NEAR EAST UNIVERSITY

Faculty of Engineering

Department of Mechanical Engineering

FORCED CONVECTION HEAT TRANSFER TO THE FOR FLOW OVER FLAT PLATES, FLOW IN TUBES, FLOW ACROSS CYLINDERS AND SPHERES

Graduation Project
ME – 400

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CHAPTER I

INTRODUCTION

1.1 GENERAL CONSIDERATION

When a surface comes in a contact with a moving fluid at a different temperature heat transfer occurs in such a manner that the fluid parcels carry the heat given off by the surface away. This mode of heat transfer is know as heat transfer by convection. Heat diffuses in the fluid primarily by molecular conduction. However, the diffusion processes may be altered by relative motion in the fluid. For example in the turbulent flows, the presence of the eddying motion in the fluid has a large effect on diffusion. Therefore convection heat transfer depends on the nature of the fluid motion and its analysis requires a detailed knowledge of the velocity field.

The convection process is called forced convection when the fluid motion is generated by an external source, such as by a blower or a pump. In many problems encountered in forced convection, the buoyancy force, caused by density differences developed due to the heat transfers it, can be neglected. However, situations can occur where buoyancy induced fluid motion can be comparable to that caused by the external forces. Under such conditions, both forced and free convection processes influence the heat transfer rate and the combined processes are refferred to as mixed convection. A new experimental work reveals that in cases where flow is driven by buoyancy, under certain conditions the heat transfer rate may be predictable from the forced convection relations.

Convection is classified as natural and forced convection, depending on how the fluid movement is initiated. In forced convection, the fluid is formed to flow over a surface or in a tube by external means such as a pump or a fan. In natural convection, any fluid motion is caused by natural means such as a buoyancy effect, which manifests itself as the rise of warmer fluid and the fall of the cooler fluid. Convection is also classified as external and internal depending on whether the fluid is forced to flow over a surface or in a channel. Convection in external and internal flows exhibits very different characteristics.

1.2 ENGINEERING HEAT TRANSFER

Heat transfer equipment such as heat exchangers, boilers, condensers, radiators, heaters, furnaces and solar collectors are designed primarily on the basis of heat transfer analysis. The heat transfer problems encountered in practice can considered two groups rating and sizing problems. The rating problems deal with the determination of the heat transfer rate for an existing system at a specified temperature difference. The sizing problems deal with the determination of the size of a system in order to transfer heat at a specified rate for a specified temperature difference.

A heat transfer processes or equipment can be studied either experimentally or analytically. The experimentally approach has the advantage that we deal with the actual physical system, and what we get is what it is, within the limits of experimental error. However this approach is expensive, time consuming, and often impractical. Besides, the system we are analyzing may not even exist. For example the size of heating system of a building must usually be determined before the building is actually built on the basis of the dimensions and specifications given. The analytic approach has the advantage that it is fast and inexpensive but the results obtained are subject to the accuracy of the assumptions and idealizations made in the analysis. In heat transfer studies often a good compromise is reached by reducing the choices to just a few by analysis and then verifying the finding experimentally and doing some fine-tuning.

1.3 BASIC CONCEPT OF HEAT TRANSFER

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. Below I give a brief description of each mode.

Conduction Heat Transfer

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.

Convection heat transfer

Convection is the made of energy transfer between a solid surface and the adjacent liquid or gas that is in the motion, and it involves the combined effects or conduction and fluid motion. The faster the fluid motion, the greater the convection heat transfers. 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 hot block by blowing cool air over its top surface. 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 that removes the heated air near the surface and replaces it by the cooler air.

Convection is called forced convection if the fluid is forced to flow over the surface by external means such as a fan, or the wind. In contrast, convection is called natural or free convection if buoyancy forces that are induced by density differences due to the variation of temperature in the fluid cause the fluid motion. For example in the absence of a fan, heat transfer from the surface of the block, will be by natural convection since any motion in the air in this case will be due to the rise of the warmer air near the surface and the fall of the cooler air to fill 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.

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 processes 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 hear transfer is observed to be proportional to the temperature difference, and is conveniently expressed by Newton's law.

Radiation Heat Transfer

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. 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 and I suffers no attenuation in a vacuum. This is exactly how the energy of the sun reaches the earth.

1.4 PHYSICAL MECHANISM OF FORCED CONVECTION

Mentioned earlier that three 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 position. 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 case of quiescent fluid.

Convection heat transfer is complicated by the fact that is 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 transfers.

To clarify this point further, consider steady heat transfer through a fluid contained between two parallel plates maintained at different temperatures. The temperature of the fluid and the plate will be the same at the points of contact because of the continuity of temperature. Assumption no fluid motion the energy of the hotter fluid molecules near the plate will be transferred to the adjacent 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 near the hot plate and inject it near the cold plate repeatedly.

Consider the cooling of a hot iron block with a fan blowing air over its top surface. We know that heat will be transferred from the hot block to the surrounding cooler air, and the block will eventually cool. We also now that 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.

Experience shows that convection heat transfer strongly depends on the fluid properties dynamic viscosity, thermal conductivity k, density p, 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. 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 the proportional to the temperature difference and is conveniently expressed by Newton law of cooling.

We use forced convection in daily life more often than we might think 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 feel.

1.5 THERMAL CONDUCTIVITY

Thermal conductivity k is a measure of materials ability to conduct heat. For example, k= 0.608 W/m. ^{0}C for water k=60 W/m. ^{0}C for iron at room temperature, which indicates that iron conducts heat almost 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.

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 be 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.

Table 1.5 Some material, of the thermal conductivity

Material	Thermal conductivity (k) W/m. ^{0}C
Silver	429
Copper	401
Gold	317
Aluminum	237
Iron	80.2
Glass	0.78
Water	0.613
Human skin	0.37
Wood	0.17
Air	0.026
Glass fiber	0.043

A layer of material of known thickness and area can be heated from one side by an electric resistance heater of known output.

1.6 VELOCITY BOUNDRY LAYER

Consider the flow of a fluid over a flat plate as shown in figure. The x coordinate measured a long the plate surface from the leading edge of the plate in the direction of the flow, and y is measured from the surface in the normal direction. The fluid approaches the plate in the x direction with a uniform velocity V_∞. For the sake of discussion we can consider the fluid to consist of adjacent layers piled on top of each other. The velocity of the particles in the first fluid layer adjacent to the plate becomes zero because of the no slip condition. This motionless layer slows down the particles of the neighboring fluid layer as a result of friction between the particles of these two adjoining fluid layers at different velocities. This fluid layer then slows down the molecules of the next layer and so on. Thus the presence of the plate is felt up to some distance from the plate beyond which the fluid velocity V_∞ remains essentially unchanged. As a result of the fluid velocity at any X location will vary from zero

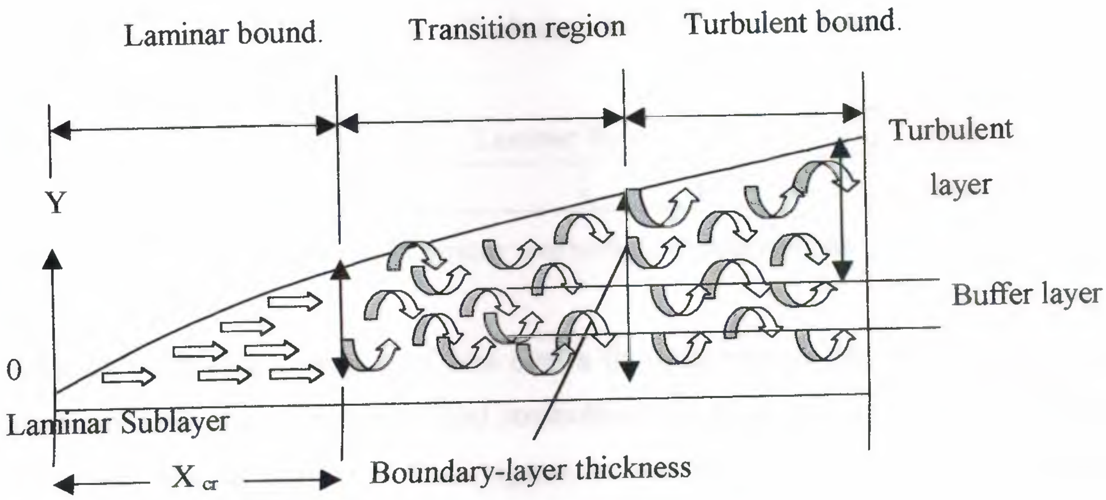


Figure 1.6 Velocity boundary layer

The region of the flow above the plate bounded by Boundary layer thickness in which the effects of the viscous shearing forces caused by fluid viscosity are felt is called the velocity boundary layer as you see figure 1.6. The viscosity of a fluid is a measure of its resistance to flow, and it as a strong function of temperature. The viscosities of liquids decrease with temperature, whereas the viscosities gases increase with temperature. The viscosities of some fluids at 20 ^{0}C are listed in table. Note that the viscosities of the different fluids differ by several orders of magnitude. Also note that it is more difficult to mo wean object in a higher viscosity fluid such as engine oil than it is in a lower- viscosity fluid such as water.

