NEAR EAST UNIVERSITY

FACULTY OF ENGINEERING
MECHANICAL ENGINEERING DEPARTMENT

ME 400 GRADUATION PROJECT

FORCED CONVECTION HEAT TRANSFER CORRELATIONS FLOW IN TUBES, OVER FLAT PLATES, ACROSS CYLINDERS, AND SPHERES

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ABSTRACT

The aim of this project is to examine both external and internal forced convection correlations for flow in tube, flow over flat plates, and flow across single cylinders, and single spheres.

In Chapter 1, brief information is given about heat transfer which is divided in three groups as conduction, convection, and radiation. The equations which are related with the every mode of heat transfer.

In Chapter 2, convection heat transfer is introduced, which is classified as natural (or free) and forced convection, depending on how the fluid motion is initiated. Brief information about forced and natural convection is given. The dimensionless of Reynolds, Nusselt, Grassohf numbers are explained respectively.

In Chapter 3, internal forced convection is explained according to flow types in tubes. The equations according to the flow conditions in tubes are given in details.

In Chapter 4, forced heat convection correlations flow over flat plates, and across cylinders and spheres are discussed. Nusselt and Reynolds numbers are given in details according to external laminar, and turbulent flow.

CHAPTER 1

INTRODUCTION TO HEAT TRANSFER

1.1 Introduction

In this Chapter brief information is given about heat transfer that is the energy transfer from the warm medium to cold medium. Shortly historical background and application areas of heat transfer are explained. The three basic mechanisms of heat transfer are presented which are 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 it involves the combined effects of conduction and fluid motion. Radiation is the energy emitted by matter in the form of electromagnetic waves as a result of the changes in the electronic configurations of the atoms or molecules. At the end the simultaneous heat transfer is discussed.

1.2 Historical Background

Heat has always been perceived to be something that produces in us a sensation of warmth, and one would think that the nature of heat is one of the first things understood by mankind. But it was only in the middle of the 19th century that we had a true physical understanding of the nature of the heat, tanks to the development at that time of the kinetic theory, which treats molecules as tiny balls that are emotion and thus possess kinetic energy. Heat is then defined as the energy associated with the random motion of atoms and molecules. Although it was suggested in the 18th and early 19th centuries that heat is the manifestation of motion at the molecular level, the prevailing view of heat until the middle of the 19th century was based on the caloric theory proposed by the French chemist Antoine Lavoisier (1743 – 1794) in 1789. The caloric theory asserts that heat is fluid-like substance called the caloric that is a mass less, colorless, odorless, and tasteless substance that can poured from one body into another. When caloric was added to a body, its temperature increased; and when caloric was removed from a body, its temperature decreased. When a body could not contain any more caloric, much the same way as when a glass of water could not dissolve anymore salt or sugar, the body was said to be saturated with caloric. This interpretation gave rise to the terms saturated liquid and saturated vapor that are still in use today.

1.3 Thermodynamics and Heat Transfer

We all know from experience as a cold canned drink left in a room warms up and a warm canned drink left in a refrigerator cools down. This is the energy transfer from the warm medium to the cold one. The energy transfer is always from the higher temperature medium to the lower temperature one, and the energy transfer stops when the two mediums reach the same temperature.

In this subject we are interested in heat, which is the form of energy that can be transfered from one system to another as a result of temperature difference. The science which is interested with the rates of energy transfers is heat transfer.We can determine the amount of heat transfer for any system under going any process using at termodynamic analysis alone. The reason is that termodynamics is concerned with the amount of heat transfer as a system under goes a process from one equilibrium state to another, and it gives no indication about how long the process will take.

The amount of heat transferred from a thermos bottle as the hot coffee inside cools from 90^0C to 80^0C can be determined by a termodynamic analysis alone. But a typical user or designer of a thermos is primarily interested in how long it will be before the hot coffee inside cools to 80^0C, and a thermodynamic analysis can not answer this question. Determining the rates of heat transfer to or from a system and thus the times of cooling or heating, as well as the variation of the temperature, is the subject of heat transfer.

Thermodynamic deals with equilibrium states and changes from one equilibrium state to another. Heat transfer, on the other hand, deals with systems that lacks thermal equilibrium, and thus it's a nonequilibrium phenomeno. Therefore, the study of heat transfer can not be based on the principles of thermodynamics alone. However, the lows of thermodynamics lay the frame work for the science of heat transfer. The first law requires that the rate of energy transfering in to the system be equal to the rate of increase of the energy of that system. The second law requires that heat be transfered in the direction of decreasing temperature.

1.3.1 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 as shown in Figure 1.1.

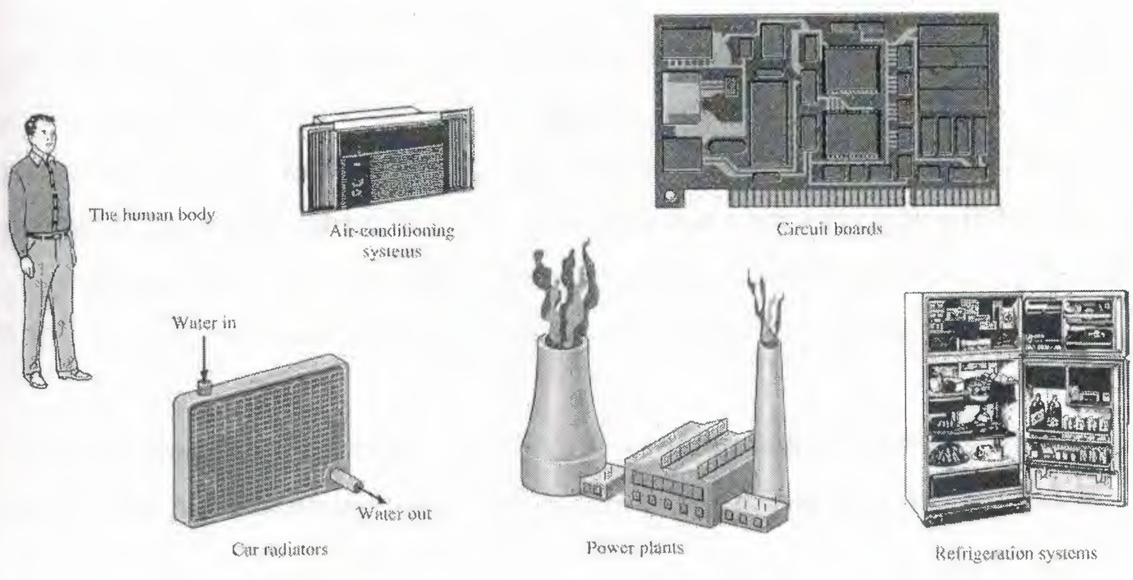

Figure 1.1 some application areas of heat transfer.

1.4 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.

1.4.1 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. The temperature difference across the wall is;

$\Delta T = T_2 - T_1$. Experiments have shown that the rate of heat transfer $\dot{Q}$ 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;

$$\dot{Q}_{cond.} = -kA\frac{\Delta T}{\Delta x} \qquad \text{(W)}$$

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 as shown in Figure 1.2. Here dt/dx is the temperature gradient, which is the slope of the temperature curve on a T-x diagram, at location x. The heat transfer area A is always normal to the direction of heat transfer.

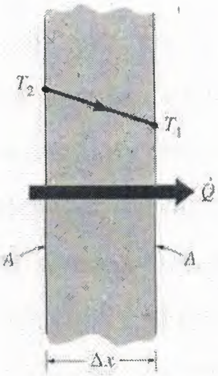

Figure 1.2 Heat conduction through a large plane wall of thickness Δx and area A.

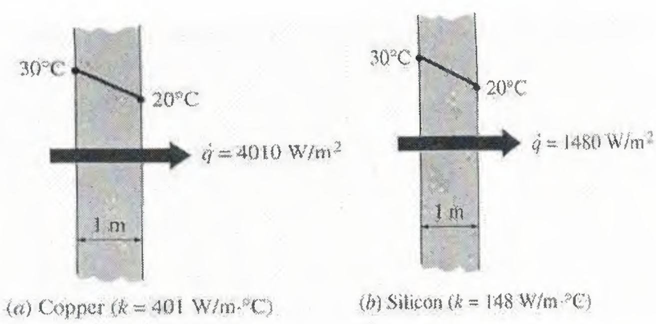

Figure 1.3 The rate of heat conduction through a solid is directly proportional to its thermal conductivity.

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 $\dot{Q}_{cond.} = kA\dfrac{\Delta T}{\Delta x}$ (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 1.1. 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.

Table 1.1 The thermal conductivities of some materials at room temperature.

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

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 $\dot{Q}_{cond.} = kA\frac{\Delta T}{\Delta x}$ (W) together with other known quantities give the thermal conductivity as shown in Figure 1.4 below.

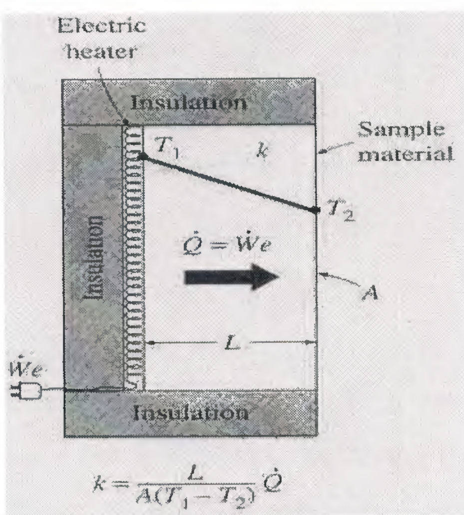

Figure 1.4 A simple experimental setup to determine the thermal conductivity of a material.

The mechanism of heat conduction in a liquid is complicated by the fact that the molecules are more closely spaced, and they exert a stronger intermolecular force field. The thermal conductivities of liquids usually lie between those of solids and gases. In solids, heat conduction is due to two effects: the lattice vibrational waves induced by the vibrational motions of the molecules positioned at relatively fixed positions in a periodic manner called a lattice, and the energy transported via the free flow of electrons in the solid. The thermal conductivity of a solid is obtained by adding the lattice and electronic components.

