INSTRUCTION MANUAL

OF

HEAT TRANSFER LAB

STEFAN BOLTZMANN'S APPARATUS

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STEFAN-BOLTZMANN'S APPARATUS

1.0 THEORY:

All substances emit thermal radiation when heated. When radiation coming out from a hot body fall over a surface of another body, a part of the radiation is absorbed, another part transmitted through the body and yet the third part is reflected back. The surface, which, absorbs all the thermal radiation falling over it is called a black body. This implies that for black body the absorptivity is unity and transmitivity & reflectivity are zero. Stefan-Boltzmann's law states that emissive power of a surface is proportional to the fourth power of its absolute temperature i.e.,

$$e \propto T^4 \qquad (1)$$

$$e = \sigma \varepsilon T^4 \qquad (2)$$

Where,

e = emissive power of a body W/m^2

T = Absolute temperature, K

σ = Stefan-Boltzmann's constant, W/m^{2}K^4

ε = emissivity of the surface

For black surface ε =1, hence, the Eq.(2) reduces to :

$$e = \sigma T^4 \qquad (3)$$

2.0 OBJECTIVES:

1. To determine Stefan-Boltzmann's constant
2. To determine radiative heat transfer coefficient

3.0 APPARATUS:

Fig.1 shows the apparatus schematically. The apparatus consists of a water-heated jacket of hemispherical shape. A copper "Test Disk" is fitted at the center of the bottom portion of the hemisphere. Three thermocouples numbering 2, 3 & 4 are fitted in the inner part of the hemisphere and a thermocouple (No.1) is fitted with the Test Disk. Hot water is obtained from the hot water tank. Hot water (boiling hot) is generated in the tank with the help of an electric heater. The hot water is then used to fill the space between outer space of the hemisphere and the vessel surrounding it. A water level indicator is used to ensure that the water level is sufficiently high over the top of the hemispherical dome. The hot water ensures a uniform temperature around the dome. A stopwatch is used to measure the temperature of the Test Disk at uniform time interval of three to five seconds.

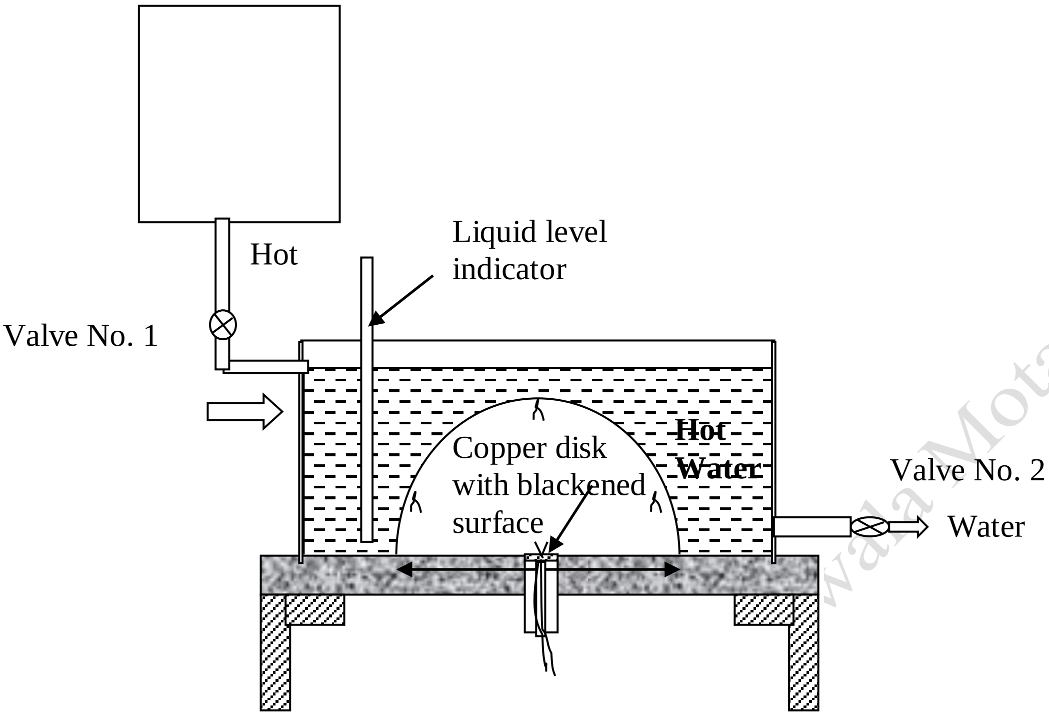


Fig.1 Schematic diagram of the experimental set-up

4.0 SUGGESTED EXPERIMENTAL SET-UP:

1. Ensure that proper amount of water is in the water tank and the valve No. 2 is closed and valve No.1 is also closed.
2. Switch 'ON" the electric heater
3. Blacken the "Test Disk" with the help of lamp black and let it cool down
4. Note the temperature of water by thermocouple No. 5. When it starts boiling put off the heater.
5. Ensure that the valve No. 2 is closed and valve No. 1 is open.
6. Ensure that water level over the hemisphere is sufficient. Use the liquid level indicator to ensure this.
7. Note down the hemisphere temperature with the help of thermocouple No 2,3 & 4.
8. Insert the Test Disk through the hole provided at the bottom of the hemispherical portion and start the stopwatch.
9. Monitor the Test Disk temperature with the help of thermocouple No. 1 at equal interval of time. Note down different readings.

5.0 RESULTS & DISCUSSIONS:

1. Fill the data sheet.
2. Plot the variation of temperature of the disk with time to find the dT/dt.
3. Now determine the Stefan Boltzman constant.

6.0 SAMPLE DATA SHEET:

Name of Experiment: **Stefan Boltzmann's Apparatus**

Name of the student: Semester Batch Session

Diameter of the Test Disk = mm
Area of Test Disk, A = m^2
Specific heat of Copper, Cp = kJ/kg°C
Mass of the Test Disk, m = g

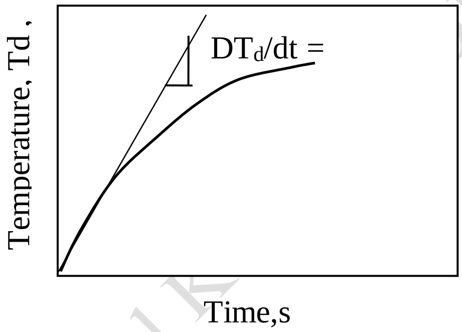
Run No.	Time (sec)	Hemisphere Temperature			Test Disc Temperature T1°C
		T2 °C	T3°C	T4 °C	

7.0 APPENDIX-1: Critical data of experimental set-up

Diameter of the Test Disk = 20 mm
Area of Test Disk, A = 3.14×10^{-4} m^2
Specific heat of Copper, Cp = 0.3831 kJ/kg°C
Mass of the Test Disk, m = 2.34g

8.0 APPENDIX-2: Sample calculation

Plot the variation of temperature of the disk with time. Let the plot be as shown below:



Hemisphere temperature, T_h K (K = °C +273.13)
Temperature of Test Disk, T_d = K
Heat Flux, q $= \varepsilon\sigma A (T_h^4 - T_d^4)$ (4)
Also, q $= m\, Cp\, (dT_d/dt)\big|_{t=0}$ (5)

From Eq.4 and Eq.5 one can write,

$\sigma = [m\, Cp\, (dT_d/dt)\big|_{t=0}] / A\, (T_h^4 - T_d^4)$ as $\varepsilon = 1$ for black body

9.0 APPENDIX-3: Experimental data

Diameter of the Test Disk = 20 mm
Area of Test Disk, A = 3.14×10^{-4} m^2

Specific heat of Copper, Cp = 0.3831 kJ/kg°C
Mass of the Test Disk, m = 2.34g

Run No.	Time (sec)	Hemisphere Temperature			Test Disc Temperature T1°C
		T2 °C	T3°C	T4 °C	
1.	0	87.9	88.1	88.1	32.8
	5				35.3
	10				37.2
	15				38.8
	20				40.0
	25				41.1
	30				42.3
	35				43.2

10.0 APPENDIX -4: Data Analysis

Determination of Stefan Boltzman Constant

Hemisphere temperature, T_h = (87.9 + 88.1 + 88.1)/3
= 88.03 °C
= 88.03 + 273.13
= 361.16 K

Temperature of Test Disk, T_d = 32.8+ 273.13
= 305.93 K

Plot the variation of temperature of the disk with time. DT_d/dt comes to be 0.44

σ = $[m\ Cp\ (dT_d/dt)] / A\ (T_h^{\ 4} - T_d^{4})$
= $0.00234 \times .3831 \times 10^3 \times 0.44 / 3.14 \times 10^{-4}\ (361.16^4 - 305.93^4)$
= $1.52 \times 10^{-7}\ W/m^2K^4$

11.0 PRECAUTIONS:

1. Never put "ON" the heater before filling the tank with water.
2. Put "OFF" the heater before charging the water from tank.

Drain the water once the experiment is completed.

THERMAL CONDUCTIVITY OF POWDERS

CONTENTS:

4.0 Theory

5.0 Objective

6.0 Apparatus

7.0 Procedure for assembling the guarded hot plate Assembly

5.0 Suggested experimental work

6.0 Results & Discussions

7.0 Procedure for de-assembling and changing of the Powder sample

8.0 Sample Data Sheet

9.0 Appendix-1: Critical data of Experimental set-up

10.0 Appendix-2: Sample experimental data

11.0 Appendix-3: Data Analysis

THERMAL CONDUCTIVITY OF POWDERS

1.0 THEORY:

Knowledge of thermal properties like thermal conductivity of a powder is important for computing heat transfer rates (losses) in many fields of engineering such as Mechanical and Chemical. Determination of thermal conductivity of powders is bit difficult as powders generally porous and can exist at different void fraction depending upon pressure exerted on it. The thermal conductivity of powder is determined by passing measured quantity of heat through a known thickness of powder and monitoring the temperature difference. The thermal conductivity, k of the powder is then computed by Eq. (1).

$$Q = -\,Ak\,\Delta t/x \qquad (1)$$

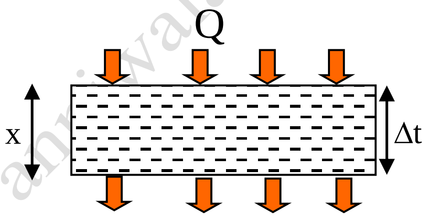
Where

Q = the rate of heat transfer, kJ/s

A = area of heat transfer, m^2

Δt = temperature drop across the powder layer of thickness “x”, ^oC

x = thickness of powder layer, m



To apply Eq.(1) for determination of thermal conductivity it is necessary that all the heat is transported through conduction across the sample. This condition necessitates that the thickness of powder layer be very small so that the conduction is across the layer.

2.0 OBJECTIVE:

To determine the thermal conductivity of a given powder sample using guarded hot plate apparatus.

3.0 APPARATUS:

The apparatus mainly consists of guarded hot plate assembly to measure thermal conductivity of powder. The container in which, the assembly is kept well insulated using thermal insulation, and the necessary instrumentation for measurement & control of voltage, current & temperature. The guarded hot plate assembly is shown in Fig.1.

