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New Scheme Based On AICTE Flexible Curricula

Mechanical Engineering, VII-Semester

ME- 701 Heat and Mass Transfer

Course Objectives:

After studying this course, students will be able to

1. Know about the basic concept of heat transfer and its modes.
2. Solve problems based on conduction, convection, and radiation.
3. Differentiate the modes of heat transfer i.e. conduction, convection, and radiation
4. Understand the working principle and types of heat exchangers.
5. Understand the concept of boiling and condensation, mass transfer.

Syllabus:

Unit-1 Basic Concepts: Modes of heat transfer, Fourier's law, Newton's law, Stefan Boltzman law; thermal resistance and conductance, analogy between flow of heat and electricity, combined heat transfer process; Conduction: Fourier heat conduction equation, its form in rectangular, cylindrical and spherical coordinates, thermal diffusivity, linear one dimensional steady state conduction through a slab, tubes, spherical shells and composite structures, electrical analogies, critical-insulation-thickness for pipes, effect of variable thermal conductivity.

Unit 2 Extended Surfaces (fins): Heat transfer from a straight and annular fin (plate) for a uniform cross section; error in measurement of temperature in a thermometer well, fin efficiency, fin effectiveness, applications; Unsteady heat conduction: Transient and periodic conduction, heating and cooling of bodies with known temperatures distribution, systems with infinite thermal conductivity, response of thermocouples.

Unit 3 Convection: Introduction, free and forced convection; principle of dimensional analysis, Buckingham 'pie' theorem, application of dimensional analysis of free and forced convection, empirical correlations for laminar and turbulent flow over flat plate and tubular geometry; calculation of convective heat transfer coefficient using data book.

Unit 4 Heat Exchangers: Types- parallel flow, counter flow; evaporator and condensers, overall heat transfers coefficient, fouling factors, log-mean temperature difference (LMTD), method of heat exchanger analysis, effectiveness of heat exchanger, NTU method;

Mass transfer: Fick's law, equi-molar diffusion, diffusion coefficient, analogy with heat transfer, diffusion of vapour in a stationary medium.

Unit 5 Thermal Radiation : Nature of radiation, emissive power, absorption, transmission, reflection and emission of radiation, Planck's distribution law, radiation from real surfaces; radiation heat exchange between black and gray surfaces, shape factor, analogical electrical network, radiation shields.

Boiling and condensation: Film wise and drop wise condensation; Nusselt theory for film wise condensation on a vertical plate and its modification for horizontal tubes; boiling heat transfer phenomenon, regimes of boiling, boiling correlations.

References:

1. Sukhatme SP; Heat and mass transfer; University Press Hyderabad
2. Holman JP; Heat transfer; TMH
3. Nag PK; heat and Mass Transfer; TMH
4. Domkudwar,Heat and Mass Transfer,Dhanpt Rai & Co.
5. Sachdeva R.C., Fundamentals of Engineering Heat and Mass Transfer, New Age Science
6. Dutta BK; Heat Transfer Principles And App; PHI Learning
7. Mills AF and Ganesan V; Heat transfer; Pearson
8. Cengel Yunus A; Heat and Mass transfer;TMH
9. Yadav R; Heat and Mass Transfer; Central India pub-Allahabad
10. Incropera FP and Dewitt DP; Heat and Mass transfer; Wiley

List of Experiments (Pl. expand it):

- 1 Conduction through a rod to determine thermal conductivity of material
- 2 Forced and free convection over circular cylinder
- 3 Free convection from extended surfaces
4. Parallel flow and counter flow heat exchanger effectiveness and heat transfer rate
5. Calibration of thermocouple
6. Experimental determination of Stefan-Boltzmann constant

Evaluation

Evaluation will be continuous an integral part of the class as well through external assessment.

StreamTechNotes

ME- 701 Heat and Mass Transfer
Unit-1 Basic Concepts:

Introduction

Heat is fundamentally transported, or “moved,” by a temperature gradient; it flows or is transferred from a high-temperature region to a low-temperature one. An understanding of this process and its different mechanisms are required to connect principles of thermodynamics and fluid flow with those of heat transfer.

Thermodynamics and Heat Transfer

Thermodynamics is concerned with the amount of heat transfer as a system undergoes a process from one equilibrium state to another, and it does not indicate how long the process will take. A thermodynamic analysis simply tells us how much heat must be transferred to realize a specified change of state to satisfy the conservation of energy principle.

In practice, we are more concerned about the rate of heat transfer (heat transfer per unit time) than we are with the amount of it. For example, we can determine the amount of heat transferred from a thermos bottle as the hot coffee inside cools from 90°C to 80°C by a thermodynamic analysis alone.

But a typical user or designer of a thermos is primarily interested in how long it will be before the hot coffee inside cools to 80°C, and a thermodynamic analysis cannot answer this question. Determining the rates of heat transfer to or from a system and thus the times of cooling or heating, as well as the variation of the temperature, is the subject of heat transfer (Figure 1).



Fig. 1 Heat transfer from the thermos

Thermodynamics deals with equilibrium states and changes from one equilibrium state to another, Heat transfer, on the other hand, deals with systems that lack thermal equilibrium, and thus it is a non-equilibrium phenomenon. Therefore, the study of heat transfer cannot be based on the principles of thermodynamics alone.

However, the laws of thermodynamics lay the framework for the science of heat transfer. The first law requires that the rate of energy transfer into a system be equal to the rate of increase of the energy of that system. The second law requires that heat be transferred in the direction of decreasing temperature (Figure 2).

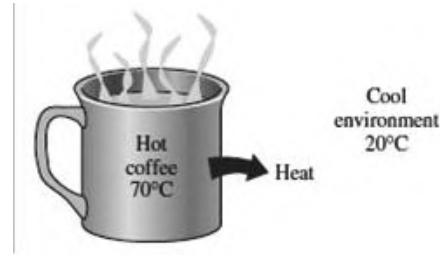


Fig. 2 Heat transfer from high temperature to low temperature

Application Areas of Heat Transfer

Many ordinary household appliances are designed, in whole or in part, by using the principles of heat transfer. Some examples:

Design of the heating and air-conditioning system, the refrigerator and freezer, the water heater, the iron, and even the computer, the TV, and the VCR

Energy-efficient homes are designed based on minimizing heat loss in winter and heat gain in summer. Heat transfer plays a major role in the design of many other devices, such as car radiators, solar collectors, various components of power plants, and even spacecraft.

The optimal insulation thickness in the walls and roofs of the houses, on hot water or steam pipes, or water heaters is again determined based on a heat transfer analysis with economic consideration (Figure 3)

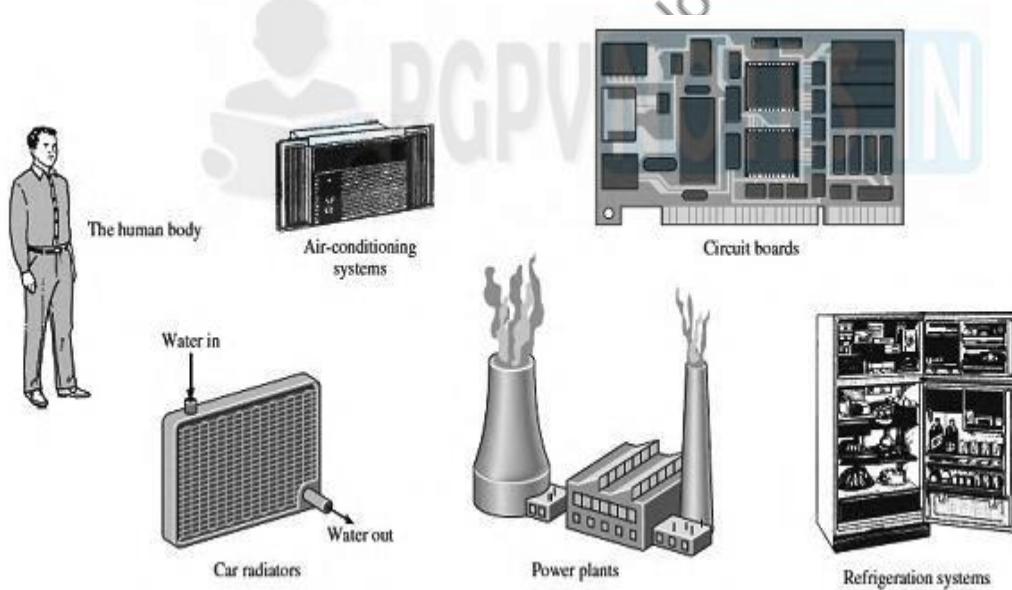


Fig. 3 Application of heat transfer

ENGINEERING HEAT TRANSFER

The heat transfer problems encountered in practice can be considered in two groups:

- i rating and
- ii Sizing problems.

The rating problems deal with the determination of the heat transfer rate for an existing system at a specified temperature difference.

The sizing problems deal with the determination of the size of a system to transfer heat at a specified rate for a specified temperature difference.

Modes of heat transfer :

Heat can be transferred in three different modes: conduction, convection, and radiation. All modes of heat transfer require the existence of a temperature difference, and all modes are from the high-temperature medium to a lower-temperature one.

Conduction : Fourier's law

Conduction is the transfer of energy from the more energetic particles of a substance to the adjacent less energetic ones as a result of interactions between the particles. Conduction can take place in solids, liquids, or gases.

In gases and liquids, conduction is due to the collisions and diffusion of the molecules during their random motion.

In solids, it is due to the combination of vibrations of the molecules in a lattice and the energy transport by free electrons.

The rate of heat conduction through a medium depends on the geometry of the medium, its thickness, and the material of the medium, as well as the temperature difference across the medium.

We know that wrapping a hot water tank with glass wool (an insulating material) reduces the rate of heat loss from the tank. Now we can say that the thicker the insulation, the smaller the heat loss.

We also know that a hot water tank will lose heat at a higher rate when the temperature of the room the tank is lowered. So that the larger the tank, the larger the surface area and thus the rate of heat loss.

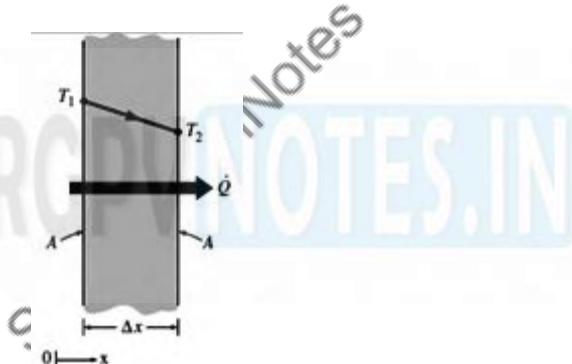


Fig. 4 Heat conduction through the large plain wall

Consider steady heat conduction through a large plane wall of thickness and area A, as shown in the figure. Experiments have shown that the rate of heat transfer through the wall is doubled when the temperature difference across the wall or the area normal to the direction of heat transfer is doubled but is halved when the wall thickness L is doubled.

Thus we conclude that the rate of heat conduction through a plane layer is proportional to the temperature difference across the layer and the heat transfer area, but is inversely proportional to the thickness of the layer. That is, Where the constant of proportionality k is the thermal conductivity of the material, which is a measure of the ability of a material to conduct heat. In the limiting case of the equation above reduces to the differential form

$$Q_{cond} = KA \frac{T_1 - T_2}{\Delta x} = -KA \frac{\Delta T}{\Delta x} \quad (w)$$

The relation above indicates that the rate of heat conduction in a direction is proportional to the temperature gradient in that direction.

Heat is conducted in the direction of decreasing temperature, and the temperature gradient becomes negative when temperature decreases with increasing x . The negative sign in Eq. ensures that heat transfer in the positive x -direction is a positive quantity.

The heat transfer area A is always normal to the direction of heat transfer.

Thermal Conductivity

The thermal conductivity of a material can be defined as the rate of heat transfer through a unit thickness of the material per unit area per unit temperature difference.

The thermal conductivity of a material is a measure of the ability of the material to conduct heat.

A high value for thermal conductivity indicates that the material is a good heat conductor, and a low value indicates that the material is a poor heat conductor or insulator.

Note that materials such as copper and silver that are good electrical conductors are also good heat conductors, and have high values of thermal conductivity.

Materials such as rubber, wood, and Styrofoam are poor conductors of heat and have low conductivity values.

Convection: Newton's law

Convection is the mode of energy transfer between a solid surface and the adjacent liquid or gas that is in motion, and it involves the combined effects of conduction and fluid motion.

The faster the fluid motion, the greater the convection heat transfers. In the absence of any bulk fluid motion, heat transfer between a solid surface and the adjacent fluid is by pure conduction.

The presence of bulk motion of the fluid enhances the heat transfer between the solid surface and the fluid, but it also complicates the determination of heat transfer rates.

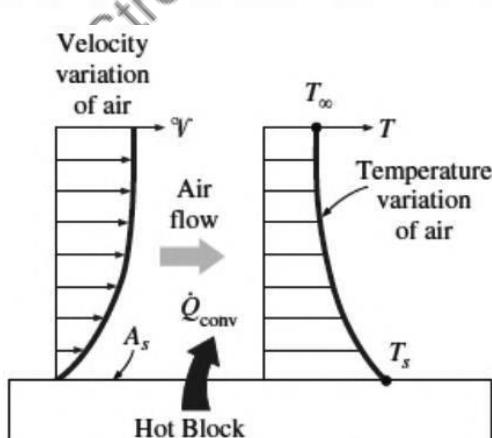


Fig. 5 Heat transfer by convection

Consider the cooling of a hot block by blowing cool air over its top surface (Figure 5).

Energy is first transferred to the air layer adjacent to the block by conduction.

This energy is then carried away from the surface by convection, that is, by the combined effects of conduction within the air that is due to the random motion of air molecules and the bulk or macroscopic motion of the air that removes the heated air near the surface and replaces it by the cooler air.

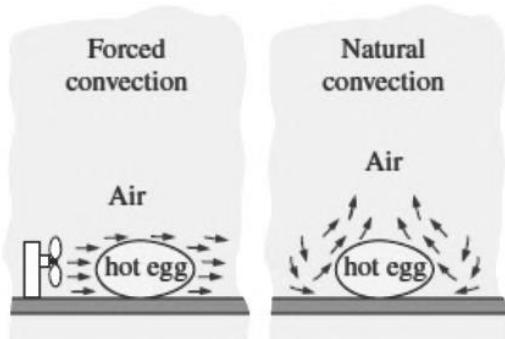


Fig. 6 Forced and Free (Natural) convection

Convection is called forced convection if the fluid is forced to flow over the surface by external means such as a fan, pump, or the wind.

In contrast, convection is called natural (or free) convection if the fluid motion is caused by buoyancy forces that are induced by density differences due to the variation of temperature in the fluid (Figure 6).

For example, in the absence of a fan, heat transfer from the surface of the hot block in figure 5 will be by natural convection since any motion in the air, in this case, will be due to the rise of the warmer (and thus lighter) air near the surface and the fall of the cooler (and thus heavier) air to fill its place.

Heat transfer between the block and the surrounding air will be by conduction if the temperature difference between the air and the block is not large enough to overcome the resistance of air to movement and thus initiate natural convection currents.

Heat transfer processes that involve a change of phase of fluid are also considered to be convection because of the fluid motion induced during the process, such as the rise of the vapour bubbles during boiling or the fall of the liquid droplets during condensation.

Despite the complexity of convection, the rate of convection heat transfer is observed to be proportional to the temperature difference and is conveniently expressed by

$$Q_{conv} = hA_s(T_s - T_\infty)(w)$$

Some people do not consider convection to be a fundamental mechanism of heat transfer since it is essentially heat conduction in the presence of fluid motion. But we still need to give this combined phenomenon a name, unless we are willing to keep referring to it as “conduction with fluid motion.”

Radiation: Stefan Boltzman law

Radiation is the energy emitted by matter in the form of electromagnetic waves (or photons) as a result of the changes in the electronic configurations of atoms or molecules.

Unlike conduction and convection, the transfer of energy by radiation does not require the presence of an intervening medium. Energy transfer by radiation is the fastest

The mechanism of the heat flow by radiation consists of three distinct phases:

1 Conversion of thermal energy of the hot source into electromagnetic waves:

All bodies above absolute zero temperature are capable of emitting radiant energy. The energy released by a radiating surface is not continuous but is in the form of successive and separate (discrete) packets or quanta of energy called photons. The photons are propagated through space as rays; the movement of the swarm of photons is described as electromagnetic waves.

2 Passage of wave motion through intervening space:

The photons, as carriers of energy, travel with unchanged frequency in straight paths with speed equal to that of light.

3 Transformation of waves into heat:

When the photons approach the cold receiving surface, there occurs reconversion of wave motion into thermal energy which is partly absorbed, reflected, or transmitted through the receiving surface. In heat transfer studies we are interested in thermal radiation, which is the form of radiation emitted by bodies because of their temperature. It differs from other forms of electromagnetic radiation such as x-rays, gamma rays, microwaves, radio waves, and television waves that are not related to temperature.

The maximum rate of radiation that can be emitted from a surface at an absolute temperature (in K) is given by the Stefan–Boltzmann law as

$$E_b = \sigma_b A T^4 (w)$$

$$\sigma_b = 5.67 * 10^{-8} \text{ W/m}^2 \text{K}^4$$

The radiation emitted by all real surfaces is less than the radiation emitted by a blackbody at the same temperature and is expressed as

$$E = \varepsilon \sigma_b A T^4 (w)$$

The net exchange of heat between the two radiating surfaces is due to the fact that one at the higher temperature radiates more and receives less energy for its absorption. An isolated body that remains at constant temperature emits just as much energy by radiation as it receives.

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

Introduction

The rate of heat conduction in a specified direction is proportional to the temperature gradient, which is the rate of change in temperature with distance in that direction. In one-dimensional steady-state heat conduction through homogenous material is given by Fourier Law of heat conduction:

$$Q = -KA \frac{dt}{dx}$$
$$q = \frac{Q}{A} = -KA \frac{dt}{dx}$$

Where,

q = heat flux, the heat conducted per unit time per unit area, Q = rate of heat flow, W

A = area perpendicular to the direction of heat flow,

dt = temperature difference between the two surfaces across which heat is passing, Kelvin K

dx = thickness of material along the path of heat flow, m

The ratio $\frac{dt}{dx}$ represents the change in temperature per unit thickness, i.e. the temperature gradient.

The negative sign indicates that the heat flow is in the direction of the negative temperature gradient, so heat transfer becomes positive.

The proportionality factor k is called the heat conductivity or thermal conductivity of material through which heat is transfer.

The Fourier law is essentially based on the following assumptions:

1. Steady state heat conduction, i.e. temperature at fixed point does not change with respect to time.
2. One dimensional heat flow.
3. Material is homogenous and isotropic, thermal conductivity has a constant value in all the directions.
4. Constant temperature gradient and a linear temperature profile.
5. No internal heat generation.

The Fourier law helps to define thermal conductivity of the material.

Hence **thermal conductivity** may be defined as the amount of heat conducted per unit time across unit area and through unit thickness, when a temperature difference of unit degree is maintained across the bounding surface.

Unit of thermal conductivity is given by:

$$K = -\frac{Qdx}{Adx}$$

$$K = -\frac{W}{m^2 deg} = \frac{W}{m - deg}$$

Thermal Resistance

In systems, which involve the flow of fluid, heat, and electricity, the flow quantity is directly proportional to the driving force and inversely proportional to the flow resistance.

In a hydraulic system, the pressure along the path is the driving potential, and the roughness of the pipe is the flow resistance.

The current flow in a conductor is governed by the voltage potential and electrical resistance of the material.

Likewise, temperature difference constitutes the driving force for heat conduction through a medium.

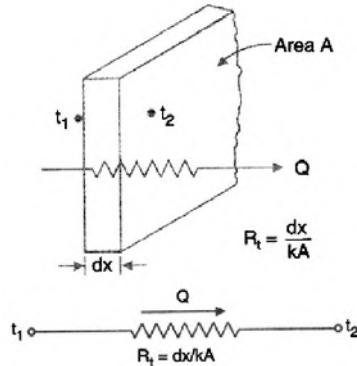


Fig. 7 Concept of thermal resistance

From Fourier's law:

$$\text{heat flow } Q = \frac{\text{temperature potential } (dt)}{\text{thermal resistance } (\frac{dx}{kA})}$$

Thermal resistance $R_t = (dx/kA)$ is expressed in the unit deg/w.

The reciprocal of thermal resistance is called thermal conductance and it represents the amount of heat conducted through a solid wall of area A and thickness dx when a temperature difference of unit degree is maintained across the bounding surfaces.

Thermal Conductivity of Materials

Thermal conductivity is a property of the material and it depends upon the material structure, moisture content and density of the material, and operating conditions of pressure and temperature.

The following remarks apply to the thermal conductivity and its variation for different materials and under different conditions:

In material thermal conductivity is due to two effects: the lattice vibration waves and flow of free electrons.

In metals the molecules are closely packed; molecular activity is rather small and so thermal conductivity is mainly due to the flow of free electrons.

In fluids, the free electron movement is negligibly small so conductivity mainly depends upon the frequency of interactions between the lattice atoms.

Thermal conductivity is highest in the purest form of a metal. Alloying of metals and the presence of other impurities reduce the conductivity of the metal.

The conductivity of hardened steel is lower than that of annealed steel.

At elevated temperatures, the thermal vibration of the lattice becomes higher and that retards the motion of free electrons. So, the thermal conductivity of metal decreases with increases in temperature except for the aluminium and uranium.

The thermal conductivity of aluminium remains almost constant within the temperature range of 130 to 370.

For uranium, heat conduction depends mainly upon the vibration movement of atoms. With an increase of temperature vibration movement increases so, conductivity also increases.

According to the kinetic theory of, the conductivity of gases is directly proportional to the density of the gas, mean molecular speed and mean free path. With the increase of temperature molecular speed increases, so conductivity of gas increases. The conductivity of gas is independent of pressure except in extreme cases, for example when conditions approach that of a perfect vacuum.

Molecular conditions associated with the liquid state are more difficult to describe, and physical mechanisms for explaining the thermal conductivity are not well understood. The thermal conductivity of non-metallic liquids generally decreases with increasing temperature. Water, glycerine, and engine oil are notable exceptions. The thermal conductivity of liquids is usually insensitive to pressure except near the critical point.

Thermal conductivity is only very weakly dependent on pressure for solids and liquids as well, and essentially dependent on pressure for gases at a pressure near-standard atmospheric.

For most materials, the dependence of thermal conductivity on temperature is almost linear.

Non-metallic solids do not conduct heat as efficiently as metals.

The ratio of the thermal and electrical conductivities is the same for all metals at the same temperature; and that the ratio is directly proportional to the absolute temperature of the metal.

General Heat Conduction Equations

The objective of conduction analysis is twofold:

- i To determine the temperature distribution within the body
- ii To make calculation of heat transfer.

Fourier law of heat conduction is essentially valid for heat flow under uni-directional and steady-state conditions, but sometimes it is necessary to consider heat flow in other directions as well.

So for heat transfer in multi-dimensional, it is necessary to develop a general heat conduction equation in rectangular, cylindrical, and spherical coordinate systems.

Cartesian (Rectangular) Co-ordinates:-

Consider the flow of heat through an infinitesimal volume element oriented in a three-dimensional coordinate system as shown in figure 8. The side's Δx , Δy and Δz have been taken parallel to the x, y, and z axis respectively.

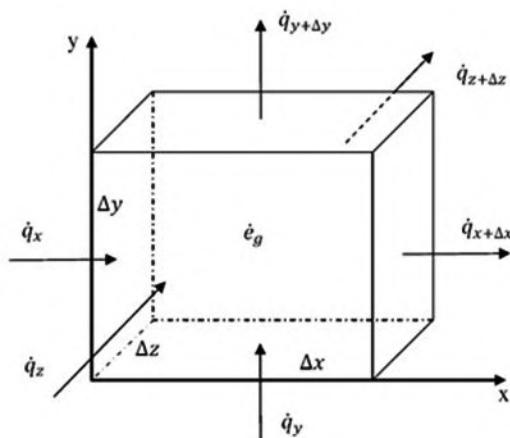


Fig. 8 Conduction analysis in Cartesian co ordinates

The general heat conduction equation can be set up by applying the Fourier equation in each Cartesian direction and then applying the energy conservation requirement.

If k represents the thermal conductivity at the left face, then the quantity of heat

Flowing into the control volume through the face during a time interval is given by:

Heat in flux:

$$q_g dx dy dz d\tau$$

$$Q_x = -k_x (dy dz) \frac{\partial t}{\partial x} d\tau$$

During the same time interval, the heat flow out of the element will be,

Heat efflux

$$Q_{x+dx} = Q_x + \frac{\partial Q_x}{\partial x} dx$$

Heat accumulated within the control volume due to heat flow in the x-direction is given by the difference between heat influx and heat efflux.

Thus the heat accumulation due to heat flow in the x-direction is

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

$$dQ_x = \frac{\partial}{\partial x} \left(K_x \frac{\partial t}{\partial x} \right) dx dy dz d\tau$$

Likewise, the heat accumulation in the control volume due to heat flow along the y-and z-directions will be:

$$dQ_y = \frac{\partial}{\partial y} \left(K_y \frac{\partial t}{\partial y} \right) dx dy dz d\tau$$

$$dQ_z = \frac{\partial}{\partial z} \left(K_z \frac{\partial t}{\partial z} \right) dx dy dz d\tau$$

Total heat accumulated due to heat transfer is given by

$$\left[\frac{\partial}{\partial x} \left(K_x \frac{\partial t}{\partial x} \right) + \frac{\partial}{\partial y} \left(K_y \frac{\partial t}{\partial y} \right) + \frac{\partial}{\partial z} \left(K_z \frac{\partial t}{\partial z} \right) \right] dx dy dz d\tau$$

$$q_g dx dy dz d\tau$$

The total heat accumulated in the control volume due to heat flow along all the co-ordinate axes and the heat generated within the control volume together increases the internal energy of the control volume.

Change in internal energy of the control volume is given by

$$\rho(dx dy dz) c \frac{\partial t}{\partial \tau} d\tau$$

According to the first law of thermodynamics heat accumulated within the control volume due to heat flow along the co-ordinate axes

$$\frac{\partial}{\partial x} \left(K_x \frac{\partial t}{\partial x} \right) + \frac{\partial}{\partial y} \left(K_y \frac{\partial t}{\partial y} \right) + \frac{\partial}{\partial z} \left(K_z \frac{\partial t}{\partial z} \right) + q_g = \rho c \frac{\partial t}{\partial \tau}$$

This expression is known as the general heat conduction equation for the Cartesian coordinate system.

Note Homogeneous and isotropic material: A homogeneous material implies that the properties, i.e., density, specific heat, and thermal conductivity of the material are the same everywhere in the material system. Isotropic means that these properties are not directional characteristics of the material, i.e., they are independent of the orientation of the surface.

Therefore for an isotropic and homogeneous material, thermal conductivity is the same $K_x = K_y = K_z = K$

$$\left(\frac{\partial^2 t}{\partial x^2}\right) + \left(\frac{\partial^2 t}{\partial y^2}\right) + \left(\frac{\partial^2 t}{\partial z^2}\right) + \frac{q_g}{k} = \frac{\rho c \partial t}{k \partial \tau} = \frac{1}{\alpha} \frac{\partial t}{\partial \tau}$$

Equation governs the temperature distribution under unsteady heat flow through a homogeneous and isotropic material.

Different cases of particular interest are:

For steady-state heat conduction, the heat flow equation reduces to:

$$\left(\frac{\partial^2 t}{\partial x^2}\right) + \left(\frac{\partial^2 t}{\partial y^2}\right) + \left(\frac{\partial^2 t}{\partial z^2}\right) + \frac{q_g}{k} = 0$$

This equation is called Poisson's equation.

In the absence of internal heat generation, the equation further reduces to:

$$\left(\frac{\partial^2 t}{\partial x^2}\right) + \left(\frac{\partial^2 t}{\partial y^2}\right) + \left(\frac{\partial^2 t}{\partial z^2}\right) = 0$$

This equation is called the Laplace equation.

Unsteady state heat flow with no internal heat generation gives:

$$\left(\frac{\partial^2 t}{\partial x^2}\right) + \left(\frac{\partial^2 t}{\partial y^2}\right) + \left(\frac{\partial^2 t}{\partial z^2}\right) = \frac{1}{\alpha} \frac{\partial t}{\partial \tau}$$

This equation is called the Fourier equation.

For one-dimensional and steady-state heat flow with no heat generation, the general heat conduction equation is reduced to:

$$\frac{\partial}{\partial x} \left(K_x \frac{\partial t}{\partial x} \right) = 0; \frac{\partial^2 t}{\partial x^2} = 0$$

Thermal diffusivity:

The thermal diffusivity of a material is the ratio of its thermal conductivity k to the thermal storage capacity (ρc). The storage capacity essentially represents the thermal capacitance or thermal inertia of the material.

