MECHANICAL ENGINEERING DEPARTMENT
LAB MANUAL SHEETS
HEAT TRANSFER

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Experiment 1

Heat Transfer from a Pin-Fin Apparatus

Aim: To calculate the value of heat transfer coefficient form the fin for natural & forced convection.

Introduction:

Extended surfaces of fins are used to increase the heat transfer rate from a surface to a fluid wherever it is not possible to increase the value of the surface heat transfer coefficient or the temperature difference between the surface and the fluid. The use of this is variety of shapes (refer fig. 1). Circumferential fins around the cylinder of a motor cycle engine and fins attached to condenser tubes of a refrigerator are a few familiar examples.

It is obvious that a fin surface sticks out from the primary heat transfer surface. The temperature difference with surrounding fluid will steadily diminish as one moves out along the fin. The design of the fins therefore required a knowledge of the temperature distribution in the fin. The main objective of this experimental set up is to study temperature distribution in a simple pin fin.

Apparatus:

A brass fin of circular cross section in fitted across a long rectangular duct. The other end of the duct is connected to the suction side of a blower and the air flows past the fin perpendicular to the axis. One end of the fin projects outside the duct and is heated by a heater. Temperature at five points along the length of the fin. The air flow rate is measured by an orifice meter fitted on the delivery side of the blower. Schematic diagram of the set-up is shown in fig. 2, while the details of the pin fin are as per fig. 3.

Theory:

Consider the fin connected at its base to a heated wall and transferring heat to the surroundings. (Refer fig. 4)

Let, A = Cross section area of the fin.

C = Circumference of the fin.

L = Length of the fin.

T_1 = Temp. of the fin at the beginning.

T_f = Duct fluid temperatures.

$\emptyset = (T - T_f)$ = Rise in temperature.

The heat is conducted along the rod and also lost to the surrounding fluid by convection.

Let, h = Heat Transfer coefficient.

K = Thermal conductivity of the fin material.

Applying the first law of thermodynamics to a controlled volume along the length of the fin at X, the resulting equation of heat balance appears as:

$$\frac{d^2 \varnothing}{d x^2} - \frac{h.\ c}{K.\ A} \varnothing = 0 \qquad \ldots\ldots\ldots\ldots\ldots..(1)$$

and the general solution of equation (1) is

$$\varnothing = C_1 . e^{mx} + C^2 . e^{-mx} \qquad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots..(2)$$

$$\text{Where, } m = \sqrt{\frac{h.\ c}{K.\ A}}$$

With the boundary conditions of $\emptyset = \emptyset_1$ at $x = 0$

Where, $\emptyset_1 = T_1 - T_F$ and assuming the fin tip to be insulated.

$\frac{d\varnothing}{dx} = 0$ at $x = L$ results in obtaining eq^n (2) in the form:

$$\frac{\varnothing}{\varnothing_1} \text{ x } \frac{T - T_F}{T_1 - T_F} = \frac{\text{Cosh } m (L - x)}{\text{Cosh } mL} \qquad \ldots\ldots\ldots\ldots\ldots(3)$$

This is the equation for the temperature distribution along the length of the fin. It is seen from the equation that for a fin of given geometry with uniform cross section, the temperature at any point can be calculated by knowing the values of T_1, T_F and X. Temperature T_1 and T_F will be known for a given situation and the value of h depends on whether the heat is lost to the surrounding by free convection or forced convection and can be obtained by using the correlation as given below:

1. For free convection condition,

$$\left.\begin{array}{ll} Nu = 1.1\ (Gr.\ Pr)^{1/6} \ldots & 10^{-1} < Gr.\ Pr. < 10^4 \\ Nu = 0.53\ (Gr.\ Pr)^{1/4} \ldots & 10^4 < Gr.\ Pr. < 10^9 \\ Nu = 0.13\ (Gr.\ Pr)^{1/4} \ldots & 10^9 < Gr.\ Pr. < 10^{12} \end{array}\right\} 4$$

2. For forced convection,

$$\begin{array}{ll} Nu = 0.615\ (R_e)^{0.466} \ldots & 40 < R_e < 4000 \\ Nu = 0.174\ (R_e)^{0.618} \ldots & 4000 < R_e < 40000 \end{array}$$

Where,

$$Nu = \frac{h.\ D}{A_{Air}}$$

$$R_e = \frac{\varrho\ VD}{\nu} = \text{Reynold's Number.}$$

$$G_r = \frac{g.\ \beta.\ D^3\ \Delta T}{\nu^2} = \text{Grashoff Number.}$$

$$P_r = \frac{Cp\ .\ \mu}{K_{Air}} = \text{Prandt 1 Number}$$

All the properties are to be evaluated at the mean film temperature. The mean film temperature is to arithmetic average of the fin temperature and air temperature. Nomenclature:

ϱ = Density of air, Kg / m^3

D = Diameter of pin-fin, m

μ = Dynamic viscosity, $N.sec/m^2$

C_p = Specific heat, KJ/Kg.k

ν = Kinematic viscosity, m2/Sec

K = Thermal conductivity of air, W/m ^{0}C

g = Acceleration due to gravity, $9.81 m/sec^2$

Tm = Average fin temperature

$$= \frac{(T1 + T2 + T4 + T5)}{5}$$

ΔT = $T_m - T_F$

$$T_{mF} = \frac{T_m + T_F}{2}$$

β = Coefficient of thermal expansion

$$= \frac{1}{T_{mF} + 273}$$

ν = Velocity of air in the duct.

The velocity of air can be obtained by calculating the volume flow rate through the duct.

$$Q \quad = Cd\ \frac{\pi}{4} \times d^2 \times \sqrt{2g\left(H.\ \frac{\varrho W}{\varrho_a}\right)}\ \ \frac{m^3}{Sec}$$

Where, H = Difference of levels in manometer, M

ϱW = Density of water 1000 Kg/m^3

ϱ_a = Density of air at T_f

Cd = 0.64

d = Diameter of the orifice = 18mm.

$$\text{Velocity of air at } T_f = \frac{Q}{\text{Duct c/s Area}} = m/sec$$

Use this velocity in the calculation of Re.

The rate of heat transfer from the fin can be calculated as,

$$Q = \sqrt{h.\ c.\ k.\ A}\ \times (T1 - Tf)\ \tanh mL \quad \ldots\ldots\ldots\ldots\ldots\ldots(6)$$

And the effectiveness of the fin can also be calculated as,

$$\eta = \frac{\tanh\ mL}{} \qquad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots(7)$$

Specifications:

1. Duct size = 150mm x 100mm.

2. Diameter of the fin = 12.7mm.

3. Diameter of the orifice = 18mm.

4. Diameter of the delivery pipe = 42mm.

5. Coefficient of discharge (or orifice meter) Cd = 0.64.

6. Centrifugal Blower 1 HP single-phase motor.

7. No. of thermocouples on fin = 5.

(1) to (5) as shown in fig. 3 and indicated on temperature indicator.

8. Thermocouple (6) reads ambient temperature inside of the duct.

9. Thermal conductivity of fin material (Brass) = 110w/m ^{o}C.

10. Temperature indicator = 0 – 300^{o}C with compensation of ambient temperature up to 50^{o}C.

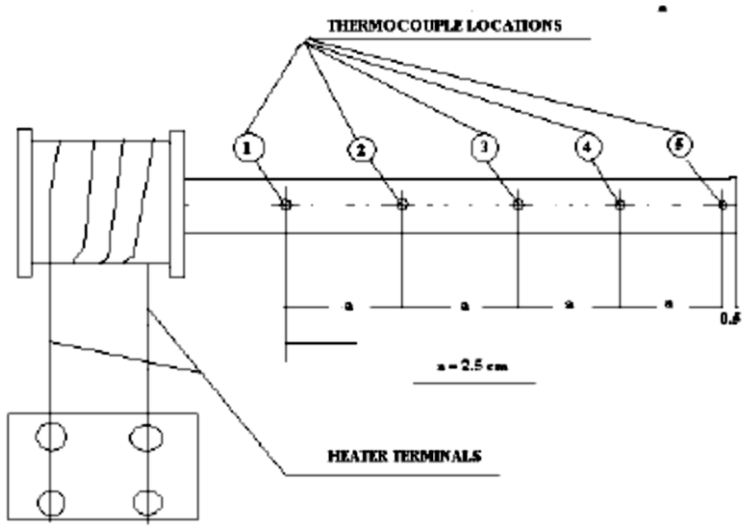
11. Dimmerstat for heat input control 230V, 2 Amps.

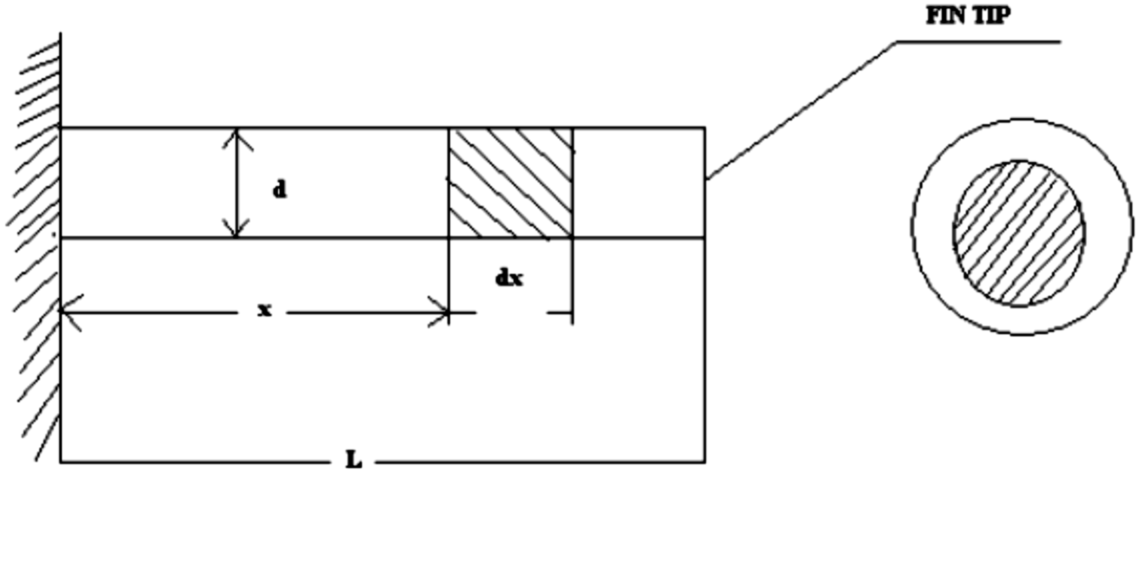
12. Heater suitable for mounting at the fin end outside the duct = 400 watts (Band type).

13. Voltmeter = 0 – 100/200 V.

14. Ammeter = 0 – 2 Amps.

Schematic Diagram





Experimental Procedure:

To study the temperature distribution along the length of a pin fin natural and forced convection, the procedure is as under.

(I) Natural Convection:

1. Start heating the fin by switching ON the heater element and adjust the voltage on dimmerstat to say 80 volt (Increase slowly from 0 to onwards) Note down the thermocouple reading 1 to 5.
2. When steady state is reached, record the final readings 1 to 5 and also record the ambient temperature reading 6.
3. Repeat the same experiment with voltage 100 volts and 120 volts.

(II) Forced Convection:

1. Start heating the fin by switching ON the heater and adjust dimmerstat voltage equal to 100 volts.
2. Start the blower and adjust the difference of level in the manometer with the help of gate valve.
3. Note down the thermocouple readings (1) to (5) at a time interval of 5 minutes.
4. When the steady state is reached, record the final reading (1) to (5) and also record the ambient temperature reading (6).
5. Repeat the same experiment with different manometer readings.

Precautions:

1. See that the dimmerstat is at zero position before switching ON the heater.
2. Operate the changeover switch of temperature indicator, gently.
3. Be sure that the steady state is reached before taking the final reading.
4. See that throughout the experiment, the blower is OFF.

Observation Table:

I) Natural Convection:

Sr. No.	V Volts	I Amps	Fin Temperatures					Ambient Temp.
			T_1 (^{0}C)	T_2 (^{0}C)	T_2 (^{0}C)	T_4 (^{0}C)	T_5 (^{0}C)	$T_6 = T_f$ (^{0}C)

II) FORCED CONVECTION:

Sr. No	V Volts	I Amps	Manometer reading (Cm.)	Fin Temperatures					Ambient Temp
				T_1 (^{0}C)	T_2 (^{0}C)	T_2 (^{0}C)	T_4 (^{0}C)	T_5 (^{0}C)	$T_6 = T_f$ (^{0}C)

Results from Experiments:

I) Natural Convection:

1. Plot the temperature distribution along the length of the fin from readings (refer fig. 5).

2. Calculate Gr and Pr and obtain Nu form equation (4) and finally get the value of ‘h’ in natural convection.

3. Calculate the value of ‘m’ and obtain the temperature at various locations along the length of the fin by using equation (3) and plot them (refer fig. 5).

