Presentation for
Heat Transfer

DEPARTMENT OF AERONAUTICAL ENGINEERING

B.TECH : V SEM

by

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INTRODUCTION TO HEAT TRANSFER, CONDUCTION

Note

- Heat and temperature are among the most misunderstood concepts in science.
- Temperature is a physical state, based on the molecular activity of an object. If you cut an object in half, each half will have the same temperature.
- Heat is a transfer of energy, which might change the state of temperature. Heat can be transferred without a change in temperature during a phase change (latent heat)
- There is no such concept as the amount of heat IN an object – heat is an energy 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

- 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

- There is no simple equation to describe convection. Here are some general statements about convection

- Heat transfer is proportional to surface area and depth of the fluid

- Heat transfer due to convection will depend on the viscosity of the fluid

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, Ɛ

- The Stefan-Boltzmann Law for heat transfer is $Q = A\varepsilon\sigma T^4$
  - 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

Heat transfer ⇄ complementary ⇄ Thermodynamics

MODES

- ✓ **Conduction**
  - needs matter
  - molecular phenomenon (diffusion process)
  - without bulk 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

The main observations and principles of heat conduction

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

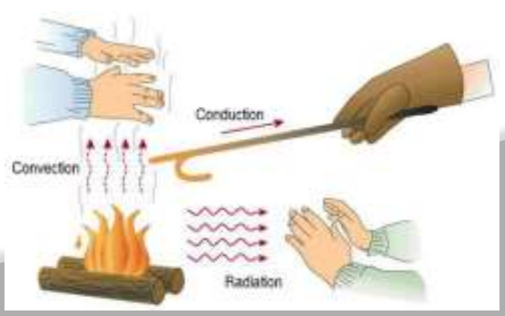
Heat transfer: The mechanism

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: $$Q = -k \frac{dT}{dy} ,$$

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 [$Wm^{-1}K^{-1}$]

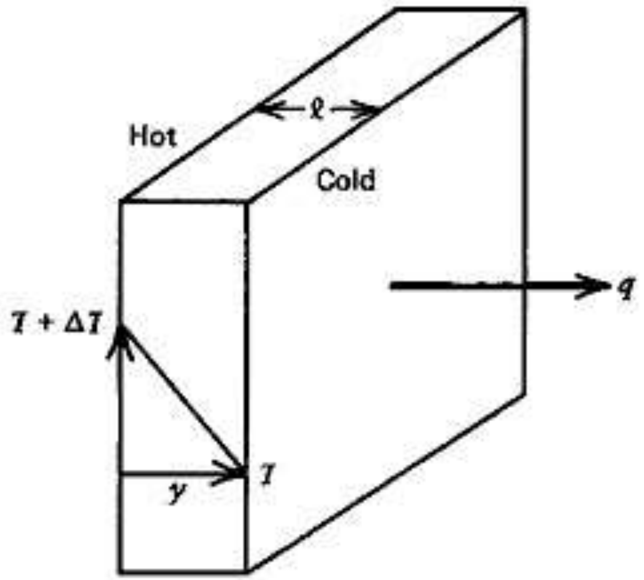
where W is read "watt", and is equal to Joule per second.

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 T:

The heat flux is given by:

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

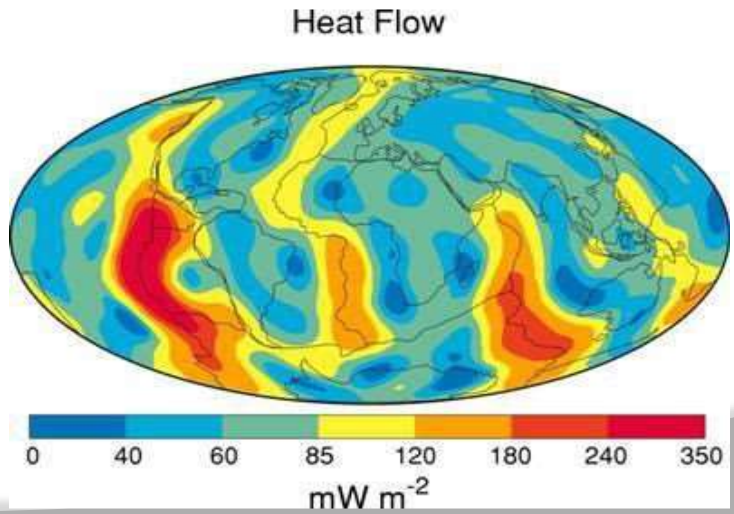


➢Highest heat loss at mid-ocean ridges and lowest at old oceanic crust.

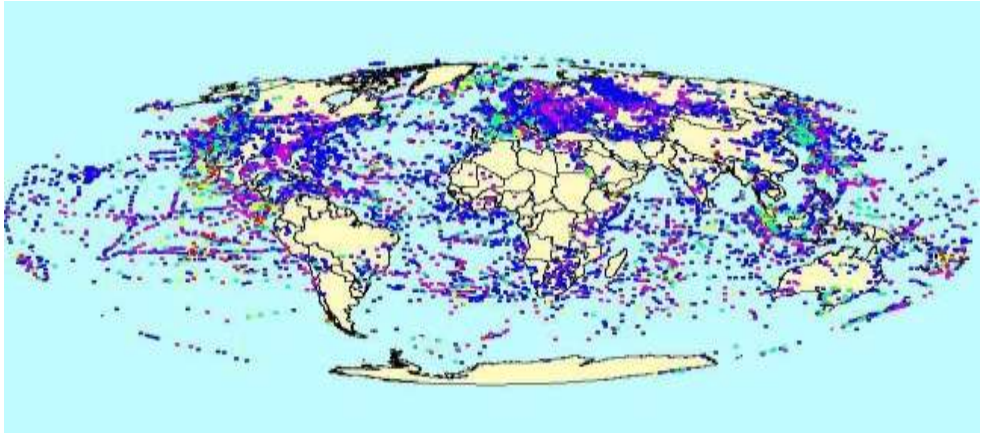
With temperature gradient of 20-30 K/km, and thermal conductivity of 2-3 $WK^{-1}m^{-1}$, the heat flux is 40-90 mWm^{-2}.

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



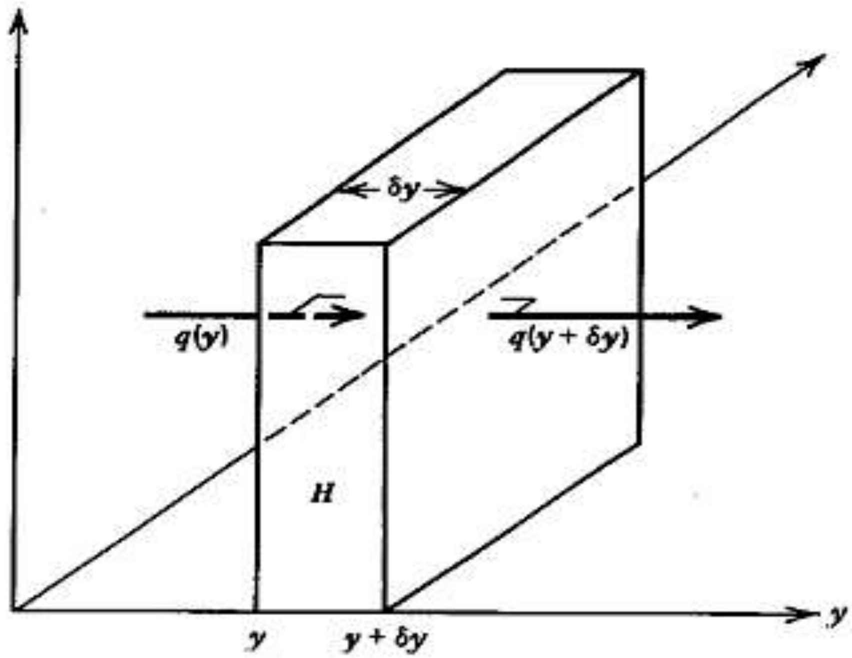
For practical reasons, the vast majority of the measurements are from continental areas in developed countries.

a slab ofainfinitesimal

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:

$$q(y+\ \ y)\quad q(y)=0.$$

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

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

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

$$q(y+\ \ y)\quad q(y)=\ \ y\frac{dq}{dy}=\ \ y\quad k\frac{d_2T}{dy^2}\div=0.$$

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

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:

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

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?

CONDUCTION

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 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)

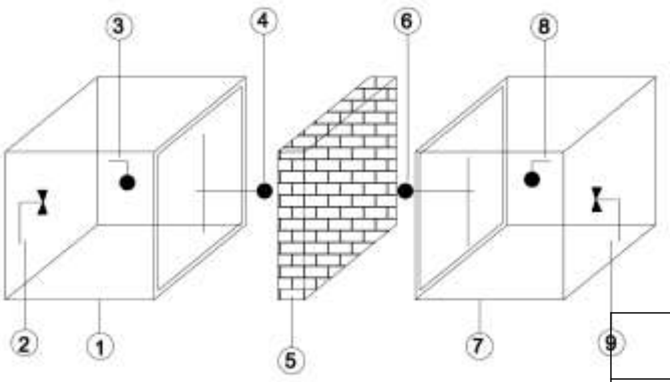
- It indicates the amount of heat that passes through a given thickness of material;
- Conductance= thermal conductivity / thickness

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

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

Example:

Determination of Thermal Conductivity Coefficient for Different Wall Systems



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	Heat

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;