It signifies the rate at which heat diffuses into the medium during a change in temperature with time. Thus, the higher value of the thermal diffusivity gives the idea of how fast the heat is conducting into the medium, whereas the low value of the thermal diffusivity shows that the heat is mostly absorbed by the material and comparatively less amount is transferred for the conduction.

Cylindrical Coordinates:-

When heat is transferred through a system having cylindrical geometries like a tube of the heat exchanger, then a cylindrical coordinate system is used.

Consider infinitesimal small element of volume

$$dV = (dr \, rd\phi \, dz)$$

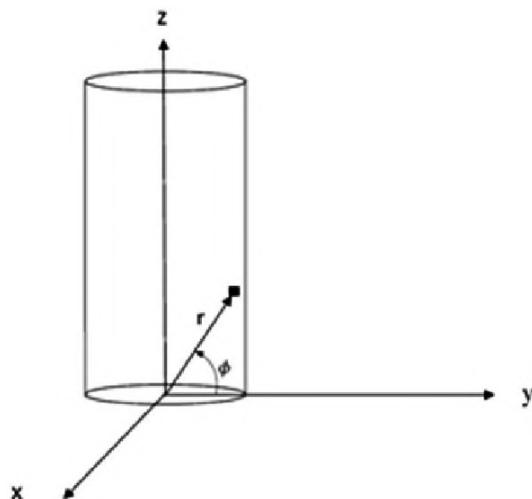


Fig. 9 (a) Cylindrical co-ordinate system

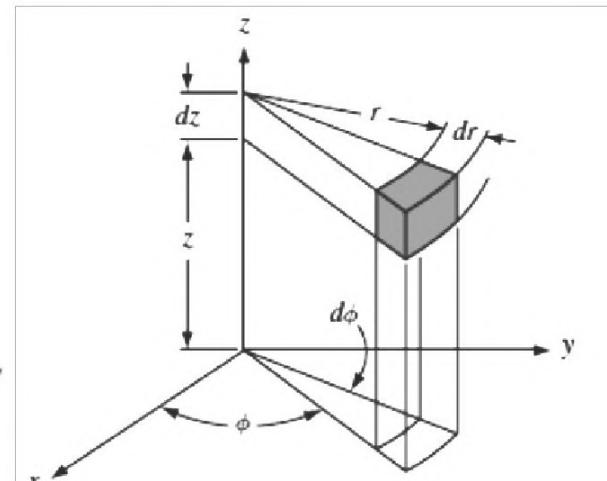


Fig. 9 (b) An element of the cylinder

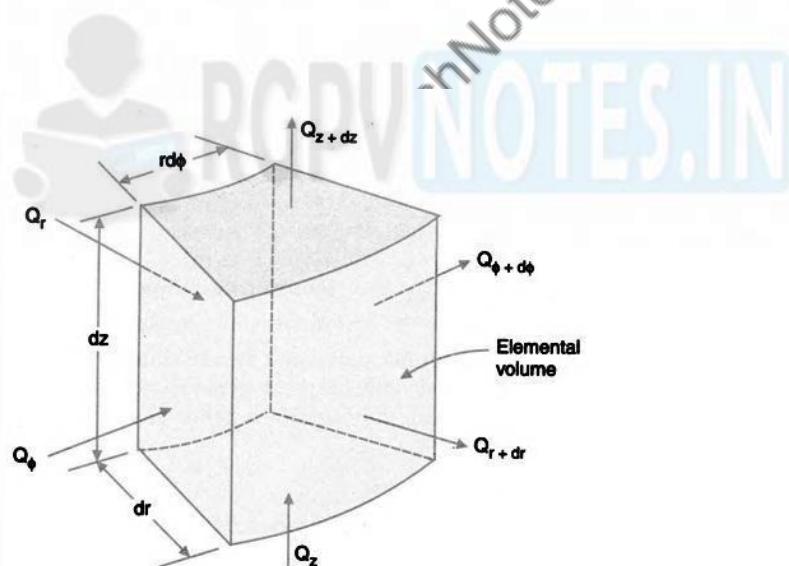


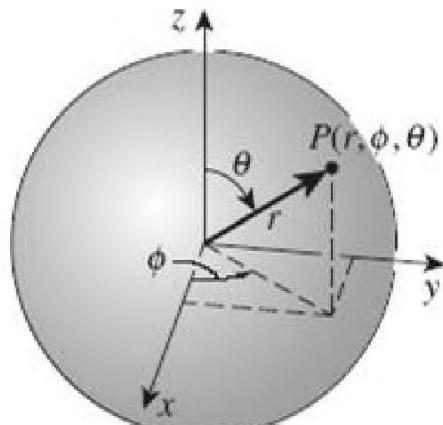
Fig. 9 (c) Heat conduction through cylindrical element

$$\left[\frac{\partial^2 t}{\partial r^2} + \frac{1}{r} \frac{\partial t}{\partial r} + \frac{1}{r^2} \frac{\partial^2 t}{\partial \phi^2} + \frac{\partial^2 t}{\partial z^2} \right] + \frac{q_g}{k} = \frac{\rho c \partial t}{k \partial \tau} = \frac{1}{\alpha} \frac{\partial t}{\partial \tau}$$

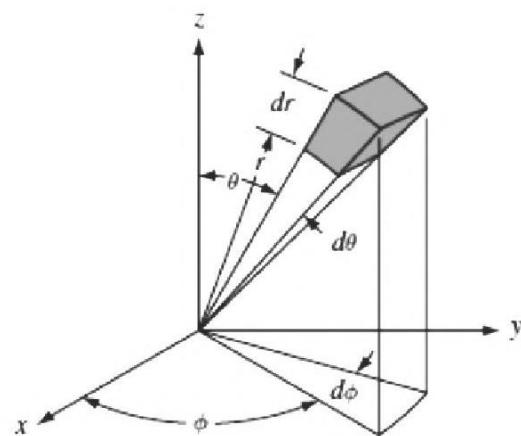
Spherical Co-ordinates:-

When heat is transferred through a system having spherical geometries like a spherical storage tank, ball of ball bearing, junction of thermocouple, then cylindrical co-ordinate system is used.

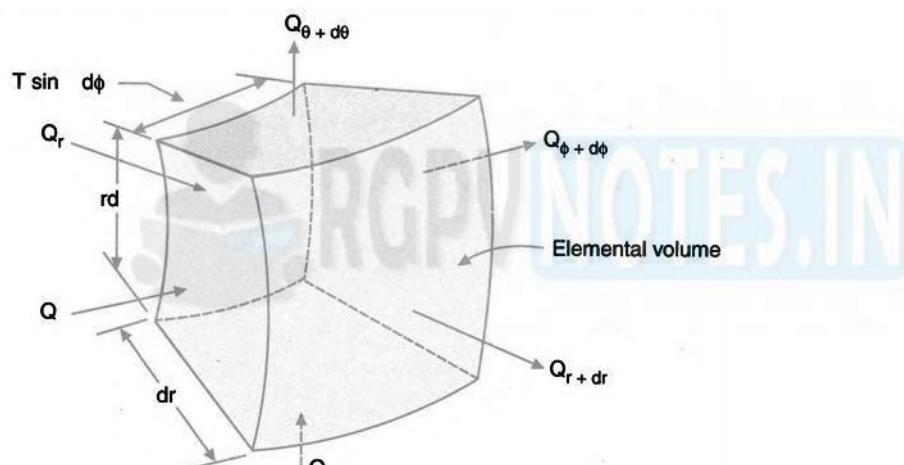
Consider an infinitesimal small element of volume



(a)



(b)



(c)

Fig. 10 (a) Spherical co-ordinate system (b) An element of sphere (c) An elemental volume

$$\left[\frac{1}{r^2 \sin^2 \theta} \frac{\partial^2 t}{\partial \phi^2} + \frac{1}{r^2 \sin^2 \theta} \frac{\partial}{\partial \theta} \left(\sin \theta \frac{\partial t}{\partial \theta} \right) + \frac{1}{r^2} \frac{\partial}{\partial r} \left(r^2 \frac{\partial t}{\partial r} \right) \right] + \frac{q_g}{k} = \frac{\rho c}{k} \frac{\partial t}{\partial \tau} = \frac{1}{\alpha} \frac{\partial t}{\partial \tau}$$

Conduction through a Plane Wall:- Analogy between flow of heat and electricity

Consider one-dimensional heat conduction through a homogeneous, isotropic wall of thickness with constant thermal conductivity and constant cross-sectional area

The wall is insulated on its lateral faces, and constant but different temperature sand is maintained at its boundary surfaces.

As per the general heat conduction equation in Cartesian co-ordinates for steady state, one dimensional with no heat generation equation.

Integrate the equation with respect to x is given by the constants of integration are evaluated by using boundary conditions and here boundary conditions are:

The temperature distribution is thus linear across the wall. Since the equation does not involve thermal conductivity so temperature distribution is independent of the material; whether it is steel, wood, or asbestos.

Heat flow can be made by substitution the value of temperature gradient into Fourier equation

The Fourier rate equation may be used directly to determine the heat flow rate.

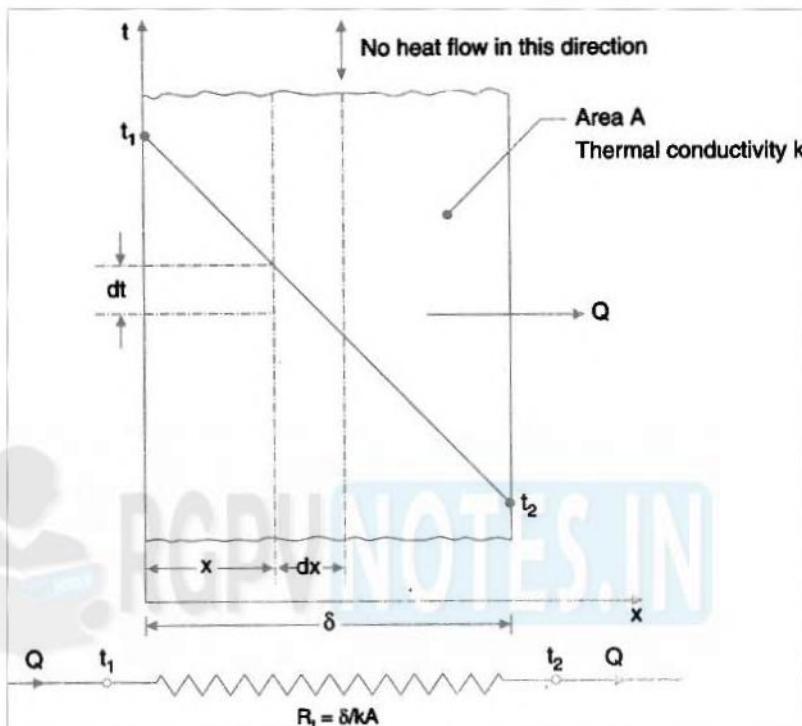


Fig. 11 Heat conduction through plane wall

To determine the temperature at any distance from the wall surface, the Fourier rate equation is integrated between the limit:

$$t = t_1 + \left(\frac{t_2 - t_1}{\delta} \right) x$$

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

From the equation when the product for a given resistance is smallest, the weight of the wall would also be so. It means for the lightest insulation for a specified thermal resistance, product of density times thermal conductivity should be smallest.

Conduction through a Composite Wall

A composite wall refers to a wall of a several homogenous layers.

Wall of furnace, boilers and other heat exchange devices consist of several layers; a layer for mechanical strength or for high temperature characteristics (fire brick), a layer of low thermal conductivity material to

restrict the flow of heat (insulating brick) and another layer for structural requirements for good appearance (ordinary brick).

Figure 12 shows one such composite wall having three layers of different materials tightly fitted to one another.

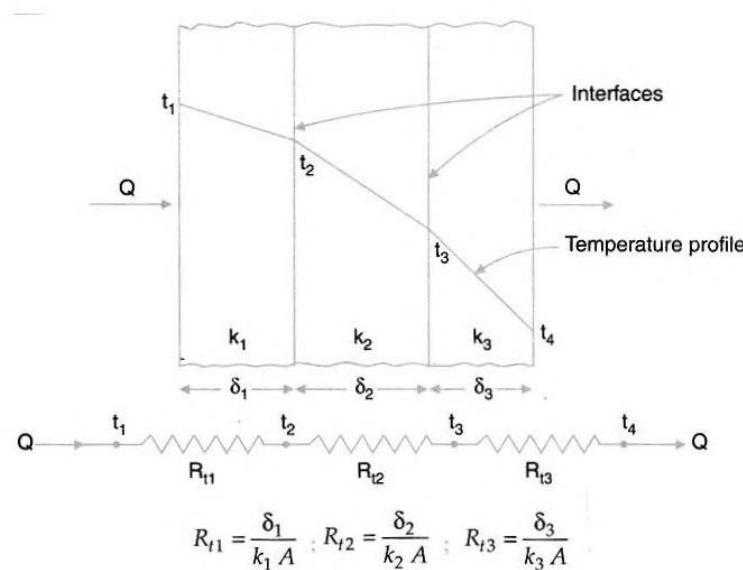


Fig. 12 Heat conduction through composite wall

$$Q = \frac{(t_1 - t_4)}{\frac{\delta_1}{k_1 A} + \frac{\delta_2}{k_2 A} + \frac{\delta_3}{k_3 A}}$$

Analysis of the composite wall assumes that there is a perfect contact between layers and no temperature drop occurs across the interface between materials.

Heat Flow between Surface and Surroundings: Cooling and Heating of Fluids

When a moving fluid comes into contact with a stationary surface, a thin boundary layer develops adjacent to the wall and in this layer, there is no relative velocity with respect to the surface.

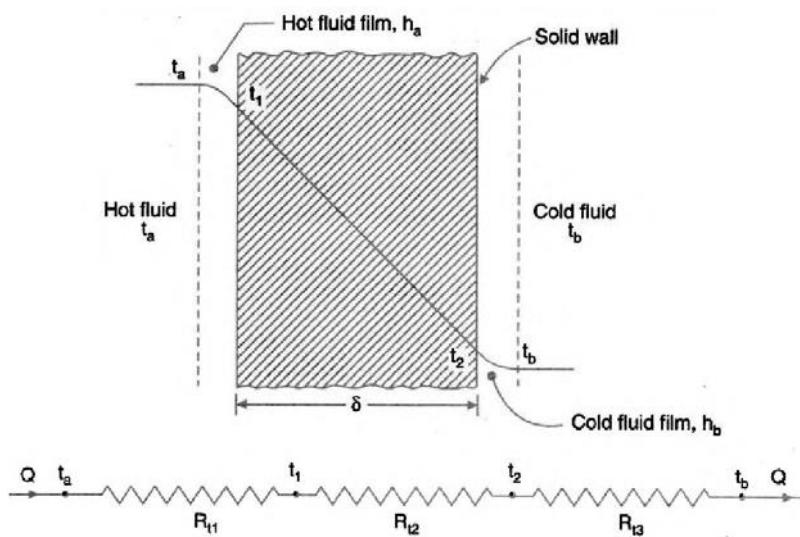


Fig. 13 Heat conduction through a wall separating two fluid

In a heat exchange process, this layer is called stagnant film and heat flow in the layer is covered both by conduction and convection processes. Since the thermal conductivity of fluids is low, the heat flow from the moving fluid of the wall is mainly due to convection.

The rate of convective heat transfer between a solid boundary and adjacent fluid is given by the Newton-Rikhman law:

$$Q = hA(t_s - t_f)$$

Where, t_s is the temperature of moving fluid, t_f is the temperature of the wall surface, A is the area exposed to heat transfer and h is the convective co-efficient.

The dimension of "h" is heat transfer by convection may be written as

$$Q = \frac{t_s - t_f}{\frac{1}{hA}} = \frac{t_s - t_f}{R_t}$$

The heat transfer through a wall separating two moving fluids involves (i) flow of heat from the fluid of high temperature to the wall, (ii) heat conduction through the wall, and (iii) transport of heat from the wall to the cold fluid.

Under steady-state conditions, the heat flow can be expressed by the equations:

$$Q = h_a A(t_a - t_1) = \frac{kA}{\delta}(t_1 - t_2) = h_b A(t_2 - t_b)$$

Where h_a and h_b represent the convective film coefficients, k is the thermal conductivity of the solid wall having a thickness. These expressions can be presented in the form:

It represents the intensity of heat transfer from one fluid to another through a wall separating them.

Numerically it equals the quantity of heat passing through the unit area of wall surface in unit time at a temperature difference of unit degree. The coefficient U has dimensions of $\text{W/m}^2\text{K}$.

By comparing the equation

$$\frac{1}{UA} = \frac{1}{h_a A} + \frac{\delta}{kA} + \frac{1}{h_b A} = R_t$$

So heat transfer coefficient is reciprocal of unit thermal resistance to heat flow.

The overall heat transfer coefficient depends upon the geometry of the separating wall, its thermal properties, and the convective coefficient at the two surfaces.

The overall heat transfer coefficient is particularly useful in the case of composite walls, such as in the design of structural walls for boilers, refrigerators, air-conditioned buildings, and in the design of heat exchangers.

Conduction through a Cylindrical Wall

Consider heat conduction through a cylindrical tube of inner radius, outer radius, and length.

The inside and outside surfaces of the tube are at constant temperatures and the thermal conductivity of the tube material is constant within the given temperature range. If both ends are perfectly insulated, the heat flow is limited to radial direction only.

Further, if the temperature at the inner surface is greater than the temperature at the outer surface, the heat flows radially outwards.

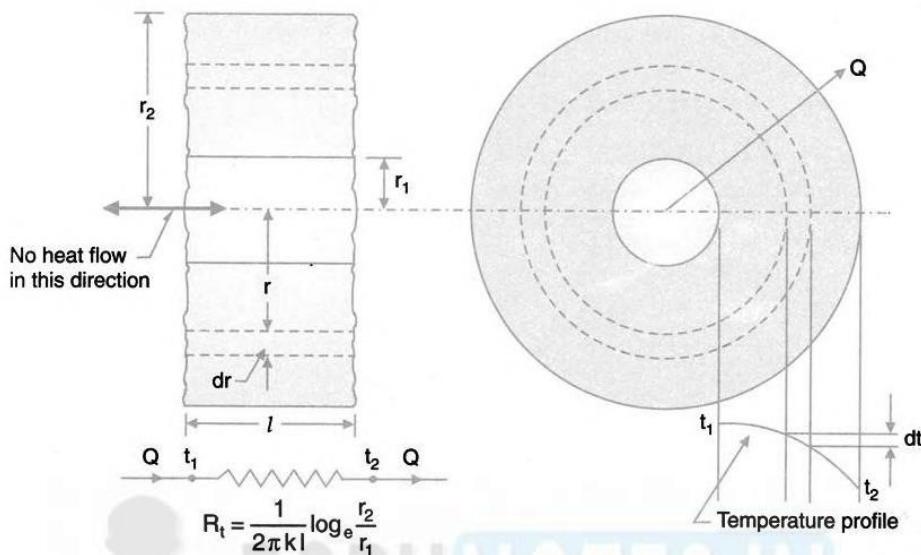


Fig. 14 Steady state heat conduction through a cylindrical wall

$$\frac{t - t_1}{t_2 - t_1} = \frac{\log_e \frac{r}{r_1}}{\log_e \frac{r_2}{r_1}}$$

From the equation it is clear that temperature distribution with radial conduction through a cylinder is logarithmic; not linear as for a plane wall. The further temperature at any point in the cylinder can be expressed as a function of radius only. Isotherms or lines of constant temperature are then concentric circles lying between the inner and outer cylinder boundaries.

The conduction heat transfer rate is determined by utilizing the temperature distribution in conjunction with the Fourier law:

$$Q = -kA \frac{dt}{dr}$$

$$Q = 2\pi kl \frac{t_1 - t_2}{\log_e \frac{r_2}{r_1}} = \frac{t_1 - t_2}{R_t}$$

In the alternative approach to estimate heat flow, consider an infinitesimally thin cylindrical element at the radius. Integrate the equation within the boundary condition for conduction in the hollow cylinder; the thermal resistance is given by:

Special Notes

Heat conduction through cylindrical tubes is found in power plant, oil refineries, and most process industries.

The boilers have tubes in them, the condensers contain banks of tubes, the heat exchangers are tubular and all these units are connected by tubes. The surface area of the cylindrical surface changes with radius. Therefore the rate of heat conduction through a cylindrical surface is usually expressed per unit length rather than per unit area as done for the plane wall.





Logarithmic Mean Area

It is advantageous to write the heat flow equation through a cylinder in the same form as that for heat flow through a plane wall.

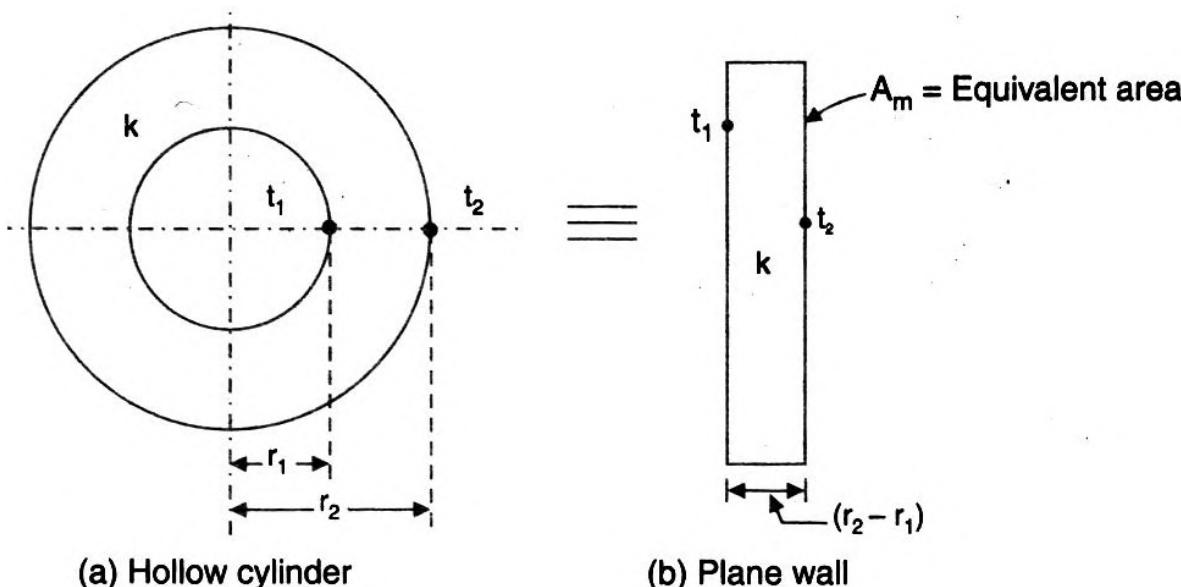


Fig. 15 Logarithmic mean area concept

$$A_m = 2\pi r_m l = \frac{2\pi(r_2 - r_1)}{\log_e \frac{r_2}{r_1}}$$
$$r_m = \frac{(r_2 - r_1)}{\log_e \frac{r_2}{r_1}}$$

Conduction through a Multilayer Cylindrical Wall

Multi-layer cylindrical walls are frequently employed to reduce heat loss from metallic pipes which handle hot fluids.

The pipe is generally wrapped in one or more layers of heat insulation.

For example, steam pipe used for conveying high-pressure steam in a steam power plant may have a cylindrical metal wall, a layer of insulation material, and then a layer of protecting plaster.

The arrangement is called lagging of the pipe system.

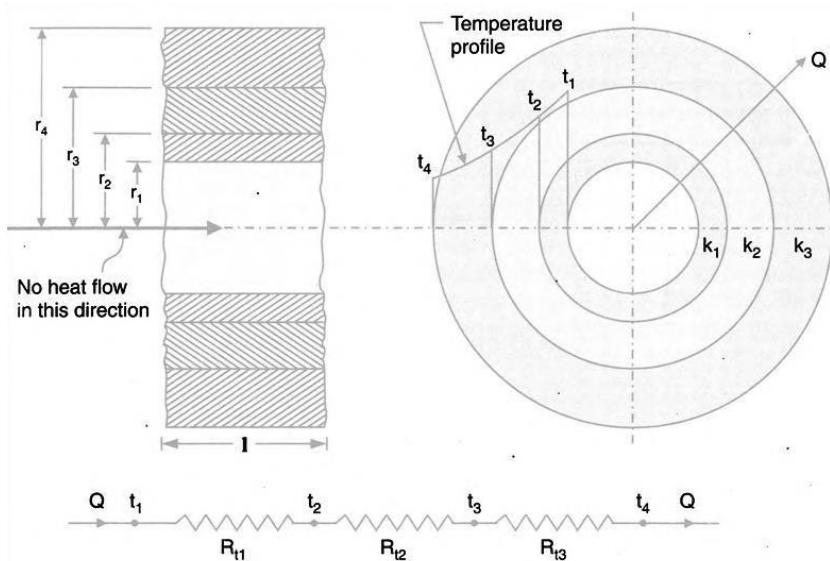


Fig. 16 Steady-state heat conduction through a composite cylindrical wall



Figure 16 shows the conduction of heat through a composite cylindrical wall having three layers of different materials.

There is a perfect contact between the layers and so an equal interface temperature for any two neighboring layers.

For steady-state conduction, the heat flow through each layer is the same and it can be described by the following set of equations:

$$Q = 2\pi k_1 l \frac{(t_1 - t_2)}{\log_e \frac{r_2}{r_1}}$$

$$= 2\pi k_2 l \frac{(t_2 - t_3)}{\log_e \frac{r_3}{r_2}}$$

$$= 2\pi k_3 l \frac{(t_3 - t_4)}{\log_e \frac{r_4}{r_3}}$$

These equations help to determine the temperature difference for each layer of the composite cylinder, The quantity in the denominator is the sum of the thermal resistance of the different layers comprising the composite cylinder.

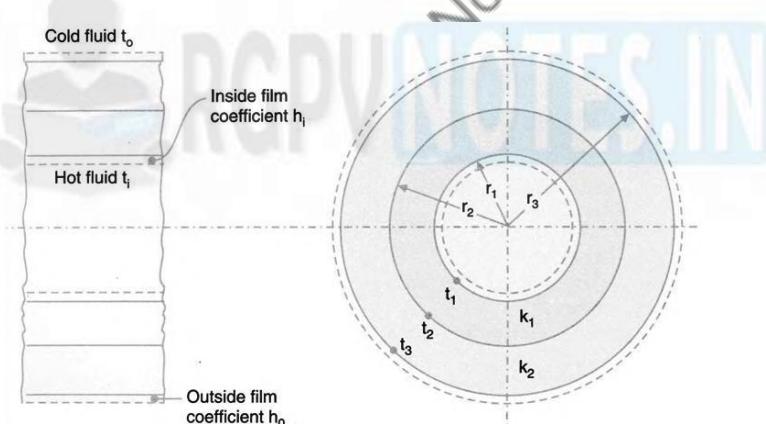


Fig. 17 Heat conduction through cylindrical wall with convection coefficient

$$R_t = \frac{1}{2\pi r_1 l h_i} + \frac{1}{2\pi k_1 l} \log_e \frac{r_2}{r_1} + \frac{1}{2\pi k_2 l} \log_e \frac{r_3}{r_2} + \frac{1}{2\pi r_3 l h_o}$$

$$Q = \frac{(t_i - t_o)}{\frac{1}{2\pi r_1 l h_i} + \frac{1}{2\pi k_1 l} \log_e \frac{r_2}{r_1} + \frac{1}{2\pi k_2 l} \log_e \frac{r_3}{r_2} + \frac{1}{2\pi r_3 l h_o}}$$

Overall Heat Transfer Coefficient U

The heat flow rate can be written as:

$$Q = UA(t_i - t_o)$$

Since the flow area varies for a cylindrical tube, it becomes necessary to specify the area on which U is based.

Thus depending upon whether the inner or outer area is specified, two different values are defined for U . Equating equations of heat transfer

$$Q = U_i A_i (t_i - t_o) = U_o A_o (t_i - t_o)$$

Overall heat transfer coefficient may be calculated by the simplified equation as follow

$$U_i A_i = U_o A_o = \frac{1}{R_t}$$

Conduction through a Sphere

Consider heat conduction through a hollow sphere of inner radius and outer radius and made of a material of constant thermal conductivity.