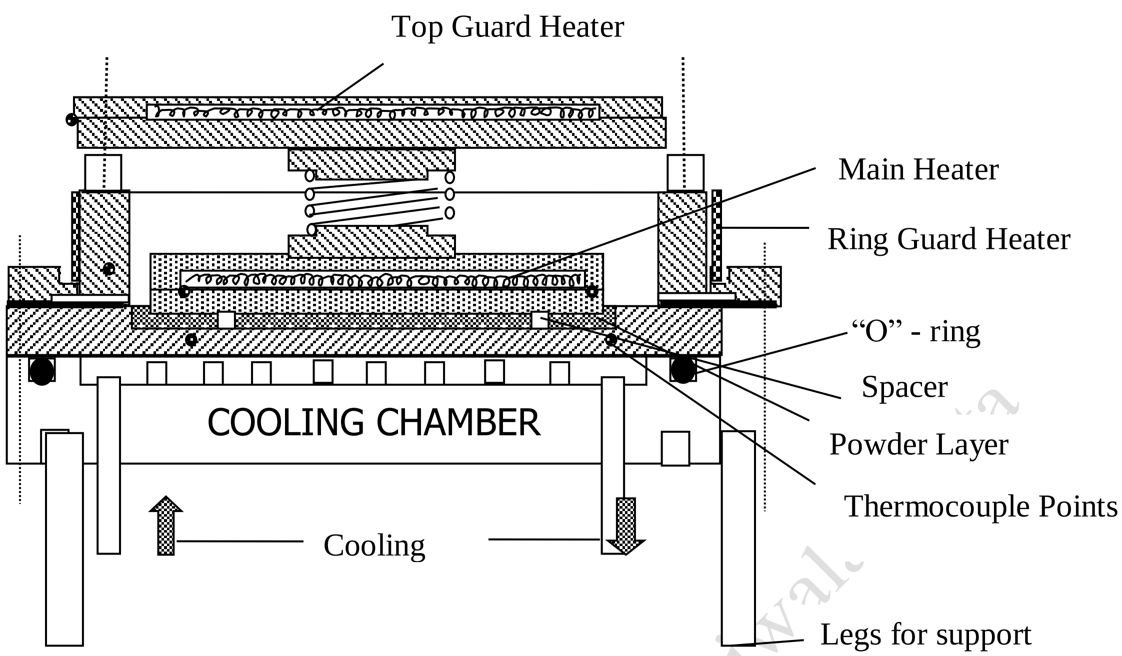


Fig.1 Guarded hot plate assembly

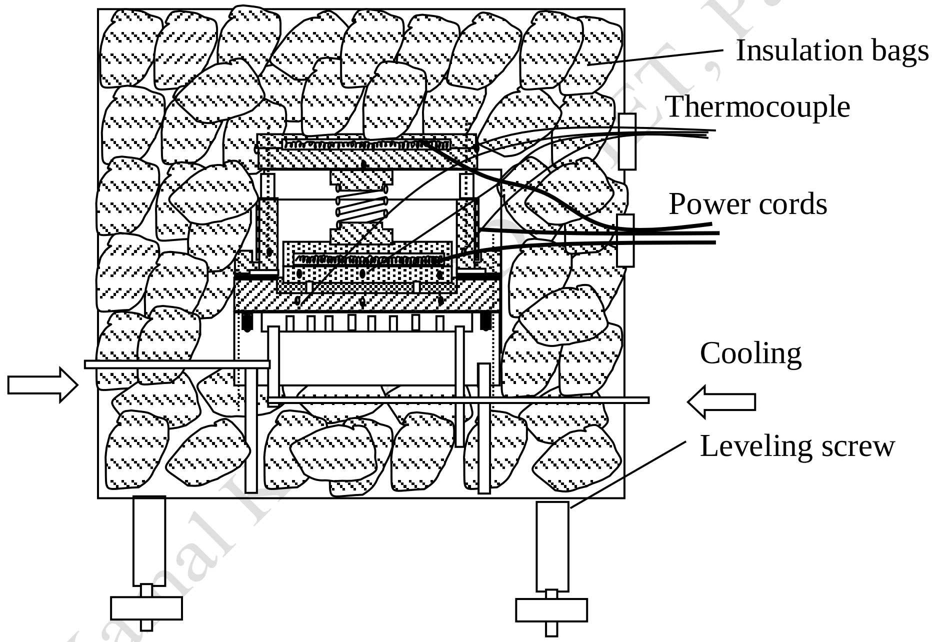


Fig. 2 Guarded Hot Plate Assembly inside the container with water, thermocouple and power connections

One top guard and second ring guard heaters are provided to stop heat leaking from the main heater and what ever heat is generated in the main heater passes through the powder layer and goes to the cooling chamber. In this way longitudinal transfer of heat in conduction mode is ensured which qualifies the Eq.(1) to be used for the computation of thermal conductivity.

The apparatus is properly instrumented to measure voltage and current fed to main heater. The Wattage to different heaters is controlled by dimmerstate. The temperature at different points are measured using copper-constantan thermocouples and are displayed using a digital temperature indicator with resolution of 0.1 °C. Thermocouples are scanned manually using a 6 point selector switch. Fig. (2) shows the Guarded Hot Plate Assembly inside the container with water, thermocouple and power connections and Fig. (3) shows the thermocouple numbers and their actual positions inside the apparatus.

It is necessary that the Guarded Hot Plate assembly be assembled every time when it is required to find out the thermal conductivity of a new powder. The procedure for proper assembly of the unit is detailed below:

4.0 PROCEDURE FOR ASSEMBLING THE GUARDED HOT PLATE ASSEMBLY:

Step1: Open up the container and remove the thermal insulation packs.

Step2: Level the container with the help of the leveling screws fitted at the bottom on a tabletop.

Step3: Introduce the actual amount of powder in the powder space with the help of the syringe. Check that the cavity is so full of powder that it is just above the spacer (provided).

Step4: Place the main heater assembly on the spacers in the powder filled cavity.

Step5: Place the top guard heater at the proper place.

Step 6: Take out the thermocouple wires and heater connection wires through the space between the top guard heater and the main heater.

Step7: Connect the heater and thermocouple terminals properly and connect the cooling water supply.

Step8: Cover the Guarded Hot Plate Assembly with insulation bags and close the cover of the container.

5.0 SUGGESTED EXPERIMENTAL WORK:

Step1: Assemble the apparatus as per the instruction given above.

Step2: Connect the thermocouples to the selector switch

Step3: Connect the three heaters (main, top guard, ring guard) to the sockets meant for these.

Step4: Bring the three dimmerstates (use to energize three heaters) to zero out put state.

Step5: Switch ON the cooling water flow to the apparatus.

Step6: Switch ON the main switch and note the initial temperature of all the 6 thermocouples.

Step7: Energize the main heater using the dimmerstate to about 35 volts. The power input to the main heater should be such that its hot surface gets the desired temperature and a low temperature drop is achieved across the powder film.

Step8: Energize top guard and ring guard heaters using the dimmerstates gradually so that the temperature shown by thermocouples 5 to 6 are properly balanced.

Step9: Wait for steady state to reach and note down the temperatures of all the 6 thermocouples and voltage & current readings for main heater, ring and top guard heater.

Step10: Step 7-9 is repeated for measuring thermal conductivity of powder at higher mean temperature.

6.0 RESULTS AND DISCUSSIONS:

1. Tabulate the temperatures of thermocouple Nos. 1 to 6. Note the difference of temperature between guard heater and main heater.
2. Compute the average temperature of main heater and cold plate and note the temperature difference.
3. Compute the thermal conductivity of the powder using Eq. (1) and compare it with the actual value of thermal conductivity of the given powder from published reference.
4. Compute the error and report it. Mention the sources of error and suggests methods to minimize it.

7.0 PROCEDURE FOR DE-ASSEMBLING & CHANGING OF THE POWDER SAMPLE:

Step1: Remove the top guard heater carefully along with thermocouple and heater connections.

Step2: Remove the main heater assembly and clean the bottom plate with the help of cotton and then rinse the surface with acetone.

Step3: Remove the powder with syringe and then by remaining powder with cotton. Clean the cavity further with acetone to remove the traces.

DO NOTS

- Do not increase voltage more than 55 Volts.
- Do not touch the surface of heating tube.
- Do not touch the insulation bags.
- The temperature of the three heaters i.e. main, top and ring must be nearly same.
- The water should not be re-circulated.

8.0 SAMPLE DATA SHEET:

Name of experiment: ***Thermal conductivity of powders***

Name of the student: Semester Batch Session

Powder =
Diameter of main heater plate, m =

Powder space diameter, m =
Spacer thickness, x, m =

Heater(s)	**Voltage, V**	**Current, I**	**Wattage = VI**
Main			
Ring Guard			
Top Guard			

Location(s)	Thermocouple number and temperature, °C			
Main Heater Plate	1		2	
Cold Plate	3		4	
Ring Guard	5			
Top Guard	6			

9.0 APPENDIX-1: Critical data of experiment:

Diameter of main heater plate, m = 0.110

Powder space diameter, m = 0.115
Spacer thickness, x, m = 0.003

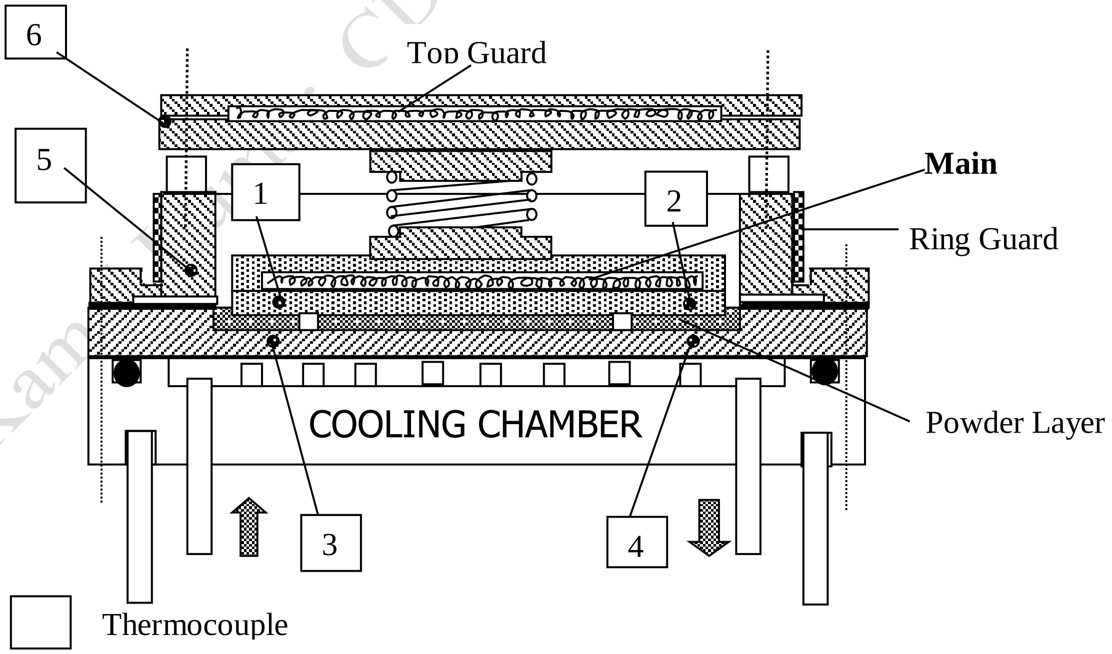


Fig. 3 Details of thermocouple positions inside the Guarded Hot

10.0 APPENDIX-2: Sample experimental data

Powder = Plaster, gypsum
Diameter of main heater plate, m = 0.110

Powder space diameter, m = 0.115
Spacer thickness, x, m = 0.003

Heater(s)	**Voltage, V**	**Current, I**	**Wattage = VI**
Main	29.8	0.28	8.344
Ring Guard	33.3	1.45	48.285
Top Guard	33.2	0.41	13.612

Location(s)	Thermocouple number and temperature, °C			
Main Heater Plate	1	39.8	2	40.3
Cold Plate	3	26.2	4	25.9
Ring Guard	5	40.6		
Top Guard	6	39.9		

11.0 APPENDIX-3: Data analysis

Rate of heat transfer Q = V x I
= 29.8 x 0.28
= 8.344 W

Average temperature of main heater, t_h

$$t_h = \frac{t_1 + t_2}{2}$$

$$= \frac{39.8 + 40.3}{2}$$

$$= 40.05^{o}C$$

Average temperature of cold plate, t_c

$$t_c = \frac{t_3 + t_4}{2}$$

$$= \frac{26.2 + 25.9}{2}$$

$$= 26.05^{o}C$$

$$\Delta t = t_h - t_c$$

$$= 40.05 - 26.05$$

$$= 14.0^{o}C$$

$$\text{Area, } A = \frac{\pi r_1^2 + \pi r_2^2}{2}$$

$$= \frac{\pi(0.055^2 + 0.0575^2)}{2}$$

$$= 9.945 \times 10^{-3} \text{ m}^2$$

Thermal conductivity of Plaster, gypsum, k can be determined by

$$k = \frac{Qx}{A\Delta t}$$

$$= \frac{8.344 \times 0.003}{9.945 \times 10^{-3} \times 14.0}$$

$$= 0.18 \text{ W/m}^o\text{C}$$

Thermal conductivity, k of Gypsum is 0.48 W/m^{o}C at 20^{o}C.
(taken from Laboratory Manual on Solar Thermal Experiments by H.P. Garg & T.C. Khandpal, Narosa Publishing House P: 297).

1. For given apparatus determine:

(a) Thermal conductivity of given insulating powder.
We are finding out the thermal conductivity of powders.

(b) Critical thickness of insulation.
(c) To determine the thermal resistance of insulating powder.
(d) To plot theoretical temp. profile by dividing the thickness in mini five parts.
(e) State all the assumption applied in above calculation.

For above four points
In the said apparatus as well as for finding out the thermal conductivity of powders, it is not possible to provide different thickness layers of powders.

Determination of thermal conductivity of powders is bit difficult as powders generally porous and can exist at different void fraction depending upon pressure exerted on it. Also for determination of thermal conductivity it is necessary that all the heat is transported through conduction across the sample. This condition necessitates that the thickness of powder layer be very small so that the conduction is across the layer.

But still you can teach the students about the critical thickness of insulation.