4. Calculate the value of heat transfer rate from the fin effectiveness by using equation (6) and equation (7).

5. Repeat the same procedure for all other sets.

II) Forced Convection:

1. plot the temperature distribution along the length of the fin from observed readings (refer fig. 6).

2. Calculate the value of ‘m’ and obtain the temperature at various locations along the length of fin by using equation (3) and plot them. (Refer fig. 6).

3. Calculate Re and Pr and obtain Nu from equation (5).

4. Calculate the value of heat transfer rate from the fin and fin effectiveness by using equation (6) and equation (7).

5. Repeat the same procedure for all other sets of observations.

Conclusion:

Heat transfer coefficient form the fin for natural & forced convection is found out to be ------------

Experiment 2

Heat Transfer through Composite Wall

Aim:

1. To determine total thermal resistance and thermal conductivity of composite wall.
2. To plot temperature gradient along composite wall structure.

Description:

The apparatus consists of a central heater sandwiched between two sheets. Three types of slabs are provided both sides of heater, which forms a composite structure. A small hand press frame is provided to ensure the perfect contact between the slabs. A dimmerstat is provided for varying the input to the heater and measurement of input is carried out by a voltmeter, ammeter.

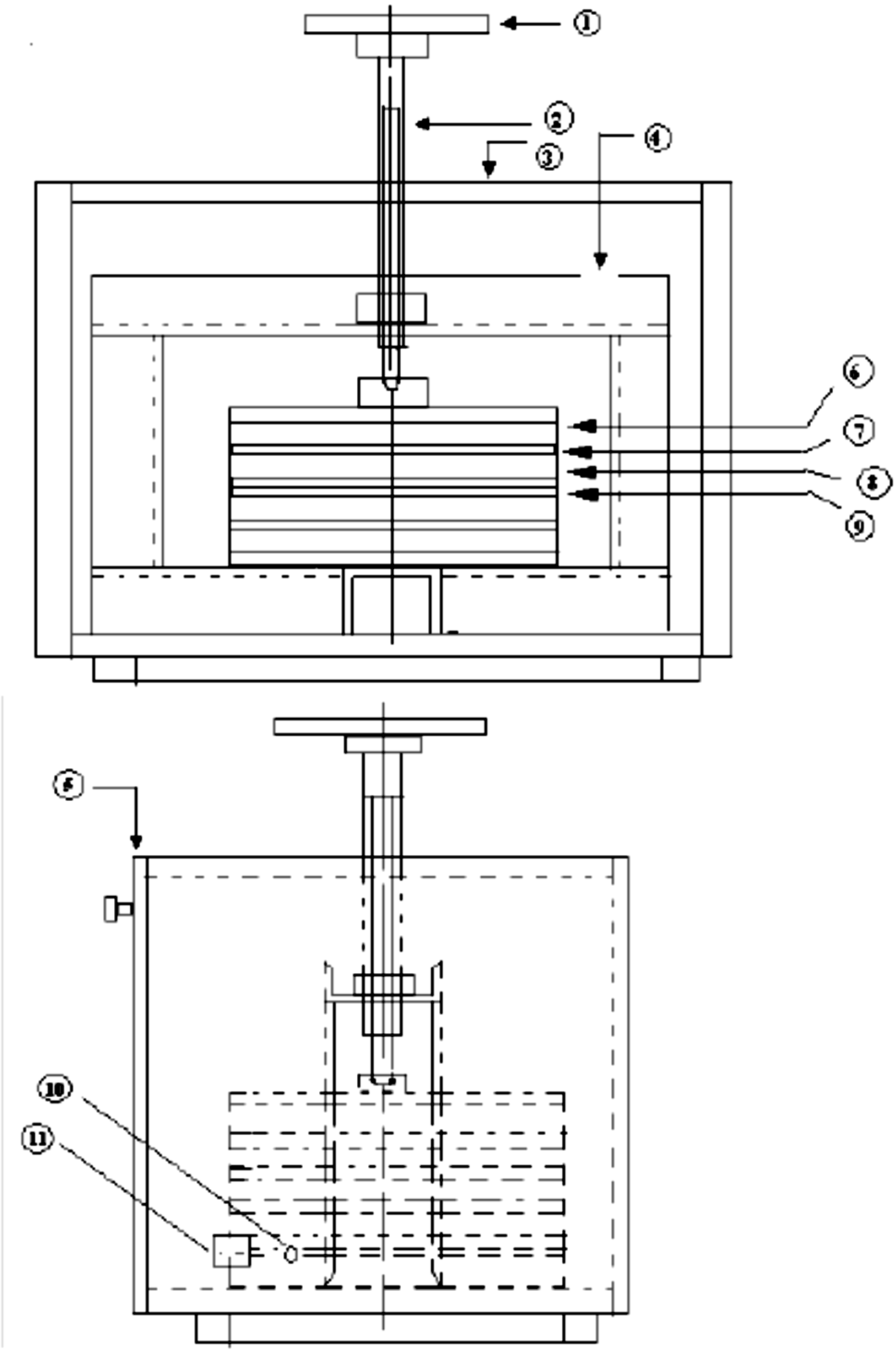
Thermocouples are embedded between interfaces of the slabs, to read the temperature at the surface.

The experiments can be conducted at various values of input and calculation can be made accordingly.

Specifications:

1. Slab assembly arranged symmetrically on both sides of heater.
2. Heater: Nochrome heater wound on mica former and insulation with control unit capacity 300 watt maximum.
3. Heater Control Unit: 0-230V. Ammeter 0-2Amps. Single phase dimmerstat (1No.).
4. Voltmeter 0-100-200V. Ammeter 0-2Amps.
5. Temperature Indicator (digital type): 0-200oC. Service required – A. C. single phase 230 V. earthed electric supply.

Schematic Diagram:



(1) Hand Wheel (2) Screw (3) Cabinet (4) Fabricated Frame (5) Acrylic Sheet (6) Press Wood Plate (7) Bakelite Plate (8) C.I. Plate (9) Heater (10) Heater Cable (11) Thermocouple Socket 12 Way

T1 To T6 Thermocouple Positions

Procedure: Arrange the plates in proper fashion (symmetrical) on both sides of the Heater plates.

1. See that plates are symmetrically arranged on both sides of the heater plates.
2. Operate the hand press properly to ensure perfect contact between the plates.
3. Close the box by cover sheet to achieve steady environmental conditions.
4. Start the supply of heater by varying the dimmerstat; adjust the input at the desired value.
5. Take readings of all the thermocouples at an interval of 10 minutes until fairly steady temperatures are achieved and rate of rise is negligible.
6. Note down the reading in observation table.

Observations and observations table:

Composite slabs: 1. Wall thickness:

a. Cast iron =

b. Hylam =

c. Wood =

2. Slab diameter = 300mm.

	SET I	SET II	SET III
READINGS 1.Voltmeter V (Volts)			
2.Ammeter I (Amps)			
Heat supplied = 0.86 VI (in MKS units) = VI (SI units)			
Thermocouple Reading ^{0}C			
T1			
T2			
T3			
T4			
T5			
T6			
T7			
T8			

Mean Readings:

$$T_A = \frac{(T1 + T2)}{2}$$

$$T_B = \frac{(T3 + T4)}{2}$$

$$T_C = \frac{(T5 + T6)}{2}$$

$$T_D = \frac{(T7 + T8)}{2}$$

Calculations:

Read the Heat supplied Q = V x I Watts (In S. I. Units) For calculating the thermal conductivity of composite walls, it is assumed that due to large diameter of the plates, heat flowing through central portion is unidirectional i. e. axial flow. Thus for calculation, central half diameter area where unidirectional flow is assumed is considered. Accordingly, thermocouples are fixed at close to center of the plates.

$$\text{Now } q = \text{Heat flux} = \frac{Q}{A} \quad [W / m^2]$$

Where A = π 4 x d^2 = half dia. of plates.

1. Total thermal resistance of composite slab

$$R \text{ total} = \frac{(T_A - T_D)}{q}$$

2. Thermal conductivity of composite slab.

$$K \text{ composite} = \frac{q \times b}{(T_A - T_D)}$$

b = Total thickness of composite slab.

3. To plot thickness of slab material against temperature gradient.

Conclusion:

1) Total Thermal resistance to found out to be --------------------

Experiment 3

Critical Heat Flux

Aim: To visualized the pool boiling over the heater wire in different regions up to the critical heat flux point at which the wire melts.

Introduction:

When heat is added to a liquid from a submerged solid surface, which is at a temperature higher than the saturation temperature of the liquid, it is usual for a part of the liquid to change phase. This change of phase is called boiling.

Boiling is of various types, the type depends upon the temperature difference the surface and the liquid. The different types are indicated in which a typical experimental boiling curve obtained in a saturated pool of liquid is down.

Description:

The apparatus consists of a cylindrical glass container housing and the test heater (Nichrome wire). Test heater is connected also to mains via a dimmer. An ammeter is connected in series while a voltmeter across it to read the current and voltage. The glass container is kept on a stand, which is fixed on a metallic platform. There is provision of illuminating the test heater wire with the help of a lamp projecting light from back and the heater wire can be viewed through a lens.

This experimental set up is designed to study the pool-boiling phenomenon up to critical heat flux point. The pool boiling over the heater wire can be visualized in the different regions up to the critical heat flux point at which the wire melts. The heat flux from the wire is slowly increased by gradually increasing the applied voltage across the test wire and the change over from natural convection to nucleate boiling can be seen.

The formation of bubbles and their growth in size and number can be visualized followed by the vigorous bubble formation and their immediate carrying over to surface and ending this in the braking of wire indicating the occurrence of critical heat flux point.

Specification:

- Glass container **:** Dia. 186.5 mm. & Height 97mm

• Nichrome wire size **:** 0.135 ϕ mm)

• Dimmer stat **:** 10 Amp, 230 volts.

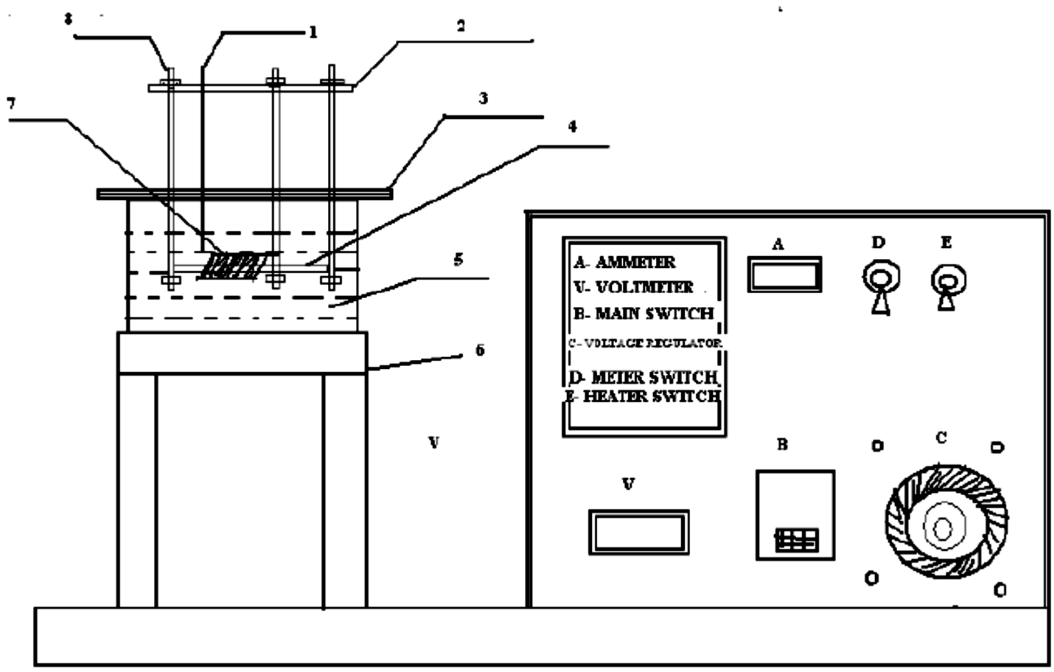
• Voltmeter **:** 0 to 75 V

• Ammeter **:** 0 to 10 AMP

• Thermometer **:** 0 to 100°

• Nichrome wire resistance **:** 6.4 ohms.

