

HEAT TRANSFER LABORATORY MANUAL

Mission and vision of the Department

Vision of Mechanical Department

To establish the state of the art learning center in Mechanical Engineering which will impart global competence, enterprising skills, professional attitude and human values in the student.

Mission of Mechanical Department

To impart quality technical education to the students.

To develop comprehensive competence in the students through various modes of learning.

To enable students for higher studies and competitive examinations.

To facilitate students and industry professionals for continuous improvement and innovation.

Program Educational Objectives:

[1] Use core competence acquired in various areas of Mechanical Engineering to solve techno-managerial issues for creating innovative products that lead to better livelihoods & economy of resources.

[2] To establish themselves as effective collaborators and innovators to address technical, managerial and social challenges.

[3] To equip students for their professional development through lifelong learning and career advancement along with organizational growth.

[4] Serve as a driving force for proactive change in industry, society and nation.

PROGRAM SPECIFIC OUTCOMES

Student should have

An ability to work professionally in mechanical systems including design, analysis, production, measurement and quality control.

An ability to work on diverse disciplinary tasks including manufacturing, materials, thermal, automobile, robotics, mechatronics, engineering software tools, automation and computational fluid dynamics.

MAHATMA GANDHI MISSION'S **JAWAHARLAL NEHRU ENGINEERING COLLEGE,**

AURANGABAD. (M.S.)

DEPARTMENT OF MECHANICAL ENGINEERING

HEAT TRANSFER LABORATORY

MANUAL



COMPILED BY: - Mr. Mahendra E. Dahake

MGM'S JNEC MECH. ENGG. DEPT.

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MR

'HEAT TRANSFER' EXPERIMENTS**SUBJECT: - Heat Transfer****CLASS: - Third Year Mechanical Engineering****LIST OF EXPERIMENTS**

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Time Allotted for each Practical Session = 02 Hrs.

EXPERIMENT NO: I

Aim: To determine the thermal conductivity of insulating powder.

Apparatus: Standard equipment for the determination of thermal conductivity of insulating powder.

The apparatus consists of two thin walled concentric copper spheres. The inner sphere houses the heating coil. The insulating powder (in our case- Chalk Powder) is packed between the two shells. The power supply to the heating coil is adjustable. Fe-Constantan thermocouples are used to measure the temperatures. Thermocouples 1 to 4 are embedded on the outer surface of the inner sphere and thermocouples 5 to 8 are embedded on the outer shell.

Theory:

It is advised that students should write the theory on the subject in their own words under the guidance of the teacher.

Fourier's Law of Heat Conduction:

Assuming no heat generation, no accumulation of heat and transfer of heat by conduction only at steady state we have,

$$q_x = Q$$

$$q_x = -KA \frac{dT}{dx}$$

Where $\frac{qx}{A}$ is the heat flux in W/m^2 while the quantity $\frac{dT}{dx}$ represents the temp. Gradient in X-direction.

Conduction through Hollow Sphere:

Fourier's Law for constant thermal conduction with distance 'dr', where r is the radius of sphere. r_1 is the radius at temp. T_1 . And r_2 is the radius at the temp. T_2 .

$$\frac{q}{A} = -K \frac{dT}{dr} \quad \& \quad A = \pi r^2$$

Integrating above equation from r_1 to r_2 .

$$\frac{q}{4\pi} \int_{r_1}^{r_2} \frac{dr}{r^2} = -K \int_{T_1}^{T_2} dT$$

$$q = \frac{4\pi k r_1 r_2 (T_1 - T_2)}{(r_1 - r_2)}$$

Insulation:

Covering the surface with another surface with another material of low thermal conductivity in order to prevent excess heat transfer to the surrounding is termed as Insulation. In order to insulate material, it is poor conductor of heat and hence to cover the surface of heat. It is used where excess heat transfer is prevented. Electrical conductors are almost always good conductor of heat viz. Copper, Aluminum and Silver. & electrical conductors are good heat insulators.

Commonly known heat insulators are Glass, Wood, Window glass, Saw dust, Chalk, Loosely packed or boards of sheet of asbestos.

HEAT TRANSFER LABORATORY MANUAL

Procedure:

1. Keep the dimmer-stat at minimum voltage position. Switch ON the electric supply.
2. Adjust the dimmer-stat to supply a maximum 40W to the heating coil. Maintain this constant throughout the experiment.
3. Wait for steady state to be attained.
4. Note down the reading in the observation table as given below.

Observations:

Radius of the inner sphere= $R_i=50$ mm.

Radius of the outer sphere= $R_o=50$ mm.

Observation Table:

Sr. No.	1	2	3	4
Voltage (v)				
Current (A)				
T_1				
T_2				
T_3				
T_4				
T_5				
T_6				
T_7				
T_8				

Formulas used:

$$k = \frac{-V.I.(R_i - R_o)}{4\pi R_i R_o (T_i - T_o)}$$

Where: k = Thermal conductivity in w/mK.

V = Voltage in volts.

I = Current in Amp.

T_i = Average surface temperature of the inner sphere= $(T_1 + T_2 + T_3 + T_4)/4$.

T_o = Average surface temperature of the outer sphere= $(T_5 + T_6 + T_7 + T_8)/4$.

Calculations:

Result: The thermal conductivity of the given insulating powder is = w/mK.

Conclusion:

Also expected in this Journal:

1. Concept: Fourier's Law of heat conduction
2. Derivation: Fourier's Law of heat conduction as applied to a sphere
3. Graph (s): None
4. Diagrams:
 - a. Set-up
 - b. Schematic of Temperature Gradient

EXPERIMENT NO: II

Aim: To determine the thermal conductivity of metal rod.

Apparatus: Standard equipment for the determination of thermal conductivity of metal rod.

The apparatus consists of a metal bar one end of which is heated by an electric heater while the other projects inside a cooling water jacket. The middle portion of the bar is insulated with asbestos filled in a cylindrical shell concentric to the rod. The temperature of the bar is measured at 9 different sections, while radial temperature distribution is measured with 4 separate thermocouples. Two thermocouples are also provided at the inlet & exit of the cooling water jacket.

The heater is provided with a dimmer-stat to control the heat input. Water under constant head is circulated through the jacket and its temperature rise & flow rate are measured.

Theory:

It is advised that students should write the theory on the subject in their own words under the guidance of the teacher.

Introduction:

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

Thermal Conductivity of a material is found to depend on the chemical composition of the substance of which it is composed. The phase (i.e. gas, liquid, 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. THERMAL CONDUCTIVITY OF SOME MATERIALS.

	Thermal Conductivity Kcal/hr-m ⁰ c	State
Solid(Metals)		
Pure Copper	330	20 ⁰ c
Brass	95	20 ⁰ c
Steel	46	20 ⁰ c
Stainless Steel	14	20 ⁰ c

MECHANISM OF THERMAL ENERGY CONDUCTION IN METALS.

Thermal energy may be conducted in solids by two modes:

1. Lattice Vibrations and
2. Transport by free electrons.

In good electrical conductors a rather large no. of free electrons moves 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 electron transport and it is for this reason that good electrical conductors are almost always good heat conductors viz. Copper, Aluminum and Silver.

With increase in the temperature, however the increased lattice vibrations come in the way of transport by free electrons & for most of the pure metals the thermal conductivity decreases with increase in the temperature.

Procedure:

1. Keep the dimmer-stat at minimum voltage position. Switch ON the electric supply.
2. Adjust the dimmer-stat to supply a voltage of 80 to 120 volts to the heating coil. Maintain this constant throughout the experiment.
3. Start the cooling water supply and adjust it to about 350 cc.
4. Wait for steady state to be attained.
5. Note down the readings in the observation table as given below.
6. Note the mass flow rate of water in kg/s.

Observations:

Length of the metal rod= 425 mm.

Test length of the metal rod= 200 mm.

Diameter of the metal rod= Inner diameter of insulation=25 mm.

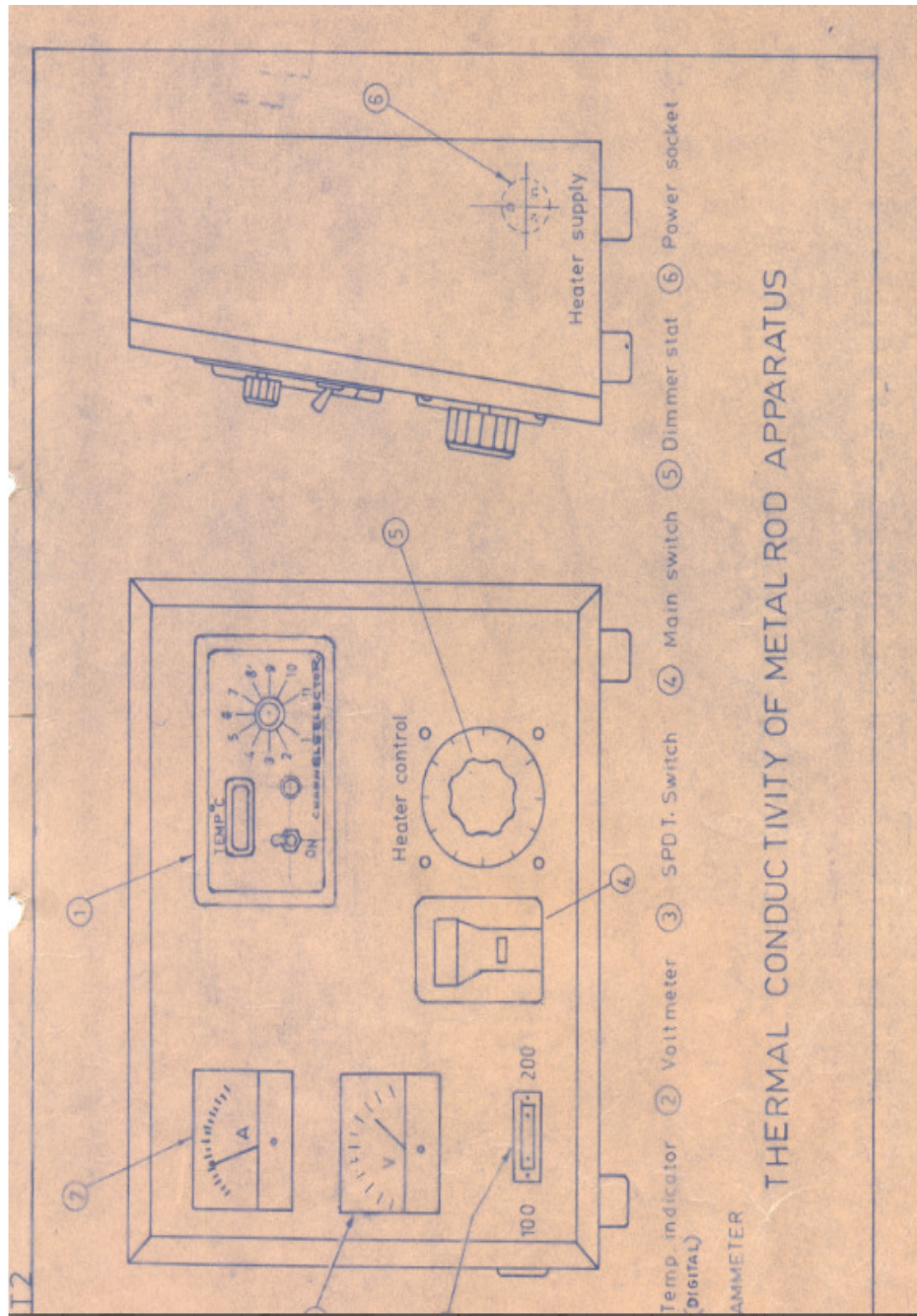
Outer diameter of the insulation= 55 mm.

