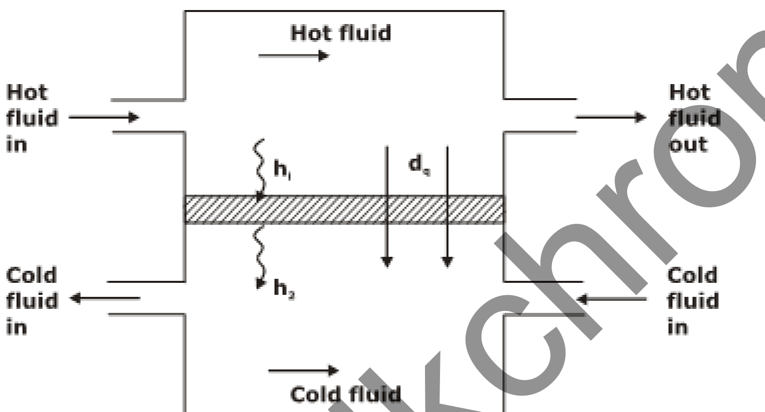


(ii) Counterflow heat exchanger:

In this heat exchanger, Hot & Cold Fluid travel in the opposite direction parallel to each other.



(iii) Cross flow heat exchanger:

In this heat exchanger, Hot & Cold fluids travel in the perpendicular direction with respect to each other.

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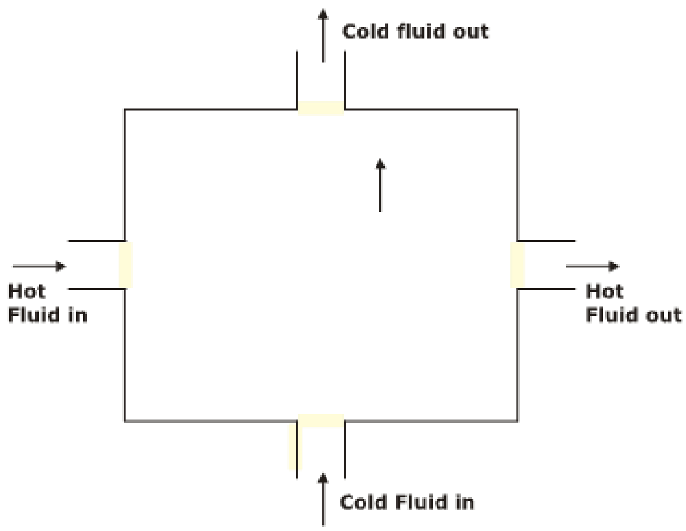
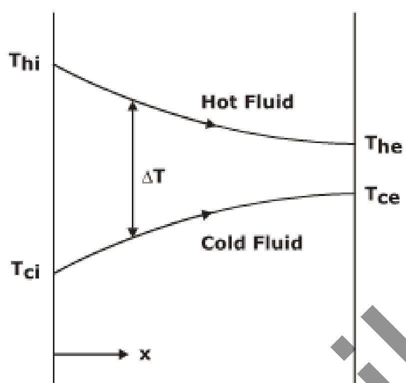
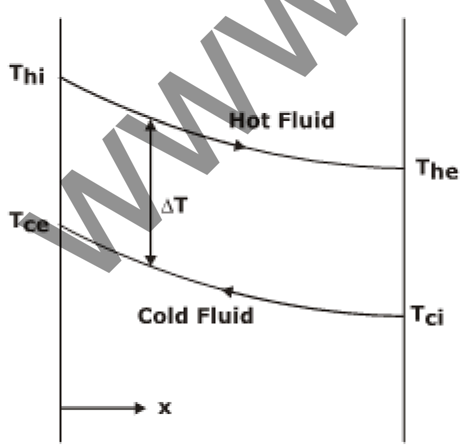


TEMPERATURE PROFILES OF HOT AND COLD FLUIDS:

(i) Parallel flow heat exchanger:



(ii) Counterflow heat exchanger:



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The variation of ΔT with respect to x is much more pronounced in parallel-flow heat exchanger as compared to that in counter-flow heat exchanger.

