HEAT TRANSFER

Course code:AME016

III. B.Tech II semester

Regulation: IARE (R-16)

BY

Dr. Ch. Sandeep

&

Dr. K. Ch. Appa Rao

Associate Professors

DEPARTMENT OF MECHANICAL ENGINEERING

INSTITUTE OF AERONAUTICAL ENGINEERING

(Autonomous)

DUNDIGAL, HYDERABAD - 500 043

CO's	Course outcomes
CO1	Understand the mechanisms of heat transfer and applying the laws to convert into mathematical model with respect to the modes and steady state process.
CO2	Derive and formulate the mathematical models for steady state heat transfer phenomenon and the applicability to different surfaces and geometries.
CO3	Understand the concepts of convective heat transfer and solving problems with various processes like free and forced convection.

CO's	Course outcomes
CO4	Explore the concept of Boundary layer and obtaining the derivation for empirical relations. Understanding the concept of condensation & boiling and radiation heat transfer.
CO5	Understand the concepts of different types of heat exchangers and applying LMTD and NTU methods for solving heat exchangers in real time problems.

UNTI– I
BASIC CONCEPTS

CLOs	Course Learning Outcome
CLO1	Understand basic concepts of heat transfer modes, Fourier Law and First law of thermodynamics.
CLO2	Remember the basic laws of energy involved in the heat transfer mechanisms.
CLO3	Understand the physical system to convert into mathematical model depending upon the mode of Heat Transfer.
CLO4	Understand the thermal response of engineering systems for application of Heat Transfer mechanism in both steady and unsteady state problems.

Heat Transfer

- Heat is a transfer of energy from one object to another due to a difference in temperature
- Temperature is a measure of the molecular energy in an object
- Heat always flows from an object of higher temp (T_H) to one of lower temp (T_L)
- We are often interested in the rate at which this heat transfer takes place

Three modes of heat transfer

- Conduction
- Convection
- Radiation

Conduction

- Molecules are in constant motion, their speed is proportionate to the temperature of the object
- When two objects come in contact, their surface molecules will transfer momentum
- An aluminum pot will conduct heat from a glass stove-top

Thermal Conductivity

- Why do tile or cement floors feel cooler than wood or carpet?
- The ability to transfer heat is an intrinsic property of a substance
- Metals are good heat conductors due to the free electrons available
- Heat transfer is energy per unit time = power

Conductive Transfer

- For two objects at T_H and the other at T_L, connected by a rod of uniform material

$$Q = kA(T_H - T_L)/L$$

Where k is the thermal conductivity of the rod, A is the cross-sectional area, and L is the length of the rod

- Home owners are concerned with the ―R-value‖ of their insulation

$R_{th} = L/k$

Impact of k

- If left alone for sufficient time, both objects will come to thermal equilibrium

- The smaller the value of k, the slower the heat transfer

- Home insulation strives to maximize this transfer time (high R-value), allowing for a temperature gradient to exist longer

Convection

- A fluid's density will change when its temperature changes (through conduction)

- This density change can create movement within the fluid

- Warmer fluid is usually less dense, and will rise

- Cooler fluid will rush in to take the place of the rising, warmer fluid
  This mixing is called convection

Types of Convection

- The previous slide describes the process of *free or natural* convection

- Using a pump or fan to assist in the mixing process is called *forced*
- *Convection*

- The daily weather is determined mostly by natural convection in the
- troposphere and the oceans

Convection in the Atmosphere

- Mixing of the atmosphere within the troposphere is mostly convection
  - Sea breeze: land warms faster, air over land rises, air from over the sea comes in
- Mechanism for energy transfer between atmospheric layers is not well understood
  - If all of the atmosphere were mixing in a convective fashion, there wouldn‘t be layers!

Radiation

- Objects tends to absorb electromagnetic waves from their surroundings.
- An ideal absorber is called a blackbody, an ideal reflector is called a white body
- Objects tend to radiate electromagnetic waves as efficiently as they absorb them
- The transfer of energy through the emission of EM waves is called radiation

Blackbody radiation

- The rate of energy radiation is related to an object's surface area A and the nature of the surface, called emissivity, $\mathcal{E}$
- The Stefan-Boltzmann Law for heat transfer is $Q = A\mathcal{E}\sigma T^4$

– Don't forget that heat transfer = energy per unit time = power

- σ is the Stefan-Boltzmann constant, which is equal to 5.67×10^{-8} $W/(m^2K^4)$

Spectral output

- The radiated EM waves from a blackbody are spread over the EM spectrum

- Early classical physics (Rayleigh-Jeans Law) predicted that radiation would increase as wavelength decreased, which was not observed. This was called the ultraviolet catastrophe

What is Heat Transfer?

“Energy in transit due to temperature difference.‖

Thermodynamics tells us:

- How much heat is transferred (δQ)
- How much work is done (δW)
- Final state of the system

Heat transfer tells us:

- How (with what modes) δQ is transferred
- At what rate δQ is transferred
- Temperature distribution inside the body

$\boxed{\text{Heat transfer}} \xrightleftharpoons{\text{complementary}} \boxed{\text{Thermodynamics}}$

MODES

- ✓ **Conduction**
  - needs matter
  - molecular phenomenon (diffusion process)
  - withoutbulk motion of matter

- ✓ **Convection**
  - heat carried away by bulk motion of fluid
  - needs fluid matter

- ✓ **Radiation**
  - does not needs matter
  - transmission of energy by electromagnetic waves

APPLICATIONS OF HEAT TRANSFER

- ✓ Energy production and conversion
  - steam power plant, solar energy conversion etc.
- ✓ Refrigeration and air-conditioning
- ✓ Domestic applications
  - ovens, stoves, toaster
- ✓ Cooling of electronic equipment
- ✓ Manufacturing/ materials processing
  - welding, casting, soldering, laser machining
- ✓ Automobiles / aircraft design
- ✓ Nature (weather, climate etc)

Heat Transfer I: The main observations and principles of heat conduction

1. Fourier law
2. Conservation of energy
3. The geothermal

Heat transfer: the sources

Sun

From the Earth interior:
- 4×10^{13} W
- 8×10^{-2} Wm^{-2}

Derives deep Processes:
- Mantle convection
- Geodynamo
- Plate tectonics
- Metamorphism
- Volcanism

From the sun:
- 2×10^{17} W
- 4×10^{2} Wm^{-2}

Derives surface processes:
- Water cycle
- Biosphere
- Rain
- Erosion

Earth

Earthquakes: 10^{11} W

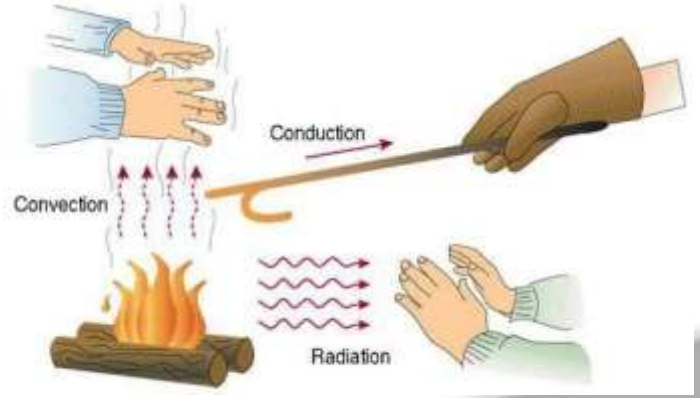
Heat transfer: the mechanisms

Three mechanisms for heat transfer: conduction, convection and radiation.

Conduction: A diffusive process wherein molecules transmit their kinetic energy to other molecules by colliding with them.

Convection: A process associated with the motion of the medium. When a hot material flows into a cold material, it will heat the region - and vise versa.

Radiation: The transfer of heat via electromagnetic radiation. Example - the Sun.



1. In the Earth, both conduction and convection are important.
2. In the lithosphere, the temperature gradient is controlled mainly by conduction.
3. Convection in the lithosphere does play a role in:

- ❖ Mid-ocean ridges in the form of hydrothermal ocean circulation.
- ❖ Volcanism and emplacement of magmatic bodies.

Heat transfer: heat flux

Heat flux is the flow per unit area and per unit time of heat. It is directly proportional to the temperature gradient.

One dimensional Fourier's law:

$$q = \quad k \frac{dT}{dy} \quad ,$$

where:
q is the heat flux
k is the coefficient of thermal conductivity
T is the temperature
y is a spatial coordinate

Units:

- q is in $[Wm^{-2}]$

- k is in $[Wm^{-1}K^{-1}]$

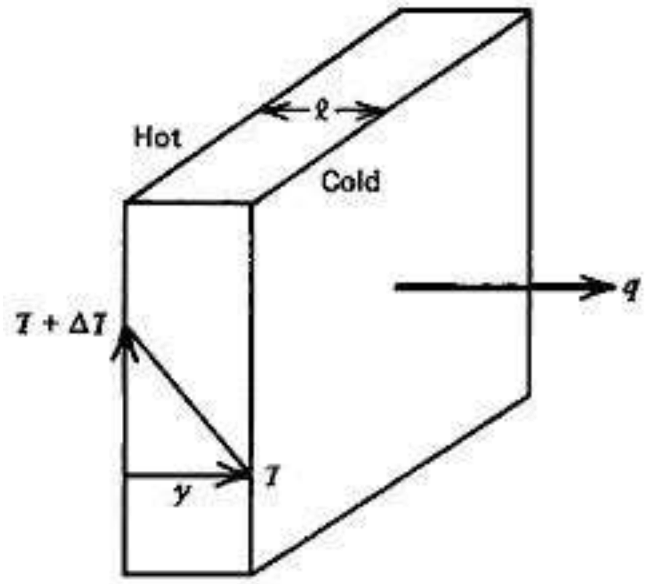
where W is read —watt‖, and is equal to Joule per second.

A substance with a large value of k is a good thermal conductor, whereas a substance with a small value of k is a poor thermal conductor or a good thermal insulator.

Example 1: a slab of thickness l, and a temperature difference of

ΔT: The heat flux is given by:

$$q = k\frac{\Delta T}{l} \quad .$$



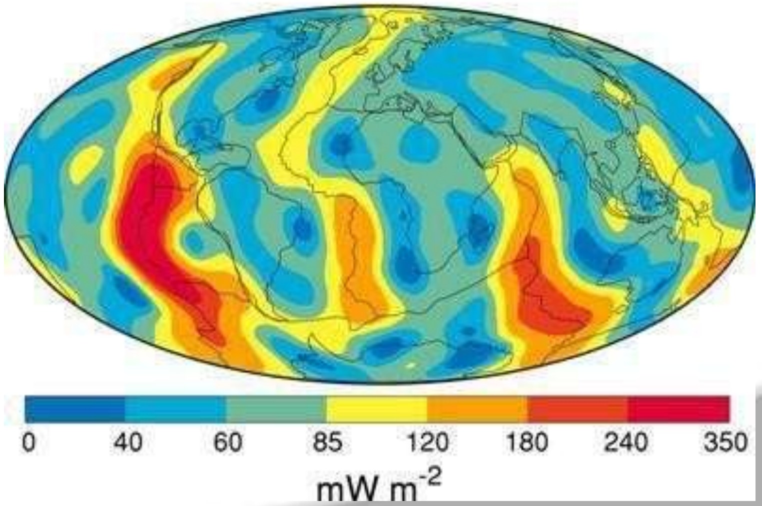
Heat transfer: world-wide heat flow

- Highest heat loss at mid-ocean ridges and lowest at old oceanic crust.

- With temperature gradient of 20-30 K/km, and thermal conductivity of 2-3 $WK^{-1}m^{-1}$, the heat flux is 40-90 mWm^{-2}.

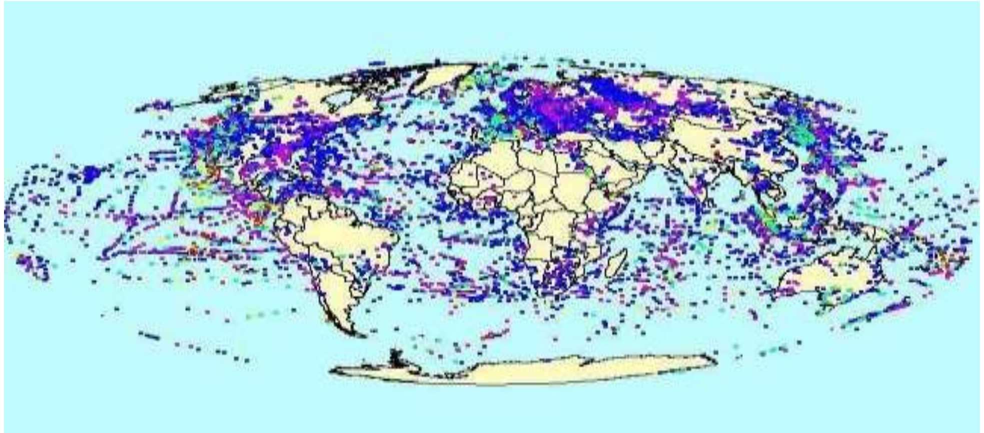
Table 7.3. *Heat loss and heat flow from the earth*

	Area (10^6 km^2)	Mean heat flow (10^{-3} W m^{-2})	Heat loss (10^{12} W)
Continents (including volcanoes)	149		8.8
Continental shelves	52		2.8
Total continents and continental shelves	201	57	11.6
Deep oceans	282		27.4
Marginal basins	27		3.0
Conductive contribution		66	20.3
Hydrothermal contribution		33	10.1
Total oceans and basins	309	99	30.4
Worldwide total	510	82	42.0



Heat transfer: measurements

Heat flow measurements: the global heat flow map on the previous slide is based on a compilation of individual measurements whose distribution is shown below.

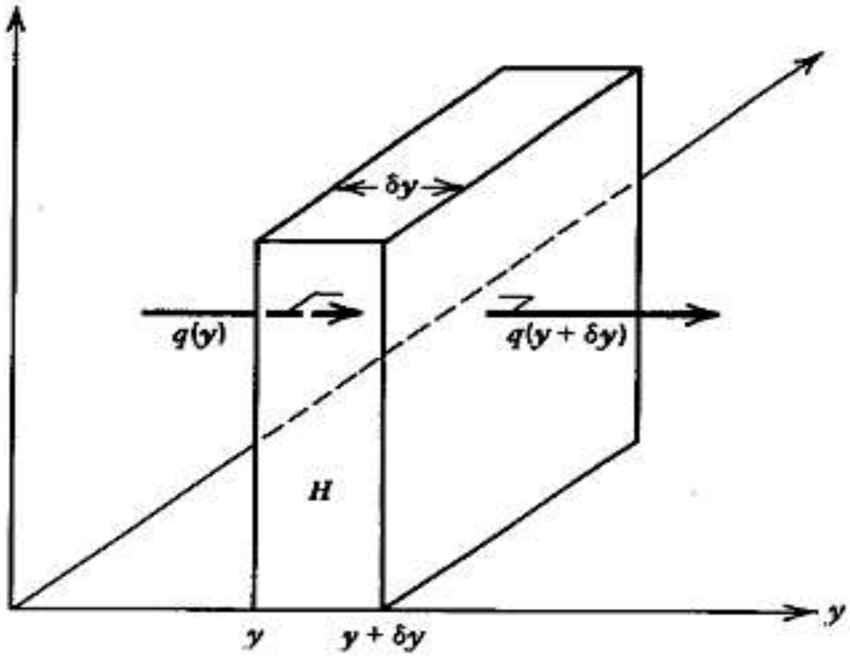


For practical reasons, the vast majority of the measurements are from continental areas in developed countries.

conservation of energy in 1-dimension

- Consider a slab of infinitesimal thickness δy; the heat flux out of the slab is q(y + δy), and the heat flux into the slab q(y).
- The net heat flow out of the slab, per unit time and per unit area of the slab's face, is:

$$q(y+\delta y) - q(y).$$



Heat transfer: conservation of energy in 1-dimension

In the absence of internal heat production, conservation of energy requires that:

$$q(y+\delta y) - q(y) = 0.$$

Since δy is infinitesimal, we can expand q(y+δy) in a Taylor series as:

$$q(y+\delta y) = q(y) + \delta y \frac{dq}{dy} + \frac{(\delta y)^2}{2} \frac{d^2 q}{dy^2} + \cdots$$

Ignoring terms higher than the first order term, leads to:

$$q(y+\delta y) - q(y) = \delta y \frac{dq}{dy} = -\delta y \, k \frac{d^2 T}{dy^2} = 0.$$

Thus:

$$\frac{d^2 T}{dy^2} = 0.$$

conservation of energy in 1-dimension

Question: in the absence of internal heat production, how does the geotherm
look like?

If there's nonzero net heat flow per unit area out of the slab, this heat must be generated internally in the slab. In that case:

$$q(y+\quad y)\quad q(y)= \quad y\frac{dq}{dy} = \quad y \quad k\frac{d^2T}{dy^2} \div = \quad y\ \ H,$$

where:

H is the heat production rate per unit mass

ρ is the density

Question: what is the source for steady-state internal heating in the Earth lithosphere?

UNIT II
ONE DIMENSIONAL STEADY STATE AND TRANSIENT CONDUCTION HEAT TRANSFER

CLOs	Course Learning Outcome
CLO5	Understand heat transfer process and systems by applying conservation of mass and energy into a system.
CLO 6	Understand the steady state condition and mathematically correlate different forms of heat transfer
CLO 7	Analyze finned surfaces, and assess how fins can enhance heat transfer
CLO 8	Remember dimensionless numbers which are used for forced and free convection phenomena.

Fourier‘s Law

“heat flux is proportional to temperature gradient‖

$$\frac{Q}{A} = q = -k\nabla T = -k\left(\frac{\partial T}{\partial x} + \frac{\partial T}{\partial y}\right)$$

units for *q* are W/m^2

where k = thermal conductivity

in general, $k = k(x,y,z,T,\ldots)$

Thermal Properties

Thermal Conductivity (k)

- It is the term used to indicate the amount of heat that will pass through a unit of area of a material at a temperature difference of one degree.
- The lower the ―k‖ value, the better the insulation qualities of the material.

