# EN 309
# THERMAL AND FLUID ENGINEERING LABORATORY
# MANUAL
**(2018-19)**

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# Index

# 1(a) IMPACT OF JET TEST RIG

**AIM**

The objective of this experiment is to verify the momentum conservation law.

**EXPERIMENTAL SETUP**

The setup mainly consists of a nozzle of about 10 mm in diameter fitted to a pipe of 50mm diameter. The jet of water issuing from nozzle strike a flat plate of about 100mm diameter placed at some distance to a nozzle (Figure 1). A force acts on the plate by which the shaft moves up. A weight W is placed on the pan P to counter-act the force F due to momentum transfer, on the plate. The plate again comes back to its original position. The force F acted by the jet on the flat plate is equal to the weight W.

The discharger, which falls in the tank after striking the flat plate, is used to measure the discharge by volumetric method.

**THEORY**

Whenever the velocity of a stream is changed either in magnitude or in direction, a force is required to bring this change.

The law of momentum conservation states that the summation of all the externally applied force on a given volume is equal to the rate of change of momentum brought in the direction of forces. In order to calculate the force caused by impact of jet onto a flat plate, principle of change in momentum is applies, i.e.,

$$\sum F_x = \rho Q(V_{in} - V_{out})$$

Where $\sum F_x$ is the summation of all the forces acting in X-direction, similar equations also be written in Y and Z direction.
$\rho$ is the density of water (= 1000 Kg/m$^3$)
$Q$ is the volumetric flow rate of fluid, m$^3$/s.

Volumetric flow rate in the above equation is calculated by taking the volume of fluid within a specified time.

$$Q = \frac{v}{t}$$

$V_{in}$ is calculated based on the velocity at the nozzle which is determined using the volumetric flow rate and the diameter of nozzle (dia = 10 mm):

$V_{nozzle} = \frac{Q}{A}$;

$V_{in}^2 = V_{nozzle}^2 - 2gS$,

Where;

g: the gravitational acceleration (9.81 $m/s^2$);
S: the distance between the jet and the plates.

$V_{out}$ generally equals $V_{in}$ cos θ, where θ represents the change in direction of the jet. Since the jet of the fluid turns by right angle when striking a flat plate
For the flat plat θ = 90◦, so that $V_{out}$ = 0.0

The predicted values for the force is thus given as

$$\sum F_x = \rho Q V_{in}$$

**PROCEDURE**

1. Switch on the power and start the motor.
2. Mark the position of vertical lever and horizontal arm when there is no weight on the pan.
3. Open the inlet valve so that jet of water strikes the flat plate at its center. The lever is deflected to one side.
4. Put a weight W on the pan so as to bring the lever in its original position as marked in step1.
5. Measure the discharge Q by volumetric method.
6. Repeat step 2 and 4 for various other discharges.

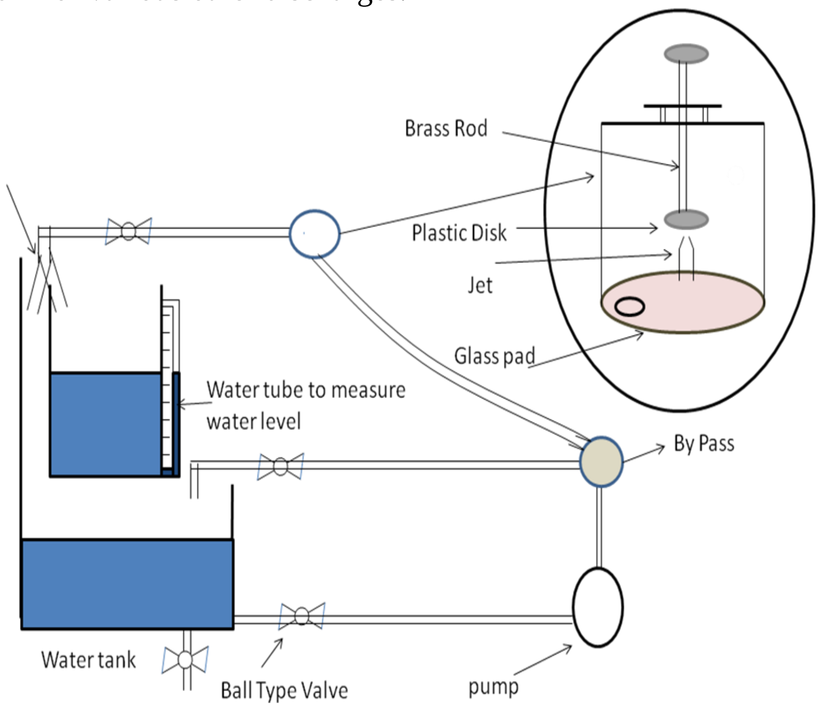



**Figure 1: IMPACT OF JET TEST RIG**

**OBSERVATION TABLE**

| Sr No. | Distance moved | Weight on pan | Volume (liters) | Time (sec) | Q, $m^3/s$ | $V_{nozzle}$, m/s | $V_{in}$ , m/s |
|---|---|---|---|---|---|---|---|
| 1. | | | | | | | |
| 2. | | | | | | | |
| 3. | | | | | | | |
| 4. | | | | | | | |
| 5. | | | | | | | |
| Average | | | | | | | |

| S no. | Inlet Area (A) | F1, Theoretical (N) | F1 actual (N) | % error |
|---|---|---|---|---|
| **1** | | | | |
| **2** | | | | |
| **3** | | | | |
| **4** | | | | |
| **5** | | | | |
| **Average** | | | | |

**FORMULAE USED**

$$F_{th} = \rho * AV^2$$

$$V = \frac{Q}{A}$$

ρ = Specific weight of water, where A is inlet area.

$$F_{th} = \rho * A * \frac{Q^2}{A^2}$$

$$F_{th} = \rho * \frac{Q^2}{A}$$

$$F_{act}\,(N) = (weight\ of\ pan + weight\ on\ pan) * g$$

Where, g is acceleration due to gravity in $m/s^2$

$$\%error = \frac{F_{act} - F_{th}}{F_{th}} * 100$$

**OBSERVATIONS**

- Diameter of nozzle = 10 mm
- Area of nozzle = $7.8 \times 10^{-5}$ $m^2$
- Sump tank = (0.3 X 0.3 X 0.9) $m^3$
- Area of tank = 0.09 $m^2$
- Weight of plate flat $w_1$ = 0.044 kg
- Weight of curve plate $w_2$ = 0.072 kg
- Weight of rod $w_3$ = 0.139 kg
- Total Weight = 0.255kg

**CONCLUSION/DISCUSSION ON THE RESULT:**

1. Write down the observations.

2. Try to explain the results from theory studied earlier

**FURTHER READING**

1. Introduction to Fluid Mechanics 8th Edition, by *Fox, Robert W. and McDonald*, Alan T. Chapter 4.

**TEACHING ASSISTANT:**

## 1(b) FALLING BALL VISCOMETER

**AIM**

The purpose of this experiment is to measure the viscosity of unknown oil with a falling ball viscometer.

Principle:

The principle of the viscometer is to determine the falling time of a sphere with known density and diameter within a fluid filled inside glass tube. The viscosity of the fluid sample is related to the time taken by the sphere to pass between two specified lines on the cylindrical tube.

**APPARATUS**

Figure 1 is a schematic of a falling ball viscometer. A sphere of known density and diameter is dropped into a large reservoir of the unknown fluid. At steady state, the viscous drag and buoyant force of the sphere is balanced by the gravitational force. In this experiment, the speed at which a sphere falls through a viscous fluid is measured by recording the sphere position as a function of time. Position is measured with a vertical scale (ruler) and time is measured with a stopwatch.

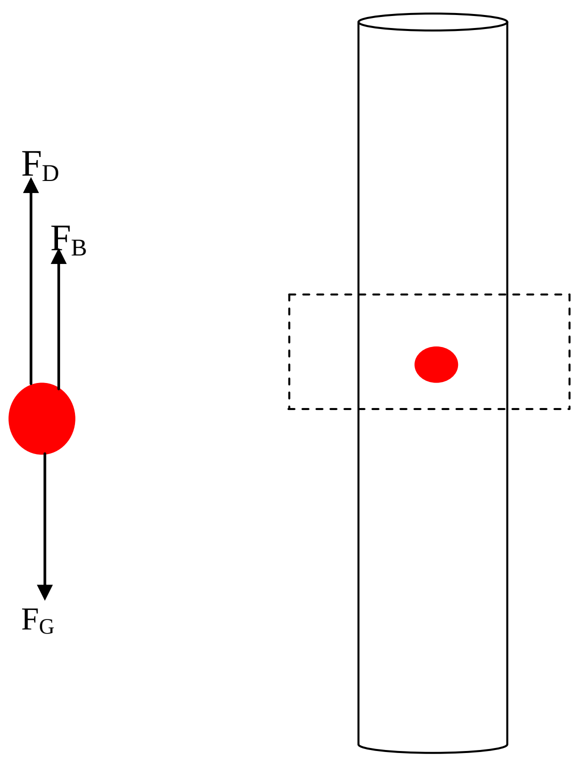



**Figure 1. Body Diagram for the falling ball viscometer.**

## THEORY

Velocity of the sphere which is falling through the tube is dependent on the viscosity of the fluid. When a sphere is placed in an infinite incompressible Newtonian fluid, it initially accelerates due to gravity. After this brief transient period, the sphere achieves a steady settling velocity (a constant terminal velocity). For the velocity to be steady (no change in linear momentum), Newton's second law requires that the net forces acting on the sphere (gravity ($F_G$), buoyancy ($F_B$), and fluid drag ($F_D$) balance) equals to zero. These forces all act vertically are defined as follows:

Gravity : $F_G = -\frac{\pi}{6}\rho_p d_p^3 g$

Buoyancy : $F_B = +\frac{\pi}{6}\rho_f d_p^3 g$

Fluid Drag : $F_D = \frac{\pi}{8}\rho_f V_p^2 d_p^2 C_D$

Where $\rho_p$ is the density of the solid sphere, $\rho_f$ is the density of the fluid, $d_p$ is the diameter of the solid sphere, g is the gravitational acceleration (9.8 $m/s^2$), $V_p$ is the velocity of the sphere, and $C_D$ is the drag coefficient. The particle accelerates to a steady velocity when the net force acting on sphere becomes zero:

$F_G - F_B - F_D = 0.$

The drag force acts upwards and is expressed in terms of a dimensionless drag coefficient. The drag coefficient is a function of the dimensionless Reynolds number, Re. The Reynolds number can be interpreted as the ratio of inertial forces to viscous forces. For a sphere settling in a viscous fluid the Reynolds number is

$$Re = \frac{\rho V_p d_p}{\mu}$$

Where $\mu$ is the viscosity of the fluid. If the drag coefficient as a function of Reynolds number is known the terminal velocity can be calculated. For the Stokes regime, Re<1, the drag coefficient can be determined either analytically. In this regime, $C_D = \frac{24}{Re}$and the settling velocity is

$$V_p = \frac{g d_p^2(\rho_p - \rho_f)}{18\mu}$$

The falling ball viscometer requires the measurement of a sphere's terminal velocity, usually by measuring the time required for sphere to fall a given distance. In this experiment we measure the position of a sphere as a function of time and determine the steady state settling velocity. From this, we can calculate the viscosity from below equation given depending on the Reynolds number. For Re<1 the viscosity would be

$$\mu = \frac{g d_p^2 (\rho_p - \rho_f) t_p}{18L}$$

**PROCEDURE**

Regardless of the Re, the settling velocity depends on the sphere diameter, the sphere density, the fluid density, and the gravitational constant.

Measure the diameter of the sphere. Measure it multiple times to gain an accurate measurement and to determine the relative error in the measurement.

The viscosity can be determined by measuring the position of the sphere as a function of time as it settles through the unknown fluid.

For each sphere

1. Place the sphere near the top of the fluid reservoir. Try to get the sphere as close as possible to the air-fluid interface.

2. Release the sphere and start the stopwatch as soon as the sphere reaches the top line marked on the glass tube and stop it as it reaches the bottom marked line.

3. As the sphere settles, record its position as a function of time. (it may be more efficient to have one person drop the sphere, one person run the stopwatch, and the third to read the time off the stopwatch).

Observation:

1. Density of sphere = 2500 $Kg/m^3$.
2. Density of fluid = 956.1 $Kg/m^3$.

**OBSERVATION TABLE**

| Sr. No | Ball Dia (m) | Ball Density ($kg/m^3$) | Ball Reynolds number $N_{Re}$ | Terminal velocity, $V_p$, m/s | Viscosity μ, Kg/m-s | Standard Daviation |
|---|---|---|---|---|---|---|
| 1 | | | | | | |
| 2 | | | | | | |
| 3 | | | | | | |
| 4 | | | | | | |
| 5 | | | | | | |
| 6 | | | | | | |
| Average | | | | | | |

**Calculation Table:**

| S NO. | Distance (m) | Time (Sec.) |
|---|---|---|
| 1 | | |
| 2 | | |
| 3 | | |
| 4 | | |
| 5 | | |
| Average | | |

**CONCLUSION/DISCUSSION ON THE RESULT:**

1. Write down the observations.
2. Try to explain the results from theory studied earlier.

**FURTHER READING**

1. Introduction to Fluid Mechanics 8th Edition, by *Fox, Robert W. and McDonald*, Alan T. , Chapter 1, PROBLEM NO. 1.2

**TEACHING ASSISTANT:**

## 2. FRICTION FACTOR IN INTERNAL PIPE TURBULENT FLOW

### AIM

The objective of this experiment is to determine the frictional losses in straight pipes.

### EXPERIMENTAL SETUP

Four pipes of different diameters and different material, control valves, sump tank U-tube manometers.

### THEORY

When the fluid flows through a pipe, it is subjected to resistance due to shear forces between fluid &wall and also between fluid layers. This resistance is called as ‘frictional resistance’. This resistance depends on various factors such as fluid properties, velocity, and wall roughness factor.