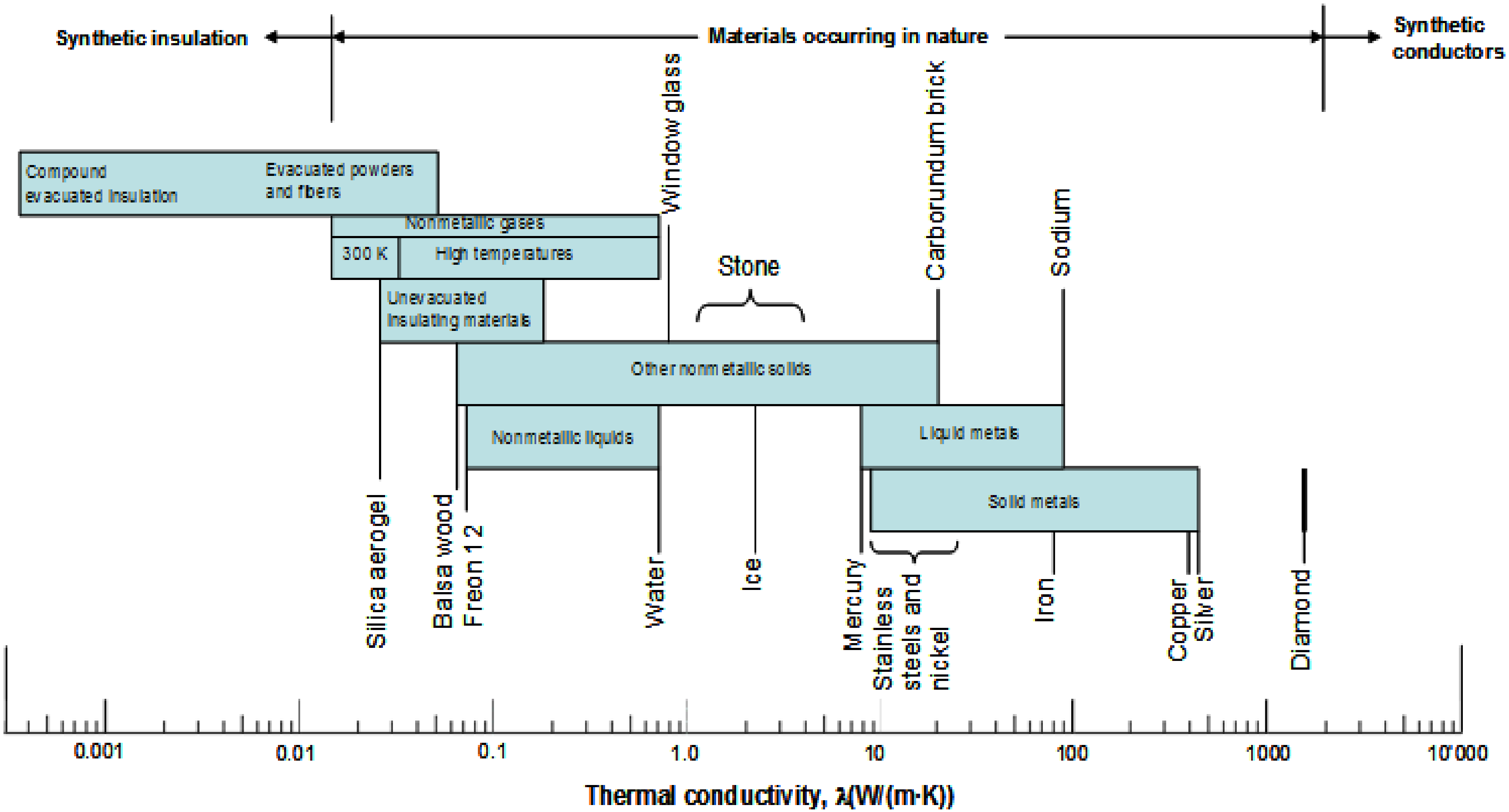
$$R = 1/c$$

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

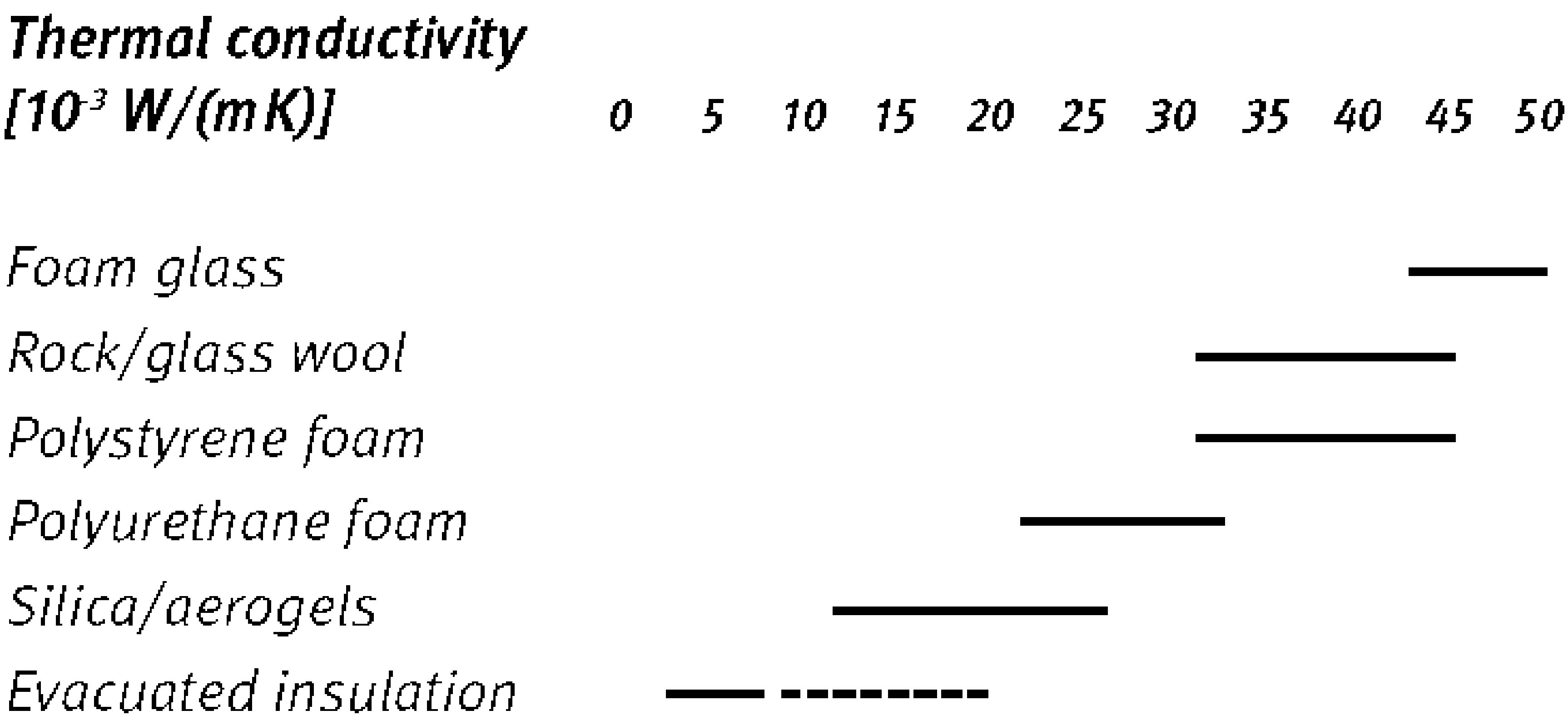
Which pan is a better conductor, one made from copper or one made from cast iron?

	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 properties of materials

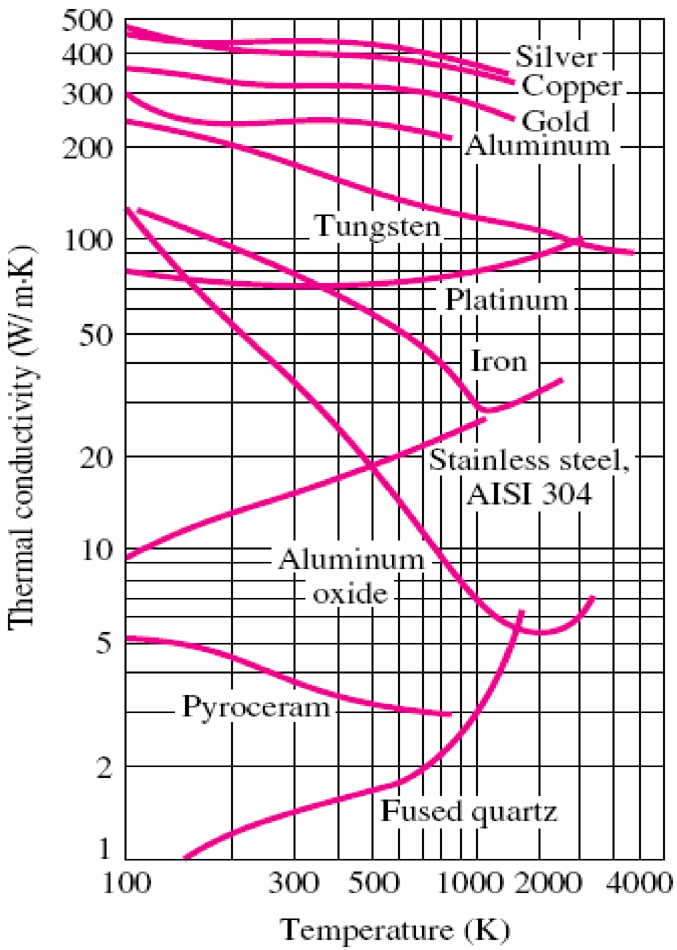
	Thermal Resistance		Thermal Conductivity[a]	
	RSI	R	K (SI)	K (US customary)
Brick, clay, 4 in (100 mm)	0.07	0.42	1.43	9.52
Built-up roofing	0.08	0.44		
Concrete block, 8 in (200 mm):				
Cinder	0.30	1.72	0.67	4.65
Lightweight aggregate	0.35	2.00	0.57	4.00
Glass, clear, ¼ in (6 mm)	0.16	0.91	0.04	0.27
Gypsum sheating, ½ in (12.5 mm)	0.08	0.43	0.16	1.16
Insulation, per 1 in (25 mm):				
Fiberboard	0.49	2.80	0.051	0.36
Glass Fiber	0.52	2.95	0.048	0.34
Expanded Polystyrene	0.75	4.23	0.033	0.24
Rigid urethane	1.05	6.00	0.024	0.17
Vermiculite	0.36	2.08	0.069	0.48
Wood shavings	0.42	2.44	0.060	0.41
Moving air	0.03	0.17		
Particle board, ½ in (12.5 mm)	0.11	0.62	0.114	0.81
Plywood, softwood, ¾ in (19 mm)	0.17	0.97	0.112	0.77
Stucco, ¾ in (19 mm)	0.02	0.11	0.95	6.82

