NEAR EAST UNIVERSITY

FACULTY OF ENGINEERING
MECHANICAL ENGINEERING DEPARTMENT

ME 400 GRADUATION PROJECT

FORCE CONVECTION HEAT TRANSFER CORRELATIONS FOR IN TUBE, SINGLE CYLINDER, SINGLE SPHERE, TUBE BUNDLES

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ABSTRACT

Convection is classified as natural (or free) or forced convection, depending on how the fluid motion is initiated. In forced convection, the fluid is forced to flow over a surface or in a tube by external means such as a pump or a fan.

The aim of this project is to examine both external and internal forced convection correlations for flow in tube, single cylinder, single sphere and tube bundles. We continue with the discussion of the dimensionless Reynolds, Prandtl, and Nusselt numbers and their physical signifinance. We then present emprical relations for flow over various geometries such as a tube, cylinder and sphere for both laminar and turbulent flow conditions.

Chapter 1 gives a brief introduction of what and how is heat transfer and the three modes of heat transfer which are conduction, convection and radiation. Chapter 2 explains the calculations for heat conduction in plane walls by using thermal resistance concept and network. Chapter 3 explains physical mechanism of natural convection, natural convection over surfaces, from finned surfaces and combined natural and forced convection. Chapter 4 explains the thermal radiation, Blackbody radiation, radiation properties and radiation is calculated by using equation $Q_{rad} = \varepsilon\sigma A\left(T_s^4 - T_{surr}^4\right)$. Chapter 5 explains the following subjects; physical mechanism of forced convection, laminar and turbulent flows, Reynolds number, flow in tubes, constant heat flux, laminar and turbulent flows in tubes, the cylinder in cross flow, the sphere in cross flow and flow across bank of tubes (tube bundles). These subjects are explained by using formulas, figures, diagrams and tables.

CHAPTER 1

INTRODUCTION TO HEAT TRANSFER

1.Introduction

1.1What and How is Heat Transfer?

The definition provides sufficient response to the question:
What is heat transfer?

Heat transfer (or heat) is energy in transit due to a temperature difference.

Whenever there exists a temperature difference in a medium or between media, heat transfer must occur.

As shown in Figure 1, we refer to different types of heat transfer processes as *modes.* When a temperature gradient exists in a stationary medium, which may be a solid or a fluid, we use the term *conduction* to refer to the heat transfer that will occur across the medium. In contrast, the term *convection* refers to heat transfer that will occur between a surface and a moving fluid when they are at different temperatures. The third mode of heat transfer is termed *thermal radiation.* All surfaces of finite temperature emit energy in the form of electromagnetic waves. Hence, in the absence of an intervening medium, there is net heat transfer by radiation between two surfaces at different temperatures. As engineers it is important that we understand the *physical mechanisms* which underlie the heat transfer modes and that we be able to use the rate equations that quantify the amount of energy being transferred per unit time.

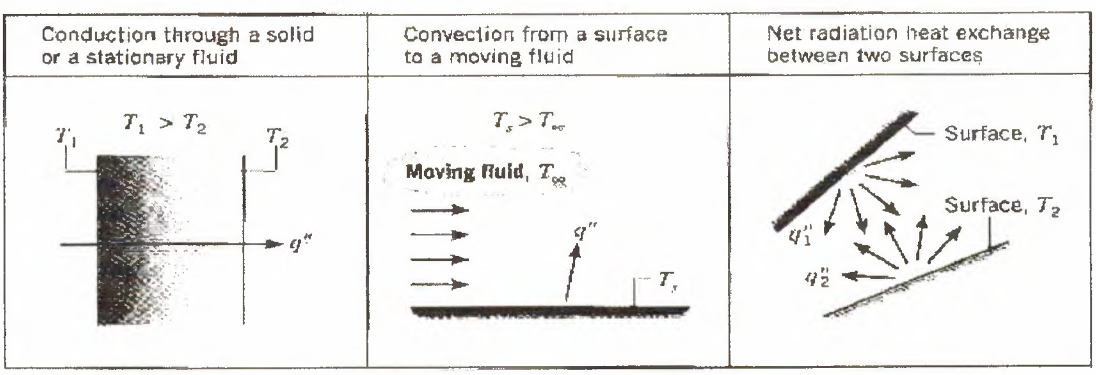


Figure 1 Conduction, convection, and radiation heat transfer modes.

1.1.1 Application Areas of Heat Transfer

Heat transfer is commonly encountered in engineering systems and other aspects of life, and one does not need to go very far to see some application areas of heat transfer. In fact, one does not need to go anywhere. The human body is constantly rejecting heat to its surroundings, and human comfort is closely tied to the rate of this heat rejection. We try to control this heat transfer rate by adjusting our clothing to the environmental conditions.

Many ordinary household appliances are designed, in whole or in part, by using the principles of heat transfer. Some examples include the electric or gas range, the heating and air-conditioning system, the refrigerator and freezer, the water heater, the iron, and even the computer, the TV, and the VCR. Of course, energy-efficient homes are designed on the basis of minimizing heat loss in winter and heat gain in summer. Heat transfer plays a major role in the design of many other devices, such as car radiators, solar collectors, various components of power plants, and even spacecraft. The optimal insulation thickness in the walls and roofs of the houses, on hot water or steam pipes, or on water heaters is again determined on the basis of a heat transfer analysis with economic consideration (Fig. 1.1).

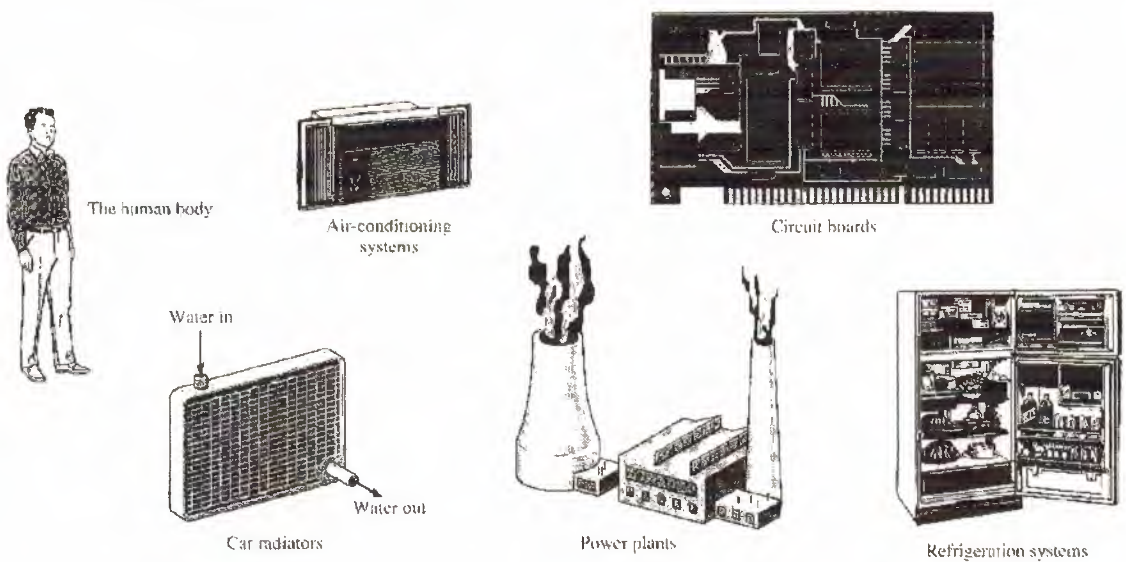


Figure 1.1 Some application areas of heat transfer.

1.2 Heat Transfer Mechanisms

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

1.2.1 Conduction

Conduction is the transfer of energy from the more energetic particles of a substance to the adjacent less energetic ones as a result of interactions between the particles. Conduction can take place in solids, liquids, or gases. In *gases* and *liquids,* conduction is due to the *collisions* and *diffusion* of the molecules during their random motion. In *solids,* it is due to the combination of *vibrations* of the molecules in a lattice and the energy transport by *free electrons.* A cold canned drink in a warm room, for example, eventually warms up to the room temperature as a result of heat transfer from the room to the drink through the aluminum can by conduction.

The *rate* of heat conduction through a medium depends on the *geometry* of the medium, its *thickness,* and the *material* of the medium, as well as the *temperature difference* across the medium. We know that wrapping a hot water tank with glass wool (an insulating material) reduces the rate of heat loss from the tank. The thicker the insulation, the smaller the heat loss. We also know that a hot water tank will lose heat at a higher rate when the temperature of the room housing the tank is lowered. Further, the larger the tank, the larger the surface area and thus the rate of heat loss.

Consider steady heat conduction through a large plane wall of thickness $\Delta x = L$ and surface area A, as shown in Figure 1.2.1. The temperature difference across the wall is $\Delta T = T_2 - T_1$. Experiments have shown that the rate of heat transfer Q through the wall is *doubled* when the temperature difference ΔT across the wall or the area A normal to the direction of heat transfer is doubled, but is *halved* when the wall thickness L is doubled. Thus we conclude that *the rate of heat conduction through a plane layer is proportional to the temperature difference across the layer and the heat transfer area, but is inversely proportional to the thickness of the layer.* That is,

$$\text{Rate of heat conduction } \alpha \frac{(Area)(Temperature\ difference)}{Thickness}$$

or,

$$Q_{cond.} = kA\frac{\Delta T}{\Delta x} \qquad \text{(W)}$$

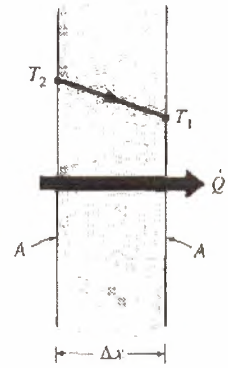


Figure 1.2.1 Heat conduction through a large plane wall of thickness Δx and area A.

Where the constant of proportionality k is the thermal conductivity of the material, which is a *measure of the ability of a material to conduct heat* (Fig. 1.2.2).

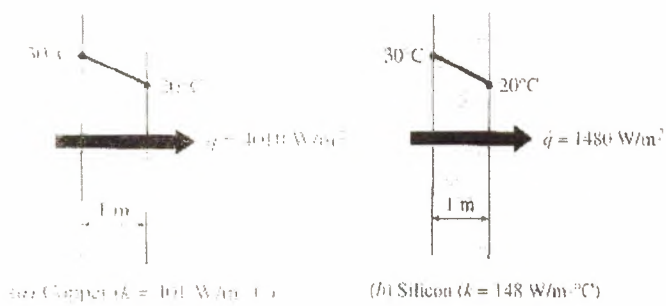


Figure1.2.2 The rate of heat conduction through a solid is directly proportional to its thermal conductivity.

1.2.2 Convection

Convection is the mode of energy transfer between a solid surface and the adjacent liquid or gas that is in motion, and it involves the combined effects of *conduction and fluid motion.* The faster the fluid motion, the greater the convection heat transfer. In the absence of any bulk fluid motion, heat transfer between a solid surface and the adjacent fluid is by pure conduction. The presence of bulk motion of the fluid enhances the heat transfer between the solid surface and the fluid, but it also complicates the determination of heat transfer rates.

Consider the cooling of a hot block by blowing cool air over its top surface (Fig. 1.2.6). Energy is first transferred to the air layer adjacent to the block by conduction. This energy is then carried away from the surface by convection; that is, by the combined effects of conduction within the air that is due to random motion of air molecules and the bulk or macroscopic motion of the air that removes the heated air near the surface and replaces it by the cooler air

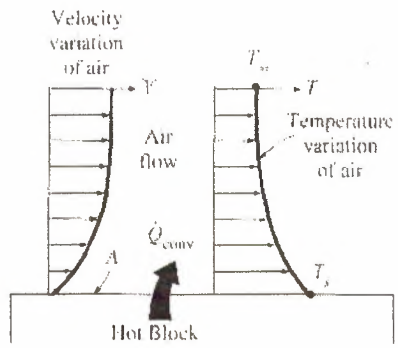


Figure 1.2.6 Heat transfer from a hot surface to air by convection.

Convection is called **forced convection** if the fluid is forced to flow over the surface by external means such as a fan, pump, or the wind. In contrast, convection is called **natural** (or **free**) **convection** if the fluid motion is caused by buoyancy forces that are induced by density differences due to the variation of temperature in the fluid (Fig. 1.2.7). For example, in the absence of a fan, heat transfer from the surface of the hot block in Figure 1.2.6 will be by natural convection since any motion in the air in this case will be due to the rise of the warmer (and thus lighter) air near the surface and the fall of the cooler (and thus heavier) air to till its place. Heat transfer between the block and the surrounding air will be by conduction if the temperature difference between the air and the block is not large enough to overcome the resistance of air to movement and thus to initiate natural convection currents.

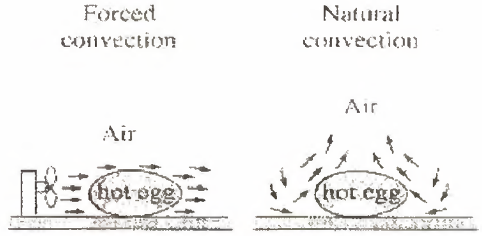


Figure 1.2.7 The cooling of a boiled egg by forced and natural convection.

Heat transfer processes that involve *change of phase* of a fluid are also considered to be convection because of the fluid motion induced during the process, such as the rise of the vapor bubbles during boiling or the fall of the liquid droplets during condensation. Despite the complexity of convection, the rate of *convection heat transfer* is observed to be proportional to the temperature difference, and is conveniently expressed by **Newton's law of cooling** as

$$Q_{convection} = hA(T_s - T_\infty) \qquad \text{(W)}$$

where h is the *convection heat transfer coefficient* in $W/m^2 \cdot {}^0C$ or $Btu/h \cdot ft^2 \cdot {}^0F$, A is the surface area through which convection heat transfer takes place, T_s is the surface temperature, and T_∞ is the temperature of the fluid sufficiently far from the surface. Note that at the surface, the fluid temperature equals the surface temperature of the solid.

The convection heat transfer coefficient h is not a property of the fluid. It is an experimentally determined parameter whose value depends on all the variables influencing convection such as the surface geometry, the nature of fluid motion, the properties of the fluid, and the bulk fluid velocity.

1.2.3 Radiation

Radiation is the energy emitted by matter in the form of *electromagnetic waves* (or *photons)* as a result of the changes in the electronic configurations of the atoms or molecules. Unlike conduction and convection, the transfer of energy by radiation does not require the presence of an *intervening medium.* In fact, energy transfer by radiation is fastest (at the speed of light) and it suffers no attenuation in a vacuum. This is exactly how the energy of the sun reaches the earth.

In heat transfer studies we are interested in *thermal radiation,* which is the form of radiation emitted by bodies because of their temperature. It differs from other forms of electromagnetic radiation such as x-rays, gamma rays, microwaves, radio waves, and television waves that are not related to temperature. All bodies at a temperature above absolute zero emit thermal radiation.

Radiation is a *volumetric phenomenon,* and all solids, liquids, and gases emit, absorb, or transmit radiation to varying degrees. However, radiation is usually considered to be a *surface phenomenon* for solids that are opaque to thermal radiation such as metals, wood, and rocks since the radiation emitted by the interior regions of such material can never reach the surface, and the radiation incident on such bodies is usually absorbed within a few microns from the surface.

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

$$Q_{emit,\max} = \sigma A T_s^4 \qquad \text{(W)}$$

where $\sigma = 5.67 \times 10^{-8}$ W/m^2. K^4 or 0.1714×10^{-8} Btu/h.ft^2.R^4 is the *Stefan-Boltzmann constant.* The idealized surface that emits radiation at this maximum rate is called **a blackbody**, and the radiation emitted by a blackbody is called blackbody radiation (Fig. 1.2.3). The radiation emitted by all real surfaces is less than the radiation emitted by a blackbody at the same temperature, and is expressed as

$$Q_{emit} = \varepsilon \sigma A T_s^4 \qquad \text{(W)}$$

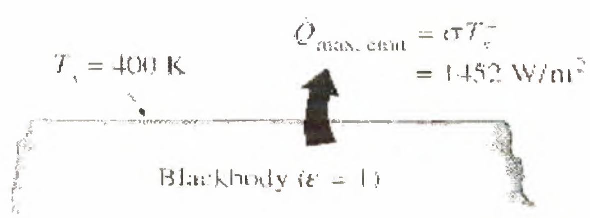


Figure 1.2.3 Blackbody radiation represents the *maximum amount of radiation that can be emitted from a surface at a specified temperature.*

where ε is the *emissivity* of the surface. The property emissivity, whose value is in the range $0 \le \varepsilon \le 1$, is a measure of how closely a surface approximates a blackbody for which $\varepsilon = 1$. The emissivities of some surfaces are given in Table 1.1.

Material	Emissivity
Aluminum foil	0.07
Anodized aluminum	0.82
Polished copper	0.03
Polished gold	0.03
Polished silver	0.02
Polished stainless steel	0.17
Black paint	0.98
White paint	0.90
White paper	0.92–0.97
Asphalt pavement	0.85–0.93
Red brick	0.93–0.96
Human skin	0.95
Wood	0.82–0.92
Soil	0.93–0.96
Water	0.96
Vegetation	0.92–0.96

Table 1.1 Emissivities of some materials at 300 K

Another important radiation property of a surface is its *absorptivity* α, which is the fraction of the radiation energy incident on a surface that is absorbed by the surface. Like emissivity, its value is in the range $0 \leq \alpha \leq 1$. A blackbody absorbs the entire radiation incident on it. That is, a blackbody is a perfect absorber $(\alpha = 1)$ as it is a perfect emitter.

In general, both ε and α of a surface depend on the temperature and the wavelength of the radiation. **Kirchhoff's law** of radiation states that the emissivity and the absorptivity of a surface are equal at the same temperature and wavelength. In most practical applications, the dependence of ε and α on the temperature and wavelength is ignored, and the average absorptivity of a surface is taken to be equal to its average emissivity. The rate at which a surface absorbs radiation is determined from (Fig. 1.2.4).

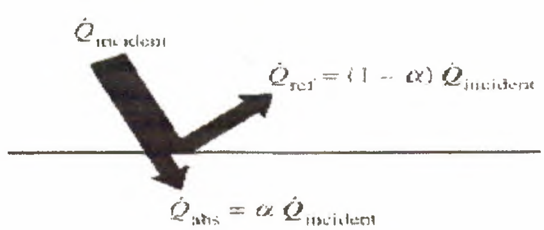


Figure 1.2.4 The absorption of radiation incident on an opaque surface of absorptivity α

$$Q_{absorbed} = \alpha Q_{incident} \qquad \text{(W)}$$

where $Q_{incident}$ is the rate at which radiation is incident on the surface and α is the absorptivity of the surface. For opaque (nontransparent) surfaces, the portion of incident radiation not absorbed by the surface is reflected back.

The difference between the rates of radiation emitted by the surface and the radiation absorbed is the *net* radiation heat transfer. If the rate of radiation absorption is greater than the rate of radiation emission, the surface is said to be *gaining* energy by radiation. Otherwise, the surface is said to be *losing* energy by radiation. In general, the determination of the net rate of heat transfer by radiation between two surfaces is a complicated matter since it depends on the properties of the surfaces, their orientation relative to each other, and the interaction of the medium between the surfaces with radiation.

When a surface of emissivity ε and surface area A at an *absolute temperature* T_s is *completely enclosed* by a much larger (or black) surface at absolute temperature T_{surr} separated by a gas (such as air) that does not intervene with radiation, the net rate of radiation heat transfer between these two surfaces is given by (Fig. 1.2.5)

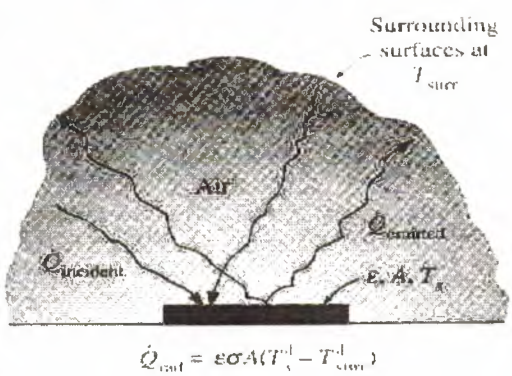


Figure 1.2.5 Radiation heat transfer between a surface and the surfaces surrounding it.

$$Q_{rad} = \varepsilon\sigma A\left(T_s^4 - T_{surr}^4\right) \qquad \text{(W)}$$

In this special cage, the emissivity and the surface area of the surrounding surface do not have any effect on the net radiation heat transfer.

1.3 Thermal Conductivity

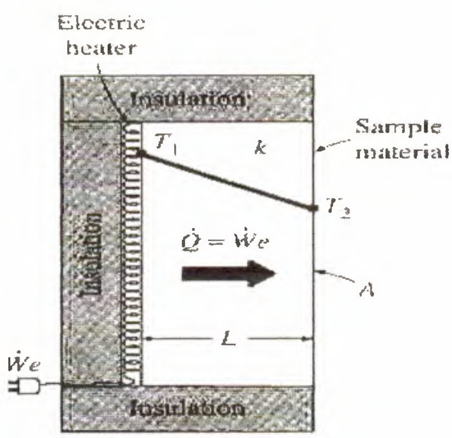
Different materials store heat differently, and the property specific heat Cp as a measure of a material's ability to store heat. For example, $Cp = 4.18$ kJ/kg . ^{0}C for water and $Cp = 0.45$ kJ/kg. ^{0}C for iron at room temperature, which indicates that water can store almost 10 times the energy that iron can per unit mass. Likewise, the thermal conductivity k is a measure of a material's ability to conduct heat. For example, $k = 0.608$ W/m. ^{0}C for water and $k = 80.2$ W/m. ^{0}C for iron at room temperature, which indicates that iron conducts heat more than 100 times faster than water can. Thus we say that water is a poor heat conductor relative to iron, although water is an excellent medium to store heat.