### PROCEDURE

Fill the sump tank with sufficient clean water. Open the outlet valve of pump and start the pump. Open the outlet valve of pipe to be tested. Remove all the air bubbles from manometer & connecting pipe. Adjust the flow such that the reading of the pressure transmitter is stable. Note down the pressure drop and the flow rate. Now increase the flow (operate outlet valve also so that there is no overflow) and take readings. Repeat the same procedure for other pipes.

### SPECIFICATIONS

Four pipes:

1) S.S. pipe with I.D. = 3.5mm
2) S.S. pipe with I.D. = 6.9 mm
3) S.S. pipe with I.D. = 9.6mm
4) S.S. pipe with I.D. = 16.7mm

Test length of pipe $L$ = 1 m

Roughness of SS pipe = 0.015 mm

### SPECIMEN CACULATIONS

**1.** Discharge: $Q = \frac{0.001}{t}$ $m^3$/sec

**2.** Velocity of flow: $V = \frac{Q}{A}$ m/sec

Where, Area, $A = \frac{\pi}{4} \times D^2$ m$^2$ ($D$ = inside diameter of pipe.)

**3.** According to Darcy-Weisbach equation, $h_f = \frac{fLV^2}{2gD}$

Where, $f$ = friction factor

Then,

$$f = \frac{2gDh_f}{LV^2}$$

4. According to Colebrook correlation $\frac{1}{\sqrt{f}} = -2.0\ log_{10}\left(\frac{\varepsilon/_D}{3.7} + \frac{2.51}{Re\sqrt{f}}\right)$

**OBSERVATION TABLE**

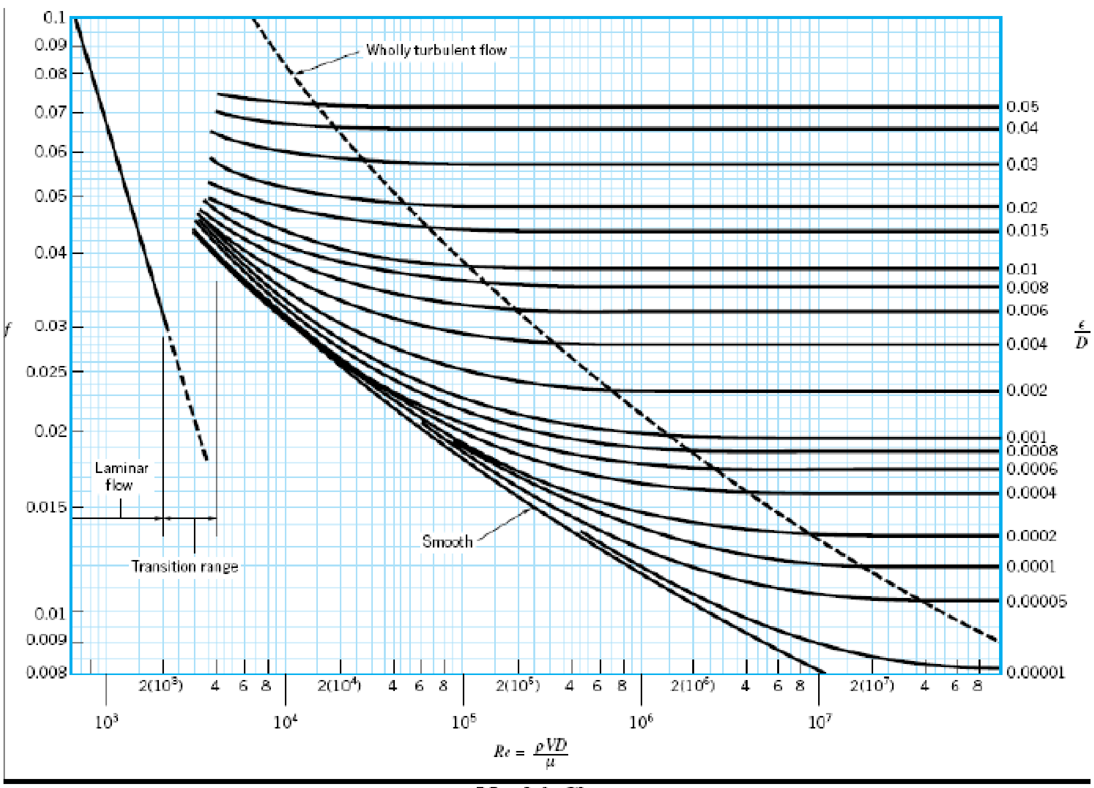
| Pipe | Sr. No. | Pressure drop (Pa) | Discharge ‘*Q*’ ($m^3$/sec) | Velocity ‘*V*’ (m/s) | *f*(expmt) | *Re* | *f* (Cole brook relation) | *f*(Moody’s chart) |
|---|---|---|---|---|---|---|---|---|
| 3.5 mm SS pipe | 1 | | | | | | | |
| | 2 | | | | | | | |
| | 3 | | | | | | | |
| | 4 | | | | | | | |
| | 5 | | | | | | | |
| 6.5 mm SS pipe | 1 | | | | | | | |
| | 2 | | | | | | | |
| | 3 | | | | | | | |
| | 4 | | | | | | | |
| | 5 | | | | | | | |
| 9.5 mm SS pipe | 1 | | | | | | | |
| | 2 | | | | | | | |
| | 3 | | | | | | | |
| | 4 | | | | | | | |
| | 5 | | | | | | | |
| 16.5 mm SS pipe | 1 | | | | | | | |
| | 2 | | | | | | | |
| | 3 | | | | | | | |
| | 4 | | | | | | | |
| | 5 | | | | | | | |

**GRAPHS:**

1. Plot friction factor, *f* (expt) vs Reynolds number, on a log-log graph.

**CONCLUSION/DISCUSSION ON THE RESULT:**

3. What is the roughness of the pipe?
4. Write down the observations.
5. Try to explain the results from theory studied earlier.




**Moody's Chart**

**FURTHER READING**

1. Introduction to Fluid Mechanics 8th Edition, by *Fox, Robert W. and McDonald*, Alan T., Chapter 8

**TEACHING ASSISTANT:**

# Insert Log-Log Graph Paper

## 3. HEAT TRANSFER STUDIES IN LAMINAR FLOW

### AIM

To determine overall heat transfer coefficient making use of transferred heat and logarithmic mean temperature difference. From overall heat transfer coefficient, determine the individual film heat transfer coefficient and verify the Sieder-Tate equation for laminar flow.

### APPARATUS

1. Stainless steel double pipe heat exchanger with facility to measure inlet and the outlet temperature of hot fluid by electronic thermometers of $0.1^0$C accuracy.
2. A stainless steel insulated tank with a heater, bottom discharge and fluid charging line at the top.
3. Hot and cold fluid circulation pump with speed variation mechanism.
4. A rotameter to measure the flow rate.

### PROCEDURE

1. Connect 15 amp. And 5 amp. plug pins to stable 230 V A.C. electric supply. Care should be taken to connect these two pins in different phases of the power supply.
2. Check the set point of the controller. The set point should be set around 60 to 80°C.
3. Connect the suction line of cold fluid circulation pump to cold water supply line.
4. We will keep cold flow rate constant and minimum as 100 LPH. It is essential for rise in temperature at least 2-4°C. The lesser we go (<100 LPH) fluctuation in flow rate is noticed. Similarly, for higher cold flow rate, it becomes difficult to reasonable temperature difference
5. As the temperature starts reaching the set value, the flow rate of hot fluid increases. Thus, it is better to wait till the temperature is close to set value.
6. Adjust the flow rate of hot fluid (known through rotameter reading) through the heat exchanger by adjusting the speed of hot fluid circulation pump. (The minimum flow rate of hot fluid should be at least being 300 LPH).
7. Note down the inlet and outlet temperatures indicated by digital thermometer on the control panel after steady state is reached. Also note down the inlet and outlet temperatures of cooling water.
8. Repeat step 7 for at least three different flow rates of hot fluid at particular set temperature.
9. Repeat the experiment again at three different set temperature

### THEORY

In a heat exchanger, heat is transferred from hot fluid to cold fluid through the metal wall. Heat transferred through the metal wall is always by conduction while on the both side of metal by convection. Generally, resistance offered to heat transfer by metal wall is negligible as compared to resistance offered by convection. The wall temperature should always be between local temperatures of the two fluids.

At low Reynolds number (Re<2100) the flow pattern is laminar and the fluid flows in an ordered manner along generally parallel “Filament like” streams which do not mix. It followsthat in this type of flow that the heat transferred to and through is essentially by convection.

When heat transferred through resistance in series, the total resistance to heat transfer is the sum of individual resistance in series. The overall heat transfer resistance in a heat exchanger can write,

$$\left(\frac{1}{U_iA_i}\right) = \left(\frac{1}{h_iA_i}\right) + \left(\frac{\Delta x}{KA_{lm}}\right) + \left(\frac{1}{h_oA_o}\right) \qquad (1)$$

OR

$$\left(\frac{1}{U_i}\right) = \left(\frac{1}{h_i}\right) + \left(\frac{\Delta x A_i}{KA_{lm}}\right) + \left(\frac{A_i}{h_oA_o}\right) \ldots\ldots\ldots\ldots \qquad (2)$$

In the above equation $h_i$is the heat transfer coefficient of hot fluid flowing through the inner tube. Since the test fluid is highly viscous and has very low thermal conductivity the inside heat transfer coefficient is expected to be very low and hence it will become controlling resistance for heat transfer. Thus, even if overall heat transfer coefficient is considered equal to inside heat transfer coefficient it will not be much in error. If flow through inner tube is in laminar flow regime $h_i$ can be predicted from Sieder-Tate equation given below.

$$\mathbf{Nu = 1.86Re^{\frac{1}{3}}Pr^{\frac{1}{3}}\left(\frac{D}{L}\right)^{\frac{1}{3}}} \qquad (3)$$

$$\mathbf{Nu = hd/K} \qquad (4)$$

On simplifying equation, we will get

$$\mathbf{h_i = C * m^{\frac{1}{3}}} \ldots\ldots\ldots\ldots. \qquad (5)$$

Where C is constant and m is mass flow rate of hot fluid.

$$\mathbf{C} = \frac{1.86K^{\frac{2}{3}}C_p{}^{\frac{1}{3}}}{(ADL)^{\frac{1}{3}}} \qquad (6)$$

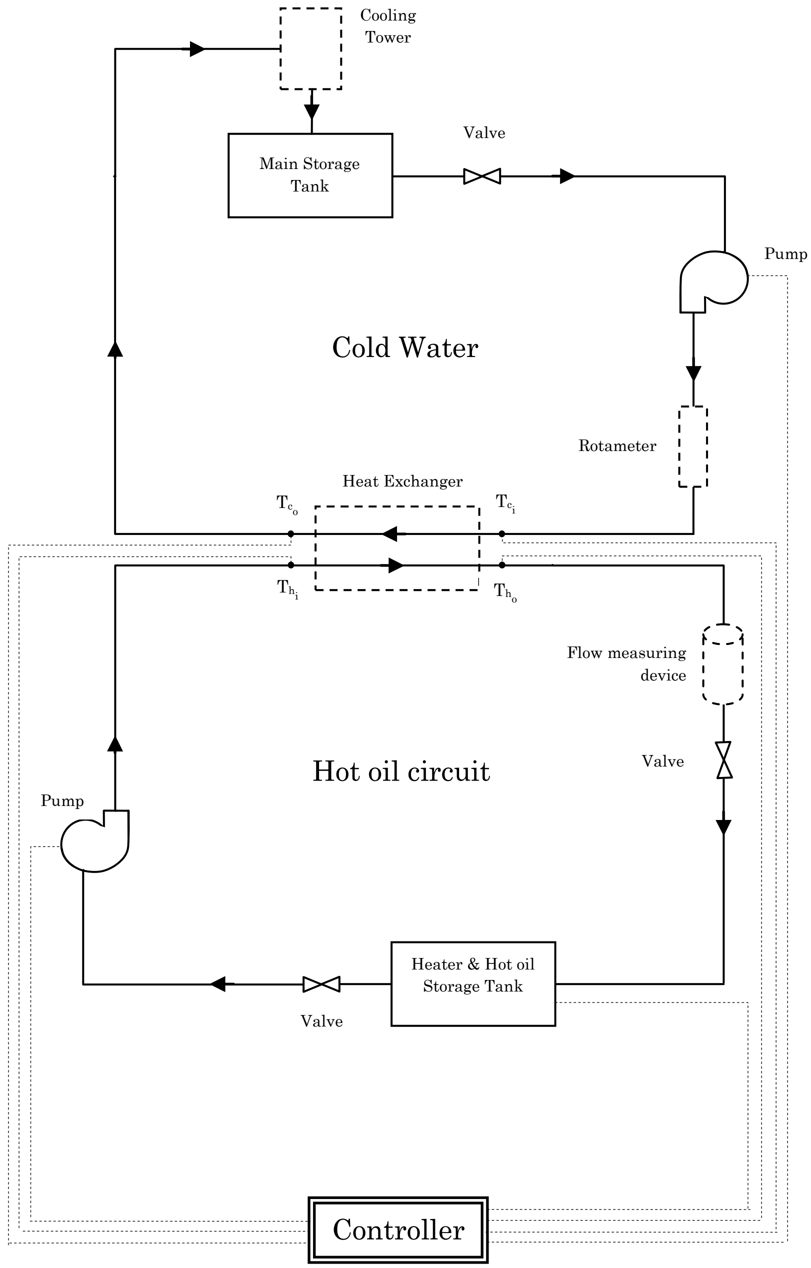
Taking log both sides for equation 5

$$\log(h_i) = \log(C) + \left(\frac{1}{3}\right)\log(m) \qquad (7)$$

**OBSERVATIONS**

1. Outside diameter of inner tube($d_2$)= 0.01 m
2. Inside diameter of inner tube($d_i$)=0.007 m
3. Length of heat exchanger (L)= 0.8 m
4. Specific heat of hot fluid= 0.625 Kcal/kg $^0$C = 2616.75 J/Kg.K
5. Thermal conductivity of the hot fluid= 0.13 W/mK
6. Kinematic Viscosity ($\vartheta$)
   At $100^0$C= 5.3 * $10^{-6}$ $m^2$/sec
   At $50^0$C =20.8 * $10^{-6}$ $m^2$/sec
   At $40^0$C =31* $10^{-6}$ $m^2$/sec
   At $90^0$C= 9.583 * $10^{-6}$ $m^2$/sec
   At $80^0$C =13.866 * $10^{-6}$ $m^2$/sec
   At $70^0$C =18.15 * $10^{-6}$ $m^2$/sec
   At $60^0$C =22.43 * $10^{-6}$ $m^2$/sec
   To find values Kinematic Viscosity for other than temperatures use linear interpolation or any existing correlation
   Dynamic viscosity = $\mu = \vartheta * \rho$ (Kg/m.s)
7. Specific Density= 0.857 g/$cm^3$ = 857 kg/$m^3$
8. Diameter of the measuring cylinder=0.0762 m