As the thickness of the lagging is increased, resistance to heat transfer by thermal conduction increases. However, the outside area from which heat is lost to the surroundings also increases, giving rise to the possibility of increased heat loss. For a cylindrical pipe/flat surface, the above possibility of heat loss being increased by the application of lagging, takes place, only if $hr/k<1$, where k is the thermal conductivity of the lagging, h is the outside film coefficient and r is the outside radius of the pipe. In practice, this situation is likely to arise only for pipes/plates of small diameter/thickness.

The extent of heat loss from the outer surface of the heater tube/plate (insulated or bare) is reflected by the temperature of the outer metal wall of the tube/plate (as heating is in constant heat flux mode). If the heat loss from the surface is more than the outer metal wall, wall temperature will be less and vice-versa.

THERMAL CONDUCTIVITY OF METAL ROD

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5.0 Results & Discussions

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THERMAL CONDUCTIVITY OF METALROD

1.0 THEORY:

If one dimensional heat flow is assumed in the metal rod under experimentation, then the heat conduction equation is:

$$Q = -kA \frac{dt}{dx} \tag{1}$$

Where, Q = rate of heat flow
A = cross sectional area through which heat flows
dt/dx = temperature gradient along x axis
k = thermal conductivity of the material

From Eq.(1) the thermal conductivity can be calculated as:

$$k = \frac{Q}{A\left(\frac{dt}{dx}\right)}$$

2.0 OBJECTIVES:

1. To determine the thermal conductivity of a metal rod using one dimensional heat conduction equation.
2. To plot the temperature distribution along the length of the metal rod.

3.0 APPARATUS:

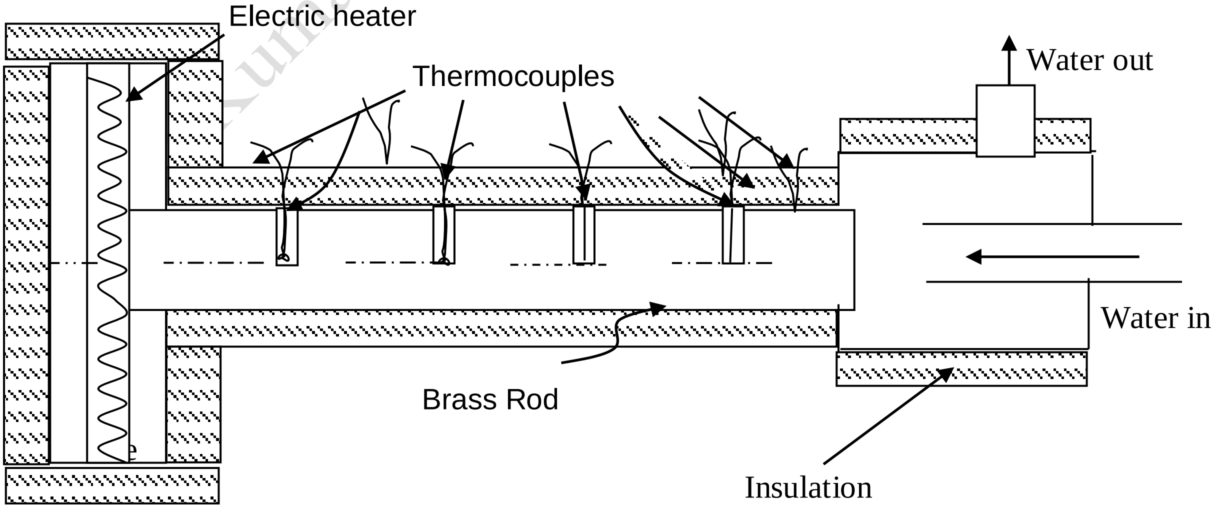

Fig.1: Schematic diagram of the apparatus for thermal conductivity of metal rod

The schematic diagram of the apparatus is shown in Fig.1.The apparatus consists of a solid brass rod of 31.7 mm diameter and length 300.0 mm. One end of the rod is fitted with a copper flange of diameter 82.0 mm and thickness 5.0 mm. An electric heater is attached to this end of rod. The other end opens to a cooling water chamber through which cooling water at a controlled rate is passed. Four thermocouples (No.1 to 4) are also inserted up to the central line of the solid brass rod to measure the axial temperature. Two thermocouples are also pasted over the insulated portion to measure its airside surface temperature for computation of the heat loss to air. The position of thermocouples is given in Table 1.

4.0 SUGGESTED EXPERIMENTAL WORK:

Step1: Draw a neat sketch of the experimental set up giving important dimensions.
Step2: Switch ON the cooling water pump.
Step 3: Switch ON the electric heater and set the voltage to 30 Volts using variac and allow the steady state to reach. Note the thermocouple temperatures from 1 to 6. Also note the reading of voltmeter and ammeter.
Step 4: Change the voltage to 35, 40 & 45 in steps and repeat the step 2 each time you change voltage.
Step 5: Note down the surface temperature of insulation and ambient temperature.

5.0 RESULTS AND DISCUSSIONS:

1. Fill up the data sheet.
2. Draw the longitudinal temperature profile for the rod for different energy input to the heater keeping the cooling water flow rate constant.
3. Compute the thermal conductivity of the brass rod using experimental data and compare it with actual values, which happen to be 111 W/mK at ambient temperature.
4. Predict the temperature profile through theoretical analysis and match it with experimental data.

6.0 SAMPLE DATA SHEET:

Name of Experiment: **Thermal conductivity of metal rod.**
Name of the student: Semester Batch Session

Diameter of rod, mm =
Length, mm =
Material =

Sl. No.	V	I	W	Thermocouple Nos. Temp °C					
				1	2	3	4	5	6

7.0 APPENDIX-1: Critical data of experiment

Diameter of rod, mm = 31.7 mm
Length, mm = 310 mm
Material = Brass
Type of insulation = felt

Table 1: Position of embeded thermocouples from the flanged end

Thermocouple No.	1	2	3	4
Distance from flange, mm	5.0	105.0	205.0	205.0

Table 2: Position of other thermocouples on the surfaces of insulation

Thermocouple No.	5	6
Distance from Flange, mm	5.0	305

8.0 APPENDIX-2: Sample Experimental Data

Diameter of rod, mm = 31.7mm
Length, mm = 310mm
Material = Brass

Sl. No.	V	I	W	Thermocouple Nos. Temp °C					
				1	2	3	4	5	6
1.	40.1	0.38	15.24	54.4	41.9	35.0	28.9	32.9	28.5

9.0 APPENDIX-3: Data Analysis

To get better results the setup should be seen at minimum possible heat flux and the colling water rate should be maintained at highest value to provide iso-thermalisation.

Heat input, Q = V x I
= 40.1 x 0.38
= 15.24 W

$$\text{Cross Sectional area} = \frac{\pi\,(0.0317)^2}{4} = 0.79 \times 10^{-2}\ \text{m}^2$$

Heat Flux, q = $15.24/0.79 \times 10^{-3} = 19291.2$ W/m^2

$$Q = -k \left(\frac{dt}{dx}\right)$$

Distance between thermocouple No. 1 & 2= 0.10m

$$19291.2 = k \; \frac{(54.4-41.9)}{0.10}$$

or k = 154.32 W/m^2

(k) from table = 111 W/m^0C

(k for brass is 111 W/m^oC at taken from Laboratory Manual on Solar Thermal Experiments by H.P. Garg & T.C. Khandpal, Narosa Publishing House)

THERMAL CONDUCTIVITY OF SLAB

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11.0 Theory

12.0 Objective

13.0 Apparatus

4.0 Suggested Experimental Work

5.0 Results & Discussions

6.0 Sample Data Sheet

7.0 Appendix-1: Critical data of Experimental set-up

8.0 Appendix-2: Sample experimental data

9.0 Appendix-3: Data Analysis

THERMAL CONDUCTIVITY OF SLAB

1.0 THEORY:

Knowledge of thermal properties like thermal conductivity of a insulating material is important for computing heat transfer rates (losses from hot surfaces) in many fields of engineering such as Mechanical and Chemical. The thermal conductivity of insulating materials are determined by passing measured quantity of heat through a known thickness of material layer and monitoring the temperature difference. The thermal conductivity of the material is then computed by Eq. (1).

$$Q = - Ak\, \Delta t/x \qquad (1)$$

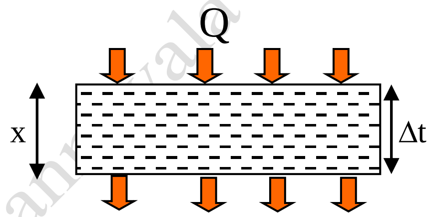
Where

Q = the rate of heat transfer, kJ/s

A = area of heat transfer, m^2

Δt = temperature drop across the material layer of thickness "x", ^oC

x = thickness of layer, m



2.0 **OBJECTIVE**:

To measure the thermal conductivity of the given insulating material sample by the guarded hot-plate apparatus and compare it with the thermal conductivity of the sample given in the standard reference.

3.0 APPARATUS:

The experimental set-up is schematically shown in Fig. 1. It consists of main heater, surrounded by guard heater (details of the heater positions are shown in Fig.2), cooling plate and insulating material specimen (Bakelite) arranged in the order shown in the Fig. 1.

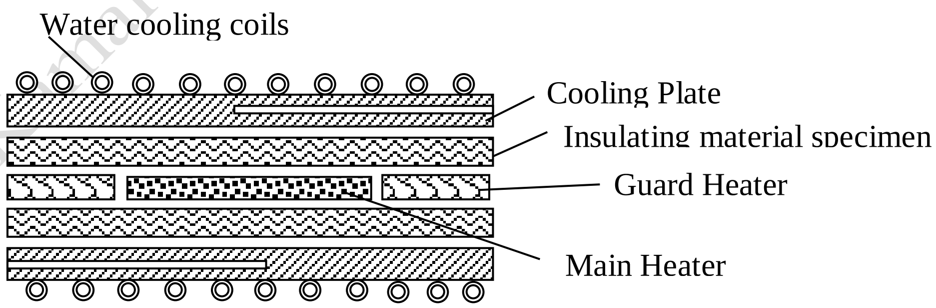


Fig. 1 Schematic diagram of the experimental set-up

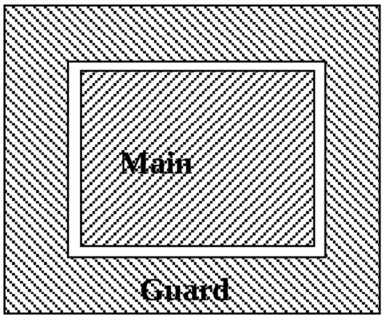


Fig.2 Relative positions of heater

The cooling plate is cooled by flowing water. The guard heater is used to stop the leakage of heat from the edges of the main heater. The apparatus is that is why called guarded hot plate apparatus for determination of thermal conductivity. The apparatus is so designed that its thermal inertia is very less.

4.0 SUGGESTED EXPERIMENTAL WORK:

Step1: Assemble the apparatus as shown in the Fig.1 in proper sequence.

Step2: Switch ON the cooling water pump and resume the cooling water flow rate to the apparatus.

Step3: Switch ON the main heater 1 and guard heater 2. Adjust the voltage of guard heater to 26.0V. Measure the temperature of the main heater. Calculate average temperature and adjust the voltage of the guard heater so that its temperature becomes almost equal to the average temperature of the main heater.

Step4: Give sufficient time for the realization of steady state.

Step 5: Measure the temperatures of the main heater, temperature of the guard heater and temperature of the top and bottom cold plates.

5.0 RESULTS AND DISCUSSIONS:

1. Draw the schematic diagram of the apparatus.
2. Collect the data carefully and fill it in the given data sheets.
3. Report the thermal conductivity of the material and its percent difference from the standard value given in the standard reference.
4. Indicate the sources of errors and possible treatments to decrease it.

6.0 SAMPLE DATA SHEET:

Name of Experiment: **Thermal Conductivity of Slab**

Name of the student: Semester Batch Session

Thickness of the insulating material, x, m =

Area of the main heater plate, A, m^2 =

Run No.	Main heater temp., °C	Guard heater temp., °C	V	I	Top cold plate temp., °C	Bottom cold plate temp., °C
	T_1	T_2			T_3	T_4

7.0 **APPENDIX-1:** Critical data of experiment:

Thickness of the insulating material, x, m = 0.010
Area of the main heater plate, A, m^2 = 0.103 x 0.103

8.0 **APPENDIX-2:** Sample experimental data

Thickness of the insulating material, x, m = 0.010
Area of the main heater plate, A, m^2 = 0.103 x 0.103

Run No.	Main heater temp., °C	Guard heater temp., °C	V	I	Top cold plate temp., °C	Bottom cold plate temp., °C
	T_1	T_2			T_3	T_4
1.	50.1	49.7	8.0	0.49	25.0	25.8

9.0 **APPENDIX-3:** Data analysis

Rate of heat transfer Q = V x I
= 8.0 x 0.49
= 3.92 W

Temperature of main heater, $t_h = 50.1^oC$

Temperature Difference across the sample

$\Delta T = T_h - T_3$
$= 50.9 – 25.0$
$= 25.9^oC$

Thermal conductivity of slab (insulating material), k on one side of heater

$$k = \frac{Qx}{2A\Delta t}$$

$$= \frac{3.92 \text{ x } 0.010}{2 \text{ x } 0.103 \text{ x } 0.103 \text{ x } 25.9}$$

$$= 0.0713 \text{ W/m}^o\text{C}$$

Thermal conductivity, k of insulating material, wool is 0.052W/m^{o}C at 30^{o}C.