Observation Table:

Sr. No.	1	2	3	4
Voltage (v)				
Current (A)				
T ₁				
T ₂				
T ₃				
T ₄				
T ₅				
T ₆				
T ₇				
T ₈				
T ₉				
T ₁₀				
T ₁₁				
T ₁₂				
T ₁₃				
T ₁₄				
Mass flow rate of water (kg/s)				

Plot the temperature gradient with the length of the rod.

Calculations:



THERMAL CONDUCTIVITY OF A METAL ROD APPARATUS

Result: The thermal conductivity of the given metal rod is as shown in the following table.

Sr. No.	Local Temperature	Thermal conductivity.
1		
2		
3		
4		
5		
6		
7		
8		
9		

Conclusion:

Also expected in this Journal:

1. Concept: Fourier's Law of heat conduction
2. Derivation: Fourier's Law of heat conduction as applied to a one-dimensional axial flow through a rod
3. Formulas used:
4. Graph (s): Temperature Gradient
5. Diagrams:
 - a. Set-up
 - b. Thermocouple positions

EXPERIMENT NO: III

Aim: To determine the thermal resistance & thermal conductivity of composite slab.

Apparatus:

Standard equipment for the determination of thermal conductivity of composite slab.

The apparatus consists of three disks of equal diameters but variable thickness arranged to form a slab of same diameter. A second set of the same arrangement is taken & an electric heater coil is sandwiched between these two slabs in such a way that the nearest to farthest sequence from the heater is the same. Two pressure plates are provided at the top & bottom of this entire assembly to ensure close contact between the adjacent plates in which case, contact resistance can be safely neglected.

A set of 8 thermocouples is incorporated to measure temperatures at every interface. A voltmeter & ammeter are provided to measure voltage and current to the heater coil. A dimmer-stat is used to vary the input power to the heater.

Theory:

A composite slab consists of slab of three different materials which are cast iron, hylum & pressed wood. Slabs are circular in cross section & heating element too. The heating element is placed between the three sets of slabs which are of different materials.

Procedure:

1. Keep the dimmer-stat at minimum voltage position. Switch ON the electric supply.
2. Adjust the dimmer-stat to supply a particular voltage. Maintain this constant throughout the experiment.
3. Wait for steady state to be attained.
4. Note down the readings in the observation table as given below.

Observations:

Diameter of disks= 300 mm.

Thickness of C.I. disk= 25 mm.

Thickness of Bakelite disk= 12 mm.

Thickness of wood disk= 12 mm.

Observation Table:

Sr. No.	1	2	3	4
Voltage (v)				
Current (A)				
T ₁				
T ₂				
T ₃				
T ₄				
T ₅				
T ₆				
T ₇				
T ₈				

Calculations:

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Result:

The total thermal resistance of the composite slab is = $\quad\quad\quad$ K/w.
The thermal conductivity of the composite slab is = $\quad\quad\quad$ w/mK.

Conclusion:

Also expected in this Journal:

1. Concept: Fourier's Law of heat conduction
2. Derivation: Fourier's Law of heat conduction as applied to a composite slab with axial heat flow only
3. Formulas used:
4. Graph (s): Temperature gradient
5. Diagrams:
 - a. Set-up

EXPERIMENT NO: IV

Aim: To determine the average surface heat transfer coefficient in natural convection and compare it with the value obtained by using appropriate correlation.

Apparatus:

The apparatus consists of a brass tube fitted in a rectangular duct in a vertical fashion. 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 of a transparent material 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.

Theory:

Convection is the heat transfer that occurs as a result of the movement of a fluid, and hence it is not possible in case of solids. It is well known that a hot metal plate will cool faster when placed in front of a fan than when exposed to still air. We say that heat is convected away and we call the process convective heat transfer. Heat will transfer by conduction through the walls of the fluid container. From the container walls, the heat will transfer by conduction and radiation into the fluid, causing convection currents to be set up in the fluid. Consider a heated plate. The temperature of the plate is T_p and the temperature of the fluid in which it is immersed is T_f . The velocity of flow increases from the wall of the plate to the surface of the fluid. At the wall, the velocity is zero. Therefore heat transfer here is only by conduction. The change of temperature over the length (temperature gradient) is dependent on the rate at which the fluid carries the heat away. This process is described by the Newton's law of cooling.

$$q = h \cdot A_s (T_p - T_f)$$

The heat transfer rate is related to the overall temperature difference between the wall and the fluid and the surface area A_s . The quantity h is called the heat transfer coefficient and it can be calculated for simple systems, but has to be experimentally determined for complex systems. It includes the effects of conduction, radiation and convection at the considered surface. For instance the convective heat transfer coefficient for free convection with water flowing on a hot vertical plate 1 ft high with a temperature difference of 30°C between the water and the plate is $4.5 \text{ W/m}^2 \cdot ^\circ\text{C}$. The following figure shows the heat transfer from the surface of a plate. Factors affecting the heat transfer coefficient are:

- Shape and inclination of the surface over which convection takes place.
- Injection of new fluid Roughness of the surface
- Whether the flow is in an open or closed space
- Mode of convection (free/forced)
- Speed of flow and Reynold's number (laminar/turbulent flow)
- Presence of Electric and magnetic fields.

Heat transfer using movement of fluids is called convection. In natural convection, the flow is induced by the differences between fluid densities which result due to temperature changes.

The heat transfer rate for convection is given by the following equation:

$$q = h \cdot A_s \cdot (T_{\text{surface}} - T_{\infty})$$

h is the convection coefficient, A_s is the surface area and T_{surface} and T_{∞} are the surface and ambient temperatures, respectively.

Density of fluid changes with temperature. In general, fluids expand as the temperature rises, and thus the density decreases (density is the mass per unit volume). Warm fluids therefore are more buoyant than cooler fluids. A hot object will heat the surrounding fluid, which rises due to buoyancy force. The warm fluid is then replaced by cool fluid. Similarly cool objects will draw heat away from the surrounding fluid which then falls due to increased density. The cool fluid is then replaced by warm fluid initiating convection currents.

For a hot horizontal plate surrounded by air, convection currents form when the air above the plate starts to rise. The air below the plate, however, cannot rise because the plate is blocking the flow. The heated fluid under the plate will eventually escape through the sides of the plate; however, the convective flow below the plate is very small compared to the flow on top. In general, natural convection is more pronounced (has a higher h) on the top side of a hot plate or the bottom side of a cold plate.

The convection coefficient is a measure of how effective a fluid is at carrying heat to and away from the surface. It is dependent on factors such as the fluid density, velocity, and viscosity. Generally, fluids with higher velocity and/or higher density have greater h . The convection coefficient for natural convection in gas is generally between $1 \text{ w/m}^2\text{K}$ and $20 \text{ w/m}^2\text{K}$; typical values for liquids fall between $100 \text{ w/m}^2\text{K}$ and $1000 \text{ w/m}^2\text{K}$.

Correlations for Heat Transfer Coefficients:

Each flow geometry requires different correlations be used to obtain heat transfer coefficients. Initially, we will look at correlations for fluids flowing in conduits. Most correlations will take the "Nusselt form":

$$Nu = aRe^b Pr^c \text{ (correction factors)}$$

The correlations that follow are limited to conduit flow without phase change. Unless otherwise specified, fluid properties should be evaluated at the "bulk average" temperature -- the arithmetic mean of the inlet and outlet temperatures:

$$T_b = \frac{T_{in} + T_{out}}{2}$$

Choosing a Correlation

When choosing a correlation, begin by asking:

1. What is the geometry? (Flow through a pipe, around an object, over a plane, etc.)
2. Is there a phase change?
3. What is the flow regime? (Check the Reynolds number to decide on laminar, transition, or turbulent flow.)
4. If the flow is laminar, is natural convection important? (The Grashof number will be used for this.)

