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SCHOOL OF MECHANICAL

DEPARTMENT OF MECHANICAL

UNIT – I – Heat Transfer Applied to IC Engines – SMEA1504

UNIT – I

BASIC CONCEPTS IN HEAT TRANSFER

1.1 Heat Energy and Heat Transfer

Heat is a form of energy in transition and it flows from one system to another, without transfer of mass, whenever there is a temperature difference between the systems. The process of heat transfer means the exchange in internal energy between the systems and in almost every phase of scientific and engineering work processes, we encounter the flow of heat energy.

1.2 Importance of Heat Transfer

Heat transfer processes involve the transfer and conversion of energy and therefore, it is essential to determine the specified rate of heat transfer at a specified temperature difference. The design of equipments like boilers, refrigerators and other heat exchangers require a detailed analysis of transferring a given amount of heat energy within a specified time. Components like gas/steam turbine blades, combustion chamber walls, electrical machines, electronic gadgets, transformers, bearings, etc require continuous removal of heat energy at a rapid rate in order to avoid their overheating. Thus, a thorough understanding of the physical mechanism of heat flow and the governing laws of heat transfer are a must.

1.3 Modes of Heat Transfer

The heat transfer processes have been categorized into three basic modes: Conduction, Convection and Radiation.

Conduction – It is the energy transfer from the more energetic to the less energetic particles of a substance due to interaction between them, a microscopic activity.

Convection - It is the energy transfer due to random molecular motion a long with the macroscopic motion of the fluid particles.

Radiation - It is the energy emitted by matter which is at finite temperature. All forms of matter emit radiation attributed to changes m the electron configuration of the constituent atoms or molecules The transfer of energy by conduction and convection requires the presence of a material medium whereas radiation does not. In fact radiation transfer is most efficient in vacuum.

All practical problems of importance encountered in our daily life Involve at least two, and sometimes all the three modes occuring simultaneously When the rate of heat flow is constant, i.e., does not vary with time, the process is called a steady state heat transfer process. When the temperature at any point in a system changes with time, the process is called unsteady or transient process. The internal energy of the system changes in such a process when the temperature variation of an unsteady process describes a particular cycle (heating or cooling of a budding wall during a 24 hour cycle), the process is called a periodic or quasi-steady heat transfer process.

Heat transfer may take place when there is a difference In the concentration of the mixture components (the diffusion thermoeffect). Many heat transfer processes are accompanied by a transfer of mass on a macroscopic scale. We know that when water evaporates, the heal transfer is accompanied by the transport of the vapour formed through an air-vapour mixture. The transport of heat energy to steam generally occurs both through molecular interaction and convection. The combined molecular and convective transport of mass is called convection mass transfer and with this mass transfer, the process of heat transfer becomes more complicated.

1.4 Thermodynamics and Heat Transfer-Basic Difference

Thermodynamics is mainly concerned with the conversion of heat energy into other useful forms of energy and IS based on (i) the concept of thermal equilibrium (Zeroth Law), (ii) the First Law (the principle of conservation of energy) and (iii) the Second Law (the direction in which a particular process can take place). Thermodynamics is silent about the heat energy exchange mechanism. The transfer of heat energy between systems can only take place whenever there is a temperature gradient and thus. Heat transfer is basically a non-equilibrium phenomenon. The Science of heat transfer tells us the rate at which the heat energy can be transferred when there IS a thermal non-equilibrium. That IS, the science of heat transfer seeks to do what thermodynamics is inherently unable to do.

However, the subjects of heat transfer and thermodynamics are highly complimentary. Many heat transfer problems can be solved by applying the principles of conservation of energy

(the First Law)

1.5 Dimension and Unit

Dimensions and units are essential tools of engineering. Dimension is a set of basic entities expressing the magnitude of our observations of certain quantities. The state of a system is identified by its observable properties, such as mass, density, temperature, etc. Further, the motion of an object will be affected by the observable properties of that medium in which the object is moving. Thus a number of observable properties are to be measured to identify the state of the system.

A unit is a definite standard by which a dimension can be described. The difference between a dimension and the unit is that a dimension is a measurable property of the system and the unit is the standard element in terms of which a dimension can be explicitly described with specific numerical values.

Every major country of the world has decided to use SI units. In the study of heat transfer the dimensions are: L for length, M for mass, e for temperature, T for time and the corresponding units are: metre for length, kilogram for mass, degree Celsius (^oC) or Kelvin (K) for temperature and second (s) for time. The parameters important In the study of heat transfer are tabulated in Table 1.1 with their basic dimensions and units of measurement.

Table 1.1 Dimensions and units of various parameters

Parameter	Dimension	Unit
Mass	M	Kilogram, kg
Length	L	metre, m
Time	T	seconds, s
Temperature	□	Kelvin, K, Celcius ^oC
Velocity	L/T	metre/second,m/s
Density	ML^{-3}	kg/m^3
Force	$ML^{-1}T^{-2}$	Newton, N = 1 kg m/s^2
Pressure	ML^2T^{-2}	N/m^2, Pascal, Pa
Energy, Work	ML^2T^{-3}	N-m, = Joule, J
Power	ML^2T^{-3}	J/s, Watt, W
Absolute Viscosity	$ML^{-1}T^{-1}$	N-s/m^2, Pa-s
Kinematic Viscosity	L^2T^{-1}	m^2/s

Thermal Conductivity	$MLT^{-3}\theta^{-1}$	W/mK, W/m°C
Heat Transfer Coefficient	$MT^{-3}\theta^{-1}$	W/m^2K, $W/m^{2o}C$
Specific Heat	$L^2T^{-2}\theta^{-1}$	J/kg K, J/kg°C
Heat Flux	MT^{-3}	W/m^2

1.6 Mechanism of Heat Transfer by Conduction

The transfer of heat energy by conduction takes place within the boundaries of a system, or a cross the boundary of t he system into another system placed in direct physical contact with the first, without any appreciable displacement of matter comprising the system, or by the exchange of kinetic energy of motion of the molecules by direct communication, or by drift of electrons in the case of heat conduction in metals. The rate equation which describes this mechanism is given by Fourier Law

$$\dot{Q} = -kAdT/dx$$

where $\dot{Q}$ = rate of heat flow in X-direction by conduction in J/S or W,

k = thermal conductivity of the material. It quantitatively measures the heat conducting ability and is a physical property of t he material that depends upon the composition of the material, W/mK,

A = cross-sectional area normal to the direction of heat flow, m^2,

dT/dx = temperature gradient at the section, as shown in Fig. 1 I The neganve sign IS Included to make the heat transfer rate Q positive in the direction of heat flow (heat flows in the direction of decreasing temperature gradient).

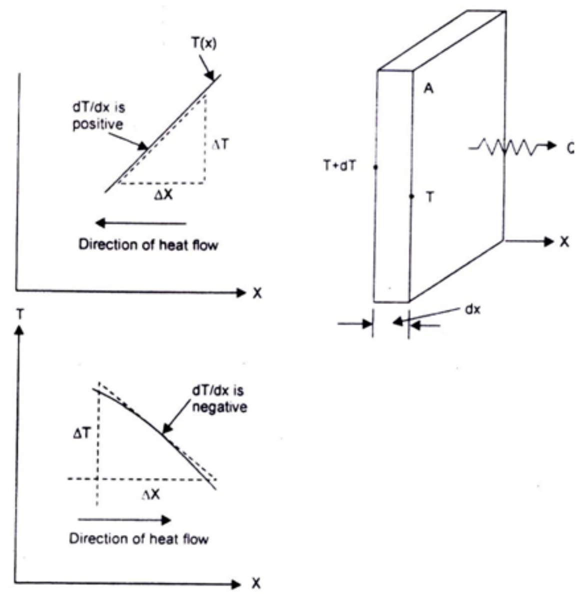


Fig 1.1: Heat flow by conduction

1.7 Thermal Conductivity of Materials

Thermal conductivity is a physical property of a substance and In general, It depends upon the temperature, pressure and nature of the substance. Thermal conductivity of materials are usually determined experimentally and a number of methods for this purpose are well known.

Thermal Conductivity of Gases: According to the kinetic theory of gases, the heat transfer by conduction in gases at ordinary pressures and temperatures take place through the transport of the kinetic energy arising from the collision of the gas molecules. Thermal conductivity of gases depends on pressure when very low «2660 Pal or very high ($> 2 \times 10^9$ Pa). Since the specific heat of gases Increases with temperature, the thermal conductivity Increases with temperature and with decreasing molecular weight.

Thermal Conductivity of Liquids: The molecules of a liquid are more closely spaced and molecular force fields exert a strong influence on the energy exchange In the collision process. The mechanism of heat propagation in liquids can be conceived as transport of energy by way of unstable elastic oscillations. Since the density of liquids decreases with increasing temperature, the thermal conductivity of non-metallic liquids generally decreases with increasing temperature, except for liquids like water and alcohol because their thermal conductivity first Increases with increasing temperature and then decreases.

Thermal Conductivity of Solids (i) Metals and Alloys: The heat transfer in metals arise due to a drift of free electrons (electron gas). This motion of electrons brings about the equalization in temperature at all points of t he metals. Since electrons carry both heat and electrical energy. The thermal conductivity of metals is proportional to its electrical conductivity and both the thermal and electrical conductivity decrease with increasing temperature. In contrast to pure metals, the thermal conductivity of alloys increases with increasing temperature. Heat transfer In metals is also possible through vibration of lattice structure or by elastic sound waves but this mode of heat transfer mechanism is insignificant in comparison with the transport of energy by electron gas. (ii) Nonmetals: Materials having a high volumetric density have a high thermal conductivity but that will depend upon the structure of the material, its porosity and moisture content High volumetric density means less amount of air filling the pores of the materials. The thermal conductivity of damp materials considerably higher than the thermal conductivity of dry material because water has a higher thermal conductivity than air. The thermal conductivity of granular material increases with temperature. (Table 1.2 gives the thermal conductivities of various materials at 0^{o}C.)

Table 1.2 Thermal conductivity of various materials at 0^{o}C.

Material	Thermal conductivity (W/m K)	Material	Thermal conductivity (W/m K)
Gases		Solids: Metals	
Hydrogen .	0175	Sliver, pure	410
Helium	0141	Copper, pure	385
A"	0024	AlumllllUm, pure	202
Water vapour (saturated)	00206	Nickel, pure	93
Carbon dioxide	00146	Iron, pure	73
(thermal conductivity of helium and hydrogen are much higher than other gases. because then molecules have small mass and higher mean travel velocity)		Carbon steel, I %C Lead, pure Chrome-nickel-steel (18% Cr, 8% Ni) Non-metals	43 35 16.3
Liquids		Quartz, parallel to axis	41.6
Mercury	821	Magnesite	4.15

Water*	0.556	Marble	2.08 to 2.94
Ammonia	0.54	Sandstone	1.83
Lubricating 011		Glass, window	0.78
SAE 40	0.147	Maple or Oak	0.17
Freon 12	0.073	Saw dust	0.059
		Glass wool	0.038

* water has its maximum thermal conductivity (k = 068 W/mK) at about 150°C

2. STEADY STATE CONDUCTION ONE DIMENSION

2.1 The General Heat Conduction Equation for an Isotropic Solid with Constant Thermal Conductivity

Any physical phenomenon is generally accompanied by a change in space and time of its physical properties. The heat transfer by conduction in solids can only take place when there is a variation of temperature, in both space and time. Let us consider a small volume of a solid element as shown in Fig. 1.2. The dimensions are: Δx, Δy, Δz along the X-, Y-, and Z-coordinates.

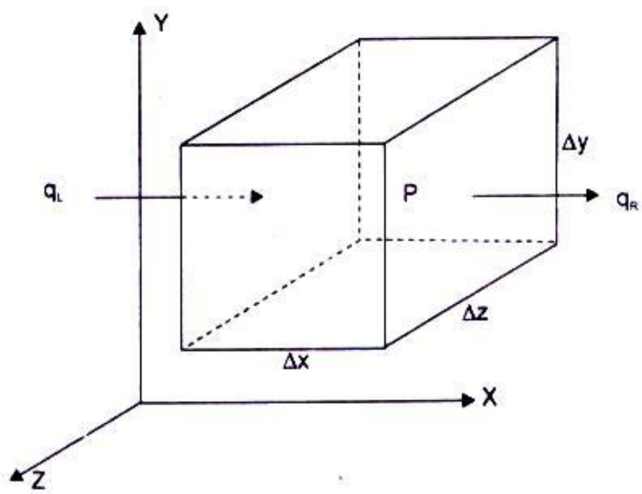


Fig 1.2 Elemental volume in Cartesian coordinates

First we consider heat conduction the X-direction. Let T denote the temperature at the point P (x, y, z) located at the geometric centre of the element. The temperature gradient at the left hand face (x - ~x12) and at the right hand face (x + Δx/2), using the Taylor's series, can be written as:

$\partial T/\partial x|_L = \partial T/\partial x - \partial^2 T/\partial x^2 .\Delta x/2$ + higher order terms.

$\partial T/\partial x|_R = \partial T/\partial x + \partial^2 T/\partial x^2 .\Delta x/2 +$ higher order terms.

The net rate at which heat is conducted out of the element 10 X-direction assuming k as constant and neglecting the higher order terms,

we get $-k\Delta y\Delta z\left[\frac{\partial T}{\partial x} + \frac{\partial^2 T}{\partial x^2}\frac{\Delta x}{2} - \frac{\partial T}{\partial x} + \frac{\partial^2 T}{\partial x^2}\frac{\Delta x}{2}\right] = -k\Delta y\Delta z\Delta x\left(\frac{\partial^2 T}{\partial x^2}\right)$

Similarly for Y- and Z-direction,

We have $-k\Delta x\Delta y\Delta z\ \partial^2 T/\Delta y^2$ and $-k\Delta x\Delta y\Delta z\ \partial^2 T/\Delta z^2$.

If there is heat generation within the element as Q, per unit volume and the internal energy of the element changes with time, by making an energy balance, we write

Heat generated within the element	Heat conducted away from the element	Rate of change of internal energy within with the element

or, $\dot{Q}_v(\Delta x\Delta y\Delta z) + k(\Delta x\Delta y\Delta z)\left(\partial^2 T/\partial x^2 + \partial^2 T/\partial y^2 + \partial^2 T/\partial z^2\right)$

$= \rho c(\Delta x\Delta y\Delta z)\ \partial T/\partial t$

Upon simplification, $\partial^2 T/\partial x^2 + \partial^2 T/\partial y^2 + \partial^2 T/\partial z^2 + \dot{Q}_v/k = \frac{\rho c}{k}\partial T/\partial t$

or, $\nabla^2 T + \dot{Q}_v/k = 1/\alpha(\partial T/\partial t)$

where $\alpha = k/\rho . c$, is called the thermal diffusivity and is seen to be a physical property of the material of which the solid is composed.

The Eq. (2.la) is the general heat conduction equation for an isotropic solid with a constant thermal conductivity. The equation in cylindrical (radius r, axis Z and longitude □) coordinates is written as: Fig. 2.I(b),

$$\partial^2 T/\partial r^2 + (1/r)\partial T/\partial r + \left(1/r^2\right)\partial^2 T/\partial\theta^2 + \partial^2 T/\partial z^2 + \dot{Q}_v/k = 1/\alpha\,\partial T/\partial t \qquad (2.1b)$$

And, in spherical polar coordinates Fig. 2.1(c) (radius, □ longitude, □ colatitudes) is

$$\frac{1}{r^2}\frac{\partial}{\partial r}\left(r^2\frac{\partial T}{\partial r}\right)+\frac{1}{r^2\sin\theta\,\partial\theta}\left(\sin\theta\frac{\partial T}{\partial\theta}\right)+\frac{1}{r^2\sin^2\theta}\frac{\partial^2 T}{\partial\phi^2}+\frac{\dot{Q}_v}{k}=\frac{1}{\alpha}\frac{\partial T}{\partial t} \qquad (2.1c)$$

Under steady state or stationary condition, the temperature of a body does not vary with time, i.e. $\partial T/\partial t = 0$. And, with no internal generation, the equation (2.1) reduces to

$$\nabla^2 T = 0$$

It should be noted that Fourier law can always be used to compute the rate of heat transfer by conduction from the knowledge of temperature distribution even for unsteady condition and with internal heat generation.

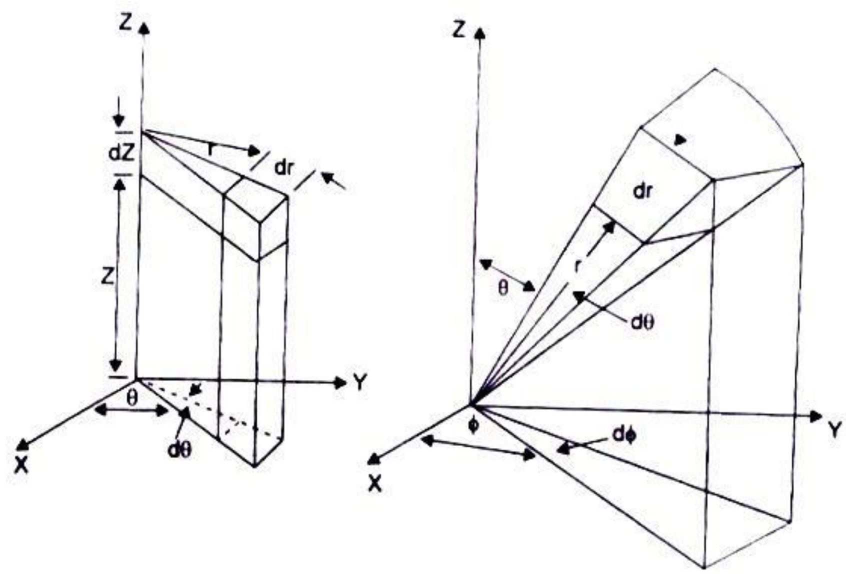


Fig1.3: Elemental volume in cylindrical coordinates (c):spherical coordinates

. One-Dimensional Heat Flow

The term 'one-dimensional' is applied to heat conduction problem when:

(i) Only one space coordinate is required to describe the temperature distribution within a heat conducting body;

(ii) Edge effects are neglected;

(iii) The flow of heat energy takes place along the coordinate measured normal to the surface.

3. Thermal Diffusivity and its Significance

Thermal diffusivity is a physical property of the material, and is the ratio of the material's ability to transport energy to its capacity to store energy. It is an essential parameter for transient processes of heat flow and defines the rate of change in temperature. In general, metallic solids have higher value, while non metallics, like paraffin, have a lower value. Materials having large thermal diffusivity respond quickly to changes in their thermal environment, while materials having lower a respond very slowly, take a longer time to reach a new equilibrium condition.

4. TEMPERATURE DISTRIBUTION IN I-D SYSTEMS

4.1 A Plane Wall

A plane wall is considered to be made out of a constant thermal conductivity material and extends to infinity in the Y- and Z-direction. The wall is assumed to be homogeneous and isotropic, heat flow is one-dimensional, under steady state conditions and losing negligible energy through the edges of the wall under the above mentioned assumptions the Eq. (2.2) reduces to

$d^2T / dx^2 = 0$; the boundary conditions are: at $x = 0, T = T_1$

Integrating the above equation, $x = L, T = T_2$

$T = C_1 x + C_2$, where C_1 and C_2 are two constants.

Substituting the boundary conditions, we get $C_2 = T_1$ and $C_1 = (T_2 - T_1)/L$ The temperature distribution in the plane wall is given by

$$T = T_1 - (T_1 - T_2)\, x/L \tag{2.3}$$

which is linear and is independent of the material.

Further, the heat flow rate, $\dot{Q}/A = -k\, dT/dx = (T_1 - T_2)k/L$, and therefore the temperature distribution can also be written as

$$T - T_1 = \left(\dot{Q}/A\right)\left(x/k\right) \tag{2.4}$$

i.e., “the temperature drop within the wall will increase with greater heat flow rate or when k is small for the same heat flow rate,"

4.2 A Cylindrical Shell-Expression for Temperature Distribution

In the cylindrical system, when the temperature is a function of radial distance only and is independent of azimuth angle or axial distance, the differential equation (2.2) would be, (Fig. 1.4)

$$d^2T/dr^2 + (1/r)\, dT/dr = 0$$

with boundary conditions: at $r = r_1$, $T = T_1$ and at $r = r_2$, $T = T_2$.

The differential equation can be written as:

$$\frac{1}{r}\frac{d}{dr}\left(r\, dT/dr\right) = 0, \text{ or, } \quad \frac{d}{dr}\left(r\ dT/dr\right) = 0$$

upon integration, $T = C_1 \ln(r) + C_2$, where C_1 and C_2 are the arbitrary constants.

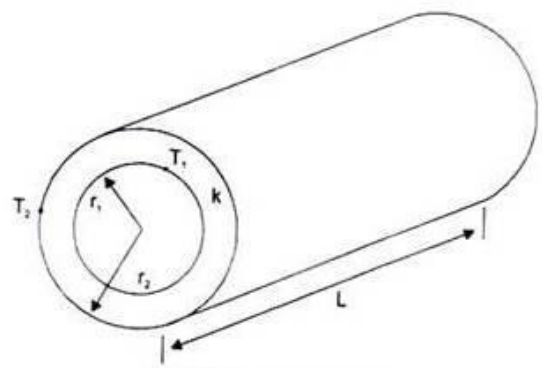


Fig 1.4: A Cylindrical shell

By applying the boundary conditions,

$$C_1 = \left(T_2 - T_1\right)/\ln\left(r_2/r_1\right)$$

and

$$C_2 = T_1 - \ln\left(r_1\right).\left(T_2 - T_1\right)/\ln\left(r_2/r_1\right)$$

The temperature distribution is given by

$$T = T_1 + \left(T_2 - T_1\right).\ \ln\left(r/r_1\right)/\ln\left(r_2/r_1\right) \text{ and}$$

$$\dot{Q}/L = -kA\ dT/dr \quad = 2\pi k\left(T_1 - T_2\right)/\ln\left(r_2/r_1\right) \qquad (2.5)$$

From Eq (2.5) It can be seen that the temperature varies 10gantJunically through the cylinder wall In contrast with the linear variation in the plane wall .

If we write Eq. (2.5) as $\dot{Q} = kA_m (T_1 - T_2)/(r_2 - r_1)$, where

$$A_m = 2\pi(r_2 - r_1)L/\ln(r_2/r_1) \quad = (A_2 - A_1)/\ln(A_2/A_1)$$

where A_2 and A_1 are the outside and inside surface areas respectively. The term A_m is called 'Logarithmic Mean Area' and the expression for the heat flow through a cylindrical wall has the same form as that for a plane wall.

4.3 Spherical and Parallelopiped Shells--Expression for Temperature Distribution

Conduction through a spherical shell is also a one-dimensional steady state problem if the interior and exterior surface temperatures are uniform and constant. The Eq. (2.2) in one-dimensional spherical coordinates can be written as

$\left(1/r^2\right)\dfrac{d}{dT}\left(r^2 dT/dr\right) = 0$, with boundary conditions,

at $\quad r = r_1,\ T = T_1;$ at $r = r_2,\ T = T_2$

or, $\quad \dfrac{d}{dr}\left(r^2 dT/dr\right) = 0$

and upon integration, T = $-C_1/r + C_2$, where c_1 and c_2 are constants. substituting the boundary conditions,

$C_1 = (T_1 - T_2)r_1 r_2/(r_1 - r_2)$, and $C_2 = T_1 + (T_1 - T_2)r_1 r_2/r_1(r_1 - r_2)$

The temperature distribution m the spherical shell is given by

$$T = T_1 - \left\{\frac{(T_1 - T_2)r_1 r_2}{(r_2 - r_1)}\right\} \times \left\{\frac{(r - r_1)}{r\ r_1}\right\} \qquad (2.6)$$

and the temperature distribution associated with radial conduction through a sphere is represented by a hyperbola. The rate of heat conduction is given by

$$\dot{Q} = 4\pi k(T_1 - T_2)r_1 r_2/(r_2 - r_1) = k(A_1 A_2)^{1/2}(T_1 - T_2)/(r_2 - r_1) \qquad (2.7)$$

where $A_1 = 4\pi_1^2$ and $A_2 = 4\pi r_2^2$

If A_l is approximately equal to A_2 i.e., when the shell is very thin,

$$\dot{Q} = kA(T_1 - T_2)/(r_2 - r_1) \text{; and } \dot{Q}/A = (T_1 - T_2)/\Delta r/k$$

which is an expression for a flat slab.

The above equation (2.7) can also be used as an approximation for parallelopiped shells which have a smaller inner cavity surrounded by a thick wall, such as a small furnace surrounded by a large thickness of insulating material, although the h eat flow especially in the corners, cannot be strictly considered one-dimensional. It has been suggested that for $(A_2/A_1) > 2$, the rate of heat flow can be approximated by the above equation by multiplying the geometric mean area $A_m = (A_1 A_2)^{½}$ by a correction factor 0.725.]

4.4 Composite Surfaces

There are many practical situations where different materials are placed m layers to form composite surfaces, such as the wall of a building, cylindrical pipes or spherical shells having different layers of insulation. Composite surfaces may involve any number of series and parallel thermal circuits.

4.5 Heat Transfer Rate through a Composite Wall

Let us consider a general case of a composite wall as shown m Fig. 1.5 There are 'n' layers of different materials of thicknesses L_1, L_2, etc and having thermal conductivities k_l, k_2, etc. On one side of the composite wall, there is a fluid A at temperature T_A and on the other side of the wall there is a fluid B at temperature T_B. The convective heat transfer coefficients on the two sides of the wall are h_A and h_B respectively. The system is analogous to a series of resistances as shown in the figure.

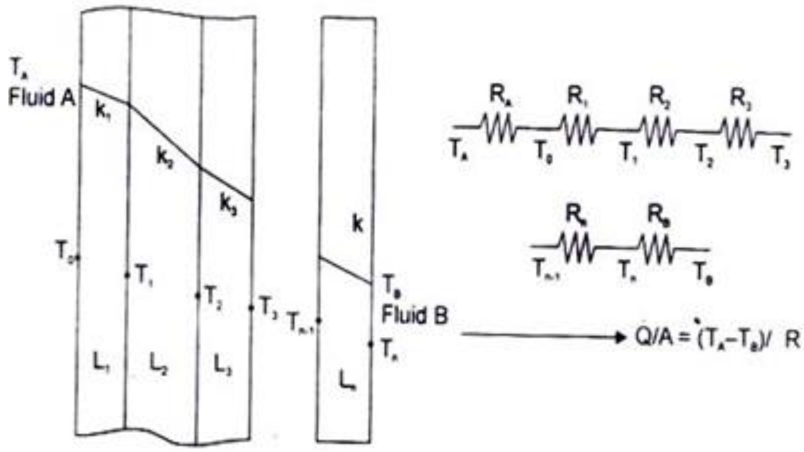


Fig 1.5 Heat transfer through a composite wall

4.6 The Equivalent Thermal Conductivity

The process of heat transfer through compos lie and plane walls can be more conveniently compared by introducing the concept of 'equivalent thermal conductivity', k_{eq}. It is defined as:

$$k_{eq} = \left(\sum_{i=1}^{n} L_i\right) \Bigg/ \sum_{i=1}^{n} (L_i / k_i) \qquad (2.8)$$

$$= \frac{\text{Total thickeness of the composite wall}}{\text{Total thermal resistance of the composite wall}}$$

And, its value depends on the thermal and physical properties and the thickness of each constituent of the composite structure.

Example 1.2 A furnace wall consists of 150 mm thick refractory brick (k = 1.6 W/mK) and 150 mm thick insulating fire brick (k = 0.3 W/mK) separated by an au gap (resistance 0 16 K/W). The outside walls covered with a 10 mm thick plaster (k = 0.14 W/mK). The temperature of hot gases is 1250°C and the room temperature is 25°C. The convective heat transfer coefficient for gas side and air side is 45 W/m2K and 20 W/m^2K. Calculate (i) the rate of heat flow per unit area of the wall surface (ii) the temperature at the outside and Inside surface of the wall and (iii) the rate of heat flow when the air gap is not there.

Solution: Using the nomenclature of Fig. 2.3, we have per m2 of the area, h_A = 45, and

$R_A = 1/h_A = 1/45 = 0.0222$; $h_B = 20$, and $R_B = 1120 = 0.05$

Resistance of the refractory brick, $R_1 = L_1/k_1 = 0.15/1.6 = 0.0937$

Resistance of the insulating brick, $R_3 = L_3/k_3 = 0.15/0.30 = 0.50$

The resistance of the air gap, $R_2 = 0.16$

Resistance of the plaster, $R_4 = 0.01/0.14 = 0.0714$

Total resistance = 0.8973, m^2K/W

Heat flow rate = $\Delta T/\Sigma R = (1250\text{-}25)/0.8973 = 13662\ W/m^2$

Temperature at the inner surface of the wall

$= T_A – 1366.2 \times 0.0222 = 1222.25$

Temperature at the outer surface of the wall

$= T_B + 1366.2 \times 0.05 = 93.31\ °C$

When the air gap is not there, the total resistance would be

0.8973 - 0.16 = 0.7373

and the heat flow rate = (1250 – 25)/0/7373 = 1661.46 W/m^2

The temperature at the inner surface of the wall

= 1250 – 1660.46 × 0.0222 = 1213.12°C

i.e., when the au gap is not there, the heat flow rate increases but the temperature at the inner surface of the wall decreases.

The overall heat transfer coefficient U with and without the air gap is

$$U = \left(\dot{Q}/A\right)/\,\Delta T$$

$= 13662 / (1250 – 25) = 1.115\ Wm^2\ °C$

and 1661.46/l225 = 1356 $W/m^{2o}C$

The equivalent thermal conductivity of the system without the air gap

$k_{eq} = (0.15 + 0.15 + 0.01)/(0.0937 + 0.50 + 0.0714) = 0.466\ W/mK.$

Example 1.2 A brick wall (10 cm thick, k = 0.7 W/m°C) has plaster on one side of the wall (thickness 4 cm, k = 0.48 W/m°C). What thickness of an insulating material (k = 0.065 W m^{o}C) should be added on the other side of the wall such that the heat loss through the wall IS reduced by 80 percent.

Solution: When the insulating material is not there, the resistances are:

$R_1 = L_1/k_1 = 0.1/0.7 = 0.143$

and $R_2 = 0.04/0.48 = 0.0833$

Total resistance = 0.2263

Let the thickness of the insulating material is L_3. The resistance would then be

$L_3/0.065 = 15.385\ L_3$

Since the heat loss is reduced by 80% after the insulation is added.

$$\frac{\dot{Q}\ \text{with insulation}}{\dot{Q}\ \text{without insulation}} = 0.2 = \frac{R\ \text{without insulation}}{R\ \text{with insulation}}$$

or, the resistance with insulation = 0.2263/0.2 = 01.1315

and, 15385 L_3 = I 1315 – 0.2263 = 0.9052

L_3 = 0.0588 m = 58.8 mm

Example 1.3 An ice chest IS constructed of styrofoam (k = 0.033 W/mK) having inside dimensions 25 by 40 by 100 cm. The wall thickness is 4 cm. The outside surface of the chest is exposed to air at 25°C with h = 10 W/m^{2}K. If the chest is completely filled with ice, calculate the time for ice to melt completely. The heat of fusion for water is 330 kJ/kg.

Solution: If the heat loss through the comers and edges are Ignored, we have three walls of walls through which conduction heat transfer Will occur.

(a) 2 walls each having dimensions 25 cm × 40 cm × 4 cm

(b) 2 walls each having dimensions 25 cm × 100 cm × 4 cm

(c) 2 walls each having dimensions 40 cm × 100 cm × 4 cm

The surface area for convection heat transfer (based on outside dimensions)

$2(33 \times 48 + 33 \times 108 + 48 \times 108) \times 10^{-4} = 2.0664\ m^2$.

Resistance due to conduction and convection can be written as

$$2\left(\frac{0.04}{0.033\times0.25\times0.4}+\frac{0.04}{0.033\times0.25\times1}+\frac{0.04}{0.033\times0.4\times1}\right)+\frac{1}{10\times2.0664}$$

$= 40 + 0.0484 = 40.0484$ K/W

$\dot{Q} = \Delta T / \Sigma R = (25 - 0.0) / 40.0484 = 0.624$ W

Inside volume of the container - $0.25 \times 04 \times 1 = 0.1\ m^3$

Mass of Ice stored = $800 \times 0.1 = 80$ kg; taking the density of Ice as 800 kg/m^3. The time required to melt 80 kg of ice is

$$t = \frac{80\times330\times1000}{0.624\times3600\times24} = 490 \text{ days}$$

Example1.4 A composite furnace wall is to be constructed with two layers of materials (k_1 = 2.5 W/m^{o}C and k_2 = 0.25 W/m^{o}C). The convective heat transfer coefficient at the inside and outside surfaces are expected to be 250 W/m^{2o}C and 50 W/m^{2o}C respectively. The temperature of gases and air are 1000 K and 300 K. If the interface temperature is 650 K, Calculate (i) the thickness of the two materials when the total thickness does not exceed 65 cm and (ii) the rate of heat flow. Neglect radiation.

Solution: Let the thickness of one material (k = 2.5 W / mK) is xm, then the thickness of the other material (k = 0.25 W/mK) will be (0.65 –x)m.

For steady state condition, we can write

$$\frac{\dot{Q}}{A} - \frac{1000-650}{\frac{1}{250}+\frac{x}{2.5}} = \frac{1000-300}{\frac{1}{250}+\frac{x}{2.5}+\frac{(0.65-x)}{0.25}+\frac{1}{50}}$$

$$\therefore\ 700(0.004+0.4x) = 350\{0.004+0.4x+4(0.65-x)+0.02\}$$

(i) 6x = 3.29 and x= 0.548 m.

and the thickness of the other material = 0.102 m.

(ii) $\dot{Q}/A$ = (350) / (0.004 + 0.4 × 0.548) = 1.568 kW/m^2

Example 1.5 A composite wall consists of three layers of thicknesses 300 rum, 200 mm and 100 mm with thermal conductivities 1.5, 3.5 and is W/mK respectively. The inside surface is exposed to gases at 1200°C with convection heat transfer coefficient as 30W/m^{2}K. The temperature of air on the other side of the wall is 30°C with convective heat transfer coefficient 10 Wm^{2}K. If the temperature at the outside surface of the wall is 180°C, calculate the temperature at other surface of the wall, the rate of heat transfer and the overall heat transfer coefficient.

Solution: The composite wall and its equivalent thermal circuits is shown in the figure.

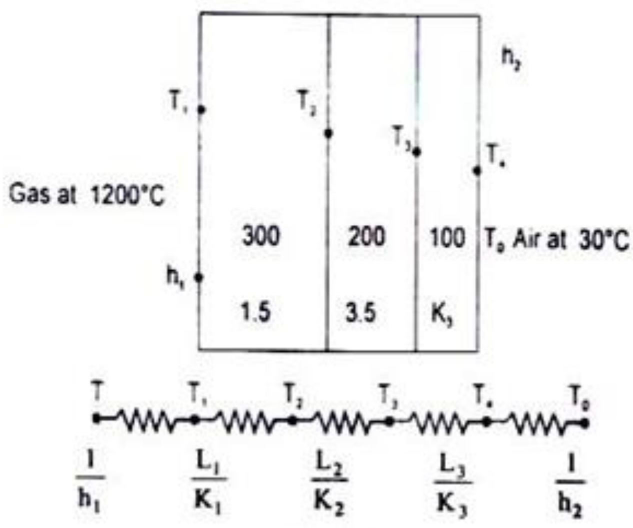


Fig 1.6

The heat energy will flow from hot gases to the cold air through the wall.

From the electric Circuit, we have

$$\dot{Q}/A = h_2(T_4 - T_0) = 10 \times (180 - 30) = 1500\,W/m^2$$

also, $\dot{Q}/A = h_1(1200 - T_1)$

$$T_1 = 1200 - 1500/30 = 1150^{o}C$$

$$\dot{Q}/A = (T_1 - T_2)/L_1/k_1$$

$$T_2 = T_1 - 1500 \times 0.3/1.5 = 850$$

Similarly, $\dot{Q}/A = (T_2 - T_3)/(L_2/k_2)$

$$T_3 = T_2 - 1500 \times 0.2/3.5 = 764.3^{o}C$$

and $\quad \dot{Q}/A = (T_3 - T_4)/(L_3/k_3)$

$$L_3/k_3 = (764.3 - 180)/1500 \text{ and } k_3 = 0.256 \text{ W/mK}$$

Check:

$$\dot{Q}/A = (1200 - 30)/\Sigma R;$$

where $\Sigma R = 1/h_1 + L_1/k_1 + L_2/k_2 + L_3/k_3 + 1/h_2$

$$\Sigma R = 1/30 + 0.3/1.5 + 0.2/3.5 + 0.1/0.256 + 1/10 = 0.75$$

and $\dot{Q}/A = 1170/0.78 = 1500 \text{ W/m}^2$

The overall heat transfer coefficient, $U = 1/\Sigma R = 1/0.78 = 1.282 \text{ W/m}^2\text{K}$

Since the gas temperature is very high, we should consider the effects of radiation also. Assuming the heat transfer coefficient due to radiation = 3.0 W/m^{2}K the electric circuit would be:

The combined resistance due to convection and radiation would be

$$\frac{1}{R} = \frac{1}{R_1} + \frac{1}{R_2} = \frac{1}{\frac{1}{h_c}} + \frac{1}{\frac{1}{h_r}} = h_c + h_r = 60 \text{W/m}^{2o}\text{C}$$

$$\therefore \dot{Q}/A = 1500 = 60(T - T_1) = 60(1200 - T_1)$$

$$\therefore T_1 = 1200 - \frac{1500}{60} = 1175^{o}C$$

again, $\therefore \dot{Q}/A = (T_1 - T_2)/L_1/k_1 \Rightarrow T_2 = T_1 - 1500 \times 0.3/1.5 = 875^{o}C$

and $T_3 = T_2 - 1500 \times 0.2/3.5 = 789.3^{o}C$

$L_3 / k_3 = (789.3 - 180)/1500; \therefore k_3 = 0.246 \text{ W/mK}$

$$\Sigma R = \frac{1}{60} + \frac{0.3}{1.5} + \frac{0.2}{1.5} + \frac{0.2}{3.5} + \frac{0.1}{0.246} + \frac{1}{10} = 0.78$$

and $U = 1/\Sigma R = 1.282 \text{ W/m}^2\text{K}$

Example 1.6 A flat roof (12 m x 20 m) of a building has a composite structure It consists of a 15 cm lime-khoa plaster covering (k = 0 17 W/m°C) over a 10 cm cement concrete (k = 0.92 W/m°C). The ambient temperature is 42°C. The outside and inside heat transfer coefficients are 30 W/m2°C and 10 W/m2 0C. The top surface of the roof absorbs 750 W/m2 of solar radiant energy. The temperature of the space may be assumed to be 260 K. Calculate the temperature of the top surface of the roof and the amount of water to be sprinkled uniformly over the roof surface such that the inside temperature is maintained at 18°C.

Solution: The physical system is shown in Fig. 1.7 and it is assumed we have one-dimensional flow, properties are constant and steady state conditions prevail.

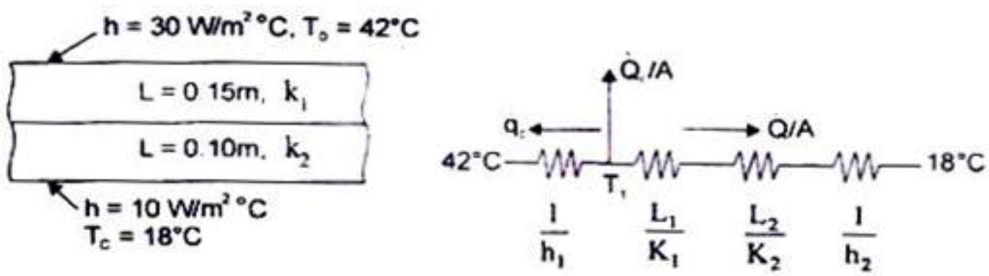


Fig 1.7

Let the temperature of the top surface be T_1°C.

Heat lost by thee top surface by convection to the surroundings is

$$\dot{Q}_c / A = h(\Delta T) = 30 \times (T_1 - 42) = (30T_1 - 1260)$$

Heat energy conducted inside through the roof = $(\Delta T / \Sigma R)$

or, $\frac{\dot{Q}}{A} = \frac{T_1 - 18}{\frac{L_1}{k_1} + \frac{L_2}{k_2} + \frac{1}{h_2}} = (T_1 - 18) / \left(\frac{0.15}{0.17} + \frac{0.1}{0.92} + \frac{1}{10} \right)$ = 0.918 (T_1 – 18)

Assuming that the top surface of the roof behaves like a black body, energy lost by radiation.

$$\dot{Q}_r / A = \sigma \left[(T_1 + 273)^4 - 260^4 \right] \quad = 5.67 \times 10^{-8} (T_1 + 273)^4 - 259.1$$

By making an energy balance on the top surface of the roof,

Energy coming in = Energy going out

750 = (30T, -1260)+ 0.918 (T_1-18) + 5.67 × 10^{-8} (T_1 + 273)^4 - 259.1

or, 2285.624 = 30.918 T_1 + 5.67 × 10^{-8} (T_1 + 273)^4

Solving by trial and error, T_1 = 53.4°C, and the total energy conducted through the roof per hour is

0.918 (53.4 – 18) × (12 × 20) × 3600 = 28077.58 kJ/hr

Assuming the latent heat of vaporization of water as 2430 kJ/kg, the quantity of water to be sprinkled over the surface such that it evaporates and consumes 28077.58 kJ/hr, is

$\dot{M}_w$ = 28077.58/2430 = 11.55 kg/hr.

Example 1.7 An electric hot plate is maintained at a temperature of 350°C and is used to keep a solution boiling at 95°C. The solution is contained in a cast iron vessel (wall thickness 25 mm, k = 50 W/mK) which is enamelled inside (thickness 0.8 mm, k = 1.05 WmK) The heat transfer coefficient for the boiling solution is 5.5 kW/m1K. Calculate (i) the overall heat transfer coefficient and (ii) heat transfer rate.

If the base of the cast iron vessel is not perfectly flat and the resistance of the resulting air film is 35 m2K1kW, calculated the rate of heat transfer per unit area. (Gate'93)

Solution: The physical system is shown in the figure below.

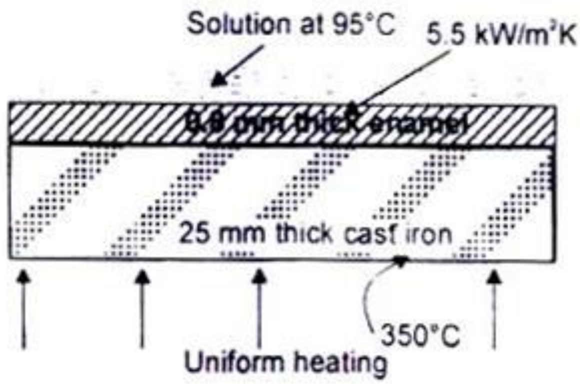


Fig 1.8

Under steady state conditions,

$$\dot{Q}/A = U(\Delta T) = \frac{(\Delta T)}{1/U}$$, where U is the overall heat transfer coefficient.

$$= \frac{(\Delta T)}{R} = \frac{(\Delta T)}{\frac{L_1}{k_1} + \frac{L_2}{k_2} + \frac{1}{h}}$$

Therefore,

$$1/U = \frac{L_1}{k_1} + \frac{L_2}{k_2} + \frac{1}{h} = \left(\frac{0.025}{50} + \frac{0.0008}{1.05} + \frac{1}{5500}\right) = 0.00144$$

$U = 692.65$ W/m^{2}K

$\dot{Q}/A = U(\Delta T)$ = 692.65 × (350 – 95) = 176.65 kW/m^2.

With the presence of air film at the base, the total resistance to heat flow would be:

0.00144 + 0.035 = 0.03644 m^{2}K/W

and the rate of heat transfer, $\dot{Q}/A$ = 255/0.03644 = 7 kW/m^2.

(Fig. 1.9 shows a combination of thermal resistance placed in series and parallel for a composite wall having one-dimensional steady state heat transfer. By drawing analogous electric circuits, we can solve such complex problems having both parallel and series thermal resistances.)

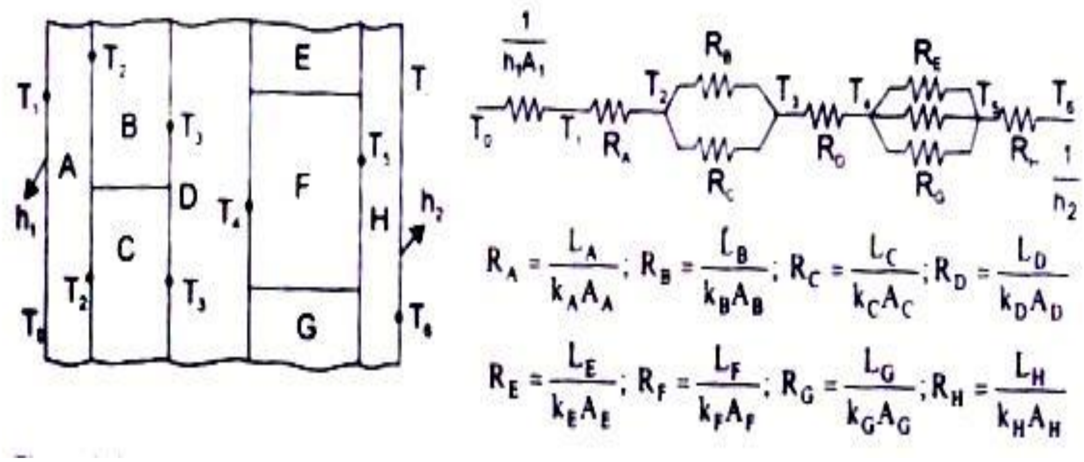


Fig. 1.9 Series and parallel one-dimensional heat transfer through a composite wall with convective heat transfer and its electrical analogous circuit

Example 1.8 A door (2 m x I m) is to be fabricated with 4 cm thick card board (k = 0.2 W /mK) placed between two sheets of fibre glass board (each having a thickness of 40 mm and k = 0.04 W/mK). The fibre glass boards are fastened with 50 steel studs (25 mm diameter, k = 40 W/mK). Estimate the percentage of heat transfer flow rate through the studs.

Solution: The thermal circuit with steel studs can be drawn as in Fig. 1.10.

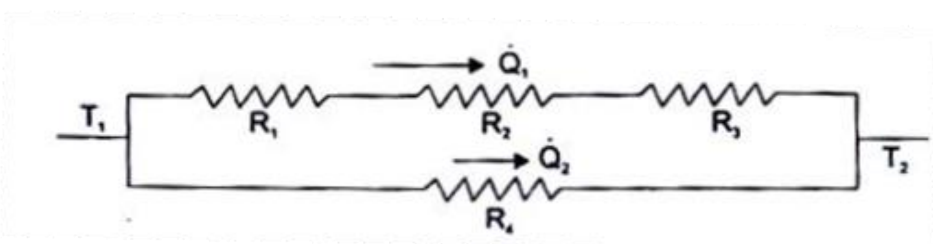


Fig 1.10

The cross-sectional area or the surface area of the door for the heat transfer is 2m^2. The cross-sectional area of the steel studs is:

$$50 \times \pi/4\,(0.025)^2 = 0.02455 \text{ m}^2$$

and the area of the door – area of the steel studs = 2.0 – 0.02455 = 1.97545

R_1, the resistance due to fibre glass board on the outside

$$= L/kA = 0.04/(0.04 \times 1.97545) = 0.506.$$

R_2, the resistance due to card board = 0.101

R_3, the resistance due to fibre glass board on the inside = 0.506

R_4, the resistance due to steel studs = L/kA = 0.121 (40 × 0.2455) = 0 1222

With reference to Fig 2.9, $\dot{Q}_1 = (T_1 - T_2)/\Sigma R = (T_1 - T_2)/1.113$

and $\dot{Q}_2 = (T_1 - T_2)/0.1222$

Therefore, $\dot{Q}_2 / (\dot{Q}_1 + \dot{Q}_2) = 8.1833/9.0818 = 0.9$

ie, 90 percent of the heat transfer will take place through the studs.