1.4.2 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 as shown in Figure 1.5. Energy is first transferred to the air layer adjacent to the block by conduction. This energy is then carried away from the surface by convection; that is, by the combined effects of conduction within the air that is due to 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.

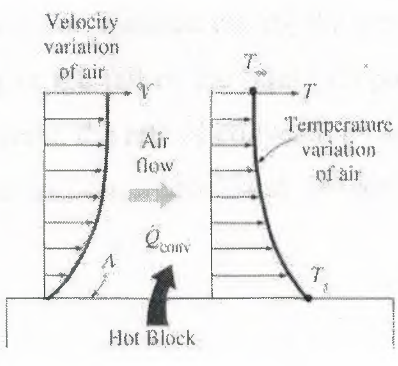

Figure 1.5 Heat transfer from a hot surface to air by convection.

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. For example, in the absence of a fan, heat transfer from the surface of the hot block in

Figure 1.5 will be by natural convection since any motion in the air in this case will be due to the rise of the warmer (and thus lighter) air near the surface and the fall of the cooler (and thus heavier) air to 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.

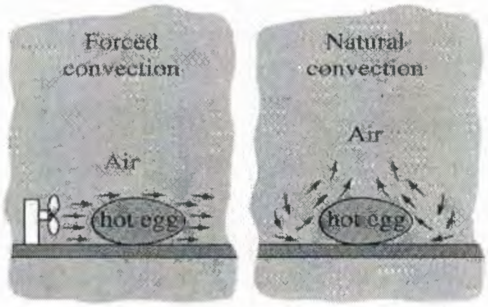

Figure 1.6 The cooling of a boiled egg by forced and natural convection.

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;

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

where h is the convection heat transfer coefficient in $W/m^2 \cdot {}^0C$, 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. Some people do not consider convection to be a fundamental mechanism of heat transfer since it is essentially heat conduction in the presence of fluid motion. Thus, it is practical to recognize convection as a separate heat transfer mechanism despite the valid arguments to the contrary.

1.4.3 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 idealized surface that emits radiation at this maximum rate is called a blackbody, and the radiation emitted by a blackbody is called blackbody radiation is shown in Figure 1.7.The radiation emitted by all real surfaces is less than the radiation emitted by a blackbody at the same temperature.

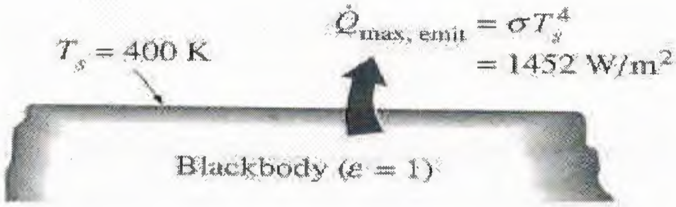

Figure 1.7 Blackbody radiation represents the maximum amount of radiation that can be emitted from a surface at a specified temperature.

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 Figure 1.9.

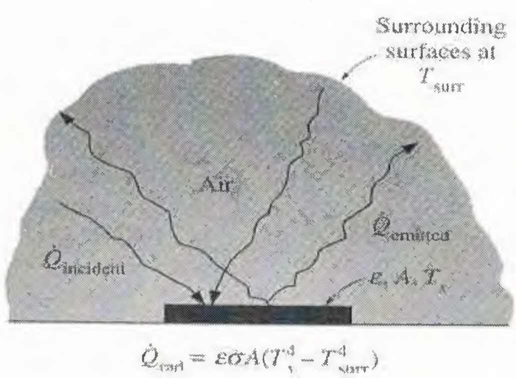

Figure 1.9 Radiation heat transfer between a surface and the surfaces surrounding it.

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

Radiation is usually significant relative to conduction or natural convection. Thus radiation in forced convection applications is normally disregarded, especially when the surfaces involved have low emissivities and low to moderate temperatures.

SUMMARY

In this chapter, the basic concept of heat transfer is introduced and discussed. Historical background and application areas of heat transfer are explained briefly.

Three different ways of heat transfer are explained which are conduction, convection, and radiation heat transfer. Brief information about conduction, convection, and radiation is given with using some figures and tables which are related with them. The equations about heat transfer mechanisms are given and explained with their parameters.

CHAPTER 2

CONVECTION HEAT TRANSFER

2.1 Introduction

In this Chapter convection heat transfer is considered, which is the mode of energy transfer between a solid surface and the adjacent liquid or gas that is in motion. Convection is classified as natural and 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 fun. In natural convection, any fluid motion is caused by natural means such as the buoyancy effect, which manifests itself as the rise of warmer fluid and the full of the cooler fluid. The dimensionless of Reynolds Prandtl, and Grashof numbers are discussed which numbers are very important for solving heat transfer problems.

2.2 Physical Mechanism of Forced Convection

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 is shown below in Figure 2.1.

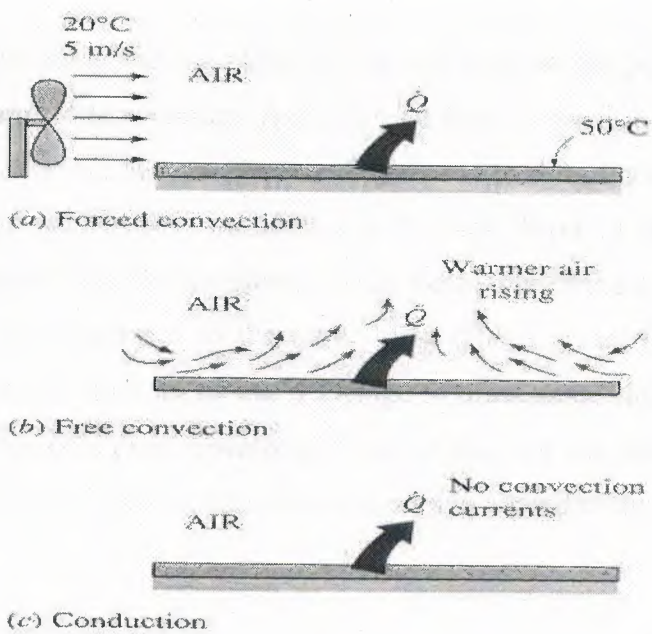

Figure 2.1 Heat transfer from a hot surface to the surrounding fluid by convection and conduction.

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 Figure2.2 below.

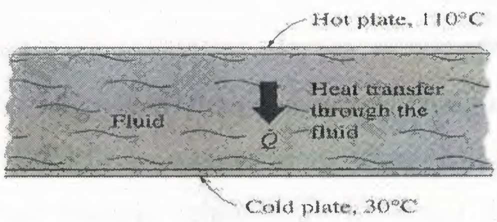

Figure 2.2 Heat transfer through a fluid sandwiched between two parallel plates.

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 2.3. 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.

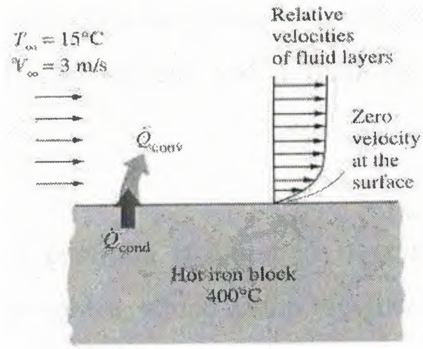

Figure 2.3 The cooling of a hot block by forced convection.

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:

$$\dot{q}_{conv} = h(T_s - T_\infty) \quad (W/m^2) \qquad \text{or;}$$

$$\dot{Q}_{conv} = hA(T_s - T_\infty) \quad (W)$$

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 2.4 below.

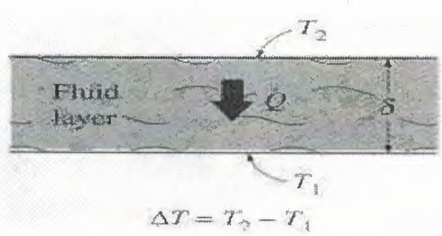

Figure 2.4 Heat transfer through a fluid layer of thickness δ and temperature difference ΔT.

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;

$$\dot{q}_{conv} = h\Delta T \qquad \text{and}$$

$$\dot{q}_{conv} = k\frac{\Delta T}{\delta}$$

Taking their ratio gives

$$\frac{\dot{q}_{conv}}{\dot{q}_{cond}} = \frac{h\Delta T}{k\Delta T / \delta} = \frac{h\delta}{k} = Nu$$

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 as shown in Figure 2.5 below.

Figure 2.5 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.

2.2.1 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 is shown in Figure 2.6.

Figure 2.6 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 2.7. 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.

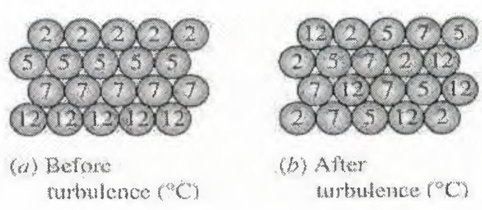

Figure 2.7 The intense mixing in turbulent flow brings fluid particles at different temperatures into close contact, and thus enhances heat transfer.

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.

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. 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.

2.2.2 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;

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

where

V_{∞} : free-stream velocity, m/s

δ : characteristic length of the geometry, m

v : 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.

2.2.3 Thermal Boundary Layer

A velocity boundary layer develops when a fluid flows over a surface as a result of the fluid layer adjacent to the surface assuming the surface velocity (i.e.. zero velocity relative to the surface). Also the velocity boundary layer is defined as the region in which the fluid velocity varies from zero to 0.99V. Likewise, a thermal boundary layer develops when a fluid at a specified temperature flow over a surface that is at a different temperature.

