UNIT: II - CONVECTION

1. Define critical Reynolds number. What is its typical value for flow over a flat plate and flow through a pipe? (May 2013, Nov/Dec 16)

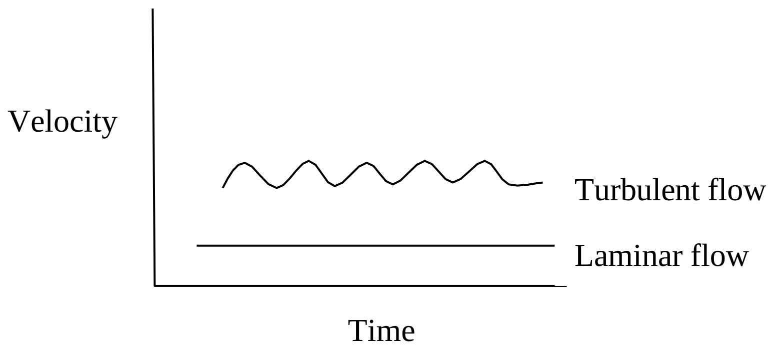
The critical Reynolds number refers to the transition from laminar to turbulent flow.

The critical Reynolds number for flow over a flat plate is $5*10^5$; the critical Reynolds number for flow through a pipe is 4000.

2. How does or Distinguish laminar flow differ from turbulent flow? (May 2013 & May 2015)

Laminar flow: Laminar flow is sometimes called stream line flow. In this type of flow, the fluid moves in layers and each fluid particle follows a smooth continuous path. The fluid particles in each layer remain in an orderly sequence without mixing with each other.

Turbulent flow: In addition to the laminar type of flow, a distinct irregular flow is frequently observed in nature. This type of flow is called turbulent flow. The path of any individual particle is zig-zag and irregular**.**



3. Differentiate viscous sub layer and buffer layer. (May 2014)

In the turbulent boundary layer, a very thin layer next to the wall where viscous effect is dominant called the viscous sub layer. The velocity profile in this layer is very nearly linear and the flow is streamlined.

In the turbulent boundary layer, next to viscous sub layer, a layer called **buffer layer** in which turbulent effects are becoming significant, but the flow is still dominated by viscous effects.

4. Define grashoff number and prandtl number. Write its significance. (May 2014 & Nov 2014 & Nov 2015-Reg 2008)(Nov 2015) (APR/MAY 2017)

Grashoff number is defined as the ratio of product of inertia force and buoyancy force to the square of viscous force.

$$Gr = \frac{\text{Inertia Force} * \text{Buoyancy Force}}{(\text{Viscous Force})^2}$$ [HMT Data Book, P.No 112]

Significance: Grashoff number has a role in free convection similar to that played by Reynolds number in forced convection.

Prandtl number is the ratio of the momentum diffusivity of the thermal diffusivity.

$$Pr = \frac{\text{Momentum Diffusivity}}{\text{Thermal Diffusivity}}$$ [HMT Data Book, P.No. 112]

Significance: Prandtl number provides a measure of the relative effectiveness of the momentum and energy transport by diffusion.

5. Define velocity boundary layer thickness. (May 2015)

The region of the flow in which the effects of the viscous shearing forces caused by fluid viscosity are felt is called velocity boundary layer. The velocity boundary layer thickness, δ, is defined as the distance from the surface at which velocity, u = 0.99V

6. Air at 27°C and 1 atmospheric flow over a flat plate at a speed of 2m/s. Calculate boundary layer thickness at a distance 40 cm from leading edge of plate. At 27°C viscosity (air) = 1.85 *10⁻⁵ kg/ms. (Nov 2012)

<u>Given Data:</u>

$T = 27^0C = 27+273 = 300K$

$P = 1 \text{ atm} = 1 \text{ bar} = 1.01325 * 10^5 \text{ N/m}^2$

$U = 2 \text{ m/s}$

$\mu = 1.85 * 10^{-5}$ kg/ms. (At 27^0C)

$R = 287$ (Gas constant)

To Find: δ at X = 40 cm = 0.4 m

<u>Solution:</u>

Step: 1 Density $\rho = P/RT$

$$= \frac{1.01325 * 10^5}{(287*300)}$$

$= 1.177\ Kg/m^3$

(Note: If Surface temperature (T_w) is given, then properties to be taken for T_f Value.)

Step: 2 Reynolds Number $Re = \rho UX/\mu$ [HMT Data Book, P.No. 112]

$$= \frac{1.177*2*0.4}{1.85*10^{-5}}$$

$= 55160$. ($Re < 5*10^5$, flow is laminar)

Step: 3 Boundary layer thickness $\delta = 5* X * (Re)^{-0.5}$

[HMT Data Book, P.No.113]

$= 5 * 0.4 * (55160)^{-0.5}$

$= 0.0085$ m

Boundary layer thickness δ at X (0.4m) = 0.0085 m

7. A square plate 40*40 cm maintained at 400K is suspended vertically in atmospheric air at 300 K. Determine the boundary layer thickness at trailing edge of the plate. (Nov 2012)

Given Data:

Length of horizontal plate X = 40 cm = 0.4m

Wide W = 40 cm = 0.40 m

Plate temperature $T_w = 400K = 127^0C$

Fluid temperature $T_\alpha = 300K = 27^0C$

$\Delta T = (T_w - T_\alpha) = 400-300 = 100$

To Find: δ at X = 40 cm = 0.4 m

Solution:

Step: 1 Film Temperature (T_f) $= \frac{T_w + T_\alpha}{2}$

$$= \frac{127+27}{2} = 77^0C = 350K$$

Step: 2 Properties of air at 77^0C (apprx 75^0C)

[HMT Data Book, P.No.34]

$\nu = 20.56 * 10^{-6}\ m^2/s$

Pr = 0.693

Step: 3 Find $\beta = 1 / T_f$ in K

$= 1 / 350$

$= 2.857 * 10^{-3}\ K^{-1}$

Step: 4 For free Convection (Note: As Velocity not given)

$$Gr = \frac{g^* \beta^* X^3 {}^* \Delta T}{\nu^2}$$ [HMT Data Book, P.No.135]

$$= \frac{9.81 * 2.857 * 10^{-3} * (0.4)^3 * (400-300)}{(20.56 * 10^{-6})^2}$$

$= 4.24 * 10^8$

Step: 5 Boundary layer thickness $\delta = 3.93 * X * (Pr)^{-0.5} * (0.952+Pr)^{0.25} * Gr^{-0.25}$

[HMT Data Book, P.No.135]

$= 3.93 * 0.4 * (0.693)^{-0.5} * (0.952+0.693)^{0.25} * (4.24*10^8)^{-0.25}$

$= 0.0155$ m

Boundary layer thickness δ at X (0.4m) = 0.0155 m

8. Define the term thermal boundary layer thickness. (Nov 2013)

The thickness of the thermal boundary layer δ_t at any location along the surface is defined as the distance from the surface at which the temperature difference equals to $0.99(T_\alpha - T_s)$, in general $T = 0.99 T_\alpha$

9. Why heat transfer coefficient for natural convection is much lesser than that for forced convection? (Nov 2013 & May 2016)

Heat transfer coefficient depends on the fluid velocity.

