WEST GODAVARI INSTITUTE OF SCIENCE & ENGINEERING

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AVAPADU, **PRAKASARAOPALEM** – 534 112, W.G.Dist., A.P

HEAT TRANSFER LAB MANUAL-R20

DEPARTMENT OF MECHANICAL ENGINEERING

III B.TECH II SEMESTER

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KAKINADA–533003, Andhra Pradesh, India
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EXP:-1 DETERMINATION OF OVERALL HEAT TRANSFER COEFFICIENT OF COMPOSITE WALL

OBJECTIVES:

- To understand the electrical analogy used for calculation of heat transfer in composite systems.
- To understand the concept of overall heat transfer coefficient

AIM: To Determine the overall heat transfer coefficient of given composite wall and comparison of theoretical overall heat transfer with experimental value.

APPARATUS: Experimental Test Rig.

INTRODUCTION:

A composite wall refers to a wall of several heterogeneous layers, for example wall of dwelling houses where bricks are given a layer of plaster on either side. Likewise walls of furnaces, boilers and other heat exchange devices consist of several layers, a layer for mechanical strength are for high temperature characteristics, a layer of low thermal conductivity material to restrict flow of heat and another layer for structural requirement for good appearance. Analysis of the composite wall assumes that there is a perfect contact between layers and no temperature difference occurs across the interface between materials.

Many Engineering applications of practical utility involve heat transfer through a medium composed of two or more materials of different thermal conductivity arranged in series or parallel. Consider for example, the walls of a refrigerator, cold storage plants, hot water tanks, which always have some kind of insulating material between the inner and outer walls. A hot fluid flowing inside a tube covered with a layer of thermal insulation is another example of a composite system because in this case the thermal conductivities of tube metal and insulation are different.

THEORY:

Consider a multi-layered wall as depicted in Figure.1. The temperature gradients in the three materials are also shown in figure. The Fourier equation may be applied directly to yield the heat flow rate. It is assumed that the interior and exterior surfaces of this system are subjected to convective heat transfer to fluids at mean coefficients h_a and h_b respectively.

Now,

$Q = Aha\ (T_a - T_l)$

$= -K_1A\ (T_2 - T_1)\ /\ L_1$

$= -K_2A\ (T_3 - T_2\ /\ L_2$

$= -K_3A\ (T_4 - T_3)\ /\ L_3$

$= Ah_6\ (T_4 - T_6)$ (or)

$Q = T_a - T_1/R_a$

$= T_1 - T_2/R_1$

$= T_2 - T_3/R_2$

$= T_3 - T_4\ /\ R_3$

$= T_4 - T_6\ /\ R_6$

Where $R_a = 1\ /Ah_a$,

$R_l = L_l/AK_1$, $R_2 = L_2\ /\ AK_2$,

$R_3 = L_3\ /\ AK_3$, $Rb = 1\ /Ah_b$

are the various thermal resistances.

Now, $T_a - T_b$

$= (T_a - T_1) + (T_l - T_2) + (T_2 - T_3) + (T_3 - T_4) + (T_4 - T_6)$

Hence from equation

$Q = T_a - T_b\ /\ 1\ /h_aA + L_l\ /\ AK_l + L_2\ /\ AK_2 + L_3\ /\ AK_3 + 1/\ h_bA$

$= T_a - T_b\ /\ \Sigma R$

Where ΣR

$= R_a + R_l + R_2 + R_3 + R_b = 1/\ UA$

Where U is the overall heat transfer coefficient. The temperature at any intermediate point of this composite structure can be obtained by above equations.For example, the interface temperature T_1 given by

$Ta - T_1\ /\ R_a = T_a - T_b\ /\ \Sigma R$ (Or)

$T_a - T_1 = (T_a - T_b)\ Ra\ /\ \Sigma R$

Or the surface temperature T_4 is given by $(T_a - T_4 = T_a - T_b)$

$(R_a + R_l + R_2 + R_3)\ /\ \Sigma R$

From equation it is noted that Heat flow

= Thermal potential difference / Thermal resistance = ΔT overall / ΣR th

This is similar to ohm's low in electric circuit theory. Equation can be generalized for an n layer wall as

$Q\ (T_a - T_b)\ /1\ /\ [1\ /\ h_a + 1\ /\ h_b + \Sigma L\ /\ KN]$

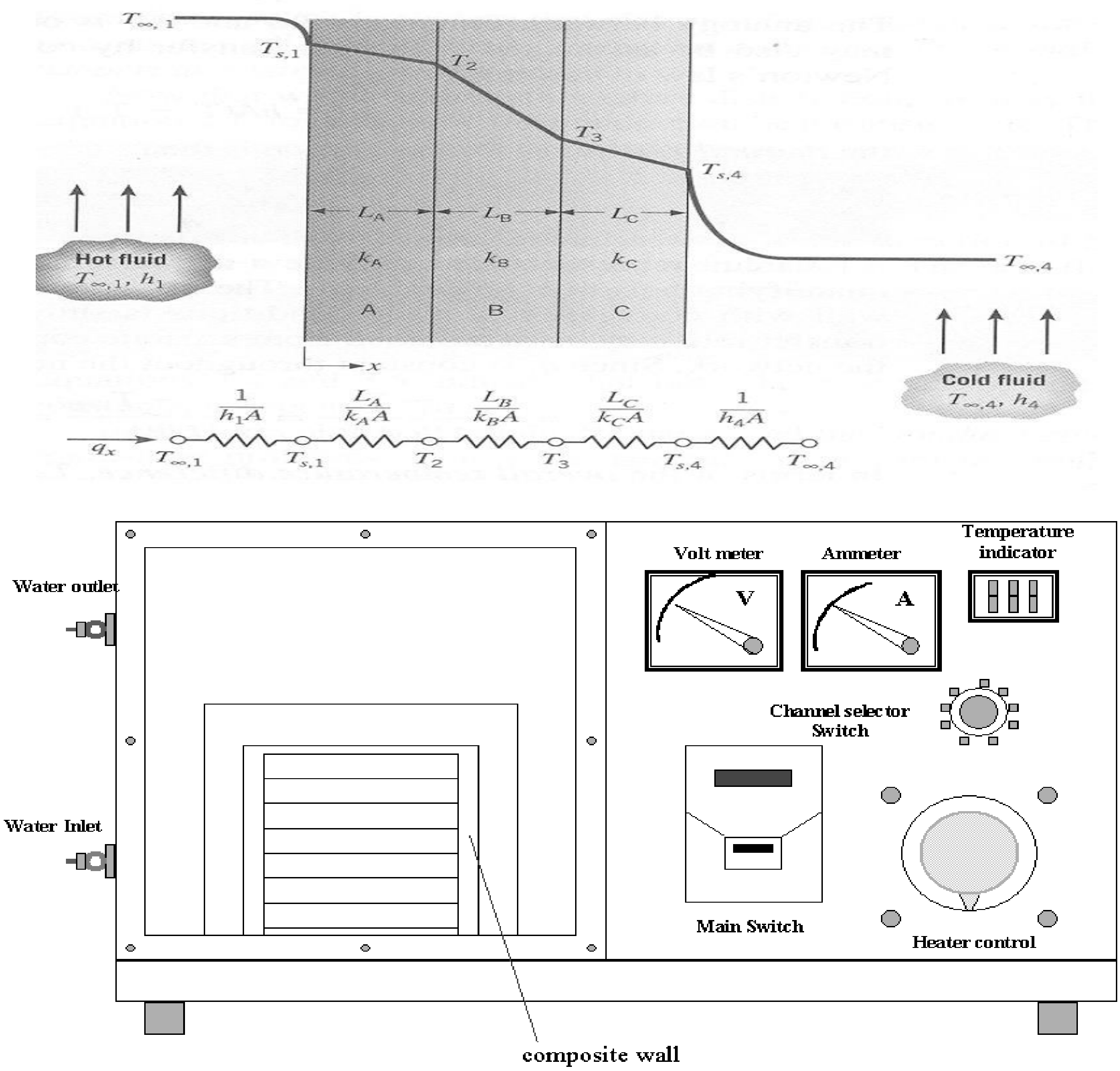


Figure.1 Experimental setup and Variation of temperature in the slab

EXPERIMENTAL SET UP:

The apparatus consists of a central heater as shown in Figure, is sand-witched between two aluminum plates. Three types of slabs are provided to ensure the perfect contact between the slabs. A dimmer stat is provided for varying the input to the heater and measurement of input is carried out by a voltmeter and ammeter. Thermocouples are embedded between interfaces of input slabs, to read temperatures at the surface.

SPECIFICATIONS:

1. Slabs size
   (a) M.S - 25 cm ϕ x 25 mm thick.
   (b) Bakelite - 25 cm ϕx 10 mm thick.

(c) Brass - 25 cm ϕx 10 mm thick.

2. Nichrome heater would on mica former and insulation with control unit capacity 200 W max.
3. Heater control unit: - 230v, 0^{-2+} A., single phase Dimmer stat - 1 No.
4. Voltmeter- 0 - 250v.
5. Ammeter- 0 –1 A
6. Multi channel digital temp indicator.

EXPERIMENTAL PROCEDURE:

1. Star the electric supply.
2. Then start heating given composite slabs, by adjusting the heater input by rotating Dimmer stat in clockwise direction.
3. Adjust water circulation through the sink and measure the quantity.
4. Allow the system to reach steady state condition.
5. When the entire temperature remains steady, note down all the observation and completes the observation table.

OBSERVATION TABLE:

S.No	Power input		Water flow rate	Temperatures				
	V	I		T_1	T_2	T_3	T_4	

MODAL CALCULATIONS:

(1) Mean reading

(a) $T_A = T_1 + T_2 / 2 =$ ^0C

(b) $T_b = T_3 + T_4 / 2 =$ ^0C

(c) $T_c = T_5 + T_6 / 2 =$ ^0C

(d) $T_d = T_7 + T_5 / 2 =$ ^0C

(2) Rate of heat supplied

Q=V x I watts = watts

Heat flux, q = Q / A watts / m_2

Where, $A = \pi / 4\ d^2$

d = half diameter of plates, A = m^2, Q = w /m^2

(3) Total thermal resistance of composite slab.

$R_{total} = (T_a - T_d / q)\ m^2\ K / W =$

(4) $U_{exp} = 1 / R_{totatl}$

(5) $U_{theoretical} = \dfrac{1}{\delta_1 / k_1 + \delta_2 / k_2 + \delta_3 / k_3}$

PRECAUTIONS:

(1) Keep the dimmer stat zero before starting.

(2) Increase the voltage and current slowly.

(3) Keep all the assembly undistributed.

(4) Do not increase voltage above maximum limit.

RESULTS AND ANALYSIS:

Table of results

Sl.No	Power input	Water flow rate	Thermal conductivity

SUGGESTIONS FOR DISCUSSION:

- Equivalent resistance of two plates connected in parallel and in series
- Terms corresponding to current, voltage and resistance in thermal analogy
- Different resistances involved in the analysis of composite systems

EXP:-2 DETERMINATION OF EMISSIVITY OF TEST PLATE

OBJECTIVES:

1. To review the basic principles of radiation heat transfer.
2. To know the relation between emissivity, Transmittivity and Reflectivity

AIM: To Determine the Emissivity of given test plate by taking the emissivity of black plate as ONE

APPARATUS: Experimental Test Rig.

INTRODUCTION:

The real surfaces like a polished metal plate do not radiate as a black body. The 'gray' nature of real surfaces can be accounted for by introducing a factor ε is called emissivity which relates radiation between gray and black body.

The concept of black body is an idealization, which serves as a standard for real black body performance. The emissivity of a surface is a measure of how it emits radiant energy in comparison with a black surface at the same temperature in reality the emissivity of material varies with temperature and the wave length of the radiation the emissivity of surface is a function of its nature and characteristics and is independent of the wavelength or the nature of the impinging radiation waves.

THEORY:

The surface of a substance highly influences its radiation characteristics, and the amount of radiant energy the surface emits, absorbs, reflects, and transmits. Polished gold, for example, has an emissivity of (measured normal to the surface) of 0.025. An unpolished gold surface, on the other hand, has a normal emissivity of 0.47. It is important to realize that such differences exist, and that surface treatment is a significant factor in how the material behaves in radiative heat transfer studies. **Emissivity** is a property that describes how radiant energy interacts with the surface of a material. Other properties that are important are **reflectivity, absorptivity,** and **transmissivity.** As implied by these names, reflectivity is the fraction of incident radiant energy reflected by the surface; absorptivity is the fraction of incident radiant energy absorbed by the surface; and, transmissivity is the fraction of incident radiant energy transmitted through a layer of the material. The radiative properties discussed in the preceding paragraphs are in general the functions of wavelength. Properties that describe surface behavior as a function of wavelength are called **monochromatic** properties. In addition, radiative properties can be the functions of direction. Such properties are referred to as **directional** properties. In the analysis of radiation heat transfer, accounting for the exact behavior of a surface or material can be complex enough to make a solution exlusive. A simplified approach must therefore be formulated. This involves the use of radiative properties that are averages over all wavelengths and all directions. These properties are called **total** or **hemispherical** properties. Use of total properties is accurate enough in a majority of cases for most engineering problems.