**SCHEMATIC FLOWSHEET**




Heat transfer in laminar flow

**OBSERVATION TABLE**

| Obs. No. | Set Temp ($^0$C) | hot fluid temperature ($^0$C) | | cold fluid temperature ($^0$C) | | Height (m) | Time (s) |
|---|---|---|---|---|---|---|---|
| | | inlet($T_1$) | outlet($T_2$) | inlet($t_1$) | outlet($t_2$) | | |
| 1 | 40 | | | | | | |
| 2 | 40 | | | | | | |
| 3 | 40 | | | | | | |
| 4 | 55 | | | | | | |
| 5 | 55 | | | | | | |
| 6 | 55 | | | | | | |
| 7 | 70 | | | | | | |
| 8 | 70 | | | | | | |
| 9 | 70 | | | | | | |

**CALCULATION**

1. Cross section area of inner tube S = $\prod d_i^2/4$= ____________ $m^2$

2. Inside heat transfer area of the heat exchanger = A=$\prod$DL = $m^2$

3. Prandtl number at hot fluid mean temperature:
   $Pr = \frac{C_p \mu}{K}$ _____________________

**SAMPLE CALCULATION FOR READING NO**___________:

1. Velocity of hot fluid
   $u = \frac{Height}{time}$=_______________________m/sec
2. Volumetric flow rate of hot fluid:
   V = velocity (u) *Area (S)_________________ $m^3$/sec
3. Mass flow rate (m) = V * $\rho$= _________________ Kg/sec
4. Heat Transferred per hour:
   $Q = (V \times \rho \times 3.6) \times C_p \times (T_1 - T_2)$ = ________________ Kcal/hr
   Q = m Cp ΔT = ( V * $\rho$) * $C_p \times (T_1 - T_2)$ = ________________ W

5. $LMTD = \frac{[(T_1 - t_1) - (T_2 - t_2)]}{\left[ln\left[\frac{(T_1 - t_1)}{(T_2 - t_2)}\right]\right]}$ = ____________ K

6. Overall heat transfer coefficient
$U = \frac{Q}{(A \times LMTD)}$ = ________________W/m$^2$ K

7. Inside heat transfer coefficient $h_i$
$h_i = \frac{1.86 K^{\frac{2}{3}} C_p^{\frac{1}{3}}}{(ADL)^{\frac{1}{3}}} * m^{\frac{1}{3}}$ = ________________________ W/m$^2$ K

8. Nusselt number:
$Nu = \frac{h_i d_i}{K}$ = _____________________

9. Reynolds number:
$Re = \frac{u d_i}{\vartheta}$ = __________________

**RESULT TABLE 1**

| Obs. No. | Volumetric flow rate of hot fluid ($m^3$/sec) | Amount of heat transferred Q (W) | Velocity of hot fluid u (m/sec) | LMTD | Overall heat transfer coefficient U (W/m$^2$ K) |
|---|---|---|---|---|---|
| 1 | | | | | |
| 2 | | | | | |
| 3 | | | | | |
| 4 | | | | | |
| 5 | | | | | |
| 6 | | | | | |
| 7 | | | | | |
| 8 | | | | | |
| 9 | | | | | |

**RESULT TABLE 2**

| Obs. No. | Inside heat transfer coefficient $h_i$ ($W/m^2 K$) | Nusselt number Nu | Reynolds number Re |
|---|---|---|---|
| 1 | | | |
| 2 | | | |
| 3 | | | |
| 4 | | | |
| 5 | | | |
| 6 | | | |
| 7 | | | |
| 8 | | | |
| 9 | | | |

**GRAPHS (Total 4)**

1. Plot the graph of log ($1/U_i$) vs. log (1/u) on linear scale.
2. Plot the graph of log(Nu) vs. log(Re) on linear scale.

**RESULTS**

**CONCLUSION/DISCUSSION ON THE RESULT**

1. Write down the observations.

2. Try to explain the results from theory studied earlier.

**FURTHER READING**

Fundamental of Heat and Mass Transfer by Frank P. Incropera and David P. Dewitt, Chapter 8.

# Insert 2 graph papers

# 4. HEAT TRANSFER STUDIES IN TURBULENT FLOW

## AIM

To determine the overall heat transfer coefficient making use of logarithmic mean temperature difference. From overall heat transfer coefficient, determine the individual film heat transfer coefficients and verify the Turbulent flow equation for heat transfer.

## APPARATUS

- Stainless steel double pipe heat exchanger with facility to measure inlet and outlet temperatures of hot fluid with electronic thermometers of accuracy 0.1°C. The inlet and outlet temperatures of cold fluid are also measured with electronic thermometers of accuracy of 0.1°C.
- A stainless steel insulated tank with a heater, bottom discharge and fluid charging line at the top. It is also provided with temperature indicator cum controller to control the hot fluid temperature.
- Hot fluid circulation pump with speed variation mechanism. Hot fluid circulation line has a rotameter to measure flow rate of hot fluid.
- Cold fluid circulation pump, with speed variation mechanism. Cold fluid circulation line has a rotameter to measure flow rate of cold fluid.

## PROCEDURE

10. Connect 15 amp. and 5 amp. plug pins to stable 230 V A.C. electric supply. Care should be taken to connect these two pins in different phases of the power supply.
11. Switch on the dual temperature indicator cum controller. Check the set point of the controller. The set point should be set around 60 to 80°C.
12. Ensure that the valve at the bottom of measuring tank is open. Open the valve on the outlet line of the hot fluid tank. Switch on the power supply to hot fluid circulation pump and slowly increase the speed of the pump by regulating the voltage supplied to it. Initially run the pump at slow speed. Check the inlet and outlet temperatures of the fluid indicated by digital thermometer. Note down the temperature difference between inlet and outlet temperatures, which gives zero error (Digital thermometers can give errors up to 1°C which is generally very difficult to bring down). After noting down the zero error in the digital thermometer, switch on all the (three) heaters of the hot fluid tank by switching on their respective main switches.
13. Connect the suction line of cold fluid circulation pump to cold water supply line.
14. We will keep cold flow rate constant and minimum as 100 LPH. It is essential for rise in temperature at least 2-4°C. The lesser we go (<100 LPH) fluctuation in flow rate is noticed. Similarly, for higher cold flow rate, it becomes difficult to reasonable temperature difference
15. As the temperature starts reaching the set value, the flow rate of hot fluid increases. Thus, it is better to wait till the temperature is close to set value.

16. Adjust the flow rate of hot fluid (known through rotameter reading) through the heat exchanger by adjusting the speed of hot fluid circulation pump. (The minimum flow rate of hot fluid should be at least being 300 LPH).
17. Note down the inlet and outlet temperatures indicated by digital thermometer on the control panel after steady state is reached. Also note down the inlet and outlet temperatures of cooling water.
18. Repeat step 7 for at least three different flow rates of hot fluid at particular set temperature.
19. Repeat the experiment again at three different set temperature

**THEORY**

In a heat exchanger, heat is transferred from hot fluid to cold fluid through metal wall which generally separates those two fluids. Heat transfer through metal wall is always by conduction while on both sides of metal wall it is generally by convection. Generally, resistance offered to heat transfer by the metal wall is negligible as compared to resistance offered by convection. The wall temperature is always between local temperatures of the two fluids. The actual value depends upon individual film heat transfer coefficient on either side.

At higher Reynolds number, the ordered flow pattern of laminar flow regime is placed by randomly moving eddies thoroughly mixing the fluid and greatly assisting heat transfer. However, this enhancement of film heat transfer coefficient is accompanied by much higher pressure drop which demands higher pumping power. Thus, although desirable, turbulent flow is usually restricted to fluids of low viscosity.

When heat is transferred through resistances in series, the total resistance to heat transfer is the sum of individual resistances in series. Thus, for heat exchanger, one can write,

$$\frac{1}{U_i \cdot A_i} = \frac{1}{h_i \cdot A_i} + \frac{\Delta X}{K \cdot A_{lm}} + \frac{1}{h_o \cdot A_o} \qquad (1)$$

OR

$$\frac{1}{U_i} = \frac{1}{h_i} + \frac{\Delta X \cdot A_i}{K \cdot A_{lm}} + \frac{A_i}{h_o \cdot A_o} \qquad (2)$$

Once the heat exchanger material and its geometry are fixed, then the metal wall resistance $\left[\frac{\Delta X}{K \cdot A_{lm}}\right]$ becomes constant. Similarly, if the flow rate of cold fluid is fixed and its mean temperature does not differ much for different flow rates of hot fluid, then the resistance by the outside film will remain almost constant. Thus, the overall heat transfer coefficient will depend upon the value of inside film heat transfer coefficient alone. If flow through inner tube is in the turbulent flow regime and Pr (0.6-100), then Dittus-Boelter equation can be used to find out inside film heat transfer coefficient.

$Nu=0.023*Re^{0.8}*Pr^{n}$ (3)

If Pr>100, we have to use the following turbulent flow equation

$$Nu = 3.66 + \frac{0.0668(\frac{D}{L}) \times Re \times Pr}{1 + 0.04(\frac{D}{L}) \times Re \times Pr^{\frac{2}{3}}}$$

If the bulk mean temperature does not differ much for different flow rates, then all the physical properties will remain nearly the same and equation(3) can be re-written as:

$Nu=constant*v^{0.8}$ (4)

Substituting equation (4) in equation (2), one can write it as:

$\frac{1}{\text{Ui}} = \frac{\text{constant 1}}{V^{0.8}} + \text{constant2}$ (5)

Thus, the graph of $\frac{1}{\text{Ui}}$ vs $\frac{1}{V^{0.8}}$(which is known as Wilson plot) should be a straight line with a slope equal to constant1 and intercept equal to constant2. From this graph, inside film heat transfer coefficient can be calculated which can be used to verify Dittus-Boelter or any turbulent flow equation.

**OBSERVATIONS**

1. Outside diameter of inner tube($d_2$)= 0.01 m
2. Inside diameter of inner tube($d_i$)=0.007 m
3. Length of heat exchanger (L)= 1 m
4. Specific heat of hot fluid= 0.625 Kcal/kg $^0$C= 2616.75 J/Kg.K
5. Thermal conductivity of the hot fluid= 0.13 W/mK
6. Kinematic Viscosity ($\vartheta$)

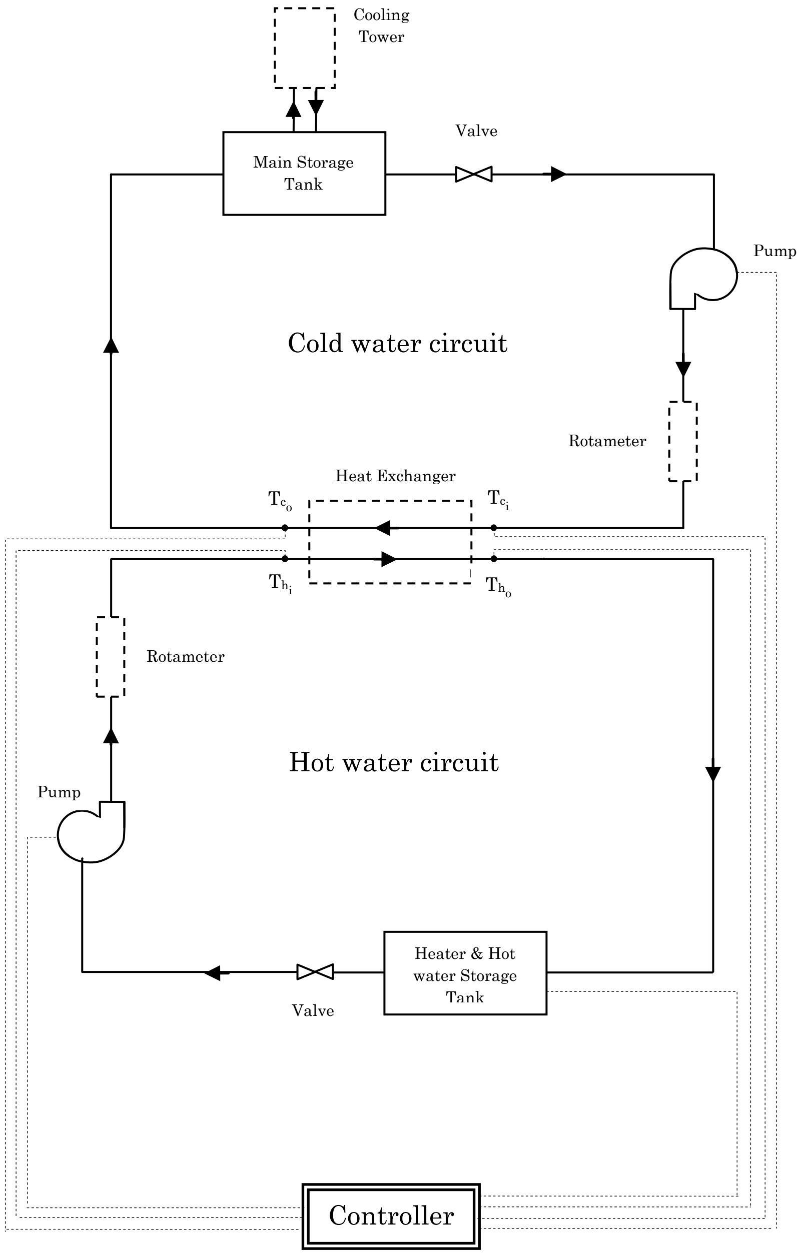
   At 100$^0$C= 5.3 * $10^{-6}$ $m^2$/sec
   At 50$^0$C =20.8 * $10^{-6}$ $m^2$/sec
   At 40$^0$C =31* $10^{-6}$ $m^2$/sec
   At 90$^0$C= 9.583 * $10^{-6}$ $m^2$/sec
   At 80$^0$C =13.866 * $10^{-6}$ $m^2$/sec
   At 70$^0$C =18.15 * $10^{-6}$ $m^2$/sec

At $60^0C = 22.43 * 10^{-6}\ m^2/sec$

To find values Kinematic Viscosity for other than temperatures use linear interpolation or any existing correlation (should be mention in journal).