1.7 LAMINAR AND TURBULENT FLOW

The convection heat transfer-rate and the surface drag coefficient depend strongly on the condition of the flow field. Therefore in the treatment of a convection problem it is essential to determine whether the boundary layer is laminar or turbulent. When fluid moves in a streamline manner parallel to the surface, the flow is called laminar. In a laminar boundary layer, the diffusion of momentum and heat is controlled solely by the molecular transport properties of the fluid, that is, viscosity and thermal conductivity. For example, if you have been around smokers, you probably noticed that the cigarette smoke rises in a smooth plume for the first few centimeters and than starts fluctuating randomly in all directions it continues its journey toward the lungs of nonsmokers.

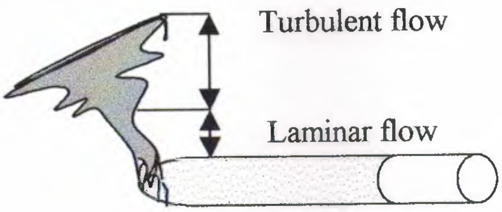


Figure 1.7 Laminar and turbulent flow regions.

Likewise, a careful inspection of flow over a flat plate reveals that the fluid flow in the boundary layer starts out as flat and streamlined but turns chaotic after some distance from the leading edge, 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. We can verify the existence of these laminar, transition and turbulent flow regimes by injecting some dye in to 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 as you see in figure 1.7

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

1.8 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, Osborn 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.

$$Re = \text{Inertia forces} / \text{viscous forces}$$

$$Re = V\,L / \nu$$

Where;

V : free stream velocity, m/s

L : characteristic length of the geometry, m

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

Note that he Reynolds number is a dimensionless quantity. Also note that kinematic viscosity ν differs from dynamic viscosity μ. 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 that 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 number, however, the viscous forces are large enough to overcome the inertia forces and to keep the fluid in line. Thus the flow is turbulent in the first case and laminar in the second.

The Reynolds number at which the flow becomes turbulent is called the critical Reynolds number. The value of the critical Reynolds number is different for different geometries. For flow over a flat plate transition from laminar to turbulent occurs at the critical Reynolds number of $Re_{critical,\ flat\ plate} = 5 \times 10^5$ this generally accepted value of the critical Reynolds number for a flat plate may vary somewhat depending on the surface roughness, the turbulence level and variation of pressure along the surface.

The critical Reynolds number for flow in a tube is generally accepted to be 2300.

$$Re < 2300 \quad \text{Laminar flow}$$
$$2300 \leq Re \leq 4000 \quad \text{Transition to turbulence}$$
$$Re > 4000 \quad \text{Turbulent flow}$$

The transition from laminar to turbulent flow in a tube is quite different than it is over a flat plate, where the Reynolds number is zero at the leading edge and increases linearly in the flow direction. Consequently, the flow starts out laminar and becomes turbulent when a critical Reynolds number is reached. Therefore, the flow over a flat plate is partly laminar and partly turbulent. In flow in a tube, however, the Reynolds number is constant. Therefore, the flow is either laminar or turbulent over practically the entire length of the tube.

1.9 THERMAL BOUNDARY LAYER

We have seen that 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. Also We defined the velocity boundary layer as the region in which the fluid velocity varies. Likewise a thermal boundary layer develops when a fluid at a specified temperature flows over a surface that is at a different temperature, as shown figure 1.9

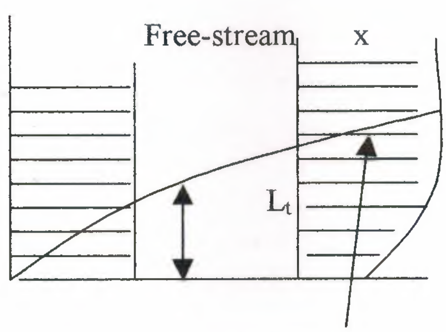


Figure 1.9 Thermal boundary layer on a flat plate

Consider the flow of a fluid at a uniform temperature of $T\infty$ over an isothermal flat plate at a temperature T_s. The fluid particles in the layer adjacent to the surface will reach thermal equilibrium with the plate and assume the surface temperature T_s.

$T_s + 0.99(T_\infty - T_S)$ These fluid particles will then exchange energy with the particles in the adjoining fluid layer, and so on. As a result temperature profile will develop in the flow field that ranges from T_s 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 thickness of the thermal boundary layer L_t at any location along the surface is defined as the distance from the surface at which the temperature difference $T-T_s$ equals $0.99(T_\infty - T_s)$. Note that for the special case of $T_s = 0$, we have $T=0.99T_\infty$ at the outer. Edge of the thermal boundary layer which is analogous to $V = 0.99V_\infty$ for the velocity boundary layer. The thickness of the thermal boundary layer increases in the flow direction since the effects of heat transfer are felt at greater distance from the surface further down stream.

1.10 PRANDTL NUMBER

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 fluid flowing over it. In flow over a heated 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 relative to the temperature profile, the development of the velocity boundary layer relative to the thermal boundary layer will have strong effect on the convection heat transfer.

Table 1.10 Prandtl number of some liquid

Fluid	**Pr**
Liquid metals	0.004 – 0.030
Gases	0.7 – 1.0
Oils	50 – 100,000
Glycerin	2000 – 100,000
Light organic fluids	5 – 50

Temperature profile, the development of the velocity boundary layer relative to the thermal boundary layer will have 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.

Pr = Molecular diffusivity of momentum/ Molecular diffusivity of heat

$$Pr = \nu / \propto$$

ν: Molecular diffusivity of momentum.

$\propto$: Molecular diffusivity of heat

1.11 CONCULUSION

In this chapter, general consideration of heat transfer, types of heat transfer, physical mechanisms of forced convection are explained. The secondly in this chapter are gives some technical knowledge. For example has been explain thermal conductivity, velocity boundary layer, Reynolds number, laminar and turbulent flow. This chapter introduces to explain heat transfer and brief description of some technical word.

CHAPTER II

FLOW OVER FLAT PLATES

2.1 GENERAL INFORMATION

So far we have discussed the physical aspects of forced convection over surface. In this section we will discuss the determination of the heat transfer rate to or from a flat plate, as well as the drag force exerted on the plate by the fluid for both laminar and turbulent flow cases. Surface that is slightly contoured such as turbine blades can also be approximately 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;

$$Nu = h\,L / k$$

$$Nu = C\,Re_L^{m}\,Pr^{n}$$

Where C, m, and n are constants L is the length of the plate in the flow direction k is thermal conductivity, h is heat transfer coefficient. 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 fluid thermal boundary layer varies from T_s at the surface to about T_∞ at the other edge of the boundary. $T_f = T_s + T_\infty / 2$ the fluid properties also vary with temperature and thus with position across the boundary layer. In order to account for the variation of the properly the fluid properties are usually evaluated at the so-called film temperature. 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. In such cases we need to know the local values of the heat transfer and friction coefficients.

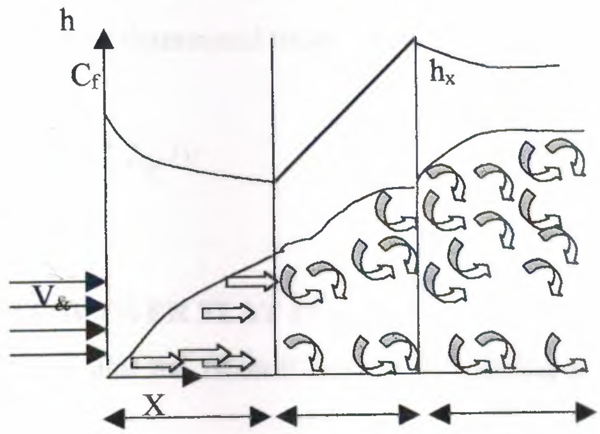


Figure 2.1 The variation of the local friction and heat transfer coefficients

2.2 LAMINAR FLOW OVER FLAT PLATES

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

$$C_{f,x} = \frac{0.664}{Re_x^{1/2}}$$

and ;

$$Nu_x = h_x x / k$$

$$Nu_x = 0.332\, Re_x^{1/2}\, Pr^{1/3} \qquad (Pr \geq 0.6)$$

Where x the distance from the leading edge of the plate and $Re = V_{\&}\, x/v$ is the Reynolds number at location x. Note that $C_{f,\,x}$ is proportional to $1/\, Re_x^{1/2}$ 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 of $x^{-1/2}$ in the flow direction. The variation of the boundary layer thickness L(D) the friction coefficient C_f and the convection heat transfer coefficient h along an isothermal flat plate is shown in figure 2.1

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 $Re_{cr} = 5 \times 10^5$ the length of the plate L_{cr} over which the flow is laminar can be determined from $Re = 5 \times 10^5$

$$Re_{cr} = V_{\infty} L_{cr} / v$$

2.3 TURBULENT FLOW OVER FLAT PLATES

The local friction coefficient and the Nusselt number at location x for turbulent flow over a flat plate are given by,

$$C_{f,x} = \frac{0.0592}{Re_x^{1/5}} \qquad \text{and} \qquad 5 \times 10^5 \leq Re_x \leq 10^7$$

and

$$Nu_x = \frac{h_x x}{k} = 0.0296\, Re_x^{4/5}\, Pr^{1/3} \qquad 0.6 \leq Pr \leq 60$$

Where again x is the distance from the leading edge of the plate and $Re_x = V_{\infty} x/v$ 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 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, as shown figure 2.1.