In natural convection, the fluid motion occurs by natural means such as buoyancy. Since the fluid velocity associated with natural convection is relatively low, the heat transfer coefficient encountered in natural convection is low.

The reason for higher heat transfer rates in forced convection is because the hot air surrounding the hot body is immediately removed by the flow of air around it. This is why forced convection heat transfer coefficient is greater than natural convection heat transfer coefficient.

10. Name four dimensions used for dimensional analysis. (Nov 2014)

1. Velocity
2. Density
3. Heat transfer coefficient
4. Thermal conductivity

11. Mention the significance of boundary layer. (Nov 2015)

Boundary layer is the layer of fluid in the immediate vicinity of a bounding surface where the effects of viscosity are significant.

12. What is Dittus Boelter equation? When does it apply? (Nov 2015)

Dittus-Boelter equation (for fully developed internal flow - turbulent flow) is an explicit function for calculating the Nusselt number. It is easy to solve but is less accurate when there is a large temperature difference across the fluid. It is tailored to smooth tubes, so use for rough tubes (most commercial applications) is cautioned.

The Dittus-Boelter equation is:

$$Nu_D = 0.023\ Re_D^{0.8}\ Pr^n \qquad \text{[HMT Data Book, P.No.126]}$$

13. What is the difference between friction factor and friction coefficient? (May 2016)

Friction factor, a dimensionless quantity used in the Darcy–Weisbach equation, for the description of friction losses in pipe flow as well as open-channel flow. Friction coefficient applied at the value of x (x=x-Local friction coefficient, x=L – Average friction coefficient)

14. Differentiate free and forced convection. (May 2016) (Nov/Dec 16)

Natural convection, or free convection, occurs due to temperature differences which affect the density, and thus relative buoyancy, of the fluid. Free convection is governed by Grashoff number and Prandtl number.

Example: Rise of smoke from a fire.

In forced convection, fluid movement results from external forces such as a fan or pump. Forced convection is typically used to increase the rate of heat exchange. It is governed by the value of the Reynolds number.

Example: Cooling of IC engines with fan in a radiator.

15. Differentiate hydrodynamic and thermal boundary layer. (May 2016)

The hydrodynamic boundary layer is a region of a fluid flow, near a solid surface, where the flow patterns (velocity) are directly influenced by viscous drag from the surface wall. The velocity of the fluid is less than 99% of free stream velocity.

The thermal boundary layer is a region of a fluid flow, near a solid surface, where the fluid temperatures are directly influenced by heating or cooling from the surface wall. The temperature of the fluid is less than 99% of free stream temperature.

16. What are the difference between natural convection and forced convection? (Nov/Dec 16)

Natural convection is a mechanism of heat transportation in which the fluid motion is not generated by an external source.

Forced convection is a mechanism, or type of heat transport in which fluid motion is generated by an external source (like a pump, fan, suction device, etc.)

1. Air at 25^0C at the atmospheric pressure is flowing over a flat plat at 3m/s. If the plate is 1m wide and the temperature T_w = 75^0C. Calculate the following at a location of 1m from leading edge.

- **a) Hydrodynamic boundary layer thickness,**
- **b) Local friction coefficient,**
- **c) Thermal heat transfer coefficient,**
- **d) Local heat transfer coefficient.**

Given Data:

Fluid temperature, $T_\alpha = 25^0C$

Velocity, U=3m/s

Wide, W = 1m

Plate surface temperature, $T_w = 75^0C$

Distance, x = 1m

To Find: δ_{hx}, C_{fx}, δ_{Tx}, h_x,

Solution: [From HMT Data Book, P.No.113]

Film temperature, $T_f = \frac{Tw + T\alpha}{2}$

$= \frac{75+25}{2} = 323K$

$T_f = 50^0C$

Properties of air at 50^0C:

[From HMT Data Book, P.No.34]

Density, ρ =1.093kg/m^3

Kinematic viscosity, v=17.95 x 10^{-6} m^2/s

Prandtl number Pr=0.698

Thermal conductivity, k =0.02826 W/mk

Reynolds number, Re =UL/v [From HMT Data Book, P.No.112]

[∵x=L=1m]

$\frac{3*1}{17.95*10^{-6}} = 1.67*10^5$

$Re=1.67*10^5<5*10^5$

Since Re=<$5*10^5$ flow is laminar.

For the plate, laminar flow.

[From HMT Data Book, P.No.113]

1. Hydrodynamic boundary layer thickness,

   $\delta_{hx} = 5*x*Re^{-0.5}$

   $= 5*x*(1.67*10^5)^{-0.5}$

   $\delta_{hx} = 0.0122m$

2. Local friction coefficient, [From HMT Data Book, P.No.113]

   $C_{fx}, = 0.664\ Re^{-0.5}$

   $= 0.664*(1.67*10^5)^{-0.5}$

   $C_{fx}, = 1.62*10^{-3}$

3. Thermal heat transfer coefficient,

[From HMT Data Book, P.No.113]

$\delta_{Tx} = \delta_{hx} * (Pr)^{-0.333}$

$=0.0122*(0.698)^{-0.333}$

$\delta_{Tx} = 0.01375$

4. Local heat transfer coefficient, h_x

[From HMT Data Book, P.No.113]

Local nusselt number $Nu_x = 0.332\ Re^{0.5} (Pr)^{0.333}$

$= 0.332\ (1.67*10^5)^{0.5} (0.698)^{0.333}$

$Nu_x = 120.415$

[From HMT Data Book, P.No.112]

$Nu_x = \frac{h_x * L}{k}$

$120.415 = \frac{h_x * 1}{0.02826}$ [∵x=L=1m]

Local heat transfer coefficient, $h_x = 3.4 W/m^2K$

Result:

a) $\delta_{hx} = 0.0122m$

b) $C_{fx}, = 1.62*10^{-3}$

c) $\delta_{Tx} = 0.01375$

d) $h_x = 3.4 W/m^2K$

2. Air at 290°C flows over a flat plate at a velocity of 6 m/s. The plate is 1m long and 0.5 m wide. The pressure of the air is 6 KN/m². If the plate is maintained at a temperature of 70°C, estimate the rate of heat removed from the plate.