7. Dynamic viscosity = $\mu = \vartheta * \rho$(Kg/m-s)
8. Specific Density= 0.857 $g/cm^3$ = 857 $kg/m^3$
9. heat transfer area of heat exchanger (A) = 0.022 $m^2$

## SCHEMATIC FLOWSHEET




**Heat transfer in turbulent flow**

**OBSERVATION TABLE**

| Obs. No. | Set Temp (°C) | Hot fluid temperature (°C) | | Cold fluid temperature (°C) | | Oil flow rate (LPH) |
|---|---|---|---|---|---|---|
| | | Inlet ($T_1$) | Outlet ($T_2$) | Inlet ($t_1$) | Outlet ($t_2$) | |
| 1 | 60 | | | | | |
| 2 | 60 | | | | | |
| 3 | 60 | | | | | |
| 4 | 70 | | | | | |
| 5 | 70 | | | | | |
| 6 | 70 | | | | | |
| 7 | 80 | | | | | |
| 8 | 80 | | | | | |
| 9 | 80 | | | | | |

**A: TABLE OF CALCULATED RESULTS**

| Obs. No. | Volumetric flow rate of hot fluid (V)$m^3$/sec | Amount of heat transferred (Q)W | Velocity of hot fluid (u) m/sec | LMTD ($\Delta T_{lm}$) (K) | Overall heat transfer coefficient (U)W/$m^2$ K |
|---|---|---|---|---|---|
| 1 | | | | | |
| 2 | | | | | |
| 3 | | | | | |

| | | | | | |
|---|---|---|---|---|---|
| 4 | | | | | |
| 5 | | | | | |
| 6 | | | | | |
| 7 | | | | | |
| 8 | | | | | |
| 9 | | | | | |

**B: TABLE OF CALCULATED RESULTS**

| Obs. No. | $(1/v^{0.8})$ | (1/U) | Inside film heat transfer coefficient $h_i$ | Nusselt number Nu | Reynolds number Re |
|---|---|---|---|---|---|
| 1 | | | | | |
| 2 | | | | | |
| 3 | | | | | |
| 4 | | | | | |
| 5 | | | | | |
| 6 | | | | | |
| 7 | | | | | |
| 8 | | | | | |
| 9 | | | | | |

**CALCULATION**

1.Cross section area of inner tube S = $\prod d_i^2/4$= ____________ $m^2$

2. Inside heat transfer area of the heat exchanger = A=$\prod$DL = $m^2$

3. Prandtl number at hot fluid mean temperature:

$Pr = \frac{C_p \mu}{K}$ ____________________

4. Volumetric flow rate of hot fluid (V)__________________$m^3$/sec

5. Velocity of hot fluid

velocity (v) = Volumetric flow rate(V) / Area (S)_________________ m/sec

6. Mass flow rate (m) = V * $\rho$ = _________________ Kg/sec

7. Heat Transferred per hour:

Qhot = m Cp$\Delta$T = ( V * $\rho$) * $C_p \times (T_1 - T_2)$ = ________________ W

Qcold = m Cp$\Delta$T = ( V * $\rho$) * $C_p \times (t_2 - t_1)$ = ________________ W

8. $LMTD = \frac{[(T_1-t_1)-(T_2-t_2)]}{\left[ln\left[\frac{(T_1-t_1)}{(T_2-t_2)}\right]\right]}$ = _____________ K

9. Overall heat transfer coefficient

$U = \frac{Q}{(A \times LMTD)}$ = _________________W/$m^2$ K (Use Qhot or Qcold value whichever is low)

10. Nusselt number: Nu=0.023*$(Re)^{0.8}$*$(Pr)^{0.3}$ (0.6<Pr>100)
Nusselt number

$Nu = 3.66 + \frac{0.0668\,(\frac{D}{L}) \times Re \times Pr}{1+0.04(\frac{D}{L}) \times Re \times Pr^{\frac{2}{3}}}$ (Pr>100)

- Theoretical value of inside film heat transfer coefficient

$$h_i = \frac{Nu.K}{d_1}$$

- Experimental value of inside film heat transfer coefficient

1/ho =((1/U)-(1/hi)-((ln(r0/ri)*Ai)/(2pi*L*K)))*(Ai/A0)

- $h_o$=(1/intercept) where the intercept is of the graph of (1/U) vs (1/$v^{0.8}$)

11. Reynolds number:
$Re = \frac{ud_i}{\vartheta}$ = ___________________

**GRAPHS (Total 3)**

1. Plot the graph of $\frac{1}{\text{Ui}}$ vs. $\frac{1}{V^{0.8}}$.
2. Plot the graph of $\log\left(\frac{1}{\text{Ui}}\right)$ vs. $\log\left(\frac{1}{v}\right)$ on linear scale
3. Plot the graph of log(Nu) vs. log (Re) on linear scale.

**CONCLUSION/DISCUSSION ON THE RESULT**

From calculation, it was observed that inside film heat transfer coefficient was less than overall heat transfer coefficient in most of the reading. The difference between overall and individual heat transfer coefficient is very less which is not significant.

**FURTHER READING**

Fundamental of Heat and Mass Transfer by Frank P. Incropera and David P. Dewitt, Chapter 8.

**TEACHING ASSISTANT**

# Insert Graph Papers

## 5. FINNED TUBE HEAT EXCHANGER

**AIM:**

To determine the efficiency of given longitudinal fin and compare it with the theoretical value for the given fin

**APPARATUS:**

1. Longitudinal finned tube heat exchanger.
2. Bare pipe (pipe without fins).
3. Steam generator to generate steam at constant pressure. The steam generator is also provided with temperature indicator cum controller and a dead weight safety valve.

**THEORY**

In a heat exchanger, the two fluids namely; hot and cold, are separated by a metal wall. Under this condition the rate of heat transfer will depend on the overall resistance to heat

$$\frac{1}{U_iA_i} = \frac{1}{h_iA_i} + \frac{x}{KA_{lm}} + \frac{1}{h_oA_o}$$

Where

$U_i$ =Overall heat transfer coefficient based on inner area [kcal/h-m$^2$-$^o$C]

$U_o$ =Overall heat transfer coefficient based on outer area [kcal/h-m$^2$-$^o$C]

$h_i$,$h_o$ =Inside and outside film heat transfer coefficient [kcal/h-m$^2$-$^o$C]

$A_i$,$A_o$ =inside and outside surface area [m$^2$]

When viscous liquids are heated in a double pipe heat exchanger or any standard tubular heat exchanger by condensing steam or hot fluid of low viscosity, the film heat transfer coefficient of the viscous liquid will be much smaller than that on the hot fluid side and will therefore, become controlling resistance for heat transfer. This condition is also present in case of air or gas heaters where the gas side film heat transfer coefficient will be very low (typically of the order of 0.01 to 0.002 times) compared to that for the liquid or condensing vapor on the other side. Since, the heat transfer coefficients of viscous fluid or gas cannot be improved much, the only alternative is to increase the area available for heat transfer on that side so that its resistance to heat transfer can be reduced. To conserve space and to reduce the cost of equipment in these cases, certain type of heat exchange surfaces, called extended surfaces, have been developed in which outside area of tube is increased many fold by fins and other appendages.

Two types of fins, are in common use viz; longitudinal fins and transverse fins. Longitudinal fins are used when the direction of flow of the fluid is parallel to the axis of tube and transverse fins are used when the direction of the flow of the fluid is across the tube. Spikes, pins, studs or spines are also used for either direction of flow.

The outside area of a finned tube consists of two parts: the area of fins and the area of bare tube not covered by the bases of fins. A unit area of fin surface is not as efficient as a unit area of bare tube surface because of the added resistance to the heat flow by conduction through the fin at its base. The expression for fin efficiencies can be derived by solving the general differential equation of heat conduction with suitable boundary conditions. *Generally three boundary conditions are used;*

1. **Fin of infinite length:** so that there is no heat dissipation from its tip, or in other words temperature at the tip of fin is same as that of the surrounding fluid.

2. **Insulated tip:** This condition even though cannot be realized in practice, but considering that the tip area is negligible as compared to the total fin area, heat dissipated from tip can be neglected and hence, [dT/dx] is assumed to be zero at the tip.

3. **Finite heat dissipation from the tip**

Even though the assumption of insulated tip is invalid, most of the fins are treated under this category, and longitudinal fine efficiency for this case is given by the expression:

$$\eta_{fin} = \frac{\tanh(mL)}{mL},$$

Where,

$$m = \sqrt{\frac{hC}{KA}}$$

h = film heat transfer coefficients from the fin surface [kcal/h-m$^2$-$^o$C]

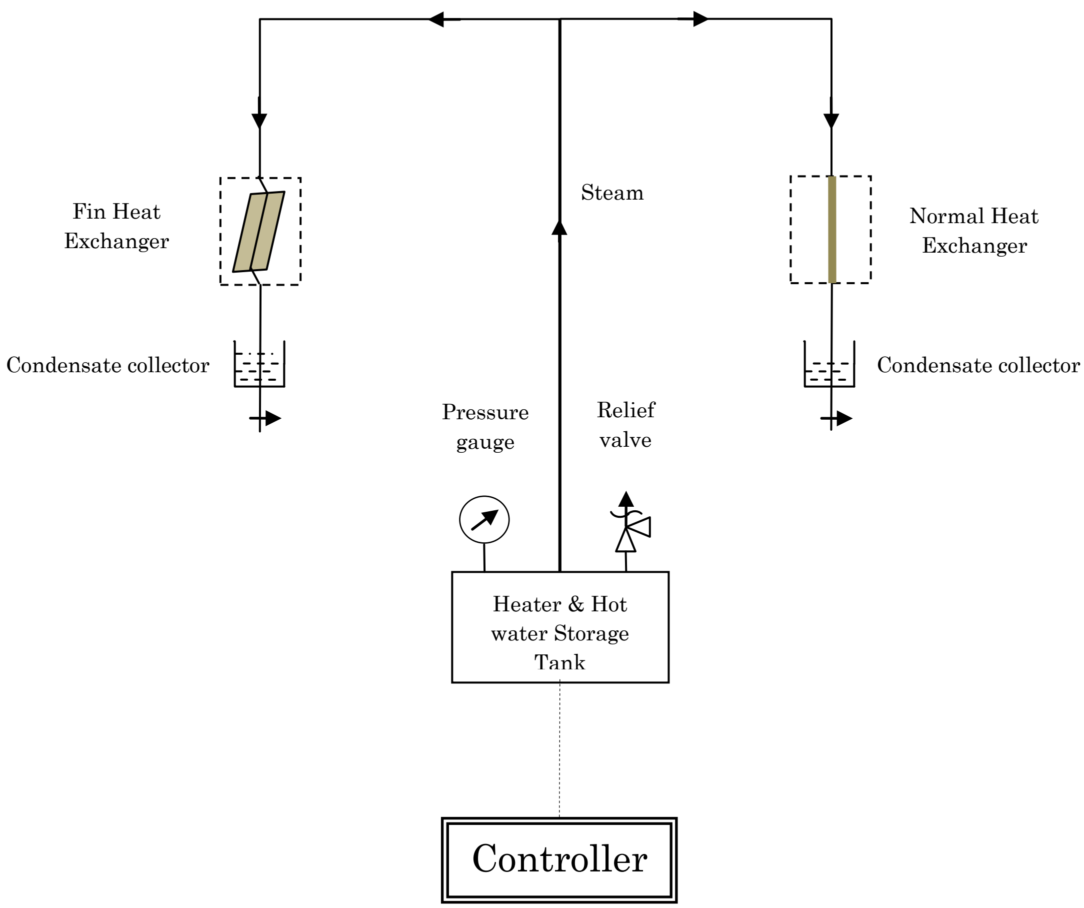
C = Circumference of the fin [m]

K = thermal conductivity of fin material [kcal/h-m$^2$-$^o$C]

A = cross sectional area of fin [m$^2$]

From the above equation, it can be seen that the fin efficiency is a function of mL, and as the value of mL increases, the fin efficiency decreases. A reasonable value of fin efficiency will be around 50 to 75% for which mL should have a value between 1 and 2. If the fin height L should be sufficient (of the order of 5 to 8 cm), then it can be seen that the value of h should be around 10 to 20 which can be given by air in natural convection. The value of film heat transfer coefficients for any other liquid in natural convection, or any gas in forced convection will be much higher than 20. Thus, the given set-up is used for heat transfer to air in natural convection region.

**SCHEMATIC FLOWSHEET**




**Fin tube heat exchanger**

**PROCEDURE:**

1. Important instructions: Follow instructions 2 and 3 without fail, otherwise electrical heater will burn out.

2. Open the drain valve provided at the bottom of steam generator and drain out the water from steam generator completely.

3. Close the drain valve and charge requisite amount of water through charging valve provided at the top of the steam generator and close it. Ensure that the dead weight safety valve is free.

4. Start the electrical heater of steam generator. Set the desired temperature on the temperature controller and start heating water in the steam generator. Steam will start forming within about 15-20 min. of switching on the heater. During this period, keep the condensate collector valve open. Once the steam generation starts, the finned tube heat exchanger as well as the bare pipe will start getting heated up and condensate will start coming out of the needle valve provided at the bottom of condensate collector. When the test section is fully heated up, steam will start coming out of the needle valve. Now regulate the needle valve in such a way that only condensate comes out of it. The pressure can be regulated between 0-1 atm. gauges as per the requirement.