Example 1.9 Find the heat transfer rate per unit depth through the composite wall sketched. Assume one dimensional heat flow.

Solution: The analogous electric circuit has been drawn in the figure.

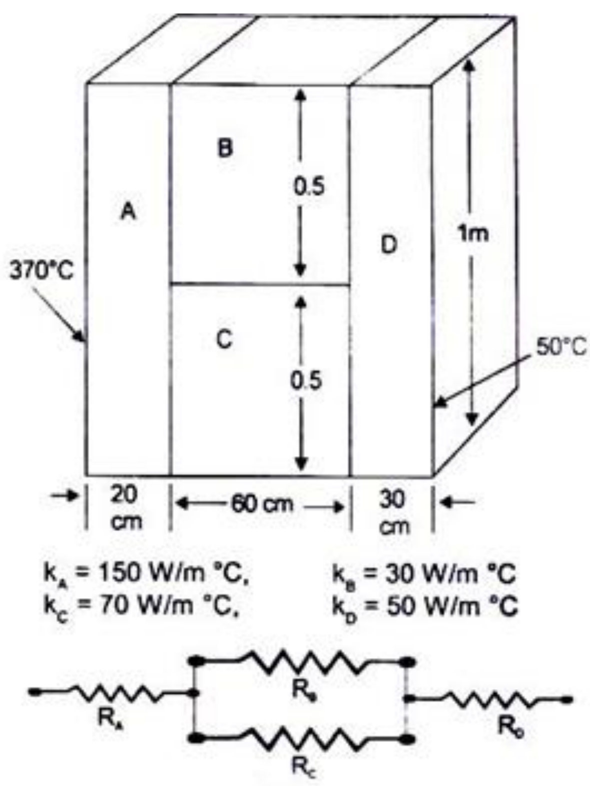


Fig 1.11

R_A = 0.2/150 = 0.00133

R_B = 0.6/(30 × 0.5) = 0.04

R_C = 0.6/(70 × 0.5) = 0.017

R_D = 0.3/50 = 0.006

$1/R_B + 1/R_C = 1/R_{BC}$ = 83.82

Therefore, R_{BC} = 1/83.82 = 0.0119

Total resistance to heat flow = 0.00133 + 0.0119 + 0006 = 0.01923

Rate of heat transfer per unit depth = (370–50)/ 0.01923 = 16.64 kW m.

The Significance of Biot Number

Let us consider steady state conduction through a slab of thickness L and thermal conductivity k. The left hand face of the wall is maintained at T constant temperature T_1 and the right hand face is exposed to ambient air at T_o, with convective heat transfer coefficient h. The analogous electric circuit will have two thermal resistances: R_1 = L/k and R_2 = l/h. The drop in temperature across the wall and the air film will be proportional to their resistances, that is, (L/k)/(1/h) = hL/k.

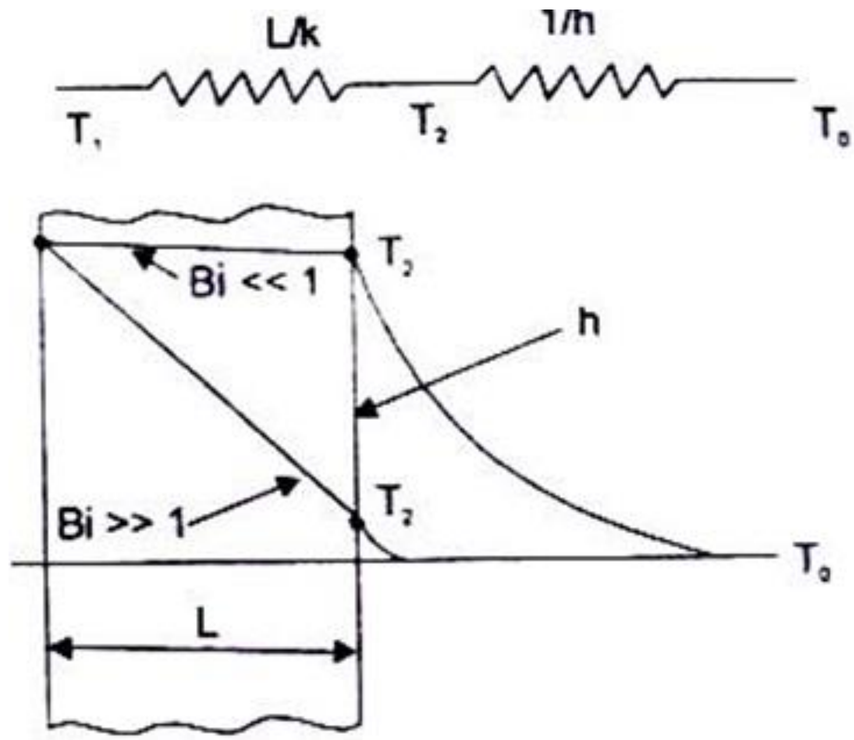


Fig 1.12: Effect of Biot number on temperature profile

This dimensionless number is called 'Biot Number' or,

$$B_i = \frac{\text{Conduction resistance}}{\text{Convection resistance}}$$

When Bi >> 1, the temperature drop across the air film would be negligible and the temperature at the right hand face of the wall will be approximately equal to the ambient temperature. Similarly, when Bi «I, the temperature drop across the wall is negligible and the transfer of heat will be controlled by the air film resistance.

5. The Concept of Thermal Contact Resistance

Heat flow rate through composite walls are usually analysed on the assumptions that - (i) there is a perfect contact between adjacent layers, and (ii) the temperature at the interface of the two plane surfaces is the same. However, in real situations, this is not true. No surface, even a so-called 'mirror-finish surface', is perfectly smooth ill a microscopic sense. As such, when two surfaces are placed together, there is not a single plane of contact. The surfaces touch only at limited number of spots, the aggregate of which is only a small fraction of the area of the surface or 'contact area'. The remainder of the space between the surfaces may be filled with air or other fluid. In effect, this introduces a resistance to heat flow at the interface. This resistance IS called 'thermal contact resistance' and causes a temperature drop between the materials at the interfaces as shown In Fig. 2.12. (That is why, Eskimos make their houses having double ice walls separated by a thin layer of air, and in winter, two thin woolen blankets are more comfortable than one woolen blanket having double thickness.)

Fig. 2.12 Temperature profile with and without contact resistance when two solid surfaces are joined together

Example 1.10 A furnace wall consists of an inner layer of fire brick 25 cm thick k = 0.4 W/mK and a layer of ceramic blanket insulation, 10 cm thick k = 0.2 W/mK. The thermal contact resistance between the two walls at the interface is 0.01 m^2K/w. Calculate the temperature drop at the interface if the temperature difference across the wall is 1200K.

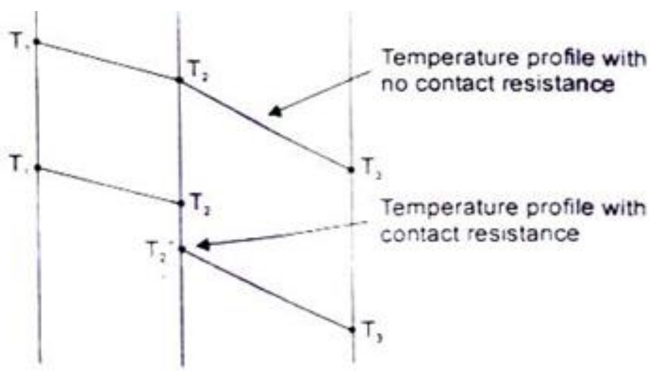


Fig 1.13: temperature profile with and without contact resistance when two solid surfaces are joined together

Solution: The resistance due to inner fire brick = L/k = 0.25/0.4 = 0.625.

The resistance of the ceramic insulation = 0.1/0.2 = 0.5

Total thermal resistance = 0.625 + 0.01 + 0.5 = 1 135

Rate of heat flow, $\dot{Q}/A$ = □t /□□ =1200/1 135 =1057.27 W/m^2

Temperature drop at the interface,

$\Delta T = \left(\dot{Q}/A\right) \times R = 1057.27 \times 0.01 = 10.57$ K

Example 1.11 A 20 cm thick slab of aluminium (k = 230 W/mK) is placed in contact with a 15 cm thick stainless steel plate (k = 15 W/mK). Due to roughness, 40 percent of the area is in direct contact and the gap (0.0002 m) is filled with air (k = 0.032 W/mK). The difference in temperature between the two outside surfaces of the plate is 200°C Estimate (i) the heat flow rate, (ii) the contact resistance, and (iii) the drop in temperature at the interface.

Solution: Let us assume that out of 40% area m direct contact, half the surface area is occupied by steel and half is occupied by aluminium.

The physical system and its analogous electric circuits is shown in Fig. 2.13.

$R_1 = \frac{0.2}{230 \times 1} = 0.00087$, $\quad R_2 = \frac{0.0002}{230 \times 0.2} = 4.348 \times 10^{-6}$

$R_3 = \frac{0.0002}{0.032 \times 0.6} = 1.04 \times 10^{-2}$, $\quad R_4 = \frac{0.0002}{15 \times 0.2} = 6.667 \times 10^{-5}$

and $R_5 = \frac{0.15}{(15 \times 1)} = 0.01$

Again $1/R_{2,3,4} = 1/R_2 + 1/R_3 + 1/R_4$

$= 2.3 \times 10^5 + 96.15 + 1.5 \times 10^4 = 24.5 \times 10^4$

Therefore, $R_{2,3,4} = 4.08 \times 10^{-6}$

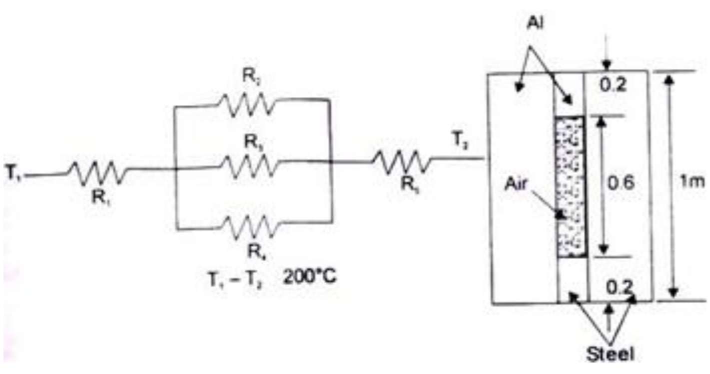


Fig 1.14

Total resistance, $\Sigma R = R_1 + R_{2,3,4} + R_5$

$= 870 \times 10^{-6} + 4.08 \times 10^{-6} + 1000 \times 10^{-6} = 1.0874 \times 10^{-2}$

Heat flow rate, $\dot{Q}$ = 200/1.087 × 10^{-2} = 18.392 kW per unit depth of the plate.

Contact resistance, $R\,R_{2,\,3,\,4} = 4.08 \times 10^{-6}$ mK / W

Drop in temperature at the interface, $\Delta T = 4.08 \times 10^{-6} \times 18392 = 0.075^oC$

6. An Expression for the Heat Transfer Rate through a Composite Cylindrical System

Let us consider a composite cylindrical system consisting of two coaxial cylinders, radii r_1, r_2 and r_2 and r_3, thermal conductivities k_l and k_2 the convective heat transfer coefficients at the inside and outside surfaces h_1 and h_2 as shown in the figure. Assuming radial conduction under steady state conditions we have:

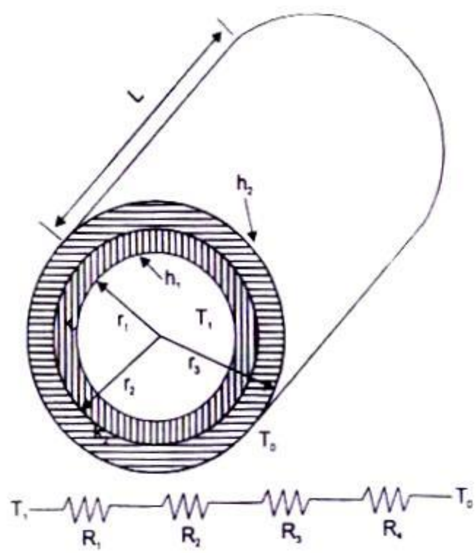


Fig 1.15

$R_1 = 1/h_1 A_1 = 1/2\ \pi_1\ Lh_1$

$R_2 = \ln(r_2/r_1)2\pi Lk_1$

$R_3 = \ln(r_3/r_2)2\pi Lk_2$

$R_4 = 1/h_2 A_2 = 1/2\pi_3 h_2 L$

And $\dot{Q}/2\pi L = (T_1 - T_0)/\Sigma R$

$$= (T_1 - T_0)/\left[\left(1/h_1 r_1 + \ln(r_2/r_1)/k_1 + \ln(r_3 + r_2)/k_2 + 1/h_2 r_3\right)\right]$$

Example 1.12 A steel pipe. Inside diameter 100 mm, outside diameter 120 mm (k 50 W/mK) IS Insulated with a 40 mm thick high temperature Insulation (k = 0.09 W/mK) and another Insulation 60 mm thick (k = 0.07 W/mK). The ambient temperature IS 25°C. The heat transfer coefficient for the inside and outside surfaces are 550 and 15 W/m^{2}K respectively. The pipe carries steam at 300^{o}C. Calculate (1) the rate of heat loss by steam per unit length of the pipe (11) the temperature of the outside surface

Solution: I he cross-section of the pipe with two layers of insulation is shown 111 Fig. 1.16. with its analogous electrical circuit.

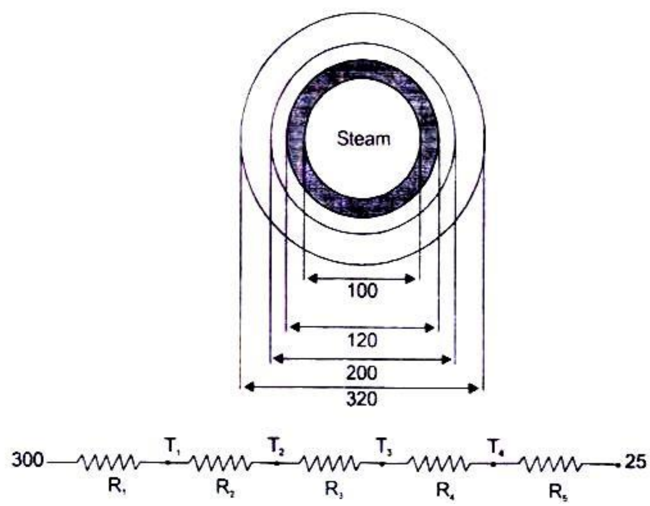


Fig1.16 Cross-section through an insulated cylinder, thermal resistances in series.

For L = 1.0 m. we have

R_1, the resistance of steam film = 1/hA = $1/(500 \times 2 \times 3.14 \times 50 \times 10^{-3})$ = 0.00579

R_2, the resistance of steel pipe = $\ln(r_2/r_l)$ / 2 π k

= ln(60/50)/2 $\pi \times 50$ = 0.00058

R_3, resistance of high temperature Insulation

$\ln(r_3/r_2)$ / 2 π k = ln(100/60) / 2 $\pi \times 0.09$ = 0.903

R_4 = 1n(r_4/r_3)/2 π k = ln(160/100)/2 π × 0.07 = 1.068

R_5 = resistance of the air film = 1/(15 × 2 π × 160 × 10^{-3}) = 0.0663

The total resistance = 2.04367

and $\dot{Q} = \Delta T / \Sigma R$ = (300 – 25) / 204367 = 134.56 W per metre length of pipe.

Temperature at the outside surface. T_4 = 25 + R_5,

$\dot{Q}$ = 25 + 134.56 × 0.0663 = 33.92^o C

When the better insulating material (k = 0.07, thickness 60 mm) is placed first on the steel pipe, the new value of R_3 would be

R_3 = ln(120 /60) / 2 π × 0.07 = 1.576 ; and the new value of R_4 will be

R_4 = ln(160/120) 2 π × 0.09 = 0.5087

The total resistance = 2.15737 and Q = 275/2.15737 = 127.47 W per m length (Thus the better insulating material be applied first to reduce the heat loss.) The overall heat transfer coefficient, U, is obtained as U = $\dot{Q} / A\,\Delta T$

The outer surface area = π × 320 × 10^{-3} × 1 = 1.0054

and U = 134.56/(275 × 1.0054) = 0.487 W/m^2 K.

Example 1.13 A steam pipe 120 mm outside diameter and 10m long carries steam at a pressure of 30 bar and 099 dry. Calculate the thickness of a lagging material (k = 0.99 W/mK) provided on the steam pipe such that the temperature at the outside surface of the insulated pipe does not exceed 32°C when the steam flow rate is 1 kg/s and the dryness fraction of steam at the exit is 0.975 and there is no pressure drop.

Solution: The latent heat of vaporization of steam at 30 bar = 1794 kJ/kg.

The loss of heat energy due to condensation of steam = 1794(0.99 – 0.975)

= 26.91 kJ/kg.

Since the steam flow rate is 1 kg/s, the loss of energy = 26.91 kW.

The saturation temperature of steam at 30 bar IS 233.84°C and assuming that the pipe material offers negligible resistance to heat flow, the temperature at the outside surface of the uninsulated steam pipe or at the inner surface of the lagging material is 233.84°C. Assuming one-dimensional radial heat flow through the lagging material, we have

$$\dot{Q} = (T_1 - T_2)/[\ln(r_2/r_l)]\ 2\pi Lk$$

or, $26.91 \times 1000\ (W) = (233.84 - 32) \times 2\pi \times 10 \times 0.99/1n(r/60)$

$\ln(r/60) = 0.4666$

$r_2/60 = \exp(0.4666) = 1.5946$

$r_2 = 95.68$ mm and the thickness = 35.68 mm

Example 1.14 A Wire, diameter 0.5 mm length 30 cm, is laid coaxially in a tube (inside diameter 1 cm, outside diameter 1.5 cm, k = 20 W/mK). The space between the wire and the inside wall of the tube behaves like a hollow tube and is filled with a gas. Calculate the thermal conductivity of the gas if the current flowing through the wire is 5 amps and voltage across the two ends is 4.5 V, temperature of the wire is 160°C, convective heat transfer coefficient at the outer surface of the tube is 12 W/m^2K and the ambient temperature is 300K.

Solution: Assuming steady state and one-dimensional radial heat flow, we can draw the thermal circuit as shown In Fig. 1 17.

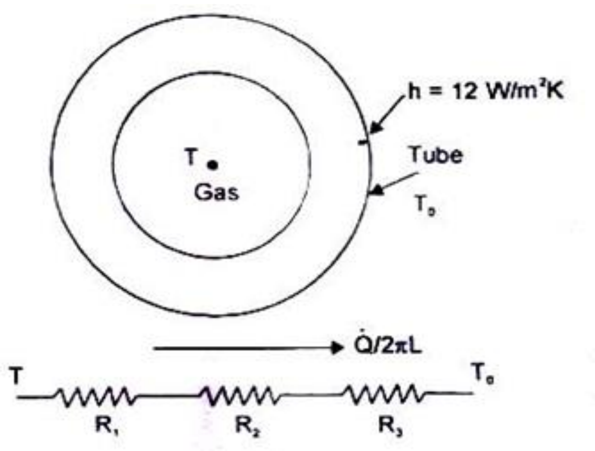


Fig 1.17

The rate of heat transfer through the system,

$\dot{Q}/2\,\pi\,L = VI/2\,\pi\,L = (4.5 \times 5)/(2 \times 3.142 \times 0.3) = 11.935$ (W/m)

R_1, the resistance due to gas = $\ln(r_2/r_l)$, k = ln(0.01/0.0005)/k = 2.996/k.

R_2, resistance offered by the metallic tube = $\ln(r_3 / r_2)$ k

= ln(1.5 /1.0) / 20 = 0.02

R_3, resistance due to fluid film at the outer surface

$l/hr_3 = 1/(l2\times1.5\times I0^{-2}) = 5.556$

and $\dot{Q}$ / 2 π L = □L/□□R = [(273 + 160) – 300]/□R

Therefore, □R = 133/11.935 = 11.l437, and

R_1 = 2.9996/k = 11.1437 – 0.02 – 5.556 = 5.568

or, k = 2.996/5.568 = 0.538 W/mK.

Example 1.15 A steam pipe (inner diameter 16 cm, outer diameter 20 cm, k = 50 W/mK) is covered with a 4 cm thick insulating material (k = 0.09 W/mK). In order to reduce the heat loss, the thickness of the insulation is Increased to 8mm. Calculate the percentage reduction in heat transfer assuming that the convective heat transfer coefficient at the Inside and outside surfaces are 1150 and 10 W/m^2K and their values remain the same.

Solution: Assuming one-dimensional radial conduction under steady state,

$\dot{Q}$ / 2□L = □T/□R

R_1, resistance due to steam film = 1/hr = 1/(1150 × 0.08) = 0.011

R_2, resistance due to pipe material = $\ln(r_2/r_1)/k$ = ln (10/8)/50 = 0.00446

R_3, resistance due to 4 cm thick insulation

= $\ln(r_3/r_2)/k$ = ln(14/10)/0.09 = 3.738

R_4, resistance due to air film = 1/hr = 1/(10 × 0.14) = 0.714.

Therefore, $\dot{Q}/2\pi L = \Delta T/(0.011 + 0.00446 + 3.738 + 0.714)$ = 0.2386□T

When the thickness of the insulation is increased to 8 cm, the values of R_3 and R_4 will change.

R_3 = ln(r_3/r_2)/k = ln(18/10)/0.09 = 6.53 ; and

R_4 = 1/hr = 1/(10 × 0.18) = 0.556

Therefore, $\dot{Q}/2\pi L = \Delta T$ / (0.011 + 0.00446 + 6.53 + 0.556)

= ΔT / 7.1 = 0.14084 ΔT

Percentage reduction in heat transfer = $\frac{(0.22386 - 0.14084)}{0.22386} = 0.37 = 37\%$

Example 1.16 A small hemispherical oven is built of an inner layer of insulating fire brick 125 mm thick (k = 0.31 W/mK) and an outer covering of 85% magnesia 40 mm thick (k = 0.05 W/mK). The inner surface of the oven is at 1073 K and the heat transfer coefficient for the outer surface is 10 W/m^{2}K, the room temperature is 20^{o}C. Calculate the rate of heat loss through the hemisphere if the inside radius is 0.6 m.

Solution: The resistance of the fire brick

$$= (r_2 - r_1)/2\pi k r_1 r_2 = \frac{0.725 - 0.6}{2\pi \times 0.31 \times 0.6 \times 0.725} = 0.1478$$

The resistance of 85% magnesia

$$= (r_3 - r_2)/2\pi k r_2 r_3 = \frac{0.765 - 0.725}{2\pi \times 0.05 \times 0.725 \times 0.765} = 0.2295$$

The resistance due to fluid film at the outer surface = 1/hA

$$= \frac{1}{10 \times 2\pi \times (0.765 \times 0.765)} = 0.2295$$

The resistance due to fluid film at the outer surface = 1/hA

$$= \frac{1}{10 \times 2\pi \times (0.765 \times 0.765)} = 0.0272$$

Rate of heat flow, $\dot{Q} = \Delta T/\Sigma R = \frac{800 - 20}{0.1478 + 0.2295 + 0.272} = 1930\text{W}$

Example 1.17 A cylindrical tank with hemispherical ends is used to store liquid oxygen at –

180^{o}C. The diameter of the tank is 1.5 m and the total length is 8 m. The tank is covered with a 10 cm thick layer of insulation. Determine the thermal conductivity of the insulating material so that the boil off rate does not exceed 10 kg/hr. The latent heat of vapourization of liquid oxygen is 214 kJ/kg. Assume that the outer surface of insulation is at 27^{o}C and the thermal resistance of the wall of the tank is negligible. (ES-94)

Solution: The maximum amount of heat energy that flows by conduction from outside to inside = Mass of liquid oxygen × Latent heat of vapourisation.

= 10 × 214 = 2140 kJ/hr = 2140 × 1000/3600 = 594.44 W

Length of the cylindrical part of the tank = 8 – 2r = 8 – 1.5 = 6.5m

since the thermal resistance of the wall does not offer any resistance to heat flow, the temperature at the inside surface of the insulation can be assumed as - 183°C whereas the temperature at the outside surface of the insulation is 27°C.

Heat energy coming in through the cylindrical part, $\dot{Q}_1 = \dfrac{\Delta T}{\dfrac{\ln(r_2/r_1)}{2\pi L k}}$

or, $\dot{Q}_1 = \dfrac{(27+183)\times 2\pi \times 6.5\,k}{\ln(8.5/7.5)} = 68531.84\ k$

Heat energy coming in through the two hemispherical ends,

$$\dot{Q}_2 = 2\times(\Delta T \times 2\pi k\ r_2 r_1)/(r_2 - r_1) = \frac{2\times 210\times 2\pi k\times 0.85\times 0.75}{0.10} = 16825.4\ k$$

Therefore, 594.44 = (68531.84 + 16825.4) k; or, k = 6.96 × 10^{-3} W/mK.

Example 1.18 A spherical vessel, made out of2.5 em thick steel plate IS used to store 10m3 of a liquid at 200°C for a thermal storage system. To reduce the heat loss to the surroundings, a 10 cm thick layer of insulation (k = 0.07 W/rnK) is used. If the convective heat transfer coefficient at the outer surface is W/m^{2}K and the ambient temperature is 25°C, calculate the rate of heat loss neglecting the thermal resistance of the steel plate.

If the spherical vessel is replaced by a 2 m diameter cylindrical vessel with flat ends, calculate the thickness of insulation required for the same heat loss.

Solution: Volume of the spherical vessel = $10\text{m}^3 = \dfrac{4\pi r^3}{3} \quad \therefore\ r = 1.336\text{ m}$

Outer radius of the spherical vessle, $r_2 = 1.3364 + 0.025 = 1.361\text{ m}$

Outermost radius of the spherical vessel after the insulation = 1.461 m.

Since the thermal resistance of the steel plate is negligible, the temperature at the inside surface of the insulation is 200°C.

Thermal resistance of the insulating material = $\left(r_3 - r_2\right)/4\pi\ k\ r_3 r_2$

$$= \frac{0.1}{4\pi \times 0.07 \times 1.461 \times 1.361} = 0.057$$

Thermal resistance of the fluid film at the outermost surface = 1/hA

$$= 1/\left[10 \times 4\pi \times \left(1.461\right)^2\right] = 0.00373$$

Rate of heat flow = $\Delta T/\Sigma R = \left(200 - 25\right)/\left(0.057 + 0.00373\right) = 2873.8\text{ W}$

Volume of the insulating material used = $\left(4/3\right)\pi\left(r_3^3 - r_2^3\right) = 2.5\text{ m}^3$

Volume of the cylindrical vessel $= 10\text{ m}^3 = \dfrac{\pi}{4}\left(d\right)^2 L; \quad \therefore L = 10/\pi = 3.183\text{m}$

Outer radius of cylinder without insulation = 1.0 + 0.025 = 1.025 m.

Outermost radius of the cylinder (with insulation) = r_3.

Therefore, the thickness of insulation = r_3 – 1.025 = □

Resistance, the heat flow by the cylindrical element

$$= \frac{\ln\left(r_3/1.025\right)}{2\pi Lk} + 1/hA = \frac{\ln\left(r_3/1.025\right)}{2\pi \times 3.183 \times 0.07} + \frac{1}{10 \times 2\pi \times r_3 \times 3.183}$$

= 0.714 ln (r_3 / 1.025) + 0.005/r_3

Resistance to heat flow through sides of the cylinder

$$= 2\delta / kA + 1/hA = \frac{2(r_3 - 1.025)}{0.07 \times \pi \times 1} + \frac{1}{10 \times 2 \times \pi}$$

$$= 9.09(r_3 - 1.025) + 0.0159$$

For the same heat loss, $\Delta T / \Sigma R$ would be equal in both cases, therefore,

$$\frac{1}{0.06073} = \frac{1}{0.714 \ln(r_3/1.025) + 0.005/r_3} + \frac{1}{9.09(r_3 - 1.025) + 0.0159}$$

Solving by trial and error, (r – 1.025j) = □ = 9.2 cm.

and the volume of the insulating material required = 2.692 m^3.

Concept of Critical Thickness of Insulation

The addition of insulation at the outside surface of small pipes may not reduce the rate of heat transfer. When an insulation is added on the outer surface of a bare pipe, its outer radius, r_0 increases and this increases the thermal resistance due to conduction logarithmically whereas t he thermal resistance to heat flow due to fluid film on the outer surface decreases linearly with increasing radius, r_0. Since the total thermal resistance is proportional to the sum of these two resistances, the rate of heat flow may not decrease as insulation is added to the bare pipe.

Fig. 2.7 shows a plot of heat loss against the insulation radius for two different cases. For small pipes or wires, the radius r_l may be less than re and in that case, addition of insulation to the bare pipe will increase the heat loss until the critical radius is reached. Further addition of insulation will decrease the heat loss rate from this peak value. The insulation thickness ($r^* - r_l$) must be added to reduce the heat loss below the uninsulated rate. If the outer pipe radius r_l is greater than the critical radius re any insulation added will decrease the heat loss.

2.7 Expression for Critical Thickness of Insulation for a Cylindrical Pipe

Let us consider a pipe, outer radius r_l as shown in Fig. 2.18. An insulation is added such that the outermost radius is r a variable and the insulation thickness is ($r - r_I$). We assume that the thermal conductivity, k, for the insulating material is very small in comparison with the thermal conductivity of the pipe material and as such the temperature T_1, at the inside surface of the insulation is constant. It is further assumed that both h and k are constant. The rate of heat flow, per unit length of pipe, through the insulation is then,

$\dot{Q}/L = 2\pi(T_1 - T_\infty)/(\ln(r/r_1)/k + 1/hr)$, where T_∞ is the ambient temperature.

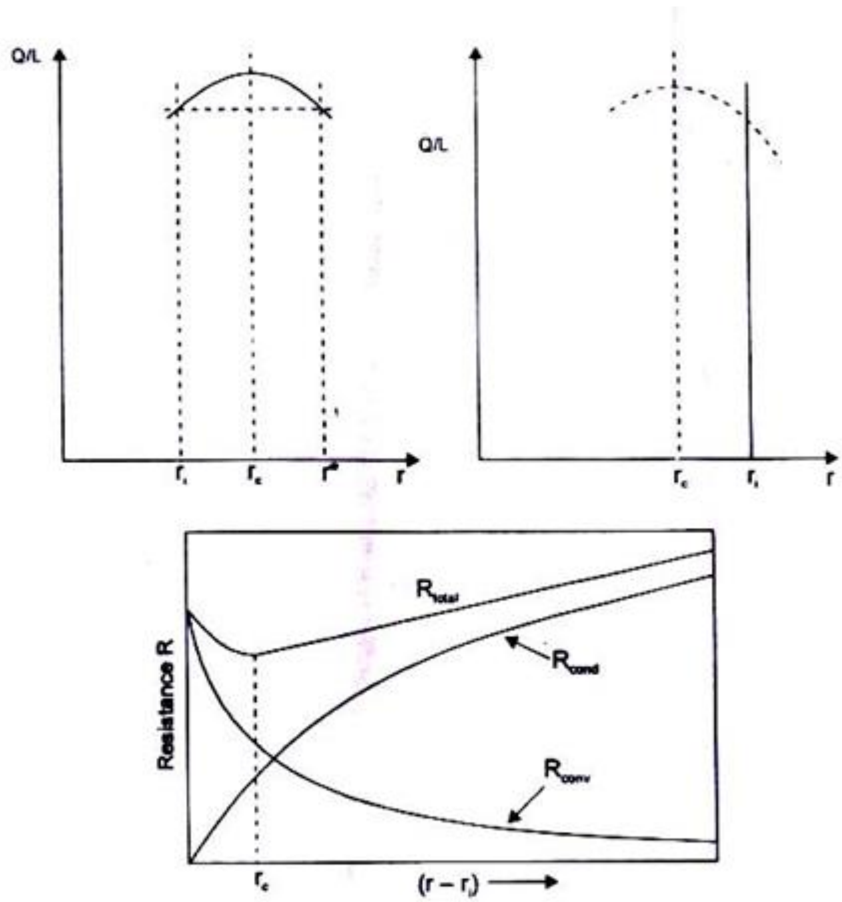


Fig 2.8 Critical thickness for pipe insulation

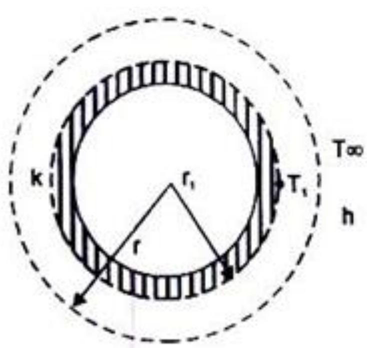


Fig 2.9 critical thickness of insulation for a pipe

An optimum value of the heat loss is found by setting $\frac{d\left(\dot{Q}/L\right)}{dr}=0$.

or, $$\frac{d\left(\dot{Q}/L\right)}{dr}=0=-\frac{2\pi\left(T_1-T_\infty\right)\left(1/kr-1/hr^2\right)}{\left(\ln\left(r/r_1\right)/k+1/hr^2\right)}$$

or, $\left(1/kr\right)-\left(1/hr^2\right)=0$ and $r = r_c = k/h$ (2.21)

where r_c denote the 'critical radius' and depends only on thermal quantities k and h.

If we evaluate the second derivative of (Q/L) at $r = r_c$, we get

$$\left.\frac{d^2(Q/L)}{dr^2}\right|_{r=r_c}=-2\pi\left(T_1-T_\infty\right)\left[\frac{\frac{k}{hr}+\mathrm{In}\left(\frac{r}{r_1}\right)\left(\frac{2k}{hr}-1\right)-2\left(1-\frac{k}{hr}\right)^2}{\frac{1}{kr}\left(\frac{k}{h}+r\,\mathrm{In}\left(\frac{r}{r_1}\right)\right)}\right]_{r=r_c}$$

$$= -\left[2\pi\left(T_1-T_\infty\right)h^2/k\right]/\left[1+1\mathrm{n}\ r_c/r_1\right]^2$$

Which is always a negative quantity. Thus, the optimum radius, $r_c = k/h$ will always give a maximum heal loss and not a minimum.

2.8. An Expression for the Critical Thickness of Insulation for a Spherical Shell

Let us consider a spherical shell having an outer radius r_1 and the temperature at that surface T_1 . Insulation is added such that the outermost radius of the shell is r, a variable. The thermal conductivity of the insulating material, k, and the convective heat transfer coefficient at the outer surface, h, and the ambient temperature T_∞ is constant. The rate of heat transfer through the insulation on the spherical shell is given by

$$\dot{Q}=\frac{\left(T_1-T_\infty\right)}{\left(r-r_1\right)/4\pi k\,r\,r_1+1/h\,4\pi r^2}$$

$$\frac{d\dot{Q}}{dr}=0=\frac{4\pi\left(T_1-T_\infty\right)\left(1/kr^2-2/hr^3\right)}{\left[\left(r-r_1\right)/k\,r\,r_1+1/hr^2\right]^2}$$

which gives, $1/Kr^2 - 2/hr^3 = 0$;

or $\quad r = r_c = 2\ k/h \qquad (2.22)$

2.9 Heat and Mass Transfer

Example 2.5 Hot gases at 175°C flow through a metal pipe (outer diameter 8 cm). The convective heat transfer coefficient at the outside surface of the insulation (k = 0.18 W /mK) IS 2.6 W m1K and the ambient temperature IS 25°C. Calculate the insulation thickness such that the heat loss is less than the uninsulated case.

Solution: (a) Pipe without Insulation

Neglecting the thermal resistance of the pipe wall and due to the inside convective heat transfer coefficient, the temperature of the pipe surface would be 175°C.

$\dot{Q}/L = h \times 2\pi r\ (T_1 - T_\infty) = 2.6 \times 2 \times 3.14 \times .04\{175-25) = 98$ W/m (b) Pipe Insulated. Outermost Radius, r*

$$\dot{Q}/L = 98 = (T_1 - T_\infty) / \left(\frac{1n(r*/4)}{2\pi \times 0.18} + \frac{100}{2.6 \times 2\pi \times r*} \right)$$

or $\frac{150}{98}$ = 08841n (r*/4)+6.12/r*; which gives r* = 13.5 cm.

Therefore, the insulation thickness must be more than 9.5 cm.

(Since the critical thickness of insulation is r_c = k/h = 0.18/2.6 = 6.92 cm, and is greater than the radius of the bare pipe, the required insulation thickness must give a radius greater than the critical radius.)

If the outer radius of the pipe was more than the critical radius, any addition of insulating material will reduce the rate of heat transfer. Let us assume that the outer radius of the pipe is 7 cm ($r > r_c$)

$\dot{Q}/L$, without insulation = hA (ΔT) = 2.6 × 2 ×3.142 × 0.07× (175-25)

= 171.55 W/m

By adding 4 cm thick insulation, outermost radius = 7.0 + 4.0 = 11.0 cm.

and $\dot{Q}/L = (175 - 25)/\left[\dfrac{\ln(11/7)}{2\pi\times0.18}+\dfrac{1}{2.6\pi\times2\times0.11}\right]$=133.58W/m.

Reduction in heat loss = $\dfrac{171.55-133.58}{171.55}$ = 0.22 or 22%.

Example 2.6 An electric conductor 1.5 mm in diameter at a surface temperature of 80°C is being cooled in air at 25°C. The convective heat transfer coefficient from the conductor surface is $16 W/m^2K$. Calculate the surface temperature of the conductor when it is covered with a layer of rubber insulation (2 mm thick, k = 0.15 W /mK) assuming that the conductor carries the same current and the convective heat transfer coefficient is also the same. Also calculate the increase in the current carrying capacity of the conductor when the surface temperature of the conductor remains at 80°C.

Solution: When there is no insulation,

$\dot{Q}/L = hA(\Delta T) = 16 \times 2 \times 3.142 \times 0.75 \times 10^{-3} = 4.147$ W/m

When the insulation is provided, the outermost radius = 0.75 + 2 = 2.75 mm

$\dot{Q}/L = 4.147 = (T_1-25)/\left(\dfrac{\ln 2.75/0.75}{2\pi\times0.15}+\dfrac{1000}{16\times2\pi\times2.75}\right)$

or $T_1 = 45.71^oC$

i.e., the temperature at the outer surface of the wire decreases because the insulation adds a resistance.

The critical radius of insulation, r c = k/h = 0.15/16 = 9.375 mm

i.e., when an insulation of thickness (9.375 - 0.75) = 8.625 mm is added, the heat transfer rate would be the maximum and the conductor can carry more current. The heat transfer rate with outermost radius equal to r_c = 9.375 mm

$\dot{Q}/L = (80 - 25)/\left(\dfrac{\ln 9.375/0.75}{2\pi\times0.15}+\dfrac{1000}{16\times2\pi\times9.375}\right)$=14.7 W/m

The rate of heat transfer is proportional to $(\text{current})^2$, the new current I_2 would be:

$I_2/I_1 = (14.7 / 4.147)^{1/2} = 1.883$

or, the current carrying capacity can be increased 1.883 times. But the maximum current capacity of wire would be limited by the permissible temperature at the centre of the wire.

The surface temperature of the conductor when the outermost radius with insulation is equal to the critical radius, is given by

$$\dot{Q}/L = 4.147 = (T\text{-}25/\left(\frac{\ln 9.375/0.75}{2\times 3.142\times 0.15}+\frac{1000}{16\times 2\times 3.142\times 9.375}\right)$$

or $\quad T = 40.83°C.$

SCHOOL OF MECHANICAL

DEPARTMENT OF MECHANICAL

UNIT – II – Heat Transfer Applied to IC Engines– SAUA1503

UNIT – 2

CONVECTION

2.1. Convection Heat Transfer-Requirements

The heat transfer by convection requires a solid-fluid interface, a temperature difference between the solid surface and the surrounding fluid and a motion of the fluid. The process of heat transfer by convection would occur when there is a movement of macro-particles of the fluid in space from a region of higher temperature to lower temperature.

2.2. Convection Heat Transfer Mechanism

Let us imagine a heated solid surface, say a plane wall at a temperature T_w placed in an atmosphere at temperature T_∞, Fig. 2.1 Since all real fluids are viscous, the fluid particles adjacent to the solid surface will stick to the surface. The fluid particle at A, which is at a lower temperature, will receive heat energy from the plate by conduction. The internal energy of the particle would Increase and when the particle moves away from the solid surface (wall or plate) and collides with another fluid particle at B which is at the ambient temperature, it will transfer a part of its stored energy to B. And, the temperature of the fluid particle at B would increase. This way the heat energy is transferred from the heated plate to the surrounding fluid. Therefore the process of heat transfer by convection involves a combined action of heat conduction, energy storage and transfer of energy by mixing motion of fluid particles.

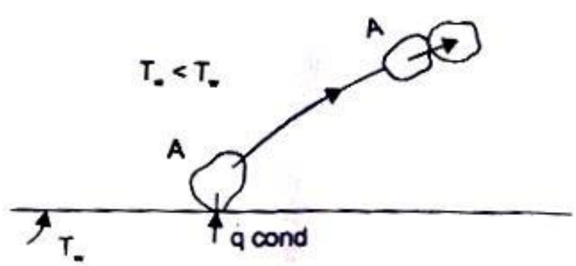


Fig. 2.1 Principle of heat transfer by convection

2.3. Free and Forced Convection

When the mixing motion of the fluid particles is the result of the density difference caused by a temperature gradient, the process of heat transfer is called natural or free convection. When the mixing motion is created by an artificial means (by some external agent), the process of heat transfer is called forced convection Since the effectiveness of heat transfer by convection depends largely on the mixing motion of the fluid particles, it is essential to have a knowledge of the characteristics of fluid flow.

2.4. Basic Difference between Laminar and Turbulent Flow

In laminar or streamline flow, the fluid particles move in layers such that each fluid p article follows a smooth and continuous path. There is no macroscopic mixing of fluid particles between successive layers, and the order is maintained even when there is a turn around a comer or an obstacle is to be crossed. If a lime dependent fluctuating motion is observed indirections which are parallel and transverse to the main flow, i.e., there is a random macroscopic mixing of fluid particles across successive layers of fluid flow, the motion of the fluid is called' turbulent flow'. The path of a fluid particle would then be zigzag and irregular, but on a statistical basis, the overall motion of the macro particles would be regular and predictable.

2.5. Formation of a Boundary Layer

When a fluid flow, over a surface, irrespective of whether the flow is laminar or turbulent, the fluid particles adjacent to the solid surface will always stick to it and their velocity at the solid surface will be zero, because of the viscosity of the fluid. Due to the shearing action of one fluid layer over the adjacent layer moving at the faster rate, there would be a velocity gradient in a direction normal to the flow.

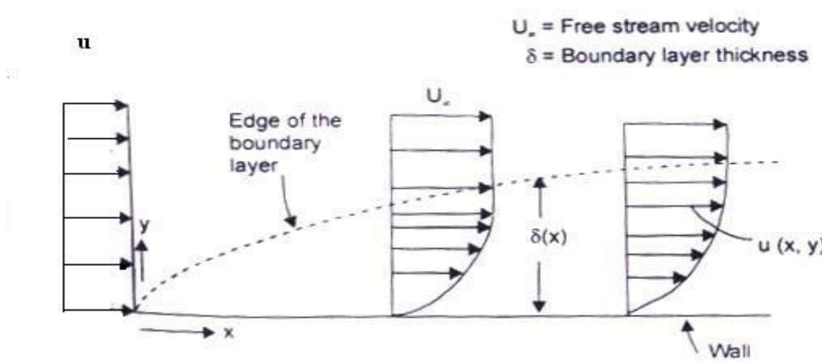


Fig 2.2: sketch of a boundary layer on a wall

Let us consider a two-dimensional flow of a real fluid about a solid (slender in cross-section) as shown in Fig. 2.2. Detailed investigations have revealed that the velocity of the fluid particles at the surface of the solid is zero. The transition from zero velocity at the surface of the solid to the free stream velocity at some distance away from the solid surface in the V-direction (normal to the direction of flow) takes place in a very thin layer called 'momentum or hydrodynamic boundary layer'. The flow field can thus be divided in two regions:

(i) A very thin layer in t he vicinity 0 f t he body w here a velocity gradient normal to the direction of flow exists, the velocity gradient du/dy being large. In this thin region, even a very small Viscosity μ of the fluid exerts a substantial Influence and the shearing stress $\tau = \mu$ du/dy may assume large values. The thickness of the boundary layer is very small and decreases with decreasing viscosity.

(ii) In the remaining region, no such large velocity gradients exist and the Influence of viscosity is unimportant. The flow can be considered frictionless and potential.

2.6. Thermal Boundary Layer

Since the heat transfer by convection involves the motion of fluid particles, we must superimpose the temperature field on the physical motion of fluid and the two fields are bound to interact. It is intuitively evident that the temperature distribution around a hot body in a fluid stream will often have the same character as the velocity distribution in the boundary layer flow. When a heated solid body IS placed in a fluid stream, the temperature of the fluid stream will also vary within a thin layer in the immediate neighborhood of the solid body. The variation in temperature of the fluid stream also takes place in a thin layer in the neighborhood of the body and is termed 'thermal boundary layer'. Fig. 2.3 shows the temperature profiles inside a thermal boundary layer.

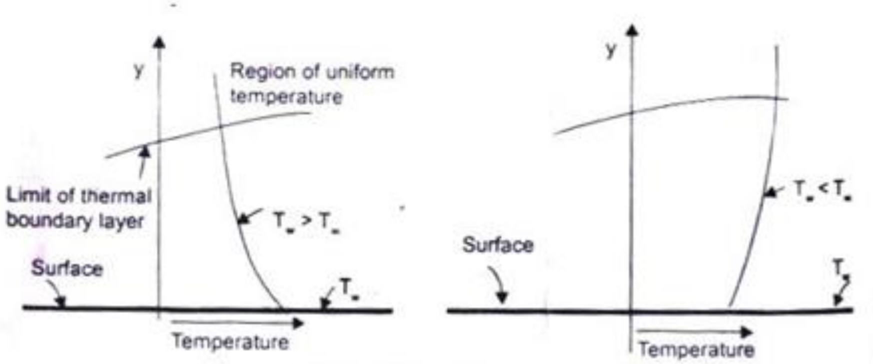


Fig2.3: The thermal boundary layer

2.7. Dimensionless Parameters and their Significance

The following dimensionless parameters are significant in evaluating the convection heat transfer coefficient:

(a) *The Nusselt Number (Nu)*-It is a dimensionless quantity defined as hL/ k, where h = convective heat transfer coefficient, L is the characteristic length and k is the thermal conductivity of the fluid The Nusselt number could be interpreted physically as the ratio of the temperature gradient in the fluid immediately in contact with the surface to a reference temperature gradient (T_s - T_∞) /L. The convective heat transfer coefficient can easily be obtained if the Nusselt number, the thermal conductivity of the fluid in that temperature range and the characteristic dimension of the object is known.

Let us consider a hot flat plate (temperature T_w) placed in a free stream (temperature $T_\infty < T_w$). The temperature distribution is shown ill Fig. 2.4. Newton's Law of Cooling says that the rate of heat transfer per unit area by convection is given by

$$\dot{Q}/A = h\left(T_w - T_\infty\right)$$

$$\frac{\dot{Q}}{A} = h(T_w - T_\infty)$$

$$= k\frac{T_w - T_\infty}{\delta_t}$$

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

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

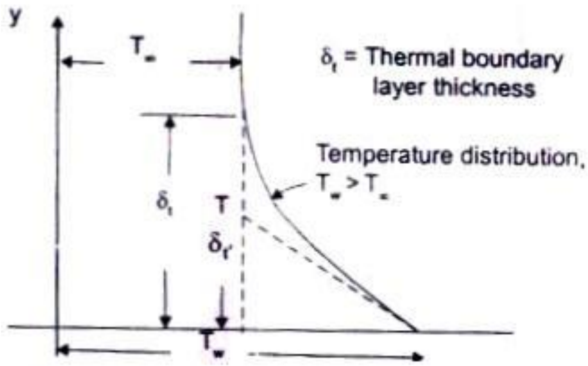


Fig. 2.4 Temperature distribution in a boundary layer: Nusselt modulus

The heat transfer by convection involves conduction and mixing motion of fluid particles. At the solid fluid interface (y = 0), the heat flows by conduction only, and is given by

$$\frac{\dot{Q}}{A} = -k\left(\frac{dT}{dy}\right)_{Y=0} \qquad \therefore h = \frac{\left(-k\, {dT}/{dy}\right)_{y=0}}{\left(T_w - T_\infty\right)}$$

Since the magnitude of the temperature gradient in the fluid will remain the same, irrespective of the reference temperature, we can write dT = d(T - T_w) and by introducing a characteristic length dimension L to indicate the geometry of the object from which the heat flows, we get

$$\frac{hL}{k} = \frac{\left({dT}/{dy}\right)_{y=0}}{\left(T_w - T_\infty\right)/L}$$, and in dimensionless form,

$$= \left(\frac{d(T_w - T)/(T_w - T_\infty)}{d\left(y/L\right)}\right)_{y=0}$$

(b) *The Grashof Number (Gr)*-In natural or free convection heat transfer, die motion of fluid particles is created due to buoyancy effects. The driving force for fluid motion is the body force arising from the temperature gradient. If a body with a constant wall temperature T_w is exposed to a qui scent ambient fluid at T_∞, the force per unit volume can be written as $\rho g \beta\left(t_w - T_\infty\right)$ where ρ = mass density of the fluid, β = volume coefficient of expansion and g is the acceleration due to gravity.