Units; US: (Btu.in) / (h.ft^2.^{o}F)

Metric: W / (m.^{o}C)

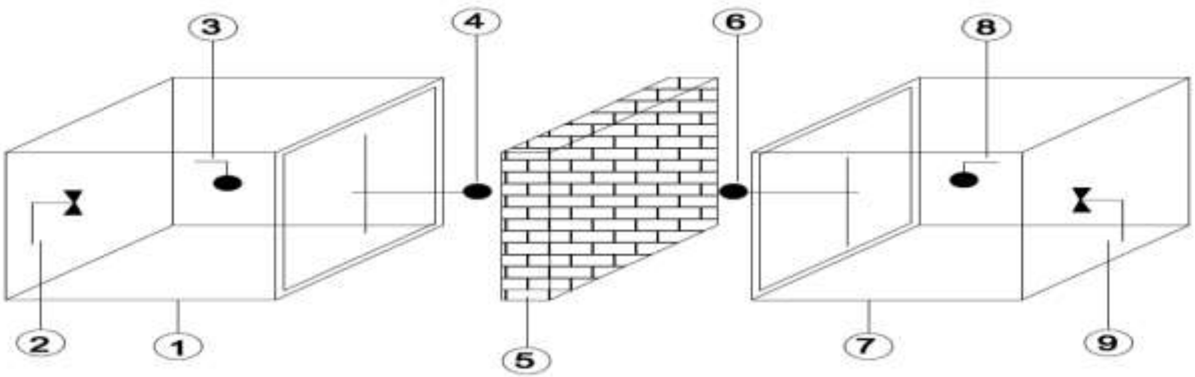
Conductance (c)

- It indicates the amount of heat that passes through a given thickness of material;
- Conductance= thermal conductivity / thickness

Units; US: Btu / (h.ft^2.^{o}F)

Metric: W/ (m^2.^{o}C)

Example:
Determination of Thermal Conductivity Coefficient for Different Wall Systems (TS EN ISO 8990)



1	Cold Chamber
2	Freeze Fan
3	Thermo- Couples (3 unit) [cold chamber]
4	Thermo- couples (9 unit) [Surface]
5	Wall specimen (1.2 x 1.2 m)
6	Thermo- couples (9 unit) [Surface]
7	Hot Chamber
8	Thermo- Couples (3 unit) [hot chamber]
9	Heather Fan

Thermal Resistance (RSI for metric unit, R for US units)

- It is that property of a material that resist the flow of heat through the material. It is the reciprocal of conductance;

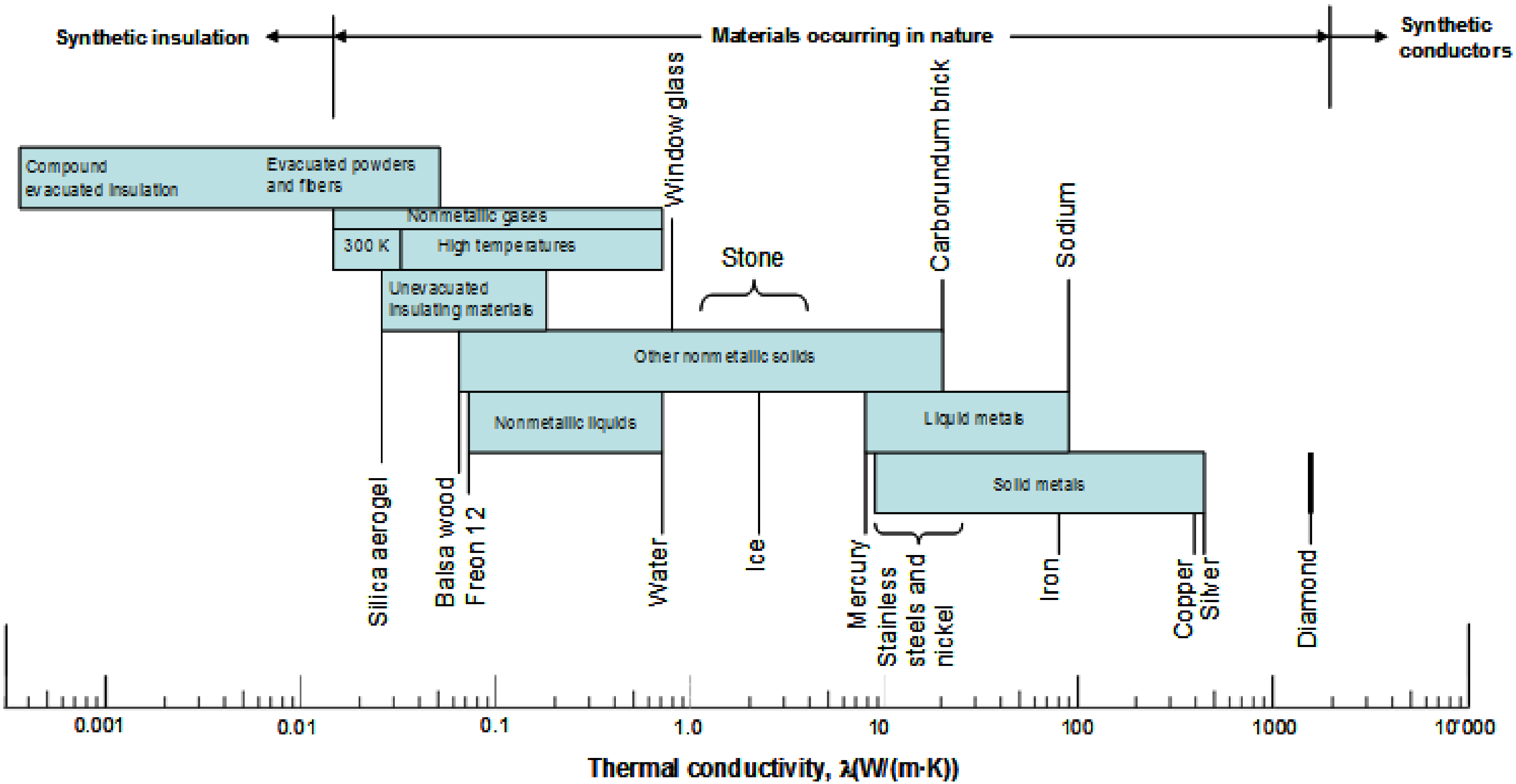
$R = 1/c$

Thermal Transmittance (U)

- It is the amount of heat that passes through all the materials in a system. It is the reciprocal of the total resistance;

$U = 1/R_t$

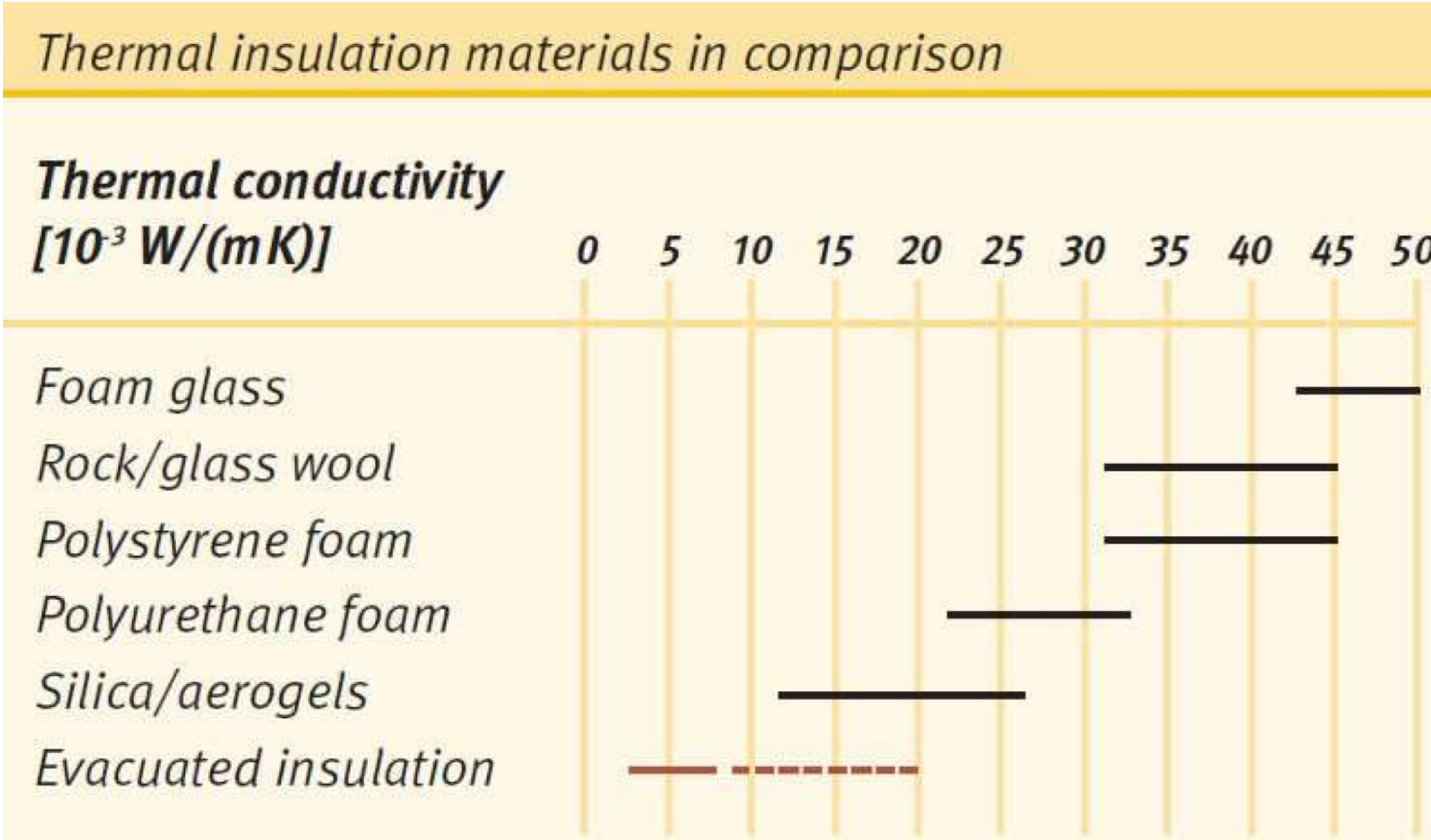
- ***Table 1 lists a few of the common materials and their thermal properties***;


Synthetic insulation
Materials occurring in nature
Synthetic conductors
Compound evacuated insulation
Evacuated powders and fibers
Nonmetallic gases
300 K
High temperatures
Unevacuated insulating materials
Window glass
Stone
Carborundum brick
Sodium
Other nonmetallic solids
Nonmetallic liquids
Liquid metals
Solid metals
Silica aerogel
Balsa wood
Freon 12
Water
Ice
Mercury
Stainless steels and nickel
Iron
Copper
Silver
Diamond
0.001
0.01
0.1
1.0
10
100
1000
10'000
Thermal conductivity, λ(W/(m·K))

Thermal conductivity is dependent on phase, temperature, density, and molecular bonding

Which pan is a better conductor, one made from copper or one made from cast iron?

	Thermal Conductivity (k) $W \cdot K^{-1} \cdot m^{-1}$
Copper	385.00
Cast Iron	80.40
Steel	50.40
Concrete	0.80
Glass	0.80
Brick	0.60
Wood	0.12-0.04
Styrofoam	0.01

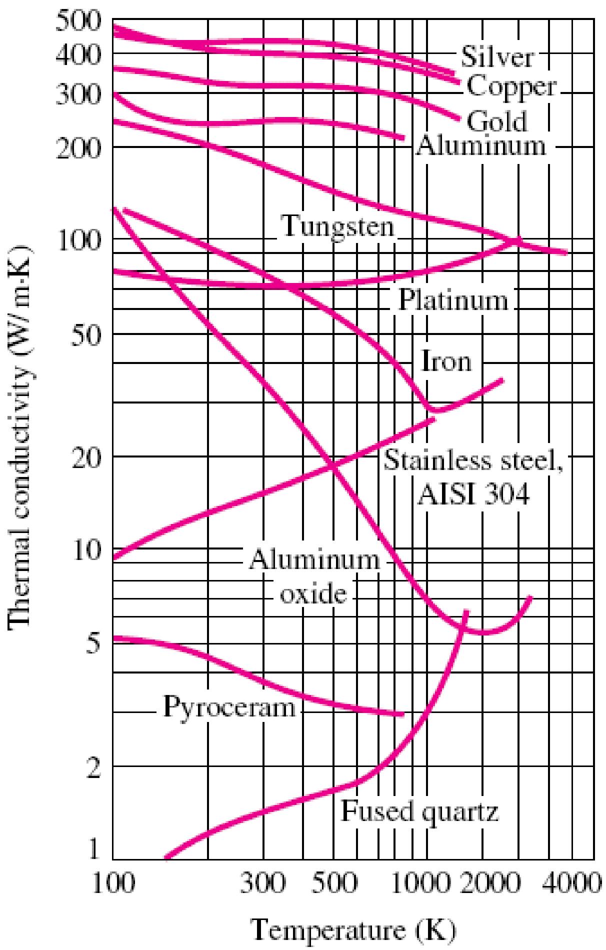

Thermal insulation materials in comparison
Thermal conductivity
[10-3 W/(m K)]
0 5 10 15 20 25 30 35 40 45 50
Foam glass
Rock/glass wool
Polystyrene foam
Polyurethane foam
Silica/aerogels
Evacuated insulation

Variable Thermal Conductivity, *k*(*T*)

▪The thermal conductivity of a material, in general, varies with temperature.

▪An average value for the thermal conductivity is commonly used when the variation is mild.

▪This is also common practice for other temperature-dependent properties such as the density and specific heat.



Variable Thermal Conductivity for One-Dimensional Cases

When the variation of thermal conductivity with temperature $k(T)$ is known, the average value of the thermal conductivity in the temperature range between T_1 and T_2 can be determined from

$$k_{ave} = \frac{\int_{T_1}^{T_2} k(T)\,dT}{T_2 - T_1} \qquad \textbf{(2-75)}$$

The variation in thermal conductivity of a material with can often be approximated as a linear function and expressed as

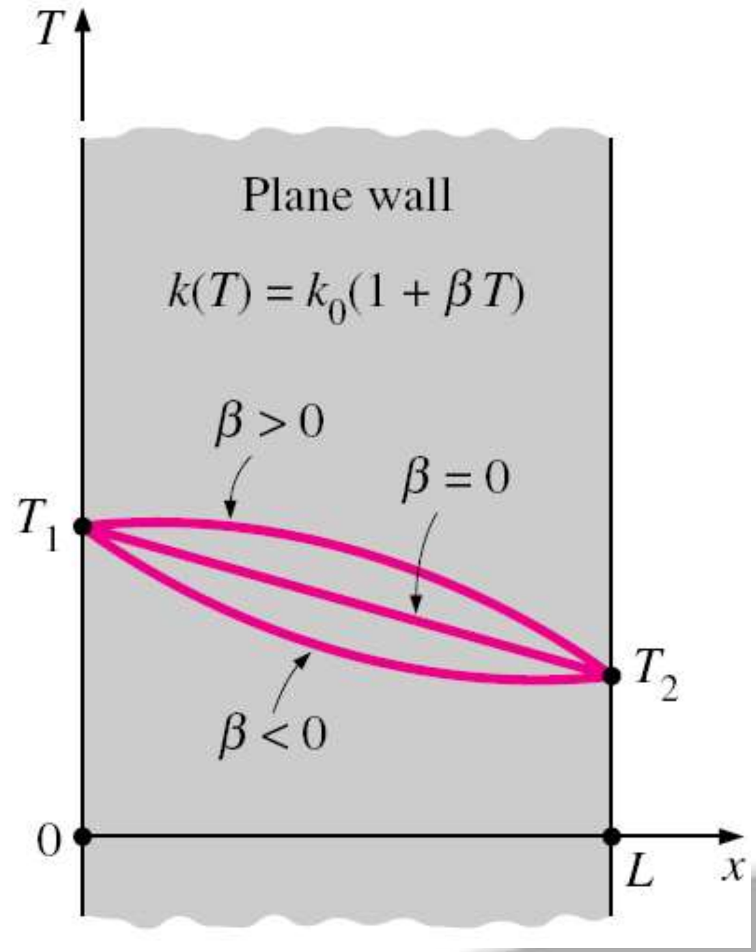
$$k(T) = k_0(1 + \beta T) \qquad \textbf{(2-79)}$$

β the **temperature coefficient of thermal conductivity**

Variable Thermal Conductivity

- For a plane wall the temperature varies linearly during steady one-dimensional heat conduction when the thermal conductivity is constant.

- This is no longer the case when the thermal conductivity changes with temperature (even linearly).



UNIT III
CONVECTIVE HEAT TRANSFER

CLOs	Course Learning Outcome
CLO 9	Understand the applications of Buckingham Pi Theorem in deriving various non dimensional numbers and their applications in heat transfer
CLO 10	Remember and use the methodology presented in tutorial to solve a convective heat transfer problems
CLO 11	Understand the various forms of free and forced convection and the application of the same in day to day problems
CLO 12	Calculate local and global convective heat fluxes using Nusselt's Theory.

Buckingham pi theorem

This note is about physical quantities R1,...,Rn. We like to measure them in a consistent system of units, such as the SI system, in which the basic units are the meter, kilogram, second, ampere, and kelvin (m, kg, s, A, K).1 As it will turn out, the existence of consistent systems of measurement has nontrivial consequences. We shall assume the fundamental units of our system of units are F1,...,Fm, so that we can write:

$$R_j = v(R_j)[R_j] = \rho_j[R_j]$$

where ρj = v(Rj) is a number, and [Rj] the units of Rj . We can write [Rj]

in terms of the fundamenta

$$[R_j] = \prod_{i=1}^{m} F_i^{a_{ij}} \qquad (j = 1, \ldots, n),.$$

It is also important for the fundamental units to be independent in the sense that

$$\prod_{i=1}^{m} F_i^{x_i} = 1 \Rightarrow x_1 = \cdots = x_m = 0.$$

We shall not be satisfied with just one system of units: The whole crux of the matter hinges on the fact that our choice of fundamental units is quite arbitrary. So we might prefer a different system of units, in which the units Fi are replaced by Fˆ i = x −1 i Fi . Here xi can be an arbitrary positive number for i = 1,...,m. We can also write our quantities in the new system thus: Rj = vˆ(Rj)[Rj + ˆ = ρˆj [Rj + ˆ .

We comp

$$R_j = v(R_j)F_1^{a_{1j}} \cdots F_m^{a_{mj}} = \underbrace{v(R_j)x_1^{a_{1j}} \cdots x_m^{a_{mj}}}_{\hat{v}(R_j)} \hat{F}_1^{a_{1j}} \cdots \hat{F}_m^{a_{mj}}$$

from which we deduce the relation

ρˆj = ρj Ym i=1 x ai j i .

For example, if $F_1 = m$ and $F_s = s$, and R_1 is a velocity, then *R1+ = ms^{-1}
$= F_1 F_2^{-1}$ and so $a_{11} = 1$, $a_{21} = -1$.

With $\hat{F}_1 = km$ and $\hat{F}_2 = h$, we find $x_1 = 1/1000$ and $x_2 = 1/3600$, and so $\hat{\rho}_1 = \rho_1 \cdot 3.6$.

Hence the example $\rho_1 = 10$, $\hat{\rho}_1 = 36$ corresponds to the relation 10m/s = 36 km/h.

Hydrodynamic &thermal boundary layer

- When a fluid flows around an object or when the object moves through a body of fluid, there exists a thin layer of fluid close to the solid surface within which shear stresses significantly influence the velocity distribution. The fluid velocity varies from zero at the solid surface to the velocity of free stream flow at a certain distance away from the solid surface.
- This thin layer of changing velocity has been called the hydrodynamic boundary layer; a concept first suggested by Ludwig Prandtl in the year 1904. Heat transfer occurs due to heat conduction and energy transport by moving fluid within this thin layer. Hence, the value of convection coefficient and heat transfer is highly dependent upon the thickness and characteristics of the boundary layer.

- **Hydrodynamic Boundary Layer: Flat Plate:**
- Consider a continuous flow of fluid along the surface of a thin plate with its sharp leading edge set parallel to flow direction.

- **The salient features of the flow situation are:**
- (i) The free stream undisturbed flow has a uniform velocity U_∞ in the x-direction. Particles of fluid adhere to the plate surface as they approach it and the fluid is slowed down considerably. The fluid becomes stagnant or virtually so in the immediate vicinity of the plate surface. Generally it is presumed that there is no slip between the fluid and the solid boundary.