Thermal insulation materials in comparison



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

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

$$k_{ave} = \frac{\int_{T_1}^{T_2} k(T)dT}{T_2 - T_1}$$

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

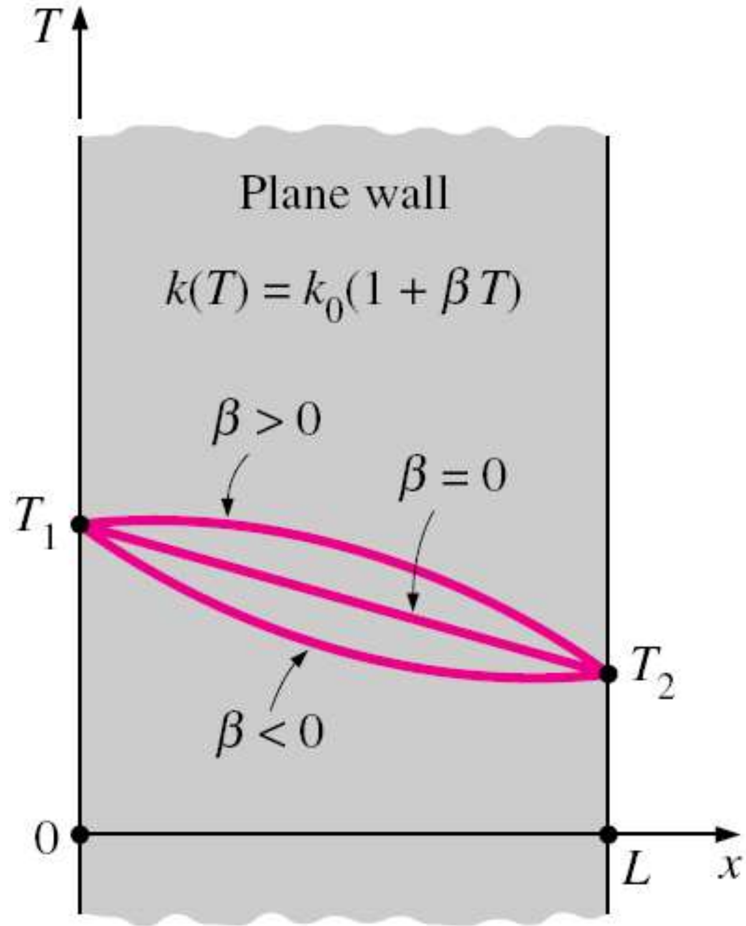
$$k(T) = k_0(1 + \beta T)$$

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).



CONVECTION,

FORCED CONVECTION

Buckingham pi theorem

It 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]

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

in terms of the fundamental units as a product of powers:

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

We shall not be satisfied with just one system of units:

The 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:

We compute

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

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+ $= ms^{-1}$

$= F_1F^{-1}_2$ 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.

(v) 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.

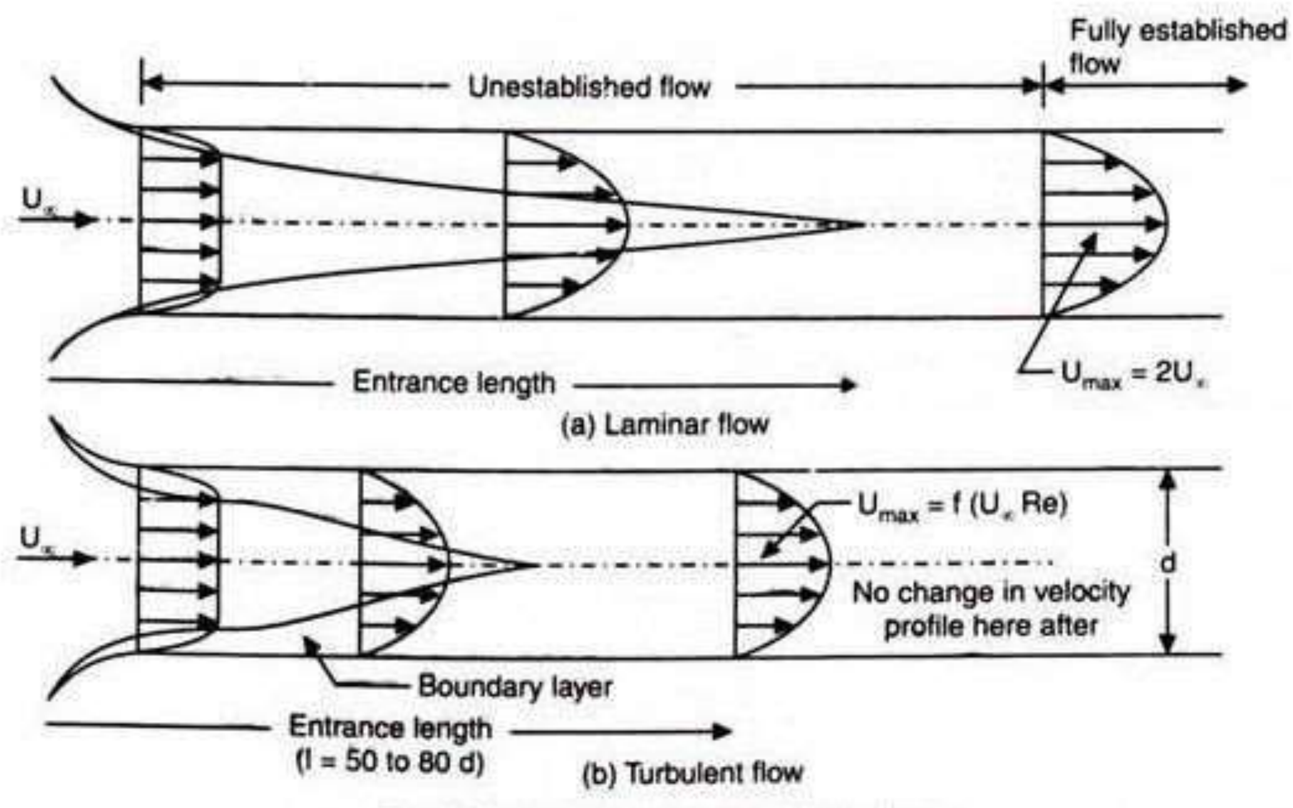
(vii) The pattern of flow in the boundary layer is judged by the Reynolds number Re = $U_\infty x/v$ where x is distance along the plate and measured from its leading edge. The transition from laminar to turbulent pattern of flow occurs at values of Reynolds number between 3×10^5 to 5×10^5. Besides this 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_∞
Entrance length
$U_{max} = 2U_\infty$
(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

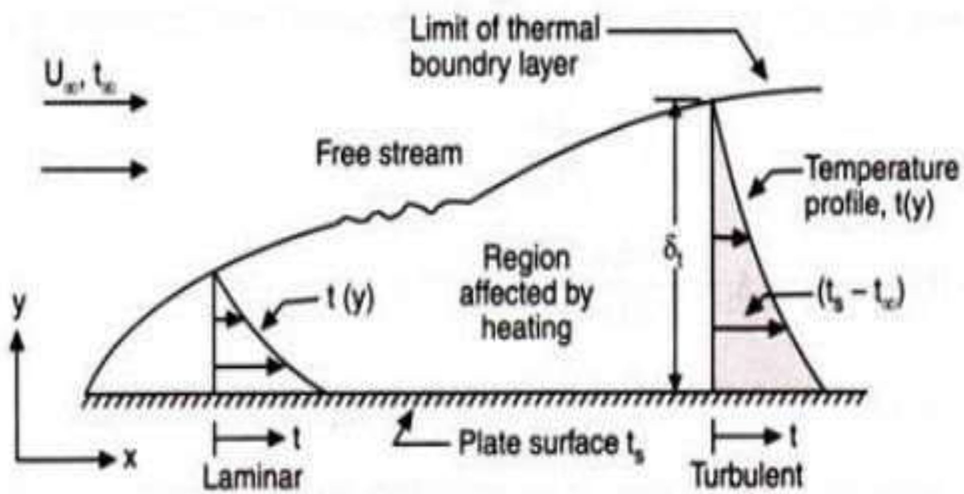
Thermal Boundary Layer:

- 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.

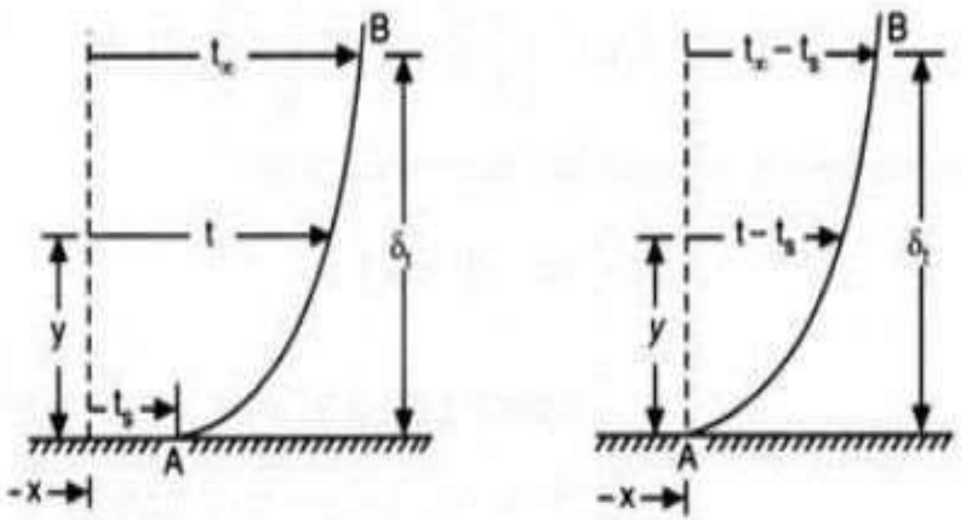
- The thickness δ_f of thermal boundary layer is arbitrarily defined as the distance y from the plate surface at which-

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



- 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.