Laminar flow and free convection:

Heat usually causes the density of a fluid to change. Less dense fluid tends to rise, while the more dense fluid falls. The result is circulation -- "natural" or "free" convection. This movement raises h values in slow moving fluids near surfaces, but is rarely significant in turbulent flow. Thus, it is necessary to check and compensate for free convection only in laminar flow problems.

The *Grashof Number* is used to assess the impact of natural convection

$$Gr = \frac{d^3 \rho_{film}^2 g \beta (T_{wall} - T_{bulk})}{\mu_{film}^2}$$

The fluid properties used to calculate the Grashof number should be evaluated at the film temperature, the arithmetic mean between the bulk and wall temperatures. This will require determining an additional set of property values.

The Grashof Number provides a measure of the significance of natural convection. When the Grashof Number is greater than 1000, heat transfer coefficients should be corrected to reflect the increase due to free circulation. Multiplicative correction factors are available to apply to the Nusselt Number or the heat transfer coefficient (do NOT use both).

$$Nu = Nu_{lam} (0.87(1 + 0.15 Gr^{1/3}))$$

$$h = h_{lam} \left(2.25 \left(\frac{1 + 0.01 Gr^{1/3}}{\log_{10} Re} \right) \right)$$

Procedure:

1. Switch on the electric supply and adjust the dimmer-stat to obtain the required heat input (say 40w, 60w, 70w).
2. Wait for steady state to be attained.
3. Note down the reading in the observation table as given below.
4. Repeat the experiment at different heat inputs. (Do not exceed 80 watts).

Observations:

O.D. of cylinder, d= 38 mm

Length of cylinder, l= 500 mm

Observation Table:

Sr. No.	1	2	3	4
Voltage (v)				
Current (A)				
T ₁				
T ₂				
T ₃				
T ₄				
T ₅				
T ₆				
T ₇				
T ₈				

Calculations:

a) Actual heat transfer coefficient:

Rate of heat transfer, $Q = V I$ watts

Average temperature of surface, $T_s = (T_1 + T_2 + T_3 + T_4 + T_5 + T_6 + T_7)/7$

Ambient temperature in the duct, $T_a = T_8$

Area of heat transfer, $A_s = \pi d l$

$Q = h_{\text{actual}} A_s (T_s - T_a)$

Hence, $h_{\text{actual}} = Q / [A_s (T_s - T_a)]$

b) Theoretical heat transfer coefficient:

$h_{\text{theoretical}}$ is calculated using the appropriate correlation between Nu, Gr and Pr from heat and mass transfer data book.

$Nu = c (Gr Pr)^n$

Result:

The actual value of heat transfer coefficient was found to be

W/m^2K and

The theoretical value of heat transfer coefficient was found to be

W/m^2K .

Conclusion:

Also expected in this Journal:

- Diagrams:
 - Set-up
 - Schematic of thermal boundary layer with velocity gradient.

EXPERIMENT NO: V

Aim: To study the pool boiling phenomenon upto critical heat flux point.

Apparatus:

The apparatus consists of a cylindrical glass container housing the heater coil for initial heating of the water. This heater coil is directly connected to the mains and the test heater (Nichrome wire) is connected to the mains via dimmer-stat. Ammeter and voltmeter are provided to measure voltage & current for the test heater.

Theory:

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 liquid to change phase. This change of phase is called boiling.

Boiling is of various types, depending upon the temperature difference between the surface and the liquid. The different types are indicated as in Fig.1, in which a typical experimental boiling curve obtained in a saturated pool of liquid is drawn.

The heat flux supplied to the surface is plotted against $(T_w - T_s)$, the difference between the temperature of the surface and saturation temperature of the liquid. It is seen that the boiling curve can be divided into three regions :

- I. Natural Convection Region
- II. Nucleate Boiling Region
- III. Film Boiling Region.

The natural convection region occurs at low temperature differences (of the order of 10°C or less). Heat transfer from the heated surface to the liquid in its vicinity causes the liquid to be superheated. This superheated liquid rises to the free liquid surface by natural convection, where vapour is produced by evaporation.

As the temperature difference $(T_w - T_s)$ 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 bubble formation as well as 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 vapour film. This is the beginning of region III viz., film boiling. In the first part of this region III-a, the vapour film is unstable, so that 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 1000°C and consequently radiative heat transfer across the vapour film is significant.

It will be observed from Fig. 1 that the heat flux does not increase in a regular manner with temperature difference. In region I, the heat flux is proportional to $(T_w - T_s)^n$ where 'n' is slightly greater than unity (about 1.3). 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 (point A). Beyond this, in region III-A, inspite of increasing temperature difference, the heat flux decreases because the thermal resistance to heat flow increases with the formation of a vapour film. The heat flux passes through a minimum (point B) at the end of region III-a. It starts to increase again with $(T_w - T_s)$ only when stable film boiling begins and radiation becomes increasingly important.

It is of interest to know how the temperature of the heating surface changes as the heat flux is steadily increased from zero. Upto the point A, natural convection boiling and then nucleate boiling occurs and the temperature of heating surface is obtained by reading off the value of $(T_w - T_s)$ from the boiling curve and adding to it the value of T_s . 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 point C. It is apparent from Fig. 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 point flux. The pool boiling curve as described above is known as Nukiyam Pool Boiling Curve.

The discussion so far has been concerned with various types of boiling which occur in saturated pool boiling. If the liquid is below the saturation temperature, we say that sub-cooled pool boiling takes place. Also in many practical situations, e.g. , steam generators, one is interested in boiling in a liquid flowing through tubes. This is called as forced convection and may also be saturated or sub-cooled and of nucleate or film type.

Thus, in order to completely specify boiling occurring in any process, one must state (1) whether it is forced convection boiling or pool boiling, (2) whether the liquid is saturated or sub-cooled, and (3) whether it is in the natural convection, nucleate or film boiling region.

Procedure:

1. Take 3 to 4 liters of distilled water in the container.
2. See that both the heaters are completely submerged
3. Make the necessary connections. Connect the test heater properly to the provided studs.
4. Switch on bulk heater and wait till water temperature is attained at a particular level, say 50 0C, 60 0C, 70 0C upto the saturation temperature.
5. Switch off bulk heater and switch on the test heater.
6. Very slowly increase the voltage across the test heater and observe the different phases of boiling until the test heater wire fuses. DO NOT EXCEED 6 AMPS UNDER ANY CONDITIONS.

Observations:

Length test heater= $l =$ mm.

Diameter of the test wire= $d =$ mm.

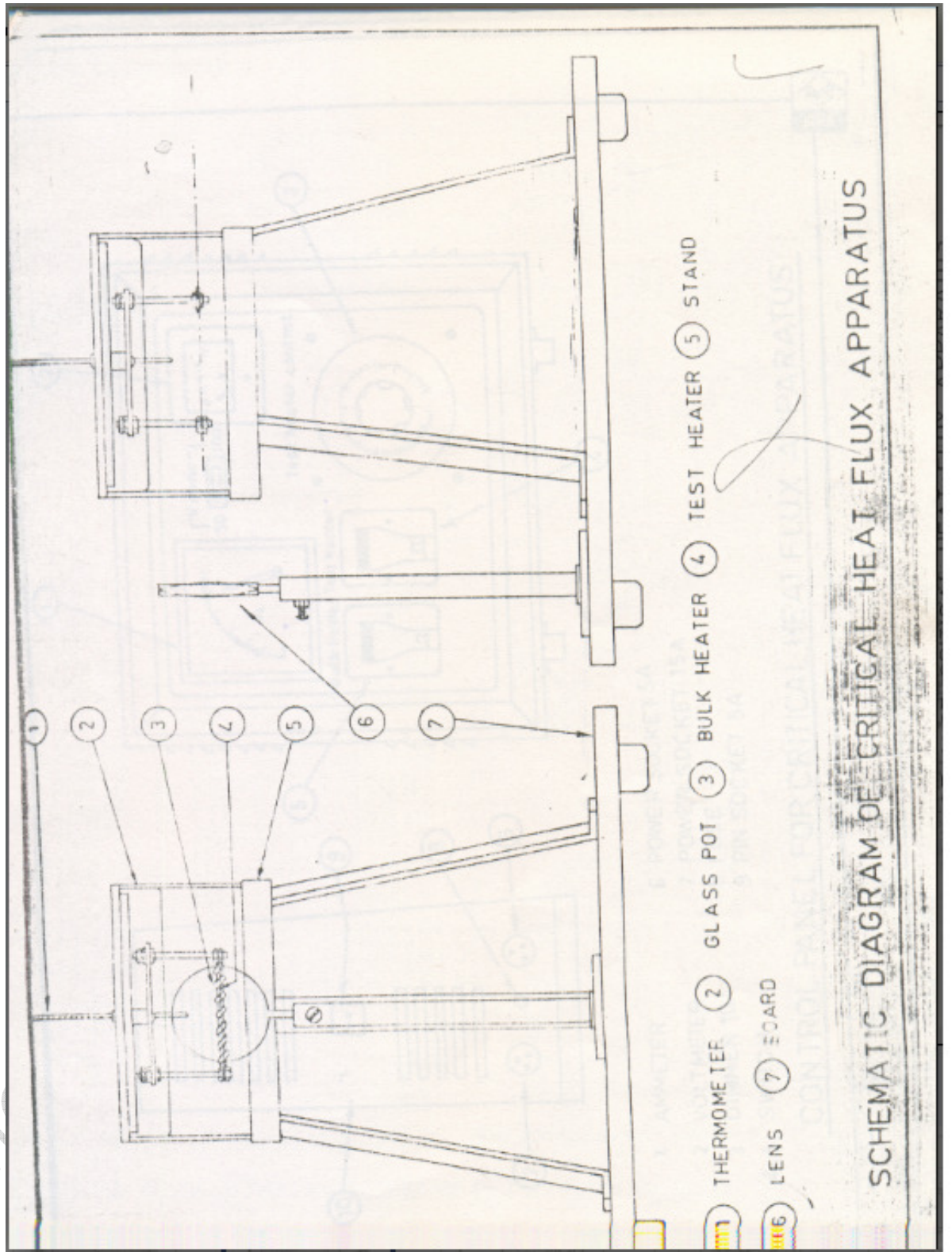
Surface area= $A = \pi dl =$ mm².