(taken from Laboratory Manual on Solar Thermal Experiments by H.P. Garg & T.C. Khandpal, Narosa Publishing House P:298).

PIN FIN IN NATURAL AND FORCED CONVECTION

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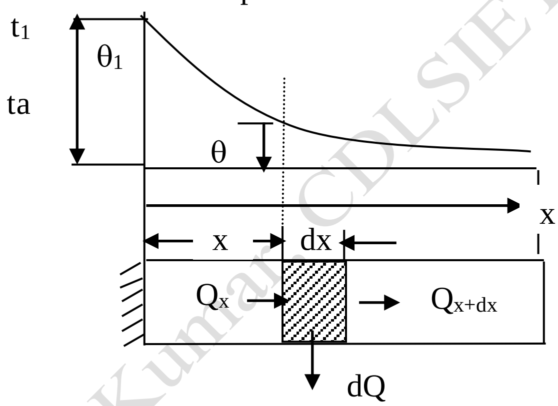
PIN FIN IN NATURAL AND FORCED CONVECTION

1.0 THEORY:

Fins may be of various cross sections (rectangular, circular, triangular and so on, including irregular geometric forms). Pin fin is a finite rod of circular cross section. Consider the propagation of heat in a straight rod of constant cross-section. Let the cross section area be denoted as ‘A’ and its parameter be ‘C’. The rod is placed in a medium of constant temperature ‘t_a’; the coefficient of heat transfer from the surface of the rod to the surrounding is assumed to be constant for the entire surface. Suppose also that the thermal conductivity ‘k’ of the rod material is quite large, and the cross-section of the rod is extremely small in relation to its length. The latter assumption allows us to neglect temperature variations over the cross-sections and to consider that the temperature changes only along the axis of the rod. To simplify calculation , t_a= constant will be considered as a reference temperature. The excess in temperature of the rod over this reference value will be denoted by θ. It is obvious that, $\theta = t - t_a$.

Where t_a = temperature of the medium and
t = rod temperature



Heat Transfer through Pin fin

For a given temperature t_1 of the base of the rod, the excess temperature of the base is $\theta_1 = t_1 - t_a$.

Consider a rod element dx long at a distance x from the base. The heat balance equation for this rod element can be written as:

$$dQ = Q_x - Q_{x+dx}$$

Where Q_x = quantity of heat entering the left face of the element per unit time.

Q_{x+dx} = quantity of heat flowing from the opposite face of the element in the same time.

dQ = quantity of heat lost per unit time to the surroundings by the outer surface of the rod.

Q_{x+dx} = quantity of heat flowing from the opposite face of the element in the same time.

dQ = quantity of heat lost per unit time to the surroundings by the outer surface of the rod.

In accordance with Fourier's law

$$Q_x = -kA \frac{d\theta}{dx}$$

and

$$Q_{x+dx} = -kA \frac{d}{dx}\left(\theta + \frac{d\theta}{dx}.dx\right)$$

or

$$Q_{x+dx} = -kA \frac{d\theta}{dx} - kA \frac{d^2\theta}{dx^2}.dx$$

Consequently,

$$Q_x - Q_{x+dx} = kA \frac{d^2\theta}{dx^2}.dx \qquad (1)$$

On the other hand, according to the Newton's law

$$dQ = h.C.(\theta)\,dx \qquad (2)$$

Equating (1) and (2), we get the following differential equation for the temperature distribution in the rod

$$\frac{d^2\theta}{dx^2} = \frac{hC}{kA}\theta = m^2\theta \qquad (3)$$

$$\text{where, } m = +\sqrt{\frac{hC}{kA}} \qquad (4)$$

It is clear from Eq. (4) that for a fin of present shape and size 'm' is constant, provided that the heat transfer coefficient 'h' is constant over the entire surface, and the thermal conductivity 'k' is constant within the considered temperature range. Then the common integral for Eq. (3) will be

$$\theta = C_1e^{mx} + C_2e^{-mx} \qquad (5)$$

Where constants C_1 and C_2 are determined from the boundary conditions, which may be given in several ways, depending on the length of the rod and other factors.

Finite rod (Pin Fin)

Pin fin is a finite rod of circular cross section. The differential equation Eq.(3) and its solution Eq.(5) are also valid for a finite rod. The boundary conditions are

$$\left.\begin{aligned} &\text{at } x = 0 \qquad \theta = \theta_1 \\ &\text{at } x = l \qquad -k\left(\frac{d\theta}{dx}\right)_{x=l} = h_l\,\theta_l \\ &\text{or} \\ &\left(\frac{d\theta}{dx}\right)_{x=l} = \frac{-h_l}{k}\,\theta_l \end{aligned}\right\} \qquad (6)$$

Where θ_l = temperature at the end of the pin fin (rod)

h_l = coefficient of heat transfer from the end of the rod.

l = length of the pin fin (rod).

When x = l, the quantity of heat imported to the end of the rod by conduction equals the quantity lost from the end of the rod to the surroundings.

The constants C_1 and C_2 in Eq.(5) are determined by using boundary conditions, Eq.(6).

$$\left.\begin{aligned} &\text{At } x = 0 \qquad \theta_l = C_1 + C_2 \\ &\text{At } x = l \qquad \left(\frac{d\theta}{dx}\right)_{x=l} = C_1.\,me^{ml} - C_2.\,me^{ml} = \frac{h_l}{k}\,\theta_l \end{aligned}\right\} \qquad (7)$$

and $\theta_l = C_1.e^{ml} + C_2.\,e^{-ml}$

From Eq. (7), C_1 and C_2 are determined as follows

$$C_1 = \frac{\theta_1\left(ml - \frac{h_l}{k} \right)}{e^{2ml}\left(m + \frac{h_l}{k}\right) + m - \frac{h_l}{k}}$$

$$C_2 = \theta_1 \frac{e^{2ml}\left(m + \frac{h_l}{k}\right)}{e^{2ml}\left(m + \frac{h_l}{k}\right) + m - \frac{h_l}{k}}$$

Substituting the value of C_1 and C_2 in Eq.(3) we obtain:

$$\theta = \theta_1 \left[\frac{e^{mx}\left(m - \frac{h_l}{k}\right)}{e^{2ml}\left(m + \frac{h_l}{k}\right) + m - \frac{h_l}{k}} + \frac{e^{-mx}.\, e^{2ml}\left(m + \frac{h_l}{k}\right)}{e^{2ml}\left(m + \frac{h_l}{k}\right) + m - \frac{h_l}{k}} \right] \qquad (8)$$

Multiplying and dividing the right side of Eq.(8) by e^{-ml} and making simple algebraic transformation, we get:

$$\theta = \theta_1 \frac{m\left\{e^{m(l-x)} + e^{-m(l-x)}\right\} + \frac{h_l}{k}\left\{e^{m(l-x)} - e^{-m(l-x)}\right\}}{m\left(e^{ml} + e^{-ml}\right) + \frac{h_l}{k}\left(e^{ml} - e^{-ml}\right)} \qquad (9)$$

Recalling that

$$\frac{e^{x} + e^{-x}}{2} = \mathrm{Cosh}(x) \text{ and } \frac{e^{x} - e^{-x}}{2} = \mathrm{Sinh}(x)$$

Equation (9) may be written in the following form:

$$\theta = \theta_1 \frac{\mathrm{Cosh}\,[m(l-x)] + \frac{h_l}{km}\,\mathrm{Sinh}\,[m(l-x)]}{h_l} \qquad (10)$$

$$\mathrm{Cosh}\,(ml) + \frac{\quad}{km}\,\mathrm{Sinh}\,(ml)$$

When the loss of heat from the end of the rod can be neglected, the boundary conditions Eq.(7) can be presented as follows:

At $x = 0$ $\qquad \theta = \theta_1$

At $x = l$ $\qquad \left(\frac{d\theta}{dx}\right)_{x=l} = 0$

The latter can be admitted for this case when h_l is small at the end of the rod, and the thermal conductivity k of the material is large and the ratio $h_l/k \rightarrow 0$, i.e. the loss of heat from the end of the rod can be neglected.

With these boundary conditions, the second terms of the numerator and denominator of Eq.(10) turns into zero and the equation acquires the appearances

$$\theta = \theta_1 \frac{\mathrm{Cosh}\,[m(l-x)]}{\mathrm{Cosh}\,(ml)} \qquad (11)$$

Where θ is in ^{o}C

Equations (10) & (11) can be used to calculated temperature in any cross section of the rod. The fraction of heat lost from the end of the rod is usually small compared to the quantity of heat lost from the surface of the fins, and Eq. (11) is usually used for practical engineering calculations.

In extreme cases, when x = l, Eq. (11) acquires the following form:

$$\theta_{x=l} = \frac{\theta_l}{\mathrm{Cosh}\,(ml)}$$

The amount of heat Q_a lost to the surroundings from the surface of a fin is equal to the amount of heat imparted to its base.

$$Q_a = -kA \left(\frac{d\theta}{dx}\right)_{x=0}$$

From Eq. (11) we find

$$\left(\frac{d\theta}{dx}\right)_{x=0} = -\theta_1 m \frac{\mathrm{Sinh}(ml)}{\mathrm{Cosh}(ml)} = -\theta_1 m \tanh\,(ml)$$

Then, $Q_a = kA\theta_1 m \tanh\,(ml)$ (12)

Substituting the value of m in Eq. (12)

$$Q_a = \theta_1 \sqrt{hCkA} \quad \tanh(ml) \tag{13}$$

Note: The value of 'h' will change when the pin fin will be operated at natural convection and forced convection. The value of 'h' in forced convection will be order of magnitude more than natural convection.

2.0 OBJECTIVE:

1. To measure temperature profiles of a pin fin heated at its bottom at natural and forced convection.

2. Estimation of temperature profiles using Eq.(10) & (11) and to compare it with experimentally observed values.

3.0 APPARATUS:

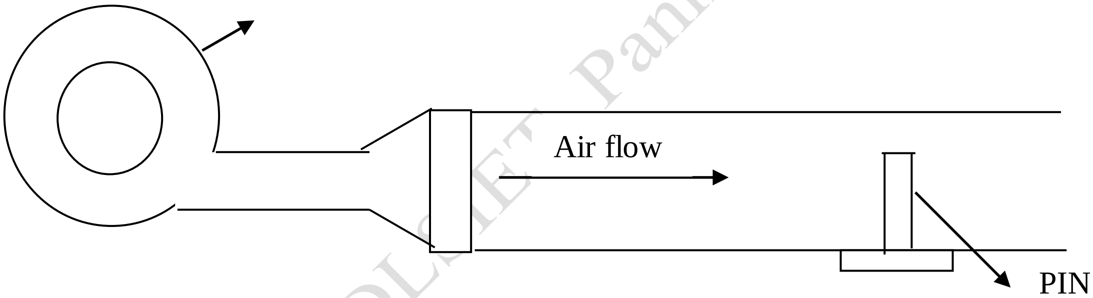


Fig.1: Schematic diagram of apparatus for pin fin in natural and forced convection

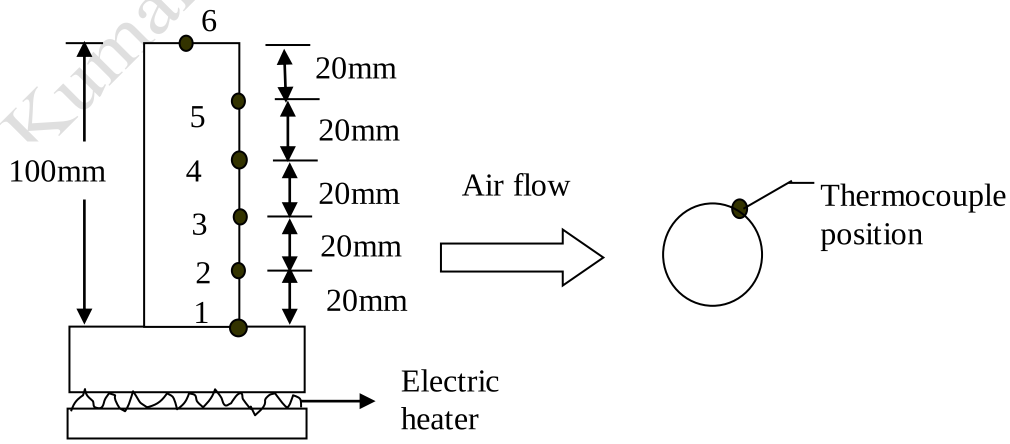


Fig. 2: Details of Pin Fin and thermocouple positions.