Given:

Fluid temperature $T_\infty = 290°C$

Velocity U = 6 m/s.

Length L = 1 m

Wide W= 0.5 m

Pressure of air P = 6 KN/m² = $6 \times 10^3 N/m^2$

Plate surface temperature T_w = 70°C

To find:

Heat removed from the plate

Solution: [From HMT Data Book, P.No.113]

Film temperature $T_f = \frac{T_w + T_\alpha}{2}$

$T_f = \frac{70 + 290}{2}$

T_f=180°C

Properties of air at 180°C (At atmospheric pressure)

[From HMT Data Book, P.No.34]

$\rho = 0.799$ Kg/m^3

$\nu = 32.49 \times 10^{-6}$ m^2/s

Pr = 0.681

K = 37.80×10^{-3} W/mK

Note: Pressure other than atmospheric pressure is given, so kinematic viscosity will vary with pressure. Pr, K, C_p are same for all pressures.

Kinematic viscosity $\nu = \nu_{atm} \frac{P_{atm}}{P_{given}}$

[∵1 bar=1×10^{5}N/m^2]

$\nu = 32.49 \times 10^{-6} \times \frac{1 \times 10^5}{6 \times 10^3}$

Kinematic viscosity ν =5.145×10^{-4}m^2/s

[From HMT Data Book, P.No.112]

Reynolds number $Re = \frac{UL}{\nu}$

$= \frac{6 \times 1}{5.145 \times 10^{-4}}$

Re = 1.10×10^4-5×10^5

Since Re<5×10^5,flow is laminar

For plate, laminar flow, UL ν

[From HMT Data Book, P.No.113]

Local nusselt number $Nu_x = 0.332\, Re^{0.5} (Pr)^{0.333}$

$= 0.332\, (1.10 \times 10^4)^{0.5} (0.681)^{0.333}$

$Nu_x=30.63$

[From HMT Data Book, P.No.112]

$$NU_x=\frac{h_x L}{K}$$

$$30.63 = \frac{h_x \times 1}{37.80 \times 10^{-3}} \qquad [\because L = 1 \text{ m}]$$

Local heat transfer coefficient $h_x = 1.15$ W/m²K

Average heat transfer coefficient $h = 2 \times h_x$

$$h = 2 \times 1.15$$

$$h = 2.31 \text{W/m}^2\text{K}$$

Heat transferred $Q = h A (T_\alpha - T_w)$

$= 2.13 \times (1 \times 0.5) \times (563-343)$

$Q=254.1W$

Heat transfer from both side of the plate = 2×254.1

= 508.2 W.

Result: Heat transfer from both side of the plate = 508.2 W

3. A large vertical plate 4 m height is maintained at 606°C and exposed to atmospheric air at 106°C. Calculate the heat transfer is the plate is 10 m wide.

Given :

Vertical plate length (or) Height, L = 4 m

Wall temperature, $T_w = 606°C$

Air temperature, $T_\infty = 106°C$

Wide, W = 10 m

To find:

a) Heat transfer, (Q)

Solution: [From HMT Data Book, P.No.113]

Film temperature $T_f = \frac{Tw + T\alpha}{2}$

$$= \frac{606 + 106}{2}$$

$T_f = 356°C$

[From HMT Data Book, P.No.34]

Properties of air at 356°C = 350°C

Density, ρ =0.566kg/m^3

Kinematic viscosity, v=55.46 x 10^{-6} m^2/s

Prandtl number Pr=0.698

Thermal conductivity, k =49.08 x 10^{-3}W/mk

Coefficient of thermal expansion $\beta = \frac{1}{T_f \text{ in K}}$

$$= \frac{1}{356 + 273} = \frac{1}{629}$$

$$\beta = 1.58 \times 10^{-3} K^{-1}$$

$$\text{Grashof number Gr} = \frac{g \times \beta \times L^3 \times \Delta T}{v^2}$$

$$\Rightarrow \text{Gr} = \frac{9.81 \times 2.4 \times 10^{-3} \times (4)^3 \times (606 - 106)}{(55.46 \times 10^{-6})^2}$$

$$\Rightarrow \text{Gr} = 1.61 \times 10^{11}$$

Gr Pr = 1.61 $\times$ 10^{11} $\times$ 0.676

Gr Pr = 1.08 $\times$ 10^{11}

Since Gr Pr > 10^9, flow is turbulent

For turbulent flow,

Nusselt number Nu = 0.10 $[\text{Gr Pr}]^{0.333}$

$$\Rightarrow \text{Nu} = 0.10 [1.08 \times 10^{11}]^{0.333}$$

$$\text{Nu} = 471.20$$

[From HMT Data Book, P.No.112]

Nusselt number $\text{Nu} = \frac{hL}{K}$

$$\Rightarrow 472.20 = \frac{h \times 4}{49.08 \times 10^{-3}}$$

Heat transfer coefficient h = 5.78 W/m^2K

Heat transfer Q = h A ΔT

$= h \times W \times L \times (T_w - T_\infty)$

$= 5.78 \times 10 \times 4 \times (606 - 106)$

Q =115600 W

$Q = 115.6 \times 10^3$ W

Result:

Heat transfer $Q = 115.6 \times 10^3$ W

4. A thin 100 cm long and 10 cm wide horizontal plate is maintained at a uniform temperature of 150°C in a large tank full of water at 75°C. Estimate the rate of heat to be supplied to the plate to maintain constant plate temperature as heat is dissipated from either side of plate.