Consider the flow of a fluid at a uniform temperature of T_∞ over an isothermal that plate at a temperature Ts. The fluid particles in the layer adjacent to the surface will reach thermal equilibrium with the plate and assume the surface temperature Ts. These fluid particles will then exchange energy with the particles in the adjoining fluid layer, and so on. As a result a temperature profile will develop in the flow field that ranges from Ts at the surface to T_∞ sufficiently far from the surface. The flow region over the surface in which the temperature variation in the direction normal to the surface is significant is the thermal boundary layer.The convection heat transfer rate anywhere along the surface is directly related to the temperature gradient at that location. Therefore, the shape of the temperature profile in the thermal boundary layer dictates the convection heat transfer between a solid surface and the fluid flowing over it. In flow over a heated (or cooled) surface, both velocity and thermal boundary layers will develop simultaneously. Noting that the fluid velocity will have a strong influence on the temperature profile, the development of the velocity boundary layer will have a strong effect on the convection heat transfer. The relative thickness of the velocity and the thermal boundary layers is best described by the dimensionless parameter Prandtl number, defined as:

Pr : Molecular diffusivitys of momentum / Molecular diffusivity of heat

$$Pr=v/\alpha=\mu Cp/k$$

It is named after Ludwig Prandtl, who introduced the concept of boundary layerin 1904 and made significant contributions to boundary layer theory.

2.3 Physical Mechanism of Natural Convection

A lot of 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.

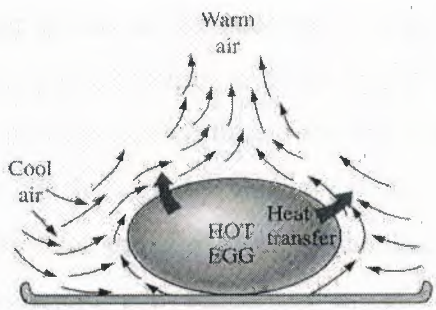

Figure 2.8 The cooling of a boiled egg in a cooler environment

by natural convection.

We know that a hot boiled egg (or a hot baked potato) on a plate eventually cools to the surrounding air temperature is shown in figure Figure 2.8 above. 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.

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 2.9 below. Note that the direction of fluid motion is reversed in this case.

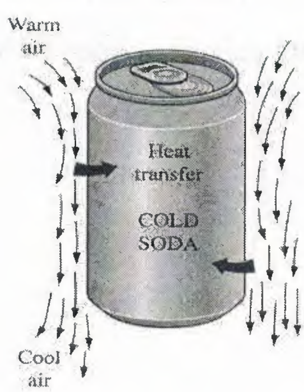

Figure 2,9 The warming up of a cold drink in a warmer environment by natural convection.

2.3.1 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,

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

Since $\Delta\rho = \rho\beta\Delta T$, it is formally expressed below and shown in Figure 2.12 below.

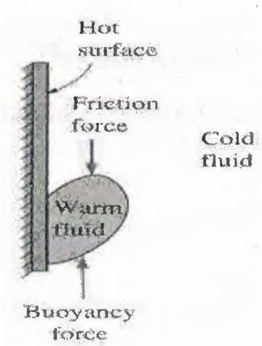

Figure 2.12 The Grashof number Gr is a measure of the relative magnitudes of the buoyancy force and the opposing friction force acting on the fluid

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

g : gravitational acceleration, m/s^2

β : coefficient of volume expansion, 1/K (β = *1*/T 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

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

The Grashof number plays important role in natural convection like Reynolds number which is important in forced convection. 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

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

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

SUMMARY

In this chapter, convection heat transfer mechanism is defined shortly which is divided in two groups as forced heat convection and natural forced convection. The physical mechanisms of natural and forced convection are explained with some figures. Reynolds, Prandtl, Nusselt, and Grashof numbers are examined respectively.

CHAPTER 3

INTERNAL FORCED CONVECTION

3.1 Introduction

In this chapter internal forced convection is explained which is related with flow in tubes. In internal flow, 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.

Flow inside tubes for both laminar and turbulent flow conditions are examined. Nusselt and Reynolds numbers are examined according to flow in tubes as laminar and turbulant. The effects of laminar flow in tubes of various cross sections to Nusselt number and friction factor are examined and given in a table respectively.

3.2 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, the friction and heat transfer coefficients that are directly related to the pressure drop and heat flux for flow through tubes will be discussed. 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 as shown in Figure. 3.1.

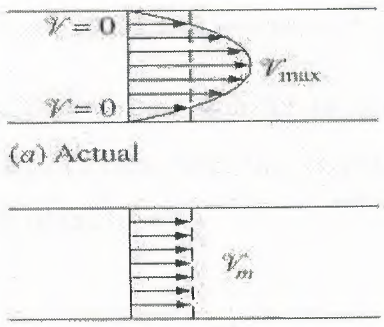

Figure 3.1 Actual and idealized velocity profiles for flow in tube(the mass flow rate of the fluid is the same for both cases.).

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

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

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.

$$E = mC_p T_m = \int_m C_p T \delta m = \int_{A_c} C_p T(\rho V dA_c) \quad \text{(kg/s)}$$

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 as shown in Figure 3.2. We 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;:

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

where T_i and T_e are the mean fluid temperatures at the inlet and exit of the tube, respectively, and $\dot{Q}$ 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.

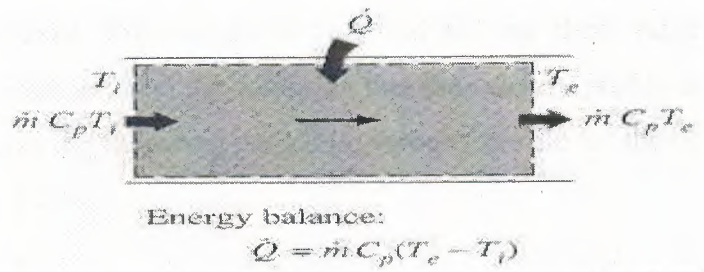

Figure 3.2 The heat transfer to a fluid flowing in profiles for flowing in a tube is equal to the increase in the energy of the fluid.

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.

3.2.1 Laminar Flow in Tubes

The 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, $f = \frac{64}{\text{Re}}$ (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

Nu = 3.66 for T_s = constant (laminar flow)

Nu = 4.36 for $q_s = constant$ (laminar flow)

Sieder and Tate as give a general relation for average Nusselt number for the hydrodynamically and thermally developing laminar flow in a circular tube:

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

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.

3.2.2 Turbulent Flow in Tubes

The 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 3.3, which is known as the Moody diagram.

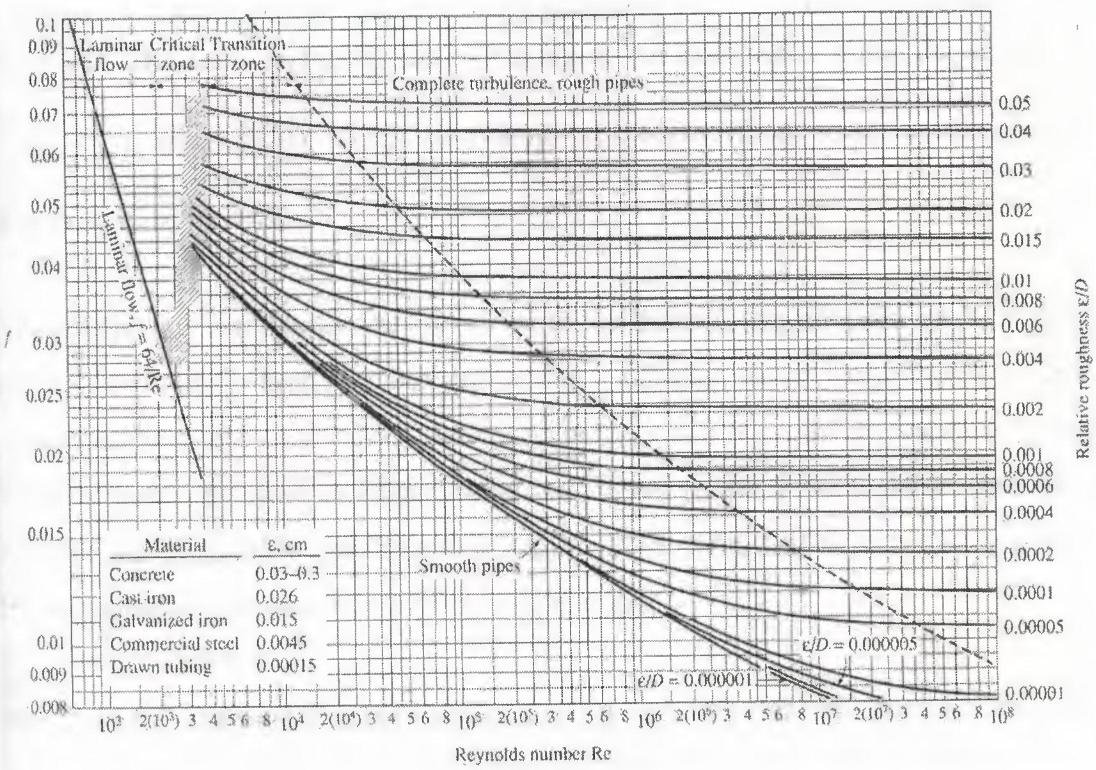

Figure 3.3 Friction factor for fully developed flow in circular tubes. (the Moody chart).

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

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

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

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

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

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

where n = 0.4 for heating and 0.3 for cooling of the fluid flowing through the tube. 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 ≤Re ≤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,

NEAR EAST UNIVERSITY

FACULTY OF ENGINEERING
MECHANICAL ENGINEERING DEPARTMENT

ME 400 GRADUATION PROJECT

FORCED CONVECTION HEAT TRANSFER CORRELATIONS FLOW IN TUBES, OVER FLAT PLATES, ACROSS CYLINDERS, AND SPHERES

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EXTERNAL FORCED CONVECTION

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First of all I would like to thank Assist. Prof. Dr. Guner ÖZMEN to be my supervisor. I successfully accomplish many difficulties about my project under her guidance

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ABSTRACT

The aim of this project is to examine both external and internal forced convection correlations for flow in tube, flow over flat plates, and flow across single cylinders, and single spheres.

In Chapter 1, brief information is given about heat transfer which is divided in three groups as conduction, convection, and radiation. The equations which are related with the every mode of heat transfer.