The apparatus is shown schematically in Fig. 1. It consists of a brass rod of 19.6 mm diameter and 100.0 mm long. At the base of the brass rod, a brass flange of 10mm thick and 75mm diameter is welded. The flange is heated by an electrical heater controlled by a variac. Thermocouples were welded on the base, on the length and on the tip of the rod. The whole assembly is put inside a duct. The air flow to the duct is supplied by a blower.

4.0 SUGGESTED EXPERIMENTAL WORK:

Step 1: Draw a neat sketch of experimental set-up giving the important dimensions.

Step 2: Switch ON the air blower and fix a predetermined flow rate of air inside the duct.

Step 3: Switch ON the electric heater and with help of variac set the voltage to 25 V. Allow sufficient time for steady state to occur. Note down the readings of voltmeter, ammeter and thermocouples.

Step 4: Change the energy inputs to the heater by variac and set the voltage to 3 0, 35, 40, and 45 V in stages for each change in voltage repeat Step 3.

Step 5: Stop the air flow rate and repeat step 3 and Step 4 for natural convection.

5.0 RESULTS AND DISCUSSIONS:

1. Plot the temperature profile of the pin fin for different base temperature (for different values of energy inputs to the electric heater) for a constant air flow rate.
2. Compute the theoretical temperature profiles by considering the value of h for forced convection and compare these temperatures profiles with that obtained by experiments.
3. Plot temperature profiles of the pin fin as a function of base temperature at Natural Convection mode (i.e. for the OFF condition of the blower).
4. Compute the theoretical temperature profile by considering the value of h for natural convection and compare it with temperature profiles that obtained by experiments.

6.0 SAMPLE DATA SHEET:

Name of experiment: **Pin fin in Natural and forced convection**

Name of the student: Semester Batch Session

Diameter of pin fin, mm =
Length of pin fin, mm =
Material of construction =
Ambient air temperature, ^{0}C =
Type of orientation =

Free/Natural Convection

Run No				Thermocouple No. Temperature, ^{o}C

	V	I	W	1	2	3	4	5	6

Forced Convection

Run No				Thermocouple No. Temperature, °C					
	V	I	W	1	2	3	4	5	6

7.0 APPENDIX –1: Critical Data of Experimental Set-up

Dimensions of Pin fin

Material of construction = Brass
Diameter = 19.6 mm
Length = 100.0 mm
Orientation = Vertical

Thermocouple Position

Thermocouple No	1	2	3	4	5	6
Distance, mm	0	20	40	60	80	100.0

- Distances are measured from base flange
- Thermocouple No.1 gives base flange temperature
- Thermocouple No.6 is placed on the tip of the pin fin
- Details of thermocouple positions are given in Fig. 2.

Dimension of Duct

Breadth = 200 mm
Width = 102 mm
Length = 1000 mm
Material = M.S.

8.0 APPENDIX- 2: EXPERIMENTAL DATA

Diameter of pin fin, mm = 19.6
Length of pin fin, mm = 100.0
Material of construction = Brass
Ambient air temperature, ^{0}C = 35.6
Orientation = Vertical

Free/Natural Convection

Run No				Thermocouple No. Temperature, ^oC					
	V	I	W	1	2	3	4	5	6
1	30.5	0.24	7.32	59.9	48.8	47.3	46.9	46.8	46.5

Forced Convection

Run No				Thermocouple No. Temperature, ^oC					
	V	I	W	1	2	3	4	5	6
1	30.5	0.24	7.32	46.8	40.5	39.9	39.7	39.7	39.6

9.0 APPENDIX-3: Data Analysis

Computation of temperature along the length of PIN FIN during Natural convection.

Average surface temp. along the length of Pin Fin = 49.37^oC

Thermal conductivity of air $k = 0.029 \; \frac{W}{mK}$

Base Temp = 59.9^oC

For a wide range of temperature $k^4 \left(\frac{\beta g \rho^2 Cp}{\mu k} \right) = 36.0$ for air.

This value can be used for easy computation of Gr.Pr values as detailed below.

$$\left(\frac{\beta g \rho^2 C}{\mu k} \right) = 36.0 / k^4$$

$$= \frac{36.0}{(0.029)^4}$$

$$= 5.1 \times 10^7$$

Determination of Heat transfer Coefficient during Natural Convection:

$Gr.\,Pr = 5.1 \times 10^7 (59.9 - 35.6) \times (0.1)^3 = 1.24 \times 10^6$

$h = C'' (\Delta T)^n \, l^{3n-1}$

(For above cases based on the value of Gr.Pr , C"= 1.37 & n= 0.25 taken from P:386 Vol-I Chemical Engg by Coulson & Richardson)

$h = 1.37 \, (24.3)^{0.25} \, (0.1)^{3 \times 0.25 - 1}$

$= 1.37 \times 2.22 \times 1.778$

$= 5.41 \frac{W}{m^2 K}$

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

$C = \pi\, d$

(k for brass is 111 W/m°C taken from laboratory mannual on Solar Thermal experiments by H.P. Garg and T.C. Khandpal)

$m = \sqrt{1.887 \times 5.41} = 3.19$

$$\text{Cos h} = \frac{e^{x} + e^{-x}}{2}$$

Computation of temperature at a distance of 20 mm from the base.

Assume no heat loss from Pin Fin tip.

$$(t-35.6) = (59.9 - 35.6)\frac{\text{Cosh}\,[(3.19) \times (0.1 - 0.02)]}{\text{Cosh}\,(3.19 \times 0.1)}$$

$$= 24.3\,\frac{\text{Cosh}\,(0.2552)}{\text{Cosh}\,(0.319)}$$

$$= 24.3\,\frac{1.033}{1.0513}$$

$$= 23.88$$

$t_{theo} = 23.88 + 35.6$
$= 59.48^{o}C$

$t_{exp} = 48.8^{o}C$

% Error = 21.9%

Computation of temperature along the length of PIN FIN during Forced convection.

$Q = hA\,(t_s - t_1)$

$7.32 = h\pi\, 0.0196 \times 0.1\,(46.8-35.6)$

$$h = 106.14\,\frac{W}{m^2K}$$

$m = \sqrt{106.14 \times 1.887} = 14.15$

Computation of temperature at 20 mm from the base

$$(t-35.6) = (46.8 - 35.6)\frac{\text{Cosh}\,[(14.15) \times (0.1 - 0.02)]}{}$$

Cosh (14.15 x 0.1)

$$= 11.2 \frac{\text{Cosh}\ (1.132)}{\text{Cosh}\ (1.415)}$$

$$= 11.2 \frac{1.712}{2.18}$$

$= 8.76$

$t_{theo} = 8.76 + 35.6$

$= 44.36^{o}C$

$t_{exp} = 40.5^{o}C$

% Error = 9.5%

A DOUBLE PIPE HEAT EXCHANGER

CONTENTS:

23.0 Theory

24.0 Objectives

25.0 Apparatus

4.0 Suggested experimental work

5.0 Results & Discussions

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8.0 Appendix-2: Experimental data

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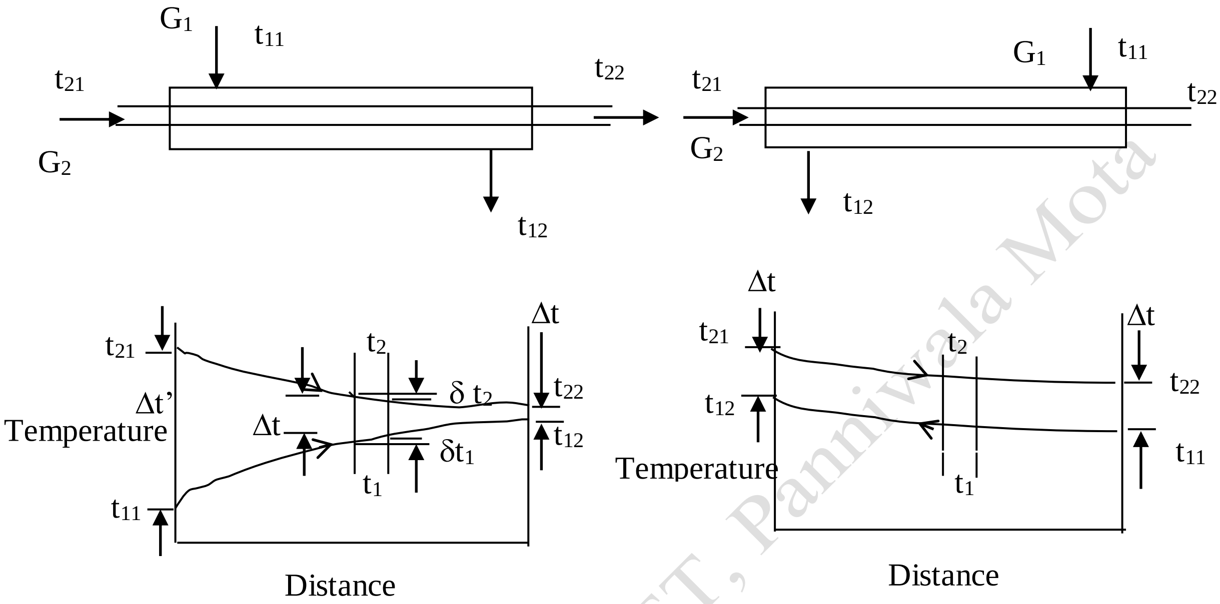
PARALLEL AND COUNTER CURRENT FLOW IN A DOUBLE PIPE HEAT EXCHANGER

1.0 THEORY:



(a) Parallel heat exchanger

(b) Counter Current heat exchanger

Fig. 1: Flow arrangements and temperature profiles of parallel and counter current heat exchangers

PARALLEL FLOW Fig. 1(a)

For an elemental area dA of a heat exchanger, the rate of heat transfer of heat dQ is given by:

$$dQ = UdA\,(t_2 - t_1) = UdA\theta \qquad (1)$$

Where t_1 and t_2 are the temperatures of the two streams and θ is the point value of the temperature difference between the streams. U is the overall heat transfer coefficient.

Also $dQ = G_1\,Cp_1\,dt_1 = -\,G_2\,Cp_2\,dt_2$

thus, $dt_2 = \dfrac{-\,dQ}{G_2Cp_2}$ and $dt_1 = \dfrac{dQ}{G_1Cp_1}$

Where G is the mass flow rate and is equal to product of volumetric flow rate (q) and density ρ.

then $dt_2 - dt_1 = d(t_2 - t_1) = d\theta = -\,dQ\left(\dfrac{1}{G_1Cp_1} + \dfrac{1}{G_2Cp_2}\right)$

Substituting from Eq.(1) for dQ

$$\frac{d\theta}{\theta} = -\,UdA\left(\frac{1}{G_1Cp_1} + \frac{1}{G_2Cp_2}\right) \qquad (2)$$

If the Eq.(2) is integrated between Δt' and Δt one gets :

$$\int_{\Delta t'}^{\Delta t} \frac{d\theta}{\theta} = \left(\frac{1}{G_1Cp_1} + \frac{1}{G_2Cp_2}\right) U\int_0^A A$$

$$\text{or } \Delta t = \Delta t' \exp\left(\frac{-\,UA}{G_1Cp_1}\left\{1+\frac{G_1Cp_1}{G_2Cp_2}\right\}\right) \qquad (2a)$$

From Eq.(2a) it follows that the variation in the temperature difference along the length of the heat transfer surface is exponential. Hence, in parallel flow heat exchanger the difference between the temperatures of hot and cold fluids drops continuously.

If the temperature drop is averaged over the whole surface of heat exchanger then Eq.(2a) becomes

$$\overline{\Delta t} = LMTD = (\Delta t)_{ln} = \frac{\Delta t' - \Delta t''}{\ln\dfrac{\Delta t'}{\Delta t''}} = \frac{\theta_1 - \theta_2}{\ln\dfrac{\theta_1}{\theta_2}} \qquad (2b)$$

The Eq. (2b) is suitable both for parallel flow and counter flow arrangements and the average (LMTD) temperature is called log mean temperature difference, $(\Delta t)_{ln}$.