- 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.

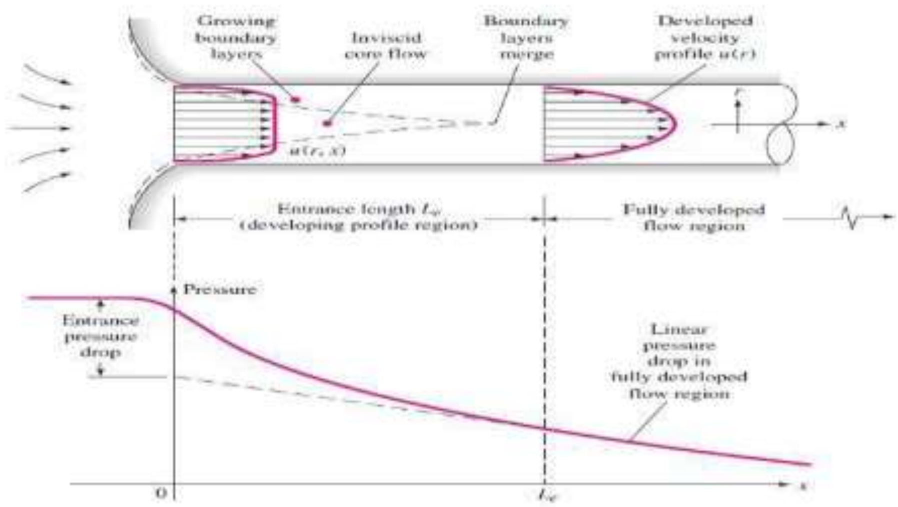
Hydrodynamic Entrance Length

- 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:

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

- 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.



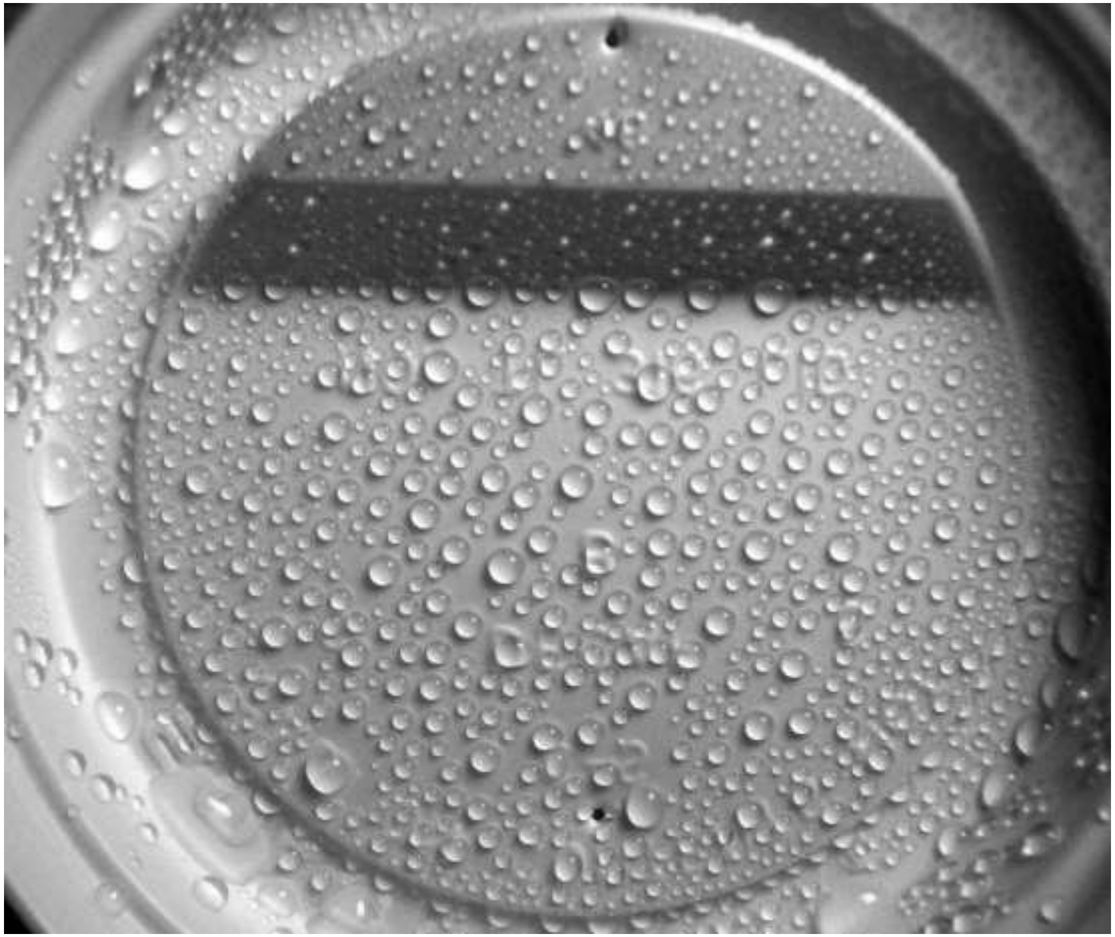
UNIT-IV

CONDENSATION & BOILING

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.

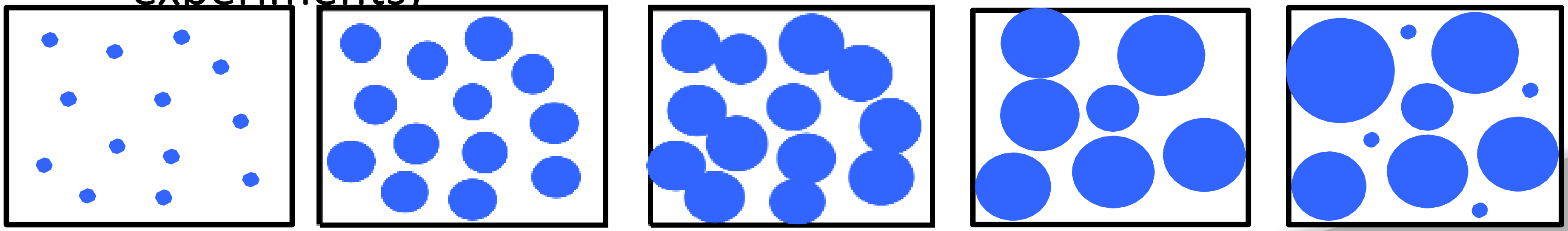
- We are approaching this problem along two lines: experimental observations and computer modeling. Unlike most previous computer models of condensation, our model is based on the physical processes involved (rather than simplified empirical models) and includes the nucleation of new drops as the simulation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

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:

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.

- how saturated vapor at temperature Ts is condensing on a vertical wall whose temperature Tw is constant and lower than the saturation temperature.

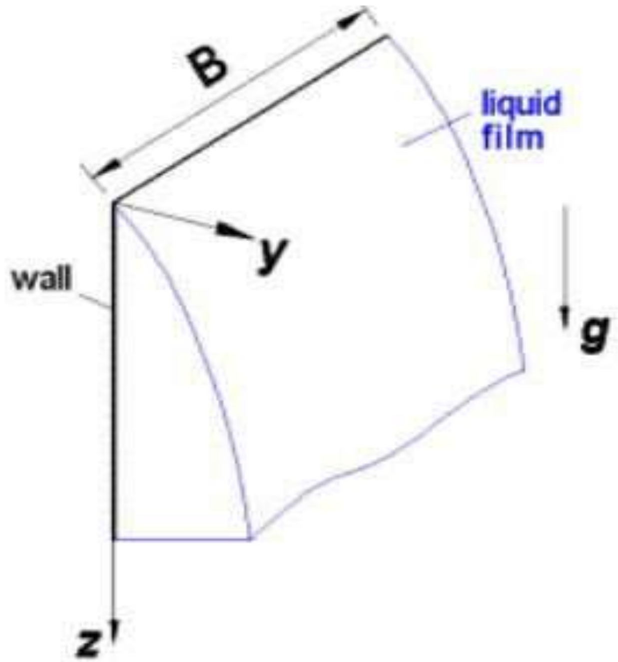


Figure 1

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).

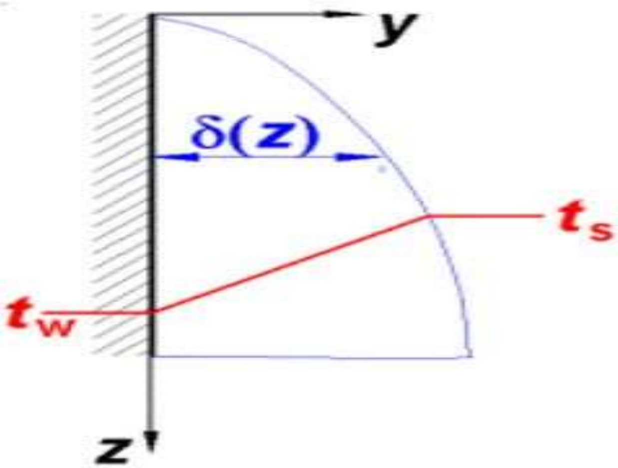


Figure 2

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

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

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} ,

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

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).