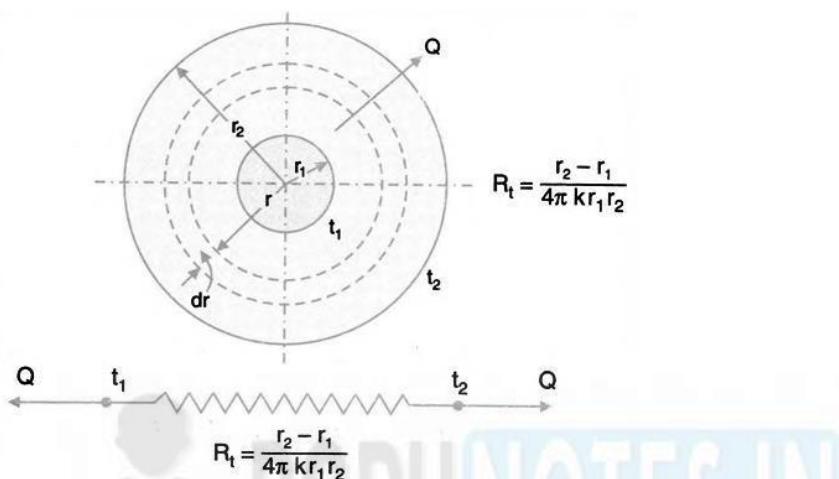


Fig. 18 Steady-state heat conduction through the sphere

The inner and outer surfaces are maintained at constant but different temperatures and respectively. If the inner surface temperature is greater than the outer surface temperature, the heat flows radially outwards. General heat conduction equation in spherical coordinates is given as for steady-state, one-directional heat flow in the radial direction and with no internal heat generation, the above equation is written as the relevant boundary conditions are using the above boundary conditions values of constants are substitute the values of constants in the equation; the temperature distribution is given as follow In non-dimensional form evidently the temperature distribution associated with radial conduction through a spherical is represented by a hyperbola.

The conduction heat transfer rate is determined by utilizing the temperature distribution in conjunction with the Fourier law.

The denominator of the equation is the thermal resistance for heat conduction through a spherical wall. In the alternative approach to determine heat flow, consider an infinitesimal thin spherical element at radius and thickness. The change of temperature across it according to the Fourier law of heat conduction.



Critical Thickness of Insulation

There is some misunderstanding about that addition of insulating material on a surface always brings about a decrease in the heat transfer rate.

But the addition of insulating material to the outside surfaces of cylindrical or spherical walls (geometries that have non-constant cross-sectional areas) may increase the heat transfer rate rather than decrease under certain circumstances.

To establish this fact, consider a thin-walled metallic cylinder of length l , radius and transporting a fluid at a temperature which is higher than the ambient temperature

Insulation of thickness and conductivity k is provided on the surface of the cylinder.

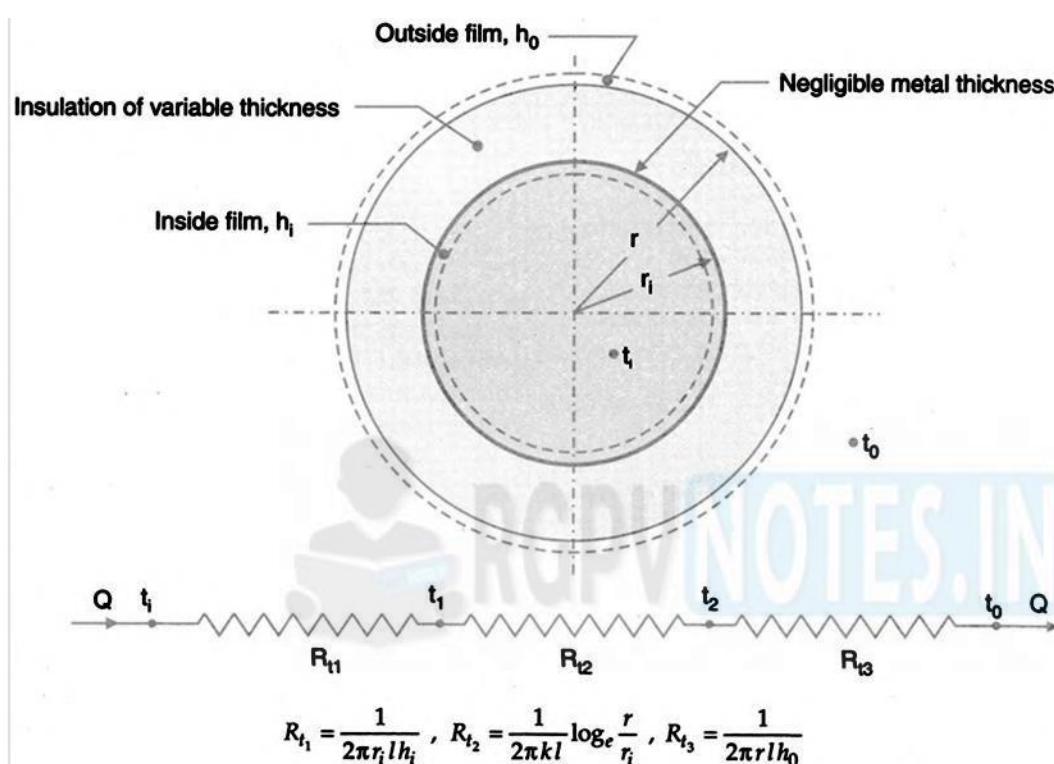


Fig. 19 Critical thickness of pipe insulation

With assumption

- Steady-state heat conduction
- One-dimensional heat flow only in the radial direction
- Negligible thermal resistance due to cylinder wall
- Negligible radiation exchange between the outer surface of insulation and surrounding

The denominator represents the sum of thermal resistance to heat flow.

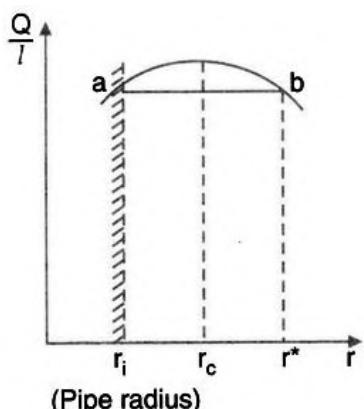
The value of h and k are constant; therefore the total thermal resistance will depend upon the thickness of insulation which depends upon the outer radius of the arrangement.

It is clear from the equation that with an increase of radius r (i.e. thickness of insulation), the conduction resistance of insulation increases but the convection resistance of the outer surface decreases.

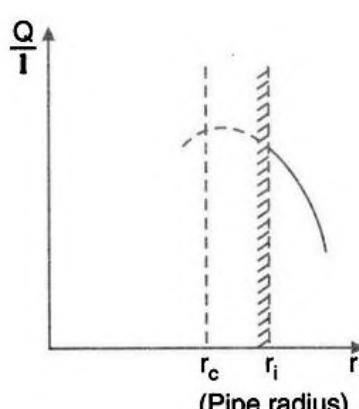
Therefore, the addition of insulation can either increase or decrease the rate of heat flow depending upon a change in total resistance with outer radius r .

To determine the effect of insulation on total heat flow, differentiate the total resistance with respect to r and equating to zero.

Because the addition of insulation decreases the convection resistance of the surface of insulation which is greater than the increase in conduction resistance of insulation.



$$(a) r_i \leq r_c = \frac{k}{h_0}$$



$$(b) r_i > r_c = \frac{k}{h_0}$$

Fig. 20 Dependence of heat loss on the thickness of insulation

Any further increase in insulation thickness decreases the heat transfer from the peak value but it is still greater than that of the bare pipe until a certain amount of insulation. So insulation greater must be added to reduce the heat loss below the bare pipe.

This may happen when insulating material of poor quality is applied to pipes and wires of a small radius. This condition is used for electric wire to increase the heat dissipation from the wire which helps to increase the current carrying capacity of the cable.



Fig. 21 Critical radius of insulation for electric wire

It is clear from figure 20 b that an increase in insulation thickness always decreases the heat loss from the pipe. This condition is used to decrease the heat loss from steam and refrigeration pipes. The critical radius of insulation for the sphere can be obtained similarly.

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Introduction

Heat transfer between a solid surface and a moving fluid is governed by the Newton's cooling law $Q=hAdT$.

Heat transfer is directly depends on heat transfer coefficient as first option and second option is regarding variation in surface area. Many times, when the first option is not in our control and the second option (i.e. increasing) is already stretched to its limit, we are left with the only alternative of increasing the effective surface area by using fins or extended surfaces.

Fins are protrusions from the base surface into the cooling fluid so that the extra surface of the protrusions is also in contact with the fluid.

Most of you have encountered cooling fins on air-cooled engines (motorcycles, portable generators, etc.), electronic equipment (CPUs), automobile radiators, air conditioning equipment (condensers), and elsewhere

Steady Flow of Heat a long Rod (Governing Differential Equation)

Consider a straight rectangular or pin fin protruding from a wall surface (figure 1 a, figure 1 b).

The characteristic dimensions of the fin are its length L , constant cross-sectional area, and the circumferential parameter P .

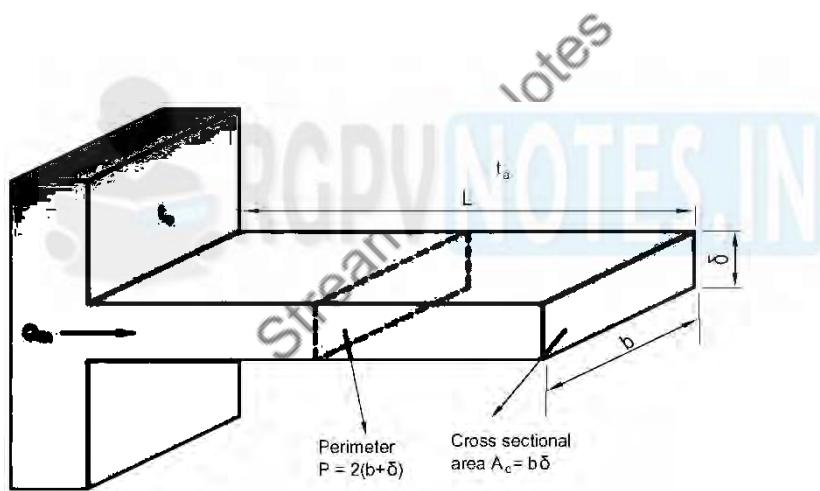


Fig. 1 a Schematic diagram of a rectangular fin protruding from a wall

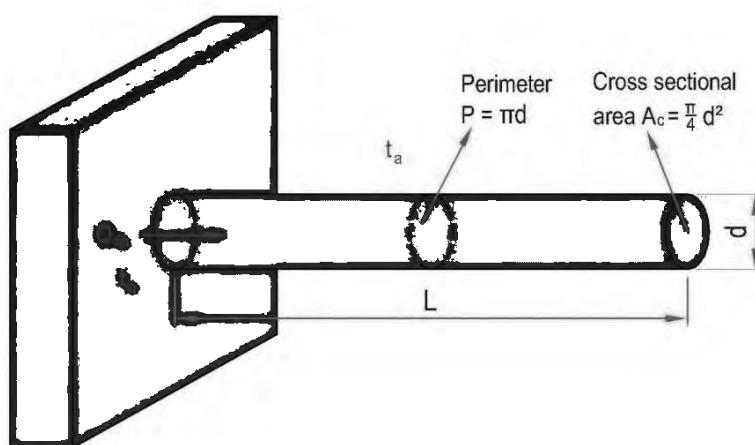


Fig. 1 b Schematic diagram of a pin fin protruding from a wall

Analysis of heat flow from the finned surface is made with the following assumptions:

- i The thickness of the fin is small compared with the length and width; temperature gradients over the cross-section are neglected and heat conduction treated one dimensional.
- ii Homogeneous and isotropic fin material; the thermal conductivity k of the fin material is constant.
- iii Uniform heat transfer coefficient h over the entire fin surface.
- iv No heat generation within the fin itself.
- v Joint between the fin and the heated wall offers no bond resistance, temperature at root or base of the fin is uniform and equal to the temperature of the wall.
- vi Negligible radiation exchange with the surroundings; radiation effects, if any, are considered as included in the convection coefficient h .
- Vii steady-state heat dissipation.

The heat from the heated wall is conducted through the fin and convected from the sides of the fin to the surroundings.

Consider infinitesimal element of the fin of thickness dx at a distance x from the base wall as shown in figure 2.

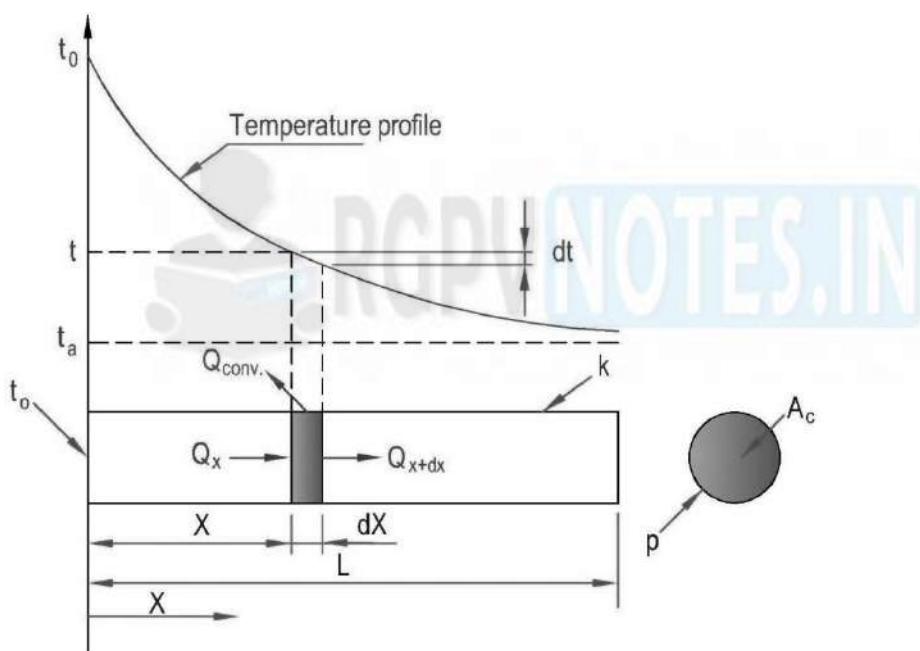


Fig. 2 Schematic diagram of a pin fin protruding from a wall

The general solution of this linear homogeneous second order differential equation is of the form

The constant and A are to be determined with the aid of relevant boundary conditions. We will treat the following four cases:

- i Heat dissipation from an infinitely long fin
- ii Heat dissipation from a fin insulated at the tip
- iii Heat dissipation from a fin losing heat at the tip

Heat Dissipation from an Infinitely Long Fin

Governing differential equation for the temperature distribution along the length of the fin is given as,

The relevant boundary conditions are

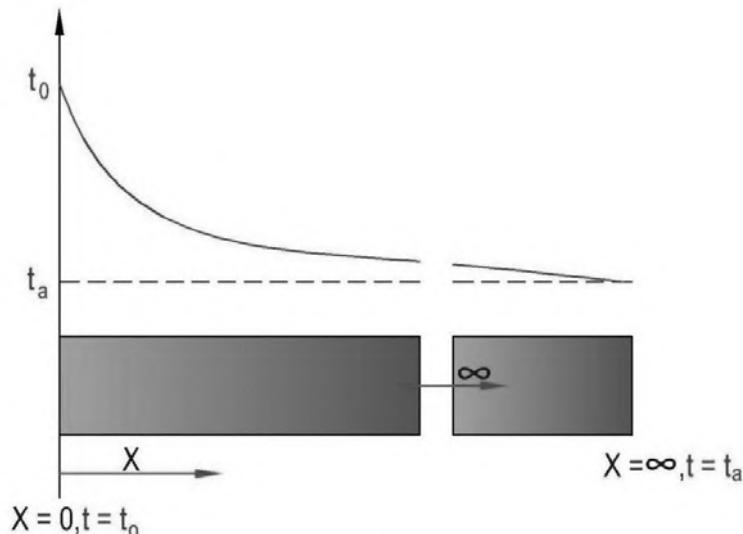


Fig. 3 Temperature distribution along the infinite long fin

The temperature at the base of the fin equals the temperature of the surface to which the fin is attached.

In terms of excess temperature or

Substitution of this boundary condition in the equation gives:

The temperature at the end of an infinitely long fin equals that of the surroundings.

Substitution of this boundary condition in the equation gives:

Heat transfer from fin

Heat transfer to the fin at the base of the fin must equal the heat transfer from the surface of the fin by convection. Heat transfer to the fin at the base is given as

From the expression for the temperature distribution

The temperature distribution would suggest that the temperature drops towards the tip of the fin.

Hence area near the fin tip is not utilized to the extent as the lateral area near the base. Obviously an increase in length beyond a certain point has little effect on heat transfer.

So it is better to use a tapered fin as it has a more lateral area near the base where the temperature difference is high.

Ingen-Hausz Experiment

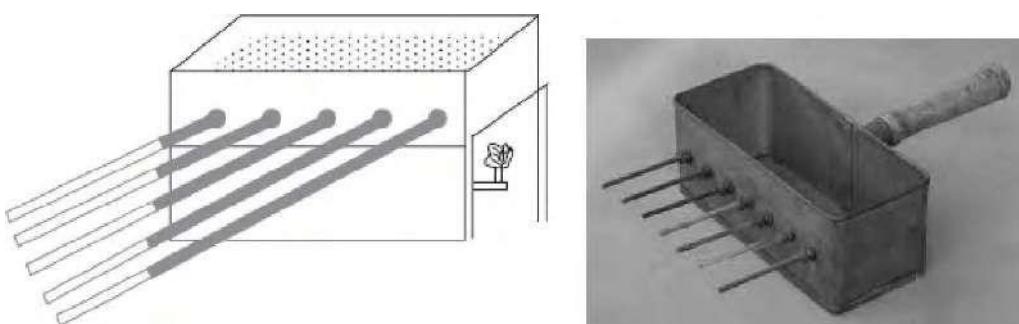


Fig. 4 Setup of Ingen-Hausz's Experiment

Heat flow rates through solids can be compared by having an arrangement consisting essentially of a box to which rods of different materials are attached (Ingen-Hausz experiment).

The rods are of the same length and area of cross-section (same size and shape); their outer surfaces are electroplated with the same material and are equally polished.

Thus, the thermal conductivity of the material of the rod is directly proportional to the square of the length up to which the wax melts on the rod.

Heat Dissipation from a Fin Insulated At the Tip

The fin is of any finite length with the end insulated and so no heat is transferred from the tip.

Therefore, the relevant boundary conditions are:

The temperature at the base of the fin equals the temperature of the surface to which the fin is attached.

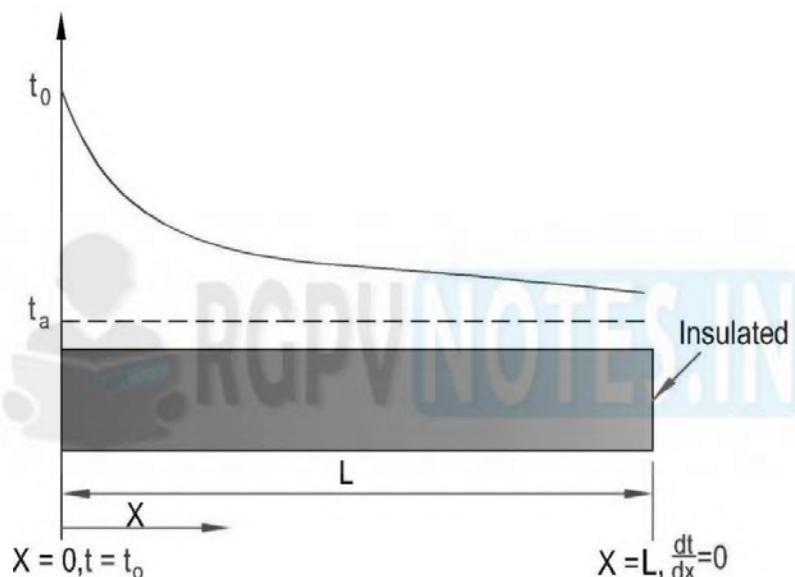


Fig. 5 Heat dissipation from a fin insulated at the tip

Heat Dissipation from a Fin Losing Heat at the Tip

The fin tips, in practice, are exposed to the surroundings. So heat may be transferred by convection from the fin tip.

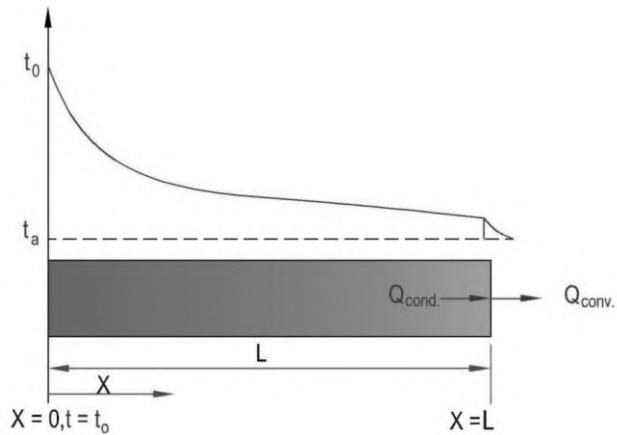


Fig. 6 Heat dissipation from fin losing heat at the tip

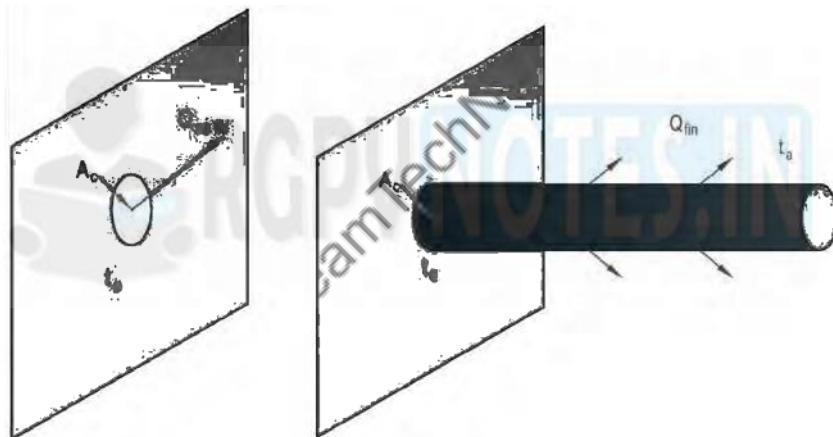


Fig. 7 Heat dissipation with and without fin

Following conclusions are given from the effectiveness of the fin

If the fin is used to improve heat dissipation from the surface, then the fin effectiveness must be greater than unity. That is, But literature suggests that the use of fins on surface is justified only if the ratio is greater than

i.To improve the effectiveness of fin, fin should be made from a high conductive material such as copper and aluminium alloys. Although copper is superior to aluminium regarding to thermal conductivity, yet fins are generally made of aluminium because of their additional advantage related to lower cost and weight.

ii.The effectiveness of fin can also be increased by increasing the ratio of perimeter to the cross-sectional area. So it is better to use more thin fins of closer pitch than fewer thicker fins at the longer pitch.

iii. A high value of film coefficient has an adverse effect on effectiveness. So fins are used with the media with low film coefficient. Therefore, in liquid-gas heat exchanger, such as a car radiator, fins are placed on the gas side.

An increase in the fin effectiveness can be obtained by extending the length of fin but that rapidly becomes a losing proposition in terms of efficiency.

Fin effectiveness: Fin effectiveness is the ratio of heat transfer from the fin to the heat transfer if the fin wasn't existing. In other words, this quantity tells us how much extra heat is being transferred by the fin. The desire is to have this ration as large as possible while keeping the additional cost of adding the fins as low as possible.

Fin efficiency: Fin efficiency is the ratio of heat transfer from the actual fin to the heat transfer of an imaginary fin of the same geometry and same conditions but with an infinite conductivity (In other words, if the entire fin surface was in a temperature equal to that of the fin base). This ratio will always be smaller than one.

Thermometric Well

An arrangement that is used to measure the temperature of gas flow through a pipeline.

A small tube called a thermometric well is welded radially into the pipeline. The well is partially filled with some liquid and the thermometer is immersed into this liquid.

When the temperature of the gas flowing through the pipe is higher than the ambient temperature, the heat flows from the hot gases towards the tube walls along the well. This may cause the temperature at the bottom of the well to become colder than the gas flowing around.

So the temperature indicated by the thermometer will not be the true temperature of the gas.

The error in the temperature measurement is estimated with the help of the theory of extended surfaces.



Fig. 8 Use of thermometric well

From the equation, it is clear that the diameter of the well does not have any effect on temperature measurement by the thermometer.

The error can be minimized by

- i Lagging the tube so that conduction of heat along its length is arrested.
- ii Increasing the value of the parameter

Transient Heat Conduction

In the preceding chapter, we considered heat conduction under steady conditions, for which the temperature of a body at any point does not change with time. This certainly simplified the analysis.

But before steady-state conditions are reached, sometimes must elapse when a solid body is suddenly subjected to a change in environment. During this transient period, the temperature changes and the analysis must take into account changes in the internal energy.

This study is a little more complicated due to the introduction of another variable namely time to the parameters affecting conduction. This means that temperature is not only a function of location, as in the steady-state heat conduction but also a function of time.

Transient heat flow is of great practical importance in industrial heating and cooling, some of the applications are given as follow

- i Heating or cooling of metal billets;
- ii Cooling of I.C. engine cylinder;
- iii Cooling and freezing of food;
- iv Brick burning and vulcanization of rubber;
- v Starting and stopping of various heat exchanger units in the power plant.

Change in temperature during unsteady-state may follow a periodic or a non-periodic variation.

Periodic variation

The temperature changes in repeated cycles and the conditions get repeated after some fixed time interval. Some examples of periodic variation are given as follow:

- i Variation of the temperature of a building during a full day period of 24 hour
- ii Temperature variation in the surface of the earth during a period of 24 hour
- iii Heat processing of regenerators whose packing are alternately heated by flue gases and cooled by air

Non-periodic variation

The temperature changes as some non-linear function of time. This variation is neither according to any definite pattern nor is in repeated cycles. Examples are:

- i Heating or cooling of an ingot in a furnace
- ii Cooling of bars, blanks, and metal billets in steelworks

Transient Conduction in Solids with Infinite Thermal Conductivity (Lumped Parameter Analysis)

Even though no materials in nature have an infinite thermal conductivity, many transient heat flow problems can be readily solved with acceptable accuracy by assuming that the internal conductive resistance of the system is so small that the temperature within the system is substantially uniform at any instant.

This simplification is justified when the external thermal resistance (Convection resistance) between the surface of the system and the surrounding medium is so large compared to the internal thermal resistance (Conduction resistance) of the system that it controls the heat transfer process.

Consider a small hot copper ball coming out of an oven (Figure 9). Measurements indicate that the temperature of the copper ball changes with time, but it does not change much with a position at any given time due to large thermal conductivity.

Thus the temperature of the ball remains uniform at all times.

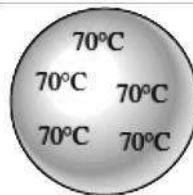


Fig. 9 Temperature distribution throughout the copper ball

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

The following points can be made from the above Consideration:

1. The body temperature falls or rises exponentially with time and the rate depends on the parameter. Theoretically, the body takes infinite time to approach the temperature of surroundings and thus attain steady-state conditions. However, the difference between and becomes extremely small after a short time, and beyond that period the body temperature becomes practically equal to the ambient

temperature. The change in temperature of a body with respect to time is shown in figure 10 for both cases (Heating and cooling)

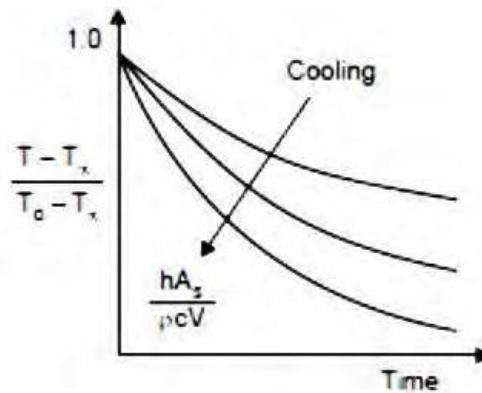


Fig. 10 Change in temperature of body with respect to time

2. The quantity has the dimensions of time and is called the thermal time constant. Its value is indicative of the rate of response of a system to a sudden change in the environmental temperature; how fast the body will respond to a change in the environmental temperature. It should be as small as possible for the fast response of the system to change in environmental temperature.

Criteria for Lumped System Analysis

Biot number is used to check the applicability of lumped parameter analysis. If the Biot number is less than 0.1, it has been proved that this model can be used without appreciable error.

The lumped parameter solution for transient conduction can be conveniently stated as

Time Constant and Response of a Thermocouple/ Error in measurement of temperature:

A Thermocouple is a sensor used to measure temperature. A thermocouple is comprised of at least two metals joined together to form two junctions.

One is connected to the body whose temperature is to be measured; this is the hot or measuring junction. The other junction is connected to a body of known temperature; this is the cold or reference junction.

Therefore the thermocouple measures the unknown temperature of the body with reference to the known temperature of the other body.

Measurement of temperature by a thermocouple is an important application of the lumped parameter analysis.

The response of a thermocouple is defined as the time required for the thermocouple to reach the source temperature when it is exposed to it.