LOGARITHMIC MEAN TEMPERATURE DIFFERENCE (LMTD) OF HEAT EXCHANGER:

Logarithmic mean temperature difference is that constant temperature difference (maintained throughout heat exchanger) which would give same amount of heat transfer.

$Q = U A \Delta T_m$

Where,

Q = total Heat Transfer Rate between hot & cold fluid

U = Overall Heat Transfer coefficient

ΔT_m = Logarithmic Mean Temperature Difference between hot and cold fluid

A = Total Heat Transfer Area of Heat Exchanger

(i) LMTD for Parallel flow heat exchanger:

At $x = 0$, $\Delta T = \Delta T_i = T_{hi} - T_{ci}$

At $x = L$, $\Delta T = \Delta T_e = T_{he} - T_{ce}$

$$\text{LMTD for parallel flow H.E.} = \frac{\Delta T_i - \Delta T_e}{\ln \frac{\Delta T_i}{\Delta T_e}}$$

(ii) LMTD for Counterflow heat exchanger:

At $x = 0$, $\Delta T = \Delta T_i = T_{hi} - T_{ce}$

At $x = L$, $\Delta T = \Delta T_e = T_{he} - T_{ci}$

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$$\text{LMTD for Counter flow H.E.} = \frac{\Delta T_i - \Delta T_e}{\ln \frac{\Delta T_i}{\Delta T_e}}$$

For the same inlet & exit temperature of hot & cold fluids, The value of LMTD for counter-flow heat exchanger is more than that for parallel flow heat exchanger.

(iii) LMTD for Cross flow heat exchanger:

LMTD for cross-flow heat exchanger can be obtained as

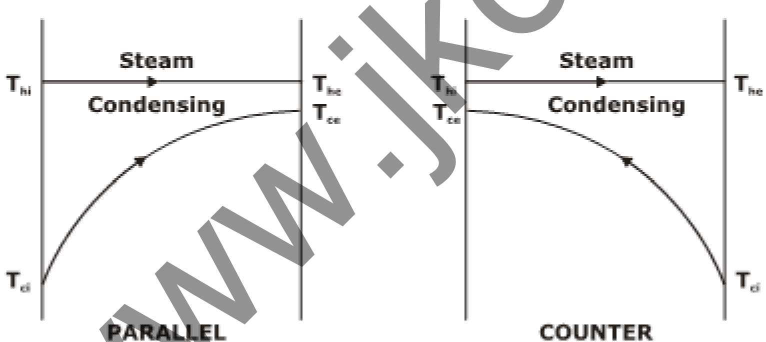
$$(\Delta T_m)_{\text{cross flow}} = (\Delta T_m)_{\text{counter flow}} \times F$$

Where, F = correction factor,

SPECIAL CASE REGARDING LMTD:

When one of the fluids is undergoing change of phase like in **steam condenser** or **evaporator** or **steam generator** then,

Steam condenser:



$(\Delta T_m)_{\text{parallel flow}} = (\Delta T_m)_{\text{counter flow}}$

Effectiveness of heat exchanger:

It is defined as the ratio between actual heat transfer rate that is taking place between hot & cold fluids to the maximum possible heat transfer rate that can occur between them,

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$$\epsilon_{HE} = \frac{q_{actual}}{q_{max\ possible}}$$

q_{actual} = Rate of enthalpy change of either fluid

$q_{actual} = m_h C_{ph} (T_{hi} - T_{he}) = m_c C_{pc} (T_{ce} - T_{ci})$

q_{max} = maximum possible heat transfer rate= $(m\ C_p)_{small} \times (T_{hi} - T_{ci})$

Where $(m\ C_p)_{small}$ is the smaller capacity rate between hot & cold fluids.

Number of Transfer Units (NTU):

It is defined as the ratio between product of UA & small capacity rate between hot & cold fluids.

$$NTU = \frac{UA}{(m\,Cp)_{small}}$$

NTU being directly proportional to the area of the heat exchanger indicates **overall size of the heat exchanger**.