5. Once the test section (finned tube heat exchanger along with bare pipe without fins) is fully heated, drain out completely the condensate, if any. Close the needle valve on condensate drain line simultaneously starting the stop-watch. Collect the condensate accumulated at an interval of 20 min. for both heat exchangers. If the quantity of condensate collected is same for 2 to 3consecutive readings (with in experimental accuracy), note down the volume of condensate collected and time interval.

**OBSERVATIONS:**

Finned Tube:

1. Height of fin (L) : 0.05 m
2. Width of fin (W) : 0.5 m
3. Thickness of fin (b) : 0.003 m
4. Number of fins (N) : 4
5. O.D. of fin tube (D) : 0.025 m.
6. Thermal conductivity of fin material (K) : 15.0 kcal/h-m-°C

Bare Tube:

1. Length of tube (l) : 0.6 m
2. O.D. of tube (d) : 0.025 m
3. $T_{ambient}$ : °C

**OBSERVATION TABLE**

| Obs. No. | Finned tube heat exchanger | | Bare tube heat exchanger | | $T_{steam}$ |
|---|---|---|---|---|---|
| | Amount of condensate collected,$m_1$(ml) | Time for collection of condensate,t(min) | Amount of condensate collected,$m_1$(ml) | Time for collection of condensate(min) | |
| 1 | | | | | |
| 2 | | | | | |
| 3 | | | | | |
| 4 | | | | | |
| 5 | | | | | |

**CALCULATIONS:**

1. Circumference of fin (C):

$$C = 2x(W+b) = \_\_\_\_\_\_\_\_\_\_ \ m$$

2. Cross-sectional area of fin (A):

$$A = Wxb = \_\_\_\_\_\_\_\_\_\_ \ m^2$$

3. Fin area available for heat transfer:

$$A_F = CxLxN = \_\_\_\_\_\_\_\_\_\_\_ \ m^2$$

4. Tube area available for heat transfer in finned tube heat exchanger:

$$A_b = [(\pi \text{ x } D) - (N \text{ x } b)] \text{ x } W = \_\_\_\_\_\_\_\_\_\_\_\_ \ m^2$$

5. Total area of finned tube heat exchanger:

$$A_t = A_F + A_b = \_\_\_\_\_\_\_\_\_\_\_\_ \ m^2$$

**Sample calculation for reading no. ____**

6. Heat given out by steam through finned tube heat exchanger ($Q_1$):

$$Q_1 = (m_1/t)\lambda = \_\_\_\_\_\_\_\_\_\_\_\_ \ kcal/h$$

7. Heat given out by steam through bare tube ($Q_2$):

$$Q_2 = (m_2/t)\lambda = \_\_\_\_\_\_\_\_\_\_ \ kcal/h$$

Where, $\lambda$ = latent heat of vaporization of water at steam pressure = 540 kcal/kg

8. Area available for heat transfer of bare tube

$$A_{bare\ tube} = (\pi dl) = \_\_\_\_\_\_\_\_\_\_\_ \ m^2$$

9. Film heat transfer co-efficient from bare tube (h):

$$h = \frac{Q_2}{A_b \Delta T} = \_\_\_\_\_\_\_\_\_\_\_\_\_\_\_ \ kcal/h\text{-}m^2\text{-}°C$$

$$\Delta T = T_{steam} - T_{ambient} = \_\_\_\_\_\_\_\_\_\_\_ \ °C$$

10. $m = \sqrt{\frac{hC}{kA}} = \_\_\_\_\_\_\_\_\_\_\_\_ \ m^{-1}$

11. $mL = \_\_\_\_\_\_\_\_\_\_\_\_\_$

12. $\eta_{Fin}(\text{Theoretical}) = \frac{\tan h(mL)}{mL} = \_\_\_\_\_\_\_\_\_\_\_\_$

13. Amount of heat actually dissipated by the fin:

$$Q_{Fin} = Q_1 - (A_b h \Delta T) = \_\_\_\_\_\_\_\_\_\_\_\_ \ kcal/h$$

14. Amount of heat that can be dissipated by deal fin:

$$Q_{ideal} = A_F h \Delta T = \_\_\_\_\_\_\_\_\_\_\_\_ \ kcal/h$$

15. Observed value of fin efficiency:

$$\eta_{Fin}(\text{Observed}) = \frac{Q_{Actual\ \ fin}}{Q_{ideal\ \ fin}} = \_\_\_\_\_\_\_\_\_\_\_\_\_\_\_$$

**RESULT:**

| Obs. No. | Amount of heat lost | | Amount of heat lost | | Fin efficiency | |
|---|---|---|---|---|---|---|
| | Through fin tube, $Q_1$ (kcal/h) | Through bare tube, $Q_2$ (kcal/h) | Actual fin, $Q_{fin}$ (kcal/h) | Ideal fin, $Q_{ideal}$ (kcal/h) | Actual | Theoretical |
| 1 | | | | | | |
| 2 | | | | | | |
| 3 | | | | | | |
| 4 | | | | | | |
| 5 | | | | | | |

**CONCLUSIONS**

1. Write down the observations.

2. Try to explain the results from theory studied earlier.

**FURTHER READING**

Fundamentals of Heat and Mass Transfer by Frank P. Incropera and David P. Dewitt
Chapter 3.

**TEACHING ASSISTANT**

## 6. HEAT TRANSFER THROUGH A SUBMERGED HELICAL COIL IN AN AGITATED VESSEL STEADY STATE

### AIM

The objectives of the experiment are

(i) To determine coil side heat transfer coefficient as a function of agitator speed (R.P.M.)

(ii) To determine the inside heat transfer coefficient as a function of flow rate

### APPARATUS

An insulated cylindrical vessel fitted with an electrical heater, a cooling coil and a variable speed fractional horse power motor with given agitator for agitation of liquid in the vessel.

- Cold fluid circulations pump with speed variation mechanism. Cold fluid circulation pipe line contains a rotameter to measure the flow rate of the cold fluid.
- Digital temperature indicators to measure inlet and outlet temperatures of cooling water with accuracy of 0.1°C.
- Digital temperature indicator cum controller to measure as well as control temperature of liquid in the vessel in which cooling coil is immersed.

### THEORY

Coils afford one of the cheapest means of obtaining heat transfer surface in the reactors. The advantage of the coils is its surface can be adjusted as per requirement. Coils are usually made by rolling lengths of copper, stainless steel or alloy tubing into helix or double helix. Inlet and outlet connections for these coils are conveniently located side by side. Helical coils of either type are frequently installed in vertical cylindrical vessel with or without an agitator. Free space is generally provided between the coil and the vessel wall which makes the entire surface of the coil available for heat transfer. When such coils are used with mechanical agitation agitator, the vertical axis of the agitator usually coincides with the axis of the cylindrical vessel in which the coil is installed. Very limited data are available in literature for prediction heat transfer coefficient from submerged coil to the surrounded fluid in natural convection. However, the heat transfer coefficients in natural convection are undoubtedly lower. A mechanical agitation can improve the heat transfer coefficients in natural convection in agitated vessel and coil. Chilton, Drew and Jebems have published an excellent correlation on both jacketed vessel and coils under batch and steady state conditions with modified Reynolds number for mechanical agitation. Although much of work was carried out on vessel of one foot diameter, checks were also obtained on vessel five times larger than the experimental setup. The deviations on runs with water were highest for the fluids tested, which included lube oils and glycerols, and were in some instances off by 17.5%. Their correlation for heat transfer to fluids in the vessel with mechanical agitation heated or cooled by submerged coils is

$$\frac{h_c * d_c}{K} = 0.87 * \left({}^{L^2 * N * \rho}/_{\mu}\right)^{2/3} * \left({}^{C_p * \mu}/_{K}\right)^{\frac{1}{3}} * \left({}^{\mu}/_{\mu_W}\right)^{0.14} \qquad [1]$$

Where

$h_c$ = Heat transfer coefficient between fluid and coil surface $\frac{KW}{m^2-k}$

$d_c$= Coil diameter (m)

L= Agitator diameter (m)

N= Agitator speed $\frac{Rev}{sec}$

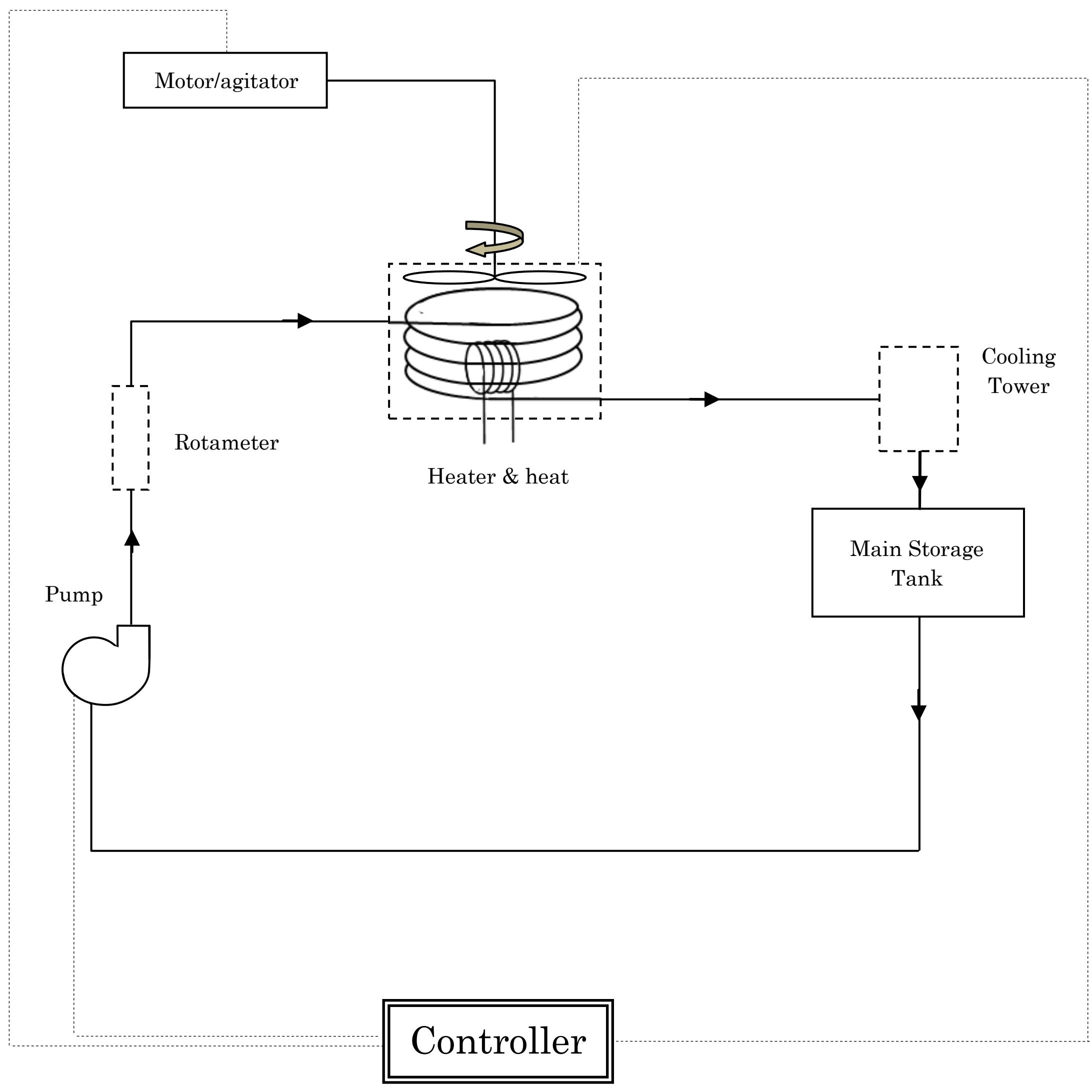
ρ= Density of fluid in the vessel ($\frac{Kg}{m^2}$)

K= Thermal conductivity of fluid in vessel ( $\frac{KW}{m^2-k}$)

**PROCEDURE:**

1) Fill the cylindrical vessel with given test liquid till the cooling coil is completely covered with the liquid.
2) Start the agitator motor and adjust the speed at desired level by making use of its speed regulator.
3) Connect the inlet of the cooling water circulation pump to cooling water supply and start the pump. Adjust the flow rate of cooling water using rotameter and speed regulator of the pump. Keep this flow rate constant throughout the entire experiment.
4) Switch on the heater in the agitated vessel and set the agitated vessel and set the desired temperature on the controller so as to keep temperature of liquid in the vessel at the constant level. Throughout the given set of readings keep this temperature constant.
5) Due to heating, the outlet temperature of water flowing through coil would rise. At steady state inlet and outlet temperature of cooling water and the temperature of the liquid in the vessel (outside the cooling coil) would attain a constant value. After steady state is attained note down inlet and outlet temperatures of cooling water as well as its flow rate. Also note down the temperature of liquid in the vessel.
6) Repeat step (5) for at least 4 different speeds of the agitator.
7) Repeat whole experiment for different flow rates also.