If Eq. (2) is integrated between $\Delta t'(\theta_1)$ and $\Delta t''(\theta_2)$ one gets:

$$\ln\frac{\theta_2}{\theta_1} = -\,UA\left(\frac{1}{G_1\,Cp_1} + \frac{1}{G_2\,Cp_2}\right)$$

Or

$$\ln\frac{t_{22}-t_{12}}{t_{21}-t_{11}} = \frac{-\,UA}{G_1\,Cp_1}\left(1+\frac{G_1\,Cp_1}{G_2\,Cp_2}\right) \qquad (3)$$

$$\frac{t_{22}-t_{12}}{t_{21}-t_{11}} = \exp\left(\frac{-\,UA}{G_1\,Cp_1}\left\{1+\frac{G_1\,Cp_1}{G_2\,Cp_2}\right\}\right)$$

$$1-\frac{t_{22}-t_{12}}{t_{21}-t_{11}} = 1-\exp\left(\frac{-\,UA}{G_1\,Cp_1}\left\{1+\frac{G_1\,Cp_1}{G_2\,Cp_2}\right\}\right)$$

$$_{21} - t_{11} - t_{22} + t_{12} \qquad \left(-\,UA \quad \left\{ \quad G_1\,Cp_1 \right\}\right)$$

$$\frac{}{t_{21} - t_{11}} = 1 - \exp \frac{}{G_1\,Cp_1}\ 1 + \frac{}{G_2\,Cp_2}$$

$$\frac{t_{21} - t_{22}) + (t_{12} - t_{11})}{t_{21} - t_{11}} = 1 - \exp\left(\frac{-UA}{G_1\,Cp_1}\left\{1 + \frac{G_1\,Cp_1}{G_2\,Cp_2}\right\}\right)$$

$$(t_{21} - t_{22}) + (t_{12} - t_{11}) = (t_{21} - t_{11})\left(1 - \exp\left\{\frac{-UA}{G_1\,Cp_1}\left\{1 + \frac{G_1\,Cp_1}{G_2\,Cp_2}\right\}\right\}\right) \qquad (4)$$

We know that

$$-G_2\,Cp_2\,(t_{21} - t_{22}) = G_1\,Cp_1\,(t_{11} - t_{12})$$

Or

$$G_2\,Cp_2\,(t_{21} - t_{22}) = G_1\,Cp_1\,(t_{12} - t_{11})$$

$$\text{or}\quad t_{21} - t_{22} = \frac{G_1Cp_1}{G_2Cp_2}(t_{12} - t_{11}) \qquad (5)$$

Substituting the value of $t_{21} - t_{22}$ from Eq. (5) in Eq. (4)

$$(t_{12} - t_{11})\left(1 + \frac{G_1Cp_1}{G_2Cp_2}\right) = (t_{21} - t_{11})\left(1 - \exp\left\{\frac{-UA}{G_1\,Cp_1}\left\{1 + \frac{G_1\,Cp_1}{G_2\,Cp_2}\right\}\right\}\right)$$

$$\frac{t_{12} - t_{11}}{t_{21} - t_{11}} = \frac{1 - \exp\left(\frac{-UA}{G_1\,Cp_1}\left\{1 + \frac{G_1\,Cp_1}{G_2\,Cp_2}\right\}\right)}{1 + \frac{G_1Cp_1}{G_2Cp_2}} \qquad (6)$$

Eq.(6) can be rewritten as

$$\delta t_2 = \Delta t' \frac{1 - \exp\left(\frac{-UA}{G_1\,Cp_1}\left\{1 + \frac{G_1\,Cp_1}{G_2\,Cp_2}\right\}\right)}{1 + \frac{G_2Cp_2}{G_1Cp_1}} \qquad (6a)$$

From Eq.(6a) it follows that the variation in temperature of hot fluid δt_2 is a certain fraction of the inlet temperature drop between the two fluids ($t_{21} - t_{11}$). The expression in Eq. (6a) is a function of

$$\frac{UA}{G_2Cp_2} \quad \text{and} \quad \frac{G_1Cp_1}{G_2Cp_2}$$

Similarly, the temperature variation of the cold fluid can be given as

$$\left(\quad \left(-UA \quad \left\{ \quad G_1Cp_1 \right\}\right)\right)$$

$$\delta t_2 = \Delta t' \frac{G_2Cp_2 \quad 1 - \exp \frac{\quad}{G_1Cp_1} \quad 1 + \frac{\quad}{G_2Cp_2}}{G_1Cp_1 \left(1 + \frac{\quad}{G_1Cp_1}\right) \quad G_2Cp_2} \qquad (6b)$$

If $G_1\ Cp_1 < G_2\ Cp_2$, $\qquad G_1\ Cp_1 = (G\ Cp)_{min}$

TRANSFER UNIT

The concept of transfer unit is useful in the design of heat exchanger, since the magnitude is less dependent on the flow rate of the fluids than the heat transfer coefficient which has been used so far. The number of transfer unit N or NTU is defined by:

$$N = NTU = \frac{UA}{(G\ Cp)_{min}} \qquad (7)$$

Where $(G\ Cp)_{min}$ is the lower of the two values $G_1\ Cp_1$ and $G_2\ Cp_2$. It is the ratio of the heat transferred for a unit temperature driving force to the heat absorbed by the fluid stream when its temperature is changed by 1°C. Thus, the number of transfer units gives a measure of the amount of heat which the heat exchanger can transfer .

From Eq.(4) $N = \dfrac{UA}{G_1\ Cp_1}$

Thus
$$\frac{t_{22} - t_{12}}{t_{21} - t_{11}} = \exp\left\{ - N \left(1 + \frac{G_1\ Cp_1}{G_2\ Cp_2} \right) \right\} \qquad (8)$$

EFFECTIVENESS OF HEAT EXCHANGER

The effectiveness η of a heat exchanger is defined as the actual rate of heat transfer Q to the maximum rate Q_{max} that is thermodynamically possible

$$\eta = \frac{Q}{Q_{max}} \qquad (9)$$

Q_{max} is the heat transfer rate, which would be achieved if it were possible to bring the outlet temperature of the stream with the lower heat capacity to the inlet temperature of other stream. Using the nomenclature given in Fig. 1 and taking stream 1 as having the lower value of G Cp:

$$Q_{max} = G_1\ Cp_1\ (t_{21} - t_{11}) \qquad (10)$$

An overall heat balance gives:

$$Q = G_1 Cp_1 (t_{12} - t_{11}) = -G_2 Cp_2 (t_{21} - t_{22}) \tag{11}$$

Thus based on stream 1:

$$\eta = \frac{G_1 Cp_1 (t_{12} - t_{11})}{G_1 Cp_1 (t_{21} - t_{11})} = \frac{t_{12} - t_{11}}{t_{21} - t_{11}} \tag{12}$$

And based on stream 2:

$$\eta = \frac{G_2 Cp_2 (t_{21} - t_{22})}{G_1 Cp_1 (t_{21} - t_{11})} \tag{13}$$

From equation (6) η base on sream 1

$$\eta = \frac{1 - \exp\left(- N\left\{1 + \dfrac{G_1 Cp_1}{G_2 Cp_2}\right\}\right)}{\dfrac{G_1 Cp_1}{G_2 Cp_2} + 1} \tag{14}$$

Or

$$\eta_{\text{parallel flow}} = \frac{\left(1 - \exp\left(- NTU\left\{1 + \dfrac{(G\,Cp)_{min}}{(GCp)_{max}}\right\}\right)\right)}{\left(1 + \dfrac{(G\,Cp)_{min}}{(G\,Cp)_{max}}\right)} \tag{15}$$

For the particular case where $G_1 Cp_1 = G_2 Cp_2$

$$\eta = \tfrac{1}{2}\left[1 - \exp(-2N)\right] \tag{16}$$

For a very large exchanger ($N \rightarrow \infty$) $\eta = 0.5$

COUNTER CURRENT FLOW Fig. 1(b)

A similar procedure may be followed for counter current flow, but it should be noted that in this case $\theta_1 = t_{11} - t_{22}$ and $\theta_2 = t_{12} - t_{21}$ the corresponding equation for the effectiveness factor η is then:

$$1 - \exp\left\{-N\left(1 - \frac{G_1 Cp_1}{G_2 Cp_2}\right)\right\}$$

$$\eta = \frac{\qquad}{1 - \dfrac{G_1\,Cp_1}{G_2\,Cp_2}\exp\left\{- N\left(1 - \dfrac{G_1\,Cp_1}{G_2\,Cp_2}\right)\right\}} \qquad (17)$$

or

$$\eta_{\text{counter flow}} = \frac{\left(1 - \exp\left(- NTU\left\{1 - \dfrac{(G\,Cp)_{min}}{(GCp)_{max}}\right\}\right)\right)}{\left(1 - \dfrac{(G\,Cp)_{min}}{(G\,Cp)_{max}}\exp\left(1 - NTU\left\{1 - \dfrac{(G\,Cp)_{min}}{(G\,Cp)_{max}}\right\}\right)\right)} \qquad (18)$$

For the case where $G_1\,Cp_1 = G_2\,Cp_2$, it is necessary to expand the exponential terms to give:

$$\eta = \frac{N}{1 + N} \qquad (19)$$

For a very large exchanger ($N \rightarrow \infty$), $\eta \rightarrow 1$ in this case.
From the above discussions it is clear that the effectiveness of counter current flow is more than the parallel flow.

Equation similar to (6a) and (6b) can also be derived for counter-current flow

$$\delta t_2 = \Delta t' \frac{1 - \exp\left(\dfrac{-UA}{G_1Cp_1}\left\{1 + \dfrac{G_1Cp_1}{G_2Cp_2}\right\}\right)}{1 - \dfrac{G_2Cp_2}{G_1Cp_1}\exp\left(\dfrac{-UA}{G_1Cp_1}\left\{1 + \dfrac{G_1Cp_1}{G_2\,Cp_2}\right\}\right)} \qquad (20)$$

$$\delta t_1 = \Delta t' \frac{G_2Cp_2}{G_1Cp_1}\;\frac{1 - \exp\left(\dfrac{-UA}{G_1Cp_1}\left\{1 + \dfrac{G_1Cp_1}{G_2Cp_2}\right\}\right)}{1 - \dfrac{G_2Cp_2}{G_1Cp_1}\exp\left(\dfrac{-UA}{G_1Cp_1}\left\{1 + \dfrac{G_1Cp_1}{G_2\,Cp_2}\right\}\right)} \qquad (21)$$

COMPARISON OF PARALLEL FLOW AND COUNTER FLOW HEAT EXCHANGERS

To evaluate the merits of one flow arrangement over other, it is sufficient to compare the rates of heat transfer in a parallel and counter flow heat exchangers, other conditions being equal. The ratio of, Qp, the quantity of

heat transferred in a parallel-flow exchanger to, Qc, the quantity of heat transferred in counter-flow apparatus, is plotted as a function of

$$\frac{G_2Cp_2}{G_1Cp_1} \text{ and } \frac{UA}{G_2Cp_2} \text{ namely:}$$

$$\frac{Qp}{Qc} = f\left(\frac{G_2Cp_2}{G_1Cp_1}, \frac{UA}{G_2Cp_2}\right)$$

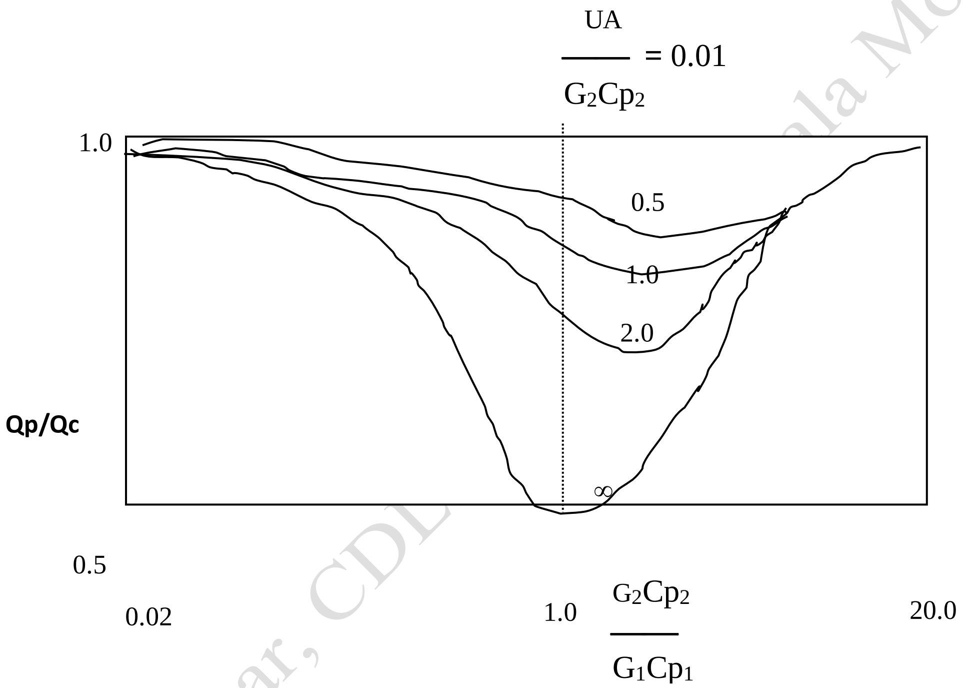


Fig.2: $\frac{Qp}{Qc} = f\left(\frac{G_2Cp_2}{G_1Cp_1}, \frac{UA}{G_2Cp_2}\right)$ **Comparison of parallel and counter flow**

From the graph (Fig.2) it follows that counter and parallel flow can be equivalent only with very large or very small values of G_2Cp_2/ G_1Cp_1 or a small values of the parameter UA/G_2Cp_2. The first condition is characteristic of cases in which the temperature of one of the heat carriers changes little. The second case is that in which the temperature drop is large in comparison with the variation in temperature of the working fluid.