Schematic Diagram:



Theory:

The heat flux supplied to the surface is plotted against (Tw - Ts) the difference between the temperature of the surface and the saturation temperature of the liquid. It is

seen that the boiling curve can be divided into three regions:

- ✓ Natural Convection Region
- ✓ Nucleate Boiling Region
- ✓ Film Boiling Region

The region of natural convection occurs at low temperature differences (of the order of 10^{o}C or less). Heat transfer from the heated surface to a liquid in its vicinity causes the liquid to be superheated.

The superheated liquid rises to the free liquid surface by natural convection, where vapour is produced by evaporation. As the temperature difference (Tw – Ts) is increased, nucleate boiling starts. In this region, it is observed that bubbles start to form at certain locations on the heated surface.

Region II consists of two parts. In the first part, II – a, the bubbles formed are very few in number. They condense in the liquid and do not reach the free surface. In the second part, II – b, the rate of bubbles formation and the number of locations where they are formed increase. Some of the bubbles now rise all the way to the free surface. With increasing temperature difference, a stage is finally reached when the rate of formation of bubbles is so high, that they start to coalesce and blanket the surface with a vapour film. This is the beginning of the region III viz film boiling.

In the first part of this region III-a, the vapour film is unstable, so that the film boiling may be occurring on a portion of the heated surface area, while nucleate boiling may be occurring on the remaining area. In the second part, III-b, a stable film covers the entire surface. The temperature difference in this region is of the order of 1000°C and consequently radiative heat transfer across the vapour film is also significant.

It will be observed that the heat flux does not increase in a regular manner with the temperature difference. In region I, the heat flux is proportional to $(Tw – Ts)^{n}$, where 'n' is slightly greater than unity. When the transition from natural convection to nucleate boiling occurs the heat flux starts to increase more rapidly with temperature difference, the value of n increasing to about 3.at the end of region II, the boiling curve reaches a peak. Beyond this, in the region II-A, in spite of increasing temperature difference, the heat flow increases with the formation of a vapour film. The heat flux passes through a minimum at the end of region III-a. it starts to increase again with (Tw – Ts) only when stable film boiling begins and radiation becomes increasingly important.

It is of interest to note how the temperature of the heating surface changes as the heat flux is steadily increased from zero. Up to point A, natural convection boiling and nucleate boiling occur and the temperature of the heating surface is obtained by reading off the value of (Tw – Ts) from the boiling curve and adding to it the value of Ts.

If the heat flux is increased even a little beyond the value of A, the temperature of the surface will shoot up to the value corresponding to the point C. it is apparent from figure 1 that the surface temperature corresponding to point C is high.

For most surfaces, it is high enough to cause the material to melt. Thus in most practical situations, it is undesirable to exceed the value of heat flux corresponding to point A. This value is therefore of considerable engineering significance and is called the critical or peak heat flux. The pool-boiling curve as described above is known as Nukiyam pool Boiling Curve. The discussions so far has been concerned with the various type of boiling which occur in saturated pool boiling. If the liquid is below the saturation temperature we say that sub-cooled pool boiling is taking place. Also in many practical situations, e.g. steam generators; one is interested in boiling in a liquid flowing through tubes. This is called forced convection boiling, may also be saturated or sub-cooled and of the nucleate or film type.

Thus in order to completely specify boiling occurring in any process, one must state

- ✓ Whether it is forced convection boiling or pool boiling,
- ✓ Whether the liquid is saturated or sub cooled, and
- ✓ Whether it is in the natural convection nucleate or film boiling region.

Procedure:

- Fill the tank with water.
- Dip the Nichrome wire into the water and make the electrical connections
- Note the current reading in steps of 1 amp till a maximum current of 10 ampere.
- Between each reading the time interval of two min is allowed for steady state to establish.
- Water temperature is noted with a thermometer at the beginning and at the end of the experiment. The average of these two is taken as the bulk liquid average temperature.

Observations:

- ✓ d = Diameter of test heater wire, = 0.135×10^{-3} mtr

- ✓ L= Length of the test heater = 0.088 mtr
- ✓ A = Surface area = πdL = 3.7322 x 10^{-5} m^2

Observations Table:

Sr. No.	Water / Bulk Temp T in ^{0}C	Voltage (V)	Ampere (I)
1			
2			

Calculation:

- **Q = heater power in Watts**

$$Q = V \times I \text{ Watts}$$

- **Q = critical heat flux in w / m^2**

$$q = \frac{Q}{A} W/m^2$$

Precautions:

- Keep the various to zero voltage position before starting the experiment.
- Take sufficient amount of distilled water in the container so that both the heaters are completely immersed.
- Connect the test heater wire across the studs tightly.
- Do not touch the water or terminal points after putting the switch in on position.
- Very gently operate the various in steps and allow sufficient time in between.
- After the attainment of critical heat flux condition, slowly decrease the voltage and bring it to zero.

Conclusion:

Heater wire in different regions up to the critical heat flux point at which the wire melts is found out to be -----------

Experiment 4

Emissivity Measurement Apparatus

Aim: To determine Emissivity of non-black test plate surface.

Theory-

Under steady state conditions:

Let - W1 = Heater input black plate.

Watts = V1 I1

W2 = Heater input to test plate.

Watts = V2 I2

$$A = \text{Area of plates} = \frac{2nd^2}{4}\ m^2$$

d = Diam. Of plate = 160mm

T_b = Temperature of black plate ^0K

T_a = Ambient temperature ^0K

E_b = Emissivity of black plate.
(To be assumed equal to unity.)

E = Emissivity of non-black test plate

σ = Stefan Boltzmann constant.

MKS = 4.876×10^{-8} Kcal/ hr-m^2 – $^0K^4$ (In MKS units)

SI = 5.67×10^{-8} w/m^2 K^4 (In SI units)

By using the Stefan Boltzmann Law:

$$(W_1 - W_2) = (E_b - E)\,\sigma A\,(T_s^4 - T_d^4)/0.86$$

In SI Units

$$(W_1 - W_2) = (E_b - E)\,\sigma A\,(T_s^4 - T_d^4)$$

Specifications:

1. Test Plate = Ø 165mm
2. Black Plate = Ø 165mm Material Aluminium.
3. Heater for (1) Nichrome strip wound on mica sheet and sandwiched between two mica sheets.
4. Heater for (2) as above capacity of heater = 200 watts each approx.
5. Dimmerstat for (1) 0 – 2A, 0 – 260V

6. Dimmerstat for (2) 0 – 2A, 0 – 260V

7. Voltmeter 0 – 100 – 200V, Ammeter 0 – 2 Amp.

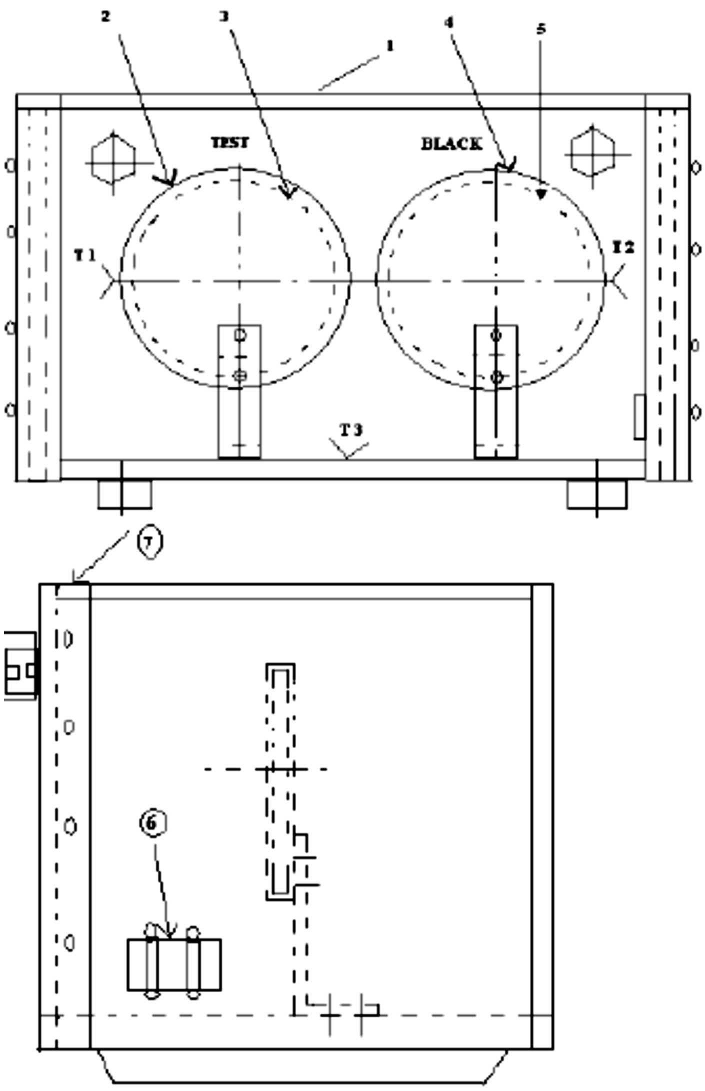
8. Enclosure size 580mm x 300mm x 300mm approximately with one side of perpex sheet.

9. Thermocouples – Choromel Alumel – (3Nos).

10. Temperature indicator 0 – 300^{o}C.

11. D. P. D. T. Switch.

Schematic Diagram:



1) Enclosure (2) Test Plate (3) Test Plate Heater (4) Black Plate (5) Black Plate Heater (6) Thermocouple Socket (7) Acrylic Cover

T 1 to T 3 Thermocouple Position

Procedure:

1. Gradually increase the input to the heater to black plate and adjust it to some value viz. 30, 50, 75 watts and adjust the heater input to test plate slightly less than the black plate 27, 35, 55 watts etc.
2. Check the temperature of the two plates with small time intervals and adjust the input of test plate only, by the dimmerstat so that the two plates will be maintained at the same temperature.
3. This will required some trial and error and one has to wait sufficiently (more than one hour or so) to obtain the steady state condition.
4. After attaining the steady state condition record the temperatures. Voltmeter and Ammeter readings for both the plates.
5. The same procedure is repeated for various surface temperatures in increasing order.

Observation Table:

Sr. No.	BLACK PLATE			TEST PLATE			ENCLOSURE TEMP.
	V1	I1	T_b	V_2	I_2	T_s	$T_a\ ^0C$

For SI Unit:

$$(W_b - W_s) = (E_b - E_s)\ \sigma\ A\ (T_s^4 - T_b^4)$$

Calculations:

$$qb = \sigma A\ E_b\ (T_s^4 - T_D^4)$$
$$qb = \sigma E\ A\ (T_s^4 - T_D^4)$$

Where,

qb = heat input to disc coated with lamp black watt.
In SI Unit qb = $V_1\ I_1$ Watts = W_b x 0.86 $V_1\ I_1 = W_b$

q_s = heat input to Specimen disc. (Kcal / hr) = W_s = 0.86
In SI unit $q_s = V_2 I_2$ Watts
= Stefan Boltzmann Constant = 4.876×10^{-8} Kcal/hr $m^2\ {}^0K^4$

In SI unit $\sigma = 5.67 \times 10^{-8}\ W/M^2\ K^4$
E = Emissivity of specimen to be determined (absorption)

In SI unit $(W_b - W_s) = (E_b - E)\ \sigma.A\ (T_s^4 - T_a^4)$

This fact could be verified by performing the experiments at various values of Ts and E can be plotted in a graph in a graph as shown in fig. 4.