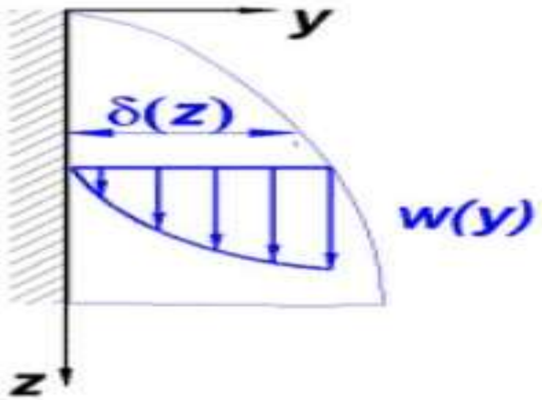


Figure 3

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:

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

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

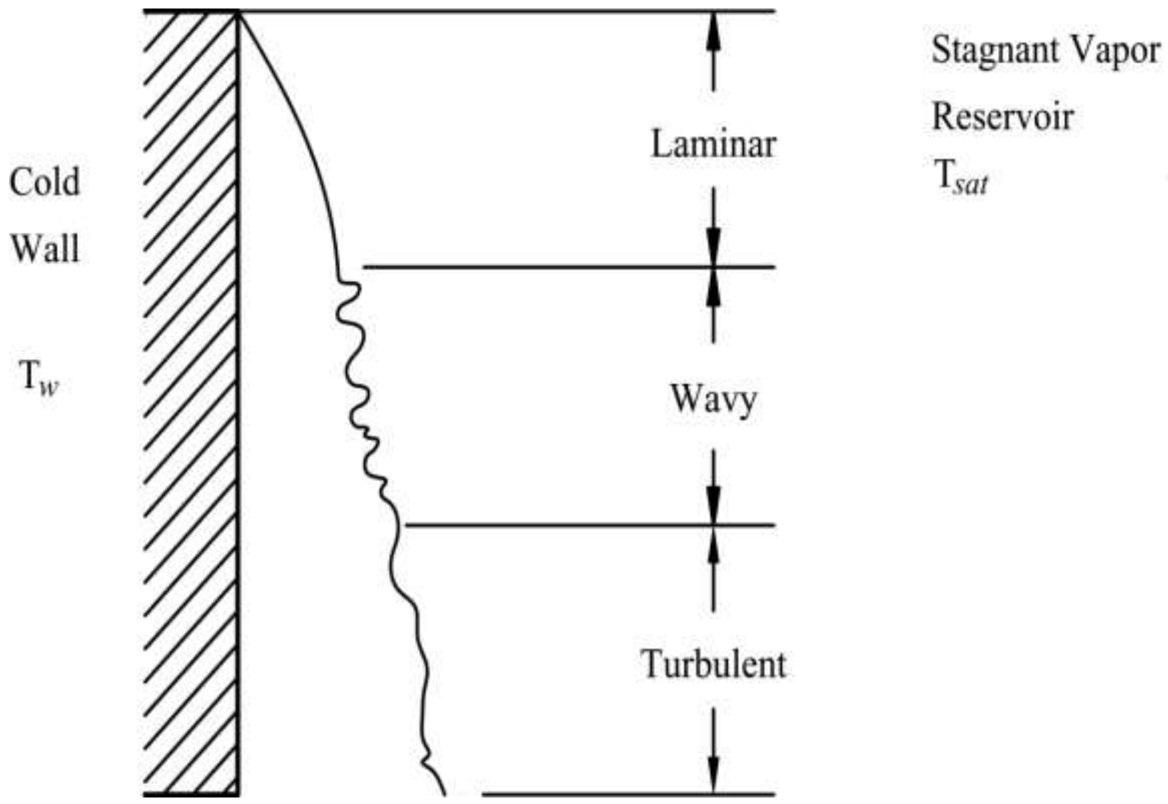
$$T_m = \frac{3}{4}T_w + \frac{3}{4}T_s$$

ρ_v is evaluated at the saturation temperature T_s.

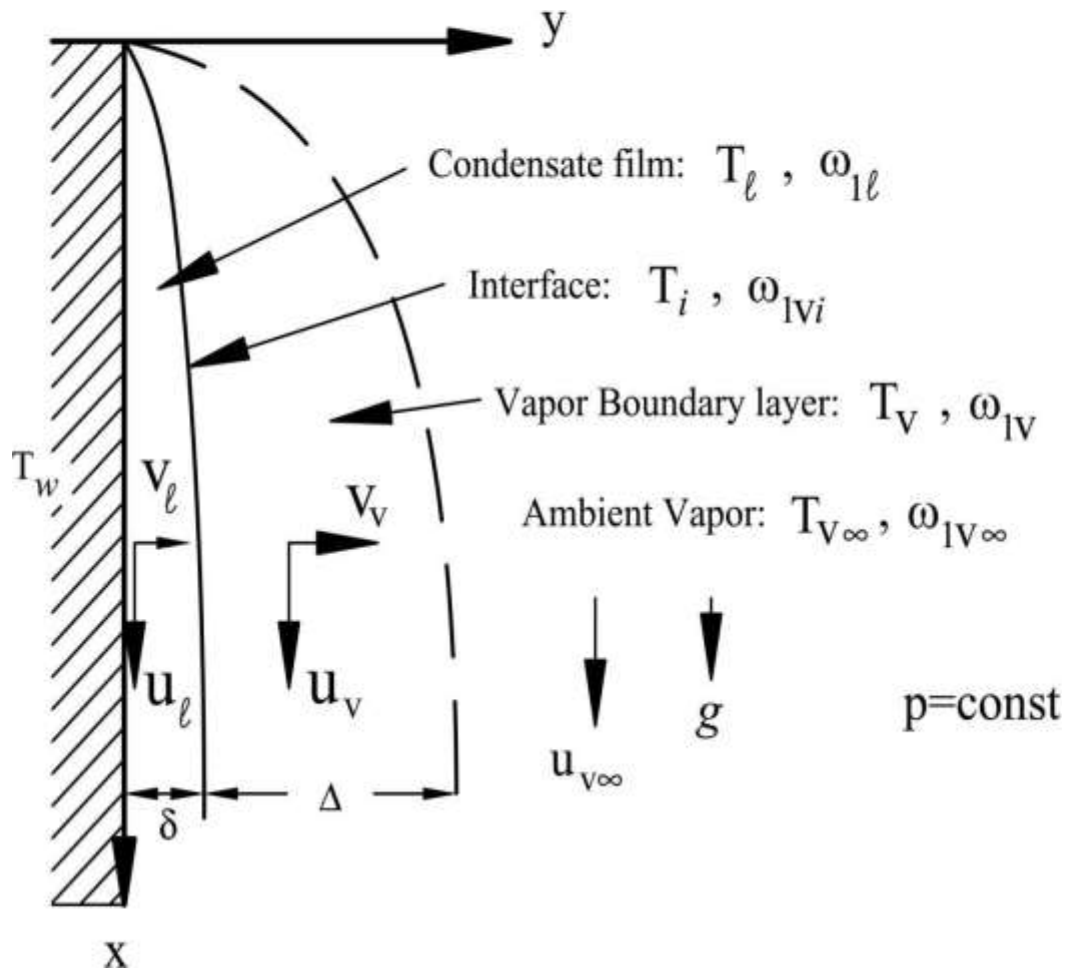
Nusselt derived a similar equation for film condensation on horizontal tubes using a numerical integration. The mean heat transfer coefficient for a single horizontal tube of diameter D is

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

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.

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

For the condensate film:

$$\frac{\partial u_1}{\partial x} + \frac{\partial v_1}{\partial y} = 0$$

$$u_1 \frac{\partial u_1}{\partial x} + v_1 \frac{\partial u_1}{\partial y} = \nu_1 \frac{\partial^2 u_1}{\partial y^2} + g - \frac{1}{\rho_1} \frac{dp}{dx}$$

$$u_1 \frac{\partial T_1}{\partial x} + v_1 \frac{\partial T_1}{\partial y} = \alpha_1 \frac{\partial^2 T_1}{\partial y^2}$$

For the vapor boundary layer:

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

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

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

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

Isobaric specific heat difference of the binary vapor

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

- Definitions of the terms in eqs.

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

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

- where

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

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

- Partial pressures of the system are determined by

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

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

Boundary conditions at the surface of the cold wall

$$u_1 = 0, \quad y = 0$$

$$v_1 = 0, \quad y = 0$$

$$T_1 = T_w, \quad y = 0$$

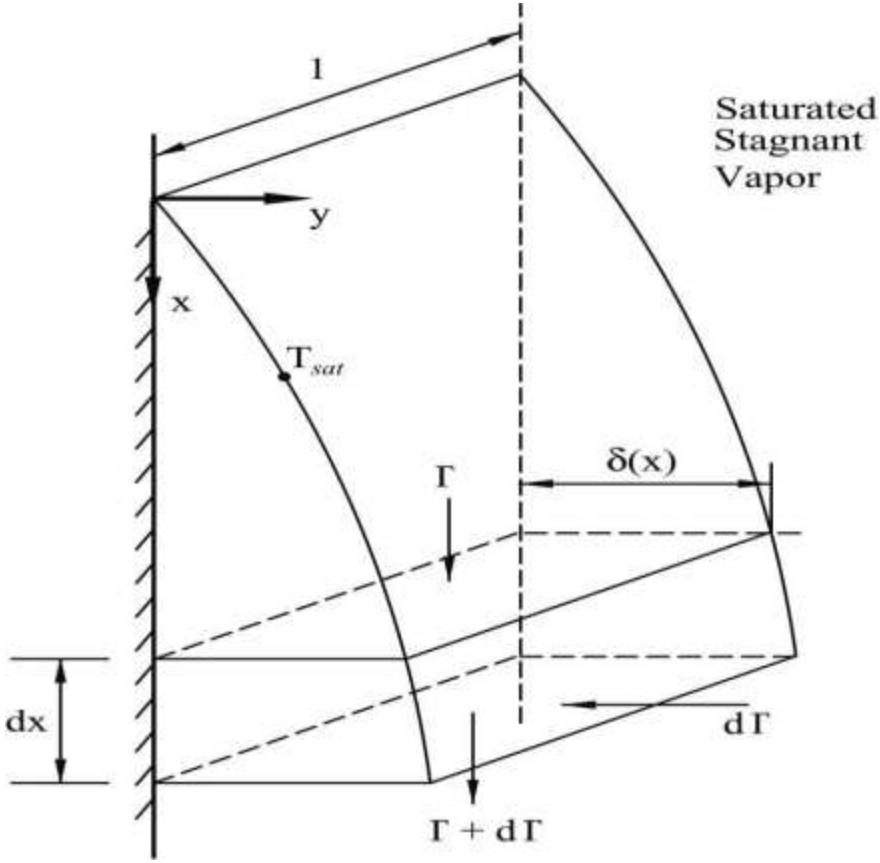
- Boundary conditions at locations far from the cold wall