Equation $Q_{cond.} = kA\dfrac{\Delta T}{\Delta x}$ (W) for the rate of conduction heat transfer under steady conditions can also be viewed as the defining equation for thermal conductivity. Thus the **thermal conductivity** of a material can be defined as *the rate of heat transfer through a unit thickness of the material per unit area per unit temperature difference.* The thermal conductivity of a material is a measure of how fast heat will flow in that material. A large value for thermal conductivity indicates that the material is a good heat conductor, and a low value indicates that the material is a poor heat conductor or *insulator.* The thermal conductivities of some common materials at room temperature are given in Table 1.2. The thermal conductivity of pure copper at room temperature is $k = 401$ W/m .^{0}C, which indicates that a 1-m-thick copper wall will conduct heat at a rate of 401 W per m^2 area per ^{0}C temperature difference across the wall. Note that materials such as copper and silver that are good electric conductors are also good heat conductors, and have high values of thermal conductivity. Materials such as rubber, wood, and Styrofoam are poor conductors of heat and have low conductivity values.

Material	k, W/m · °C
Diamond	2300
Silver	429
Copper	401
Gold	317
Aluminum	237
Iron	80.2
Mercury (l)	8.54
Glass	0.78
Brick	0.72
Water (l)	0.613
Human skin	0.37
Wood (oak)	0.17
Helium (g)	0.152
Soft rubber	0.13
Refrigerant-12	0.072
Glass fiber	0.043
Air (g)	0.026
Urethane, rigid foam	0.026

Table 1.2 The thermal conductivities of some materials at room temperature.

A layer of material of known thickness and area can be heated from one side by an electric resistance heater of known output. If the outer surfaces of the heater are well insulated, all the heat generated by the resistance heater will be transferred through the material whose conductivity is to be determined. Then measuring the two surface temperatures of the material when steady heat transfer is reached and substituting them into Equation $Q_{cond.} = kA\dfrac{\Delta T}{\Delta x}$ (W) together with other known quantities give the thermal conductivity (Fig. 1.3).



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

Figure 1.3 A simple experimental setup to determine the thermal conductivity of a material.

1.4 Thermal Diffusivity

The product ρC_p, which is frequently encountered in heat transfer analysis, is called the heat capacity of a material. Both the specific heat C_p and the heat capacity ρC_p represent the heat storage capability of a material. But C_p expresses it *per unit mass* whereas ρC_p expresses it *per unit volume,* as can be noticed from their units J/kg .^{0}C and J/m^3 .^{0}C, respectively.

Another material property that appears in the transient heat conduction analysis is the **thermal diffusivity,** which represents how fast heat diffuses through a material and is defined as

$$\alpha = \frac{Heat\ conducted}{Heat\ stored} = \frac{k}{\rho C_p} \quad (\mathrm{m^2/s})$$

Note that the thermal conductivity k represents how well a material conducts heat, and the heat capacity ρC_p represents how much energy a material stores per unit volume. Therefore, the thermal diffusivity of a material can be viewed as the ratio of the *heat conducted* through the material to the *heat stored* per unit volume. A material that has a high thermal conductivity or a low heat capacity will obviously have a large thermal diffusivity. The larger the thermal diffusivity, the faster the propagation of heat into the medium. A small value of thermal diffusivity means that the material mostly absorbs heat and a small amount of heat will be conducted further.

The thermal diffusivities of some common materials at 20°C are given in Table 1.3.

Material	α, m^2/s
Silver	149×10^{-6}
Gold	127×10^{-6}
Copper	113×10^{-6}
Aluminum	97.5×10^{-6}
Iron	22.8×10^{-6}
Mercury (l)	4.7×10^{-6}
Marble	1.2×10^{-6}
Ice	1.2×10^{-6}
Concrete	0.75×10^{-6}
Brick	0.52×10^{-6}
Heavy soil (dry)	0.52×10^{-6}
Glass	0.34×10^{-6}
Glass wool	0.23×10^{-6}
Water (l)	0.14×10^{-6}
Beef	0.14×10^{-6}
Wood (oak)	0.13×10^{-6}

Table 1.3 The thermal diffusivities of some materials at room temperature.

Note that the thermal diffusivity ranges from $\alpha = 0.14 \times 10^{-6}$ m^2/s for water to 174×10^{-6} m^2/s for silver, which is a difference of more than a thousand times. Also note that the thermal diffusivities of beef and water are the same. This is not surprising, since meat as well as fresh vegetables and fruits are mostly water, and thus they possess the thermal properties of water.

1.5 Thermal insulation

Thermal insulations are *materials or combinations of materials that are used primarily to provide resistance to heat flow* (Fig. 1.5). You are probably familiar with several kinds of insulation available in the market. Most insulations are heterogeneous materials made of low thermal conductivity materials, and they involve air pockets. This is not surprising since air has one of the lowest thermal conductivities and is readily available. The *Styrofoam* commonly used as a packaging material for TVs, VCRs, computers, and just about anything because of its lightweight is also an excellent insulator.

Temperature difference is the driving force for heat flow, and the greater the temperature difference, the larger the rate of heat transfer. We can slow down the heat flow between two mediums at different temperatures by putting "barriers" on the path of heat flow. Thermal insulations serve as such barriers, and they play a major role in the design and manufacture of all energy- efficient devices or systems, and they are usually the cornerstone of all energy conservation projects.

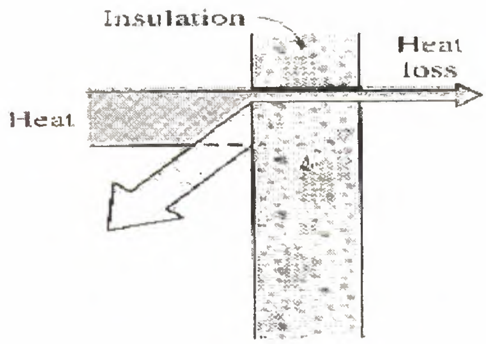


Figure 1.5 Thermal insulation retards heat transfer by acting as a barrier in the path of heat flow.

Heat is generated *in furnaces* or *heaters* by burning a fuel such as coal, oil, or natural gas or by passing electric current through a *resistance heater.* Electricity is rarely used for heating purposes since its unit cost is much higher. The heat generated is absorbed by the medium in the furnace and its surfaces, causing a temperature rise above the ambient temperature. This temperature difference drives heat transfer from the hot medium to the ambient, and insulation reduces the amount of heat loss and thus saves fuel and money. Therefore, insulation *pays for itself* from the energy it saves. Insulating properly requires a one-time capital investment, but its effects are dramatic and long term. The payback period of insulation is usually under two years. That is, the money insulation saves during the first two years is usually greater than its initial material and installation costs. On a broader perspective, insulation also helps the environment and fights air pollution and the greenhouse effect by reducing the amount of fuel burned and thus the amount of CO2 and other gases released into the atmosphere (Fig. 1.5.1).

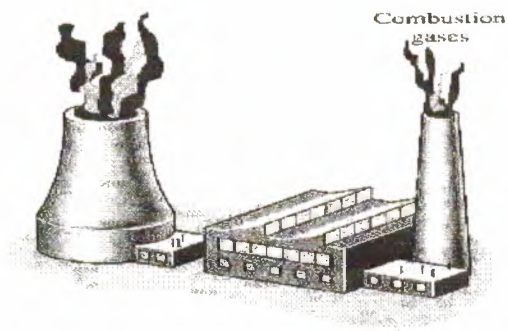


Figure 1.5.1 Insulation also helps the environment by reducing the amount of fuel burned and the air pollutants released.

Saving energy with insulation is not limited to hot surfaces. We can also save energy and money by insulating *cold surfaces* (surfaces whose temperature is below the ambient temperature) such as chilled water lines, cryogenic storage tanks, refrigerated trucks, and air-conditioning ducts. The source of "coldness" is *refrigeration,* which requires energy input, usually electricity. In this case, heat is transferred from the surroundings to the cold surfaces, and the refrigeration unit must now work harder and longer to make up for this heat gain and thus it must consume more electrical energy. A cold canned drink can be kept cold much longer by wrapping it in a blanket. A refrigerator with well-insulated walls will consume much less electricity than a similar refrigerator with little or no insulation. Insulating a house well will result in reduced cooling load, and thus reduced electricity consumption for air- conditioning.

1.5.1 Reasons for Insulating

If you examine the engine compartment of your car, you will notice that the firewall between the engine and the passenger compartment as well as the inner surface of the hood is insulated. The reason for insulating the hood is not to conserve the waste heat from the engine but to protect people from burning themselves by touching the hood surface, which will be too hot if not insulated. As this example shows, the use of insulation is not limited to energy conservation. Various reasons for using insulation can be summarized as follows:

Energy Conservation Conserving energy by reducing the rate of heat flow is the primary reason for insulating surfaces. Insulation materials that will perform satisfactorily in the temperature range of -268°C to 1000°C (-450 ^{0}F to 1800 ^{0}F) are widely available.

Personnel Protection and Comfort A surface that is too hot poses a danger to people who are working in that area of accidentally touching the hot surface and burning themselves (Fig. 1.5.2). To prevent this danger and to comply with the OSHA (Occupational Safety and Health Administration) standards, the temperatures of hot surfaces should be reduced to below 60°C (140 ^{0}F) by insulating them. Also, the excessive heat coming off the hot surfaces creates an unpleasant environment in which to work, which adversely affects the performance or productivity of the workers, especially in summer months.

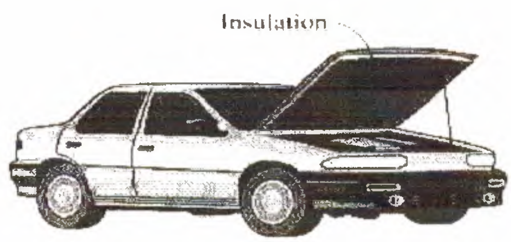


Figure 1.5.2 The hood of the engine compartment of a car is insulated to reduce its temperature and to protect people from burning themselves.

Maintaining Process Temperature Some processes in chemical industry are temperature-sensitive, and it may become necessary to insulate the process tanks and flow sections heavily to maintain the same temperature throughout

Reducing Temperature Variation and Fluctuations The temperature in an enclosure may vary greatly between the midsection and the edges if the enclosure is not insulated. For example, the temperature near the walls of a poorly insulated house is much lower than the temperature at the midsections. Also, the temperature in an uninsulated enclosure will follow the temperature changes in the environment closely and fluctuate. Insulation minimizes temperature nonuniformity in an enclosure and slows down fluctuations.

Condensation and Corrosion Prevention Water vapor in the air condenses on surfaces whose temperature is below the dew point, and the outer surfaces of the tanks or pipes that contain a cold fluid frequently fall below the dew-point temperature unless they have adequate insulation. The liquid water on exposed surfaces of the metal tanks or pipes may promote corrosion as well as algae growth.

Keeping valuable combustibles in a safety box that is well insulated may minimize fire Protection Damage during a fire. Insulation may lower the rate of heat flow to such levels that the temperature in the box never rises to unsafe levels during fire.

Freezing Protection Prolonged exposure to subfreezing temperatures may cause water in pipes or storage vessels to freeze and burst as a result of heat transfer from the water to the cold ambient. Thy bursting of pipes as a result of freezing can cause considerable damage. Adequate insulation will slow down the heat loss from the water and prevent freezing during limited exposure to subfreezing temperatures. For example, covering vegetables during a cold night will protect them from freezing, and burying water pipes in the ground at a sufficient depth will keep them from freezing during the entire winter. Wearing thick gloves will protect the fingers from possible frostbite. Also, a molten metal or plastic in a container will solidify on the inner surface if the container is not properly insulated.

Reducing Noise and Vibration An added benefit of thermal insulation is its ability to dampen noise and vibrations (Fig. 1.5.3). The insulation materials differ in their ability to reduce noise and vibration, and the proper kind can be selected if noise reduction is an important consideration.

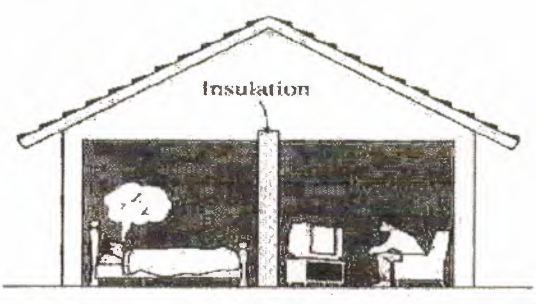


Figure 1.5.3 Insulation material absorb vibration and sound waves, and are used to minimize sound transmission.

1.5.2 Superinsulators

You may be tempted to think that the most effective way to reduce heat transfer is to use insulating materials that are known to have very low thermal conductivities such as urethane or rigid foam (k = 0.026 W/m.^{0}C) or fiberglass s (k = 0.035 W/m.^{0}C). After all, they are widely available, inexpensive, and easy to install. Looking at the thermal conductivities of materials, you may also notice that the thermal conductivity of air at room temperature is 0.026 W/m.^{0}C, which is lower than the conductivities of practically all of the ordinary insulating materials. Thus you may think that a layer of enclosed air space is as effective as any of the common insulating materials of the same thickness. Of course, heat transfer through the air will probably be higher than what a pure conduction analysis alone would indicate because of the natural convection currents that are likely to occur in the air layer. Besides, air is transparent to radiation, and thus heat will also be lost from the surface by radiation. The thermal conductivity of air is practically independent of pressure unless the pressure is extremely high or extremely low. Therefore, we can reduce the thermal conductivity of air and thus the conduction heat transfer through the air by evacuating the air space. In the limiting cage of absolute vacuum, the thermal conductivity will be zero since there will be no particles in this case to "conduct" heat from one surface to the other, and thus the conduction heat transfer will be zero. Noting that the thermal conductivity cannot be negative, an absolute vacuum must be the ultimate insulator.

The purpose of insulation is to reduce "total" heat transfer from a surface, not just conduction. A vacuum totally eliminates conduction but offers zero resistance to radiation, whose magnitude can be comparable to conduction or natural convection in gases. Thus, a vacuum is no more effective in reducing heat transfer than sealing off one of the lanes of a two-lane road is in reducing the flow of traffic in a one-way road.

Insulation against radiation heat transfer between two surfaces is achieved by placing "barriers" between the two surfaces, which are highly reflective thin metal sheets. Radiation heat transfer between two surfaces is inversely proportional to the number of such sheets placed between the surfaces. Very effective insulations are obtained by using closely packed layers of highly reflective thin metal sheets such as aluminum foil separated by fibers made of insulating material such as glass fiber. The result is an insulating material which is one thousand times less than the conductivity of air or any common insulating material. These specially built insulators are called **superinsulators,** and they are commonly used in space applications.

SUMMARY

In this chapter, the basic concept of heat transfer is introduced and discussed. The science of *heat transfer* deals with the rate of heat transfer, which is the main quantity of interest in the design and evaluation of heat transfer equipment. The sensible and latent forms of internal energy can be transferred from one medium to another as a result of temperature difference, and are referred to as *heat* or *thermal energy*. Thus, *heat transfer* is the exchange of the sensible and latent forms of internal energy between two mediums as a result of temperature difference. The amount of heat transferred per unit time is called *heat transfer rate* and is denoted by Q. The rate of heat transfer per unit surface area is called *heat flux*, q.

Heat can be transferred in three different ways: Conduction, convection, and radiation. *Conduction* is the transfer of energy from the more energetic particles of a substance to the adjacent less energetic ones as a result of interactions between the particles, and is expressed by *Fourier' s law of heat conduction.*

Convection is the mode of heat transfer between a solid surface and the adjacent liquid or gas that is in motion, and involves the combined effects of conduction and fluid motion. Newton's law of cooling expresses the rate of convection heat transfer.

Radiation is the energy emitted by matter in the form of electromagnetic waves (or photons) as a result of the changes in the electronic configurations of the atoms or molecules. The Stefan-Boltzmann law gives the maximum rate of radiation that can be emitted from a surface at an absolute temperature Ts.

Thermal insulations are materials or a combination of materials that are used primarily to provide resistance to heat flow. Thermal insulations are used for various reasons, such as energy conservation, personnel protection and comfort, maintenance of process temperature, reduction of temperature variation and fluctuations, condensation and corrosion prevention, fire protection, freezing protection, and reduction of noise and vibration. Insulation materials are classified as fibrous, cellular, granular, and reflective, and they are available in the form of loose-fill, blankets and batts, rigid insulations, insulating cements, formed-in-place insulations, and reflective insulations. Important considerations in the selection of insulations are the purpose, environment, ease of handling and installation, and cost. Optimum thickness of insulation is usually determined on the basis of minimum combined cost of insulation and heat lost.

CHAPTER 2

CONDUCTION

2. Introduction

2.1 Steady Heat Conduction In Plane Walls

Consider steady heat conduction through the walls of a house during a winter day. We know that heat is continuously lost to the outdoors through the wall. We intuitively feel that heat transfer through the wall is in the *normal direction* to the wall surface, and no significant heat transfer takes place in the wall in other directions (Fig. 2.1).

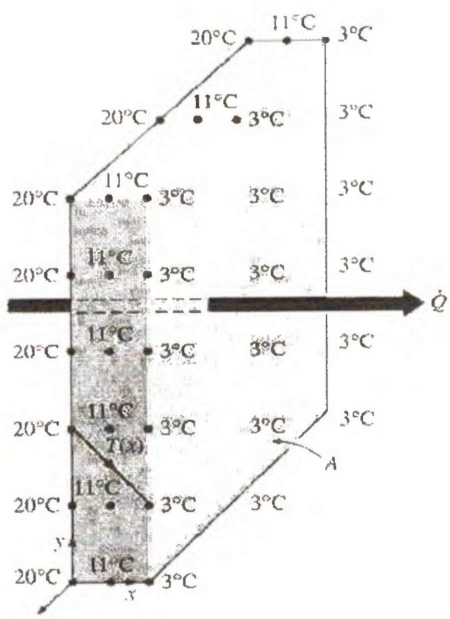


Figure 2.1 Heat flow through a wall is one-dimensional when the temperature of the wall varies in one direction only.

Recall that heat transfer in a certain direction is driven by the *temperature gradient* in that direction. There will be no heat transfer in a direction in which there is no change in temperature. Temperature measurements at several locations on the inner or outer wall surface will confirm that a wall surface is nearly *isothermal.* That is, the temperatures at the top and bottom of a wall surface as well as at the right or left ends are almost the same. Therefore, there will be no heat transfer through the wall from the top to the bottom, or from left to right, but there will be considerable temperature difference between the inner and the outer surfaces of the wall, and thus significant heat transfer in the direction from the inner surface to the outer one.

The small thickness of the wall causes the temperature gradient in that direction to be large. Further, if the air temperatures in and outside the house remain constant, then heat transfer through the wall of a house can be modeled as *steady* and *one-dimensional.* The temperature of the wall in this case will depend on one direction only (say the x-direction) and can be expressed as $T(x)$.

Noting that heat transfer is the only energy interaction involved in this case and there is no heat generation, the *energy balance* for the wall can be expressed as (Rate of heat transfer into the wall) – (Rate of heat transfer out of the wall) = (Rate of change of the energy of the wall)

or,

$$Q_{in} - Q_{out} = \frac{dE_{wall}}{dt}$$

But $dE_{wall}/dt = 0$ for *steady* operation, since there is no change in the temperature of the wall with time at any point. Therefore, the rate of heat transfer into the wall must be equal to the rate of heat transfer out of it. In other words, *the rate of heat transfer through the wall must be constant,* $Q_{cond,\ wall}$ = constant.

Consider a plane wall of thickness L and average thermal conductivity k. The two surfaces of the wall are maintained at constant temperatures of T_1 and T_2. For one-dimensional steady heat conduction through the wall, we have $T(x)$. Then Fourier's law of heat conduction for the wall can be expressed as

$$Q_{cond,wall} = -kA\frac{dT}{dx} \quad \text{(W)}$$

where the rate of conduction heat transfer $Q_{cond,wall}$, wall and the surface area A are constant, Thus we have dT/dx = constant, which means that *the temperature through the wall varies linearly with x.* That is, the temperature distribution in the wall under steady conditions is a *straight line* (Fig. 2.2).

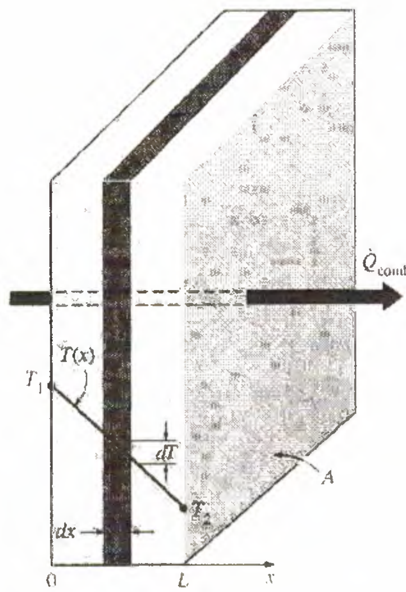


Figure 2.2 Under steady conditions, the temperature distribution in a plane wall is a straight line.