Referring to the lumped-parameter solution for transient heat conduction;

The sensitivity of the thermocouple is defined as the time required by the thermocouple to reach 63.2% of its steady-state value. According to the definition of sensitivity

The time constant represents the speed of response, i.e., how fast the thermocouple tends to reach the steady-state value. A large time constant corresponds to slow system response, and a small-time constant represents a fast response. A low value of the time constant can be achieved for a thermocouple by

- i Decreasing light metals the wire diameter
- ii Using light metals of low density and low specific heat
- iii Increasing the heat transfer coefficient

Depending upon the type of fluid used, the response times for different sizes and materials of thermocouple wires usually lie between 0.04 to 2.5 seconds.

Note: - Once the time constant is measured, we have to wait for the time to measure the temperature within 63.2% of accuracy.

Transient Heat Conduction in Solids with Finite Conduction and Convective Resistance ($0 < Bi < 100$)

In the lumped parameter analysis we assume that the conductivity of the material is infinite or variation of temperature within the body is negligible. But sometimes there may be a variation of temperature with time and position.

Consider a plane wall of thickness $2L$, a long cylinder of radius r_o , and a sphere of radius r_o initially at a uniform temperature T_i , as shown in figure 11.

Note that all three cases possess geometric and thermal symmetry: the plane wall is symmetry about its centre plane ($x = 0$), the cylinder is symmetry about its centreline ($r = 0$), and the sphere is symmetry about its centre point ($r = 0$).

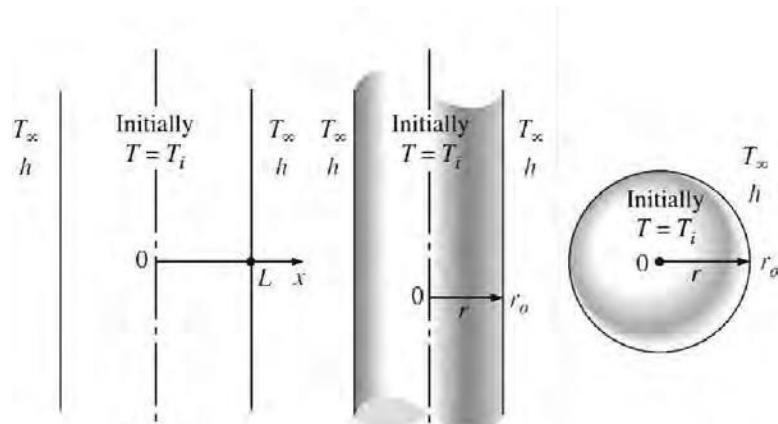


Fig. 11 Transient heat conduction in large wall, cylinder and sphere

Temperature profile of plane wall

The variation of temperature profile with respect to time in plane wall is shown in figure 12.

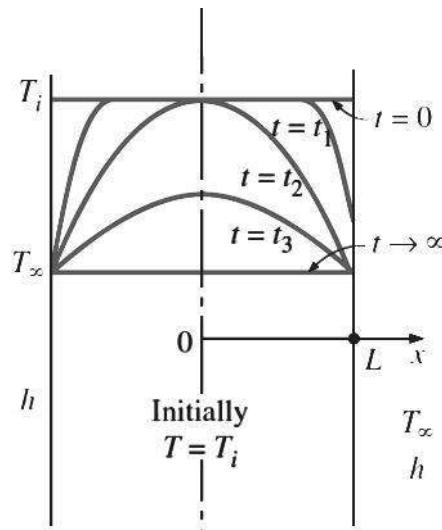


Fig. 12 Transient heat conduction in a large wall, cylinder, and sphere

When the wall is first exposed to the surrounding medium the entire wall is at its initial temperature.

But the wall temperature at the surface starts to drop as a result of heat transfer from the wall to the surrounding medium. This creates a temperature gradient in the wall.

The temperature profile within the wall remains symmetric at all times about the centre plane. The temperature profile gets flatter and flatter as times passes as a result of heat transfer and finally becomes uniform at. The controlling differential equation for the transient heat conduction is:

The solution of the controlling differential equation in conjunction with initial boundary conditions would give an expression for temperature variation both with time and position.

The temperatures at other locations are worked out by multiplying the mid-plane temperature by correction factors read from correction charts.

Numerical: Aluminums fins 1.5 cm wide and 10 mm thick are placed on a 2.5 cm diameter tube to dissipate the heat. The tube surface temperature is 170°C ambient temperature is 20°C. Calculate the heat loss per fin. Take $h = 130 \text{ W/m}^2 \text{ C}$ and $K = 200 \text{ W/m}^2 \text{ C}$ for aluminums.

Given:

$$\text{Width of the fin } b = 1.5 \text{ cm} = 1.5 \times 10^{-2} \text{ m}$$

Thickness $t = 10 \text{ mm} = 10 \times 10^{-3} \text{ m}$

Diameter of the tube $d = 2.5 \text{ cm} = 2.5 \times 10^{-2} \text{ m}$

Surface temperature $T_b = 170^\circ\text{C} + 273 = 443 \text{ K}$

Ambient temperature $T_\infty = 20^\circ\text{C} + 273 = 293 \text{ K}$

Heat transfer co-efficient $h = 130 \text{ W/m}^2\text{C}$

Thermal conductivity $K = 200 \text{ W/m}^\circ\text{C}$

Solution

Assume fin end is insulated, so this is short fin end insulated type problem.

Heat transfer [short fin, end insulated]

$$Q = (hPKA)^{1/2} (T_b - T_\infty) \tan h(mL) \quad \dots\dots(1) \quad [\text{From NoHMT.41] data book}]$$

Where

A – Area = Breadth ‘thickness

$$A = \text{Breadth} \times \text{thickness}$$

$$= 1.5 \times 10^{-2} \times 10 \times 10^{-3}$$

$$\boxed{A = 1.5 \times 10^{-4} \text{ m}^2}$$

$$P - \text{Perimeter} = 2(b + t)$$

$$= 2[(1.5 \times 10^{-2}) + (10 \times 10^{-3})]$$

$$\boxed{P = 0.05 \text{ m}}$$

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

$$= \sqrt{\frac{130 \times 0.05}{200 \times 1.5 \times 10^{-4}}}$$

$$\boxed{m = 14.7 \text{ m}^{-1}}$$

$$(1) \Rightarrow Q = [130 \times 0.05 \times 200 \times 1.5 \times 10^{-4}]^{1/2} \times (443 - 293) \times \tan h(14.7 \times 1.5 \times 10^{-2})$$

$$\boxed{Q = 14.3 \text{ W}}$$

END

UNIT –III

Unit 3 Convection: Introduction, free and forced convection; principle of dimensional analysis, Buckingham ‘pie’ theorem, application of dimensional analysis of free and forced convection, empirical correlations for laminar and turbulent flow over a flat plate and tubular geometry; calculation of convective heat transfer coefficient using data book.

Introduction to Convection: Thermal convection occurs when a temperature difference exists between a solid surface and a fluid flowing past it.

Free Stream

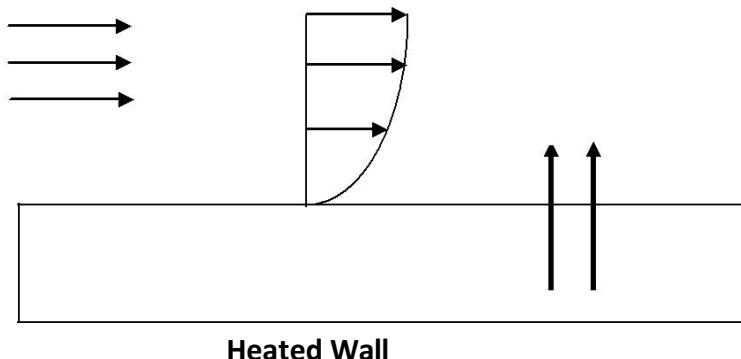


Fig. 1 Convection Phenomena

It is well known that a hot plate of metal will cool faster when placed in front of a fan than when exposed to still air.

For example, we know that the velocity at which the air blows over the hot plate influences the heat transfer rate. But does it linearly influence the cooling? i.e. if the velocity is doubled, will the heat transfer rate be doubled?

Relation with conduction:

As shown in Fig.1 the velocity of the fluid layer at the wall will be zero, the heat must be transferred by **conduction** at that point.

Thus we might compute the heat transfer using Fourier's equation of conduction i.e. with the thermal conductivity of the fluid and the fluid temperature gradient at the wall.

Why then, if the heat flows by conduction in this layer, do we speak of “Convection” heat transfer and need to consider the velocity of the fluid?

The answer is that the temperature gradient is dependent on the rate at which the fluid carries the heat away; a high velocity produces a large temperature gradient, and so on.

It must be remembered that the physical mechanism of heat transfer at the wall is a conduction process.

Newton-Rikhman Law or Newton's Law of Cooling:

The value of film co-efficient is dependent upon:

1. Surface conditions: Roughness & Cleanliness
2. Geometry and orientation of surface: Plate, Tube, and Cylinder placed horizontally or vertically.
3. Thermo-physical properties of the fluid: Density, Viscosity, Specific heat, Co-efficient of expansion, and thermal conductivity.
4. Nature of fluid flow: Laminar or Turbulent
5. Boundary layer configuration
6. Existing thermal conditions.

The film co-efficient depends on the viscosity of fluid because the viscosity influences the velocity profile and correspondingly the energy transfer rate in the region near the wall.

Free and Forced Convection

Concerning the cause of fluid flow, two types of convection are distinguished:

1. Free Convection or Natural Convection and
2. Forced Convection.

1. Free Convection or Natural Convection

When a surface is maintained in still fluid at a temperature higher or lower than that of the fluid, a layer of fluid adjacent to the hot or cold surface gets heated or cooled by conduction.

A density difference is created between this adjacent layer and the still fluid surrounding it.

The density difference introduces a buoyant force causing a flow of fluid near the surface.

Heat transfer under such conditions is known as Free or Natural Convection.

Thus, "Free or Natural convection is the process of heat transfer which occurs due to movement of the fluid particles by density changes associated with temperature differential in a fluid."

This mode of heat transfer occurs very commonly, some of the examples are:

House heating system, the cooling of transmission lines, electric transformers, and rectifiers.

2. Forced Convection

The flow of fluid is caused by a pump, a fan, or by the atmospheric winds.

These mechanical devices speed up the heat transfer rate.

In free convection flow velocities encountered are lower compared to flow velocities in forced convection, consequently, the value of the convection coefficient is lower, and for a given rate of heat transfer, a larger area could be required.

Examples of forced convection are cooling of I.C. Engines, Air conditioners, Heat exchangers, etc.

Dimensional Analysis

"Dimensional analysis is a mathematical technique which makes use of the study of the dimensions for solving several engineering problems."

Dimensional analysis has become an important tool for analyzing fluid flow problems. It is especially useful in presenting experimental results in a concise form.

There are two methods used in the dimensional analysis: Rayleigh's Method and Buckingham's - Theorem.

Buckingham's Theorem

"If there are variables (independent and dependent variables) in a physical phenomenon and if these variables contain fundamental dimensions, then the variables are arranged into dimensionless terms."

System of Dimensions:

In the area of heat transfer, two more dimensions namely the temperature difference and the heat are also taken as fundamental quantities.

Here heat can be expressed in terms of MLT. So the fundamental quantities are mass, length, time, and temperature; designated by the M, L, T & θ respectively.

Temperature is specially used incompressible flow and heat transfer phenomena.

Dimensionless Numbers & Their Physical Significance

1. Reynolds Number (Re)

It is defined as a ratio of inertia force to viscous force.

It indicates the relative importance of the inertial and viscous effects in fluid motion.

At low Reynolds number, the viscous effect dominates and the fluid motion is laminar.

At a high Reynolds number, the inertial effects lead to turbulent flow.

Reynolds number constitutes an important criterion of kinematic and dynamic similarity in forced convection heat transfer.

2. Prandtl Number (Pr)

"It is the ratio of kinematic viscosity to thermal diffusivity of the fluid".

The kinematic viscosity represents the momentum transport by molecular friction and thermal diffusivity represents the heat energy transport through conduction.

Pr provides a measure of the relative effectiveness of **momentum** and **energy** transport by diffusion.

For highly viscous oils, Pr is quite large (100 to 10000) and that indicates rapid diffusion of momentum by viscous action compared to the diffusion of energy.

For gases, Pr is about 1, which indicates that both momentum and heat dissipate through the field at about the same rate.

The liquid (liquid sodium or liquid potassium) metals have Pr = 0.003 to 0.01 and that indicates more rapid diffusion of energy compared to the momentum diffusion rate.

The Prandtl number is connecting link between the velocity field and the temperature field, and its value strongly influences the relative growth of velocity and thermal boundary layers.

3. Nusselt Number (Nu)

Nu established the relation between convective film coefficient (h), the thermal conductivity of the fluid (K) and a significant length parameter (L) of the physical system.

To understand the physical significance of the Nu, consider a fluid layer of thickness and temperature differences shown in Fig.2.

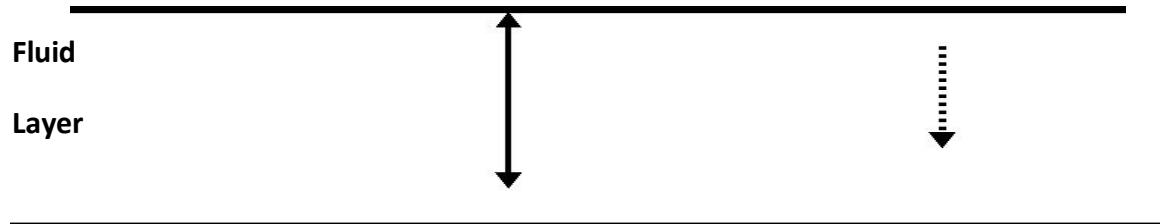


Fig. 2 Heat transfer through the fluid layer

Heat transfer through the fluid layer is by **convection** when the fluid involves some motion and by **conduction** when the fluid layer is motionless.

4. Grashoff Number (Gr)

It indicates the relative strength of the buoyant to viscous forces.

The Grashoff number represents the ratio of buoyant force and Inertia force to the square of the viscous force.

Grashoff number has a role in free convection.

Free convection is usually suppressed at sufficiently small Gr, begins at some critical value of Gr, and then becomes more and more effective as Gr increases.

5. Stanton Number (St)

"It is the ratio of heat transfer coefficient to the flow of heat per unit temperature rise due to the velocity of fluid".

It should be noted that the Stanton number can be used only in co-relating forced-convection data (since the expression contains velocity).

6. Peclet Number (Pe)

"It is the ratio of mass heat flow rate by convection to the flow rate by conduction under a unit temperature gradient and through a thickness".

The Peclet number is a function of Reynolds number and Prandtl number.

7. Graetz Number (G)

"It is the ratio of heat capacity of fluid flowing through the pipe per unit length to the conductivity of pipe material."

Graetz number is merely a product of a constant and the Peclet number.

Dimensional Analysis Applied to Forced Convection & Free Convection

Key Notes:

In natural or free convection, the flow is produced by buoyant effects resulting from the temperature difference. These effects are included in the Grashoff number.

Reynolds number is important in the case of forced convection and similarly, the Grashoff number is important in the case of free convection.

Mathematical analysis of convective heat problems is complicated due to the large number of variables involved.

The majority of the convective problems are, therefore, analyzed through the technique of dimensional analysis supported by experimental investigations. The dimensional analysis helps to develop certain correlations for the convective coefficient.

The constants and exponents appearing in these correlations for a particular situation are worked out through experiments.

Use "Heat & Mass Transfer by Dr. D. S. Kumar" to see different empirical co-relations for free and forced convection for different cases. (Equations should be given in examination so no need to remember)

Some of the important terminology associated with this topic is explained below:

Bulk Temperature & Mean Film Temperature

The physical properties (μ , ρ , C_p , k) of fluid are temperature dependent.

The accuracy of the results obtained by using theoretical relations and the dimensionless empirical co-relations would depend upon the temperature chosen for the evaluation of these properties.

No uniform procedure has been attained in the selection of this reference temperature.

However, it is customary to evaluate the fluid properties either based on **bulk temperature** or the **mean film temperature**.

Mean Bulk Temperature:

The mean bulk temperature (t_b) denotes the equilibrium temperature that would result if the fluid at a cross-section was thoroughly mixed in an adiabatic container.

For internal flow (Heat exchangers), the fluid flowing through the tubes may be heated or cooled during its flow passage. The bulk temperature is then taken to be the arithmetic mean of the temperatures at the inlet to and at the exit from the heat exchanger tube; i.e.

Mean Film Temperature:

It is the arithmetic mean of the surface temperature of a solid and the undisturbed temperature of the fluid which flows over the surface.

Thermal and Hydrodynamic Boundary Layer

The concept of the boundary layer was first introduced by L. Prandtl in 1904 and since then it has been applied to several fluid flow problems.

A. Hydrodynamic Boundary Layer: Flat Plate

"When a fluid flows around an object, there exists a thin layer of fluid close to the solid surface within which shear stresses significantly influence the velocity distribution. The fluid velocity varies from zero at the solid surface to the velocity of free streamflow at a certain distance away from the solid surface. This thin layer of changing velocity has been called the **hydrodynamic boundary layer**."

Consider the parallel flow of a fluid over a flat plate as shown in Fig. 3.

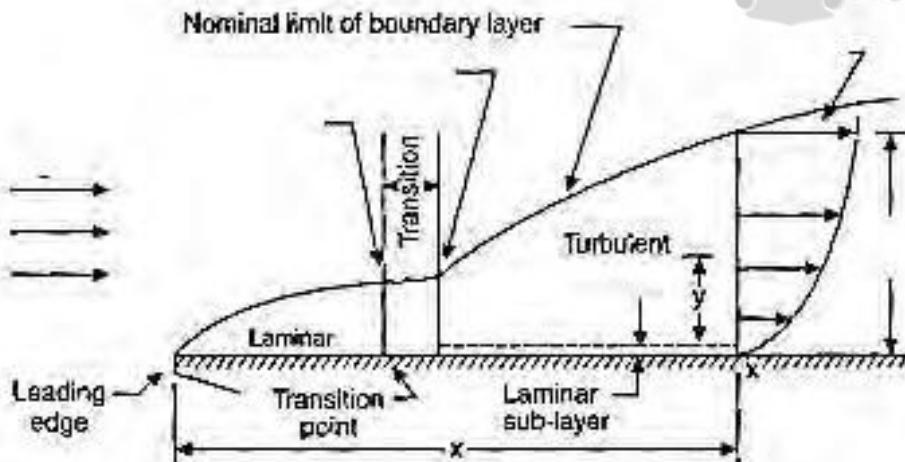


Fig. 3 Development of a boundary layer on a flat plate

The edge facing the direction of flow is called the leading edge. The rear edge is called the trailing edge.

The velocity of the fluid particles in the first fluid layer adjacent to the plate becomes zero because of the no-slip condition.

This motionless layer slows down the particles of the neighboring fluid layer as a result of friction between the particles of these two adjoining fluid layers at different velocities.

This fluid the thickness of the boundary layer increases with distance from the leading edge; as more and more fluid is slowed down by the viscous effects, becomes unstable and breaks into a turbulent boundary layer.

In the turbulent boundary layer, a very thin layer near the smooth surface remains laminar, called the **laminar sub-layer**.

For the flow over a flat surface, if Reynolds No. is less than 5×10^5 , the flow is laminar and velocity distribution is parabolic.

The boundary layer thickness:

"It is arbitrarily defined as that distance from the plate surface in which the velocity reaches 99% of the velocity of the free stream.

The hypothetical line divides the flow over a plate into two regions:

(a) **The boundary layer region**, in which the viscous effects and the velocity changes are significant, and (b) **The irrotational flow region**, in which the frictional effects are negligible and the velocity remains essentially constant.

B. Thermal Boundary Layer

Whenever a flow of fluid takes place over a heated or cold surface, a temperature field is set up in the field next to the surface. The zone or thin layer wherein the temperature field exists is called the **thermal boundary layer**.

The temperature gradient results due to heat exchange between the plate and the fluid.

Consider the flow of a fluid at a uniform temperature over a hot flat plate temperature as shown in Fig. 4.

The fluid particles in the layer adjacent to the surface will reach thermal equilibrium with the plate and assume the surface temperature. These fluid particles will then exchange energy with the

particles in the adjoining fluid layer and so on. As a result, a temperature profile will develop in the flow field that ranges from at the surface to sufficiently far from the surface.

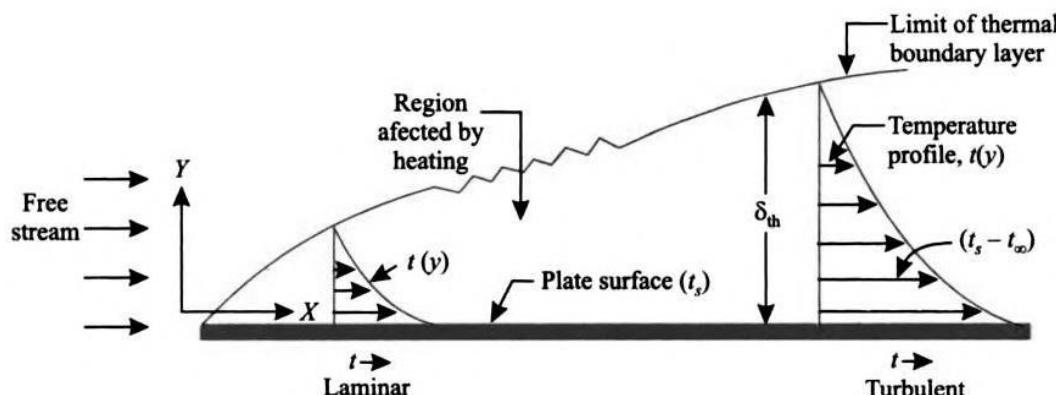


Fig. 4 Thermal boundary layer during the flow of cold fluid over a warm plate

The flow region over the surface in which the temperature variation in the direction normal to the surface is significant is the **thermal boundary layer**.

The thickness of the thermal boundary layer is defined as the distance from the surface at equal at any location along the surface in which the temperature difference.

The thickness of the thermal boundary layer increases in the flow direction since the effects of heat transfer are felt at greater distances from the surface further downstream.

If the approaching free stream temperature is above the plate surface temperature, the thermal boundary layer will have the shape as depicted in Fig. 5.

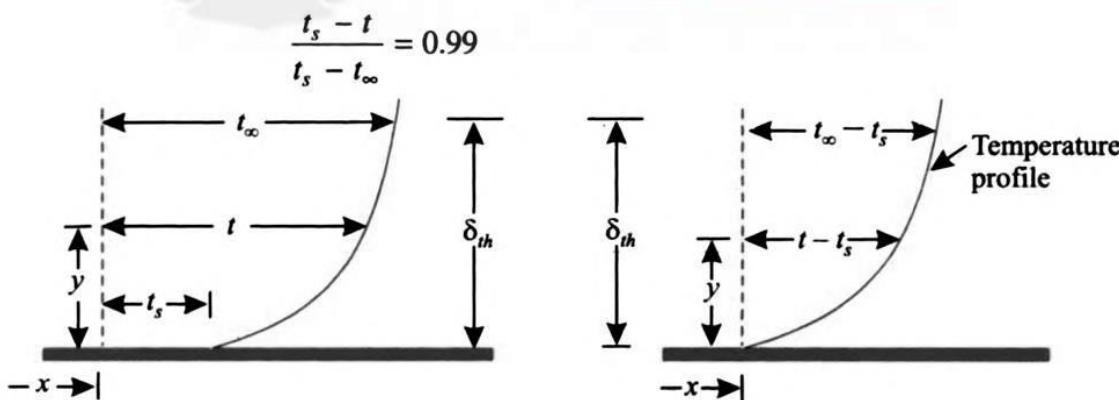


Fig. 5 Temperature profile in T.B.L. when warm fluid flows over a cold plate

The temperature of the fluid changes from a minimum at the plate surface to the temperature of the mainstream at a certain distance from the surface.

At point A, the temperature of the fluid is the same as the surface temperature.

The fluid temperature increases gradually until it acquires the free stream temperature.

The distance measured perpendicularly to the thickness of the thermal boundary layer at a distance plate.

Relation between Thermal & Velocity Boundary Layer

The velocity profile of the velocity boundary layer is dependent primarily upon the viscosity of the fluid.

The temperature profile of the thermal boundary layer depends upon the flow velocity, specific heat, viscosity, and thermal conductivity of the fluid.

Boiling and Condensation

Introduction

When the temperature of a liquid at a specified pressure is raised to the saturation temperature (T_{sat}), at that pressure **Boiling** occurs.

Likewise, when the temperature of the vapour is lowered to saturation temperature (T_{sat}),

Condensation occurs.

Boiling and Condensation are considered to be forms of convection heat transfer since they involve fluid motion, such as the rise of the bubbles to the top and the flow of condensate to the bottom.

Boiling and Condensation differ from other forms of convection, in that they depend on the latent heat of vaporization (h_{fg}) of the fluid and the surface tension (σ) at the liquid-vapour interface, in addition to the properties of the fluid in each phase.

During a phase change, a large amount of heat (due to the large latent heat of vaporization released or absorbed) can be transferred essentially at a constant temperature.

The phenomena are quite difficult to describe due to change in fluid properties(density, specific heat, thermal conductivity, viscosity, etc.) and due to considerations of surface tension, latent heat of vaporization, surface characteristics, and other features of two-phase flow.

Heat transfer co-efficient h associated with boiling and condensation are typically much higher than those encountered in other forms of convection processes that involve a single phase.

Boiling

Boiling is the convective heat transfer process that involves a phase change from liquid to vapor state.

Boiling is a liquid to vapour phase change process just like evaporation, but there are significant differences between the two. **Evaporation** occurs at the liquid–vapour interface when the vapor pressure is less than the saturation pressure of the liquid at a given temperature. Examples of evaporation are drying of clothes, the evaporation of sweat to cool the human body, and the rejection of waste heat in wet cooling towers. Note that evaporation involves no bubble formation or bubble motion.

Boiling, on the other hand, occurs at the solid-liquid interface when a liquid is brought into contact with a surface maintained at a temperature T_s sufficiently above the saturation temperature T_{sat} of the liquid. At 1 atm, for example, liquid water in contact with a solid surface at 110°C will boil since the saturation temperature of water at 1 atm is 100°C.

The boiling process is characterized by the rapid formation of vapor bubbles at the solid-liquid interface that detaches from the surface when they reach a certain size and attempt to rise to the free surface of the liquid.

Applications of Boiling

Steam production. Absorption of heat in refrigeration and air-conditioning systems. Greater importance has recently been given to boiling heat transfer because of developments of nuclear reactors, space-crafts, and rockets, where large quantities of heat are produced in a limited space and are to be dissipated at very high rates.

Types of Boiling

A. Classification of boiling based on the presence of bulk fluid motion

1. Pool Boiling

The liquid above the hot surface is stationary. Fig. 6(a)

The only motion near the surface is because of free convection and the motion of the bubbles under the influence of buoyancy.

The pool boiling occurs in steam boilers. Pool boiling of a fluid can also be achieved by placing a heating coil in the fluid.

2. Forced Convection Boiling / Flow Boiling

The fluid motion is induced by external means such as a pump. Fig. 6(a)

The liquid is pumped and forced to move in a heated pipe or over a surface in a controlled manner. The free convection and the bubble-induced mixing also contribute towards the fluid motion.

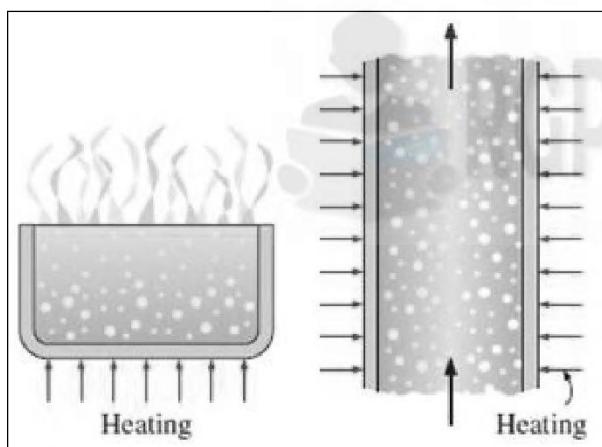


Fig. 6 (a) Pool Boiling and Flow Boiling

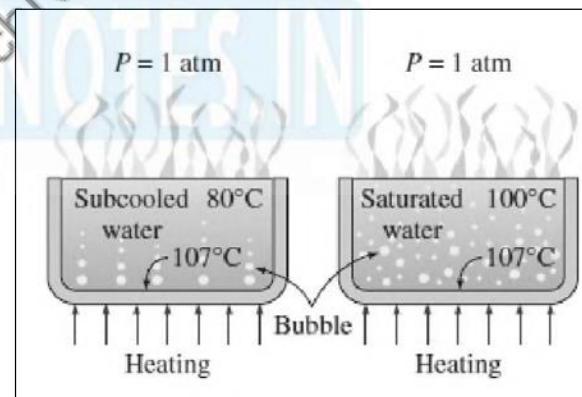


Fig. 6 (b) Sub-cooled Boiling and Saturated Boiling

B. Classification of boiling based on the presence of bulk liquid temperature

1. Sub-cooled or Local Boiling

The temperature of the liquid is below the saturation temperature and boiling takes place only in the vicinity of the heated surface. Fig. 6(b)

The vapor bubbles travel a short path and then vanish; apparently, they condense in the bulk of the liquid which is at a temperature less than a boiling point or saturation temperature.