In Chapter 2, convection heat transfer is introduced, which is classified as natural (or free) and forced convection, depending on how the fluid motion is initiated. Brief information about forced and natural convection is given. The dimensionless of Reynolds, Nusselt, Grassohf numbers are explained respectively.

In Chapter 3, internal forced convection is explained according to flow types in tubes. The equations according to the flow conditions in tubes are given in details.

In Chapter 4, forced heat convection correlations flow over flat plates, and across cylinders and spheres are discussed. Nusselt and Reynolds numbers are given in details according to external laminar, and turbulent flow.

CHAPTER 1

INTRODUCTION TO HEAT TRANSFER

1.1 Introduction

In this Chapter brief information is given about heat transfer that is the energy transfer from the warm medium to cold medium. Shortly historical background and application areas of heat transfer are explained. The three basic mechanisms of heat transfer are presented which are 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 it involves the combined effects of conduction and fluid motion. Radiation is the energy emitted by matter in the form of electromagnetic waves as a result of the changes in the electronic configurations of the atoms or molecules. At the end the simultaneous heat transfer is discussed.

1.2 Historical Background

Heat has always been perceived to be something that produces in us a sensation of warmth, and one would think that the nature of heat is one of the first things understood by mankind. But it was only in the middle of the 19^{th} century that we had a true physical understanding of the nature of the heat, tanks to the development at that time of the kinetic theory, which treats molecules as tiny balls that are emotion and thus possess kinetic energy. Heat is then defined as the energy associated with the random motion of atoms and molecules. Although it was suggested in the 18^{th} and early 19^{th} centuries that heat is the manifestation of motion at the molecular level, the prevailing view of heat until the middle of the 19^{th} century was based on the caloric theory proposed by the French chemist Antoine Lavoisier (1743 – 1794) in 1789. The caloric theory asserts that heat is fluid-like substance called the caloric that is a mass less, colorless, odorless, and tasteless substance that can poured from one body into another. When caloric was added to a body, its temperature increased; and when caloric was removed from a body, its temperature decreased. When a body could not contain any more caloric, much the same way as when a glass of water could not dissolve anymore salt or sugar, the body was said to be saturated with caloric. This interpretation gave rise to the terms saturated liquid and saturated vapor that are still in use today.

1.3 Thermodynamics and Heat Transfer

We all know from experience as a cold canned drink left in a room warms up and a warm canned drink left in a refrigerator cools down. This is the energy transfer from the warm medium to the cold one. The energy transfer is always from the higher temperature medium to the lower temperature one, and the energy transfer stops when the two mediums reach the same temperature.

In this subject we are interested in heat, which is the form of energy that can be transfered from one system to another as a result of temperature difference. The science which is interested with the rates of energy transfers is heat transfer.We can determine the amount of heat transfer for any system under going any process using at termodynamic analysis alone. The reason is that termodynamics is concerned with the amount of heat transfer as a system under goes a process from one equilibrium state to another, and it gives no indication about how long the process will take.

The amount of heat transferred from a thermos bottle as the hot coffee inside cools from 90^0C to 80^0C can be determined by a termodynamic analysis alone. But a typical user or designer of a thermos is primarily interested in how long it will be before the hot coffee inside cools to 80^0C, and a thermodynamic analysis can not answer this question. Determining the rates of heat transfer to or from a system and thus the times of cooling or heating, as well as the variation of the temperature, is the subject of heat transfer.

Thermodynamic deals with equilibrium states and changes from one equilibrium state to another. Heat transfer, on the other hand, deals with systems that lacks thermal equilibrium, and thus it's a nonequilibrium phenomeno. Therefore, the study of heat transfer can not be based on the principles of thermodynamics alone. However, the lows of thermodynamics lay the frame work for the science of heat transfer. The first law requires that the rate of energy transfering in to the system be equal to the rate of increase of the energy of that system. The second law requires that heat be transfered in the direction of decreasing temperature.

1.3.1 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 as shown in Figure 1.1.

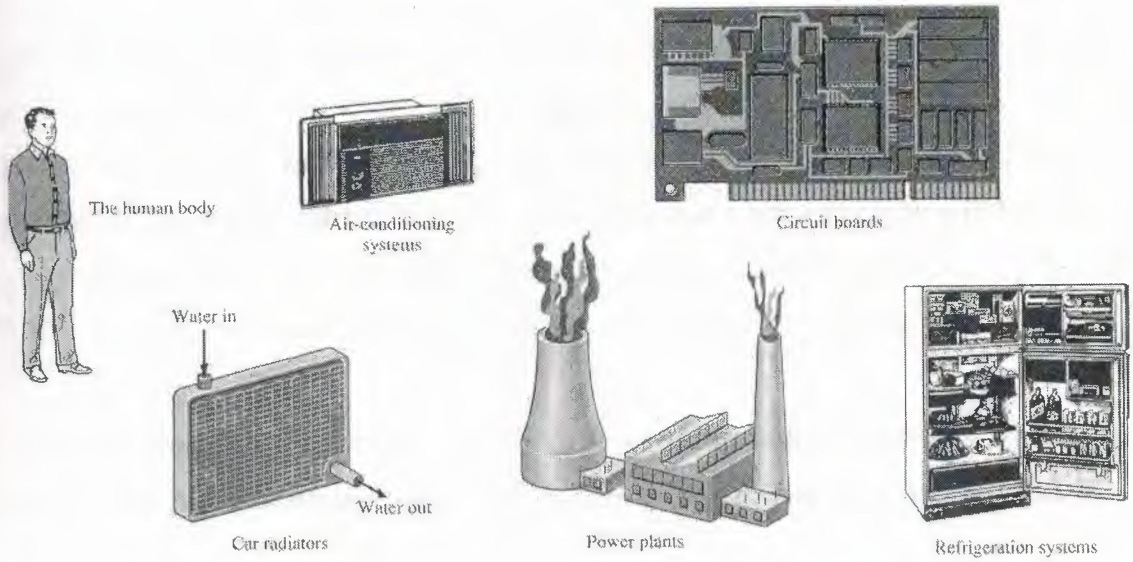

Figure 1.1 some application areas of heat transfer.

1.4 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.

1.4.1 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. The temperature difference across the wall is;

$\Delta T = T_2 - T_1$. Experiments have shown that the rate of heat transfer $\dot{Q}$ 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;

$$\dot{Q}_{cond.} = -kA\frac{\Delta T}{\Delta x} \qquad \text{(W)}$$

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 as shown in Figure 1.2. Here dt/dx is the temperature gradient, which is the slope of the temperature curve on a T-x diagram, at location x. The heat transfer area A is always normal to the direction of heat transfer.

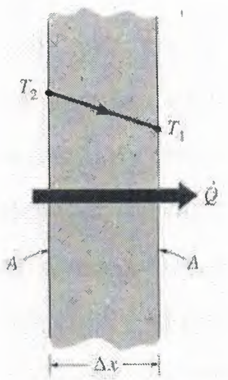

Figure 1.2 Heat conduction through a large plane wall of thickness Δx and area A.

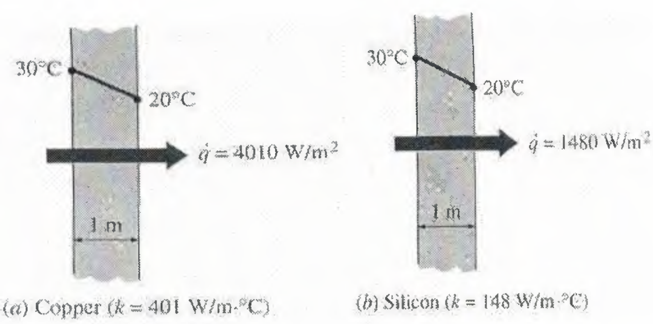

Figure 1.3 The rate of heat conduction through a solid is directly proportional to its thermal conductivity.

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 $\dot{Q}_{cond.} = kA\frac{\Delta T}{\Delta x}$ (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 1.1. 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.

Table 1.1 The thermal conductivities of some materials at room temperature.

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

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 $\dot{Q}_{cond.} = kA\frac{\Delta T}{\Delta x}$ (W) together with other known quantities give the thermal conductivity as shown in Figure 1.4 below.

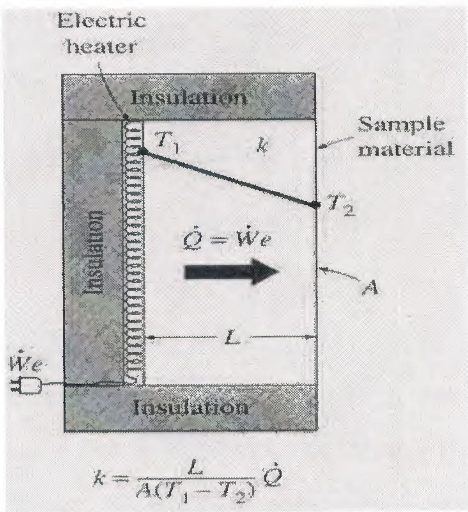

Figure 1.4 A simple experimental setup to determine the thermal conductivity of a material.

The mechanism of heat conduction in a liquid is complicated by the fact that the molecules are more closely spaced, and they exert a stronger intermolecular force field. The thermal conductivities of liquids usually lie between those of solids and gases. In solids, heat conduction is due to two effects: the lattice vibrational waves induced by the vibrational motions of the molecules positioned at relatively fixed positions in a periodic manner called a lattice, and the energy transported via the free flow of electrons in the solid. The thermal conductivity of a solid is obtained by adding the lattice and electronic components.

1.4.2 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 as shown in Figure 1.5. Energy is first transferred to the air layer adjacent to the block by conduction. This energy is then carried away from the surface by convection; that is, by the combined effects of conduction within the air that is due to 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.