Separating the variables in the above equation and integrating from

$x = 0$, where $T(0) = T_1$, to $x = L$, where $T(L) = T_2$, we get

$$\int_{x=0}^{L} Q_{cond,wall}\,dx = -\int_{T=T_1}^{T_2} kA\,dT$$

Performing the integration and rearranging gives

$$Q_{cond,wall} = kA\frac{T_1 - T_2}{L} \text{ (W)}$$

which is identical to Equation $Q_{cond.} = kA\frac{\Delta T}{\Delta x}$. Again, *the rate of heat conduction through a plane wall is proportional to the average thermal conductivity, the wall area, and the temperature difference, but is inversely proportional to the wall thickness.* Also, once the rate of heat conduction is available, the temperature $T(x)$ at any location x can be determined by replacing T_2 in Equation $Q_{cond,wall} = kA\frac{T_1 - T_2}{L}$ by T, and L by x.

2.1.1 The Thermal Resistance Concept

Equation $\dot{Q}_{cond,wall} = kA\dfrac{T_1 - T_2}{L}$ for heat conduction through a plane wall can be rearranged as

$$\dot{Q}_{cond,wall} = \frac{T_1 - T_2}{R_{wall}} \qquad \text{(W)}$$

where

$$R_{wall} = \frac{L}{kA} \qquad (°\text{C/W})$$

is the *thermal resistance* of the wall against heat conduction or simply the conduction resistance of the wall. Note that the thermal resistance of a medium depends on the *geometry* and the *thermal properties* of the medium.

The equation above for heat flow is analogous to the relation for *electric current flow **I***, expressed as

$$I = \frac{V_1 - V_2}{R_e}$$

where $Re = L/\sigma_e A$ is the *electric resistance* and $V_1 - V_2$ is the *voltage difference* across the resistance (σ_e is the electrical conductivity). Thus, the *rate of heat transfer* through a layer corresponds to the *electric current*, the *thermal resistance* corresponds to *electrical resistance*, and the *temperature difference* corresponds to *voltage difference* across the layer (Fig. 2.3).

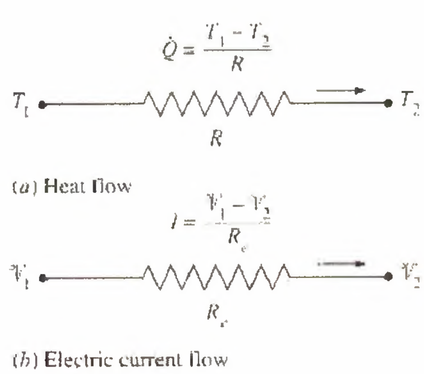


Figure 2.3 Analogy between thermal and electrical resistance concepts.

Consider convection heat transfer from a solid surface of area A and temperature T_s to a fluid whose temperature sufficiently far from the surface is T_{∞}, with a convection heat transfer coefficient h. Newton's law of cooling for convection heat transfer rate $Q_{conv}=hA\ (T_s - T_{\infty})$ can be rearranged as

$$Q_{cond,wall} = \frac{T_s - T_{\infty}}{R_{conv}} \qquad \text{(W)}$$

where

$$R_{conv} = \frac{1}{hA} \qquad (^{\circ}\text{C/W})$$

is the *thermal resistance* of the surface against heat convection, or simply the **convection resistance** of the surface (Fig. 2.4).

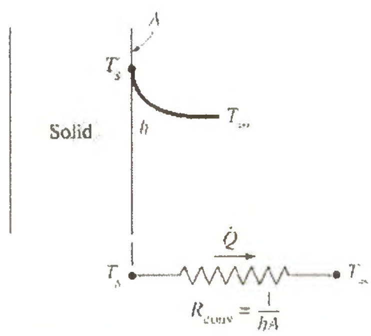


Figure 2.4 Schematic for convection resistance at a surface.

Note that when the convection heat transfer coefficient is very large ($h \to \infty$), the convection resistance becomes *zero* and $Ts \approx T\infty$. That is, the surface offers *no resistance to convection,* and thus it does not slow down the heat transfer process. This situation is approached in practice at surfaces where boiling and condensation occur. Also note that the surface does not have to be a plane surface. Equation $R_{conv} = \dfrac{1}{hA}$ for convection resistance is valid for surfaces of any shape, provided that the assumption of h = constant and uniform is reasonable.

When a gas surrounds the wall, the *radiation effects,* which we have ignored so far, can be significant and may need to be considered. The rate of radiation heat transfer between a surface of emissivity ε and area A at temperature T_s and the surrounding surfaces at some average temperature T_{surr} can be expressed as

$$Q_{rad} = \varepsilon\sigma A(T_s^4 - T_{surr}^4) = h_{rad}A(T_s - T_{surr}) = \frac{T_s - T_{surr}}{R_{rad}} \qquad \text{(W)}$$

where

$$R_{rad} = \frac{1}{h_{rad}A} \qquad \text{(K/W)}$$

is the *thermal resistance* of a surface against radiation, or the *radiation resistance,* and

$$h_{rad} = \frac{Q_{rad}}{A(T_s - T_{surr})} = \varepsilon\sigma(T_s^2 + T_{surr}^2)(T_s + T_{surr}) \qquad \text{(W/m}^2\text{.K)}$$

is the radiation heat transfer coefficient. Note that both T_s and T_{surr} *must* be in K in the evaluation of h_{rad}. The definition of the radiation heat transfer coefficient enables us to express radiation conveniently in an analogous manner to convection in terms of a temperature difference. But h_{rad} depends strongly on temperature while h_{conv} usually does not.

A surface exposed to the surrounding air involves convection and radiation simultaneously, and the total heat transfer at the surface is determined by adding (or subtracting, if in the opposite direction) the radiation and convection components. The convection and radiation resistances are parallel to each other, as shown in Figure 2.5, and may cause some complication in the thermal resistance network.

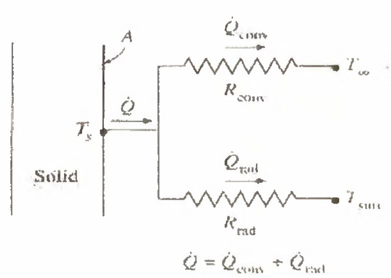


Figure 2.5 Schematic for convection and radiation resistance at a surface.

When $T_{surr} \approx T_\infty$, the radiation effect can properly be accounted for by replacing h in the convection resistance relation by

$$h_{combined} = h_{conv} + h_{rad} \qquad \text{(W /m2 . K)}$$

where $h_{combined}$ is the combined heat transfer coefficient. This way all the complications associated with radiation are avoided.

2.1.2 Thermal Resistance Network

Now consider steady one-dimensional heat flow through a plane wall of thickness L and thermal conductivity k that is exposed to convection on both sides to fluids at temperatures $T_{\infty 1}$ and $T_{\infty 2}$ with heat transfer coefficients h_1 and h_2, respectively, as shown in Figure 2.6.

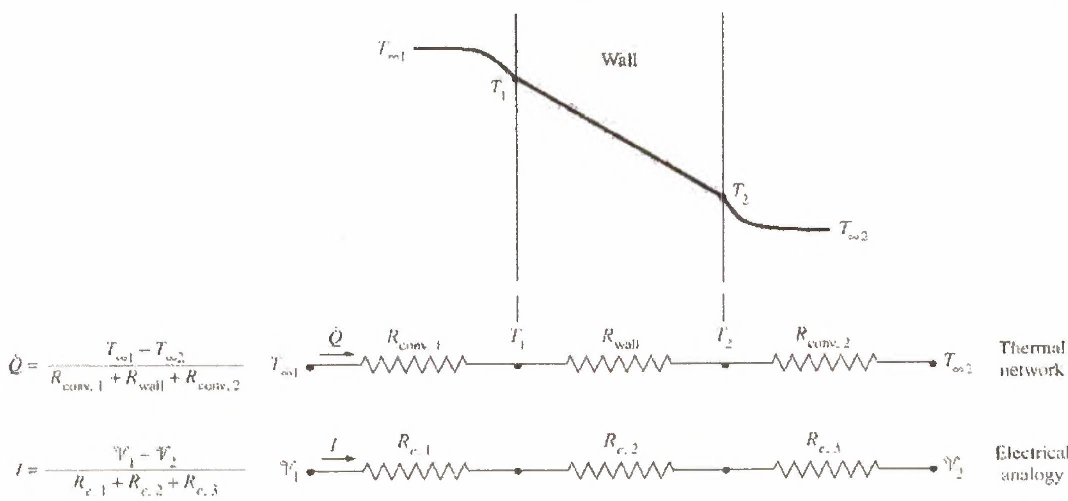


Figure 2.6 The thermal resistance network for heat transfer through a plane wall subjected to convection on both sides, and the electrical analogy.

Assuming $T_{\infty 2} < T_{\infty 1}$, the variation of temperature will be as shown in the figure. Note that the temperature varies linearly in the wall, and asymptotically approaches $T_{\infty 1}$ and $T_{\infty 2}$ in the fluids as we move away from the wall.

Under steady conditions we have (Rate of convection into wall) = (Rate of conduction through wall) =(Rate of convection from the wall)

or

$$Q = h_1 A(T_{\infty 1} - T_1) = kA\frac{T_1 - T_2}{L} = h_2 A(T_2 - T_{\infty 2})$$

which can be rearranged as

$$Q = \frac{T_{\infty 1} - T_1}{1/h_1 A} = \frac{T_1 - T_2}{L/kA} = \frac{T_2 - T_{\infty 2}}{1/h_2 A} = \frac{T_{\infty 1} - T_1}{R_{conv,1}} = \frac{T_1 - T_2}{R_{wall}} = \frac{T_2 - T_{\infty 2}}{R_{conv,2}}$$

Adding the numerators and denominators yields

$$Q = \frac{T_{\infty 1} - T_{\infty 2}}{R_{total}} \qquad \text{(W)}$$

where

$$R_{total} = R_{conv,1} + R_{wall} + R_{conv,2} = \frac{1}{h_1 A} + \frac{L}{kA} + \frac{1}{h_2 A} \qquad ({}^0\text{C/W})$$

Note that the heat transfer area A is constant for a plane wall, and the rate of heat transfer through a wall separating two mediums is equal to the temperature difference divided by the total thermal resistance between the mediums. Also note that the thermal resistances are in *series,* and the equivalent thermal resistance is determined by simply *adding* the individual resistances, just like the electrical resistances connected in series. Thus, the electrical analogy still applies. We summarize this as *the rate of steady heat transfer between two surfaces is equal to the temperature difference divided by the total thermal resistance between those two surfaces.*

Another observation that can be made from Equation $Q = \frac{T_{\infty 1} - T_{\infty 2}}{R_{total}}$ is that the ratio of the temperature drop to the thermal resistance across any layer is constant, and thus the temperature drop across any layer is proportional to the thermal resistance of the layer. The larger the resistance, the larger the temperature drop. In fact, the equation $Q = \Delta T/R$ can be rearranged as

$$\Delta T = QR \qquad ({}^0\text{C})$$

which indicates that the *temperature drop* across any layer is equal to the *rate of heat transfer* times the *thermal resistance* across that layer (Fig. 2.7).

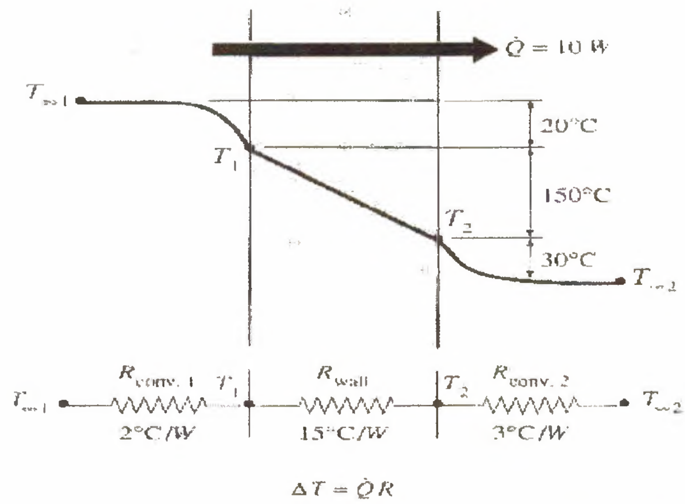


Figure 2.7 The temperature drop across a layer is proportional to its thermal resistance.

You may recall that this is also true for voltage drop across an electrical resistance when the electric current is constant.

It is sometimes convenient to express heat transfer through a medium in an analogous manner to Newton's law of cooling as

$$Q = UA\Delta T \qquad \text{(W)}$$

where U is the **overall heat transfer coefficient.** A comparison of Equations

$$Q = \frac{T_{\infty 1} - T_{\infty 2}}{R_{total}}$$ and $Q = UA\Delta T$ reveals that

$$UA = \frac{1}{R_{total}}$$

Therefore, for a unit area, the overall heat transfer coefficient is equal to the inverse of the total thermal resistance.

Note that we do not need to know the surface temperatures of the wall in order to evaluate the rate of steady heat transfer through it. All we need to know is the convection heat transfer coefficients and the fluid temperatures on both sides of the wall. The *surface temperature* of the wall can be determined as described above using the thermal resistance concept, but by taking the surface at which the temperature is to be determined as one of the terminal surfaces. For example, once Q is evaluated, the surface temperature T_1 can be determined from

$$Q = \frac{T_{\infty 1} - T_1}{R_{conv,1}} = \frac{T_{\infty 1} - T_1}{1/h_1 A}$$

2.1.3 Generalized Thermal Resistance Networks

The *thermal resistance* concept or the *electrical analogy* can also be used to solve steady heat transfer problems that involve parallel layers or combined series-parallel arrangements. Although such problems are often two- or even three-dimensional, approximate solutions can be obtained by assuming one- dimensional heat transfer and using the thermal resistance network.

Consider the composite wall shown in Figure 2.10, which consists of two parallel layers. The thermal resistance network, which consists of two parallel resistances, can be represented as shown in the figure.

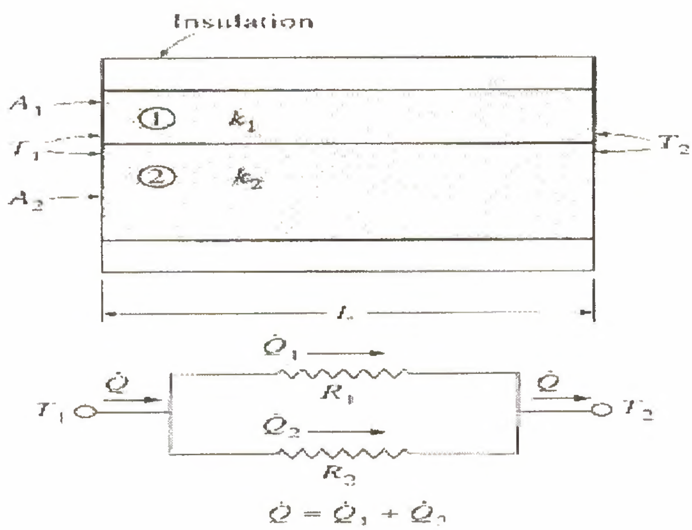


Figure 2.10 Thermal resistance network for two parallel layers.

Noting that the total heat transfer is the sum of the heat transfers through each layer, we have

$$Q = Q_1 + Q_2 + = \frac{T_2 - T_1}{R_1} + \frac{T_1 - T_2}{R_2} = (T_1 - T_2)\left(\frac{1}{R_1} + \frac{1}{R_2}\right)$$

Utilizing electrical analogy, we get

$$Q = \frac{T_1 - T_2}{R_{total}}$$

where

$$\frac{1}{R_{total}} = \frac{1}{R_1} + \frac{1}{R_2} \rightarrow R_{total} = \frac{R_1 R_2}{R_1 + R_2}$$

since the resistances are in parallel.

Now consider the combined series-parallel arrangement shown in Figure 2.11.

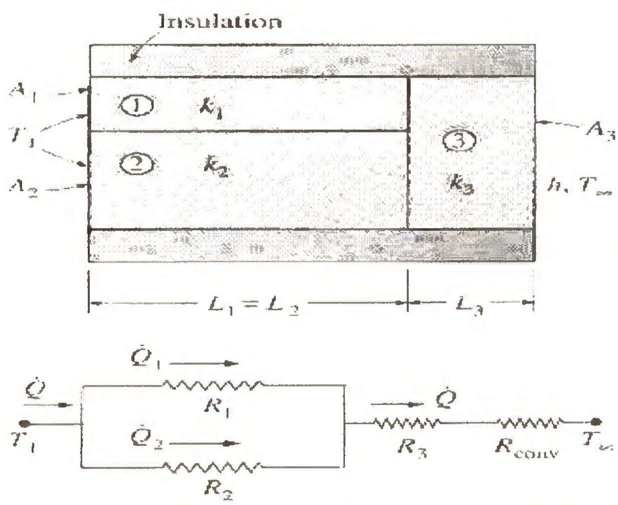


Figure 2.11 Thermal resistance network for combined series – parallel arrangement.

The total rate of heat transfer through this composite system can again be expressed as

$$Q = \frac{T_1 - T_\infty}{R_{total}}$$

where

$$R_{total} = R_{12} + R_3 + R_{conv} = \frac{R_1 R_2}{R_1 + R_2} + R_3 + R_{conv}$$

and

$$R_1 = \frac{L_1}{k_1 A_1}, \quad R_2 = \frac{L_2}{k_2 A_2}, \quad R_3 = \frac{L_3}{k_3 A_3}, \quad R_{conv} = \frac{1}{hA_3}$$

Once the individual thermal resistances are evaluated, the total resistance and the total rate of heat transfer can easily be determined from the relations above.

The result obtained will be somewhat approximate, since the surfaces of the third layer will probably not be isothermal, and heat transfer between the first two layers is likely to occur.

Two assumptions commonly used in solving complex multidimensional heat transfer problems by treating them as one-dimensional (say, in the x-direction) using the thermal resistance network are (1) any plane wall normal to the x-axis is *isothermal* (i.e., to assume the temperature to vary in the x-direction only) and (2) any plane parallel to the x-axis is *adiabatic* (i.e., to assume heat transfer to occur in the x-direction only). These two assumptions result in different resistance networks, and thus different (but usually close) values for the total thermal resistance and thus heat transfer. The actual result lies between these two values. In geometries in which heat transfer occurs predominantly in one direction, either approach gives satisfactory results.

SUMMARY

In this chapter, one-dimensional heat transfer through a simple or composite body exposed to convection from both sides to mediums at temperatures $T_{\infty 1}$, and $T_{\infty 2}$, can be expressed as

$$Q = \frac{T_{\infty 1} - T_{\infty 2}}{R_{total}} \qquad \text{(W)}$$

where R_{total} is the total thermal resistance between the two mediums. For a plane wall exposed to convection on both sides, the total resistance is expressed as

$$R_{total} = R_{conv,1} + R_{wall} + R_{conv,2} = \frac{1}{h_1 A} + \frac{L}{kA} + \frac{1}{h_2 A} \qquad (^0\text{C/W})$$

This relation can be extended to plane walls that consist of two or more layers by adding an additional resistance for each additional layer. The elementary thermal resistance relations can be expressed as follows:

Conduction resistance (plane wall): $R_{wall} = \dfrac{L}{kA}$

Once the rate of heat transfer is available, the *temperature drop* across any layer can be determined from

$$\Delta T = QR \qquad (^0\text{C})$$

The thermal resistance concept can also be used to solve steady heat transfer problems involving parallel layers or combined series-parallel arrangements.

CHAPTER 3

NATURAL CONVECTION

3. Introduction

3.1 Physical Mechanism Of Natural Convection

Many familiar heat transfer applications involve *natural convection* as the *primary* mechanism of heat transfer. Some examples are cooling of electronic equipment such as power transistors, TVs, and VCRs; heat transfer from electric baseboard heaters or steam radiators; heat transfer from the refrigeration coils and power transmission lines; and heat transfer from the bodies of animals and human beings. Natural convection in gases is usually accompanied by radiation of comparable magnitude except for low-emissivity surfaces.

We know that a hot *boiled egg* (or a hot *baked potato)* on a plate eventually cools to the surrounding air temperature (Fig. 3.1). The egg is cooled by transferring heat by *convection* to the air and by radiation to the surrounding surfaces. Disregarding heat transfer by radiation, the physical mechanism of cooling a hot egg (or any hot object) in a cooler environment can be explained as follows:

As soon as the hot egg is exposed to cooler air, the temperature of the outer surface of the eggshell will drop somewhat, and the temperature of the air adjacent to the shell will rise as a result of heat conduction from the shell to the air. Consequently, a thin layer of warmer air will soon surround the egg, and heat will then be transferred from this warmer layer to the outer layers of air. The cooling process in this case would be rather *slow* since the egg would always be *blanketed* by warm air, and it would have no direct contact with the cooler air farther away. We may not notice any *air motion* in the vicinity of the egg, but careful measurements indicate otherwise.