Capacity Rate Ratio (C):

$$C = \frac{(m\,C_p)_{small}}{(m\ C_p)_{large}}$$

C will become zero when one of the fluids is undergoing change of phase like **steam condenser.**

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

Effectiveness for the parallel flow heat exchanger:

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

Effectiveness for the counterflow heat exchanger:

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

SPECIAL CASES:

CASE 1: when one of the fluids is undergoing change of phase, then C is equal to zero.

Hence,

$$\epsilon_{parallel\ flow} = \epsilon_{counter\ flow} = 1 - e^{-NTU}$$

CASE 2: The other extreme value of C=1, when both hot & cold fluids have equal capacity rates ($m_h Cp_h = m_c C_{pc}$)

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

$$\epsilon_{counterl\ flow} = \frac{NTU}{1 + NTU}$$

Free and Forced Convection

Convection: Convection is the mechanism of heat transfer that occurs between a solid surface & the surrounding fluid in the presence of bulk fluid motion.

Convection is classified as **Natural (or free) and Forced convection** depending on how the fluid motion is initiated.

CONVECTION	
Forced	Free or Natural
(velocity is evident)	(No velocity is evident but flow occurs due to buoyant forces arising out of density changes of fluid)

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The rate of convection heat transfer is expressed by Newton's law of cooling and it is given as **Forced Convection**

$$q_{conv} = h(T_s - T_\infty)\ W/m^2$$
$$Q_{conv} = hA(T_s - T_\infty)\ W$$

Where h is the heat transfer coefficient and A is the area of the contact.

The convective heat transfer coefficient h strongly depends on the fluid properties and roughness of the solid surface, and the type of the fluid flow (laminar or turbulent).

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

Boundary layer:

Hydrodynamic Boundary Layer:

Hydrodynamic Boundary Layer is defined as a thin region formed on the plate inside which velocity gradients are seen in a normal direction to the plate.

$$\text{Local Reynold's Number } (Re_x) = \frac{V_\infty x \rho}{\mu} = \frac{V_\infty x}{\upsilon}$$

(at any x measured from **leading edge**)

If $Re_x < 5 \times 10^5$, then flow is **LAMINAR**

If $Re > 6.5$ to 7×10^5, Then flow is **TURBULENT**

CASE 1: Flow over flat plates:

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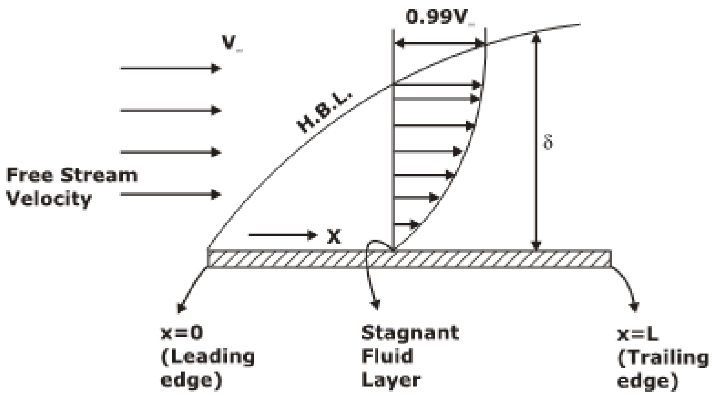

Hydrodynamic boundary layer:

δ → Thickness of Hydrodynamic Boundary Layer at any distance x measured from leading-edge [i.e., x = 0] of the plate.

$\delta = f(x)$

Thermal boundary layer:

Similar to velocity boundary layer, a thermal boundary layer develops when a fluid at specific temperature flows over a surface which is at different temperature.