- Thus, there exists a region where the flow velocity changes from that of solid boundary to that of mainstream fluid, and in this region the velocity gradients exist in the fluid. Consequently the flow is rotational and shear stresses are present. This thin layer of changing velocity has been called the hydrodynamic boundary layer.

(ii) The condition $\partial u/\partial y \neq 0$ is true for the zone within the boundary layer, whilst the conditions for flow beyond the boundary layer and its outer edge are-

$\partial u/\partial y = 0$ and $u = U_\infty$

Thus all the variation in fluid velocity is concentrated in a comparatively thin layer in immediate vicinity of the plate surface.

(iii) The concepts of boundary layer thickness and outer edge of the boundary layer are quite fictitious as there is no abrupt transition from the boundary layer to the flow beyond or outside it. Velocity within the boundary layer approaches the free stream velocity asymptotically. Usually the boundary layer thickness δ is taken to be the distance from the plate surface to a point at which the velocity is within 1 percent of the asymptotic limit, i.e., $u = 0.99\ U_\infty$. The parameter 8 then becomes a nominal measure of the thickness of boundary layer, i.e., of the region in which the major portion of velocity deformation takes place.

(iv) The thickness of the boundary layer is variable along the flow direction; it is zero at leading edge of the plate and increases as the distance x from the leading edge is increased. This aspect may be attributed to the viscous forces which dissipate more and more energy of the fluid stream as the flow proceeds. Consequently, a large group of the fluid particles is slowed down. The boundary layer growth is also governed by other parameters such as the magnitude of the incoming velocity and the kinematic viscosity of the flowing fluid.

For higher incoming velocities, there would be less time for viscous forces to act and accordingly there would be less quantum of boundary layer thickness at a particular distance from the leading edge. Further, the boundary layer thickness is greater for the fluids with greater kinematic viscosity.

(v) For some distance from the leading edge, the boundary layer is laminar and the velocity profile is parabolic in character. Flow within the laminar boundary layer is smooth and the streamlines are essentially parallel to the plate. Subsequently the laminar boundary layer becomes unstable and the laminar flow undergoes a change in its flow structure at a certain point, called transition point, in the flow field. Within a transition zone, the flow is unstable and is referred to as transition flow. After going through a transition zone of finite length, the boundary layer entirely changes to turbulent boundary layer.

(vi) The turbulent boundary layer does not extend to the solid surface. Underlying it, an extremely thin layer, called laminar sub-layer, is formed wherein the flow is essentially of laminar character. Outside the boundary layer, the main fluid may be either laminar or turbulent.

(vii) The pattern of flow in the boundary layer is judged by the Reynolds number Re = $U_\infty x/v$ where x is distance along the plate and measured from its leading edge. The transition from laminar to turbulent pattern of flow occurs at values of Reynolds number between 3×10^5 to 5×10^5. Besides this critical Reynolds number, the co-ordinate points at which deterioration of the laminar layer begins and stabilized turbulent flow sets in is dependent on the surface roughness, plate curvature and the pressure gradient, and the intensity of turbulence of the free stream flow.

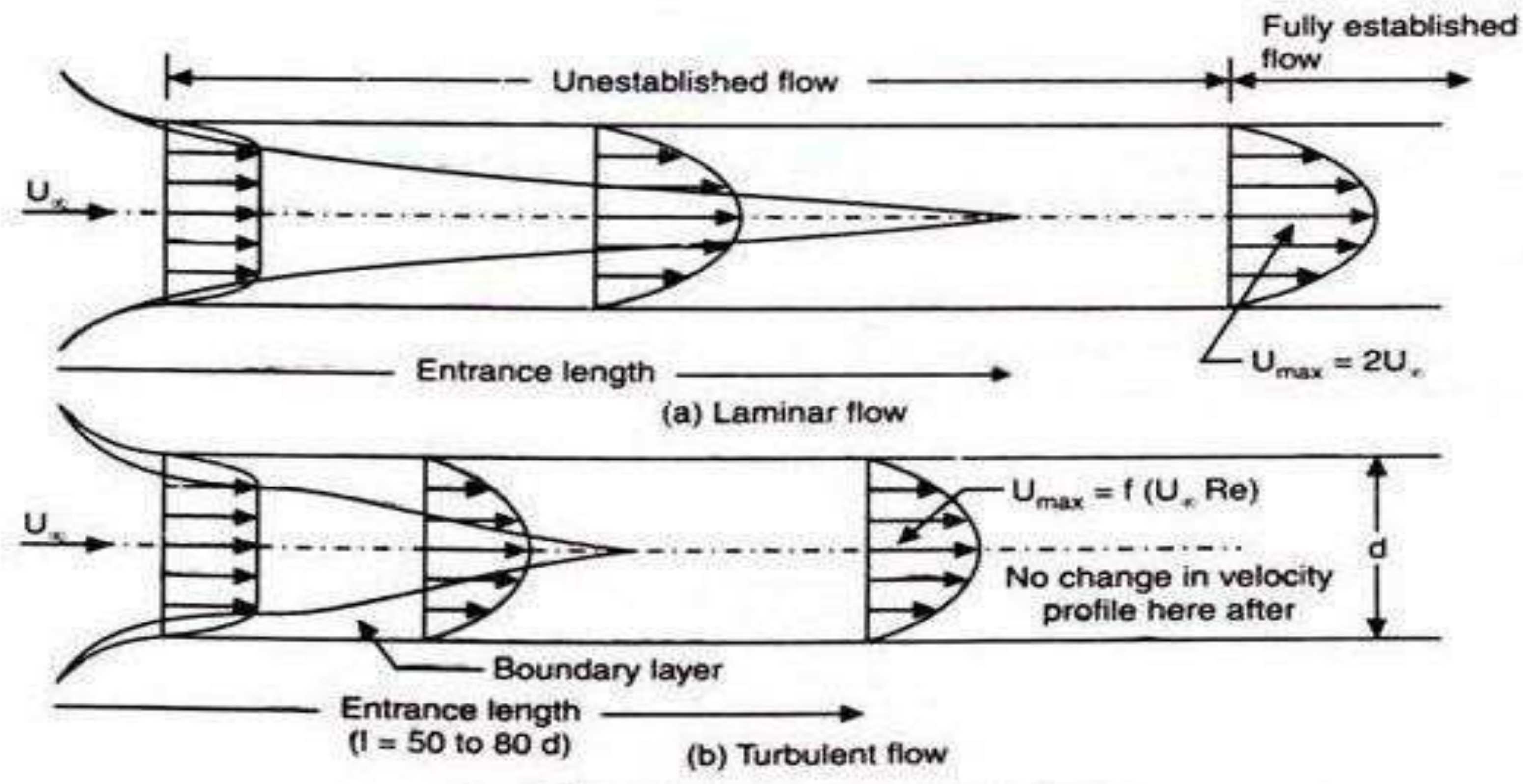
(viii) In a laminar boundary layer, the velocity gradient becomes less steep as one proceeds along the flow. It is because now the change in velocity from no slip at the plate surface to free stream value in the potential core occurs over a greater transverse distance. Nevertheless in a turbulent boundary layer, there occurs an interchange of momentum and energy amongst the individual layers comprising the boundary layer. Consequently, a turbulent boundary layer has a fuller velocity profile and a much steeper velocity gradient at the plate surface when compared to those for a laminar boundary layer.

.

(ix) Velocity gradient and hence the shear stress has a higher value at the plate surface. For a laminar boundary layer the velocity gradient becomes smaller along the flow direction and so does the shear stress. However for a turbulent boundary layer the shear stress at the plate surface again takes up a high value consistent with the steeper velocity gradient

(x) Development of boundary layer for pipe flow proceeds in a fashion similar to that for flow along a flat plate. However boundary layer is limited to the pipe radius because of the flow being within a confined passage. Boundary layers from the pipe walls meet at the center of the pipe and the entire flow acquires the characteristics of a boundary layer.

(xi) Beyond this point, the velocity profile does not change and it is said to constitute a fully-developed flow. Further, the velocity gradient and the wall shear stresses are greatest at the pipe entrance and drop to a steady value at and beyond the region of fully-developed flow.



Fully established flow
Unestablished flow
U_∞
$U_{max} = 2U_\infty$
Entrance length
(a) Laminar flow
$U_{max} = f(U_\infty Re)$
d
No change in velocity profile here after
Boundary layer
Entrance length (l = 50 to 80 d)
(b) Turbulent flow

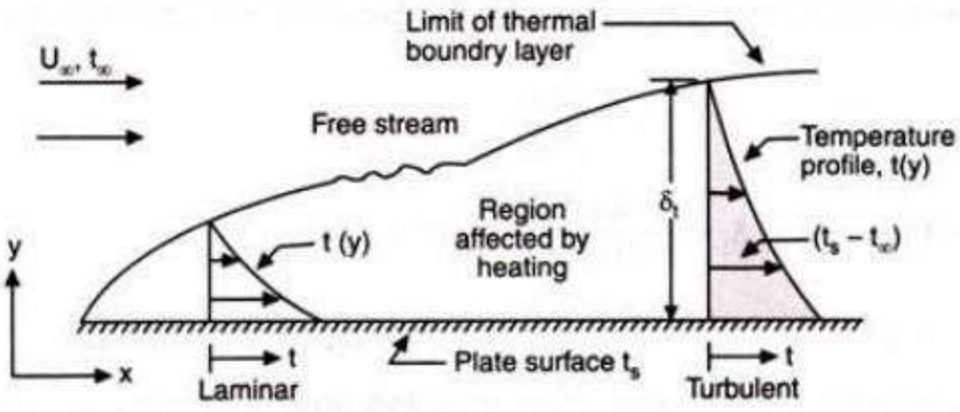
Thermal Boundary Layer:

- When a fluid flows past a heated or cold surface, a temperature field is set up in the fluid next to the surface. If the plate surface is hotter than the fluid,.

- Usually the temperature field encompasses a very small region of fluid, i.e., the region of fluid being heated by the plate is confined to a thin layer near the surface. This zone or thin layer wherein the temperature field exists is called the thermal boundary layer. The temperature gradient results due to heat exchange between the plate and the fluid.

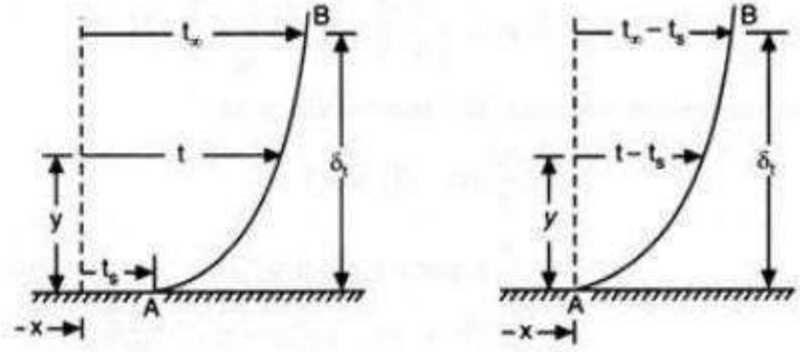
- The thickness δ_f of thermal boundary layer is arbitrarily defined as the distance y from the plate surface at which-

$$\frac{t_s - t}{t_s - t_\infty} = 0.99$$



- The convection of energy reduces the outward conduction in the fluid and consequently the temperature gradient decreases away from the surface. Further, the temperature gradient is infinite at the leading edge of the plate and approaches zero as the layer develops downstream. Moreover in the turbulent boundary layer, the action of eddies flattens the temperature profile.

- At point A, the temperature of the fluid is the same as the surface temperature t_s. The fluid temperature increases gradually until it acquires the free stream temperature t_∞. The distance AB, measured perpendicularly to the plate surface, denotes the thickness of thermal boundary at a distance x from the leading edge of the plate.



The concept of thermal boundary layer is analogous to that of hydrodynamic boundary layer; the parameters affecting their growth are, however, different. The velocity profile of the hydrodynamic boundary layer is dependent primarily upon the viscosity of the fluid.

Thermal entrance length

- A fully developed heat flow in a pipe can be considered in the following situation. If the tube wall of the pipe is constantly heated or cooled so the heat flux from the wall to the fluid via convection is a fixed value, then the bulk temperature of the fluid increases steadily at a fixed rate along the flow direction. An example can be a pipe covered entirely by an electrical heating pad, and the flow is introduced after a uniform heat flux from the pad is achieved.

- At a distance away from the entrance of the fluid, the fully developed heat flow is achieved when the heat transfer coefficient of the fluid becomes constant, and the temperature profile has the same shape along the flow. This distance is defined as the thermal entrance length, which is important for engineers to design efficient heat transfer processes.

Quantitative measurement

- Quantitatively, If x is chosen to be the axis parallel to the pipe and x= 0 is chosen as the commencing point of the pipe flow, the thermal entrance length is defined as the distance (x >0) required for the Nusselt Number Nu associated with the pipe flow to decrease to within 5% of its value for a fully developed heat flow.

- Depends on different flow conditions (laminar, turbulent, shapes of entrance, etc.), the Nusselt number has different dependence on Reynolds number and the friction factor of the flow.

Hydrodynamic Entrance Length

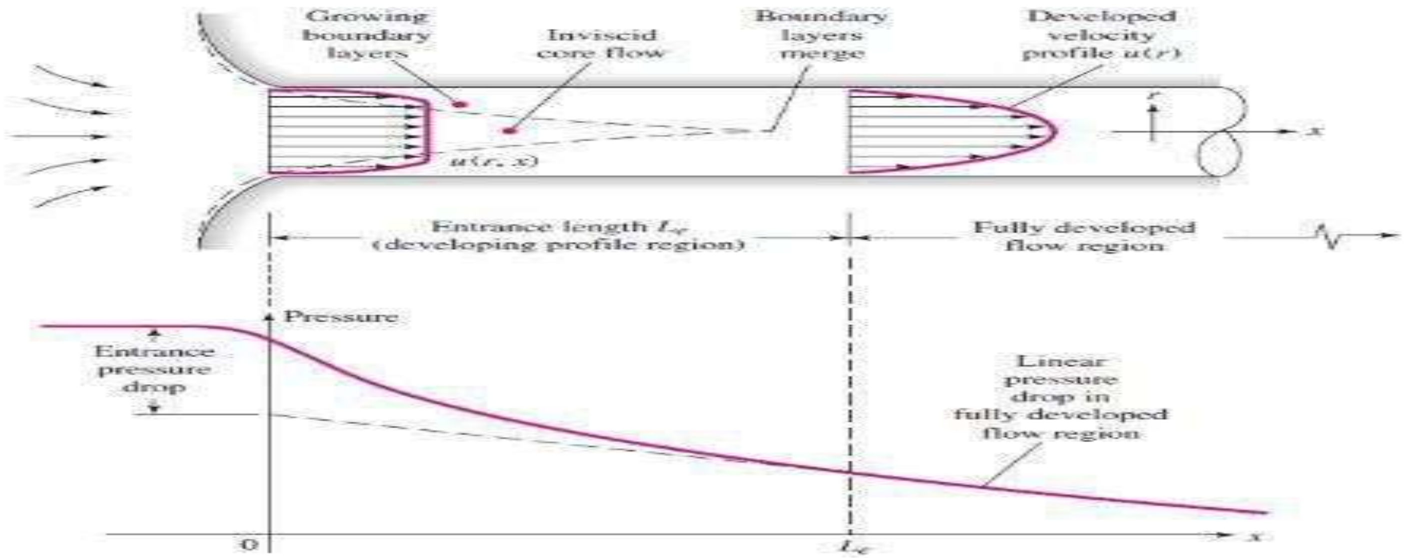
- For internal flow regime an entrance region is typical. In this region a nearly in viscid upstream flow converges and enters the tube. To characterize this region the hydrodynamic entrance length is introduced and is approximately equal to:

$$\frac{L_e}{D} = \begin{cases} 0.05\,\mathrm{Re} & \text{for laminar flow} \\ 1.36\,\mathrm{Re}^{1/4} \approx 10 & \text{for turbulent flow} \end{cases}$$

- the maximum hydrodynamic entrance length, at $Re_{D,crit} = 2300$ (laminar flow), is $L_e = 138d$, where D is the diameter of the pipe. This is the longest development length possible. In turbulent flow, the boundary layers grow faster, and L_e is

relatively shorter.

- For any given problem, L_e / D has to be checked to see if L_e is negligible when compared to the pipe length. At a finite distance from the entrance, the entrance effects may be neglected, because the boundary layers merge and the inviscid core disappears. The tube flow is then fully developed.



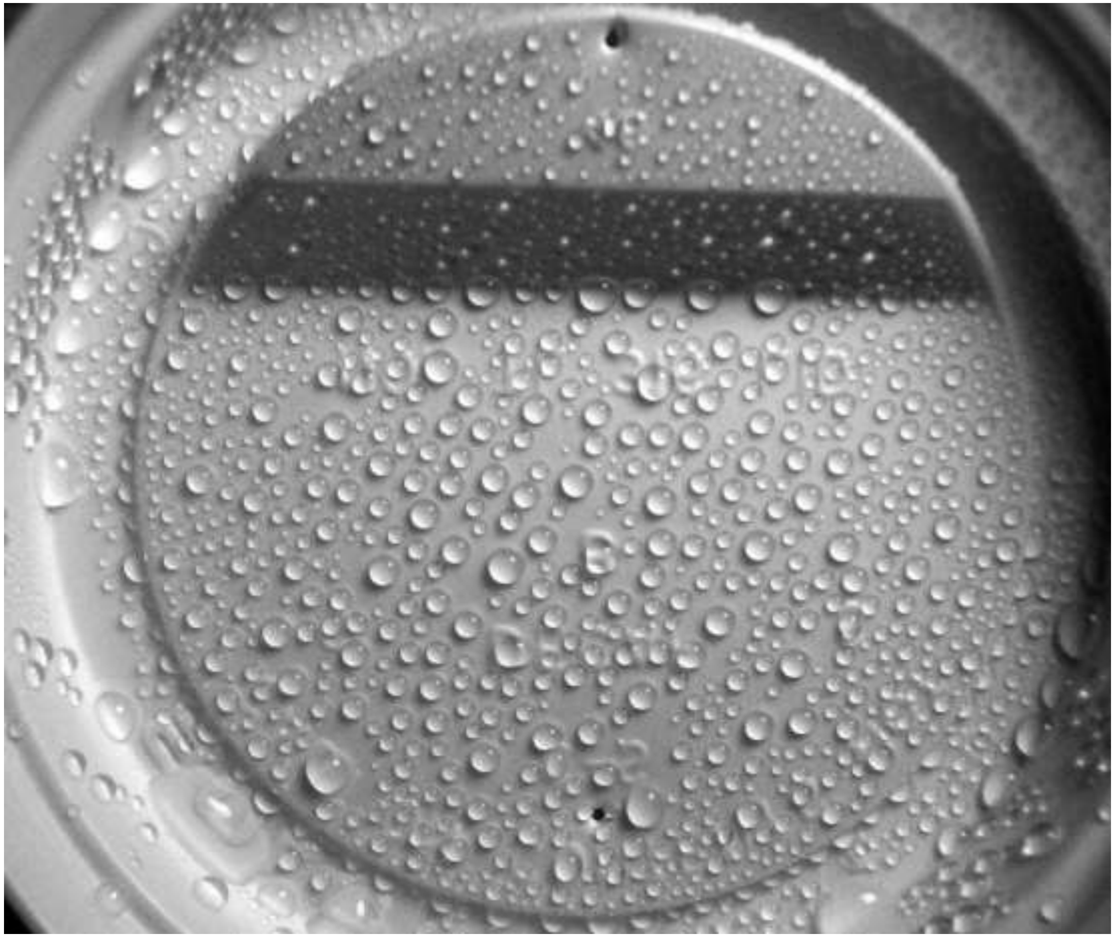
UNIT IV
HEAT TRANSFER WITH PHASE CHANGE

CLOs	Course Learning Outcome
CLO 13	Apply numerical methods to obtain approximate solutions to Taylors, Eulers, Modified Eulers
CLO 14	Runge-Kutta methods of ordinary differential equations.
CLO 15	Understand Nusselt’s theory of condensation for the application in film and drop wise condensation
CLO 16	Correlate the empirical relations in terms of vertical and horizontal cylinders during film condensation

Drop-wise condensation

- When water condenses on a hydrophobic surface, a semi-ordered array of water droplets is formed, as on the plastic coffee cup lid shown at right. Such ―dropwise‖ condensation produces higher heat transfer rates than ―filmwise‖ condensation, and is of interest for heat exchangers and condensers. Understanding the formation of drops also has applications for waterproofing, meteorology (rain, dew), and adhesion. The formation and growth of liquid droplets on a solid surface remains incompletely understood.