The temperature of the air adjacent to the egg is higher, and thus its density is lower, since at constant pressure the *density* of a gas is inversely proportional to its *temperature*. Thus, we have a situation in which a high-density or “heavy” gas surrounds some low-density or “light” gas, and the natural laws dictate that *the light gas rise*. This is no different than the oil in a vinegar-and-oil salad dressing rising to the top (note that $\rho_{oil} < \rho_{vinegar}$). This phenomenon is characterized incorrectly by the phrase "heat rises," which is understood to mean *heated air rises*. The cooler air nearby replaces the space vacated by the warmer air in the vicinity of the egg, and the presence of cooler air in the vicinity of the egg speeds up the cooling process. The rise of warmer air and the flow of cooler air into its place continue until the egg is cooled to the temperature of the surrounding air. The motion that results from the continual replacement of the heated air in the vicinity of the egg by the cooler air nearby is called a **natural convection current**, and the heat transfer that is enhanced as a result of this natural convection current is called **natural convection heat transfer**. Note that in the absence of natural convection currents, heat transfer from the egg to the air surrounding it would be by *conduction* only, and the rate of heat transfer from the egg would be much *lower*.

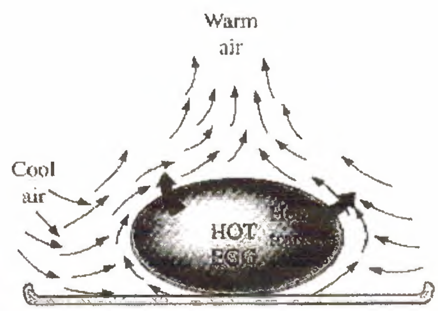


Figure 3.1 The cooling of a boiled egg in a cooler environment by natural convection.

Natural convection is just as effective in the heating of cold surfaces in warmer environment as it is in the cooling of hot surfaces in a cooler environment, as shown in Figure 3.2. Note that the direction of fluid motion is reversed in this case.

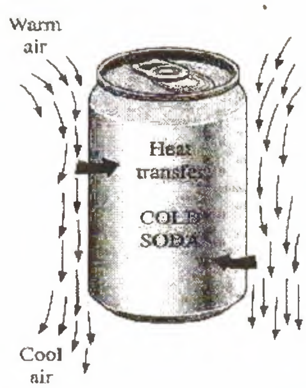


Figure 3.2 The warming up of a cold drink in a warmer environment by natural convection.

In a gravitational field, there seems to be a net force that pushes upward a light fluid placed in a heavier fluid. The upward force exerted by a fluid on a body completely or partially immersed in it is called the buoyancy force. The magnitude of the buoyancy force is equal to the weight of the *fluid displaced* by the body. That is,

$$F_{buoyancy} = \rho_{fluid}\, g V_{body}$$

where ρ_{fluid} is the average density of the *fluid* (not the body), g is the gravitational acceleration, and V_{body} is the volume of the portion of the body immersed in the fluid (for bodies completely immersed in the fluid, it is the total voltime of the body). In the absence of other forces, the net vertical force acting on a body is the difference between the weight of the body and the buoyancy force. That is,

$$\begin{aligned} F_{net} &= W - F_{buoyancy} \\ &= \rho_{body}\, g V_{body} - \rho_{fluid}\, g V_{bod} \\ &= (\rho_{body} - \rho_{fluid})\, g V_{body} \end{aligned}$$

Note that this force is *proportional* to the difference in the *densities* of the fluid and the body immersed in it. Thus, a body immersed in a fluid will experience a "weight loss" in an amount equal to the weight of the fluid it displaces. This is known as *Archimedes' principle.*

To have a better understanding of the buoyancy effect, consider an egg dropped into water. If the average density of the egg is greater than the density of water (a sign of freshness), the egg will settle at the bottom of the container. Otherwise, it will rise to the top. When the density of the egg equals the density of water, the egg will settle somewhere in the water while remaining completely immersed, acting like a "weightless object" in space. This occurs when the upward buoyancy force acting on the egg equals the weight of the egg, which acts downward.

The *buoyancy effect* has far-reaching implications in life. For one thing, without buoyancy, heat transfer between a hot (or cold) surface and the fluid surrounding it would be by *conduction* instead of by *natural convection.* The natural convection currents encountered in the oceans, lakes, and the atmosphere owes their existence to buoyancy. Also, light boats as well as heavy warships made of steel float on water because of buoyancy (Fig. 3.3).

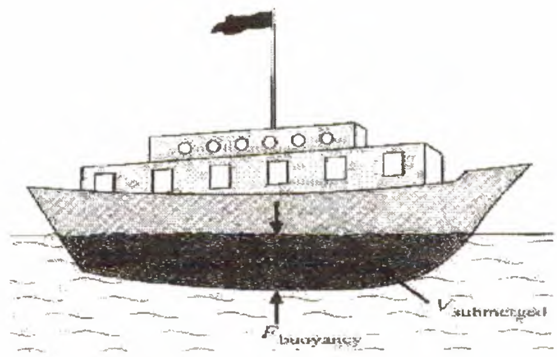


Figure 3.3 It is the buoyancy force that keeps the ships afloat in water,

($W=F_{Buoyancy}$ for floating objects)S

Ships are designed on the basis of the principle that the entire weight of a ship and its contents is equal to the weight of the water that the submerged volume of the ship can contain. (Note that a larger portion of the hull of a ship will sink in fresh water than it does in salty water.) The "chimney effect" that induces the upward flow of hot combustion gases through a chimney is also due to the buoyancy effect, and the upward force acting on the gases in the chimney is proportional to the difference between the densities of the hot gases in the chimney and the cooler air outside. Note that there is *no* *gravity* in space, and thus there will be no natural convection heat transfer in a spacecraft, even if the spacecraft is filled with atmospheric air.

In heat transfer studies, the primary variable is *temperature,* and it is desirable to express the net buoyancy force (Eq. $F_{\text{net}} = W - F_{\text{buoyancy}}$) in terms of temperature differences.

We can show easily that the volume expansion coefficient β of an *ideal gas* ($P = \rho RT$) at a temperature T is equivalent to the inverse of the temperature:

$$\beta_{ideal\,gas} = \frac{1}{T} \qquad (1/\mathrm{K})$$

where T is the ***absolute*** temperature. Note that a large value of β for a fluid means a large change in density with temperature, and that the product $\beta\Delta T$ represents the fraction of volume change of a fluid that corresponds to a temperature change ΔT at constant pressure. Also note that the buoyancy force is proportional to the *density difference,* which is proportional to the *temperature difference* at constant pressure. Therefore, the larger the temperature difference between the fluid adjacent to a hot (or cold) surface and the fluid away from it, the *larger* the buoyancy force and the *stronger* the natural convection currents, and thus the *higher* the heat transfer rate.

The magnitude of the natural convection heat transfer between a surface and a fluid is directly related to the *mass flow rate* of the fluid. The higher the mass flow rate, the higher the heat transfer rate. In fact, it is the very high flow rates that increase the heat transfer coefficient by orders of magnitude when forced convection is used. In natural convection, no blowers are used, and therefore the flow rate cannot be controlled externally. The dynamic balance of buoyancy and friction in this case establishes the flow rate.

As we have discussed earlier, the *buoyancy force* is caused by the density difference between the heated (or cooled) fluid adjacent to the surface and the fluid surrounding it, and does the warmer fluid occupy proportional to this density difference and the volume. It is also well known that whenever two bodies in contact (solid-solid, solid-fluid, or fluid-fluid) move relative to each other, a *friction force* develops at the contact surface in the direction opposite to that of the motion. This opposing force slows down the fluid and thus reduces the flow rate of the fluid. Under steady conditions, the airflow rate driven by buoyancy is established at the point where these two effects *balance* each other. The friction force increases as more and more solid surfaces are introduced, seriously disrupting the fluid flow and heat transfer. For that reason, heat sinks with closely spaced fins are not suitable for natural convection cooling.

Most heat transfer correlations in natural convection are based on experimental measurements. The instrument used in natural convection experiments most often is the *Mach-Zehnder interferometer*, which gives a plot of isotherms in the fluid in the vicinity of a surface. The operation principle of interferometers is based on the fact that at low pressure, the lines of constant temperature for a gas correspond to the lines of constant density, and that the index of refraction of a gas is a function of its density. Therefore, the degree of refraction of light at some point in a gas is a measure of the temperature gradient at that point. An interferometer produces a map of interference fringes, which can be interpreted as lines of *constant temperature* as shown in Figure 3.4. The smooth and parallel lines in *(a)* indicate that the flow is *laminar*, whereas the eddies and irregularities in *(b)* indicate that the flow is *turbulent*.

Note that the lines are closest near the surface, indicating a *higher temperature gradient*.

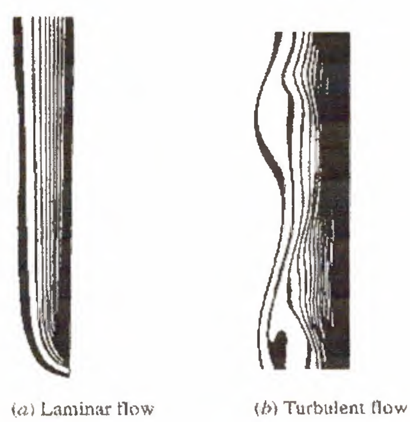


Figure 3.4 Isotherms in natural convection over a hot plate in air.

The Grashof Number

We mentioned in the preceding chapter that the flow regime in forced convection is governed by the dimensionless *Reynolds number,* which represents the ratio of inertial forces to viscous forces acting on the fluid. The flow regime in natural convection is governed by another dimensionless number, called the Grashof number, which represents the ratio of the *buoyancy force* to the *viscous force* acting on the fluid. That is,

$$Gr = \frac{Buoyancy\ force}{Viscous\ force} = \frac{g\Delta\rho V}{\rho \upsilon^2} = \frac{g\beta\Delta TV}{\upsilon^2}$$

Since $\Delta\rho = \rho\beta\Delta T$, it is formally expressed as (Fig. 3.5)

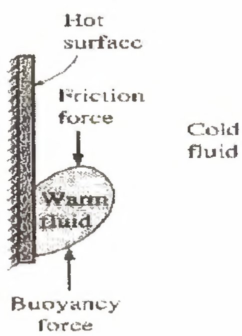


Figure 3.5 The Grashof number Gr is a measure of the relative magnitudes of the *buoyancy force* and the opposing *friction force* acting on the fluid.

$$Gr = \frac{g\beta(T_s - T_\infty)\delta^3}{\upsilon^2}$$

where

g = gravitational acceleration, m/s^2

β = coefficient of volume expansion, 1/K (β = *1/K* for ideal gases)

T_s = temperature of the surface, ^{0}C

T_∞= temperature of the fluid sufficiently far from the surface, ^{0}C

δ= characteristic length of the geometry, m

ν = kinematic viscosity of the fluid, m^2/s

The Grashof number plays the role-played by the Reynolds number in forced convection in natural convection. As such, the Grashof number provides the main criterion in determining whether the fluid flow is laminar or turbulent in natural convection. For vertical plates, for example, the critical Grashof number is observed to be about 10^9. Therefore, the flow regime on a vertical plate becomes turbulent at Grashof numbers greater than 10^9. The heat transfer rate in natural convection from a solid surface to the surrounding fluid is expressed by Newton's law of cooling as

$$Q_{conv} = hA(T_s - T_\infty) \qquad \text{(W)}$$

where A is the heat transfer surface area and h is the average heat transfer coefficient on the surface.

3.2 Natural Convection Over Surfaces

Natural convection heat transfer on a surface depends on the geometry of the surface as well as its orientation. It also depends on the variation of temperature on the surface and the thermophysical properties of the fluid involved.

The *velocity* and *temperature profiles* for natural convection over a vertical hot plate immersed in a quiescent fluid body are given in Figure 4.6.

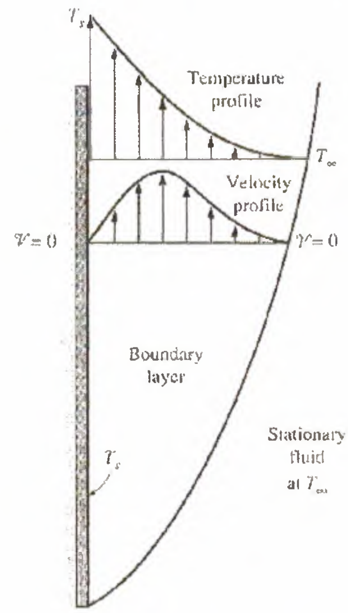


Figure 3.6 Typical velocity and temperature profiles for natural convection flow over a hot vertical plate at temperature T_s inserted in a fluid at temperature T_∞.

As in forced convection, the thickness of the boundary layer increases in the flow direction. Unlike forced convection, however, the fluid velocity is *zero* at the *outer edge* of the velocity boundary layer as well as at the surface of the plate. This is expected since the fluid beyond the boundary layer is stationary. Thus, the fluid velocity increases with distance from the surface, reaches a maximum, and gradually decreases to zero at a distance sufficiently far from the surface. The *temperature* of the fluid will equal the plate temperature at the surface and gradually decrease to the temperature of the surrounding fluid at a distance sufficiently far from the surface, as shown in the figure. In the case of *cold surfaces,* the shape of the velocity and temperature profile s remains the same but their direction is reversed.

Natural Convection Correlations

Although we understand the mechanism of natural convection well, the complexities of fluid motion make it very difficult to obtain simple analytical relations for heat transfer by solving the governing equations of motion and energy. Some analytical solutions exist for natural convection, but such solutions lack generality since they are obtained for simple geometries under some simplifying assumptions. Therefore, with the exception of some simple cases, heat transfer relations in natural convection are based on experimental studies. Of the numerous such correlations of varying complexity and claimed accuracy available in the literature for any given geometry, we present below the *simpler* ones for two reasons: first, the accuracy of simpler relations is usually within the range of uncertainty associated with a problem, and second, we would like to keep the emphasis on the physics of the problems instead of formula manipulation.

The simple empirical correlations for the average *Nusselt number* Nu in natural convection are of the form,

$$Nu = \frac{h\delta}{k} = C(Gr\,\mathrm{Pr})^n = CRa^n$$

where Ra is the Rayleigh number, which is the product of the Grashof and Prandtl numbers:

$$Ra = Gr\,\mathrm{Pr} = \frac{g\beta(T_s - T_\infty)\delta^3}{\nu^2}\mathrm{Pr}$$

The values of the constants C and n depend on the *geometry* of the surface and the *flow regime,* which is characterized by the range of the Rayleigh number. The value of n is usually $\frac{1}{4}$ for laminar flow and $\frac{1}{3}$ for turbulent flow. The value of the constant C is normally less than 1. Also given in this table are the characteristic lengths of the geometries and the ranges of Rayleigh number in which the relation is applicable. All fluid properties are to be evaluated at the film temperature $T_f = \frac{1}{2}(T_s + T_\infty)$.

These relations have been obtained for the case of isothermal surfaces but could also be used approximately for the case of nonisothermal surfaces by assuming the surface temperature to be constant at some average value.

3.3 Natural Convection From Finned Surfaces

Finned surfaces of various shapes, called *heat sinks,* are frequently used in the cooling of electronic devices. Energy dissipated by these devices is transferred to the heat sinks by *conduction* and from the heat sinks to the ambient air by *natural* or *forced convection,* depending on the power dissipation requirements. Natural convection is the preferred mode of heat transfer since it involves *no moving parts,* like the electronic components themselves. However, in the natural convection mode, the components are more likely to run at a *higher temperature* and thus undermine reliability. A properly selected heat sink may considerably *lower* the operation temperature of the components and thus reduce the risk of failure.

A question that often a rises in the selection of a heat sink is whether to select one with *closely packed* fins or *widely spaced* fins for a given base area (Fig. 4.7).

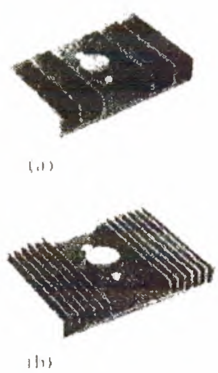


Figure 3.7 Heat sinks with (a) widely spaced and (b) closely packed fins.

A heat sink with closely packed fins will have greater surface area for heat transfer but a smaller heat transfer coefficient because of the extra resistance the additional fins will introduce to fluid flow through the interfin passages. A heat sink with widely spaced fins, on the other hand, will have a higher heat transfer coefficient but a smaller surface area. Therefore, there must be an *optimum spacing* that maximizes the natural convection heat transfer from the heat sink for a given base area *WL,* where *W* and *L* are the width and height of the base of the heat sink, respectively, as shown in Figure 4.8.

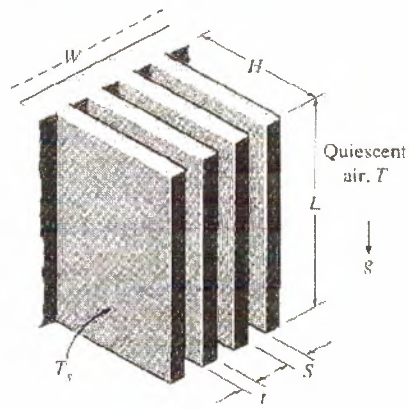


Figure 3.8 Various dimensions of a finned surface oriented vertically.

When the fins are essentially isothermal and the fin thickness t, is small relative to the fin spacing S, the optimum fin spacing for a vertical, heat sink is determined by Bar-Cohen and Rohsenow to be

$$S_{opt} = 2.714\frac{L}{Ra^{1/4}}$$

where the fin length L in the vertical direction is taken to be the characteristic length in the evaluation of the Rayleigh number. The heat transfer coefficient for the optimum spacing case was determined to be

$$h = 1.31\frac{k}{S_{opt}}$$

Then the rate of heat transfer by natural convection from the fins can be determined from

$$Q = h\,(2nLH)(T_s - T_\infty),$$

where $n = W/(S + t) \approx W/S$ is the number of fins on the heat sink and T_s is the surface temperature of the fins.

As we mentioned earlier, the magnitude of the natural convection heat transfer is directly related to the *mass flow rate* of the fluid, which is established by the dynamic balance of two opposing effects: *buoyancy* and *friction.*

The fins of a heat sink introduce both effects: *inducing extra buoyancy* as a result of the elevated temperature of the fin surfaces and *slowing down the fluid* by acting as an added obstacle on the flow path. As a result, increasing the number of fins on a heat sink can either enhance or reduce natural convection, depending on which effect is dominant. The buoyancy-driven airflow rate is established at the point where these two effects balance each other. The friction force increases as more and more solid surfaces are introduced, seriously disrupting fluid flow and heat transfer. Under some conditions, the increase in friction may more than offset the increase in buoyancy. This in turn will tend to reduce the flow rate and thus the heat transfer. For that reason, heat sinks with closely spaced fins are not suitable for natural convection cooling.

When the heat sink involves closely spaced fins, the narrow channels formed tend to block or "suffocate" the fluid, especially when the heat sink is long. As a result, the blocking action produced overwhelms the extra buoyancy and downgrades the heat transfer characteristics of the heat sink. Then, at a fixed power setting, the heat sink runs at a higher temperature relative to the no-shroud case. When the heat sink involves widely spaced fins, the shroud does not introduce a significant increase in resistance to flow, and the buoyancy effects dominate. As a result, heat transfer by natural convection may improve, and at a fixed power level the heat sink may run at a lower temperature.

When extended surfaces such as fins are used to enhance natural convection heat transfer between a solid and a fluid, the flow rate of the fluid in the vicinity of the solid adjusts itself to incorporate the changes in buoyancy and friction. It is obvious that this enhancement technique will work to advantage only when the increase in buoyancy is greater than the additional friction introduced. One does not need to be concerned with pressure drop or pumping power when studying natural convection since no pumps or blowers are used in this case. Therefore, an enhancement technique in natural convection is evaluated on heat transfer performance alone.

The failure rate of an electronic component increases almost exponentially with operating temperature. The cooler the electronic device operates, the more reliable it is. A role of thumb is that semiconductor failure rate is halved for each 10°C reduction in junction operating temperature. The desire to lower the operating temperature without having to resort to forced convection has motivated researchers to investigate enhancement techniques for natural convection. Sparrow and Prakash have demonstrated that, under certain conditions, the use of discrete plates in lieu of continuous plates of the same surface area increases heat transfer considerably. In other experimental work, using transistors as the heat source, Çengel and Zing have demonstrated that temperature recorded on the transistor case dropped by as much as 30°C when a shroud was used, as opposed to the corresponding no- shroud case.

3.4 Combined Natural And Forced Convection

The presence of a temperature gradient in a fluid in a gravity field always gives rise to natural convection currents, and thus heat transfer by natural convection. Therefore, forced convection is always accompanied by natural convection.

We mentioned earlier that the convection heat transfer coefficient, natural or forced, is a strong function of the fluid velocity. Heat transfer coefficients encountered in forced convection are typically much higher than those encountered in natural convection because of the higher fluid velocities associated with forced convection. As a result, we tend to ignore natural convection in heat transfer analyses that involve forced convection, although we recognize that natural convection always accompanies forced convection. The error involved in ignoring natural convection is negligible at high velocities but may be considerable at low velocities associated with forced convection. Therefore, it is desirable to have a criterion to assess the relative magnitude of natural convection in the presence of forced convection.

For a given fluid, it is observed that the parameter Gr/Re^2 represents the importance of natural convection relative to forced convection. This is not surprising since the convection heat transfer coefficient is a strong function of the Reynolds number Re in forced convection and the Grashof number Gr in natural convection.