EXPERIMENTAL SETUP:

The apparatus uses comparator method for determining the emissivity of test plate. It consists of two aluminum plates of equal physical dimensions. Mica heaters are provided inside the plates. The plates are mounted in an enclosure to provide undisturbed surroundings.

One of the plates are blackened outside for use as a comparator (because black surfaces has ε = 1). Another plate is having natural surface finish. Input to heaters can be controlled by separate dimmer stats. Heater input is measured on common ammeter and voltmeter. One thermocouple is provided on surface of each plate to measure the surface temperature with digital temperature indicator. By adjusting input to the heaters both the plates are brought to the same temperature, so that conduction and convection losses from both the plates are equal and difference in input is due to different emissivities.

Holes are provided at back side bottom and at the top of enclosure for natural circulation of air over the plates. The plate enclosure is provided with perspex acrylic sheet at the front

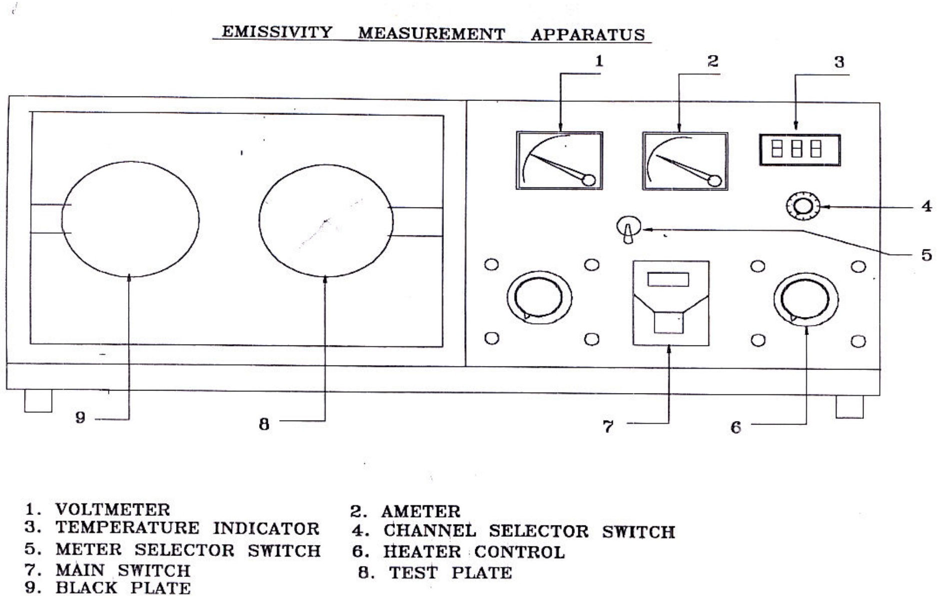


Fig.1 Experimental set up

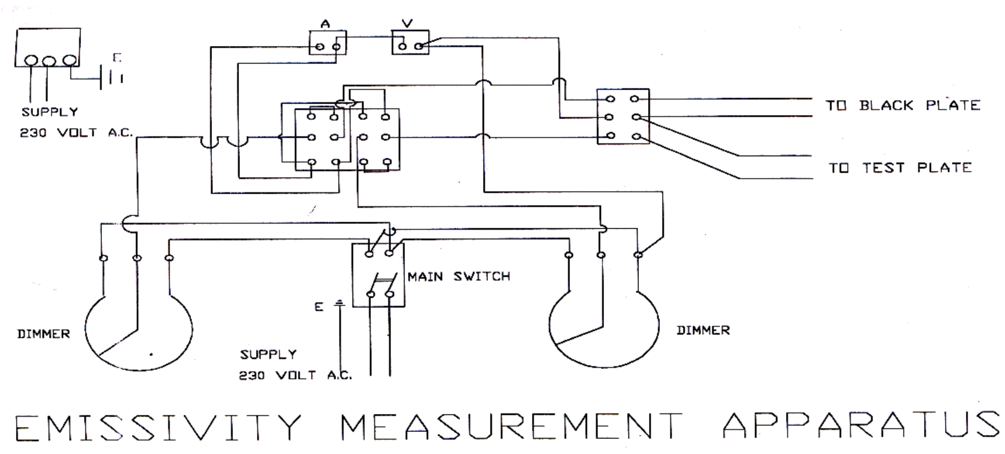


Fig.2 Circuit diagram of emissivity measuring apparatus

EXPERIMENTAL PROCEDURE:

1) Switch on electric supply by keep both dimmer knobs at zero position.
2) Put the toggle switch towards test plate and give some power input by adjusting dimmer stat of test plate.
3) Change the toggle switch towards black plate and give some power input such that it is slightly more than the power input given to the test plate.
4) Check the temperatures after 10 minutes and adjust the power input by adjusting dimmer stat so that temperatures of both plates are equal and steady.
5) After attainment of steady state note down the plate temperatures.
6) Repeat the experiment for different power inputs.

OBSERVATIONS:

Plate	Input		Surface Temperature(^0C)	Enclosure Temperature (^0C)
	V	I		
Test Plate				
Black Plate				

CALCULATIONS

1) Enclosure temperature T_e = $(T_3 + 273.15)K$
2) Plate surface temperature $T = T_1 = T_2 =$ ^0C
   $T_S = (T+273.15)^0K =$ ^0K
3) Heat input to black plate $W_b = V \times I =$ W

4) Heat input to test plate $W_T = V \times I =$ W
5) Surface area of plates

$A = (2\pi D^2/4) + \pi Dt =$

Where D = diameter of the plate = 0.16m

t = thickness of the plate = 0.009m.

6) $W_b - W_t = \sigma A (T^4_S - T^4_E) (l-\varepsilon)$

$\sigma = 5.667 \times 10^{-8}\ W/m^2k^4$

RESULT: Emissivity of the test plate =

PRECAUTIONS:

1) Operate all the switches and knobs gently.
2) Black plate should be perfectly blackened.
3) Never put hands or papers over the holes provided at the top of enclosure.

RESULTS AND ANALYSIS:

Table of results

Sl.No	Power input	Emissivity of black body	Emissivity of test body

Draw the plot between Power input to the plate and emissivity

SUGGESTIONS FOR DISCUSSION:

1. Whether the emissivity of the test plate is less than or greater than that of the test plate
2. How the emissivity of test plate varies with temperature
3. How do the surface characteristics influence the emissivity?

EXP:-3 DETERMINATION OF HEAT TRANSFER COEFFICIENT IN FORCED CONVECTION

OBJECTIVES:

- To know the order of magnitude of convection heat transfer coefficient in Forced convection.
- To understand the effect Prandtl number influence boundary layer development in the entry region

AIM: To Determine the Heat transfer coefficient in Forced Convection and comparison of Experimental Heat Transfer Coefficient value with Theoretical Heat Transfer Coefficient.

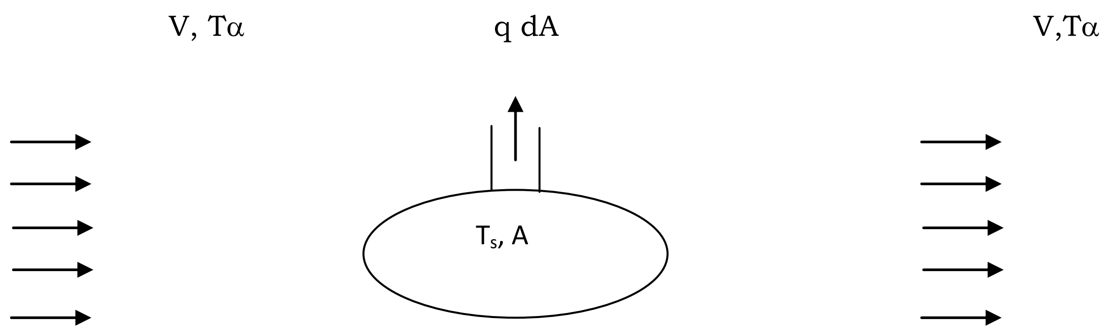
APPARATUS:

Experimental Test Rig.

THEORY:

When a fluid flows inside a duct or over a solid body and the temperatures of the fluid and solid surfaces are different, heat transfer between the fluid and solid surface is considered as convective heat transfer. The transfer of heat here is inseparably linked with moment of fluid molecules.

In laminar flow heat transfer occurs only by conduction as there are no eddies to carry heat by convection across the isothermal surface. The heat transfer in convection is totally depending up on the property heat transfer coefficient. Tenon may be explained below.



The local heat flux q, for an arbitrarily shaped surface of area A at temperature Ts, over which flows a fluid of velocity, V and of temperature Tα is given by

$$q = h^*(Ts-T\alpha) \qquad \text{-----------------------------} \qquad (1)$$

Where h is the local heat transfer coefficient. This equation is referred to as Newton's law of cooling or heating. The simplicity of (1) is misleading because the convective heat transfer coefficient is actually the complicated function of the nature of fluid flow, thermal properties of the fluid and the configuration of the system. Due to the variation of flow conditions from point to point, the values of q&h along the surface also vary and that is why the adjective local is applied to them. The total heat transfer rate may be obtained by integrating (1), over the entire surface, assuming a uniform value of Ts.

$$Q=\int q\, dA= (Ts - T\alpha) \int h\, dA \quad \text{-------------------------------------(2)}$$

Defining h as the average of total heat transfer coefficient for the entire surface, we may write (2) as

$$Q=h*A*(Ts-T\alpha) \quad \text{-------------------------------------(3)}$$

Comparing (2)&(3) the average and local convection coefficients are related by

$$h_a = (l/A) \int h\, dA.$$

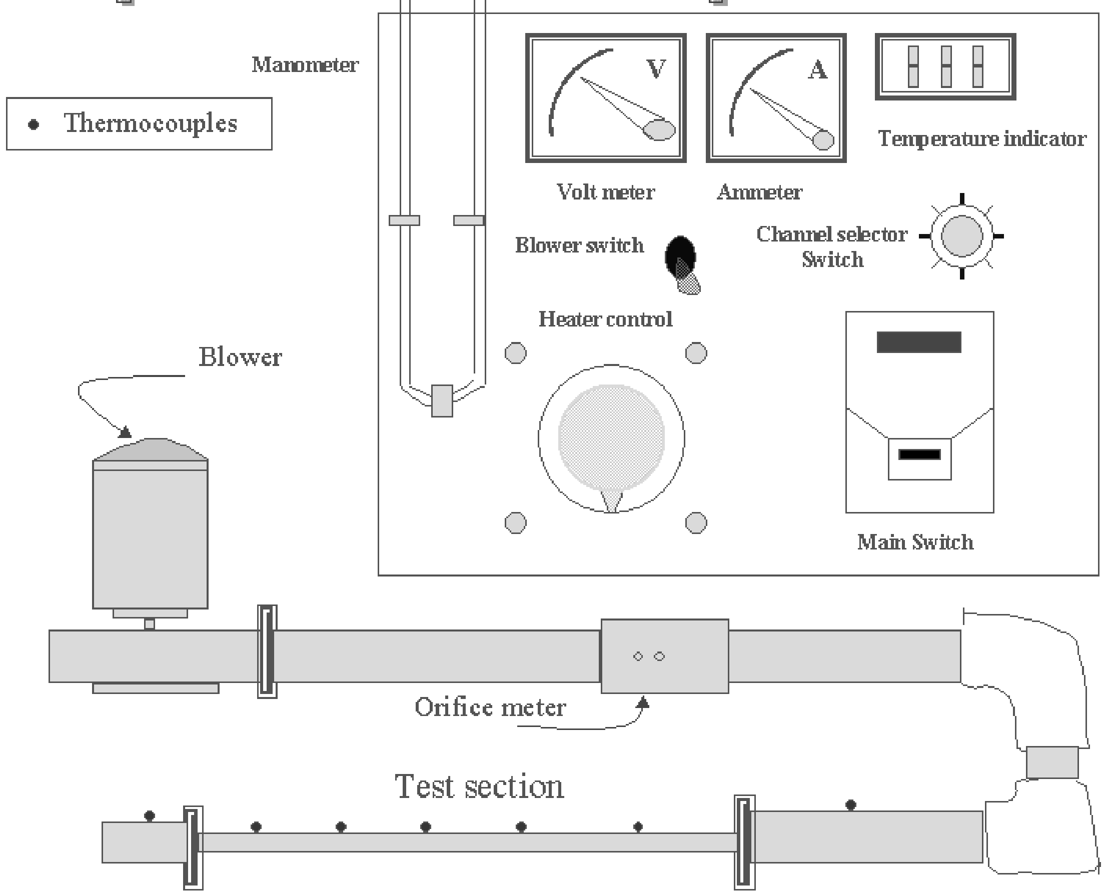
Note that in the case of simple flat plate of unit width where h is a function of x alone. $h^1 = (1/L) \int h\, dx$.