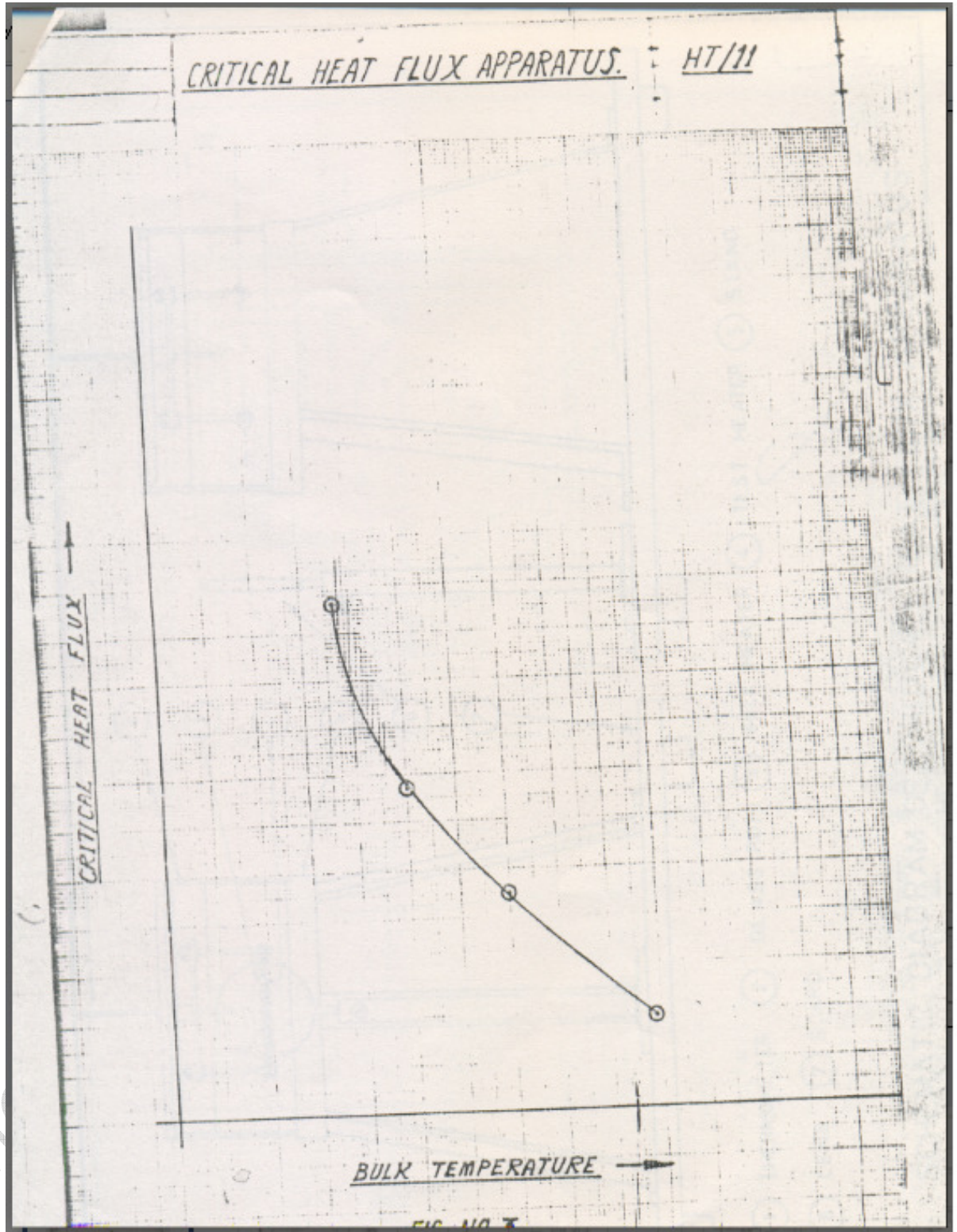
Observation Table:

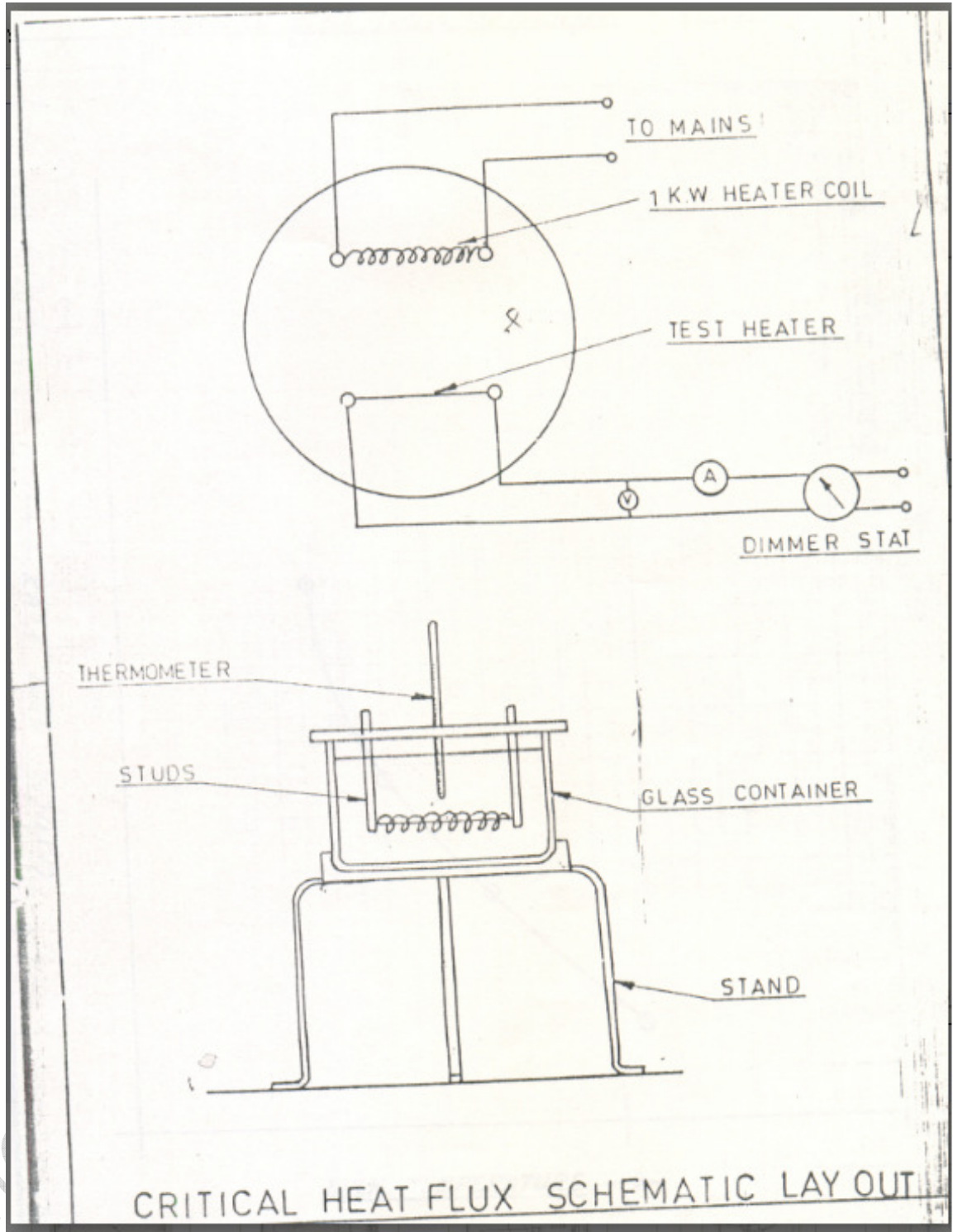
Bulk water temperature	Current (A)	Voltage (v)
40 °C		
50 °C		
60 °C		
70 °C		
80 °C		
90 °C		
95 °C		
Saturation Temp (°C)		