Natural convection may *help* or *hurt* forced convection heat transfer, depending on the relative directions of *buoyancy-induced* and the *forced convection* motions (Fig. 3.9):

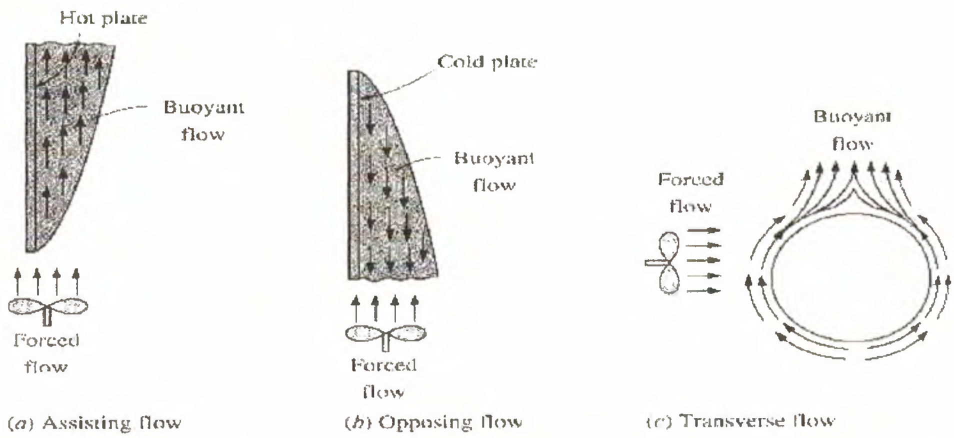


Figure 3.9 Natural convection can *enhance* or *inhibit* heat transfer, depending on the relative directions of *buoyancy y – induced motion* and the *forced convection motion.*

1. In *assisting flow,* the buoyant motion is in the *same* direction as the forced motion. Therefore, natural convection assists forced convection and *enhances* heat transfer. An example is upward forced flow over a hot surface,

2. In *opposing flow,* the buoyant motion is in the *opposite* direction to the forced motion. Therefore, natural convection resists forced convection and *decreases* heat transfer. An example is upward forced flow over a cold surface,

3. In *transverse flow,* the buoyant motion is *perpendicular* to the forced motion. Transverse flow enhances fluid mixing and thus *enhances* heat transfer. An example is horizontal forced flow over a hot or cold cylinder or sphere.

When determining heat transfer under combined natural and forced convection conditions, it is tempting to add the contributions of natural and forced convection in assisting flows and to subtract them in opposing flows. However, the evidence indicates differently. A review of experimental data suggests a correlation of the form

$$Nu_{combined} = (Nu^{n}_{forced} \pm Nu^{n}_{natural})^{1/n}$$

where Nu_{forced} and $Nu_{natural}$ are determined from the correlations for *pure forced* and *pure natural convection,* respectively. The plus sign is for *assisting* and *transverse* flows and the minus sign is for *opposing* flows. The value of the exponent *n* varies between 3 and 4, depending on the geometry involved. It is observed that $n = 3$ correlate experimental data for vertical surfaces well. Larger values of *n* are better suited for horizontal surfaces.

A question that frequently arises in the cooling of heat-generating equipment such as electronic components is whether to use a run (or a pump if the cooling medium is a liquid) that is, whether to utilize *natural* or *forced* convection in the cooling of the equipment. The answer depends on the maximum allowable operating temperature. Recall that the convection heat transfer rate from a surface at temperature T_s in a medium at T_∞ is given by

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

where h is the convection heat transfer coefficient and A is the surface area. Note that for a fixed value of power dissipation and surface area, h and T_s are *inversely proportional*. Therefore, the device will operate at a *higher* temperature when h is low (typical of natural convection) and at a *lower* temperature when h is high (typical of forced convection).

Natural convection is the preferred mode of heat transfer since no blowers or pumps are needed and thus all the problems associated with these, such as noise, vibration, power consumption, and malfunctioning, are avoided. Natural convection is adequate for cooling *low-power-output* devices, especially when they are attached to extended surfaces such as heat sinks. For *high-power-output* devices, however, we have no choice but to use a blower or a pump to keep the operating temperature below the maximum allowable level. For *very-high-power-output* devices, even forced convection may not be sufficient to keep the surface temperature at the desirable levels. In such cases, we may have to use *boiling* and *condensation* to take advantage of the very high heat transfer coefficients associated with phase change processes.

SUMMARY

In this chapter, we have considered *natural convection* heat transfer where any fluid motion occurs by natural means such as buoyancy. The fluid velocities associated with natural convection are low. Therefore, the heat transfer coefficients encountered in natural convection are usually much lower than those encountered in forced convection.

The upward force exerted by a fluid on a body completely or partially immersed in it is called the *buoyancy force,* whose magnitude is equal to the weight of the fluid displaced by the body.

The instrument used in natural convection experiments most of ten is the *Mach-Zehnder interferometer,* which gives a plot of isotherms in the fluid in the vicinity of a surface. The flow regime in natural convection is governed by a dimensionless number called the *Grashof number,* which represents the ratio of the buoyancy force to the viscous force acting on the fluid.

The Grashof number provides the main criterion in determining whether the fluid flow is laminar or turbulent in natural convection. Newton's law of cooling expresses the heat transfer rate in natural convection from a solid surface to the surrounding fluid.

Most heat transfer relations in natural convection are based on experimental studies, and the simple empirical correlations for the average *Nusselt number* Nu in natural convection.

Ra is the *Rayleigh number,* which is the product of the Grashof and Prandtl numbers.

CHAPTER 4

RADIATION

4. Introduction

4.1 Thermal Radiation

Although all electromagnetic waves have the same general features, waves of different wavelength differ significantly in their behavior. The electromagnetic radiation encountered in practice covers a wide range of wavelengths, varying from less than 10^{-10} μm for cosmic rays to more than 10^{-10} μm for electrical power waves. The **electromagnetic spectrum** also includes gamma rays, X-rays, ultraviolet radiation, visible light, infrared radiation, thermal radiation, microwaves, and radio waves.

Different types of electromagnetic radiation are produced differently through different mechanisms. For example, *gamma rays* are produced by nuclear reactions, *X-rays* by the bombardment of metals with high-energy electrons, *microwaves* by special types of electron tubes such as klystrons and magnetrons, and *radio waves* by the excitation of some crystals or by the flow of alternating current through electric conductors.

The short-wavelength gamma rays and X-rays are primarily of concern to nuclear engineers, while the long-wavelength microwaves and radio waves are of concern to electrical engineers. The type of electromagnetic radiation that is pertinent to heat transfer is the **thermal radiation** emitted as a result of vibrational and rotational motions of molecules, atoms, and electrons of a substance. Temperature is a measure of the strength of these activities at the microscopic level, as discussed in Chapter 1, and the rate of thermal radiation emission increases with increasing temperature. All matter whose temperature is above absolute zero continuously emits thermal radiation. That is, everything around us, such as walls, furniture, and our friends constantly emits (and absorbs) radiation (Fig. 4.1).

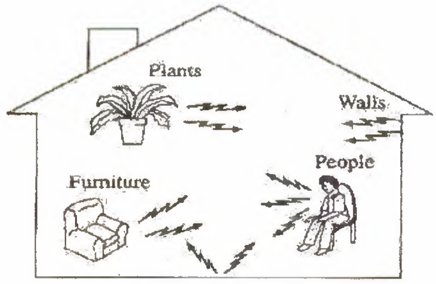


Figure 4.1 Everything around us constantly emits thermal radiation.

Thermal radiation is also defined as the portion of the electromagnetic spectrum that extends from about 0.1 to 100 μm, since the radiation emitted by bodies because of their temperature falls almost entirely into this wavelength range. Thus, thermal radiation includes the entire visible and infrared (IR) radiation as well as a portion of the ultraviolet (UV) radiation.

What we call **light** is simply the *visible* portion of the electromagnetic spectrum that lies between 0.40 and 0.76 μm. Light is characteristically no different than other electromagnetic radiation, except that it happens to trigger the sensation of seeing in the human eye. Light or the visible spectrum consists of narrow bands of color from violet (0.40-0.44 μm) to red (0.63-0.76 μm). The color of a surface depends on its ability to *reflect* certain wavelengths. For example, a surface that reflects radiation in the wavelength range 0.63-0.76 μm while absorbing the rest of the visible radiation appears red to the eye. A surface that reflects all of the light appears *white*, while a surface that absorbs all of the light incident on it appears *black*.

A body that emits some radiation in the visible range is called a light source. The sun is obviously our primary light source. The electromagnetic radiation emitted by the sun is known as **solar radiation**, and nearly all of it falls into the wavelength band 0.3-3 μm. Almost half of solar radiation is light (i.e., it falls into the visible range), with the remaining being ultraviolet and infrared.

4.2 Blackbody Radiation

A body at a temperature above absolute zero emits radiation in all directions over a wide range of wavelengths. The amount of radiation energy emitted from a surface at a given wavelength depends on the material of the body and the condition of its surface as well as the surface temperature. Therefore, different bodies may emit different amounts of radiation per unit surface area, even when they are at the same temperature. Thus, it is natural to be curious; about the *maximum* amount of radiation that can be emitted by a surface at a given temperature. Satisfying this curiosity requires the definition of an idealized body, called a *blackbody,* to serve as a standard against which the radiative properties of real surfaces may be compared.

A **blackbody** is defined as *a perfect emitter and absorber of radiation.* At a specified temperature and wavelength, no surface can emit more energy than a blackbody. A blackbody absorbs *all* incident radiation, regardless of wavelength and direction. Also, a blackbody emits radiation energy uniformly in all directions (Fig. 4.4). That is, a blackbody is a *diffuse* emitter. The term *diffuse* means "independent of direction. "

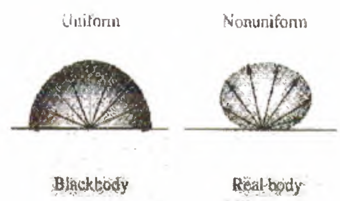


Figure 4.4 A blackbody is said to be a *diffuse* emitter since it emits radiation energy uniformly in all directions.

The radiation energy emitted by a blackbody per unit time and per unit surface area was determined experimentally by Joseph Stefan in 1879 and is expressed as

$$E_b = \sigma T^4 \qquad (\mathrm{W/m^2})$$

where σ = 5.67 X 10-8 W/m^2. K^4 is the *Stefan-Boltzmann constant* and T is the absolute temperature of the surface in K. This relation was theoretically verified in 1884 by

Ludwig Boltzmann. Equation $E_b = \sigma T^4$ is known as the **Stefan Boltzmann law** and E_b is called the **blackbody emissive power**. Note that, the emission of thermal radiation is proportional to the *fourth power* of the absolute temperature.

4.3 Radiation Properties

Most materials encountered in practice, such as metals, wood, and bricks, are *opaque* to thermal radiation, and radiation is considered to be a *surface phenomenon* for such materials. That is, thermal radiation is emitted or absorbed within the first few microns of the surface, and thus we speak of radiation properties of *surfaces* for opaque materials.

Some other materials, such as glass and water, allow visible radiation to penetrate to considerable depths before any significant absorption takes place. Radiation through such *semitransparent* materials obviously cannot be considered to be a surface phenomenon since the entire volume of the material interacts with radiation. On the other hand, both glass and water are practically opaque to infrared radiation. Therefore, materials can exhibit different behavior at different wavelengths, and the dependence on wavelength is an important consideration in the study of radiation properties such as emissivity, absorptivity, reflectivity, and transmissivity of materials.

In the preceding section, we defined a *blackbody* as a perfect emitter and absorber of radiation and said that no body can emit more radiation than a blackbody at the same temperature. Therefore, a blackbody can serve as a convenient *reference* in describing the emission and absorption characteristics of real surfaces.

Emissivity

The emissivity of a surface is defined as *the ratio of the radiation emitted by the surface to the radiation emitted by a blackbody at the same temperature.*

The emissivity of a surface is denoted by ε, and it varies between zero and one, $0 \leq \varepsilon \leq 1$. Emissivity is a measure of how closely a surface approximates a blackbody, for which $\varepsilon = 1$.

Absorptivity, Reflectivity, and Transmissivity

Everything around US constantly emits radiation, and the emissivity represents the emission characteristics of those bodies. This means that every body, including our own, is constantly bombarded by radiation coming from all directions over a range of wavelengths. *The radiation energy incident* on *a surface per unit surface area per unit time* is called **irradiation** and is denoted by G. When radiation strikes a surface, part of it is absorbed, part of it is reflected, and the remaining part, if any, is transmitted, as illustrated in Figure 4.5. *The fraction of irradiation absorbed by the surface* is called the **absorptivity** α, *the fraction reflected by the surface* is called the **reflectivity** ρ. and *the fraction transmitted* is called the **transmissivity** τ

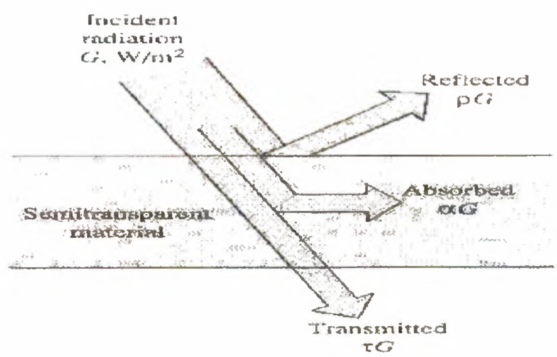


Figure 4.5 The absorption, reflection, and transmission of incident radiation by a semitransparent material.

SUMMARY

In this chapter, radiation propagates in the form of electromagnetic waves. All matter whose temperature is above absolute zero continuously emits *thermal radiation* as a result of vibrational and rotational motions of molecules, atoms, and electrons of a substance. Temperature is a measure of the strength of these activities at the microscopic level.

A *blackbody* is defined as a *perfect emitter and absorber of radiation.* At a specified temperature and wavelength, no surface can emit more energy than a blackbody. A blackbody absorbs *all* incident radiation, regardless of wavelength and direction. The radiation energy emitted by a blackbody per unit time and per unit surface area is called the *blackbody emissive power* and is expressed by the *Stefan-Boltzmann law.*

The *emissivity* of a surface is defined as the ratio of the radiation emitted, by the surface to the radiation emitted by a blackbody at the same temperature.

The radiation energy incident on a surface per unit surface area per unit time is called irradiation and is denoted by G. When irradiation strikes a surface, part of it is absorbed, part of it is reflected, and the remaining part, if any, is transmitted. The fraction of irradiation absorbed by the surface is called the *absorptivity* α. The fraction of irradiation reflected by the surface is called the *reflectivity* ρ, and the fraction transmitted is called the *transmissivity* τ.

CHAPTER 5

FORCED CONVECTION HEAT TRANSFER FOR FLOW IN PIPES, SINGLE CYLINDER, SINGLE SPHERE, TUBE BUNDLES

5.Introduction

5.1 Physical Mechanism Of Forced Convection

We mentioned earlier that there are three basic mechanisms of heat transfer: conduction, convection, and radiation. Conduction and convection are similar in that both mechanisms require the presence of a material medium. But they are different in that convection requires the presence of *fluid motion.*

Heat transfer through a *solid* is always by *conduction,* since the molecules of a solid remain at relatively fixed positions. Heat transfer through a *liquid* or *gas,* however, can be by *conduction* or *convection,* depending on the presence of any bulk fluid motion. Heat transfer through a fluid is by *convection* in the presence of bulk fluid motion and by *conduction* in the absence of it. Therefore, conduction in a fluid can be viewed as the *limiting case* of convection, corresponding to the cage of quiescent fluid (Fig. 5.1).

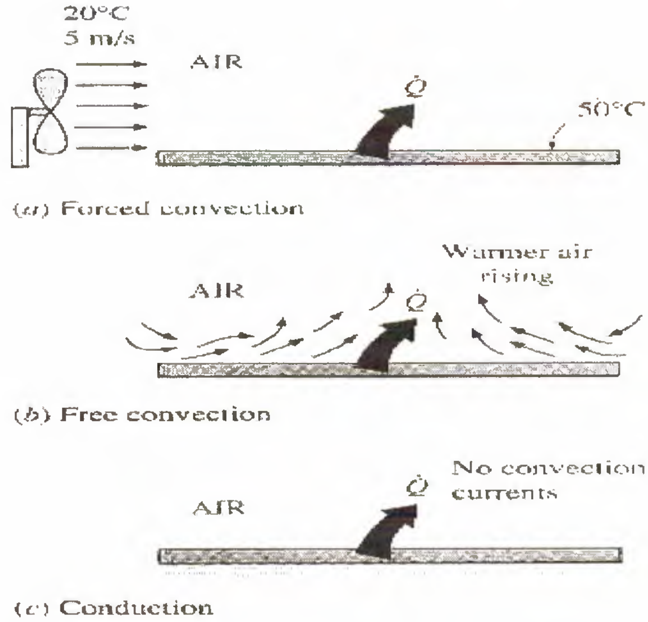


Figure 5.1 Heat transfer from a hot surface to the surrounding fluid by convection and conduction.

Convection heat transfer is complicated by the fact that it involves *fluid motion* as well as *heat conduction.* The fluid motion *enhances* heat transfer, since it brings hotter and cooler chunks of fluid into contact, initiating higher rates of conduction at a greater number of sites in a fluid. Therefore, the rate of heat transfer through a fluid is much higher by convection than it is by conduction. In fact, the higher the *fluid velocity,* the higher the *rate of heat transfer.*

To clarify this point further, consider steady heat transfer through a fluid contained between two *parallel plates* maintained at different temperatures, as shown in Figure 5.2.

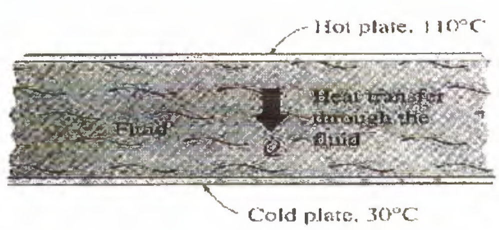


Figure 5.2 Heat transfer through a fluid sandwiched between two parallel plates.

The temperatures of the fluid and the plate will be the same at the points of contact because of the *continuity of temperature.* Assuming no fluid motion, the energy of the hotter fluid molecules near the hot plate will be transferred to the adjacent cooler fluid molecules. This energy will then be transferred to the next layer of the cooler fluid molecules. This energy will then be transferred to the next layer of the cooler fluid, and so on, until it is finally transferred to the other plate. This is what happens during *conduction* through a fluid. Now let us use a syringe to draw some fluid near the hot plate and inject it near the cold plate repeatedly. You can imagine that this will speed up the heat transfer process considerably, since some energy is *carried* to the other side as a result of fluid motion.

Consider the cooling of a *hot iron block* with a fan blowing air over its top surface, as shown in Figure 5.3. We know that heat will be transferred from the hot block to the surrounding cooler air, and the block will eventually cool. We also know that the block will cool faster if the fan is switched to a higher speed. Replacing air by water will enhance the convection heat transfer even more.

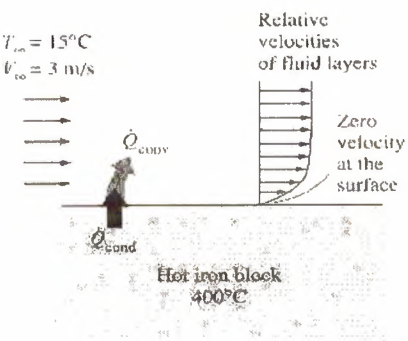


Figure 5.3 The cooling of a hot block by forced convection.

Experience shows that convection heat transfer strongly depends on the fluid properties *dynamic viscosity* μ, *thermal conductivity k, density* ρ, and *specific heat* C_p, as well as the *fluid velocity V.* It also depends on the *geometry* and *roughness* of the solid surface, in addition to the *type of fluid flow* (such as being streamlined or turbulent). Thus, we expect the convection heat transfer relations to be rather complex because of the dependence of convection on so many variables. This is not surprising, since convection is the most complex mechanism of heat transfer.

Despite the complexity of convection, the rate of *convection heat transfer* is observed to be proportional to the temperature difference and is conveniently expressed by **Newton's law of cooling** as

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

or

$$Q_{conv} = hA(T_s - T_\infty) \quad (\mathrm{W})$$

where

h = convection heat transfer coefficient, W /m^2 . ^{0}C

A = heat transfer surface area, m^2

T_s = temperature of the surface, ^{0}C

T_∞ = temperature of the fluid sufficiently far from the surface, ^{0}C

Judging from its units, the convection heat transfer coefficient can be defined as *the rate of heat transfer between a solid surface and a fluid per unit surface area per unit temperature difference.*

In convection studies, it is common practice to nondimensionalize the governing equations and combine the variables, which group together into *dimensionless numbers* in order to reduce the number of total variables. It is also common practice to *nondimensionalize* the heat transfer coefficient h with the **Nusselt number,** defined as

$$Nu = \frac{h\delta}{k}$$

where k is the thermal conductivity of the fluid and δ is the *characteristic length.* The Nusselt number is named after Wilhelm Nusselt, who made significant contributions to convective heat transfer in the first half of the 20th century, and it is viewed as the *dimensionless convection heat transfer coefficient.*

To understand the physical significance of the Nusselt number, consider a fluid layer of thickness δ and temperature difference $\Delta T = T_2 - T_1$, as shown in Figure 5.4.

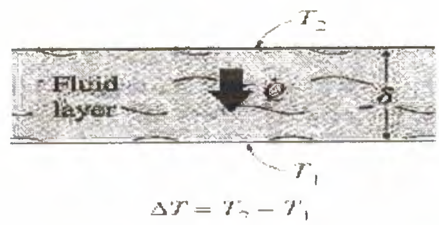


Figure 5.4 Heat transfer through a fluid layer of thickness δ and temperature difference ΔT.