Conclusion:-

Emissivity of non-black test plate surface is found out to be ----

Experiment 5

Heat Transfer through the Lagged Pipe

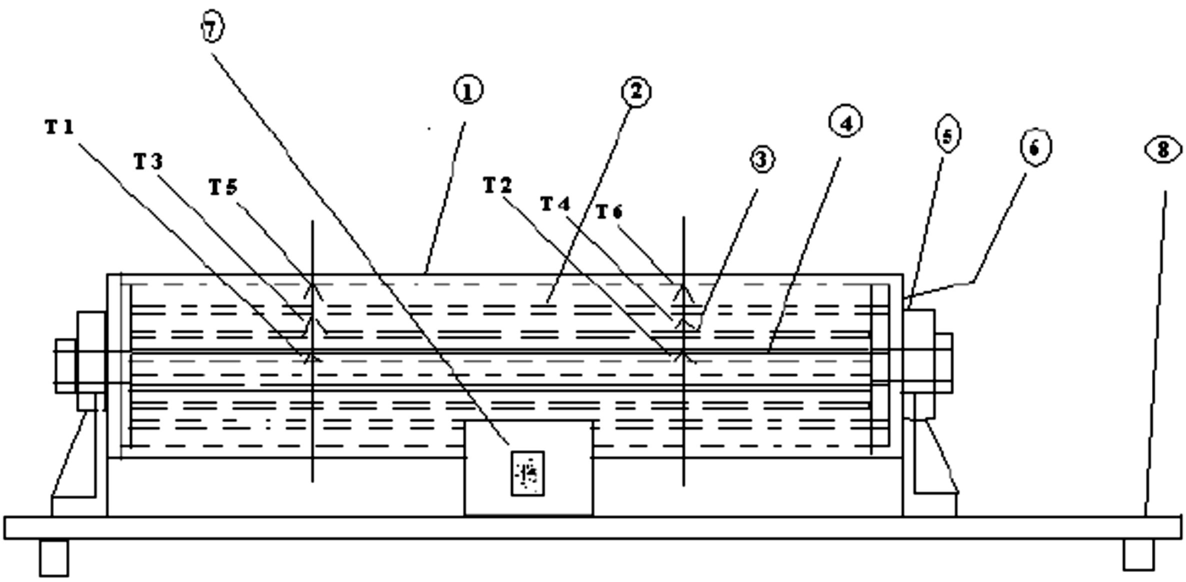
Aim:

1. To determine heat flow rate through the lagged pipe and compare it with the heater for known value of thermal conductivity of lagging material.
2. To determine the thermal conductivity of lagging material by assuming the heater input to be the heat flow rate through lagged pipe.

Description:

The apparatus consists of three concentric pipes mounted on suitable stand. The inside pipe consists of a heater, which is wound with Nichrome wire on the insulation. Between first two cylinders the insulating material with which lagging is to be done is filled compactly. Between second and third cylinders another material used for lagging is filled. The thermocouples are attached to the surface of cylinders approximately to measure the temperatures. The input to the heater is varied through a dimmerstat and measured on voltmeter and ammeter. The experiments can be conducted at various values of input and calculations can be made accordingly. Similarly the experiments can be made for double or single lagging removing appropriate pipes.

Schematic Diagram:



1) Outer 2) Middle Pipe 3) Inside Pipe 4) Heater 5) Support 6) Connection Strir 7) Thermocouple Socket 8) Board T1 to T6 Thermocouple Position

Procedure:

Arrange the pipes in proper fashion with heater assembly (Arranged normally). Fill the lagging material in pipes uniformly and by gentle pushing, press the lagging material (filled normally).

See that material gets packed uniformly.

Close both ends of pipes and keep the assembly on stands.

Start the supply of heater and by varying dimmerstat adjusts the input for desired value (Range 60 to 120 watts) by using voltmeter and ammeter.

Take readings of all the 6 thermocouples at an interval of 5 minutes until the steady state is reached.

Note down steady readings in observation table.

Limits and Precautions:

Keep dimmer stat to zero position before start.

Increase voltage gradually.

Keep the assembly undisturbed while testing.

While removing or changing the lagging material, do not disturb the thermocouples.

Do not increase power above 100 Watts.

Operate selector switch of temperature indicator gently.

Observations:

Di = Inner Diameter of pipe = 58.2 mm = 0.0582 m

Dm = Middle Diameter of pipe = 105 mm = 0.105 m

Do = Outer Diameter of pipe = 156 mm = 0.156 m

L = Length of Pipe = 0.890 m

Observation Table:

Sr. No.	Voltmeter V	Ammeter I	Thermocouple Readings in ^0C					
			T_1	T_2	T_3	T_4	T_5	T_6
1								
2								

Calculations:

Qa = Actual heat Input in Watts

Qa ct = V x I

Where,

V = Voltage in volts

I = Current in Amperes

Mean temperature readings in ^{o}C

$$\mathrm{T\ (inside)} = \frac{T_1 + T_2}{2} \quad ^{o}C$$

$$\mathrm{T\ (middle)} = \frac{T_3 + T_4}{2} \quad ^{o}C$$

$$\mathrm{T\ (outer)} = \frac{T_5 + T_6}{2} \quad ^{o}C$$

K_1 = Thermal Conductivity of Inner material in W / m K

$$\mathrm{K_1} = \frac{\mathrm{Q_a} \times \ln(\mathrm{r_m}/\mathrm{r_i})}{2 \times \pi \times \mathrm{L}(\mathrm{T_i} - \mathrm{T_m})}$$

Where,

r_i = Radius of Inner Pipe in meter

r_m = Radius of middle pipe in meter

L= Length of Pipe in meter

K_2 = Thermal Conductivity of Outer material in W / m K

$$\mathrm{K_1} = \frac{\mathrm{Q_a} \times \ln(\mathrm{r_o}/\mathrm{r_m})}{2 \times \pi \times \mathrm{L}(\mathrm{T_m} - \mathrm{T_o})}$$

Where, r_o = Radius of Outer Pipe in meter

K = Thermal Conductivity of Combined Lagging Material in W / m K

$$\mathrm{K_1} = \frac{\mathrm{r_i} \times \ln(\mathrm{r_o}/\mathrm{r_i})}{[r_m/K_1 \times \ln(r_m/r_i)] + [r_o/K_2 \times \ln(r_o/r_m)]}$$

Q = Flow Rate of Heat Transfer in Watts

$$\mathbf{K} = \frac{2 \times \pi \times L\ (T_i - T_o)}{[\ \ln (r_m / r_i) / K_1\ ] + [\ \ln (r_o / r_m) / K_2]}$$

Conclusion:

1) Heat flow rate through the lagged pipe and compare it with the heater for known value of thermal conductivity of lagging material is found out to be -----

2) Thermal conductivity of lagging material by assuming the heater input to be the heat flow rate through lagged pipe is found out to be ------

Experiment 6

Thermal Conductivity of Insulating Powder

Aim: To find out the thermal conductivity of power.

Description:

The apparatus consists of two thin walled concentric copper spheres. The inner sphere houses the heating coil. The insulating powder (Asbestos powder – Lagging Material) is packed between the two shells. The powder supply to the heating coil is by using a dimmerstat and is measured by Voltmeter and Ammeter. Choromel Alumel thermocouples are used to measure the temperatures. Thermocouples (1) to (4) are embedded on inner sphere and (5) to (10) are as shown in the fig. Temperature readings in turn enable to find out the Thermal Conductivity of the insulating powder as an isotropic material and the value of Thermal Conductivity can be determined.

Consider the transfer of heat by conduction through the wall of a hollow sphere formed by the insulating powdered layer packed between two thin copper spheres (Ref. Fig. 1)

Let, ri = Radius of inner sphere in meters.

ro = Radius of outer sphere in meters.

Ti = Average Temperature of the inner sphere in ^oC

To = Average Temperature of the outer sphere in ^oC

Where,

$$Ti = \frac{T1 + T2 + T3 + T4}{4}$$

and

$$To = \frac{T5 + T6 + T7 + T8 + T9 + T10}{6}$$

Note that T1 to T10 denote the temperature of thermocouples (1) to (10).

Form the experimental values of q, Ti and To the unknown thermal conductivity K can be determined as:

$$K = \frac{q\,(ro - ri)}{4\,\pi ri \times ro\ (Ti + To)}$$

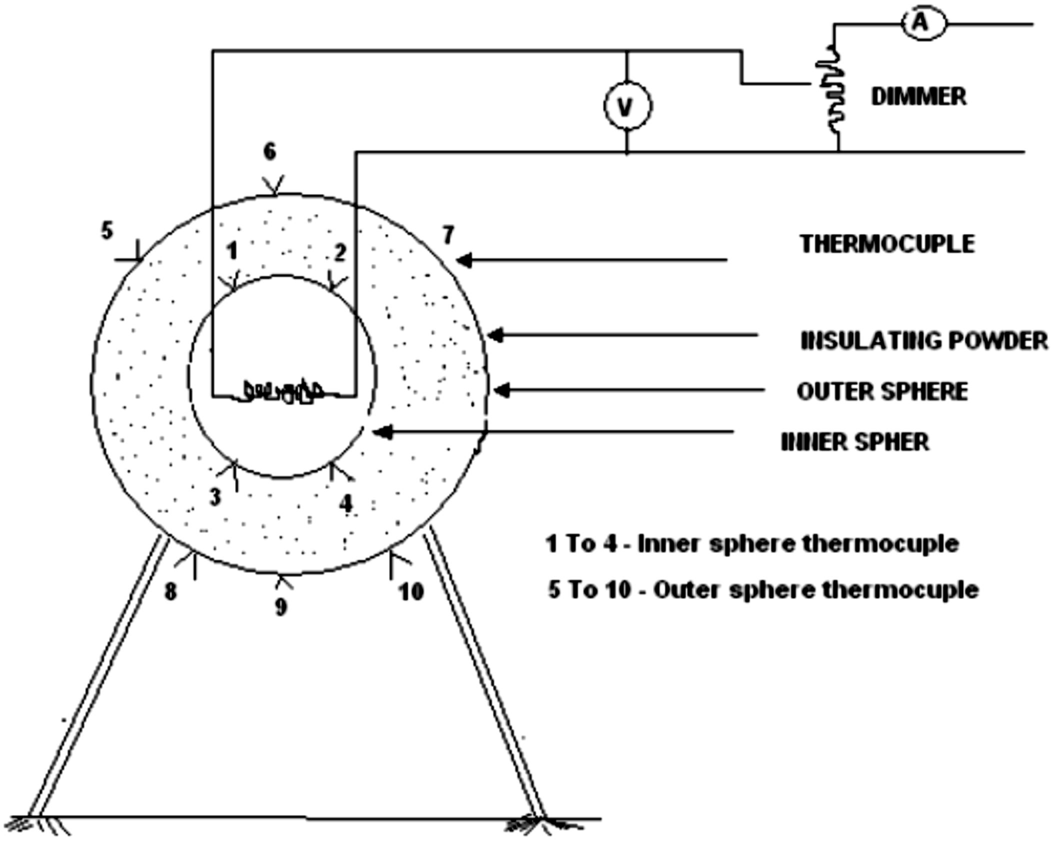
Specifications:

1. Radius of the inner copper sphere, ri = 50mm
2. Radius of the outer copper sphere, ro = 100mm
3. Voltmeter (0 – 100 – 200 V).
4. Ammeter (0 – 2 Amps.)
5. Temperature Indicator 0 – 300 ^{o}C calibrated for chromel alumel.
6. Dimmerstat 0 – 2A, 0 – 230 V.
7. Heater coil - Strip Heating Element sandwiched between mica sheets – 200 watts.
8. Chromel Alumel Thermocouples – No. (1) to (4) embedded on inner sphere to measure Ti.
9. Chromel Alumel Thermocouples – No. (5) to (10) embedded on outer sphere to measure To.
10. Insulating Powder – Asbestos magnesia commercially available powder and packed between the two spheres.

Experimental Procedure:

1. Start main switch of control panel.
2. Increase slowly the input to heater by the dimmerstat starting from zero volt position.
3. Adjust input equal to 40 Watts Max. by Voltmeter and Ammeter. Wattage W = VI
4. See that this input remains constant throughout the experiment.
5. Wait till fairly steady state condition is reached. This can be checked by reading temperatures of thermocouples (1) to (10) and note changes in their readings with time.
6. Note down the readings in the observations table as given below :

Schematic Diagram:



Observation Table:

1. Voltmeter reading (V) = Volts.
2. Ammeter reading (I) = Amps.
3. Heater input (VI) = Watts.

Inner Sphere:

Thermocouple No.	1	2	3	4	
	T1	T2	T3	T4	Mean Temp. Ti $Ti = \dfrac{T1 + T2 + T3 + T4}{4}$
Temp. ^{0}C					

Outer Sphere:

Thermocouple No.	5	6	7	8	9	10	
	T5	T6	T7	T8	T9	T10	Mean Temp. Ti $Ti = \dfrac{T5 + T6 + \ldots + T10}{4}$
Temp. ^0C							

Calculation:

W = V x I Watts.

Ti = Inner sphere mean temp. ^oC

To = Outer sphere mean temp. ^oC

ri = Radius of inner copper sphere = 50 mm.

ro = Radius of outer copper sphere = 100 mm.