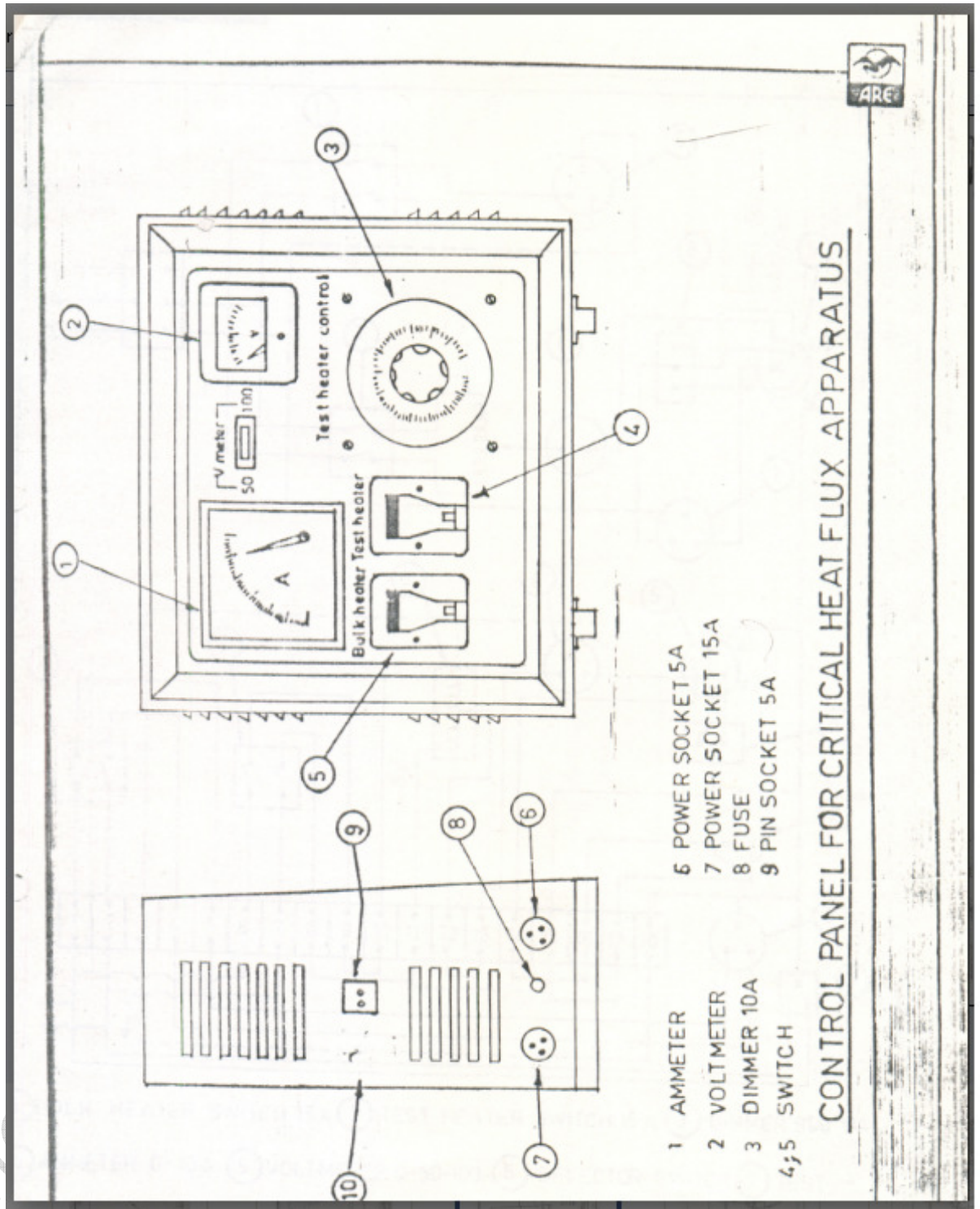
Calculations:

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Result:

The thermal conductivity of the given metal rod is as shown in the following table.

Sr. No.	Bulk Water Temperature	Critical heat flux
1	40 °C	
2	50 °C	
3	60 °C	
4	70 °C	
5	80 °C	
6	90 °C	
7	95 °C	
8	Saturation Temp (°C)	

Conclusion:

Also expected in this Journal:

1. Concept: Boiling Phenomenon, pool boiling curve & its various regions
2. Formulas used:
3. Graph (s): Variation of critical flux with pool temperature
4. Diagrams:
 - a. Set-up
 - b. Pool Boiling Curve

EXPERIMENT NO: VI

Aim: To determine the following for parallel flow and counter flow double pipe heat exchangers:

- I) LMTD
- II) Effectiveness
- III) Overall heat transfer coefficients

Apparatus:

It consists of a tube in tube type of heat exchanger. A heater is provided for heating the water. The hot water flows through the inner tube and the cold water flows through the annular space between the inner and outer tubes. A pipe and valve arrangement is provided for reversing the direction of the cold water. Thermo wells are fitted at the inlets and outlets of both hot and cold water. The temperatures of the hot and cold water at these points are measured with mercury in glass thermometers. The volumetric flow rates of the hot and cold water are measured using a measuring jar and a stopwatch.

Theory:

A device used to transfer heat between fluids that are at different temperatures and separated by a solid wall is termed a heat exchanger. Common uses for heat exchangers are found in waste heat removal, air-conditioning, power production and chemical processing. The general representation of a heat exchanger is as shown in the figure above.

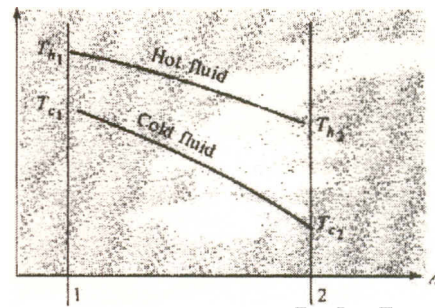
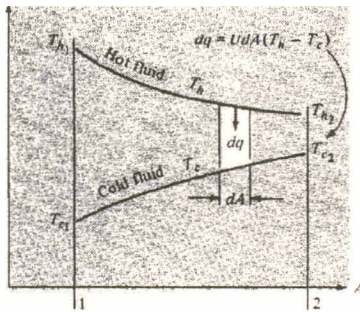
Types

Heat exchangers are normally classified according to type of construction and flow arrangement. Heat exchangers can have several different flow arrangements. One arrangement is where the fluids both enter the heat exchanger from the same side and flow parallel to each other until the exit point. This is called Parallel Flow heat exchanger. Another common type is the Counter Flow in which the opposite happens. Here, one fluid enters the heat exchanger at the exit point of the other fluid, and exits at that fluid's entrance point. A third type of heat exchanger is the Cross Flow heat exchanger, in which one fluid flows perpendicular to the other. The main difference between the flow arrangements lies in the temperature distribution along the length of the heat exchanger, and the relative amounts of heat transfer under given temperature specifications for specified heat exchanger surfaces. A counter flow heat exchanger requires a minimum area; a parallel flow heat exchanger requires a maximum area while a cross-flow heat exchanger requires an area in between.

Parallel Flow

In a parallel flow heat exchanger, both the hot and cold fluid flow in the same direction. They enter together at one end, flow through in the same direction and leave together at the other end. The temperature difference between the two fluids decreases asymptotically along the increasing length of the exchanger. The outlet temperature of the cold fluid never exceeds that of the hot fluid.

The temperature profiles for parallel flow and counter flow heat exchangers is shown in the figures below: Here the subscripts '1' and '2' stand for inlet and outlet respectively and the subscripts 'h' and 'c' stand for hot and the cold fluids respectively.



Counter Flow

Counter flow heat exchangers are the opposite of parallel flow heat exchangers. In a counter flow design, the entrance of the hot fluid of the heat exchanger is the exit of the cold fluid, and at the other end of the exchanger, the hot fluid exit is the cold fluids entrance. In contrast to the parallel flow, exchanger, this configuration provides for heat transfer between the hotter portions of the two fluids at one end, as well as between the colder portions at the other. The difference in temperature between the two fluids along the length of the exchanger is nowhere near as large as it is for the inlet of the parallel flow design. The outlet temperature of the cold fluid may exceed the outlet temperature of the hot fluid.

- The heat exchanger is insulated from its surroundings, in which case the only heat exchange is between the hot and cold fluids.
- Axial conduction along the tubes is negligible.
- Potential and kinetic energy changes are negligible.
- The fluid specific heats are constant.
- The overall heat transfer coefficient is constant

Common designs incorporate concentric tubes, also known as double-pipe arrangement. This consists of one pipe placed concentrically inside another of larger diameter. This design can have parallel or counter flow configurations. This is largely used for sensible heating or cooling of process fluids where small heat transfer areas are required.

For a hollow cylinder exposed to a convection environment on its inner and outer surfaces, the electrical resistance analogy would appear as in the following figure where, again T_A and T_B are the two fluid temperatures. Note that the area for convection is not the same in both fluids in this case, these areas depending on the inside tube diameter and the wall thickness. In this case the overall heat transfer would be expressed by

$$q = \frac{T_A - T_B}{\left(\frac{1}{h_i A_i} + \frac{\ln(r_o/r_i)}{2\pi kL} + \frac{1}{h_o A_o} \right)}$$

In accordance with the thermal network shown in the figure, the terms A_i and A_o represent the inside and cut side surface areas of the inner tube. The overall heat transfer coefficient may be based on either the inside or the outside area of the tube. Accordingly

$$U_i = \frac{1}{\left(\frac{1}{h_i} + \frac{A_i \ln(r_o/r_i)}{2\pi kL} + \frac{A_i}{h_o A_o}\right)}$$

$$U_o = \frac{1}{\left(\frac{A_o}{h_i A_i} + \frac{A_o \ln(r_o/r_i)}{2\pi kL} + \frac{1}{h_o}\right)}$$

Fouling Factors

After a period of operation the heat transfer surfaces for a heat exchanger may become coated with various deposits present in the flow systems, or the surfaces may become corroded as a result of the interaction between the fluids and the material used for construction of the heat exchanger. In either event, this coating represents an additional resistance to the heat flow, and thus results in decreased performance. The overall effect is usually represented by a fouling factor or fouling resistance R_f , which must be included with the other thermal resistances making up the overall heat transfer coefficient.

Logarithmic Mean Temperature Difference

For a double pipe heat exchanger with a hot and a cold fluid flowing through it, we calculate the heat transfer in the arrangement with a simple formula $q = UA\Delta T_m$

Where U = Overall heat transfer coefficient.