Heat transfer through the fluid layer will be by *convection* when the fluid involves some motion and by *conduction* when the fluid layer is motionless. Heat flux (the rate of heat transfer per unit time per unit surface area) in either case will be

$$q_{conv} = h\Delta T$$

and

$$q_{conv} = k\frac{\Delta T}{\delta}$$

Taking their ratio gives

$$\frac{q_{conv}}{q_{cond}} = \frac{h\Delta T}{k\Delta T / \delta} = \frac{h\delta}{k} = Nu$$

which is the *Nusselt number*. Therefore, the Nusselt number represents the enhancement of heat transfer through a fluid layer as a result of convection relative to conduction across the same fluid layer. The larger the Nusselt number, the more effective the convection. A Nusselt number Nu = 1 for a fluid layer represents heat transfer by pure conduction.

We use forced convection in daily life more of ten than you might think (Fig. 5.5).

Figure 5.5 We resort to forced convection whenever we need to increase the rate of heat transfer.

We resort to forced convection whenever we want to increase the rate of heat transfer from a hot object. For example, we turn on the *fan* on hot summer days to help our body cool more effectively. The higher the fan speed, the better we feel. We *stir* our soup and *blow* on a hot slice of pizza to make them cool faster. The air on *windy* winter days feels much colder than it actually is. The simplest solution to heating problems in electronics packaging is to use a large enough fan.

Laminar and Turbulent Flows

If you have been around smokers, you probably noticed that the cigarette smoke rises in a smooth plume for the first few centimeters and then starts fluctuating randomly in all directions as it continues its journey toward the lungs of nonsmokers (Fig. 5.6).

Figure 5.6 Laminar and turbulent flow regimes of cigarette smoke.

Likewise, a careful inspection of flow over a flat plate reveals that the fluid flow in the boundary layer starts out as fiat and streamlined but turns chaotic after some distance from the leading edge, as shown in Figure 5.7. The flow regime in the first case is said to be **laminar,** characterized by *smooth streamlines* and *highly ordered motion,* and **turbulent** in the second case, where it is characterized by *velocity fluctuations* and *highly disordered motion.* The **transition** from laminar to turbulent flow does not occur suddenly; rather, it occurs over some region in which the flow hesitates between laminar and turbulent flows before it becomes fully turbulent.

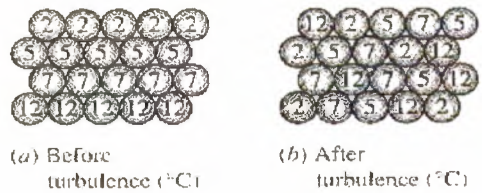


Figure 5.7 The intense mixing in turbulent flow brings fluid particles at different temperatures into close contact, and thus enhances heat transfer.

We can verify the existence of these laminar, transition, and turbulent flow regimes by injecting some dye into the flow stream. We will observe that the dye streak will form a *smooth* line when the flow is laminar, will have *bursts of fluctuations* in the transition regime, and will *zigzag rapidly and randomly* when the flow becomes fully turbulent.

Note that the velocity profile is approximately parabolic in laminar flow and becomes flatter in turbulent flow, with a sharp drop near the surface. The turbulent boundary layer can be considered to consist of three layers. The very thin layer next to the wall where the viscous effects are dominant is the **laminar sublayer.** The velocity profile in this layer is nearly linear, and the flow is streamlined. Next to the laminar sublayer is the **buffer layer**, in which the turbulent effects are significant but not dominant of the diffusion effects, and next to it is the **turbulent layer**, in which the turbulent effects dominate.

The *intense mixing* of the fluid in turbulent flow as a result of rapid fluctuations enhances heat and momentum transfer between fluid particles, which increases the friction force on the surface and the convection heat transfer rate (Fig. 5.7). It also causes the boundary layer to enlarge. Both the friction and heat transfer coefficients reach maximum values when the flow *becomes fully turbulent.* So it will come as no surprise that a special effort is made in the design of heat transfer coefficients associated with turbulent flow. The enhancement in heat transfer in turbulent flow does not come for free, however. It may be necessary to use a larger pump or fan in turbulent flow to overcome the larger friction forces accompanying the higher heat transfer rate.

Reynolds Number

The transition from laminar to turbulent flow depends on the *surface geometry, surface roughness, free-stream velocity, surface temperature,* and *type of fluid,* among other things. After exhaustive experiments in the 1880s, Osbom Reynolds discovered that the flow regime depends mainly on the ratio of the *inertia forces* to *viscous forces* in the fluid. This ratio is called the **Reynolds number** and is expressed for external flow as (Fig. 5.8)

$$\mathrm{Re} = \frac{Inertia\ forces}{Viscous\ forces} = \frac{V_\infty \delta}{v}$$

where

V_∞ = free-stream velocity, m/s

δ = characteristic length of the geometry, m

$v = \mu/\rho$ = kinematic viscosity of the fluid, m^2/s

Note that the Reynolds number is a *dimensionless* quantity. Also note that *kinematic viscosity* v differs from dynamic viscosity μ by the factor ρ. Kinematic viscosity has the unit m^2/s, which is identical to the unit of thermal diffusivity, and can be viewed as *viscous diffusivity.* The characteristic length is the distance from the leading edge x in the flow direction for a flat plate and the diameter D for a circular cylinder or sphere.

At *large* Reynolds numbers, the inertia forces, which are proportional to the density and the velocity of the fluid, are large relative to the viscous forces, and thus the viscous forces cannot prevent the random and rapid fluctuations of the fluid. At *small* Reynolds numbers, however, the viscous forces are large enough to overcome the inertia forces and to keep the fluid "in line." Thus the flow is *turbulent* in the first case and *laminar* in the second.

The Reynolds number at which the flow becomes turbulent is called the **critical Reynolds number.** The value of the critical Reynolds number is different for different geometries. For flow over a *flat plate,* transition from laminar to turbulent occurs at the critical Reynolds number of

$$\mathrm{Re}_{critical,\ flat\ plate} \approx 5 \times 10^5$$

This generally accepted value of the critical Reynolds number for a flat plate may vary somewhat depending on the surface roughness, the turbulence level, and the variation of pressure along the surface.

5.2 Flow In Tubes

Liquid or gas flow through or *pipes* or *ducts* is commonly used in practice in heating and cooling applications. The fluid in such applications is forced to flow by a fan or pump through a tube that is sufficiently long to accomplish the desired heat transfer. In this section, we will discuss the *friction* and *heat transfer coefficients* that are directly related to the *pressure drop* and *heat flux* for flow through tubes. These quantities are then used to determine the pumping power requirement and the length of the tube.

There is a *fundamental* difference between external and internal flows. In *external flow,* which we have considered so far, the fluid had a free surface, and thus the boundary layer over the surface was free to grow indefinitely. In *internal flow,* however, the fluid is completely confined by the inner surfaces of the tube, and thus there is a limit on how much the boundary layer can grow.

General Considerations

The fluid velocity in a tube changes from *zero* at the surface to a *maximum* at the tube center. In fluid flow, it is convenient to work with an *average* or *mean* velocity V_m, which remains constant in incompressible flow when the cross-sectional area of the tube is constant. The mean velocity in actual heating and cooling applications may change somewhat because of the changes in density with temperature. But, in practice, we evaluate the fluid properties at some average temperature and treat them as constants. The convenience in working with constant properties usually more than justifies the slight loss in accuracy.

The value of the mean velocity V_m is determined from the requirement that the *conservation of mass* principle be satisfied (Fig. 5.8).

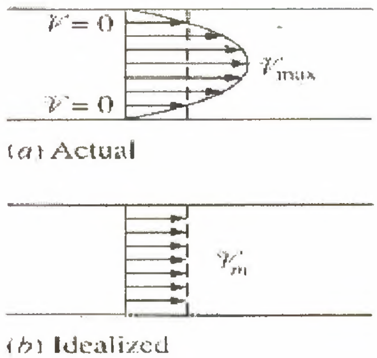


Figure 5.8 Actual and idealized velocity profiles for flow in tube (the mass flow rate of the fluid is the same for both cases.)

That is, the mass flow rate through the tube evaluated using the mean velocity V_m from

$$m = \rho V_m A_c \quad \text{(kg/s)}$$

will be equal to the actual mass flow rate. Here ρ is the density of the fluid and A_c is the cross-sectional area, which is equal to $A_c = \frac{1}{4}\pi D^2$ for a circular tube.

When a fluid is heated or cooled as it flows through a tube, the temperature of a fluid at any cross-section changes from T_s at the surface of the wall at that cross-section to some maximum (or minimum in the case of heating) at the tube center. In fluid flow it is convenient to work with an *average* or *mean* temperature T_m that remains constant at a cross-section. The mean temperature *Tm will change* in the flow direction, however, whenever the fluid is heated or cooled.

where C_p is the specific heat of the fluid and m is the mass flow rate. Note that the product mC_pT_m at any cross-section along the tube represents the *energy flow* with the fluid at that cross-section. You will recall that in the absence of any work interactions (such as electric resistance heating), the conservation of energy equation for the steady flow of a fluid in a tube can be expressed as (Fig. 5.9)

$$Q = mC_p(T_e - T_i) \quad \text{(kJ/s)}$$

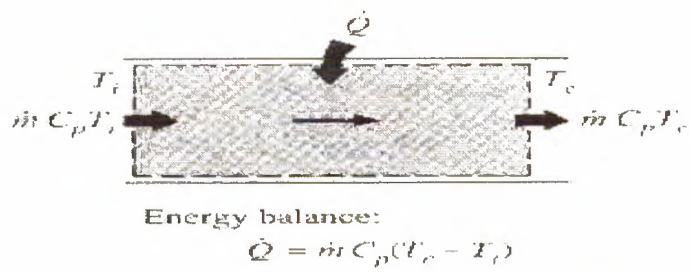


Figure 5.9 The heat transfer to a fluid flowing in profiles for flowing in a tube is equal to the increase in the energy of the fluid.

where T_i and T_e are the mean fluid temperatures at the inlet and exit of the tube, respectively, and Q is the rate of heat transfer to or from the fluid. Note that the temperature of a fluid flowing in a tube remaining constant in the absence of any energy interactions through the wall of the tube.

Perhaps we should mention that the friction between the fluid layers in a tube does cause a slight rise in fluid temperature as a result of the mechanical energy being converted to sensible heat energy. But *this frictional heating* is too small to warrant any consideration in calculations, and thus is disregarded. For example, in the absence of any heat transfer, no noticeable difference will be detected between the inlet and exit temperatures of a fluid flowing in a tube. Thus, it is reasonable to assume that any temperature change in the fluid is due to heat transfer.

The thermal conditions at the surface of a tube can usually be approximated with reasonable accuracy to be *constant surface temperature (Ts* = constant) or *constant surface heat flux (qs* = constant). For example, the constant surface temperature condition is realized when a phase change process such as boiling or condensation occurs at the outer surface of a tube. The constant surface heat flux condition is realized when the tube is subjected to radiation or electric resistance heating uniformly from all directions.

The convection heat flux at any location on the tube can be expressed as

$$q = h(T_s - T_m) \text{ (W/m}^2\text{)}$$

where h is the *local* heat transfer coefficient and T_s and T_m are the surface and the mean fluid temperatures at that location. Note that the mean fluid temperature T_m of a fluid flowing in a tube must change during heating or cooling. Therefore, when h = constant, the surface temperature T_s must change when q_s = constant, and the surface heat flux q_s must change when T_s = constant. Thus we may have either T_s = constant or q_s = constant at the surface of a tube, but not both. Below we consider convection heat transfer for these two common cases.

Constant Surface Heat Flux *(q_s = constant)*

In the case of q_s = constant, the rate of heat transfer can also be expressed as

$$Q = q_s A = mC_p(T_e - T_i) \qquad \text{(W)}$$

Then the mean fluid temperature at the tube exit becomes

$$T_e = T_i + \frac{q_s A}{mC_p}$$

Note that the mean fluid temperature increases *linearly* in the flow direction in the case of constant surface heat flux, since the surface area increases linearly in the flow direction *(A* is equal to the perimeter, which is constant, times the tube length).

The surface temperature in this case can be determined from $q = h(T_s - T_m)$, Note that when h is constant, T_s - T_m = constant, and thus the surface temperature will also increase *linearly* in the flow direction. Of course, this is true when the variation of the specific heat C_p with T is disregarded and C_p is assumed to remain constant.

Laminar Flow in Tubes

We mentioned earlier that flow in smooth tubes is laminar for Re < 2300. The theory for laminar flow is well developed, and both the friction and heat transfer coefficients for fully developed laminar flow in smooth circular tubes can be determined analytically by solving the governing differential equations. Combining the conservation of *mass* and *momentum* equations in the axial direction for a tube and solving them subject to the no-slip condition at the boundary and the condition that the velocity profile is symmetric about the tube center give the following *parabolic* velocity profile for the hydro-dynamically developed laminar flow:

$$V(r) = 2V_m\left(1 - \frac{r^2}{R^2}\right)$$

where V_m is the *mean fluid velocity* and R is the *radius* of the tube. Note that the maximum velocity occurs at the tube center $(r = 0)$, and it is

$V_{max} = 2V_m$.

But we also have the following practical definition of shear stress:

$$\tau_s = C_f \frac{\rho V_m^2}{2}$$

where C_f is the friction coefficient.

The *friction factor f*, which is the parameter of interest in the pressure drop calculations, is related to the friction coefficient C_f by $f = 4C_f$ Therefore,

$$f = \frac{64}{\mathrm{Re}} \qquad \text{(Laminar Flow)}$$

Note that the friction factor f is related to the *pressure drop* in the fluid, whereas the friction coefficient C_f is related to the *drag force* on the surface directly. Of course, these two coefficients are simply a constant multiple of each other.

The Nusselt number in the fully developed laminar flow region in a circular tube is determined in a similar manner from the conservation of energy equation to be

$\mathrm{Nu} = 3.66$ for $T_s = constant$ (laminar flow)

$\mathrm{Nu} = 4.36$ for $q_s = constant$ (laminar flow)

$$Nu = 1.86\left(\frac{\mathrm{Re}\,\mathrm{Pr}\,D}{L}\right)^{1/3}\left(\frac{\mu_b}{\mu_s}\right)^{0.14} \qquad (\mathrm{Pr} > 0.5)$$

All properties are evaluated at the bulk mean fluid temperature, except for μ_s which is evaluated at the surface temperature.

The Reynolds and Nusselt numbers for flow in these tubes are based on the hydraulic diameter D_h defined as

$$D_h = \frac{4A_c}{p}$$

where A_c is the cross-sectional area of the tube and p is its perimeter. The hydraulic diameter is defined such that it reduces to ordinary diameter D for circular tubes since $Ac = \pi D^2/4$ and $p = \pi D$. Once the Nusselt number is available, the convection heat transfer coefficient is determined from $h = k\,Nu/D_h$. It turns out that for a fixed surface area, the *circular tube* gives the most heat transfer for the least pressure drop, which explains the over- whelming popularity of circular tubes in heat transfer equipment. The effect of *surface roughness* on the friction factor and the heat transfer coefficient in laminar flow is negligible.

Turbulent Flow in Tubes

We mentioned earlier that flow in smooth tubes is turbulent at Re > 4000. Turbulent flow is commonly utilized in practice because of the higher heat transfer coefficients associated with it. Most correlations for the friction and heat transfer coefficients in turbulent flow are based on experimental studies because of the difficulty in dealing with turbulent flow theoretically.

For *smooth* tubes, the friction factor in fully developed turbulent flow can be determined from

$$f = 0.184\,\mathrm{Re}^{-0.2} \qquad \text{(Smooth tubes)}$$

The friction factor for flow in tubes with *smooth* as well as *rough surfaces* over a wide range of Reynolds numbers is given in Figure 5.9, which is known as the Moody diagram.

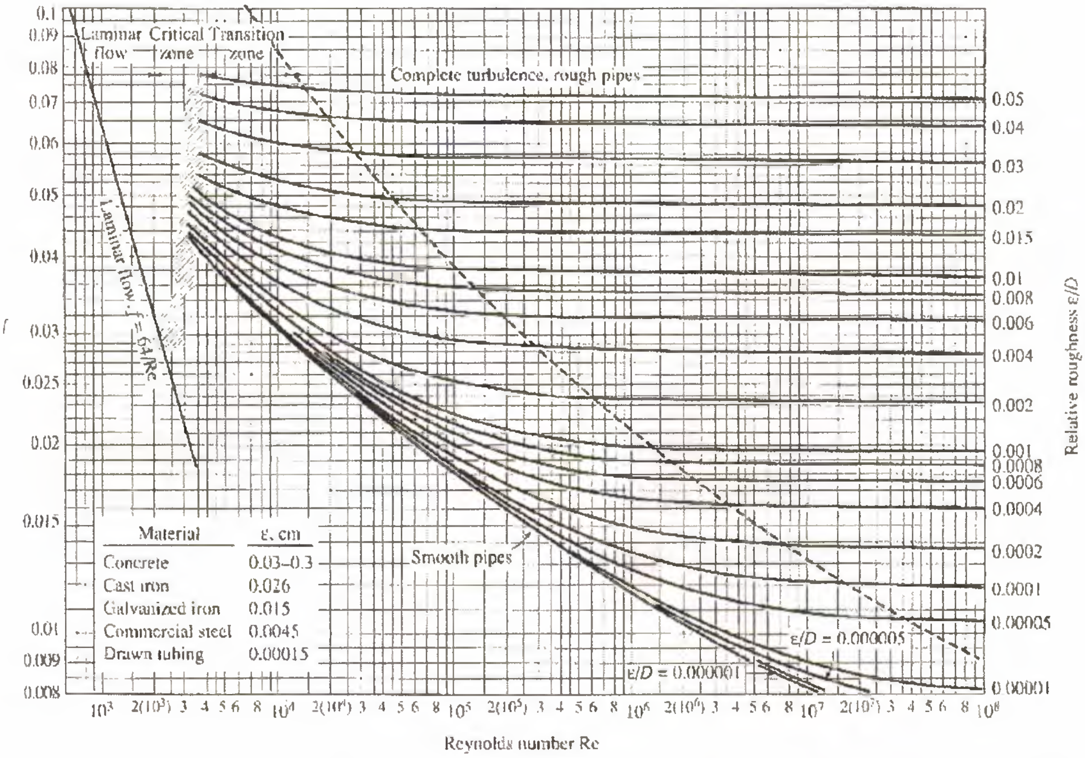


Figure 5.9 Friction factor for fully developed flow in circular tubes (the Moody chart)

Note that the friction factor and thus the pressure drop for flow in a tube can vary several times as a result of surface roughness.

The Nusselt number in turbulent flow is related to the friction factor through the famous **Chilton-Colburn analogy** expressed as

$$\text{Nu} = 0.125 f \,\text{Re}\,\text{Pr}^{1/3} \qquad \text{(turbulent flow)}$$

Substituting the f relation from Equation $f = 0.184\,\text{Re}^{-0.2}$ into Equation $\text{Nu} = 0.125 f \,\text{Re}\,\text{Pr}^{1/3}$ gives the following relation for the Nusselt number for *fully developed turbulent flow in smooth tubes:*

$$\text{Nu} = 0.023\,\text{Re}^{0.8}\,\text{Pr}^{1/3} \qquad (0.7 \le \text{Pr} \le 160)(\text{Re} > 10.000)$$

which is known as the **Colburn equation.** The accuracy of this equation can be improved by modifying it as

$$\text{Nu} = 0.023\,\text{Re}^{0.8}\,\text{Pr}^{n} \qquad (0.7 \le \text{Pr} \le 160)(\text{Re} > 10.000)$$

where n = 0.4 for *heating* and 0.3 for *cooling* of the fluid flowing through the tube. This equation is known as the **Dittus-Boulter equation**, and it is preferred to the Colburn equation. The fluid properties are evaluated at the *bulk mean fluid temperature* T_b = $\frac{1}{2}(T_i + T_e)$, which is the arithmetic average of the mean fluid temperatures at the inlet and the exit of the tube.

The relations above are not very sensitive to the *thermal conditions* at the tube surfaces and can be used for both T_s = constant and q_s = constant cases. Despite their simplicity, the correlations above give sufficiently accurate results for most engineering purposes. They can also be used to obtain rough estimates of the friction factor and the heat transfer coefficients in the transition region 2300 $\leq$Re $\leq$4000, especially when the Reynolds number is closer to 4000 than it is to 2300.

The Nusselt number for *rough surfaces* can also be determined from Equation Nu = 0.125 f Re $\mathrm{Pr}^{1/3}$ by substituting the friction factor f value from the Moody chart. Note that tubes with rough surfaces have much higher heat transfer coefficients than tubes with smooth surfaces. Therefore, tube surfaces are of ten intentionally *roughened, corrugated,* or *finned* in order to *enhance* the convection heat transfer coefficient and thus the convection heat transfer rate (Fig. 5.9). Heat transfer in turbulent flow in a tube has been increased by as much as 400 percent by roughening the surface. Roughening the surface, of course, also increases the friction factor and thus the power requirement for the pump or the fan.

The turbulent flow relations above can also be used for *noncircular tubes* with reasonable accuracy by replacing the diameter D in the evaluation of the Reynolds number by the hydraulic diameter $D_h = 4\,A_c/p$.