- We are approaching this problem along two lines: experimental observations and computer modeling. Unlike most previous computer models of condensation, our model is based on the physical processes involved (rather than simplified empirical models) and includes the nucleation of new drops as the simulation progresses



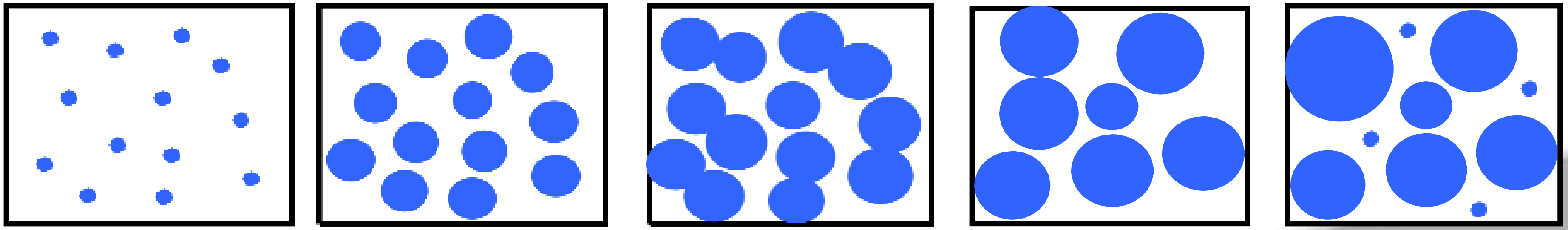
- Most prior experiments and modeling have used constant temperature. However, many natural systems where dropwise condensation occurs have varying temperature, and in many cases condensation occurs as an initially hot system cools. To better understand these systems, we included an exponential temperature decrease in the computer model, and the initially hot water in the experiments was allowed to cool naturally.

•Once the computer model has been calibrated to the experimental results, the model can be used to investigate the effect of changing various parameters, which may not be directly accessible experimentally.

Stages in Droplet Evolution

- Processes involved in dropwise condensation are:

1) Initial nucleation of drops on previously bare surface
2) Growth of drops
3) Coalescence of drops which grow large enough to touch each other
4) Additional nucleation of new drops in areas which become available due to coalescence (renucleation)
5) Loss of drops due to gravity (not included in current experiments)



- We modeled the initial nucleation as a random process (homogeneous nucleation). Droplet growth was modeled by an equation including contributions from vapor and from adsorbed molecules. Renucleation was modeled as either a random process, or as being linked to droplet coalescence.

•We initially assumed that renucleation was a random process which could occur in any area which had been cleared by coalescence. However, experimental observations revealed that renucleation did not appear to be random, but was closely associated with droplet coalescence. Coalescence not only provided the open space required for new drops to form, but also seemed to trigger the formation of new drops in neighboring areas.

- In this sequence of images, four large drops (marked by asterisks in image **a**) and several smaller drops coalesce into a single larger drop (marked by an asterisk in image **b**). In image **c** many new drops are visible due to renucleation in the area opened up by coalescence. These drops became visible from 0.2-1.0 seconds after coalescence. Note that a nearby open area (red arrow in **a**) did not experience renucleation until the coalescence occurred, while a more distant open area (purple arrows in **a** and **c**) remained clear of renucleation.

- Similar events were observed many times. Areas which appeared large enough to support renucleation remained empty until a nearby coalescence event occurred. After a coalescence event, renucleation occurred in the newly cleared area and also in nearby areas which had previously been cleared, including areas which did not appear to be directly linked to the coalescence event. Nearly all observed renucleation was associated with nearby coalescence events.

- Using the computer model, we simulated renucleation both as random and as being associated with coalescence. However, the two models produced essentially similar results.

- For the experiments, a film of Saran® wrap was stretched across a holder and placed over a beaker of water that had been heated to the desired temperature. A microscope above the film was used to observe and record the water drops that condensed on the film as the water cooled. Drops could be observed down to about 5 μm in diameter at the highest magnification, and video was captured at 30 frames per second.

- Saran ® wrap is primarily poly(vinylidene chloride) and has a water contact angle of 58°.

Drop wise condensation

1. By specially treating the condensing surface the contact angle can be changed & the surface become ‗non – wet table‘ .As the stream condenses, a large number of generally spherical beads cover the surface.

2. As the condensation proceeds ,the bead become larger, coalesce, and then strike downwards over the surface. The moving bead gathers all the static bead along its downward in its trail. The ‗bear‘ surface offers very little resistance to the transfer of heat and very high heat fluxes therefore possible.

3. Unfortunately, due to the nature of the material used in the construction of condensing heat exchangers, film wise condensation is normal.(Although many bare metal surfaces are ‗non-wet table‘ this not is true of the oxide film which quickly covers the bare material

Film wise condensation

1. Unless specially treated, most materials are wet table as condensation occurs a film condensate spreads over the surface.

2.The thickness of the film depends upon a numbers of factors, e.g. the rate of condensation ,the viscosity of the condensate and whether the surface is horizontal or vertical, etc. Fresh vapour condenses on to the outside of the film &heat is transferred by conduction through the film to the met-al surface beneath

3. As the film thickness it flows downward & drips from the low points leaving the film intact & at an equilibrium thickness.

4.The film of liquid is barrier to transfer of the heat and its resistance accounts for most of the difference between the effectiveness of film wise and drop wise condensation.

Nusselt‘s theory of condensation on a vertical plate:

- Lecture 18 1 ChE 333 – Heat Transfer Condensation and the Nusselt's Film Theory Condensation is a rather complicated process. It was Wilhelm Nusselt's idea to reduce the complexity of the real process to a rather simple model, namely that the only resistance for the removal of the heat released during condensation occurs in the condensate film. The following gives an explanation of the Nusselt theory at the example of condensation on a vertical wall.

- Condensation occurs if a vapor is cooled below its (pressure dependent) saturation temperature. The heat of evaporation which is released during condensation must be removed by heat transfer, e.g. at a cooled wall. Figure 1 shows how saturated vapor at temperature Ts is condensing on a vertical wall whose temperature Tw is constant and lower than the saturation temperature.

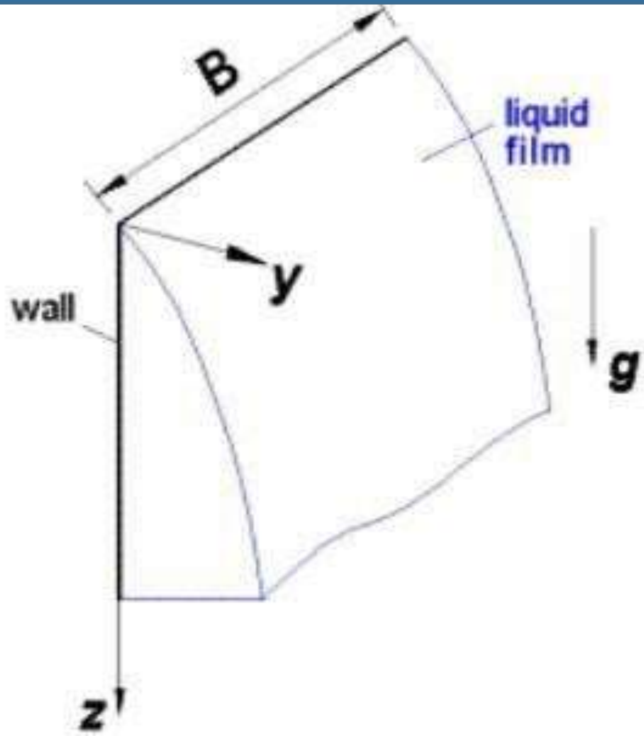


Figure 1

A condensate film develops which flows downwards under the influence of gravity. As condensation occurs over the whole surface the thickness of the film increases.

For laminar film flow heat can be transferred from the film surface to the wall only by heat conduction through the film (Figure 2).

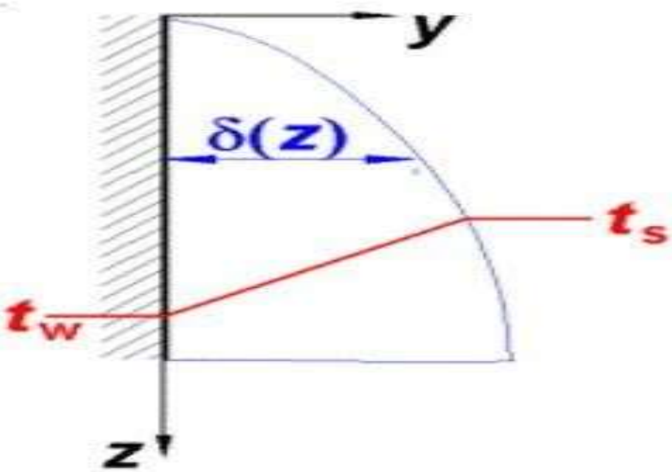


Figure 2

The local heat flux at position z through the film due to conduction is

$$q_z = \frac{k}{\delta(z)}(T_s - T_w)$$

where k is the thermal conductivity of the condensate (which is assumed to be constant) and is the film thickness at position z.

From the definition of the local heat transfer coefficient h_{loc} ,

$$q_z = h_{loc}(T_s - T_w)$$

it follows that

$$h_{loc} = \frac{k}{\delta(z)}$$

The problem is reduced to the calculation of the film thickness profile . If is known integration of equations (1) and (3) over the whole surface yields the total heat flow and the mean heat transfer coefficient.

The first step in determining is the calculation of the velocity profile w(y) in the condensate film (see Figure 3).

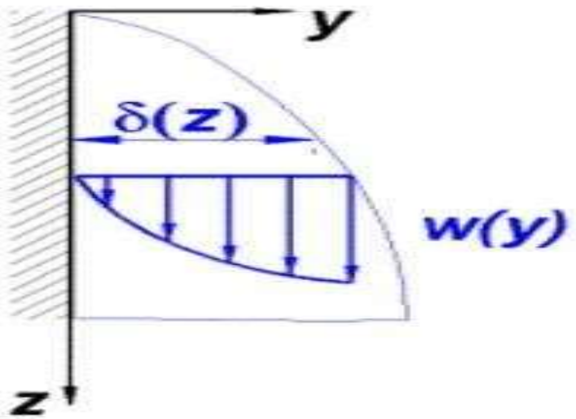


Figure 3

w(y) can either be determined by applying the Navier-Stokes equation or directly from a force balance for a fluid element in the film (force exerted by the shear stress equals force of gravity minus buoyancy

$$\mu \frac{d^2 w}{d y^2} = -(\rho - \rho_v) g$$

$$\left. \frac{dw}{dy} \right|_{y=\delta} = 0 \;;\; w(0) = 0$$

and the solution is

$$w(y) = -\frac{(\rho - \rho_v) g}{\mu} \left(\delta y - \frac{y^2}{2} \right)$$

Using the velocity profile equation (5) we can now calculate the condensat mass flow rate by integrating from y=0 to y= δ:

The result is:

$$\dot{m} = \frac{\rho(\rho - \rho_v) g B}{\mu} \frac{\delta^3}{3}$$

By differentiating equation (6) we can also determine the change in the mass flow rate with the film thickness:

$$\frac{d\dot{m}}{d\delta} = \frac{\rho(\rho - \rho_v)gB}{\mu}\delta^2$$

The change of the condensate mass flow rate results from the condensation of vapor and requires the heat flow

$$d\dot{Q} = \Delta H_v \, d\dot{m} = qBdz$$

to be removed (= enthalpy of evaporation). Using equations (1) and (7) the differential equation for the film thickness as a function of the coordinate z is:

$$\delta^2 \frac{d\delta}{dz} = \frac{k\,\mu}{\rho(\rho - \rho_v)g}(T_s - T_w)$$

Integration of equation (8) with the boundary condition , $\delta(0) = 0$, yields

$$\delta = \left[\frac{k\,\mu\,(T_s - T_w)}{\Delta H_v \rho(\rho - \rho_v)g}\right]^{1/4}$$

The film thickness increases with the fourth root of the coordinate z.

By substituting δ, according to equation (9) into equation (3) the local heat transfer coefficient follows:

$$h_{loc} = \frac{k}{\delta} = \left[\frac{\Delta H_v \rho(\rho - \rho_v)gk^3}{4\mu\,(T_s - T_w)z}\right]^{1/4}$$

Finally, the mean heat transfer coefficient for a wall of height L can be calculated by integrating the local heat transfer coefficient, h_{loc}, from z = 0 to z = L:

$$h_m = \frac{1}{L}\int_0^L h_{loc} dz = 0.943\left[\frac{\Delta H_v \rho(\rho - \rho_v) g k^3}{4\mu (T_s - T_w) L}\right]^{1/4}$$

As we can see from this equation, the heat transfer coefficients are large for small temperature differences ts-tw and heights L. In both cases the condensate film is thin and hence the heat transfer resistance is low. Equation (11) can also be used for film condensation at the inner or outer walls of vertical tubes if the tube diameter is large compared to the film thickness. All fluid properties in equation (11) with the exception of the vapor density are best evaluated at the mean temperature

$$T_m = \frac{3}{4}T_w + \frac{3}{4}T_s$$

ρ_v is evaluated at the saturation temperature T_s.

Nusselt derived a similar equation for film condensation on horizontal tubes using a numerical integration. The mean heat transfer coefficient for a single horizontal tube of diameter D is

$$h_m = 0.728\left[\frac{\Delta H_v \rho(\rho - \rho_v) g k^3}{\mu (T_s - T_w) D}\right]^{1/4}$$

In spite of the simplifications heat transfer coefficients from the Nusselt theory are surprising accurate. Measured heat transfer coefficients are up to +25% higher than the values calculated from above equations. The main reason for the deviation is the formation of waves on the film surface which isn't considered in the Nusselt theory. These waves lead to an improvement in the heat transfer. Whitaker recommends for the rippling falling condensate film a value 20% larger so that we might use

$$h_m = 1.137\left[\frac{\Delta H_v \rho(\rho - \rho_v) g k^3}{\mu (T_s - T_w) L}\right]^{1/4}$$

The flow in the film could become turbulent at Re > 1800, in which case in the coefficient might be represented as

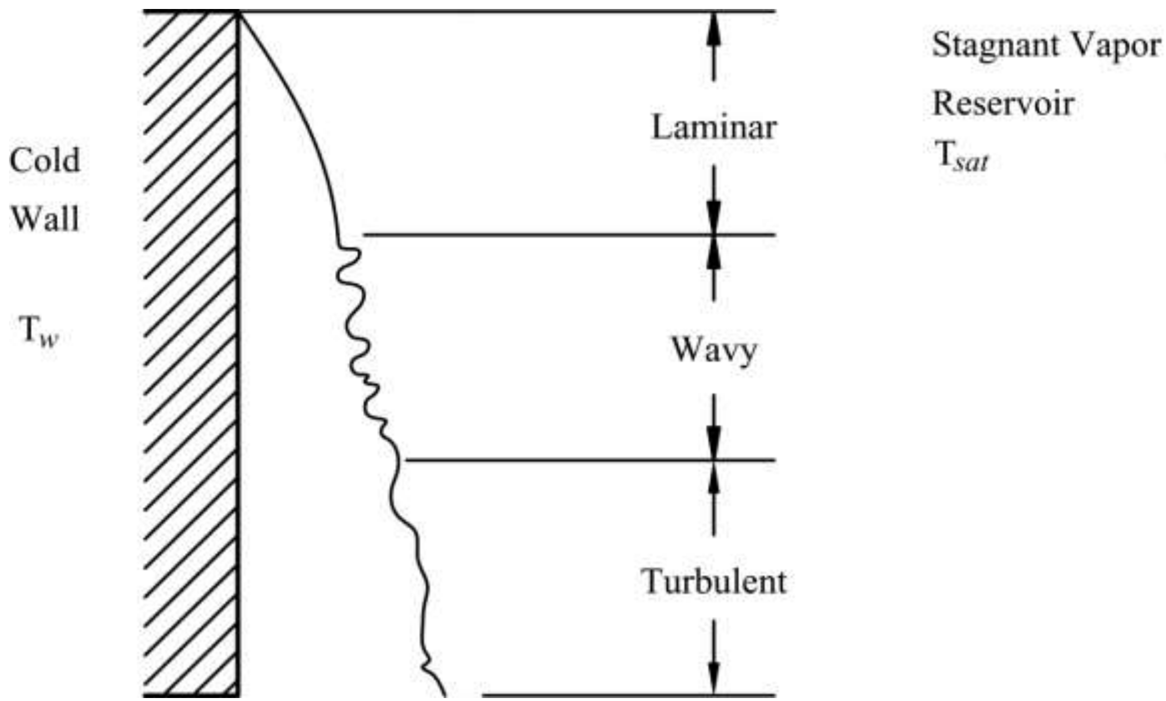
$$Re = \frac{4}{3}\frac{g}{v^2}\left[\frac{4 v k (T_s - T_w) L}{\Delta H_v \rho g}\right]$$

so that the heat transfer coefficient is

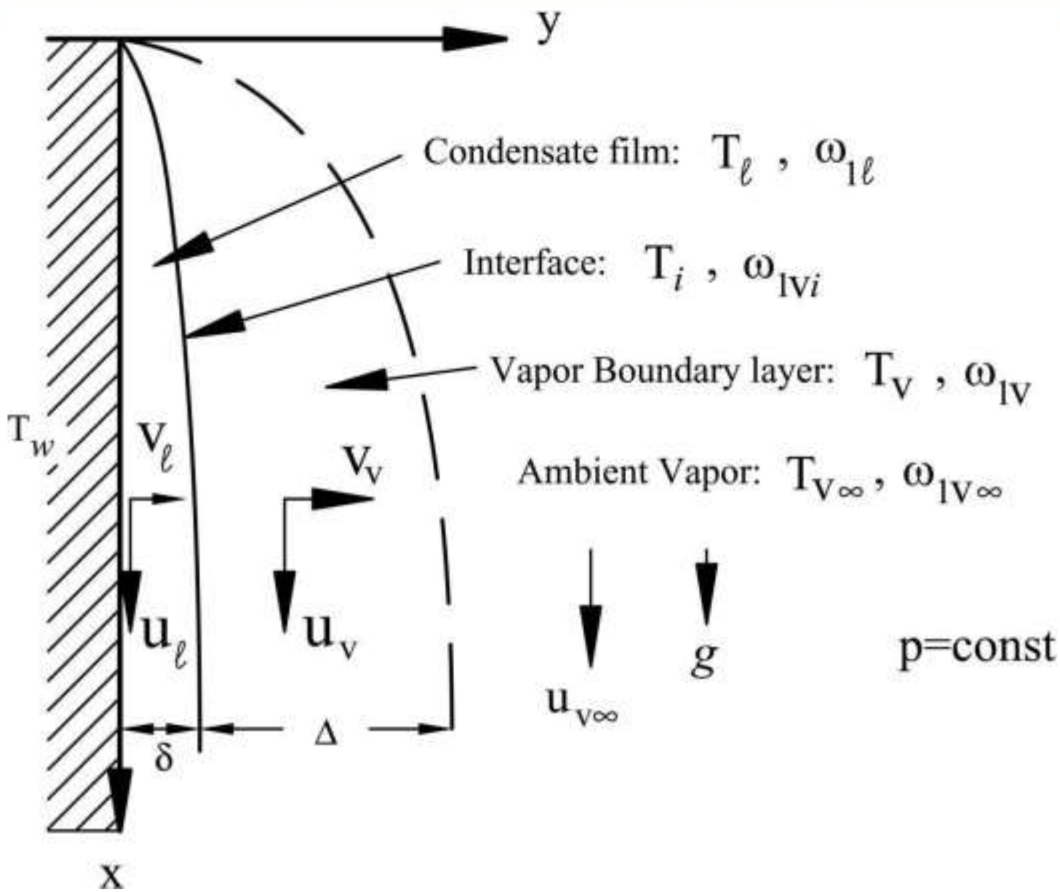
$$h_m = 0.0076\left[\frac{\rho(\rho - \rho_v) g}{\mu^2}\right]^{1/3} Re^{-0.4}$$

Filmwise Condensation

Regimes of Filmwise Condensation



Flow regimes of film condensate on a vertical wall.