A = surface area for heat transfer consistent with definition of U T_m = suitable mean temperature difference across the heat exchanger

LMTD is determined by the following equation

$$\Delta T_m = \frac{(T_{ho} - T_{co}) - (T_{hi} - T_{ci})}{\ln[(T_{ho} - T_{co}) / (T_{hi} - T_{ci})]}$$

Where the subscripts i and o stand for the inlet and outlet positions; and h and c stand for the hot and cold fluid. The above equation is valid for parallel flow heat exchangers. **For counter flow heat exchangers, the inlet and outlet temperature values for the cold fluid get reversed in the equation.**

Heat Exchanger Effectiveness

One way of measuring a heat exchanger's performance is to calculate its effectiveness. The heat exchanger effectiveness is defined as the ratio of the actual heat transfer to the heat transfer attainable in an infinitely long counter flow exchanger. So, by defining the effectiveness in this way, we have a much simpler expression for our effectiveness:

$$\varepsilon = \frac{c_c(T_{c_{out}} - T_{c_{in}})}{c_{min}(T_{H_{in}} - T_{c_{in}})} = \frac{c_H(T_{H_{in}} - T_{H_{out}})}{c_{min}(T_{H_{in}} - T_{c_{in}})}$$

If, for any given exchanger, the effectiveness is known we can write the heat transfer equation as:

$$q = \varepsilon \cdot c_{min}(T_{H_{in}} - T_{c_{in}})$$

Procedure:

1. With the help of the valves provided both the hot and cold water are made to flow in the same direction (Parallel flow arrangement).
2. Adjust the flow rate of hot and cold water with the help of the valves.
3. Measure the flow rate of the hot and cold water with the help of a measuring jar and a stopwatch. Note the flow rates.
4. Switch ON the heater and wait for steady state to be attained.
5. Note the inlet and outlet temperatures of the hot and cold water.
6. Also note the voltage and current supplied to the heating element.
7. Repeat the experiment after reversing the direction of the hot water. (Counter flow arrangement)

Observations:

Diameter of inner pipe, $d_i = 8.1$ mm

Diameter of outer pipe, $d_o = 13.14$ mm

Length of pipes, $l = 2$ m

Observation table:

Type of flow	Cold Water			Hot water		
	Flow rate	T_{ci}	T_{co}	Flow rate	T_{hi}	T_{ho}
	(lit/s)	($^{\circ}$ C)	($^{\circ}$ C)	(lit/s)	($^{\circ}$ C)	($^{\circ}$ C)
Parallel flow						
Counter flow						

Calculations:

Results:

For parallel flow arrangement

1. LMTD= °C
2. U_i actual= W/m^2K
3. U_i theoretical= W/m^2K
4. U_o actual= W/m^2K
5. U_o theoretical= W/m^2K
6. ϵ =

For Counter flow arrangement:

1. LMTD= °C
2. U_i actual= W/m^2K
3. U_i theoretical= W/m^2K
4. U_o actual= W/m^2K
5. U_o theoretical= W/m^2K
6. ϵ =

Conclusion:

Also expected in this Journal:

- 1) Diagrams:
 - Set-up
 - Different flow arrangements
 - Set-up ckt

EXPERIMENT NO: VII

Aim: To determine the value of emissivity for a gray body.

Apparatus:

The apparatus consists of two plates of the same diameter, mounted on electric heaters. The supply to these heaters can be varied with the help of two separate dimmer-stats, which are connected to the heaters through a double pole double through switch. One of these plates is blackened with lamp black while the other is kept gray. The plates are mounted with their axes horizontal & parallel to each other and kept inside a wooden box. One side of this box is covered with transparent acrylic sheet for viewing. Three thermocouples are employed to find the temperatures of Black plate, Test Plate and surrounding air.

Theory:

It is advised that students should write the theory on the subject in their own words under the guidance of the teacher.

All substance at all temp. Emit thermal radiation. Thermal radiation is an electromagnetic wave & does not require any material medium for propagation all bodies can emit radiation and have also the capacity to absorb all or a part of radiation coming from surrounding towards it.

An idealized black surface is one which absorbs all the incident radiation with reflectivity and transmissivity equal to zero. The radiant energy per unit time per unit area from the surface of body is called emissive power and is usually denoted by 'e'. The emissivity of the surface is the ratio of emissive power of surface to emissive power of black surface at same temperature.

$$E = \frac{e}{e_b}$$

For the black body absorbtivity =1 and by the knowledge of Kirchoff's law emissivity of the black body becomes unity. It is obvious from Stefan-Boltzmann law that the prediction of emissive power of a surface requires knowledge about the values of its emissivity and therefore much experimental research in radiation has been concentrated on measuring values of emissivity as a surface temperature. The present experimental set up is designed and fabricated to measure the property of emissivity of the test plate surface at various temperatures.

Following table gives values of emissivity for some common materials for reference.

TEMPERATURE EMISSIVITY

Metal	Polished Copper steel, Stainless steel	20° c	0.15 increases with temp.
	Nickel, Aluminum	90° c – 540° c	0.2 to 0.33
Non-metal	Brick wood marble water	20° c – 100° c	0.8 to 1

Procedure:

- 1) Keep the dimmer-stat at minimum voltage position. Switch ON the test plate heater while keeping the black plate heater off.
- 2) Adjust the dimmer-stat to the black plate for a particular voltage. Then adjust the dimmer-stat to the test plate approximately 10 to 15 volts lower than the black plate dimmer-stat.
- 3) Son the Black plate will attain a constant temperature. Adjust the Test plate dimmer-stat ONLY to bring the temperature of the test plate to the same value. This will require some trial & error.
- 4) After a couple of hours when the temperatures of both the plates become equal, note down the following readings in the observation table as given below.

Observations:

Diameter of both the plates= 160 mm.

Observation Table:

Sr. No.	1	2	3	4
W ₁ (watts)				
W ₂ (watts)				
T ₁				
T ₂				
T ₃				

Calculations:

Result: The emissivity of the test plate is .
Conclusion:

Also expected in this Journal:

- 1) Concept: Radiation, Emissivity, Shape factor
- 2) Formulas used:
- 3) Graph (s): None
- 4) Diagrams:
 - a. Set-up

EXPERIMENT NO: VIII

Aim: To determine the value of Stefan-Boltzmann constant experimentally.

Apparatus:

The apparatus consists of an inverted hemispherical vessel which is blackened from the concave side and there is an arrangement for storing hot water at the convex side. The concave end is further sealed except for a small opening at the bottom. 4 thermocouples are fitted on the hemispherical shell. One is fixed on the blackened test plate (Copper). There is provision for heating water up and sending it for storage in the tank.

Theory:

Radiation heat transfer is concerned with the exchange of thermal radiation energy between two or more bodies. Thermal radiation is defined as Electromagnetic radiation in the wavelength range of .1 to 100 microns (which encompasses the visible light regime.), and arises as a result of a temperature difference between two bodies. No medium need exists between two bodies for heat transfer to take place (as is need by conduction and convection.) Rather the intermediaries are photons which travel at the speed of light. The heat transferred into or out of an object by thermal radiation is a function of a several components. These include its surface reflectivity, emissivity, surface area, temperature and geometric orientation with respect to other thermally participating objects. In turn, an object's surface reflectivity and emissivity is a function of its surface conditions (roughness, finish etc.) and composition.

In conduction and convection energy is transferred through a material medium, but energy may also be transferred through regions where perfect vacuum exist. Radiation is an electromagnetic phenomenon of varying wavelengths closely allied to the transmission of light and radio. It proceeds in straight lines at the speed of light. A classic example of energy transmission by radiation is the Sun which transmits abundant energy to the earth by this means.

Radiation is independent of mass except for nuclear reactions. It is much a surface through a wide wavelength band. Electromagnetic radiation propagated as a result of temperature difference is termed as thermal radiation. An ideal thermal radiator is called a black body which will emit energy at a rate proportional to the fourth power of the absolute temperature of the body and directly proportional to the surface area.

Where σ the Stefan-Boltzmann constant & has a value of 5.669×10^{-8} w/m²K⁴. This equation governs only the radiation emitted by a black body.

The net radiation exchange between two surfaces is proportional to diff. in absolute temperature to the fourth power

$$\frac{q}{A} \propto \sigma (T_1^4 - T_2^4)$$

No object is perfectly black body. It cannot radiate all the heat given to it. To account for this Gray nature of surface, we introduce a factor called emissivity (ϵ). Again radiation travels in a straight line so the radiation leaving one surface does not completely reach the other surface, but some of it is lost to the surrounding. So we introduce another factor called view factor (F_G).

HEAT TRANSFER LABORATORY MANUAL

$$q = \epsilon F_G A \sigma (T_1^4 - T_2^4)$$

Where ϵ – emissivity factor. And F_G = Geometric view factor.

Procedure:

1. Heat up the water by the electric heater and after a certain temperature is attained, send it into the convex bottom tank. .
2. Wait till thermocouples 1 to 4 show fairly similar readings.
3. Now insert the test disk into the slot provided and immediately start taking readings against time.
4. Plot the temperature-time graph and find out its slope at time=0.

Observations:

Test disk diameter= 20 mm.

Test disk weight= 6 gm.