The average friction coefficient and the Nusselt number over the entire plate in turbulent flow are determined by substituting the relations and we get.

$$C_f = \frac{0.074}{Re_x^{1/5}} \qquad 5 \times 10^5 \leq Re_L \leq 10^7$$

and

$$Nu = hL/k \qquad \begin{array}{c} 0.6 \leq Pr \leq 60 \\ 5 \times 10^5 \leq Re \leq 10^7 \end{array}$$

$$Nu = 0.0337 Re_L^{4/5} Pr^{1/3}$$

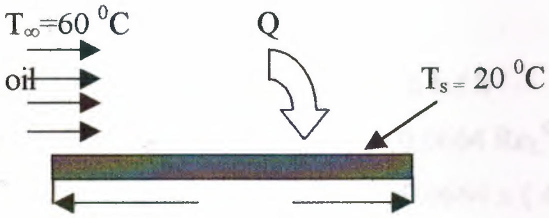
The two relations above give the average friction and heat transfer coefficients for the entire plate only when the flow is turbulent over the entire plat, or when the laminar flow region of the plate is too small relative to the turbulent flow region.

Example: Flow of hot oil over a flat plates

Engine oil at 60 ^{0}C flows over a flat plate whose temperature is 20 ^{0}C with a velocity of 2 m/s determine the total drag force and rate of the heat transfer per unit width of the entire plate.

$V_\infty = 2$ m/s

$T_\infty = 60\ ^0C$



$L = 5m\ V_\infty = 2$ m/s

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.

Assumptions: Steady operating conditions exist. The critical Reynolds number is $Re_{cr} = 5 \times 10^5$ Radiation effects negligible.

The properties of engine oil at the film temperature of as fallow;

$T_f = (T_s + T_\infty) / 2 = (20+60)/20 = 40\ ^0C$

$P = 876$ kg/m^3

$k = 0.144$ W/m. ^{0}C Can be taken from table for $T_f = 40\ ^0C$

$Pr = 2870$

$$Re_L = V_\infty L / v$$

$$= 2m/s.\ 5m / 242 \times 10^{-6}\ m^2/s$$

$$= 4.13 \times 10^4$$ which less than the critical Reynolds number

Thus we have laminar flow over the entire plate, and the average friction coefficient is

Determined from: $v_f = 1.328 \, Re_L^{-0.5}$

$$= 1.328 \times (4.13 \times 10^4)^{-0.5}$$

$$= 0.00653$$

The Nusselt number is determined using the laminar flow relations for a flat plate.

$$Nu = hL/k$$

$$= 0.0664 \, Re_L^{0.5} \, Pr^{1/3}$$

$$= 0.0664 \times (4.13 \times 10^4) \times 2870^{1/3}$$

$$= 1918$$

Then h and $\dot{Q}$ to be calculated as fallows:

$$h = k \, Nu / L$$

$$= 0.144.1918 / 5$$

$$= 55.2 \ W/m^2 . {}^0C$$

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

$$= (55.2 \ W/m^2 . {}^0C)(5x1)(60-20){}^0C$$

$$= 11.040 \ W$$

2.4 CONCULUSION

In this chapter, discussed very important topics that are related to the flow over flat plates. Gives some information about flow over flat plates and explained differences between laminar and turbulent flow. Fluid flow over a flat plate starts out as smooth and streamlined but turns chaotic after some distances from the leading edge. The flow regime is said in the first case to be laminar, by smooth streamlines and highly ordered motion, and the turbulent flow. The flow regime depends mainly on the ratio of the inertia forces to viscous forces in the fluid. This ratio is called Reynolds number.

CHAPTER III

FLOW ACROSS CYLINDER AND SPHERES

3.1 GENERAL KNOWLEDGE

In the preceding section we considered fluid flow over flat surface. In this section, we consider flow over cylinders and spheres, which are frequently encountered in practice. For example the tubes in a tube and shell heat exchanger involve both internal flow through the tubes and external flow over the tubes and both flows must be considered in the analyses of heat transfer between the two fluids. Below we consider external flow only. The characteristic length for a circular cylinder or sphere is taken to be the external diameter D.

$$Re = V_{\infty}D / \nu$$

Thus the Reynolds number is defined as where V is the uniform velocity of the fluid as it approaches the cylinder or sphere. The critical Reynolds number for flow across a circular cylinder or sphere is $Re_{cr} \approx 2 \times 10^5$. That is the boundary layer remains laminar for $Re < 2x\ 10^5$ and becomes turbulent for $Re > 2x\ 10^5$.

Cross flow over a cylinder exhibits complex flow patterns. The fluid approaching the cylinder will branch out and encircle the cylinder forming a boundary layer that wraps around the cylinder. The fluid particles on the mid plane will strake the cylinder at the stagnation point brining the fluid to the complete stop and thus raising the pressure at that point. The pressure decreases in the flow direction while the fluid velocity increases.

At very low free stream velocities the fluid completely wraps around the cylinder and two arms of the fluid meet on the rear side of the cylinder in an orderly manner. Thus the fluid follows the curvature of the cylinder. At higher velocities the fluid still hugs the cylinder on the frontal side but it is too fast to remain attached to the surface as it approaches the top of the cylinder. As a result the boundary layer detaches from the surface forming a wake behind the cylinder. This point is called the separation point. As you see figure 3.1. Random vortex formation and pressures much lower than the stagnation point pressure characterize flow in the wake region.

The flow separation phenomenon is analogous to fast vehicles jumping off on hills. At low velocities the wheels of the vehicle always remain in contact with the road surface. But at high velocities the vehicle is too fast to follow the curvature of the road and takes off at the hill, losing contact with the road.

Flow separation occurs at about $\theta \approx 80^0$ when the boundary layer is laminar and at about $\theta \approx 140^0$ when it is turbulent. The delay of separation in turbulent flow is caused by the rapid fluctuations of the fluid in the transverse direction.

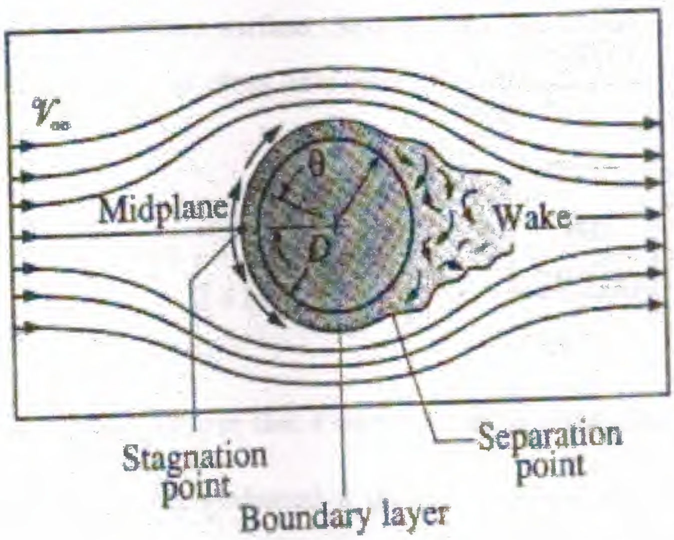


Figure 3.1 Typical flow patterns in cross flow over a cylinder.

3.2 THE HEAT TRANSFER COEFFICIENT

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

The complicated flow pattern across a cylinder discussed earlier greatly influences heat transfer. The variation of the Nusselt number around the periphery of a cylinder subjected to cross flow of air.

On the local heat transfer coefficients are insightful; however, they are of little value in heat transfer calculations since the calculation of heat transfer requires the average heat transfer coefficient over the entire surface. Several relations available in the literature for the average Nusselt number for cross flow over a cylinder.

$$Nu_{cyl} = \frac{hD}{k} = 0.3 + \frac{0.62\mathrm{Re}^{1/2}\mathrm{Pr}^{1/3}}{\left[1+(0.4/\mathrm{Pr})^{2/3}\right]^{1/4}}\left[1+\left(\frac{\mathrm{Re}}{28{,}200}\right)^{5/8}\right]^{4/5}$$

This relation is quite comprehensive in that it correlates all available data well for Re

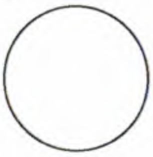
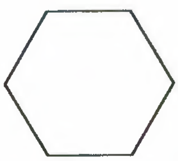
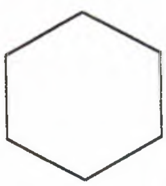
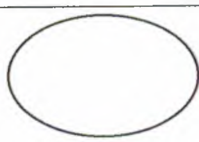
Pr > 0.2. The fluid properties are evaluated at the film temperature $T_f = \frac{1}{2}(T_\infty + T_s)$, which is the average of the free-system and surface temperatures.