Physical model and coordinate system for condensation of a binary vapor mixture.

90

- The governing equations for the laminar film condensation of a binary vapor mixture can begiven by taking the above assumptions into account and using boundary layer analysis, i.e.,

For the condensate film:

$$\frac{\partial u_l}{\partial x} + \frac{\partial v_l}{\partial y} = 0 \qquad (8.58)$$

$$u_l \frac{\partial u_l}{\partial x} + v_l \frac{\partial u_l}{\partial y} = \nu_l \frac{\partial^2 u_l}{\partial y^2} + g - \frac{1}{\rho_l}\frac{dp}{dx} \qquad (8.59)$$

$$u_l \frac{\partial T_l}{\partial x} + v_l \frac{\partial T_l}{\partial y} = \alpha_l \frac{\partial^2 T_l}{\partial y^2} \qquad (8.60)$$

91

For the vapor boundary layer:

$$\frac{\partial u_v}{\partial x}+\frac{\partial v_v}{\partial y}=0 \tag{8.61}$$

$$u_v\frac{\partial u_v}{\partial x}+v_v\frac{\partial u_v}{\partial y}=\nu_v\frac{\partial^2 u_v}{\partial y^2}+g\left(1-\frac{\rho_{v\infty}}{\rho_v}\right) \tag{8.62}$$

$$u_v\frac{\partial T_v}{\partial x}+v_v\frac{\partial T_v}{\partial y}=\alpha_v\frac{\partial^2 T_v}{\partial y^2}+Dc_{p12}\frac{\partial \omega_{1v}}{\partial y}\frac{\partial T_v}{\partial y} \tag{8.63}$$

$$u_v\frac{\partial \omega_{1v}}{\partial x}+v_v\frac{\partial \omega_{1v}}{\partial y}=D\frac{\partial^2 \omega_{1v}}{\partial y^2} \tag{8.64}$$

Isobaric specific heat difference of the binary vapor

$$c_{p12}=\frac{c_{p1v}-c_{p2v}}{c_{p1v}\omega_{1v}+c_{p2v}\omega_{2v}}=\frac{c_{p1v}-c_{p2v}}{c_{pv}} \tag{8.65}$$

- Definitionsof the terms in eqs. (8.58-8.64)

$$\omega_{1v} = \frac{\rho_{1v}}{\rho_v} \tag{8.66}$$

$$\omega_{2v} = \frac{\rho_{2v}}{\rho_v} \tag{8.67}$$

- where

$$\rho_v = \rho_{1v} + \rho_{2v} \tag{8.68}$$

$$\omega_{1v} + \omega_{2v} = 1 \tag{8.69}$$

- Partial pressures of the system are determined by

$$\frac{p_1}{p} = \left(1 + \frac{M_1 \omega_{2v}}{M_2 \omega_{1v}}\right)^{-1} \tag{8.70}$$

$$\frac{p_2}{p} = \left(1 + \frac{M_2 \omega_{1v}}{M_1 \omega_{2v}}\right)^{-1} \tag{8.71}$$

- Boundary conditions at the surface of the cold wall

$$u_l = 0, \quad y = 0 \tag{8.72}$$

$$v_l = 0, \quad y = 0 \tag{8.73}$$

$$T_l = T_w, \quad y = 0 \tag{8.74}$$

- Boundary conditions at locations far from the cold wall

$$u_v = u_{v\infty}, \quad y \to \infty \tag{8.75}$$

$$T_v = T_{v\infty}, \quad y \to \infty \tag{8.76}$$

$$y \to \infty \tag{8.77}$$

$$\omega_{1v} = \omega_{1v\infty},$$

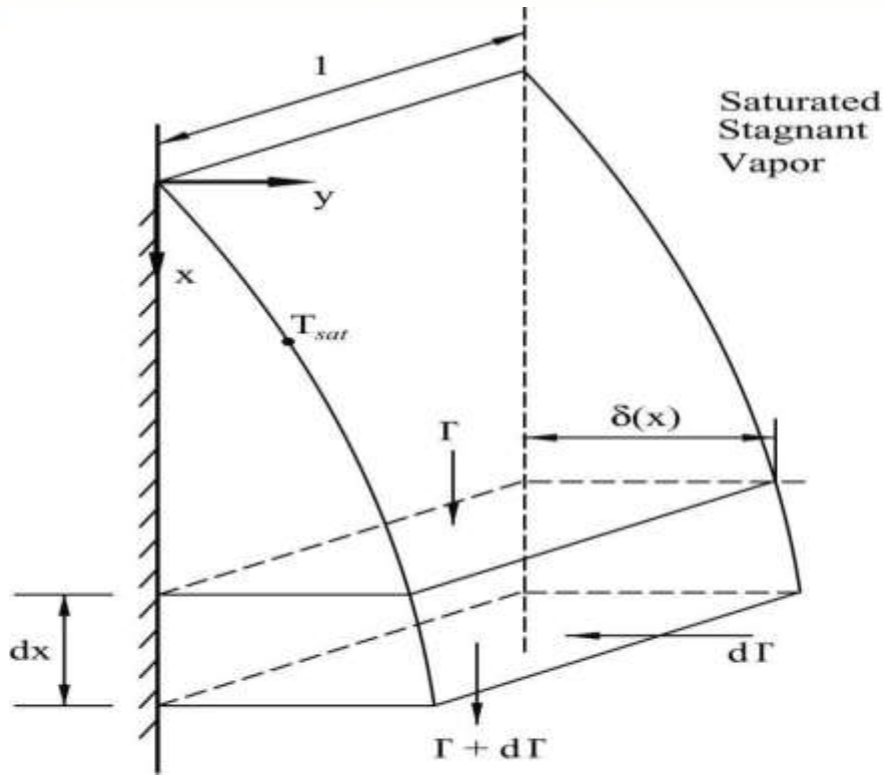


Figure 8.13 Overview of the control volume under consideration in the Nusselt analysis.

- Pressure in the liquid film

$$\frac{dp}{dx} = \rho_v g \tag{8.91}$$

- Substituting eq. (8.91) into eq. (8.59)

$$\rho_l\left(u\frac{\partial u}{\partial x} + v\frac{\partial u}{\partial y}\right) = \mu_l\frac{\partial^2 u}{\partial y^2} + g(\rho_l - \rho_v) \tag{8.92}$$

- Neglecting the inertia term, eq. (8.92) becomes

$$\frac{\partial^2 u}{\partial y^2} = \frac{g}{\mu_l}(\rho_v - \rho_l) \tag{8.93}$$

- Integrating twice and applying boundary conditions

$$u(x, y) = \frac{(\rho_l - \rho_v)g}{\mu_l}\left(y\delta - \frac{y^2}{2}\right) \tag{8.94}$$

- Mass flow rate per unit width of surface

$$\Gamma = \rho_l \int_0 u\,dy = \frac{\rho_l(\rho_l - \rho_v)g\delta^3}{3\mu_l} \tag{8.95}$$

Heat flux across the film thickness

$$q' = \frac{k_l (T_{sat} - T_w)}{\delta} \tag{8.96}$$

Heat transfer rate per unit width for the control volume

$$dq' = \frac{k_l \Delta T}{\delta} dx \tag{8.97}$$

where $\Delta T = T_{sat} - T_w$.

Latent heat effects of condensation dominate the process

$$dq' = h_{lv} d\Gamma \tag{8.98}$$

$d\Gamma$ is found by differentiating the expression for mass flow rate per unit surface eq. (8.95)

$$d\Gamma = \frac{\rho_l (\rho_l - \rho_v) g \delta^2}{\mu_l} d\delta \tag{8.99}$$

Substituting into eq. (8.98) (8.100)

$$\delta^3 \frac{d\delta}{dx} = \frac{k_l \mu_l \Delta T}{\rho_l (\rho_l - \rho_v) g h_{lv}}$$

Substituting the velocity profile eq. (8.94) and using a linear temperature profile

$$\frac{T_{sat}-T}{T_{sat}-T_w}=1-\frac{y}{\delta} \qquad (8.111)$$

To evaluate eq. (8.110) an energy balance

$$\frac{k_l \Delta T_l}{\delta}=h'_{lv}\frac{d\Gamma}{dx} \qquad (8.112)$$

where

$$h'_{lv}=h_{lv}\left\{1+\frac{3}{8}\left[\frac{c_{pl}(T_{sat}-T_w)}{h_{lv}}\right]\right\} \qquad (8.113)$$

Rohsnow included convection and liquid subcooling effects to develop

$$h'_{lv}=h_{lv}\left\{1+0.68\left[\frac{c_{pl}(T_{sat}-T_w)}{h_{lv}}\right]\right\} \qquad (8.114)$$

Assuming the film condensation is laminar (as will be verified later), the heat transfer coefficient can be obtained from eq. (8.105), i.e.,

$$\bar{h} = 0.943\left[\frac{\rho_l(\rho_l - \rho_v)gk_l^3h_{lv}}{\mu_l\Delta TL}\right]^{1/4}$$

$$= 0.943 \times \left[\frac{965.3\times(965.3-0.5974)\times 9.8\times 0.675^3\times 2308.4\times 10^3}{0.315\times 10^{-3}(100-80)\times 1}\right]^{1/4}$$

$$= 5340.2\ \text{W/m}^2\text{-K}$$

The heat transfer rate is then

$$q = \bar{h}Lb(T_{sat} - T_w) = 5340.2\times 1\times 1.5\times(100-80) = 1.602\times 10^5\ \text{W}$$

The condensation rate is

$$m = \frac{q}{h'_{lv}} = \frac{1.602\times 10^5}{2308.4\times 10^3} = 0.0694\ \text{kg/s}$$

The assumption of laminar film condensation is now checked by obtaining the Reynolds number defined in eq. (8.106), i.e.,

$$Re_\delta = 588$$

which is greater than 30 and below 1800. This means that the assumption of laminar film condensation is invalid and it is necessary to consider the effect of waves on the film condensation.

It should be kept in mind the above Reynolds number of 588 is obtained by assuming laminar film condensation. For film condensation with wavy effects, the Reynolds number should be obtained from eq. (8.121), i.e.,

$$\mathrm{Re}_\delta = \left[4.81 + \frac{3.7 L k_1 (T_{sat} - T_w)}{\mu_1 h'_{1v}} \left(\frac{g}{\nu_1^2}\right)^{1/3}\right]^{0.82}$$

$$= \left[4.81 + \frac{3.7 \times 1 \times 0.675 \times (100 - 80)}{0.315 \times 10^{-3} \times 2308.4 \times 10^3} \left(\frac{9.8}{(0.326 \times 10^{-6})^2}\right)^{1/3}\right]^{0.82}$$

$$= 730.9$$

Black-body radiation

- Is the thermal electromagnetic radiation within or surrounding a body in thermodynamic equilibrium with its environment, or emitted by a black body (an opaque and non-reflective body). It has a specific spectrum and intensity that depends only on the body's temperature, which is assumed for the sake of calculations and theory to be uniform and constant.

- The thermal radiation spontaneously emitted by many ordinary objects can be approximated as black-body radiation. A perfectly insulated enclosure that is in thermal equilibrium internally contains black-body radiation and will emit it through a hole made in its wall, provided the hole is small enough to have negligible effect upon the equilibrium.

- A black-body at room temperature appears black, as most of the energy it radiates is infra-red and cannot be perceived by the human eye.

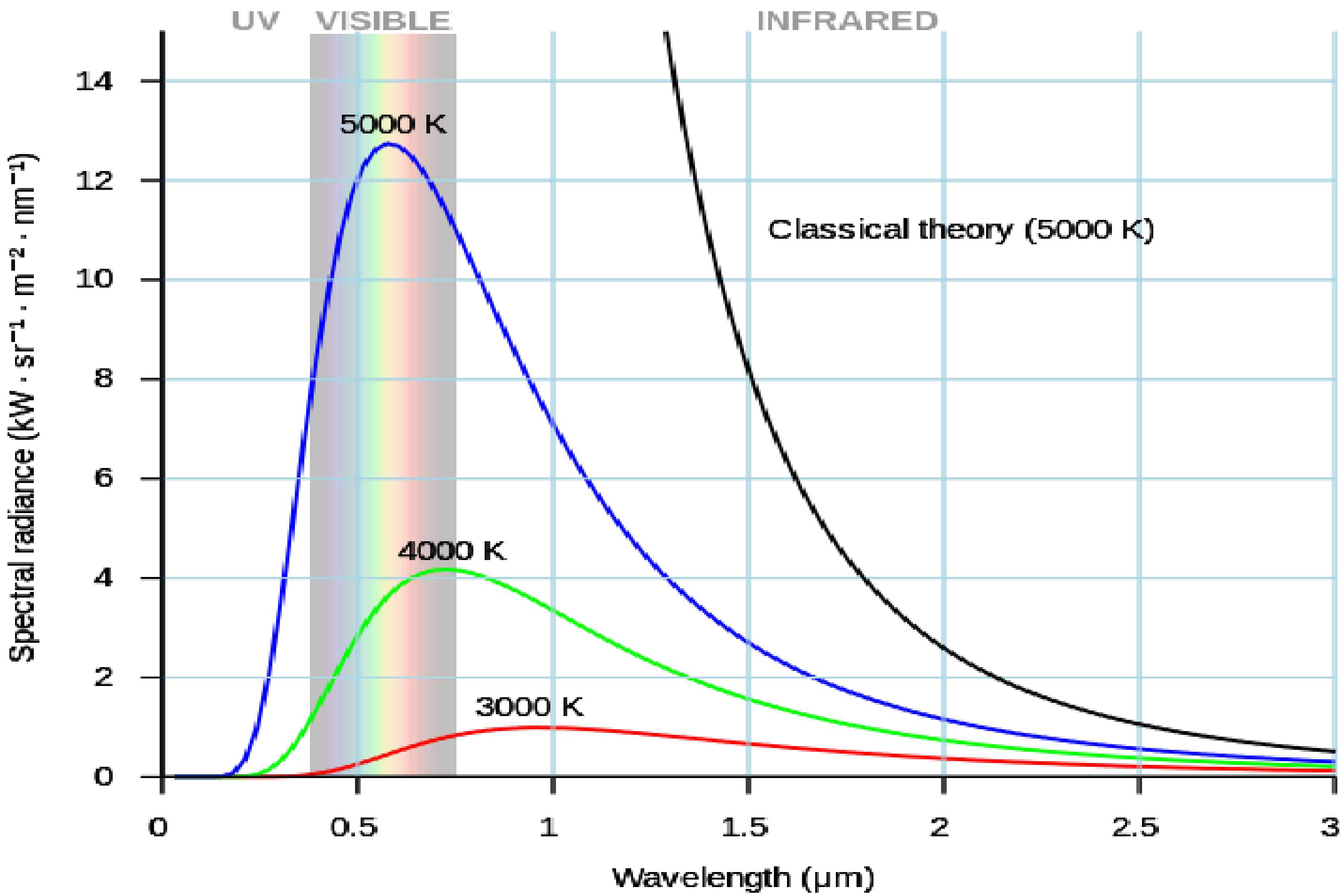
- Because the human eye cannot perceive light waves at lower frequencies, a black body, viewed in the dark at the lowest just faintly visible temperature, subjectively appears grey, even though its objective physical spectrum peaks in the infrared range.

- When it becomes a little hotter, it appears dull red. As it temperature increases further it becomes yellow, white, and ultimately blue-white.

- Although planets and stars are neither in thermal equilibrium with their surroundings nor perfect black bodies, black-body radiation is used as a first approximation for the energy they emit.

- Black holes are near-perfect black bodies, in the sense that they absorb all the radiation that falls on them. It has been proposed that they emit black-body radiation (called Hawking radiation), with a temperature that depends on the mass of the black hole.

- The term black body was introduced by GustavKirchhoff in 1860. Black-body radiation is also called thermal radiation, cavity radiation, complete radiation or temperature radiation.


UV
VISIBLE
INFRARED
5000 K
4000 K
3000 K
Classical theory (5000 K)
Spectral radiance (kW · sr⁻¹ · m⁻² · nm⁻¹)
Wavelength (μm)
0
2
4
6
8
10
12
14
0
0.5
1
1.5
2
2.5
3

Radiation Shape Factor

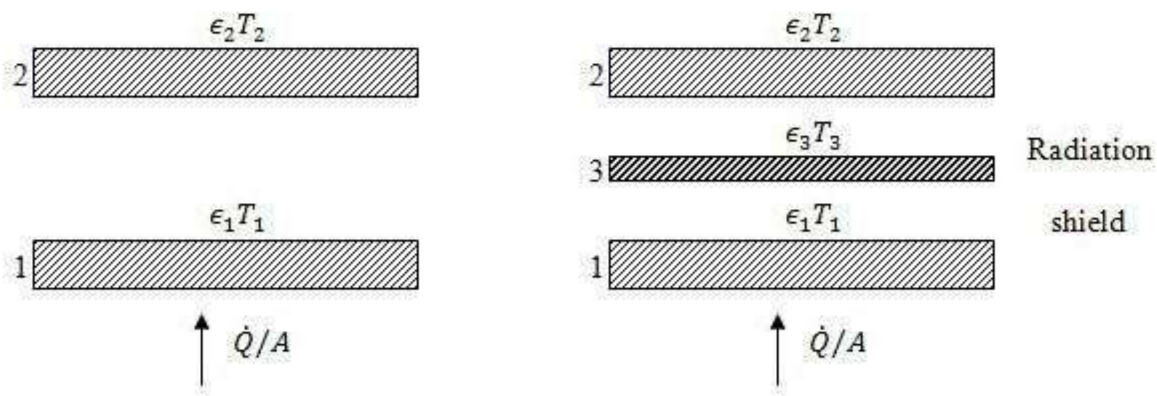
- The easiest method to calculate radiative heat transfer between two

- In the above image G represent irradiation which is the total radiation that come in contact with a surface per unit time and unit area. While J represents the radiosity which is the total amount of radiation that is reflected off a surface per unit time and unit area.