Given :

Length of horizontal plate, L = 100 cm = 1m

Wide, W = 10 cm = 0.10 m

Plate temperature, T_w = 150°C

Fluid temperature, T_∞ = 75°C

To find:

a) Heat loss (Q) from either side of plate

Solution:

Film temperature, $T_f = \frac{T_w + T_\alpha}{2}$ [From HMT Data Book, P.No.113]

$= \frac{150+75}{2} = 323K$

$T_f = 112.5^0C$

Properties of water at 112.5^0C

$P = 951 Kg/m^3$

$V = 0.264 \times 10^{-6}\ m^2/s$

$P_r = 1.55$

$K = 683 \times 10^{-3} W/mK$

Coefficient of thermal expansion $\beta = \frac{1}{T_f \text{ in K}} = \frac{1}{112.5+273} = 2.59 \times 10^{-3} k^{-1}$

Grashof Number $Gr = \frac{g \times \beta \times L^3 \times \Delta T}{v^2}$

For horizontal plate,

Characteristic length $L_c = \frac{W}{2} = \frac{0.10}{2}$

$L_c = 0.05$ m

$$Gr = \frac{9.81 \times 2.59 \times 10^{-3} \times (0.05)^3 \times (150-75)}{(0.264 \times 10^{-6})^2}$$

$Gr = 3.41 \times 10^9$

$GrPr = 3.14 \times 10^9 \times 1.55$

$Gr\,Pr = 5.29 \times 10^9$

Gr Pr value is in between 8×10^6 and 10^{11}

i.e., $8 \times 10^6 < Gr\,Pr < 10^{11}$

For horizontal plate, upper surface heated:

Nusselt number $Nu = 0.15\,(Gr\,Pr)^{0.333}$

[From HMT Data Book, P.No.114]

$Nu = 0.15\,(5.29 \times 10^9)^{0.333}$

$Nu = 259.41$

Nusselt number $Nu = \frac{h_u L_c}{K}$

$259.41 = \frac{h_u \times 0.05}{683 \times 10^{-3}}$

$h_u = 3543.6\ W/m^2K$

Upper surface heated, heat transfer coefficient $h_u = 3543.6\ W/m^2K$

For horizontal plate, lower surface heated:

Nusselt number $Nu = 0.27\,[Gr\,Pr]^{0.25}$

$Nu = 0.27\,[5.29 \times 10^9]^{0.25}$

$Nu = 72.8$

[From HMT Data Book, P.No.113]

Nusselt number $Nu = \frac{h_1 L_C}{K}$

$72.8 = \frac{h_1 L_C}{K}$

$72.8 = \frac{h_1 \times 0.05}{683 \times 10^{-3}}$

$h_1 = 994.6 W/m^2K$

Lower surface heated, heat transfer coefficient $h_1 = 994.6\ W/m^2K$

Total heat transfer $Q = (h_u + h_1) \times A \times \Delta T$

$= (h_u + h_1) \times W \times L \times (T_w - T_\infty)$

$= (3543.6 + 994.6) \times 0.10 \times (150 - 75)$

$Q = 34036.5\ W$

Result:

Total heat transfer Q = 34036.5 W

5. Explain in detail about the boundary layer concept.

The concept of a boundary layer as proposed by prandtl forms the starting point for the simplification of the equation of motion and energy.

When a real i.e., viscous fluid, flow along a stationary solid boundary, a layer of fluid which comes in contact with boundary surface and undergoes retardation this retarded layer further causes retardation for the adjacent layer of the fluid. So small region is developed in the immediate vicinity of the boundary surface in which the velocity of the flowing fluid increases rapidly from zero at boundary surface and approaches the velocity of main stream.

Types of boundary layer

1. **Velocity boundary layer (or)hydrodynamic boundary layer**
2. **Thermal boundary layer**

Velocity boundary layer (or) hydrodynamic boundary layer

In the Velocity boundary layer, velocity of the fluid is less than 99% of free steam velocity.

The fluid approaches the plate in x direction with uniform velocity u_∞. The fluid particles in the fluid layer adjacent to the surface get zero velocity. This motionless layer acts to retard the motion of particles in the adjoining fluid layer as a result of friction between the particles of these two adjoining fluid layers at two different velocities. This fluid layer then acts to retard the motion of particles of next fluid layer and so on, until a distance y =d from the surface reaches, where these effects become negligible and the fluid velocity u reaches the free stream velocity u_∞ as a result of frictional effects between the fluid layers.

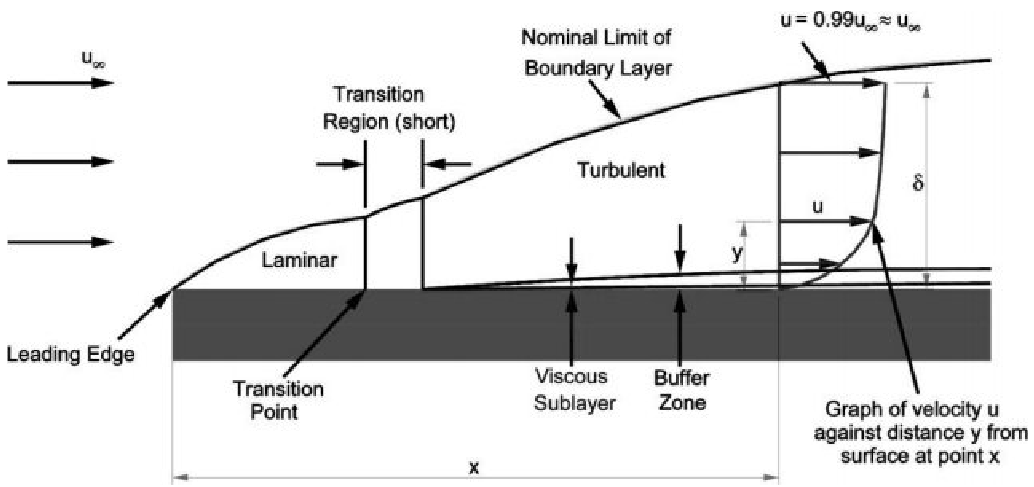
Thermal boundary Layer:

In the Thermal boundary layer, temperature of the fluid is less than 99% of free steam temperature.