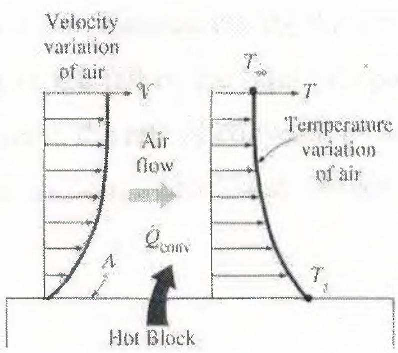

Figure 1.5 Heat transfer from a hot surface to air

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. For example, in the absence of a fan, heat transfer from the surface of the hot block in

Figure 1.5 will be by natural convection since any motion in the air in this case will be due to the rise of the warmer (and thus lighter) air near the surface and the fall of the cooler (and thus heavier) air to 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.

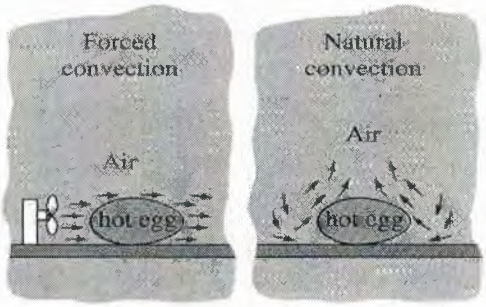

Figure 1.6 The cooling of a boiled egg by forced and natural convection.

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;

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

where h is the convection heat transfer coefficient in $W/m^2 \cdot {}^0C$, 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. Some people do not consider convection to be a fundamental mechanism of heat transfer since it is essentially heat conduction in the presence of fluid motion. Thus, it is practical to recognize convection as a separate heat transfer mechanism despite the valid arguments to the contrary.

1.4.3 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 idealized surface that emits radiation at this maximum rate is called a blackbody, and the radiation emitted by a blackbody is called blackbody radiation is shown in Figure 1.7.The radiation emitted by all real surfaces is less than the radiation emitted by a blackbody at the same temperature.

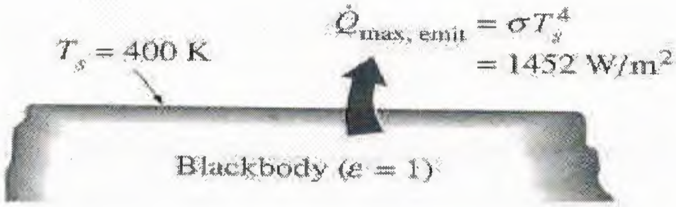

Figure 1.7 Blackbody radiation represents the maximum amount of radiation that can be emitted from a surface at a specified temperature.

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 Figure 1.9.

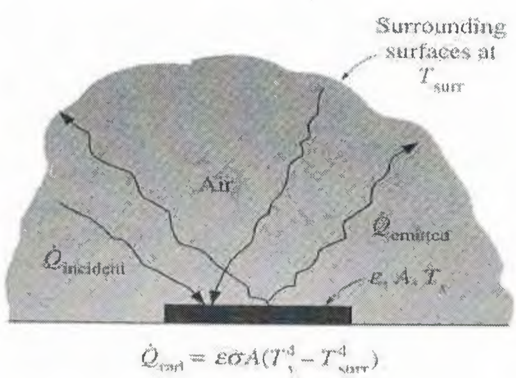

Figure 1.9 Radiation heat transfer between a surface and the surfaces surrounding it.

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

Radiation is usually significant relative to conduction or natural convection. Thus radiation in forced convection applications is normally disregarded, especially when the surfaces involved have low emissivities and low to moderate temperatures.

SUMMARY

In this chapter, the basic concept of heat transfer is introduced and discussed. Historical background and application areas of heat transfer are explained briefly.

Three different ways of heat transfer are explained which are conduction, convection, and radiation heat transfer. Brief information about conduction, convection, and radiation is given with using some figures and tables which are related with them. The equations about heat transfer mechanisms are given and explained with their parameters.

CHAPTER 2

CONVECTION HEAT TRANSFER

2.1 Introduction

In this Chapter convection heat transfer is considered, which is the mode of energy transfer between a solid surface and the adjacent liquid or gas that is in motion. Convection is classified as natural and 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 fun. In natural convection, any fluid motion is caused by natural means such as the buoyancy effect, which manifests itself as the rise of warmer fluid and the full of the cooler fluid. The dimensionless of Reynolds Prandtl, and Grashof numbers are discussed which numbers are very important for solving heat transfer problems.

2.2 Physical Mechanism of Forced Convection

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 is shown below in Figure 2.1.

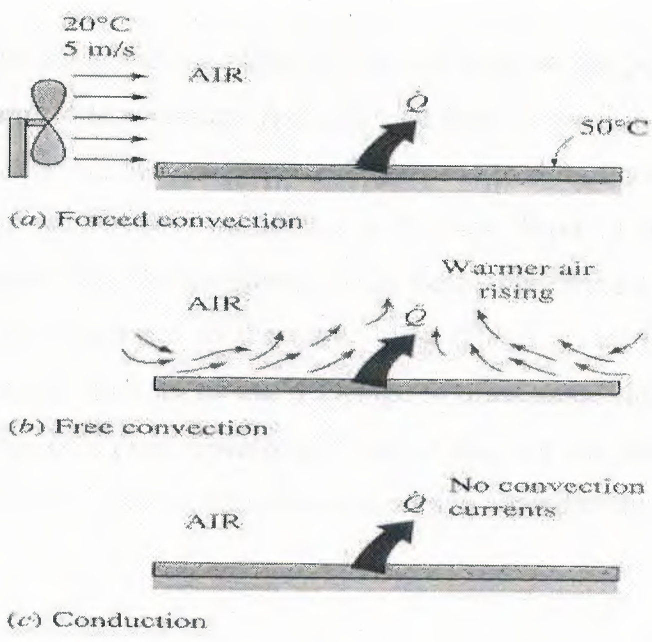

Figure 2.1 Heat transfer from a hot surface to the surrounding fluid by convection and conduction.

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 Figure2.2 below.

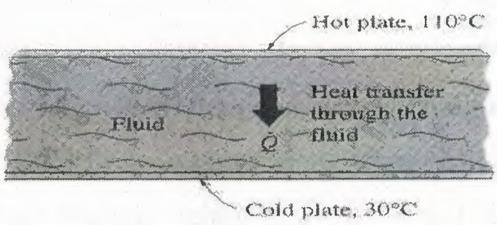

Figure 2.2 Heat transfer through a fluid sandwiched between two parallel plates.

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 2.3. 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.

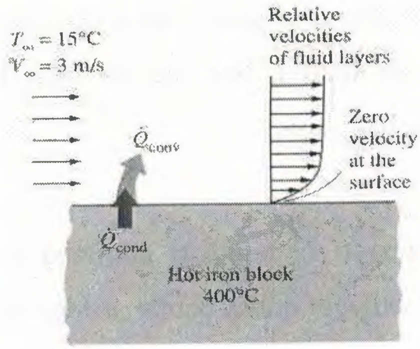

Figure 2.3 The cooling of a hot block by forced convection.

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:

$$\dot{q}_{conv} = h(T_s - T_\infty) \quad (\text{W/m}^2) \qquad \text{or;}$$

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

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 2.4 below.

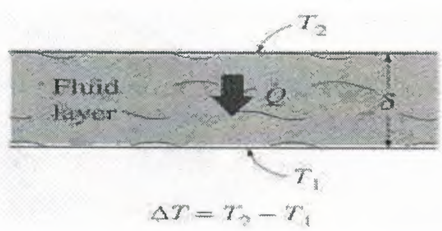

Figure 2.4 Heat transfer through a fluid layer of thickness δ and temperature difference ΔT.

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;

$$\dot{q}_{conv} = h\Delta T \qquad \text{and}$$

$$\dot{q}_{conv} = k\frac{\Delta T}{\delta}$$

Taking their ratio gives

$$\frac{\dot{q}_{conv}}{\dot{q}_{cond}} = \frac{h\Delta T}{k\Delta T/\delta} = \frac{h\delta}{k} = Nu$$

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 as shown in Figure 2.5 below.

Figure 2.5 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.

2.2.1 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 is shown in Figure 2.6.

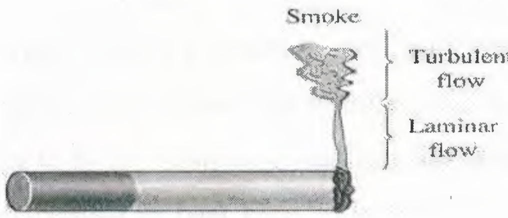

Figure 2.6 Laminar and turbulent flow regimes of cigarette smoke

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 2.7. 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.

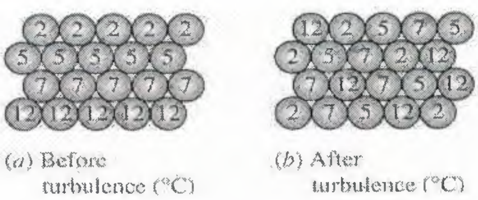

Figure 2.7 The intense mixing in turbulent flow brings fluid particles at different temperatures into close contact, and thus enhances heat transfer.

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.

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. 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.

2.2.2 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;

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

where

V_∞ : free-stream velocity, m/s

δ : characteristic length of the geometry, m

v : 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.

2.2.3 Thermal Boundary Layer

A velocity boundary layer develops when a fluid flows over a surface as a result of the fluid layer adjacent to the surface assuming the surface velocity (i.e.. zero velocity relative to the surface). Also the velocity boundary layer is defined as the region in which the fluid velocity varies from zero to 0.99V. Likewise, a thermal boundary layer develops when a fluid at a specified temperature flow over a surface that is at a different temperature.