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

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

$$y \to \infty$$

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



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

- Pressure in the liquid film

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

-

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

-

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

-

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

-

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

The assumption of laminar film condensation is now checked by obtaining the Reynolds number defined

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

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.

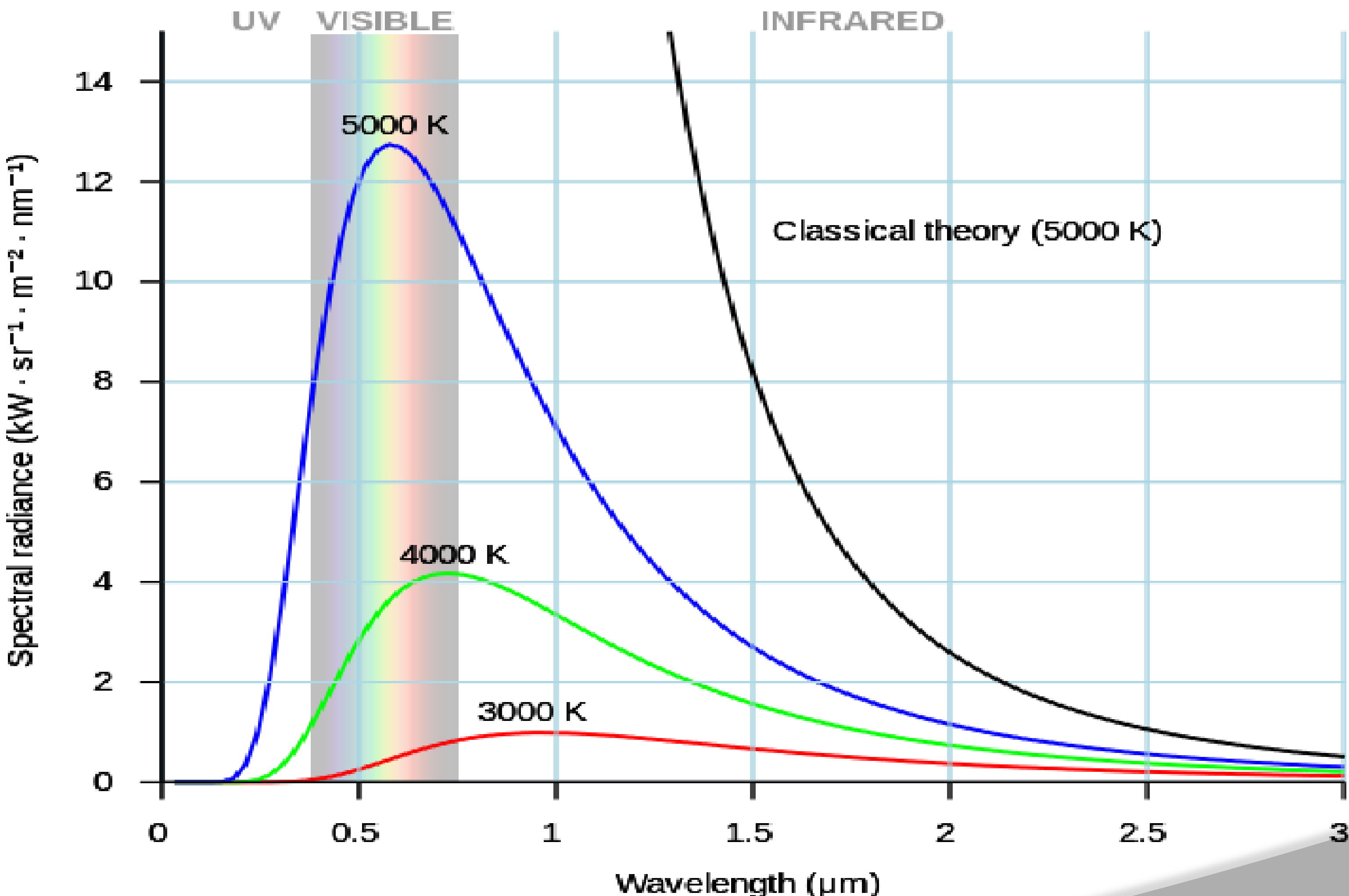
Because the human eye cannot perceive light waves at lower frequencies, a black body, viewed in the dark at the lowest just faintly visible temperature, subjectively appears grey, even though its objective physical spectrum peaks in the infrared range.

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

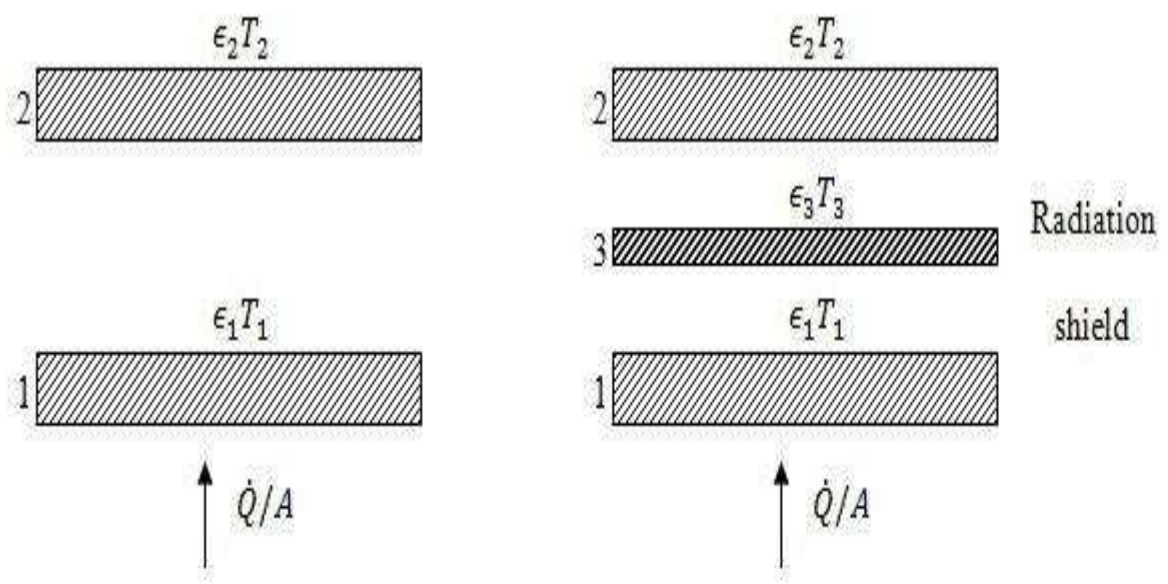
- The easiest method to calculate radiative heat transfer between two

- 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 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,

$$\alpha_m + \tau_m = 1 = \epsilon_m + \tau_m$$

The energy leaving surface 1 which is transmitted through the medium and reaches the surface 2 is,

$$J_1 A_1 F_{12} \tau_m$$

and that which leaves surface 2 and arrives at surface 1 is,

$$J_2 A_2 F_{21} \tau_m$$

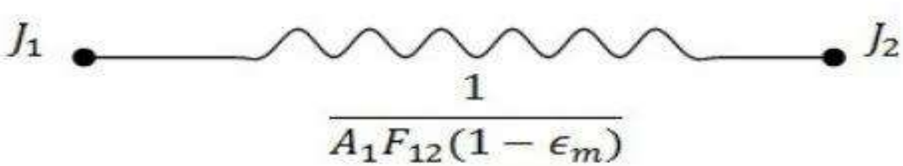
Therefore, the net exchange in the transmission process is therefore,

$$\dot{Q}_{12} = A_1 F_{12} \tau_m (J_1 - J_2)$$

Using eq.

$$\dot{Q}_{12} = \frac{(J_1 - J_2)}{\left(\frac{1}{A_1 F_{12}(1-\epsilon_m)}\right)}$$