Using Equation :

q = 0.86 W Kcal/hr (In MKS units)

$$K = \frac{0.86W\,(ro - ri)}{4\pi ri \times ro\,(Ti - To)}$$

q = V x I w / m – k (In SI units)

$$K = \frac{q\,(ro - ri)}{4\pi ri \times ro\,(Ti - To)}$$

Conclusion:

Thermal conductivity of powder is found out to be ----

Experiment 7

Thermal Conductivity of Metal Rod

Aim : To determine thermal conductivity of metal rod.

Introduction :

Thermal conductivity is the physical property of the material denoting the ease with a particular substance can accomplish the transmission of thermal energy by molecular motion.

Thermal conductivity of material is found to depend on the chemical composition of the substance or substance of which it is a composed, the phase (i. e. gas, liquid or solid) in which it exists, its crystalline structure if a solid, the temperature and pressure to which it is subjected, and whether or not it is a homogeneous material.

Table 1 lists the values of thermal conductivity for some common metal :

Metal	Thermal Conductivity kcal / hr – m - ^{0}c	State
SOLID'S Pure Copper	330	20 degree
Brass	95	- - do - -
Steel (0.5%C)	46	- - do - -
S. S.	14	- - do --

Mechanism of Thermal Energy Conduction In Metals:

Thermal energy may be conducted in solids by two modes :

1. Lattice Vibration.
2. Transport by free electrons.

In good electrical conductors a rather large number of free electrons move about in the lattice structure of the material. Just as these electrons may transport electric charge, they may also carry thermal energy from a high temperature region to a low temperature region. In fact, these electrons are frequently referred as the electron gas. Energy may also be transmitted as vibrational energy in the lattice structure of the material. In general, however, this latter mode of energy transfer is not as large as the electrons transport and it is for this reason that good electrical conductors are almost always good heat conductor viz. Copper, Aluminium and silver. With increase in the temperature, however the increased lattice vibrations come in the way of the transport by free electrons for most of the pure metals the thermal conductivity decreases with increase in the temperature.

Fig. 1 shows the trend of vibration of thermal conductivity with temperature for some metals.

Apparatus:

The experimental set up consists of the metal bar, one end of which is heated by an electric heater while the other end of the bar projects inside the cooling water jacket. The middle portion of the bar is surrounded by a cylindrical shell filled with the asbestos insulating powder. The temperature of the bar is measured at eight different sections { Fig. 2 (1) to (4) } while the radial temperature distribution is measured by separate thermocouples at two different sections in the insulating shell.

The heater is provided with a dimmerstat for controlling the heat input. Water under constant heat condition is circulated through the jacket and its flow rate and temperature rise are noted.

Specification:

1. Length of the metal bar (total) : 410 mm
2. Size of the metal bar (diameter) : 25 mm
3. Test length of the bar : 200mm
4. No. of thermocouple mounted on the Bar (Positions are shown fig. 2) : 9
5. No. of thermocouples in the insulation shell (shown in fig. 2) : 2
6. Heater coil (Bald type) : Nichrome.
7. Water jacket diameter : 80mm
8. Temperature indicator, 13 channel : 200 Degree.
9. Dimmerstat for heater coil : 2A / 230 V.
10. Voltmeter 0 to 300 volts.
11. Ammeter 0 to 2 Amps.

12. Measuring flask for water flow rate.

13. Stop clock.

Theory :

The heater will heat the bar at its end and heat will be conducted through the bar to other end.

After attaining the steady state Heat flowing out of bar.

Heat flowing out of bar = Heat gained by water

$$Q_w = m_w \times Cp_w \times (T_{out} - T_{in}) = m_w Cp_w(\Delta T_w) = m_w Cp_w(T_{out} - T_{in})$$

Where, m_w : Mass flow rate of the cooling water In Kg / hr

C_p : Specific Heat of water (Given 1)

T : $(T_{out} - T_{in})$ for water

T : $(T_{out} - T_{in})$ for water

Thermal Conductivity of Bar

1. Heat Conducted through the Bar (Q)

$$Q = Qw + \frac{2\pi \; KL\,(To - T\,I)}{\text{Log e}\,\{ro / ri\}}$$

Where, Qw : Heat conducted through water

K : Thermal conductivity of Asbestos powder is 0.3 Kcal / hr – mdegree

ro & ri : Radial distance of thermocouple in insulating shell.

2. Thermal conductivity of Bar (K)

$$Q = K \{dt / dx\} \times A$$

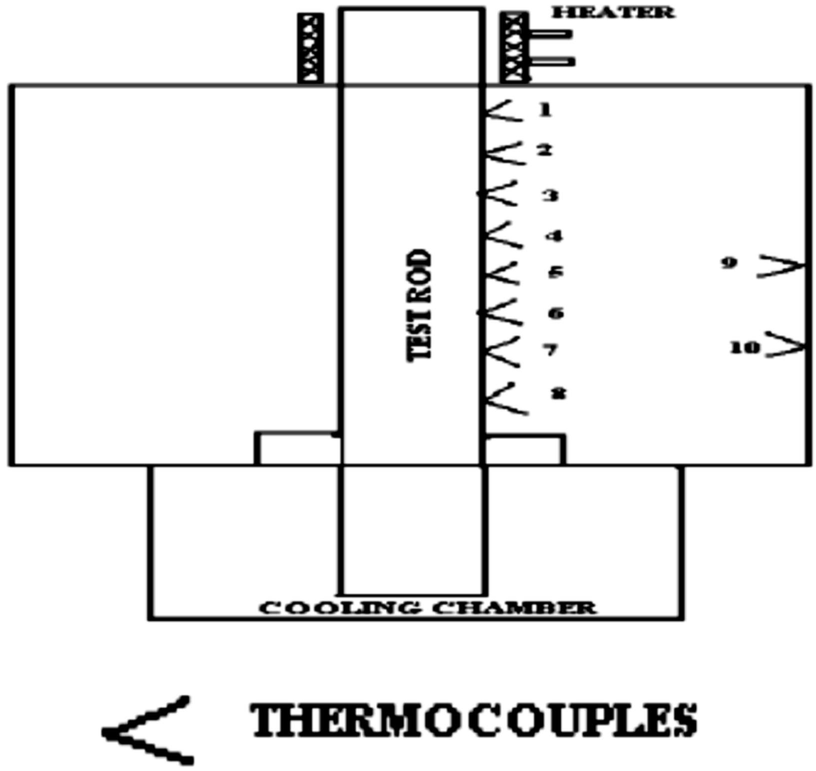
Where, dt : Change in temperature. (T1 – T9)

dx : Length across temperature. (0.2)

A : Area of the bar (p / 4 x d2).

$$\pi/4 \times (0.025)2 = 4.9 \times 10^{-4}\ m^2$$

Schematic Diagram:



Procedure:

1. Start the electric supply.

2. Adjust the temperature in the temperature indicator by means of rotating the knob for compensation of temperature equal to room temperature. (Normally this is per adjusted).

3. Give input to the heater by slowly rotating the dimmerstat and adjust it to voltage equal to 80 V, 120 V etc.

4. Start the cooling water supply through the jacket and adjust it about 350cc per minute.

5. Go on checking the temperature at some specified time interval say 5 minute and continue this till a satisfactory steady state condition is reached.

6. Note the temperature reading 1 to 13.

7. Note the mass flow rate of water in Kg/minute and temperature rise in it.

Observation Table :

Sr. No.	Mass Flow Rate in Kg/Min	Temperature in Degree Centigrade T1, T2, T3, T4,..........................T13
1. 2. 3. 4. 5.		

Observations:

Mass flow rate of water (m) : Kg/min

Water inlet temperature (T12) : Degree Centigrade

Water outlet temperature (T13) : Degree Centigrade

Rod Temperature (T1 to T9) : Degree Centigrade

Radial distance of Thermocouples (ro) : 40mm

in insulating shell. (ri) : 25mm

Specific heat of water (Cp) : 1 Kcal/Kg^{o}K = 4.186 KJ/KgK

Thermal conductivity of Asbestos powder (K) : 0.3 Kcal/hr-m-^{o}C

0.3 x 4.18 KJ/KgK

Length of bar (L) : 200mm

Demeter of bar (d) : 50mm

Area of the bar (A) : 4.9 x 10^{-4} m^2

Plot the temperature distribution along the length of the bar using observed values.

Calculations:

1. Heat flowing out of bar. Qbar = Qw

Qw = m x Cp x (ΔTw) (Kcal/hr)

Where, m : Mass flow rate of the cooling water In Kg/hr

Cp :Specific Heat of water (Given 1)

ΔTw : (Tout – Tin) for water

2. Heat conducted through the Bar (Q)

$$Q = Qw + \frac{2n\ KL\ (T10 - T11)}{Log\ e\ \{ro / ri\}}\ (Kcal / Hr)$$

Where, Qw : Heat conducted through water

K : Thermal conductivity of Asbestos powder is

0.3 Kcal/ hr-m-degree.

ro & ri : Radial distance of thermocouple in insulating shell.

3. Thermal conductivity of Bar (K)

$Q = K \{dt/dx\} \times A\ (Kcal/Hr\text{-}m\text{-}^{o}C)$

Where, dt : Change in temperature. (T1 – T9)

dx : Length Across temperature. (0.2)

A : Area of the bar (n/4 x d2).

$n/4 \times (0.025)2 = 4.9 \times 10^{-4}\ m^{2}$

Conclusion:-

1) Thermal conductivity of metal rod is found out to be -----

Experiment 8

Heat Transfer in Natural Convection

Aim: To determine the surface heat transfer coefficient for a vertical tube losing heat by natural convection.

Theory:-

When a hot body is kept in still atmosphere, heat is transferred to the surrounding fluid by natural convection. The fluid layer in contact with the hot body gets heated; rise up due to the decrease in its density and the cold fluid rushes in to take place. The process is continuous and the heat transfer takes place due to the relative motion of hot cold fluid particles.

The heat transfer coefficient is given by:

$$h = \frac{q - q_1}{As \times (Ts - Ta)} \qquad \text{.............................(1)}$$

Where, h = Average surface heat transfer coefficient ($W/m^2\ {}^oC$)

q = Heat transfer rate (Watts)

As = Area of heat transferring surface = π. d. l (m^2)

Ts = Average surface temperature

$$= \frac{(T1 + T2 + T3 + T4 + T5 + T6 + T7)}{7}\ {}^oC$$

Ta = Ambient temperature in the duct = T8 ^oC

$$q1 = \text{Heat loss by radiation} = \sigma \,.\, A.\ \varepsilon \,.\, (T_s^4 - T_a^4)$$

Where, σ = Stefan Boltzmann constant = $5.667 \times 10^{-8}\ W/m^2.K^4$

A = Surface area of pipe = (0.05966) m^2 = πDL

ε = Emissive of pipe material = 0.6

T_s and T_a Surface and ambient temperatures in ^oK respectively

The surface heat transfer coefficient, of a system transferring heat by natural convection depends on the shape, dimensions and orientation of the fluid and the temperature difference between heat transferring surface and the fluid. The dependence of 'h' on all the above-mentioned parameters is generally expressed in terms on Nondimensional groups as follow:

$$\frac{h \times L}{k} = A \times \left[\frac{g.\ L^3.\ \beta.\ \Delta T}{v^2} \times \frac{Cp.\ \mu}{k} \right]^n \quad ....(2)$$

Where, $\frac{h \times L}{k}$ .... is called the Nusselt number,

$\frac{g.L^3.\ \beta.\ \Delta T}{v^2}$ .... is called to Grashof Number and

$\frac{Cp.\ \mu}{k}$ .... is the Prandtl Number

A and n are constants depending on the shape and orientation of the heat transferring surface.

Where, L = A characteristic dimension of the surface.

K = Thermal conductivity of fluid.

v = Kinematics Viscosity of fluid.

μ = Dynamic Viscosity of fluid.

Cp = Specific heat of fluid.

β = Coefficient of volumetric expansion for the fluid.

g = Acceleration due to gravity.