2. Saturated Boiling

The temperature of the liquid exceeds the saturation temperature. Fig. 6(b)

The vapor bubbles generated at the solid surface (solid-liquid interface) are transported through the liquid by buoyancy effects and eventually escape from the surface (liquid-vapor interface).

The actual evaporation process then sets in.



Boiling Regimes

Whether the boiling phenomenon corresponds to pool boiling or forced circulation boiling, there are some definite regimes of boiling associated with progressively increasing heat flux.

Nukiyama (1934) was the first to identify different regimes of pool boiling using the apparatus of Fig. 7. These different regimes can be illustrated by considering an electrically heated horizontal nichrome/Platinum wire submerged in a pool of liquid at saturation temperature.

Fig. 7 shows the relationship between heat flux and the temperature excess ($T_s - T_{sat}$); Where,

Different boiling regimes are:

A. Natural Convection Boiling

B. Nucleate Boiling

C. Film Boiling

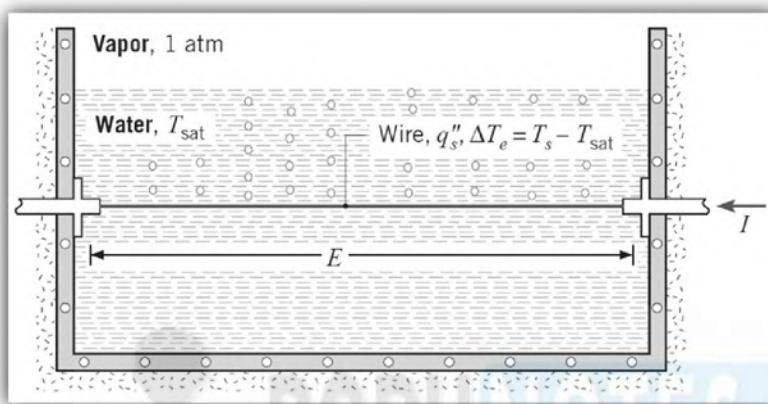


Fig. 7 Nukiyama's power controlled heating apparatus for demonstrating the boiling curve

A. Natural / Free Convection Boiling (up to point A on boiling curve)

The boiling takes place in a thin layer of liquid which adjoins the heated surface.

The liquid near the wall becomes superheated, i.e. temperature of the liquid exceeds the saturation temperature at the given pressure.

The superheated liquid rises to the liquid-vapor interface where evaporation takes place.

The fluid motion is by free convection effects.

The heat transfer rate increases, but gradually, with growth in a temperature excess.

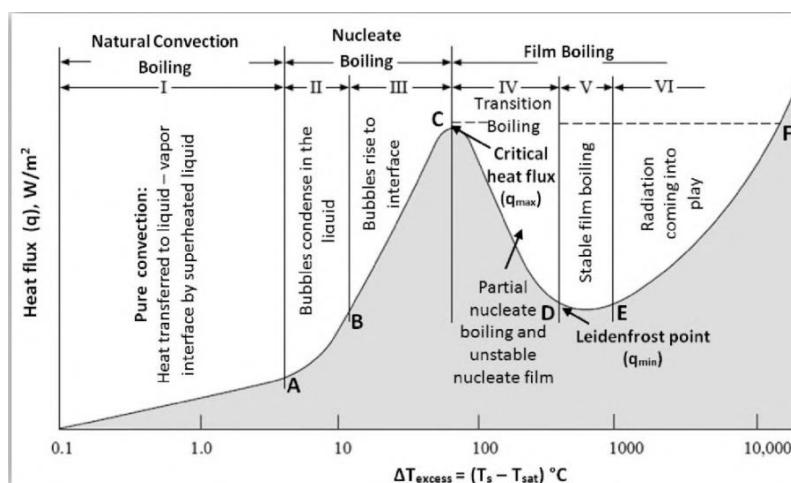


Fig. 8 boiling curve for saturated water at atmospheric pressure

B. Nucleate Boiling (between points A & C on boiling curve)

When the liquid is overheated to saturation temperature, vapor bubbles are formed at certain favorable spots called the **Nucleation or Active sites**. Point **A** is referred to as the **onset of nucleate boiling, ONB**.

The nucleate boiling regimes can be separated into two distinct regions:

A – B:- Isolated bubbles are formed at various nucleation sites, on the heated surface but these bubbles get condensed in the liquid after detaching from the surface.

B – C:- Heater temperature is further increased. Bubbles form at very high rates and they form continuous columns of vapor in the liquid.

The liquid is quite hot and the bubbles do not condense in it.

These bubbles rise to the free surface, where they break up and release their vapor content and that helps in rapid evaporation.

The space vacated by the rising bubbles is filled by the liquid in the vicinity of the heated surface, and the process is repeated.

The agitation or stirring caused by the entrainment of the liquid to the heated surface and rapid evaporation is responsible for the increased heat transfer coefficient and heat flux in the nucleate boiling region.

The heat flux hence reaches a maximum at point **C**, which is called the **critical /maximum heat flux, q_{max}** .

Nucleate boiling is the most desirable boiling regime in practice because high heat transfer rates can be achieved in this regime with relatively small values of ΔT_{excess} .

C. Film Boiling (beyond point C on Boiling curve) & Transition Boiling (between point C & D)

As the heater temperature and thus ΔT_{excess} is increased past point **C**, the heat flux decreases as shown in Fig. 9.

This is because a bubble formation is very rapid; the bubbles blanket the heating surface and prevent the incoming fresh liquid from taking its place.

A large fraction of the heating surface is covered by a vapor film, which acts as insulation due to the low thermal conductivity of the vapor.

In the transition boiling regime, both nucleate and film boiling partially occurs.

Nucleate boiling at point **C** is completely replaced by film boiling at point **D**.

Operation in the transition boiling regime, which is also called the unstable film boiling regime, is avoided in practice.

Beyond point D

In this region, the heated surface is completely covered by a continuous stable vapor film.

The temperature differences are so large that radiant heat flux becomes significant, and the heat flux curve begins to rise upward with increasing ΔT_{excess} . That marks the region of stable film boiling.

The phenomenon of stable film boiling is referred to as the “**Leidenfrost effect**” and point **D**, where the heat flux reaches a minimum, is called the Leidenfrost point.



Burn out point (Point F)

To move beyond point C, where q_{\max} occurs, we must increase the heated surface temperature (T_s).

To increase T_s , however, we must increase the heat flux. But the fluid cannot receive this increased energy beyond point C, and the heated surface temperature (T_s) rises even further.

If the surface temperature exceeds the temperature limit of the wall material, burn out (structural damage & failure) of the wall occurs.

Bubble Growth

The bubble formation in nucleate boiling is greatly influenced by the nature and condition of the heating surface and surface tension at the solid-liquid interface (Shape, size, and inclination of bubbles, however, do not have much effect on the heat transfer rate).

The surface tension signifies the wetting capability of the surface with the liquid (i.e. low surface tension \rightarrow Highly wetted surface) and that influences the angle of contact between the bubble and solid surface.

Any contamination of the surface would affect its wetting characteristics and influence the size and shape of the vapor bubbles.

If the surface tension of the liquid is low, it tends to wet the surface (fully wetted surface), so that the bubble is readily pushed by the liquid and rises. The vapor bubbles tend to become globular or oval in shape as shown in Fig. 9(a) and they are disengaged from the surface.

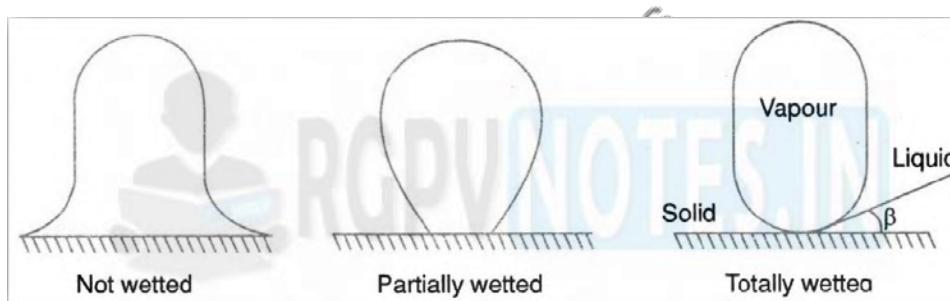


Fig. 9(a) Wetting characteristics for typical vapor bubbles

In the case of liquid having intermediate surface tension (partially wetted surface) a momentary balance may exist between the bubbles and solid surface so that it is necessary to form larger bubbles before the buoyant force can free them from the surface; the shape of the bubble is shown in Fig. 9(a).

On the unwetted surface, the bubbles spread out as shown in Fig. 9(a) forming a wedge between the water and heating surface, thereby allowing hydrostatic forces to resist the action of buoyancy.

The formation of the bubble with the fully wetted surface as shown in Fig. 9(a) gives a high heat transfer rate compared with the bubble shapes shown in Fig. 9(a) because the area covered by the insulating vapor film is the smallest.

Experimental evidence does indicate that the vapor bubbles are not always in thermodynamic equilibrium with the surrounding liquid.

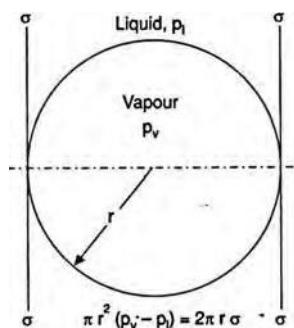


Fig. 9(b) Force balance for a spherical bubble

The vapor inside the bubble is not necessarily at the same temperature as the liquid and the vapor pressure P_v inside the bubble exceeds the liquid pressure P_l acting from outside of the bubble. Fig. 9(b) indicates one such spherical bubble with various forces acting on it.

- i. The resultant pressure ($P_v - P_l$) acts on area πr^2 and the pressure force equals $\pi r^2(P_v - P_l)$.
- ii. The surface tension σ of the vapor-liquid interface acts on the interface length $2\pi r$ and the surface tension force equals $2\pi r\sigma$.

Factors affecting the nucleate pool boiling

1) Material, shape, and condition of the surface:

Under identical conditions of pressure and temperature difference, the boiling heat transfer coefficient is different for different metals; copper has a high value compared to steel. Further, a rough surface gives a better heat transmission than when the surface is either smooth or has been coated to weaken its tendency to get wetted.

2) Pressure:

The temperature difference between the heating surface and the bulk and hence the rate of bubble growth is affected by pressure. The maximum allowable heat flux for a boiling liquid increases with pressure until critical pressure is reached and thereafter it declines.

3) Liquid properties:

Experiments have shown that the bubble size increases with the dynamic viscosity of the liquid. With the increase in bubble size, the frequency of bubble formation decreases, and that results in a reduced rate of heat transfer.

Condensation

"Condensation occurs when the temperature of a vapor is reduced below its saturation temperature corresponding to the vapor pressure."

This is usually done by bringing the vapor into contact with a solid surface whose temperature, T_s is below the saturation temperature T_{sat} of the vapor.

The latent energy of the vapor is released, heat is transferred to the surface, and the condensate is formed.

The condensation can also occur on the free surface of a liquid or even in gas when the temperature of the liquid or the gas to which the vapor is exposed is below T_{sat} .

In this chapter, we will consider surface condensation only.

Depending upon the behavior of condensate upon the cooled surface, the condensation process has been categorized into two distinct modes: (A) Film-wise condensation and (B) Dropwise condensation.

Dropwise and Film-wise Condensation

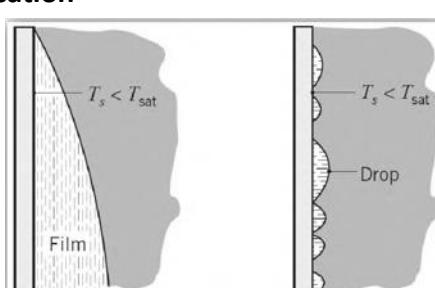


Fig. 10 Film-wise and Dropwise Condensation

A. Film-wise condensation

The liquid condensate wets the solid surface, spread out, and forms a continuous film over the entire surface.

The liquid flows down the cooling surface under the action of gravity and the layer continuously grows in thickness because of newly condensing vapor.

The continuous film offers resistance and restricts the further transfer of heat between the vapor and the surface.

Film condensation only occurs when a vapor relatively free from impurities, is allowed to condense on a clean surface.

Film condensation is generally a characteristic of clean, uncontaminated surfaces.

B. Dropwise condensation

The liquid condensate collects in droplets and does not wet the solid cooling surface.

The droplets develop in cracks, pits, and cavities on the surface, grow in size, break away from the surface, knock off other droplets, and eventually run off the surface without forming a film.

A part of the condensation surface is directly exposed to the vapor without an insulating film of condensate liquid.

There is no film barrier to heat flow and higher heat transfer rates are experienced.

Dropwise condensation has been observed to occur either on highly polished surfaces or surfaces contaminated with impurities like fatty acids and organic compounds.

Dropwise condensation gives a coefficient of heat transfer generally 5 to 10 times larger than with film condensation.

It is therefore common practice to use surface coatings that inhibit wetting and hence simulate drop-wise condensation.

Silicon, Teflon, and an assortment of waxes and fatty acids are often used for this purpose.

However such coatings gradually lose their effectiveness due to oxidation, fouling, or outright removal and film condensation eventually occur.

Although it is desirable to achieve drop-wise condensation in industrial applications, it is often difficult to maintain this condition.

Condenser design calculations are often based on the assumption of film condensation.

Influence of the presence of non-condensable gases

The presence of non-condensable gas such as air in a condensing vapor produces a detrimental (negative) effect on the heat transfer coefficient.

It has been observed that even with a few percent by volume of air in steam the condensation heat transfer coefficient is reduced by more than 50%.

This is because when a vapor (containing non-condensable gas) condenses, the non-condensable gas is left at the surface.

Any further condensation at the surface will occur only after the incoming vapor has diffused through this non-condensable gas collected in the vicinity of the surface.

The non-condensable gas adjacent to the surface acts as a thermal resistance to the condensation process. The rate of condensation decreases greatly when the condensable vapor is contaminated with even very small amounts of non-condensable gases.

As the presence of non-condensable gas in a condensing vapor is undesirable, the general practice in the design of a condenser should be to vent the non-condensable gas to the maximum extent possible.

UNIT –IV

Unit 4 Heat Exchangers: Types- parallel flow, counter flow; evaporator and condensers, overall heat transfers coefficient, fouling factors, log-mean temperature difference (LMTD), method of heat exchanger analysis, effectiveness of heat exchanger, NTU method;

Mass transfer: Fick's law, equi-molar diffusion, diffusion coefficient, analogy with heat transfer, diffusion of vapour in a stationary medium.

Introduction

"Heat exchanger is process equipment designed for the effective transfer of heat energy between two fluids; a hot fluid and a coolant". The purpose may be either to remove heat from a fluid or to add heat to a fluid.

Examples of heat exchangers:

- Intercoolers and pre-heaters,
- Condensers and boilers in steam plant,
- Condensers and evaporators in refrigeration unit,
- Regenerators,
- Automobile radiators,
- Oil coolers of heat engine,
- Evaporator of an ice plant and milk-chillers of a pasteurizing plant,

The heat transferred in the heat exchanger may be in the form of latent heat (i.e. in boilers & condensers) or sensible heat (i.e. in heaters & coolers).

Types of Heat Exchangers

Many types of heat exchangers have been developed to meet the widely varying applications. Heat exchangers are typically classified according to:

A. Nature of heat exchange process:

1. **Direct contact or open heat exchanger:** Complete physical mixing of hot and cold fluid and reach a common temperature, Simultaneous heat and mass transfer. Use is restricted, where mixing between two fluids is harmful.

Examples: (I) Water cooling towers - in which a spray of water falling from the top of the tower is directly contacted and cooled by a stream of air flowing upward and (II) Jet condensers.

2. **Regenerators:** In a regenerator the hot fluid is passed through a certain medium called "matrix", serves as heat storage.

The heat is transferred and stored in solid matrix and subsequently transferred to the cold fluid.

The effectiveness of regenerator is depends upon the heat capacity of the regenerating material.

In a fixed matrix configuration, the hot and cold fluids pass alternately through a stationary matrix, and for continuous operation two or more matrices are necessary, as shown in Fig. 1(a). One commonly used arrangement for the matrix is the “packed bed”. Another approach is the rotary regenerator in which a circular matrix rotates and alternately exposes a portion of its surface to the hot and then to the cold fluid, as shown in Fig. 1(b).

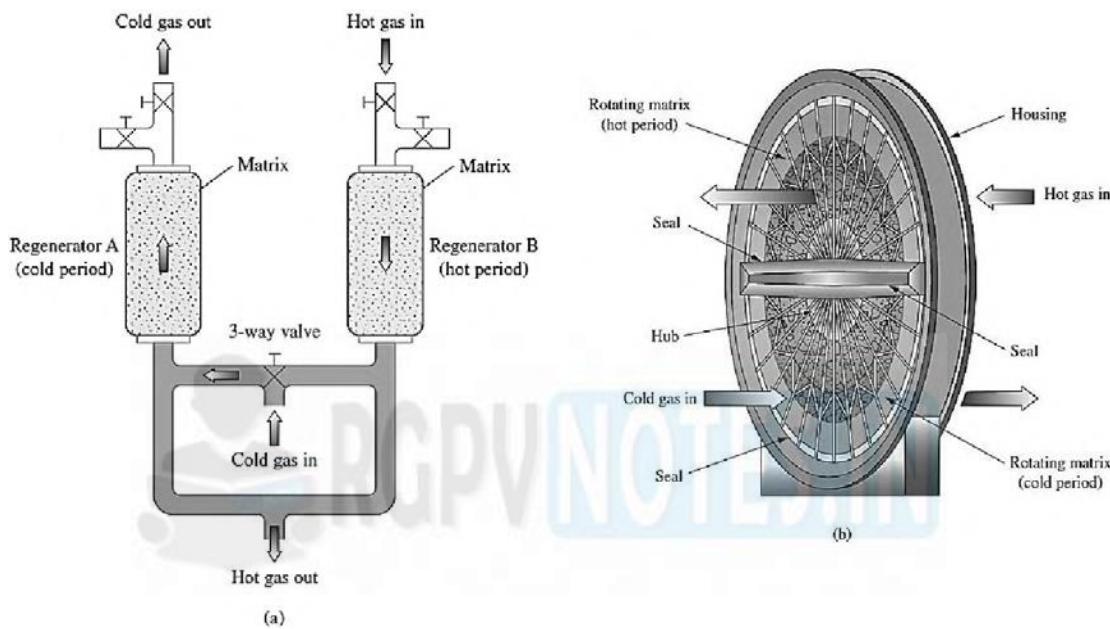


Fig.1 (a) Fixed dual-bed regenerator (b) Rotary regenerator

3. Recuperator:

In this type of heat exchanger the hot and cold fluids are separated by a wall and heat is transferred by a combination of convection to and from the wall and conduction through the wall. The wall can include extended surfaces, such as fins.

Majority of the industrial applications have recuperator type heat exchangers.

B. Relative direction of motion of fluids

1. Parallel flow

- Hot and cold both the fluids flow in the same direction

2. Counter flow

- Flow of fluids is opposite in direction to each other
- Gives maximum heat transfer rate

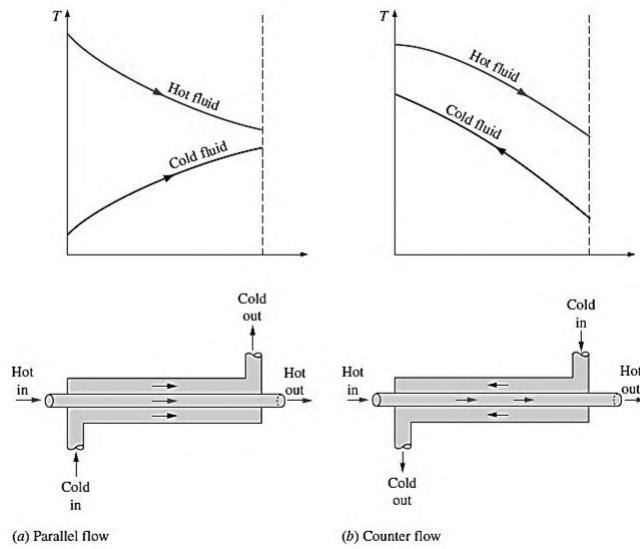


Fig.2 Different flow regimes and temperature profiles in a double-pipe heat exchanger

3. Cross flow arrangement

- Two fluids are directed perpendicular to each other.
 - Examples: Automobile radiator and cooling unit of air-conditioning duct.
 - The flow of the exterior fluid may be by forced or by natural convection.
 - Fig.3 shows different configurations used in cross-flow heat exchangers.

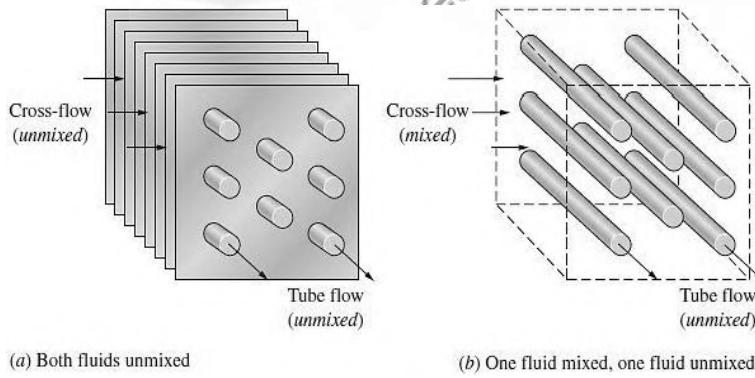


Fig.3 Different flow configurations in cross-flow heat exchangers

C. Mechanical design of heat exchange surface

1. Concentric tube heat exchanger

- Two concentric pipes.
 - Each carrying one of the fluids.
 - The direction of flow may correspond to parallel or counter flow arrangement as shown in Fig. 2

3 Shell & tube heat exchanger

- One of the fluids is carried through a bundle of tubes enclosed by a shell and other fluid is forced through shell and flows over the outside surface of tubes.
- The direction of flow for either or both fluids may change during its passage through the heat exchanger.

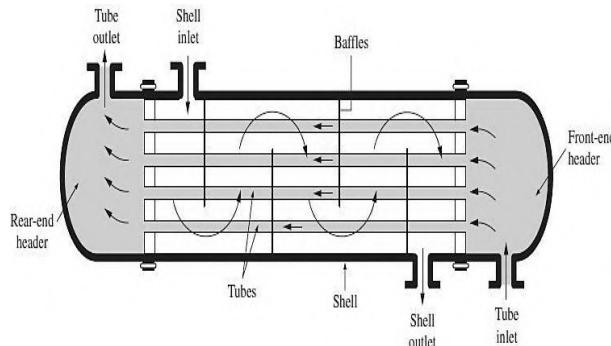


Fig.4 Shell & tube heat exchanger with one shell pass and one tube pass (1-1 exchanger)

3. Mechanical design of heat exchange surface

- Single-pass: Two fluids may flow through the exchanger only once as shown in Fig.4.
- Multi-pass: One or both fluids may traverse the exchanger more than once as shown in Fig.5.
- Baffles are provided within a shell which cause the fluid surrounding the tubes (shell side fluid) to travel the length of shell a no. of times.
- An exchanger having n – shell passes and m – tubes passes is designed as $n-m$ exchanger.
- A multiple shell & tube exchanger is preferred to ordinary counter flow design due to its low cost of manufacture, easy dismantling for cleaning and repair and reduced thermal stresses due to expansion.

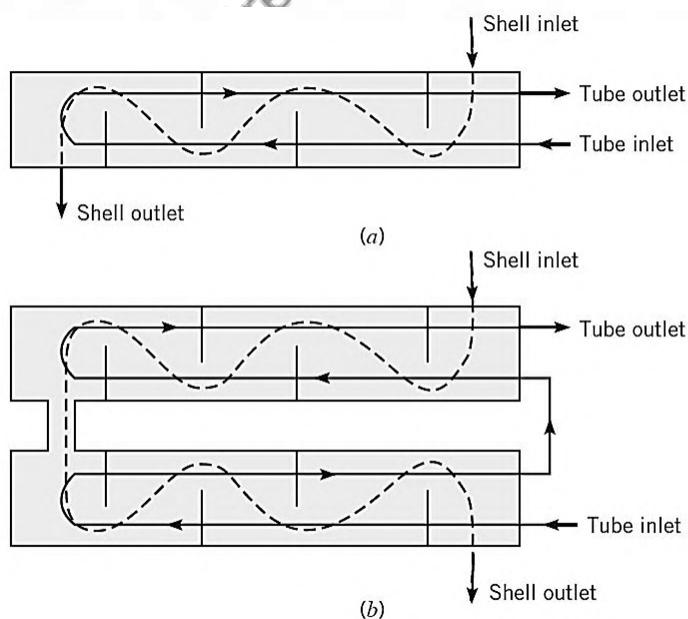


Fig. 5 Shell & tube heat exchangers (a) one shell pass and two tubes passes (b) Two shell passes and four tube passes.

4. Physical state of heat exchanging fluids

The direction of flow is immaterial in these cases and the LMTD will be the same for both parallel flow, counter flow and other flow types. Refer Fig. 6.

a. Condenser

The temperature of hot fluid will remain constant throughout the heat exchanger. (Only latent heat is transferred)

b. Evaporator

The temperature of cold fluid will remain constant throughout the heat exchanger. (Only latent heat is transferred)

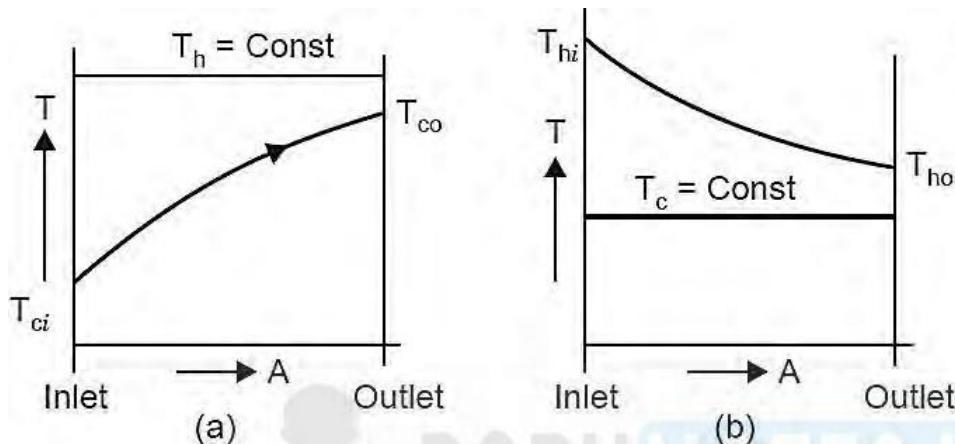


Fig. 6 (a) Condensing (b) Evaporating

Heat Exchanger Analysis

Fig. 7 represents the block diagram of a heat exchanger. The governing parameters are:

- Overall heat transfer co-efficient (U) due to various modes of heat transfer
- Heat transfer surface area
- Inlet and outlet fluid temperatures

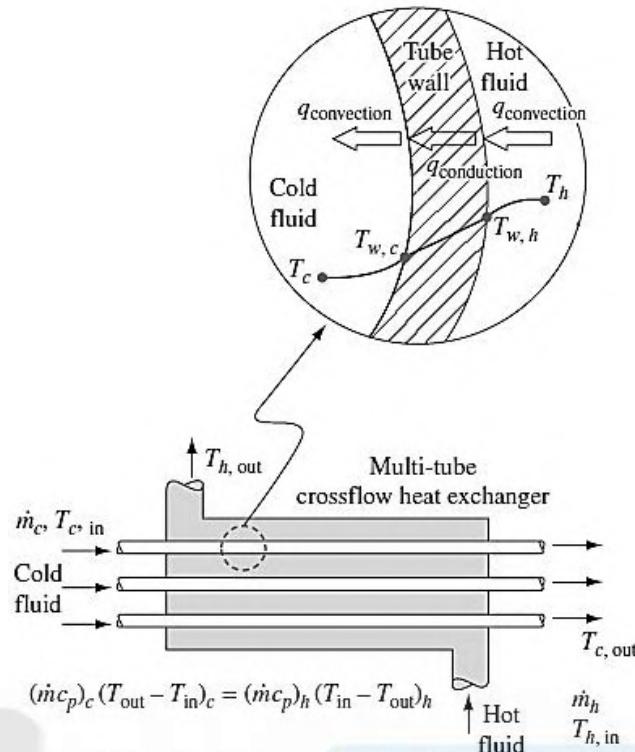


Fig. 7 Overall energy balance in heat exchanger

Overall Heat Transfer Co-efficient

- A heat exchanger is essentially a device in which energy is transferred from one fluid to another across a good conducting solid wall.
- The rate of heat transfer between two fluids is given by,

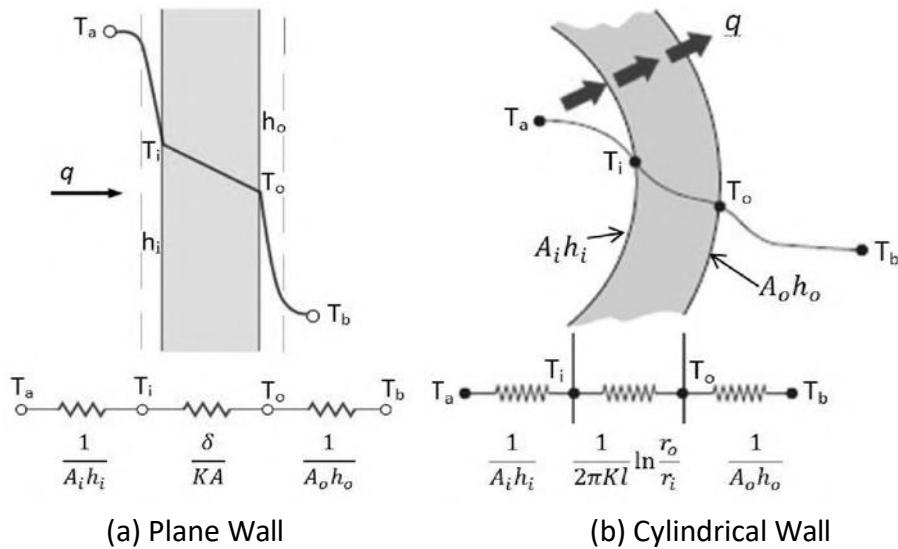


Fig. 8 Thermal resistance network for (a) plane and (b) cylindrical separating wall

When the two fluids of the heat exchanger are separated by a plane wall as shown in Fig. 8 (a), the thermal resistance comprises:

- (i) Convection resistance due to the fluid film at the inner surface
- (ii) Wall conduction resistance
- (iii) Convection resistance due to fluid film at the outer surface

Fouling Factor

Equations are essentially valid only for clean and un-corroded surface. However during normal operation the tube surfaces get covered by deposits of ash, soot (smoke), dirt and scale etc. This phenomenon of rust formation and deposition of fluid impurities is called Fouling.