Consider the flow of a fluid at a uniform temperature of T_∞ over an isothermal that plate at a temperature Ts. The fluid particles in the layer adjacent to the surface will reach thermal equilibrium with the plate and assume the surface temperature Ts. These fluid particles will then exchange energy with the particles in the adjoining fluid layer, and so on. As a result a temperature profile will develop in the flow field that ranges from Ts at the surface to T_∞ sufficiently far from the surface. The flow region over the surface in which the temperature variation in the direction normal to the surface is significant is the thermal boundary layer.The convection heat transfer rate anywhere along the surface is directly related to the temperature gradient at that location. Therefore, the shape of the temperature profile in the thermal boundary layer dictates the convection heat transfer between a solid surface and the fluid flowing over it. In flow over a heated (or cooled) surface, both velocity and thermal boundary layers will develop simultaneously. Noting that the fluid velocity will have a strong influence on the temperature profile, the development of the velocity boundary layer will have a strong effect on the convection heat transfer. The relative thickness of the velocity and the thermal boundary layers is best described by the dimensionless parameter Prandtl number, defined as:

Pr : Molecular diffusivitys of momentum / Molecular diffusivity of heat

$$Pr = v/\alpha = \mu Cp/k$$

It is named after Ludwig Prandtl, who introduced the concept of boundary layerin 1904 and made significant contributions to boundary layer theory.

2.3 Physical Mechanism of Natural Convection

A lot of 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.

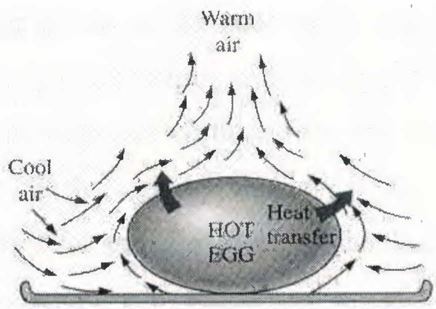

Figure 2.8 The cooling of a boiled egg in a cooler environment

by natural convection.

We know that a hot boiled egg (or a hot baked potato) on a plate eventually cools to the surrounding air temperature is shown in figure Figure 2.8 above. 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.

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 2.9 below. Note that the direction of fluid motion is reversed in this case.

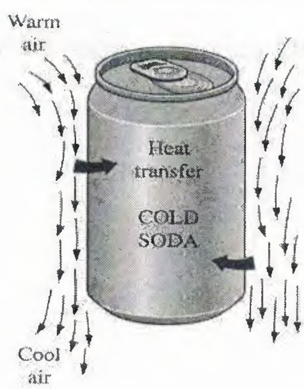

Figure 2,9 The warming up of a cold drink in a warmer environment by natural convection.

2.3.1 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 below and shown in Figure 2.12 below.

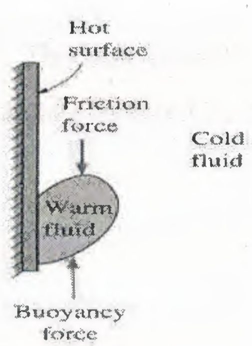

Figure 2.12 The Grashof number Gr is a measure of the relative magnitudes of the buoyancy force and the opposing friction force acting on the fluid

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

g : gravitational acceleration, m/s^2

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

T_s : temperature of the surface, ^0C

T_∞: temperature of the fluid sufficiently far from the surface, ^0C

δ: characteristic length of the geometry, m

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

The Grashof number plays important role in natural convection like Reynolds number which is important in forced convection. 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

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

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

SUMMARY

In this chapter, convection heat transfer mechanism is defined shortly which is divided in two groups as forced heat convection and natural forced convection. The physical mechanisms of natural and forced convection are explained with some figures. Reynolds, Prandtl, Nusselt, and Grashof numbers are examined respectively.

CHAPTER 3

INTERNAL FORCED CONVECTION

3.1 Introduction

In this chapter internal forced convection is explained which is related with flow in tubes. In internal flow, 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.

Flow inside tubes for both laminar and turbulent flow conditions are examined. Nusselt and Reynolds numbers are examined according to flow in tubes as laminar and turbulant. The effects of laminar flow in tubes of various cross sections to Nusselt number and friction factor are examined and given in a table respectively.

3.2 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, the friction and heat transfer coefficients that are directly related to the pressure drop and heat flux for flow through tubes will be discussed. 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 as shown in Figure. 3.1.

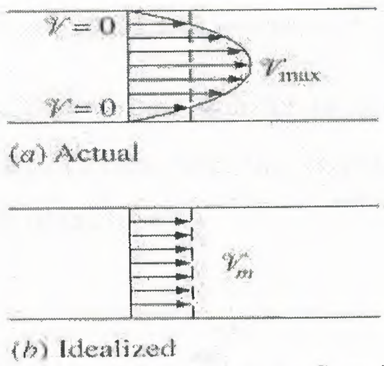

Figure 3.1 Actual and idealized velocity profiles for flow in tube(the mass flow rate of the fluid is the same for both cases.).

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

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

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.

$$E = mC_pT_m = \int_{m} C_pT\delta m = \int_{A_c} C_pT(\rho V dA_c) \quad \text{(kg/s)}$$

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 as shown in Figure 3.2. We 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;:

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

where T_i and T_e are the mean fluid temperatures at the inlet and exit of the tube, respectively, and $\dot{Q}$ 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.

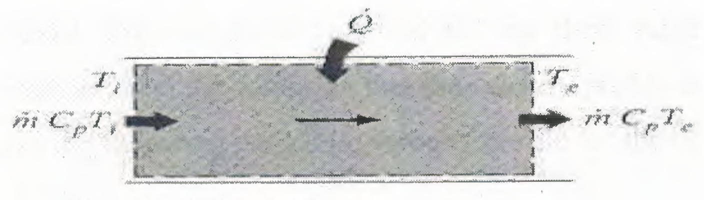

Figure 3.2 The heat transfer to a fluid flowing in profiles for flowing in a tube is equal to the increase in the energy of the fluid.

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.

3.2.1 Laminar Flow in Tubes

The 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, $f = \frac{64}{\mathrm{Re}}$ (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

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

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

Sieder and Tate as give a general relation for average Nusselt number for the hydrodynamically and thermally developing laminar flow in a circular tube:

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

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.

3.2.2 Turbulent Flow in Tubes

The 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 3.3, which is known as the Moody diagram.

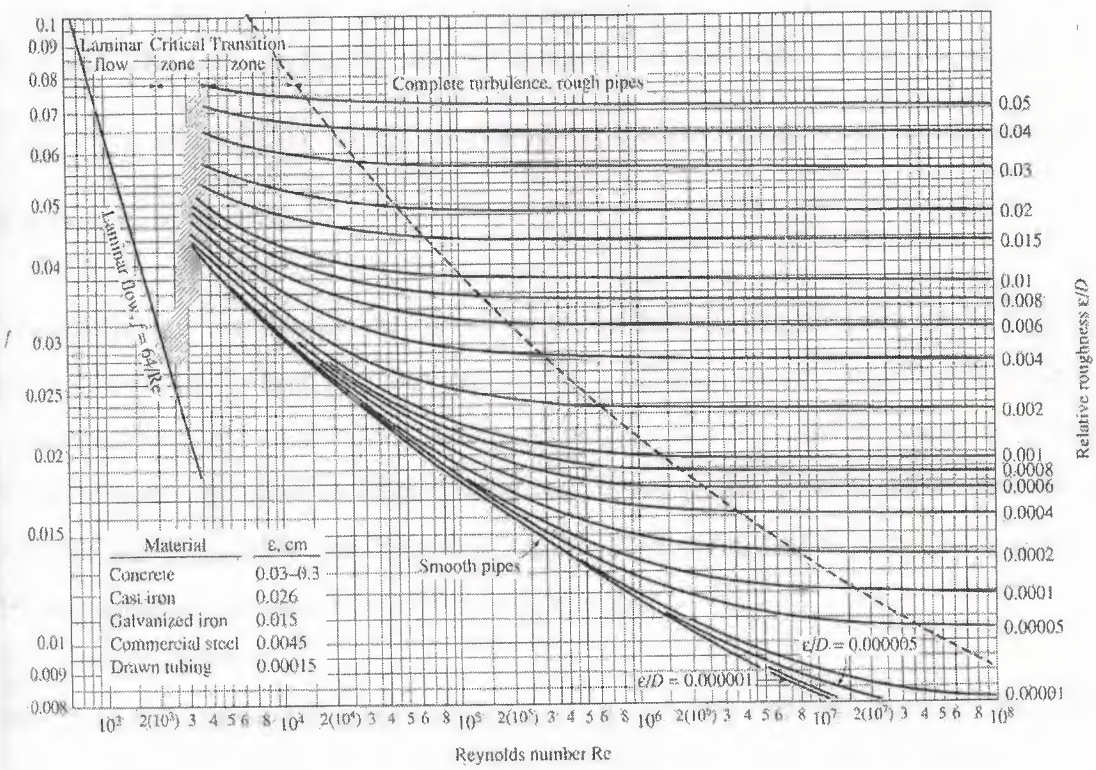

Figure 3.3 Friction factor for fully developed flow in circular tubes.
(the Moody chart).

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

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

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

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

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

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

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 $Nu = 0.125 f Re Pr^{1/3}$ 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. 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 $D_h = 4\, A_c / p$.

Example 3.1 Heat Loss from the Ducts of a Heating System in the Attic

Hot air at athmospheric pressure and 80^0C enters an 8m long uninsulated square duct of crossection 0.2m x 0.2m that passes through the attic of a house at a rate of $0.15 m^3/s$ is shown in figure below. The duct is observed to be nearly isothermal at 60^0C. Determine the exit temperature of the air and the rate of heat loss from the duct to the attic space.

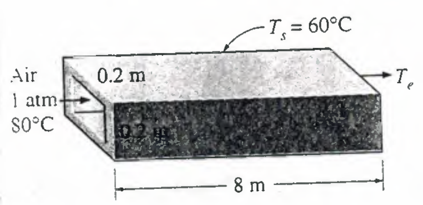

Figure 3.4 Schematic for Example 3-1.

Solution: Heat loss from uninsulated square ducts of a heating system in the attic is considered. The exit temperature and the rate of heat loss are to be determined.

Assumptions: 1 Steady operating conditions exits. **2** The inner surfaces of the duct are smooth. **3** Air is an ideal gas.