$\Delta T = [Ts - Ta]$

$$\text{For gases, } \beta = \frac{1}{(T_f + 273)} \ / \ ^0k$$

$$T_f = \frac{(Ts + Ta)}{2}$$

For a vertical cylinder losing heat by natural convection, the constant A and n of equation(2) have been determined and the following empirical correlation obtained.

$$\frac{h \times L}{k} = 0.59(Gr.\,Pr)^{0.25} \text{ for } 10^4 < Gr.\,Pr < 10^8 \qquad \ldots\ldots\ldots\ldots(3)$$

$$hL/k = 0.13(Gr.\,Pr)^{1/3} \text{ for } 10^8 < Gr.\,Pr < 10^{12} \ldots\ldots\ldots\ldots \qquad (4)$$

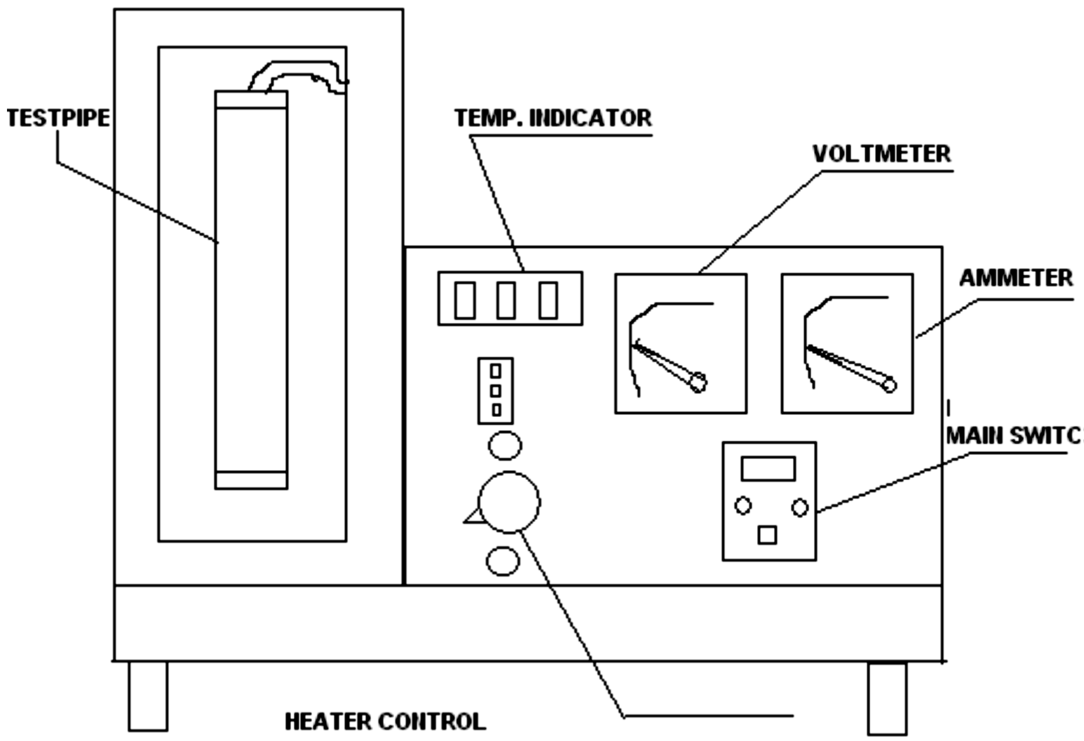
L = Length of the cylinder.

All the properties of the determined at the mean film temperature (Tf).

Specification:

1. Diameter of the tube (d) = 38 mm
2. Length of tube (L) = 500 mm
3. Duct size 200mm x 200mm x 800mm. Length
4. Multichannel Digital Temperature Indicator 0 – 300 ^oC using Chromel / Alumel thermocouple.
5. Ammeter 0 – 2 Amp. and Voltmeter 0 – 200 Volts.
6. Dimmerstat 2 Amp. 240 Volts.

Schematic Diagram:



Procedure:

1. Put ON the supply and adjust the dimmerstat to obtain the required heat input (Say 40W, 60W, 70W etc.)

2. Wait till the steady state is reached, which is confirmed from temperature reading (T_1 to T_7)

3. Measure surface temperature at the various point i.e. T_1 to T_7.

4. Note the ambient temperature i.e. T_8.

5. Repeat the experiment at different heat inputs **(Do not exceed 80w)**.

Observations

1. O. D. Cylinder = 38mm.

2. Length of cylinder = 500mm.

3. Input to heater = V x I Watts.

Observations Table:

Sr. No.	Volt	Amp	TEMPERATURE, ^{0}C							
			T1	T2	T3	T4	T5	T6	T7	T8

Calculations:

1) Calculate the value of average surface heat transfer coefficient, neglecting end losses using equation (1).

2) Calculate and plot (fig. 4) the variation of local heat transfer coefficient along the length of the tube using:

$$T = T_1 \text{ to } T_7 \text{ and } h = \frac{q}{As\,(Ts + Ta)}$$

3) Compare the experimentally obtained value with the prediction of the correlation equations (3) or (4).

Note – The heat loss due to radiation and conduction is not considered, but they are present, which give different between actual and theoretical values.

Results and Discussion:

Some typical results are shown in fig. 4 and 2 different heater inputs. The heat transfer coefficient is having a maximum value at the beginning as expected because of the just starting of the building of the boundary layer and it decreases as expected in the upward direction due to thickening of layer and which is laminar one. This trend is maintained up to half of the length (approx.) and beyond that there is little variation and turbulent boundary layers. The last point shown somewhat increase in the value of heat transfer coefficient which is attributed to end loss causing a temperature drop.

The comparison of average heat transfer coefficient is also made with predicted values are somewhat less than experimental values due to the heat loss by radiation.

$$\therefore \textbf{ Heat loss by radiation} = \sigma.\ A.\ \varepsilon.\ (T_s^4 - T_a^4)$$

Where, s = Stefan Boltzmann constant = 5.667×10^{-8} W/ m^2. K^4

A = Surface area of pipe = 0.59 m^2

e = Emissivity of pipe material = 0.6

T_s and T_a = Surface and ambient temperature in K respectively.

Conclusion:-

Heat coefficient for a vertical tube losing heat by natural convection is found out to be …..

Experiment 9

Parallel Flow / Counter Flow Heat Exchanger

Aim: To determine heat transfer rate and overall heat transfer coefficient of Parallel flow and counter flow heat exchanger.

Introduction:

Heat exchanger is a device used for affecting the process of heat exchange between two fluids that are at different temperatures. It is useful in many engineering processes like those in Refrigeration and Air conditioning system, power system, food processing systems, chemical reactor and space or aeronautical applications. The necessity for doing this arises in multitude of industrial applications. Common examples of best exchangers are the radiator of a car, the condenser at the back of the domestic refrigerator, and the steam boiler of a thermal power plant.

Description & Construction:

The simple example of transfer type of heat exchanger can be in the form of a tube in tube type arrangement as shown in the figure. One fluid flowing through the inner tube and the other through the annulus surroundings it. The heat transfer takes place across the walls of the inner tube. The experiments are conducted by keeping the identical flow rates [approx.] while running the unit as a parallel flow heat exchanger and counter flow exchanger.

The temperatures are measured with the help of the temperature sensor. The readings are recorded when steady state is reached. The outer tube is provided with adequate insulation to minimize the heat losses.

The PF & CF heat exchanger consist of following components:

1. Main Frame
2. Heat Exchanger
3. Temperature Indicator
4. Hot water Generator
5. Rotameter for hot & cold water flow rate measurement
6. Temperature Sensors

The total assembly is supported on a main frame. The apparatus consists of a ‘tube in tube’ type concentric tube heat exchanger. The hot fluid is water, which is obtained from the hot water generator it is attached at the bottom of assembly to supply the hot fluid i.e., water with the help of pump through the inner tube while the cold fluid is flowing through annulus. Pump set is connected to the hot water generator to suck the water from it & deliver as per requirement. Different valves are provided in the system to regulate the flow of fluid to the system. The hot water & cold water admitted at the same end & the opposite end, named parallel & counter flow heat exchanger accordingly, is done by valve operation.

The concentric type heat exchanger is connected in system, which transfers thermal energy between two fluids at different temperature.

Specification:

Inner Tube Material : copper

Outer Diameter (do) : 12.5 mm

Inner Diameter (di) : 10.6 mm

Outer Tube Material : G.I.

Inner Diameter (Di) : 33 mm

Outer Diameter (Do) : 29 mm

Length of the heat (L)

Exchanger : 1600 mm

Heater : 3kw x 01 No.

Thermostat : 1 (Range10-110^{o}C)

Temperature Indicator : 6 Channel (0 to 200^{o}C), 0.1 ^{o}C Resolution

MCB : 16 Amp for Heater, 6 Amp for Pump

Type of Pump : ¼ HP, 230 VAC (optional)

Type of Heat Exchangers:

Heat exchangers are classified in three categories.

1. Transfer Type According to flow arrangement
2. Parallel flow
3. Counter flow

4. Cross flow Storage Type

1. Direct Transfer Type

2. Shell and tube heat Exchanger

3. Concentric tube Heat Exchanger

A Transfer type heat exchanger is the one in which both fluids pass simultaneously flow through the device and heat is transferred through separating walls. In practice most of the heat exchangers used are transfer type ones. The transfer type heat exchangers are further classified according to flow arrangements as Parallel Flow, in which fluids flow in the same direction.

Counter flow, in which they flow in opposite direction.

Procedure:

1. Make all connections as shown in the fig. & check for any leakage in the circuit.
2. Make the oil well at the places where thermocouples are inserted for sensing the temperature of water.
3. Set the temperature of the heater tank to some fix temp say around 55 to 60 ^{o}C.
4. Once the temperature of hot water is reached start the flow of water through hot and cold water side and adjust it as per requirement.
5. For Parallel flow the flow of hot & cold water should be on same side & for counter flow the flow of both the fluids should be on opposite side. Make this adjustment with the help of valves
6. Wait to stabilized the temperature on the indicator.
7. As the temperature get stabilized take down the readings for different four channels by using switch on the panel.
8. Readings for the flow rates can be taken from the Rotameter attached at the front of the instrument.
9. Take down the readings by varying the flow rates.
10. Observe flow rate of hot water to be less than flow rate of cold water.
11. Once the experiment is completed drain the water remains in concentric tube. By opening the cocks given at side & below the shell.

Precautions:

1. Do not put on heater unless water flow is continuous.
2. Once the flow is fixed, do not change it until note down the readings for that flow.
3. The thermocouples should keep in pockets.
4. There should make the oil well in pockets of thermocouple.
5. Equipment should be earthed prop.
6. Once the experiment is completed drain out the water remain in both the tubes.

Observation:

Given data:

Inner Tube

Inner tube material = Copper

Outer Diameter (do) = 12.5 mm = 0.0125 m

Inner Diameter (di) = 10.6 mm = 0.0106 m

Outer Tube

Outer tube Material = G.I.

Inner Diameter (Di) = 33 mm = 0.033 m

Outer Diameter (Do) = 29 mm = 0.029 m

Length of the heat (L)

Exchanger = 1600 mm = 1.6 m

Constants:

1. Cpc = Specific heat of cold water = 4.174 KJ / KG k
2. Cph = Specific heat of hot water = 4.174 KJ / KG k

Observation Table – I:

Parallel Flow Run:

SR. NO.	HOT WATER SIDE			COLD WATER SIDE		
	Flow rate m_h. (Kg/hr)	Inlet Temp. Thi (°C)	Outlet Temp. Tho (°C)	Flow rate m_c. (Kg/hr)	Inlet Temp. Tci (°C)	Outlet Temp. Tco (°C)

Note: Tci in parallel is becoming Tco in counter Flow while making necessary Correction.