The surface deposits increase thermal resistance with a corresponding drop in the performance of the heat exchange equipment. Since the thickness and thermal conductivity of the scale deposits are difficult to determine, the effect of scale on heat flow is considered by specifying an "Equivalent Scale Heat Transfer Co-efficient",

If h_i and h_o denote the heat transfer co-efficient for the scale formed on the inside and outside surfaces respectively, then the thermal resistance due to scale formation on the inside surface is,

Important Points

- The overall heat transfer co-efficient (U) depends upon the flow rate and properties of the fluid, the material thickness and surface condition of tubes and the geometrical configuration of the heat exchanger.
- High conducting liquids such as water and liquid metals give higher values of heat transfer co-efficient (h) and overall heat transfer co-efficient (U).
- For an efficient and effective design, there should be no high thermal resistance in the heat flow path; all the resistance in the heat exchanger must be low.

Logarithmic Mean Temperature Difference (LMTD)

- During heat exchange between two fluids, the temperature of the fluids, change in the direction of flow and consequently there occurs a change in the thermal head causing the flow of heat.
- In a parallel flow system, the thermal head (temperature potential) causing the flow of heat is maximum at inlet and it goes on diminishing along the flow path and becomes minimum at the outlet.
- In a counter flow system, both the fluids are in their coldest state at the exit.
- To calculate the rate of heat transfer by the expression, an average value of the temperature difference (i.e. LMTD) between the fluids has to be determined.

Assumptions made to derive expression for LMTD:

1. The overall heat transfer co-efficient, U is constant.
2. The flow conditions are steady.
3. The specific heats and mass flow rate of both fluids are constant.

4. There is no loss of heat to surrounding i.e. the heat exchanger is perfectly insulated.
5. There is no change of phase either of the fluid during the heat transfer.
6. The changes in potential and kinetic energies are negligible.
7. Axial conduction along the tubes of the heat exchanger is negligible.

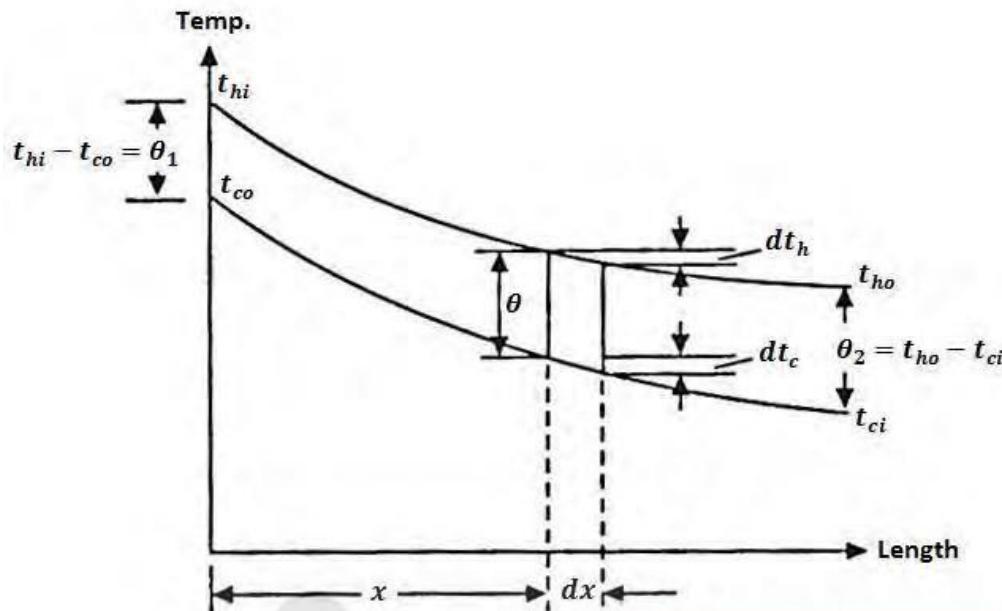


Fig. 9 Temperature changes of fluids during counter flow arrangement

- LMTD for Counter Flow Heat Exchanger
- LMTD for Parallel Flow Heat Exchanger
- Arithmetic Mean Temperature Difference (AMTD)
- When the temperature variation of the fluids is relatively small, then temperature variation curves are approximately straight lines (as in condenser and evaporator) and sufficiently accurate results are obtained by taking the arithmetic mean temperature difference (AMTD).

Correction Factors for Multi-pass Arrangements

- The relation for LMTD is essentially applicable for the single pass heat exchangers.
- The effect of multi-tubes, several shell passes or cross flow in an actual flow arrangement is considered by identifying a correction factor F such that depends on geometry of the heat exchanger and the inlet and outlet temperatures of hot and cold fluid streams.
- Correction factors for several common arrangements have been given in Figs. 10 to 13.
- The data is presented as a function of two non-dimensional temperature ratios P and R. the parameter P is the ratio of the rise in temperature of the cold fluid to the difference in the inlet temperatures of the two fluids and the parameter R defines the ratio of the temperature drop of the hot fluid to temperature rise in the cold fluid.
- Since no arrangement can be more effective than the conventional counter flow, the correction factor F is always less than unity for shell and tube heat exchanger.

- Its value is an indication of the performance level of a given arrangement for the given terminal fluid temperatures.
- When a phase change is involved, as in condensation or boiling, the fluid normally remains at essentially constant temperature. For these conditions, P or R becomes zero and we obtain

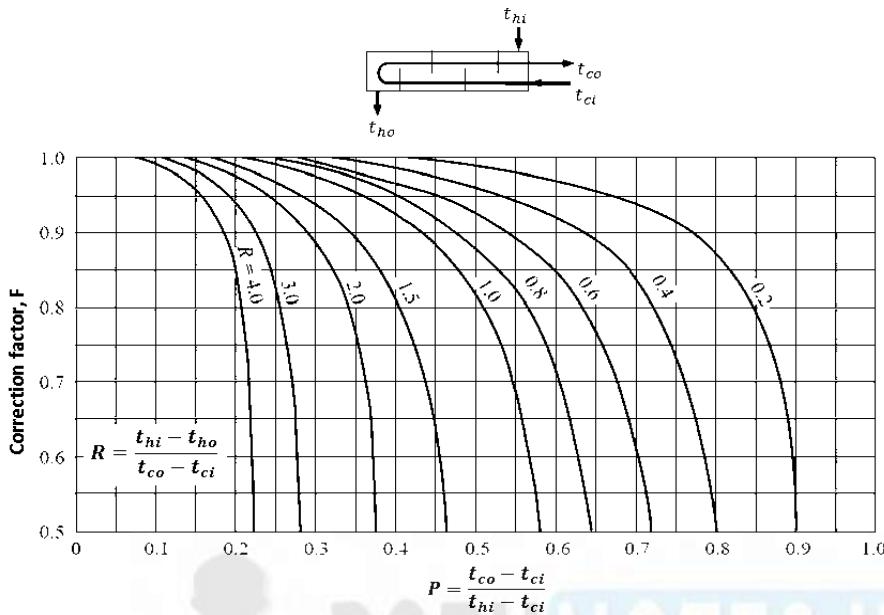


Fig. 10 Correction-factor plot for exchanger with one shell pass and two, four, or any multiple of tube passes

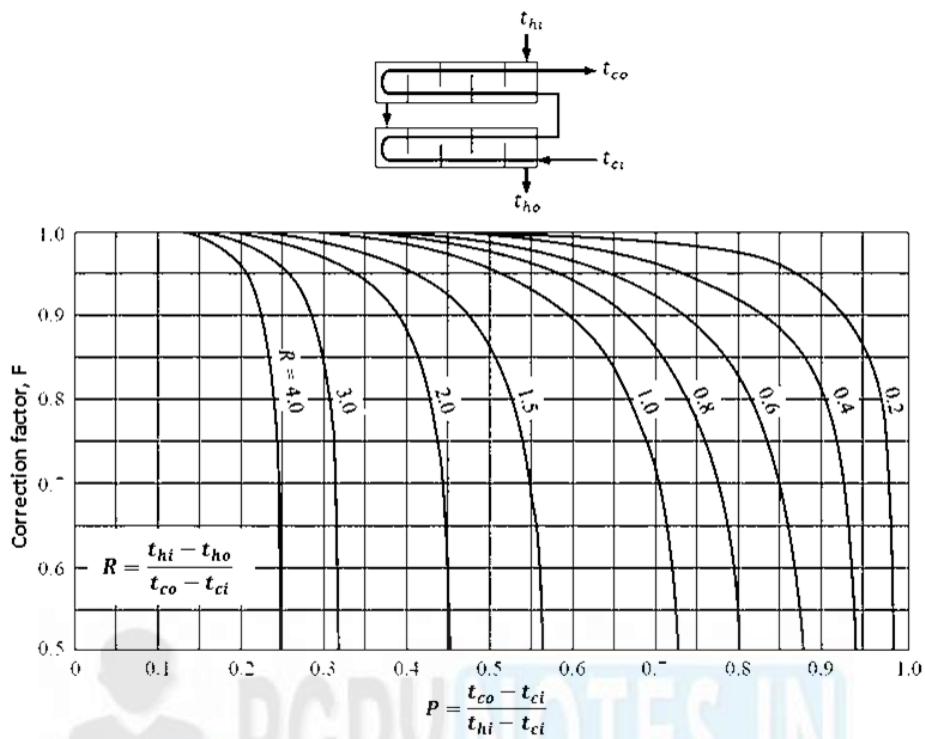


Fig. 11 Correction-factor plot for exchanger with two shell passes and four eight or any multiple of tube passes

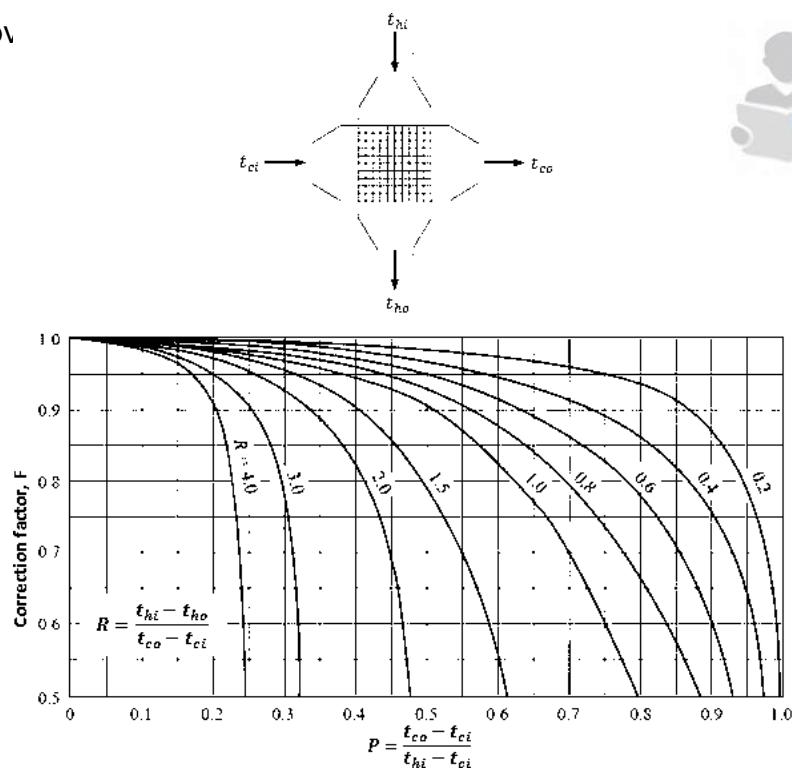


Fig. 12 Correction factor plot for single pass cross-flow heat exchanger with both fluids unmixed

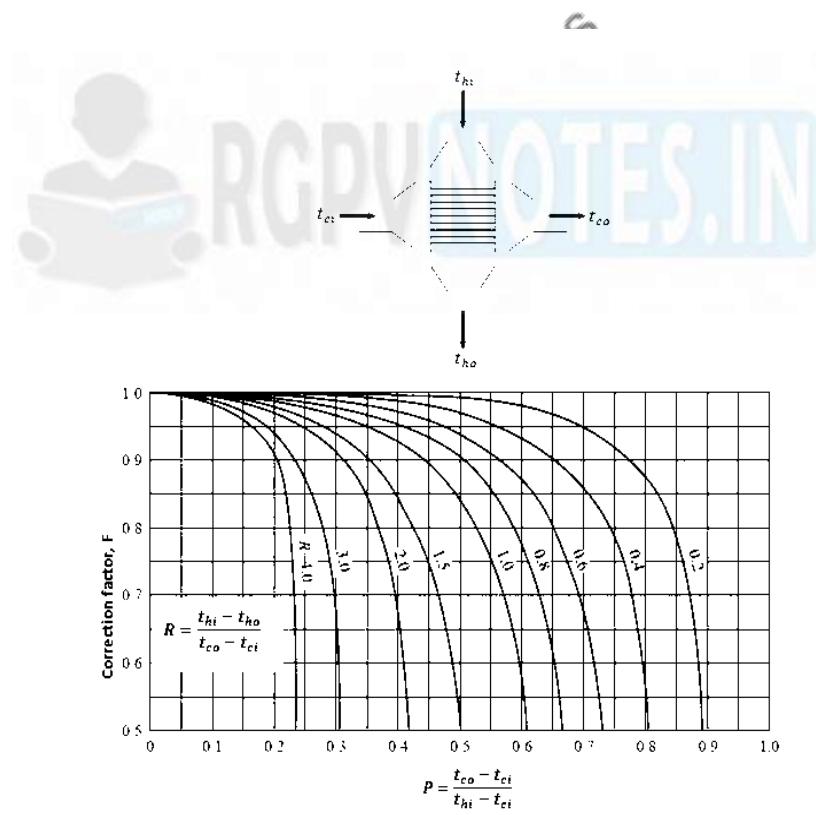


Fig. 13 Correction factor plot for single-pass flow heat exchanger, one fluid mixed and the other unmixed

Effectiveness and Number of Transfer Units (NTU)

- The concept of LMTD for estimating/analysing the performance of a heat exchanger unit is quite useful only when the inlet and outlet temperature of the fluids are either known or can be determined easily from the relevant data.
- In normal practice the useful design is however based on known fluid inlet temperatures and estimated heat transfer co-efficient. The unknown parameters may be the outlet conditions and heat transfer or the surface area required for a specified heat transfer.
- An analysis/estimate of the heat exchanger can be made more conveniently by the NTU approach, which is based on the capacity ratio, effectiveness and number of transfer units.

Capacity Ratio (C):

The product of mass and specific heat () of a fluid flowing in a heat exchanger is termed as the Capacity rate. It indicates the capacity of the fluid to store energy at a given rate. "The ratio of minimum to maximum capacity rate is defined as Capacity ratio(C).

Capacity rate of the hot fluid,

Capacity rate of the cold fluid,

Number of Transfer Units (NTU):

- NTU is a dimensionless parameter.
- It is a measure of the (heat transfer) size of the heat exchanger.
- The larger the value of NTU, the closer the heat exchanger reaches its thermodynamic limit of operation.

Mass Transfer: The basic fundamental law of mass transfer is Fick's law.

Fick's Laws: The simplest description of diffusion is given by Fick's laws, which were developed by Adolf Fick in the 19th century:

The molar flux due to diffusion is proportional to the concentration gradient.

The rate of change of concentration at a point in space is proportional to the second derivative of concentration with space.

Fick's First Law of Diffusion:

Writing the first law in a modern mathematical form:

$$N_i = D_i \nabla C_i$$

Where for species i, N_i is the molar flux ($\text{mol m}^{-2} \text{s}^{-1}$), D_i is the diffusion coefficient ($\text{m}^2 \text{s}^{-1}$), and C_i is the concentration (mol m^{-3}).

$$\frac{\partial C_i}{\partial t} + \nabla \cdot N_i = 0$$

We can derive Fick's second law directly:

$$\frac{\partial C_i}{\partial t} = D_i \nabla^2 C_i$$

This assumes that D_i is a constant, which is only true for dilute solutions. This is usually a good assumption for diffusion in solids; diffusion of chemicals in a dilute solution, water, or other typical liquid solvents; and diffusion of dilute (trace) species in the gas phase, such as carbon dioxide in air.

Fick's Second Law of Diffusion

Fick's second law of diffusion is a linear equation with the dependent variable being the concentration of the chemical species under consideration. Diffusion of each chemical species occurs independently. These properties make mass transport systems described by Fick's second law easy to simulate numerically.

When modeling diffusion, it is often a good idea to begin with the assumption that all diffusion coefficients are equal and independent of temperature, pressure, etc. This simplification ensures the linearity of the mass transport equations in the modeled domain and often allows simpler correlations to known analytical limits. This assumption can be relaxed once the behavior of a system with all equal diffusion coefficients is well understood.

Dimensional analysis of Fick's second law reveals that, in diffusive processes, there is a fundamental relation between the elapsed time and the square of the length over which diffusion takes place. Understanding this relation is very important for an accurate numerical simulation of diffusion.

Multi-Component Diffusion

For concentrated solutions or gas mixtures where more than one chemical species is present in significant mass fractions, it is no longer the case that the diffusion coefficient can be treated as constant or composition-independent. The interactivity of the different species' molecules with each other is too prevalent for a physical description to ignore these inter-molecular dependencies. Therefore, the diffusion coefficient becomes a tensor and the equation for diffusion is altered to relate the mass flux of one chemical species to the concentration gradients of all chemical species present. The necessary equations are formulated as the Maxwell-Stefan description of diffusion; they are often applied to describe gas mixtures, such as gas in a reactor or the mix of oxygen, nitrogen, and water in a fuel cell cathode.

In Maxwell-Stefan diffusion, the sensible choices of dependent variables are not the species concentrations, but rather the species mole or mass fractions (x_i and ω_i respectively). These are constrained as:

$$\sum_i x_i = 1, \sum_i \omega_i = 1$$

and related to the concentration and each other as:

$$xi = Ci / \sum j j Cjj$$

$$\omega_i = M_i C_i / \sum M_j j C_j j$$

Here, M_i , is the relative molar mass (kg mol^{-1}) of species i .

The diffusive mass flux of each species is, in turn, expressed based on the gradients of the mole or mass fractions, using multi-component diffusion coefficients. These are symmetric, so that an n -component system requires $n(n-1)/2$ independent coefficients to parameterize the rate of diffusion of its components. For four-component or more complex mixtures, these quantities are often unknown. Simplifications can be applied to the Maxwell-Stefan equations in order to employ the equivalent Fick's law diffusivity.

Most often, systems involving concentrated mixtures require convection and momentum conservation (fluid flow) to be solved with diffusion.

Analogy of mass transfer with heat transfer:

Mass (or heat) transfer between flowing fluid and reactor internals is usually described in terms of dimensionless numbers:

For mass transfer: Sherwood number (Sh), is a function of Reynolds number Re , and Schmidt number $Sc = \mu\cdot\eta - 1 \cdot D - 1$ that characterizes diffusion properties of flowing media. D is the mass diffusivity (m^2/s), μ is the dynamic viscosity of the fluid, ρ is the density of the fluid.

For heat transfer: Nusselt number Nu , is a function of Reynolds number Re , and Prandtl number $Pr = Cp/\mu\cdot k$ that characterizes thermal properties of the fluid. μ dynamic viscosity, k : thermal conductivity, Cp : specific heat,

Sometimes, only one of the transport characteristics—heat or mass transfer—is known. Then, the Chilton-Colburn analogy may be used to derive the missing equation. The analogy is strictly valid for the turbulent flow. However, it is commonly applied to the flow-through packed columns, monoliths, or solid foams. The analogy assumes identical mechanisms of heat and mass transfer, thus the dependence of Sherwood and Nusselt numbers on the Reynolds number has to be identical. The usual form of the Chilton-Colburn analogy is:

$$j = Nu/Re \times Pr^{1/3} = Sh/ Re Sc^{1/3}.$$

The so-called Colburn factor j is assumed to be a function of Reynolds number and is identical for heat and mass transfer.

Diffusion of vapour in a stationary medium: Stationary media with specified surface concentration in the diffusion of gas from containers, there is diffusion of gas from inside to the outside without the metal molecules diffusing into the gas. In these cases the concentration of gas at the surfaces should be known. The solubility of the gas in the surface determines the concentration at the surface. These cases are similar to conduction through the medium. In these cases the temperature potential in conduction is replaced by concentration potential ($C_{a1} - C_{a2}$) for component A. The flow rate can be obtained as in the case of conduction.

$$Na = (C_{a1} - C_{a2})/R,$$

Where R is the resistance of diffusion.

End





Heat and Mass Transfer (ME-701)

UNIT –V

Unit 5 Thermal Radiation: Nature of radiation, emissive power, absorption, transmission, reflection and emission of radiation, Planck's distribution law, radiation from real surfaces; radiation heat exchange between black and gray surfaces, shape factor, analogical electrical network, radiation shields.

Boiling and condensation: Film-wise and drop wise condensation; Nusselt theory for film-wise condensation on a vertical plate and its modification for horizontal tubes; boiling heat transfer phenomenon, regimes of boiling, boiling correlations.

Introduction Radiation Process and Properties

- Consider a hot object that is suspended in an evacuated chamber whose walls are at room temperature (Figure 1). The hot object will eventually cool down and reach thermal equilibrium with its surroundings.
- That is, it will lose heat until its temperature reaches the temperature of the walls of the chamber.
- Heat transfer between the object and the chamber could not have taken place by conduction or convection, because these two mechanisms cannot occur in a vacuum.
- Therefore, heat transfer must have occurred through another mechanism that involves the emission of the internal energy of the object. This mechanism is radiation.

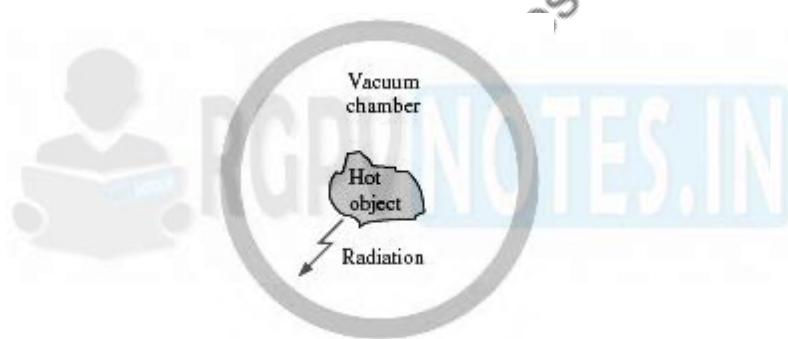


Fig. 1 Hot object in a vacuum chamber

Note: - Radiation differs from the other two heat transfer mechanisms in that it does not require the presence of a material medium to take place. In fact, energy transfer by radiation is fastest (at the speed of light) in a vacuum. Also, radiation transfer occurs in solids as well as liquids and gases. In most practical applications, all three modes of heat transfer occur concurrently at varying degrees. But heat transfer through an evacuated space can occur only by radiation. For example, the energy of the sun reaches the earth by radiation.

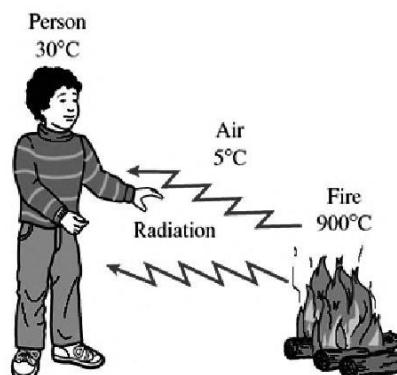


Fig. 2 Radiation heat transfer from hot to a cold body

Unlike conduction and convection, heat transfer by radiation can occur between two bodies, even when they are separated by a medium colder than both as shown in figure 2.

Salient Features and Characteristics of Radiation

- Radiation is the propagation and emission of energy in the form of electromagnetic waves.
- The electromagnetic waves are emitted as a result of vibrational and rotational movements of the molecular, atomic, or subatomic particles comprising the matter. When the body is excited by an oscillating electrical signal, electronic or neutronic bombardment, chemical reaction, etc, emission of radiation occurs.
- One form of radiation is differing from the other form of radiation by its frequency and wavelength. The relation between frequency and wavelength
- The general phenomenon of radiation covers the propagation of electromagnetic waves of all wavelengths, from short-wavelength gamma rays to long-wavelength microwaves.

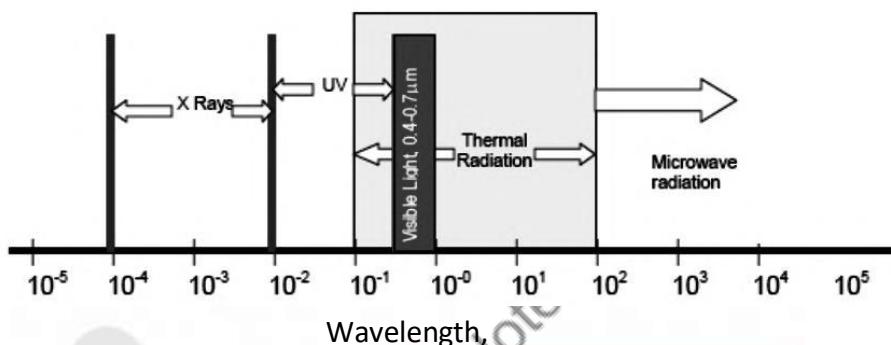


Fig. 3 Electromagnetic wave spectrum

- Thermal radiation is that electromagnetic radiation emitted by a body as a result of its temperature.
- Thermal radiation is limited to a range of wavelengths between 0.1 to 100 μm , which includes the entire visible and infrared and a part of the ultraviolet spectrum.
- Light is simply the visible portion of the electromagnetic spectrum that lies between 0.40 and 0.76 m.
- A body that emits some radiation in the visible range is called a light source. The sun is our primary light source.
- The radiation emitted by bodies at room temperature falls into the infrared region of the spectrum, which extends from 0.76 to 100 μm .
- The ultraviolet radiation includes the low-wavelength end of the thermal radiation spectrum and lies between the wavelengths 0.01 and 0.40 μm . Ultraviolet rays are to be avoided since they can kill micro-organisms and cause serious damage to humans and other living beings.
- About 12 percent of solar radiation is in the ultraviolet range. The ozone (O_3) layer in the atmosphere acts as a protective blanket and absorbs most of this ultraviolet radiation.
- Thermal radiation exhibits characteristics similar to those of light and follows the optical laws.
- Thermal radiation is continuously emitted by all matter whose temperature is above absolute zero.
- Body at higher temperature emits energy at a greater rate than bodies at low temperature.
- Normally a body radiating heat is simultaneously receiving heat from other bodies as incident radiation.
- Net heat exchange between two radiating surfaces is because one at high temperature radiates more and receives less energy for its absorption.
- An isolated body that remains at constant temperature emits just as much energy radiation as it receives.
- Heat transfer by radiation depends upon the level of temperature, unlike conduction and convection.