## SCHEMATIC FLOWSHEET




**Heat transfer in agitated vessel**

## OBSERVATIONS

| | | |
|---|---|---|
| 1) Length of the coil immersed in the agitated vessel (L) | 1.6 | m |
| 2) Inside diameter of the coil tube ($d_i$) | 9.45 | mm |
| 3) Outside diameter of the coil tube ($d_o$) | 12.7 | mm |

4) Outside area of coil available for heat transfer=($\pi$*$d_o$*L) = $mm^2$

5) Temperature of fluid in the vessel during the test run (T) = 60-70 °C

6) Coil Diameter = 19.05cm

7) Conductivity of tube (Stainless Steel) = 43.5 W/m-k

**OBSERVATION TABLE 1**

| **Obs. No.** | **Inlet temperature of cold fluid $T_1$(°C)** | **Output temperature of cold fluid $T_2$(°C)** | **Flow rate of cold fluid $Q_{Water}$(LPH)** | **R.P.M. of agitator motor N** |
|---|---|---|---|---|
| 1 | | | Constant | |
| 2 | | | ” | |
| 3 | | | ” | |
| 4 | | | ” | |
| 5 | | | ” | |
| 6 | | | ” | |

**OBSERVATION TABLE 2**

| Obs. No. | Inlet temperature of cold fluid T1(°C) | Output temperature of cold fluid T2(°C) | Flow rate of cold fluid QWater(LPH) | R.P.M. of agitator motor N |
|---|---|---|---|---|
| 1 | | | | Constant |
| 2 | | | | ” |
| 3 | | | | ” |

| | | | | |
|---|---|---|---|---|
| 4 | | | | ” |
| 5 | | | | ” |
| 6 | | | | ” |

**CALCULATIONS**

1) Mass flow rate of water (ṁ) = (Volumetric flow rate in LPH*$\rho_{water}$)=……….…..=…………$\frac{Kg}{s}$

2) Amount of heat transferred Q=ṁ*$C_p$*($T_2$-$T_1$)=…………………..……….…=……………..……..…$\frac{KJ}{s}$

3) Log mean temperature difference(LMTD) $\Delta T_{lm}= \frac{(T-T_1)-(T-T_2)}{\ln\frac{(T-T_1)}{(T-T_2)}}$=………………………………k

4) Overall heat transfer coefficient (U) $U = \frac{Q}{A*\Delta T_{lm}}$=………………..………..=……………………….$\frac{KW}{m^2-k}$

5) Velocity of water through tube of coil V=$\frac{Q_{water}}{A}$=……………………=……………………$\frac{m}{s}$

6) Reynolds number of water through tube $N_{Re}$=$\frac{d_i*V*\rho}{\mu_{water}}$=………………..=…………………..

7) Prandtl number of water $N_{Pr}$=$\frac{C_p*\mu}{K}$=…………………………….=………………………………

8) Coil inside heat transfer coefficient $h_i$=0.023$(N_{Re})^{0.8}(N_{Pr})^{0.4}[1+3.5*\left({d_i}/{d_{coil}}\right)]\frac{K}{d_i}$

=………………………………………=…………………………………………….$\frac{KW}{m^2-k}$

9) Outside film heat transfer coefficient $\frac{1}{h_o} = \frac{1}{U_o} - \frac{A_o}{A_i h_i} - \frac{Ao}{2\pi kL} ln\frac{d_o}{d_i}$

=…………………………………..=………………………………..

**Or** $h_o$=……………………….=……………………………….$\frac{KW}{m^2-k}$

μ = Viscosity of fluid in the vessel $\frac{Kg}{m*s}$

$\mu_w$= viscosity of fluid in vessel at coil wall temperature $\frac{Kg}{m*s}$

**CALCULATION TABLE 1**

| Obs. No. | Amount of heat transferred Q $\frac{KJ}{s}$ | LMTD $\Delta T_{lm}$ k | Overall heat transfer coefficient U $\frac{KW}{m^2-k}$ | Inside heat transfer coefficient $h_i$ $\frac{KW}{m^2-k}$ | Outside heat transfer coefficient $h_o$ $\frac{KW}{m^2-k}$ |
|---|---|---|---|---|---|
| 1 | | | | | |
| 2 | | | | | |
| 3 | | | | | |
| 4 | | | | | |
| 5 | | | | | |
| 6 | | | | | |

**CALCULATION TABLE 2**

| Obs. No. | Amount of heat transferred Q $\frac{KJ}{s}$ | LMTD $\Delta T_{lm}$ k | Overall heat transfer coefficient U $\frac{KW}{m^2-k}$ | Inside heat transfer coefficient $h_i$ $\frac{KW}{m^2-k}$ | Outside heat transfer coefficient $h_o$ $\frac{KW}{m^2-k}$ |
|---|---|---|---|---|---|
| 1 | | | | | |
| 2 | | | | | |
| 3 | | | | | |
| 4 | | | | | |
| 5 | | | | | |
| 6 | | | | | |

It can be noticed from equation [1] that for the given coil and given fluid in the vessel heat transfer coefficient from liquid t in the vessel to the coil wall will be proportional to $N^{2/3}$. As far as the inside film heat transfer coefficient for the coil is concerned because of increased turbulence due to circulatory path the film heat transfer coefficient will be greater than that calculated for straight pipe. **McAdams** suggests that $h_{coil}=h_{straight\ tube}\ [1+(\frac{3.5\ d_{tube}}{d_{coil}})]$ where $h_{straighttube}$ is given by **Dittus-Boelter** or **Sieder-Tate** equations depending upon the flow regime in the tube. Once the inside heat transfer coefficient of the coil is calculated the outside heat transfer coefficient can be calculated from the equation given below.

$$\frac{1}{h_o} = \frac{1}{U_o} - \frac{A_o}{A_i h_i} - \frac{Ao}{2\pi kL} ln\frac{d_o}{d_i} \qquad [2]$$

Since outside heat transfer coefficient is proportional to $N^{2/3}$a plot of log (N) vs log ($h_o$) should have a slope of 2/3.

**GRAPHS:** Plot graph of log ($h_o$) vs log (N)

Plot graph of log ($h_i$) vs log (U)

**RESULTS:** Slope of log (N) vs log ($h_o$)

Slope of log ($h_i$) vs log (U)

**CONCLUSION**

1. Write down the observations.

2. Try to explain the results from theory studied earlier.

**FURTHER READING**

Fundamental of Heat and Mass Transfer by Frank P. Incropera and David P. Dewitt, Chapter 3.

**TEACHING ASSISTANT**

# Insert 2 Graph Papers

## 7. FLOW MEASUREMENT BY VENTURI METER AND ORIFICE METER

**Objectives:**

- To find the coefficient of discharge of a venturi meter
- To find the coefficient of discharge of an orifice meter

**Theory:**

Venturi meter

The venturi meter has a converging conical inlet, a cylindrical throat and a diverging recovery cone (Fig.1). It has no projections into the fluid, no sharp corners and no sudden changes in contour.

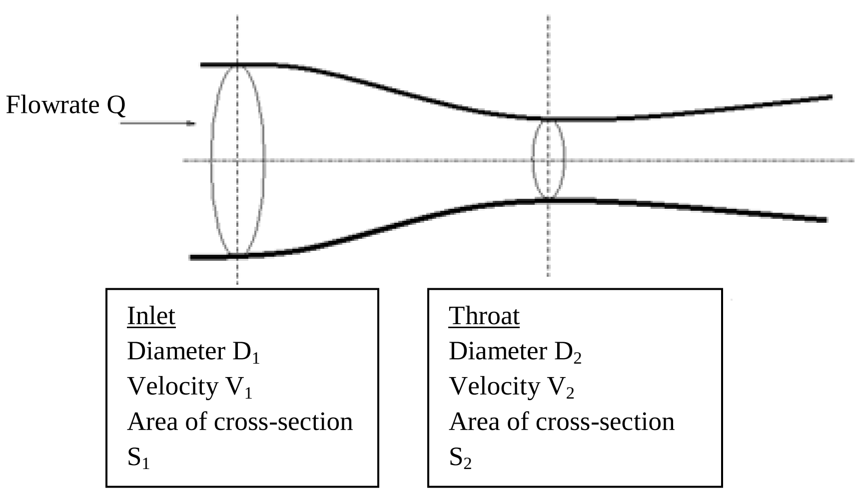



**Fig. 1 Venturi meter**

The converging inlet section decreases the area of the fluid stream, causing the velocity to increase and the pressure to decrease. At the centre of the cylindrical throat, the pressure will be at its lowest value, where neither the pressure nor the velocity will be changing. As the fluid enters the diverging section, the pressure is largely recovered lowering the velocity of the fluid. The major disadvantages of this type of flow detection are the high initial costs for installation and difficulty in installation and inspection.

The *Venturi effect* is the reduction in fluid pressure that results when a fluid flows through a constricted section of pipe. The fluid velocity must increase through the constriction to satisfy the equation of continuity, while its pressure must decrease due to conservation of energy: the gain in kinetic energy is balanced by a drop in pressure or a pressure gradient force. An equation for the drop in pressure due to Venturi effect may be derived from a combination of Bernoulli's principle and the equation of continuity.

The equation for venturi meter is obtained by applying Bernoulli equation and equation of continuity assuming an incompressible flow of fluids through manometer tubes. If $V_1$ and $V_2$ are the average upstream and downstream velocities and $\rho$ is the density of the fluid, then using Bernoulli's equation we get,

$$\alpha_2 {V_2}^2 - \alpha_1 {V_1}^2 = \frac{2(P_1 - P_2)}{\rho} \qquad (1)$$

where $\alpha_1$ and $\alpha_2$ are kinetic energy correction factors at two pressure tap positions.

Assuming the density of fluid to be constant, the equation of continuity can be written as:

$$V_1 = \left(\frac{D_2}{D_1}\right)^2 V_2 \qquad (2)$$

where $D_1$ and $D_2$ are the diameters of the pipe and the throat respectively.

Eliminating $V_1$ from equation (1) and equation (2) we get,

$$V_2 = \frac{1}{\sqrt{\alpha_2 - \alpha_2 \beta^4}} \sqrt{\frac{2(P_1 - P_2)}{\rho}} \qquad (3)$$

where $\beta$ is the ratio of the diameter of throat to that of diameter of pipe.

If we assume a small friction loss between the two pressure taps, the above equation (3) can be corrected by introducing an empirical factor $C_v$ (Coefficient of discharge) and written as:

$$V_2 = \frac{C_v}{\sqrt{1-\beta^4}}\sqrt{\frac{2(P_1-P_2)}{\rho}} \tag{4}$$

The small effect of the kinetic energy factors *α1* and *α2* is also taken into account in the definition of $C_v$.

Volumetric flow rate $Q$ can be calculated as:

$$Q = V_2 S_2 = \frac{C_v S_2}{\sqrt{1-\beta^4}}\sqrt{\frac{2(P_1-P_2)}{\rho}} \tag{5}$$

where, $S_2$ is the cross sectional area of the throat in $m^2$.

Substituting $(P_1 - P_2) = \rho g \Delta H$ in equation (5) we get,

$$Q = V_2 S_2 = \frac{C_v S_2}{\sqrt{1-\beta^4}}\sqrt{2g\Delta H} \tag{6}$$

where $\Delta H$ is the manometric height difference × (specific gravity of manometric fluid – specific gravity of water).

<u>Orifice meter</u>

An orifice meter is essentially a cylindrical tube that contains a plate with a thin hole in the middle of it. The thin hole essentially forces the fluid to flow faster through the hole in order to maintain flow rate. The point of maximum convergence (vena contracta) usually occurs slightly downstream from the actual physical orifice. This is the reason why orifice meters are less accurate than venturi meters, as we cannot use the exact location and diameter of the point of maximum convergence in calculations. Beyond the vena contracta point, the fluid expands again and velocity decreases as pressure increases.

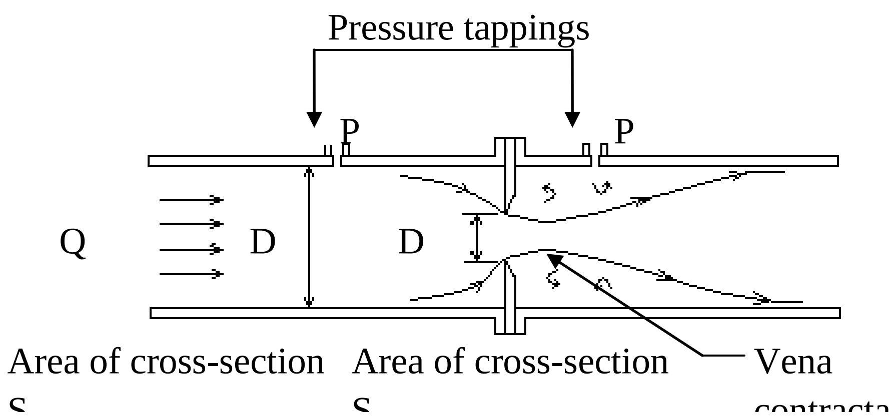



**Fig. 2 Orifice meter**

Figure 2 shows an orifice meter with the variable position of vena contracta with respect to the orifice plate. By employing the continuity equation and Bernoulli’s principle, the volumetric flow rate through the orifice meter can be calculated as described previously for venturi meter.

Hence,

$$Q = V_2 S_2 = \frac{C_O S_2}{\sqrt{1-\beta^4}} \sqrt{2g\Delta H} \tag{7}$$

where $C_o$ is the orifice discharge coefficient, $S_2$ is the area of cross-section of the orifice, $V_2$ is the flow velocity through the orifice, $\beta$ is the ratio of the diameter of orifice to that of the diameter of pipe, $\Delta H$ is the manometric height difference × (specific gravity of manometric fluid – specific gravity of water), and g is the acceleration due to gravity.

Figure 3 depicts the schematic layout of the test setup consisting of the venturi meter and the orifice meter.

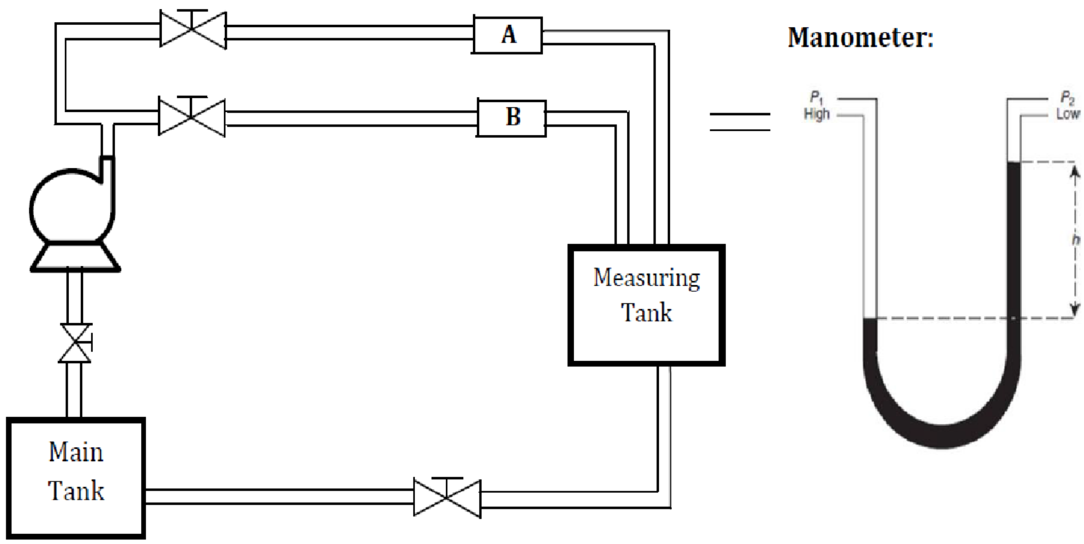



**Figure 3: Schematic Diagram for Venturimeter and Orifice meter**
**A: Venturimeter B: Orifice meter**

**Procedure:**

1. Check all the clamps for tightness.
2. Check whether the water level in the main tank is sufficient for the suction pipe of pump to be completely immersed.
3. For measurement through venturi, open the outlet valve of the venturi meter line and close the valve of the orifice meter line.
4. For a good amount of variation in discharge, close the by-pass valve of pump also.
5. Now switch on the pump.
6. Open the gate valve and start the flow.