Calculation for Parallel Flow:

1. Qh = Heat transfer rate from hot water in KJ / sec
2. Qh = mh x Cph x (Thi – Tho)
3. Qc = Heat transfer rate from cold water in KJ /sec
4. Qc = mc x Cpc x (Tco – Tci)
5. Q = Total heat transfer rate in KJ /sec

$$Q = \frac{Qh + Qc}{2}$$

ΔTm = Logarithmic mean temperature difference in ^{o}K

$$\Delta Tm = \frac{T_{in} - T_{out}}{ln(T_i/T_o)}$$

Where,

T_{in} = Thi - Tci in ^{o}C

T_{out} = Tho - Tco in ^{o}C

A_i = Area of inner side tube in m^2

$A_i = \pi \times di \times L$

Where,

d_i = diameter of inner tube in meter

A_o = Area of outer side tube in m^2

$A_o = \pi \times d_o \times L$

Where,

d_o = diameter of outer tube in meter

U_o = Overall heat transfer coefficient based on outer area in W/m^2 K

$$U_o = \frac{Qc}{A_o \times \Delta Tm}$$

U_i = Overall heat transfer coefficient based on inner area in W/m^2 K

$$U_i = \frac{Qh}{A_i \times \Delta Tm}$$

C = Capacity ratio

$$C = \frac{Cmin}{Cmax}$$

Where,

$Cmin = m_h \times Cp_h$

$Cmax = m_C \times Cp_C$

ε = Effectiveness of heat exchanger

$$\varepsilon = \frac{mc \times Cpc \times (Tco - Tci)}{(mCp)min \times (Thi - Tci)} = \frac{mh \times Cph \times (Thi - Tho)}{(mCp)min \times (Thi - Tci)}$$

When, (mCp) min = mh × Cph < mc × Cpc

$$\varepsilon = \frac{\text{Thi} - \text{Tho}}{\text{Thi} - \text{Tci}}$$

(mCp) min = mc × Cpc < mh × Cph ,

$$\varepsilon = \frac{\text{Tco} - \text{Tci}}{\text{Thi} - \text{Tci}}$$

NTU(i) = No. of transfer unit for inner surface

$$\text{NTU (i)} = \frac{\text{Ui} \times \text{Ai}}{\text{Cmin}}$$

NTUo = No. of transfer unit for outer surface

$$\text{NTU (o)} = \frac{\text{Uo} \times \text{Ao}}{\text{Cmin}}$$

Observation Table – II:

Counter Flow Run :

SR. NO.	Hot Water Side			Cold Water Side		
	Flow rate m_h. (Kg/hr)	Inlet Temp. Thi (°C)	Outlet Temp. Tho (°C)	Flow rate m_c. (Kg/hr)	Inlet Temp. Tci (°C)	Outlet Temp. Tco (°C)

Note: Tci in parallel is becoming Tco in counter Flow while making necessary Correction.

Calculation For Counter Flow:

Qh = Heat transfer rate from hot water in KJ / sec

$Qh = m_h \times Cp_h \times (Thi - Tho)$

Qc = Heat transfer rate from cold water in KJ /sec

Qc = mc x Cpc x (Tco – Tci)

Q = Total heat transfer rate in KJ /sec

$$\mathrm{Q} = \frac{\mathrm{Qh} + \mathrm{Qc}}{2}$$

ΔTm = Logarithmic mean temperature difference in ^{o}K

$$\Delta Tm = \frac{\mathrm{Tin} - \mathrm{Tout}}{\ln(\mathrm{Ti}/\mathrm{To})}$$

Where,

Tin = Thi - Tci in ^{o}C

Tout = Tho - Tco in ^{o}C

Ai = Area of inner side tube in m^2

$$Ai = \pi \times di \times L$$

Where,

di = diameter of inner tube in meter

Ao = Area of outer side tube in m^2

$$Ao = \pi \times do \times L$$

Where,

do = diameter of outer tube in meter

Uo = Overall heat transfer coefficient based on outer area in W/m^2 K

$$\mathrm{Uo} = \frac{\mathrm{Qc}}{\mathrm{Ao} + \Delta\mathrm{Tm}}$$

Ui = Overall heat transfer coefficient based on inner area in W/m^2 K

$$\mathrm{Ui} = \frac{\mathrm{Qh}}{\mathrm{Ai} + \Delta\mathrm{Tm}}$$

ε = Effectiveness of heat exchanger

$$\varepsilon = \frac{mc \times Cpc \times (Tco - Tci)}{m_h \times Cp_h \times (Thi - Tci)} \text{with } m_h < m_c$$

C = Capacity ratio

Where,

Cmin = m_h x Cp_h

Cmax = m_C x Cp_C

NTU(i) = No. of transfer unit for inner surface

$$\text{NTU (i)} = \frac{\text{Ui} \times \text{Ai}}{\text{Cmin}}$$

NTUo = No. of transfer unit for outer surface

$$\text{NTU (o)} = \frac{\text{Uo} \times \text{Ao}}{\text{Cmin}}$$

Result Sheet – I:

Sample Calculation for Parallel Flow

Given Data :

Inner Tube

Inner tube material = Copper

Outer Diameter (do) = 12.5 mm = 0.0125 m

Inner Diameter (di) = 10.5 mm = 0.0105 m

Outer Tube

Outer tube Material = G.I.

Outer Diameter (D_i) = 33 mm = 0.033 m

Inner Diameter (D_o) = 28 mm = 0.028 m

Length of the heat (L) = 1500 mm = 1.5 m

Exchanger

Constants:

Cpc = Specific heat of cold water = 4.174 KJ / KG k

Cph = Specific heat of hot water = 4.174 KJ / KG k

Observation:

mh = Mass flow rate of hot water = 70 LPH = 0.0194 Kg / sec

mC = Mass flow rate of cold water = 95 LPH = 0.0263 Kg / sec

Observation Table – I

SR. NO.	Hot Water Side			Cold Water Side		
	Flow rate m_h. (Kg/sec)	Inlet Temp. Thi (°C)	Outlet Temp. Tho (°C)	Flow rate m_c. (Kg/sec)	Inlet Temp. Tci (°C)	Outlet Temp. Tco (°C)
1	0.0194	62.5	50.5	0.0263	30.5	38.3

Note: Tci in parallel is becoming Tco in counter Flow while making necessary Correction.

Calculation for Parallel Flow

Qh = Heat transfer rate from hot water in KJ / sec

Qh = m_h x Cp_h × (Thi – Tho)

Qh = 0.0194 x 4.174 x (62.5 – 50.5)

Qh = 0.97370	KJ / sec or KW
Qh = 973.70	Watts

Qc = Heat transfer rate from cold water in KJ /sec

Qc = mc x Cpc x (Tco – Tci)

Qc = 0.0263 x 4.174 x (38.3 – 30.5)

Qc = 0.85725	KJ / sec or KW
Qc = 857.25	Watts

Q = Tota heat transfer rate in KJ /sec

$$Q = \frac{Qh + Qc}{2}$$

$$Q = \frac{973.70 + 857.25}{2}$$

$$\boxed{Q = 915.47 \quad \text{Watts}}$$

ΔTm = Logarithmic mean temperature difference in ^{o}K

$$\Delta Tm = \frac{\text{Tin} - \text{Tout}}{\ln(\text{Ti}/\text{To})}$$

Where,

Tin = Thi - Tci = 62.5 – 30.5 = 32 ^oC or ^oK

Tout = Tho - Tco = 50.5 – 38.3 = 12.2 ^oC or ^oK

$$\Delta Tm = \frac{32 - 12.2}{\ln(32/12.2)}$$

$$\boxed{\mathbf{\Delta Tm = 20.53\ ^oC\ or\ ^0K}}$$

Ai = Area of inner side of outer tube in m2

$$Ai = \pi \times di \times L$$

$$Ai = \pi \times 0.028 \times 1.5$$

Where,

di = Inner diameter of outer tube = 0.028 m

$$\boxed{\mathbf{Ai = 0.1319\ m^2}}$$

Ao = Area of outer side tube in m^2

$$Ao = \pi \times do \times L$$

$$Ao = \pi \times 0.0125 \times 1.5$$

Where,

do = Outer diameter of inner tube = 0.0125 m

$$\boxed{\mathbf{Ao = 0.0589\ m^2}}$$

Uo = Overall heat transfer coefficient based on outer area in W / m2 K

$$\text{Uo} = \frac{\text{Qc}}{\text{Ao} + \Delta\text{Tm}}$$

$$\text{Uo} = \frac{857.25}{0.1319 + 20.53}$$

$$\boxed{Uo = 316.57 \quad W/m^2\,K}$$

Ui = Overall heat transfer coefficient based on inner area in W/m^2 K

$$\mathrm{Ui} = \frac{\mathrm{Qh}}{\mathrm{Ai} + \Delta\mathrm{Tm}}$$

$$\mathrm{Ui} = \frac{973.70}{0.0589 + 20.53}$$

$$\boxed{Ui = 805.23 \quad W/m^2\,K}$$

ε = Effectiveness of heat exchanger

$$\varepsilon = \frac{mc \times Cpc \times (Tco - Tci)}{(mCp)min \times (Thi - Tci)} = \frac{mh \times Cph \times (Thi - Tho)}{(mCp)min \times (Thi - Tci)}$$

When, (mCp) min = mh × Cph < mc × Cpc

$$\varepsilon = \frac{\mathrm{Thi} - \mathrm{Tho}}{\mathrm{Thi} - \mathrm{Tci}}$$

(mCp) min = mc × Cpc < mh × Cph ,

$$\varepsilon = \frac{\mathrm{Tco} - \mathrm{Tci}}{\mathrm{Thi} - \mathrm{Tci}}$$

Here,

m_h x Cp_h = 0.0194 x 4.174 = 0.0809

m_C x Cp_C = 0.0263 x 4.174 = 0.109

m_h x Cp_h < m_C x Cp_C

$$\varepsilon = \frac{mc \times Cpc \times (Tco - Tci)}{m_h \times Cp_h \times (Thi - Tci)} \text{with } m_h < m_c$$

$$\varepsilon = \frac{0.0263 \times 4.174 \times (62.5 - 50.5)}{0.0194 \times 4.174 \times (62.5 - 38.3)}$$

$$\boxed{\text{Effectiveness } (\varepsilon) = 0.67}$$

C = Capacity ratio

$$C = \frac{Cmin}{Cmax}$$

Where,

Cmin = $m_h \times Cp_h$ = 0.0194 × 4.174

= 0.0809 KW = 80.97 Watts

Cmax = $m_C \times Cp_C$ = 0.0263 × 4.174 = 0.10977

= 0.10977 KW = 109.77 Watts

$$C = \frac{0.0809}{0.109}$$

$$\boxed{\textbf{Capacity Ratio (C) = 0.7369}}$$

NTU(i) = No. of transfer unit for inner surface

$$\text{NTU (i)} = \frac{\text{Ui} \times \text{Ai}}{\text{Cmin}}$$

$$\text{NTU (i)} = \frac{805.23 \times 0.0589}{80.97}$$

$$\boxed{\textbf{NTU (i) = 1.15}}$$

NTUo = No. of transfer unit for outer surface

$$\text{NTU (o)} = \frac{\text{Uo} \times \text{Ao}}{\text{Cmin}}$$

$$\text{NTU (o)} = \frac{316.57 \times 0.1319}{80.9}$$

$$\boxed{\textbf{NTU (o) = 0.51}}$$

Result Sheet - Ii

Sample Calculation for Counter Flow

Observation:

m_h = Mass flow rate of hot water = 67 LPH = 0.0186 Kg / sec

m_C = Mass flow rate of cold water = 95 LPH = 0.0263 Kg / sec

Observation Table - I

SR. NO.	Hot Water Side			Cold Water Side		
	Flow rate m_h. (Kg/sec)	Inlet Temp. Thi (°C)	Outlet Temp. Tho (°C)	Flow rate m_c. (Kg/sec)	Inlet Temp. Tci (°C)	Outlet Temp. Tco (°C)
1	0.0186	63.4	50.6	0.0263	38.2	30.9

Note: Tci in parallel is becoming Tco in counter Flow while making necessary Correction.