Thus the equivalent circuit diagram is shown in fig



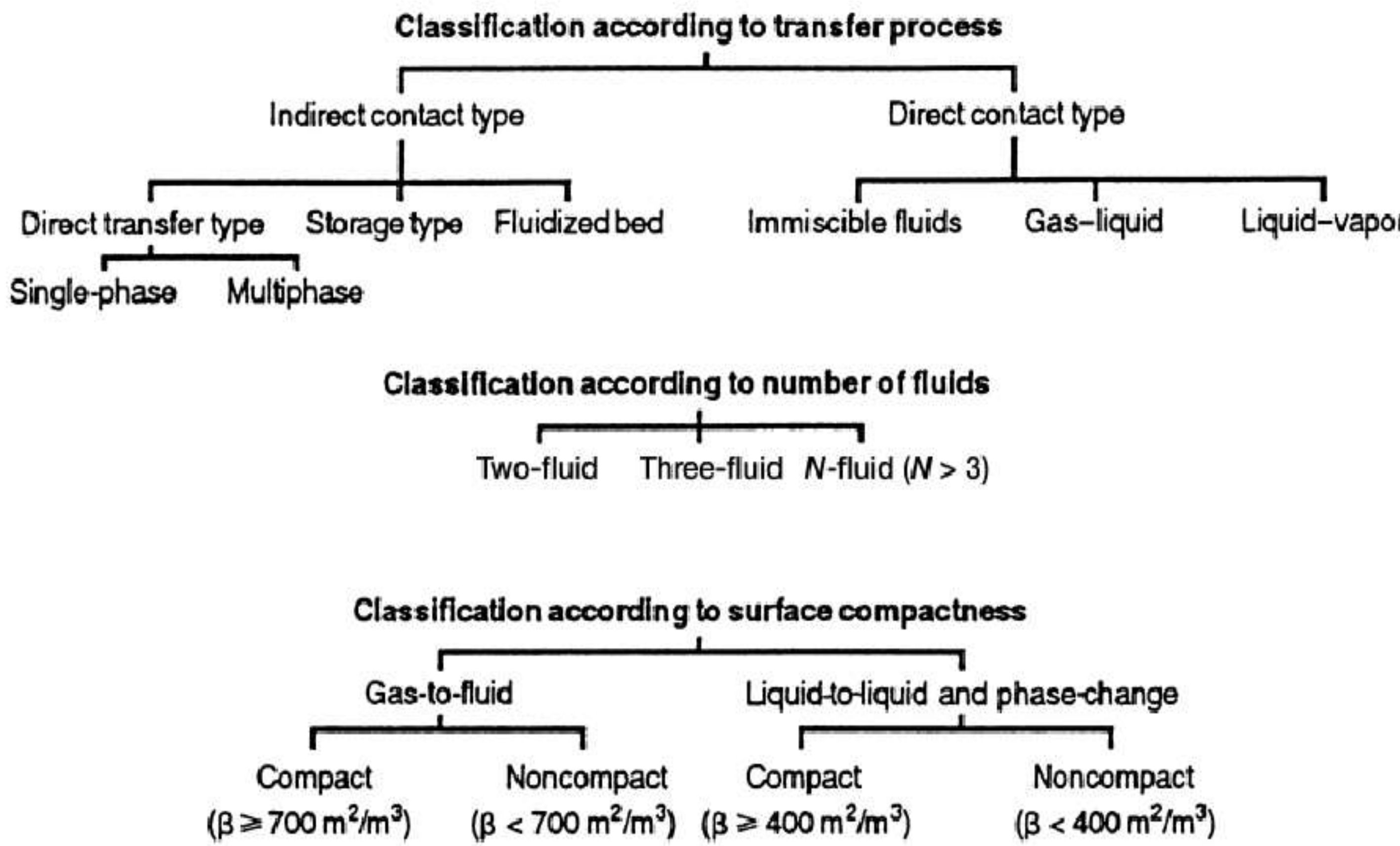
HEAT EXCHANGERS

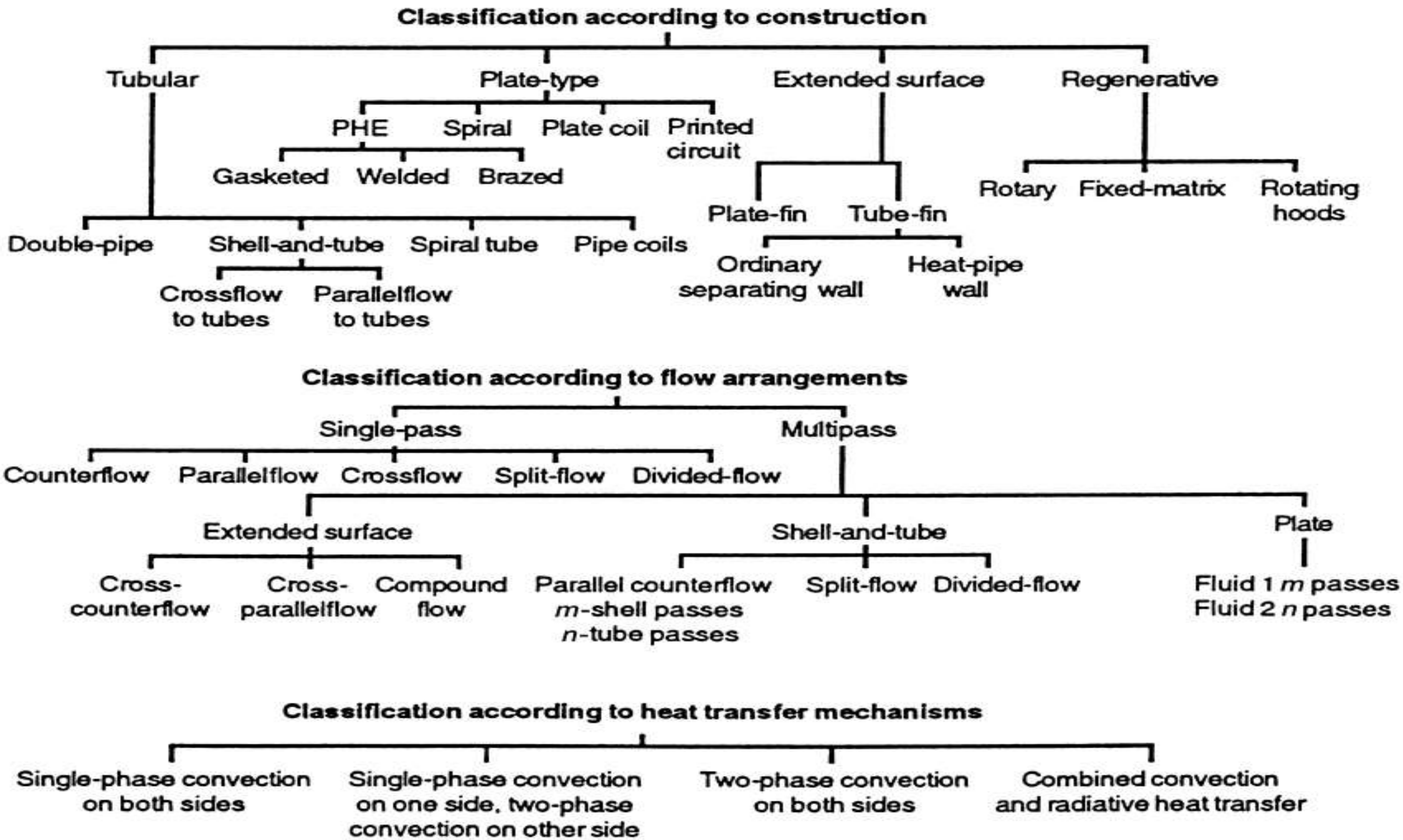
Classification of Heat Exchangers

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.

Typical applications involve heating or cooling of a fluid stream of concern and evaporation or condensation of single- or multi component fluid streams.

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 contact type
Direct transfer type
Storage type
Fluidized bed
Immiscible fluids
Gas–liquid
Liquid–vapor
Single-phase
Multiphase
Classification according to number of fluids
Two-fluid
Three-fluid
N-fluid (N > 3)
Classification according to surface compactness
Gas-to-fluid
Liquid-to-liquid and phase-change
Compact
(β ≥ 700 m²/m³)
Noncompact
(β < 700 m²/m³)
Compact
(β ≥ 400 m²/m³)
Noncompact
(β < 400 m²/m³)


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

CLASSIFICATIONS ACCORDING TO TRANSFER PROCESSES:

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

Indirect-Contact Heat Exchangers:

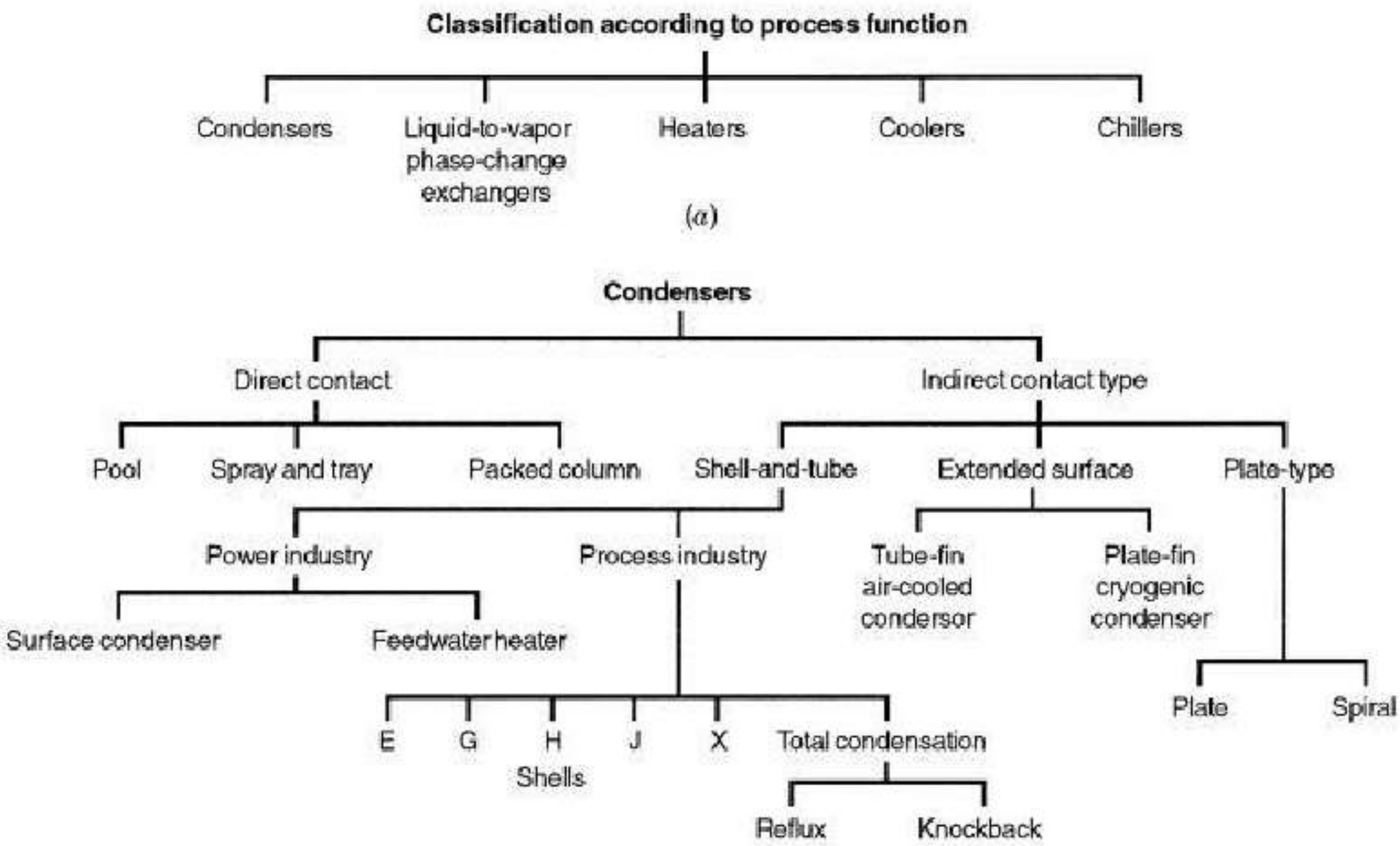
In an indirect-contact heat exchanger, the fluid streams remain separate and the heat transfers continuously through an impervious dividing wall or into and out of a wall in a transient manner.