7. Remove any bubbles present in the U-tube manometer through air cock valve. Operate the air cock valve slowly and cautiously to avoid mercury run-away through water.
8. Wait till the flow attains a steady state.
9. Close the gate valve of the measuring tank and note the initial water level in the tank. Measure the time taken for the water level in the tank to reach a certain level and then calculate the flow rate. Also note the manometer difference. Before taking any measurements, make sure the flow is stable.
10. Repeat the procedure by changing the discharge by slowly opening the by-pass valve and take the six readings.
11. Repeat the same procedure for orifice meter.

**Observations and calculations:**

A. Venturi meter
Length of the venturi meter = 16mm
Entrance diameter, $D_1$ = 16mm
Throat diameter, $D_2$= 7mm
Length = 8mm

Cross-sectional area of the throat, $S_2 = \pi (D_2)^2/4 =$
$\beta = D_2/D_1 =$

| Collector Tank Readings | | | | Manometer Reading | | | | Coefficient of discharge, $C_v$ |
|---|---|---|---|---|---|---|---|---|
| Initial water level (cm) | Final water level (cm) | Time taken (sec) | Flow rate, Q ($m^3$/sec) | $h_1$ (cm) | $h_2$ (cm) | $h_1$-$h_2$ (cm) | $\Delta H = 12.6 \times (h_1-h_2) \times 10^2$ (m) | $C_v = \frac{Q\sqrt{1-\beta^4}}{S_2\sqrt{2g\Delta H}}$ |
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Average value of $C_v$ =

B. Orifice meter

Length of the orifice meter = 13mm
Entrance diameter, $D_1$ = 16mm
Diameter of the orifice, $D_2$= 8mm

Cross-sectional area of the orifice, $S_2 = \pi (D_2)^2/4$ =
$\beta = D_2/D_1$ =

| Collector Tank Readings | | | | Manometer Reading | | | | Coefficient of discharge, $C_O$ |
|---|---|---|---|---|---|---|---|---|
| Initial water level (cm) | Final water level (cm) | Time taken (sec) | Flow rate, Q ($m^3$/sec) | $h_1$ (cm) | $h_2$ (cm) | $h_1$-$h_2$ (cm) | $\Delta H = 12.6 \times (h_1-h_2) \times 10^{-2}$ (m) | $C_O = \frac{Q\sqrt{1-\beta^4}}{S_2\sqrt{2g\Delta H}}$ |
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Average value of $C_O$ =

**Further reading:**

- McCabe, W.L., Smith, J.C., and Harriott, P., 1993, *Unit Operations of Chemical Engineering*, McGraw-Hill Inc., Singapore, Chap. 8.
- White, F.M., 2016, *Fluid Mechanics*, McGraw-Hill Education, New York, Chap. 6.

**TEACHING ASSISTANT**

## 8.LOSSES DUE TO PIPE FITTINGS

**Aim:** To determine the losses across the fittings in a pipe network

**Theory:**

The resistance to flow in a pipe network causes loss in the pressure head along the flow. The overall head loss across a pipe network consists of the following:

- Major losses ($h_{major}$), and
- Minor losses ($h_{minor}$)

(i) Major losses

Major losses refer to the losses in pressure head of the flow due to friction effects. Such losses can be evaluated by using the *Darcy-Weisbach* equation:

$$h_{major} = f\frac{Lv^2}{2gD} \tag{1}$$

where *f* is the Darcy friction factor, L is the length of the pipe segment, v is the flow velocity, D is the diameter of the pipe segment, and g is acceleration due to gravity. Equation (1) is valid for any fully-developed, steady and incompressible flow.

The friction factor *f* can be calculated by the following empirical formula, known as the *Blasius formula*, valid for turbulent flow in smooth pipes with $Re_D < 10^5$:

$$f = 0.316(Re_D)^{-0.25} \tag{2}$$

(ii) Minor losses

In a pipe network, the presence of pipe fittings such as bends, elbows, valves, sudden expansion or contraction causes localized loss in pressure head. Such losses are termed as minor losses. Minor losses are expressed using the following equation:

$$h_{minor} = K\frac{v^2}{2g} \quad (3)$$

where *K* is called the *Loss Coefficient* of the pipe fitting under consideration.

Minor losses are also expressed in terms of the equivalent length of a straight pipe ($L_{eq}$) that would cause the same head loss as the fitting under consideration:

$$h_{minor} = K\frac{v^2}{2g} = f\frac{L_{eq}\,v^2}{2gD}$$

or

$$L_{eq} = K\frac{D}{f} \quad (4)$$

In the present study, we shall determine the head losses across sudden enlargement, sudden contraction, sharp bend (90 ° elbow), smooth bend, and a straight section.

*Loss of head due to sudden enlargement:* This is the energy loss due to sudden enlargement. Sudden enlargement in the diameter of pipe results in the formation of eddies in the flow at the corners of the enlarged pipe (Fig.1). This results in the loss of head across the fitting.

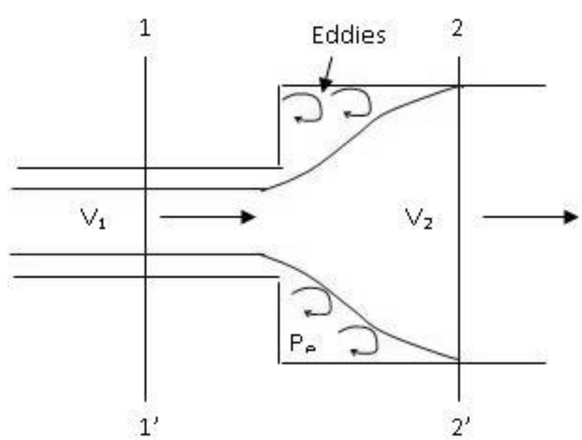


**Fig. 1 Sudden Expansion**

*Loss of head due to sudden contraction:* This is the energy loss due to sudden contraction. In reality, the head loss does not take place due to the sudden contraction but due to the sudden enlargement, which takes place just after vena-contracta (Fig. 2).

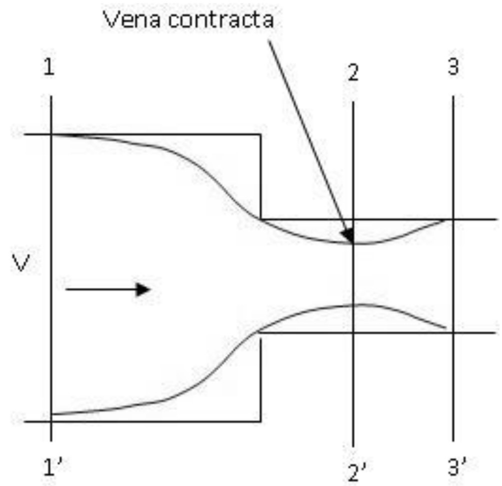


**Fig. 2 Sudden Contraction**

*Loss of head due to bend in pipe:* This is the energy loss due to bend. When a bend is provided in the pipeline, there is a change in direction of the velocity of flow (figures 3 and 4). Due to this, the flow separates from the walls of the bend and formation of eddies takes place.

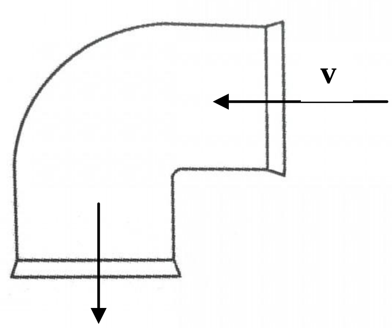


**Fig. 3 Sharp Bend (90° elbow)**

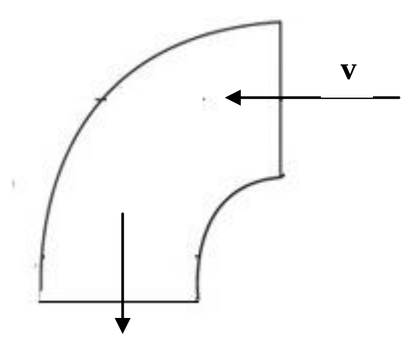


**Fig. 4 Smooth Bend**

Figure 5 shows the schematic layout of the pipe network to be used in the present study.

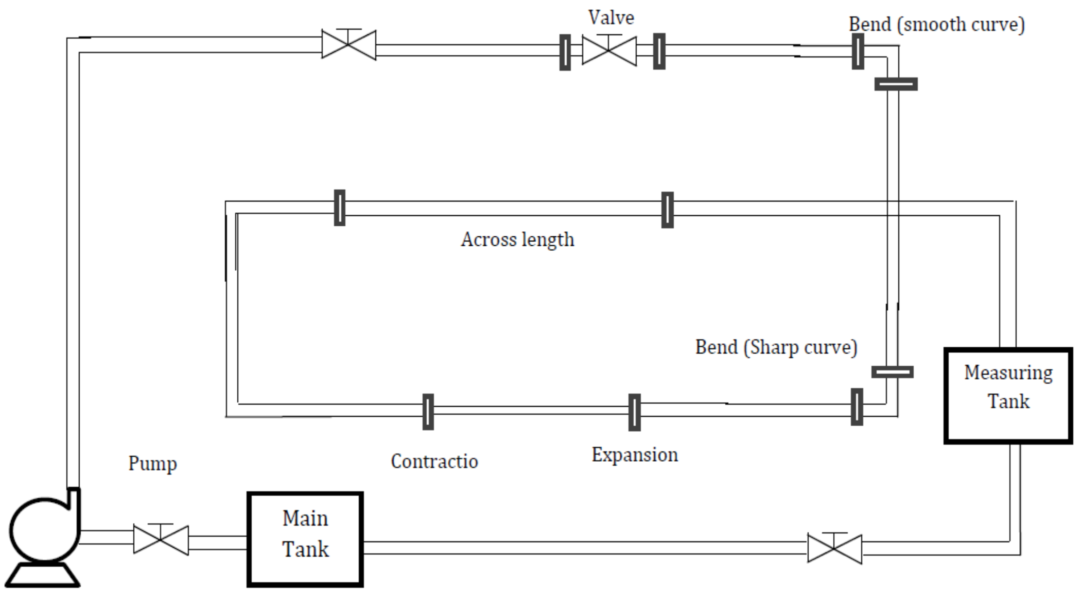


**Fig. 5 Schematic layout of pipe network with fittings**

**Procedure:**

1. Start the pump and wait until water flows through all the sections of the piping network and attains a steady state.
2. When steady state is achieved, measure the readings in the manometers across the fittings of interest.
3. With the help of a stop-watch, measure the time required to fill water in the measuring tank to a certain height and then calculate the flow rate.
4. In case of straight section, calculate the Darcy's friction factor $f$ for a given flow rate by using equations (1) and (2) and compare the values thus obtained.

5. For each pipe fitting, find the loss coefficient $K$ from eq. (3). Also, calculate the Darcy's friction factor $f$ from eq. (2) and substitute in eq. (4) to obtain equivalent length for the fitting. Compare the values of $K$ obtained from the experiment with the standard values for a given fitting.