Calculation For Counter Flow:

Qh = Heat transfer rate from hot water in KJ / sec

$Qh = m_h \times Cp_h \times (Thi - Tho)$

$Qh = 0.0186 \times 4.174 \times (63.4 - 50.6)$

Qh = 0.99374 KJ / sec or KW
Qh = 993.74 Watts

Qc = Heat transfer rate from cold water in KJ /sec

$Qc = mc \times Cpc \times (Tco - Tci)$

$Qc = 0.0263 \times 4.174 \times (38.2 - 30.9)$

Qc = 0.80136 KJ / sec or KW
Qc = 801.36 Watts

Q = Total heat transfer rate in KJ /sec

$$Q = \frac{Qh + Qc}{2}$$

$$Q = \frac{993.74 + 801.36}{2}$$

Q = 897.55 Watts

ΔTm = Logarithmic mean temperature difference in ^{o}K

$$\Delta Tm = \frac{\text{Tin} - \text{Tout}}{\ln(\text{Ti}/\text{To})}$$

Where,

Tin = Thi - Tci = 63.4 – 30.9 = 32.5 ^{o}C or ^{o}K

Tout = Tho - Tco = 50.6 – 38.2 = 12.4 ^{o}C or ^{o}K

$$\Delta Tm = \frac{32.5 - 12.4}{\ln(32.5/12.4)}$$

$$\boxed{\mathbf{\Delta Tm = 20.86\ ^{o}C\ or\ ^{0}K}}$$

Ai = Area of inner side of outer tube in m^2

$$Ai = \pi \times di \times L$$

$$Ai = \pi \times 0.028 \times 1.5$$

Where,

di = Inner diameter of outer tube = 0.028 m

$$\boxed{\mathbf{Ai = 0.1319\ \ m^2}}$$

Ao = Area of outer side tube in m^2

$$Ao = \pi \times do \times L$$

$$Ao = \pi \times 0.0125 \times 1.5$$

Where,

do = Outer diameter of inner tube = 0.0125 m

$$\boxed{\mathbf{Ao = 0.0589\ \ m^2}}$$

Uo = Overall heat transfer coefficient based on outer area in W / m2 K

$$\text{Uo} = \frac{\text{Qc}}{\text{Ao} + \Delta\text{Tm}}$$

$$\text{Uo} = \frac{801.36}{0.0589 + 20.86}$$

$$\boxed{Uo = 652.22 \quad W/m^2 K}$$

Ui = Overall heat transfer coefficient based on inner area in W/m^2 K

$$\text{Ui} = \frac{\text{Qh}}{\text{Ai} + \Delta\text{Tm}}$$

$$\text{Ui} = \frac{993.74}{0.1319 + 20.86}$$

$$\boxed{Ui = 361.17 \quad W/m^2 K}$$

ε = Effectiveness of heat exchanger

$$\varepsilon = \frac{mc \times Cpc \times (Tco - Tci)}{m_h \times Cp_h \times (Thi - Tci)} \text{with } m_h < m_c$$

$$\varepsilon = \frac{mc \times Cpc \times (Tco - Tci)}{(mCp)min \times (Thi - Tci)} = \frac{mh \times Cph \times (Thi - Tho)}{(mCp)min \times (Thi - Tci)}$$

When, (mCp) min = mh × Cph < mc × Cpc

$$\varepsilon = \frac{\text{Thi} - \text{Tho}}{\text{Thi} - \text{Tci}}$$

(mCp) min = mc × Cpc < mh × Cph ,

$$\varepsilon = \frac{\text{Tco} - \text{Tci}}{\text{Thi} - \text{Tci}}$$

Here,

m_h x Cp_h = 0.0186 × 4.174 = 0.0776

m_C x Cp_C = 0.0263 × 4.174 = 0.109

$\therefore$ m_h x Cp_h < m_C x Cp_C

$$\varepsilon = \frac{mc \times Cpc \times (Tco - Tci)}{m_h \times Cp_h \times (Thi - Tci)} \text{with } m_h < m_c$$

$$\varepsilon = \frac{0.0263 \times 4.174 \times (63.4 - 50.6)}{0.0194 \times 4.174 \times (63.8 - 38.2)}$$

$$\boxed{\text{Effectiveness } (\varepsilon) = 0.706}$$

C = Capacity ratio

$$C = \frac{Cmin}{Cmax}$$

Where,

$Cmin = m_h \times Cp_h = 0.0186 \times 4.174 = 0.0776$

$Cmax = m_C \times Cp_C = 0.0263 \times 4.174 = 0.109$

$$C = \frac{0.0776}{0.109}$$

$$\boxed{\text{Capacity Ratio (C)} = 0.711}$$

NTU(i) = No. of transfer unit for inner surface

$$\text{NTU (i)} = \frac{\text{Ui} \times \text{Ai}}{\text{Cmin}}$$

$$\text{NTU (i)} = \frac{361.16 \times 0.1319}{77.63}$$

$$\boxed{\text{NTU (i)} = 0.613}$$

NTUo = No. of transfer unit for outer surface

$$\text{NTU (o)} = \frac{\text{Uo} \times \text{Ao}}{\text{Cmin}}$$

$$\text{NTU (o)} = \frac{652.22 \times 0.0589}{77.63}$$

$$\boxed{\text{NTU (o)} = 0.494}$$

Conclusion:

Heat transfer coefficient of Parallel flow and counter flow heat exchanger is found out to be ----

Experiment 10

Heat Transfer in Forced Convection

Aim: To determine the heat transfer coefficient in forced convection of air in a tube.

Introduction:

In many practical situations and equipment, we invariably deal with flow of fluids in tubes e.g. boiler, super heaters and condensers of a power plant, automobile radiators, water and air heaters or coolers etc. the knowledge and evolution of forced convection heat transfer coefficient for fluid flow in tubes is essentially a prerequisite for an optional design of all thermal system

Convection is the transfer of heat within a fluid by mixing of one portion of fluid with the other. Convection is possible only in a fluid medium and is directly linked with the transport of medium itself.

In forced convection, fluid motion is principally produced by some superimposed velocity field like a fan, blower or a pump, the energy transport is said due to *forced convection.*

Description:

The apparatus consists of a blower unit fitted with the test pipe. The test section is surrounded by a Nichrome band heater. Four thermocouples are embedded on the test section and two thermocouples are placed in the air stream at the entrance and exit of the test section to measure the air temperature. Test pipe is connected to the delivery side of the blower along with the orifice to measure flow of air through the pipe. Input to the heater is given through a dimmerstat and measured by meters.

It is to be noted that only a part of the total heat supplied is utilized in heating the air. A temperature indicator with cold junction compensation is provided to measure temperatures of pipe wall at various points in the test section. Airflow is measured with the help of orifice meter and the water manometer fitted on the board.

Specification:

Pipe diameter (D_o) : 33 mm

Pipe diameter (D_i) : 28 mm

Length of test section (L) : 400 mm

Blower : 35 No. FHP motor

Orifice Diameter (d) : 14 mm

Dimmer stat : 0 to 2 amp, 230 volt, AC

Temperature indicator : Digital type and range 0 - 200 °c

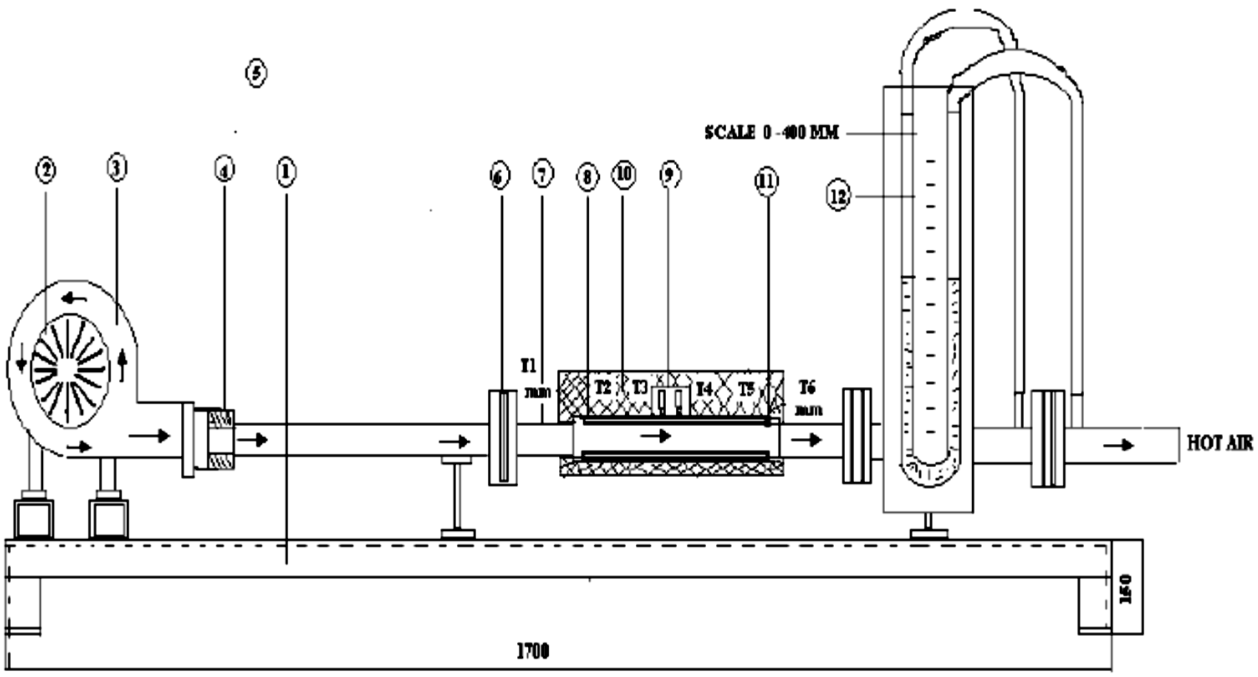
Voltmeter : 0 -100 /200v

Ammeter : 0 – 2 amp

Heater : Nichrome wire heater wound on

Test Pipe (Band Type) 400 watt

Schematic Diagram:



1) ‘C’ Channel 2) Motor 3) Blower 4) Adapter 6) Orifice 7) Air Inlet Temperature 8) Mica Covered Heater 9) Heater Socket 10) Foam Packing 11) Stainless Still Cladding 12) Monometer

T1:- Air Inlet Temperature

T6:- Air Outlet Temperature

T2 - T4:- Pipe Wall Temperature

Procedure:

1. Switch ON the mains system
2. Switch ON blower.
3. Adjust the flow by means of gate valve to some desired difference in the manometer level.
4. Switch ON heater
5. Start the heating of the test section with the help of dimmerstat and adjust desired heat input with the help of Voltmeter and Ammeter.
6. Take readings of all the six thermocouples at an interval of 10 min until the steady state is reached.
7. Note down the heater input.

Precaution

1. Keep the dimmer stat at zero position before switching ON the power supply.
2. Increase the voltmeter gradually.
3. Do not stop the blower in between the testing period.
4. Do not disturb thermocouples while testing. Operate selector switch of the thermocouple gently. Don't exceed 200 watts
5. Operate selector switch of the temperature indicator gently.

Observation:

1. Outer diameter of the pipe (D_o) = 33 mm
2. Inner diameter of the test pipe (D_i) = 28 mm
3. Length of the test section (L) = 400 mm
4. Diameter of the orifice (d) = 14 mm

Sr No	Voltage [V] (Volts)	Current [I] (Amps)	Temperature in °c						Manometer reading of water h in meter
			T_1°C	T_2°C	T_3°C	T_4°C	T_5°C	T_6°C	

Calculation:

Ao = Area of Cross Section Orifice in m^2

$$C = \frac{\pi}{4} \times d^2$$

Q = Volume flow rate in m^3 / sec

$$Q = Cd \times Ao \times \sqrt{2 \times g \times h(\rho_w/\rho_a)}$$

Where,

Cd = Coefficient of discharge of orifice = 0.68

Ao = area of cross section of orifice in m^2

ρ_w = Density of water = 1000 Kg/m^3

ρ_a = density of air at ambient temp. = 1.03 Kg/m^3

h = manometer reading in meter

ma = mass flow rate of air in Kg /sec

$$ma = Q \, x \, \rho_a$$

Where,

ρ_a = Density of air at Ambt. temp. = 1.03 Kg/m^3

ΔT = Temperature rise in air in ^oC or ^oK

$\Delta T = (T_6 - T_1)$

Qa = Heat carried away by Air in kJ/sec or Watts

Qa = ma × Cp × ΔT

Where,

Cp = specific heat of air= 1.005 KJ / °K Kg

Ta = Average Temperature of Air in 0C

$$Ta = \frac{(T_1 + T_6)}{2}$$

Ts = Average Surface Temperature in ^oC

$$Ts = \frac{T_1 + T_2 + T_3 + T_4}{4}$$

As = Test Section Surface Area in m^2

$$As = \pi \times D_i \times L$$

Where,

D_i = Inner diameter of the test pipe in meter

L = Length of the test section in meter

h = Heat Transfer Coefficient in W / m^2k

$$h = \frac{Q}{A(Ts - Ta)}$$

Ac = Cross Test Section Area in m^2

$$Ac = \frac{\pi}{4} \times Di^2$$

V = Mean Velocity of Flow through tube in m / sec

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

Re = Reynold's Number

$$Re = \frac{V\ D_1}{\nu}$$

Where,

ν = Kinematic Viscosity at bulk mean

Temp. i.e. (T1 + T6) in m^2/ s

Pr = Prandtle Number

Pr = 0.7 at Avg. Temperature

Nu = Nusselt Number

Nu = 0.023 x $Re^{0.8}$ x $Pr^{0.3}$

h = heat transfer coefficient calculated by using the correlations

Nu = 0.023 $Re^{0.8}$ $Pr^{0.3}$ ………… For Re >10000

Nu = 0.036 $Re^{0.8}$ $Pr^{0.3}$ ………… For Re >2300

$$Nu = \frac{h \times D}{K}$$

Where,

K = thermal conductivity of air at avg. temp. in w / m k

Conclusion:-

Heat transfer coefficient in forced convection of air in a tube is found out to be -----