For flow over sphere,

$$Nu_{sph} = \frac{hD}{k} = 2 + \left[0.4\,\mathrm{Re}^{1/2} + 0.06\,\mathrm{Re}^{2/3}\right]\mathrm{Pr}^{0.4}\left(\frac{\mu_\infty}{\mu_s}\right)^{1/4}$$

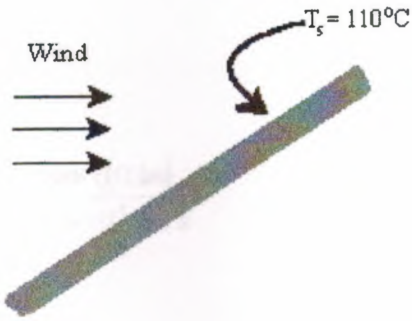
Which is valid for $3.5 \leq Re \leq 80{,}000$ and $0.7 \leq Pr \leq 380$. The fluid properties in this case are evaluated at the free-stream temperature T_{∞}, except for μ_s, which is evaluated at the surface temperature T_s. Although the two relations above are considered to be quite accurate, the results obtained from them can be of by as much as 30 percent.

Table 3.3 Empirical correlations for the average Nusselt number for forced convection over circular and noncircular cylinders in cross flow.

Cross-section of the cylinder	Fluid	Range of Re	Nusselt number
Circle 	Gas or liquid	0.4 – 4 4 – 40 40 – 4000 4000 – 40,000 40,000 – 400,000	$Nu=0.989Re^{0.330}Pr^{1/3}$ $Nu=0.911Re^{0.385}Pr^{1/3}$ $Nu=0.683Re^{0.466}Pr^{1/3}$ $Nu=0.193Re^{0.618}Pr^{1/3}$ $Nu=0.027Re^{0.805}Pr^{1/3}$
Square 	Gas	5000 – 100,000	$Nu=0.102Re^{0.675}Pr^{1/3}$
Square (Tilted 45°) 	Gas	5000 – 100,000	$Nu=0.246Re^{0.588}Pr^{1/3}$
Hexagon 	Gas	5000 – 100,000	$Nu=0.153Re^{0.638}Pr^{1/3}$
Hexagon (Tilted 45°) 	Gas	5000 – 19,500 19,500 – 100,000	$Nu=0.160Re^{0.638}Pr^{1/3}$ $Nu=0.0385Re^{0.782}Pr^{1/3}$
Vertical Plate 	Gas	4000 – 15,000	$Nu=0.228Re^{0.731}Pr^{1/3}$
Ellipse 	Gas	2500 – 15,000	$Nu=0.248Re^{0.612}Pr^{1/3}$

Example : Heat Loss from a Steam Pipe in Windy Air

A long 10-cm-diameter steam pipe whose external surface temperature is 110°C passes through some open area that is not protected against the winds. Determine the rate of heat loss from the pipe per unit of its length when the air is at 1 atm pressure and 4°C and the wind is blowing across the pipe at a velocity of 8 m/s.



Schematic for Example

Solution : A steam pipe is exposed to windy air. The rate of heat loss from the steam is to be determined.

Assumptions: Steady operating conditions exist. Radiation effects are negligible. Air is an ideal gas.

Properties: The properties of air at the film temperature can be taken from table.

$$T_f = \frac{1}{2}(T_\infty + T_s)$$

=(110+4)/2

= 57°C

=330K

$k = 0.0283\ W/m \cdot {}^{o}C$,

$Pr=0.708$ Can be taken from table for $T_f = 57\ {}^{0}C$

$\nu = 1.86 \times 10^{-5}\ m^2/s$

Analysis: This is an external flow problem, since we are interested in the heat transfer from the pipe to the air that is flowing outside the pipe. The Reynolds number of the flow is

$$Re = \frac{V_\infty D}{\nu}$$

$$= 0.3 + \frac{(8m/s)(0.1m)}{1.86\times10^{-5} m^2/s}$$

$$= 43{,}011$$

Then the Nusselt number in this case can be determined from

$$Nu = \frac{hD}{k} = 0.3 + \frac{0.62\,Re^{1/2}\,Pr^{1/3}}{\left[1+(0.4/Pr)^{2/3}\right]^{1/4}}\left[1+\left(\frac{Re}{28{,}200}\right)^{5/8}\right]^{4/5}$$

$$= 0.3 + \frac{0.62(43{,}011)^{1/2}(0.708)^{1/3}}{\left[1+(0.4/0.708)^{2/3}\right]^{1/4}}\left[1+\left(\frac{43{,}011}{28{,}200}\right)^{5/8}\right]^{4/5}$$

$$= 196.3$$

Then h to be calculated as fallow;

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

$$= \frac{0.0283 W/m \cdot {}^{\circ}C}{0.1m}(196.3)$$

$$= 55.6 W/m^2 \cdot {}^{\circ}C$$

Then the rate of heat transfer from the pipe per unit of its length becomes

$$A = pL$$
$$= \pi DL$$
$$= \pi(0.1 \text{ m})(1 \text{ m})$$
$$A = 0.314 \text{ m}^2$$

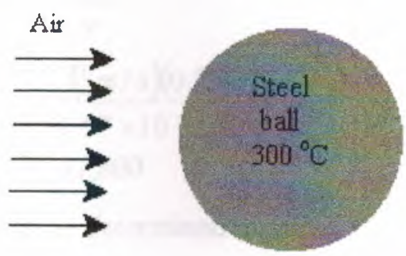
And the heat transfer rate can be calculated as fallow;

$$\dot{Q} = hA(T_s - T_\infty)$$
$$= (55.6 W/m^2 \cdot °C)(0.314 m^2)(110 - 4)°C$$

$$\mathbf{=1851W}$$

Example : Cooling of a Steel Ball by Forced Air

A 25-cm-diameter stainless steel ball $\rho = 8055 kg/m^3, C_p = 480 J/kg \cdot °C$ is removed from the oven at a uniform temperature of 300 °C . The ball is then subjected to the flow of air at 1 atm pressure and 27 °C with a velocity of 3 m/s. The surface temperature of the ball eventually drops to 200°C. Determine the average convection heat transfer coefficient during this cooling process and estimate how long this cooling process will take.



Schematic for Example

Solution: A hot stainless steel ball is cooled by forced air. The average convection heat transfer coefficient and the cooling time are to be determined.

Assumption: Steady operating conditions exist. Radiation effects are negligible. Air is an ideal gas. The outer surface temperature of the ball is uniform at all times.

The surface temperature of the ball during cooling is changing. Therefore, the convection heat transfer coefficient between the ball and the air will also change. To avoid this complexity, we take the surface temperature of the ball to be constant at average temperature of (300 + 200)/2 = 250°C in the evaluation of the heat transfer coefficient and use the value obtained for the entire cooling process.

Properties: The dynamic viscosity of air at the surface temperature is $\mu_s = \mu_{@250°C} = 2.96 \times 10^{-5} kg/m \cdot s$.

The properties of air at the free-stream temperature of 27°C and 1 atm are given table.

$k = 0.0261$ W/m · °C $\qquad \nu = 1.57 \times 10^{-5}$ m²/s

$\mu = 1.85 \times 10^{-5}$ kg/m · s $\qquad$ Pr = 0.712

Analysis: This is an external flow problem since the air flows outside the ball. The Reynolds number of the flow is determined from

$$\begin{aligned} \mathrm{Re} &= \frac{V_\infty D}{\nu} \\ &= \frac{(3m/s)(0.25m)}{1.57\times10^{-5}\,m^2/s} \\ &= 47{,}800 \end{aligned}$$

Then the Nusselt number can be determined from

$$Nu = \frac{hD}{k} = 2 + \left[0.4\mathrm{Re}^{1/2} + 0.6\mathrm{Re}^{2/3}\right]\mathrm{Pr}^{0.4}\left(\frac{\mu_\infty}{\mu_s}\right)^{1/4}$$

$$= 2 + \left[0.4(47{,}800)^{1/2} + 0.06(47{,}800)^{2/3}\right](0.712)^{0.4}\left(\frac{\mu_\infty}{\mu_s}\right)^{1/4}$$

$$\mathrm{Nu} = 131$$

Then the average convection heat transfer coefficient becomes

$$h = \frac{k}{D}Nu = \frac{0.0261 W/m\cdot{}^\circ C}{0.25m}(131)$$

$$h = 13.6\ \mathrm{W/m^2\cdot{}^\circ C}$$

In order to estimate the time of cooling of the ball from 300°C to 200°C, we determine the average rate of heat transfer from Newton's law of cooling by using the average surface temperature. That is,

$$A = \pi D^2$$

$$A = \pi (0.25\text{m})^2$$

$$\text{A} = 0.196 \text{ m}^2$$

$$\begin{aligned}\dot{Q} &= \text{h A } (T_s - T_\infty) \\ &= (13.6 \text{ W/m}^2 \cdot {}^\circ\text{C})(0.196 \text{ m}^2)(250 - 27)\ {}^\circ\text{C} \\ &= 594 \text{ W}\end{aligned}$$

Next we determine the total heat transferred from the ball, which is simply the change in energy of the ball as it cools from 300°C to 200°C:

$$\begin{aligned}m &= \rho V \\ &= \rho \frac{1}{6} \pi D^3 \\ &= \left(8085 kg/m^3\right)\frac{1}{6}\pi(0.25m)^3 \\ &= 66.1 kg\end{aligned}$$

$$\begin{aligned}Q_{total} &= mC_p(T_2 - T_1) \\ &= (66.1 \text{ kg})(480 \text{ J/kg} \cdot {}^\circ\text{C})(300 - 200)\ {}^\circ\text{C} \\ &= 3{,}172{,}800 \text{ J}\end{aligned}$$