- Heat transfer by conduction and convection from the body at a temperature of 1000 to surrounding at a temperature of 800 practically remains the same for the body at a temperature of 900 to surrounding at a temperature of 700.
- Whereas in the case of radiation heat transfer, heat transfer is not the same even if the temperature differences are the same.
- Net heat transfer by radiation at elevated temperature is greater than heat transfer at low temperature.

Absorptivity, Reflectivity, and Transmissivity

When thermal radiation is incident on a surface, a part of the radiation may be reflected by the surface, a part may be absorbed by the surface and a part may be transmitted through the surface as shown in figure 4.

These fractions of reflected, absorbed, and transmitted energy is interpreted as system properties called reflectivity, absorptivity, and Transmissivity, respectively.

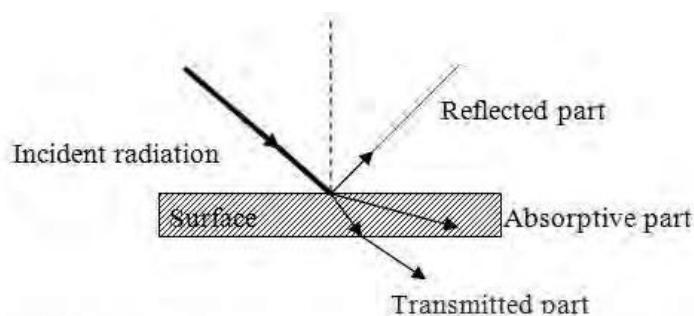


Fig. 4: Reflection, absorption, and transmitted energy thus using energy conservation,
A blackbody is defined as a perfect emitter and absorber of radiation. At a specified temperature and wavelength, no surface can emit more energy than a blackbody.

A blackbody absorbs all incident radiation, regardless of wavelength and direction. Also, a blackbody emits radiation energy uniformly in all directions per unit area normal to the direction of emission. For black body = 1, and

- In actual practice, there does not exist a perfectly black body that will absorb all incident radiations. Snow, with its absorptivity of 0.985, is nearly black to the thermal radiation.
- The absorptivity of a surface depends upon the direction of incident radiation, the temperature of the surface, composition, and structure of the irradiated surface, and the spectral distribution of incident radiation.
- When a surface absorbs a certain fixed percentage of impinging radiation, the surface is called a gray body. A surface whose properties are independent of the wavelength is known as a gray surface.
- A gray body is defined such that the monochromatic emissivity of the body is independent of wavelength. For gray body
- The condition of constant absorptivity too is not satisfied by the real materials and as such even a gray body remains a hypothetical concept like the black body.
- A body that reflects all the incident thermal radiations is called an absolutely white body or specular body. For white body= 1, and
- Regular (specular) reflection implies that the angle between the reflected beam and the normal to the surface equals the angle made by the incident radiation with the same normal.
- In a diffused radiation, the incident beam is reflected in all directions.
- Most of the engineering materials have rough surfaces, and these rough surfaces give diffused reflections.



- A body that allows all the incident radiations to pass through it is called a transparent body or diathermanous.
- Transmissivity varies with the wavelength of incident radiation. Material may be non-transparent for a certain wavelength transparent for another. This type of material is called selective transmitter
- A thin glass plate transmits most of the thermal radiations from the sun but absorbs in equally great measure the thermal radiations emitted from the low-temperature interior of a building.
- That's the reason to use the glass in the greenhouse to trap the solar radiation in low-temperature space.
- For opaque bodies, It means that good absorbers are bad reflectors or vice-versa.
- The electrons, atoms, and molecules of all solids, liquids, and gases above absolute zero temperature are constantly in motion, and thus radiation is constantly emitted, as well as being absorbed or transmitted throughout the entire volume of matter.
- That is, radiation is a volumetric phenomenon.
- Radiation in opaque solid is considered a surface phenomenon since the radiation emitted only by the molecules at the surface can escape the solid as shown in figure 5.

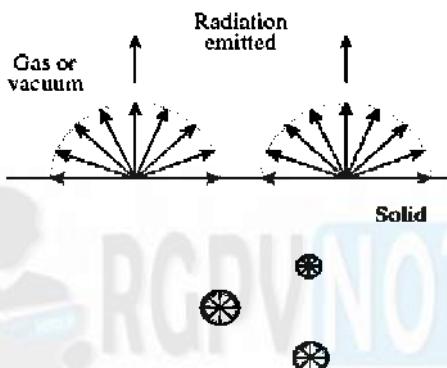


Fig. 5 Radiation in opaque solid

Black Body Concept

- Consider a large cavity with a small opening maintained at a constant temperature as shown in figure 6.
- The inner surface of the cavity is coated with a black lamp. A beam of thermal radiation entering the hole strikes the inner surface. Since the inner surface has high absorptivity, the major portion of the radiation is absorbed and only a small fraction is reflected.
- The weak reflected beam does not find any way out and again strikes the inner surface. Here it is again partly absorbed and partly reflected.
- Likewise, the reflected radiation is successively absorbed, and finally, when escapes out is, it has only a negligible amount of energy associated with it.

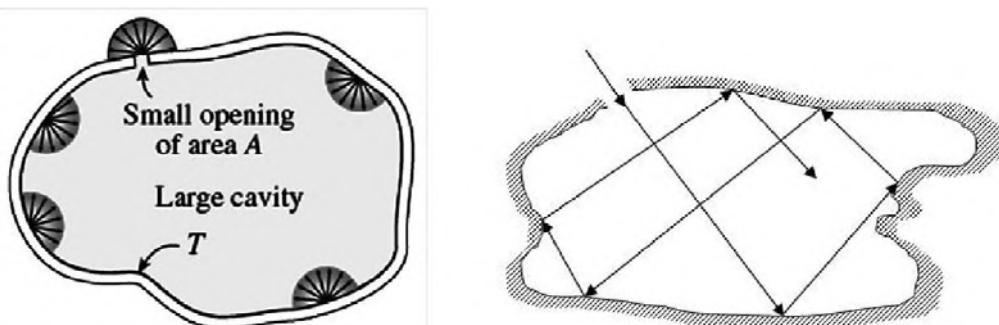


Fig.6 Black body concept



- Although a blackbody would appear black to the eye, a distinction should be made between the idealized blackbody and an ordinary black surface.
- Any surface that absorbs light (the visible portion of radiation) would appear black to the eye, and a surface that reflects it completely would appear white.
- Considering that visible radiation occupies a very narrow band of the spectrum from 0.4 to 0.76 μm , we cannot make any judgments about the blackness of a surface on the basis of visual observations.
- For example, snow and white paint reflect light and thus appear white. But they are essentially black for infrared radiation since they strongly absorb long-wavelength radiation. Surfaces coated with lampblack paint approach idealized blackbody behaviour.

Spectral and Spatial Energy Distribution

- **Spectral Energy Distribution:** The radiation emitted by the body consists of electromagnetic waves of various wavelengths. The distribution of radiation with wavelength is called spectral energy distribution as shown in figure 7.

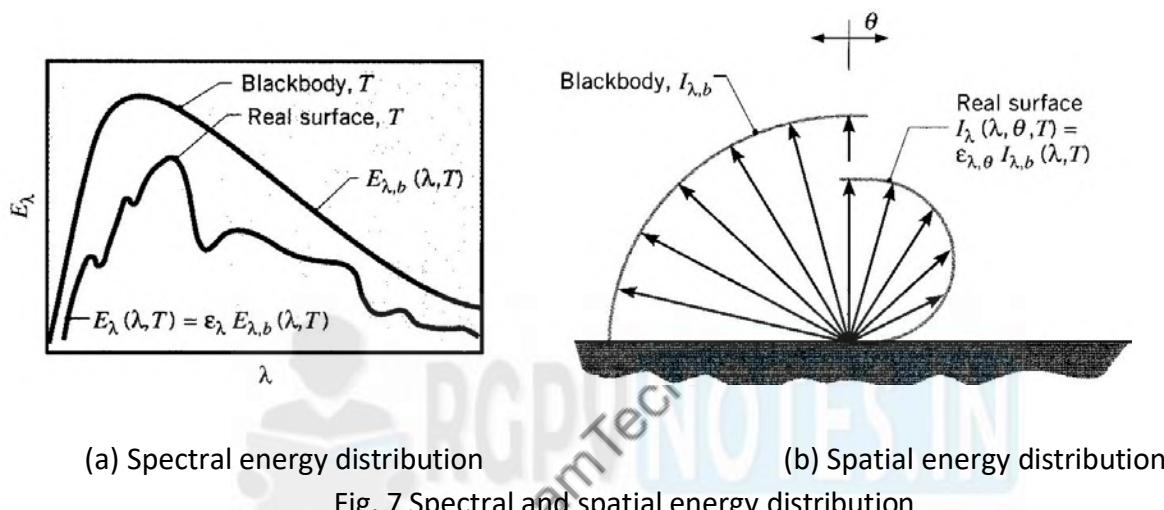


Fig. 7 Spectral and spatial energy distribution

- **Spatial (Directional) Energy Distribution:** A surface emits radiation in all directions. The intensity of radiation is different in a different direction. The distribution of radiation along the direction is called spatial distribution.

Wavelength Distribution of Black Body Radiation: Plank's Law

- The energy emitted by a black surface varies in accordance with wavelength, temperature, and surface characteristics of the body.
- Spectral blackbody emissive power (**monochromatic emissive power**)= “amount of radiation energy emitted by a blackbody at an absolute temperature T per unit time, per unit surface area, and per unit wavelength about the wavelength
- Plank suggested the following law for the spectral distribution of emissive power:
- The variation of the distribution of the monochromatic emissive power with wavelength is called spectral energy distribution, and this has been shown in figure 8
- The following important features can be noted from this plot:
 - The emitted radiation is a continuous function of wavelength. At any specified temperature, it increases with wavelength, reaches a peak, and then decreases with increasing wavelength.
 - At any wavelength, the amount of emitted radiation increases with increasing temperature.
 - As temperature increases, the pick of the curves shifts to the left to the shorter wavelength region. Consequently, a larger fraction of the radiation is emitted at shorter wavelengths at higher temperatures.

iv The radiation emitted by the sun, which is considered to be a blackbody at 5780 K (or roughly at 5800 K), reaches its peak in the visible region of the spectrum. Therefore, the sun is in tune with our eyes.

v On the other hand, surfaces at $T < 800$ K emit almost entirely in the infrared region and thus are not visible to the eye unless they reflect light coming from other sources.

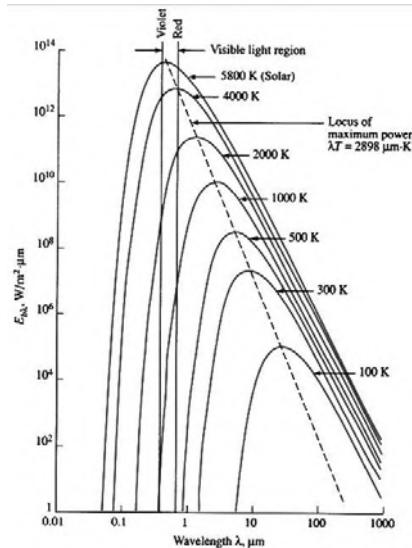


Fig. 8 Radiation of black body as a function of wavelength and temperature

Total Emissive Power: Stefan-Boltzmann law

- The total emissive power E of a surface is defined as the total radiant energy emitted by the surface in all directions over the entire wavelength per unit surface area per unit time.
- The basic rate equation for radiation transfer is based on Stefan-Boltzmann's law which states that the amount of radiant energy emitted per unit area of the black surface is proportional to the fourth power of its absolute temperature.
- The total emissive power of the black body can be obtained by integrating the monochromatic emissive power over the entire wavelength By simplifying the equation
- The Stefan-Boltzmann law helps us to determine the number of radiations emitted in all directions and over the entire wavelength spectrum from a simple knowledge of the temperature of the black body.
- Normally a body radiating heat is simultaneously receiving heat from other bodies as
- Radiation. Consider that surface 1 at temperature T_1 is completely enclosed by another black surface at temperature T_2 . The net radiation heat flux is then given by

Wien's Displacement law

- Figure 9 shows that as the temperature increases the peaks of the curve also increases and shift towards the shorter wavelength.
- The wavelength, at which the monochromatic emissive power is a maximum, is found by differentiating the Plank's Equation with respect to λ and equating to zero.
- **Wien's displacement law** may be stated as "The product of absolute temperature and the wavelength at which the emissive power is maximum, is constant"
- It can be easily found out that the wavelength corresponding to the peak of the plot
- It means that maximum spectral radiation intensity shifts towards the shorter wavelength with rising temperature.
- The peak of solar radiation, for example, occurs at which is near the middle of the visible range.



- The peak of the radiation emitted by a surface at room temperature ($T = 298\text{ K}$) occurs at 9.72 m , which is well into the infrared region of the spectrum.
- An electrical resistance heater starts radiating heat soon after it is plugged in, and we can feel the emitted radiation energy by holding our hands facing the heater. But this radiation is entirely in the infrared region and thus cannot be sensed by our eyes. The heater would appear dull red.
- When its temperature reaches about 1000 K since it will start emitting a detectable amount (about $1\text{ W/m}^2 \cdot \text{m}$) of visible red radiation at that temperature.
- As the temperature rises, even more, the heater appears bright red and is said to be red hot. When the temperature reaches about 1500 K , the heater emits enough radiation in the entire visible range of the spectrum to appear almost white to the eye, and it is called white-hot.
- Although it cannot be sensed directly by the human eye, infrared radiation can be detected by infrared cameras, which transmit the information to microprocessors to display visual images of objects at night.
- Rattlesnakes can sense the infrared radiation or the “body heat” coming off warm-blooded animals, and thus they can see at night without using any instruments.
- A surface that reflects all of the light appears white, while a surface that absorbs the entire light incident on it appears black. (Then how do we see a black surface?)
- It should be clear from this discussion that the colour of an object is not due to emission, which is primarily in the infrared region unless the surface temperature of the object exceeds about 1000 K .
- Instead, the colour of a surface depends on the absorption and reflection characteristics of the surface and is due to selective absorption and reflection of the incident visible radiation coming from a light source such as the sun or an incandescent light bulb.
- A piece of clothing containing a pigment that reflects red while absorbing the remaining parts of the incident light appears “red” to the eye (Fig. 9). Leaves appear “green” because their cells contain the pigment chlorophyll, which strongly reflects green while absorbing other colours.

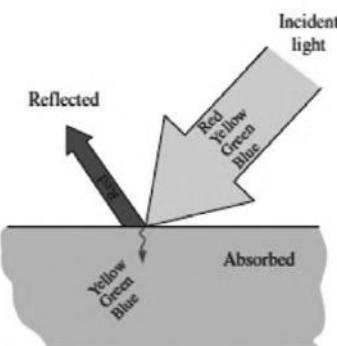


Fig. 9 Reflection of incident light from the surface

Relation between Emissivity and Absorptivity of the Body:

Kirchhoff's Law

- Consider two surfaces, one absolutely black at temperature and the other non-black at temperature T . The surfaces are arranged parallel to each other and so close that the radiation of one falls totally on the other.

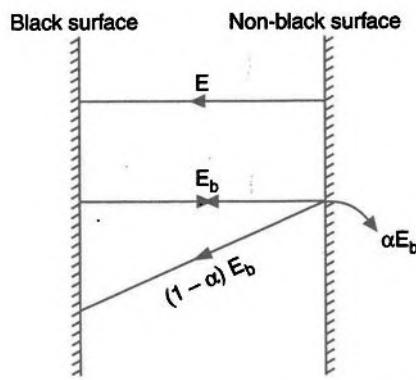


Fig. 10 Heat transfer between black and non-black surface

- The radiant energy E emitted by the non-black surface impinges on the black surface and gets fully absorbed. Likewise, the radiant energy emitted by the black surface strikes the non-black surface. If the non-black surface has absorptivity, it will absorb
 - “The ratio of the emissive power of a certain non-black body E to the emissive power of the black body, both bodies being at the same temperature, is called the **emissivity** of the body”.
 - Emissivity is used to find out the emissive power of the grey surface.

Kirchhoff's law can be stated as: “The emissivity and absorptivity of a real surface are equal for radiation with identical temperature and wavelength.” It means that the perfect absorber is also a perfect radiator

Plane and Solid Angle

- A plane angle is defined by a region by the rays of a circle and is measured as the ratio of the element of the arc of length l on the circle to the radius r of the circle.

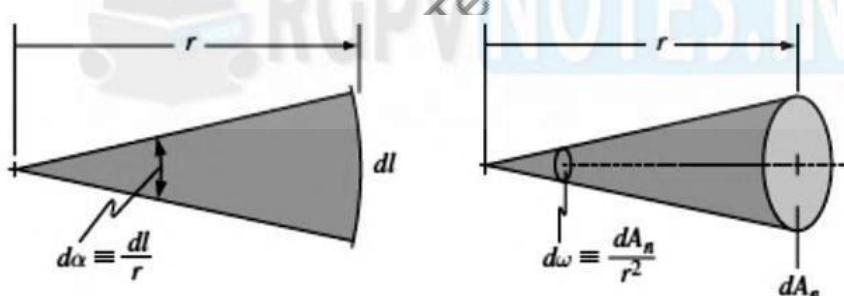


Fig. 11 Plane and solid angle

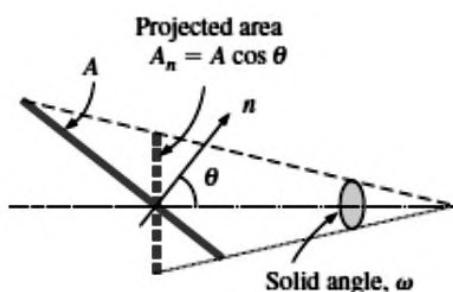


Fig. 12 Relationship between A and A_n

- Let us try to quantify the size of a slice of pizza. One way of doing that is to specify the arc length of the outer edge of the slice and to form the slice by connecting the endpoints of the arc to the centre.
- A more general approach is to specify the angle of the slice at the centre, as shown in Figure 13, this angle is called plain angle

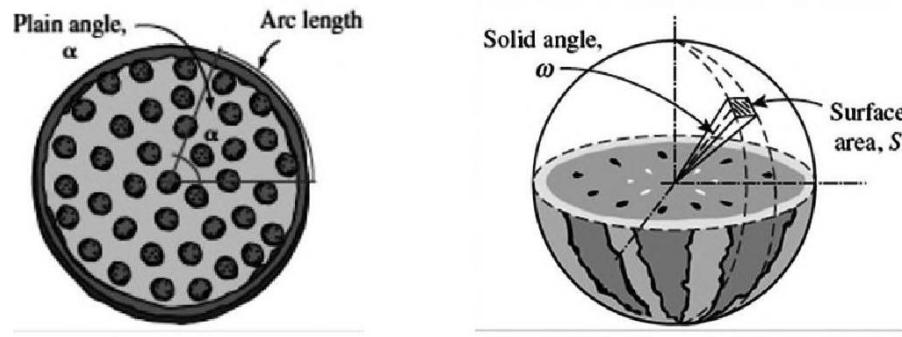


Fig. 13 (a) Slice of pizza of plain angle

(b) Slice of watermelon of solid angle

- Now consider a watermelon, and let us attempt to quantify the size of a slice. Again we can do it by specifying the outer surface area of the slice (the green part), or by working with angles for generality.
- Connecting all points at the edges of the slice to the centre in this case will form a three-dimensional body (like a cone whose tip is at the centre), and thus the angle at the centre in this case is properly called the **solid angle**.

The intensity of Radiation and Lambert's Cosine Law

- Intensity of radiation** is the energy emitted (of all wavelengths) in a particular direction per unit surface area and through a unit solid angle".

- The area is the projected area of the surface on a plane perpendicular to the direction of radiation.
- The intensity of radiation varies with the angle normal to the surface and is given by Lambert's cosine law.
- Lambert's cosine law** "the intensity of radiation in a direction from the normal to black emitter is proportional to the cosine of the angle".

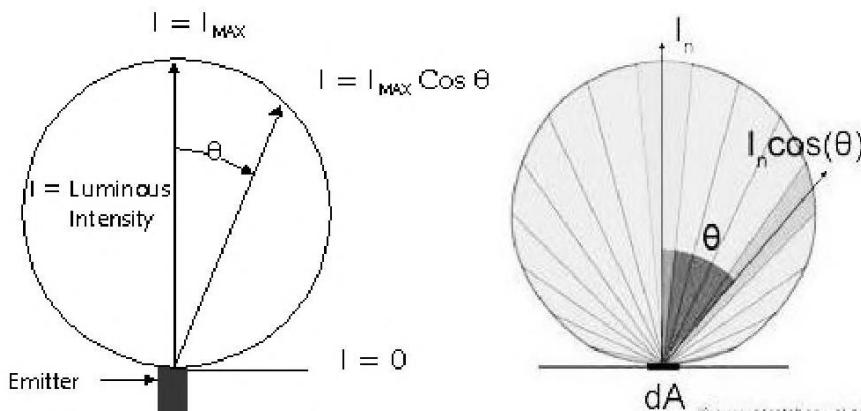


Fig. 14 Lambert cosine law

Fig. 15 Radiation emitted at an angle

Relation between the Normal Intensity and Emissive Power

- Consider the emission of radiation by a differential area element dA of a surface, as shown in Figure 16. Radiation is emitted in all directions into the hemispherical space.

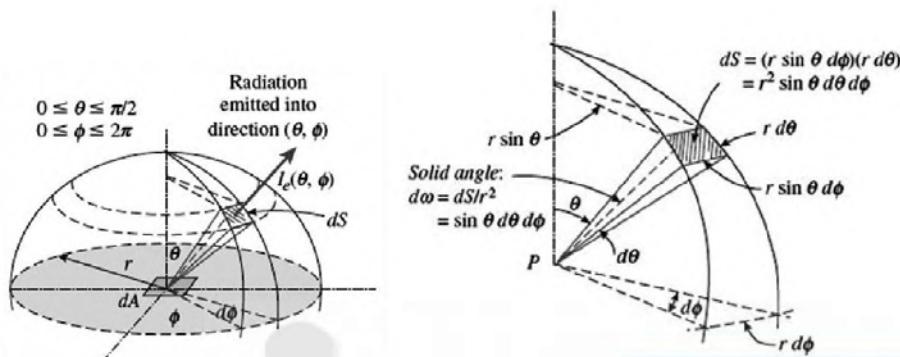


Fig. 16 Emission of radiation from differential element dA into a hemispherical shape

Introduction

- Till now we have discussed fundamental aspects of various definitions and laws. Now we will study the heat exchange between two or more surfaces which is of practical importance.
- The two surfaces which are not in direct contact exchange the heat due to radiation phenomena. The factors that determine the rate of heat exchange between two bodies are the temperature of the individual surfaces, their emissivities, as well as how well one surface can see the other surface. The last factor is known as the view factor, shape factor, angle factor, or configuration factor.

Heat Exchange between Two Black Surfaces: Shape Factor

- Consider heat exchange between the elementary area and two black radiating bodies having areas and respectively.
- The elementary areas are at a distance r apart and the normal's to the areas make angles and with the line joining them. The surface is at temperature and the surface is at temperature.

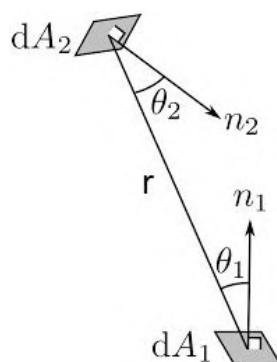


Fig. 17 Radiant heat exchange between two black surfaces

- The above result is known as a reciprocity theorem. It indicates that the net radiant interchange may be evaluated by computing one way configuration factor from either surface to the other.

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Shape factor algebra and salient features of the shape factor

- The salient features for complex geometries can be derived in terms of known shape factors for other geometries. For that, the complex shape is divided into sections for which the shape factors are either known or can be readily evaluated.
- The known configuration factor is worked out by adding and subtracting known factors of related geometries. The method is based on the definition of shape factor, the reciprocity principle, and the energy conservation law.
- The inter-relation between various shape factors is called factor algebra.

Salient features of shape factor:

- The value of the shape factor depends only on the geometry and orientation of surfaces with respect to each other. Once the shape factor between two surfaces is known, it can be used for calculating the radiant heat exchange between the surfaces at any temperature.
- This reciprocal relation is particularly useful when one of the shape factor is unity.
- All the radiation streaming out from an inner sphere (surface 1) is intercepted by the enclosing outer sphere (surface 2). As such the shape factor of the inner sphere (surface 1) with respect to the enclosure is unity and the shape factor of the outer sphere (surface 2) can be obtained by using reciprocal relation.

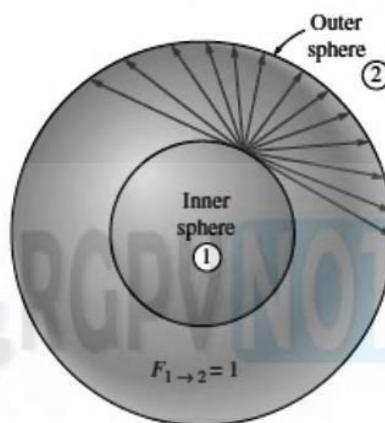
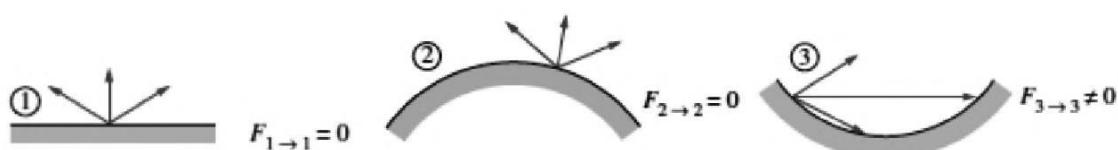


Fig. 18 Two concentric spheres

- The radiant energy emitted by one part of the concave surface is intercepted by another part of the same surface. Accordingly, a concave surface has a shape factor with respect to itself. The shape factor with respect to itself is denoted by



(a) Flat surface (b) Convex surface (c) Concave surface

Fig. 19 Shape factor of the surface with respect to itself

- For a flat or convex surface, the shape factor with respect to itself is zero.
- We have already discussed that the view factors are related to each other is given by
- This relation is known as the reciprocity relation or the reciprocity rule.

Summation Rule

- Any radiating surface will have a finite area and therefore will be enclosed by many surfaces.
- For radiation heat transfer analysis, the radiating surface is considered as a part of the enclosure.
- Even openings are treated as imaginary surfaces with radiation properties equivalent to those of the opening.

- The conservation of energy principle requires that the entire radiation leaving any surface i of an enclosure be intercepted by the surfaces of the enclosure.
- Therefore, the sum of the view factors from surface i of an enclosure to all surfaces of the enclosure, including to itself, must equal unity. This is known as the summation rule for an enclosure and is expressed as (Figure 6.4)



Fig. 20 Radiation leaving the surface i of an enclosure intercepted completely by the surface of the enclosure

The Superposition Rule

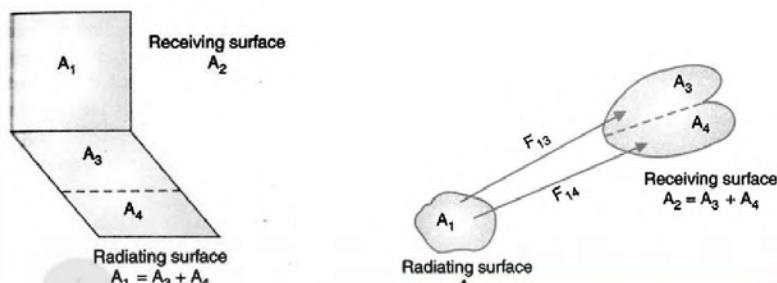


Fig. 21 Superposition rule

- Thus if the transmitting surface is subdivided, the shape factor for that surface with respect to the receiving surface is not equal to the sum of the individual shape factors.
- Apparently, the shape factor from a radiating surface to a subdivided receiving surface is simply the sum of the individual shape factors.

The Symmetry Rule

Identical surfaces that are oriented in an identical manner with respect to another surface will intercept identical amounts of radiation leaving that surface.