The equation below can be used to determine the value for J.

Radiation shield

Till now we have discussed about the radio active heat transfer from one surface to another without any interfering surface in between. Here we will discuss about an interfering shield in between, which is termed as radiation shield. A radiation shield is a barrier wall of low emissivity placed between two surfaces which reduce the radiation between the bodies. In fact, the radiation shield will put additional resistance to the radio active heat transfer between the surfaces



Considering fig and the system is at steady state, and the surfaces are flat (F_{ij} because each plate is in full view of the other). Moreover, the surface are large enough and may be considered and the equivalent blackbody radiation energy$_b$ may be written as E = σT .

$$\left.\frac{\dot{Q}}{A}\right|_{net} = \left.\frac{\dot{Q}_{13}}{A_1}\right|_{net} = \left.\frac{\dot{Q}_{32}}{A_3}\right|_{net} = \frac{\sigma\left(T_1^4 - T_3^4\right)}{\frac{1}{\epsilon_1}+\frac{1}{\epsilon_3}-1} = \frac{\sigma\left(T_3^4 - T_2^4\right)}{\frac{1}{\epsilon_3}+\frac{1}{\epsilon_2}-1}$$

In order to have a feel of the role of the radiation shield, consider that the emissivities of all the three surfaces are equal. Then it can be seen that the heat flux is just one half of that which would be experienced if there were no shield present. In similar line we can deduce that when n- shields are arranged between the two surfaces then,

$$\left(\frac{\dot{Q}}{A}\right)_{net_{with\ shield}} = \frac{1}{n+1}\left(\frac{\dot{Q}}{A}\right)_{without\ shield}$$

Electrical network for radiation through absorbing and transmitting medium

the previous discussions were based on the consideration that the heat transfer surfaces were separated by a completely transparent medium. However, in real situations the heat transfer medium absorbs as well as transmits. The examples of such medium are glass, plastic film, and various gases.Consider two non- transmitting surfaces (same as in fig. 7.8) are separated by a transmitting and absorbing medium. The medium may be considered as a radiation shield which see themselves and others. If we distinguish the transparent medium by *m* and if the medium is non-reflective (say gas) then using Kirchhoff‘s law,

UNIT-V
HEAT EXCHANGERS

CLOs	Course Learning Outcome
CLO 17	Understand the concepts of black and gray body radiation heat transfer.
CLO 18	Understand the concept of shape factor and evolve a mechanism for conducive radiation shields
CLO 19	Understand the various classifications of heat exchangers based on arrangement and correlate the effects of fouling
CLO 20	Understand the LMTD and NTU methods and apply the same for solving real time problems in heat exchangers

Classification of Heat Exchangers

INTRODUCTION:

1. A heat exchanger is a device that is used to transfer thermal energy (enthalpy) between two or more fluids, between a solid surface and a fluid, or between solid particulates and a fluid, at different temperatures and in thermal contact. In heat exchangers, there are usually no external heat and work interactions.
2. Typical applications involve heating or cooling of a fluid stream of concern and evaporation or condensation of single- or multi component fluid streams.

3. Such exchangers are referred to as direct transfer type, or simply recuperates In contrast, exchangers in which there is intermittent heat exchange between the hot and cold fluids—via thermal energy storage and release through the exchanger surface or matrix— are referred to as indirect transfer type, or simply regenerators.

Classification according to transfer process

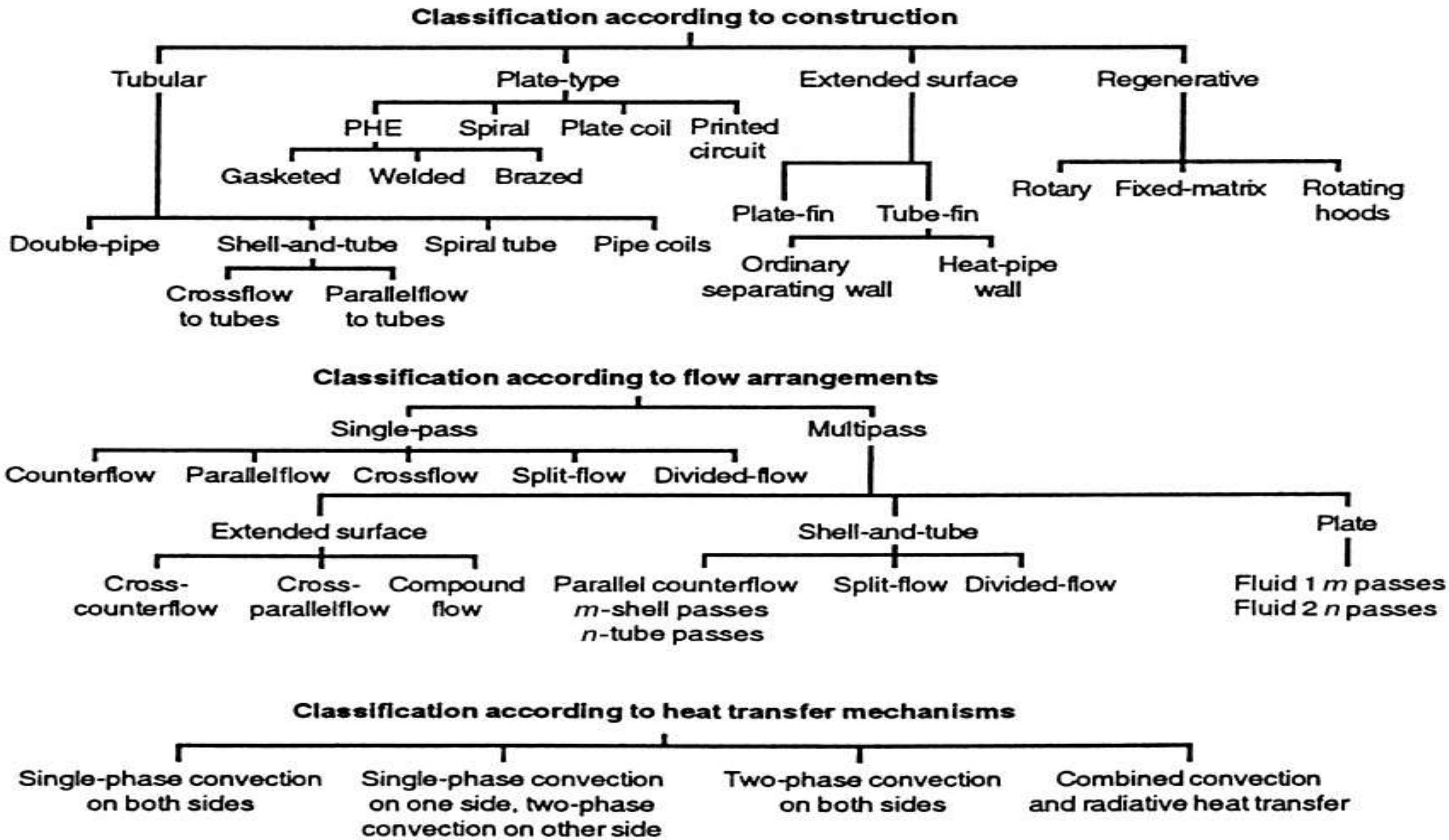
- Indirect contact type
  - Direct transfer type
    - Single-phase
    - Multiphase
  - Storage type
  - Fluidized bed
- Direct contact type
  - Immiscible fluids
  - Gas–liquid
  - Liquid–vapor

Classification according to number of fluids

- Two-fluid
- Three-fluid
- N-fluid ($N > 3$)

Classification according to surface compactness

- Gas-to-fluid
  - Compact ($\beta \geq 700\ m^2/m^3$)
  - Noncompact ($\beta < 700\ m^2/m^3$)
- Liquid-to-liquid and phase-change
  - Compact ($\beta \geq 400\ m^2/m^3$)
  - Noncompact ($\beta < 400\ m^2/m^3$)


Classification according to construction
Tubular
Plate-type
Extended surface
Regenerative
PHE
Spiral
Plate coil
Printed circuit
Gasketed
Welded
Brazed
Plate-fin
Tube-fin
Rotary
Fixed-matrix
Rotating hoods
Double-pipe
Shell-and-tube
Spiral tube
Pipe coils
Ordinary separating wall
Heat-pipe wall
Crossflow to tubes
Parallelflow to tubes
Classification according to flow arrangements
Single-pass
Multipass
Counterflow
Parallelflow
Crossflow
Split-flow
Divided-flow
Extended surface
Shell-and-tube
Plate
Cross-counterflow
Cross-parallelflow
Compound flow
Parallel counterflow m-shell passes n-tube passes
Split-flow
Divided-flow
Fluid 1 m passes
Fluid 2 n passes
Classification according to heat transfer mechanisms
Single-phase convection on both sides
Single-phase convection on one side, two-phase convection on other side
Two-phase convection on both sides
Combined convection and radiative heat transfer

CLASSIFICATIONS ACCORDING TO TRANSFER PROCESSES:

Heat exchangers are classified according to transfer processes into indirect- and direct contact types.

Indirect-Contact Heat Exchangers:

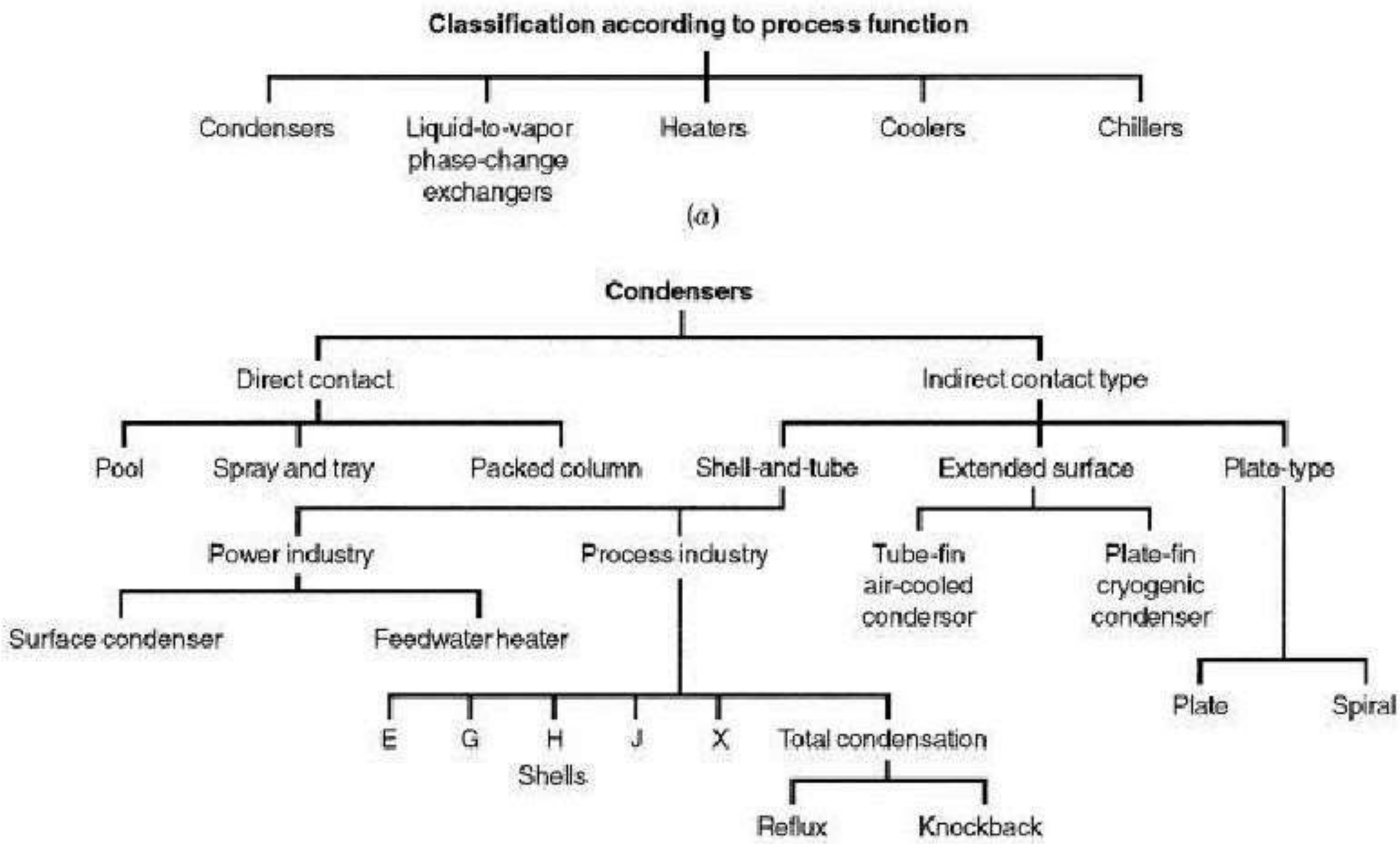
- In an indirect-contact heat exchanger, the fluid streams remain separate and the heat transfers continuously through an impervious dividing wall or into and out of a wall in a transient manner.
- Thus, ideally, there is no direct contact between thermally interacting fluids. This type of heat exchanger, also referred to as a surface heat exchanger, can be further classified into direct-transfer type, storage type, and fluidized-bed exchangers.

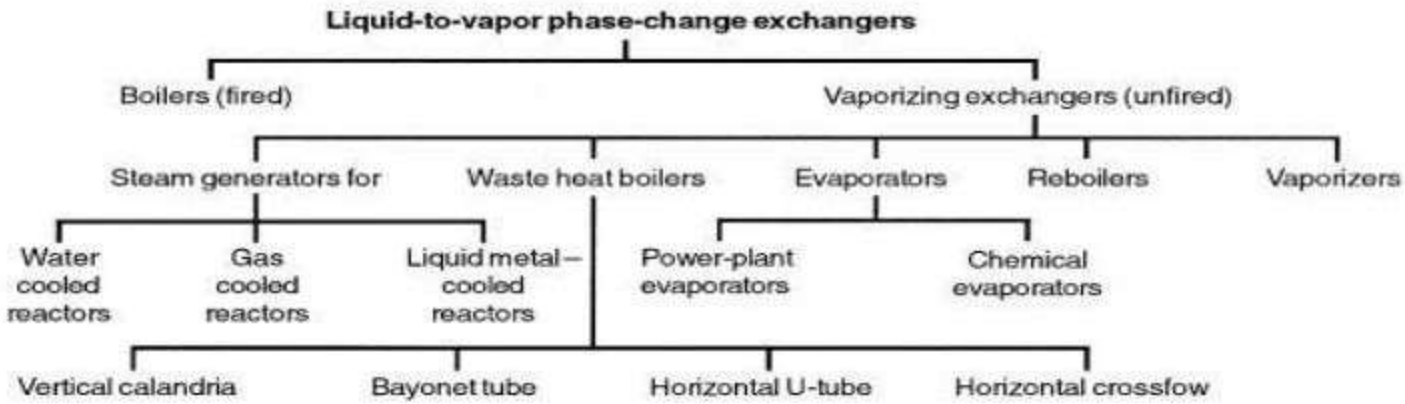
Direct-Transfer Type Exchangers:

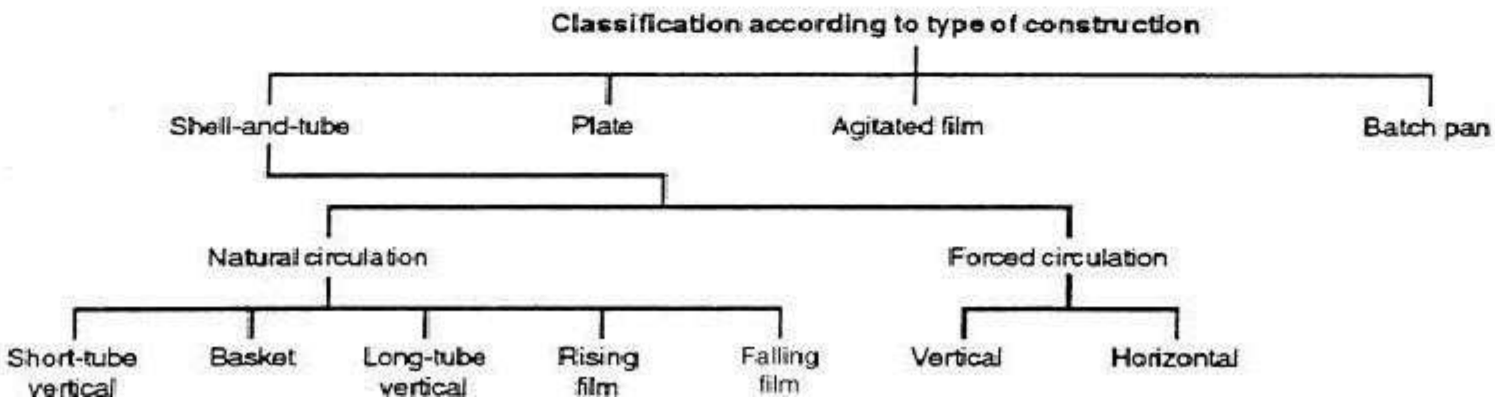
➢In this type, heat transfers continuously from the hot fluid to the cold fluid through a dividing wall. Although a simultaneous flow of two (or more) fluids is required in the exchanger, there is no direct mixing of the two (or more) fluids because each fluid flows in separate fluid passages.

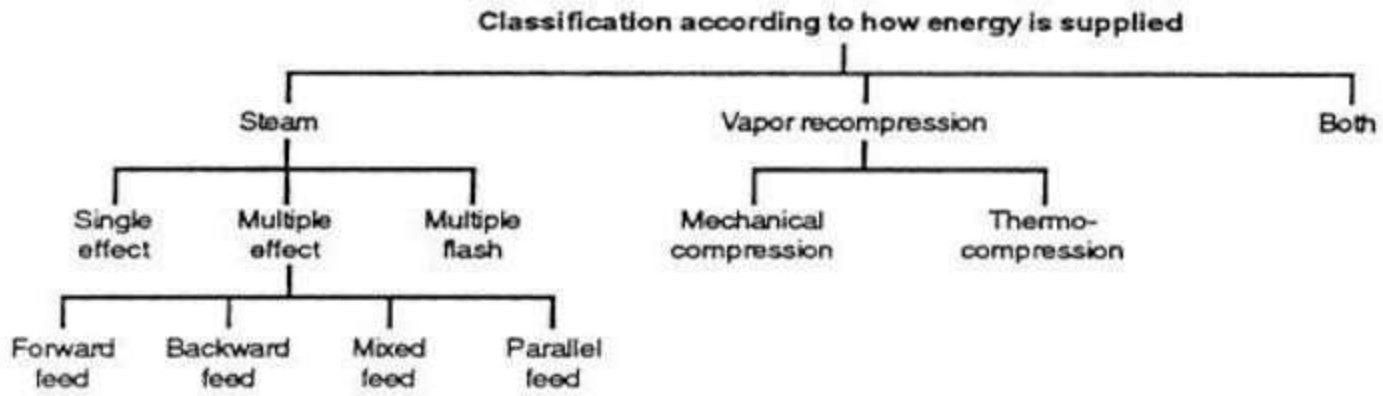
➢In general, there are no moving parts in most such heat exchangers. This type of exchanger is designated as a recuperative heat exchanger or simply as a recuperate.