In other cases less heat is transferred in a parallel flow than in a counter flow, other conditions being equal.

Preference, therefore, should be given to counter flow over parallel flow exchanger. It must be kept in mind, however, that if one of the fluid is at a high temperature, then the heat exchanger surface of the counter flow exchanger will be subjected to higher temperature condition that of parallel flow heat exchanger.

2.0 OBJECTIVES:

1. To determine the LMTD and overall heat transfer coefficient of the heat exchanger during parallel and counter current flow configuration.
2. To estimate the effectiveness of the heat exchanger under parallel and counter current flow conditions.

3.0 APPARATUS:

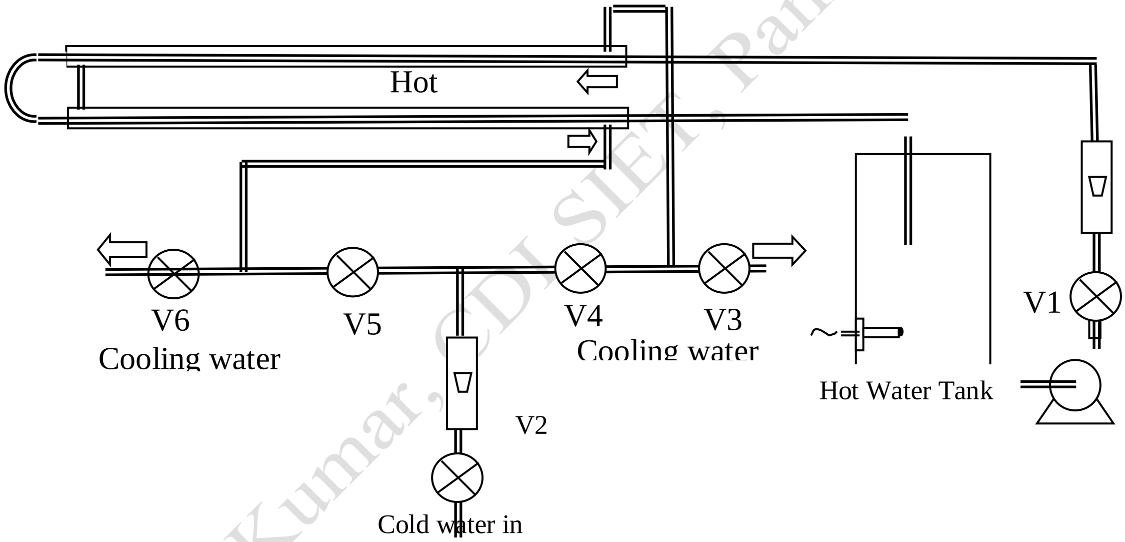


Fig 3: Schematic diagram of Apparatus for Parallel and Counter Current flow in a double pipe heat exchanger

Fig.3 shows the schematic diagram of the experimental set up. Hot water is generated in the hot water tank using an electric heater. The hot water is then pumped to the inner tube using a centrifugal pump and is metered using a rotameter. Hot water flow rate is controlled by valve V1. Cooling water (cw) is taken from water supply line. The flow rate of cw is controlled by the valve V2 and is metered by a rotameter before it is fed to the outer tube of the heat exchanger. Valves V3, V4, V5 & V6 are used to direct the water flow in parallel and counter current directions. Four RTDs (Resistance Temperature Ditector) are used to measure the inlet and outlet temperatures of cooling as well as hot water streams.

4.0 SUGGESTED EXPERIMENTAL WORK:

Step1: **Fill the hot water tank with distilled water (or good quality water).**

Step2: **Start the electric heater.**

Step3: **Adjust cooling water flow in counter-current model by opening the valves V4 and V6 and closing valves V3 and V5.**

Step4: **Adjust the flow rate of hot water in the inner tube using valve V1.**

Step5: **Allow sufficient time for steady state to occur.**

Step6: **At steady state note down the flow rates of cooling as well as hot water along with its inlet and outlet temperatures.**

Step7: **Close valve V4 and V6 and open valve V3 and V5. Now the flow becomes parallel. Repeat Steps 5 & 6.**

Step8: **Change hot water flow rate by manipulating valve V1 and repeat steps 3, 5 & 6 without altering the flow rate of cooling water. Repeat steps 7, 5 and 6.**

Step9: **Repeat step 8 for several values of hot water flow rate.**

Step10: **The whole exercise i.e. Steps 3 to 9 can be repeated for different values of cooling water flow rates.**

5.0 RESULTS & DISCUSSIONS:

5. Tabulate the temperature and flow rate data properly in the data sheet. Note down the ambient temperature.
6. Draw the schematic diagram of the experimental set -up.
7. Compute the LMTD (log mean temperature difference), rate of heat transferred by hot and cold water (Q) and overall heat transfer coefficients (U) for both parallel and counter current flow configurations.
8. Compute the effectiveness factor ’η’ for parallel and counter current flow configurations.
9. Plot the temperature profiles for cold as well as hot water along the length of the heat exchanger. Using Eq. (6a), (6b) and (20) & (21) for parallel and counter current flow configurations.
6. Compare the amount of heat transferred in both the flow configurations and provide reasons for the difference.
7. Mention the sources of error and give suggestions for improvement.

6.0 SAMPLE DATA SHEET:

Name of experiment: **Parallel & Counter Current flow in a double pipe heat exchanger.**

Name of the student: Semester Batch Session

Tube material =

Length of tube, mm =

Inside diameter of inner tube, mm =

Outside diameter of inner tube, mm =

Run. No.	***Counter- Current flow*** ***temperature °C***						***Parallel flow*** temperature °C					
	t_{11}	t_{12}	t_{21}	t_{22}	q_1 (lpm)	q_2 (lpm)	t_{11}	t_{12}	t_{21}	t_{22}	q_1 (lpm)	q_2 (lpm)

7.0 APPENDIX-1: Critical Data of experiments

Tube material = Brass

Length of tube, mm **= 1000 x 2**

Inside diameter of inner tube, mm = 6.3

Outside diameter of inner tube, mm **= 8.0**

Table 1

For Parallel flow, the temperatures t_{11}, t_{12}, t_{21} and t_{22} are represented by thermocouple Nos given below.

Thermocouple No.	**1**	**2**	**3**	**4**
Presentation of Temperature	t_{21}	t_{22}	t_{11}	t_{12}

Table 2

For Counter-current flow, the temperatures t_{11}, t_{12}, t_{21} and t_{22} are represented by thermocouple Nos given below.

Thermocouple No.	**1**	**2**	**3**	**4**

Presentation of Temperature	t_{21}	t_{22}	t_{12}	t_{11}

8.0 APPENDIX-2: Sample Experimental data

Tube material = Brass

Length of tube, mm = 1000 x 2

Inside diameter of inner tube, mm = 6.3

Outside diameter of inner tube, mm = 8.0

Run. No.	***Counter- Current flow temperatures °C***						***Parallel flow*** temperatures °C					
	t_{11}	t_{12}	t_{21}	t_{22}	q_1 (lpm)	q_2 (lpm)	t_{11}	t_{12}	t_{21}	t_{22}	q_1 (lpm)	q_2 (lpm)
1.	35.6	53.9	80.0	60.9	2.0	2.0	34.8	54.1	83.4	65.2	2.0	2.0

9.0 APPENDIX-3: Data Analysis

PARALLEL FLOW

Computation of LMTD, $(\Delta t)_{ln}$

$$(\Delta t)_{ln} = \textbf{logmean temperature difference}$$

$$= \frac{(t_{21}-t_{11}) - (t_{22}-t_{12})}{\ln \dfrac{t_{21}-t_{11}}{t_{22}-t_{12}}}$$

$$= \frac{(83.4 - 34.8) - (65.2 - 54.1)}{\ln \dfrac{(83.4 - 34.8)}{(65.2 - 54.1)}} = \frac{48.6 - 11.1}{\ln \dfrac{48.6}{11.1}}$$

$$= \frac{37.5}{} = 25.34^{o}C$$

Computation of Q

The heat transferred, Q is computed based on the hot stream as it is surrounded by the cold stream and there are less chances of its picking up heat from environment.

Density of water at mean hot water temp., [(83.4 + 65.2)/2 = 74.3°C] = 976.0 kg/m^3 ; (Cp at 72.1°C = 4.189 kJ/kgK)

Q = Volumetric flow rate x density x (temp. difference).

$$Q = \frac{2.0 \times 10^{-3} \times 976.0 \times 4.189 \times 10^{3}\,(83.4 - 65.2)}{60}$$

$$= 2480.33 \text{ W}$$

Computation of U

$$A = \pi\,(0.008) \times 1 = 0.050256 \text{ m}^2$$

$$Q = UA\,(\Delta t)_{ln}$$

$$\text{or } U = \frac{Q}{A\,(\Delta t)_{ln}} = \frac{2480.33}{25.34 \times 0.050256} = 1947.67 \frac{\text{W}}{\text{m}^2\text{K}}$$

$$N = NTU = \frac{UA}{(G\,Cp)_{min}} = \frac{1947.67 \times 0.050256 \times 60}{2.0 \times 10^{-3} \times 976.0 \times 4.189 \times 10^{3}}$$

$$= 0.718$$

η for parallel flow from Eq.(16)

$$\eta_{parallel} = \frac{1}{2}\,[1 - \exp(-2N)]$$

$$\eta_{parallel} = \frac{1}{2}\,[1 - 0.238] = 0.381$$

COUNTER-CURRENT FLOW

Computation of LMTD, $(\Delta t)_{ln}$

$$(\Delta t)_{ln} = \frac{(80.0 - 53.9) - (60.9 - 35.6)}{\ln \dfrac{80.0 - 53.9}{60.9 - 35.6}} = \frac{26.1 - 25.3}{\ln \dfrac{26.1}{25.3}}$$

$$= 0.8/0.0311 = 25.72\ ^{o}C$$

Please note that $(\Delta t)_{ln}$ for counter flow is slightly more than parallel flow

Computation of Q

$$Q = \frac{2.0 \times 10^{-3} \times 976.0 \times 4.189 \times 10^{3} \times (80.0 - 60.9)}{60}$$

$$= 2602.99\ W$$

Computation of U

$$U = \frac{2602.99}{25.72 \times 0.050256} = 2013.79 \frac{W}{m^2K}$$

$$N = NTU = \frac{UA}{(G\ Cp)_{min}} = \frac{2013.79 \times 0.050256 \times 60}{2.0 \times 10^{-3} \times 976.0 \times 4.189 \times 10^{3}}$$

$$= 0.743$$

η for Counter flow Eq. (19), $\eta = \dfrac{NTU}{1 + NTU} = \dfrac{N}{1 + N}$

$$\eta = \frac{0.743}{1+0.743} = 0.426$$

Please note that effectiveness of counter flow is more than parallel flow. However, exchanger surface of the counter

flow exchanger will be subjected to higher temperature condition that of parallel flow heat exchanger.

FORCED CONVECTION

CONTENTS:

26.0 Theory

27.0 Objectives

28.0 Apparatus

29.0 Suggested Experimental Work

30.0 Results & Discussions

31.0 Sample Data Sheet

32.0 Appendix-1: Critical Data of Experiment

33.0 Appendix-2: Sample Experimental data

9.0 Appendix-3: Data Analysis

FORCED CONVECTION

1.0 THEORY:

The rate of transfer of heat to the air flowing over the surface of the hot tube is given by the equation:

$$Q = hA(t_s - t_i) \quad (1)$$

Where Q is heat transfer rate, h is heat transfer coefficient, A is surface area of the tube, t_s is temperature of the tube outer surface and t_i is the temperature of the free air stream. The mode of heat transfer is mainly due to forced convection.