In the above calculation, we assumed that the entire ball is at 200°C, which is not necessarily true. The inner region of the ball will probably be at a higher temperature than its surface. With this assumption, the time of cooling is determined to be

$$\Delta t \approx \frac{Q}{\dot{Q}_{ave}}$$

$$= \frac{3{,}172{,}800J}{594 J/s} = 5341 s$$

3.3 CONCULUSION

In this chapter I explained flow across cylinder and spheres. The firstly I gave general information about flow across cylinder and spheres. And than I gave the heat transfer coefficient with in this phenomena , and also I gave the average Nusselt numbers for cross flow over a cylinder and sphere which is valid. The average Nusselt numbers for cross flow over a cylinder and sphere can be determined from

$$Nu_{cyl} = \frac{hD}{k} = 0.3 + \frac{0.62\,\mathrm{Re}^{1/2}\,\mathrm{Pr}^{1/3}}{\left[1+(0.4/\mathrm{Pr})^{2/3}\right]^{1/4}}\left[1+\left(\frac{\mathrm{Re}}{28{,}200}\right)^{5/8}\right]^{4/5}$$

Which is valid for Re Pr > 0.2, and

$$Nu_{sph} = \frac{hD}{k} = 2 + \left[0.4\,\mathrm{Re}^{1/2} + 0.06\,\mathrm{Re}^{2/3}\right]\mathrm{Pr}^{0.4}\left(\frac{\mu_\infty}{\mu_s}\right)^{1/4}$$

Which is valid for $3.5 \leq \mathrm{Re} \leq 80{,}000$ and $0.7 \leq \mathrm{Pr} \leq 380$. The fluid properties are evaluated at the film temperature $T_f = \frac{1}{2}(T_\infty + T_s)$, in the case of a cylinder, and at the free-stream temperature T_{oo} .

CHAPTER IV

FLOW IN TUBES

4.1 GENERAL CONSIDERATION

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

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. That is, the mass flow rate through the tube evaluated using the mean velocity V_m from $\dot{m} = \rho V_m A_c$ (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.

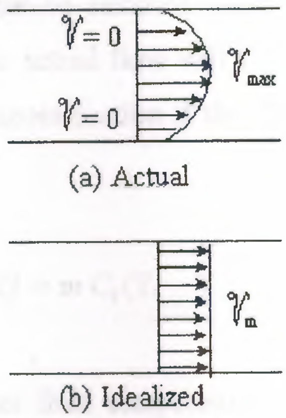


Figure 4.1 Actual and idealized velocity profiles for flow in a tube.

When a fluid is heated or cooled as it flows through 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 T_m will change in the flow direction, however whenever the fluid is heated or cooled. As you see figure 4.1.

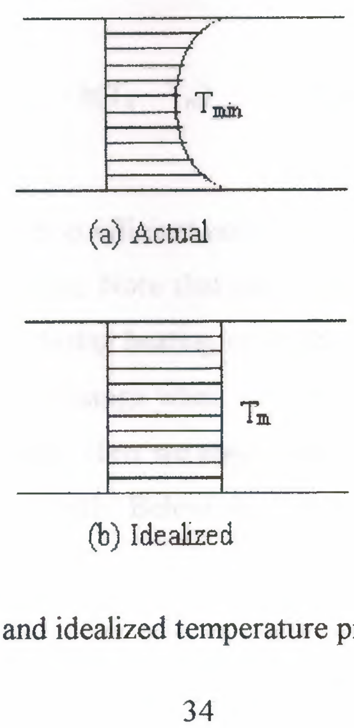


Figure 4.2 Actual and idealized temperature profiles for flow in a tube.

The value of the mean temperature T_m is determined from the requirement that the conservation of energy principle be satisfied. That is, the energy transported by the fluid through a cross-section in actual flow will be equal to the energy that would be transported through the same cross-section if the fluid were at a constant temperature T_m.

$$\dot{Q} = m\, C_p(T_e - T_i) \quad , \qquad kJ/s$$

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 remains constant in the absence of any energy interactions through the wall of the tube.

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

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

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

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

4.2 CONSTANT SURFACE HEAT FLUX

In the case of $\dot{q}_s$ = constant, the rate of heat transfer can also be expressed as

$$\dot{Q} = \dot{q}_s A$$
$$= \dot{m} C_p (T_e - T_i)$$

Then the mean fluid temperature at the tube exit becomes

$$T_e = T_i + \frac{\dot{q}_s A}{\dot{m} C_p}$$

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

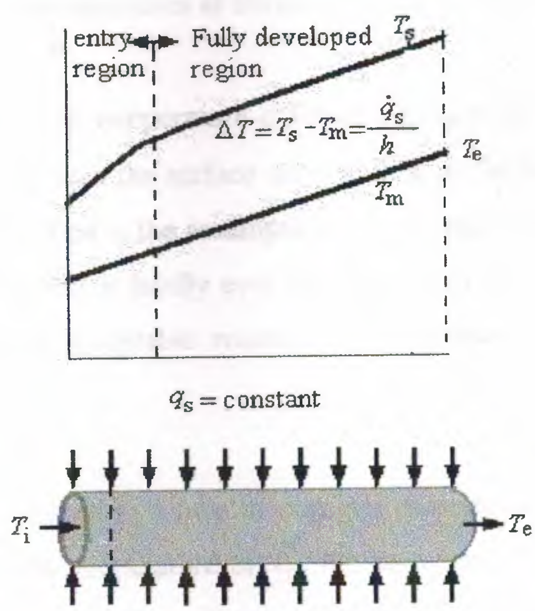


Figure 4.3 Variation of the tube surface and the mean fluid temperatures along the tube for the case of constant surface heat flux.

4.3 CONSTANT SURFACE TEMPERATURE

From Newton's law of cooling, the rate of heat transfer to or from a fluid flowing in a tube can be expressed as

$$\dot{Q} = hA\Delta T_{ave}$$
$$= hA(T_s - T_m)_{ave}$$

Where h is the average convection heat transfer coefficient, A is the heat transfer surface area (it is equal to πDL for a circular pipe of length L), and ΔT_{ave} is some appropriate average temperature difference between the fluid and the surface. Below we discuss two suitable ways of expressing ΔT_{ave}.

In the constant surface temperature (T_s = constant) case, ΔT_{ave} can be expressed approximately by the arithmetic mean temperature difference ΔT_{am} .

Where $T_b = \frac{1}{2}(T_i + T_e)$ is the bulk mean fluid temperature, which are the arithmetic average of the mean fluid temperatures at the inlet and the exit of the tube.

Note that the arithmetic mean temperature difference ΔT_{am} is simply the average of the temperature difference between the surface and the fluid at the inlet and the exit of the tube. Inherent in this definition is the assumption that the mean fluid temperature varies linearly along the tube, which is hardly ever the case when T_s = constant. This simple approximation often gives acceptable results, but not always. Therefore, we need a better way to evaluate ΔT_{ave}.

Consider the heating of a fluid in a tube of constant cross-section whose inner surface is maintained at a constant temperature of T_s. We know that the mean temperature of the fluid T_m will increase in the flow direction as a result of heat transfer. The energy balance on a differential control volume gives

$$\dot{m}C_p dT_m = h(T_s - T_m)dA$$

Solving for mC_p gives

$$\dot{m}C_p = \frac{hA}{\ln \frac{T_s - T_e}{T_s - T_i}}$$

Substituting this into we obtain

$$\dot{Q} = hA\Delta T_{\ln}$$

Where; is the logarithmic mean temperature difference.

$$\Delta T_{\ln} = \frac{T_e - T_i}{\ln \frac{T_s - T_e}{T_s - T_i}}$$

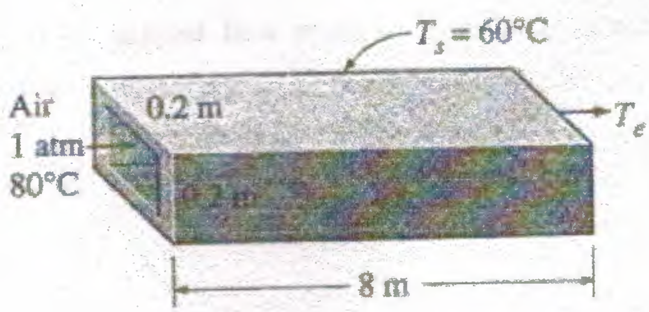
$$= \frac{\Delta T_e - \Delta T_i}{\ln(\Delta T_e / \Delta T_i)}$$

Is the logarithmic mean temperature difference. Note that $\Delta T_i = T_s - T_i$ and $\Delta T_e = T_s - T_e$ are the temperature differences between the surface and the fluid at inlet and the exit of the tube, respectively. The $\Delta T_{\ln}$ relation above appears to be prone to misuse, but it is practically failsafe, since using T_i in place of T_e and vice versa in the numerator and/or the denominator will, at most, affect the sign, not the magnitude. Also, it can be used for both heating ($T_s > T_i$ and T_e) and cooling ($T_s < T_i$ and T_e) of a fluid in a tube.