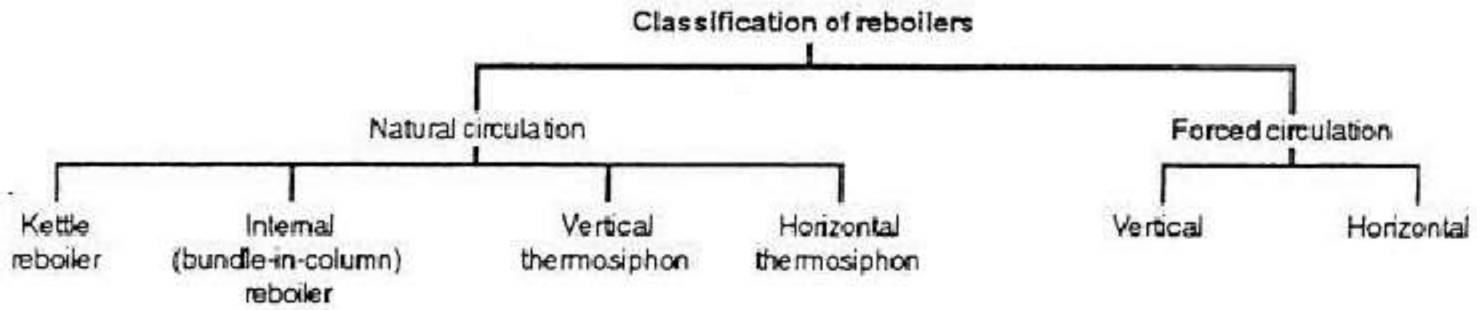
{ Some examples of direct transfer type heat exchangers are tubular, plate-type, and extended surface exchangers.


Classification according to process function
Condensers
Liquid-to-vapor phase-change exchangers
Heaters
Coolers
Chillers
(a)
Condensers
Direct contact
Indirect contact type
Pool
Spray and tray
Packed column
Shell-and-tube
Extended surface
Plate-type
Power industry
Process industry
Tube-fin air-cooled condersor
Plate-fin cryogenic condenser
Surface condenser
Feedwater heater
Plate
Spiral
E
G
H
J
X
Total condensation
Shells
Reflux
Knockback


Liquid-to-vapor phase-change exchangers
Boilers (fired)
Vaporizing exchangers (unfired)
Steam generators for
Waste heat boilers
Evaporators
Reboilers
Vaporizers
Water cooled reactors
Gas cooled reactors
Liquid metal – cooled reactors
Power-plant evaporators
Chemical evaporators
Vertical calandria
Bayonet tube
Horizontal U-tube
Horizontal crossfow


Classification according to type of construction
Shell-and-tube
Plate
Agitated film
Batch pan
Natural circulation
Forced circulation
Short-tube vertical
Basket
Long-tube vertical
Rising film
Falling film
Vertical
Horizontal


Classification according to how energy is supplied
Steam
Vapor recompression
Both
Single effect
Multiple effect
Multiple flash
Mechanical compression
Thermo- compression
Forward feed
Backward feed
Mixed feed
Parallel feed


Classification of reboilers
Natural circulation
Forced circulation
Kettle reboiler
Internal (bundle-in-column) reboiler
Vertical thermosiphon
Horizontal thermosiphon
Vertical
Horizontal

Storage Type Exchangers:

1. In a storage type exchanger, both fluids flow alternatively through the same flow passages, and hence heat transfer is intermittent. The heat transfer surface (or flow passages) is generally cellular in structure and is referred to as a matrix, or it is a permeable (porous) solid material, referred to as a packed bed.
2. The actual time that hot gas takes to flow through a cold regenerator matrix is called the hot period or hot blow, and the time that cold gas flows through the hot regenerator matrix is called the cold period or cold blow. For successful operation, it is not necessary to have hot- and cold-gas flow periods of equal duration.
3. There is some unavoidable carryover of a small fraction of the fluid trapped in the passage to the other fluid stream just after switching of the fluids; this is referred to as carryover leakage.

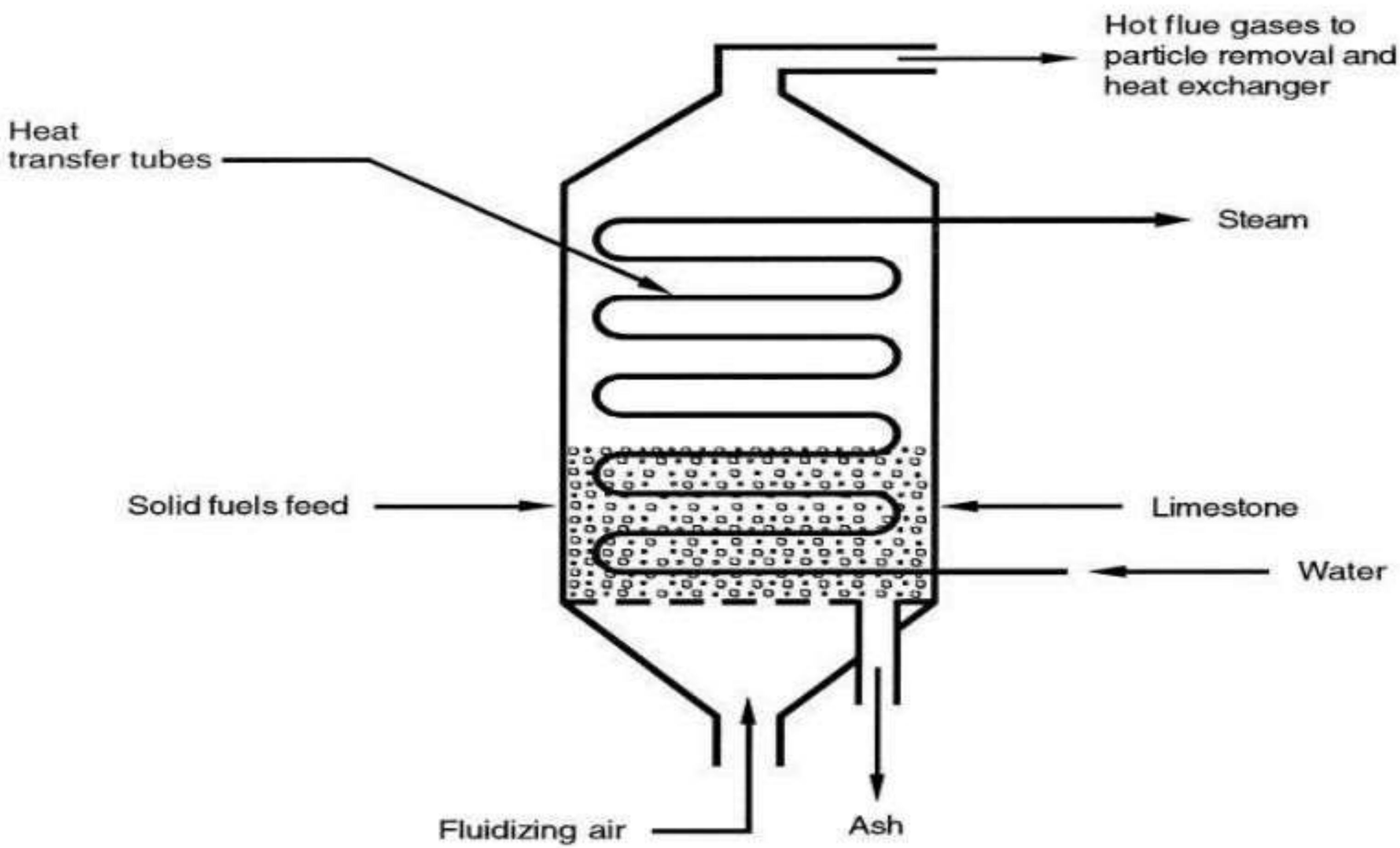
Fluidized-Bed Heat Exchangers:

In a fluidized-bed heat exchanger, one side of a two-fluid exchanger is immersed in a bed of finely divided solid material, such as a tube bundle immersed in a bed of sand or coal particles.

If the upward fluid velocity on the bed side is low, the solid particles will remain fixed in position in the bed and the fluid will flow through the interstices of the bed. If the upward fluid velocity is high, the solid particles will be carried away with the fluid. At a ‗ proper' value of the fluid velocity, the upward drag force is slightly higher than the weight of the bed particles.

This results in a uniform temperature for the total bed (gas and particles) with an apparent thermal conductivity of the solid particles as infinity.

- As a result, the solid particles will float with an increase in bedvo lume, and the bed behaves as a liquid. This characteristic of the bed is referred to as a fluidized condition. Under this condition, the fluid pressure drop through the bed remains almost constant, independent of the flow rate, and a strong mixing of the solid particles occurs.


Hot flue gases to
particle removal and
heat exchanger
Heat
transfer tubes
Steam
Solid fuels feed
Limestone
Water
Fluidizing air
Ash

Direct-Contact Heat Exchangers:

- In a direct-contact exchanger, two fluid streams come into direct contact, exchange heat, and are then separated. Common applications of a direct-contact exchanger involve mass transfer in addition to heat transfer, such as in evaporative cooling and rectification; applications involving only sensible heat transfer are rare.
  - The enthalpy of phase change in such an exchanger generally represents

a significant portion of the total energy transfer. The phase change generally enhances the heat transfer rate.

- Compared to indirect contact recuperates and regenerators, in direct-

contact heat exchangers, (1) very high heat transfer rates are achievable, (2) the exchanger construction is relatively inexpensive, and (3) the fouling problem is generally nonexistent, due to the absence of a heat transfer surface (wall) between the two fluids. However, the applications are limited to those

Immiscible Fluid Exchangers

- In this type, two immiscible fluid streams are brought into direct contact. These fluids may be single-phase fluids, or they may involve condensation or vaporization. Condensation of organic vapors and oil vapors with water or air are typical examples.
- Gas–Liquid Exchangers. In this type, one fluid is a gas (more commonly, air) and the other a low-pressure liquid (more commonly, water) and are readily separable after the energy exchange. In either cooling of liquid (water) or humidification of gas (air) applications, liquid partially evaporates and the vapor is carried away with the gas.
- In these exchangers, more than 90% of the energy transfer is by virtue of mass transfer (due to the evaporation of the liquid), and convective heat transfer is a minor mechanism. A ‗ wet‘ (water) cooling tower with forced- or natural-draft airflow is the most common application. Other applications are the air-conditioning spray chamber, spray drier, spray tower, and spray pond.

Liquid–Vapour Exchangers

1. In this type, typically steam is partially or fully condensed using cooling water, or water is heated with waste steam through direct contact in the exchanger.
2. Non condensable and residual steam and hot water are the outlet streams. Common examples are de super heaters and open feed water heaters in power plants.

CLASSIFICATIONS

CLASSIFICATIONS ACCORDING TO NUMBER OF FLUIDS

1.Most processes of heating, cooling, heat recovery, and heat rejection involve transfer of heat between two fluids. Hence, two-fluid heat exchangers are the most common.

2.Three fluid heat exchangers are widely used in cryogenics and some chemical processes (e.g., air separation systems, a helium–air separation unit, purification and liquefaction of hydrogen, ammonia gas synthesis). Heat exchangers with as many as 12 fluid streams have been used in some chemical process applications.

CLASSIFICATION ACCORDING TO SURFACE COMPACTNESS:

Compared to shell-and-tube exchangers, compact heat exchangers are characterized by a large heat transfer surface area per unit volume of the exchanger, resulting in reduced space, weight, together with low fluid inventory.

A gas-to-fluid exchanger is referred to as a compact heat exchanger if it incorporates a heat transfer surface having a surface area density greater than about 700 m2 /m3 (213 ft2 /ft3) { or a hydraulic diameter Dh 6 mm (1 4 in.) for operating in a gas stream and 400 m2 /m3 (122 ft2 /ft3) or higher for operating in a liquid or phase-change stream. A laminar flow heat exchanger (also referred to as a meso heat exchanger) has a surface area density greater than about 3000 m2 /m3 (914 ft2 /ft3) or 100 mm Dh 1 mm.

- The term micro heat exchanger is used if the surface area density is greater than about 15,000 m^2/m^3 (4570 ft^2/ft^3) or 1 mm Dh 100 mm. A liquid/two-phase fluid heat exchanger is referred to as a compact heat exchanger if the surface area density on any one fluid side is greater than about 400 m^2/m^3

Gas-to-Fluid Exchangers:

1. The heat transfer coefficient h for gases is generally one or two orders of magnitude lower than that for water, oil, and other liquids. Now, to minimize the size and weight of a gas to liquid heat exchanger, the thermal conductance's (hA products) on both sides of the exchanger should be approximately the same.

2. Hence, the heat transfer surface on the gas side needs to have a much larger area and be more compact than can be realized ractically with the circular tubes commonly used in shell-and-tube exchangers.

3. Thus, for an approximately balanced design (about the same hA values), a compact surface is employed on the gas side of gas-to-gas, gas-to-liquid, and gas-to-phase change heat exchangers.

CLASSIFICATION ACCORDING TO CONSTRUCTION FEATURES

- Heat exchangers are frequently characterized by construction features. Four major construction types are tubular, plate-type, extended surface, and regenerative exchangers. Heat exchangers with other constructions are also available, such as scraped surface exchanger, tank heater, cooler cartridge exchanger, and others (Walker, 1990). Some of these may be classified as tubular exchangers, but they have some unique features compared to conventional tubular exchangers.

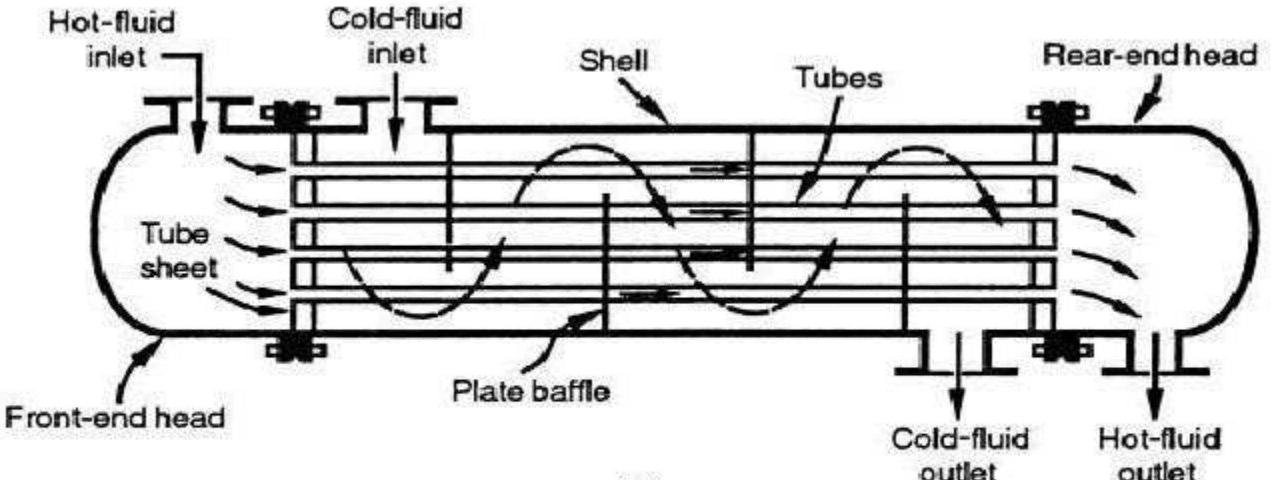
- Although the "-NTU and MTD methods are identical for tubular, plate-type, and extended-surface exchangers, the influence of the following factors must be taken into account in exchanger design: corrections due to leakage and bypass streams in a shell- and-tube exchanger, effects due to a few plates in a plate exchanger, and fin efficiency in an extended-surface exchanger. Similarly, the "-NTU method must be modified to take into account the heat capacity of the matrix in a regenerator.

Tubular Heat Exchangers:

- These exchangers are generally built of circular tubes, although elliptical, rectangular, or round/flat twisted tubes have also been used in some applications. There is considerable flexibility in the design because the core geometry can be varied easily by changing the tube diameter, length, and arrangement.
- Tubular exchangers can be designed for high pressures relative to the environment and high-pressure differences between the fluids. Tubular exchangers are used primarily for liquid-to-liquid and liquid-to-phase change (condensing or evaporating) heat transfer applications.
- They are used for gas-to-liquid and gas-to-gas heat transfer applications primarily when the operating temperature and/ or pressure is very high or fouling is a severe problem on at least one fluid side and no other types of exchangers would work.