$T = f(x,y)$

δ_t = Thickness of thermal Boundary Layer

$\delta_t = f(x)$

The Thermal boundary layer thickness is defined as the distance measured form the solid boundary in y direction at which

$$\frac{T_s - T}{T_s - T_\infty} = 0.99$$

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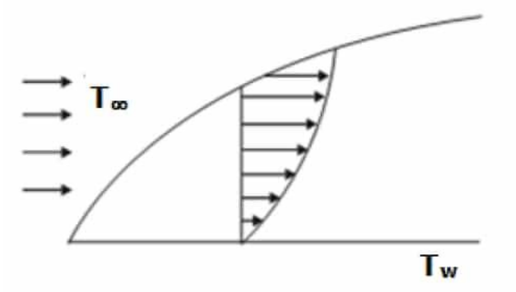
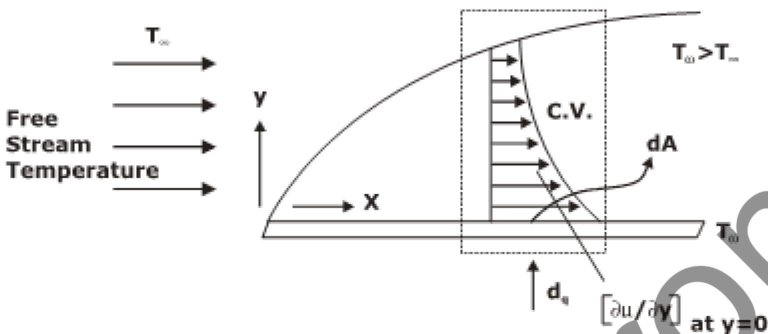


Thermal boundary layer (T_∞ > T_w):

Energy balance for thermal boundary layer in a control volume:



T = f (x, y)

Assuming steady-state conditions for the control volume,

dq = Heat conducted through the stagnant fluid layer at y = 0 = Heat convected between the plate & free stream fluid

$$dq = -K_f dA\left(\frac{\partial T}{\partial y}\right)_{at\,y=0} = h_x dA\left(T_w - T_\infty\right)$$

$$h_x = \frac{-K_f\left(\frac{\partial T}{\partial y}\right)_{at\,y=0}}{\left(T_w - T_\infty\right)}\ w/m^2k$$

Where, h_x = Local convective heat transfer coefficient at that x,

variation of h_x with x :

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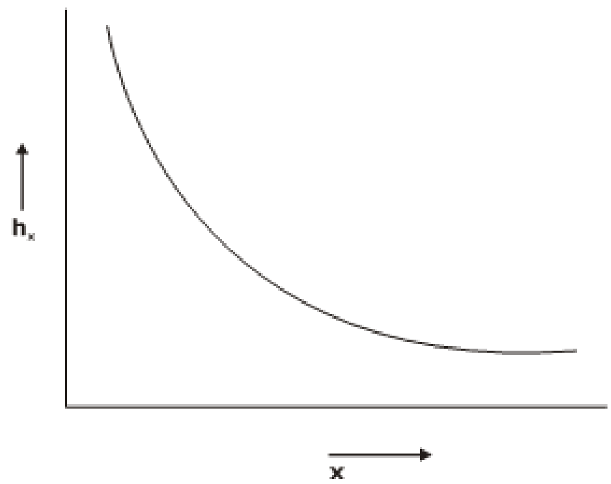

Nusselt Number (Nu):

$$[Nu] = \frac{\text{conductive heat transfer resistance}}{\text{convective heat transfer resistance}} = \frac{hL}{k_f}$$

Prandtl Number (Pr):

$$[Pr] = \frac{\text{Momentoum diffusivity}}{\text{thermal diffusivity}} = \frac{\nu}{\alpha} = \frac{\mu C_p}{k}$$

Peclet Number (Pe):

It is the product of Reynold number and Prandtl number.

Peclet number = Reynold number × Prandtl number

Stanton Number (St):

It is the ratio of nusselt number to the Peclet number.

$$\text{Stanton number} = \frac{\text{Nusselt number}}{\text{peclet number}}$$

$$St = \frac{\text{Nusselt number}}{\text{Reynold number} \times \text{Prandtl number}}$$

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

Relation between thermal boundary layer and hydrodynamic boundary layer:

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$$\frac{\delta_t}{\delta} = \frac{1}{1.026}(\Pr)^{-\frac{1}{3}}$$

Reynold Colburn analogy:

$$St \times \Pr^{\frac{2}{3}} = \frac{C_f}{2}$$

Flow Over Flat Plate:

Laminar Flow:

Local Nusselt number at the location x for laminar flow over a flat plate is,

$$Nu_x = \frac{h_x x}{k} = 0.332\ Re_x^{1/2} \Pr^{1/3}$$

Average Nusselt number for the entire length of the plate,

$$\overline{Nu} = \frac{\bar{h}L}{k} = 0.664\ Re_L^{1/2} \Pr^{1/3}$$

$\overline{Nu}$ = Average Nusselt Number

The local friction coefficient at the location x for **laminar flow over a flat plate** is

$$c_{f_x} = \frac{0.664}{\sqrt{R_{e_x}}}$$

where x is the distance from the **leading edge of the plate** and

Average local friction coefficient for entire length of the plate

$$c_{f_L} = \frac{1.328}{\sqrt{R_{e_L}}}$$

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$$Re_x = \frac{\rho V x}{\mu} = \frac{vx}{\nu}$$

Critical Reynolds number for flow over flat plate is 5×10^5

Turbulent Flow:

Local Nusselt number at location x for turbulent flow over a flat isothermal plate are:

$$Nu_x = 0.0296 (Re_x)^{0.8} (pr)^{\frac{1}{3}}$$

$$h_x = \frac{Nu_x \times k_f}{x} = 0.0296 \times \frac{k_f}{x} \times (Re_x)^{0.8} (pr)^{\frac{1}{3}}$$

Average Nusselt number:

$$Nu_L = 0.037 (Re_L)^{0.8} (Pr)^{\frac{1}{3}}$$

$$h_L = \frac{Nu_L \times k}{L} = 0.037 \times \frac{k}{L} \times (Re_L)^{0.8} (pr)^{\frac{1}{3}}$$

Local friction coefficient:

$$\frac{\delta}{x} = \frac{0.371}{(R_{e_x})^{\frac{1}{5}}}$$

$$c_{f_x} = \frac{0.059}{(R_{e_x})^{\frac{1}{5}}}$$

Average friction coefficient:

$$c_{f_L} = \frac{0.074}{(R_{e_L})^{\frac{1}{5}}}$$

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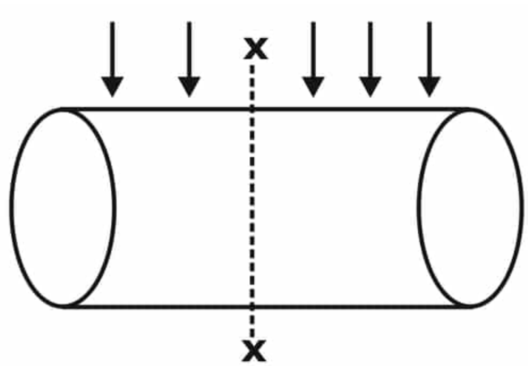
Pipe Flow:

Laminar Flow:

The **bulk mean temperature** (T_m) at a given cross-section of the pipe of flowing fluid is defined as the constant temperature which takes into account the variation of temperature of fluid layers with respect to radius at that cross-section of the pipe and hence indicates the total thermal energy or enthalpy carried by the fluid through that cross-section.



Constant heat flux:

$$Nu = \frac{hL}{K_f} = 4.364$$

No dependence on Re and Pr.

Constant surface Temperature:

$$Nu = \frac{hL}{K_f} = 3.66$$

Turbulent Flow:

$$Nu = 0.023\,(Re_L)^{0.8}(Pr)^{1/3}$$

$$h = Nu \times \frac{k}{D} = 0.023\frac{k}{D}(Re_L)^{0.8}(Pr)^{1/3}$$

n = 0.3 for cooling and n = 0.4 for heating

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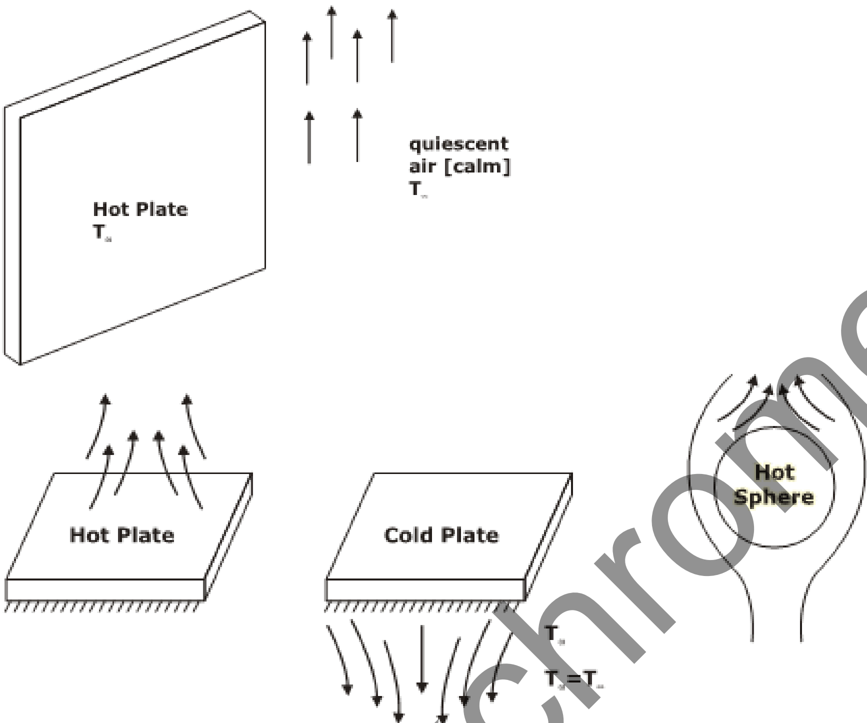
Free Convection or Natural Convection:

No velocity evident but the flow occurs due to buoyancy forces arising out of density changes of fluid.

Free convection over different geometries:



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In any free convection heat transfer,

h = f (g, β, ΔT, L, μ, ρ, C_p, k)

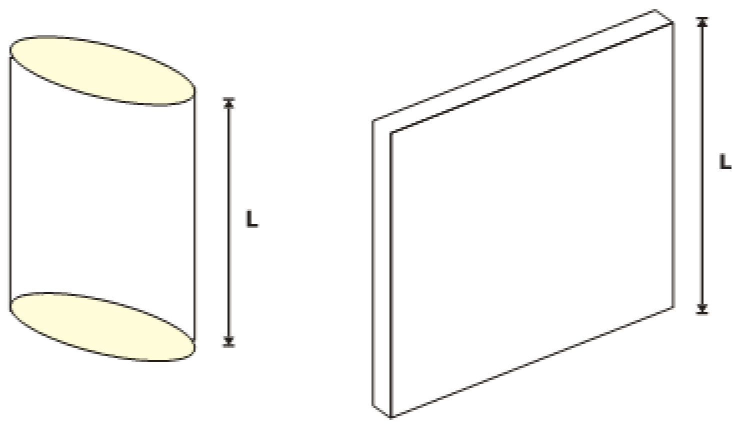
μ, ρ, Cp, k = Thermophysical Properties of fluid

β = Isobaric volume expansion coefficient of fluid [per kelvin]

$$\beta = \frac{1}{v}\left(\frac{\partial v}{\partial T}\right)_{p=c} \quad \text{[per kelvin]}$$

Characteristic dimension for various dimensionless numbers:

For vertical plates & vertical cylinders.



For horizontal plate for horizontal cylinder

Grash of Number (Gr):

where

g = gravitational acceleration, m/s^2

β = coefficient of volume expansion, 1/K

δ = characteristic length of the geometry, m

v = kinematics viscosity of the fluid, m^2/s

Grash of number (Gr) replaces Reynolds Number (Re) in free convection heat transfer.

Therefore, in any free convection heat transfer,

Heat transfer coefficient:

Nu = f (Gr Pr)

product of Gr & Pr is called Rayleigh Number (Ra)

$Nu = C\,(Gr\,Pr)^m$

C & m are constants which vary from case to case.

m = 1/4 for Laminar Flow

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m = 1/3 for Turbulent Flow

The flow during free convection heat transfer is decided as Laminar or Turbulent based on the value of (Gr Pr) product.