The ratio of inertia force × Buoyancy force/(viscous force)^2 can be written as

$$\mathrm{Gr} = \frac{\left(\rho V^2 L^2\right) \times \rho g \beta \left(T_w - T_\infty\right) L^3}{\left(\mu V L\right)^2}$$

$$= \frac{\rho^2 g \beta \left(T_w - T_\infty\right) L^3}{\mu^2} = g \beta L^3 \left(T_w - T_\infty\right) / \nu^2$$

The magnitude of Grashof number indicates whether the flow is laminar or turbulent. If the Grashof number is greater than 10^9, the flow is turbulent and for Grashof number less than 10^8, the flow is laminar. For $10^8 < \mathrm{Gr} < 10^9$, It is the transition range.

(c) *The Prandtl Number (Pr)* - It is a dimensionless parameter defined as

$\mathrm{Pr} = \mu C_p / k = \nu / \alpha$

Where μ is the dynamic viscosity of the fluid, v = kinematic viscosity and α = thermal diffusivity.

This number assumes significance when both momentum and energy are propagated through the system. It is a physical parameter depending upon the properties of the medium It is a measure of the relative magnitudes of momentum and thermal diffusion in the fluid: That is, for Pr = I, the r ate of diffusion of momentum and energy are equal which means that t he calculated temperature and velocity fields will be Similar, the thickness of the momentum and thermal boundary layers will be equal. For Pr <<I (in case of liquid metals), the thickness of the thermal boundary layer will be much more than the thickness of the momentum boundary layer and vice versa. The product of Grashof and Prandtl number is called Rayleigh number. Or, Ra = Gr × Pr.

2.8. Evaluation of Convective Heat Transfer Coefficient

The convective heat transfer coefficient in free or natural convection can be evaluated by two methods:

(a) Dimensional Analysis combined with experimental investigations

(b) Analytical solution of momentum and energy equations 10 the boundary layer.

Dimensional Analysis and Its Limitations

Since the evaluation of convective heat transfer coefficient is quite complex, it is based on a combination of physical analysis and experimental studies. Experimental observations become necessary to study the influence of pertinent variables on the physical phenomena.

Dimensional analysis is a mathematical technique used in reducing the number of experiments to a minimum by determining an empirical relation connecting the relevant variables and in grouping the variables together in terms of dimensionless numbers. And, the method can only be applied after the pertinent variables controlling t he phenomenon are Identified and expressed In terms of the primary dimensions. (Table 1.1)

In natural convection heat transfer, the pertinent variables are: h, ρ, k, μ, $C_{p,}$ L, (ΔT), β and g. Buckingham π's method provides a systematic technique for arranging the variables in dimensionless numbers. It states that the number of dimensionless groups, π's, required to describe a phenomenon involving 'n' variables is equal to the number of variables minus the number of primary dimensions 'm' in the problem.

In SI system of units, the number of primary dimensions are 4 and the number of variables for free convection heat transfer phenomenon are 9 and therefore, we should expect (9 - 4) = 5 dimensionless numbers. Since the dimension of the coefficient of volume expansion, β, is θ^{-1}, one dimensionless number is obviously $\beta(\Delta T)$. The remaining variables are written in a functional form:

$$\phi\left(h, \rho, k, \mu, C_p, L, g\right) = 0.$$

Since the number of primary dimensions is 4, we arbitrarily choose 4 independent variables as primary variables such that all the four dimensions are represented. The selected primary variables are: ρ,g, k. L Thus the dimensionless group,

$$\pi_1 = \rho^a g^b k^c L^d h = \left(ML^{-3}\right)^a \left(LT^{-2}\right)^b . \left(MLT^{-3}\theta^{-1}\right) = M^0 L^0 T^0 \theta^0$$

Equating the powers of M, L, T, θ on both sides, we have

$$\left.\begin{array}{l} M: a + c + 1 = 0 \text{ \} Upon solving them,} \\ L: -3a + b + c + d = 0 \\ T: -2b -3c -3 = 0 \\ \theta: -c - 1 = 0 \end{array}\right\}\text{Up on solving them,}$$

c = 1, b = a = 0 and d = 1.

and π_1 = hL/k, the Nusselt number.

The other dimensionless number

$\pi_2 = p^a g^b k^c L^d C_p = (ML^{-3})^a (LT^{-2})^b (MLT^{-3}\theta^{-1})^c (L)^d (MT^{-1}\theta^{-1}) = M^0L^0T^0\theta^0$ Equating the powers of M,L,T and θ and upon solving, we get

$\pi_3 = \mu^2/\rho^2 g L^3$

By combining π_2 and π_3, we write $\pi_4 = [\pi_2 \times \pi_3]^{1/2}$

$= \left[\rho^2 g L^3 C_p^2 / k^2 \times \mu^2 / gL^3\right]^{1/2} = \dfrac{\mu C_p}{k}$ (the Prandtl number)

By combining π_3 with $(\beta\Delta T)$, we have $\pi_5 = (\beta\Delta T) * \dfrac{1}{\pi_3}$

$= \beta(\Delta T) \times \dfrac{\rho^2 g L^3}{\mu^2} = g\beta(\Delta T)L^3/\nu^2$ (the Grashof number)

Therefore, the functional relationship is expressed as:

$$\phi(\text{Nu}, \text{Pr}, \text{Gr}) = 0; \text{Or}, \text{Nu} = \phi_1(\text{Gr}\,\text{Pr}) = \text{Const} \times (\text{Gr} \times \text{Pr})^m \qquad (2.1)$$

and values of the constant and 'm' are determined experimentally.

Table 2.1 gives the values of constants for use with Eq. (2.1) for isothermal surfaces.

Table 2.1 Constants for use with Eq. 2.1 for Isothermal Surfaces

Geometry	$G_{r_f}\ p_{r_f}$	*C*	*m*

Vertical planes and cylinders	$10^4 - 10^9$	0.59	1/4
	$10^9 - 10^{13}$	0.021	2/5
	$10^9 - 10^{13}$	0.10	1/3
Horizontal cylinders	$0 - 10^{-5}$	0.4	0
	$10^4 - 10^9$	0.53	1/4
	$10^9 - 10^{12}$	0.13	1/3
	$10^{10} - 10^{-2}$	0.675	0.058
	$10^{-2} - 10^2$	1.02	0.148
	$10^2 - 10^4$	0.85	0.188
	$10^4 - 10^7$	0.48	1/4
	$10^7 - 10^{12}$	0.125	1/3
Upper surface of heated plates or lower surface of cooled plates	$8 \times 10^6 - 10^{11}$	0.15	1/3
- do -	$2 \times 10^4 - 8 \times 10^6$	0.54	1/4
Lower surface of heated plates or upper surface of cooled plates	$10^5 - 10^{11}$	0.27	1/4
Vertical cylinder height = diameter characteristic length = diameter	$10^4 - 10^6$	0.775	0.21
Irregular solids, characteristic length = distance the fluid particle travels in boundary layer	$10^4 - 10^9$	0.52	1/4

Analytical Solution-Flow over a Heated Vertical Plate in Air

Let us consider a heated vertical plate in air, shown in Fig. 2.5. The plate is maintained at uniform temperature T_w .The coordinates are chosen in such a way that x - is in the stream wise direction and y - is in the transverse direction. There will be a thin layer of fluid adjacent to the hot surface of the vertical plate within

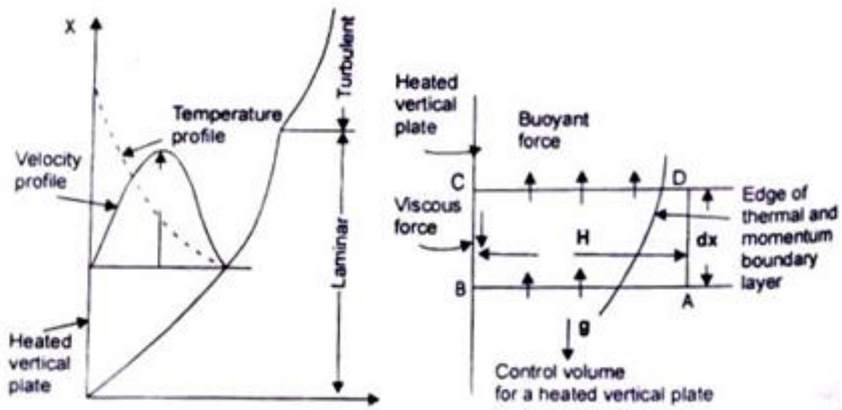


Fig. 2.5 Boundary layer on a heated vertical plate

Which the variations in velocity and temperature would remain confined. The relative thickness of the momentum and the thermal boundary layer strongly depends upon the Prandtl number. Since in natural convection heat transfer, the motion of the fluid particles is caused by the temperature difference between the temperatures of the wall and the ambient fluid, the thickness of the two boundary layers are expected to be equal. When the temperature of the vertical plate is less than the fluid temperature, the boundary layer will form from top to bottom but the mathematical analysis will remain the same.

The boundary layer will remain laminar upto a certain length of the plate ($Gr < 10^8$) and beyond which it will become turbulent ($Gr > 10^9$). In order to obtain the analytical solution, the integral approach, suggested by von-Karman is preferred.

We choose a control volume ABCD, having a height H, length dx and unit thickness normal to the plane of paper, as shown in Fig. 25. We have:

(b) Conservation of Mass:

Mass of fluid entering through face AB = $\dot{m}_{AB} = \int_0^H \rho u dy$

Mass of fluid leaving face CD = $\dot{m}_{CD} = \int_0^H \rho u dy + \frac{d}{dx}\left[\int_0^H \rho u dy\right] dx$

$\therefore$ Mass of fluid entering the face DA = $\frac{d}{dx}\left[\int_0^H \rho u dy\right]dx$

(ii) Conservation of Momentum:

Momentum entering face AB = $\int_0^H \rho u^2 dy$

Momentum leaving face CD = $\int_0^H \rho u^2 dy + \frac{d}{dx}\left[\int_0^H \rho u^2 dy\right]dx$

$\therefore$ Net efflux of momentum in the + x-direction = $\frac{d}{dx}\left[\int_0^H \rho u^2 dy\right]dx$

The external forces acting on the control volume are:

(a) Viscous force = $\mu \left.\frac{du}{dy}\right|_{y=0} dx$ acting in the –ve x-direction

(b) Buoyant force approximated as $\left[\int_0^H \rho g \beta (T - T_\infty) dy\right]dx$

From Newton's law, the equation of motion can be written as:

$$\frac{d}{dx}\left[\int_0^\delta \rho u^2 dy\right] = -\mu \left.\frac{du}{dy}\right|_{y=0} + \int_0^\delta \rho g \beta (T - T_\infty) dy \qquad (2.2)$$

because the value of the integrand between δ and H would be zero.

(iii) Conservation of Energy:

$\dot{Q}_{AB}$, convection + $\dot{Q}_{AD}$, convection + $\dot{Q}_{BC}$, conduction = $\dot{Q}_{CD}$ convection

$$\text{or, } \int_0^H \rho u C T dy + C T_\infty \left[\frac{d}{dx}\int_0^H \rho u dy\right]dx - k\left.\frac{dT}{dy}\right|_{y=0} dx$$

$$= \int_0^H \rho u C T dy + \frac{d}{dx}\left[\int_0^H \rho u T C dy\right]dx$$

$$\text{or } \frac{d}{dx}\int_0^\delta \rho u (T_\infty - T) dy \frac{k}{\rho C}\left.\frac{dT}{dy}\right|_{y=0} = \alpha \left.\frac{dT}{dy}\right|_{y=0} \qquad (2.3)$$

The boundary conditions are:

or,

(2.3)

Velocity profile	Temperature profile
$u = 0$ at $y = 0$	$T = T_w$ at $y = 0$
$u = 0$ at $y = \delta$	$T = T_\infty$ at $y = \delta_1 \equiv \delta$
$du/dy = 0$ at $y = \delta$	$dT/dy \equiv 0$ at $y = \delta_1 \equiv \delta$

Since the equations (2.2) and (2.3) are coupled equations, it is essential that the functional form of both the velocity and temperature distribution are known in order to arrive at a solution.

The functional relationships for velocity and temperature profiles which satisfy the above boundary conditions are assumed of the form:

$$\frac{u}{u_*} = \frac{y}{\delta}\left(1 - \frac{y}{\delta}\right)^2 \tag{2.4}$$

Where u_* is a fictitious velocity which is a function of x; and

$$\frac{(T - T_\infty)}{(T_w - T_\infty)} = \left(1 - \frac{y}{\delta}\right)^2 \tag{2.5}$$

After the Eqs. (5.4) and (5.5) are inserted in Eqs. (5.2) and (5.3) and the operations are performed (details of the solution are given in Chapman, A.J. Heat Transfer, Macmillan Company, New York), we get the expression for boundary layer thickness as:

$$\delta / x = 3.93 \mathrm{Pr}^{-0.5} (0.952 + \mathrm{Pr})^{0.25} \mathrm{Gr}_x^{-0.25}$$

Where Gr, is the local Grashof number = $g\beta x^3 (T_w - T_\infty)/\nu^2$

The heat transfer coefficient can be evaluated from:

$$\dot{q}_w = -k\left.\frac{dT}{dy}\right|_{y=0} = h\left(T_w - T_\infty\right)$$

Using Eq. (5.5) which gives the temperature distribution, we have

$h = 2k/\delta$ or, $hx/k = Nu_x = 2x/\delta$

The non-dimensional equation for the heat transfer coefficient is

$$Nu_x = 0.508\ Pr^{0.5}\,(0.952 + Pr)^{-0.25}\ Gr_x^{0.25} \qquad (2.7)$$

The average heat transfer coefficient, $\bar{h} = \frac{1}{L}\int_0^L h_x dx = 4/3 h_{x=L}$

$$Nu_L = 0.677\ Pr^{0.5}\,(0.952 + Pr)^{-0.25}\ Gr^{0.25} \qquad (2.8)$$

Limitations of Analytical Solution: Except for the analytical solution for flow over a flat plate, experimental measurements are required to evaluate the heat transfer coefficient. Since in free convection systems, the velocity at the surface of the wall and at the edge of the boundary layer is zero and its magnitude within the boundary layer is so small. It is very difficult to measure them. Therefore, velocity measurements require hydrogen-bubble technique or sensitive hot wire anemometers. The temperature field measurement is obtained by interferometer.

Expression for 'h' for a Heated Vertical Cylinder in Air

The characteristic length used in evaluating the Nusselt number and Grashof number for vertical surfaces is the height of the surface. If the boundary layer thickness is not to large compared with the diameter of the cylinder, the convective heat transfer coefficient can be evaluated by the equation used for vertical plane surfaces. That is, when $D/L \geq 25/\left(Gr_L\right)^{0.25}$

Example 2.1 A large vertical flat plate 3 m high and 2 m wide is maintained at 75°C and is exposed to atmosphere at 25°C. Calculate the rate of heat transfer.

Solution: The physical properties of air are evaluated at the mean temperature. i.e. T = (75 + 25)/2 = 50°C

From the data book, the values are:

$\rho = 1.088\ kg/m^3$; $\qquad$ $C_p = 1.00$ kJ/kg.K;

$\mu = 1.96 \times 10^{-5}$ Pa-s   k = 0.028 W/mK.

Pr = $\mu C_p/k = 1.96 \times 10^{-5} \times 1.0 \times 10^{3} /0.028 = 0.7$

$$\beta = \frac{1}{T} = \frac{1}{323}$$

Gr = $\rho^2 g\beta(\Delta T)L^3/\mu^2$

$$= \frac{(1.088)^2 \times 9.81 \times 1 \times (3)^3 \times 50}{323 \times (1.96 \times 10^{-5})^2}$$

$= 12.62 \times 10^{10}$

Gr.Pr = 8.834×10^{10}

Since Gr.Pr lies between 10^9 and 10^{13}

We have from Table 2.1

$$Nu = \frac{hL}{k} = 0.1(Gr.Pr)^{1/3} = 441.64$$

$\therefore$ h = 441.64 × 0.028/3 = 4.122 W/m^{2}K

$$\dot{Q} = hA(\Delta T) = 4.122 \times 6 \times 50 = 1236.6 W$$

We can also compute the boundary layer thickness at x = 3m

$$\delta = \frac{2x}{Nu_x} = \frac{2 \times 3}{441.64} = 1.4 \text{ cm}$$

Example 2.2 A vertical flat plate at 90°C. 0.6 m long and 0.3 m wide, rests in air at 30°C. Estimate the rate of heat transfer from the plate. If the plate is immersed in water at 30°C. Calculate the rate of heat transfer

Solution: (a) *Plate in Air*: Properties of air at mean temperature 60°C

Pr = 0.7, k = 0.02864 W/ mK, v = 19.036×10^{-6} m^2/s

Gr = $9.81 \times (90 - 30)(0.6)^3/ [333 (19.036 \times 10^{-6})^2]$

$= 1.054 \times 10^9$; Gr × Pr $1.054 \times 10^9 \times 0.7 = 7.37 \times 10^8 < 10^9$

From Table 5.1: for $Gr \times Pr < 10^9$, $Nu = 0.59\ (Gr.\ Pr)^{1/4}$

$\therefore\ h = 0.02864 \times 0.59\ (7.37 \times 10^8)^{1/4}/0.6 = 4.64\ W/m^2K$

The boundary layer thickness, $\delta = 2\ k/h = 2 \times 0.02864/4.64 = 1.23$ cm

and $\dot{Q} = hA\ (\Delta T) = 4.64 \times (2 \times 0.6 \times 0.3) \times 60 = 100$ W.

Using Eq (2.8). $Nu = 0.677\ (0.7)^{0.5}\ (0.952 + 0.7)^{0.25}\ (1.054 \times 10^9)^{0.25}$,

Which gives $h = 4.297\ W/m^2K$ and heat transfer rate, $\dot{Q}$ 92.81 W

Churchill and Chu have demonstrated that the following relations fit very well with experimental data for all Prandtl numbers.

For $Ra_L < 10^9$, $Nu = 0.68 + (0.67\ Ra_L^{0.25})/\ [1 + (0.492/Pr)^{9/16}]^{4/9)}$ (5.9)

$Ra_L > 10^9$, $Nu = 0.825 + (0.387\ Ra_L^{1/6})/[1 + (0.492/Pr)^{9/16}]^{8/27}$ (5.10)

Using Eq (5.9): $Nu = 0.68 + [0.67(7.37 \times 10^8)^{0.25}]\ /\ [1 + (0.492/0.7)^{9/16}]^{4/9}$

$= 58.277$ and $h = 4.07\ W/m^2k$; $\dot{Q} = 87.9$ W

(b) Plate in Water: Properties of water from the Table

$Pr = 3.01$, $\rho^2\ g\ \beta\ C_p/\mu k = 6.48 \times 10^{10}$;

$Gr.Pr = \rho^2\ g\ \beta\ C_p\ L^3(\Delta T)/\ \mu k = 6.48 \times 10^{10} \times (0.6)^3 \times 60 = 8.4 \times 10^{11}$

Using Eq (5.10): $Nu = 0.825 + [0.387\ (8.4 \times 10^{11})^{1/6}]/\ [1 + (0.492/3.01)^{9/16})]^{8/27} = 89.48$ which gives h = 97.533 and Q = 2.107 kW.

2.9. Modified Grashof Number

When a surface is being heated by an external source like solar radiation incident on a wall, a surface heated by an electric heater or a wall near a furnace, there is a uniform heat flux distribution along the surface. The wall surface will not be an isothermal one. Extensive experiments have been performed by many research workers for free convection from vertical and inclined surfaces to water under constant heat flux conditions. Since the temperature difference (ΔT) is not known beforehand, the Grashof number is modified by multiplying it by Nusselt number. That is,

$$Gr_x^* = Gr_x . Nu_x = (g\ \beta\ \Delta T / \nu^2) \times (hx/k) = g\ \beta\ x^4\ q/k\,\nu^2 \qquad (2.11)$$

Where q is the wall heat flux in Wm^2. ($q = h\,(\Delta T)$)

It has been observed that the boundary layer remains lam mar when the modified Rayleigh number, $Ra^* = Gr_x^*$'. Pr is less than 3×10^{12} and fully turbulent flow appears for $Ra^* > 10^{14}$. The local heat transfer coefficient can be calculated from:

$$q \text{ constant and } 10^5 < Gr_x^* < 10^{11}: Nu_x = 0.60\,(Gr_x^* . Pr)^{0.2} \qquad (2.12)$$

$$q \text{ constant and } 2 \times 10^{13} < Gr_x^* < 10^{16}: Nu_x = 0.17\,(Gr_x^* . Pr)^{0.25} \qquad (2.13)$$

Although these results are based on experiments for water, they are applicable to air as well. The physical properties are to be evaluated at the local film temperature.

Example 2.3 Solar radiation of intensity 700W/m' is incident on a vertical wall, 3 m high and 3 m wide. Assuming that the wall does not transfer energy to the inside surface and all the incident energy is lost by free convection to the ambient air at 30oe, calculate the average temperature of the wall

Solution: Since the surface temperature of the wall is not known, we assume a value for $h = 7\ W/m^2 K$.

$\Delta T = \dot{q} / h = 700/7 = 100^oC$ and the film temperature = $(30 + 130)/2 = 80^oC$

The properties of air at 273 +80 = 353 are: $\beta = 1/353$, Pr = 0.697

$k = 0.03\ W/mK$, $\nu = 20.76 \times 10^{-6}\ m^2/s$.

Modified Grashof number, $Gr_x^* = 9.81. (1/353) \cdot (3)^4 \times 700/[0.03 \times (20.76 \times 10^{-6})^2] = 1.15 \times 10^{14}$

From Eq. (2.13), $h = (k/x)\,(0.17)\,(Gr_x^*\ Pr)^{0.25}$

$= (0.03/3)\,(0.17)\,(1.15 \times 10^{14} \times 0.697)^{1/4}$

$= 5.087\ W/m^2K$, a different value from the assumed value.

Second Trial: $\Delta T = \dot{q} / h = 700/5.087 = 137.66$ and film temperature

= 98.8°C

The properties of air at (273 + 98.8) °C are: β =1/372, k = 0.0318 W/mK

Pr = 0.693, v = 23.3 × 10^{-6} m^2/s

Gr_x^* = 9.81. (1/372)· $(3)^4$ × 700/ [0.318(23.3 × $10^{-6})^2$] = 8.6 × 10^{13}

Using Eq (2.13), h = (k/x) (0.17) (Gr_x^* Pr$)^{1/4}$ = 5.015 W/m^2k, an acceptable value. In turbulent heat transfer by convection, Eq. (5.13) tells us that the local heat transfer coefficient h_x does not vary with x and therefore, the average and local heat transfer coefficients are the same.

2 Laminar Flow Forced Convection Heat Transfer

2.1Forced Convection Heat Transfer Principles

The mechanism of heat transfer by convection requires mixing of one portion of fluid with another portion due to gross movement of the mass of the fluid. The transfer of heat energy from one fluid particle or a molecule to another one is still by conduction but the energy is transported from one point in space to another by the displacement of fluid.

When the motion of fluid is created by the imposition of external forces in the form of pressure differences, the process of heat transfer is called ‘forced convection’. And, the motion of fluid particles may be either laminar or turbulent and that depends upon the relative magnitude of inertia and viscous forces, determined by the dimensionless parameter Reynolds number. In free convection, the velocity of fluid particle is very small in comparison with the velocity of fluid particles in forced convection, whether laminar or turbulent. In forced convection heat transfer, $Gr/Re^2 << 1$, in free convection heat transfer, $GrRe^2 >> 1$ and we have combined free and forced convection when $Gr/Re^2 \approx 1$.

2.2. Methods for Determining Heat Transfer Coefficient

The convective heat transfer coefficient in forced flow can be evaluated by: (a) Dimensional Analysis combined with experiments;

(b) Reynolds Analogy – an analogy between heat and momentum transfer; (c) Analytical Methods – exact and approximate analyses of boundary layer equations.

2.3. Method of Dimensional Analysis

As pointed out in Chapter 5, dimensional analysis does not yield equations which can be solved. It simply combines the pertinent variables into non-dimensional numbers which facilitate the interpretation and extend the range of application of experimental data. The relevant variables for forced convection heat transfer phenomenon whether laminar or turbulent, are

(b) The properties of the fluid – density p, specific heat capacity C_p, dynamic or absolute viscosity μ, thermal conductivity k.

(ii) The properties of flow – flow velocity Y, and the characteristic dimension of the system L.

As such, the convective heat transfer coefficient, h, is written as h = f (ρ, V, L, μ, Cp, k) = 0 (5.14)

Since there are seven variables and four primary dimensions, we would expect three dimensionless numbers. As before, we choose four independent or core variables as ρ,V, L, k, and calculate the dimensionless numbers by applying Buckingham π's method:

$$\pi_1 = \rho^a V^b L^c K^d h = \left(ML^{-3}\right)^a \left(LT^{-1}\right)^b (L)^c \left(MLT^{-3}\theta^{-1}\right)^d \left(MT^{-3}\theta^{-1}\right)$$

$$= M^o L^o T^o \theta^o$$

Equating the powers of M, L, T and θ on both sides, we get

M : a + d + 1 = O

L : - 3a + b + c + d = 0

T : - b – 3d – 3 = 0 By solving them, we have

θ : - d – 1= 0. D = -1, a = 0, b = 0, c = 1.

Therefore, π_1 = hL/k is the Nusselt number.

$$\pi_2 = \rho^a V^b L^c K^d \mu = \left(ML^{-3}\right)^a \left(LT^{-1}\right)^b (L)^c \left(MLT^{-3}\theta^{-1}\right)^d \left(ML^{-1}T^{-1}\right)$$

$$= M^o L^o T^o \theta^o$$

Equating the powers of M, L, T and on both sides, we get

M : a + d +1 = 0

L : - 3a + b + c + d = 1 = 0

T : - b – 3d – 1 = 0

θ : - d = 0.

By solving them, d = 0, b = - 1, a = - 1, c = - 1

and $\pi_2 = \mu / \rho VL; \text{or}, \pi_3 = \frac{1}{\pi_2} = \frac{\rho VL}{\mu}$

(Reynolds number is a flow parameter of greatest significance. It is the ratio of inertia forces to viscous forces and is of prime importance to ascertain the conditions under which a flow is laminar or turbulent. It also compares one flow with another provided the corresponding length and velocities are comparable in two flows. There would be a similarity in flow between two flows when the Reynolds numbers are equal and the geometrical similarities are taken into consideration.)

$$\pi_4 = \rho^a V^b L^c k^d C_p = \left(ML^{-3}\right)^a \left(LT^{-1}\right)^b \left(L\right)^c \left(MLT^{-3}\theta^{-1}\right)^d \left(L^2T^{-2}\theta^{-1}\right)$$

$M^o L^o T^o \theta^o$

Equating the powers of M, L, T, on both Sides, we get

M : a + d = 0; L : - 3a + b + c + d + 2 = 0

T : - b – 3d – 2 = 0; θ : - d – 1 = 0

By solving them,

d =- 1,a = l, b = l, c = l,

$$\pi_4 = \frac{\rho VL}{k} C_p; \quad \pi_5 = \pi_4 \times \pi_2$$

$$= \frac{\rho VL}{k} C_p \times \frac{\mu}{\rho VL} = \frac{\mu C_p}{k}$$

$\therefore$ π_5 is Prandtl number.

Therefore, the functional relationship is expressed as:

Nu = f (Re, Pr); or Nu = C Re^m Pr^n (5.15)

Where the values of c, m and n are determined experimentally.

2.4. Principles of Reynolds Analogy

Reynolds was the first person to observe that there exists a similarity between the exchange of momentum and the exchange of heat energy in laminar motion and for that reason it has been termed 'Reynolds analogy'. Let us consider the motion of a fluid where the fluid is flowing over a plane wall. The X-coordinate is measured parallel to the surface and the V-coordinate is measured normal to it. Since all fluids are real and viscous, there would be a thin layer, called momentum boundary layer, in the vicinity of the wall where a velocity gradient normal to the direction of flow exists. When the temperature of the surface of the wall is different than the temperature of the fluid stream, there would also be a thin layer, called thermal boundary layer, where there is a variation in temperature normal to the direction of flow. Fig. 2.6 depicts the velocity distribution and temperature profile for the laminar motion of the fluid flowing past a plane wall.

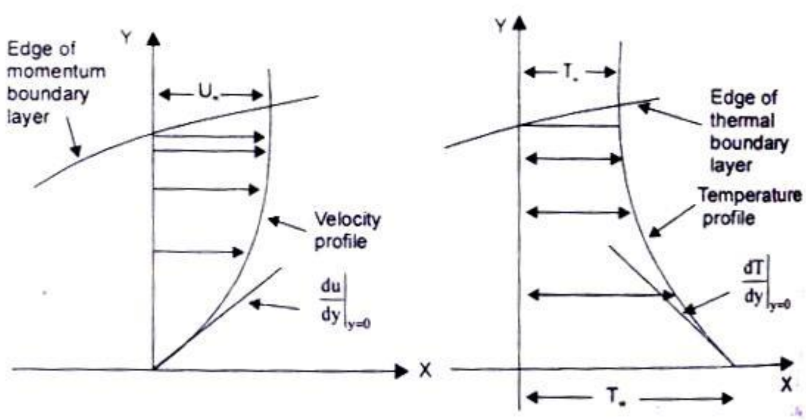


Fig. 2.6 velocity distribution and temperature profile for laminar motion of the fluid over a plane surface

In a two-dimensional flow, the shearing stress is given by $\tau_w = \mu \left.\frac{du}{dy}\right|_{y=0}$

and the rate of heat transfer per unit area is given by $\frac{\dot{Q}}{A} = \frac{\tau_w k}{\mu}\frac{dT}{du}$

For $Pr = \mu C_p/k = 1$, we have $k/\mu = C_p$ and therefore, we can write after separating the variables,

$$\frac{\dot{Q}}{A\tau_w C_p} du = -dT \tag{5.16}$$

Assuming that Q and τ_w are constant at any station x, we integrate equation (5.16) between the limits: u = 0 when $T = T_w$, and $u = U_\infty$ when $T = T_\infty$, and we get,

$$\dot{Q}/\left(A\tau_w C_p\right) \times U_\infty = \left(T_w - T_\infty\right)$$

Since by definition, $\dot{Q}/A = h_x\left(T_w - T_\infty\right)$, and $\tau_w = C_{fx} \times \rho U_\infty{}^2/2$,

Where C_{fx}, is the skin friction coefficient at the station x. We have

$$C_{fx}/2 = h_x/\left(C_p \rho U_\infty\right) \tag{5.17}$$

Since $h_x/C_p\rho U_\infty = \left(h_x.x/k\right) \times \left(\mu/\rho \times U_\infty\right) \times \left(k/\mu.C_p\right) = Nu_x/\left(Re.Pr\right)$,

$$Nu_x/Re.Pr = C_{fx}/2 = \text{Stantonnumer}, St. \tag{5.18}$$

Equation (5.18) is satisfactory for gases in which Pr is approximately equal to unity. Colburn has shown that Eq. (5.18) can also be used for fluids having Prandtl numbers ranging from 0.6 to about 50 if it is modified in accordance with experirnental results.

$$\text{Or, } \frac{Nu_x}{Re_x\, Pr}.Pr^{2/3} = St_x\, Pr^{2/3} = C_{fx}/2 \tag{5.19}$$

Eq. (5.19) expresses the relation between fluid friction and heat transfer for laminar flow over a plane wall. The heat transfer coefficient could thus be determined by making measurements of the frictional drag on a plate under conditions in which no heat transfer is involved.

Example 2.4 Glycerine at 35°C flows over a 30cm by 3Ocm flat plate at a velocity of 1.25 m/s. The drag force is measured as 9.8 N (both Side of the plate). Calculate the heat transfer for such a flow system.

Solution: From tables, the properties of glycerine at 35°C are:

ρ = 1256 kg/m^3, C_p = 2.5 kJ/kgK, μ = 0.28 kg/m-s, k = 0.286 W/mK, Pr = 2.4 Re= $\rho VL/\mu$ = 1256 × 1.25 × 0.30/0.28 = 1682.14, a laminar flow.*

Average shear stress on one side of the plate = drag force/area

= 9.8/(2 × 0.3 × 0.3) = 54.4

and shear stress = $C_f \rho U^2/2$

$\therefore$ The average skin friction coefficient, Cr/ 2 = $\dfrac{\tau}{\rho U^2}$

= 54.4/(1256 ×1.25 × 1.25) = 0.0277

From Reynolds analogy, $C_f/2 = St.\ Pr^{2/3}$

or, h = $\rho\, C_p\, U \times C_f/2 \times Pr^{-2/3} = \dfrac{1256 \times 2.5 \times 1.25 \times 0.0277}{(2.45)^{0.667}}$ =59.8 kW/m^{2}K.

2.5. Analytical Evaluation of ‘h’ for Laminar Flow over a Flat Plat – Assumptions

As pointed out earlier, when the motion of the fluid is caused by the imposition of external forces, such as pressure differences, and the fluid flows over a solid surface, at a temperature different from the temperature of the fluid, the mechanism of heat transfer is called ‘forced convection’. Therefore, any analytical approach to determine the convective heat transfer coefficient would require the temperature distribution in the flow field surrounding the body. That is, the theoretical analysis would require the use of the equation of motion of the viscous fluid flowing over the body along with the application of the principles of conservation of mass and energy in order to relate the heat energy that is convected away by the fluid from the solid surface.

For the sake of simplicity, we will consider the motion of the fluid in 2 space dimension, and a steady flow. Further, the fluid properties like viscosity, density, specific heat, etc are constant in the flow field, the viscous shear forces m the Y –direction is negligible and there are no variations in pressure also in the Y –direction.

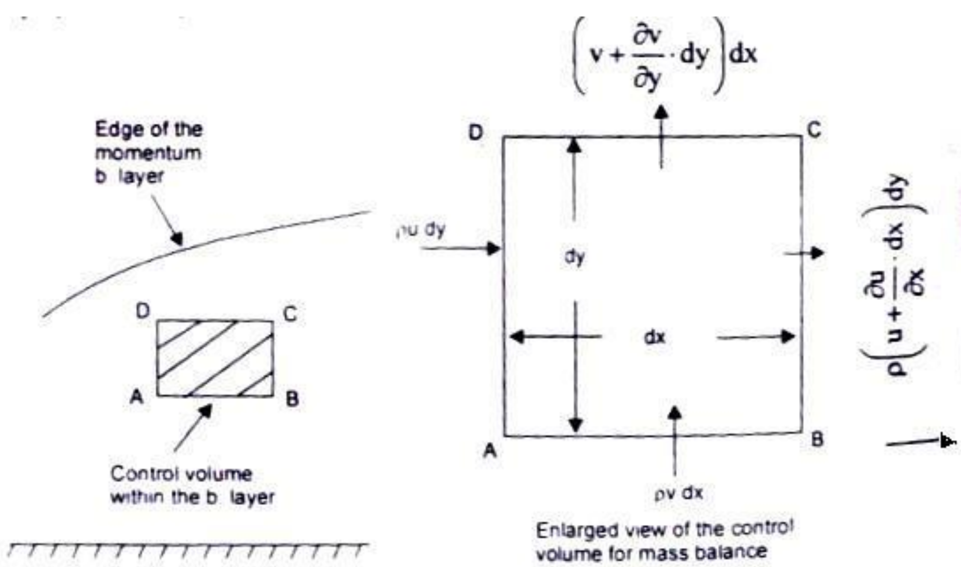
2.6. Derivation of the Equation of Continuity–Conservation of Mass

We choose a control volume within the laminar boundary layer as shown in Fig. 6.2. The mass will enter the control volume from the left and bottom face and will leave the control volume from the right and top face. As such, for unit depth in the Z-direction,

$$\dot{m}_{AD} = \rho u dy \; ; \quad \dot{m}_{BC} = \rho\left(u + \frac{\partial u}{dx}.dx \right)dy;$$

$$\dot{m}_{AB} = \rho v dx \; ; \quad \dot{m}_{CD} = \rho\left(v + \frac{\partial u}{dy}.dy \right)dx;$$

For steady flow conditions, the net efflux of mass from the control volume is zero, therefore,



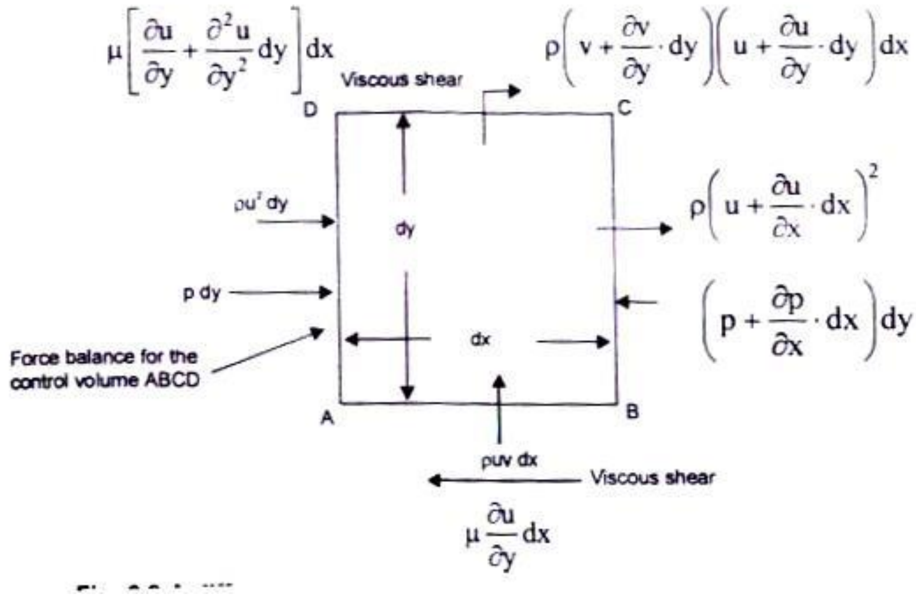


Fig. 2.7 a differential control volume within the boundary layer for laminar flow over a plane wall

$$\rho udy + \rho xdx = \rho udy + \rho\frac{\partial u}{\partial x}dxdy + \rho vdx + \rho\frac{\partial v}{\partial x}.dxdy$$

or, $\partial u/\partial x + \partial v + \partial y = 0,$ the equation of continuity. (2.20)

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SCHOOL OF MECHANICAL

DEPARTMENT OF MECHANICAL

UNIT – III – Heat Transfer Applied to IC Engines – SAUA1503

UNIT III

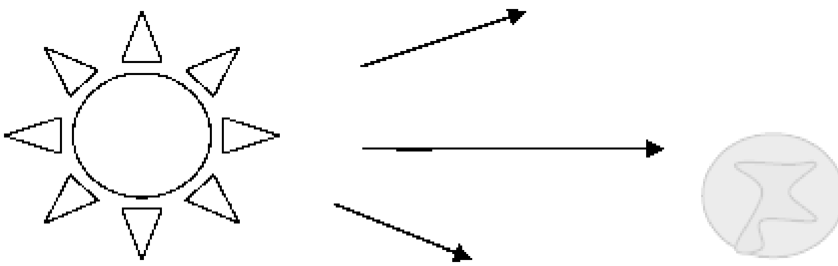
RADIATION HEAT TRANSFER

3.1RADIATION

Definition:

Radiation is the energy transfer across a system boundary due to a ΔT, by the mechanism of photon emission or electromagnetic wave emission.

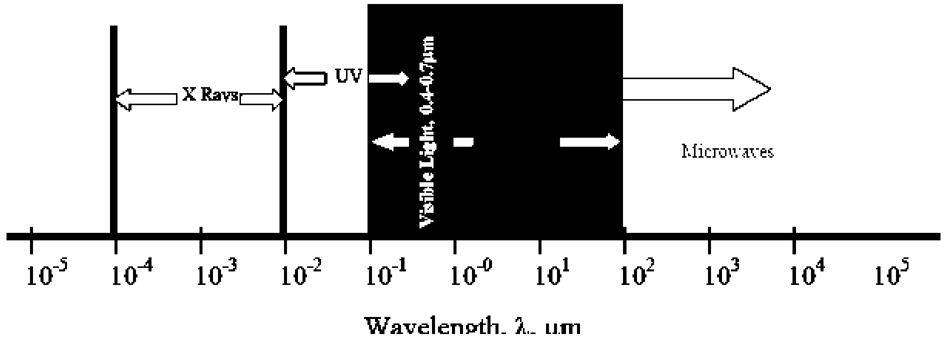
Because the mechanism of transmission is photon emission, unlike conduction and convection, there need be no intermediate matter to enable transmission.



The significance of this is that radiation will be the only mechanism for heat transfer whenever a vacuum is present.

3.2 Electromagnetic Phenomena.

We are well acquainted with a wide range of electromagnetic phenomena in modern life. These phenomena are sometimes thought of as wave phenomena and are, consequently, often described in terms of electromagnetic wave length, λ. Examples are given in terms of the wave distribution shown below:



One aspect of electromagnetic radiation is that the related topics are more closely associated with optics and electronics than with those normally found in mechanical engineering courses. Nevertheless, these are widely encountered topics and the student is familiar with them through every day life experiences.

From a viewpoint of previously studied topics students, particularly those with a background in mechanical or chemical engineering will find the subject of Radiation Heat Transfer a little unusual. The physics background differs fundamentally from that found in the areas of continuum mechanics. Much of the related material is found in courses more closely identified with quantum physics or electrical engineering, i.e. Fields and Waves. At this point, it is important for us to recognize that since the subject arises from a different area of physics, it will be important that we study these concepts with extra care.

3.3Stefan-Boltzman Law

Both Stefan and Boltzman were physicists; any student taking a course in quantum physics will become well acquainted with Boltzman's work as he made a number of important contributions to the field. Both were contemporaries of Einstein so we see that the subject is of fairly recent vintage. (Recall that the basic equation for convection heat transfer is attributed to Newton)

$$E_b = \sigma \cdot T_{abs}^{\ 4}$$

where: E_b = Emissive Power, the gross energy emitted from an ideal surface per unit area, time.

σ = Stefan Boltzman constant, $5.67 \cdot 10^{-8}$ W/m$^2 \cdot$K^4

T_{abs} = Absolute temperature of the emitting surface, K.

Take particular note of the fact that absolute temperatures are used in Radiation. It is suggested, as a matter of good practice, to convert all temperatures to the absolute scale as an initial step in all radiation problems.

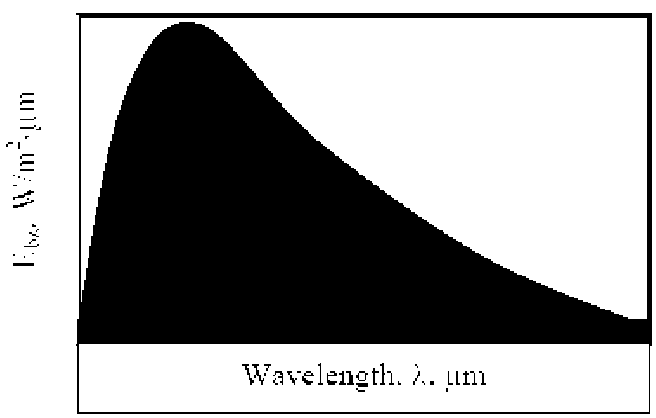
You will notice that the equation does not include any heat flux term, q". Instead we have a term the emissive power. The relationship between these terms is as follows. Consider two infinite plane surfaces, both facing one another. Both surfaces are ideal surfaces. One surface is found to be at temperature, T1, the other at temperature, T2. Since both temperatures are at temperatures above absolute zero, both will radiate energy as described by the Stefan-Boltzman law. The heat flux will be the net radiant flow as given by:

$$q'' = E_{b1} - E_{b2} = \sigma \cdot T_1^4 - \sigma \cdot T_2^4$$

3.4Plank's Law

While the Stefan-Boltzman law is useful for studying overall energy emissions, it does not allow us to treat those interactions, which deal specifically with wavelength, λ. This problem was overcome by another of the modern physicists, Max Plank, who developed a relationship for wave-based emissions.

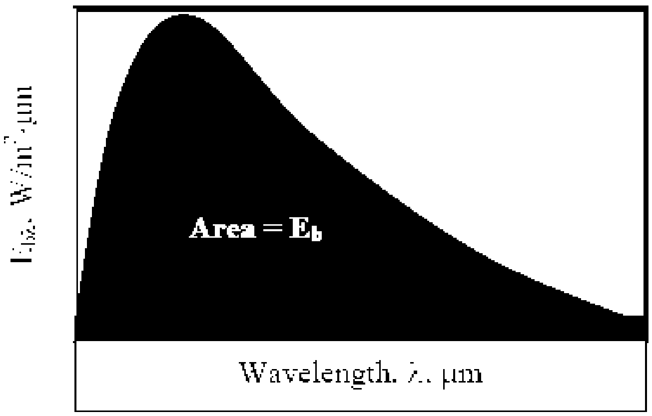
$$E_{b\lambda} = f(\lambda)$$



We haven't yet defined the Monochromatic Emissive Power, $E_{b\lambda}$. An implicit definition is provided by the following equation:

$$E_b = \int_0^\infty E_{b\lambda} \cdot d\lambda$$

We may view this equation graphically as follows:



A definition of monochromatic Emissive Power would be obtained by differentiating the integral equation:

$$E_{b\lambda} \equiv \frac{dE_b}{d\lambda}$$

The actual form of Plank's law is:

$$E_{b\lambda} = \frac{C_1}{\lambda^5 \cdot \left[e^{C_2/\lambda \cdot T} - 1\right]}$$

$$C_1 = 2 \cdot \pi \cdot h \cdot c_o^2 = 3.742 \cdot 10^8 \ W \cdot \mu m^4/m^2$$
$$C_2 = h \cdot c_o/k = 1.439 \cdot 10^4 \ \mu m \cdot K$$

Where: h, c_o, k are all parameters from quantum physics. We need not worry about their precise definition here.