5.3. The Cylinder In Cross Flow

5.3.1 Flow Considerations

Another common external flow involves fluid motion normal to the axis of a circular cylinder. As shown in Figure 5.10, the free stream fluid is brought to rest at the *forward stagnation point,* with an accompanying rise in pressure. From this point, the pressure decreases with increasing x, the streamline coordinate, and the boundary layer develops under the influence of a *favorable pressure gradient* ($dp/dx < 0$). However, the pressure must eventually reach a minimum, and toward the rear of the cylinder further boundary layer development occurs in the presence of an *adverse pressure gradient* ($dp/dx > 0$).

In Figure 5.10 the distinction between the upstream velocity V and the free stream velocity u_∞ should be noted. Unlike conditions for the flat plate in parallel flow, these velocities differ, with u_∞ now depending on the distance x from the stagnation point. From Euler's equation for an inviscid flow [12], $u_\infty(x)$ must exhibit behavior opposite to that of $p(x)$. That is, from $u_\infty = 0$ at the stag- nation point, the fluid accelerates because of the favorable pressüre gradient ($du_\infty/dx > 0$ when $dp/dx < 0$), reaches a maximum velocity when $dp/dx = 0$, and decelerates as a result of the adverse pressure gradient ($du_\infty/dx < 0$ when $dp/dx > 0$). At this location, termed the *separation point,* fluid near the surface lacks sufficient momentum to overcome the pressure gradient, and continued downstream movement is impossible.

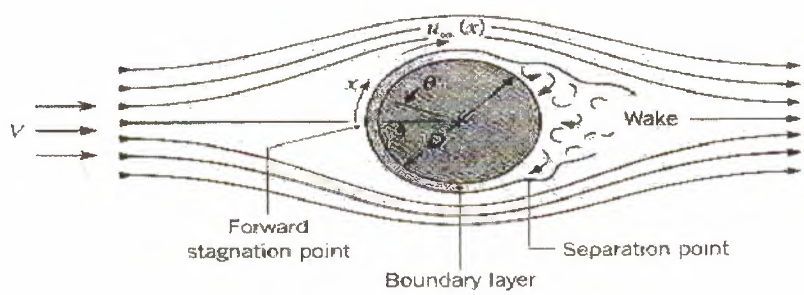


Figure 5.10 Boundary layer formation and separation on circular cylinder in cross flow.

Since the oncoming fluid also precludes flow back upstream, *boundary layer separation* must occur. This is a condition for which the boundary layer detaches from the surface, and a *wake* is formed in the downstream region. Flow in this region is characterized by vortex formation and is highly irregular. The *separation point* is the location for which $(\partial u/\partial y)_s = 0$.

The occurrence of *boundary layer transition,* which depends on the Reynolds number, strongly influences the position of the separation point. For the circular cylinder the characteristic length is the diameter, and the Reynolds number is defined as

$$\mathrm{Re}_D \equiv \frac{\rho V D}{\mu} = \frac{VD}{\nu}$$

Since the momentum of fluid in a turbulent boundary layer is larger than in the laminar boundary layer, it is reasonable to expect transition to delay the occurrence of separation. if $Re_D \leq 2 \text{ X } 10^5$, the boundary layer remaining laminar, and

separation occurs at $\theta = 80°$ (Figure 5.11).

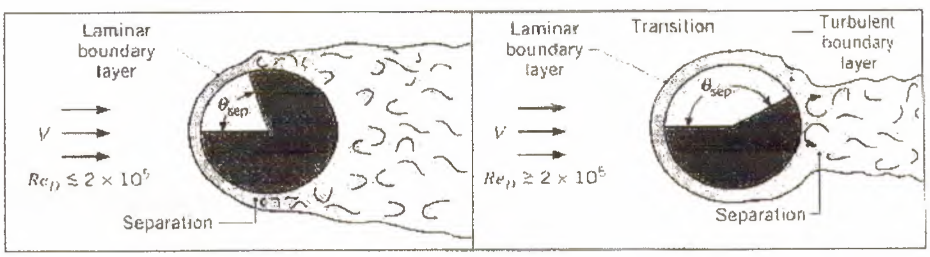


Figure 5.11 The effect of turbulence on separation.

However, if $Re_D \geq 2 \text{ X } 10^5$, boundary layer transition occurs, and separation is delayed to $\theta = 140°$. The foregoing processes strongly influence the drag force F_D acting on the cylinder. This force has two components, one of which is due to the boundary layer surface shear stress *(friction drag).* The other component is due to a pressure differential in the flow direction resulting from formation of the wake *(form,* or *pressure, drag).* A dimensionless *drag coefficient* C_D maybe defined as

$$C_D \equiv \frac{F_D}{A_f(\rho V^2/2)}$$

where A_f is the cylinder frontal area (the area projected perpendicular to the free stream velocity). The drag coefficient is a function of Reynolds number and results are presented in Figure 5.11.

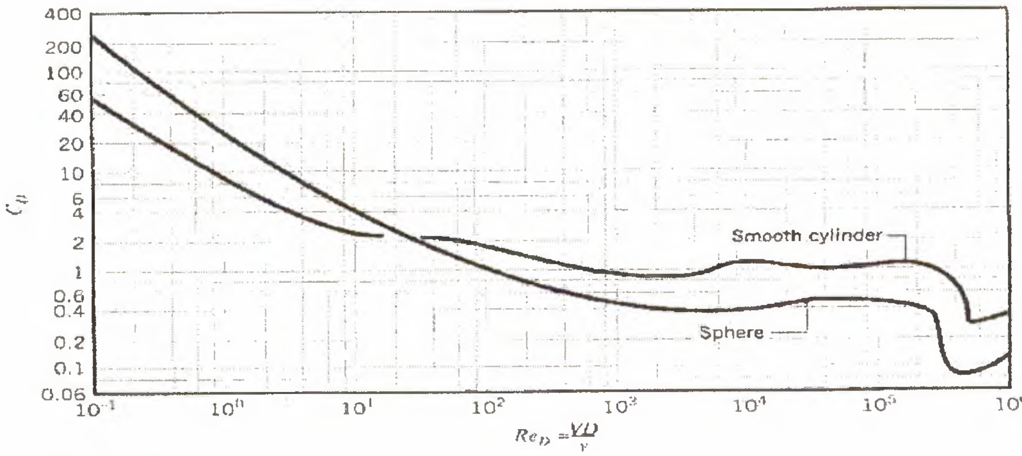


Figure 5.11 Drag coefficients for smooth, circular cylinder in flow and for a sphere.

For $Re_D < 2$ separation effects are negligible, and conditions are dominated by motion drag. However, with increasing Reynolds number, the effect of separation, and therefore form drag, becomes more important. The large reduction in C_D that occurs for $Re_D > 2 \times 10^5$ is due to boundary layer transition, which delays separation, thereby reducing the extent of the wake region and the magnitude of the form drag.

5.3.2 Convection Heat Transfer

Experimental results for the variation of the local Nusselt number with θ are shown in Figure 7.9 for the cylinder in across flow of air. Not unexpectedly, the results are strongly influenced by the nature of boundary layer development on the surface. Consider the conditions for $Re_D \leq 10^5$. Starting at the stagnation point, Nu_θ decreases with increasing θ due to laminar boundary layer development. However, a minimum is reached at $\theta = 80°$,~where separation occurs and Nu_θ increases with θ due to mixing associated with vortex formation in the wake. In contrast, for $Re_D \sim 10^5$ the variation of Nu_θ is characterized by two minima. The decline in Nu_θ from the value at the stagnation point is again due to laminar boundary layer development, but the sharp increase that occurs between 80° and 100° is now due to boundary layer transition to turbulence. With further development of the turbulent boundary layer, Nu_θ again begins to decline. Eventually separation occurs ($\theta = 140°$), and Nu_θ increases as a result of mixing in the wake region. The increase in Nu_θ with in- creasing Re_D is due to a corresponding reduction in the boundary layer thickness.

Correlations may be obtained for the local Nusselt number, and at the forward stagnation point for $Pr \sim 0.6$, boundary layer analysis [5] yields an expression of the form

$$Nu_D(\theta = 0) = 1..15 \mathrm{Re}_D^m \mathrm{Pr}^{1/3}$$

However, from the standpoint of engineering calculations, we are more interested in overall average conditions. The empirical correlation due to Hilpert [13]

$$Nu_D \equiv \frac{hD}{k} = C \mathrm{Re}_D^m \mathrm{Pr}^{1/3}$$

is widely used, where the constants C and m are listed in Table 5.1. Equation may also be used for *gas flow* over cylinders of noncircular cross section with the characteristic length D and the constants obtained from Table 5.2. in working with Equations all properties are evaluated at the film temperature.

Re_D	C	m
0.4–4	0.989	0.330
4–40	0.911	0.385
40–4000	0.683	0.466
4000–40,000	0.193	0.618
40,000–400,000	0.027	0.805

Table 5.1 Constants of Equation $Nu_D \equiv \frac{hD}{k} = C \mathrm{Re}_D^m \mathrm{Pr}^{1/3}$ for the circular cylinder in cross flow [13, 14].

Geometry	Re_D	C	m
Square			
	5×10^3–10^5	0.246	0.588
	5×10^3–10^5	0.102	0.675
Hexagon			
	5×10^3–1.95×10^4 1.95×10^4–10^5	0.160 0.0385	0.638 0.782
	5×10^3–10^5	0.153	0.638
Vertical plate			
	4×10^3–1.5×10^4	0.228	0.731

Table 5.2 Constants of Equation $Nu_D \equiv \dfrac{hD}{k} = C\,\mathrm{Re}_D^m\,\mathrm{Pr}^{1/3}$ for noncircular cylinders in cross flow of a gas [16].

Other correlations have been suggested for the circular cylinder in cross flow [8, 16, 17]. The correlation due to Zhukauskas [16] is of the form

$$Nu_D = C\,\mathrm{Re}_D^m\,\mathrm{Pr}^n(\mathrm{Pr}/\mathrm{Pr}_s)^{1/4} \qquad (0.7 < \mathrm{Pr} < 500)\ (1 < \mathrm{Re} < 10^6)$$

where all properties are evaluated at $T\infty$ except Pr_s which is evaluated at T_s. Values of C and m are listed in Table 5.3.

Re_D	C	m
1–40	0.75	0.4
40–1000	0.51	0.5
10^3–2×10^5	0.26	0.6
2×10^5–10^6	0.076	0.7

Table 5.3 Constants of Equation $Nu_D = C\,\mathrm{Re}_D^m\,\mathrm{Pr}^n(\mathrm{Pr}/\mathrm{Pr}_s)^{1/4}$ for the circular cylinder in cross flow [16].

If $Pr \leq 10$, $n = 0.37$; if $Pr > 10$, $n = 0.36$. Churchill and Bernstein [17] have proposed a single comprehensive equation that covers the entire range of Re_D for which data are available, as well as a wide range of Pr. The equation is recommended for all $Re_D\,Pr > 0.2$ and has the form

$$Nu = 0.3 + \frac{0.62\,\mathrm{Re}_D^{1/2}\,\mathrm{Pr}^{1/3}}{\left[1 + (0.4/\mathrm{Pr})^{2/3}\right]^{1/4}}\left[1 + \left(\frac{\mathrm{Re}_D}{282{,}000}\right)^{5/8}\right]^{4/5}$$

where all properties are evaluated at the film temperature.

5.4. The Sphere In Cross Flow

Boundary layer effects associated with flow over a sphere are much like those for the circular cylinder, with transition and separation both playing prominent roles. In the limit of very small Reynolds numbers *(creeping flow),* the coefficient is inversely proportional to the Reynolds number and the specific relation is termed *Stokes' law*

$$C_D = \frac{24}{\mathrm{Re}_D} \qquad \mathrm{Re} < 0.5$$

Numerous heat transfer correlations have been proposed, and Whitaker [8] recommends an expression of the form

$$Nu_D = 2 + (0.4\mathrm{Re}_D^{1/2} + 0.06\mathrm{Re}_D^{2/3})\mathrm{Pr}^{0.4}\left(\frac{\mu}{\mu_s}\right)^{1/4}$$

$$\begin{bmatrix} 0.71 < \mathrm{Pr} < 380 \\ 3.5 < \mathrm{Re}_D < 7.6 \times 10^4 \\ 1.0 < (\mu / \mu_s) < 3.2 \end{bmatrix}$$

All properties except μ_s are evaluated at T_∞

A special cage of convection heat transfer from spheres relates to transport from freely falling liquid drops, and the correlation of Ranz and Marshall [19] is often used

$$Nu = 2 + 0.6\mathrm{Re}_D^{1/2}\,\mathrm{Pr}^{1/3}$$

In the limit $Re_D \to 0$, Equations reduce to $Nu_D = 2$, which corresponds to heat transfer by conduction from a spherical surface to a stationary, infinite medium around the surface.

5.5. Flow Across Bank of Tubes (Tube Bundles)

Heat transfer to or from a bank (or bundle) of tubes in cross flow is relevant to numerous industrial applications, such as steam generation in a boiler or air cooling in the coil of an air conditioner. The geometric arrangement is shown schematically in Figure 5.12.

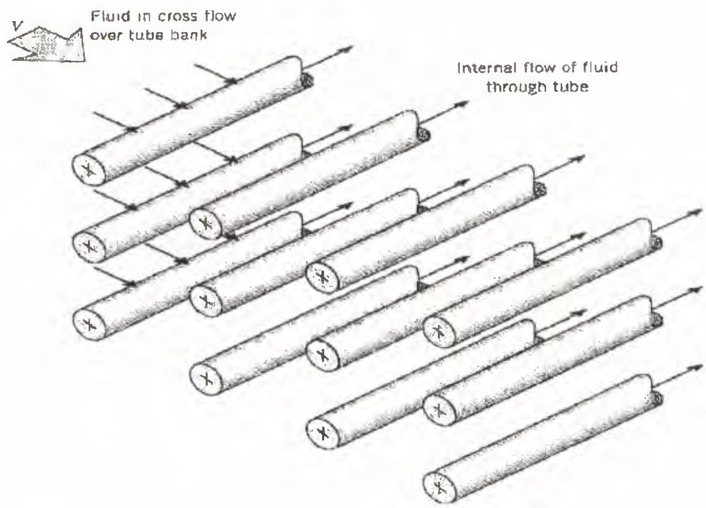


Figure 5.12 Schematic of a tube in cross flow.

Typically, one fluid moves over the tubes, while a second fluid at a different temperature passes through the tubes. In this section we are specifically interested in the convection heat transfer associated with cross flow over the tubes.

The tube rows of a bank are either *staggered* or *aligned* in the direction of the fluid velocity V (Figure 5.13).

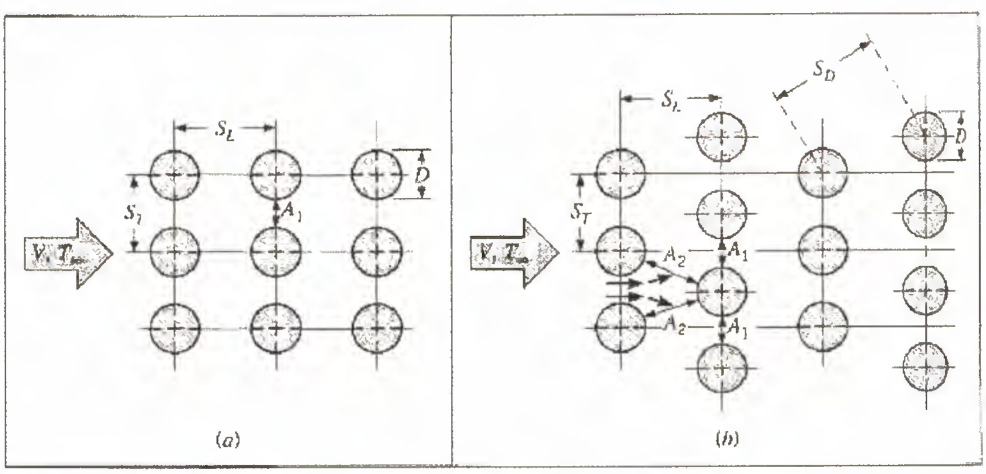


Figure 5.13 Tube arrangements in a bank. (a) Aligned. (b) Staggered.

The configuration is characterized by the tube diameter D, and by the *transverse pitch* S_T and *longitudinal pitch* S_L measured between tube centers. Flow conditions within the bank are dominated by boundary layer separation effects and by wake interactions, which in turn influence convection heat transfer.

The heat transfer coefficient associated with a tube is determined by its position in the bank. The coefficient for a tube in the first row is approximately equal to that for a single tube in cross flow, whereas larger heat transfer coefficients are associated with tubes of the inner rows. The tubes of the first few rows act as a turbulence grid, which increases the heat transfer coefficient for tubes in the following rows. In most configurations, however, heat transfer conditions stabilize, such that little change occurs in the convection coefficient for a tube beyond the fourth or fifth row.

Generally, we wish to know the *average* heat transfer coefficient for the *entire* tube bundle. For airflow across tube bundles composed of *10 or more rows* ($N_L \geq 10$), Grimison [20] has obtained a correlation of the form

$$Nu_D = C_1 \mathrm{Re}^m_{D,\max} \qquad \begin{bmatrix} N_L \geq 10 \\ 2000 < \mathrm{Re}_{D,\max} < 40{,}000 \\ \mathrm{Pr} = 0.7 \end{bmatrix}$$

where C_1 and m are listed in Table 5.4 and

$$\mathrm{Re}_{D,\max} \equiv \frac{\rho V_{\max} D}{\mu}$$

	S_T/D							
	1.25		1.5		2.0		3.0	
S_L/D	C_1	m	C_1	m	C_1	m	C_1	m
Aligned								
1.25	0.348	0.592	0.275	0.608	0.100	0.704	0.0633	0.752
1.50	0.367	0.586	0.250	0.620	0.101	0.702	0.0678	0.744
2.00	0.418	0.570	0.299	0.602	0.229	0.632	0.198	0.648
3.00	0.290	0.601	0.357	0.584	0.374	0.581	0.286	0.608
Staggered								
0.600	—	—	—	—	—	—	0.213	0.636
0.900	—	—	—	—	0.446	0.571	0.401	0.581
1.000	—	—	0.497	0.558	—	—	—	—
1.125	—	—	—	—	0.478	0.565	0.518	0.560
1.250	0.518	0.556	0.505	0.554	0.519	0.556	0.522	0.562
1.500	0.451	0.568	0.460	0.562	0.452	0.568	0.488	0.568
2.000	0.404	0.572	0.416	0.568	0.482	0.556	0.449	0.570
3.000	0.310	0.592	0.356	0.580	0.440	0.562	0.428	0.574

Table 5.4 Constants of Equation $Nu_D = C_1 \mathrm{Re}^m_{D,\max}$ and $Nu_D = 1.13 C_1 \mathrm{Re}^m_{D,\max} \mathrm{Pr}^{1/3}$ for airflow over a tube bank of 10 or more rows [20].

It has become common practice to extend this result to other fluids through insertion of the factor $1.13\mathrm{Pr}^{1/3}$, in which case

$$Nu_D = 1.13 C_1 \mathrm{Re}^m_{D,\max} \mathrm{Pr}^{1/3} \quad \begin{bmatrix} N_L \geq 10 \\ 2000 < \mathrm{Re}_{D,\max} < 40{,}000 \\ \mathrm{Pr} = 0.7 \end{bmatrix}$$

All properties appearing in these equations are evaluated at the film temperature. If $N_L<10$ a correction factor may be applied such that

$$\overline{Nu_D}\big|_{(N_L<10)} = C_2 \overline{Nu_D}\big|_{(N_L\geq 10)}$$

N_L	1	2	3	4	5	6	7	8	9
Aligned	0.64	0.80	0.87	0.90	0.92	0.94	0.96	0.98	0.99
Staggered	0.68	0.75	0.83	0.89	0.92	0.95	0.97	0.98	0.99

Table 5.5 Correction factor C_2 of Equation $\overline{Nu_D}\big|_{(N_L<10)} = C_2 \overline{Nu_D}\big|_{(N_L\geq 10)}$ for $N_L<10$ [21].

where C_2 is given in Table 5.5.

Note that the Reynolds number $Re_{D,\max}$ for the foregoing correlations is based on the *maximum fluid velocity* occurring within the tube bank. For the aligned arrangement, V

max occurs at the transverse plane A_1 of Figure *5.13a,* and from the mass conservation requirement for an incompressible fluid

$$V_{\max} = \frac{S_T}{S_T - D} V$$

For the staggered configuration, the maximum velocity may occur at either the transverse plane A_1 or the diagonal plane A_2 of Figure *5.13b.* It will occur at A_1

if the rows are spaced such that

$$2(S_D - D) < (S_T - D)$$

The factor of 2 results from the bifurcation experienced by the fluid moving from the A_1 to the A_2 planes. Hence V max occurs at A_2 if

$$S_D = \left[S_L^2 + \left(\frac{S_T}{2} \right)^2 \right]^{1/2} < \frac{S_T + D}{2}$$

in which case it is given by

$$V_{\max} = \frac{S_T}{2(S_D - D)} V$$

If V max occurs at A_1 for the staggered configuration, it may again be computed from Equation $V_{\max} = \frac{S_T}{S_T - D} V$.