Invariably, the problem of convection is to determine the local heat flux or the total heat transfer rate, which may be determined from (1)&(3) provided that the local and total heat transfer coefficients are known. The aim of any convection analysis is to get these coefficients first.

EXPERIMENTAL SETUP:

The apparatus shown in Fig: 1 consists of a circular pipe through which cold fluid is passed and air is being forced through the circular pipe by means of a blower. Pipe is heated by means of a band heater. Outside pipe temperature is measured with thermocouples attached to the pipe surface. Manometer is also fitted to measure the air circulated in the pipe. Heater input is measured by using voltmeter and ammeter.

Experimental Set Up



SPECIFICATIONS:

1. Test pipe =33mm inner diameter 500mm long;
2. Band heater for pipe.
3. Multi channel digital temperature indicator 0-300°c using chromel / alumel thermocouples.
4. Dimmer stat 2 amp 240 volts for heater input control.
5. Voltmeter 0-200 volts.
6. Ammeter 0-2 A.
7. Blower to force air through test pipe.
8. Orifice meter with water manometer.

EXPERIMENTAL PROCEDURE:

1. Star the electric supply.
2. Give heat input to heater by adjusting the heater input by rotating Dimmer stat in clockwise direction.

3. Start blower to allow air to circulate through the system.
4. Adjust the quantity of air circulated through the system by adjusting the gate valve.
5. Allow the system to reach steady state.
6. When all the temperatures remain steady we should note down all the observation and complete the observation table.

OBSERVATION TABLE:

S.No.	V	A	Temperature ^{0}C							Manometer Difference (m)
			T_1	T_2	T_3	T_4	T_5	T_6	T_7	

MODEL CALCULATIONS:

For full gate opening

1. Air inlet temperature T_l= °C
2. Air outlet temperature=T_7= °C
3. Density of air $\rho_a = (1.293 \times 273) / (273 + T_1)$ Kg/m^3
4. Diameter of orifice= 22 mm.

   Manometer difference = water head = h_w m

   Air head ha = $h_w (\rho_w / \rho_a)$

   ρ_w = density of water = 1000 Kg/m^3

   Air volume flow rate, $Q=C_d*(2*g*ha)^{0.5} *a_0$ m^3/s

   Where, Cd = 0.64

   a_o= cross sectional areas of orifice.
5. Mass flow rate of air

   $m_a=Q* \rho_a$ kg/s

   Velocity of air

   $V= Q/a_p$ m/s

   Where a_p = C.S area of pipe.

   $= 8.33 \times 10^{-4}$ m^2
6. Heat gained by air

   $q = ma *Cpa * (T_7-T_1)$.

   Where Cpa = specific heat of air

   = 1 kJ/kgK
7. Average inside surface temperature

$T_s = (T_2+T_3+T_4+T_5+T_6)/5$ °C

8. Bulk mean temperature of air

   $T_m = (T_1+T_7)/2$ °C

9. Average surface heat transfer coefficient, $h_{expt} = q / A (T_s - T_m)$

Where, A = inside surface area of the pipe

$= \pi * d_i * L = 0.0518\ m^2$

10. Reynolds number, $Re = V*d_i / \upsilon$

   If Re<2000, flow is laminar

For laminar flow, $Nu = h*d/K_{air} = 4.36$

If Re>2000flow is turbulent

$Nu = h*d/K_{air} = 0.023\ (Re)^{0.8}\ (Pr)^{n}$

Where n = 0.4 when fluid is being heated

n = 0.3 when fluid is being cooled

PRECAUTIONS:

1. While putting "ON' the supply keep dimmer stat at zero position and switch off blower.
2. Operate all the switches and controls gently.
3. Do not obstruct the flow of air while experiment is going on.

RESULTS AND ANALYSIS:

Table of results

Sl.No	Power input	$h_{theoritical}$	$h_{experimental}$

Draw the following plots

(i) $h_{theoritical}$ Vs Power input

(ii) $h_{experimental}$ Vs Power input

SUGGESTIONS FOR DISCUSSION:

- Nature of fully developed flow, and it's variation from flow in the entry region.
- Mode of convection heat transfer (Natural/Forced) in which heat transfer coefficient is more and why it happens
- Variation of heat transfer co efficient with temperature
- If the theoretical value deviates from experimental, the reasons for the deviation

EXP_4 DETERMINATION OF THERMAL CONDUCTIVITY OF INSULATING POWDER

OBJECTIVES:

1. To differentiate the conductors and insulators
2. To know the range of thermal conductivity of insulators

AIM: To Determine the Thermal Conductivity of given Insulating Powder.

APPARATUS: Experimental Test Rig.

INTRODUCTION:

INSULATION**:** It is defined as a material which retards the heat flow effectively. Conduction of heat is flow of energy in the form of heat from one molecule to another due to drift of valance electrons. In order to reduce losses of heat, various types of insulation are used in practice. Various powders for example asbestos powder, plaster of paris etc., are also used fur heat insulation. In order to determine the appropriate thickness of insulation, knowledge of thermal conductivity of insulating material is essential. Generally thermal conductivity of insulating powder is determined by using concentric spheres.

Heat is transferred through insulation by conduction, convection, and radiation or by the combination of these three. There is no perfect insulator in practice. The desirable properties of insulating materials either for heating systems or cooling systems are low weight, resistant to wear, resistant to fire, resistant to moisture absorption and free from odor and long life. The types of insulating materials used are classified as insulations for low temperatures, medium temperatures, and high temperatures. The insulations are also classified as loose fill insulations and slab insulations.

The insulations are commonly used for air conditioning systems, refrigerators, furnaces, boilers and steam pipes.

THEORY:

Consider the transfer of heat by conduction through the wall of a hollow sphere formed of insulating powder (refer fig. 1).

Let r_i = radius of inner sphere, m.

r_o= radius of outer sphere, m.

T_i = average inner sphere surface temperature, ^{0}C.

T_0 = average outer sphere surface temperature, ^{0}C.

Consider a thin spherical layer of thickness ‘dr’ at radius r and temperature difference of dT across the layer. According to the Fourier Law of heat conduction, Heat transfer rate,

$Q = - k \,.\, 4\pi \,.\, r^2 \,(dT / dr)$

$$dT = \frac{q}{4\pi k}\left(-\frac{dr}{r^2}\right)$$

Where k = thermal conductivity of insulating powder.

Integrating the above equation between, r_i to r_0 and T_i to T_0

$$\frac{q}{4\pi k}\int_{r_i}^{r_0} -\frac{dr}{r^2} = \int_{T_i}^{T_0} dt$$

$$\frac{q}{4\pi k\left[\frac{1}{r_0}-\frac{1}{r_i}\right]} = (T_0 - T_i)$$

$$q = \frac{4\pi k r_0 r_i (T_0 - T_i)}{r_i - r_0}$$

From the measured value of q_i T_i and T_0, thermal conductivity of insulating powder can be determined as

$$k = \frac{q(r_i - r_0)}{4\pi r r_0 r_i (T_0 - T_i)}$$

EXPERIMENTAL SEUP:

The experimental setup as shown in Figure1 consists of a smaller (inner) sphere, inside which is fitted a mica electric heater shown in Fig2. Similar sphere is fitted at the center of the outer sphere. The insulating powder whose 'k' value is to be determined is filled in the gap between the two spheres. The heat generated by the heater flows through the powder to the outer sphere. The outer sphere loses heat to the atmosphere. The input to the heater is controlled by a dimmer stat and is measured on voltmeter and ammeter. Four thermocouples are provided on outer the surface of the inner sphere and six thermocouples are on the inner surface of the outer sphere which are connected to a multi channel digital temperature indicator. Average of outer & inner sphere temperatures gives the temperature difference across the layer of powder.

SPECIFICATIONS:

1. Inner sphere - 100 mm OD, halved construction
2. Outer sphere – 200 mm ID, halved construction
3. Heater - Mica flat heater, fitted inside inner sphere.
4. Controls _
    a) Main switch - 30A. DPDT switch
    b) Dimmer stat - 0-230V, 2A capacity
5. Measurements -
    a) Voltmeter0-200V
    b) Ammeter 0-1 amp
    c) Multi channel digital temperature indicator uses Chromel/Alumel thermocouples.

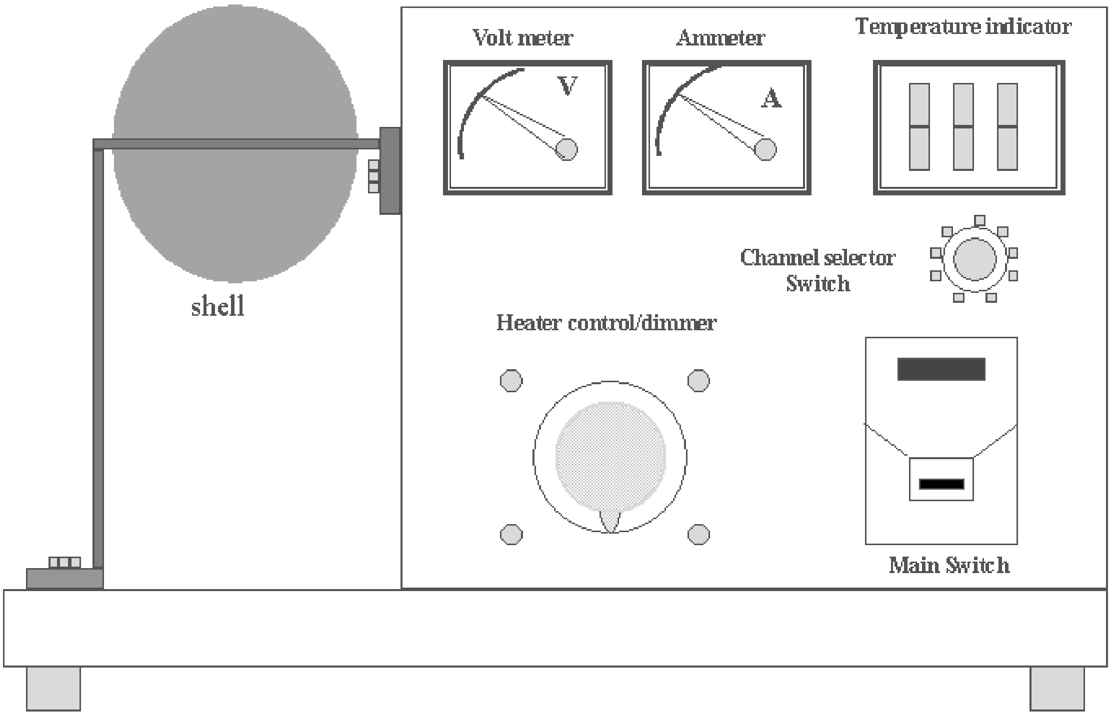


Fig. 1 Experimental Setup for finding the thermal conductivity of insulating powder.