The process of heat transfer for a single tube in infinite cross flow is characterised by a number of peculiarities. Smooth unseparated flow of fluid past the cylinder takes place only at

$$\frac{dV_o\rho}{\mu} < 5$$

Where V_o is velocity of free stream and d is the diameter of tube. With Re>5 the cylinder in cross flow is not a streamline body. The boundary layer forming at the front half of the tube separates from its surface at the rear half and two symmetrical eddies form behind the cylinder. These eddies straighten out at a distance from the cylinder with further increase of Reynolds number. Then eddies separate periodically from the tube and are swept downstream by the fluid flow, forming an eddy path behind the cylinder. Upto Re $\cong 10^3$ the frequency of eddy separation increases, then the region where Reynolds numbers range from 10^3 to 2×10^5 it becomes practically constant and is characterized by the Strouhal Number Sh = fd/V_o= 0.2 where f is frequency. Detailed experimental studies of the circumferential mean rate of heat transfer from a cylinder may be described by following empirical equations:

at $5 < Re < 10^3$

$$Nu = 0.5\, Re^{0.5}\, Pr^{0.38}\, (Pr/P_{rw})^{0.25} \quad (2)$$

at $10^3 < Re < 2 \times 10^5$

$$Nu = 0.25\, Re^{0.6}\, Pr^{0.38}\, (Pr/P_{rw})^{0.25} \quad (3)$$

at $3 \times 10^5 < Re < 2 \times 10^6$

$$Nu = 0.023\, Re^{0.8}\, Pr^{0.37}\, (Pr/P_{rw})^{0.25} \quad (4)$$

Where P_{rw} is Prandtl No. at the wall.

The equations from (2) to (4) apply only to the case where the angle φ formed between the direction of flow and the tube axis and called the angle of attack is 90°. The dependence of the rate of heat transfer in tubes on the angle of attack is of great practical interest. The results of such investigations, carried out by Soviet scientists A.S. Sinelnikov and A. Chashchikhin, are shown in following Fig. Here, the angle of attack φ is plotted along the abscissa and along the ordinate is the value φ which is the ratio of the heat flux at an angle of attack φ to the heat flux at an angle φ = 90°, i.e,

$$\acute{\varepsilon}\varphi = \frac{\alpha\varphi}{\alpha\varphi=90}$$

The curve shows that the φ drops sharply with the decrease in the angle of attack.

The process of heat transfer in prismatic solids of rectangular square, oval and any other cross section is even more intricate than in the case of round cylinders. Here, the already known factors attacking heat transfer are supplemented with a new factor- orientation of the prismatic body in respect to the flow.

The mode in which the fluid flows past such an object and heat transfer depends on the shape of the body and its arrangement in respect to the flow. Therefore, the published literary data may be applied only to geometrically similar bodies. The least deviation may lead to an absurdity, and reliable data may be obtained only by experimentation.

2.0 OBJECTIVES:

1. To determine the convective heat transfer coefficient for forced convection due to flow of air across the heated tube.
2. To compare the experimental result with the theoretical computed results based on empirical relations.

3.0 APPARATUS:

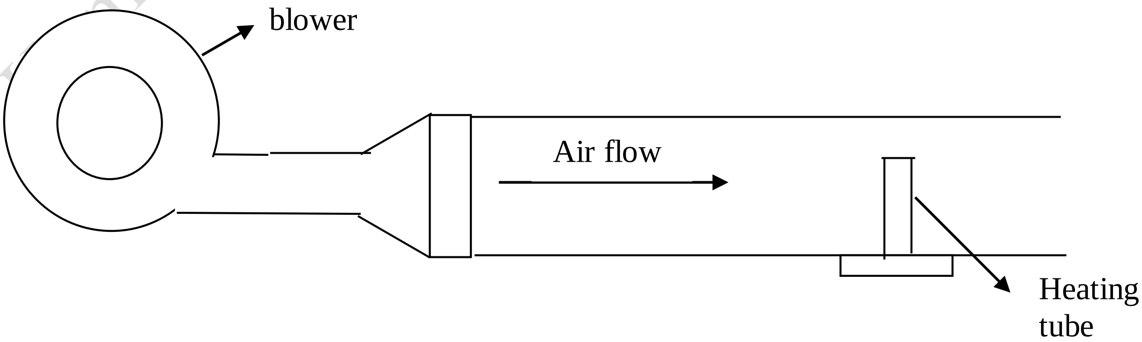


Fig 1: Schematic sketch of apparatus for forced convection

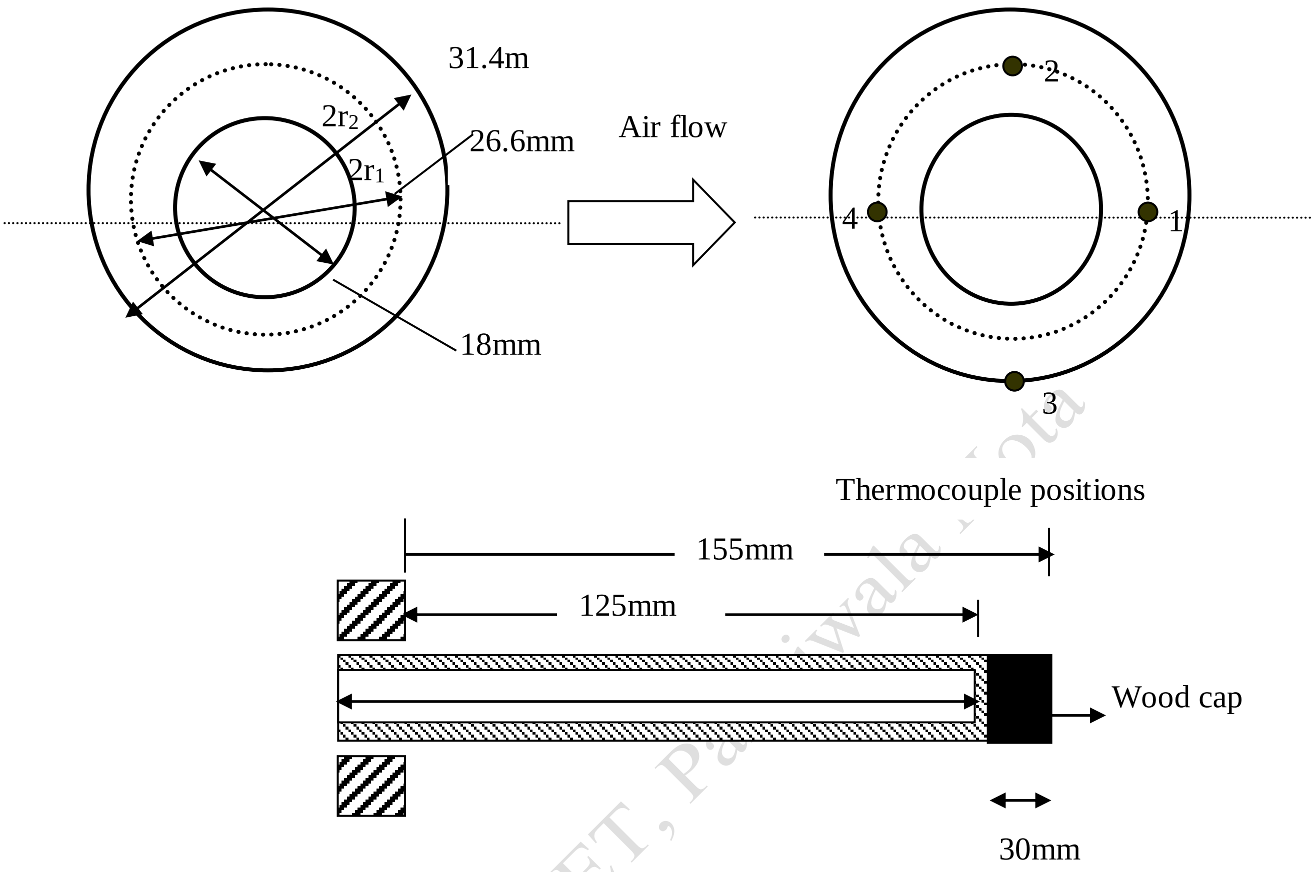


Fig 2: Details of heating tube and thermocouple positions

The experimental set-up is shown in Fig.1. It consists of an air blower, an air duct and a brass heating tube fitted in vertical orientation. The brass tube is heated by an electric heater embeded inside it. Four 3 mm diameter holes are drilled in the annular section of the tube to measure wall temperature. In fact, these measured temperatures are used to predict the outer surface temperature of the brass tube, which is necessary for predicting forced convective heat transfer coefficient. The details of the thermocouple positions are given in Fig.2

4.0 SUGGESTED EXPERIMENTAL WORK:

Step 1: Draw a neat sketch of experimental set up giving important dimensions.

Step 2: Switch ON the air blower.

Step 3: Switch ON the electric heater and set the voltage to 20 V using the variac and allow the steady state to reach. Note down the ammeter readings along with the readings of all the four thermocouples.

Step 4: Change the voltage to 25, 30, 35 and 40 V in steps and repeat step 3 each time you change the voltage.

DO NOTS
• Do not increase voltage more than 50 Volts. • Do not touch the surface of heating tube.

5.0 RESULTS AND DISCUSSIONS:

1. Tabulate the data properly as per the data sheet..
2. Find out the heat transfer coefficient for each energy input (Watts) given to the heater.
3. Estimate the heat transfer coefficient using heat transfer correlation and compare it with experimentally obtained value.

6.0 SAMPLE DATA SHEET

Name of Experiment: **Forced Convection**

Name of the Student: Semester Batch
Session

Orientation =
Material of construction =
Diameter of heating tube, mm =
Length of heated portion, mm =

Depth of thermocouple hole, mm =
Diameter of thermocouple hole, mm =
Ambient air temperature, ^oC =

Run No.	V	I	W	Thermocouple Temperature ^oC				Air Velocity m/s
				1	2	3	4	

7.0 APPENDIX-1: Critical data of experiment

Orientation = Vertical
Material of construction = Brass
Diameter of heating tube, mm = 31.4
Length of heated portion, mm = 125
Depth of thermocouple hole, mm = 71
Diameter of thermocouple hole, mm = 3
Pitch circle diameter of thermocouple holes, mm = 26.6
Details of thermocouples position are given in Fig. 2.

8.0 APPENDIX-2: Sample Experimental data

Orientation = Vertical (Can be set at any angle)

Material of construction = Brass
Diameter of heating tube, mm = 31.4
Length of heated portion, mm = 125
Depth of thermocouple hole, mm = 71
Diameter of thermocouple hole, mm = 3
Pitch circle diameter of thermocouple holes, mm = 26.6
Ambient Temperature = 30.1 ^oC

Run No.	V	I	W	Thermocouple Temperature ^oC				Air Velocity m/s (fixed)
				1	2	3	4	
1.	18.5	0.70	12.95	47.6	47.0	46.5	46.0	4.6

9.0 APPENDIX-3: Data Analysis

Computation for forced convection heat transfer coefficient

Q = VI
Q = 18.5 x 0.70
= 12.95 W
Area of heating tube = π dl
Area = π x 0.0314 x 0.125
= 0.01233 m^2.

Computation of corrected temperature for Thermocouple No.1

$$Q = \frac{2\pi lk\,(t_1-t_2)}{\ln (r_2/r_1)}$$

$$12.95 = \frac{2\,\pi \times 0.125 \times 99.4\,(46.6 - t_2)}{l_n\left(\frac{31.4}{26.6}\right)} = \frac{78.068\,(47.6 - t_2)}{0.16589}$$

NOTE: Thermal conductivity, k, of Brass (70-30) at 0^oC – 96.936 W/ m^0C and 100^0C- 103.68 W/m^0C
(taken from Chemical Engineering hand book by R.H. Perry & Cecil,
page No. 3-220, table No. 3-299)

0.028 = (47.6 - t_2) or t_2 = 47.572^oC

Similarly corrected temperatures for all thermocouples were computed.

Experimental Computation of heat transfer coefficient at first thermocouple position.

$$h_1 = \frac{12.95}{0.01233\,(47.572-30.1)} = 60.113\ \frac{W}{m^2K}$$

Theoretical Computation of heat transfer coefficient for forced convection.

Physical properties of air (taken from Heat Transfer by V.P. Isachenko, V.A. Osipova, A.S. Sukomel, page No. 480) at 300 K.

$$k \text{ of air} = 0.0276\ \frac{W}{mK}$$

$$C_p = 1.005\ \frac{kJ}{kgK}$$

$$\rho = 1.128\ kg/m^3$$

$$\mu = 19.1 \times 10^{-6}\ \frac{Ns}{m^2}$$

$$Pr = 0.699$$

Velocity of air, V = 4.6 m/sec

$$R_e = \frac{dV\rho}{\mu} = \frac{0.0314 \times 4.6 \times 1.128}{19.1 \times 10^{-6}} = 0.008530278 \times 10^6 = 8530.278$$

For $10^3 < Re < 2 \times 10^5$

$$Nu = 0.25\, Re^{0.6}\, P_r^{0.38} \qquad (\text{taking } P_r/P_{rw} = 1.0)$$

$$Nu = 0.25\, (8530.278)^{0.6}\, (0.699)^{0.38}$$

$$= 0.25 \times 228.34 \times 0.873 = 49.84$$

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

$$49.84 = \frac{h \times 0.0314}{0.0276}$$

$$h_{theo} = 43.8\ \frac{W}{m^2K}$$