More recent results have been obtained [8,16], and Zhukauskas [16] has proposed a correlation of the form

$$\overline{Nu_D} = C \, \mathrm{Re}_{D,\max}^{m} \, \mathrm{Pr}^{0.36} \left(\frac{\mathrm{Pr}}{\mathrm{Pr}_s} \right)^{1/4}$$

$$\begin{bmatrix} N_L \geq 20 \\ 0.7 < \mathrm{Pr} < 500 \\ 1000 < \mathrm{Re}_{D,\max} < 2 \times 10^6 \end{bmatrix}$$

where all properties except Pr_s, are evaluated at the arithmetic mean of the fluid inlet and outlet temperatures and the constants C and *m,* are listed in Table 5.6.

Configuration	$Re_{D,\,max}$	C	m
Aligned	10–10^2	0.80	0.40
Staggered	10–10^2	0.90	0.40
Aligned	10^2–10^3	Approximate as a single	
Staggered	10^2–10^3	(isolated) cylinder	
Aligned ($S_T/S_L > 0.7$)[a]	10^3–2×10^5	0.27	0.63
Staggered ($S_T/S_L < 2$)	10^3–2×10^5	$0.35(S_T/S_L)^{1/5}$	0.60
Staggered ($S_T/S_L > 2$)	10^3–2×10^5	0.40	0.60
Aligned	2×10^5–2×10^6	0.021	0.84
Staggered	2×10^5–2×10^6	0.022	0.84

[a]For $S_T/S_L < 0.7$, heat transfer is inefficient and aligned tubes should not be used.

Table 5.6 Constants of Equation $\overline{Nu}_D = C\,\mathrm{Re}_{D,\max}^{m}\,\mathrm{Pr}^{0.36}\left(\dfrac{\mathrm{Pr}}{\mathrm{Pr}_s}\right)^{1/4}$ for the tube bank in cross flow [16].

The need to evaluate fluid properties at the arithmetic mean of the inlet $(T_i = T_\infty)$ and outlet (T_0) temperatures is dictated by the fact that the fluid temperature will decrease or increase, respectively, due to heat transfer to or from the tubes. If the fluid temperature change, $|T_i - T_0|$, is large, significant error could result from evaluation of the properties at the inlet temperature. if $N_L < 20$, a correction factor may be applied such that

$$\overline{Nu}_D\big|_{(N_L<20)} = C_2\,\overline{Nu}_D\big|_{(N_L\geq 20)}$$

where C_2 is given in Table 5.7.

N_L	1	2	3	4	5	7	10	13	16
Aligned	0.70	0.80	0.86	0.90	0.92	0.95	0.97	0.98	0.99
Staggered	0.64	0.76	0.84	0.89	0.92	0.95	0.97	0.98	0.99

Table 5.7 Correction factor C_2 of Equation $\overline{Nu}_D\big|_{(N_L<20)} = C_2\,\overline{Nu}_D\big|_{(N_L\geq 20)}$ for $N_L<20$ ($Re_D>10^3$) [16].

Flow around tubes in the first row of a tube bank corresponds to that for a single (isolated) cylinder in cross flow. However, for subsequent rows, flow depends strongly on the tube bank arrangement (Figure 5.14).

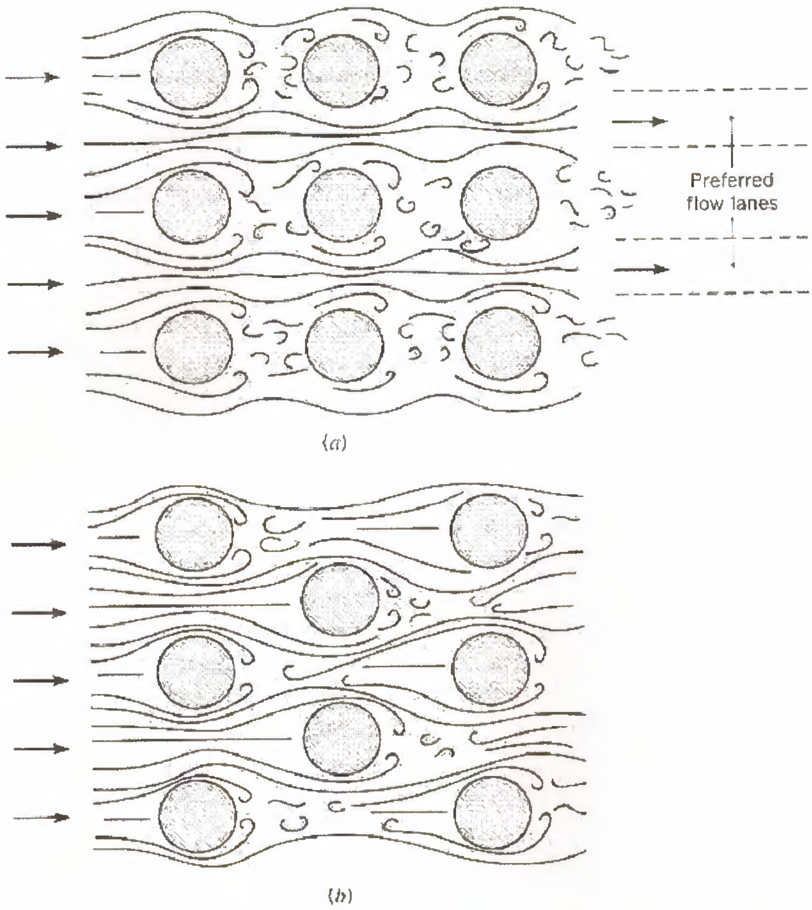


Figure 5.14 Flow conditions for (a) aligned and (b) staggered

Aligned tubes beyond the first row are in the turbulent wakes of upstream tubes, and for moderate values of S_L convection coefficients associated with downstream rows are enhanced by turbulation of the flow. Typically, the convection coefficient of a row increases with increasing row number until approximately the fifth row, after which there is little change in the turbulence and hence in the convection co- efficient. However, for small values of S_T/S_L, upstream rows, in effect, shield downstream rows from much of the flow, and heat transfer is adversely affected. That is, the preferred flow path is in lanes between the tubes and much of the tube surface is not exposed to the main flow. For this reason, operation of aligned tube banks with $S_T/S_L < 0.7$ (Table 5.6) is undesirable. For the staggered array, however, the path of the main flow is more tortuous and a greater portion of the surface area of downstream tubes remains in this path. In general, heat transfer enhancement is favored by the more tortuous flow of a staggered arrangement, particularly for small Reynolds number *(Re_D* < 100).

Since the fluid may experience a large change in temperature as it moves through the tube bank, the heat transfer rate could be significantly over predicted by using $\Delta T = T_s - T_\infty$ as the temperature difference in Newton's law of cooling. As the fluid moves through the bank, its temperature approaches T_s and $|\Delta T|$ decreases.

SUMMARY

Convection is the mode of heat transfer that involves conduction as well as bulk fluid motion. The rate of convection heat transfer in external flow is expressed by *Newton'* s *law of cooling* as

$$Q_{conv} = hA(T_s - T_\infty) \quad \text{(W)}$$

where T_s is the surface temperature and T_∞ is the free-stream temperature. The heat transfer coefficient h is usually expressed in the dimensionless form as the *Nusselt number*

$$Nu = \frac{h\delta}{k}$$

Fluid flow over a flat plate starts out as smooth and streamlined but turns chaotic after some distance from the leading edge. The flow regime is said in the first cage to be *laminar,* characterized by smooth streamlines and highly ordered motion, and to be *turbulent* in the second cage, where it is characterized by velocity fluctuations and highly disordered motion. The intense mixing in turbulent flow enhances both the drag force and the heat transfer. The flow regime depends mainly on the ratio of the inertia forces to viscous forces in the fluid. This ratio is called the *Reynolds number* and is expressed as

$$\mathrm{Re} = \frac{V_\infty \delta}{v}$$

where V is the free-stream velocity V_m for external flow and the mean fluid velocity V_{m} for internal flow. Also, δ is the characteristic length of the geometry, which is the distance from the leading edge for a flat plate, the outer diameter for flow over cylinders or spheres, and the inner diameter for flow inside circular tubes.

The Reynolds number at which the flow becomes turbulent is called the *critical Reynolds number.* The value of the critical Reynolds number is about 5×10^5 for flow over a flat plate, 2×10^5 for flow over cylinders and spheres, and 2300 for flow inside tubes.

For flow in a tube, the mean velocity V_m is the average velocity of the fluid. The mean temperature T_m at a cross-section can be viewed as the average temperature at that cross-section. The mean velocity V_m remains constant, but the mean temperature T_m

changes along the tube unless the fluid is not heated or cooled. The heat transfer to a fluid during steady flow in a tube can be expressed as

$$Q = mC_p(T_e - T_i) \qquad \text{(kJ/s)}$$

where T_i and T_e are the mean fluid temperatures at the inlet and exit of the tube.

The conditions at the surface of a tube can usually be approximated with reasonable accuracy to be *constant surface temperature (T_s* = constant) or *constant surface heat flux (q_s* = constant). In the cage of q_s = constant, the rate of heat transfer can be expressed as

$$Q = q_s A = mC_p(T_e - T_i) \qquad \text{(W)}$$

Then mean fluid temperature at the tube exit becomes

$$T_e = T_i + \frac{q_s A}{mC_p}$$

In fully developed laminar flow, the friction factor is determined to be f = 64/Re. The Nusselt number is determined to be Nu = 3.66 for the cage of T_s = constant and Nu = 4.36 for the case of q_s = constant. The average Nusselt number for the hydrodynamically and/or thermally developed laminar flow in a circular tube is given as

$$Nu = 1.86\left(\frac{\mathrm{Re}\,\mathrm{Pr}\,D}{L}\right)^{1/3}\left(\frac{\mu_b}{\mu_s}\right)^{0.14} \qquad (\mathrm{Pr} > 0.5)$$

The recommended relations for the friction factor f and the Nusselt number for fully developed turbulent flow in smooth circular tubes are

$$f = 0.184\,\mathrm{Re}^{-0.2} \qquad \text{(Smooth tubes)}$$

and

$$\mathrm{Nu} = 0.023\,\mathrm{Re}^{0.8}\,\mathrm{Pr}^{1/3} \qquad (0.7 \le \mathrm{Pr} \le 160)(\mathrm{Re} > 10.000)$$

where n = 0.4 for *heating* and 0.3 for *cooling* of the fluid flowing through the tube. The fluid properties are evaluated at the *bulk mean fluid temperature* $T_b = \frac{1}{2}(T_i + T_e)$, which is the arithmetic average of the mean fluid temperatures at the inlet and the exit of the tube.

The average Nusselt numbers for cross flow over a *cylinder* and *sphere* can be determined from

$$Nu = 0.3 + \frac{0.62\,\mathrm{Re}_D^{1/2}\,\mathrm{Pr}^{1/3}}{\left[1+(0.4/\mathrm{Pr})^{2/3}\right]^{1/4}}\left[1+\left(\frac{\mathrm{Re}_D}{282{,}000}\right)^{5/8}\right]^{4/5}$$

which is valid for Re Pr > 0.2, and

$$Nu_D = 2 + (0.4\,\mathrm{Re}_D^{1/2} + 0.06\,\mathrm{Re}_D^{2/3})\,\mathrm{Pr}^{0.4}\left(\frac{\mu}{\mu_s}\right)^{1/4}$$

Generally, the average heat transfer coefficient for the entire tube bundle. For airflow across tube bundles composed of 10 or more rows, Grimison has obtained a correlation of the form.

$$Nu_D = C_1\,\mathrm{Re}_{D,\max}^{m} \qquad \begin{bmatrix} N_L \geq 10 \\ 2000 < \mathrm{Re}_{D,\max} < 40{,}000 \\ \mathrm{Pr} = 0.7 \end{bmatrix}$$

where C_1 and m are listed in Table 5.4 and

$$\mathrm{Re}_{D,\max} \equiv \frac{\rho V_{\max} D}{\mu}$$

CONCLUSION

In Chapter 1, the basic concept of heat transfer is introduced and discussed. The science of *heat transfer* deals with the rate of heat transfer, which is the main quantity of interest in the design and evaluation of heat transfer equipment. The sensible and latent forms of internal energy can be transferred from one medium to another as a result of temperature difference, and are referred to as *heat* or *thermal energy*. Thus, *heat transfer* is the exchange of the sensible and latent forms of internal energy between two mediums as a result of temperature difference. Heat can be transferred in three different ways: Conduction, convection, and radiation. *Conduction* is the transfer of energy from the more energetic particles of a substance to the adjacent less energetic ones as a result of interactions between the particles. *Convection* is the mode of heat transfer between a solid surface and the adjacent liquid or gas that is in motion, and involves the combined effects of conduction and fluid motion. *Radiation* is the energy emitted by matter in the form of electromagnetic waves (or photons) as a result of the changes in the electronic configurations of the atoms or molecules. The Stefan-Boltzmann law gives the maximum rate of radiation that can be emitted from a surface at an absolute temperature Ts. *Thermal insulations* are materials or a combination of materials that are used primarily to provide resistance to heat flow. Thermal insulations are used for various reasons, such as energy conservation, personnel protection and comfort, maintenance of process temperature, reduction of temperature variation and fluctuations. Insulation materials are classified as fibrous, cellular, granular, and reflective, and they are available in the form of loose-fill, blankets and batts, rigid insulations, insulating cements, formed-in-place insulations, and reflective insulations. Important considerations in the selection of insulations are the purpose, environment, ease of handling and installation, and cost.

In Chapter 2, one-dimensional heat transfer through a simple or composite body exposed to convection from both sides to mediums at temperatures $T_{\infty 1}$, and $T_{\infty 2}$, can be expressed as

$$Q = \frac{T_{\infty 1} - T_{\infty 2}}{R_{total}} \qquad \text{(W)}$$

where R_{total} is the total thermal resistance between the two mediums. For a plane wall exposed to convection on both sides, the total resistance is expressed as

$$R_{total} = R_{conv,1} + R_{wall} + R_{conv,2} = \frac{1}{h_1 A} + \frac{L}{kA} + \frac{1}{h_2 A} \qquad (^0\text{C/W})$$

This relation can be extended to plane walls that consist of two or more layers by adding an additional resistance for each additional layer. The elementary thermal resistance relations can be expressed as follows:

Conduction resistance (plane wall): $$R_{wall} = \frac{L}{kA}$$

Convection resistance: $$R_{conv} = \frac{1}{hA}$$

Once the rate of heat transfer is available, the *temperature drop* across any layer can be determined from

$$\Delta T = QR \qquad (^0\text{C})$$

The thermal resistance concept can also be used to solve steady heat transfer problems involving parallel layers or combined series-parallel arrangements.

In Chapter 3, we had considered *natural convection* heat transfer where any fluid motion occurs by natural means such as buoyancy. The fluid velocities associated with natural convection are low. Therefore, the heat transfer coefficients encountered in natural convection are usually much lower than those encountered in forced convection. The upward force exerted by a fluid on a body completely or partially immersed in it is called the *buoyancy force,* whose magnitude is equal to the weight of the fluid displaced by the body. The instrument used in natural convection experiments most often is the *Mach-Zehnder interferometer,* which gives a plot of isotherms in the fluid in the vicinity of a surface. The flow regime in natural convection is governed by a dimensionless number called the *Grashof number,* which represents the ratio of the buoyancy force to the viscous force acting on the fluid. The Grashof number provides the main criterion in determining whether the fluid flow is laminar or turbulent in natural convection. Newton's law of cooling expresses the heat transfer rate in natural convection from a solid surface to the surrounding fluid. Most heat transfer relations in natural convection are based on experimental studies, and the simple empirical correlations for the average *Nusselt number* Nu in natural convection. Ra is the *Rayleigh number,* which is the product of the Grashof and Prandtl numbers.

In Chapter 4, radiation propagates in the form of electromagnetic waves. All matter whose temperature is above absolute zero continuously emits *thermal radiation* as a result of vibrational and rotational motions of molecules, atoms, and electrons of a substance. Temperature is a measure of the strength of these activities at the microscopic level. A *blackbody* is defined as a *perfect emitter and absorber of radiation.* At a specified temperature and wavelength, no surface can emit more energy than a blackbody. A blackbody absorbs *all* incident radiation, regardless of wavelength and direction. The radiation energy emitted by a blackbody per unit time and per unit surface area is called the *blackbody emissive power* and is expressed by the *Stefan-Boltzmann law.* The *emissivity* of a surface is defined as the ratio of the radiation emitted, by the surface to the radiation emitted by a blackbody at the same temperature.

The radiation energy incident on a surface per unit surface area per unit time is called irradiation and is denoted by G. When irradiation strikes a surface, part of it is absorbed, part of it is reflected, and the remaining part, if any, is transmitted. The fraction of irradiation absorbed by the surface is called the *absorptivity* α. The fraction of irradiation reflected by the surface is cal led the *reflectivity* ρ, and the fraction transmitted is called the *transmissivity* τ.

In Chapter 5, Convection is the mode of heat transfer that involves conduction as well as bulk fluid motion. The rate of convection heat transfer in external flow is expressed by *Newton'* s *law of cooling* as

$$Q_{conv} = hA(T_s - T_\infty) \quad \text{(W)}$$

where T_s is the surface temperature and T_∞ is the free-stream temperature. The heat transfer coefficient h is usually expressed in the dimensionless form as the *Nusselt number*

$$Nu = \frac{h\delta}{k}$$

Fluid flow over a flat plate starts out as smooth and streamlined but turns chaotic after some distance from the leading edge. The flow regime is said in the first cage to be *laminar,* characterized by smooth streamlines and highly ordered motion, and to be *turbulent* in the second cage, where it is characterized by velocity fluctuations and highly disordered motion. The intense mixing in turbulent flow enhances both the drag force and the heat transfer. The flow regime depends mainly on the ratio of the inertia forces to viscous forces in the fluid. This ratio is called the *Reynolds number* and is expressed as

$$\mathrm{Re} = \frac{V_\infty \delta}{\nu}$$

where V is the free-stream velocity V_m for external flow and the mean fluid velocity V_m for internal flow. Also, δ is the characteristic length of the geometry, which is the distance from the leading edge for a flat plate, the outer diameter for flow over cylinders or spheres, and the inner diameter for flow inside circular tubes. The Reynolds number at which the flow becomes turbulent is called the *critical*

Reynolds number. The value of the critical Reynolds number is about 5×10^5 for flow over a flat plate, 2×10^5 for flow over cylinders and spheres, and 2300 for flow inside tubes. For flow in a tube, the mean velocity V_m is the average velocity of the fluid. The mean temperature T_m at a cross-section can be viewed as the average temperature at that cross-section. The mean velocity V_m remains constant, but the mean temperature T_m changes along the tube unless the fluid is not heated or cooled. The heat transfer to a fluid during steady flow in a tube can be expressed as

$$Q = mC_p(T_e - T_i) \qquad \text{(kJ/s)}$$

where T_i and T_e are the mean fluid temperatures at the inlet and exit of the tube.

The conditions at the surface of a tube can usually be approximated with reasonable accuracy to be *constant surface temperature* (T_s = constant) or *constant surface heat flux* (q_s = constant). In the cage of q_s = constant, the rate of heat transfer can be expressed as

$$Q = q_s A = mC_p(T_e - T_i) \qquad \text{(W)}$$

Then mean fluid temperature at the tube exit becomes

$$T_e = T_i + \frac{q_s A}{mC_p}$$

In fully developed laminar flow, the friction factor is determined to be f = 64/Re. The Nusselt number is determined to be Nu = 3.66 for the cage of T_s = constant and Nu = 4.36 for the case of q_s = constant. The average Nusselt number for the hydrodynamically and/or thermally developed laminar flow in a circular tube is given as

$$Nu = 1.86\left(\frac{\mathrm{Re}\,\mathrm{Pr}\,D}{L}\right)^{1/3}\left(\frac{\mu_b}{\mu_s}\right)^{0.14} \qquad (\mathrm{Pr} > 0.5)$$

The recommended relations for the friction factor f and the Nusselt number for fully developed turbulent flow in smooth circular tubes are

$$f = 0.184\,\mathrm{Re}^{-0.2} \qquad \text{(Smooth tubes)}$$

and

$$\mathrm{Nu} = 0.023\ \mathrm{Re}^{0.8}\ \mathrm{Pr}^{1/3} \qquad (0.7 \le \mathrm{Pr} \le 160)(\mathrm{Re} > 10.000)$$

where n = 0.4 for *heating* and 0.3 for *cooling* of the fluid flowing through the tube. The fluid properties are evaluated at the *bulk mean fluid temperature* $T_b = \frac{1}{2}(T_i + T_e)$, which is the arithmetic average of the mean fluid temperatures at the inlet and the exit of the tube. The average Nusselt numbers for cross flow over a *cylinder* and *sphere* can be determined from

$$Nu = 0.3 + \frac{0.62\,\mathrm{Re}_D^{1/2}\,\mathrm{Pr}^{1/3}}{\left[1 + (0.4/\mathrm{Pr})^{2/3}\right]^{1/4}}\left[1 + \left(\frac{\mathrm{Re}_D}{282{,}000}\right)^{5/8}\right]^{4/5}$$

which is valid for Re Pr > 0.2, and

$$Nu_D = 2 + (0.4\,\mathrm{Re}_D^{1/2} + 0.06\,\mathrm{Re}_D^{2/3})\,\mathrm{Pr}^{0.4}\left(\frac{\mu}{\mu_s}\right)^{1/4}$$

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