If Gr Pr < 10^9 Then, flow is Laminar

If Gr Pr > 10^9 Then, flow is Turbulent

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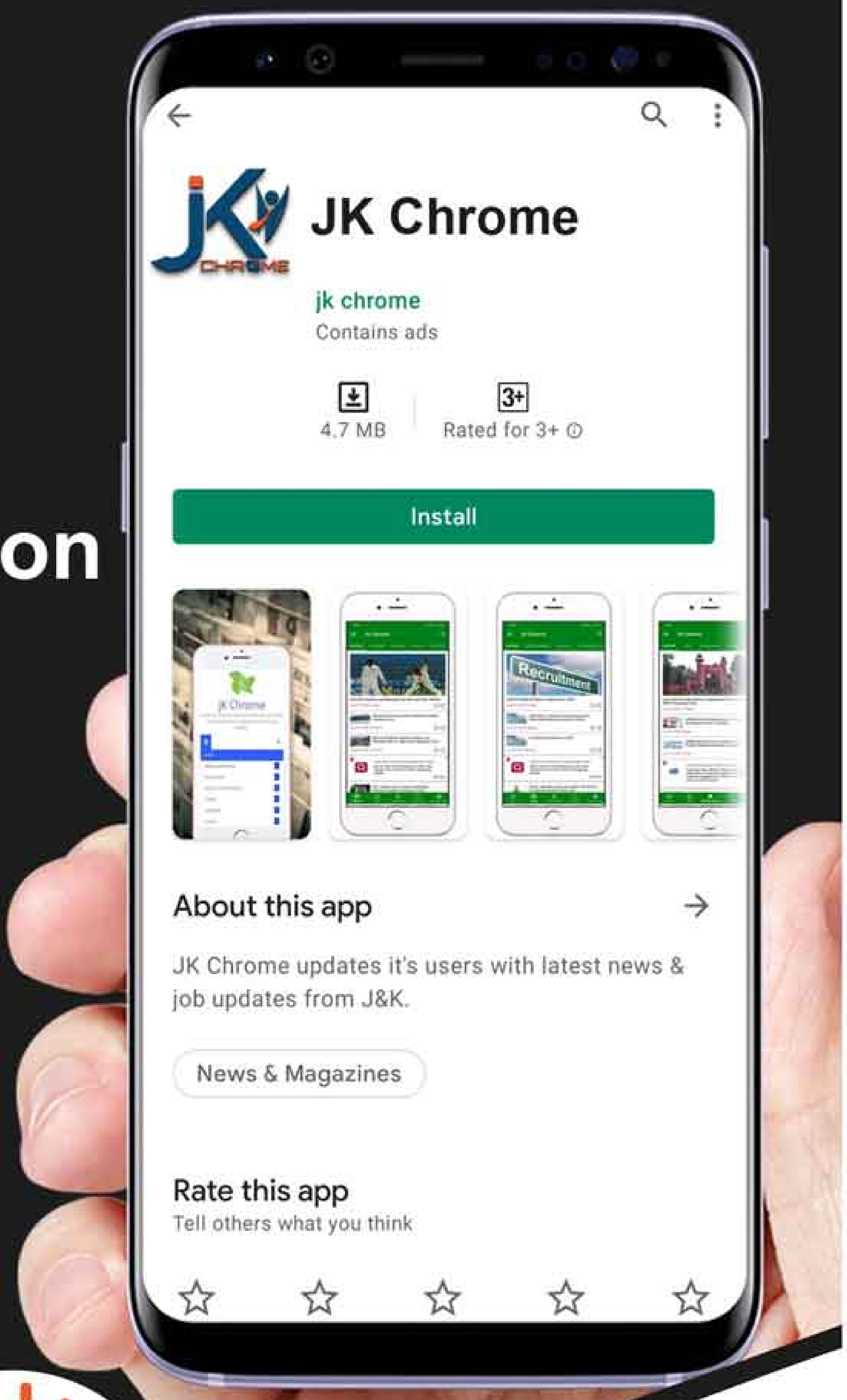

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