The logarithmic mean temperature difference $\Delta T_{\ln}$ is obtained by tracing the actual temperature profile of the fluid along the tube, and is an exact representation of the average temperature difference between the fluid and the surface. It truly reflects the exponential decay of the local temperature difference. When T_e differs from T_i by no more that 40 percent, the error in using the arithmetic mean temperature difference is less than 1 percent. But the error increases to undesirable levels when ΔT_e differs from ΔT_i by greater amounts. Therefore, we should always use the logarithmic mean temperature difference when determining the convection heat transfer in a tube whose surface is maintained at a constant temperature.

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

Hot air at atmospheric pressure and 80°C enters an 8-m-long annulated square duct of cross-section 0.2m x 0.2m that passes through the attic of a house at a rate of 0.15 m^3/s The duct is observed to be nearly isothermal at 60°C. Determine the exit temperature of the air and the rate of heat loss from the duct to the attic space.



Schematic for Example .

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

Assumptions: Steady operating conditions exist. The inner surfaces of the ducts are smooth. Air is an ideal gas.

Properties: We do not know the exit temperature of the air in the duct, and thus we cannot determine the bulk mean temperature of air, which is the temperature at which the properties are to be determined. The mean temperature of the air at the inlet is 80°C or 353 K, 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 350 K for air for the purpose of evaluating the properties of air. At this temperature and 1 tam are given from table.

$$\rho = 1.009 kg / m^3$$

$$k = 0.0297 W / m \cdot {}^o C$$

$$\nu = 2.06x10^{-5} m^2 / s$$

$$C_p = 1008 J / kg \cdot {}^o C$$

$$\Pr = 0.706$$

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

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

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

$$\mathrm{Re} = \frac{V_m D_h}{\nu} = \frac{(3.75 m / s)(0.2m)}{2.06x10^{-5} m^2 / s} = 36{,}408$$

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

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

Which is 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,

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

$$= \frac{0.0297 W / m^{o}C}{0.2m}(92.3)$$

$$= 13.7 W / m^2.^{o}C$$

Next, we determine the exit temperature of air from

$$T_e = T_s - (T_s - T_i)e^{-hA/\dot{m}C_p}$$

$$\mathbf{=71.2\ ^0C}$$

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

$$\Delta T_{in} = \frac{T_e - T_i}{\ln \frac{T_s - T_e}{T_s - T_i}}$$

$$= \frac{71.2 - 80}{\ln \frac{60 - 71.2}{60 - 80}} = 15.2^{o}C$$

$$\dot{Q} = hA\Delta T_{\ln}$$

$$= (13.7 W / m^2.^{o}C)(6.4 m^2)(15.2^{o}C)$$

$$\mathbf{=1368\ W}$$

Therefore, the air will lose heat at a rate of 1368 W 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

$$T_b = \frac{T_i + T_e}{2}$$

$$= \frac{80 + 71.2}{2}$$

$$= 75.6^\circ C = 348.6K$$

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.

4.4 FLOW REGIMES IN A TUBE

The flow in a tube can be laminar or turbulent, depending on the flow conditions. The type of flow in a tube can be verified experimentally by injecting a small amount of colored fluid into the main flow, as Reynolds did in the 1880s. The streak of colored fluid is straight and smooth in laminar flow, as shown in Figure 4.8, but fluctuates rapidly and randomly in turbulent flow.

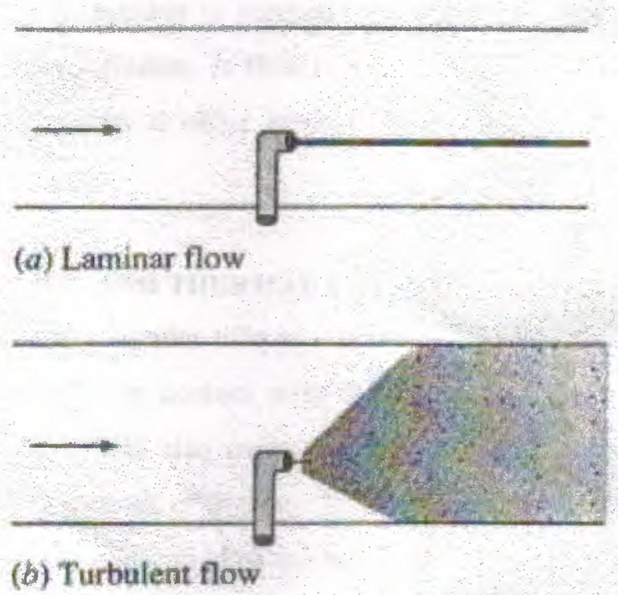


Figure 4.8 The behavior of colored fluid injected into the flow in laminar and turbulent flows in a tube.

The Reynolds number for flow in a circular tube of diameter D is defined as

$$\mathrm{Re} = \frac{V_m D}{\nu}$$

Where V_m is the mean fluid velocity and ν is the kinematics viscosity of the fluid. The Reynolds number again provides a convenient criterion for determining the flow regime in a tube, although the roughness of the tube surface and the fluctuations in the flow has considerable influence. The critical Reynolds number for flow in a tube is generally accepted to be 2300. Therefore,

$Re < 2300$	Laminar flow
$2300 \leq Re \leq 4000$	Transition to turbulence
$Re > 4000$	Turbulent flow

The transition from laminar to turbulent flow in a tube is quite different than it is over a flat plate, where the Reynolds number is zero at the leading edge and increases linearly in the flow direction. Consequently, the flow starts out laminar and becomes turbulent when a critical Reynolds number is reached. Therefore, the flow over a flat plate is partly laminar and partly turbulent. In flow in a tube, however, the Reynolds number is constant. Therefore, the flow is either laminar or turbulent over practically the entire length of the tube.

4.5 HYDRODYNAMIC AND THERMAL ENTERY LENGTHS

Consider a fluid entering a circular tube at a uniform velocity. As in external flow, the fluid particles in the layer in contact with the surface of the tube will come to a complete stop. This layer will also cause the fluid particles in the adjacent layers to slow down gradually as a result of friction. To make up for this velocity reduction, the velocity of the fluid at the midsection of the tube will have to increase to keep the mass flow rate through the tube constant. As a result, a velocity boundary layer develops along the tube. The thickness of this boundary layer increases in the flow direction until the boundary layer reaches the tube center and thus fills the entire tube, as shown in Figure 4.9. The region from the tube inlet to the point at which the boundary layer merges at the centerline is called the hydrodynamic entry region, and the length of this region is called the hydrodynamic entry length L_h. the region beyond the hydrodynamic entry region in which the velocity profile is fully developed and remains unchanged is called the hydro dynamically developed region. The velocity profile in the hydro dynamically developed region is parabolic in laminar flow and somewhat flatter in turbulent flow.

In laminar flow in a tube, the magnitude of the dimensionless Prandtl number Pr is a measure of the relative growth of the velocity and thermal boundary layers. For fluids with $Pr \approx 1$, such as gases, the two boundary layers essentially coincide with each other. For fluids with $Pr > 1$, such as oils, the velocity boundary layer outgrows the thermal boundary layer. As a result, the hydrodynamic entry length is smaller than the thermal entry length. The opposite is true for fluids with Pr < 1 such as liquid metals.

The hydrodynamic and thermal entry lengths in laminar flow are given approximately as

$$L_{h,laminar} \approx 0.05 \operatorname{Re} D$$

$$L_{t,laminar} \approx 0.05 \operatorname{Re} \Pr D$$

In turbulent flow, the hydrodynamic and thermal entry lengths are known to be independent of Re or Pr and are generally taken to be

$$L_{h,turbulent} \approx L_{t,turbulent} \approx 10D$$

The friction coefficient is related to the shear stress at the surface, which is related to the slope of the velocity profile at the surface. Nothing that the velocity profile remains unchanged in the hydro dynamically developed region, the friction coefficient also remains constant in that region. A similar argument can be given for the heat transfer coefficient in the thermally developed region. Thus, we conclude that the friction and the heat transfer coefficients in fully developed flow region remain constant.

4.6 LAMINAR FLOW IN TUBES

We mentioned earlier that flow in smooth tubes is laminar for Re < 2300. The theory for laminar flow is well developed, and both the friction and the 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 $V_{max} = 2V_m$. Knowing the velocity profile, the shear stress at the wall becomes

$$\tau = -\mu \left.\frac{dV}{dr}\right|_{r=E}$$

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

$$= \frac{8\mu V_m}{D}$$

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.

$$C_f = \frac{8\mu V_m}{D}\frac{2}{\rho V_m^2}$$

$$= \frac{16\mu}{\rho V_m D}$$

$$= \frac{16}{\text{Re}}$$

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

$$f = \frac{64}{\text{Re}} \qquad \text{Laminar flow}$$

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

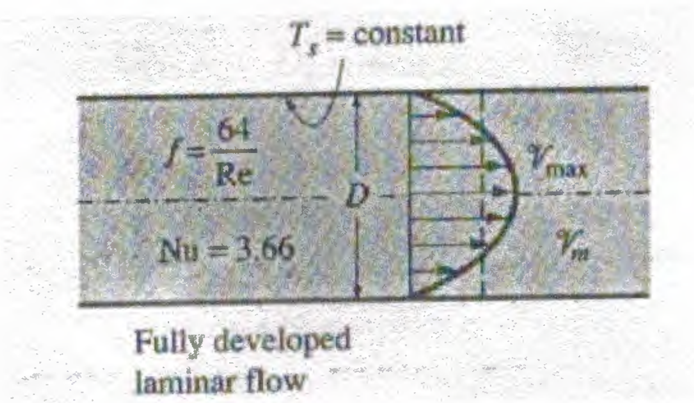


Figure 4.6 In laminar flow in a tube with constant surface temperature, both the friction factor and the heat transfer rate remain constant in the fully developed region.