Properties: We do not know the exit temperature of the air in the duct, and thus we can not determine the bulk mean temperature of air, which is the temperature at which the properties are to be determined. The mean temperature of air at the inlet is 80^0C or

353K, and we expect this temperature to drop somewhat as a result of heat loss through the duct whose surface is at a lower temperature. Thus it is reasonable to assume a bulk mean temperature of 350K for air for the purpose of evalating the properties of air. At this temperature and 1atm we read from Properties of gases table at 1atm.

$\rho = 1.009\ kg/m^3$ $Cp = 1008\ J/kg\ ^0C$

$k = 0.0297\ W/m\ ^0C$ $Pr = 0.706$

$v = 2.06 \times 10^{-5}\ m^2/s$

Analysis: This is an internal flow problem since the air is flowing in a duct. The characteristic length (which is the hydraulic diameter), the mean velocity, and the Reynolds number in this case are

$$D_h = \frac{4A_c}{\rho} = \frac{4a^2}{4a} = a = 0.2m$$

$$V_{in} = \frac{\dot{V}}{A_c} = \frac{0.15m^3/s}{(0.2m)^2} = 3.75m/s$$

$$\mathrm{Re} = \frac{V_m D_h}{v} = \frac{(3.75m/s)(0.2m)}{2.06 \times 10^{-5} m^2/s} = 36.408$$

which is greater than 4000. Therefore, the flow is turbulrnt and the entry lengths in this case are roughly

$$L_h \approx L_t \approx 10D_h = 10 \times (0.2m) = 2m$$

which is the much shorter than the total length of the duct. Therefore, we can assume fully developed turbulent flow in the entire duct and determine the Nusselt number from

$$Nu = \frac{hD_h}{k} = 0{,}023\,\mathrm{Re}^{0,8}\,\mathrm{Pr}^{0.3} = 0.023(36{,}408)^{0,8}(0{,}706)^{0,3} = 92{,}3$$

Then

$$h = \frac{k}{D_h} Nu = \frac{0,0297 W/m,^{O}C}{0,2m} (92,3) = 13,7 W/m^2.^{o}C$$

$$A = pL = 4aL = 4 \times (0.2m)(8m) = 6.4m^2$$

$$m = \rho \dot{V} = (1.009 kg/m^3)(0.15 m^3/s) = 0.151 kg/s$$

next, we determine the exit temperature of air from

$$T_e = T_s - (T_s - T_i) e^{-hA/mC_P}$$

$$= 60^0C - \left[(60-80)^0C\right] \exp\left[-\frac{(13.7W/m^{2\,0}C)(6.4m^2)}{(0.151kg/s)(1008J/kg^0C}\right] = 71.2^0C$$

Then the logarithmic mean temperature difference and the rate of heat loss from the air become

$$\Delta T_{in} = \frac{T_e - T_i}{\ln \frac{T_s - T_e}{T_s - T_i}} = \frac{71.2 - 80}{\ln \frac{60 - 71.2}{60 - 80}} = 15.2^0C$$

$$\dot{Q} = hA\Delta T_{in} = (13.7W/m^{2\,0}C)(6.4m^2)(15.2^0C) = 1368W$$

Therefore, the air wills lose heat at a rate of 1368W as it flows through the duct in the attic.

Discussion: Having calculated the exit temperature of the air, we can now determine the actual bulk mean fluid temperature from

$$T_b = \frac{T_i + T_e}{2} = \frac{80 + 71.2}{2} = 75.6^0C = 348.6K$$

Which is sufficiently close to the assumed value of 350 K at which we evaluated the properties of air. Therefore, it is not necessary to re-evaluate the properties at this T_b and to repeat the calculations.

SUMMARY

In this chapter internal forced convection is introduced and discussed. Nusselt number is examined according to flow in tubes for each laminar and turbulent flow. An example is solved about heat loss from the ducts of a heating system in the attic, with this example the solution way of internal forced convection problems is explained.

CHAPTER 4

EXTERNAL FORCED CONVECTION

4.1 Introduction

In this chapter external forced convection is explained which is related with flow over flat plates, across cylinders, and spheres. In internal flow, the fluid has a free surface, and thus the boundary layer over the surface is free to grow indefinitely. Flow over flat plates, across cylinders, and spheres for both laminar and turbulent flow conditions are examined.

Nusselt and Reynolds numbers are examined according to laminar, turbulent, and combined laminar and turbulent flow over flat plates. Empirical correlations for the average Nusselt number are given according to cross flow for forced convection over circular and noncircular cylinders.

4.2 Flow Over Flat Plates

In this section the determination of the heat transfer rate to or from a plate is discuussed, as well as drag force exerted on the plate by the fluid for both laminar and turbulent flow cases. Surfaces that are slightly contoured such as turbine blades can also be approximated as flat plates with reasonable accuracy. The friction and the heat transfer coefficients for a flat plate can be determined theoretically by solving the conservation of mass, momentum, and energy equations approximately or numerically. They can also be determined experimentally and expressed by empirical correlations. In either approach, it is found that the average Nusselt number can be expressed in terms of the Reynolds and Prandtl numbers in the form

$$Nu = \frac{hL}{k} = C\,\mathrm{Re}_L^m\,\mathrm{Pr}^n\;K$$

where C, m, and n are constants and L is the length of the plate in the flow direction. The local Nusselt number at any point on the plate will depend on the distance of that point from the leading edge.

The fluid temperature in the thermal boundary layer varies from T_s at the surface to about T_∞ at the outer edge of the boundary. The fluid properties also vary with temperature, and thus with position across the boundary layer. In order to account for the variation of the properties with temperature properly, the fluid properties are usually evaluated at the so-called film temperature, defined as;

$$T_f = \frac{T_s + T_\infty}{2}$$

which is the arithmetic average of the surface and the free-stream temperatures. The fluid properties are then assumed to remain constant at those values during the entire flow.

The local friction and heat transfer coefficients vary along the surface of the flat plate as a result of the changes in the velocity and thermal boundary layers in the flow direction. We are usually interested in the heat transfer and drag force on the entire surface, which can be determined using the average heat transfer and friction coefficients. But sometimes we are also interested in the heat flux and the drag force at a certain location.

4.2.1 Laminar Flow Over Flat Plates

The local friction coefficient and Nusselt number at location x for laminar flow over flat plate are given by;

$$C_{f,x} = \frac{0.664}{\mathrm{Re}_x^{1/2}}$$

and

$$Nu_x = \frac{h_x x}{k} = 0.332\,\mathrm{Re}_x^{1/2}\,\mathrm{Pr}^{1/3} \qquad (\mathrm{Pr} \geq 0.6)$$

where x is the distance from the leading edge of the plate and $\mathrm{Re}_x = V_\infty x/v$ is the Reynolds number at location x. Note that $C_{f,x}$ is proportional to $1/\mathrm{Re}_x^{1/2}$ and thus to $x^{1/2}$. Likewise, $Nu_x = h_x x/k$ is proportional to $x^{1/2}$ and thus h_x is proportional to $x^{-1/2}$. Therefore, both $C_{f,x}$ and h_x are supposedly infinite at the leading edge (x=0) and decrease by a factor $x^{-1/2}$ in the flow direction.

The average friction coefficient and the Nusselt number over entire plate are;

$$C_f = \frac{1.328}{\mathrm{Re}_L^{1/2}}$$

and

$$Nu = \frac{hL}{k} = 0.664\,\mathrm{Re}_L^{1/2}\,\mathrm{Pr}^{1/3} \qquad (\mathrm{Pr} \geq 0.6)$$

The relations above give the average friction and heat transfer coefficients for the entire plate when the flow is laminar over the entire plate. Taking the critical Reynolds number to be $\mathrm{Re}_{cr} = 5x10^5$, the length of the plate x_{cr} over which the flow is laminar can be determined from

$$\mathrm{Re}_{cr} = 5x10^5 = \frac{V_\infty x_{cr}}{v}$$

thus, the relations above can be used for $x \leq x_{cr}$.

4.2.2 Turbulant Flow Over Flat Plates

the local friction coefficent and Nusselt number at location x for turbulent flow over a flat plat are given by

$$C_{f,x} = \frac{0.0592}{\mathrm{Re}_x^{1/5}} \qquad (5\times10^5 \le \mathrm{Re}_x \le 10^7)$$

and

$$Nu_x = \frac{h_x x}{k} = 0{,}0296\,\mathrm{Re}_x^{4/5}\,\mathrm{Pr}^{1/3} \qquad (0{,}6 \le \mathrm{Pr} \le 60)\ \ (5\times10^5 \le \mathrm{Re}_x \le 10^7)$$

where again x is the distance from the leading edge of the plate and $\mathrm{Re}_x = V_\infty X/v$ is the Reynolds number at location x. the local friction and heat transfer coefficients are higher in turbulent flow than they are in laminar flow because of the intense mixing that occurs in the turbulent boundary layer. Note that the both $C_{f,x}$ and h_x reach their highest values when the flow becomes fully turbulent, and then decrease by a factor of $x^{-1/5}$ in the flow direction.

The average friction coefficient and the Nusselt number over the entire plate in turbulent flow are ;

$$C_{f,x} = \frac{0.074}{\mathrm{Re}_x^{1/5}} \qquad (5\times10^5 \le \mathrm{Re}_x \le 10^7)$$

and

$$Nu_x = \frac{hL}{k} = 0{,}037\,\mathrm{Re}_x^{4/5}\,\mathrm{Pr}^{1/3} \qquad (0{,}6 \le \mathrm{Pr} \le 60)\ \ (5\times10^5 \le \mathrm{Re}_x \le 10^7)$$

The two relations above give average friction and heat transfer coefficients for the entire plate only when the flow is turbulent over the entire plate, or when the laminar flow region of the plate is too small relative to the turbulent flow region (that is, $x_{cr} \le L$).

Example 4.1 Flow of Hot Oil over a Flat Plate

Engine oil at 60^0C flows over a 5m long flat plate whose temperatures 20^0C with a velocity of 2m/s (fig. 6-15). Determine the total drug force and the rate of heat transfer per unit width of the entire plate.