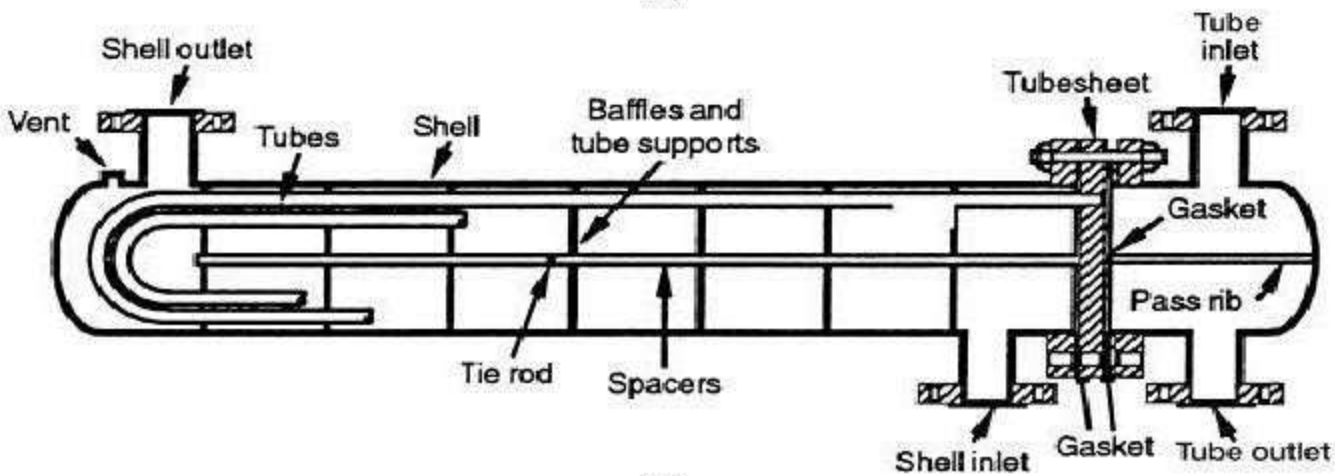
Shell-and-Tube Exchangers

- Is generally built of a bundle of round tubes mounted in a cylindrical shell with the tube axis parallel to that of the shell. One fluid flows inside the tubes, the other flows across and along the tubes. The major components of this exchanger are tubes (or tube bundle), shell, frontend head, rear-end head, baffles, and tube sheets, and are described briefly later in this subsection.
- A variety of different internal constructions are used in shell-and- tube exchangers, depending on thedesired heat transfer and pressure drop performance and the methods employed to reduce thermal stresses, to prevent leakages,to provide for ease of cleaning, to contain operating pressures and temperatures, to control corrosion, to accommodate highly asymmetric flows, and so on



Hot-fluid inlet
Cold-fluid inlet
Shell
Tubes
Rear-end head
Tube sheet
Plate baffle
Front-end head
Cold-fluid outlet
Hot-fluid outlet

(a)



Shell outlet
Vent
Tubes
Shell
Baffles and tube supports
Tubesheet
Tube inlet
Gasket
Pass rib
Tie rod
Spacers
Shell inlet
Gasket
Tube outlet

Double-Pipe Heat Exchangers

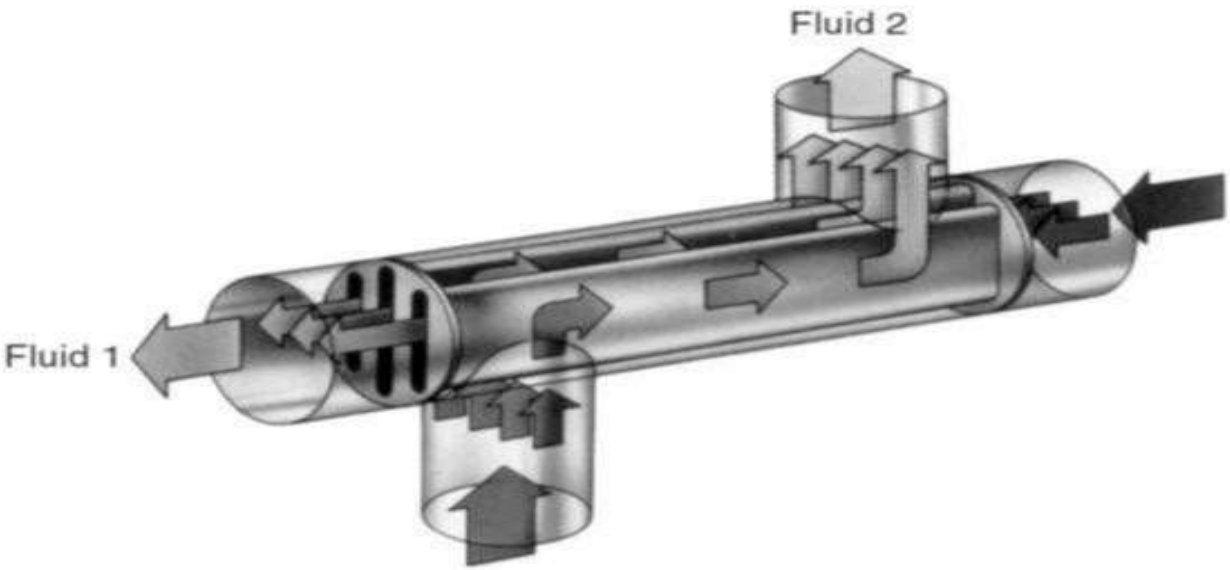
- This exchanger usually consists of two concentric pipes with the inner pipe plain or finned. One fluid flows in the inner pipe and the other fluid flows in the annulus between pipes in a counter flow direction for the ideal highest performance for the given surface area. However, if the application requires an almost constant wall temperature, the fluids may flow in a parallelflow direction.This is perhaps the simplest heat exchanger.
- Flow distribution is no problem, and cleaning is done very easily by disassembly. because containment in the small-diameter pipe or tubing is less costly than containment in a large-diameter shell.

- Double-pipe exchangers are generally used for small-capacity applications where the total heat transfer surface area required is 50 m2 (500 ft2) or less because it is expensive on a cost per unit surface area basis. Stacks of double-pipe or multi tube heat exchangers are also used in some process applications with radial or longitudinal fins.

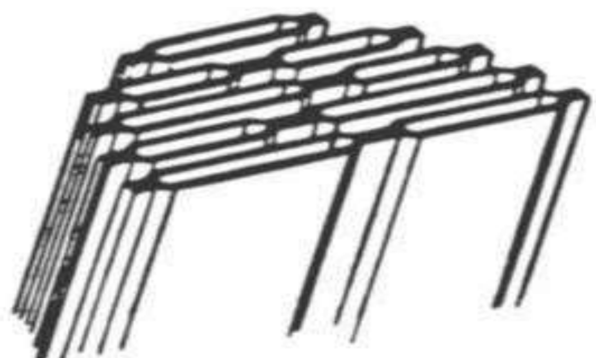
Spiral Tube Heat Exchangers.

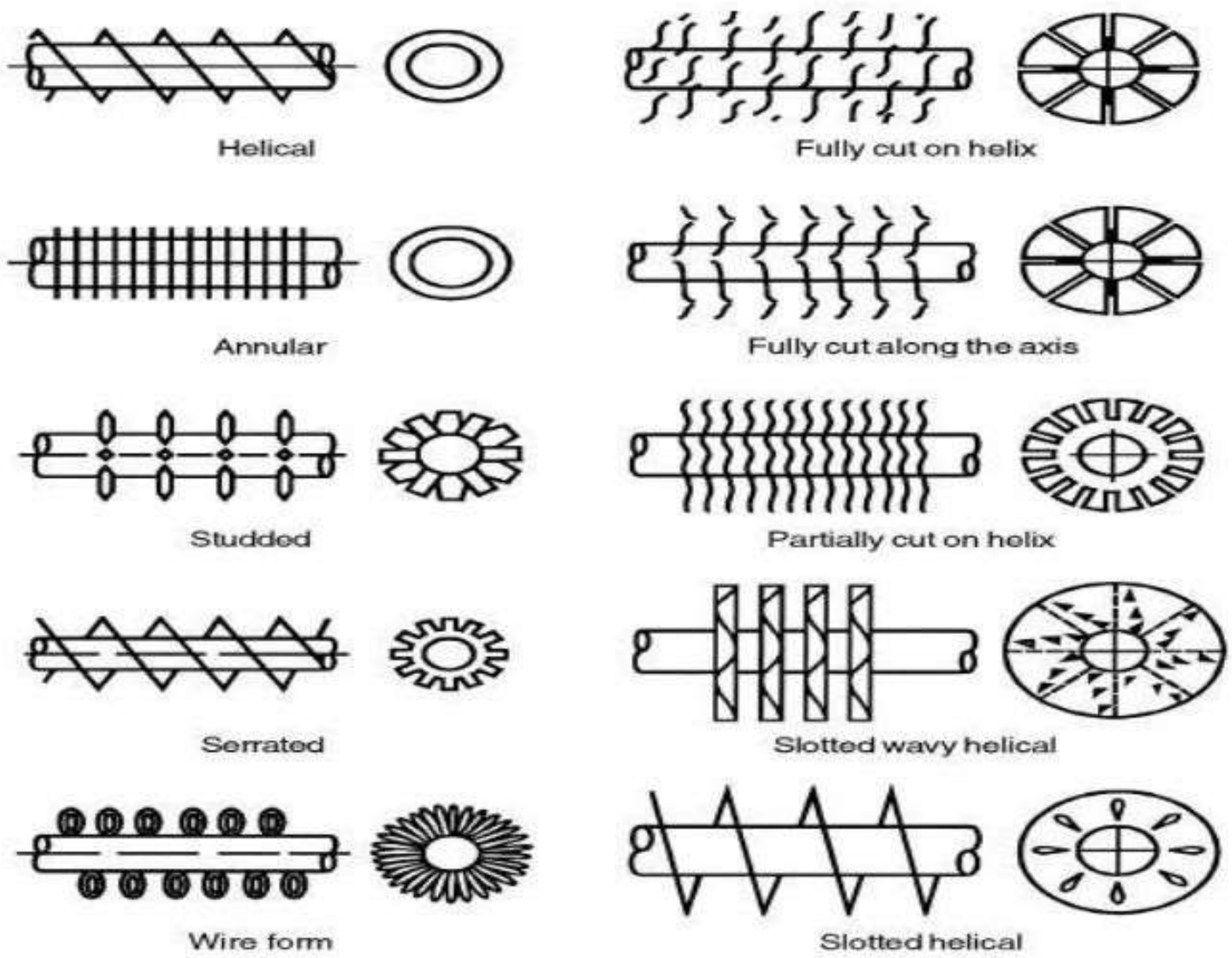
- These consist of one or more spirally wound coils fitted in a shell. Heat transfer rate associated with a spiral tube is higher than that for straight tube. In addition, a considerable amount of surface can be accommodated in a given space by spiraling. Thermal expansion is no problem, but cleaning is almost impossible.

- **Plate-Type Heat Exchangers** Plate-type heat exchangers are usually built of thin plates (all prime surface). The plates are either smooth or have some form of corrugation, and they are either flat or would in in an heat exchanger. Generally, these cannot accommodate very high pressure temperature temperatures, or pressure and temperature differences. Plate heat exchangers (PHEs){ can be classified as gasketed, welded (one or both fluid passages), or brazed, depending on the leak tightness required. Other plate-type exchangers are spiral plate, lamella, and plate coil exchangers.



(a)




Helical
Fully cut on helix
Annular
Fully cut along the axis
Studded
Partially cut on helix
Serrated
Slotted wavy helical
Wire form
Slotted helical

An air-cooled exchanger is a tube-fin exchanger in which hot process fluids (usually liquids or condensing fluids) flow inside the tubes, and atmospheric air is circulated outside by forced or induced draft over the extended surface. If used in a cooling tower with the process fluid as water, it is referred to as a dry cooling tower. Characteristics of this type of exchanger are shallow tube bundles (short airflow length) and large face area, due to the design constraint on the fan power.

Heat Pipe Heat Exchangers

This type of exchanger is similar to a tube-fin exchanger with individually finned tubes or flat (continuous) fins and tubes. However, the tube is a heat pipe, and hot and cold gases flow continuously in separate parts of the exchanger, as shown in Fig. 1.36. Heat is transferred from the hot gas to the evaporation section of the heat pipe by convection; the thermal energy is then carried away by the vapor to the condensation section of the heat pipe, where it transfers heat to the cold gas by convection.

- Regenerators The regenerator is a storage-type heat exchanger, as described earlier. The heat transfer surface or elements are usually referred to as a matrix in the regenerator. To have continuous operation, either the matrix must be moved periodically into and out of the fixed streams of gases, as in a rotary regenerator, or the gas flows must be diverted through valves to and from the fixed matrices as in a fixed matrix regenerator.
- The latter is also sometimes referred to as a periodic-flow regenerator, a swing regenerator, or a reversible heat accumulator. Thus, in a rotary regenerator, the matrix (disk or rotor) rotates continuously with a constant fraction of the core (having disk sector angle h) in the hot-fluid stream and the remaining fraction (having the disk sector angle c) in the cold-fluid stream; the outlet fluid temperatures vary across the flow area and are independent of time. the matrix. In a fixed-matrix regenerator, the hot and cold fluids are ducted through the use of valves to the different matrices

Rotary Regenerators:

- Rotary regenerators are through 1.40. Depending on the applications, rotary regenerators are variously referred to as a heat wheel, thermal wheel, Munter wheel, or Ljungstrom wheel. When the gas flows are laminar, the rotary regenerator is also referred to as a laminar flow wheel. In this exchanger, any of the plain plate-fin surface geometries could be used in the matrix made up of thin metal sheets.

- Interrupted passage surfaces (such as strip fins, louver fins) are not used because a transverse (to the main flow direction) flow leakage is present if the two fluids are at different pressures. This leak mixes the two fluids (contaminates the lower pressure fluid) and reduces the heat exchanger effectiveness. Hence, the matrix generally has continuous interrupted) flow passages. Flat or wavy spacers are used to stack the ‗ fins". The fluid is unmixed at any cross section for these surfaces. The herringbone or skewed passage matrix does not require spacers for stacking the ‗ fins". The design Reynolds number range with these types of surfaces is 100 to 1000.

Fixed-Matrix Regenerator.

- This type is also referred to as a periodic-flow, fixed-bed, valve, or stationary regenerator.
- For continuous operation, this exchanger has at least two identical matrices operated in parallel, but usually three or four, to reduce the temperature variations in outlet-heated cold gas in high-temperature applications.
- In contrast, in a rotary or rotating hood regenerator, a single matrix is sufficient for continuous operation

Fixed-matrix regenerators have two types of heat transfer elements: checker work and pebble beds. Checker work or thin-plate cellular structure are of two major categories:

- non compact regenerators used for high-temperature applications [925 to 16008C (1700 to 29008F)] with corrosive gases, such as a Cowper stove for a blast furnace used in steel industries, and air pre heaters for coke manufacture and glass melting tanks made of refractory material.
- highly compact regenerators used for low to high-temperature applications, such as in cryogenic process for air separation, in refrigeration, and in Stirling, Ericsson, Gifford, and Vuileumier cycle engines.
- The regenerator, a key thermodynamic element in the Stirling engine cycle, has only one matrix, and hence it does not have continuous fluid flows as in other regenerators. For this reason, we do not cover the design theory of a Stirling regenerator.

CLASSIFICATION ACCORDING TO FLOW ARRANGEMENTS

Common flow arrangements of the fluids in a heat exchanger are classified . The choice of a particular flow arrangement is dependent on the required exchanger effectiveness, available pressure drops, minimum and maximum velocities allowed, fluid flow paths, packaging envelope, allowable thermal stresses, temperature levels, piping and plumbing considerations, and other design criteria.

- Let us first discuss the concept of multi passing, followed by some of the basic ideal flow arrangements for a two fluid heat exchanger for single- and multi pass heat exchangers.

Multi passing:

- The concept of multi passing applies separately to the fluid and heat exchanger.
- A fluid is considered to have made one pass if it flows through a section of the heat exchanger through its full length.

Counter flow Exchanger:

- In a counter flow or countercurrent exchanger, the two fluids flow parallel to each other but in opposite directions within the core.{ The temperature variation of the two fluids in such an exchanger may be idealized as one-dimensional.
- As shown later, the counter flow arrangement is thermodynamically superior to any other flow arrangement.
- It is the most efficient flow arrangement, producing the highest temperature change in each fluid compared to any other two-fluid flow arrangements for a given overall thermal conductance (UA), fluid flow rates (actually, fluid heat capacity rates), and fluid inlet temperatures.

Parallel flow Exchanger

- In a parallel flow (also referred to as concurrent or concurrent parallel stream) exchanger, the fluid streams enter together at one end, flow parallel to each other in the same direction, and leave together at the other end with the dashed arrows reversed would then depict parallel flow.
- Fluid temperature variations, idealized as one-dimensional. This arrangement has the lowest exchanger effectiveness among single-pass exchangers for given overall thermal conductance (UA) and fluid flow rates (actually, fluid heat capacity rates) and fluid inlet temperatures; however, some multi pass exchangers may have an even lower effectiveness, as discussed later.

Log-Mean Temperature Difference

For the entire pipe:

$$\frac{T_{m,o}-T_s}{T_{m,i}-T_s}=\frac{\Delta T_o}{\Delta T_i}=\exp\left(-\frac{\bar{h}(PL)}{\dot{m}C_P}\right) \quad \text{or } \dot{m}C_P=-\frac{\bar{h}A_s}{\ln\left(\frac{\Delta T_o}{\Delta T_i}\right)}$$

$$q=\dot{m}C_P(T_{m,o}-T_{m,i})=\dot{m}C_P((T_s-T_{m,i})-(T_s-T_{m,o}))$$

$$=\dot{m}C_P(\Delta T_i-\Delta T_o)=\bar{h}A_s\frac{\Delta T_o-\Delta T_i}{\ln\left(\frac{\Delta T_o}{\Delta T_i}\right)}=\bar{h}A_s\Delta T_{lm}$$

where $\Delta T_{lm}=\dfrac{\Delta T_o-\Delta T_i}{\ln\left(\frac{\Delta T_o}{\Delta T_i}\right)}$ is called the log mean temperature difference.

This relation is valid for the entire pipe.

Cross flow Exchanger:

In this type of exchanger, the two fluids flow in directions normal to each other. For the inlet and outlet sections only. Thermodynamically, the effectiveness for the cross flow exchanger falls in between that for the counter flow and parallel flow arrangements. The largest structural temperature difference exists at the ‗ corner‘ of the entering hot and cold fluids. This is one of the most common flow arrangements used for extended surface heat exchangers, because it greatly simplifies the header design at the entrance and exit of each fluid. If the desired heat exchanger effectiveness is high (such as greater than 80%), the size penalty for the cross flow exchanger may become excessive. In such a case, a counter flow unit is preferred.

1. Seven idealized combinations of flow arrangements for a single- pass cross flow exchanger are shown symbolically in Fig. 1.55. The flow arrangements are: (a) Both fluids unmixed.
2. A cross flow plate-fin exchanger with plain fins on both sides represents the ‗ both fluids unmixed‘ case. (b) One fluid unmixed, the other mixed.
3. A crossflow plate-fin exchanger with fins on one side and a plain gap on the other side would be treated as the unmixed–mixed case.
4. Symbolic presentation of various degrees of mixing in a single- phase cross flow exchanger. (c) Both fluids mixed. This case is practically less important, and represents a limiting case of some multi pass shell-and-tube exchangers as presented later. (d) One fluid unmixed and coupled in identical order, the other partially mixed.

- A tube-fin exchanger with flat fins represents the case of tube fluid partially mixed, the fin fluid unmixed. When the number of tube rows is reduced to one, this exchanger reduces to the case of out-of- tube (fin) fluid unmixed the tube fluid mixed (case b).
- When the number of tube rows approaches infinity (in reality greater than four), the exchanger reduces to the case of both fluids unmixed (case a). (e) One fluid partially unmixed, the other partially mixed. The case of one fluid (fluid 1) partially unmixed (i.e., mixed only between tube rows) and the other (fluid 2) partially mixed is of less practical importance for single-pass cross flow exchangers.
- However, as mentioned later it represents the side-by-side multi pass

cross flow arrangement. When the number of tube rows is reduced to one, this exchanger is reduced to the case of out-of-tube fluid unmixed, the tube fluid mixed.

- When the number of tube rows approaches infinity, the exchanger reduces to the case of out-of-tube fluid mixed, the tube fluid unmixed. (f) One fluid unmixed and coupled in inverted order, the other partially mixed. Here, the term inverted order refers to the fact that a fluid coupled in such order leaves the first row at the point where the other fluid enters (leaves) the first row and enters the other row where the second fluid leaves (enters) that row. This case is also of academic interest for single-pass crossflow exchangers. (g) One fluid mixed, the other partially mixed. This is the case realized in plain tubular crossflow exchangers with a few tube rows.

Multi pass Shell-and-Tube Exchangers

- When the number of tube passes is greater than one, the shell-and- tube exchangers with any of the TEMA shell types (except for the F shell) represent a multi pass exchanger.
- Since the shell-side fluid flow arrangement is unique with each shell type, the exchanger effectiveness is different for each shell even though the number of tube passes may be the same. For illustrative purposes, in the following subsections, two tube passes (as in a U-tube bundle) are considered for the multi pass shell-and-tube exchangers having E, G, H, and J shells.
- However, more than two tube passes are also common, as will be mentioned later. The ideal flow arrangement in the F shell with two tube passes is a pure counterflow arrangement as considered with single-pass exchangers and as can be found by unfolding the tubes.

Thank you