This equation may be solved at any T, λ to give the value of the monochromatic emissivity at that condition. Alternatively, the function may be substituted into the integral $E_b = \int_0^\infty E_{b\lambda} \cdot d\lambda$ to find the Emissive power for any temperature. While performing this integral by hand is difficult, students may readily evaluate the integral through one of several computer programs, i.e. MathCad, Maple, Mathmatica, etc.

$$E_b = \int_0^\infty E_{b\lambda} \cdot d\lambda = \sigma \cdot T^4$$

3.5 Emission over Specific Wave Length Bands

Consider the problem of designing a tanning machine. As a part of the machine, we will need to design a very powerful incandescent light source. We may wish to know how much energy is being emitted over the

Ultraviolet band (10^{-4} to 0.4 μm), known to be particularly dangerous.

$$E_b(0.0001 \rightarrow 0.4) = \int_{0.001 \cdot \mu m}^{0.4 \cdot \mu m} E_{b\lambda} \cdot d\lambda$$

With a computer available, evaluation of this integral is rather trivial. Alternatively, the text books provide a table of integrals. The format used is as follows:

$$\frac{E_b(0.001 \rightarrow 0.4)}{E_b} = \frac{\int_{0.001 \cdot \mu m}^{0.4 \cdot \mu m} E_{b\lambda} \cdot d\lambda}{\int_0^{\infty} E_{b\lambda} \cdot d\lambda} = \frac{\int_0^{0.4 \cdot \mu m} E_{b\lambda} \cdot d\lambda}{\int_0^{\infty} E_{b\lambda} \cdot d\lambda} - \frac{\int_{0.}^{0.0001 \cdot \mu m} E_{b\lambda} \cdot d\lambda}{\int_0^{\infty} E_{b\lambda} \cdot d\lambda} = F(0 \rightarrow 0.4) - F(0 \rightarrow 0.0001)$$

Referring to such tables, we see the last two functions listed in the second column. In the first column is a parameter, λ·T. This is found by taking the product of the absolute temperature of the emitting surface, T, and the upper limit wave length, λ. In our example, suppose that the incandescent bulb is designed to operate at a temperature of 2000K. Reading from the table:

λ,

λ, μm	*T, K*	*λ·T, μm·K*	*F(0→λ)*
0.0001	2000	0.2	0
0.4	2000	600	0.000014
F(0.4→0.0001μm) = F(0→0.4μm)- F(0→0.0001μm)			**0.000014**

This is the fraction of the total energy emitted which falls within the IR band. To find the absolute energy emitted multiply this value times the total energy emitted:

$$E_{bIR} = F(0.4 \rightarrow 0.0001 \mu m) \cdot \sigma \cdot T^4 = 0.000014 \cdot 5.67 \cdot 10^{-8} \cdot 2000^4 = \mathbf{12.7\ W/m^2}$$

3.6 Solar Radiation

The magnitude of the energy leaving the Sun varies with time and is closely associated with such factors as solar flares and sunspots. Nevertheless, we often choose to work with an average value. The energy leaving the sun is emitted outward in all directions so that at any particular distance from the Sun we may imagine the energy being dispersed over an imaginary spherical area. Because this area increases with the distance squared, the solar flux also decreases with the distance squared. At the average distance between Earth and Sun this heat flux is 1353 W/m2, so that the average heat flux on any object in Earth orbit is found as:

$$G_{s,o} = S_c \cdot f \cdot \cos\theta$$

Where S_c = Solar Constant, 1353 W/m^2

f = correction factor for eccentricity in Earth Orbit, ($0.97<f<1.03$)

θ = Angle of surface from normal to Sun.

Because of reflection and absorption in the Earth's atmosphere, this number is significantly reduced at ground level. Nevertheless, this value gives us some opportunity to estimate the potential for using solar energy, such as in photovoltaic cells.

Some Definitions

In the previous section we introduced the Stefan-Boltzman Equation to describe radiation from an ideal surface.

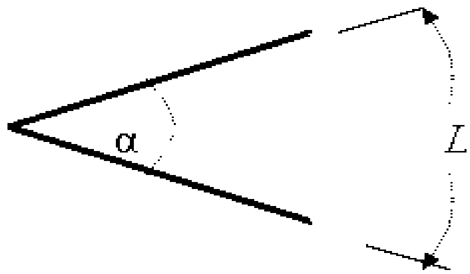
$$E_b = \sigma \cdot T_{abs}^{4}$$

This equation provides a method of determining the total energy leaving a surface, but gives no indication of the direction in which it travels. As we continue our study, we will want to be able to calculate how heat is distributed among various objects.

For this purpose, we will introduce the radiation intensity, I, defined as the energy emitted per unit area, per unit time, per unit solid angle. Before writing an equation for this new property, we will need to define some of the terms we will be using.

3.7 Angles and Arc Length

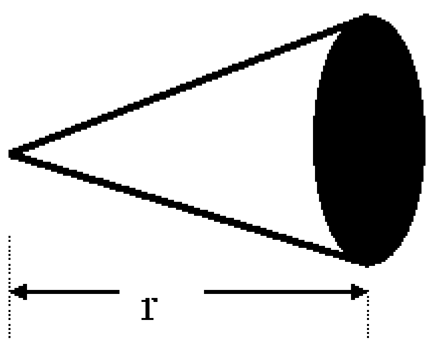
We are well accustomed to thinking of an angle as a two dimensional object. It may be used to find an arc length:



$$L = r \cdot \alpha$$

Solid Angle

We generalize the idea of an angle and an arc length to three dimensions and define a solid angle, Ω, which like the standard angle has no dimensions. The solid angle, when multiplied by the radius squared will have dimensions of length squared, or area, and will have the magnitude of the encompassed area.

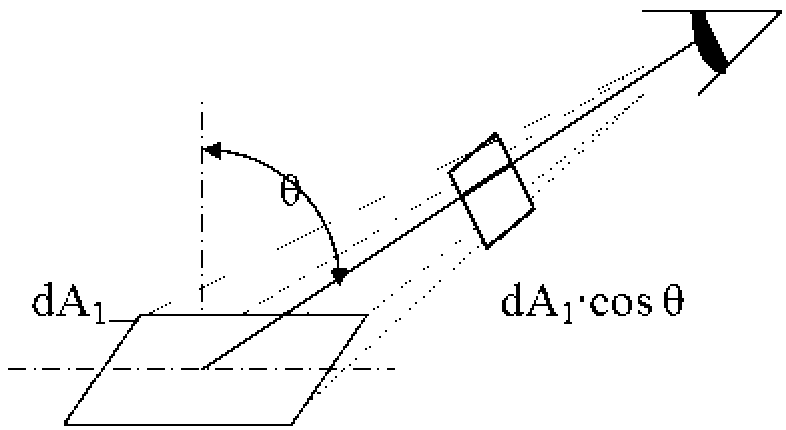


$$A = r^2 \cdot d\Omega$$

3.8 Projected Area

The area, dA1, as seen from the prospective of a viewer, situated at an angle θ from the normal to the surface, will appear somewhat smaller, as cos θ·dA1. This smaller area is termed the projected area.

$$A_{projected} = \cos\theta \cdot A_{normal}$$



3.9 Intensity

The ideal intensity, Ib, may now be defined as the energy emitted from an ideal body, per unit projected area, per unit time, per unit solid angle.

$$I = \frac{dq}{\cos\theta \cdot dA_1 \cdot d\Omega}$$

3.10 Spherical Geometry

Since any surface will emit radiation outward in all directions above the surface, the spherical coordinate system provides a convenient tool for analysis. The three basic coordinates shown are R, φ, and θ, representing the radial, azimuthal and zenith directions.

In general dA1 will correspond to the emitting surface or the source. The surface dA2 will correspond to the receiving surface or the target. Note that the area proscribed on the hemisphere, dA2, may be written as:

$$dA_2 = [(R \cdot \sin\theta) \cdot d\varphi] \cdot [R \cdot d\theta]$$

or, more simply as:

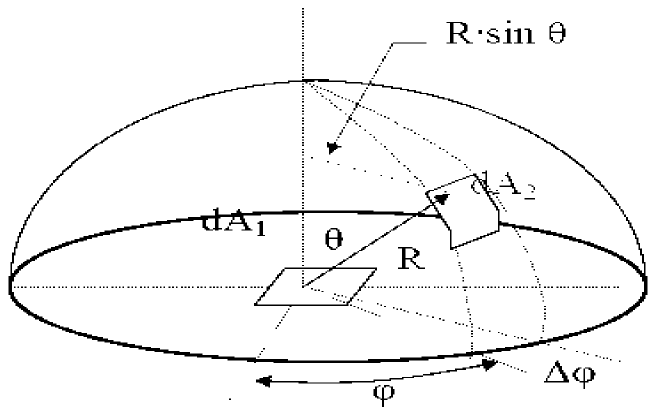
$$dA_2 = R^2 \cdot \sin\theta \cdot d\varphi \cdot d\theta]$$

Recalling the definition of the solid angle,

$dA = R^2 \cdot d\Omega$

we find that:

$d\Omega = R^2 \sin\theta \cdot d\theta \cdot d\varphi$



3.11 Real Surfaces

Thus far we have spoken of ideal surfaces, i.e. those that emit energy according to the Stefan-Boltzman law:

$$E_b = \sigma \cdot T_{abs}^{4}$$

Real surfaces have emissive powers, E, which are somewhat less than that obtained theoretically by Boltzman. To account for this reduction, we introduce the emissivity, ε.

$$\varepsilon \equiv \frac{E}{E_b}$$

so that the emissive power from any real surface is given by:

$$E = \varepsilon \cdot \sigma \cdot T_{abs}^{4}$$

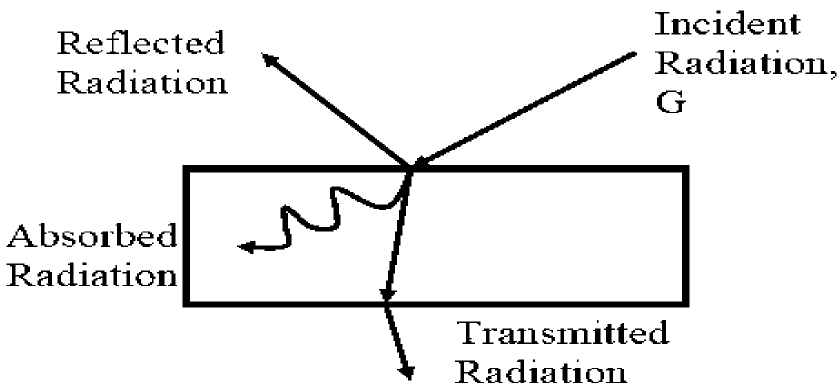
Receiving Properties

Targets receive radiation in one of three ways; they absorption, reflection or transmission. To account for these characteristics, we introduce three additional properties:

• Absorptivity, α, the fraction of incident radiation absorbed.

• Reflectivity, ρ, the fraction of incident radiation reflected.

• Transmissivity, τ, the fraction of incident radiation transmitted.



We see, from Conservation of Energy, that:

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

In this course, we will deal with only opaque surfaces, $\tau = 0$, so that:

$\alpha + \rho = 1$ Opaque Surfaces

3.12 Relationship Between Absorptivity, α, and Emissivity, ε

Consider two flat, infinite planes, surface A and surface B, both emitting radiation toward one another. Surface B is assumed to be an ideal emitter, i.e. $\varepsilon_B = 1.0$. Surface A will emit radiation according to the Stefan-Boltzman law as:

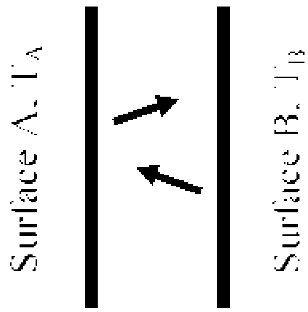
$$E_A = \varepsilon_A \cdot \sigma \cdot T_A^4$$

and will receive radiation as:

$$G_A = \alpha_A \cdot \sigma \cdot T_B^4$$

The net heat flow from surface A will be:

$$q'' = \varepsilon_A \cdot \sigma \cdot T_A^4 - \alpha_A \cdot \sigma \cdot T_B^4$$



Now suppose that the two surfaces are at exactly the same temperature. The heat flow must be zero according to the 2nd law. If follows then that:

$\alpha_A = \varepsilon_A$

Because of this close relation between emissivity, ε, and absorptivity, α, only one property is normally measured and this value may be used alternatively for either property.

Let's not lose sight of the fact that, as thermodynamic properties of the material, α and ε may depend on temperature. In general, this will be the case as radiative properties will depend on wavelength, λ. The wave length of radiation will, in turn, depend on the temperature of the source of radiation. The emissivity, ε, of surface A will depend on the material of which surface A is composed, i.e. aluminum, brass, steel, etc. and on the temperature of surface A. The absorptivity, α, of surface A will depend on the material of which surface A is composed, i.e. aluminum, brass, steel, etc. and on the temperature of surface B.

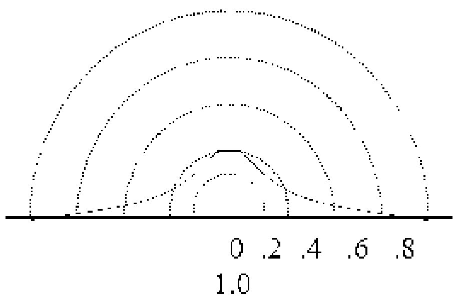
In the design of solar collectors, engineers have long sought a material which would absorb all solar radiation, ($\alpha = 1$, Tsun ~ 5600K) but would not re-radiate energy as it came to temperature ($\varepsilon << 1$, $T_{collector}$~ 400K). NASA developed an anodized chrome, commonly called "black chrome" as a result of this research.

3.13 Black Surfaces

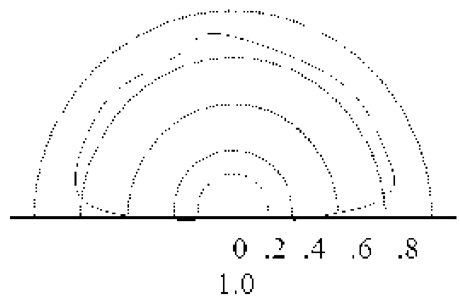
Within the visual band of radiation, any material, which absorbs all visible light, appears as black. Extending this concept to the much broader thermal band, we speak of surfaces with $\alpha = 1$ as also being "black" or "thermally black". It follows that for such a surface, $\varepsilon = 1$ and the surface will behave as an ideal emitter. The terms ideal surface and black surface are used interchangeably.

3.14 Lambert's Cosine Law:

A surface is said to obey Lambert's cosine law if the intensity, I, is uniform in all directions. This is an idealization of real surfaces as seen by the emissivity at different zenith angles:



Dependence of Emissivity on Zenith Angle. Typical Metal.



Dependence of Emissivity on Zenith Angle. Typical Non-Metal.

The sketches shown are intended to show is that metals typically have a very low emissivity, ε, which also remain nearly constant, expect at very high zenith angles, θ. Conversely, non-metals will have a relatively high emissivity, ε, except at very high zenith angles. Treating the emissivity as a constant over all angles is

Generally a good approximation and greatly simplifies engineering calculations.

3.15 Relationship between Emissive Power and Intensity

By definition of the two terms, emissive power for an ideal surface, Eb, and intensity for an ideal surface, I_b

$$E_b = \int_{hemisphere} I_b \cdot \cos\theta \cdot d\Omega$$

Replacing the solid angle by its equivalent in spherical angles:

$$E_b = \int_0^{2\cdot\pi} \int_0^{\pi/2} I_b \cdot \cos\theta \cdot \sin\theta \cdot d\theta \cdot d\varphi$$

Integrate once, holding I_b constant:

$$E_b = 2\cdot\pi\cdot I_b \cdot \int_0^{\pi/2} \cos\theta \cdot \sin\theta \cdot d\theta$$

Integrate a second time. (Note that the derivative of sin θ is cos θ·dθ.)

$$E_b = 2 \cdot \pi \cdot I_b \cdot \left. \frac{\sin^2 \theta}{2} \right|_0^{\pi/2} = \pi \cdot I_b$$

$$E_b = \pi \cdot I_b$$

3.16 Radiation Exchange

During the previous lecture we introduced the intensity, I, to describe radiation within a particular solid angle.

$$I = \frac{dq}{\cos \theta \cdot dA_1 \cdot d\Omega}$$

This will now be used to determine the fraction of radiation leaving a given surface and striking a second surface.

Rearranging the above equation to express the heat radiated:

$$dq = I \cdot \cos \theta \cdot dA_1 \cdot d\Omega$$

Next we will project the receiving surface onto the hemisphere surrounding the source. First find the projected area of surface dA_2, $dA_2 \cdot \cos \theta_2$. (θ_2 is the angle between the normal to surface 2 and the position vector, R.) Then find the solid angle, Ω, which encompasses this area.

Substituting into the heat flow equation above:

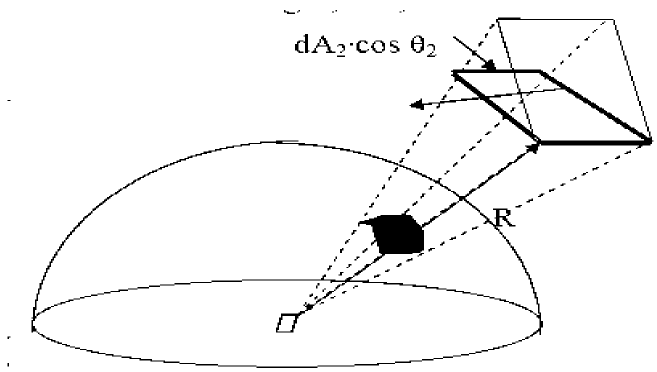
$$dq = \frac{I \cdot \cos \theta_1 \cdot dA_1 \cdot \cos \theta_2 dA_2}{R^2}$$

To obtain the entire heat transferred from a finite area, dA1, to a finite area, dA, we integrate over both surfaces:

$$q_{1 \to 2} = \int_{A_2} \int_{A_1} \frac{I \cdot \cos \theta_1 \cdot dA_1 \cdot \cos \theta_2 dA_2}{R^2}$$

To express the total energy emitted from surface 1, we recall the relation between emissive power, E, and intensity, I.

$q_{emitted} = E_1 \cdot A_1 = \pi \cdot I_1 \cdot A_1$



3.17 View Factors-Integral Method

Define the view factor, $F_{1\text{-}2}$, as the fraction of energy emitted from surface 1, which directly strikes surface 2.

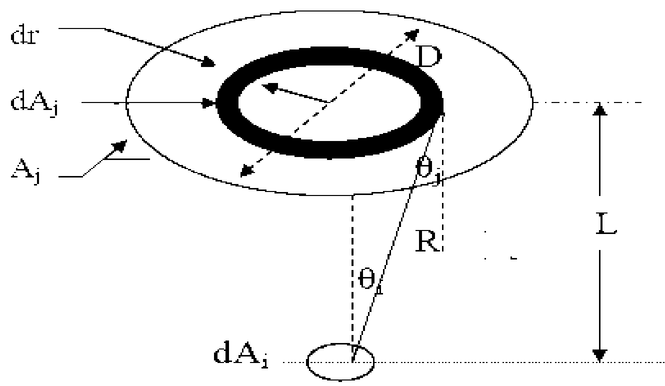
$$F_{1\to 2} = \frac{q_{1\to 2}}{q_{emitted}} = \frac{\displaystyle\int_{A_2}\int_{A_1} \frac{I\cdot\cos\theta_1\cdot dA_1\cdot\cos\theta_2 dA_2}{R^2}}{\pi\cdot I\cdot A_1}$$

after algebraic simplification this becomes:

$$F_{1\to 2} = \frac{1}{A_1}\cdot\int_{A_2}\int_{A_1}\frac{\cos\theta_1\cdot\cos\theta_2\cdot dA_1\cdot dA_2}{\pi\cdot R^2}$$

Example 3.1 Consider a diffuse circular disk of diameter D and area Aj and a plane diffuse surface of area A

i << Aj. The surfaces are parallel, and Ai is located at a distance L from the center of Aj. Obtain an expression for the view factor F_{ij}



The view factor may be obtained from:

$$F_{1\to 2} = \frac{1}{A_1} \cdot \int_{A_2}\int_{A_1} \frac{\cos\theta_1 \cdot \cos\theta_2 \cdot dA_1 \cdot dA_2}{\pi \cdot R^2}$$

Since dA_i is a differential area

$$F_{1\to 2} = \int_{A_1} \frac{\cos\theta_1 \cdot \cos\theta_2 \cdot dA_1}{\pi \cdot R^2}$$

Substituting for the cosines and the differential area:

$$F_{1\to 2} = \int_{A_1} \frac{\left(L/R\right)^2 \cdot 2\pi \cdot r \cdot dr}{\pi \cdot R^2}$$

After simplifying:

$$F_{1\to 2} = \int_{A_1} \frac{L^2 \cdot 2 \cdot r \cdot dr}{R^4}$$

Let $\rho^2 \equiv L^2 + r^2 = R^2$. Then $2\cdot\rho\cdot d\rho = 2\cdot r\cdot dr$.

$$F_{1\to 2} = \int_{A_1} \frac{L^2 \cdot 2 \cdot \rho \cdot d\rho}{\rho^4}$$

After integrating,

$$F_{1\rightarrow 2} = -2 \cdot L^2 \cdot \left.\frac{\rho^{-2}}{2}\right|_{A_2} = -L^2 \cdot \left[\frac{1}{L^2 + \rho^2}\right]_0^{D/2}$$

Substituting the upper & lower limits

$$F_{1\rightarrow 2} = -L^2 \cdot \left[\frac{4}{4 \cdot L^2 + D^2} - \frac{1}{L^2}\right]_0^{D/2} = \frac{D^2}{4 \cdot L^2 + D^2}$$

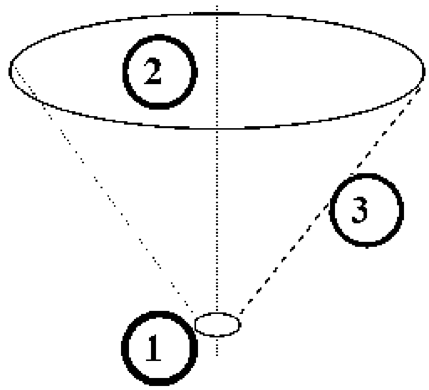
This is but one example of how the view factor may be evaluated using the integral method. The approach used here is conceptually quite straight forward; evaluating the integrals and algebraically simplifying the resulting equations can be quite lengthy.

Enclosures

In order that we might apply conservation of energy to the radiation process, we must account for all energy leaving a surface. We imagine that the surrounding surfaces act as an enclosure about the heat source which receives all emitted energy. Should there be an opening in this enclosure through which energy might be lost, we place an imaginary surface across this opening to intercept this portion of the emitted energy. For an N surfaced enclosure, we can then see that:

$\sum_{j=1}^{N} F_{i,j} = 1$ This relationship is known as "Conservation Rule".

Example: Consider the previous problem of a small disk radiating to a larger disk placed directly above at a distance L.



The view factor was shown to be given by the relationship:

$$F_{1\to2} = \frac{D^2}{4\cdot L^2 + D^2}$$

Here, in order to provide an enclosure, we will define an imaginary surface 3, a truncated cone intersecting circles 1 and 2.

From our conservation rule we have:

$$\sum_{j=1}^{N} F_{i,j} = F_{1,1} + F_{1,2} + F_{1,3}$$

Since surface 1 is not convex F1,1 = 0. Then:

$$F_{1\to3} = 1 - \frac{D^2}{4\cdot L^2 + D^2}$$

3.18 Reciprocity

We may write the view factor from surface i to surface j as:

$$A_i \cdot F_{i\to j} = \int_{A_j}\int_{A_i} \frac{\cos\theta_i \cdot \cos\theta_j \cdot dA_i \cdot dA_j}{\pi \cdot R^2}$$

Similarly, between surfaces j and i:

$$A_j \cdot F_{j\to i} = \int_{A_j}\int_{A_i} \frac{\cos\theta_j \cdot \cos\theta_i \cdot dA_j \cdot dA_i}{\pi \cdot R^2}$$

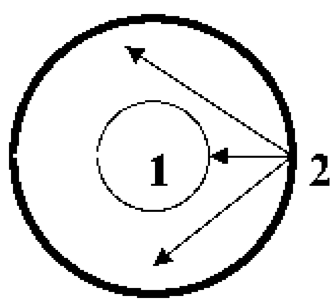
Comparing the integrals we see that they are identical so that:

$$A_i \cdot F_{i\to j} = A_j \cdot F_{j\to i}$$

This relation is known as “Reciprocity”.

Example:4.2 Consider two concentric spheres shown to the right. All radiation leaving the outside of surface 1 will strike surface 2. Part of the radiant energy leaving the inside surface of object 2 will strike surface 1, part will return to surface 2. To find the fraction of energy leaving surface 2 which strikes surface 1, we apply reciprocity:

$$A_2 \cdot F_{2,1} = A_1 \cdot F_{1,2} \Rightarrow F_{2,1} = \frac{A_1}{A_2} \cdot F_{1,2} = \frac{A_1}{A_2} = \frac{D_1}{D_2}$$



3.19 Associative Rule

Consider the set of surfaces shown to the right: Clearly, from conservation of energy, the fraction of energy leaving surface i and striking the combined surface j+k will equal the fraction of energy emitted from i and striking j plus the fraction leaving surface i and striking k.

$$F_{i\Rightarrow(j+k)} = F_{i\Rightarrow j} + F_{i\Rightarrow k}$$

3.20 Radiosity

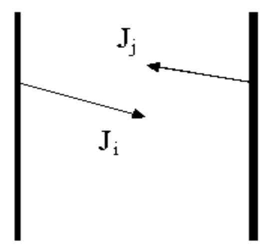
We have developed the concept of intensity, I, which let to the concept of the view factor. We have discussed various methods of finding view factors. There remains one additional concept to introduce before we can consider the solution of radiation problems.

$$J \equiv \varepsilon \cdot E_b + \rho \cdot G$$

Radiosity, J, is defined as the total energy leaving a surface per unit area and per unit time. This may initially sound much like the definition of emissive power, but the sketch below will help to clarify the concept.

3.21 Net Exchange Between Surfaces

Consider the two surfaces shown. Radiation will travel from surface i to surface j and will also travel from j to i.



$$q_{i \to j} = J_i \cdot A_i \cdot F_{i \to j}$$

likewise,

$$q_{j \to i} = J_j \cdot A_j \cdot F_{j \to j}$$

The net heat transfer is then:

$$q_{j \to i \ (net)} = J_i \cdot A_i \cdot F_{i \to j} - J_j \cdot A_j \cdot F_{j \to j}$$

From reciprocity we note that $F_{1 \to 2} \cdot A_1 = F_{2 \to 1} \cdot A_2$ so that

$$q_{j \to i \ (net)} = J_i \cdot A_i \cdot F_{i \to j} - J_j \cdot A_i \cdot F_{i \to j} = A_i \cdot F_{i \to j} \cdot (J_i - J_j)$$

3.22 Net Energy Leaving a Surface

The net energy leaving a surface will be the difference between the energy leaving a surface and the energy received by a surface:

$$q_{1\rightarrow} = [\varepsilon \cdot E_b - \alpha \cdot G] \cdot A_1$$

Combine this relationship with the definition of Radiosity to eliminate G.

$$J \equiv \varepsilon \cdot E_b + \rho \cdot G \Rightarrow G = [J - \varepsilon \cdot E_b]/\rho$$

$$q_{1\rightarrow} = \{\varepsilon \cdot E_b - \alpha \cdot [J - \varepsilon \cdot E_b]/\rho\} \cdot A_1$$

Assume opaque surfaces so that $\alpha + \rho = 1 \Rightarrow \rho = 1 - \alpha$, and substitute for ρ.

$$q_{1\rightarrow} = \{\varepsilon \cdot E_b - \alpha \cdot [J - \varepsilon \cdot E_b]/(1 - \alpha)\} \cdot A_1$$

Put the equation over a common denominator:

$$q_{1\rightarrow} = \left[\frac{(1-\alpha)\cdot \varepsilon \cdot E_b - \alpha \cdot J + \alpha \cdot \varepsilon \cdot E_b}{1-\alpha}\right] \cdot A_1 = \left[\frac{\varepsilon \cdot E_b - \alpha \cdot J}{1-\alpha}\right] \cdot A_1$$

If we assume that $\alpha = \varepsilon$ then the equation reduces to:

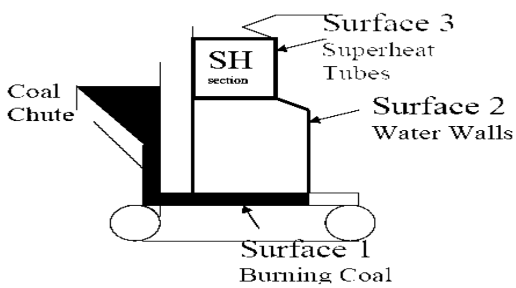
$$q_{1\rightarrow} = \left[\frac{\varepsilon \cdot E_b - \varepsilon \cdot J}{1-\varepsilon}\right] \cdot A_1 = \left[\frac{\varepsilon \cdot A_1}{1-\varepsilon}\right] \cdot (E_b - J)$$

3.23 Electrical Analogy for Radiation

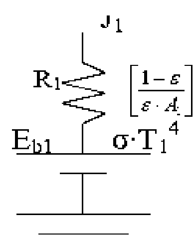
We may develop an electrical analogy for radiation, similar to that produced for conduction. The two analogies should not be mixed: they have different dimensions on the potential differences, resistance and current flows.

	Equivalent Current	Equivalent Resistance	Potential Difference
Ohms Law	I	R	ΔV
Net Energy Leaving Surface	$q_{1\rightarrow}$	$\left[\dfrac{1-\varepsilon}{\varepsilon \cdot A}\right]$	E_b - J
Net Exchange Between Surfaces	$q_{i\rightarrow j}$	$\dfrac{1}{A_1 \cdot F_{1\rightarrow 2}}$	$J_1 - J_2$

Example 4.3: Consider a grate fed boiler. Coal is fed at the bottom, moves across the grate as it burns and radiates to the walls and top of the furnace. The walls are cooled by flowing water through tubes placed inside of the walls. Saturated water is introduced at the bottom of the walls and leaves at the top at a quality of about 70%. After the vapor is separated from the water, it is circulated through the superheat tubes at the top of the boiler. Since the steam is undergoing a sensible heat addition, its temperature will rise. It is common practice to subdivide the super-heater tubes into sections, each having nearly uniform temperature. In our case we will use only one superheat section using an average temperature for the entire region.

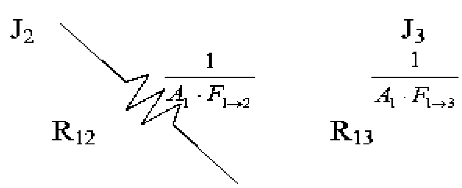


Energy will leave the coal bed, Surface 1, as described by the equation for the net energy leaving a surface. We draw the equivalent electrical network as seen to the right:

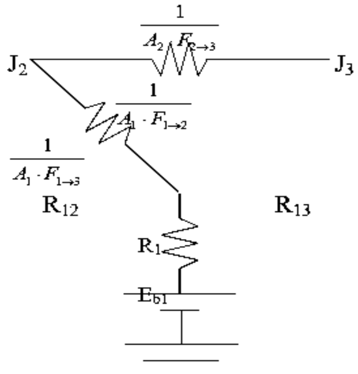


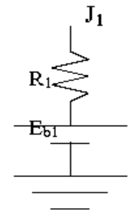
The heat leaving from the surface of the coal may proceed to either the water walls or to the super-heater section. That part of the circuit is represented by a potential difference between Radiosity:

It should be noted that surfaces 2 and 3

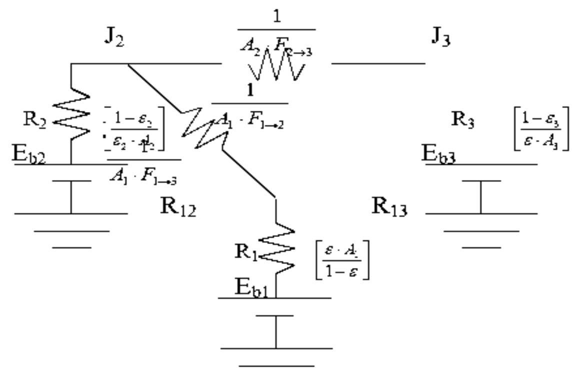


will also radiate to one another.





It remains to evaluate the net heat flow leaving (entering) nodes 2 and 3.



I

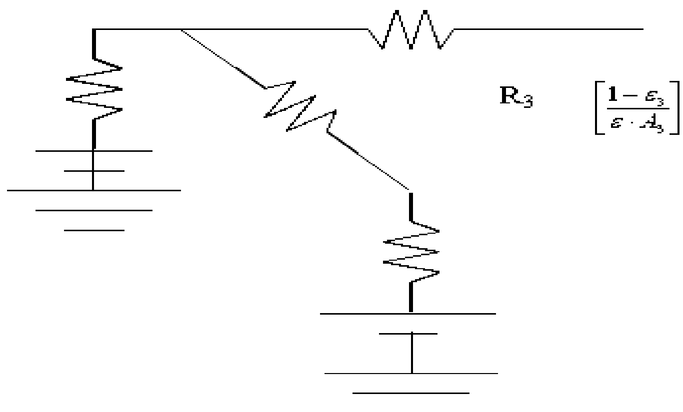
Alternate Procedure for Developing Networks

- Count the number of surfaces. (A surface must be at a "uniform" temperature and have uniform properties, i.e. ε, α, ρ.)
- Draw a radiosity node for each surface.
- Connect the Radiosity nodes using view factor resistances, $1/A_i \cdot F_{i \to j}$.
- Connect each Radiosity node to a grounded battery, through a surface resistance, $\left[1-\varepsilon \big/ \varepsilon \cdot A\right]$.

This procedure should lead to exactly the same circuit as we obtain previously.

3.24 Simplifications to the Electrical Network

• Insulated surfaces. In steady state heat transfer, a surface cannot receive net energy if it is insulated. Because the energy cannot be stored by a surface in steady state, all energy must be re-radiated back into the enclosure. *Insulated surfaces are often termed as re-radiating surfaces*.



Electrically cannot flow through a battery if it is not grounded.

Surface 3 is not grounded so that the battery and surface resistance serve no purpose and are removed from the drawing.

• Black surfaces: A black, or ideal surface, will have no surface resistance:

$$\left[\frac{1-\varepsilon}{\varepsilon \cdot A}\right] = \left[\frac{1-1}{1 \cdot A}\right] = 0$$

In this case the nodal Radiosity and emissive power will be equal.

This result gives some insight into the physical meaning of a black surface. Ideal surfaces radiate at the maximum possible level. Non-black surfaces will have a reduced potential, somewhat like a battery with a corroded terminal. They therefore have a reduced potential to cause heat/current flow.

• Large surfaces: Surfaces having a large surface area will behave as black surfaces, irrespective of the actual surface properties:

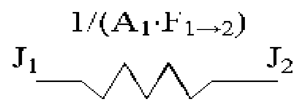
$$\left[\frac{1-\varepsilon}{\varepsilon \cdot A}\right] = \left[\frac{1-\varepsilon}{\varepsilon \cdot \infty}\right] = 0$$

Physically, this corresponds to the characteristic of large surfaces that as they reflect energy, there is very little chance that energy will strike the smaller surfaces; most of the energy is reflected back to another part of the same large surface. After several partial absorptions most of the energy received is absorbed.

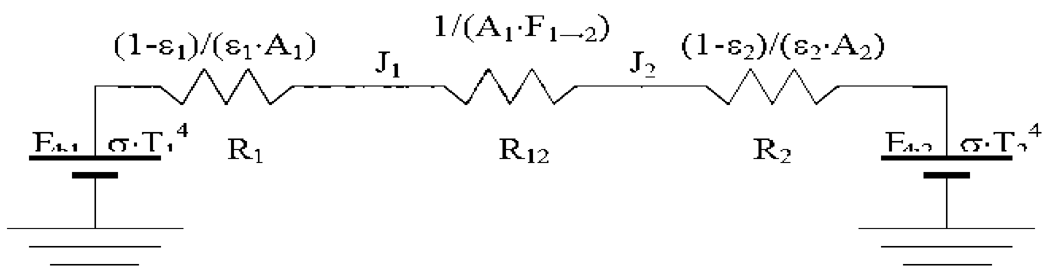
3.28 Solution of Analogous Electrical Circuits.

• Large Enclosures

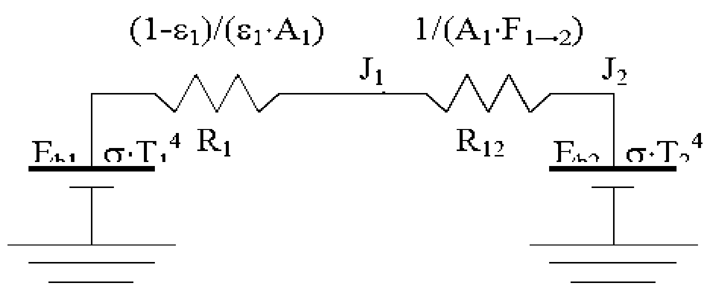
Consider the case of an object, 1, placed inside a large enclosure, 2. The system will consist of two objects, so we proceed to construct a circuit with two radiosity nodes



Now we ground both Radiosity nodes through a surface resistance.



Since A_2 is large, $R_2 = 0$. The view factor, $F_{1\to2} = 1$



Sum the series resistances:

$$R_{Series} = (1-\varepsilon_1)/(\varepsilon_1 \cdot A_1) + 1/A_1 = 1/(\varepsilon_1 \cdot A_1)$$

Ohm's law:

$$i = \Delta V/R$$

or by analogy:

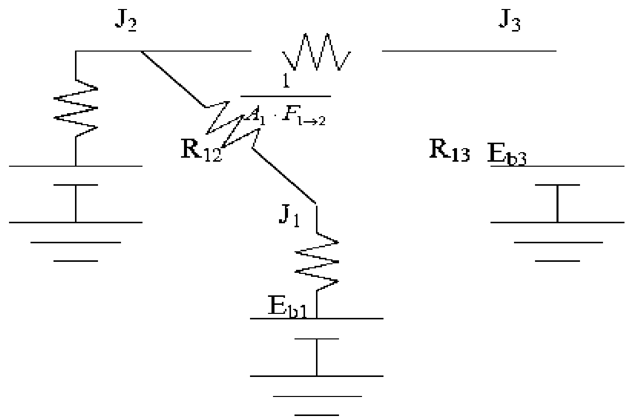
$$q = \Delta E_b/R_{Series} = \varepsilon_1 \cdot A_1 \cdot \sigma \cdot (T_1^4 - T_2^4)$$

You may recall this result from Thermo I, where it was introduced to solve this type of radiation problem.

- Networks with Multiple Potentials

Systems with 3 or more grounded potentials will require a slightly different solution, but one which students have previously encountered in the Circuits course.

The procedure will be to apply Kirchoff's law to each of the Radiosity junctions.



$$\sum_{i=1}^{3} q_i = 0$$

In this example there are three junctions, so we will obtain three equations. This will allow us to solve for three unknowns.

Radiation problems will generally be presented on one of two ways:

1. The surface net heat flow is given and the surface temperature is to be found.

2. The surface temperature is given and the net heat flow is to be found.

Returning for a moment to the coal grate furnace, let us assume that we know (a) the total heat being produced by the coal bed, (b) the temperatures of the water walls and (c) the temperature of the super heater sections.

Apply Kirchoff's law about node 1, for the coal bed:

$$q_1 + q_{2\to 1} + q_{3\to 1} = q_1 + \frac{J_2 - J_1}{R_{12}} + \frac{J_3 - J_1}{R_{13}} = 0$$

Similarly, for node 2:

$$q_2 + q_{1\to 2} + q_{3\to 2} = \frac{E_{b2} - J_2}{R_2} + \frac{J_1 - J_2}{R_{12}} + \frac{J_3 - J_2}{R_{23}} = 0$$

(Note how node 1, with a specified heat input, is handled differently than node 2, with a specified temperature.

And for node 3:

$$q_3 + q_{1\to 3} + q_{2\to 3} = \frac{E_{b3} - J_3}{R_3} + \frac{J_1 - J_3}{R_{13}} + \frac{J_2 - J_3}{R_{23}} = 0$$

The three equations must be solved simultaneously. Since they are each linear in J, matrix methods may be used:

$$\begin{bmatrix} -\frac{1}{R_{12}} - \frac{1}{R_{13}} & \frac{1}{R_{12}} & \frac{1}{R_{13}} \\ \frac{1}{R_{12}} & -\frac{1}{R_2} - \frac{1}{R_{12}} - \frac{1}{R_{13}} & \frac{1}{R_{23}} \\ \frac{1}{R_{13}} & \frac{1}{R_{23}} & -\frac{1}{R_3} - \frac{1}{R_{13}} - \frac{1}{R_{23}} \end{bmatrix} \cdot \begin{bmatrix} J_1 \\ J_2 \\ J_3 \end{bmatrix} = \begin{bmatrix} -q_1 \\ -\frac{E_{b2}}{R_2} \\ -\frac{E_{b3}}{R_3} \end{bmatrix}$$

The matrix may be solved for the individual Radiosity. Once these are known, we return to the electrical analogy to find the temperature of surface 1, and the heat flows to surfaces 2 and 3.

Surface 1: Find the coal bed temperature, given the heat flow:

$$q_1 = \frac{E_{b1} - J_1}{R_1} = \frac{\sigma \cdot T_1^4 - J_1}{R_1} \Rightarrow T_1 = \left[\frac{q_1 \cdot R_1 + J_1}{\sigma}\right]^{0.25}$$

Surface 2: Find the water wall heat input, given the water wall temperature:

$$q_2 = \frac{E_{b2} - J_2}{R_2} = \frac{\sigma \cdot T_2^4 - J_2}{R_2}$$

Surface 3: (Similar to surface 2) Find the water wall heat input, given the water wall temperature:

$$q_3 = \frac{E_{b3} - J_3}{R_3} = \frac{\sigma \cdot T_3^4 - J_3}{R_3}$$

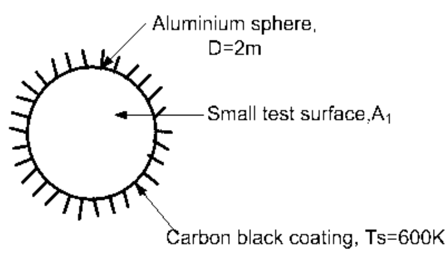
Worked out problems

1. A spherical aluminum shell of inside diameter D=2m is evacuated and is used as a radiation test chamber. If the inner surface is coated with carbon black and maintained at 600K, what is the irradiation on a small test surface placed in the chamber? If the inner surface were not coated and maintained at 600K, what would the irradiation test?

Known: Evacuated, aluminum shell of inside diameter D=2m, serving as a radiation test chamber.

Find: Irradiation on a small test object when the inner surface is lined with carbon black and maintained at 600K.what effect will surface coating have?

Schematic:



Assumptions: (1) Sphere walls are isothermal, (2) Test surface area is small compared to the enclosure surface.

Analysis: It follows from the discussion that this isothermal sphere is an enclosure behaving as a black body. For such a condition, the irradiation on a small surface within the enclosure is equal to the black body emissive power at the temperature of the enclosure. That is

$$G_1 = E_b(T_s) = \sigma T_s^4$$

$$G_1 = 5.67 \times 10^{-8} W/m^2.K(600K)^4 = 7348W/m^2$$

The irradiation is independent of the nature of the enclosure surface coating properties.

Comments: (1) The irradiation depends only upon the enclosure surface temperature and is independent of the enclosure surface properties.

(2) Note that the test surface area must be small compared to the enclosure surface area. This allows for inter-reflections to occur such that the radiation field, within the enclosure will be uniform (diffuse) or isotropic.

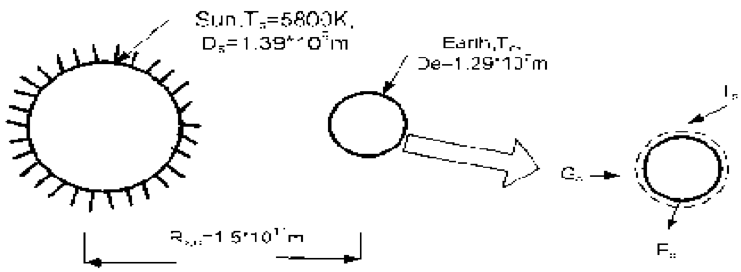
(3) The irradiation level would be the same if the enclosure were not evacuated since; in general, air would be a non-participating medium.

2 Assuming the earth's surface is black, estimate its temperature if the sun has an equivalently blackbody temperature of 5800K.The diameters of the sun and earth are 1.39*109 and 1.29*107m, respectively, and the distance between the sun and earth is 1.5*1011m.

Known: sun has an equivalently blackbody temperature of 5800K. Diameters of the sun and earth as well as separation distances are prescribed.

Find: Temperature of the earth assuming the earth is black.

Schematic:



Assumptions: (1) Sun and earth emit black bodies, (2) No attenuation of solar irradiation enroute to earth, and (3) Earth atmosphere has no effect on earth energy balance.

Analysis: performing an energy balance on the earth

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

$$A_{e,p} . G_S = A_{e,s} . E_b(T_e)$$

$$(\pi D_e^2 / 4) G_S = \pi D_e^2 \sigma T_e^4$$

$$T_e = (G_S / 4\sigma)^{1/4}$$

Where $A_{s,p}$ and $A_{e,s}$ are the projected area and total surface area of the earth, respectively. To determine the irradiation GS at the earth's surface, perform an energy bounded by the spherical surface shown in sketch

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

$$\pi D_S^2 . \sigma T_S^4 = 4\pi [R_{S,e} - D_e / 2]^2 G_S$$

$$\pi (1.39 \times 10^9 m)^2 \times 5.67 \times 10^{-8} W / m^2 . K (5800 K)^4 =$$

$$4\pi [1.5 \times 10^{11} - 1.29 \times 10^7 / 2]^2 m^2 \times G_S$$

$$G_S = 1377.5 W / m^2$$

Substituting numerical values, find

$$T_e = (1377.5 W / m^2 / 4 \times 5.67 \times 10^{-8} W / m^2 . K^4)^{1/4} = 279 K$$

Comments:

(1) The average earth's temperature is greater than 279 K since the effect of the atmosphere is to reduce the heat loss by radiation.

(2) Note carefully the different areas used in the earth energy balance. Emission occurs from the total spherical area, while solar irradiation is absorbed by the projected spherical area.

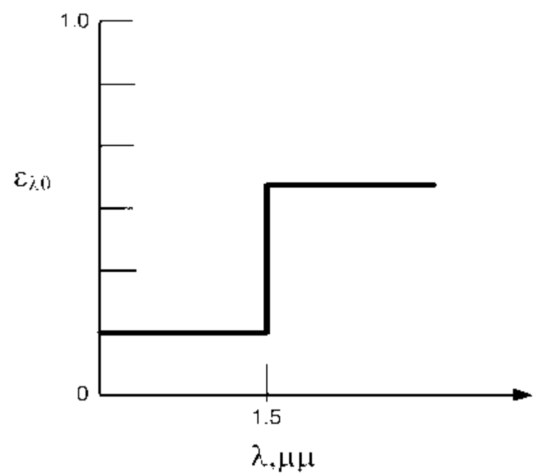
3 The spectral, directional emissivity of a diffuse material at 2000K has the following distribution.

Determine the total, hemispherical emissivity at 2000K.Determine the emissive power over the spherical range 0.8 t0 2.5 µm and for the directions 0≤θ≤30°.

Known: Spectral, directional emissivity of a diffuse material at 2000K.

Find: (1) The total, hemispherical emissivity, (b) emissive power over the spherical range 0.8 t0 2.5 μm and for the directions 0≤θ≤30°.

Schematic:



Assumptions: (1) Surface is diffuse emitter.