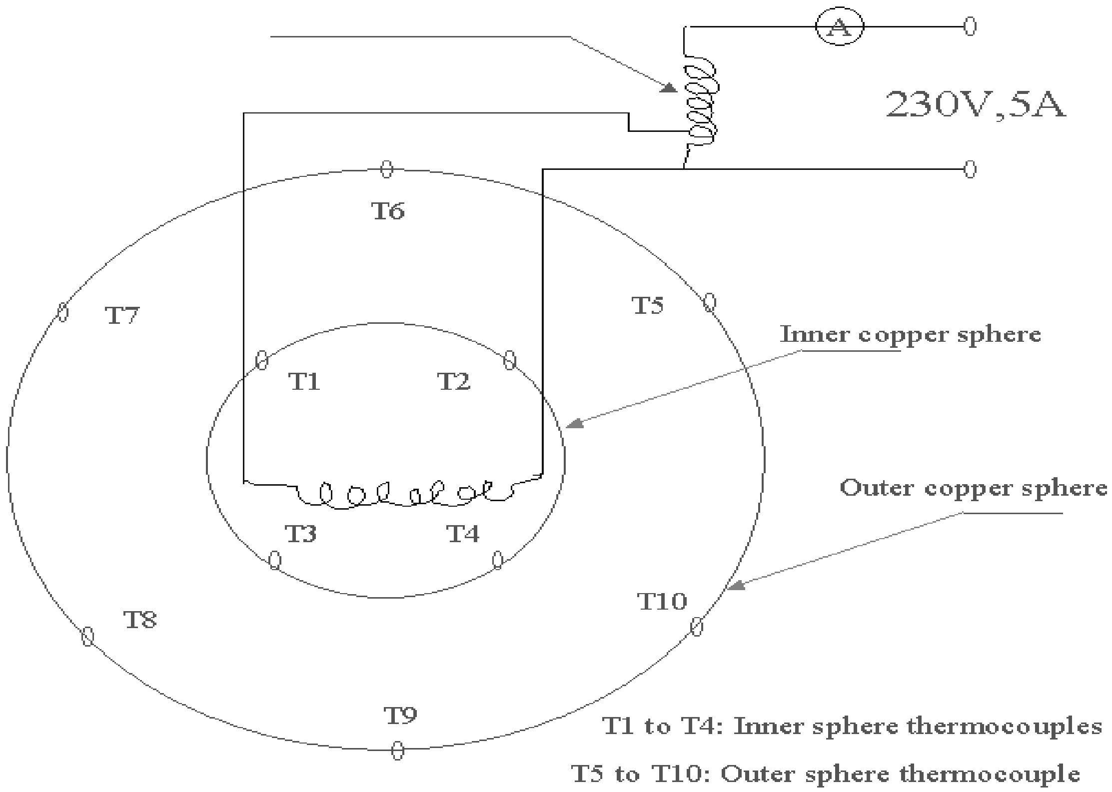


Fig.2: Circuit Diagram of the Experimental Setup.

EXPERIMENTAL PROCEDURE:

1. Keep dimmer stat knob at zero position and switch on the equipment.
2. Slowly rotate the dimmer stat knob, so that voltage is applied across the heater. Let the temperatures rise.
3. Wait until steady state is reached.
4. Note down the temperatures and input of heater in terms of volts and current.
5. Repeat the procedure for different heat inputs.

OBSERVATION TABLE:

S.No.	Temperature ^0C										Heater Input	
	T_1	T_2	T_3	T_4	T_5	T_6	T_7	T_8	T_9	T_{10}	V	I

MODAL CALCULATIONS:

1. Heat input q = V x I
2. Average inner sphere surface temperature, T = $\frac{T_1 + T_2 + T_3 + T_4}{4}$
3. Average outer spbere surface temperature, T_0 = $\frac{T_5 + T_6 + T_7 + T_8 + T_9 + T_{10}}{6}$
4. Inner sphere radius = 50 mm
5. Outer sphere radius = 100 mm

Now k = $\frac{q(r_i - r_0)}{4\pi r r_0 r_i (T_0 - T_i)}$ at $\frac{T_i + T_0}{2}$ ^{0}C

PRECAUTIONS:

1. Operate switches and controls gently.
2. If thermal conductivity of the powder other than supplied is to be determined, then gently dismantle the outer sphere and remove the powder, taking care that heater connections and thermocouples are not disturbed.

RESULTS AND ANALYSIS:

Table of results

Sl.No	Power input (W)	Thermal Conductivity(W/mK)

Plot a graph between power input and thermal conductivity

SUGGESTIONS FOR DISCUSSION

- Variation of thermal conductivity of insulators with temperature
- Different parameters which influence the heat transfer rate through insulators
- Whether the addition of insulation increases or decreases heat transfer rate?

EXP:- 5 DETERMINATION OF OVERALL HEAT TRANSFER COEFFICIENT OF LAGGED PIPE

OBJECTIVES:

- To understand the concept of critical radius of insulation
- To understand the application of insulators in industry

AIM: To Determine the Thermal Conductivity of given lagged material.

APPARATUS: Experimental Test Rig.

INTRODUCTION:

Generally insulating materials are provided around the tube carrying hot fluid in order to reduce heat losses. But there is no such insulating material, which is 100% effective to prevent the flow of heat under temperature gradient. However due to advent of new technology, it is necessary to prevent the heat flow from the system to the surroundings. In order to compensating this, lagged pipe concept came into picture.

The pipe, which is generally wrapped in one or more layers of heat insulating materials and then layer of protecting plaster on a metal wall. This arrangement is called lagging of pipe system.

This arrangement of piping is advisable for conveying high-pressure fluids such as steam, hot gases etc., in power plant industries.

The main aim of this experiment is to determine the thermal conductivity of lagging material and heat flow through lagged pipe.

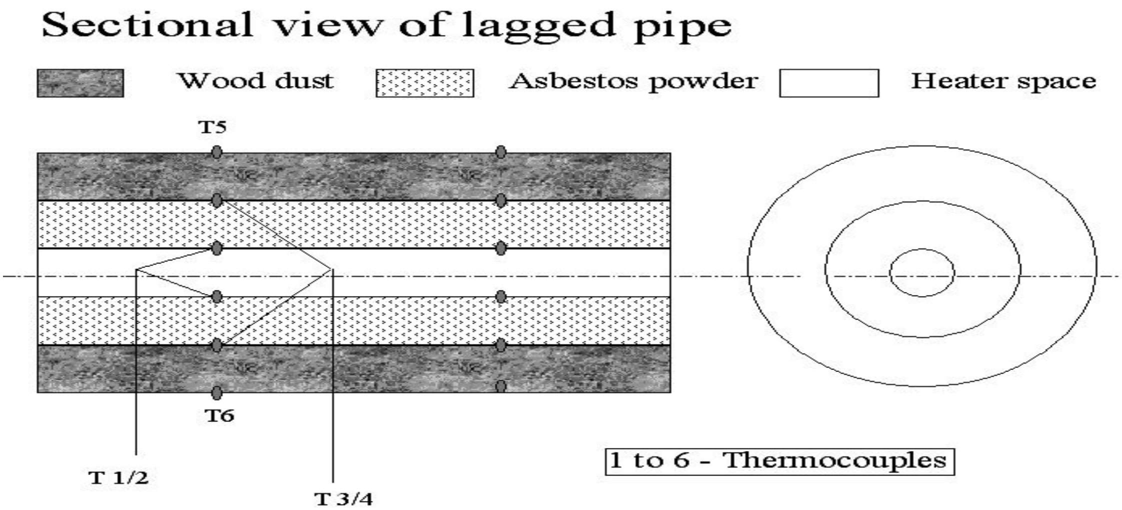


Fig (1)

EXPERIMENTAL SETUP

The sectional view of lagged pipe as shown in fig.(1) consists of three concentric pipes mounted on suitable stand. The hollow space of the innermost pipe consists of the heater. 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 third cylinder is concentric to another outer cylinder. Water flows between these two cylinders. The thermocouples are attached to the surface of cylinders approximately to measure the temperatures.

SPECIFICATIONS:

1) Pipes -
    a) GI Pipe inside 6 cm. Dia. (O.D)
    b) GI Pipe Middle 8.5 cm (Mean dia)
    c) GI pipe outer 10.7 cm. (I.D.)
    d) Length of pipes 1 meter.
2) Heater - nichrome wire heater (cartridge type) placed centrally having suitable capacity.
3) Control panel comprising of
    a) Single phase dimmer stat 0-230 1 No.
    b) Voltmeter 0-250 V 1 No.
    c) Ammeter 0-2 A 1 No.
4) Multi channel digital temperature indicator
   Range 0-300 ^{0}C using Cr / Al Thermocouples 1 No.

EXPERIMENTAL PROCEDURE:

1. Star the electric supply.
2. Then start heating given lagged material, by adjusting the heater input by rotating dimmer stat in clockwise direction.
3. Adjust water circulation through the sink and measure the quantity.
4. Allow the system to reach steady state condition.
5. When all the temperatures remain steady, note down all the observation and complete the observation table.

OBSERVATION TABLE:

Table: the readings of the experiment is as follows

Voltmeter	Ammeter	Heat supplied	Thermocouple Reading					
			T_1	T_2	T_3	T_4	T_5	T_6

NOMENCLATURE:

r_i = Inner pipe radius

r_o = Outer pipe radius

r_m = Mean radius of middle pipe

L = Length of the pipe = 1 mtr.

K = Thermal conductivity w / m ‘K

MODAL CALCULATIONS:

The pipe is so long compared with diameter that heat flows in radial direction only the following calculations were done to find the heat flow rate and thermal conductivity of lagging material-2.

a) Now from known value of heat flow rate, value of combined thermal conductivity of lagging material can be calculated

$$q = \frac{2.L.\pi.k.\,(T_i - To)}{\log_e (r_o / r_i)} = \qquad W / m\text{-}K$$

$$K = \frac{q.\log_e (r_o / r_i)}{2.\,L.\pi.k.\,(T_i - To)} = \qquad W / m\text{-}K$$

the space between the pipes of 6cm and 8.5cms contain asbestos powder, the space between pipes of 8.5cms and 10.5cms contain saw dust

K_1 (Thermal conductivity of asbestos powder)

K_2 (Thermal conductivity of saw dust)

$$K_1 = \frac{q.\log_e (r_m / r_i)}{2.L.\pi.(T_i - T_m)} = \qquad W / m\text{-}K$$

$$K_2 = \frac{q.\log_e (r_o / r_m)}{2.L.\pi.(T_m - T_o)} = \qquad W/m\text{-}K$$

b) Now, to find out the theoretical heat flow rate through the composite cylinder.

$$q = \frac{T_{inside} - T_{outside}}{\frac{1}{2\pi L}\left|\frac{1}{K_1} Log_e \frac{r_m}{r_i} + \frac{1}{K_2} Log_e \frac{r_o}{r_m}\right|}$$

From the experiment it is determined that thermal conductivity of lagged pipe is 0.1764 (asbestos powder) and 0.1282 (saw dust), the heat transfer rate across the lagged pipe is 11.7 watts.

PRECAUTIONS:

1) Keep the dimmer stat at zero position before start. 2) Increase voltage gradually.
3) Keep the assembly undisturbed while testing.
4) Do not increase voltage above 150 Volts.
5) Operate selector switch of temperature gently.

RESULTS AND ANALYSIS:

Table of results

Sl.No	Power input	Thermal conductivity

Plot the graph between thermal conductivity Vs Power for both the lagging materials (Asbestos, Saw dust)

SUGGESTIONS FOR DISCUSSION:

- Variation of thermal conductivity of asbestos and saw dust with temperature
- Reasons for increase in heat transfer with addition of insulation at smaller values of insulation thickness

EXP:- 6 DETERMINATION OF HEAT TRANSFER COEFFICIENT IN NATURAL CONVECTION

OBJECTIVES:

1. To demonstrate the basic principles of natural convection heat transfer
2. To demonstrate the boundary layer character of external natural Convection

AIM: To Determine the Heat transfer coefficient in Natural Convection between the brass tube and air

APPARATUS:Experimental Test Rig.

INTRODUCTION:

Natural or free convection is observed as a result of motion of the fluid due to density changes caused by temperature difference arising from the heating process. A hot radiator used for heating a room is one example of practical device which transfer heat by free convection The movement of fluid in free convection, whether it is a gas or a liquid, result from the buoyancy forces imposed on fluid when its density in the proximity of heat transfer surface i.e., decreases as result of heating process.

This present experimental setup is designed and fabricated to study the natural convection phenomenon from a vertical cylinder in terms of the variation of local heat transfer coefficient along the length and also the average heat transfer coefficient and its comparison with the value obtained by using an appropriate correlation.

THEORY:

Energy transfer by free convection occurs in many engineering applications such as heat transfer from a hot radiator, refrigeration coils, transmission lines, electric transformer and electronic equipment. The seasonal thermal inversion of lakes is also caused by buoyancy induced free convection motion.

In contrast, to the forced convection, natural convection phenomenon is due to the temperature difference between the surface and fluid and is not created by any external agency. 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 a hot body gets heated, rising up due to the decrease in its density and the cold fluid rushes in take place. The process is continuous and the heat transfer takes place due to the relative motion of hot and cold fluid particles. We know that the heat transfer coefficient is given by

$$h = Q / As (Ts - Ta)$$

h = average surface heat transfer coefficient ($W/m^2\ ^0C$)

q = heat transfer rate (watts)

As= Area of heat transferring surface

= $\pi d\ l\ (m^2)$

Ts = average surface temperature

$$= \frac{T_1 + T_2 + T_3 + T_4 + T_5 + T_6 + T_7}{7}\ ^0C$$

Ta = Ambient temperature in duct = $T_8\,^0C$

The surface heat transfer coefficient, of a system transferring heat by natural convection depends on the shape, dimension 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 of non-dimensional groups as follows:

$$Nu = A \times (Gr.Pr)^n$$

Where Nusselt number $Nu = h \times L / K$

Grashofer number $Gr = g L^3 \beta \Delta T / \upsilon^2$

Prandllt number $Pr = C_p.\mu / K$

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

Where,

L = characteristic dimension of the surface.