If the fluid flowing on a surface has a different temperature than the surface, the thermal boundary layer developed is similar to the velocity boundary layer. Consider a fluid at a temperature T_∞ flows over a surface at a constant temperature Ts. The fluid particles in adjacent layer to the plate get the same temperature that of surface. The particles exchange heat energy with particles in adjoining fluid layers and so on. As a result, the temperature gradients are developed in the fluid layers and a temperature profile is developed in the fluid flow, which ranges from Ts at the surface to fluid temperature T_∞ sufficiently far from the surface in y direction.

Velocity boundary layer on a flat plate:

It is most essential to distinguish between laminar and turbulent boundary layers. Initially, the boundary layer development is laminar as shown in figure for the flow over a flat plate. Depending upon the flow field and fluid properties, at some critical distance from the leading edge small disturbances in the flow begin to get amplified, a transition process takes place and the flow becomes turbulent. In laminar boundary layer, the fluid motion is highly ordered whereas the motion in the turbulent boundary layer is highly irregular with the fluid moving to and from in all directions. Due to fluid mixing resulting from these macroscopic motions, the turbulent boundary layer is thicker and the velocity profile in turbulent boundary layer is flatter than that in laminar flow.



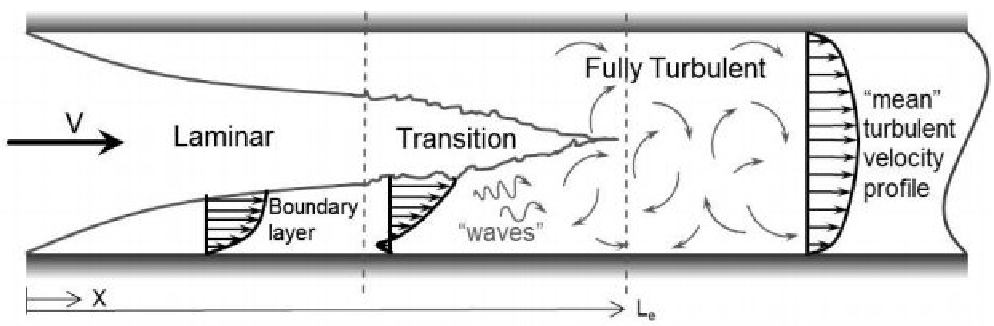
Velocity boundary layer on a tube:

Laminar Boundary Layer Flow

The laminar boundary is a very smooth flow, while the turbulent boundary layer contains swirls or "eddies." The laminar flow creates less skin friction drag than the turbulent flow, but is less stable. Boundary layer flow over a wing surface begins as a smooth laminar flow. As the flow continues back from the leading edge, the laminar boundary layer increases in thickness.

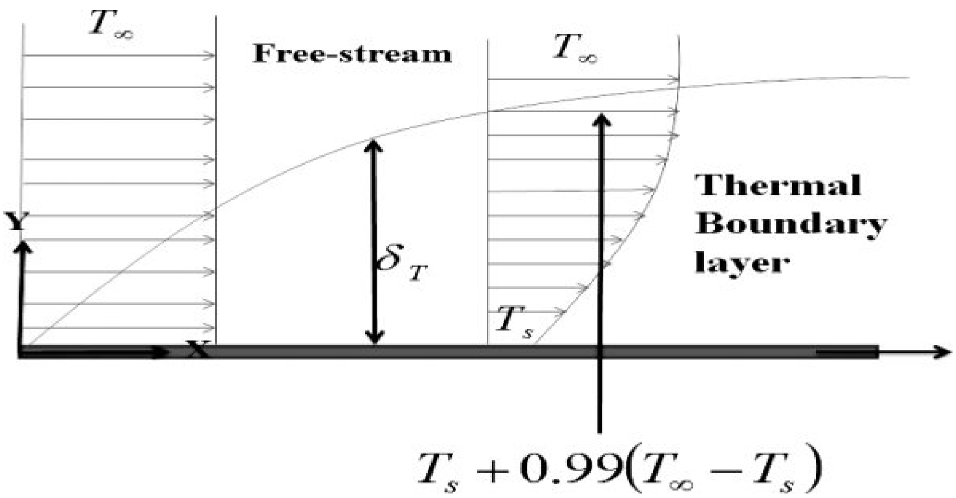
Turbulent Boundary Layer Flow

At some distance back from the leading edge, the smooth laminar flow breaks down and transitions to a turbulent flow. From a drag standpoint, it is advisable to have the transition from laminar to turbulent flow as far aft on the wing as possible, or have a large amount of the wing surface within the laminar portion of the boundary layer. The low energy laminar flow, however, tends to break down more suddenly than the turbulent layer.

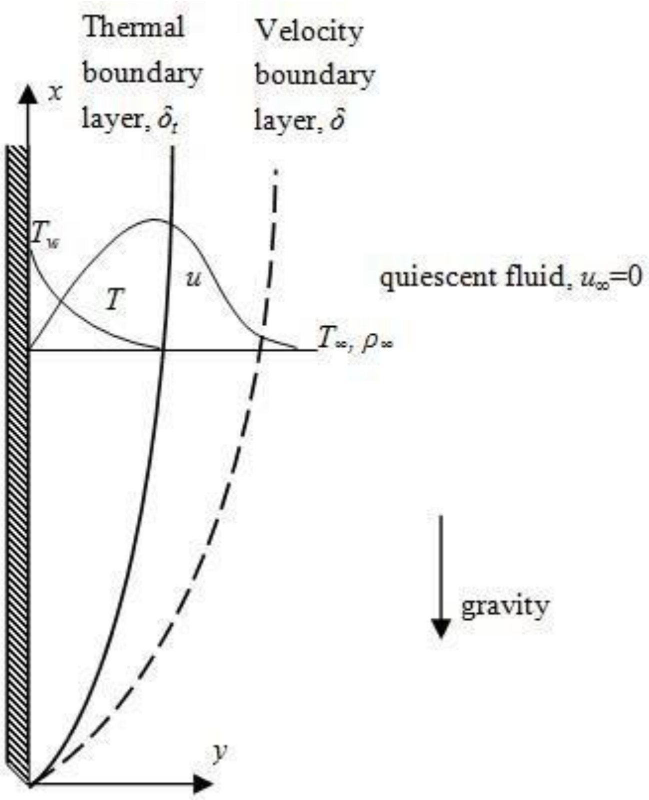


Thermal boundary Layer on a flat plate:

Consider a fluid of uniform temperature T_α approaching a flat plate of constant temperature T_s in the direction parallel to the plate. At the solid/liquid interface the fluid temperature is T_s since the local fluid particles achieve thermal equilibrium at the interface. The fluid temperature T in the region near the plate is affected by the plate, varying from T_s at the surface to T_α in the main stream. This region is called the thermal boundary layer.