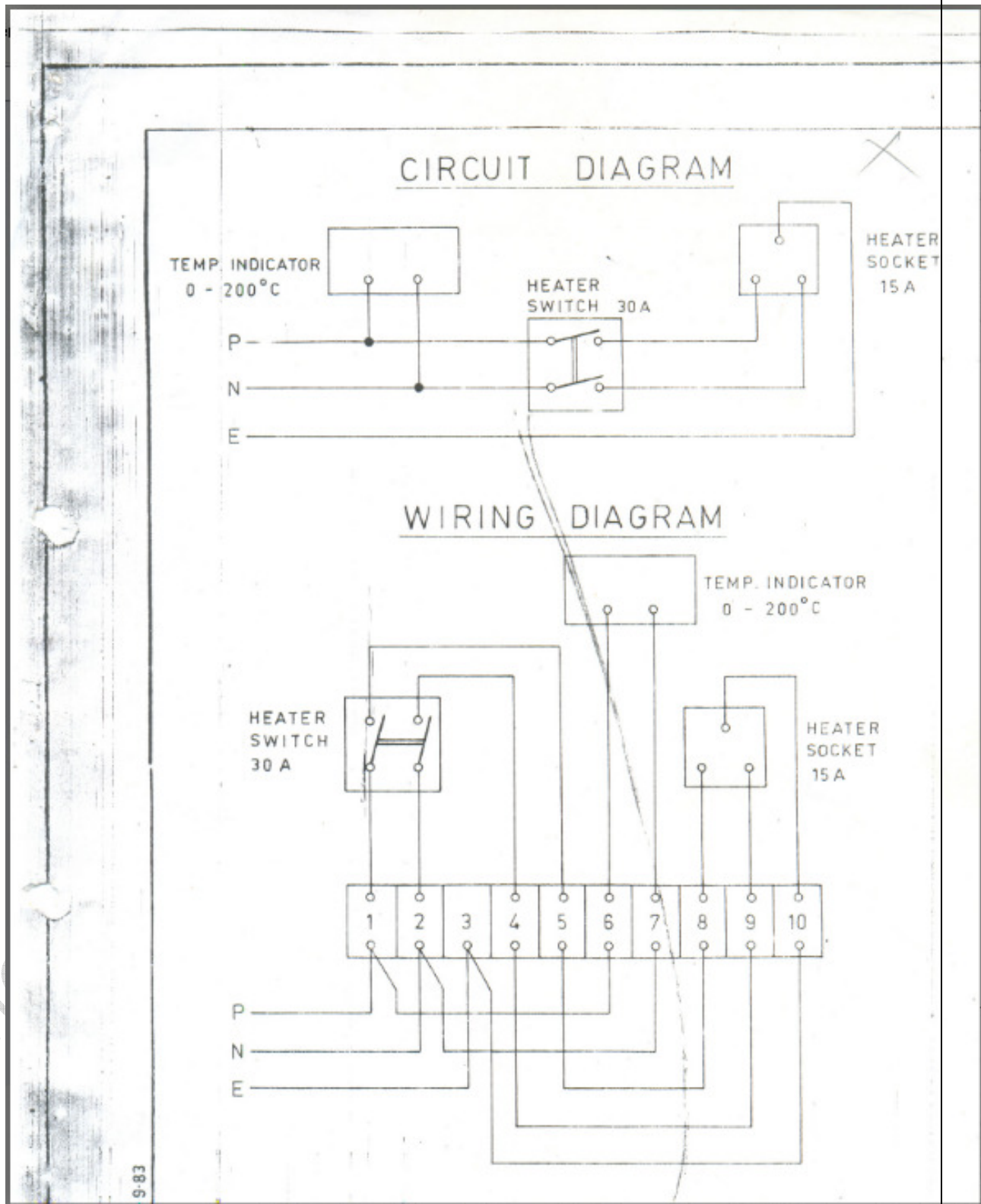
Specific heat of the test disk= 418.68j/keg.

Observation Table:

T ₁		T ₂		T ₃		T ₄	
(°c)		(°c)		(°c)		(°c)	
Time	T ₅	Time	T ₅	Time	T ₅	Time	T ₅
1s		20s		55s			
2s		25s		60s			
3s		30s		65s			
4s		35s		70s			
5s		40s		75s			
10s		45s		80s			
15s		50s		85s			

Plot the temperature gradient against time. Find out the slope of the curve at time=0.

Calculations:



CIRCUIT DIAGRAM AND WIRING DIAGRAM FOR STEFAN BOLTZ-MANN CONSTANT

Result: The experimental value of Stefan-Boltzmann constant was found to be w/m^2K^4

Conclusion:

Also expected in this Journal:

- 1) Concept: Radiation, Stefan-Boltzmann constant
- 2) Formulas used:
- 3) Graph (s): Temperature against time with tangent drawn at time=0, slope of the tangent shown
- 4) Diagrams:
 - a. Set-up

EXPERIMENT NO: IX

Aim: To determine the heat transfer coefficient in forced convection

Apparatus:

Standard Forced Convection set up.

The apparatus consists of a metal pipe provided with an electric heater. This assembly is covered with insulation so as to avoid any unaccounted heat losses. A blower is fitted to this pipe to provide a constant supply of high velocity air.

Thermocouples are provided to measure necessary temperatures. The heater coil is fitted with voltmeter and ammeter. An orifice meter with a U-tube differential type manometer is fitted to measure the velocity of the air.

Theory:

It is advised that students should write the theory on the subject in their own words under the guidance of the teacher.

Procedure:

- 1) Keep the dimmer-stat at minimum voltage position. Switch ON the electric supply to the heater.
- 2) Adjust the dimmer-stat to supply a constant voltage to the heating coil. Start the blower supply.
- 3) Wait for steady state to be attained.
- 4) Note down the readings in the observation table as given below.

Observations:

O.D. of the pipe= 32 mm

I.D. of the pipe= 28 mm

Length of test section= 40 cm

Orifice diameter= $d = 14$ mm

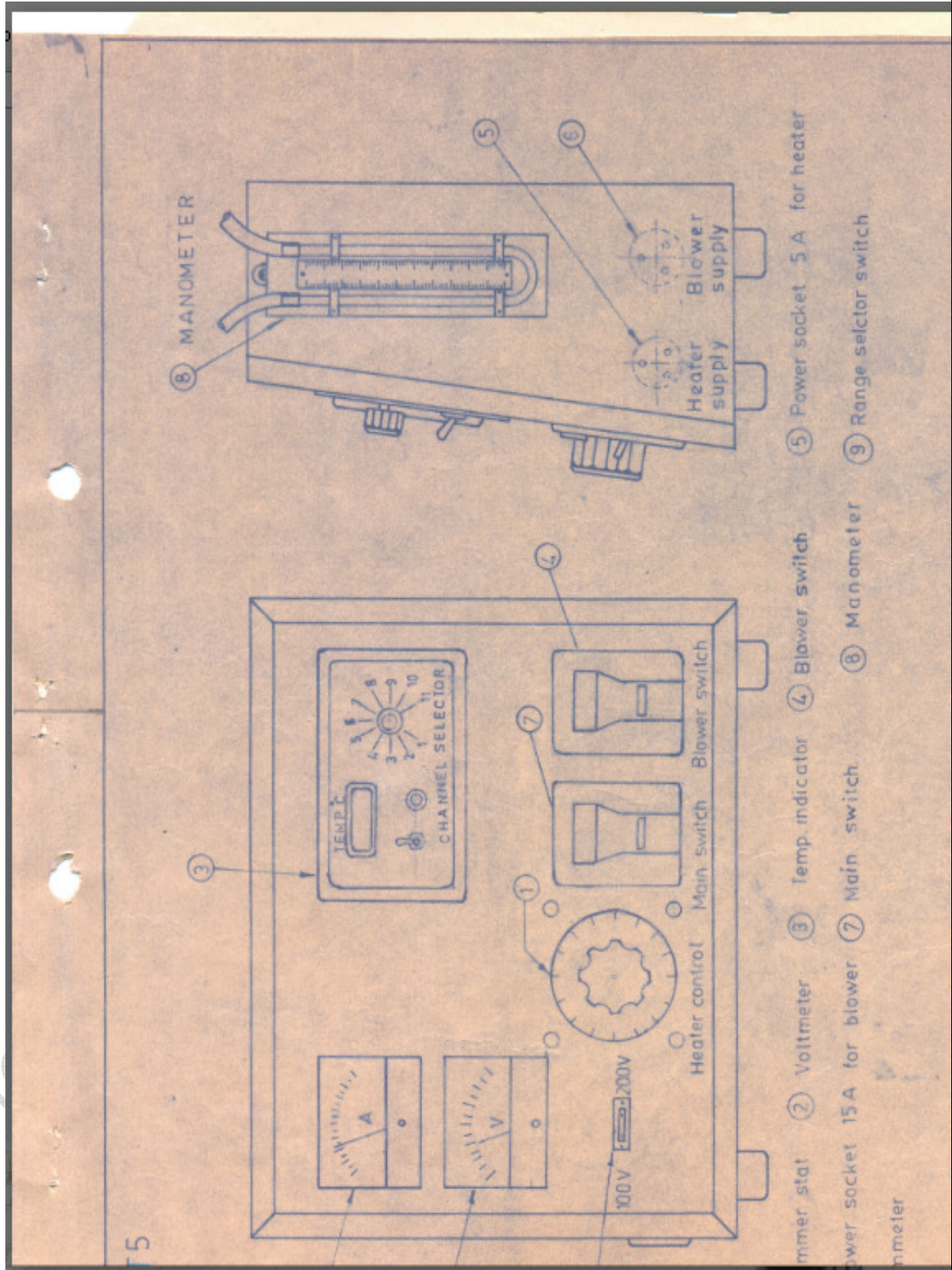
Coefficient of discharge= 0.64

Observation Table:

Sr. No.	1	2	3	4
Voltage (v)				
Current (A)				
T ₁				
T ₂				
T ₃				
T ₄				
T ₅				
T ₆				
Manometer reading (mm)				

Calculations:

MGM'S JNEC MECH. ENGG. DEPT.



Result: The heat transfer coefficient in forced convection was found to be w/m^2K

Conclusion:

Also expected in this Journal:

- a. Concept: Heat Transfer in forced convection, Important correlations
- b. Formulas used:
- c. Diagrams:
 - i. Set-up
 - ii. Location of the thermocouples

EXPERIMENT NO: X

Aim: To determine the heat transfer coefficient for natural & forced convection from pin-fin.

Apparatus:

The apparatus consists of a pin-fin provided with an electric heater at one end. This assembly is kept in a duct and provided with thermocouples along the length. A blower is fitted to this duct to provide a constant supply of high velocity air. This blower is fitted in such a way that its inlet is at the end of the duct. The heater coil is fitted with voltmeter and ammeter. An orifice meter with a U-tube differential type manometer is fitted to the delivery pipe to measure the velocity of the air.

Theory:

In order to increase the rate of cooling or heating, the heat flow area is increased by attachment of fins or solid materials to the wall of heat exchangers on that side which the coefficient of heat transfer is small.

These fins are widely used in engineering heat transfer applications like transformer motors, IC engines, etc.. External surfaces in the form of rods are known as pin fins or spines and one in the form of continuous plates are known as fins. They may be located longitudinally or circumferentially. The cross sectional shape of extended surface in a plane normal to the wall is known as the profile of spine or fin.