K = thermal conductivity of the fluid

υ = kinmatic viscosity of the fluid

μ = dynamic viscosity of fluid

Cp = specific heat of fluid

β = coefficient of volumetric expansion for the fluid

g = acceleration due to gravity

$\Delta T = [T_s - T_a]$

For gases $\beta = 1/(T_f + 273)\ ^0K$

$$T_f = (Ts + Ta)/2$$

For a vertical cylinder losing heat by natural convection, the constant A and n in the above equation are given by

A = 0.59 and n = 0.25 for $10^4 < Gr.Pr < 10^8$

A = 0.13 and n = 0.333 for $10^8 < Gr.Pr < 10^{12}$

All the properties of the fluid are determined at the mean film temperature (T_f).

EXPERIMENTAL SETUP:

The apparatus consists of a brass tube fitted in a rectangular vertical duct. The duct is open at the top and bottom and forms an enclosure and serves the purpose of undisturbed surrounding. One side of the duct is made up perspex for visualization. An electric heating element is kept in the vertical tube, which in turn heats the tube surface. The heat is lost from the tube to the surrounding air by natural convection. The temperature of the vertical tube is measured by seven thermocouples. The heat input to the heater is measured by an ammeter and a voltmeter and is varied by a dimmer stat. The tube surface is polished to minimize the radiation losses. Fig.1 and Fig.2 shows the tube inside the duct and tube with heating coil respectively.

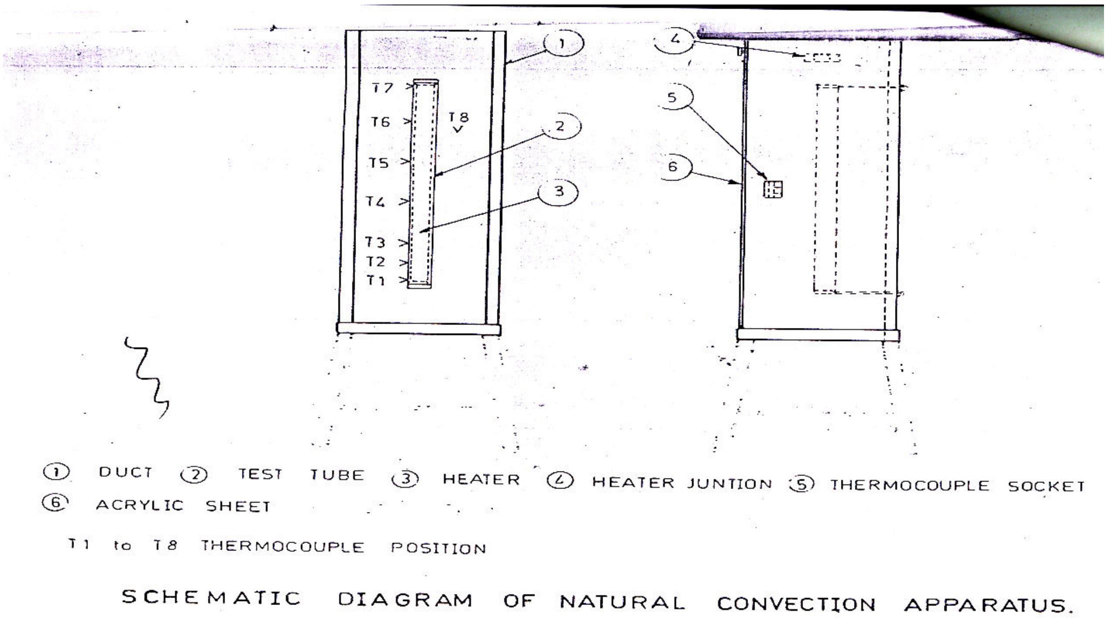


Fig.1 Experimental set up

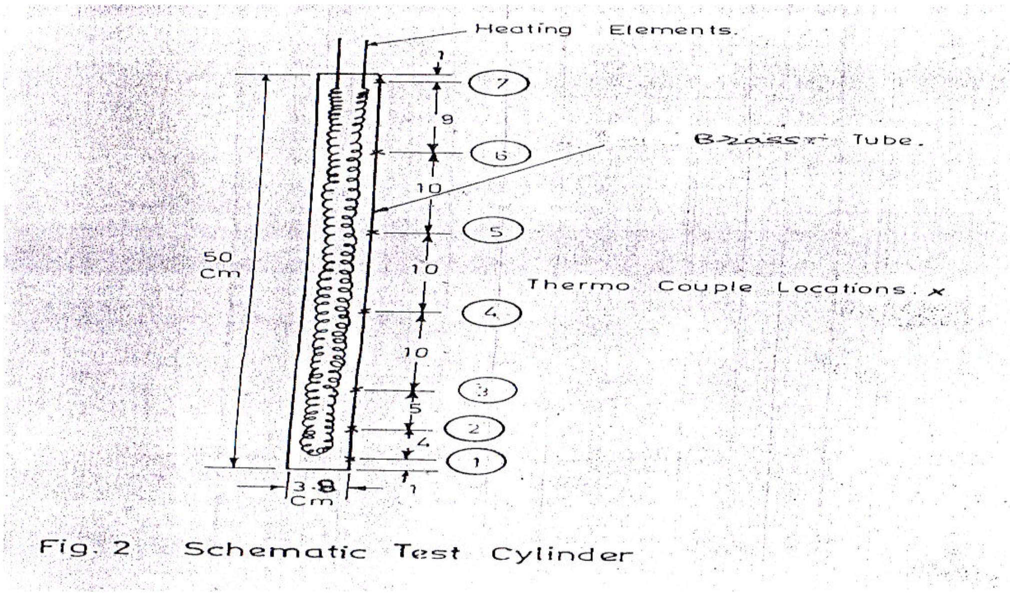


Fig.2 Schematic test cylinder

SPECIFICATIONS:

1. Diameter of the tube (d) = 38 mm
2. Length of tub (L) = 500 mm
3. Duct size 200 mm x 200 mm x 800 mm. Length
4. Multi channel digital temperature indicator 0 - 300°c using chromel/Alumel Thermocouple.
5. Ammeter 0 - 2 Amp and voltmeter 0 - 200 volts.
6. Dimmer stat 2Amp 240 Volts.

OBSEVATIONS TABLE:

S.No.	Current	Voltage	Temperature							
			T_1	T_2	T_3	T_4	T_5	T_6	T_7	T_8

CALCULATIONPROCEDURE:

EXPERIMENTAL

1. Put ON the supply and adjust the dimmer stat to obtain the required heat input.
2. Wait till the steady state is reached, which is confirmed from temperature readings (T_1 toT_7)
3. Measure surface temperature at various points T_1 to T_7.
4. Note the ambient temperature T_8:
5. Repeat the experiment at different heat inputs.

THEORITICAL

1) Take the properties of Pr, υ, K of dry air at $T_{avg} = (T_1+T_2+T_3+T_4+T_5+T_6) / 5$

2) Find the Grashof number

$$Gr = (g\beta D^3 \Delta T) / \upsilon^2 \text{ where } \beta = 1 / (Tavg + 273)$$

3) Based on the range of Gr. Pr select one of the formulae and calculate the Nussult Number (Nu)

$Nu = 1.1 (Gr.Pr)^{1/6}$ -------------for $10^{-1}<Gr.Pr< 10^{4}$

$Nu = 0.53 (Gr.Pr)^{1/4}$ -------------for $10^{4}<Gr.Pr< 10^{9}$

$Nu = 0.13 (Gr.Pr)^{1/3}$ -------------for $10^{9}<Gr.Pr< 10^{12+}$

4) Then Nu = h D / K gives the heat transfer coefficient

PRECAUTIONS:

1. Keep the dimmer stat to zero volt position before putting on main switch and Increase it slowly.

2. Operate the change over switch of temperature indicator gently from one Position to other, i.e. from 1 to 8 positions.

RESULTS AND ANALYSIS

Table of results

Sl.No	Power input	$h_{theoritical}$	$h_{experimental}$

1. Plot the local heat transfer coefficient versus distance from the leading edge.
2. Calculate the average Nusselt number and Rayleigh number for each power setting.
3. Plot the Nusselt number versus Rayleigh number for each case

SUGGESTIONS FOR DISCUSSION

1. How does the experimental data compare to the published correlations?
2. What are some possible errors in the experiment?
3. What can you tell about the structure of the boundary layer from the local heat transfer coefficient?
4. How one can identify a transition from laminar to turbulent flow?

EXP:-7 DETERMINATION OF EFFECTIVENESSS OF PARALLEL AND COUNTER FLOW HEAT EXCHANGER

OBJECTIVES:

1. To understand the basic operations of heat exchanger
2. To demonstrate the basic equations of heat exchange operations

AIM: To determine the effectiveness of Parallel Flow and Counter Flow Heat Exchanger.

APPARATUS: Heat Exchanger with Parallel Flow and Counter Flow arrangement and four thermometers range (-10 – 110^{0}C).

INTRODUCTION:

A heat exchanger is a device used for affecting the process of heat exchange between two fluids that are at different temperatures. Heat exchangers are useful in many engineering processes like those in refrigerating and air conditioning systems, power systems, Processing systems, chemical reactors and space or Aeronautical applications.

Heat exchangers are of basically of three types. (1) Transfer type, in which both fluids pass through the exchanger and heat gets transferred through the separating walls between the fluids, (2) Storage type- in this, firstly the hot fluid passes through a medium having high heat capacity and then cold fluid is passed through the medium to collect the heat. Thus hot and cold fluids are alternately passed through the medium, (3) Direct contact type- in this type, the fluids are not separated but they mix with each other and heat passes directly from one fluid to the other.

Heat exchangers may be classified in several ways. One classification is according to the fluid flow arrangement or the relative direction of the hot and cold fluids. The fluids may be separated by a plane wall but more commonly by a concentric tube (double pipe) arrangement. If both the fluids move in the same direction, the arrangement is called a parallel flow type. In the counter flow arrangement the fluids move in parallel but opposite directions. In a double pipe heat exchanger, either the hot or cold fluid occupies the annular space or the other fluid moves through the inner pipe. Since both fluid streams transverse the exchanger only once, this arrangement is called a single pass heat exchanger.

There is some special type of heat exchanger used in industrial applications. Some of them have tubes with fins, pins or spiral grooves on the outer surface. Normally a liquid flows through the tubes and a gas with a low heat transfer coefficient over the extended sulfates. Such a special heat exchangers are called compact heat exchangers.

THEORY:

LMTD METHOD OF HEAT EXCHANGER ANALYSIS

The thermal analysis of any heat exchanger involves variables like inlet and outlet fluid temperatures, the overall heat transfer coefficient, total surface area for heat transfer and the total heat transfer rate. Since the hot fluid is transferring a part of its energy to the cold fluid, there win an increase in enthalpy of the cold fluid and a corresponding decrease in enthalpy of the hot fluid. This may be expressed as

$$Q = m_h c_h (T_{hi} - T_{ho}) \quad \text{-----------------} \ (1)$$

$$Q = m_c c_c (T_{co} - T_{ci}) \quad \text{-----------------} \ (2)$$

Where m = mass flow rate.

c = constant pressure specific heat.

The subscripts c & h indicate the cold and hot fluids, whereas the subscripts i and 0 refer to the fluid inlet and outlet conditions, respectively.

If we denote the temperature difference between hot and cold fluids by

$$\Delta T = T_h - T_c \quad \text{------------------} \ (3)$$

Since ΔT is varies with position in the heat exchanger, the actual rate equation for heat transfer will be

$$Q = U\,A\,\Delta T_m \quad \text{------------------} \ (4)$$

Where ΔT_m is a suitable mean temperature difference across the heat exchanger. This average or mean value must be determined before use can be made in eq. (4). We shell now present a method for the determination of the mean temperature difference. Since the final expression obtained by this method will be in the form of a logarithmic relation, this method is referred to as Logarithmic Mean Temperature Difference (LMTD) method of analysis.

Let us consider a parallel flow heat exchanger as depicted in Fig. 1.

Fig. 1: Temperature distribution for a parallel flow heat exchanger.

- The overall heat transfer coefficient is uniform through out the exchanger.
- The potential and kinetic energy changes are negligible.
- The specific heats of the fluids are constant.
- The heat exchange takes place only between the two fluids.
- The temperatures of both the fluids are constant over a given cross section and may be represented by their bulk temperatures.