Velocity and Temperature boundary layer (Profile) for a vertical plate



6. In a long annulus (3.125 cm ID and 5 cm OD) the air is heated by maintaining the temperature of the outer surface of inner tube at 50°C. The air enters at 16°C and leaves at 32°C. Its flow rate is 30 m/s. Estimate the heat transfer coefficient between air and the inner tube.

Given : Inner diameter D_i = 3.125 cm = 0.03125 m

Outer diameter D_o = 5 cm = 0.05 m

Tube wall temperature T_w = 50°C

Inner temperature of air T_{mi} = 16°C

Outer temperature of air t_{mo} = 32°C

Flow rate U = 30 m/s

To find: Heat transfer coefficient (h)

Solution:

Step 1. Mean temperature $T_m = \frac{T_{m1}+T_{m2}}{2}$

$= T_m = \frac{16+32}{2}$

24°C

Properties of air at 24°C [From HMT Data book page no. 34]

ρ = 1.185 Kg/m³

$\nu = 15.53 \times 10^{-6}$ m²/s

Pr = 0.702

k = 0.02634 W/mK

Step 2. Hydraulic or Equivalent diameter

$D_h = \frac{4A}{P} = \frac{4 \times \frac{\pi}{4}[D_o^2 - D_i^2]}{\pi[D_o+D_i]}$

$= \frac{[D_o+D_i][D_o-D_i]}{[D_o+D_i]}$

= Do - Di

= 0.05 – 0.03125

D_h = 0.01875 m

Step 3. Reynolds number, $Re = \frac{UD_h}{\nu}$

$= \frac{30X0.01875}{15.53X10^{-6}}$

$Re = 36.2 \times 10^3$

Since Re > 2300, flow is turbulent.

For turbulent flow, general equation is (Re > 10000).

$Nu = 0.023 (Re)^{0.8} (Pr)^n$

[From HMT Data book, Page No. 126]

This is heating process. So n = 0.4. $[T_{mo} > T_{mi}]$

Step 4. $Nu = 0.023 \times (36.2 \times 10^3)^{0.8} (0.702)^{0.4}$

$Nu = 88.59$

Step 5. $Nu = \frac{hD_h}{k}$

$$88.59 = \frac{h X 0.01875}{26.34 X 10^{-3}}$$

$h = 124.4 \ W/m^2K$

Heat transfer coefficient, $h = 124.4 \ W/m^2K$

7. In a surface condenser, water flows through staggered tubes while the air is passed in cross flow over the tubes. The temperature and velocity of air are 30°C and 8 m/s respectively. The longitudinal and transverse pitches are 22 mm and 20 mm respectively. The tube outside diameter is 18 mm and tube surface temperature is 90°C. Calculate the heat transfer coefficient.

Given:

Fluid temperature, T_∞ = 30°C

Velocity, U = 8 m/s

Longitudinal pitch, S_l = 22 mm = 0.022 mm

Transverse pitch, S_t = 20 mm = 0.020 m

Diameter, D = 18 mm = 0.018 m

Tube surface temperature, T_w = 90°C

To find:

Step 1. Heat transfer coefficient.

Solution:

We know that,

Film temperature, $T_f = \frac{T_w + T_\infty}{2}$

$= \frac{90+30}{2}$

$T_f = 60°C$

Properties of air at 60°C

[From HMT data book, Page No. 34]

$\nu = 18.97 \times 10^{-6}\ m^2/s$

$Pr = 0.696$

$K = 0.02896\ W/mK$

Step 2. Maximum velocity, $U_{max} = U \times \frac{S_t}{S_t - D}$

$U_{max} = 8 \times \frac{0.020}{0.020-0.018}$ $U_{max} = 80\ m/s$

Step 3. Reynolds Number, $Re = \frac{U_{max} X D}{\nu}$

$= \frac{80\ X\ 0.018}{18.97 X 10^{-6}}$

$Re = 7.5 \times 10^4$

$\frac{S_t}{D} = \frac{0.020}{0.018} = 1.11$

$\boxed{\frac{S_t}{D} = 1.11}$

$\frac{S_l}{D} = \frac{0.022}{0.018} = 1.22$

$\boxed{\frac{S_t}{D} = 1.22}$

$\frac{S_t}{D} = 1.11$, $\frac{S_l}{D} = 1.22$, corresponding C, n values are 0.518 and 0.556 respectively.

[From HMT data book, page No. 123]

$C = 0.518$	$n = 0.556$

Step 4. Nusselt Number, $Nu = 1.13\ (Pr)^{0.333}\ [\ C\ (Re)^{n}]$

[From HMT data dook, Page No. 123]

$Nu = 1.13x(0.696)^{0.333}x[\ 0.518\ x\ (7.5\ x\ 10^{4})^{0.556}]$

$Nu = 266.3$

Step 5. Nusselt Number, $Nu = \frac{hD}{k}$

$$266.3 = \frac{hX0.018}{28.96\ X\ 10^{-3}}$$

Heat transfer coefficient, $h = 428.6\ W/m^2K$

8. A thin 100 cm long and 10 cm wide horizontal plate is maintained at a uniform temperature of 150°C in a large tank full of water at 75°C. Estimate the rate of heat to be supplied to the plate to maintain constant plate temperature as heat is dissipated from either side of plate.