**Observations:**

Diameter of the collecting tank, $D_c$ = 0.28 m

Diameter of the larger cross-section pipe, $D_1$ = 14.3 mm

Diameter of the smaller cross-section pipe, $D_2$ = 9.22 mm

1. <u>Straight section</u>

Length of the pipe between the pressure tapping, L =

| Collector tank readings | | | | Manometer Readings | | | |
|---|---|---|---|---|---|---|---|
| Initial water level (cm) | Final water level (cm) | Time taken (sec) | Flow rate, Q ($m^3$/sec) | $h_1$ (cm) | $h_2$ (cm) | $h_1$-$h_2$ (cm) | Head loss, $\Delta h$ = 12.6 × ($h_1$-$h_2$) ×$10^{-2}$ (m) |
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2. <u>Sudden expansion</u>

Length of the pipe between the pressure tapping, L =

| Collector tank readings | | | | Manometer Readings | | | |
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| Initial water level (cm) | Final water level (cm) | Time taken (sec) | Flow rate, Q ($m^3$/sec) | $h_1$ (cm) | $h_2$ (cm) | $h_1$-$h_2$ (cm) | Head loss, $\Delta h$ = 12.6 × ($h_1$-$h_2$) ×$10^{-2}$ (m) |
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3. <u>Sudden contraction</u>

Length of the pipe between the pressure tapping, L =

| Collector tank readings | | | | Manometer Readings | | | |
|---|---|---|---|---|---|---|---|
| Initial water level (cm) | Final water level (cm) | Time taken (sec) | Flow rate, Q ($m^3$/sec) | $h_1$ (cm) | $h_2$ (cm) | $h_1$-$h_2$ (cm) | Head loss, $\Delta h = 12.6 \times (h_1-h_2) \times 10^{-2}$ (m) |
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4. <u>Sharp bend</u>

Length of the pipe between the pressure tapping, L =

| Collector tank readings | | | | Manometer Readings | | | |
|---|---|---|---|---|---|---|---|
| Initial water level (cm) | Final water level (cm) | Time taken (sec) | Flow rate, Q ($m^3$/sec) | $h_1$ (cm) | $h_2$ (cm) | $h_1$-$h_2$ (cm) | Head loss, $\Delta h = 12.6 \times (h_1-h_2) \times 10^{-2}$ (m) |
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5. Smooth bend

Length of the pipe between the pressure tapping, L =

| Collector tank readings | | | | Manometer Readings | | | |
|---|---|---|---|---|---|---|---|
| Initial water level (cm) | Final water level (cm) | Time taken (sec) | Flow rate, Q ($m^3$/sec) | $h_1$ (cm) | $h_2$ (cm) | $h_1$-$h_2$ (cm) | Head loss, $\Delta h = 12.6 \times (h_1-h_2) \times 10^{-2}$ (m) |
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**Results:**

1. Straight section

| Flow rate, Q ($m^3$/sec) | Head loss, $\Delta h$ (m) | Flow velocity, v (m/s) | Reynolds number, $Re_D$ | Darcy's Friction factor, *f* (Darcy-Weisbach eq.) | Darcy's Friction factor, *f* (Blasius formula) | % difference |
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2. <u>Sudden expansion</u>

| Flow rate, Q ($m^3$/sec) | Head loss, $\Delta h$ (m) | Flow velocity, v[1] (m/s) | Loss coefficient, $K$ | Reynolds number, $Re_D$ | Darcy's Friction factor, $f$ (Blasius formula) | Equivalent length, $L_{eq}$ (m) |
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Average value of $K$ (from the above table) =

Standard value of $K$ =

% Difference =

3. <u>Sudden contraction</u>

| Flow rate, Q ($m^3$/sec) | Head loss, $\Delta h$ (m) | Flow velocity, v[*] (m/s) | Loss coefficient, $K$ | Reynolds number, $Re_D$ | Darcy's Friction factor, $f$ (Blasius formula) | Equivalent length, $L_{eq}$ (m) |
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[*] Flow velocity based on smaller pipe diameter

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Average value of *K* (from the above table) =

Standard value of *K* =

% Difference =

4. <u>Sharp bend</u>

| Flow rate, Q ($m^3$/sec) | Head loss, $\Delta$h (m) | Flow velocity, v (m/s) | Loss coefficient, *K* | Reynolds number, $Re_D$ | Darcy's Friction factor, *f* (Blasius formula) | Equivalent length, $L_{eq}$ (m) |
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Average value of *K* (from the above table) =

Standard value of *K* =

% Difference =

2. Smooth bend

| Flow rate, Q ($m^3/sec$) | Head loss, $\Delta h$ (m) | Flow velocity, v (m/s) | Loss coefficient, *K* | Reynolds number, $Re_D$ | Darcy's Friction factor, *f* (Blasius formula) | Equivalent length, $L_{eq}$ (m) |
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Average value of *K* (from the above table) =

Standard value of *K* =

% Difference =

**Further reading:**


Books

- McCabe, W.L., Smith, J.C., and Harriott, P., 1993, *Unit Operations of Chemical Engineering*, McGraw-Hill Inc., Singapore, Chap. 5.
- White, F.M., 2016, *Fluid Mechanics*, McGraw-Hill Education, New York, Chap. 6.

Websites

- www.metropumps.com/ResourcesFrictionLossData.pdf
- http://nptel.ac.in/courses/101103004/module5/lec6/2.html


**TEACHING ASSISTANT**

## 9.PLATE TYPE HEAT EXCHANGER

**AIM:**

To determine the overall heat transfer coefficient in a plate type heat exchanger at different hot fluid flow rate

**EXPERIMENTAL SETUP:**

1. A Stainless-steel plate type heat exchanger with facility to measure hot and cold fluid temperature.
2. A stainless steel insulated tank with a heater to act as a reservoir for the hot fluid
3. Hot fluid circulation pump with a speed control potentiometer
4. Cold fluid inlet from the water supply taps
5. Thermocouples in order to sense the inlet and outlet temperature of hot as well as cold fluid
6. Rotameter

**THEORY:**

The plate heat exchanger normally consists of corrugated plates assembled into a frame. The hot fluid flows in one direction in alternating chambers while the cold fluid flows in true counter-current flow in the other alternating chambers. A schematic diagram of the flow is shown in Figure 1. The fluids are directed into their proper chambers either by a suitable gasket or a weld depending on the type of exchanger chosen. Plate heat exchangers are best known for having overall heat transfer coefficients (U-values) in excess of 3–5 times the U-value in a shell and tube designed for the same service.

Plate heat exchanger is an attractive option when more expensive materials of construction can be employed. The significantly higher U-value results in far less area for a given application. The higher U-values are obtained by inducing turbulence between the plate surfaces. Owing to this they are also known to minimize the fouling.

**Heat Transfer Correlation:**

Generally the heat transfer correlation for a fluid flow past a solid surface is expressed in a dimensionless form

$$Nu = Nu\,(\mathrm{Re}, \mathrm{Pr}) \qquad (1)$$

where Nu is the non dimensionless Nusselt number expressed as

$$Nu = \frac{hD}{k} \tag{2}$$

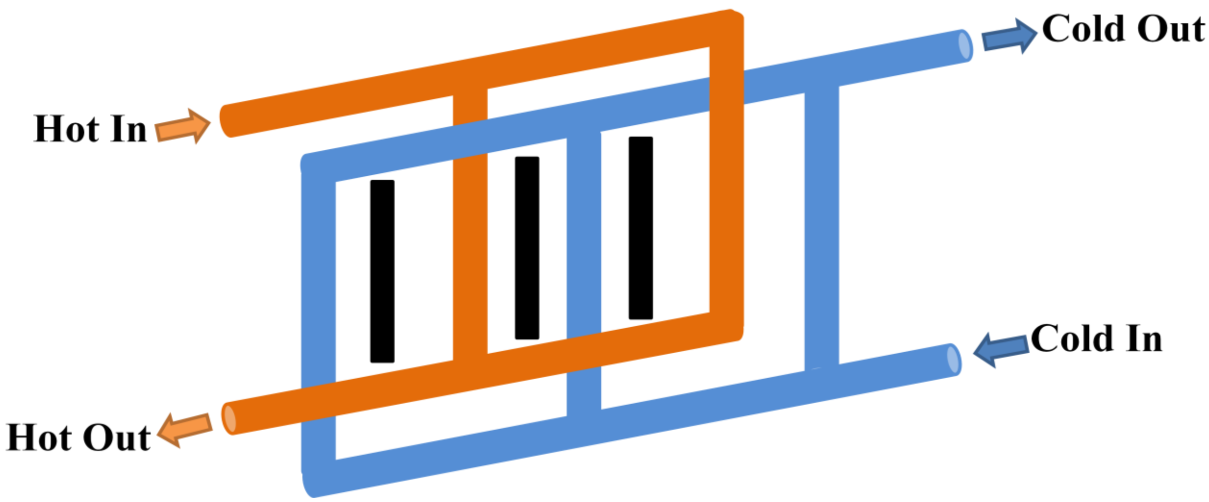


Figure 1: Schematics for Plate Type Heat Exchanger

For a heat transfer in a laminar fluidflow past a solid surface, with constant fluid properties, the steady state temperature profile is a function of Re, and Pr only.

For fully developed laminar flows (internal flows), we expect the Nusselt number Nu to be constant, howeverfor a developing flow it is expressed as:

$$Nu = C_1 \operatorname{Re}^{\alpha} \operatorname{Pr}^{\beta} \tag{3}$$

The value of$\beta \approx 0.33$, C=0.664. The value of$\alpha$is found to be around 0.3 for developing laminar flow and around0.5 for turbulent flow. The transition from laminar to turbulent region occurs between $2300 \leq Re \leq 4000$ for corrugated plates. It can be expected to be higher for plain plates. The heat transfer coefficient appearing in the Nusset number can be calculated from the overall heat transfercoefficientU, which is given by

$$\frac{1}{U} = \frac{1}{h_h} + \frac{\Delta x}{K_p} + \frac{1}{h_o}, \tag{4}$$

Where, $h_h$ is the hot fluid heat transfer coefficient and$h_0$is the cold fluid heat transfer coefficient, $K_p$ is the thermal conductivity of the metal plate and$\Delta x$is its thickness. Once the heat exchanger material and itsgeometry are fixed, then the metal wall resistance ($\Delta x/K_p$) becomes constant.

Similarly, if the flow rate ofcold fluid is fixed and its mean temperature does not differ much for different flow rates of hot fluid, thenthe resistance of the cold fluid will remain almost constant. Thus, the overall heat transfer coefficient willdepend upon the value of the hot fluid heat transfer coefficient alone. If the bulk mean temperature does notdiffer much for different flow rates, then all the physical properties will remain nearly the same and Eq. (4)can be re-written in combination with Eq. (3) as

$$\frac{1}{U} = \frac{1}{h_h} + C = \frac{m}{u^{\alpha}} + C \qquad (5)$$

WhereCis constant. $h_h$ can therefore be evaluated from the intercept of the plot of 1/U vs. $1/u^{\alpha}$.Thena plot of 1/U vs. $1/=u^{\alpha}$will provide the intercept valueC, which is then used to calculate the heat transfercoefficient from Eq. (5). The Nusselt number correlation can then be found. For the sake of simplicity, it isoften assumed that$\alpha$=0.5. This can be verified if the plot of 1/U vs. $1/u^{0.5}$ is a straight line for a largerange in the small limit.

**PROCEDURE:**

1. Set the pump to maximum hot fluid flow rate (≈550 lph), and measure the temperature difference between the outlet and inlet of the hot fluid.
2. Set the temperature of the inlet hot fluid in the dual temperature indicator cum controller. The setpoint should be set between 60 to $80^{O}$C.
3. Provide cooling water supply to the plate heat exchanger minimum as 100 LPH. This will ensure that the temperature difference is maintained at least 2–$3^{O}$C. Keep this flow rate constantthroughout the experiment.
4. Connect the 15 A and 5 A plug pins to a stable 230 V A.C. electric supply. Care should be taken toconnect these two pins in different phases of the power supply. Switch on the heater power supply.
5. Adjust the flow rate of hot fluid through the heat exchanger by adjusting the speed of hot fluid circulation pump. This step should be done only when the steady state is reached.
6. To measure the oil flow rate, we have to close the valve beneath the fixed volume container for certain height. By measuring the time required for the hot fluid toachieve certain height, we can calculate the velocity and successively the volumetric flow rate of hot fluid.
7. Repeat step 6 at least for 10 different flow rates.

**SPECIFICATION:**

Height of the plate = 120 mm
Width of the plate = 65 mm
Gap between two plates = 1 mm
Number of plates = 7
Number of hot fluid chambers = 3
Number of cold fluid chambers = 4
Zero error of hot fluid digital thermometers $\delta T$ =
Oil in tank = Servo Engine 32 Grade Oil
Flash Point (COC) = 190$^o$C
Kinematic Viscosity = 32 cS @ 45$^o$C, 5.4 cS @ 100$^o$C.
Density of Oil = 857 Kg/m$^3$
Diameter of flow measuring cylinder: 0.0762m
Thermal conductivity of the hot fluid: 0.13W/mk
Thermal conductivity of stainless steel 16 W/m.K

**OBSERVATION TABLE:**

| Observation No. | Flow rate V lph | Hot fluid temperature ($^o$C) | | Cold fluid temperature ($^o$C) | |
|---|---|---|---|---|---|
| | | Inlet ($T_1$) | Outlet ($T_2$) | Inlet ($t_1$) | Outlet ($t_2$) |
| 1 | | | | | |
| 2 | | | | | |
| 3 | | | | | |
| 4 | | | | | |
| 5 | | | | | |
| 6 | | | | | |
| 7 | | | | | |
| 8 | | | | | |
| 9 | | | | | |
| 10 | | | | | |

**CALCULATIONS**

1. Total heat transfer area of heat exchanger $A = NHW =$

2. Cup mean temperature (use any typical value) $T_m = (T_1+T_2)/2 =$

3. Density of Oil $\rho = 857$ kg/m$^3$

4. Specific heat of Ethylene glycol atTm $C_p =$

5. Viscosity of oil at Tm $\mu =$

6. Thermal conductivity of Ethylene glycol atTm K =

7. Prandtl Number for hot fluid $Pr = \frac{C_p \mu}{K} =$

8. Equivalent diameter $De = \frac{2Wb}{W+b} =$

**SAMPLE CALCULATIONS**

1.Flow Rate V =

2. Velocity of hot fluid in a chamber $u = \frac{V}{WbN_h} =$

3. Total heat transferred $Q = \rho C_p V (T_1 - T_2) =$

4. Brinkman number $Br = \frac{\mu u^2}{K(T_1 - T_2)} =$

5. Log Mean Temperature Difference (LMTD)= $\Delta T_{LMTD} = \frac{(T_1 - t_2) - (T_2 - t_1)}{\ln[\frac{(T_1 - t_2)}{(T_2 - t_1)}]} =$

6. Overall heat transfer coefficient $U = \frac{Q}{A \Delta T_{LM}} =$

7. Reynolds number $Re = \frac{D_e u \rho}{\mu} =$

8.Intercept of 1/U vs $1/u^{0.5}$plot C =

Here C = $\frac{1}{h_0} + \frac{\Delta x}{K}$

9. Hot fluid heat transfer coefficient $\frac{1}{h_i} = \frac{1}{U} - C =$

10. Nusselt number $Nu = \frac{h_i D_e}{K}$

**GRAPHS:**

1. Plot of 1/Uvs $1/u^{0.5}$

2. Plot of Nu number vs Reynolds number

**CONCLUSION/DISCUSSION ON THE RESULT**

1. Overall heat transfer coefficient and individual heat transfer coefficient in plate type heat exchanger was calculated.

**FURTHER READING**

G H Hewitt, G L Shires, and T R Bott, “Process Heat Transfer”, CRC Press, NY, 1994

**TEACHING ASSISTANT**

# Insert 2 Graph Papers

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# Appendix

(Ref: http://www.metropumps.com/ResourcesFrictionLossData.pdf)