Analysis: (a) Since the surface is diffuse,ελ,θ is independent of direction; from Eq. $\varepsilon_{\lambda,\theta} = \varepsilon_{\lambda}$

$$\varepsilon(T) = \int_0^{\infty} \varepsilon_{\lambda}(\lambda) E_{\lambda,b}(\lambda, T) d\lambda / E_b(T)$$

$$\varepsilon(T) = \int_0^{1.5} \varepsilon_1) E_{\lambda,b}(\lambda, 2000) d\lambda / E_b) + \int_0^{1.5} \varepsilon_2 E_{\lambda,b}(\lambda, 2000) d\lambda / E_b$$

Written now in terms of $F_{(0\to\lambda)}$, with $F_{(0\to1.5)}$ =0.2732 at λT=1.5*2000=3000μm.K, find

$$\varepsilon(2000K) = \varepsilon_1 F_{(0\to1.5)} + \varepsilon_2 \left[1 - F_{(0\to1.5)}\right] = 0.2 \times 0.2732 + 0.8[1 - 0.2732] = 0.636$$

(b) For the prescribed spectral and geometric limits,

$$\Delta E = \int_{0.8}^{2.5} \int_0^{2\pi} \int_0^{\pi/6} \varepsilon_{\lambda,\theta} I_{\lambda,b}(\lambda, T) \cos\theta \sin\theta \, d\theta \, d\phi \, d\lambda$$

where $I_{\lambda,e}(\lambda,\theta,\phi) = \varepsilon_{\lambda,\theta} I_{\lambda,b}(\lambda,T)$. Since the surface is diffuse, $\varepsilon_{\lambda,\theta} = \varepsilon_{\lambda}$, and nothing $I_{\lambda,b}$ is independent of direction and equal to $E_{\lambda,b/}\pi$, we can write

$$\Delta E = \left\{ \int_0^{2\pi\pi/6} \int_0 \cos\theta \sin\theta \, d\theta \, d\phi \right\} \frac{E_b(T)}{\pi} \frac{\int_{0.8}^{1.5} \varepsilon_1 E_{\lambda,b}(\lambda,T) d\lambda}{E_b(T)} + \frac{\int_{1.5}^{2.5} \varepsilon_2 E_{\lambda,b}(\lambda,T) d\lambda}{E_b(T)}$$

Or in terms $F_{(0\to\lambda)}$ values,

$$\Delta E = \left\{ \phi \Big|_0^{2\pi} \times \frac{\sin^2\theta}{2} \Big|_0^{\pi/6} \right\} \frac{\sigma T^4}{\pi} \left\{ \varepsilon_1 [F_{(0\to1.5)} - F_{(0\to0.8)}] \right\} + \varepsilon_2 [F_{(0\to2.5)} - F_{(0\to1.5)}] \}$$

From table $\lambda T = 0.8 \times 2000 = 1600 \mu m.K$ $\quad F_{(0\to0.8)} = 0.0197$

$\lambda T = 2.5 \times 2000 = 5000 \mu m.K$ $\quad F_{(0\to2.5)} = 0.6337$

$$\Delta E = 2\pi \times \frac{\sin^2 \pi/6}{2} \frac{5.67 \times 10^{-8} 2000^4}{\pi} \frac{W}{m^2} \{0.2[0.2732 - 0.0197] + [0.80.6337 - 0.2732]\}$$

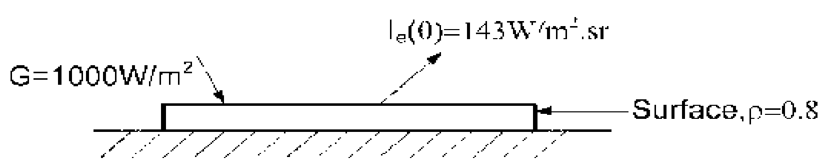
$$\Delta E = 0.25 \times (5.67 \times 10^{-8} \times 2000^4 W/m^2 \times 0.339 = 76.89 W/m^2$$

4. A diffusely emitting surface is exposed to a radiant source causing the irradiation on the surface to be 1000W/m^2.The intensity for emission is 143W/m^2.sr and the reflectivity of the surface is 0.8.Determine the emissive power ,E(W/m^2),and radiosity ,J(W/m^2),for the surface. What is the net heat flux to the surface by the radiation mode?

Known: A diffusely emitting surface with an intensity due to emission of Is=143W/m^2.sr and a reflectance ρ=0.8 is subjected to irradiation=1000W/m^2.

Find: (a) emissive power of the surface, E (W/m^2), (b) radiosity, J (W/m^2), for the surface, (c) net heat flux to the surface.

Schematic:



Assumptions: (1) surface emits in a diffuse manner.

Analysis: (a) For a diffusely emitting surface, $I_s(\theta) = I_e$ is a constant independent of direction. The emissive power is

$$E = \pi I_e = \pi sr \times 143\,W/m^2.sr = 449\,W/m^2$$

Note that π has units of steradians (sr).

(b) The radiosity is defined as the radiant flux leaving the surface by emission and reflection,

$$J = E + \rho G = 449W/m^2 + 0.8 \times 1000W/m^2 = 1249W/m^2$$

(c) The net radiative heat flux to the surface is determined from a radiation balance on the surface.

$$q''_{net} = q''_{rad,in} - q''_{rad,out}$$

$$q''_{net} = G - J = 1000W/m^2 - 1249W/m^2 = -249W/m^2$$

Comments: No matter how the surface is irradiated, the intensity of the reflected flux will be independent of direction, if the surface reflects diffusely.

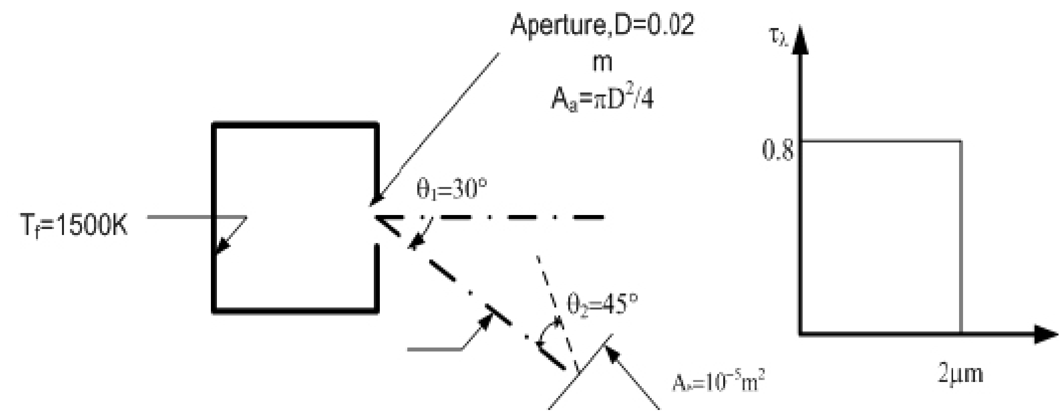
5. Radiation leaves the furnace of inside surface temperature 1500K through an aperture 20mm in diameter. A portion of the radiation is intercepted by a detector that is 1m from the aperture, as a surface area $10^{-5}m^2$, and is oriented as shown.

If the aperture is open, what is the rate at which radiation leaving the furnace is intercepted by the detector? If the aperture is covered with a diffuse, semitransparent material of spectral transmissivity $\tau\lambda$=0.8 for $\lambda \leq 2\mu m$ and $\tau\lambda$=0 for $\lambda > 2\mu m$, what is the rate at which radiation leaving the furnace is intercepted by the detector?

Known: Furnace wall temperature and aperture diameter. Distance of detector from aperture and orientation of detector relative to aperture.

Find: Rate at which radiation leaving the furnace is intercepted by the detector, (b) effect of aperture window of prescribed spectral transmissivity on the radiation interception rate.

Schematic:



Assumptions:

(1) Radiation emerging from aperture has characteristics of emission from a black body, (2) Cover material is diffuse, (3) Aperture and detector surface may be approximated as infinitesimally small.

Analysis: (a) the heat rate leaving the furnace aperture and intercepted by the detector is

$q = I_e A_s \cos\theta\, w_{a-a}$ Heat and Mass Transfer

$$I_e = \frac{E_b(T_f)}{\pi} = \frac{\sigma T_f^4}{\pi} = \frac{5.67\times10^{-8}(1500)^4}{\pi} = 9.14\times10^4 W/m^2.sr$$

$$w'_{s-a} = \frac{A''}{r^2} = \frac{A_s.\cos\theta^2}{r^2} = \frac{10^{-5}m^2\cos45^\circ}{(1m)^2} = 0.70710^{-5}.sr$$

Hence

$$q = 9.14\times10^4 W/m^2.sr[\pi(0.02)m^2/4]\cos30^\circ\times0.707\times10^{-5}sr = 1.76\times10^{-4}W$$

(b) With the window, the heat rate is

$$q = \tau(I_e A_a \cos\theta_1 w_{a-a})$$

where τ is the transmissivity of the window to radiation emitted by the furnace wall.

$$\tau = \frac{\int_0^\infty \tau_\lambda G_\lambda d_\lambda}{\int_0^\infty G_\lambda d_\lambda} = \frac{\int_0^\infty \tau_\lambda E_{\lambda,b}(T_f) d\lambda}{\int_0^\infty E_{\lambda,b} d\lambda} = 0.8\int_0^2 (E_{\lambda,b}/E_b) d\lambda = 0.8F_{(0\to 2\mu m)}$$

with $\lambda T = 2\mu m \times 1500K = 3000\mu m.K$, from table $F(0 \to 2\mu m) = 0.273$.

hence with $0.273 \times 0.8 = 0.218$, find

$$q = 0.218 \times 1.76 \times 10^{-4}\,W = 0.384 \times 10^{-4}\,W$$

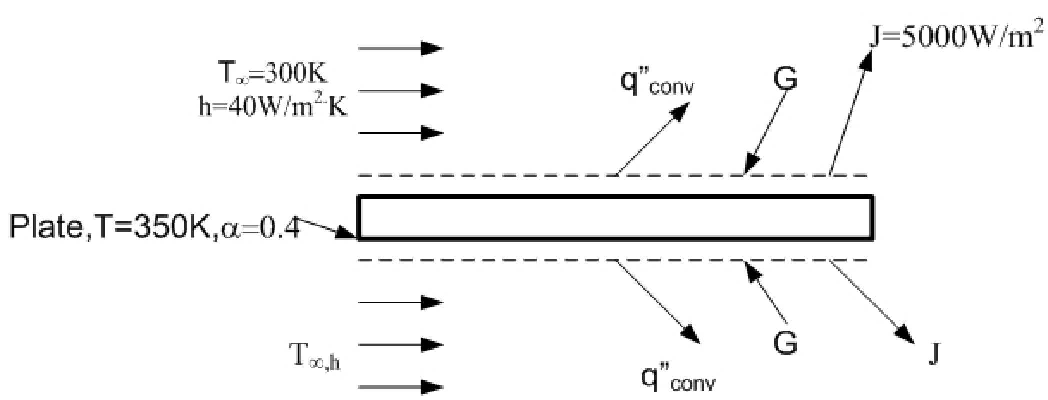
6.A horizontal semitransparent plate is uniformly irradiated from above and below, while air at T=300K flows over the top and bottom surfaces. providing a uniform convection heat transfer coefficient of h=40W/m2.K.the total, hemispherical absorptivity of the plate to the irradiation is 0.40.Under steady-state conditions measurements made with radiation detector above the top surface indicate a radiosity(which includes transmission, as well as reflection and emission) of J=5000W/m2,while the plate is at uniform temperature of T=350K

Determine the irradiation G and the total hemispherical emissivity of the plate. Is the plate gray for the prescribed conditions?

Known: Temperature, absorptivity, transmissivity, radiosity and convection conditions for a semi-transparent plate.

Find: Plate irradiation and total hemispherical emissivity.

Schematic:



Assumptions: From an energy balance on the plate

E_{in}- E_{out}

2G=2q"$_{conv}$+2J

Solving for the irradiation and substituting numerical values,

G=40W/m^2.K (350-300) K+5000W/m^2=7000W/m^2

From the definition of J

$$J = E + \rho G + \tau G = E + (1-\alpha)G$$

Solving for the emissivity and substituting numerical values,

$$\varepsilon = \frac{J-(1-\alpha)G}{\sigma T^4} = \frac{(5000\,W/m^2) - 0.6(7000\,W/m^2)}{5.67\times10^{-8}\,W/m^2.K^4\,(350\,K)^4} = 0.94$$

Hence

$$\alpha \neq \varepsilon$$

And the surface is not gray for the prescribed conditions.

Comments: The emissivity may also be determined by expressing the plate energy balance as

$$2\alpha G = 2q''_{conv}\,2E$$

hence

$$\varepsilon\sigma T^4 = \alpha G - h(T - T_\infty)$$

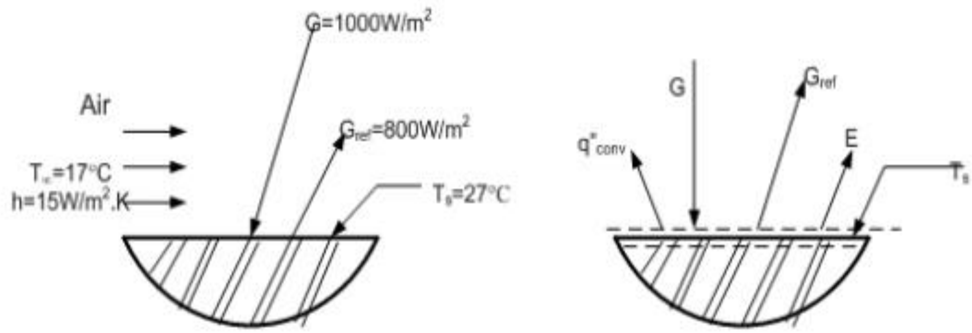
$$\varepsilon = \frac{0.4(7000W/m^2) - 40W/m^2.K(50K)}{5.67\times10^{-8}W/m^2.K4(350K)^4} = 0.94$$

7 An opaque, gray surface at 27°C is exposed to irradiation of 1000W/m2, and 800W/m2 is reflected. Air at 17°C flows over the surface and the heat transfer convection coefficient is 15W/m2.K.Determine the net heat flux from the surface.

Known: Opaque, gray surface at 27°C with prescribed irradiation, reflected flux and convection process.

Find: Net heat flux from the surface.

Schematic:



Assumptions:

1) Surface is opaque and gray,

2) Surface is diffuse,

3) Effects of surroundings are included in specified irradiation.

Analysis: From an energy balance on the surface, the net heat flux from the surface is

$$q''_{net} = E''_{out} - E''_{in}$$

$$q''_{net} = q''_{conv} + E + G_{ref} - G = h(T_s - T_\infty) + \varepsilon\sigma T_s^4 + G_{ref} - G$$

$$\varepsilon = \alpha = 1 - \rho = 1 - (G_{ref}/G) = 1 - (800/1000) = 1 - 0.8 = 0.2$$

where $\rho = G_{ref}/G$**. the net heat flux from the surface**

$$q''_{net} = 15W/m^2.K(27-17)K + 0.2 \times 5.67 \times 10^{-8} W/m^2.K^4(27+273)^4 K^4 + 800W/m^2 - 1000W/m^2$$

$$q''_{net} = (150 + 91.9 + 800 - 1000)W/m^2 = 42W/m^2$$

Comments: (1) For this situation, the radiosity is

$$J = G_{ref} + E = (800 + 91.9)W/m^2 = 892W/m^2$$

The energy balance can be written involving the radiosity (radiation leaving the surface) and the irradiation (radiation to the surface).

$$q''_{net,out} = J - G + q''_{conv} = (892 - 1000 + 150) W/m^2 = 42 W/m^2$$

Note the need to assume the surface is diffuse, gray and opaque in order that Eq (2) is applicable.

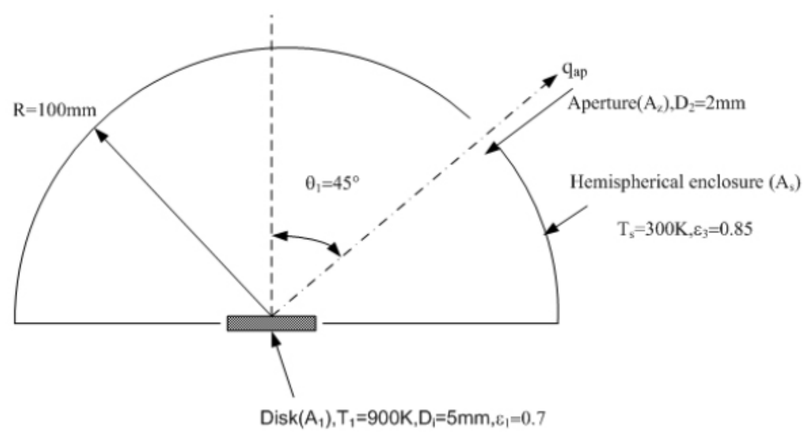
8. A small disk 5 mm in diameter is positioned at the center of an isothermal, hemispherical enclosure. The disk is diffuse and gray with an emissivity of 0.7 and is maintained at 900 K. The hemispherical enclosure, maintained at 300 K, has a radius of 100 mm and an emissivity of 0.85.

Calculate the radiant power leaving an aperture of diameter 2 mm located on the enclosure as shown.

Known: Small disk positioned at center of an isothermal, hemispherical enclosure with a small aperture.

Find: radiant power [μW] leaving the aperture.

Schematic:



Assumptions: (1)Disk is diffuse-gray,(2) Enclosure is isothermal and has area much larger than disk,(3) Aperture area is very small compared to enclosure area, (4) Areas of disk and aperture are small compared to radius squared of the enclosure.

Analysis: the radiant power leaving the aperture is due to radiation leaving the disk and to irradiation on the aperture from the enclosure. That is

$$q_{ap} = q_{1\to 2} + G_2 . A_2$$

The radiation leaving the disk can be written in terms of the radiosity of the disk. For the diffuse disk

$$q_{1\to 2} = \frac{1}{\pi} J_1 . A_1 \cos\theta_1 . w_{2-1}$$

and with $\varepsilon = \alpha$ for the gray behavior, the radiosity is

$$\mathbf{J_1} = \varepsilon_1 E_b(T_1) + \rho G_1 = \varepsilon_1 \sigma T_1^4 + (1-\varepsilon_1)\sigma T_3^4$$

Where the irradiatin G_1 is the emissive power of the black enclosure, $E_b(T_3)$;

$G_1 = G_2 = E_b(T_3)$. The solid angle $\omega_{2\text{-}1}$ follows

$$\omega_{2-1} = A_2 / R^2$$

Combining equations. (2), (3) and (4) into eq.(1) with G2=σT^4_3, the radiant power is

$$q_{ap} = \frac{1}{\pi}\sigma[\varepsilon_1 T_1^4 + (1-\varepsilon_1)T_3^4] A_1 \cos\theta_1 . \frac{A_2}{R^2} + A_2 \sigma T_3^4$$

$$q_{ap} = \frac{1}{\pi} 5.67\times 10^{-8} W/m^2.K^4 [0.7(900K)^4 + (1-0.7)(300K)^4 \frac{\pi}{4}(0.005m)^2 \cos 45° \times$$

$$\frac{\pi/4(0.002m)^2}{(0.100m)^2} + \frac{\pi}{4}(0.002m)25.67\times 10^{-8} W/m^2.K^4(300K)^4$$

$$q_{ap} = (36.2 + 0.19 + 1443)\mu W = 1479\mu W$$

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DEPARTMENT OF MECHANICAL

UNIT – IV – Heat Transfer Applied to IC Engines – SAUA1503

4.1 Engine Heat Transfer

• Heat transfer is a parasitic process that contributes to a loss in fuel conversion efficiency

• The process is a "surface" effect

• Relative importance reduces with: – Larger engine displacement – Higher load

Engine Heat Transfer: Impact

• Efficiency and Power: Heat transfer in the inlet decrease volumetric efficiency. In the cylinder, heat losses to the wall is a loss of availability.

• Exhaust temperature: Heat losses to exhaust influence the turbocharger performance. In-cylinder and exhaust system heat transfer has impact on catalyst light up.

• Friction: Heat transfer governs liner, piston/ ring, and oil temperatures. It also affects piston and bore distortion. All of these effects influence friction. Thermal loading determined fan, oil and water cooler capacities and pumping power.

• Component design: The operating temperatures of critical engine components affects their durability; e.g. via mechanical stress, lubricant behavior

Mixture preparation in SI engines: Heat transfer to the fuel significantly affect fuel evaporation and cold start calibration

• Cold start of diesel engines: The compression ratio of diesel engines are often governed by cold start requirement

• SI engine octane requirement: Heat transfer influences inlet mixture temperature, chamber, cylinder head, liner, piston and valve temperatures, and therefore end-gas temperatures, which affect knock. Heat transfer also affects build up of in-cylinder deposit which affects knock.

4.2 Engine heat transfer environment

Gas temperature: ~300 – 3000^{o}K

Heat flux to wall: Q/A<0 to 10 MW/m^2

Materials limit: – Cast iron ~ 400^{o}C – Aluminum ~ 300^{o}C – Liner (oil film) ~200^{o}C

Hottest components – Spark plug > Exhaust valve > Piston crown > Head – Liner is relatively cool because of limited exposure to burned gas

• Source – Hot burned gas – Radiation from particles in diesel engines

4.3 Energy flow diagram for an IC engine

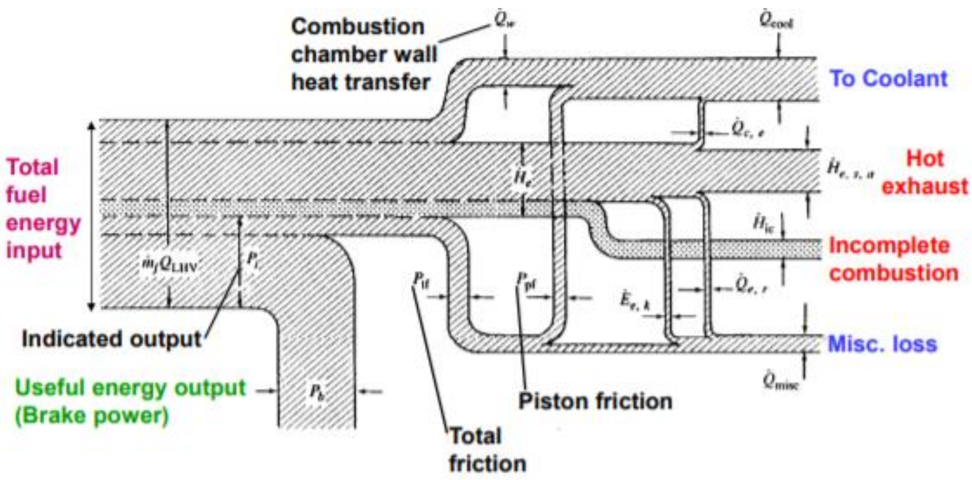


Fig 4.1

4.4 Energy flow distribution for SI and Diesel

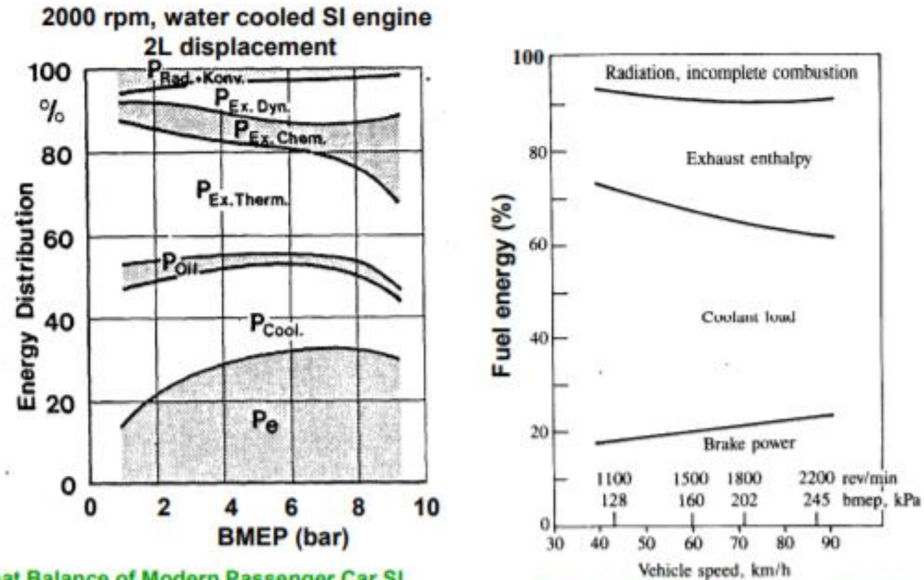
Energy balance for automotive engines at maximum power

	P_b	$\dot{Q}_{cool}$	$\dot{Q}_{misc}$	$\dot{H}_{e,ic}$	$\dot{m}h_{e,s}$
	(percentage of fuel heating value)				
SI engine	25–28	17–26	3–10	2–5	34–45
Diesel	34–38	16–35	2–6	1–2	22–35

Sources: From Khovakh,[3] Sitkei,[4] and Burke *et al.*[5]

Update for modern engines:
SI engine in the low 30's
Diesel in the low 40's

4.5 Energy distribution in SI engine

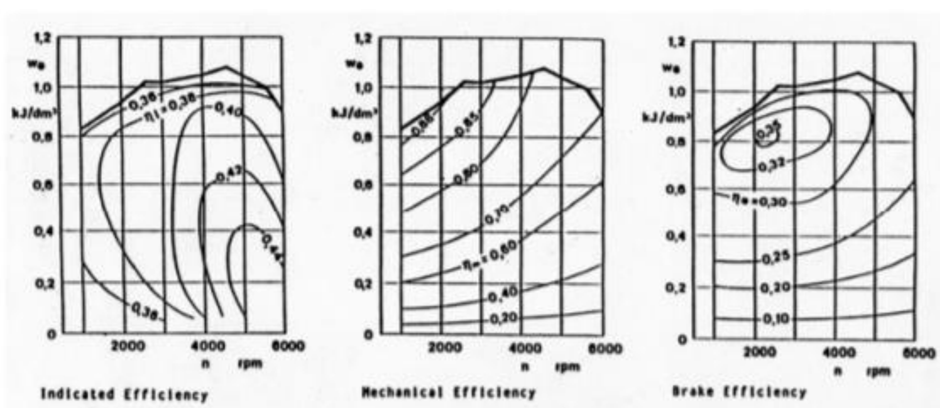


"Heat Balance of Modern Passenger Car SI Engines", Gruden, Kuper and Porsche, in *Heat and Mass Transfer in Gasoline and Diesel Engines*, ed. by Spalding and Afgan

Fig. 12-4 SI engine energy distribution under road load condition, 6 cylinder engine; SAE Paper 770221, 1977

Fig 4.2

4.6 Efficiency of Passenger Car SI Engines



Source: D. Gruden, P.F., and F. Porsche AG. R & D Center Weissach, 1989.

Fig 4.3

4.7 Heat transfer process in engines

Areas where heat transfer is important

– Intake system: manifold, port, valves

– In-cylinder: cylinder head, piston, valves, liner

– Exhaust system: valves, port, manifold, exhaust pipe

– Coolant system: head, block, radiator – Oil system: head, piston, crank, oil cooler, sump

• Information of interest – Heat transfer per unit time (rate) – Heat transfer per cycle (often normalized by fuel heating value) – Variation with time and location of heat flux (heat transfer rate per unit area)

Schematic of temperature distribution and heat flow across the combustion chamber wall

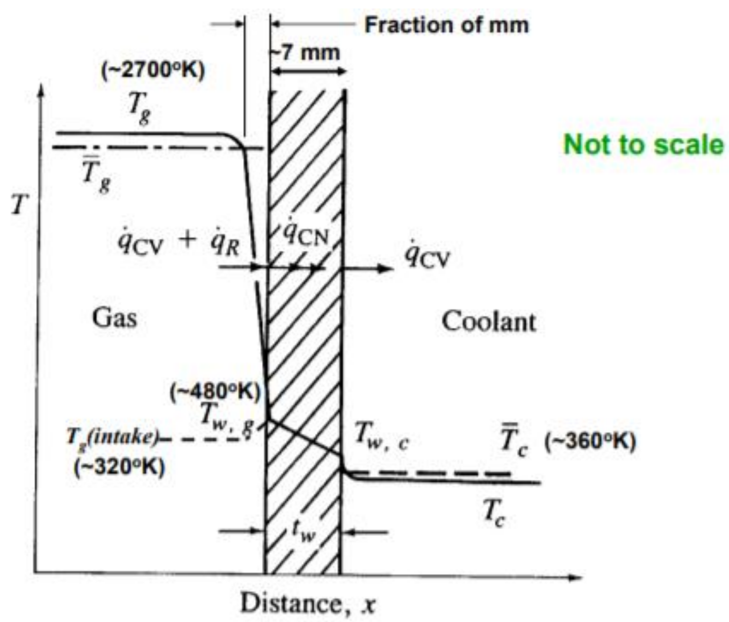


Fig 4.4

4.8 Combustion Chamber Heat Transfer

Turbulent convection: hot gas to wall

$$\dot{Q} = Ah_g(\overline{T}_g - T_{wg})$$

Conduction through wall

$$\dot{Q} = A\frac{\kappa}{t_w}(T_{wg} - T_{wc})$$

Turbulent convection: wall to coolant

$$\dot{Q} = Ah_c(T_{wc} - \overline{T}_c)$$

Overall heat transfer

$$\dot{Q} = Ah(\overline{T}_g - \overline{T}_c)$$

Overall thermal resistance: three resistance in series

$$\frac{1}{h} = \frac{1}{h_g} + \frac{t_w}{\kappa} + \frac{1}{h_c}$$

(κ_{alum} ~180 W/m-k
$\kappa_{cast\ iron}$ ~ 60 W/m-k
$\kappa_{stainless\ steel}$ ~18 W/m-k)

Turbulent Convective Heat Transfer CorrelationApproach: Use Nusselt- Reynolds number correlations similar to those for turbulent pipe or flat plate flows. e.g. In-cylinder:

$$Nu = \frac{hL}{\kappa} = a(Re)^{0.8}$$

h = Heat transfer coefficient
L = Characteristic length (e.g. bore)
Re= Reynolds number, $\rho UL/\mu$
U = Characteristic gas velocity
κ = Gas thermal conductivity
μ = Gas viscosity
ρ = Gas density
a = Turbulent pipe flow correlation coefficient

Radiative Heat Transfer

- Important in diesels due to presence of hot radiating particles (particulate matters) in the flame
- Radiation from hot gas relatively small

$$\dot{Q}_{rad} = \varepsilon \cdot \sigma \cdot T^4_{particle}$$

σ = Stefan Boltzman Constant (5.67×10^{-8} W/m^2-K^4)
ε = Emissivity
where

$$T_{cyl.\ ave} < T_{particle} < T_{max\ burned\ gas}$$

- Radiation spectrum peaks at λ_{max}
  $\lambda_{max}T$ = constant (λ_{max} = 3 μm at 1000K)

Typically, in diesels: $\overline{Q}_{rad} \approx 0.2\overline{Q}_{total}$ **(cycle cum)**

$\dot{Q}_{rad,max} \approx 0.4\dot{Q}_{total,max}$ **(peak value)**

4.9 IC Engine heat transfer

Heat transfer mostly from hot burned gas

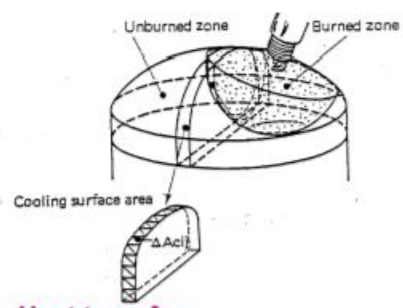
– That from unburned gas is relatively small

– Flame geometry and charge motion/turbulence level affects heat transfer rate

• Order of Magnitude – SI engine peak heat flux ~ 1-3 MW/m2 – Diesel engine peak heat flux ~ 10 MW/m2

• For SI engine at part load, a reduction in heat losses by 10% results in an improvement in fuel consumption by 3% – Effect substantially less at high load

SI Engine Heat Transfer



- Heat transfer dominated by that from the hot burned gas
- Burned gas wetted area determine by cylinder/ flame geometry
- Gas motion (swirl/ tumble) affects heat transfer coefficient

Heat transfer

Burned zone: sum over area "wetted" by burned gas $\dot{Q}_b = \sum_i A_{ci,b} h_b (T_b - T_{w,i})$

Unburned zone: sum over area "wetted" by unburned gas $\dot{Q}_u = \sum_i A_{ci,u} h_u (T_u - T_{w,i})$

Note: Burned zone heat flux >> unburned zone heat flux

Heat Transfer Summary

1. Magnitude of heat transfer from the burned gas much greater than in any phase of cycle
2. Heat transfer is a significant performance loss and affects engine operation

¬ Loss of available energy

¬ Volumetric efficiency loss

¬ Effect on knock in SI engine

¬ Effect on mixture preparation in SI engine cold start

¬ Effect on diesel engine cold start

3. Convective heat transfer depends on gas temperature, heat transfer coefficient, which depends on charge motion, and transfer area, which depends on flame/combustion chamber geometry

4. Radiative heat transfer is smaller than convective one, and it is only significant in diesel engines

HEAT EXCHANGERS

4.10 Heat Exchangers: Regenerators and Recuperators

A heat exchanger is an equipment where heat energy is transferred from a hot fluid to a colder fluid. The transfer of heat energy between the two fluids could be carried out (i) either by direct mixing of the two fluids and the mixed fluids leave at an intermediate temperature determined from the principles of conservation of energy, (ii) or by transmission through a wall separating the two fluids. The former types are called direct contact heat exchangers such as water cooling towers and jet condensers. The latter types are called regenerators, recuperator surface exchangers.

In a regenerator, hot and cold fluids alternately flow over a surface which provides alternately a sink and source for heat flow. Fig. 10.1 (a) shows a cylinder containing a matrix that rotates in such a way that it passes alternately through cold and hot gas streams which are sealed from each other. Fig. 10.1 (b) shows a stationary matrix regenerator ill which hot and cold gases flow through them alternately.

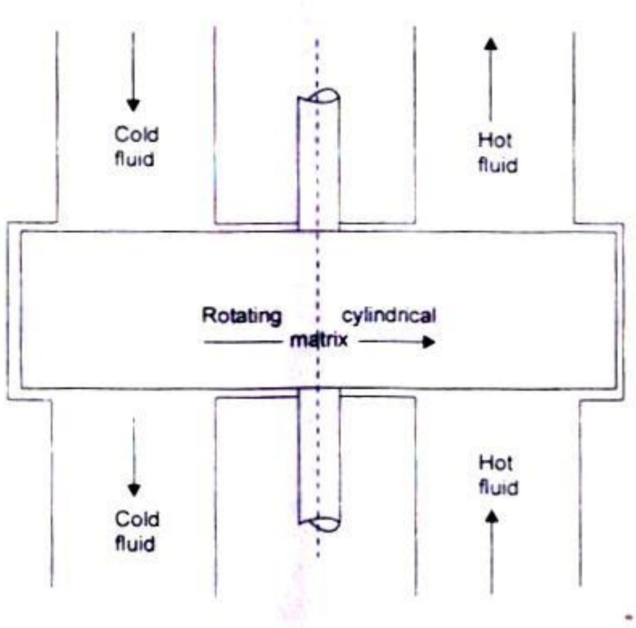


Fig. 4.5 (a) Rotating matrix regenerator

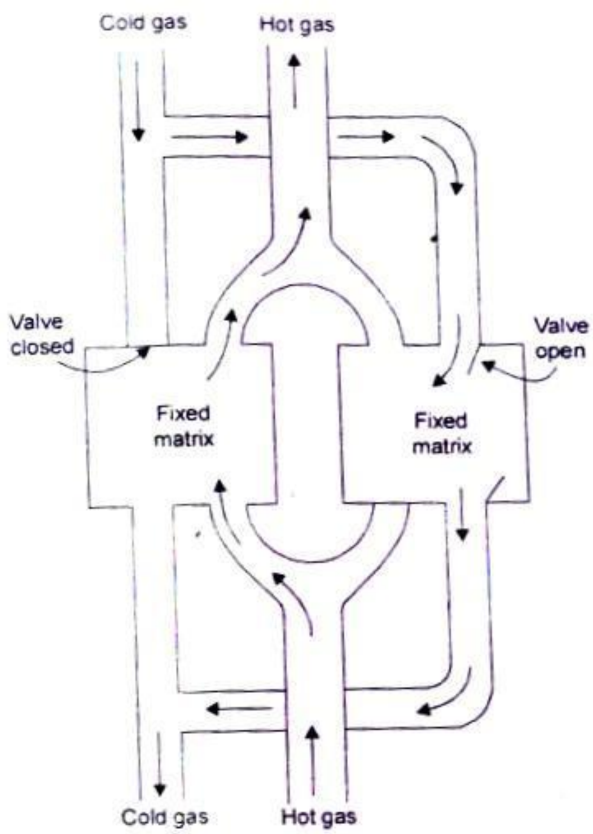


Fig. 4.5 (b) Stationary matrix regenerator

In a recuperator, hot and cold fluids flow continuously following he same path. The heat transfer process consists of convection between the fluid and the separating wall, conduction through the wall and convection between the wall and the other fluid. Most common heat exchangers are of recuperative type having a Wide variety of geometries:

4.11 Classification of Heat Exchangers

Heat exchangers are generally classified according to the relative directions of hot and cold fluids:

(a) Parallel Flow – the hot and cold fluids flow in the same direction. Fig 3.2 depicts such a heat exchanger where one fluid (say hot) flows through the pipe and the other fluid (cold) flows through the annulus.

(b) Counter Flow – the two fluids flow through the pipe but in opposite directions. A common type of such a heat exchanger is shown in Fig. 3.3. By comparing the temperature distribution of the two types of heat exchanger

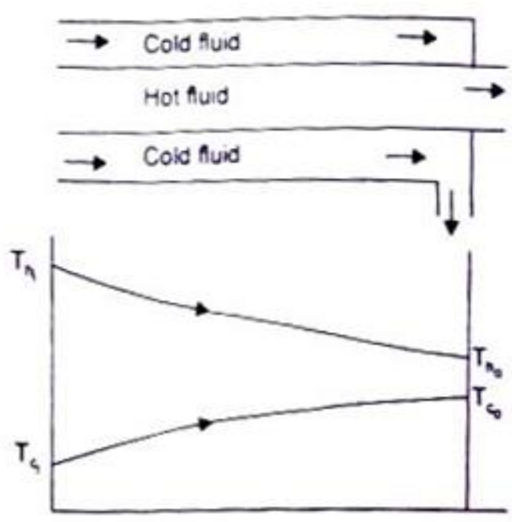


Fig 4.6a Parallel flow heat exchanger with temperature distribution

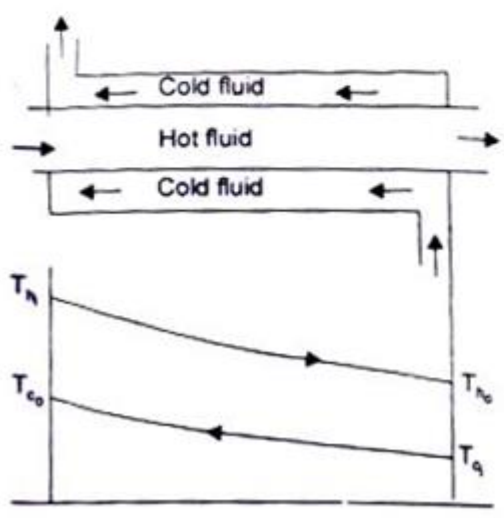


Fig 4.6b Counter-flow heat exchanger with temperature distribution

we find that the temperature difference between the two fluids is more uniform in counter flow than in the parallel flow. Counter flow exchangers give the maximum heat transfer rate and are the most favoured devices for heating or cooling of fluids.

When the two fluids flow through the heat exchanger only once, it is called one-shell-pass and one-tube-pass as shown in Fig. 3.2 and 3.3. If the fluid flowing through the tube makes one pass through half of the tube, reverses its direction of flow, and makes a second pass through the remaining half of the tube, it is called 'one-shell-pass, two-tube-pass' heat exchanger,fig 3.4. Many other possible flow arrangements exist and are being used. Fig. 10.5 depicts a 'two-shell-pass, four-tube-pass' exchanger.

(c) Cross-flow - A cross-flow heat exchanger has the two fluid streams flowing at right angles to each other. Fig. 3.6 illustrates such an arrangement An automobile radiator is a good example of cross-flow exchanger. These exchangers are 'mixed' or 'unmixed' depending upon the mixing or not mixing of either fluid in the direction transverse to the direction of the flow stream and the analysis of this type of heat exchanger is extremely complex because of the variation in the temperature of the fluid in and normal to the direction of flow.

(d) Condenser and Evaporator - In a condenser, the condensing fluid temperature remains almost constant throughout the exchanger and temperature of the colder fluid gradually increases from the inlet to the exit, Fig. 3.7 (a). In an evaporator, the temperature of the hot fluid gradually decreases from the inlet to the outlet whereas the temperature of the colder fluid remains the same during the evaporation process, Fig. 3.7(b). Since the temperature of one of the fluids can be treated as constant, it is immaterial whether the exchanger is parallel flow or counter flow.

(e) Compact Heat Exchangers - these devices have close arrays of finned tubes or plates and are typically used when atleast one of the fluids is a gas. The tubes are either flat or circular as shown in Fig. 10.8 and the fins may be flat or circular. Such heat exchangers are used to a chieve a very large ($\geq$ 700 m^2/mJ) heat transfer surface area per unit volume. Flow passages are typically small and the flow is usually laminar.

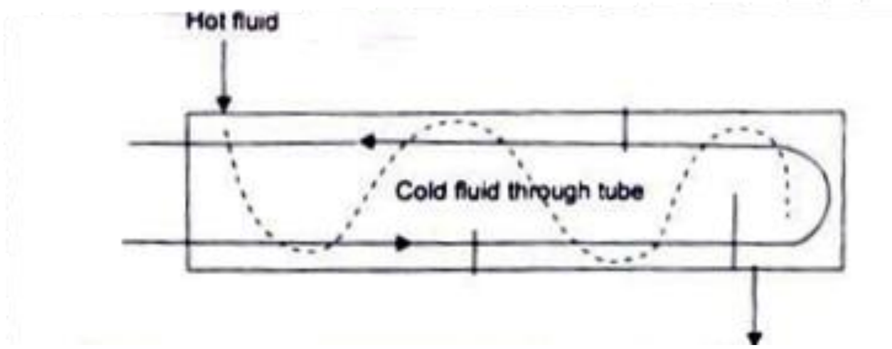


Fig 4.7: multi pass exchanger one shell pass, two shell pass

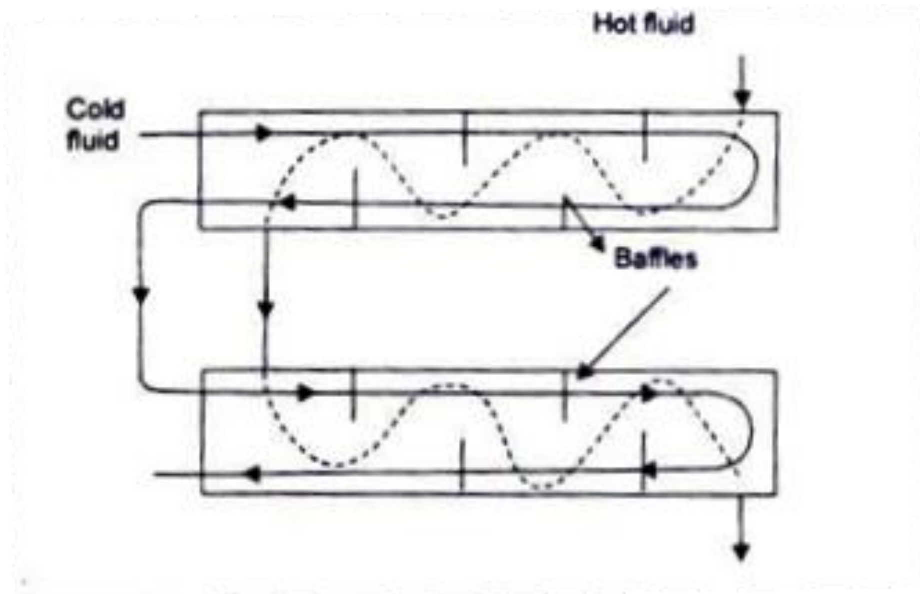


Fig 4.8: Two shell passes, four-tube passes heat exchanger (baffles increases the convection coefficient of the shell side fluid by inducing turbulance and a cross flow velocity component)

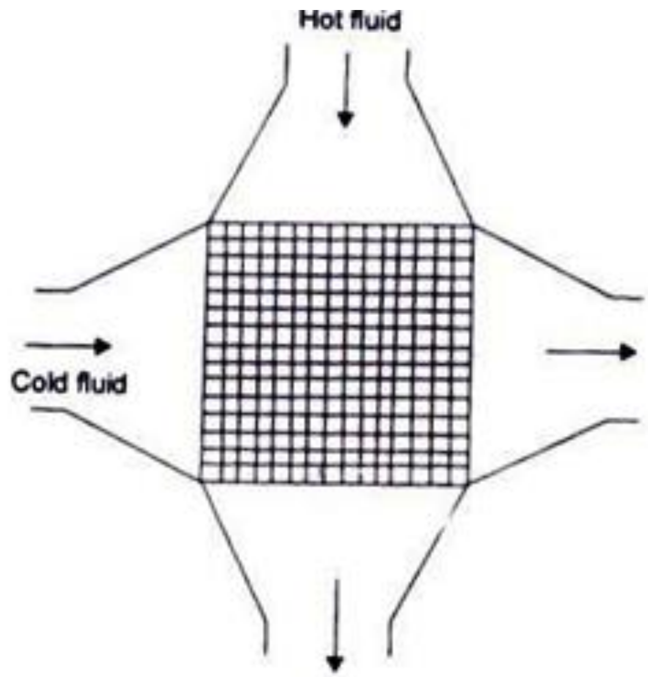


Fig 4.9: A cross-flow exchanger

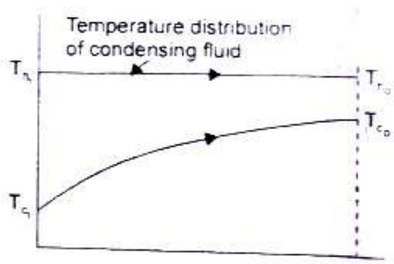


Fig. 10.7 (a) A condenser

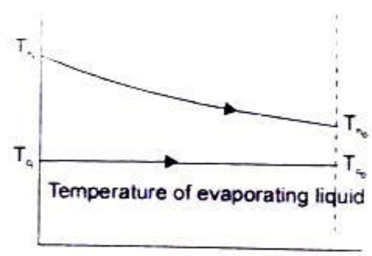


Fig. 10.7 (b) An evaporator

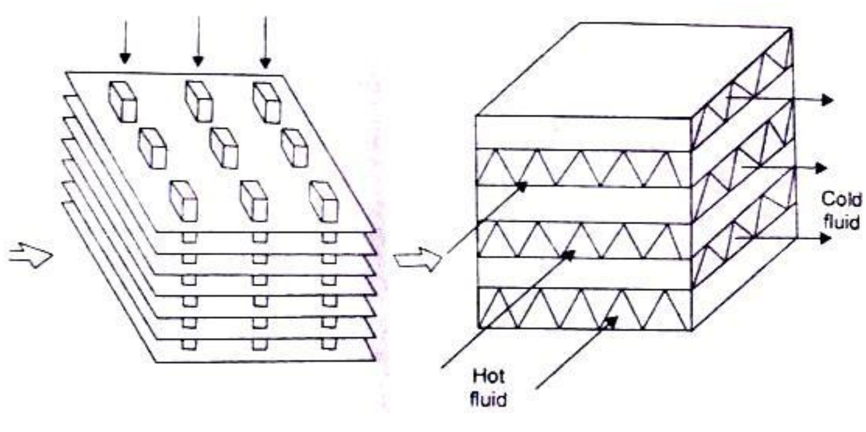


Fig. 4.10 Compact heat exchangers: (a) flat tubes, continuous plate fins, (b) plate fin (single pass)

4.12 Expression for Log Mean Temperature Difference - Its Characteristics

Fig. represents a typical temperature distribution which is obtained in heat exchangers. The rate of heat transfer through any short section of heat exchanger tube of surface area dA is: $dQ = U\,dA(T_h - T_c) = U\,dA\,\square T$. For a parallel flow heat exchanger, the hot fluid cools and the cold fluid is heated in the direction of increasing area. therefore, we may write

$d\dot{Q} = -\dot{m}_h\ c_h\ dT_h = \dot{m}_c\ c_c\ dT_c$ and $d\dot{Q} = -\dot{C}_h\,dT_h = \dot{C}_c dT_c$ where $\dot{C} = \dot{m} \times c$, and is called the 'heat capacity rate.'