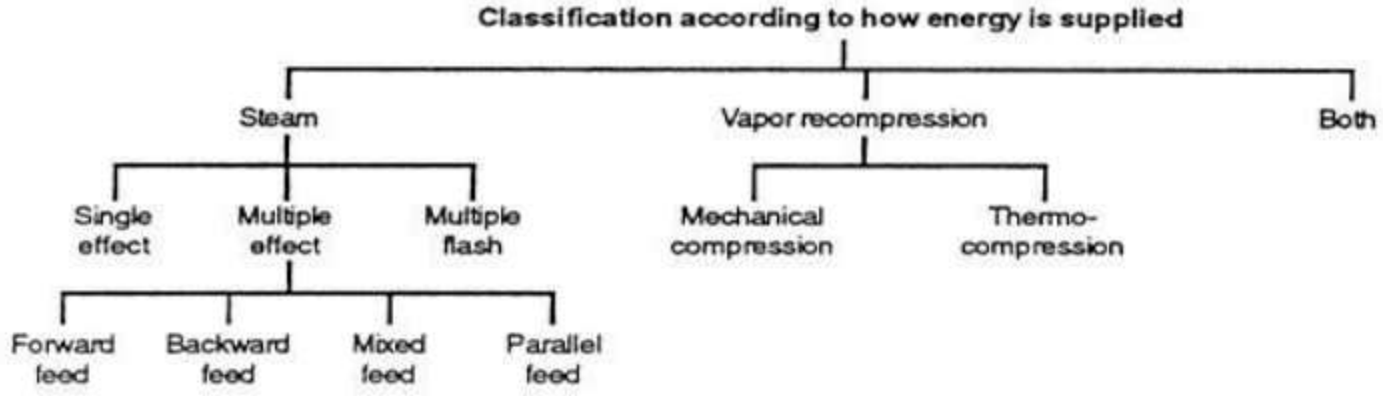
Thus, ideally, there is no direct contact between thermally interacting fluids. This type of heat exchanger, also referred to as a surface heat exchanger, can be further classified into direct-transfer type, storage type, and fluidized-bed exchangers.

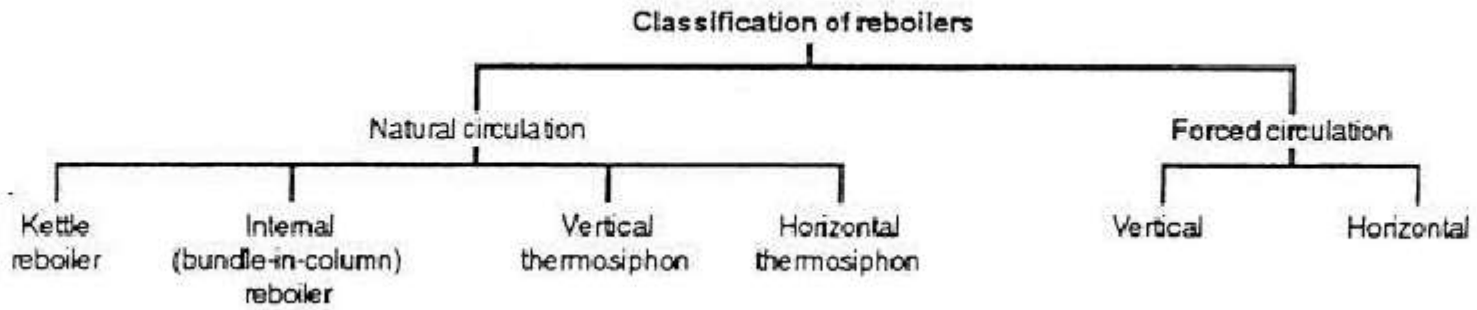
Direct-Transfer Type Exchangers:

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 condensor
Plate-fin cryogenic condenser
Surface condenser
Feedwater heater
Plate
Spiral
E
G
H
J
X
Total condensation
Shells
Reflux
Knockback


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:

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.

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.

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.

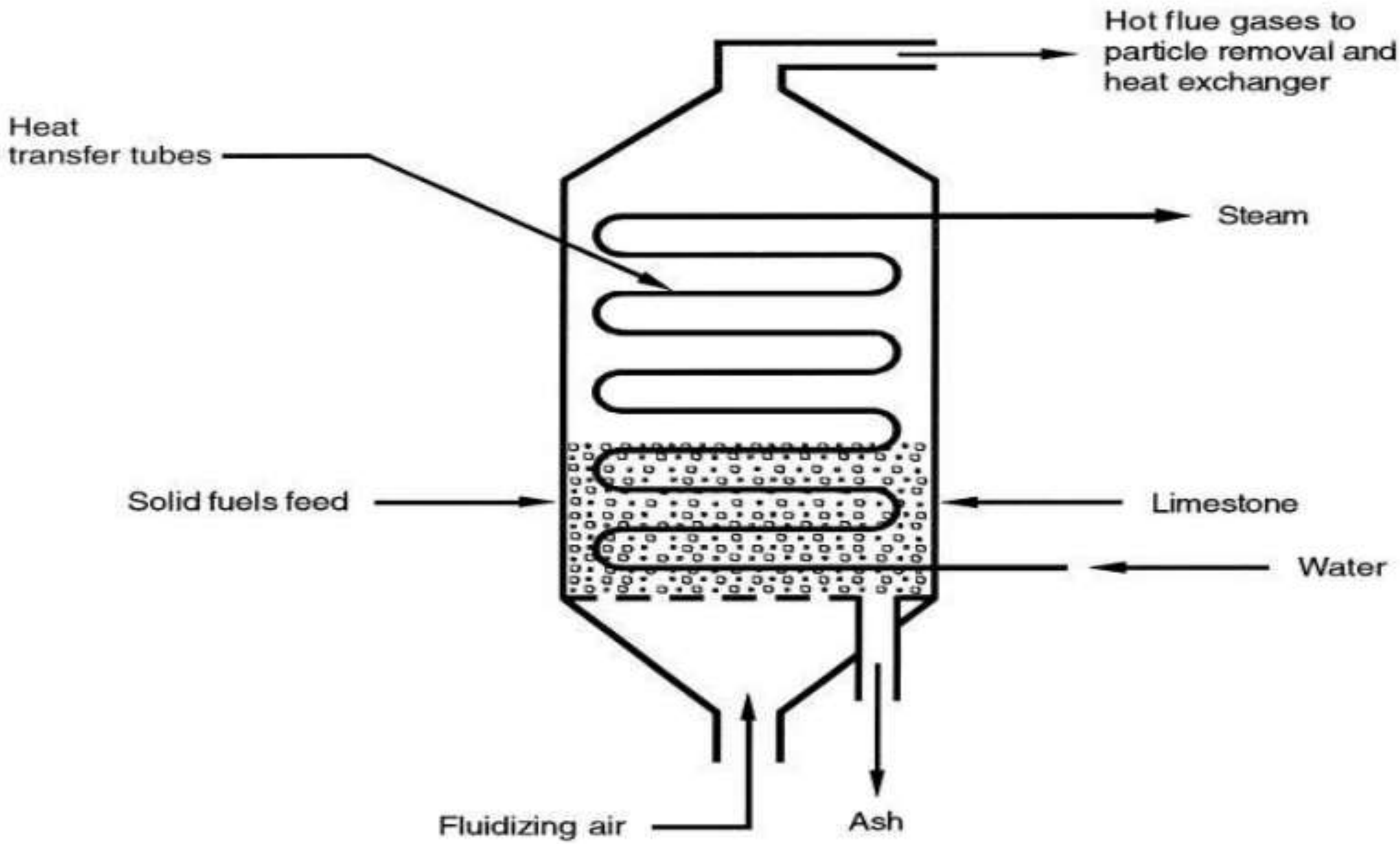
Fluidized-Bed Heat Exchangers:

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.

As a result, the solid particles will float with an increase in bed vo 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.

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


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 cases where a direct contact of two fluid streams is permissible.

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.