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.

$Nu = 3.66$	for T_s = constant	Laminar flow
$Nu = 4.36$	for $\dot{q}_s$ = constant	Laminar flow

A general relation for the average Nusselt number for the hydro dynamically and / or thermally developing laminar flow in a circular tube is given as

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

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

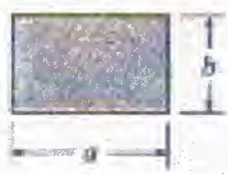
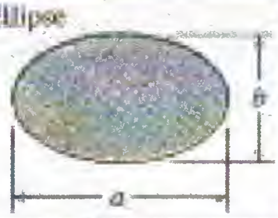
The Nusselt number Nu and the friction factor f are given in Table 4.2 for fully developed laminar flow in tubes of various cross-sections. The Reynolds and Nusselt numbers for flow in these tubes are based on the hydraulic diameter D_h defined as

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

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 $A_c = \pi D^2/4$ and $p = \pi D$. Once the Nusselt number is available, the convection heat transfer coefficient is determined from $h = k\mathrm{Nu}/D_h$. It turns out that for a fixed surface area, the circular tubes gives the most heat transfer for the least pressure drop, which explains the overwhelming 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.

Table 4.2 Nusselt number and friction factor for fully developed in laminar flow in tubes of various cross-section.

Cross-section of tube	a/b or $\theta°$	Nusselt number		Friction factor f
		T_s = const.	$\dot{q}_s$ = const.	
Circle 	—	3.66	4.36	64.00/Re
Hexagon 	—	3.35	4.00	60.20/Re
Square 	—	2.98	3.61	56.92/Re
Rectangle 	a/b			
	1	2.98	3.61	56.92/Re
	2	3.39	4.12	62.20/Re
	3	3.96	4.79	68.36/Re
	4	4.44	5.33	72.92/Re
	6	5.14	6.05	78.80/Re
	8	5.60	6.49	82.32/Re
	∞	7.54	8.24	96.00/Re
Ellipse 	a/b			
	1	3.66	4.36	64.00/Re
	2	3.74	4.56	67.28/Re
	4	3.79	4.88	72.96/Re
	8	3.72	5.09	76.60/Re
	16	3.65	5.18	78.16/Re
Triangle 	θ			
	10°	1.61	2.45	50.80/Re
	30°	2.26	2.91	52.28/Re
	60°	2.47	3.11	53.32/Re
	90°	2.34	2.98	52.60/Re
	120°	2.00	2.68	50.96/Re

4.7 TURBULENT FLOW IN TUBES

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

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

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

The friction factor for flow in tubes with smooth has well as rough roughens over a wide range of Reynolds numbers, which is known as the Moody diagram. Note that the friction factor and thus the pressure drop for flow in a tube can vary several times as a result of surface roughness.

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

$$Nu = 0.125 f\,\mathrm{Re}\,\mathrm{Pr}^{1/3} \quad \text{Turbulent flow}$$

Substituting the f relation from Equation 4.27 into Equation 4.28 gives the following relation for the Nusselt number for fully developed turbulent flow in smooth tubes:

$$Nu = 0.023\,\mathrm{Re}^{0.8}\,\mathrm{Pr}^{1/3} \qquad \begin{pmatrix} 0.7 \le \mathrm{Pr} \le 160 \\ \mathrm{Re} > 10{,}000 \end{pmatrix}$$

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

$$Nu = 0.023 \mathrm{Re}^{0.8} \mathrm{Pr}^{n} \qquad \begin{pmatrix} 0.7 \le \mathrm{Pr} \le 160 \\ \mathrm{Re} > 10{,}000 \end{pmatrix}$$

Where $n = 0.4$ for heating and 0.3 for cooling of the fluid flowing through the tube. The fluid properties are evaluated at the bulk mean fluid temperature $T_b = \frac{1}{2}(T_i + T_e)$, which are 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 condition 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

$$2300 \le \mathrm{Re} \le 4000$$

Especially when the Reynolds number is closer to 4000 than it is to 2300.

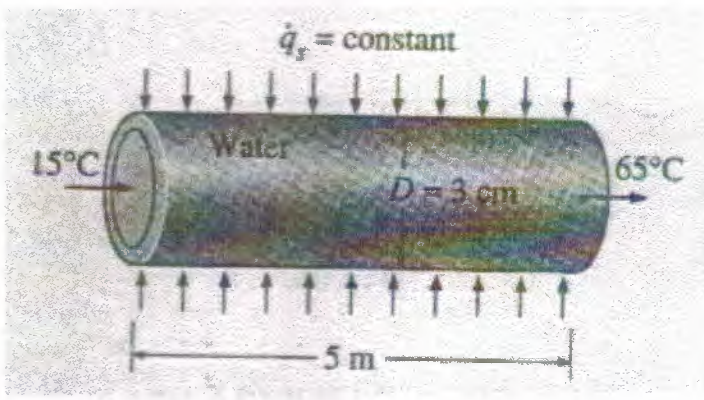
The Nusselt number for rough surface can also be determined. Note that tubes with rough surfaces have much higher heat transfer coefficients than tubes with smooth surfaces. Therefore, tube surfaces are often intentionally roughened, corrugated or finned in order to enhanced the convection heat transfer 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 $D_h = 4A_c/p$.

Example: Heating of water by resistance heaters in a tubes

Water is to be heated from 15°C to 65°C as it flows through a 3-cm-internal 5-m-long tube. The tube is equipped with an electric resistance heater that provides uniform heating throughout the surface of the tube. The outer surface of the heater is well insulated, so that in steady operation all the heat generated in the heater is transferred to the water in the tube. If the system is to provide hot water at a rate of 10 L/min, determine the power rating of the resistance heater. Also, estimate the inner surface temperature of the pipe at the exit.

Solution: Water is to be heated in a tube equipped with an electric resistance heater on its surface. The power rating of the heater and the inner surface temperature are to be determined.



Schematic for Example

Assumptions: Steady flow conditions exist. The surface heat flux is uniform. The inner surfaces of the tube are smooth.

Properties: The properties of water at the bulk mean temperature of

$$T_b = (T_i + T_c)/2$$
$$= (15 + 65)/2 = 40°C$$

$\rho = 992.1\ kg/m^3$

$k = 0.631\ W/m\ ^{\cdot o}C$

$\nu = \mu/\rho = 0.658 \times 10^{-6}\ m^2/s$ This value are given from table for T_b=40 ^0C

$C_p = 4179\ J/kg\ ^{\cdot o}C$

$Pr = 4.32$

Analysis: This is an internal flow problem since the water is flowing in a pipe. The cross-sectional and heat transfer surface areas are

$$A_c = \frac{1}{4}\pi D^2$$
$$= \frac{1}{4}\pi(0.03m)^2$$
$$= 7.0690 \times 10^{-4} m^2$$

$$A = pL$$
$$= \pi DL$$
$$= \pi(0.03m)(5m)$$
$$= 0.471 m^2$$

The volume flow rate of water is given as V = 10 L/min = 0.01 m^3/min. Then the mass flow rate of water becomes

$$\dot{m} = \rho V$$
$$= (992.1 kg/m^3)(0.01 m^3/\min)$$
$$= 9.921 kg/\min$$
$$= 0.1654 kg/s$$

To heat the water at this mass flow rate from 15°C to 65°C, heat must be supplied to the water at a rate of

$$\dot{Q} = \dot{m}C_p(T_e - T_i)$$
$$= (0.1654 kg/s)(4.179 kJ/kg \cdot °C)(65 - 15)°C$$
$$= 34.6 kJ/s = 34.6 kW$$

All of this energy must come from the resistance heater. Therefore, the power rating of the heater must be **34.6kW**.