Thus, $d(\Delta T) = d(T_h - T_c) = dT_h - dT_c = -(1/C_h + 1/C_c)\,d\dot{Q}$ (3.1)

For a counter flow heat exchanger, the temperature of both hot and cold fluid decreases in the direction of increasing area, hence

$d\dot{Q} = -\dot{m}_h\ c_h\ dT_h = -\dot{m}_c\ c_c\,dT_c$, and $d\dot{Q} = -C_h\,dT_h = -C_c dT_c$

or, $d(\Delta T) = dT_h - dT_c = (1/C_h - 1/C_c)\,d\dot{Q}$ (3.2)

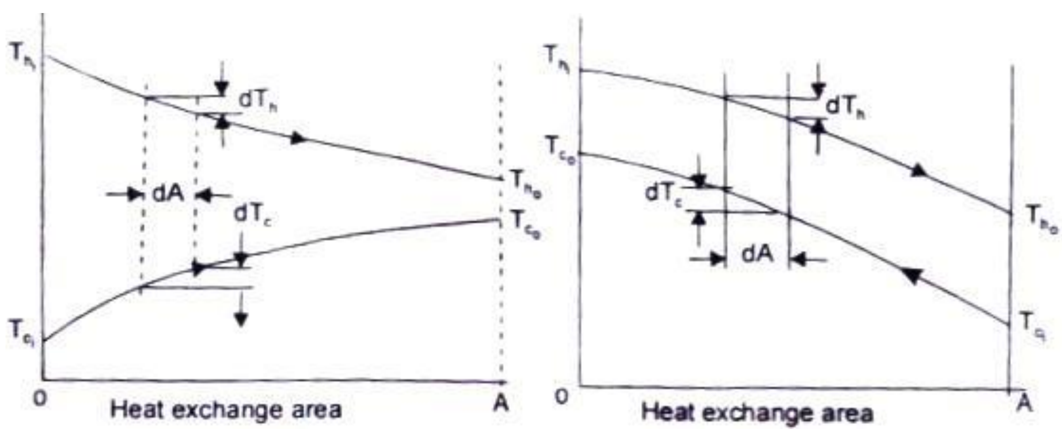


Fig. 4.11 Parallel flow and Counter flow heat exchangers and the temperature distribution with length

Integrating equations (3.1) and (3.2) between the inlet and outlet. and assuming that the specific heats are constant, we get

$-(1/C_h \pm 1/C_c)\dot{Q} = \Delta T_o - \Delta T_i$ (3.3)

The positive sign refers to parallel flow exchanger, and the negative sign to the counter flow type. Also, substituting for dQ in equations (10.1) and (10.2) we get

$$-\left(1/C_h \pm 1/C_c\right)UdA = d\left(\Delta T\right)/\Delta T \qquad (3.3a)$$

Upon integration between inlet i and outlet 0 and assuming U as a constant,

We have $-\left(1/C_h \pm 1/C_c\right)U\,A = \ln\left(\Delta T_0/\Delta T_i\right)$

By dividing (10.3) by (10.4), we get

$$\dot{Q} = UA\left[\left(\Delta T_o - \Delta T_i\right)/\ln\left(\Delta T_o/\Delta T_i\right)\right] \qquad (3.5)$$

Thus the mean temperature difference is written as

Log Mean Temperature Difference,

$$\text{LMTD} = \left(\Delta T_0 - \Delta T_i\right)/\ln\left(\Delta T_o/\Delta T_i\right) \qquad (3.6)$$

(The assumption that U is constant along the heat exchanger is never strictly true but it may be a good approximation if at least one of the fluids is a gas. For a gas, the physical properties do not vary appreciably over moderate range of temperature and the resistance of the gas film is considerably higher than that of the metal wall or the liquid film, and the value of the gas film resistance effectively determines the value of the overall heat transfer coefficient U.)

It is evident from Fig.1 0.9 that for parallel flow exchangers, the final temperature of fluids lies between the initial values of each fluid whereas m counter flow exchanger, the temperature of the colder fluid at exit is higher than the temperature of the hot fluid at exit. Therefore, a counter flow exchanger provides a greater temperature range, and the LMTD for a counter flow exchanger will be higher than for a given rate of mass flow of the two fluids and for given temperature changes, a counter flow exchanger will require less surface area.

4.13 Special Operating Conditions for Heat Exchangers

(i) Fig. 3.7a shows temperature distributions for a heat exchanger (condenser) where the hot fluid has a much larger heat capacity rate, $\dot{C}_h = m_h c_h$ than that of cold fluid, $\dot{C}_c = \dot{m}_c c_c$ and therefore, the temperature of the hot fluid remains almost constant throughout the exchanger and the temperature of the cold fluid increases. The LMTD, in this case is not affected by whether the exchanger is a parallel flow or counter flow.

(ii) Fig. 3.7b shows the temperature distribution for an evaporator. Here the cold fluid expenses a change in phase and remains at a nearly uniform temperature $\left(\dot{C}_c \to \infty\right)$. The same effect would be achieved without phase change if $\dot{C}_c >> \dot{C}_h$, and the LMTD will remain the same for both parallel flow and counter flow exchangers.

(iii) In a counter flow exchanger, when the heat capacity rate of uoth the fluids are equal, $\dot{C}_c = \dot{C}_h$, the temperature difference is the same all along the length of the tube. And in that case, LMTD should be replaced by $\Delta T_a = \Delta T_b$, and the temperature profiles of the two fluids along Its length would be parallel straight lines.

(Since $d\dot{Q} = -\dot{C}_c dT_c = -\dot{C}_h\ dT_h$; $dT_c = -d\dot{Q}/\dot{C}_c$, and $dT_h = -d\dot{Q}/\dot{C}_h$

and, $dT_c - dT_h = d\theta = -dQ\left(1/\dot{C}_c - 1/\dot{C}_h\right) = 0$ (because $\dot{C}_c = \dot{C}_h$)

Or, d□ = 0, gives □ = constant and the temperature profiles of the two fluids

along Its length would be parallel straight lines.)

4.14 LMTD for Cross-flow Heat Exchangers

LMTD given by Eq (10.6) is strictly applicable to either parallel flow or counter flow exchangers. When we have multipass parallel flow or counter flow or cross flow exchangers, LMTD is first calculated for single pass counter flow exchanger and the mean temperature difference is obtained by multiplying the LMTD with a correction factor F which takes care of the actual flow arrangement of the exchanger. Or,

$$\dot{Q} = U\ A\ F\ (LMTD) \tag{3.7}$$

The correction factor F for different flow arrangements are obtained from charts given in Fig. 3.10 (a, b, c, d).

4.15 Fouling Factors in Heat Exchangers

Heat exchanger walls are usually made of single materials. Sometimes the walls are bimettalic (steel with aluminium cladding) or coated with a plastic as a protection against corrosion, because, during normal operation surfaces are subjected to fouling by fluid impurities, rust formation, or other reactions between the fluid and the wall material. The deposition of a film or scale on the surface greatly increases the resistance to heat transfer between the hot and cold fluids. And, a scale coefficient of heat transfer h, is defined as:

$$R_s = 1/h_s A,\ ^{o}C/W \text{ or } K/W$$

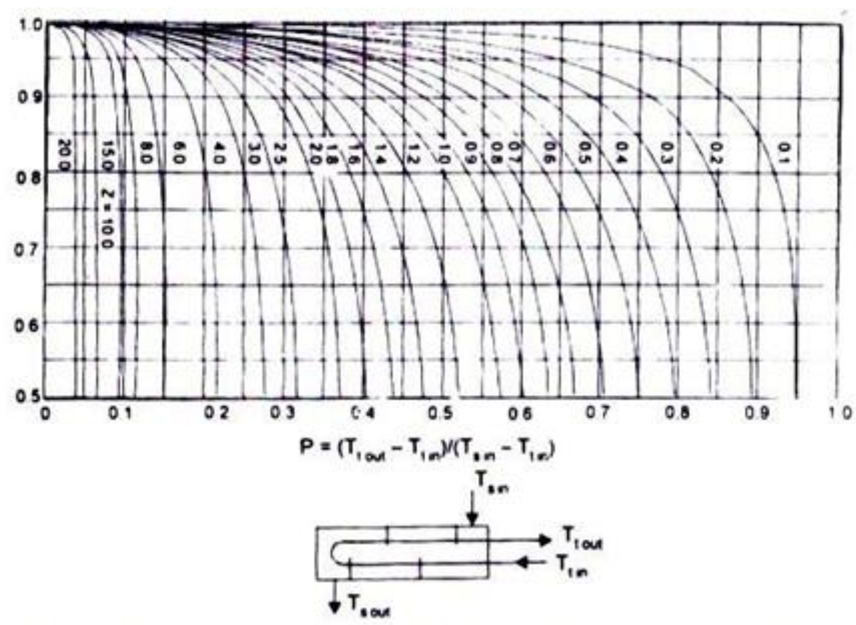


Fig 4.12(a) correction factor to counter flow LMTD for heat exchanger with one shell pass andtwo, or a muliple of two,tube passes

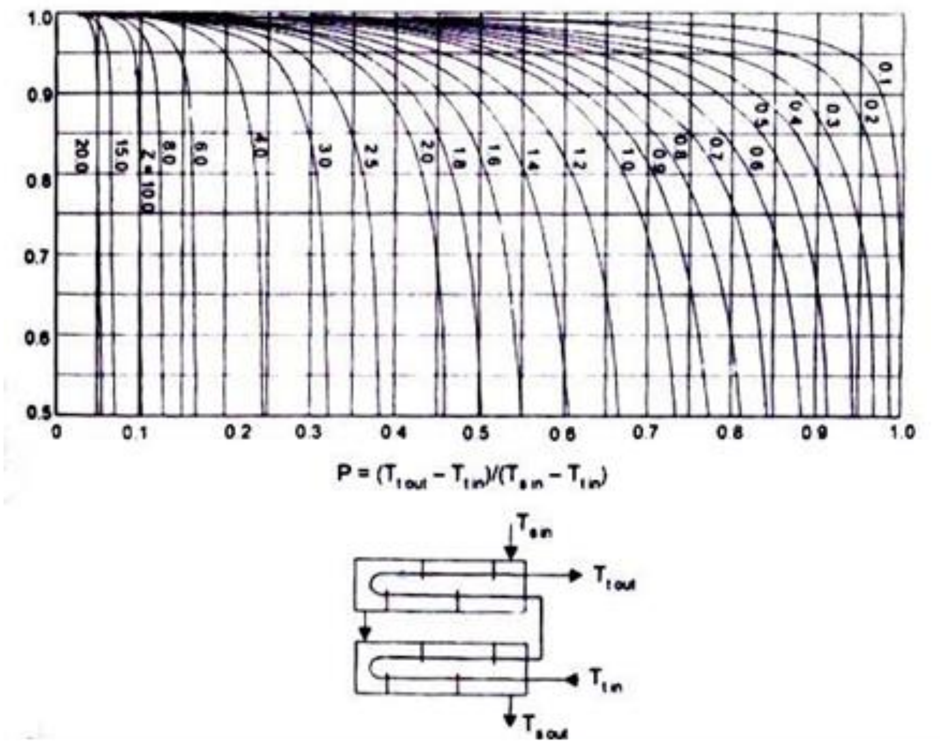


Fig 4.12 (b) Correction factor to counter flow LMTD for heat exchanger with two shell passes and a multiple of two tube passes

where A is the area of the surface before scaling began and l/h_s, is called 'Fouling Factor'. Its value depends upon the operating temperature, fluid velocity, and length of service of the heat exchanger. Table 10.1 gives the magnitude of l/h, recommended for inclusion in the overall heat transfer coefficient for calculating the required surface area of the exchanger

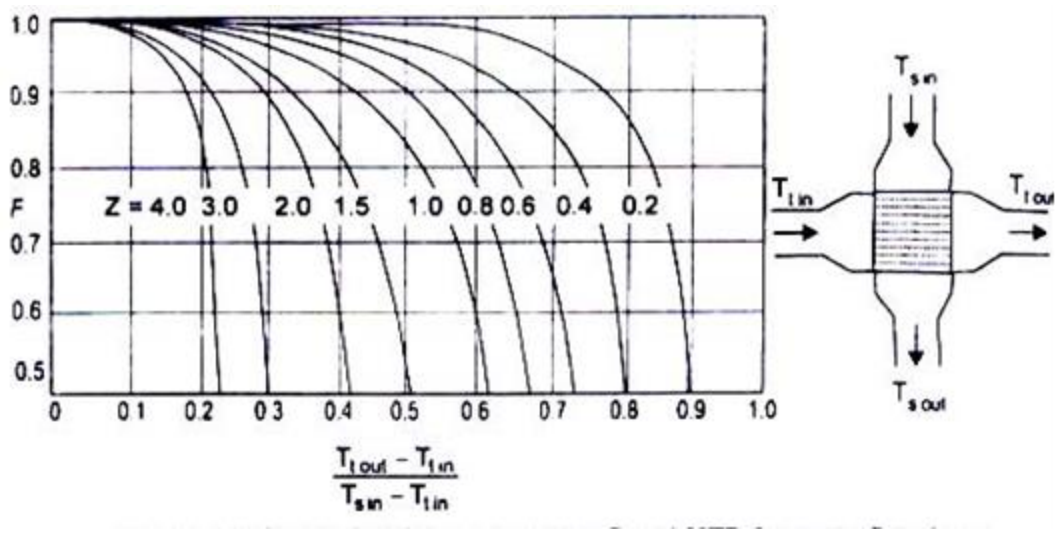


Fig4.12(c) Correction factor to counter flow LMTD for cross flow heat exchangers, fluid on shell side mixed, other fluid unmixed one tube pass..

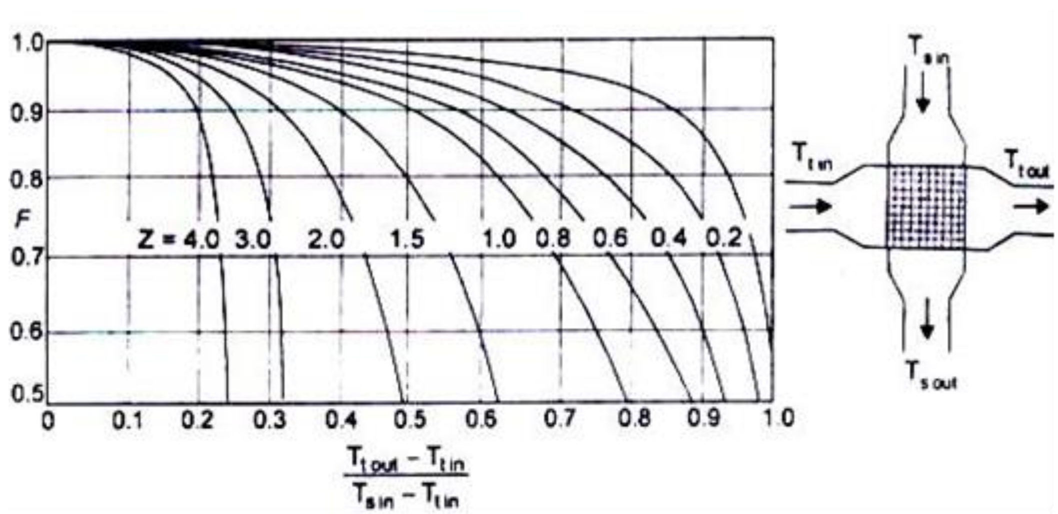


Fig. 4.12 (d) Correction factor to counter flow LMTD for cross flow heat exchangers, both fluids unmixed, one tube pass..

Table 4.1 Representative fouling factors ($1/h_s$)

Type of fluid	Fouling factor	Type of fluid	Fouling Factor
Sea water below 50°C	000009 m'K/W	Refrigerating liquid	0.0002 m'K/W
above 50°C	0.002		
Treated feed water	0.0002	Industrial air	0.0004
Fuel oil	0.0009	Steam, non-oil-bearing	0.00009
Quenching oil	0.0007	Alcohol vapours	0.00009

However, fouling factors must be obtained experimentally by determining the values of U for both clean and dirty conditions in the heat exchanger.

4.16 The Overall Heat Transfer Coefficient

The determination of the overall heat transfer coefficient is an essential, and often the most uncertain, part of any heat exchanger analysis. We have seen that if the two fluids are separated by a plane composite wall the overall heat transfer coefficient is given by:

$$1/U = (1/h_i) + (L_1/k_1) + (L_2/k_2) + (1/h_o) \tag{3.8}$$

If the two fluids are separated by a cylindrical tube (inner radius r_i, outer radius r_0), the overall heat transfer coefficient is obtained as:

$$1/U_i = (1 + h_i) + (r_i/k)\ln(r_o/r_i) + (r_i/r_o)(1/h_o) \tag{3.9}$$

where h_i, and h_o are the convective heat transfer coefficients at the inside and outside surfaces and V, is the overall heat transfer coefficient based on the inside surface area. Similarly, for the outer surface area, we have:

$$1/U_o = (1/h_o) + (r_o/k)\ln(r_o + r_i) + (r_o + r_i)(1/h_i) \tag{3.10}$$

and U_iA_i will be equal to U_oA_o; or, $U_ir_i = U_or_o$.

The effect of scale formation on the inside and outside surfaces of the tubes of a heat exchanger would be to introduce two additional thermal resistances to the heat flow path. If h_{si} and h_{so} are the two heat transfer coefficients due to scale formation on the inside and outside surface of the inner pipe, the rate of heat transfer is given by

$$Q=\left(T_i-T_o\right)/\left[\left(1/h_iA_i\right)+1/h_{si}A_i+\ln\left(r_o+r_i\right)/2\pi Lk+1/h_{so}A_o+\left(1/h_oA_o\right)\right] \qquad (3.11)$$

where T_i, and T_o are the temperature of the fluid at the inside and outside of the tube. Thus, the overall heat transfer coeffiCIent based on the inside and outside surface area of the tube would be:

$$1/U_i=1/h_i+1/h_{si}+\left(r_i/k\right)\ln\left(r_o/r_i\right)+\left(r_i/r_o\right)\left(1/h_{so}\right)+\left(r_i/r_o\right)\left(1/h_o\right); \qquad (3.12)$$

and

$$1/U_o=\left(r_o/r_i\right)\left(1/h_i\right)+\left(r_o/r_i\right)\left(1/h_{si}\right)+\ln\left(r_o/r_i\right)\left(r_0/k\right)+1/h_{so}+1/h_o$$

Example 4.1 In a parallel flow heat exchanger water flows through the inner pipe and is heated from 25°C to 75°C. Oil flowing through the annulus is cooled from 210°C to 110°C. It is desired to cool the oil to a lower temperature by increasing the length of the tube. Estimate the minimum temperature to which the oil can be cooled.

Solution: By making an energy balance, heat received by water must be equal to 4he heat given out by oil.

$$\dot{m}_wc_w\left(75-25\right)=\dot{m}_oc_o\left(210-110\right);\dot{C}_w/\dot{C}_o=100/50=2.0$$

In a parallel flow heat exchanger, the minimum temperature to which oil can be cooled will be equal to the maximum temperature to which water can be heated,

Fig. 10.2: $\left(T_{ho}=T_{co}\right)$

therefore, C_w (T –25) = C_o (210 – T);

(T – 25)/(210 – T) = 1/2 = 0.5; or, T = 260/3 = 86.67°C.

or the same capacity rates the oil can be cooled to 25°C (equal to the water inlet temperature) in a counter-flow arrangement.

Example 4.2 Water at the rate of 1.5 kg/s IS heated from 30°C to 70°C by an oil (specific heat 1.95 kJ/kg C). Oil enters the exchanger at 120°C and leaves the exchanger at 80°C. If the overall heat transfer coefficient remains constant at 350 W /m^2°C, calculate the heat exchange area for (i) parallel-flow, (ii) counter-flow, and (iii) cross-flow arrangement.

Solution: Energy absorbed by water,

$$\dot{Q} = \dot{m}_w c_w \left(\Delta T\right) = 1.5 \times 4.182 \times 40 = 250.92 \text{ kW}$$

(i) Parallel flow: Fig. 10.9; $\Delta T_a = 120 - 30 = 90$; $\Delta T_b = 80 - 70 = 10$

LMTD = (90 – 10)/ln(90/10) = 36.4;

Area = Q/U (LMTD) = 250920 /(350 × 36.4) = 19.69 m^2.

(ii) Counter flow: Fig 10.9; ΔT_a. = 120 – 70 = 50, ΔT_b = 80 – 30 = 50

Since ΔT_a. = ΔT_b,. LMTD should be replaced by ΔT = 50

Area A = $\dot{Q}/U\left(\Delta T\right) = 250920/\left(350 \times 50\right) = 14.33 \text{ m}^2$

(iii) Cross flow: assuming both fluids unmixed - Fig. 10.10d

using the nomenclature of the figure and assuming that water flows through the tubes and oil flows through the shell,

$$P = \left(T_{to} - T_{ti}\right)/\left(T_{si} - T_{ti}\right) = \left(70 - 30\right)/\left(120 - 30\right) = 0.444$$

$$Z = \left(T_{si} - T_{so}\right)/\left(T_{to} - T_{ti}\right) = \left(120 - 80\right)/\left(70 - 30\right) = 1.0$$

and the correction factor, F = 0.93

$\dot{Q} = UAF\left(\Delta T\right)$; or Area A = 250920/(350 × 0.93 × 50) = 15.41 m^2.

Example 4.3 0.5 kg/s of exhaust gases flowing through a heat exchanger are cooled from 400°C to 120°C by water initially at 25°C. The specific heat capacities of exhaust gases and water are 1.15 and 4.19 kJ/kgK respectively, and the overall heat transfer coefficient from gases to water is 150 W/m^{2}K. If the cooling water flow rate is 0.7 kg/s, calculate the surface area when (i) parallel-flow (ii) cross-flow with exhaust gases flowing through tubes and water is mixed in the shell.

Solution: The heat given out by the exhaust gases is equal to the heat gained by water.

or, $0.5 \times 1.15 \times (400 - 120) = 0.7 \times 4.19 \times (T - 25)$

Therefore, the temperature of water at exit, T = 79.89°C

For paralle1-flow: $\Delta T_a = 400 - 25 = 375$; $\Delta T_b = 120 - 79.89 = 40.11$

LMID = (375 – 40.11)/ln(375/40.11) = 149.82

$\dot{Q}$ = 0.5 × 1.15 × 280 = 161000 W;

Therefore Area A = 161000/(150 × 149.82) = 7.164 m^2

For cross-flow: $\dot{Q}$ = U A F (LMTD);

and LMTD is calculated for counter-flow system.

$$\Delta T_a = (400 - 79.89) = 320.11; \; \Delta T_b = 120 - 25 = 95$$

$$\text{LMTD} = (320.11 - 95)/\ln(320.11/95) = 185.3$$

Using the nomenclature of Fig 10.10c,

$$P = (120 - 400)/(25 - 400) = 0.747$$

$$Z = (25 - 79.89)/(120 - 400) = 0.196 \qquad \therefore \; F = 0.92$$

and the area $A = 161000/(150 \times 0.92 \times 185.3) = 6.296 \text{ m}^2$

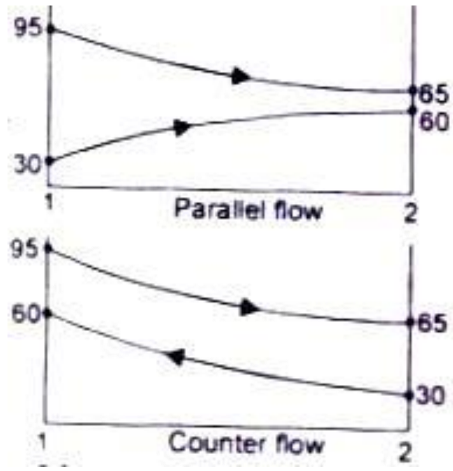
Example 4.4 In a certain double pipe heat exchanger hot water flows at a rate of 5000 kg/h and gets cooled from 95°C to 65°C. At the same time 5000 kg/h of cooling water enters the heat exchanger. The overall heat transfer coefficient is 2270 W/m^2K. Calculate the heat transfer area and the efficiency assuming two streams are in (i) parallel flow (ii) counter flow. Take C_p for water as 4.2 kJ/kgK, cooling water inlet temperature 30°C.

Solution: By making an energy balance:

Heat lost by hot water = 5000 ×4.2 ×(95 –65)

= heat gained by cold water = 5000 ×4.2 ×(T –30) 30

T = 60^{o}C



(i) Parallel now

$$\theta_1 = (95 - 30) = 65$$

$$\theta_2 = (65 - 60) = 5$$

LMTD = $(65 - 5)/\ln(65/5) = 23.4$

Area, $A = \dot{Q}/(U \times LMTD) = \dfrac{500 \times 4.2 \times 10^3 \times 30}{3600 \times 2270 \times 23.4} = 3.295\ m^2$

(ii) Counter flow: θ_1 = (95 – 60) = 35

θ_2 = (65 – 30) = 35

LMTD = □T = 35

Area A = 500 × 4200 × 30/(3600 × 2270 × 35) = 2.2 m^2

$\in$, Efficiency.= Actual heat transferred/Maximum heat that could be transferred. Therefore, for parallel flow, $\in$ = (95 – 65)/(95 – 60) = 0.857

For counter flow, $\in$ = (95 - 65)/(95 – 30) = 0.461.

Counter now

Example 4.5 The flow rates of hot and cold water streams running through a double pipe heat exchanger (inside and outside diameter of the tube 80 mm and 100 mm) are 2 kg/s and 4 kg/so The hot fluid enters at 75°C and comes out at 45^{o}C. The cold fluid enters at 20°C. If the convective heat transfer at the inside and outside surface of the tube is 150 and 180 W /m^{2}K, thermal conductivity of the tube material 40 W/mK, calculate the area of the heat exchanger assuming counter flow.

Solution: Let T is the temperature of the cold water at outlet.

By making an energy balance, $\dot{Q} = \dot{m}_h c_h \left(T_{h1} - T_{h2}\right) = \dot{m}_c c_c \left(T_{c2} - T_{c1}\right)$

since $c_h = c_c$, $4.2\ \text{kJ/kgK};\ 2 \times (75 - 45) = 4 \times (T - 20);\ T = 35^{o}C$

and $\dot{Q} = 252\ \text{kW}$

for counter flow: $\theta_1 = (75 - 35) = 40;\ \theta_2 = (45 - 20) = 25$

LMTD = (40 – 25) / ln (40/25) = 31.91

overall heat transfer coefficient based in the inside surface of tube

$1/U = (1/h_i) + (r_i/k)\ln(r_o/r_i) + (r_o r_i)(1/h_o)$

$= 1/150 + (0.04/40)\ln(50/40) + (50/40)(1/180) = 0.0138$

and U = 72.28

area $A = \dot{Q}/(U \times \text{LMTD}) = 252 \times 10^3/(72.28 \times 31.91) = 109.26\ \text{m}^2$

Example 4.6 Water flows through a copper tube (k = 350 W/mK, inner and outer diameter 2.0 cm and 2.5 cm respectively) of a double pipe heat exchanger. Oil flows through the annulus between this pipe and steel pipe. The convective heat transfer coefficient on the inside and outside of the copper tube are 5000 and 1500 W /m2K. The fouling factors on the water and oil sides are 0.0022 and 0.00092 K1W. Calculate the overall heat transfer coefficient with and without the fouling factor.

Solution: The scales formed on the inside and outside surface of the copper tube introduces two additional resistances in the heat flow path. Resistance due to inside convective heat transfer coefficient

$$1/h_i A_i = 1/5000\ A_i$$

Resistance due to scale formation on the inside = $1/h_s A_i$ = 0.0022

Resistance due to conduction through the tube wall = $\ln(r_o/r_i)/2\pi Lk$

$$= \ln(2.5/2.0)/2\pi \times L \times 350 = 1.014 \times 10^{-4}/L$$

Resistance due to convective heat transfer on the outside

$$1/h_o A_o = 1/1500 A_o$$

Resistance due to scale formation on the outside = $1/h_s A_o = 0.00092$

Since, $Q = \Delta T \sum R = U_i A_1 (\Delta T) = \Delta T/(1/U_i A_i)$; we have

(a) With fouling factor:-

Overall heat transfer coefficient based on the inside pipe surface

$$U_i = 1/\left(1/5000 + \pi \times 0.02(0.0022 + 0.00092) + 0.02\pi \times 1.014 \times 10^{-4} + 8.33 \times 10^{-4}\right)$$

= 809.47 W/m^{2}K per metre length of pipe

(b) Without fouling factor

$$U_i = 1/\left(1/5000 + 0.02\pi \times 1.014 \times 10^{-4} + 8.33 \times 10^{-4}\right)$$

= 962.12 W/m^{2}K per m of pipe length.

The heat transfer rate will reduce by (962.12 – 809.47)1962.12 = 15.9 percent when fouling factor is considered.

Example 4.7 In a surface condenser, dry and saturated steam at 50^{o}C enters at the rate of 1 kg/so The circulating water enters the tube, (25 nun inside diameter, 28 mm outside diameter, k = 300 W/mK) at a velocity of 2 m/s. If the convective heat transfer coefficient on the outside surface of the tube is 5500 W/m^{2}K, the inlet and outlet temperatures of water are 25^{o}C and 35^{o}C respectively, calculate the required surface area.

Solution: For calculating the convective heat transfer coefficient on the inside surface of the tube, we calculate the Reynolds number on the basis of properties of water at the mean temperature of 30°C. The properties are:

μ = 0.001 Pa-s, □ = 1000 kg/m^3, k = 0.6 W/mK, h_{fg} at 50°C = 2375 kJ/kg

Re = □VD/□ = 10^3 × 2 × 0.025/0.001 = 50,000, a turbulent flow. Pr = 7.0.

The heat transfer coefficient at the inside surface can be calculated by:

Nu = 0.023 $Re^{0.8}$ 8 $Pr^{0.3}$ = 0.023 $(50000)^{0.8}$ $(7)^{0.3}$ = 236.828

and h_i = 236.828 × 0.6/0.025 = 5684 W/m^2K.

The overall heat transfer coefficient based on the outer diameter,

U = 1/(0.028/(0.025 × 5684) + 1/5500 + 0.014 ln(28/25)/300)

= 2603.14 W/m^2K

□T_a. = (50 - 25) = 25; □T_b = (50 - 35) = 15;

□□□□LMTD = (25 -15)/ln(25/15) = 19.576.

Assuming one shell pass and one tube pass, Q = UA (LMTD)

or A = 2375 × 10^3/(2603.14 x 19.576) = 46.6 m^2

Mass of Circulating water = Q/(c_p □T) = 2375/(4.182 × 10) = 56.79 kg/s

also, m_w = □ × area × V × n, where n is the number of tubes.

n = 56.79 × 4/(2 × □ × 0.025 × 0.025 × 1000) = 58 tubes

Surface area, 46.6 = n × □ × d × L

and L = 46.6/(58 × □ × 0.025) = 10.23 m.

Hence more than one pass should be used.

Example 4.8 A heat exchanger is used to heat water from 20°C to 50°C when thin walled water tubes (inner diameter 25 mm, length IS m) are laid beneath a hot spring water pond, temperature 75°C. Water flows through the tubes with a velocity of 1 m/s. Estimate the required overall heat transfer coefficient and the convective heat transfer coefficient at the outer surface of the tube.

Solution: Water flow rate, $\dot{m}$ = □ × V × A = 10^3 × 1 × (□/4) $(0.025)^2$

= 0.49 kg/so

Heat transferred to water, Q = $\dot{m}c$ (□T) = 0.49 × 4200 × 30 = 61740 W.

Since the temperature of the water in the hot spring is constant,

$\theta_1 = (75-20) = 55;\ \theta_2 = (75-50) = 25;$

LMTD = (55 – 25) / ln(55/25) = 38

Overall heat transfer coefficient, U = Q/ (A × LMTD)

= 61740/(38 × □ × 0.025 × 15) = 1378.94 W/m^2K.

The properties of water at the mean temperature (20 + 50)/2 = 35°C are:

$\mu = 0.001\ \text{Pa}-\text{s},\ k = 0.6\ \text{W/mk}$ and Pr = 7.0

Reynolds number, $\text{Re} = \rho Vd/\mu = 1000 \times 1.0 \times 0.25/0.001 = 25000$, turbulent flow.

Nu = 0.023 $(\text{Re})^{0.8}$ $(\text{Pr})^{0.33}$ = 0.023 $(25000)^{0.8}$ × $(7)^{0.33}$ = 144.2

and h_i = 144.2 × k/d = 144.2 × 0.6/0.025 = 3460.8 W/m^2K

Neglecting the resistance of the thin tube wall,

1/U = 1/h_i + 1/h_o; $\therefore\ 1/h_o = 1/1378.94 = 1/3460.8$

or, $h_o = 2292.3\ \text{W}/\text{m}^2\text{K}$

Example 4.9 A hot fluid at 200°C enters a heat exchanger at a mass rate of 10000 kg/h. Its specific heat is 2000 J/kg K. It is to be cooled by another fluid entering at 25°C with a mass flow rate 2500 k g/h and specific heat 400 J/kgK. The overall heat transfer coefficient based on outside area of 20 m2 is 250 W/m^2K. Find the exit temperature of the hot fluid when the fluids are in parallel flow.

Solution: From Eq(10.3a), $-U\ dA\left(1/C_h + 1/C_c\right) = d\left(\Delta T\right)/\Delta T$

Upon integration,

$$-U\,A\left(1/C_h + 1/C_c\right) = \ln\left(\Delta T\right)\Big|_1^2 \ = \ln\left(T_{h_0} - T_{c_0}\right)/\left(T_{h_i} - T_{c_i}\right)$$

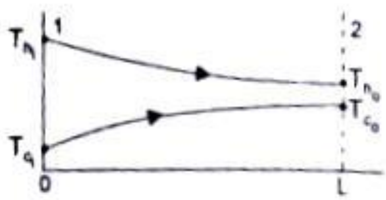
The values are: $U = 250\ W/m^2K$

$A = 20\ m^2$

$l/C_h = 3600/(10000 \times 2000) = 1.8 \times 10^{-4}$

$1/C_c = 3600/(2500 \times 400) = 3.6 \times 10^{-3}$

$-UA(1/C_h + l/C_c) = -250 \times 20\ (1.8 \times 10^{-4} + 3.6 \times 10^{-3}) = -18.9$



; or, $T_{h_0} = T_{c_0}$

By making an energy balance,

$$10000 \times 2000\left(200 - T_{h_0}\right) = 2500 \times 400\left(T_{c_0} - 25\right)$$

$$= 2500 \times 400\left(T_{h_0} - 25\right) \text{ and } 21\ T_{h_0} = 20 \times 200 + 25$$

or, $T_{h_0} = 191.67^{o}C$

Example 4.10 Cold water at the rate of 4 kg/s is heated from 30°C to 50°C in a shell and tube heat exchanger with hot water entering at 95°C at a rate of 2 kg/s. The hot water flows through the shell. The cold water flows through tubes 2 cm inner diameter, velocity of flow 0.38 m/s. Calculate the number 0 f tube passes, the number 0 f tubes per pass if the maximum length of the tube is limited to 2.0 m and the overall heat transfer coefficient is 1420 W/m^2K.

Solution: Let T be the temperature of the hot water at exit. By making an energy balance: $4c\left(50-30\right) = 2c\left(95-T\right); \quad \therefore T = 55^{o}C$

For a counter-flow arrangement:

$\Delta T_a = \left(95-50\right) = 45$, $\Delta T_b = \left(55-30\right) = 25$,

$\therefore$ LMTD $= (45-25)/\ln(45/25) = 34$; $Q = mC(\Delta T) = 4 \times 4.182 \times 20 = 334.56$ kW

Since the cold water is flowing through the tubes, the number of tubes, n is given by

$\dot{m} = n \times \rho \times$ Area × velocity; the cross-sectional area 3.142 × 10^{-2} m^2

4 = n × 1000 × 3.142 × 10^{-4} × 0.38; □ n = 33.5, or 34 (say)

Assuming one shell and two tube pass, we use Fig. 10.9(a).

$P(50-30)/(95-30) = 0.3;\quad Z = (95-55)/(50-30) = 2.0$

Therefore, the correction factor, F = 0.88

Q = UAF LMTD; 34560 = 1420 × A × 0.88 × 34; or A = 7.875 m^2.

For 2 tube pass, the surface area of 34 tubes per pass = 2 L□d 34

L = 1.843 m

Thus we will have 1 shell pass, 2 tube; 34 tubes of 1.843 m in length.

Example 4.11 A double pipe heat exchanger is used to cool compressed air (pressure A bar, volume flow rate 5 mJ/mm at I bar and 15°C) from 160°C to 35°C. Air flows with a velocity of 5 m/s through thin walled tubes, 2 cm inner diameter. Cooling water flows through the annulus and its temperature rises from 25°C to 40°C. The convective heat transfer coefficient at the inside and outside tube surfaces are 125 W/m^2K and 2000 W/m^2K respectively. Calculate (i) mass of water flowing through the exchanger, and (ii) number of tubes and length of each tube.

Solution: Air is cooled from 1600C to 35°C while water is heated from 25°C to 400C and therefore this must be a counter flow arrangement.

Temperature difference at section 1 : $\left(T_{h_i} - T_{c_0}\right) = (160-40) = 120$

Temperature difference at section 2 : $\left(T_{h_0} - T_{c_i}\right) = (35-25) = 10$

LMTD = (120 –10)/ln 120/10) = 44.27

Mass of air flowing, $\dot{m}$ = □ × Volume = $\left(10^5/287 \times 288\right)(5/60) = 0.1$ kg/s

Heat given out by air = Heat taken in by water,

$\therefore\ 0.1 \times 1.005 \times (160 - 35) = \dot{m}_w \times 4.182 \times (40 - 25)$; Or $\dot{m}_w = 0.20\ \text{kg/s}$

Density of air flowing through the tube, □ = p/RT. The mean temperature of air flowing through the tube is (160 + 35)/2 = 97.5 ^{o}C = 370.5K

□□□□□□ = 4 × 10^5/(287 × 370.5) = 3.76 kg/m^3. If n is the number of tubes, from the conservation of mass, $\dot{m} = \rho AV$; 0.1 = 3.76 × (□/4) $(0.02)^2$ × 5 × n

□□□n = 16.9 ≡ 17 tubes; $\dot{Q}$ = UA (LMTD)

U = 1/(1/2000 + 1/125) = 117.65, Area for heat transfer A = □DLn

Q = UA(LMTD); 0.1 × 1005 × 125 = 117.65 × 3.142 × 0.02 × L × 17 × 44.27 and L = 2.26 m.

Example 4.12 A refrigerant (mass rate of flow 0.5 kg/s, S = 907 J/kgK k = 0.07 W/mK, □ = 3.45 × 10^{-4} Pa-s) at –20°C flows through the annulus (inside diameter 3 c m) of a double pipe counter flow heat exchanger used to cool water (mass flow rate 0.05 kg/so k = 0.68 W/mK, □ = 2.83 × 10^{-4} Pa-s) at 98°C flowing through a thin walled copper tube of 2 cm inner diameter. If the length of the tube is 3m, estimate (i) the overall heat transfer coefficient, and (ii) the temperature of the fluid streams at exit.

Solution: Mass rate of flow, $\dot{m} = \rho AV = \rho(\pi/4)D^2V$;

$\rho VD = 4\dot{m}/\pi D$ and, Reynolds number, $Re = \rho VD/\mu = 4\dot{m}/\pi D\mu$

Water is flowing through the tube of diameter 2 cm,

$\therefore\ Re = 4 \times 0.05 / \left(3.142 \times 0.02 \times 2.83 \times 10^{-4}\right) = 1.12 \times 10^4$, turbulent flow.

$$Nu = 0.023\ Re^{0.8}(Pr)^{0.33} = 0.023\left(1.12 \times 10^4\right)^{0.8}(1.8)^{0.33}$$

= 48.45; and $h_i = Nu \times k / D = 48.45 \times 0.68 / 0.02 = 1647.3\ \text{W/m}^2\text{K}$

Refrigerant is flowing through the annulus. The hydraulic diameter is

D_o - D_i, and the Reynolds number would be, $Re = 4m/\mu\pi\left(D_0 + D_i\right)$

$Re = 4 \times 0.5/\left(3.45 \times 10^{-4} \times 3.142 \times \left(0.02 + 0.03\right)\right)$ = 3.69 × 10^4, a turbulent flow.

$Nu = 0.023\left(Re\right)^{0.8}\left(Pr\right)^{0.33}$,

where $Pr = \mu\ c/k = 3.45 \times 10^{-4} \times 907/0.07 = 4.47$

$= 0.023\left(3.69 \times 10^4\right)^{0.8}\left(4.47\right)^{0.33} = 169.8$

$\therefore\ h_o = nu \times k/\left(D_o - D_i\right) = 169.8 \times 0.07/0.01 = 1188.6\ W/m^2K$

and, the overall heat transfer coefficient, U = 1/(1/1647.3 + 1/1188.6)

= 690.43 W/m^2K

For a counter flow heat exchanger, from Eq. (10.4), we have,

$\left(1/C_c - 1/C_h\right)UA = \ln\left(\Delta T_0/\Delta T_i\right) = \ln\left[\left(T_{h_0} - T_{c_i}\right)/\left(T_{h_i} - T_{c_0}\right)\right]$

$C_c = 0.5 \times 907 = 453.5;\ C_h = 0.05 \times 4182 = 209.1$

$1/C_c - 1/C_h UA = \left(1/453.5 - 1/209.1\right) \times 690.43 \times 3.142 \times 0.02 \times 3\ = –0.335$

$\therefore\ \left(T_{h_0} - T_{c_i}\right)/\left(T_{h_i} - T_{c_0}\right) = \exp\left(-0.335\right) = 0.715$

or, $\left(T_{h_0} + 20\right)/\left(98 - T_{c_0}\right) = 0.715$; By making an energy balance,

$453.5\left(T_{c_0} + 20\right) = 209.1\left(98 - T_{h_0}\right)$

which gives $T_{c_0} = 3.12^oC; T_{h_0} = 47.8^oC$

4.17 Heat Exchangers Effectiveness - Useful Parameters

In the design of heat exchangers, the efficiency of the heat transfer process is very important. The method suggested by Nusselt and developed by Kays and London is now being extensively used. The effectiveness of a heat exchanger is defined as the ratio of the actual heat transferred to the maximum possible heat transfer.

Let $\dot{m}_h$ and $\dot{m}_c$ be the mass flow rates of the hot and cold fluids, c_h and c_c be the respective specific heat capacities and the terminal temperatures be Th and T h for the hot fluid at inlet and outlet, T_{h_i} and T_{h_0} for the cold fluid at inlet and outlet. By making an energy balance and assuming that there is no loss of energy to the surroundings, we write

$$\dot{Q} = \dot{m}_h\ c_h\left(T_{h_i} - T_{h_0}\right) = \dot{C}_h\left(T_{h_i} - T_{c_0}\right), \text{ and}$$

$$= \dot{m}_c c_c\left(T_{c_0} - T_{c_i}\right) = \dot{C}_c\left(T_{c_0} - T_{c_i}\right) \qquad (3.13)$$

From Eq. (10.13), it can be seen that the fluid with smaller thermal capacity, C, has the greater temperature change. Further, the maximum temperature change of any fluid would be $\left(T_{h_i} - T_{c_i}\right)$ and this Ideal temperature change can be obtained with the fluid which has the minimum heat capacity rate. Thus,

$$\text{Effectiveness, } \in = \dot{Q}/C_{min}\left(T_{h_i} - T_{c_i}\right) \qquad (3.14)$$

Or, the effectiveness compares the actual heat transfer rate to the maximum heat transfer rate whose only limit is the second law of thermodynamics. An useful parameter which also measures the efficiency of the heat exchanger is the 'Number of Transfer Units', NTU, defined as

NTU = Temperature change of one fluid/LMTD.

Thus, for the hot fluid: NTU = $\left(T_{h_i} - T_{h_0}\right)/\text{LMTD}$, and

for the cold fluid: $\text{NTU} = \left(T_{c_0} - T_{c_i}\right)/\text{LMTD}$

Since $\dot{Q} = UA(\text{LMTD}) = C_h\left(T_{h_i} - T_{h_0}\right) = \dot{C}_c\left(T_{c_0} - T_{c_i}\right)$

we have $\text{NTU}_h = UA/C_h$ and $\text{NTU}_c = UA/C_c$

The heat exchanger would be more effective when the NTU is greater, and therefore,

$$\text{NTU} = AU/C_{min} \qquad (3.15)$$

An another useful parameter in the design of heat exchangers is the ratio the minimum to the maximum thermal capacity, i.e., $R = C_{min}/C_{max}$,

where R may vary between I (when both fluids have the same thermal capacity) and 0 (one of the fluids has infinite thermal capacity, e.g., a condensing vapour or a boiling liquid).

4.18 Effectiveness - NTU Relations

For any heat exchanger, we can write: $\in = f\left(\text{NTU}, C_{min}/C_{max}\right)$. In order to determine a specific form of the effectiveness-NTU relation, let us consider a parallel flow heat exchanger for which $C_{min} = C_h$. From the definition of effectiveness (equation 10.14), we get

$$\varepsilon = \left(T_{h_i} - T_{h_0}\right)/\left(T_{h_i} - T_{c_i}\right)$$

and, $C_{min}/C_{max} = C_h/C_c = \left(T_{c_0} - T_{c_i}\right)/\left(T_{h_i} - T_{h_0}\right)$ for a parallel flow heat exchanger, from Equation 10.4,

$$\ln\left(T_{h_0} - T_{c_0}\right)/\left(T_{h_i} - T_{c_i}\right) = -UA\left(1/C_h + 1/C_c\right) = \frac{-UA}{C_{min}}\left(1 + C_{min}/C_{max}\right)$$

or, $\left(T_{h_0} - T_{c_0}\right)/\left(T_{h_i} - T_{c_i}\right) = \exp\left[-\text{NTU}\left(1 + C_{min}/C_{max}\right)\right]$

But, $\left(T_{h_0} - T_{c_0}\right)/\left(T_{h_i} - T_{c_i}\right) = \left(T_{h_0} - T_{h_i} + T_{h_i} - T_{c_0}\right)/\left(T_{h_i} - T_{c_i}\right)$

$$= \left[\left(T_{h_0} - T_{h_i}\right) + \left(T_{h_i} - T_{c_i}\right) - \left\{R\left(T_{h_i} - T_{h_0}\right)\right\}\right]/\left(T_{h_i} - T_{c_i}\right)$$

$$= \in + 1 - R \in = 1 - \in\left(1 + R\right)$$

Therefore, $\in = \left[1 - \exp\left\{-\text{NTU}(1 + R\right\}\right]/\left(1 + R\right)$

$$\text{NTU} = -\ln\left[1 - \in\left(1 + R\right)\right]/\left(1 + R\right)$$

Similarly, for a counter flow exchanger, $\varepsilon = \dfrac{\left[1 - \exp\left\{-\text{NTU}(1 - R)\right\}\right]}{\left[1 - R\exp\left\{-\text{NTU}(1 - R)\right\}\right]}$;

and, $\text{NTU} = \left[1/\left(R - 1\right)\right]\ln\left[\left(\in - 1\right)/\left(\in R - 1\right)\right]$

Heat Exchanger Effectiveness Relation

Flow arrangement	relationship
Concentric tube	
Parallel flow	$\in = \dfrac{1-\exp\left[-N\left(1+R\right)\right]}{\left(1+R\right)}$; $R = C_{min} / C_{max}$
Counter flow	$\in = \dfrac{1-\exp\left[-N\left(1-R\right)\right]}{1-R\ \exp\left[-N\left(1-R\right)\right]}$; $R < 1$
	$\in = N/\left(1+N\right)$ for $R = 1$
Cross flow (single pass)	
Both fluids unmixed	$\in = 1-\exp\left[\left(1/R\right)\left(N\right)^{0.22}\left\{\exp\left(-R\left(N\right)^{0.78}\right)-1\right\}\right]$
C_{max} mixed , C_{min} unmixed	$\in = \left(1/R\right)\left[1-\exp\left\{-R\left(1-\exp\left(-N\right)\right)\right\}\right]$
C_{min} mixed, C_{max} unmixed	$\in = 1-\exp\left[-R^{-1}\left\{1-\exp\left(-RN\right)\right\}\right]$
All exchangers (R = 0)	$\in = 1-\exp\left(-N\right)$

Kays and London have presented graphs of effectiveness against NTU for Various values of R applicable to different heat exchanger arrangements, Fig.