Given:

Length of horizontal plate L = 100 cm = 1m

Wide W = 10 cm = 0.10 m

Plate temperature T_w = 150°C

Fluid temperature T_∞ = 75°C

To find: Heat loss (Q) from either side of plate:

Solution:

Step 1. Film temperature, $T_f = \frac{T_w + T_\infty}{2}$

$$= \frac{150+75}{2}$$

T_f = 112.5°C

Properties of water at 112.5°C:

[From HMT data book, Page No. 22]

$\rho = 951\ Kg/m^3$

$\nu = 0.264\ x\ 10^{-6}\ m^2/s$

Pr = 1.55

k = 0.683 W/mK

$\beta_{(for\ water)}$ = 0.8225 x 10^{-3} K^{-1}

[From HMT data book, Page No. 30]

Step 2. Grashof Number, Gr = $\frac{g \times \beta \times L_c^3 \times \Delta T}{\nu^2}$

For horizontal plate,

Characteristic length, L_c = $\frac{W}{2} = \frac{0.10}{2}$

L_c = 0.05 m

Gr = $\frac{9.81\ X\ 0.8225\ X\ 10^{-3}\ X\ (0.05)^3\ X\ (150-75)}{(0.264\ X\ 10^{-6})^2}$

Gr = 1.0853 x 10^9

GrPr = 1.0853 x 10^9 x 1.55

GrPr = 1.682 x 10^9

GrPr value is in between 8 x 10^6 and 10^{11}

i.e., 8 x 10^6 < GrPr < 10^{11}

For horizontal plate, upper surface heated:

Step 3. Nusselt Number, Nu = 0.15 $(GrPr)^{0.333}$

[From HMT data book, Page No. 136]

Nu = 0.15 [1.682 x 10^9]0.333

Nu = 177.13

Step 4. Nusselt Number, Nu = $\frac{h_u L_c}{k}$

177.13 = $\frac{h_u\ X\ 0.05}{0.683}$

h_u = 2419.7 W/m^{2}K

Upper surface heated, heat transfer coefficient

h_u = 2419.7 W/m^{2}K

For horizontal plate, lower surface heated:

Step 5. Nusselt Number Nu = 0.27 $[GrPr]^{0.25}$

[From HMT data book, Page No. 136]

Nu = 0.27 $[1.682 \times 10^9]^{0.25}$

Nu = 54.68

Step 6. Nusselt Number, Nu = $\frac{h_l L_c}{k}$

$$54.68 = \frac{h_u \, X \, 0.05}{0.683}$$

$$h_l = 746.94 \text{ W/m}^2\text{K}$$

Lower surface heated, heat transfer coefficient, h_l = 746.94 W/m^2K

Step 7. Total heat transfer, Q = $(h_u + h_l)$ x A x ΔT

= $(h_u + h_l)$ x W x L x $(T_w - T_\infty)$

= (2419.7 + 746.94) X 0.10 X (150-75)

Heat transfer, Q = 23749.8 W

9. Atmospheric air at 275 K and a free stream velocity of 20 m/s flows over a flat plate 1.5 m long that is maintained at a uniform temperature of 325 K. Calculate the average heat transfer coefficient over the region where the boundary layer is laminar, the average heat transfer coefficient over the entire length of the plate and the total heat transfer rate from the plate to the air over the length 1.5 m and width 1 m. Assume transition occurs at Re_c = 2 x 10^5.

Given: Fluid temperature, T_∞ = 275 K = 2ºC

Velocity, U = 20 m/s

Length, L = 1.5 m

Plate surface temperature, T_w = 325 K = 52°C

Width, W = 1 m

Critical Reynolds number, Re_c = 2 x 10^5

To find: 1. Average heat transfer coefficient, h_l [Boundary layer is laminar]

2. Average heat transfer coefficient, h_t [Entire length of the plate]

3. Total heat transfer rate, Q.

Solution:

Step 1. Film temperature, $T_f = \frac{T_w + T_\infty}{2}$

$$= \frac{52+2}{2}$$

$T_f = 27°C$

Properties of air at 27°C ≈ 25°C

[From HMT data book, Page No. 34]

$\rho = 1.185\ Kg/m^3$

$\nu = 15.53 \times 10^{-6}\ m^2/s$

$Pr = 0.702$

$k = 0.02634\ W/mK$

Case (i): Reynolds number, $Re = \frac{UL}{\nu}$

Transition occurs at $Re_c = 2 \times 10^5$

ie., Flow is laminar upto Reynolds number value is 2×10^5, after that flow is turbulent.

$$2 \times 10^5 = \frac{20 \times L}{15.53 \times 10^{-6}}$$

$L = 0.155\ m$

For flat plate, laminar flow,

Step 2. Local Nusselt number, $Nu_x = 0.332\ (Re)^{0.5}(Pr)^{0.333}$

[From HMT data book, Page No. 113]

$$Nu_x = 0.332\ (2 \times 10^5)^{0.5}(0.702)^{0.333}$$

$$Nu_x = 131.97$$

Step 3. Local Nusselt Number, $Nu_x = \frac{h_x L}{k}$

$$131.97 = \frac{h_x \times 0.155}{0.02634}$$

$$h_x = 22.42\ W/m^2K$$

Local heat transfer coefficient, $h_x = 22.42\ W/m^2K$ W/m^2K

Step 4. Average heat transfer coefficient, $h = 2 \times h_x$

$= 2 \times 22.42$

$= 44.84\ W/m^2K$

Case (ii):

Step 5. Reynolds number, Re_L [For entire length] $= \frac{UL}{\nu}$

$$= \frac{20 \times 1.5}{15.53 \times 10^{-6}}$$

$= 1.93 \times 10^6 > 5 \times 10^6$

Since $Re_L > 5 \times 10^5$, flow is turbulent.

For flat plate, laminar-turbulent combined flow,

Step 6. Average Nusselt number, $Nu = (Pr)^{0.333} [0.037 (Re_L)^{0.8} - 871]$

$Nu = (0.702)^{0.333}[0.037 (1.93 \times 10^6)^{0.8} - 871]$

$Nu = 2737.18$

Step 7. Nusselt number, $Nu = \frac{hL}{k}$

$2737.18 = \frac{h \times 1.5}{0.02634}$

$h = 48.06 \ W/m^2K$

Average heat transfer coefficient for turbulent flow, $h_t = 48.06 \ W/m^2K$

Step 8. Total heat transfer rate, $Q = h_t \times A \times \Delta T$

$= h_t \times W \times L \times (T_w - T_\infty)$

$= 48.06 \times 1 \times 1.5 \times (52-2)$

$Q = 3604.5 \ W$

10. A steam pipe 10 cm outside diameter runs horizontally in a room at 23°C. Take the outside surface temperature of pipe as 165°C. Determine the heat loss per metre length of the pipe. [Dec 2004]

Given: Diameter of the pipe, D = 10 cm = 0.10 m

Ambient air temperature, $T_\infty = 23°C$

Wall temperature, $T_w = 165°C$

To find: Heat loss per metre length.