The surface temperature T_s of the tube at any location can be determined from

$$\dot{q}_s = h(T_s - T_m) \rightarrow T_s = T_m + \frac{\dot{q}_s}{h}$$

Where h is the heat transfer coefficient and T_m is the mean temperature of the fluid at that location. The surface heat flux is constant in this case, and its value can be determined from

$$\dot{q}_s = \frac{\dot{Q}}{A}$$
$$= \frac{34.6 kW}{0.471 m^2}$$
$$= 73.46 kW/m^2$$

To determine the heat transfer coefficient, we first need to find the mean velocity of water and the Reynolds number:

$$V_m = \frac{\dot{V}}{A_c}$$

$$= \frac{0.010m^3/\min}{7.069x10^{-4}m^2}$$

$$= 14.15m/\min = 0.236m/s$$

$$\text{Re} = \frac{V_m D}{v}$$

$$= \frac{(0.236m/s)(0.03m)}{0.658x10^{-6}m^2/s} = 10{,}760$$

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

$$L_h \approx L_t \approx 10D$$

$$= 10(0.03m) = 0.3m$$

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

$$Nu = \frac{hD}{k}$$

$$= 0.023\,\text{Re}^{0.8}\,\text{Pr}^{0.4}$$

$$= 0.23(10{,}760)^{0.8}(4.34)^{0.4} = 69.5$$

Then h value calculated as fallow;

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

$$= \frac{0.631 W / m^{o}C}{0.03m}(69.5)$$

$$= 1462 W / m^2 \cdot^{o} C$$

And the surface temperature of the pipe at the exit becomes

$$T_s = T_m + \frac{\dot{q}_s}{h}$$

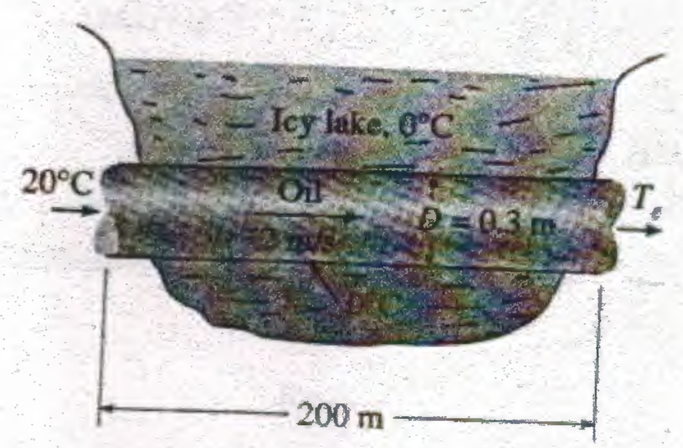
$$= 65^{o} C + \frac{73{,}460 kW / m^2}{1462 W / m^2 \cdot^{o} C}$$

$$= 115\ ^{o}C$$

Discussion: Note that the inner surface temperature of the pipe will be 50°C higher than the mean water temperature at the pipe exit. This temperature difference of 50°C between the water and the surface will remain constant throughout the fully developed flow region.

Example: Flow oil in a pipeline through the Icy waters of a lake

Consider the flow of oil at 20°C in a 30-cm-diameter pipeline at an average velocity of 2 m/s. A 200-m-long section of the pipeline passes through icy waters of lake at 0°C. Disregarding the thermal resistance of the pipe material, determine (a) the temperature of the oil when the pipe leaves the lake, (b) the rate of heat transfer from the oil, and (c) the pumping power required to overcome the pressure losses and to maintain the flow of the oil in the pipe.



Schematic for Example

Solution: Oil flows in a pipeline that passes through icy waters of a lake at 0°C. The exit temperature of the oil, the rate of heat loss, and the pumping power needed to overcome pressure losses are to be determined.

Assumptions: Steady operating conditions exist. The surface temperature of the pipe is very nearly 0°C. The thermal resistance of the pipe is negligible. The inner surfaces of the pipeline are smooth. The flow is hydrodynamic ally developed when the pipeline reaches the lake.

Properties: We do not know the exit temperature of the oil, and thus we cannot determine the bulk mean temperature, which is the temperature at which the properties of oil are to be evaluated. The mean temperature of the oil at the inlet is 20°C, and we expect this temperature to drop somewhat as a result of heat loss to the icy waters of the lake. We evaluate the properties of the oil at the inlet temperature, but we will repeat the calculations, if necessary, using properties at the evaluated bulk mean temperature. At 20°C we read.

$$\rho = 888k / m^3$$

$$k = 0.145W / m.^oC$$

$$\mu = 0.800kg / m s$$

$$v = 901x10^{-6} m^2 / s$$

$$C_p = 1880J / kg.^oC$$

$$\Pr = 10{,}400$$

Analysis: This is an internal flow problem since the oil is flowing in a pipe. The Reynolds number in this case is

$$\mathrm{Re} = \frac{V_m D_h}{v}$$

$$= \frac{(2m/s)(0.3m)}{901x10^{-6} m^2 / s} = 666$$

Which is less than the total length of the pipe. This is typical of fluids with high Prandtl numbers. Therefore, we assume thermally developing flow and determine the Nusselt number from

$$Nu = \frac{hD}{k}$$

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

$$= 1.86\left(\frac{666x10{,}400x0.3}{200m}\right)^{1/3}\left(\frac{0.8}{3.85}\right)^{0.14} = 32.6$$

Where the dynamic viscosity μ_s is determined at the surface temperature of 0°C. Note that this Nusselt number is considerably higher than the fully developed value of 3.66. Then h value calculated as fallow;

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

$$= \frac{0.145 W/m.^{o}C}{0.3m}(32.6)$$

$$= 15.8 W/m^2.^{o}C$$

Also,

$$A = pL$$

$$= \pi DL$$

$$= \pi(0.3m)(200m)$$

$$= 188.5 m^2$$

$$\dot{m} = \rho A_c V_m$$

$$= (888 kg/m^3)\left[\frac{1}{4}\pi(0.3m)^2\right](2m/s)$$

$$= 125.5 kg/s$$

Next we determine the exit temperature of oil from

$$T_e = T_s - (T_s - T_i)e^{-hA/\dot{m}C_p}$$

$$= 0^{o}C - \left[(0-20)^{o}C\right]\exp\left[-\frac{(15.8W/m^2.^{o}C)(188.5m^2)}{(125.5kg/s)(1880J/kg.^{o}C)}\right]$$

= 19.75 °C

Thus, the mean temperature of oil drops by a mere 0.25°C as it crosses the lake. This makes the bulk mean oil temperature 19.875°C, which is practically identical to the inlet mean temperature of 20°C. Therefore, we do not need to re-evaluate the properties at this bulk temperature and repeat the calculations.

(b) The logarithmic mean temperature difference and the rate of heat loss from the oil are

$$\Delta T_{\ln} = \frac{T_e - T_i}{\ln \dfrac{T_s - T_e}{T_s - T_i}}$$

$$= \frac{19.75 - 20}{\ln \dfrac{0 - 19.75}{0 - 20}}$$

$$= 19.875^{o}C$$

$$\dot{Q} = hA\Delta T_{\ln}$$

$$= (15.8W/m^2.^{o}C)(188.5m^2)(19.875^{o}C)$$

=59,190 W

Therefore, the oil will lose heat at a rate of 59,190 W as it flows through the pipe in the icy waters of the lake. Note that ΔT_{in} is identical to the arithmetic mean temperature in this case, since $\Delta T_i \approx \Delta T_e$.

(c) The laminar flow of oil is hydro dynamically developed. Therefore, the friction factor can be determined from

$$f = \frac{64}{\text{Re}}$$

$$= \frac{64}{666} = 0.0961$$

Then the pressure drop in the pipe and the required pumping power become

$$\Delta P = f \frac{L}{D}\frac{\rho V_m^2}{2}$$

$$= 0.0961 \frac{200m}{0.3m}\frac{(888kg/m^3)(2m/s)}{2}$$

$$= 113{,}780 N/m^2$$

$$\dot{W}_{pump} = \frac{\dot{m}\Delta P}{\rho}$$

$$= \frac{(125.5kg/s)(113{,}780N/m^2)}{888kg/m^3}$$

=16.1 kW

Discussion: We will need a 16.1-kW pump just to overcome the friction in the pipe as the oil flows in the 200-m-long pipe through the lake.

4.8 CONCULUSION

In this chapter we consider flow in tubes. Then we analyzed the constant surface heat flux, constant surface temperature, hydrodynamic and thermal entry lengths, laminar, and turbulent flow. We considered 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 $\dot{Q} = \dot{m} C_p (T_e - T_i)$ 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 in this case of $\dot{q}_s$ = constant. After its, some example which is related flow in tubes.

SUMMARY

The aim of this research discuss the forced convection heat transfer to the, for flow over flat plates, flow across cylinders and spheres, and flow in tubes.

We began with the basic fundamentals of heat transfer by defining the some technical word. With a brief description of Reynolds number and I explained differences between laminar and turbulent flow. Newton's law of cooling as expresses the rate of convection heat transfer in external flow $\dot{Q}_{conv} = hA\,(T_s - T_\infty)$

Where T_s is the surface temperature and T_∞ is the free-stream temperature. As we saw in chapter one

As for the second chapter we found flow over flat plates. In this chapter we have discussed laminar and turbulent flow, flow over flat plates. Also I gave one example for this phoneme.

In the next chapter we explained main characteristics of the flow across cylinder and spheres, when it is laminar and when it is turbulent. I gave also information about heat transfer coefficient. Flow across cylinders and spheres in general involve flow separation, which it is difficult. Therefore such flows must be studied experimentally. And they developed experimentally some empirical correlations. We saw that for example Nusselt number is different for cylinder and spheres. I solved two example, which, is related with this subject.

The last chapter, we considered very important topics that are related flow in tubes. After explaining some technical word which, is related with flow in tubes and than explaining certain keys differences between laminar and turbulent flows in tube. 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 $\dot{Q} = \dot{m}\,C_p\,(T_e - T_i)$

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 $\dot{q}_s$ = constant. In this case of $\dot{q}_s$ = constant, the rate of heat transfer can be expressed as we explained in chapter.

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