The heat transfer between the cold and hot fluids for a differential element of length, dx, is

$dQ = U\ dA\ \Delta T = U\ dA\ (T_h - T_e)$ ------------------------ (5)

where $\Delta T = T_h - T_e$. is the local temperature difference between the hot and cold fluids and dA = width* dx.

An energy balance over the hot and cold fluids for this differential element gives

$dQ = -\ m_h c_h d_{Th} = -c_h\ d_{Th}$ ------------------------ (6)

and $dQ = m_c c_c d_{Tc} = c_c\ d_{Th}$ ------------------------ (7)

where c_c & c_h are the cold and hot fluid capacity rates.

$d\Delta T = -dQ(1/c_h + 1/c_c)$ -------------------------- (8)

integrating along the heat exchanger length between sections 1 &2

$$\int_{\Delta t_1}^{\Delta t_2} dAd\Delta T / \Delta T = -U(\frac{1}{c_h} + \frac{1}{c_c})\int_1^2 dA$$

$\ln(\Delta T_2/\Delta T_1) = -UA(\frac{1}{c_h} + \frac{1}{c_c})$ ---------------------------- (9)

substituting for C_h and C_c from equations 1 and 2 respectively

$$\ln(\Delta T_2 / \Delta T_1) = -UA\{(T_{h,l} - T_{h,0}) + (T_{c,0} - T_{c,l})$$

$Q = UA(\Delta T_2 - \Delta T_1) / \ln(\Delta T_2 / \Delta T_1)$--------------------- (10)

Where

$\Delta T_1 = T_{h,l} - T_{c,l}$

$\Delta T_2 = T_{h,0} - T_{c,o}$

On comparing this result, we may see that the appropriate average temperature difference is a log mean temperature difference. So, we may write

$Q = UA\Delta T_{lm}$ ------------------- (11)

Where $\Delta T_{lm} = \Delta T_2 - \Delta T_1 / \ln(\Delta T_2/\Delta T_1)$ ---------- (12)

A counter flow heat exchanger, is one in which the fluid moves in parallel but opposite directions.The change in temperature difference between the two fluids is greatest at the entrance of a parallel flow heat exchanger but it may not be so in a counter flow arrangement.

The analysis of a counter flow heat exchanger can be done exactly in the same manner as outlined in the previous section for a parallel flow exchanger. Equations 1 & 2 are in fact valid for any heat exchanger. By taking' the differential element for a counter flow exchanger the figure un-proceeding as before it can be easily shown that the equations 10 & 11 are valid in this case too.

Hence $Q=UA\Delta T_{lm}= UA*(\Delta T_1 - \Delta T_2) / \ln (\Delta T_1 / \Delta T_2)$

Where $T1 = T_{h,i} - T_{c,o}$. $T_2=T_{h,o} - T_{c,l}$ finally the equations may be developed into a final equation as below.

$$\Delta T_{lm} = \Delta T_1 = \Delta T_2$$

EFFECTIVENESS OF PARALLEL FLOW HEAT EXCHANGER

Let us now determine the specific form of the effectiveness, NTU relation for a parallel flow heat exchanger. Assuming $C_{min} = C_{max}$

$$\varepsilon = T_{co} - Tdl / T_{Hl} - T_{cl} \qquad \text{---------------------} (14)$$

From equations 1 &2 we get

$$c_{min} / c_{max} = m_h c_h / m_c c_c = T_{hi}-T_{ho} / T_{co} - T_{ci} \text{ -------------} (15)$$

Rearranging the equation 9 in the form of

$$Ln (\Delta t_2/t_1) = \ln (t_{ho}-t_{co})/(t_{hi}-t_{ci}) = -UA/c_{min}(1+c_{min}/c_{max})$$

Which on substitution of the value of T_{ho}

$$T_{ho}- T_{co} / T_{hi} - T_{ci} = (T_{hi}-T_{ci})-c_{min}/c_{max} (T_{co}-T_{ci}) - (T_{co}-T_{ci}) / [T_{hi}-T_{ci}]$$

$$= 1-\varepsilon(1+c_{min}/c_{max})$$

going back to equation 15 we get

$$\varepsilon=1-\exp \{-NTU [1 + (cmin/c_{max})]\} \qquad \text{---------------------} (16)$$

EFFECTIVENESS OF COUNTER FLOW HEAT EXCHANGER

From the analysis like that made in the previous section, the following relation for effectiveness in a counter flow heat exchanger can be obtained

$$\varepsilon = 1-\exp\{-NTU[1-(c_{min}/c_{max})\}/1-(c_{min}/c_{max}\{\exp-NTU[1-(c_{min} / c_{max}]\} \text{ ----} (16)$$

OVERALL HEAT TRANSFER COEFFICIENT

The thermal design of a heat exchanger involves the calculation of the surface area required to transfer heat at a given rate for fluid flows and fluid temperatures

$$Q=UA\Delta T_m \qquad \text{--------------------------} (17)$$

Where ΔT_m is an average temperature effective difference for entire heat exchanger.

EXPERIMENTAL SETUP

The experimental setup shown in figuer consists of two concentric tubes in which fluids pass. The hot fluid is hot water, which is obtained from an electric geyser. Hot water flows through the inner tube, in one direction. Cold fluid is cold water, which flows through the annulus. Control valves are provided so that direction of cold water can be kept parallel or opposite to that of hot water. Thus, the heat exchanger can be operated either as paralle1 or counter flow heat exchanger. The temperatures are measured with thermometer. Thus, the heat transfer rate, heat transfer coefficient, LMTD and effectiveness of heat exchanger can be calculated for both parallel and counter flow.

SPECIFICATIONS:

(1) Heat exchanger - (a) Inner tube - ϕ12 mm OD. ϕ 11 mm ID copper tube.

(b) Outer tube - ϕ 25 mm NB G. I. Pipe.

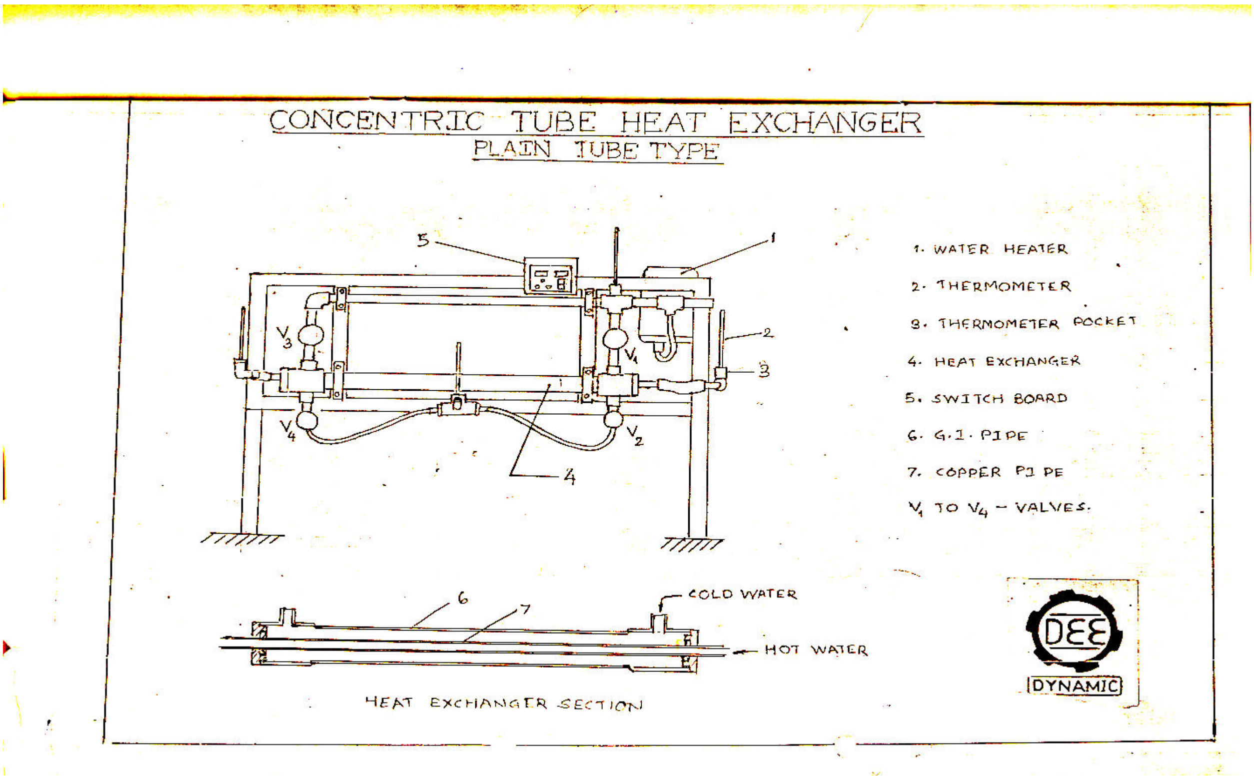
(c) Length of HX is 1 m.

(2) Electric heater - 3 KW Capacity to supply hot water.

(3) Valves for flow and direction control- 5 No’s.

(4) Thermometers to measure temperatures - 10 to 110°C - 4 No's.

(5) Measuring flask and stop clock for flow measurement.



EXPERIMENTAL PROCEDURE:

1. Start the water supply. Adjust the water supply on hot and cold sides.
2. Keep the valves V_2 & V_3 dosed and V_1 & V_4 opened so that arrangement is parallel flow.

3. Switch ON the geyser. Temperature of water will start rising. After temperatures become steady, note down the readings in the observation table.
4. Repeat the experiment by changing the flow. Now open the valves V_2 & V_3 and then close the valves V_1 & V_4. The arrangement is now counter flow. Wait until the steady state is reached and note down the readings.

OBSERVATIONS TABLE:

Type of Flow	Hot Water			Cold Water		
	Time for 1 Lit Water in Sec	Temperature ^0C		Time for one Liter of Water in seconds	Temperature ^0C	
		In	Out		In	Out
Parallel flow						
Counter flow						

MODAL CALCULATIONS:

I

1. Hot water inlet temperature for parallel flow = ^0C
2. Hot water inlet temperature for counter flow = ^0C
3. Hot water outlet temperature for parallel flow = ^0C
4. Hot water outlet temperature for counter flow = ^0C

II

Hot water flow rate, m_h

Let time required for 1 lt of water be X_h seconds

Mass of 1 lt. Water = 1 Kg

$M_h = 1/X_h$ kg/sec = for parallel flow = 0.6489 for counter flow = 0.0344

III

Heat given by hot water (inside heat transfer rate)

$Q_h = M_h * C_p * (T_{hi} - T_{ho})$ watts

Where C_p = specific heat of water = 4200J/ kgK

Parallel q/h = 0.06489 * 4200 * (38.5-36) = 681.37 watts

Counter flow = 722.63 watts

IV

Similarly for cold water parallel flow counter flow

T_{co} =

T_{ci} =

Heat collected by cold water (outside heat transfer rate)

M_c = for parallel flow

= for counter flow

r_c = for parallel flow watts

= for counter flow watts

V

Logarithmic mean temperature difference (LMTD)

LMTD = $\Delta T_m = T_i - T_o / \text{in} (T_i / T_o)$

Parallel flow = Counter flow =

Where for parallel flow $T_i = T_{hi} - T_{ci}$

$T_o = T_{ho} - T_{co}$

Parallel flow Counter flow

T_i =

T_o =

VI

Overall heat transfer = U

(a) Inside overall heat transfer coefficient = U_i

Inside diameter = 0.011m

Inside surface area of the tube = $A_i = \pi * D_i * L$ = m^2

Now $T_h = U_i \Delta T_m * A_i$

$U_i = T_h / (\Delta T_m * A_i)$ $W/m^2\,^0C$

Parallel flow = $W/m^2\,^0C$ Counter flow = $W/m^2\,^0C$

(b) Outside overall heat transfer coefficient = U_o

Outside diameter of the tube = m

Outside area of the tube = $A_o = \pi * D_o * L$ = m^2

Similarly $T_c = U_o * \Delta T_m * A_o$

U_o Parallel flow = $W/m^2\,^0C$ Counter flow = $W/m^2\,^0C$

VII

Effectiveness of heat exchanger ε = Rate of heat exchange in H.X / Max. Possible heat transfer rate ε. = $m_h * c_p * (T_{hi} - T_{ho}) / m * c_p * (T_{hi} - T_{ci})$

Parallel flow ε = counter flow ε =

PRECAUTIONS:

(1) Never switch ON the geyser unless there is water supply through it.