Solution:

Step 1. Film temperature, $T_f = \frac{T_w + T_\infty}{2}$

$= \frac{165+23}{2}$

$T_f = 94°C$

Properties of air at 94°C≈95°C

[From HMT data book, Page No. 34]

$\rho = 0.959 \ Kg/m^3$

$\nu = 22.615 \times 10^{-6} \ m^2/s$

Pr = 0.689

k = 0.03169 W/mK

Step 2. Coefficient of thermal expansion, $\beta = \frac{1}{T_f \text{ in } K}$

$= \frac{1}{94+273}$

$= 2.72 \text{ X } 10^{-3} \text{ K}^{-1}$

$\beta = 2.72 \text{ X } 10^{-3} \text{ K}^{-1}$

Step 3. Grashof Number, $Gr = \frac{g \times \beta \times D^3 \times \Delta T}{\nu^2}$

[From HMT data book, Page No. 135]

$Gr = \frac{9.81 \text{ X } .72 \text{ X } 10^{-3} \text{ X } (0.10)^3 \text{ X } (165-23)}{(22.615 \text{ X } 10^{-6})^2}$

$Gr = 7.40 \times 10^6$

$GrPr = 7.40 \times 10^6 \times 0.689$

$GrPr = 5.09 \times 10^6$

For horizontal cylinder, Nusselt number, $Nu = C\,[GrPr]^m$

[From HMT data book, Page No. 138]

$GrPr = 5.09 \times 10^6$, corresponding C = 0.48, and m = 0.25

$Nu = 0.48\,[5.09 \times 10^6]^{0.25}$

Nu = 22.79

tep 4. Nusselt number, $Nu = \frac{hD}{k}$

$22.79 = \frac{h X 0.10}{0.03169}$

$h = 7.22 \text{ W/m}^2\text{K}$

Step 5. Heat loss, $Q = hA\Delta T$

$= h \times \pi DL(T_w - T_\infty)$

$\frac{Q}{L} = h \times \pi \times D \times (T_w - T_\infty)$

$= 7.22 \times \pi \times 0.10 \times (165 - 23)$

$\frac{Q}{L} = 322.08$ W/m

Heat loss per metre length, $\frac{Q}{L} = 322.08$ W/m

1. **Consider the flow of oil at 20° C in a 30cm diameter pipeline at an average velocity of 2 m/s. a 200m long section of the pipeline passes through icy waters of a lake at 0° C. Measurements indicate that the surface temperature of the pipe is very nearly 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.**

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

1. Steady operating conditions exist. 2. The surface temperature of the pipe is very nearly 0° C. 3. The thermal resistance of the pipe is negligible.4. The inner surfaces of the pipeline are smooth. 5. The flow is hydrodynamically 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 from HMT data book

ρ =888 kg/m³ $\quad$ U =901 $\times 10^{-6}$ m²/s
k = 0.145 W/m °C $\quad$ Cp =1880 J/kg _ °C
Pr = 10,400

$$R_e = \frac{UD}{\nu} = \frac{2\times0.3}{901\times10^{-6}} = 666$$

which is less than the critical Reynolds number of 2300. Therefore, the flow is laminar, and we assume thermally developing flow and determine the nusselt number from

$$Nu = \frac{hD}{k} = 3.66 + \frac{0.065\,(D/L)R_eP_r}{1+0.04[(D/L)R_eP_r]^{2/3}}$$

$$= 3.66 + \frac{0.065\,(0.3/200)\times 666\times 10400}{1+0.04[(0.3/200)666\times 10400]^{2/3}} = 37.3$$

This nusselt number is considerably higher than the fully developed value of 3.66 then

$$h = \frac{k}{D}\,Nu = \frac{0.0145}{0.3}(37.3) = 18.0\frac{W}{m^2}\ °C$$

also we determine the exit temperature of air from

$Te = Ts - (Ts - Ti)\exp(-h\,As\,/m\,Cp)$

here

$A_s = PL = \pi DL = \pi\ (0.3\ m)(200\ m) = 188.5\ m^2$

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

Substitute As and m in Te

$Te = 60 - (60 - 80)\exp(-13.5\times 6.4/0.151\times 1008) = 71.3\ °C$

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

$$\Delta T_{ln} = \frac{T_i - T_e}{ln\frac{T_s - T_e}{T_s - T_i}} = -15.2°C$$

$Q = h\,As\,\Delta T_{ln} = (13.5\ W/m2\ °C)(6.4\ m2)(-15.2°C) =$ **-1313 W**

Therefore, air will lose heat at a rate of 1313 W as it flows through the duct in the attic.

2. In condenser water flows through two hundred thin walled circular tubes having inner diameter 20mm and length 6 m. the mass flow rate of water is 160 kg/s. the water enters at 30° C and leaves at 50 ° C. Calculate the average heat transfer coefficient.

Given :

Inner diameter D = 20mm

Length L = 6 m

Mass flow rate m = 160 kg/s

Inlet water temperature T_{mi} = 30 ° C

Outlet water temperature, T_{mo} = 50 ° C

To find: Heat transfer coefficient (h)

Solution:

Bulk mean temperature $T_m = \frac{T_{mi} + T_{mo}}{2} = \frac{30+50}{2} = 40$ °C

Properties of water at 40 °C [from HMT data boo page no 21]

$\rho = 995\ kg/m^3$

$\nu = 0.657 \times 10^{-6}\ m^2/s$

Pr = 4.340

k = 0.628 W/mK

C_p = 4178 J/kg K

Reynolds Number $R_e = UD/\nu$

$m = \rho A U$

Velocity $U = m/\rho A$

$= \frac{(\frac{160}{200})}{995 \times \frac{\pi}{4} 0.020^2} = 2.55$ m/s (no of tubes = 200)

$R_e = UD/\nu$

$= \frac{2.55 \times 0.020}{0.657 \times 10^{-6}}$ =77625.57

Since R_e >2300, flow is turbulent

For turbulent flow, general equation is (R_e > 10000)

Nu = 0.023 [from HMT data boo page no 125]$\times Re^{0.8} Pr^n$

This is heating process so n = 0.4 ($T_{mo} > T_{mi}$)

$Nu = 0.023 \times 77625.57^{0.8} 4.340^{0.4}$

Nu = 337.8

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

$337.8 = \frac{h \times 0.020}{0.628}$

Heat transfer coefficient h = 10606.9 w/m²K