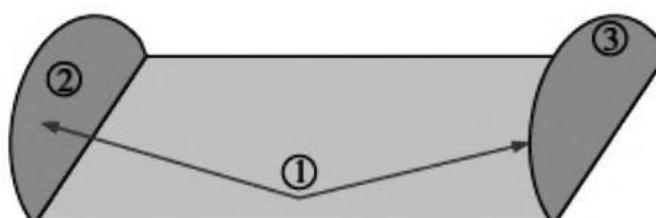


Fig. 22 Symmetry rule

So, the symmetry rule can be expressed as two or more surfaces that possess symmetry about a third surface will have identical view factors from that surface. From figure 6.6

Electrical Network Approach for Radiation Heat Exchange

- Solution of the radiation heat transfer problem can be obtained by reducing the actual system to an equivalent electrical network and then solving that network. To understand the concept, first, some terminology should be defined.
- Radiosity:** It indicates the total radiant energy leaving a surface per unit time per unit surface area. It is the sum of the radiation emitted from the surface and the reflected portion of any radiation incident upon it.
- Irradiation:** It indicates the total radiant energy incident upon a surface per unit time per unit area. Some of it may be reflected to become a part of the radiosity of the surface.

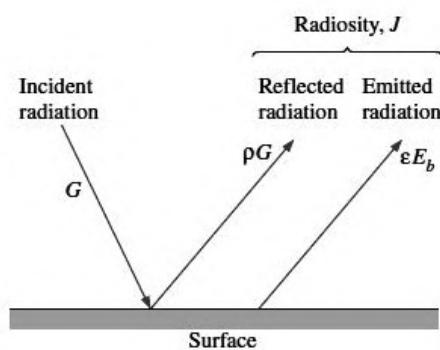


Fig. 23 Surface radiosity and irradiation

Radiation Heat Exchange between Non-Black Bodies

Small-Object in a Large Cavity

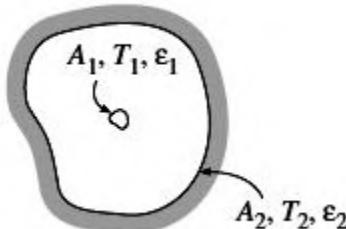


Fig. 24 Small object in a large cavity (enclosure)

Infinite Large Parallel Plates

- All the radiations emitted by plane one reach and are absorbed by other planes, and areas of the two planes are infinite. So,

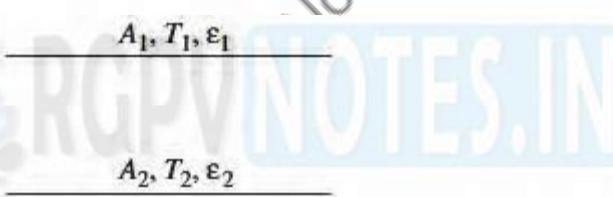
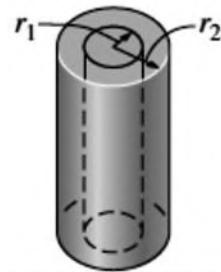
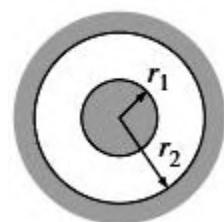


Fig. 25 Infinite large parallel plates

Infinite Long Concentric Cylinders or Sphere



(a) Concentric cylinder



(b) Concentric sphere

Fig. 26 Infinite long concentric cylinder and sphere

Radiation Shields

- Radiation heat transfer between two surfaces can be reduced greatly by inserting a thin, highly reflectivity (low-emissivity) sheet of material between the two surfaces. Such highly reflective thin plates or shells are called radiation shields.
- Consider two infinite parallel plates as shown in figure 27. The radiation network for the radiation heat transfer consists of two surface resistances and one space resistance as shown in figure 28.

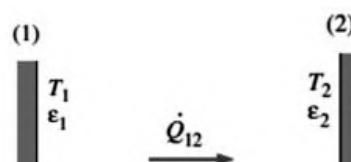


Fig. 27 Heat exchange between two infinite parallel planes without radiation shields

The radiation network of this geometry is constructed by drawing a surface resistance associated with each surface and connecting these surface resistances with space resistances, as shown in figure 28.

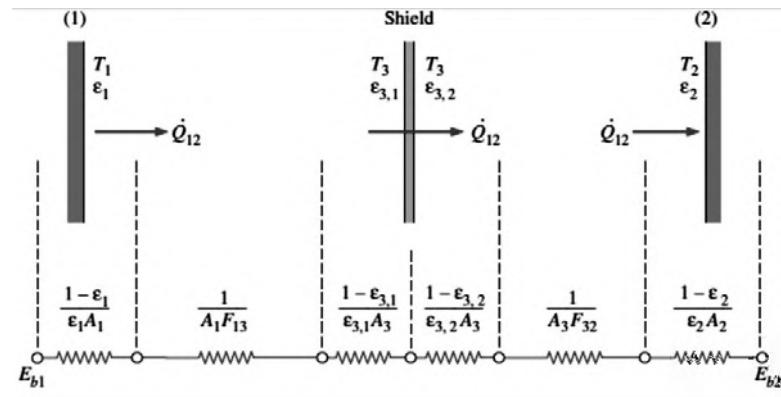


Fig. 28 Radiation heat exchange between two infinite parallel plates with radiation shield

Radiation Heat Transfer in Three-Surface Enclosure:

- Consider an enclosure consisting of three opaque, diffuse, and grey surfaces as shown in figure 29.

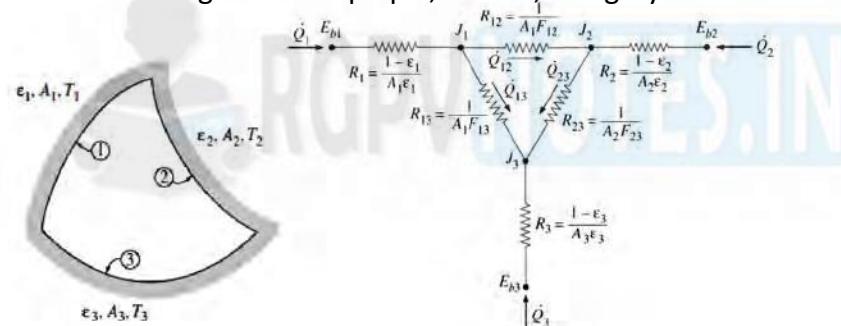


Fig.29 three opaque, diffuse, and grey surfaces

- The radiation network of this geometry is obtained by drawing a surface resistance associated with each of the three surfaces and connect these surface resistances with space resistances as shown in figure 29.
- The three equations for the determination of the radiosity, and are obtained from the requirement that the algebraic sum of the currents at each node must equal zero.

Boiling and Condensation

Introduction

When the temperature of a liquid at a specified pressure is raised to the saturation temperature (T_{sat}), at that pressure **Boiling** occurs.

Likewise, when the temperature of a vapour is lowered to saturation temperature (T_{sat}), **Condensation** occurs.

Boiling and Condensation are considered to be forms of convection heat transfer since they involve fluid motion, such as the rise of the bubbles to the top and the flow of condensate to the bottom.

Boiling and Condensation differ from other forms of convection, in that they depend on the latent heat of vaporization (h_{fg}) of the fluid and the surface tension (σ) at the liquid vapour interface, in addition to the properties of the fluid in each phase.

During a phase change, large amount of heat (due to large latent heat of vaporization released or absorbed) can be transferred essentially at constant temperature.

The phenomenon's are quite difficult to describe due to change in fluid properties(density, specific heat, thermal conductivity, viscosity, etc.) and due to considerations of surface tension, latent heat of vaporization, surface characteristics and other features of two phase flow.

Heat transfer co-efficient h associated with boiling and condensation are typically much higher than those encountered in other forms of convection processes that involve a single phase.

Boiling

Boiling is the convective heat transfer process that involves a phase change from liquid to vapor state.

Boiling is a liquid to vapour phase change process just like evaporation, but there are significant differences between the two. **Evaporation** occurs at the liquid–vapour interface when the vapor pressure is less than the saturation pressure of the liquid at a given temperature. Examples of evaporation are: drying of clothes, the evaporation of sweat to cool human body and the rejection of waste heat in wet cooling towers. Note that evaporation involves no bubble formation or bubble motion.

Boiling, on the other hand, occurs at the solid–liquid interface when a liquid is brought into contact with a surface maintained at a temperature T_s sufficiently above the saturation temperature T_{sat} of the liquid. At 1 atm, for example, liquid water in contact with a solid surface at 110°C will boil since the saturation temperature of water at 1 atm is 100°C.

The boiling process is characterized by the rapid formation of vapor bubbles at the solid–liquid interface that detach from the surface when they reach a certain size and attempt to rise to the free surface of the liquid.

Applications of Boiling

Steam production.

Absorption of heat in refrigeration and Air-conditioning systems.

Greater importance has recently been given to the boiling heat transfer because of developments of nuclear reactors, space-crafts and rockets, where large quantities of heat are produced in a limited space and are to be dissipated at very high rates.

Types of Boiling

A. Classification of boiling on the basis of the presence of bulk fluid motion



1. Pool Boiling

The liquid above the hot surface is stationary.

The only motion near the surface is because of free convection and the motion of the bubbles under the influence of buoyancy.

The pool boiling occurs in steam boilers. Pool boiling of a fluid can also be achieved by placing a heating coil in the fluid.

2. Forced Convection Boiling / Flow Boiling

The fluid motion is induced by external means such as a pump.

The liquid is pumped and forced to move in a heated pipe or over a surface in a controlled manner.

The free convection and the bubble-induced mixing also contribute towards the fluid motion.

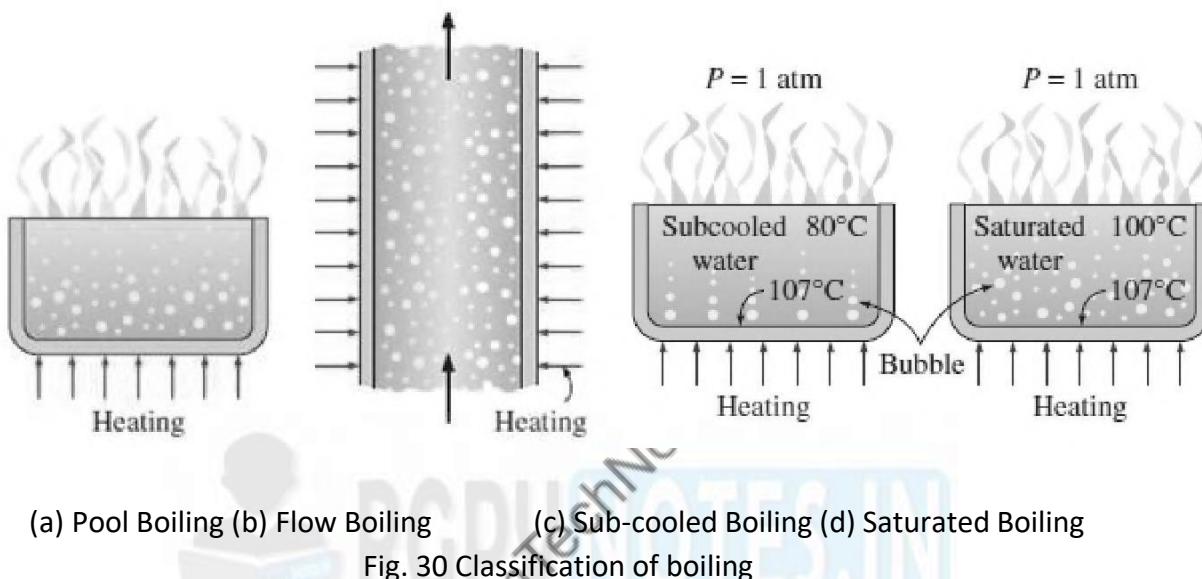


Fig. 30 Classification of boiling

B. Classification of boiling on the basis of the presence of bulk liquid temperature

1. Sub-cooled or Local Boiling

The temperature of the liquid is below the saturation temperature and boiling takes place only in the vicinity of the heated surface fig.30.

The vapor bubbles travel a short path and then vanish; apparently, they condense in the bulk of the liquid which is at a temperature less than a boiling point or saturation temperature.

2. Saturated Boiling

The temperature of the liquid exceeds the saturation temperature.

The vapor bubbles generated at the solid surface (solid-liquid interface) are transported through the liquid by buoyancy effects and eventually escape from the surface (liquid-vapor interface).

The actual evaporation process then sets in.

Boiling Regimes

Whether the boiling phenomenon corresponds to pool boiling or forced circulation boiling, there are some definite regimes of boiling associated with progressively increasing heat flux.

Nukiyama (1934) was the first to identify different regimes of pool boiling using the apparatus of Fig. 31. These different regimes can be illustrated by considering an electrically heated horizontal nichrome/Platinum wire submerged in a pool of liquid at saturation temperature.

Fig. 31 shows the relationship between heat flux and the temperature excess ($T_s - T_{sat}$); Where,

**Different boiling regimes are:**

- A. Natural Convection Boiling
- B. Nucleate Boiling
- C. Film Boiling

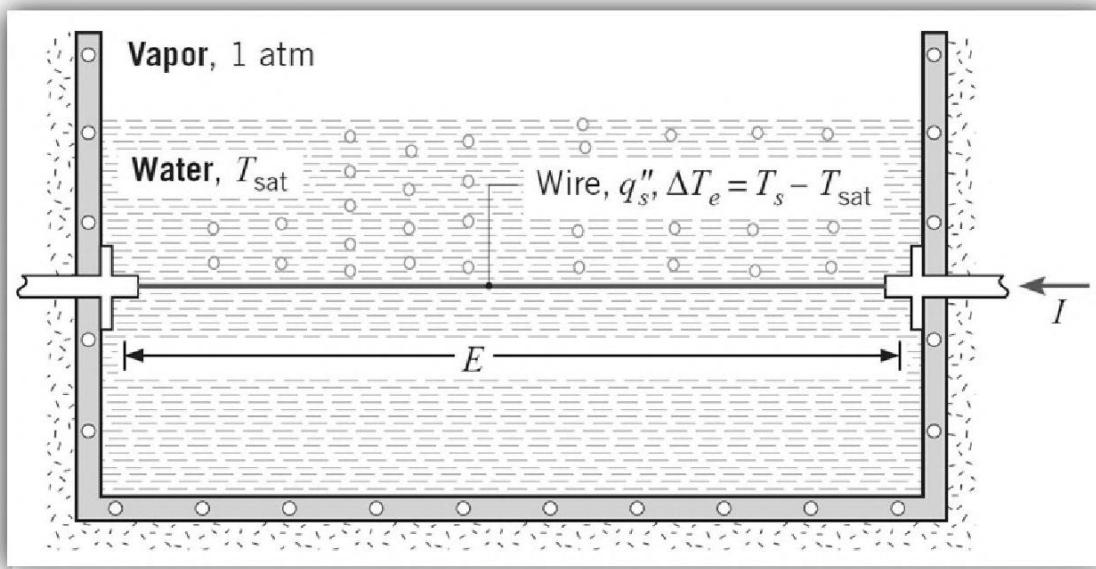


Fig. 31 Nukiyama's power controlled heating apparatus for demonstrating the boiling curve

A. Natural / Free Convection Boiling (up to point A on boiling curve)

The boiling takes place in a thin layer of liquid which adjoins the heated surface.

The liquid in the immediate vicinity of the wall becomes superheated, i.e. temperature of the liquid exceeds the saturation temperature at the given pressure.

The superheated liquid rises to the liquid-vapor interface where evaporation takes place.

The fluid motion is by free convection effects.

The heat transfer rate increases, but gradually, with growth in a temperature excess.

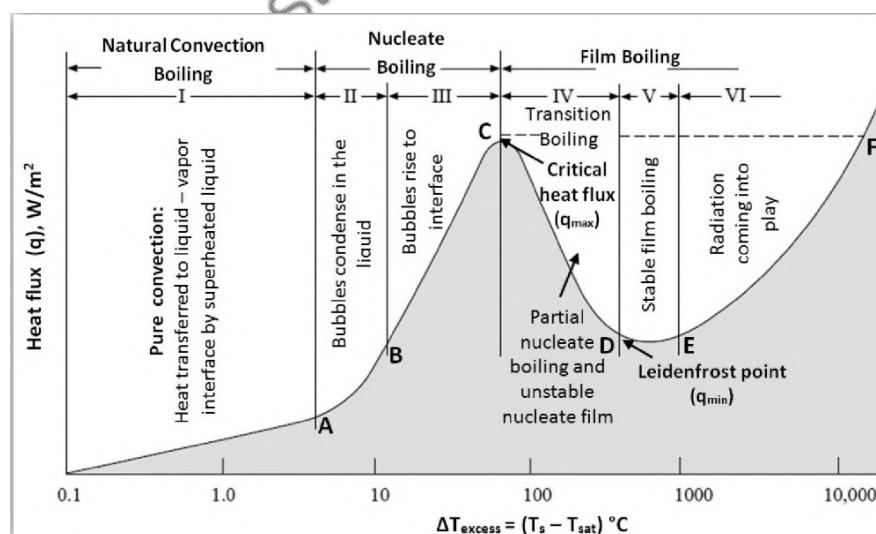


Fig. 32 boiling curve for saturated water at atmospheric pressure

B. Nucleate Boiling (between point A & C on boiling curve fig.32)

When the liquid is overheated in relation to saturation temperature, vapor bubbles are formed at certain favourable spots called the **Nucleation or Activesites**. Point A is referred as the **onset of nucleate boiling, ONB**.

The nucleate boiling regimes can be separated into two distinct regions:

A – B:-

Isolated bubbles are formed at various nucleation sites, on the heated surface but these bubbles get condensed in the liquid after detaching from the surface.

B – C:-

Heater temperature is further increased. Bubbles form at very high rates and they form continuous columns of vapor in the liquid.

The liquid is quite hot and the bubbles do not condense in it.

These bubbles rise to the free surface, where they break up and release their vapor content and that helps in rapid evaporation.

The space vacated by the rising bubbles is filled by the liquid in the vicinity of the heated surface, and the process is repeated.

The agitation or stirring caused by the entrainment of the liquid to the heated surface and rapid evaporation is responsible for the increased heat transfer coefficient and heat flux in the nucleate boiling region.

The heat flux hence reaches a maximum at point C, which is called the **critical /maximum heat flux, q_{max}** .

Nucleate boiling is the most desirable boiling regime in practice because high heat transfer rates can be achieved in this regime with relatively small values of ΔT_{excess} .

C. Film Boiling (beyond point C on Boiling curve) Transition Boiling (between point C & D)

As the heater temperature and thus ΔT_{excess} is increased past point C, the heat flux decreases as shown in Fig. 3.

This is because a bubble formation is very rapid; the bubbles blanket the heating surface and prevent the incoming fresh liquid from taking its place.

A large fraction of the heating surface is covered by a vapor film, which acts as insulation due to the low thermal conductivity of the vapor.

In the transition boiling regime, both nucleate and film boiling partially occurs.

Nucleate boiling at point C is completely replaced by film boiling at point D.

Operation in the transition boiling regime, which is also called the unstable film boiling regime, is avoided in practice.

Beyond point D

In this region, the heated surface is completely covered by a continuous stable vapor film.

The temperature differences are so large that radiant heat flux becomes significant, and the heat flux curve begins to rise upward with increasing ΔT_{excess} . That marks the region of stable film boiling.

The phenomenon of stable film boiling is referred to as the "**Leidenfrost effect**" and point D, where the heat flux reaches a minimum, is called the Leidenfrost point.

Burn out point (Point F)

In order to move beyond point C, where q_{\max} occurs, we must increase the heated surface temperature (T_s).

To increase T_s , however, we must increase the heat flux. But the fluid cannot receive this increased energy beyond point C, and the heated surface temperature (T_s) rises even further.

If the surface temperature exceeds the temperature limit of the wall material, burn out (structural damage & failure) of the wall occurs.

Bubble Growth

The bubble formation in nucleate boiling is greatly influenced by the nature and condition of the heating surface and surface tension at the solid-liquid interface (Shape, size, and inclination of bubbles, however, do not have much effect on the heat transfer rate).

The surface tension signifies the wetting capability of the surface with the liquid (i.e. low surface tension → Highly wetted surface) and that influences the angle of contact between the bubble and solid surface. Any contamination of the surface would affect its wetting characteristics and influence the size and shape of the vapor bubbles.

If the surface tension of the liquid is low, it tends to wet the surface (fully wetted surface), so that the bubble is readily pushed by the liquid and rises. The vapor bubbles tend to become globular or oval in shape as shown in Fig. 33(a) (iii) and they are disengaged from the surface.

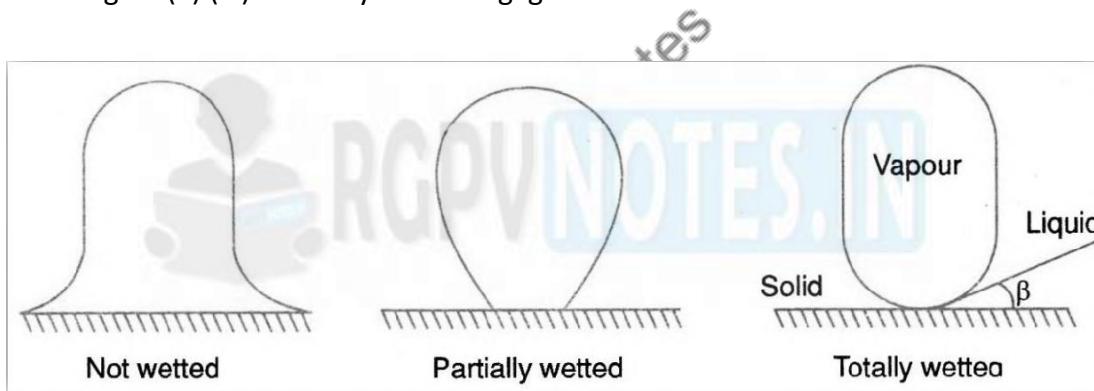


Fig. 33(a) Wetting characteristics for typical vapor bubbles

In the case of liquid having intermediate surface tension (partially wetted surface) a momentary balance may exist between the bubbles and solid surface so that it is necessary to form larger bubbles before the buoyant force can free them from the surface; the shape of the bubble is shown in Fig. 33(a) (ii).

On the unwetted surface, the bubbles spread out as shown in Fig. 33(a) (i); forming a wedge between the water and heating surface, thereby allowing hydrostatic forces to resist the action of buoyancy.

The formation of the bubble with the fully wetted surface as shown in Fig. 33(a) (iii) gives a high heat transfer rate compared with the bubble shapes shown in Fig. 33(a) (i) and (ii); because the area covered by the insulating vapor film is the smallest.

Experimental evidence does indicate that the vapor bubbles are not always in thermodynamic equilibrium with the surrounding liquid.

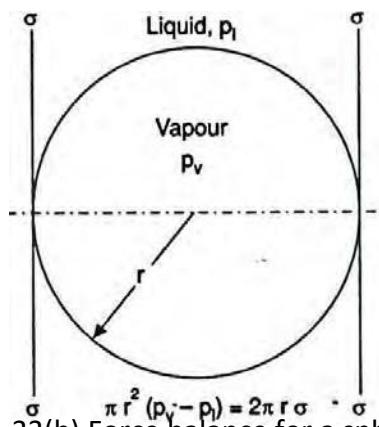


Fig. 33(b) Force balance for a spherical bubble

The vapor inside the bubble is not necessarily at the same temperature as the liquid and the vapor pressure P_v inside the bubble exceeds the liquid pressure P_l acting from outside of the bubble. Fig. 33(b) indicates one such spherical bubble with various forces acting on it.

i. The resultant pressure ($P_v - P_l$) acts on area πr^2 and the pressure force equals

$$\pi r^2 (P_v - P_l)$$

ii. The surface tension σ of the vapor-liquid interface acts on the interface length $2\pi r$ and the surface tension force equals $2\pi r \sigma$.

Factors affecting the nucleate pool boiling

1) Material, shape, and condition of the surface:

Under identical conditions of pressure and temperature difference, the boiling heat transfer coefficient is different for different metals; copper has a high value compared to steel. Further, a rough surface gives a better heat transmission than when the surface is either smooth or has been coated to weaken its tendency to get wetted.

2) Pressure:

The temperature difference between the heating surface and the bulk and hence the rate of bubble growth is affected by pressure. The maximum allowable heat flux for a boiling liquid increases with pressure until critical pressure is reached and thereafter it declines.

3) Liquid properties:

Experiments have shown that the bubble size increases with the dynamic viscosity of the liquid. With the increase in bubble size, the frequency of bubble formation decreases, and that results in a reduced rate of heat transfer.

Condensation

"Condensation occurs when the temperature of a vapor is reduced below its saturation temperature corresponding to the vapor pressure."

This is usually done by bringing the vapor into contact with a solid surface whose temperature, T_s is below the saturation temperature T_{sat} of the vapor.

The latent energy of the vapor is released, heat is transferred to the surface, and the condensate is formed.

The condensation can also occur on the free surface of a liquid or even in gas when the temperature of the liquid or the gas to which the vapor is exposed is below T_{sat} .

In this chapter, we will consider surface condensation only.

Depending upon the behavior of condensate upon the cooled surface, the condensation process has been categorized into two distinct modes: (A) Film-wise condensation and (B) Dropwise condensation fig.34.

Dropwise and Film-wise Condensation

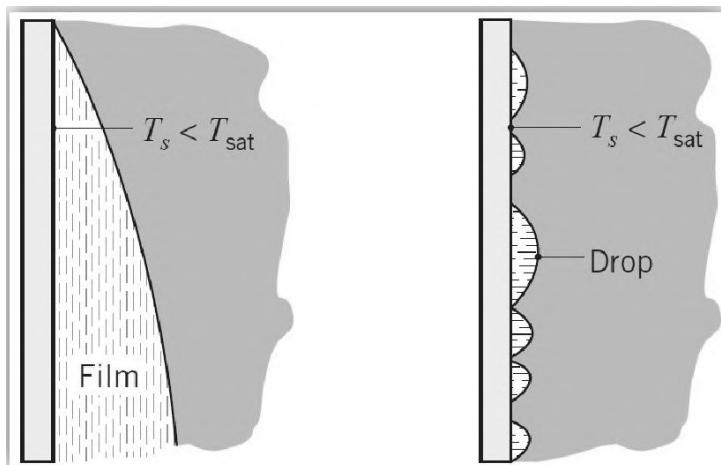


Fig. 34 Film-wise and Dropwise Condensation

A. Film-wise condensation

The liquid condensate wets the solid surface, spread out, and forms a continuous film over the entire surface.

The liquid flows down the cooling surface under the action of gravity and the layer continuously grows in thickness because of newly condensing vapor.

The continuous film offers resistance and restricts the further transfer of heat between the vapor and the surface.

Film condensation only occurs when a vapor relatively free from impurities, is allowed to condense on a clean surface.

Film condensation is generally a characteristic of clean, uncontaminated surfaces.

B. Dropwise condensation

The liquid condensate collects in droplets and does not wet the solid cooling surface.

The droplets develop in cracks, pits, and cavities on the surface, grow in size, break away from the surface, knock off other droplets, and eventually run off the surface without forming a film.

A part of the condensation surface is directly exposed to the vapor without an insulating film of condensate liquid.

Evidently, there is no film barrier to heat flow and higher heat transfer rates are experienced.

Dropwise condensation has been observed to occur either on highly polished surfaces or on surfaces contaminated with impurities like fatty acids and organic compounds.

Dropwise condensation gives a coefficient of heat transfer generally 5 to 10 times larger than with film condensation.

It is therefore common practice to use surface coatings that inhibit wetting and hence simulate drop-wise condensation.

Silicon, Teflon, and an assortment of waxes and fatty acids are often used for this purpose.

However such coatings gradually lose their effectiveness due to oxidation, fouling, or outright removal and film condensation eventually occur.

Although it is desirable to achieve drop-wise condensation in industrial applications, it is often difficult to maintain this condition.

Condenser design calculations are often based on the assumption of film condensation.

Influence of the presence of non-condensable gases

The presence of non-condensable gas such as air in a condensing vapor produces a detrimental (negative) effect on the heat transfer coefficient.

It has been observed that even with a few percent by volume of air in steam the condensation heat transfer coefficient is reduced by more than 50%.

This is owing to the fact that when a vapor (containing non-condensable gas) condenses, the non-condensable gas is left at the surface.

Any further condensation at the surface will occur only after the incoming vapor has diffused through this non-condensable gas collected in the vicinity of the surface.

The non-condensable gas adjacent to the surface acts as a thermal resistance to the condensation process. The rate of condensation decreases greatly when the condensable vapor is contaminated with even very small amounts of non-condensable gases.

As the presence of non-condensable gas in a condensing vapor is undesirable, the general practice in the design of a condenser should be to vent the non-condensable gas to the maximum extent possible.