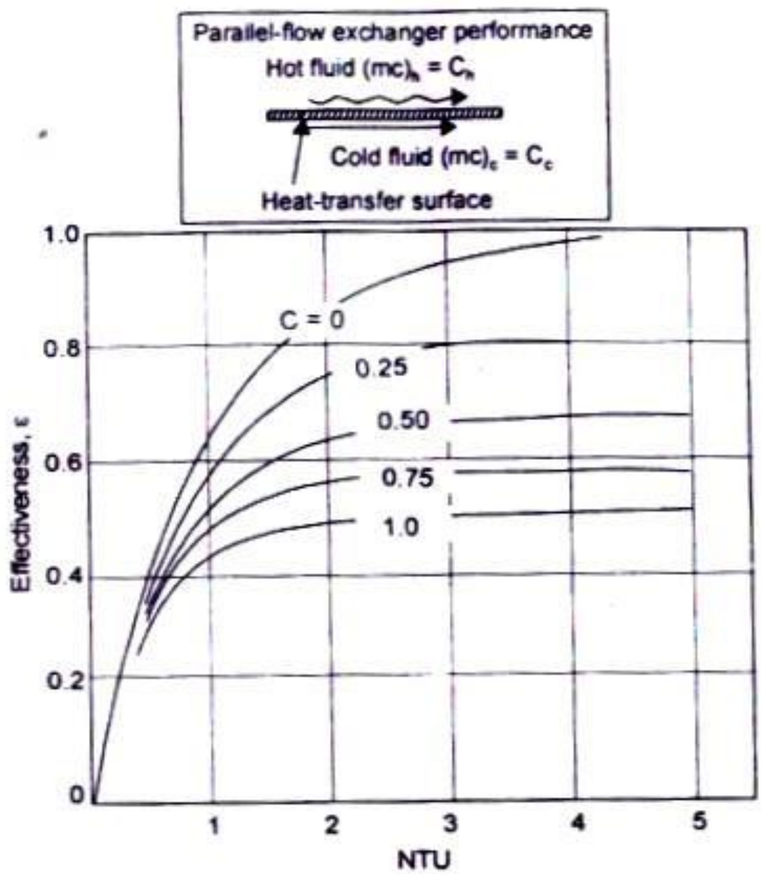


Fig 4.12 Heat exchanger effectiveness for parallel flow

Example A single pass shell and tube counter flow heat exchanger uses exhaust gases on the shell side to heat a liquid flowing through the tubes (inside diameter 10 mm, outside diameter 12.5 mm, length of the tube 4 m). Specific heat capacity of gas 1.05 kJ/kgK, specific heat capacity of liquid 1.5 kJ/kgK, density of liquid 600 kg/m^3, heat transfer coefficient on the shell side and on the tube sides are: 260 and 590 W/m^{2}K respectively. The gases enter the exchanger at 675 K at a mass flow rate of 40 kg/s and the liquid enters at 375 K at a mass flow rate of 3 kg/s. If the velocity of liquid is not to exceed 1 m/s, calculate (i) the required number of tubes, (ii) the effectiveness of the heat exchanger, and (iii) the exit temperature of the liquid. Neglect the thermal resistance of the tube wall.

Solution: Volume flow rate of the liquid = 3/600 = 0.005 m/s. For a velocity of 1 mls through the tube, the cross-sectional area of the tubes will be 0.005 m^2. Therefore, the number of tubes would be

$$n(0.005\times 4)/(3.142\times 0.01)^2 = 63.65 = 64 \text{ tubes}$$

The overall heat transfer coefficient based on the outside surface area of the tubes, after neglecting the thermal resistance of the tube wall, is

$$U = 1/(1/h_0 + r_o/r_i h_i) = 1/[1/260 + 12.5/(10\times 590)] = 167.65\,W/m^2K$$

$$C_{max} = 40\times 1.05 = 42;\ C_{min} = 3\times 1.5 = 4.5;\ R = 4.5/42 = 0.107$$

$$NTU = AU/C_{min} = 3.142\times 0.0125\times 4\times 64\times 167.65/(4.5\times 1000) = 0.374$$

From Fig. 10.12, for R = 0.107, and NTU = 0.3 74, E = 0.35 approximately Therefore, $0.35 = (T_{c_0} - 375)/(675-375)$ or $T_{c_0} = 207^oC$

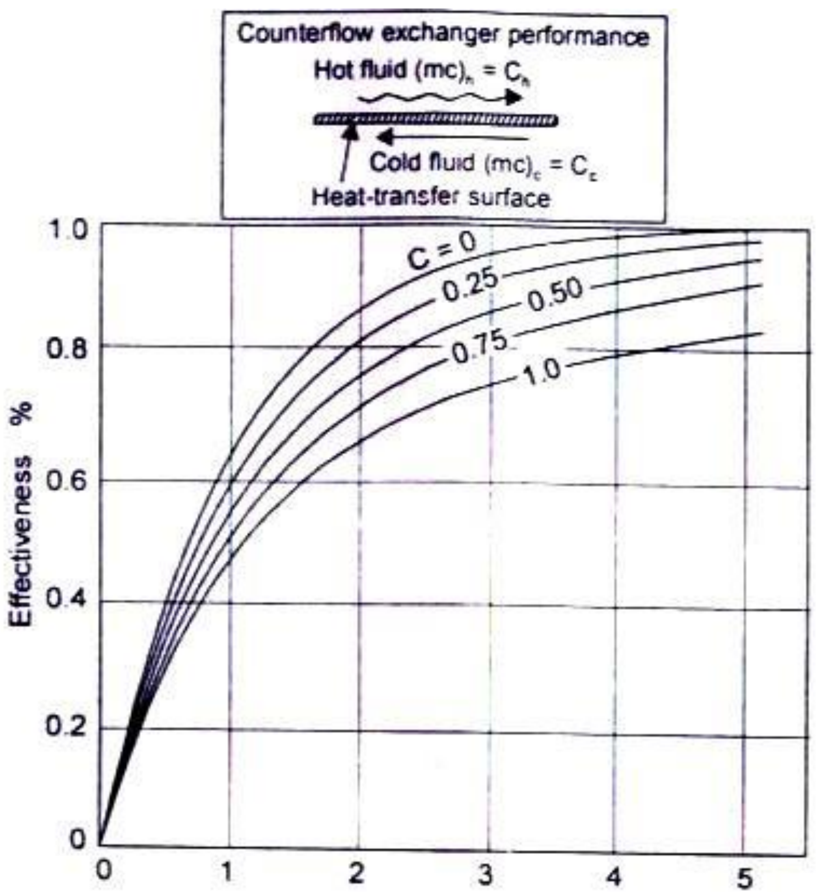


Fig 4.13 Heat exchanger effectiveness for counter flow

Example Air at 25°C, mass flow rate 20 kg/min, flows over a cross-flow heat exchanger and cools water from 85°C to 50°C. The water flow rate is 5 kg/mm. If the overall heat transfer coefficient IS 80 W/m^{2}K and air is the mixed fluid, calculate the exchanger effectiveness and the surface area.

Solution: Let the specific heat capacity of air and water be 1.005 and 4.182 kJ/ kgK. By making an energy balance:

$$\dot{m}_c \times c_c \times \left(T_{c_0} - T_{c_i}\right) = \dot{m}_h \times c_h \times \left(T_{h_i} - T_{h_0}\right)$$

or, 5 × 4182 × (85 – 50) = 20 × 1005 × $\left(T_{c_0} - 25\right)$

i.e., the air will come out at 61.4 ^{o}C.

Heat capacity rates for water and air are:

$$C_w = 4182 \times 5/60 = 348.5; \quad C_a = 1005 \times 20/60 = 335$$

$$R = C_{min}/C_{max} = 335/348.5 = 0.96$$

The effectiveness on the basis of minium heat capacity rate is

$$\in = \left(61.4 - 25\right)/\left(85 - 25\right) = 0.6$$

From Fig. 10.13, for R = 0.96 and $\in$ = 06, NTU = 2.5

Since NTU = AU/C_{min}; A = 2.5 × 335/80 = 10.47 m^2

Since all the four terminal temperatures are easily obtained, we can also use the LMTD approach. Assuming a simple counter flow heat exchanger,

LMTD = (25 – 23.6)/ ln (25/23.6) = 24.3

The correction factor for using a cross-flow heat exchanger with one fluid mixed and the other unmixed, from Fig. 10.10(d), F = 0.55

$\dot{Q}$ = U A F (LMTD)

Therefore, A = 348.5 × 35/(80 × 0.55 × 24.3) = 11.4 m^2

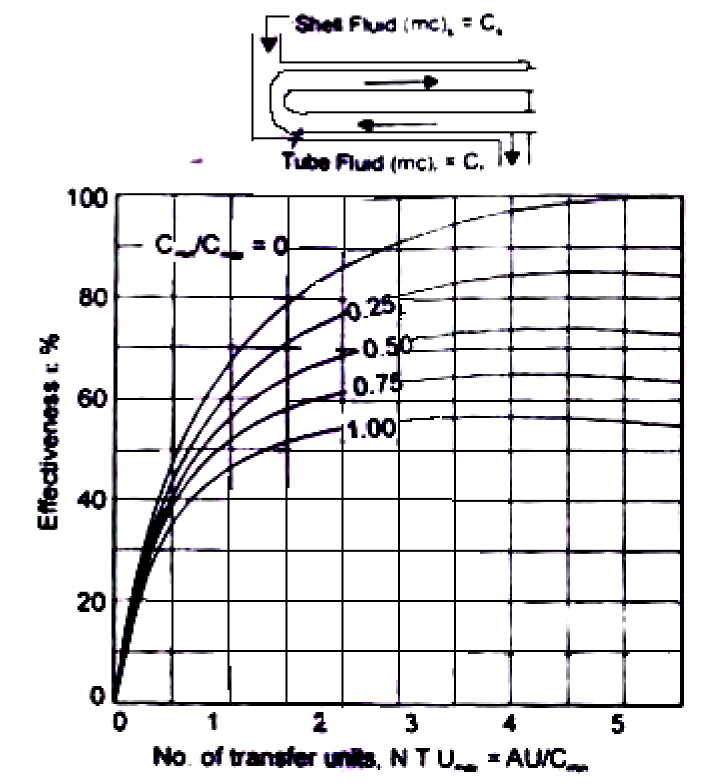
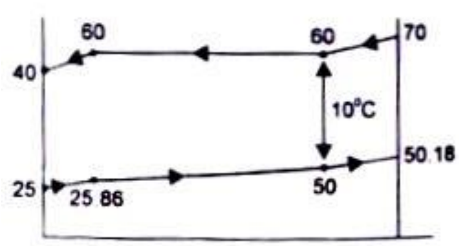


Fig. 4.13 Heat exchanger effectiveness for shell and tube heat exchanger with one shell pass and two, or a multiple of two, tube passes

Example Steam at 20 kPa and 70°C enters a counter flow shell and tube exchanger and comes out as subcooled liquid at 40°C. Cooling water enters the condenser at 25°C and the temperature difference at the pinch point is 10°C. Calculate the (i) amount of water to be circulated per kg of steam condensed, and (ii) required surface area if the overall heat transfer coefficient is 5000 W/m^{2}K and is constant.

Solution: The temperature profile of the condensing steam and water is shown in the 40 accompanying sketch.



The saturation temperature corresponding to 20 kPa is 60°C and as such the temperature of the cooling water at the pinch pint is 50°C. The condensing unit may be considered as a combination of three sections:

(i) desuperheater - the superheated steam is condensed to saturated steam from 70°C to 60°C.

(ii) the condenser - saturated steam is condensed into saturated liquid.

(iii) subcooler - saturated liquid at 60°C is cooled to 40°C.

Assuming that the specific heat capacity of superheated steam is 1.8 kJ/kgK, heat given out in the desuperheater section is 1.8 × (70 - 60) = 18000 Jlkg. Heat given out in the condenser section = 2358600 J/kg (= hfg)

Heat given out in the subcooler = 4182 × (60 - 40) = 83640 J/kg

By making an energy balance, for subcooler and condenser section, we have

$\dot{m}_w \times 4182 \times (50 - 25) = (83640 + 2358600)$;

∴. Mass of water circulated, $\dot{m}_w$ = 23.36 kg/kg steam condensed.

The temperature of water at exit

= 25 + (83640 + 2358600 + 18000)/ (23.36 × 4182) = 50.18 °C

LMTD for desuperheater section

= [(70 - 50.18) - (60 - 50))/ln(20.18/1 0) = 14.5

LMTD for condenser section = [(60 - 50) - (60 - 25.86))/ln(1 0/34.14)

= 19.66

LMTD for subcooler section = [(34.14 - 15)/1n (34.14/15) = 23.27

Since U is constant through out,

Surface area for subcooler section = 83640/(5000 × 23.27) = 0.7188 m^2

Surface area for condenser section = 2358600/(5000 × 19.66) = 23.9939 m^2

Surface area for desuperheater section = 18000/(5000 × 14.5) = 0.2483 m^2

$\therefore$. Total surface area = 24.96 m^2 and average temperature difference = 19.71°C.

Example In an economiser (a cross flow heat exchanger, both fluids unmixed) water, mass flow rate 10 kg/s, enters at 175°C. The flue gas mass flow rate 8 kg/s, specific heat 1.1 kJ/kgK, enters at 350°C. Estimate the temperature of the flue gas and water at exit, if U = 500 W/m^2K, and the surface area 20 m^2 What would be the exit temperature if the mass flow rate of flue gas is (i) doubled, and (ii) halved.

Solution: The heat capacity rate of water = 4182 × 10 = 41820 W /K

The heat capacity rate of flue gas = 1100 × 8 = 8800 W/K

C_{min}/C_{max} = 8800/41820 = 0.21

$$NTU = AU/C_{min} = 500 \times 20/8800 = 1.136$$

From Fig. 10.14. for NTU = 1.136 and $C_{min}/C_{max} = 0.21,\ \in = 0.62$

Therefore, 0.62 = (350 - T)/(350 - 175) and T = 241.5°C

The temperature of water at exit, T_w = 175 + 8800 × (350 - 241.5)/41820

= 197.83^{o}C

When the mass flow rate of the flue gas is doubled. C_{gas} = 17600 W/K

$C_{min}/C_{max} = 0.42,\ NTU = AU/C_{min} = 0.568$

$\in$ = 0.39 = (350 - T)/(350 - 175);

T = 281. 75°C, an increase of 40°C

and T_w = 175 + 28.72 = 203.72°C, an increase of about 6°C.

When the mass flow rate of the flue gas is halved, C_{min} = 4400 W/K

$C_{min}/C_{max} = 0.105$, NTU = 2.272, and from the figure, $\in$ = 0.83, an increase and T_g = 204.75 and T_w = 190.3^{o}C

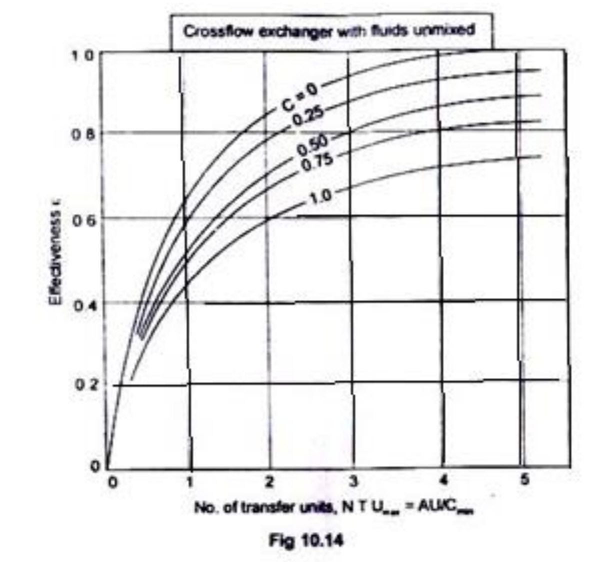


Fig 4.14

Example In a tubular condenser, steam at 30 kPa and 0.95 dry condenses on the external surfaces of tubes. Cooling water flowing through the tubes has mass flow rate 5 kg/s, inlet temperature 25°C, exit temperature 40°C. Assuming no subcooling of the condensate, estimate the rate of condensation of steam, the effectiveness of the condenser and the NTU.

Solution: Since there is no subcooling of the condensate, the steam will lose its latent heat of condensation = 0.95 × h_{fg} = 0.95 × 2336100 = 2.22 × 10^6 J/kg. At pressure, 30kPa, saturation temperature is 69.124^{o}C

Steam condensation rate × 2.22 × 10^6 = Heat gained by water

= 5 × 4182 × (40 – 25) = 313650 J

Therefore, m, = 313650/2.22 × 106 = 847.7 kg/hour.

When the temperature of the evaporating or condensing fluid remains constant, the value of LMTD is the same whether the system is having a parallel flow or counter flow arrangement, therefore,

LMTD = [(69.124 - 25) - (69.124-40)]/ln(44.124/29.124) = 36.1

Q = UA(LMTD)

Therefore, UA = 5 × 4182 × (40 - 25)/36.1 = 8688.36 W/K

NTU = UA/C_{min} = 8688.36/(5 × 4182) = 0.4155

Effectiveness= Actual temp. difference; Maximum possible temp. difference

= (40 - 25)/(69.124 - 25) = 34%.

Example A single shell 2 tube pass steam condenser IS used to cool steam entering at 50°C and releasing 2000 MW of heat energy. The cooling water, mass flow rate 3 × 104 kg/s, enters the condenser at 25°C. The condenser has 30,000 thin walled tube of 30 mm diameter. If the overall heat transfer coefficient is 4000 Wim^{2}K, estimate the (I) rise m temperature of the cooling water, and (II) length of the tube per pass.

Solution: By making an energy balance:

Heat released by steam = heat taken in by cooling water,

or, 2000 × 10^6 = 3 × 10^4 × 4182 × (□T); □□□□T = 15.94°C.

Since in a condenser, heat capacity rate of condensing steam is usually very large m comparison with the heat capacity rate of cooling water, the effectiveness

$$\in = \left(T_{c_0} - T_{c_i}\right) / \left(T_{h_i} - T_{c_i}\right) = 15.94/(50-25) = 0.6376$$

$$\text{And, for } C_{min}/C_{max} = 0, \ \in = 1 - \exp(-NTU)$$

$$\therefore \ \exp(-NTU) = 1.0 - 0.6376 = 0.3624$$

$$\text{And, } NTU = 1.015 = AU/C_{min} = (2 \times 3.142 \times 0.03 \times L \times 30000) \times 4000/(1.25 \times 108)$$

L = 5.546 m

4.19 Heat Exchanger Design-Important Factors

A comprehensive design of a heat exchanger involves the consideration of the thermal, mechanical and manufacturing aspect. The choice of a particular design for a given duty depends on either the selection of an existing design or the development of a new design. Before selecting an existing design, the analysis of his performance must be made to see whether the required performance would be obtained within acceptable limits.

In the development of a new design, the following factors are important:

(a) Fluid Temperature - the temperature of the two fluid streams are either specified for a given inlet temperature, or the designer has to fix the outlet temperature based on flow rates and heat transfer considerations. Once the terminal temperatures are defined, the effectiveness of the heat exchanger would give an indication of the type of flow path-parallel or counter or cross-flow.

(b) Flow Rates - The maximum velocity (without causing excessive pressure drops, erosion, noise and vibration, etc.) in the case of liquids is restricted to 8 m1s and m case of gases below 30 m/s. With this restriction, the flow rates of the two fluid streams lead to the selection of flow passage cross-sectional area required for each of the two fluid steams.

(c) Tube Sizes and Layout - Tube sizes, thickness, lengths and pitches have strong influence on heat transfer calculations and therefore, these are chosen with great care. The sizes of tubes vary from 1/4" O.D. to 2" O.D.; the more commonly used sizes are: 5/8", 3/4" and 1" O.D. The sizes have to be decided after making a compromise between higher heat transfer from smaller tube sizes and the easy clean ability of larger tubes. The tube thickness will depend pressure, corrosion and cost. Tube pitches are to be decided on the basis of heat transfer calculations and difficulty in cleaning. Fig. 3.16 shows several arrangements for tubes in bundles. The two standard types of pitches are the square and the triangle. the usual number 0 f tube passes in a given shell ranges from one to eight. In multipass designs, even numbers of passes are generally used because they are Simpler to design.

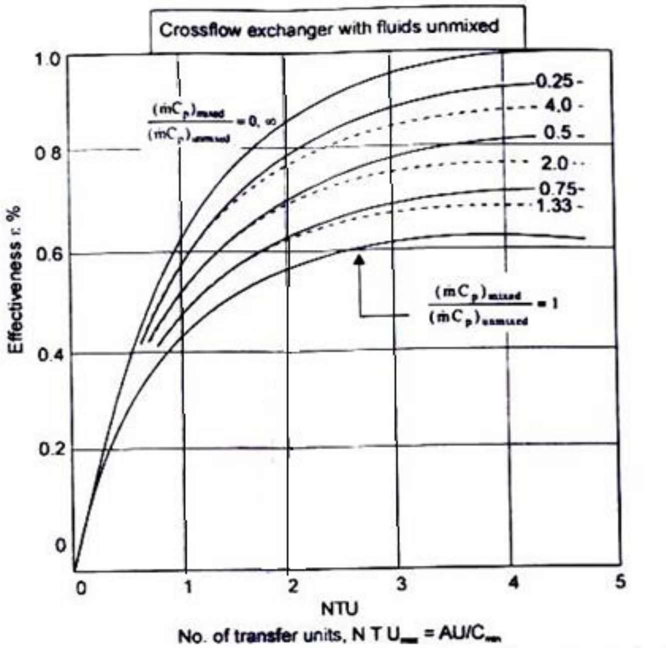


Fig 4.15: Heat exchanger effectiveness for crossflowwith one fluid mixed and the other unmixed

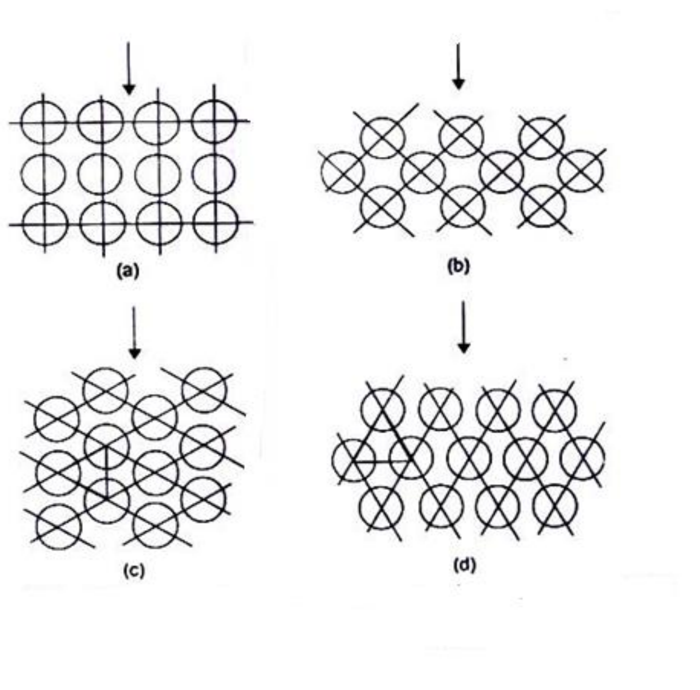


Fig 4.16 Several arrangements of tubes in bundles : (a) I line arrangement with square pitch, (b) staggered arrangement with triangular pitches (c) and(d) stagged arrangement with triangular pitches

Fig 4.17 shows three types of transverse baffles used to increase velocity on the shell side. The choice of baffle spacing and baffle cut is a variable and the optimum ratio of baffle cuts and spacing cannot be specified because of many uncertainties and insufficient data.

(d)Dirt F actor and Fouling - the accumulation of dirt or deposits affects significantly the r ate of heat transfer and the pressure drop. P roper allowance for the fouling factor and dirt factor should receive the greatest attention design because they cannot be avoided. A heat exchanger requires frequent cleaning. Mechanical cleaning will require removal of the tube bundle for cleaning. Chemical cleaning will require the use of non-corrosive materials for the tubes.

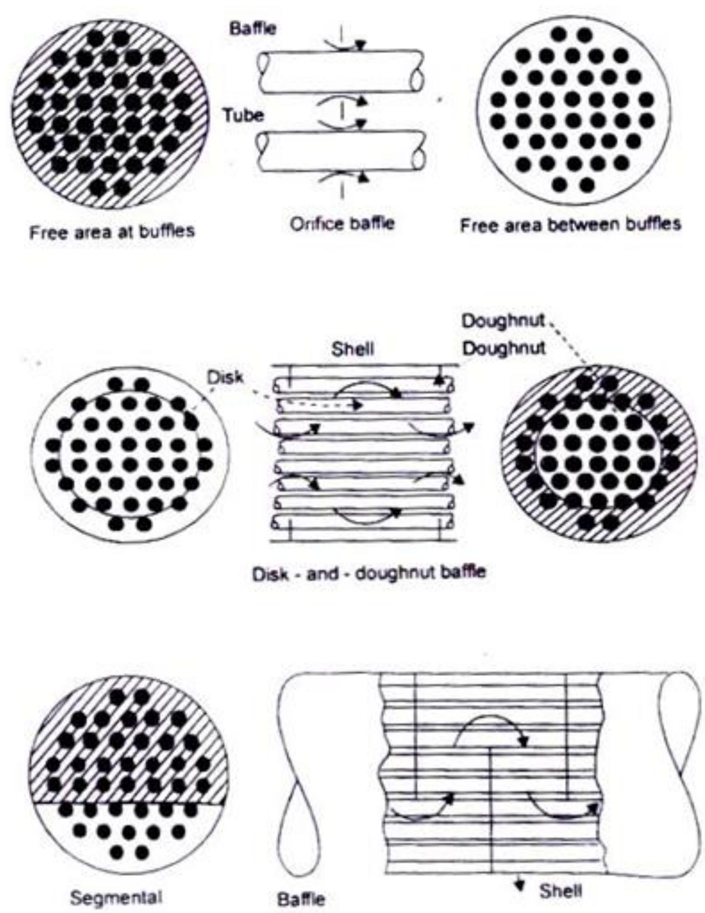


Fig. 4.17 Three types of transverse baffles

(e) Size and Installation - In designing a heat exchanger, It is necessary that the constraints on length, height, width, volume and weight is known at the outset. Safety regulations should also be kept in mind when handling fluids under pressure or toxic and explosive fluids.

(f) Mechanical Design Consideration - While designing, operating temperatures, pressures, the differential thermal expansion and the accompanying thermal stresses require attention.

And, above all, the cost of materials, manufacture and maintenance cannot be Ignored.

Example In a counter flow concentric tube heat exchanger cooling water, mass flow rate 0.2 kg/s, enters at 30°C through a tube inner diameter 25rnnl. The oil flowing through the annulus, mass flow rate 0.1 kg/s, diameter 45 rum, has temperature at inlet 100°C. Calculate the length of the tube if the oil comes out at 60°C. The properties of oil and water are:

Oil: C_p – 2131 J/kgK, □ = 3.25 × 10^{-2} Pa-s, k = 0.138 W/mK,

Water; C_p = 4178 J/kg K, □ = 725 × 10^{-6} Pa-s,

k = 0.625 W/mK, Pr = 4.85

Solution: By making an energy balance: Heat given out by oil = heat taken in by water.

$$0.1\times 2131\times(100-60)=0.2\times 4187\times\left(T_{c_0}-30\right)$$

$$T_{c_0}=40.2^{o}C$$

$$\text{LMTD} = \left[\left(T_{h_i}-T_{c_0}\right)-\left(T_{h_0}-T_{c_i}\right)\right]/\ln\left[\left(T_{h_i}-T_{c_0}\right)/\left(T_{h_0}-T_{c_0}\right)\right]$$

$$=\left[(100-40.2)-(60-30)\right]/\ln(59.8/30)=43.2^{o}C$$

Since water is flowing through the tube,

$$Re=4\dot{m}/\pi D\mu=\frac{4\times 0.2}{3.142\times 0.025\times 725\times 10^{-6}} = 14050, \text{ a turbulent flow.}$$

□□□□Nu = 0.023 $Re^{0.8}$ $Pr^{0.4}$, fluid being heated.

= 0.023 $(14050)^{0.8}$ $(4.85)^{0.4}$ = 90; $\therefore$ h_i = 90 × 0.625/0.025 = 2250 W/m^2K

The oil is flowing through the annulus for which the hydraulic diameter is:

(0.045 - 0.025) = 0.02 m

$$Re = 4\dot{m}/\pi\left(D_o + D_i\right)\mu = 4\times0.1/\left(3.142\times0.07\times3.25\times10^{-2}\right) = 56.0$$

laminar flow.

Assuming Uniform temperature along the Inner surface of the annulus and a perfectly insulated outer surface.

Nu = 5.6, by interpolation (chapter 6)

$h_o = 5.6 \times 0.138/0.02 = 38.6$ W/m^{2}K.

The overall heat transfer coefficient after neglecting the tube wall resistance,

U = 1/(1/2250 + 1/38.6) = 38 W/m^{2}K

$\dot{Q} = UA\left(LMTD\right)$, where where A = $\pi D_i \times L$

L = (0.1 × 2131 × 40)/(38 × 3.142 × 0.025 × 43.2) = 66.1 m requires more than one pass.

Example A double pipe heat exchanger has an effectiveness of 0.5 for the counter flow arrangement and the thermal capacity of one fluid is twice that of the other fluid. Calculate the effectiveness of the heat exchanger if the direction of flow of one of the fluids is reversed with the same mass flow rates as before.

Solution: For a counter flow arrangement and R = 0.5, $\in$ = 0.5

$$NTU = \left[1/\left(R-1\right)\right]\ln\left(\in R - 1\right) = -2.0\ \ln\left(0.5/0.75\right) = 0.811$$

For parallel flow, $\in\ = \left[1-\exp\left\{-NTU(1+R)\right\}\right]/\left(1+R\right)$

$$= \left[1-\exp\left(-0.811\times1.5\right)\right]/1.5 = 0.469$$

Example Oil is cooled in a cooler from 65°C to 54°C by circulating water through the cooler. The cooling load is 200 kW and water enters the cooler at 27°C. If the overall heat transfer coefficient, based on the outer surface area of the tube is 740 W/m^{2}K and the temperature rise of cooling water is 11°C, calculate the mass flow rate of water, the effectiveness and the heat transfer area required for a single pass In a parallel flow and in a counter flow arrangement.

Solution: Cooling load = 200 kW = mass of water × sp. heat × temp. rise Mass of water = 200/(4.2 × 11) = 4.329 kg/s

(i) Parallel flow:

From the temperature profile:

LMTD = (38 – 16)/ln(38/16) = 25.434 Q = U A (LMTD);

Area A = $200 \times 10^3/(740 \times 25.434) = 10.626\ m^2$

Effectiveness, $\in$ = (38 - 27)/(54 - 27) = 0.407.

(ii) Counter flow:

From the temperature profile:

LMTD = mean temperature difference = 27°C

Area A = $200 \times 10^3/(740 \times 27) = 10\ m^2$

Effectiveness, E = (38 - 27)/(65 - 27) = 0.289.

Example Oil (mass flow rate 1.5 kg/s C_p = 2 kJ/kgK) is cooled in a single pass shell and tube heat exchanger from 65 to 42°C. Water (mass flow rate 1 kg/s, C_p = 4.2 kJ/kgK) has an inlet temperature of 28°C. If the overall heat transfer coefficient is 700 W/m^2K, calculate heat transfer area for a counter flow arrangement using $\in$ - NTU method.

Solution: Heat capacity rate of 011; 1.5 × 2.0 = 3 kW/K

Heat capacity rate of water = 1 × 4.2; 4.2 kW/K

$C_{min} = 3.0 \text{ kW/K}$ and $R = C_{min}/C_{max} = 3/4.2 = 0.714$

For a counter flow arrangement, $NTU = \left[1/\left(R-1\right)\right]\ln\left[\left(\in -1\right)/\left(\in R-1\right)\right]$

Effectiveness, $\in = \left(65-42\right)/\left(65-28\right) = 0.6216$

and NTU = 1.346 = AU/C_{min}; A = 1.346 × 3000 / 700 = 5.77 m^2

By making an energy balance, we can compute the water temperature at outlet.

or 3.0 × (65 - 42); 4.2 × (T - 28), T; 44.428

LMTD for a counter flow arrangement:

LMTD; (20.572 - 14)/ln (20.572/14) = 17.076

Area, A = $\dot{Q}/U \times \left(LMTD\right) = 3\times10^3 \times \left(65-42\right)/\left(700\times17.076\right) = 5.77 \text{ m}^2$

Example A fluid (mass flow rate 1000 kg/min, sp. heat capacity 3.6 kJ/kgK) enters a heat exchanger at 700 C. Another fluid (mass flow rate 1200 kg/mm, sp. heal capacity 4.2 kJ/kgK) enters al 100 C. If the overall heat transfer coefficient is 420 W/m^2K and the surface area is 100m2, calculate the outlet temperatures of both fluids for both counter flow and parallel flow arrangements.

Solution: Heat capacity rate for the hot fluid

$$1000\times3.6\times10^3\times60 = 60\times10^3 W/K$$

Heat capacity rate for the cold fluid = 1200 × 4.2 × 10^3/60 = 84 × 10^3 W/K

R = C_{min}/C_{max} = 60/84; 0.714, NTU = U A/C_{min} = 420 × 100/60000 = 0.7

(i) For counter flow heat exchanger:

$$\in = \left[1-\exp\left\{-N\left(1-R\right)\right\}\right]/\left[1-\mathrm{Re\,xp}\left\{-N\left(1-R\right)\right\}\right]$$

$$\left[1-\exp\left\{-0.7\left(1-0.714\right)\right\}\right]/1\left[1-0.714\ \exp\left\{-0.7\left(1-0.714\right)\right\}\right] = 0.4367$$

Since heat capacity rate of the hot fluid IS lower,

$$\in = \left(700 - T_{h_0}\right) / \left(700 - 100\right)$$

and $T_{h_0} = 700 - 0.4367 \times 600 = 438^{o}C$

By making an energy balance, $60 \times 10^3 \left(700 - 438\right) = 84 \times 10^3 \left(T_{c_0} - 100\right)$

or, $T_{c_0} = 60 \times 262 / 84 + 100 = 87.14^{o}C$

(ii) For parallel flow heat exchanger

$\in = \left[1 - \exp\left\{-N\left(1+R\right)\right\}\right] / \left(1+R\right) = \left[1 - \exp\left\{0.7\left(1+0.714\right)\right\}\right] / \left(1.714\right)$

$\in = 0.4077$, a lower value

and $\left(T_{h_i} - T_{h_0}\right) / \left(T_{h_i} - T_{c_0}\right) = 0.4077 = \left(700 - T_{h_0}\right) / \left(700 - T_{c_0}\right)$

By making an energy balance: $60 \times 10^3 \times \left(700 - T_{h_0}\right) = 84 \times 10^3 \times \left(T_{c_0} - 100\right)$

or, $\left(700 - T_{c_0}\right) = \left(700 - T_{h_0}\right) / 0.4077$

and $84 \times \left(T_{c_0} - 100\right) / 60 = \left(1.4T_{c_0} - 140\right)$

Therefore, $T_{c_0} = 237.5^{o}C$

and $T_{h_0} = 511.4^{o}C$

Example Steam enters the surface condenser at 100°C and water enters at 25°C with a temperature rise of 25°C. Calculate the effectiveness and the NTU for the condenser. If the water temperature at inlet changes to 35 C, estimate the temperature rise for water.

Solution: Effectiveness, $\in$ = 25/(100 25) = 0.33

For R = 0, $\in$ = 1 - exp(- N)

or, N = -ln(1 - $\in$) = 0.405

Since other parameters remain the same,

$$25/(100 - 25) = \Delta T/(100 - 35)$$

and $\Delta T = 21.66$; or, $T_{c_0} = 35 + 21.66 = 56.66°C$.

4.20 Increasing the Heat Transfer Coefficient

For a heat exchanger, the heat load is equal to Q = UA (LMTD). The effectiveness of the heat exchanger can be increased either by increasing the surface area for heat transfer or by increasing the heat transfer coefficient. Effectiveness versus NTU(AU/C_{min}) curves, Fig. 10.10 - 15, reveal that by increasing the surface area beyond a certain limit (the knee of the curves), there is no appreciable improvement in the performance of the exchangers. Therefore, different methods have been employed to increase the heat transfer coefficient by increasing turbulence, improved mixing, flow swirl or by the use of extended surfaces. The heat transfer enhancement techniques is gaining industrial importance because it is possible to reduce the heat transfer surface area required for a given application and that leads to a reduction in the size of the exchanger and its cost, to increase the heating load on the exchanger and to reduce temperature differences.

The 'different techniques used for increasing the overall conductance U are: (a) Extended Surfaces - these are probably the most common heat transfer enhancement methods. The analysis of extended surfaces has been discussed in Chapter 2. Compact heat exchangers use extended surfaces to give the required heat transfer surface area in a small volume. Extended surfaces are very effective when applied in gas side heat transfer. Extended surfaces find their application in single phase natural and forced convection pool boiling and condensation.

(b) Rough Surfaces - the inner surfaces of a smooth tube is artificially roughened to promote early transition to turbulent flow or to promote mixing between bulk flow and the various sub-layer in fully developed turbulent flow. This method is primarily used in single phase forced convection and condensation.

(c) Swirl Flow Devices - twisted strips are inserted into the flow channel to impart a rotational motion about an axis parallel to the direction of bulk flow. The heat transfer coefficient increases due to increased flow velocity, secondary flows generated by swirl, or increased flow path length in the flow channel. This technique is used in flow boiling and single phase forced flow.

(d) Treated Surfaces - these are used mainly in pool boiling and condensation.

Treated surfaces promote nucleate boiling by providing bubble nucleation sites. The rate of condensation increases by promoting the formation of droplets, instead of a liquid film on the condensing surface. This can be accomplished by coating the surface with a material that makes the surface non-wetting.

All of these techniques lead to an increase in pumping work (increased frictional losses) and any practical application requires the economic benefit of increased overall conductance. That is, a complete analysis should be made to determine the increased first cost because of these techniques, increased heat exchanger heat transfer performance, the effect on operating costs (especially a substantial increase in pumping power) and maintenance costs.

4.21 Fin Efficiency and Fin Effectiveness

Fins or extended surfaces increase the heat transfer area and consequently, the amount of heat transfer is increased. The temperature at the root or base of the fin is the highest and the temperature along the length of the fm goes on decreasing Thus, the fin would dissipate the maximum amount of heat energy if the temperature all along the length remains equal to the temperature at the root. Thus, the fin efficiency is defined as:

η_{fin} = (actual heat transferred) / (heat which would be transferred if the entire fin area were at the root temperature)

In some cases, the performance of the extended surfaces is evaluated by comparing the heat transferred with the fin to the heat transferred without the fin. This ratio is called 'fin effectiveness' E and it should be greater than 1, if the rate of heat transfer has to be increased with the use of fins.

For a very long fin, effectiveness E = $\dot{Q}_{\text{with fin}} / \dot{Q}_{\text{without fin}}$

$= (\text{hpkA})^{1/2}\theta_0$ $/\text{hA}\,\theta_0$ $= (\text{kp/hA})^{1/2}$

And $\eta_{\text{fin}} = (\text{hpkA})^{1/2}\theta_0$ $(\text{hpL}\,\theta_0) = (\text{hpkA})^{1/2} / (\text{hpL})$

$$\frac{E}{\eta_{\text{fin}}} = \frac{(\text{kp}/\text{hA})^{1/2}}{(\text{hpkA})^{1/2}} \times \text{hpL} = \frac{\text{pL}}{\text{A}} = \frac{\text{Surface area of fin}}{\text{Cross-sectional area of the fin}}$$

i.e., effectiveness increases by increasing the length of the fin but it will decrease the fin efficiency.

Expressions for Fin Efficiency for Fins of Uniform Cross-section

1. Very long fins: $(\text{hpkA})^{1/2}(T_0 - T_\infty)/\left[\text{hpL}(T_0 - T_\infty)\right] = 1/\text{mL}$

2 For fins having insulated tips:

$$\frac{(\text{hpkA})^{1/2}(T_0 - T_\infty)\tan h(\text{mL})}{\text{hpL}(T_0 - T_\infty)} = \frac{\tan h\,(\text{mL})}{\text{mL}}$$

Example The total efficiency for a finned surface may be defined as the ratio of the total heat transfer of the combined area of the surface and fins to the heat which would be transferred if this total area were maintained at the root temperature T_0 . Show that this efficiency can be calculated from

$\eta_t = 1 - A_f / A(1 - \eta_t)$ where η_t = total effltiency, A_f = surface area of all fins, A = total heat transfer area, η_f = fin efficiency

Solution: Fin efficiency,

$$\eta_f = \frac{\text{Actual heat transfered}}{\text{Heat that would be transferred if the en tire fin were at the root temperature}}$$

or, $\eta_f = \dfrac{\text{Actual heat transfer}}{hA_f(T_0 - T_\infty)}$

$\therefore$ Actual heat transfer from finned surface = $\eta_f hA_f(T_0 - T_\infty)$

Actual heat transfer from un finned surface which are at the root temperature: $h(A\text{-}A_f)$

$(T_0 - T_\infty)$

Actual total heat transfer = $h(A - A_f)(T_0 - T_\infty) + \eta_f h A_f (T_0 - T_\infty)$

By the definition of total efficiency,

$$\eta_t = \left[h\left(A - A_f\right)\left(T_0 - T_\infty\right) + \eta_f h A_f \left(T_0 - T_\infty\right)\right] / \left[hA\left(T_0 - T_\infty\right)\right]$$

$$= \frac{\left(A - A_f\right) + \eta_f h A_f}{A} = 1 - A_f / A + \eta_f A_f / A$$

$$= 1 - \left(A_f / A\right) + \left(1 - \eta_f\right).$$

4.23. Extended Surfaces do not always Increase the Heat Transfer Rate

The installation of fins on a heat transferring surface increases the heat transfer area but it is not necessary that the rate of heat transfer would increase. For long fins, the rate of heat loss from the fin is given by $(hpkA)^{1/2}\theta_0 = kA(hp/kA)^{1/2}\theta_0 = kAm\,\theta_0$. When h/ mk = 1, Q = $hA\,\theta_0$ which is equal to the heat loss from the primary surface with no extended surface. Thus, when h = mk, an extended surface will not increase the heat transfer rate from the primary surface whatever be the length of the extended surface.

For $h/mk > 1$, $Q < hA\,\theta_0$ and hence adding a secondary surface reduces the heat transfer, and the added surface will act as an insulation. For $h/mk < 1$, $Q > hA\,\theta_0$, and the extended surface will increase the heat transfer, Fig. 2.3l. Further, $h/mk = (h^2.kA/k^2hp)^{1/2} = (hA/kP)^{1/2}$, i.e. when $h/mk < 1$, the heat transfer would be more effective when h/k is low for a given geometry.

4.24 An Expression for Temperature Distribution for an Annular Fin of Uniform Thickness

In order to increase the rate of heat transfer from cylinders of air-cooled engines and in certain type of heat exchangers, annular fins of uniform cross-section are employed. Fig. 2.32 shows such a fin with its nomenclature.

In the analysis of such fins, it is assumed that:

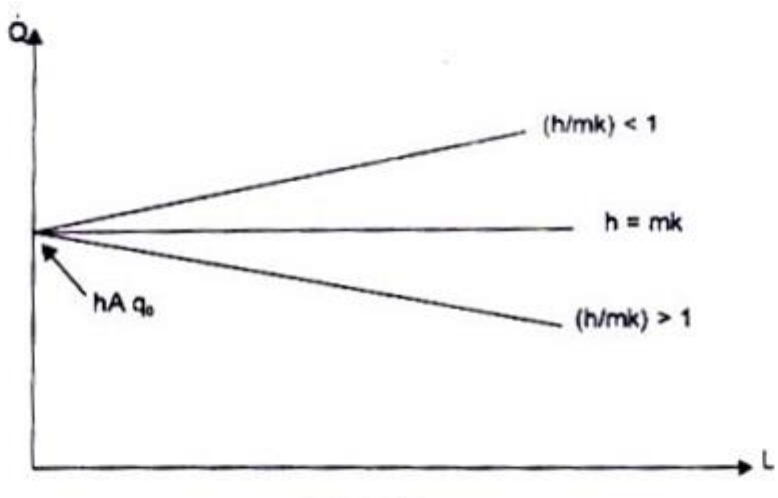
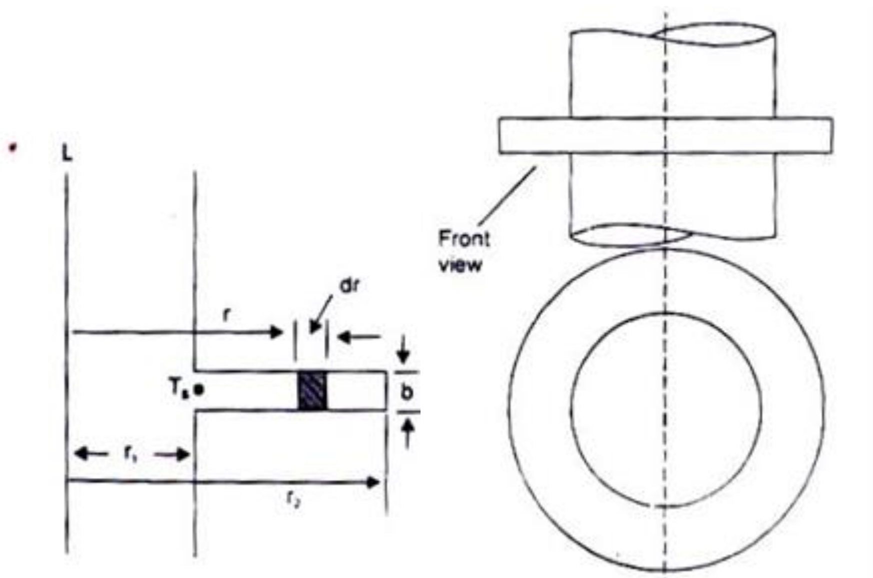


Fig 4.18

(For increasing the heat transfer rate by fins, we should have (i) higher value of thermal conductivity, (ii) a lower value of h, fins are therefore generally placed on the gas side, (iii) perimeter/cross-sectional area should be high and this requires thin fins.)

(i) the thickness b is much smaller than the radial length (r_2 - r_1) so that one-dimensional radial conduction of heat is valid;

(ii) steady state condition prevails.



Annular fin of uniform thickness Top view of annular fin

Fig 4.19

We choose an annular element of radius r and radial thickness dr. The cross-sectional area for radial heat conduction at radius r is $2\pi rb$ and at radius r + dr is 2π (r + dr)b. The surface area for convective heat transfer for the annulus is $2(2\pi$ r.dr). Thus, by making an energy balance,

$$-k2\pi rb\frac{dT}{dr} = -k2\pi(r+dr)b\left(\frac{dT}{dr}+\frac{d^2T}{dr^2}dr\right)+h\times 4\pi r.dr(T-T_\infty)$$

or, $d^2T / dr^2 + (1/r)dT / dr - 2h/kb(T - T_\infty) = 0$

Let, $\theta = (T - T_\infty)$ the above equation reduces to

$d^2\theta / dr^2 + (l/r)\, d\theta/dr - (2h\ kb)\ \theta = 0$

The equation is recognised as Bessel's equation of zero order and the solution is $\theta = C_1 I_0(nr) + C_2 K_0(nr)$, where $n=(2h/kb)^{1/2}$, I_0 is the modified Bessel function, 1st kind and K_0 is the modified Bessel function, 2nd kind, zero order, The constants C_1 and C_2 are evaluated by applying the two boundary conditions:

at $r = r_l$, $T = T_s$ and $\theta = T_s - T_\infty$

at $r = r_2$, $dT / dr = 0$ because $b << (r_2 - r_l)$

By applying the boundary conditions, the temperature distribution is given by

$$\frac{\theta}{\theta_0} = \frac{I_0(nr)K_1(nr_2)+K_0(nr)I_1(nr_2)}{I_0(nr_1)K_1(nr_2)+K_0(nr_1)I_1(nr_2)} \tag{3.16}$$

I_1 (nr) and K_1 (nr) are Bessel functions or order one.

And the rate of heat transfer is given by:

$$Q = 2\pi knb\,\theta_0 r_1\frac{K_1(nr_1)I_1(nr_2)-I_1(nr_1)K_1(nr_2)}{K_0(nr_1)I_1(nr_2)+I_0(nr_1)K_1(nr_2)} \tag{3.17}$$

Table 2.1 gives selected values of the Modified Bessel Functions of the First and Second kinds, order Zero and One. (The details of solution can be obtained from: C.R. Wylie, Jr: Advanced Engineering Mathematics, McGraw-Hill Book Company, New York.)