(2) If the red indicator on geyser goes off during operation, increase the water supply,

because it indicates that water temperature exceeds the set limit.

(3) Ensure steady water flow rate and temperatures before noting down the readings, as fluctuating water supply can give erratic results.

RESULTS AND ANALYSIS:

TABLE OF RESULTS

Type of flow	Heat transfer in Watts		Effectiveness	LMTD ^0C	U_i W/m^2 ^0C	U_o W/m^2 ^0C
	Inside	Outside				
Parallel Flow						
Counter Flow						

(i) Plot the overall heat transfer coefficient versus water velocity

(ii)Plot the effectiveness versus number of transfer units for this exchanger and compare it to the theoretical relationship.

SUGGESTIONS FOR DISCUSSION

1. How does the experimental value of the overall heat transfer coefficient compared with expected values for this type of heat exchanger?

2. How does the effectiveness - NTU relationships compare with theory?

3. Discuss the differences in performance for the parallel flow and counter flow heat exchangers.

4. What errors may be present in your experimental analysis?

EXP:-8 DETERMINATION OF EFFECTIVENESS AND EFFICIENCY OF PIN-FIN

OBJECTIVES:

1. To understand the mechanisms of heat transfer through the fin
2. To understand the insulated tip condition used in fin analysis
3. To know the conditions for maximum heat transfer through the fins

AIM: To Determine the Efficiency and Effectiveness of Pin-Fin in Natural Convection and Forced Convection modes.

APPARATUS: Experimental test rig.

INDRODUCTION:

To study the rate of heat transfer by convection Newton's law of cooling or heating is very helpful which states that rate of heat transfer is directly proportional to the surface area and temperature difference. It is given by, Q = h A (Ts- Ta). Where h is heat transfer coefficient, Ts is surface temperature and Ta is ambient temperature.

The above relation reveals that the convective heat flow from a surface can be enhanced by increasing the heat transfer coefficient 'h' or the surface area 'A' or the temperature difference (Ts- Ta). The convective coefficient is a function of the geometry, fluid properties and the flow rate. Control of 'h' through these parameters help to obtain its optimum value. With regard to the effect of excess temperature (Ts- Ta), difficulties are encountered when the ambient temperature Ta is too high particularly in hot weather conditions. The surface area exposed to the surroundings can be increased by the attachment of protrusions to the surface which are called as fins.

A fin is an extended surface used to increase the heat transfer rates from the surface to the surrounding fluid wherever is not possible to increase the value of the surface heat transfer coefficient or the temperature difference between the surface and the fluid. Fin can be fabricated in variety of forms. The various types of fins are (1). Longitudinal fins. (2) Circumferential fins. (3) Pin fins.

Common application of finned surface are with economizer in steam power plants, convectors for steam and hot water heating systems, electrical transformers and with air-cooled cylinders of aircraft engines, i.c. Engines and air compressors. The objective of this experiment is to find the effectiveness and efficiency of the pin fin.

THEORY:

The problem of determination of heat flow through a fin requires the knowledge of temperature distribution through it. This can be obtained by considering the fin as a metallic plate connected at its base to heated wall and transferring heat to a fluid by convection. The heat flow through the fin is by conduction and from the fin to surrounding fluid is by convection. Thus the temperature distribution in a fin will depend upon the properties of both the fin material and the surrounding fluid. For the analysis of heat flow through the fin the following assumptions were made.

1. A fin material is homogenous and isentropic; hence the thermal conductivity of the fin material is uniform.
2. The temperature at any cross section of the fin is uniform. In other words the heat conduction is one-dimensional.
3. The heat transfer coefficient 'h' is uniform over the entire surface.
4. There is no internal heat generation in the fin.
5. Contact thermal resistance is negligible.
6. Radiation is negligible.
7. The heat conduction is steady state.

Let A = cross sectional area of the fin (m)

P = circumference of the fin (m)

L = Height of the fin (m)

T_1 = Base temperature of the fin (^0C)

T_f = Duct fluid temperature (^0C)

θ = temperature difference between fin and fluid = $T - T_f$

h = Heat transfer coefficient, W/m ^0C

K_f = thermal conductivity of fin material

= 110 W/mK for Brass

= 46 W/mK for Mild Steel

= 232 W/mK for Aluminium

In practice, the heat transfer area at the fin tip is generally small when compared to that through the surface area of the fin. Under these conditions the flow of heat at the tip is negligible and this condition can be mathematically expressed as $(d\theta/dx)_{x=L} = 0$. The mathematical formulation of this problem becomes:

$$(d^2\theta / dx^2) - m^2\theta = 0 \text{ in } 0 \le x \le L$$

$$\theta = T_o - T_\infty = \theta_o \text{ at } x = 0$$

$$d\theta / dx = 0 \qquad \text{at } x = L$$

To solution of the above differential equation is

$$\theta = C_1 e^{-rnx} + C_2\, e^{rnx}$$

Application of first boundary condition gives

$$\theta_o = C_1 + C_2 \text{ --- } (1)$$

and applying the second boundary condition, we get

$$d\theta / dx = -m\, C_1\, e^{-rnx} + m\, C_2\, e^{rnx}$$

$$0 = -m\, C_1\, e^{-mL} + m\, C_2\, e^{mL}$$

$$C_1 = C_2\, e^{2\,m\,L} \text{ --- } (2)$$

Combining equations (1) and (2)

$$\theta_o = C_2\,(1 + e^{2mL})$$

Or $C_2 = \theta_o / (1 + e^{2mL})$ and $C_1 = \theta_o / (e^{-2mL} + 1)$

Or $\theta = \theta_o\, [(e^{-mx} / (1 + e^{2mL})) + (e^{mx} / (e^{-2mL} + 1))]$ ------- (3)

Equation (3) can be written in a more compact form by multiplying the numerator and denominator of the first term on R.H.S by e^{mL} and those of second term by e^{-mL}.

$$\theta = \theta_o\, [(e^{-m(x-L)} / (e^{-mL} + e^{mL})) + (e^{m(x-L)} / (e^{-mL} + e^{mL}))] \quad \text{(Or)}$$

$$\theta = \theta_o\, [(e^{-m(x-L)} + e^{m(x-L)}) / (e^{-mL} + e^{mL})] = \theta_0 \cosh m\,(L-x) / (\cosh mL)$$

Since $\cosh \beta = (e^{\beta} + e^{-\beta}) / 2$

$$(T - T_\infty) / (To - T_\infty) = \cosh [m\,(L-x)] / (\cosh (mL)) \text{ ------------------------ } (4)$$

The equation (4) is the equation. for temperature distribution along the length of the fin. Temperature T_1 and T_f will be known from the given situation and the value of 'h' depends upon mode of convection i.e., natural or forced.

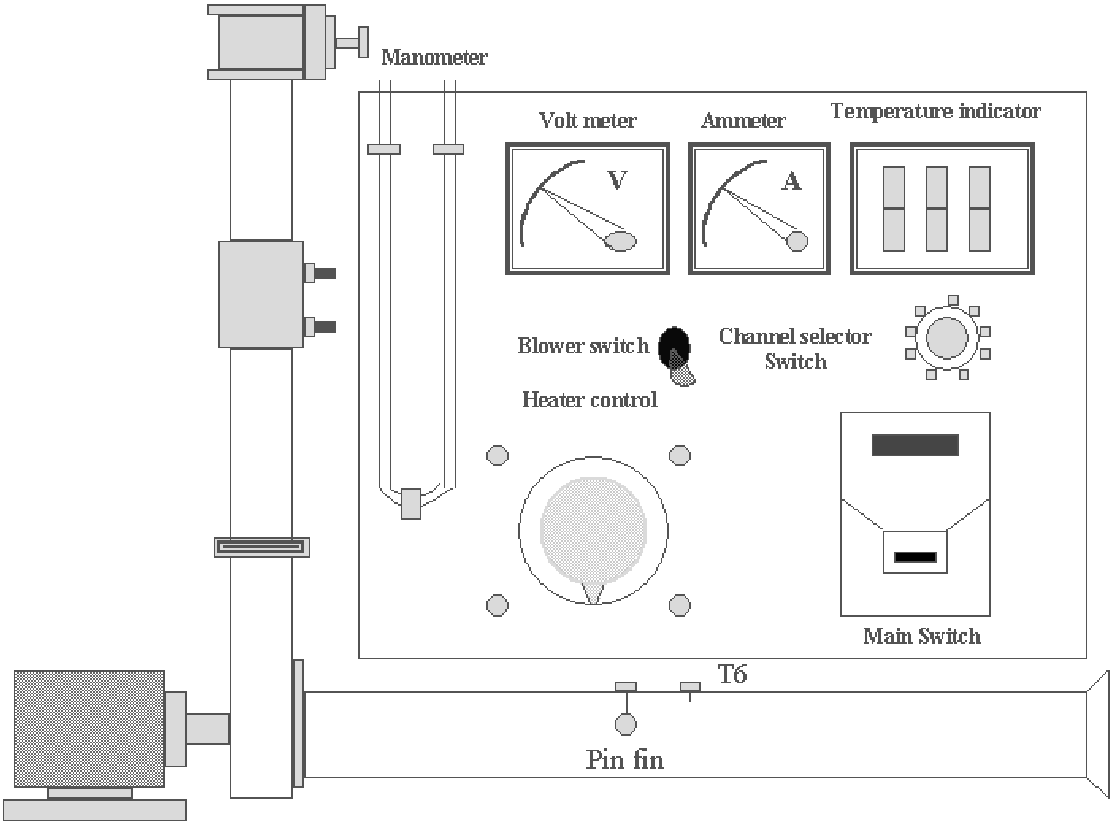


Fig. 1 Experimental Setup for finding the efficiency and effectiveness of pin fin

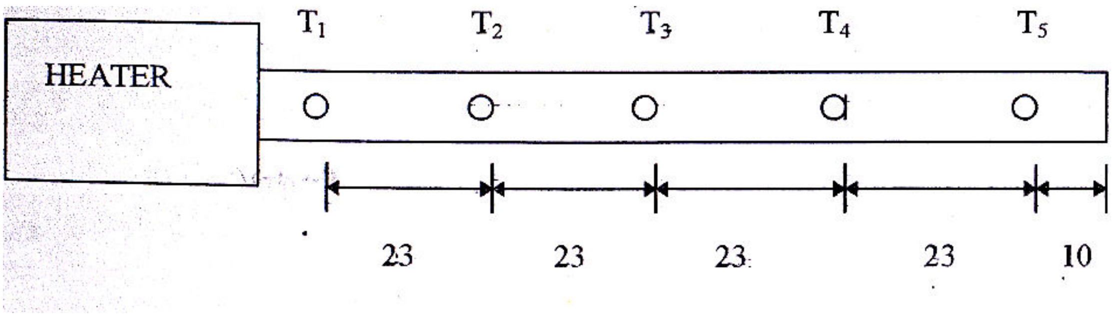


Fig.2 Fin with thermo couple positions

EXPERIEMNTAL SETUP:

The experimental set up as shown in Fig1 consists of a simple pin fin, which is fitted, in a rectangular duct. The duct is attached to suction end of blower. An electric heater heats one end of the fin. Five thermocouples are mounted on the surface along the length of the fin and one thermocouple is left openly in the duct. These thermocouples measure the duct fluid temperature and temperatures on the surface of the fin. One manometer with water as manometric fluid is provided to measure the air flow rate in case of forced convection.

SPECIFICATIONS

1) Fins - 12 mm O.D., effective length 102 mm with 5 Nos. of thermocouple positions along the length, made of brass, mild steel and aluminum - one each. Fin is screwed in heater block, which is heated by a band heater
2) Duct - 150 X 100 mm cross-section, 1000 mm long connected to suction side of blower.
3) F.H.P. centrifugal blower with orifice and flow control valve on discharge side.
4) Orifice - dia. 22 mm, coefficient of discharge C_d = 0.64
5) Measurements and controls;
   a) Dimmer stat to control heater input, 0-230 V, 2 amp
   b) Voltmeter 0-250 V, for heater supply voltage
   c) Ammeter 0-2 amp, for heater current
   d) Multi channel digital temperature indicator.
   e) Water manometer connected to orifice meter.

PROCEDURE:

THE EXPERIMENTAL PROCEDURE IN NATURAL CONVECTION MODE

1) Switch on the main supply
2) Adjust dimmer stat and apply some voltage and current.
3) Wait until the steady state is reached.
4) Note down the surface temperatures of the fin and duct fluid temperature.
5) Repeat the experiment at different power inputs.