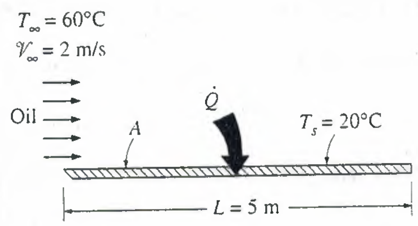

Figure 4.1 shematic for example 4-1

Solution: hot engine oil flows over a flat plate. The total drag force and the rate of heat transfer per unit width of the plate are to be determined.

Assumption: 1 Steady operating conditions exist. **2** The critical Reynolds number is $Re_{cr} = 5x10^5$ **3** Radiation effects are negligible.

Properties: The properties of engine oil at the film temperature of $T_f = (T_s + T_\infty)/2$ = $(20+60)/2=40^0$C are (Table a 10)

ρ = 876 kg/m^3 Pr = 2870

k = 0.144 W/m ^{0}C $\nu = 242 \times 10^{-6}\ m^2/s$

Analysis: Noting that L = 5m Reynolds number at the and of the plate is

$$\mathrm{Re}_L = \frac{V_\infty L}{\upsilon} = \frac{(2m/s)(5m)}{242\times10^{-6}m^2/s} = 4.13\times10^4$$

which is less than the critical Reynolds number, Thus we have laminar flow over the entire plate, and the average friction coefficient is determined from

$$C_f = 1.328\,\mathrm{Re}_L^{-0.5} = 1.328\times(4.13\times10^4)^{-0.5} = 0.00653$$

than the drag force acting on the plate per unit width becomes

$$F_D = C_f A \frac{\rho V_\infty^2}{2} = 0.00653 \times (5 \times 1m^2) \frac{(876kg/m^3)(2m/s)^2}{2} = 57.2N$$

This force corresponds to the weight of a mass of about 6kg. Therefore, a person who applies an equal and opposite force to the plate to keep it from moving will feel like he or she is spending as much power as is necessary to hold a 6kg mass from dropping.

Similarly, the Nusselt number is determined using the laminar flow relations for a flat plate,

$$Nu = \frac{hL}{k} = 0.664 \mathrm{Re}_L^{0.5} \mathrm{Pr}^{1/3} = 0.664 \times (4.13 \times 10^4)^{0.5} \times 2870^{1/3} = 1918$$

then

$$h = \frac{k}{L} Nu = \frac{0.144W/m^0\mathrm{C}}{5m}(1918) = 55.2W/m^{2\,0}\mathrm{C}$$

and

$$\dot{Q} = hA(T_\infty - T_s) = (55.2W/m^{2\,0}C)(5 \times 1m^2)(60-20)^0C = 11.040W$$

Discussion: Note that hate transfer is always from the higher temperatures medium to the lower temperature one. In this case, it's from the oil to the plate. Both the drag force and the heat transfer rate are per m width of the plate. The total quantities for the entire plate can be obtained by multiplying these quantities by the actual width of the plate.

4.3 Flow Across The Cylinders

Another common external flow involves fluid motion normal to the axis of a circular cylinder. As shown in Figure 4.2, 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 4.2 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 reaches a maximum velocity when $dp/dx = 0$, and decelerates as a result of the adverse pressure gradient (duldx < 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.

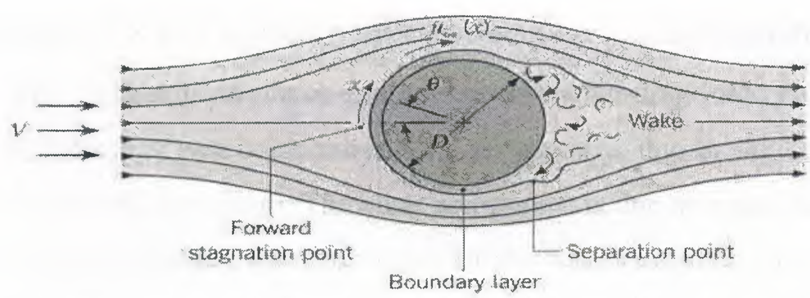

Figure 4.2 Boundary layer formation and separationon circular cylinder in cross flow.

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 $\mathrm{Re}_D \leq 2 \times 10^5$, the boundary layer remaining laminar, and separation occurs at $\theta = 80°$ is shown in Figure 4.3.

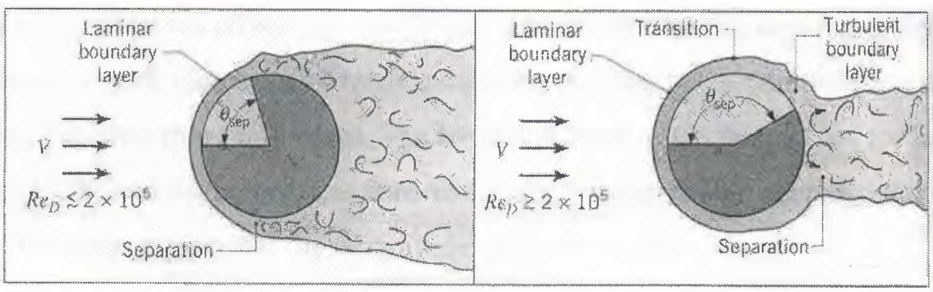

Figure 4.3 The effect of turbulence on separation.

However, if $\mathrm{Re}_D \geq 2 \times 10^5$, 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 4.4..

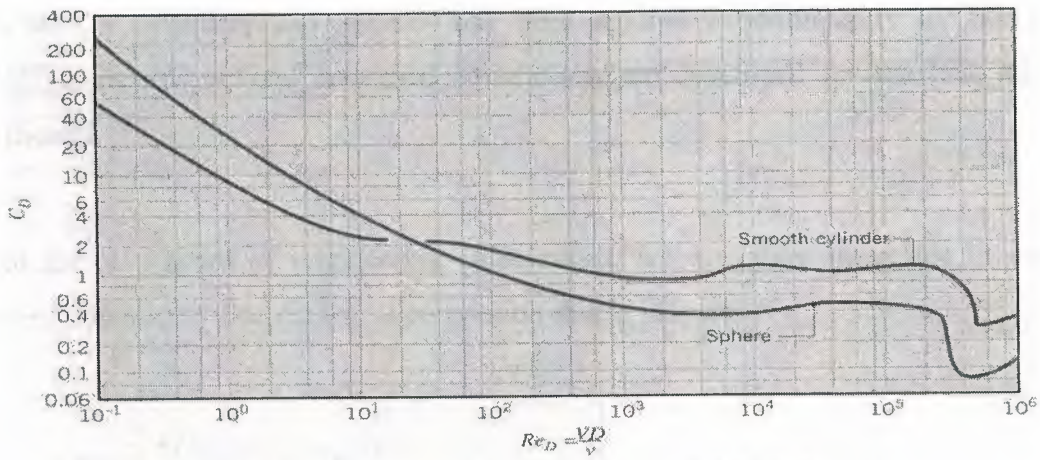

Figure 4.4 Drag coefficients for smooth, circularcylinder in flow and for a sphere.

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.

4.3.1 The Heat Transfer Coefficient

Flow across cylinders and spheres, in general, involve flow seperation, which is difficult to handle analytically. Therefore, such flows must be studied experimentally. Indeed, flow across cylinders and spheres has been studied experimentally by numerous investigators, and several empirical correlations are developed for the heat transfer coefficient.

From the standpoint of engineering calculations, we are more interested in overall average conditions. The empirical correlation due to Hilpert

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

is widely used, where the constants C and m are listed in Table 4.1. 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 4.2. in working with equations all properties are evaluated at the film temperature.

Table 4.1 Constants of forthe circular cylinder in cross flow.

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 4.2 Constants for noncircular cylinders in cross flow of a gas.

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	0.160	0.638
	1.95×10^4–10^5	0.0385	0.782
	5×10^3–10^5	0.153	0.638
Vertical plate			
	4×10^3–1.5×10^4	0.228	0.731

Churchill and Bernstein 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 the fluid properties are evaluated at the film temperature, which is the average of the free stream and surface temperatures. The equation is formed as;

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

4.4 Flow Across The Spheres

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

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

$$\begin{bmatrix} 0.71 < \mathrm{Pr} < 380 \\ 3.5 < \mathrm{Re}_D < 7.6\times10^4 \\ 1.0 < (\mu/\mu_s) < 3.2 \end{bmatrix}$$

All properties are evaluated at free stream temperature T_∞ except μ_s, which is evaluated at the surface temperature T_s. Although above is considered to be quite accurate, the results obtained from this relation can be off by as much as 30 percent.

SUMMARY

In this Chapter external forced convection is discussed. Flow over plates and across the cylinders and spheres is explained breafly. Nusselt and Reynolds numbers are discussed for both laminar and turbulent flow. An example is solved about flow of hot oil over a flat plate is solved.

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 and is expressed by a Fourier's law of heat conduction as:

$$\dot{Q}_{cond} = -kA\frac{dT}{dx} \qquad \text{(W)}$$

where k is the thermal conductivity of the material, A is the area normal to the direction of heat transfer, and dT/dx temperature gradient.

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. The rate of convection heat transfer expressed by Newton's law of cooling as:

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

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. When a surface emmisivity ε and surface area A at an absolute temperature T_s is completely enclosed by a much larger surface at absolute temperature T_{surr} separated by a gas that does not intervene with radiation, the net rate of radiation heat transfer between two surfaces is given by:

$$\dot{Q}_{rad} = \varepsilon\sigma A(T_s^4 - T_{surr}^4) \qquad \text{(W)}$$

In Chapter 2, convection heat transfer mechanism is discussed which is divided in two groups as forced and natural heat convection. Convection heat transfer is complicated by the fact that 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. 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 fun.Natural convection heat transfer is explained 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 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.

In Chapter 3, internal forced convection is explained which is related with flow in tubes. In internal flow, 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. Flow inside tubes for both laminar and turbulent flow conditions are examined. 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;

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

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:

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

In Chapter 4, external forced convection is introduced. External flow over flat plates, across the cylinders and spheres is discussed. 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. The Reynolds number at which the flow becomes turbulent is called the critical Reynolds number.

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