The efficiency of circumferential fins is also obtained from curves for efficiencies

$\left(\text{along Y-axis}\right) \square \left(r_2 + \frac{b}{2} - r_1\right)^{\frac{3}{2}} \left(\frac{2h}{Kb}\left(r_2 - r_1\right)\right)^{\frac{1}{2}}$ for different values of $\left(r_2 + \frac{b}{2}\right) / r_1$.

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UNIT – V – Heat Transfer Applied to IC Engines – SAUA1503

UNIT V

MEASUREMENTS OF INSTANTANEOUS HEAT-TRANSFER RATES

5.1 Fast-response thermocouple

The instantaneous combustion chamber surface temperatures were measured by J-type coaxial fast response thermocouples. The thermocouple consists of a thin wire of constantan coated with a ceramic insulation of high dielectric strength, swaged securely in a tube made of iron. A vacuum-deposited metallic plate forms a metallurgical bond with the two thermocouple elements, thus forming the thermocouple junction with 1–2 mm thickness over the sensing end of the probe. The response time is of the order of a microsecond. Figure 5.1 is an illustration of the construction of a coaxial thermocouple

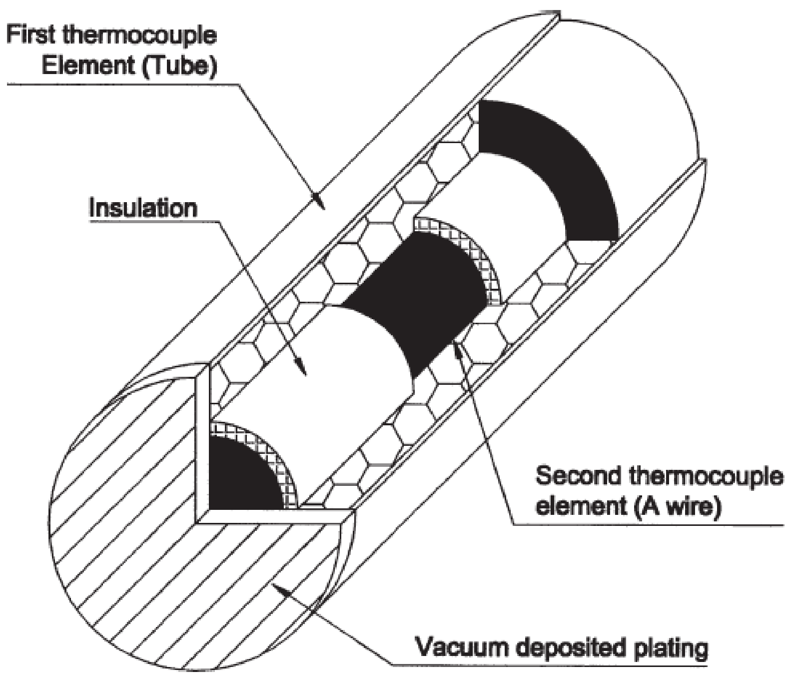


Fig 5.1 Construction of the coaxial thermocouple

5.2 Instrumented piston and cylinder head

Two fast-response thermocouple probes were custom manufactured to fit into the instrumentation sleeves machined in the cylinder head. The probes are located close to the periphery of the combustion chamber and their respective positions are denoted as H1 and H2

(Fig.5.2). The sensing area is flush with the surface. A DISI aluminium piston was instrumented with fast-response thermocouples press-fitted into the piston crown. The sensing area is flush with the piston surface. Figure 5.2 illustrates the locations of the thermocouple junctions. The location of probe P1 is on the intake valve side, near the fuel injector. Probe P2 is located under the spark plug. Probes P3, P4, and P5 are located in the bowl, along the fuel spray trajectory. All probes include a second reference thermocouple (backside junction) located 4mm from the tip (d54mm) to facilitate determination of the steady state component of heat flux.

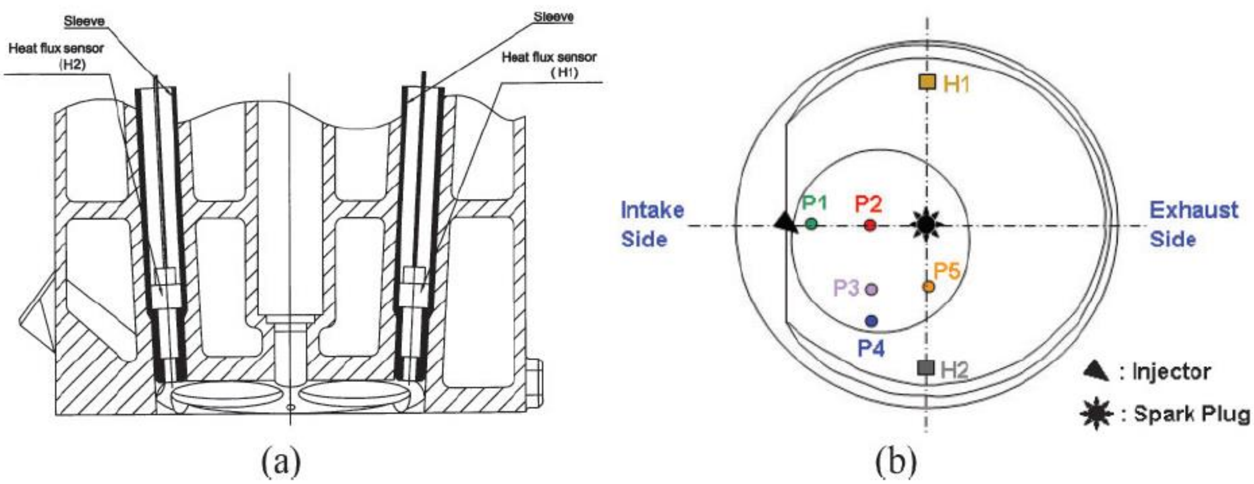


Fig 5.2 Thermocouple locations on (a) the cylinder head and (b) the piston top

5.3 Telemetry linkage system

Measurement of the piston surface temperature in a firing engine relies on the ability to transmit thermocouple signals from the moving piston to the data acquisition system outside the engine. The simultaneous crank-angle-resolved measurements from multiple locations on the piston, with front and backside junctions at each location, require extremely high transmission rates. In addition, thermocouple signals are very weak, and thus susceptible to electric noise. The mechanical telemetry linkage system is much better suited to dealing with these challenges and was therefore adopted.

A two-bar linkage system with fork joints was designed and made from titanium because of its good strength-to-weight ratio. Major components are illustrated in Fig.5.3. The linkage system consists of an adaptor, an aluminium bracket, two links, and an anchor plate. The electric connectors for the signal wires are located at the aluminium bracket inserted into the adaptor. Cage-guided needle roller bearings allow low-friction swivel motion at joints. The anchor plate attached to the side cover of the crankcase carries two electric connectors for transferring the signals to the data acquisition system. Electric continuity of the signal wires is achieved by passing the signal wires directly through the dowel pins. This converts bending motion to torsion, therefore increasing the life of wires. The hook shaped brass tubes are added at the pin joints of the links A and B to guide the wires in a way that releases the stress concentration of the signal wires. The brass tubes are attached to the links by steel wires that run through pre machined holes at the links. Tube encloses the signal wires at the pin joint between the link B and the anchor plate owing to its superior properties at high temperature and in an oily environment.

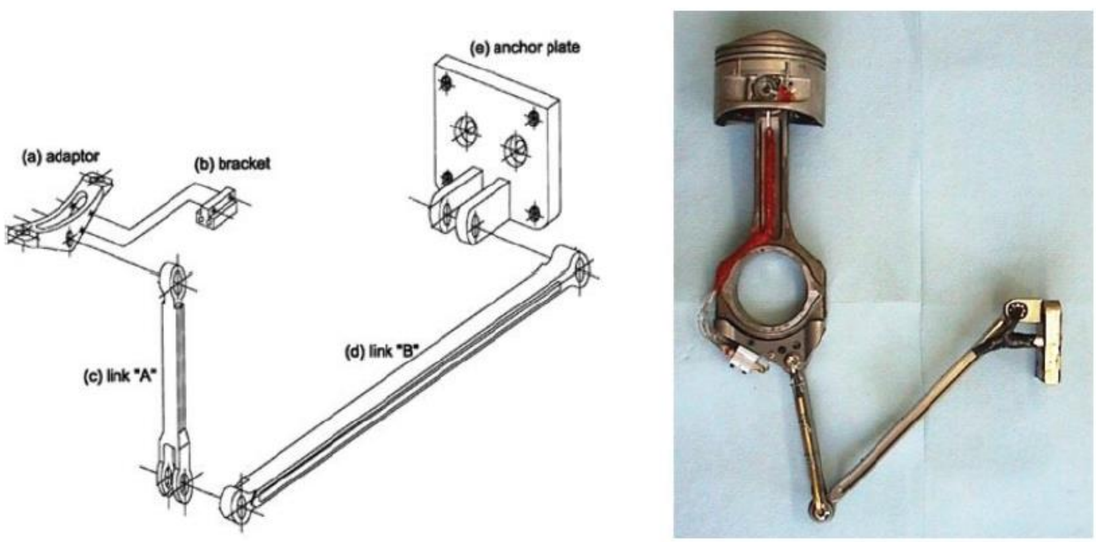


Fig 5.4 Linkage System

5.4 Connecting the thermocouples to data acquisition; the law of intermediate metals

The thermocouple wire has a low endurance when exposed to frequent flexing. Hence, it is not practical to run the thermocouple wire through the linkage, and usage of proper wire is crucial for developing a durable mechanical telemetry system. Using a wire made of a different material, e.g. stainless steel, introduces additional junctions in the circuit. The ‘Law of intermediate metals’ states that this is possible without distorting the signal, as ‘a third metal inserted between the two dissimilar metals of a thermocouple junction will have no effect upon the output voltage as long as the two junctions formed by the additional metal are at the same temperature’. Past studies showed that the temperature of the underside of the piston can be maintained constant at given operating conditions; hence, establishing a reference thermocouple junction at that location and measuring the reference temperature with a thermistor allows the temperature

of the hot junction to be determined. The special stainless steel wire with high resistance to flexing can then be used to transfer the signal from the reference junction, through the telemetry and out of the engine. Special isothermal aluminium plates were machined and installed on the inner side of skirts. The inner skirt surface was also machined in a way that ensures close contact and consistency of cold junction conditions (Fig. 5.5).

The ‘Ultra miniature stainless steel medical wire’ was selected for transferring the signal from the isothermal plate onwards. This is a finely stranded stainless steel wire with a nominal outside diameter of 0.356mm, insulated with clear non-hygroscopic fluorocarbon and applicable in the range from280^0 C to 200^0 C.

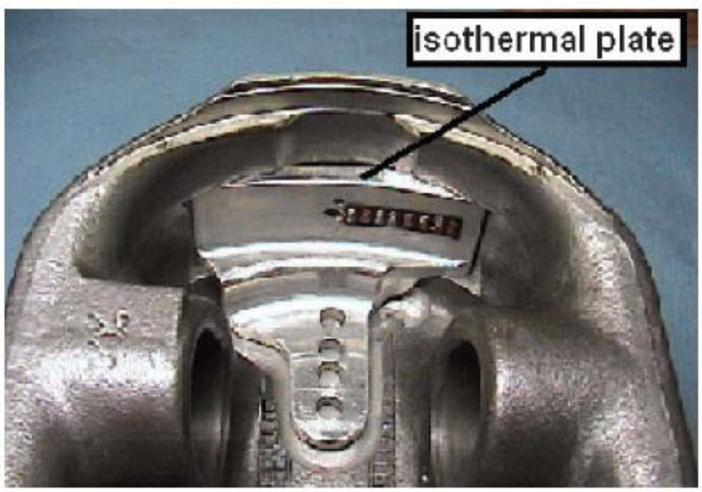


Fig. 5 Piston and isothermal plate

Fig 5.5

5.5 Heat flux calculation

The heat flux at the surface of the piston or the combustion chamber can be found by solving a one dimensional time-dependent heat conduction equation with time-varying temperature for the surface boundary condition and a constant temperature at a depth beneath the surface. The experimental surface temperature variation can be expressed as

$$T_{\mathrm{w}}(t)=T_{\mathrm{m}}+\sum_{n=1}^{N}[A_n \cos(n\omega t)+B_n \sin(n\omega t)] \qquad (1)$$

where An and Bn are Fourier coefficients, n is a harmonic number, and v is the angular frequency of the temperature cycle. Tm is the time-averaged experimental surface temperature. The heat flux can be expressed as

$$q=\frac{k}{\delta}(T_{\mathrm{m}}-T_{\delta})+ k\sum_{n=1}^{N}\varphi_n[(A_n+B_n)\cos(n\omega t)-(A_n-B_n)\sin(n\omega t)] \qquad (2)$$

where $\varphi_n=\sqrt{n\omega/2\alpha}$ and α is the thermal diffusivity of the wall material (and is equal to k/rcp where k, r, and cp are thermal conductivity, density, and specific heat respectively). In this study, the number N of harmonic components required for high accuracy was 40

5.6 Global heat loss calculation

The heat release calculations are critical for subsequent comparisons of the local surface measurements with global heat transfer parameters in the combustion chamber. After the net mass fraction is determined, the heat loss calculated using a selected heat transfer correlation is scaled to match the total energy released during the cycle. The latter is determined from the fuel energy content multiplied by the combustion efficiency. The heat flux determined after such scaling of the heat transfer model represents accurately the global process affecting the bulk gas in the chamber. Comparing the local measurements with the global profile will indicate how well the classic correlations behave when applied to a DISI engine.The analysis is based on a single-zone ideal-gas model of the combustion process. The algorithm is based on the work of Gatowski et al. and the final form of the model is

$$\frac{\mathrm{d}Q_{\mathrm{ch}}}{\mathrm{d}t}=\frac{\gamma}{\gamma-1}p\frac{\mathrm{d}V}{\mathrm{d}t}+\frac{1}{\gamma-1}V\frac{\mathrm{d}p}{\mathrm{d}t}-\frac{\mathrm{d}Q_{\mathrm{loss}}}{\mathrm{d}t} \qquad (3)$$

where $\mathrm{d}Q_{\mathrm{ch}}/\mathrm{d}t$ is the chemical heat release rate, γ is the specific heat ratio, p is cylinder pressure, V is cylinder volume, and $\mathrm{d}Q_{\mathrm{ioss}}/\mathrm{d}t$ is the heat loss rate.

The global heat loss is

$$\frac{dQ_{loss}}{dt} = hA_s(T - T_w) \qquad (4)$$

where h is the convective heat transfer coefficient, As is the cylinder surface area, T is the gas temperature, and Tw is the estimated cylinder wall temperature. The convective heat transfer coefficient is estimated using one of the well-known global heat transfer models for internal combustion engines. Correlations proposed by Woschni and by Hohenberg were considered.

Once the convective heat transfer coefficient has been estimated with the global heat transfer models, the heat loss scaling factor b is determined such that the sum of the integrated net heat release energy and the heat loss is equal to the expected gross energy release from the fuel according to

$$\beta = \frac{\eta_c E_{fuel} - \int Q_n}{\int Q_{loss}} \qquad (5)$$

where η_c is the combustion efficiency, E_{fuel} is the energy content in the fuel (m_{fuel} LHV), Q_n is the net heat release rate, and Q_{loss} is the heat loss rate. The expected energy release $\eta_c E_{fuel}$ from the mass of the fuel sets the upper bound for the cumulative heat release, and the calculated peak gross heat release $\int Q_n + \int Q_{loss}$ should ideally reach this value. The crevice loss is neglected. The heat loss scaling factor is a good measure of the validity of the heat release calculation.

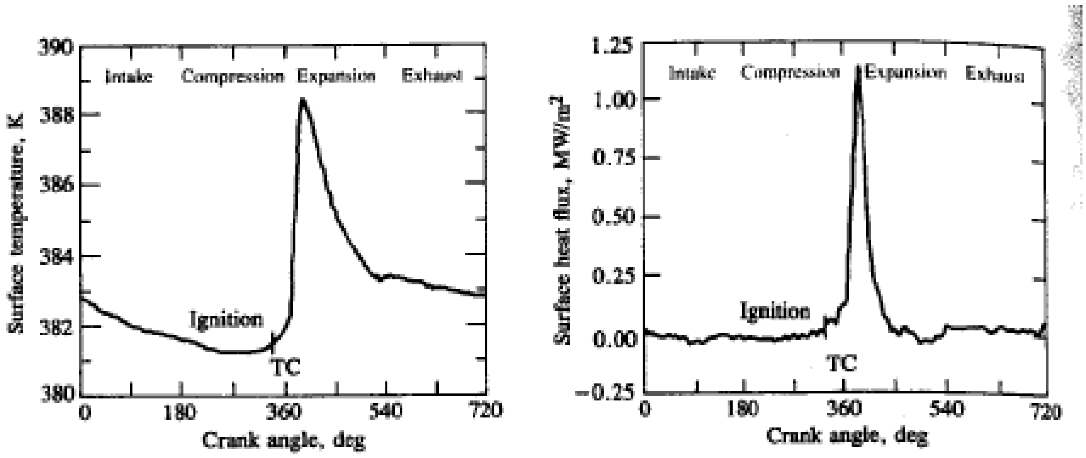


Fig 5.6

Surface temperature measured with thermocouple in cylinder head, and surface heat flux calculated from surface temperature, as a function of crank angle. Spark-ignition engine operated at part-load.[34]

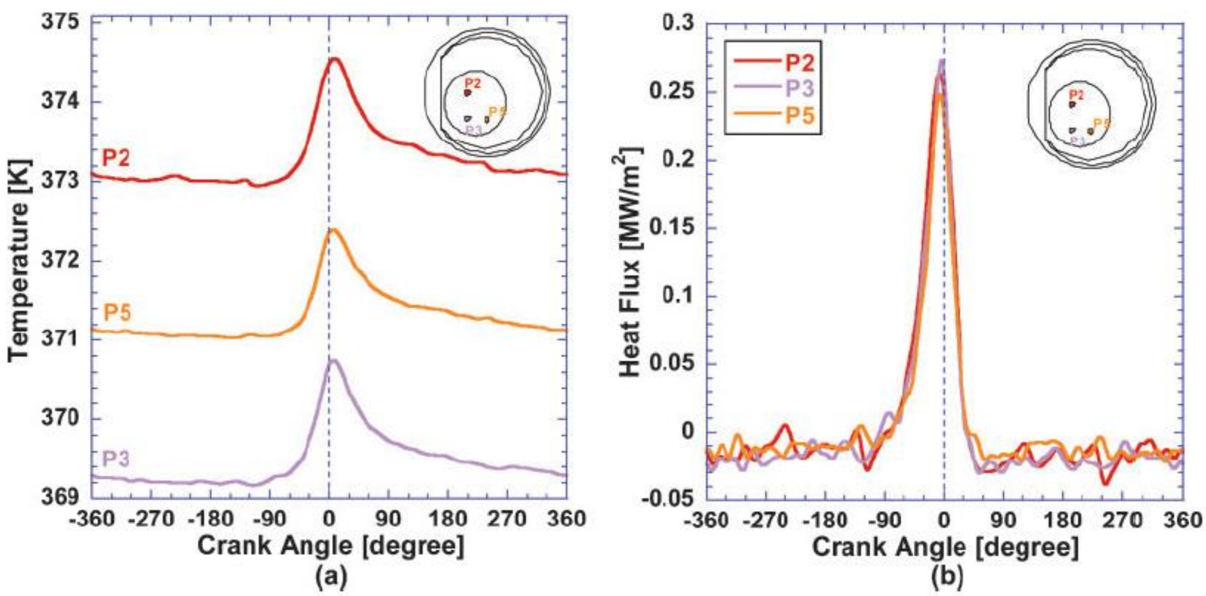


Fig 5.7

Histories of (a) the temperature and (b) the heat flux averaged over 50 cycles for 600 r/min, motoring operation

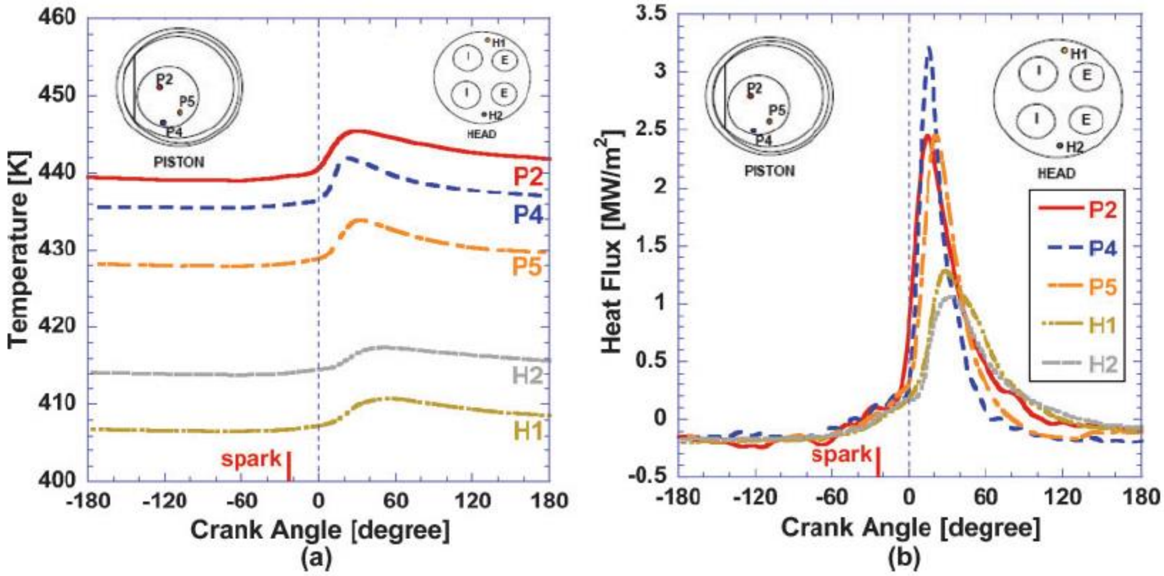


Fig 5.8 Histories of (a) the temperature and (b) the heat flux for the 2000 r/min, homogeneous mode

5.7 Thermal loading and component temperatures

The heat flux to the combustion chamber walls varies with engine design and operating conditions. Also, the heat flux to the various parts of the combustion chamber is not the same. As a result of this non uniform heat flux and the different thermal impedances between locations on the combustion chamber surface and the cooling fluid, the temperature distribution within engine components is non uniform. This section reviews the variation in temperature and heat flux in the components that comprise the combustion chamber

Component Temperature Distributions

Normally, the heat flux is highest in the center of the cylinder head, in the exhaust valve seat region, and to the center of the piston. It is lowest to the cylinder walls. Cast-iron pistons run about 40 to**80•‹C** hotter than aluminum pistons. With flat-topped pistons (typical of sparkignition engines) the center of the crown is hottest and the outer edge cooler by **20** to **50•C.** Diesel engine piston crown surface temperatures are about **50•C** higher than SI engine equivalent temperatures the maximum piston temperatures with DI diesel engine pistons are at the lip of the bowl. In ID1 diesel engines, maximum piston temperatures occur where the pre chamber jet impinges on the piston crown.

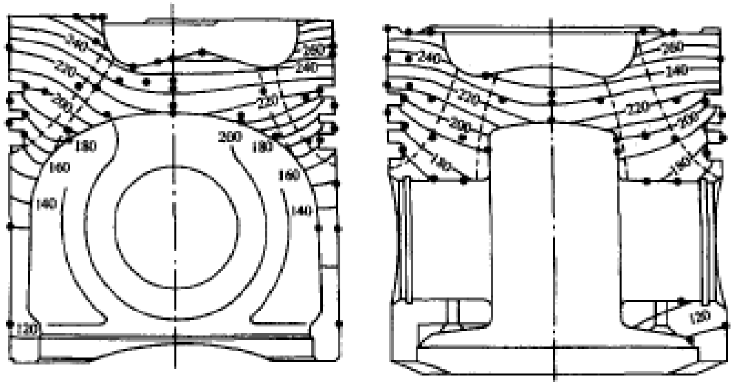


Fig 5.9

Isothermal contours (solid lines) and heat flow paths (dashed lines) determined from measured temperature distribution in piston of high-speed DI diesel engine. Bore 125 mm, stroke 110 mm, $r_c = 17$, 3000 rev/min, and full load.[40]

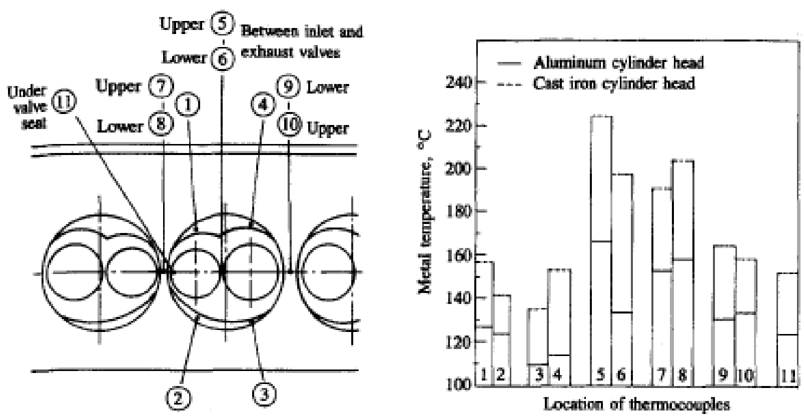


Fig 5.10

Variation of cylinder head temperature with measurement location in spark-ignition engine operating at 2000 rev/min, wide-open throttle with coolant water at 95°C and 2 atm.[41]

Figure 5.10 shows the temperatures at various locations on a four-cylinder SI engine cylinder head. The maximum temperatures occur where the heat flux is high and access for cooling is difficult. Such locations are the bridge between the valves and the region between the exhaust valves of adjacent cylinders. Figure 5.11 shows how the average heat flux and temperature vary along the length of a DI diesel engine liner. Because the lower regions of the liner are only exposed to combustion products for part of the cycle after significant gas expansion has occurred, the heat flux and temperature decrease significantly with distance from cylinder head.

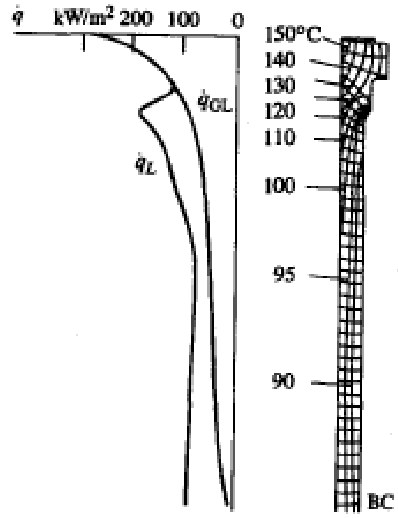


Fig 5.11

Temperature and heat flux distribution in the cylinder liner of a high-speed DI diesel engine at 1500 rev/min and bmep = 11 bar. $\dot{q}_L$ is heat flux into the liner; $\dot{q}_{GL}$ is heat flux from the gas to the liner. Difference is friction-generated heat flux.[42]

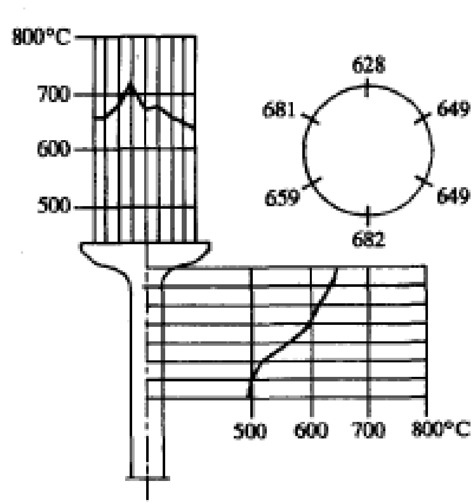


Fig 5.12

Temperature distribution in one of the four exhaust valves of a two-stroke-cycle uniflow DDA 4-53 DI diesel engine. Bore = 98 mm, stroke = 114 mm.[43]

Note that the heat generated by friction between the piston and the liner, the difference between q_{gl}, (the gas to liner heat flux) and q_L (the total heat **flux** into the liner), is a significant fraction of the liner thermal loading. The exhaust valve is cooled through the stem and the guide, and the valve seat. In small-size valves the greater part of the heat transfer occurs through thestem; with large-size valves, the valve seat carries the higher thermal load. Temperature distributions in engine components can be calculated from a knowledge of the heat fluxes across the component surface using finite element analysis techniques. For steady-state engine operation, the depth within a component to which the unsteady temperature fluctuations (caused by the variations in heat flux during the cycle) penetrate is small, so a quasi-steady solution is satisfactory. Results from such calculations for a spark-ignition engine piston illustrate the method.44 A mean heat-transfer coefficient from the combustion chamber gases to the piston crown and a mean chamber gas temperature were defined

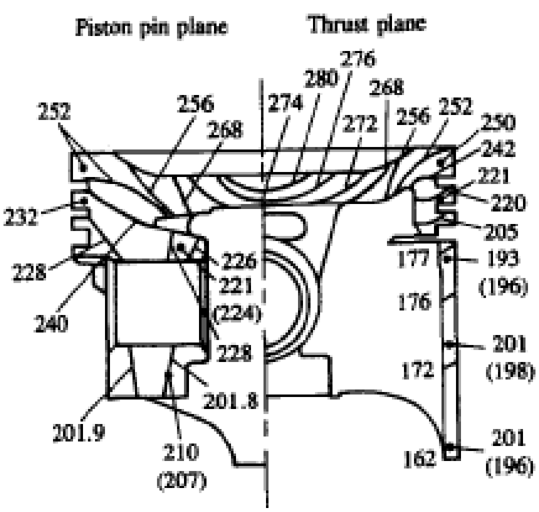


Fig 5.13

Measured (dots) and calculated temperature (°C) distributions in piston pin and thrust planes of the piston of a four-cylinder 2.5-dm^3 spark-ignition engine at 4600 rev/min and wide-open throttle.[44]

These define the time-averaged heat flux into the piston. Heat-transfer coefficients for the different surfaces of the piston (underside of dome, ring-land areas, ring regions, skirt outer and inner surfaces, wrist pin bearings, etc.) were estimated. The actual piston shape was approximated with a three-dimensional grid for one quadrant of the piston. A standard finite element analysis of the heat flow through the piston yields the temperature distribution within the piston. The thermal stresses can therefore be calculated and added to the mechanical stress field to determine the total stress distribution. This can be used to define the potential fatigue regions in the actual piston design. Figure 5.13 shows the temperature distribution calculated with this approach, compared with measurements (indicated by dots). The agreement is acceptable, except in the piston skirt where the heat-transfer rate between the skirt and cylinder liner has been overestimated.

Detailed measurements of the temperature distribution in the piston allow the relative amounts of heat which flow out of the different piston surfaces to be estimated. Figure 5.14 shows examples of such estimates for a DI diesel engine at no-load and full-load. About 70 percent of the heat flows out through the ring zone, and much smaller amounts through the pin boss zone, underside of the crown and skirt. In larger diesel engines and highly loaded diesel engines, one or more cooling channels are incorporated into the piston crown. This reduces the heat flow out through the ring area significantly.

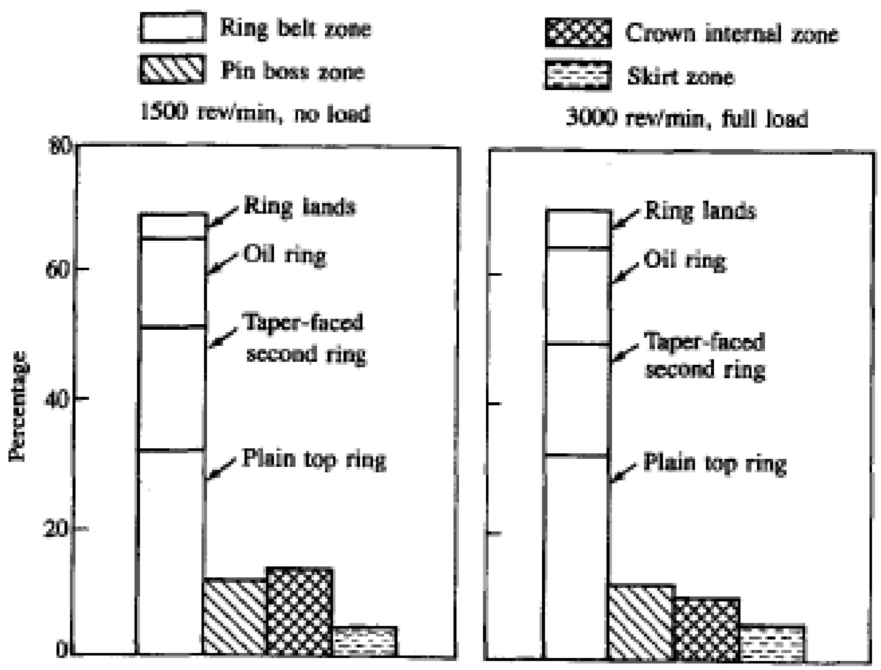


Fig 5.14

Heat outflow from various zones of piston as percentage of heat flow in from combustion chamber. High-speed DI diesel engine, 125 mm bore, 110 stroke, $r_c = 17$.[40]

Effect of Engine Variables

The following variables affect the magnitude of the heat flux to the different surfaces of the engine combustion chamber and the temperature distribution in the components that comprise the chamber: engine speed; engine load; overall equivalence ratio; compression ratio; spark or injection timing; swirl and squish motion; mixture inlet temperature; coolant temperature and composition; wall material; wall deposits. Of these variables, speed and load have the greatest effect

Equation (12.19), derived from the Nusselt-Reynolds number relation

$$h_c = \text{constant} \times B^{-0.2} p^{0.8} T^{-0.55} w^{0.8}$$

and the relation for heat-transfer rate per unit area [Eq. (12.2)]

$$\dot{q} = h_c(T - T_w)$$

are useful for predicting trends as engine operating and design variables change. The effect of the above variables on engine and component heat flux will now be summarized. The comments which follow apply primarily to spark- ignition engines. In compression-ignition engines, the distribution of heat flux and temperature varies greatly with the size of cylinder and form of the combustion chamber.

SPEED, LOAD, AND EQUIVALENCE RATIO. Predictions of spark-ignition engine heat transfer as a function of speed and load are shown in Fig. 12-25. The cycle heat transfer is expressed as a percent of the fuel's chemical energy (mass of fuel × Q_{LHV}). The heat transfer to the total combustion chamber surface (excluding the exhaust port) was calculated using a thermodynamic-based cycle simulation (see Sec. 14.4). The relative importance of heat losses *per cycle* decreases as speed and load increase: the *average* heat transfer *per unit time*, however, increases as speed and load increase.

Since speed and load affect p, T, and w in Eq. (12.19), simpler correlations have been developed to predict component heat fluxes from experimental data. Time-averaged heat fluxes at several combustion chamber locations, determined from measurements of the temperature gradient in the chamber walls, have been fitted with the empirical expression

$$\dot{q} = \text{constant} \times \left(\frac{\dot{m}_f}{A_p}\right)^n$$

with n between 0.5 and 0.75 (the value of n depending on engine type and location within the combustion chamber). Results for a modern four-cylinder SI

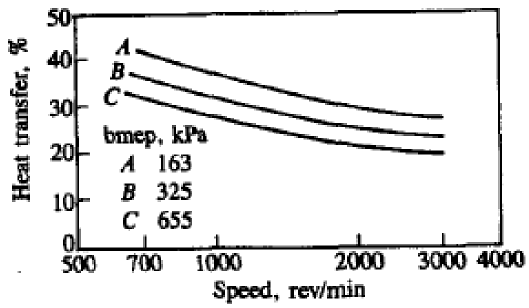


Fig 5.15

Predicted average heat-transfer rate (as percent of fuel flow rate × Q_{LHV}) to combustion chamber walls of an eight-cylinder 5.7-dm^3 spark-ignition engine as a function of speed and load. Stoichiometric operation; MBT timing.[46]

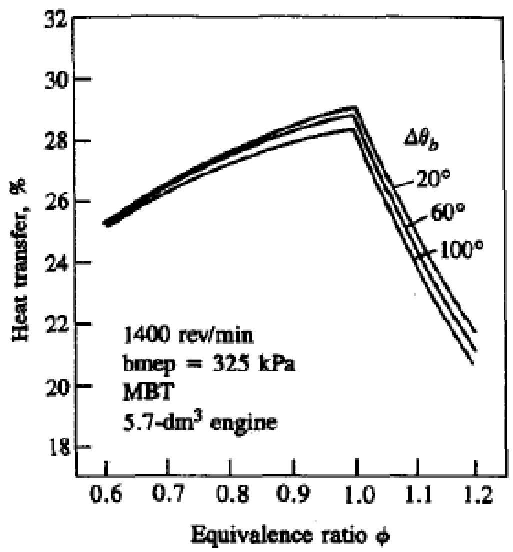


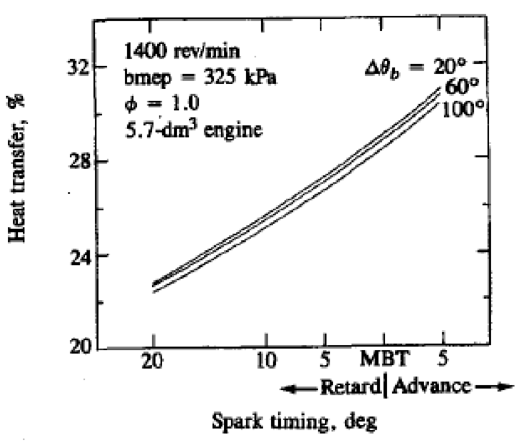
Fig 5.16

Predicted average heat-transfer rate (as percent of fuel flow rate × Q_{LHV}) to combustion chamber walls of an eight-cylinder 5.7-dm^3 spark-ignition engine as a function of equivalence ratio and burn rate ($\Delta\theta_b$ = combustion duration).[46]

engine and several diesel engine designs, with appropriate values of n, can be found in Refs. 47 to 49. While this correlation is not dimensionless and does not satisfy Eq. (12.19), it provides a convenient method for reducing the experimental data. The heat flux to the cylinder head and liner for a spark-ignition engine were well correlated by Eq. (12.38) with $n = 0.6$. The flux distribution over the cylinder head at a fuel flow rate per unit piston area of 0.195 kg/s · m^2 for several different DI diesel engines were comparable in magnitude. The effect of speed at wide-open throttle on component temperatures for a spark-ignition engine can be found in Ref. 47. Exhaust valve, piston crown and top ring groove, and nozzle throat temperatures for a Comet prechamber diesel as a function of fuel flow rate can be found in Ref. 49.

The peak heat flux in an SI engine occurs at the mixture equivalence ratio for maximum power $\phi \approx 1.1$, and decreases as ϕ is leaned out or enriched from this value.[50] The major effect is through the gas temperature in Eqs. (12.2) and (12.19). However, as a fraction of the fuel's chemical energy, the heat transfer per cycle is a maximum at $\phi = 1.0$ and decreases for richer and leaner mixtures, as shown by the thermodynamic-based cycle-simulation predictions in Fig. 12-26. In CI engines, the air/fuel ratio variation is incorporated directly in the load variation effects.

COMPRESSION RATIO. Increasing the compression ratio in an SI engine decreases the total heat flux to the coolant until $r_c \approx 10$; thereafter heat flux increases slightly as r_c increases.[50] The magnitude of the change is modest; e.g., a 10 percent decrease in the maximum heat flux (at the valve bridge) occurs for an increase in r_c from 7.1 to 9.4.[47] Several gas properties change with increasing compression ratio (at fixed throttle setting): cylinder gas pressures and peak burned gas temperatures increase; gas motion increases; combustion is faster; the surface/volume ratio close to TC increases; the gas temperature late in the expansion stroke and during the exhaust stroke is reduced. Measured mean exhaust temperatures confirm the last point, which probably dominates the trend at lower



Predicted average heat-transfer rate (as percentage of fuel flow rate × Q_{LHV}) to combustion chamber walls of an eight-cylinder 5.7-dm^3 spark-ignition engine as a function of spark timing and burn rate ($\Delta\theta_b$ = combustion duration).[46]

compression ratios. As the compression ratio increases further, the other factors (which all increase heat transfer) become important.

The effect of changes in compression ratio on component temperatures depends on location. Generally, head and exhaust valve temperatures decrease with increasing compression ratio, due to lower expansion and exhaust stroke temperatures. The piston and spark plug electrode temperatures increase, at constant throttle setting, due to the higher peak combustion temperatures at higher compression ratios. If knock occurs (see Sec. 9.6), increases in heat flux and component temperatures result; see below.

SPARK TIMING. Retarding the spark timing in an SI engine decreases the heat flux as shown in Fig. 12-27. A similar trend in CI engines with retarding injection timing would be expected. The burned gas temperatures are decreased as timing is retarded because combustion occurs later when the cylinder volume is larger. Temperature trends vary with component. Piston and spark plug electrode temperatures change most with timing variations; exhaust valve temperature increases as timing is retarded due to higher exhausting gas temperatures.[47]

SWIRL AND SQUISH. Increased gas velocities, due to swirl or squish motion, will result in higher heat fluxes. Equation (12.19) indicates that the effect on local heat flux, relative to quiescent engine designs, should be proportional to (local gas velocity)$^{0.8}$. There is no direct evidence to support this correlation but there is evidence that use of a shrouded value to increase gas velocities within the cylinder increases the total heat transfer.[50]

INLET TEMPERATURE. The heat flux increases linearly with increasing inlet temperature; the gas temperatures throughout the cycle are increased. An increase of 100 K gives a 13 percent increase in heat flux.[51]

COOLANT TEMPERATURE AND COMPOSITION. Increasing liquid coolant temperature increases the temperature of components directly cooled by the liquid coolant. Figure 12-28 shows the result of a 50-K rise in coolant temperature in a spark-ignition engine. The exhaust valve and spark plug temperatures are unchanged. The smaller response of the metal temperatures to coolant temperature change occurs at higher heat flux locations (such as the valve bridge), and indicates that heat transfer to the coolant has entered the nucleate-boiling regime in that region. The response is greater where heat fluxes are lower (e.g., the cylinder liner), indicating that there heat transfer to the coolant is largely by forced convection. When nucleate boiling occurs (i.e., when steam bubbles are formed in the liquid at the metal surface, although the bulk temperature of the coolant is below the saturation temperature), the metal temperature is almost independent of coolant temperature and velocity. Addition of antifreeze (ethylene glycol) to coolant water changes the thermodynamic properties of the coolant.

WALL MATERIAL. While the common metallic component materials of cast iron and aluminum have substantially different thermal properties, they both operate with combustion chamber surface temperatures (200 to 400°C) that are low relative to burned gas temperatures. There is substantial interest in using materials that could operate at much higher temperatures so that the heat losses from the working fluid would be reduced. Ceramic materials, such as silicon nitride and

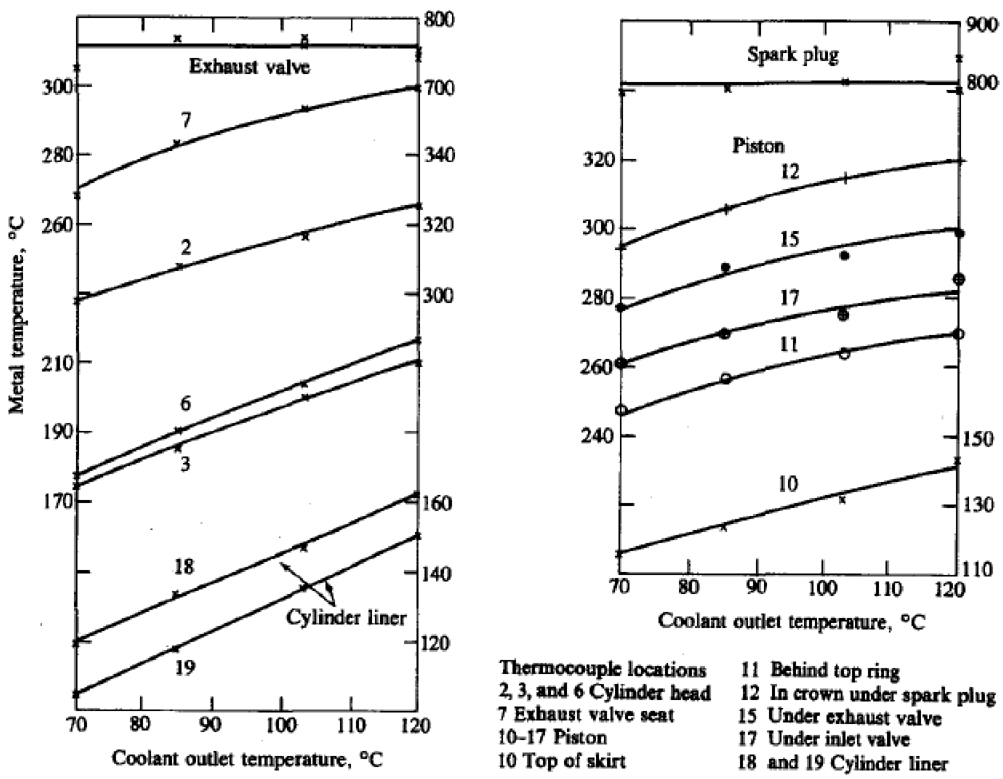


Fig 5.18

Effect of coolant temperature on cylinder head, liner, exhaust valve, valve seat, piston, and spark plug metal temperatures. Spark-ignition engine at 5520 rev/min and wide-open throttle. $r_c = 8.5$.[47]

Thermal properties of wall materials

Material	Thermal conductivity k, W/m·K	Density ρ, kg/m^3	Specific heat c, J/kg·K	Thermal diffusivity α, m^2/s	$k\rho c$	Skin depth δ, mm	Peak temperature swing, K
Cast iron	54	7.2×10^3	480	1.57×10^{-5}	1.8×10^8	2.8	18
Aluminum	155	2.75×10^3	915	6.2×10^{-5}	3.9×10^8	5.4	12
Reaction-bonded silicon nitride	5–10	2.5×10^3	710	2.8×10^{-6}	1.3×10^7	1.2	70
Sprayed zirconia	1.2	5.2×10^3	732	3.2×10^{-7}	4.6×10^6	0.39	95

zirconia, have lower thermal conductivity than cast iron, would operate at higher temperatures, and thereby insulate the engine. The thermal properties of some of these materials are listed in Table 12.2. With these thermally insulating materials it is possible to reduce the heat transfer through the wall by a substantial amount.

This approach is most feasible for diesel engines where there is the possibility of eliminating the conventional engine coolant system and improving engine efficiency. Since the coolant-side heat transfer is essentially steady during each cycle, a high enough thermal resistance in the wall material can bring the net heat transfer close to zero. However, there is still substantial heat transfer between the working fluid in the cylinder and the combustion chamber walls. Figure 12-29 illustrates these heat-transfer processes by comparing the mean gas temperature to the piston surface temperature for metal and ceramic combustion chamber wall materials. The results come from a thermodynamic simulation of a turbocompounded diesel engine system operating cycle. From Eq. (12.2) the heat transfer is *from* the gas when $T_g > T_w$ and *to* the gas when $T_g < T_w$. With the ceramic material at about 800 K surface temperature, the *net* heat transfer is much reduced compared with the metal case. However, there is substantial heat transfer to the gas from the ceramic walls during intake (which reduces volumetric efficiency) and compression (which increases compression stroke work), and still substantial heat transfer from the gas during combustion and expansion.

The heat transfer from the hot walls to the incoming charge makes thermally insulating materials unattractive for spark-ignition engines. Such heat transfer would increase the unburned mixture temperature leading to earlier onset of knock (see Sec. 9.6).

KNOCK. Knock in an SI engine is the spontaneous ignition of the unburned "end-gas" ahead of the flame as the flame propagates across the combustion chamber. It results in an increase in gas pressure and temperature above the normal combustion levels (see Sec. 9.6). Knock results in increased local heat fluxes to regions of the piston, the cylinder head, and liner in contact with the end-gas. Increases to between twice and three times the normal heat flux in the end-gas region have been measured.[13, 52] It is thought that the primary knock damage to the piston crown in this region is due to the combination of extremely high local pressures and higher material temperatures.