THE EXPERIMENTAL PROCEDURE IN FORCED CONVECTION MODE

1) Start the blower
2) Adjust dimmer stat and apply some voltage and current.
3) Wait until the steady state is reached and tabulate the temperatures and the manometer difference.
4) Repeat the experiment for different inputs at different airflow rates

OBSERVATIONS:

S.No	Input		Fin Temperature					Duct Fluid Temperature (^0C) T_6
	Voltage	Current	T_1	T_2	T_3	T_4	T_5	

Table - 1: The readings in natural convection mode

S.No	Power input		Manometer difference		Fin Temperature					Duct Fluid Temperature (^0C) T_6
	Voltage	Current	h_1	h_2	T_1	T_2	T_3	T_4	T_5	

Table -2. The readings in forced convection mode

CALCULATIONS:

EXPERIMENTAL

1) T_m = Average fin temperature = $(T_1+T_2+T_3+T_4+T_5) / 5$
2) $\Delta T = T_m - T_6$
3) Heat supplied = Power input = Q = V.I
4) $Q = h \cdot A \cdot \Delta T$ gives $h = Q / A \cdot \Delta T$
5) Actual heat transfer through the fin $Q_a = (hpk_f A)^{0.5} .(T_1 - T_6).\tanh mL$
6) Maximum possible heat transfer through the fin $Q_m = h \cdot \pi Dl \cdot (T_1 - T_6)$.
7) Heat transfer with out fin $Q_{w.o.f} = h \cdot (\pi / 4) D^2 \cdot (T_1 - T_6)$.
8) Efficiency = Q_a / Q_m and Effectiveness = $Q_a / Q_{w.o.f}$.

THEORITICAL

<u>Natural convection</u>

1) Take the properties of Pr, υ, K of dry air at $T_{avg} = (T_1+T_2+T_3+T_4+T_5+ T_6) / 5$
2) Find the Grashof number

   $Gr = (g\beta D^3 \Delta T) / \upsilon^2$ where $\beta = 1 / (Tavg + 273)$

3) Based on the range of Gr . Pr select one of the formulae and calculate the Nussult Number (Nu)

   $Nu = 1.1 (Gr.Pr)^{1/6}$ -------------for $10^{-1} < Gr.Pr < 10^{4}$

$Nu = 0.53\ (Gr.Pr)^{1/4}$ -------------for $10^{4} < Gr.Pr < 10^{9}$

$Nu = 0.13\ (Gr.Pr)^{1/3}$ -------------for $10^{9} < Gr.Pr < 10^{12+}$

4) Then $Nu = h\ D / K$ gives the heat transfer coefficient

5) Actual heat transfer through the fin $Q_a = (hpk_f\ A)^{0.5}\ .(T_1 - T_6).\tanh mL$

6) Maximum possible heat transfer through the fin $Q_m = h\ .\ \pi Dl\ .\ (T_1 - T_6)$.

7) Heat transfer with out fin $Q_{w.o.f} = h\ .\ (\pi / 4)\ D^2\ .\ (T_1 - T_6)$.

8) Efficiency = Q_a / Q_m and Effectiveness = $Q_a / Q_{w.o.f}$.

Forced convection

1) In forced convection heat transfer is the function of Reynolds number and Prandtl Number.

2) Find the Reynolds number which is given by

$Re = (V_{tmf}\ .\ D) / \upsilon$

$V_{tmf} = V\ .\ (T_{mf} + 273) / (T_f + 273)$

$Q = C_d\ (\pi / 4)\ d^2\ (2gH(\rho_w - \rho_a)^{0.5}\ m^3/sec$.

$V = Q$ / duct cross sectional area

$= Q / (0.15 \times 0.1)$ m/sec

3) Find the Nussult number by selecting one of the formulae given below based on range of Reynolds number

$Nu = 0.615\ (Re)^{0.466}$ -------------- for $40 < Re < 4000$

$Nu = 0.174\ (Re)^{0.466}$ -------------- for $4000 < Re < 40000$

4) Then $Nu = h\ D / K$ gives the heat transfer coefficient

5) Actual heat transfer through the fin $Q_a = (hpk_f\ A)^{0.5}\ .(T_1 - T_6).\tanh mL$

6) Maximum possible heat transfer through the fin $Q_m = h\ .\ \pi Dl\ .\ (T_1 - T_6)$.

7) Heat transfer with out fin $Q_{w.o.f} = h\ .\ (\pi / 4)\ D^2\ .\ (T_1 - T_6)$.

8) Efficiency = Q_a / Q_m and Effectiveness = $Q_a / Q_{w.o.f}$.

PRECAUTIONS

1) Do not obstruct the suction of the duct or discharge pipe.

2) Open the duct cover over the fin during the experiment in natural convection mode.

3) Fill up water in the manometer and close duct cover during the experiment in for forced convection mode.

4) Proper earthing to the unit is necessary.

5) While replacing the fins, be careful for fixing the thermocouples.

RESULTS AND ANALYSIS:

Table of results

Sl.No	Power input	Mode of heat transfer (Forced/Natural)	Heat Transfer coefficient	Heat transferred	Efficiency	Effectiveness

Draw the following plots

1. Power input Vs Efficiency
2. Power input Vs Effectiveness

SUGGESTIONS FOR DISCUSSION

- The variation of efficiency and effectiveness with temperature
- Circumstances under which the addition of fins may actually decrease heat transfer?
- Whether the use of a larger number of small diameter fins or a single large diameter fin will enhance the effectiveness.

EXP:-9 DETERMINATION OF THERMAL CONDUCTIVITY OF METAL ROD

OBJECTIVES:

- Variation of thermal conductivity of metals with temperature.
- Effect of lattice structure on the thermal conductivity of metals

AIM: To Determine the thermal conductivity of copper metal rod.

APPARATUS: Experimental Test Rig.

THEORY:

MECHANISM OF THERMAL ENERGY CONDUCTION IN METALS:

Thermal energy can be conducted in solids by free electrons and by lattice vibrations. Large number of free electrons moves about in the lattice structure of the material, in the good conductivity. These electrons carry thermal energy from high temperature to lower temperature region in a similar way they transport electron charge, in fact these electrons frequently referred as electrons gas. Energy may also be transferred as vibrational energy in the lattice structure of the material. In general however, this mode of energy transfer is not as large as electron transport and hence good electrical conductors are always good heat conductors. E.g., Copper, silver.

However, with increase in temperature lattice vibration come in the way of transport by free electrons and for most of the metals thermal conductivity decreases with increase in temperature. The governing equation to find the thermal conductivity is Fourier's equations which is given by the,

$$Q = - k A \frac{dT}{dx}$$

Thermal conductivity of some materials.

Solids (metals)	T.C. (w/m^0C)	Stall (^0C)
Pure copper	380	20
Brass	110	20
Steel	54	20
Stainless steel	17	20

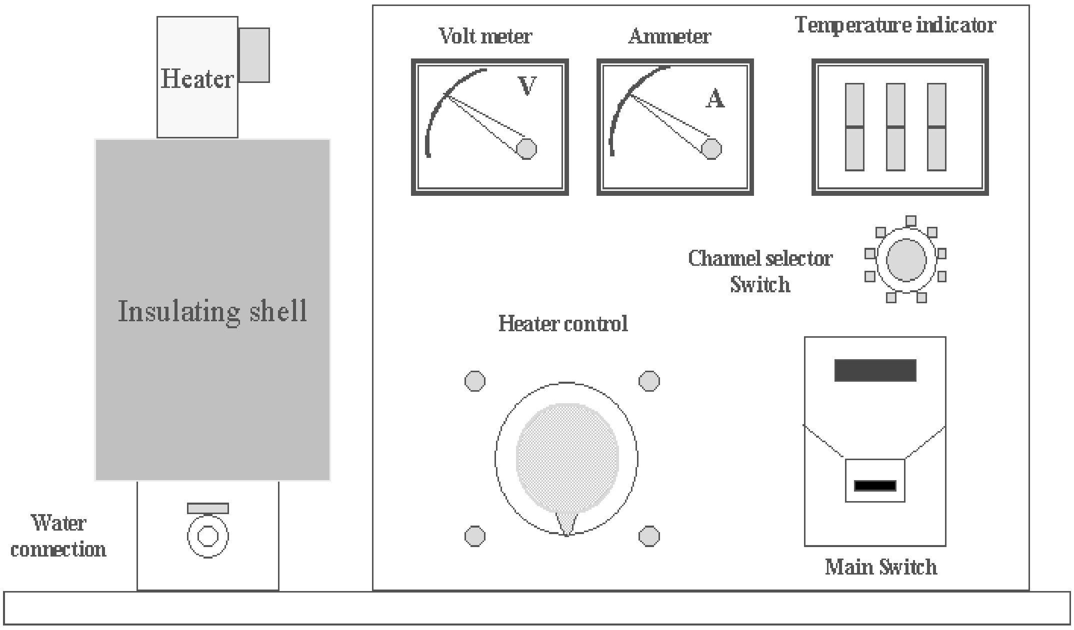


Figure.1 Experimental setup

EXPERIMENTAL SETUP:

The schematic diagram of experimental set up as shown in Figure.1 consists of a copper bar, one end of which is heated by an electrical heater and the other end is cooled by water. The middle portion i.e., test section of the bar is covered by a shell containing insulation.

The bar temperature is measured at 8 different sections while 2 thermocouples measure the temperature at the shell, two thermocouples are provided to measure water inlet and out let temperature.

A dimmer stat is provided for the heater to control its input. Constant water flow is circulated through the heat sink. A gate valve provided to control the water flow.

Specifications:

1. Metal bar - Cu, 25 mm O.D, approximately 430 mm long with insulator shell along the heat lost length and water control heat sink at the other end.
2. Test length of the bar 240 mm.
3. Thermocouples- chromel / alumel –12No.s.
4. Band nichrom heater to heat the bar.
5. Dimmer stat to control the heater input 2amp, 230 V.
6. Voltmeter & ammeter to measure heater T */P.*
7. Multi channel digital temperature indicator 0.1^{0}C, Least count, 0-200°c with channel selection switches.
8. Measuring flask to measures water flow.

THE EXPERIMENTAL PROCEDURE:

1. Star the electric supply.
2. Then start heating the bar by adjusting the heater input by rotating Dimmerstat in clockwise direction.
3. Start cooling water supply through the heat sink and adjust it to around 350- 400 cc per minute.
4. Bar temperature will start rising, then we go on checking the temperature at time intervals of 10min.
5. When all the temperatures attain steady state we should note down all the observation and complete the observation table.

OBSERVATIONS:

Test Bar Temperature								Shell		Water Temp.		
T_1	T_2	T_3	T_4	T_5	T_6	T_7	T_8	T_9	T_{10}	T_{11}	T_{12}	Flow Rate

MODAL CALCULATIONS:

Heat is flowing through the bar from heater end to water heat sink. When steady state is reached, heat passing through the section CC is equal to the heat absorbed by the water circulated in the sink.

1) <u>Heat passing through section CC</u>

$Q = m.Cp.\Delta T$ watts

Where,

m = mass flow rate of cooling water, Kg / s.

Cp = Specific heat of water

= 4180 J / *Kg* 'C

ΔT = (Water outlet temp.) - (Water inlet temp.) 'C

=

Heat flux $q_{cc} = Q/A =$ KJ/m^2

Now,

$q_{cc} = [- K_{cc} [dt/dx] A]_{cc}$

A = Cross sectional area *of* the bar = 0.00049 m^2

$K_{cc} = q_{cc} / (dt/dx)$;

= W / m°C

2) Heat passing through section BB

$q_{bb} = q_{cc}$ + radial heat loss between CC & BB.

$$= qcc + \frac{2\Pi k L_1 (T_6 - T_{10})}{\log_e (r_o / r_i)}$$

Where,

k = thermal conductivity *of* insulation

= W / m°C

L_1 = length *of* insulation cylinder

= m

r_o = outer radius

= m.

r_i = inner radius

= m.

= *W/m-K*

3) Similarly. heat passing through section AA.

$q_{aa} = q_{bb}$ + Radial heat loss between BB & AA.

$$= qcc + \frac{2\Pi k L_2 (T_6 - T_{10})}{\log_e (r_o / r_i)}$$

where, L2 = 0.09 m

= *W/m-K*

PRECAUTIONS:

(1) Keep the dimmer stat zero before starting.

(2) Increase the voltage and current slowly.

(3) Keep all the assembly undistributed.

(4) Do not increase voltage above maximum limit

RESULTS AND ANALYSIS:

Table of results

Sl.No	Power input	Water flow rate	Thermal conductivity

Draw the plot between power input and thermal conductivity

SUGGESTIONS FOR DISCUSSION

- Relation between thermal conductivity of real gases with pressure
- Relation between thermal diffusivity and thermal conductivity
- Comparison of good electrical conductors are good thermal conductors