Procedure:

- 1) Keep the dimmer-stat to the heater at minimum voltage position. Switch ON the electric supply.
- 2) Adjust the dimmer-stat to supply a particular voltage to the heater. Maintain this constant throughout the experiment.
- 3) Wait for steady state to be attained in natural convection mode.
- 4) Note down the readings in the observation table as given below for natural convection mode.
- 5) Now turn the blower on.
- 6) Again, note down the readings in the observation table as given below for forced convection mode.

Observations:

Density of manometric fluid= 1000 kg/m^3 .

Diameter of orifice= 15mm.

Diameter of pin= 12.7 mm.

Duct size= $10 \times 15 \text{ cm}^2$.

Coefficient of discharge= 0.64.

Diameter of delivery pipe= 28 mm.

Length of pipe= 40 cm.

Observation Table:

Sr. No.	Natural Convection	Forced Convection
Voltage (v)		
Current (A)		
T ₁		
T ₂		
T ₃		
T ₄		
T ₅		
T ₆		
Manometric reading	—	

Plot the temperature gradient with the length of the pin.

Calculations:

MGM'S JNEC MECH. ENGG. DEPT.

Result: Heat transfer coefficient by-

1. Natural convection:
 - a. Experimentally $w/m^2K.$
 - b. Theoretically $w/m^2K.$
2. Forced convection:
 - a. Experimentally $w/m^2K.$
 - b. Theoretically $w/m^2K.$

Conclusion:

Also expected in this Journal:

- a. Concept: Heat transfer through pin-fin
- b. Derivation: Heat transfer through pin-fin (All 3 conditions)
- c. Graph (s): Temperature gradient along the length of fin
- d. Diagrams:
 - i. Set-up
 - ii. Location of thermocouples

EXPERIMENT NO: XI

Aim: To determine the following for a shell & tube heat exchangers:

- I) LMTD
- II) Effectiveness
- III) Overall heat transfer coefficients

Apparatus:

The apparatus consists of a shell & tube type heat exchanger (two pass-tube side). The tube side is supplied with hot water while the cold side with cold water. Thermocouples are provided at the entry & exit of both the water circuits. There is provision of a measuring flask & a stop-watch to measure flow rate on either side. The water is heated with the help of two geysers. Water is continuously supplied to both the circuits.

Theory:

A device used to transfer heat between fluids that are at different temperatures and separated by a solid wall is termed a *heat exchanger*. Common uses for heat exchangers are found in waste heat removal, air-conditioning, power production and chemical processing. Heat exchangers may be constructed in different fashions.

Common designs incorporate concentric tubes, also known as *double-pipe arrangement*. This consists of one pipe placed concentrically inside another of larger diameter. This design can have parallel or counter flow configurations. This is largely used for sensible heating or cooling process fluids where small heat transfer rates are required. Other designs incorporate a *shell-in-tube* construction.

Shell and Tube Heat Exchangers

They are built of round tubes mounted in large cylindrical shells with the tube axis parallel to the shell. They are widely used as oil coolers, power condensers, preheaters in power plants and steam generators in nuclear power plants. Specific shell and tube designs differ by the number of shell and tube passes. Baffles are commonly used in heat exchangers to increase the convection coefficient.

Resistance to Heat Transfer

The ability to transfer heat from one fluid to another through some form of solid wall is dependent on a variety of factors:

- Surface area between the two fluids.
- Rate of heat transfer through a dividing wall which separates the two fluids.
- Rate of heat transfer from one layer to the next (if more than one layer is present) in the dividing wall.
- Rate of heat transfer from the hot fluid to the dividing wall.
- Rate of heat transfer from the dividing wall to the cold fluid.

Surface Area

Increasing a common surface area between two fluids will increase overall heat transfer rate between the fluids. This could be accomplished by incorporating more tubes in a shell and tube heat exchanger.

Typical Values for Overall Heat Transfer 'U' are-

- Plate Heat Exchanger, liquid to liquid U range 1000>4000Wm⁻²K⁻¹
- Shell and tube, liquid inside and outside tubes U range 150>1200 Wm⁻²K⁻¹
- Spiral Heat Exchanger, liquid to liquid U range 700>2500Wm⁻²K⁻¹

The general representation of a heat exchanger is as shown in the figure below.

U= Overall heat transfer coefficient Wm⁻²K⁻¹

A= Heat Exchanger Surface Area m²

h₁=Heat transfer coefficient-hot side (Wm⁻²K⁻¹)

h₂= Heat transfer coefficient-cold side (Wm⁻²K⁻¹)

t=Temperature (°C)

θ=Temperature difference (°C)

Overall Heat Transfer Coefficient

For a hollow cylinder exposed to a convection environment on its inner and outer surfaces, the electrical resistance analogy would appear as in the following figure, where, again T_A and T_B are the two fluid temperatures. Note that the area for convection is not the same in both fluids in this case, these areas depending on the inside tube diameter and the wall thickness. In this case, the overall heat transfer would be expressed by

$$q = \frac{T_A - T_B}{\left(\frac{1}{h_1 A_1} + \frac{\ln\left(\frac{r_o}{r_i}\right)}{2\pi k L} + \frac{1}{h_2 A_2}\right)}$$

In accordance with the thermal network shown in the figure, the terms A_i and A_o represent the inside and outside surface areas of the inner tube. The overall heat transfer coefficient may be based on either the inside or the outside area of the tube. Accordingly,

$$U_i = \frac{1}{\left(\frac{1}{h_i} + \frac{A_i \ln\left(\frac{r_o}{r_i}\right)}{2\pi k L} + \frac{A_i}{A_o} * \frac{1}{h_o}\right)}$$

$$U_o = \frac{1}{\left(\frac{A_o}{A_i} * \frac{1}{h_i} + \frac{A_i \ln\left(\frac{r_o}{r_i}\right)}{2\pi k L} + \frac{1}{h_o}\right)}$$

Fouling Factors

After a period of operation, the heat transfer surfaces for a heat exchanger may become coated with various deposits present in the flow systems or the surfaces may become corroded as a result of the interaction between the fluids and the material used for construction of heat exchanger. In either event, this coating represents an additional resistance to the heat flow, and thus results in decreased performance. The overall effect is usually represented by a *fouling factor* or fouling resistance R_f, which must be included with other resistances making up the overall heat transfer coefficient.

Logarithmic Mean Temperature Difference

For a double pipe heat exchanger with a hot and cold fluid flowing through it, we calculate the heat transfer in the arrangement with a simple formula

$$Q=UA\Delta T_m$$

Where, U=overall heat transfer coefficient

A=surface area for heat transfer consistent with definition of U

T_m=suitable mean temperature difference across the heat exchanger

LMTD is determined by following equation

$$\Delta T_m = \frac{(T_{ho} - T_{co}) - (T_{hi} - T_{ci})}{\ln [(T_{ho} - T_{co}) / (T_{hi} - T_{ci})]}$$

Where the subscripts *l* and *o* stand for the inlet and outlet positions and *h* and *c* stand for the hot and cold fluids.

The above equation is valid for parallel flow heat exchangers. For counter flow heat exchangers, the inlet and outlet temperature values for the cold fluid get reversed in the equation.

LMTD Correction Factor For Countercurrent Shell and Tube Heat Exchanger

Log Mean Temperature difference (LMTD) is defined for countercurrent heat transfer. A double pipe heat exchanger provides true countercurrent flow. Other equipment geometries will depart from countercurrent flow. When this happens, the effective MTD is lower than the LMTD. A LMTD correction factor has been defined to account for this circumstance. The LMTD correction factor (F) is defined and used as follows :

$$F = (\text{Effective MTD}) / (\text{LMTD})$$

$$\text{Heat Exchange Duty } Q = (\text{Overall Coefficient } U) * (\text{Area } A) * (F) * (\text{LMTD})$$

F is related to heat transfer geometry and temperatures. This section contains formulas for countercurrent multi-tube pass shell and tube heat exchangers.

Temperatures And LMTD Definition

T₁= Shell IN t₁= Tube IN

T₂= Shell OUT t₂= Tube OUT

$$LMTD = \frac{(T_1 - t_2) - (T_2 - t_1)}{\ln [(T_1 - t_2) / (T_2 - t_1)]} \quad (1)$$

The definition of LMTD is the same for all heat exchangers. Shell and tube designations are immaterial in this definition.

Auxiliary Variables

$$P = \frac{t_2 - t_1}{T_1 - t_1} \quad (2)$$

$$R = \frac{(T_1 - T_2)}{(t_2 - t_1)} \quad (3)$$

The LMTD correction factor F is obtained from the relevant charts for the values of P & R.

Heat Exchanger Effectiveness

One way of measuring a heat exchanger's performance is to calculate its effectiveness. The heat exchanger effectiveness is defined as the ratio of the actual heat transfer to the heat transfer attainable in an infinitely long counter-flow exchanger. So, by defining the effectiveness in this way, we have a much simpler expression for our effectiveness:

$$\epsilon = \frac{C_c(T_{c\text{out}} - T_{c\text{in}})}{C_{\text{min}}(T_{\text{hin}} - T_{c\text{in}})} = \frac{C_h(T_{\text{hin}} - T_{h\text{out}})}{C_{\text{min}}(T_{\text{hin}} - T_{c\text{in}})}$$

If, for any given exchanger, the effectiveness is known, we can write the heat transfer equation as:

$$q = \epsilon C_{\text{min}}(T_{\text{hin}} - T_{c\text{in}})$$

Procedure:

- 1) Start the tube-side water-flow. After ensuring that the water is running, start one of the geysers.
- 2) Start the cold water supply after a while. Keep the flow-rate at the cold water side higher than the hot water side. Maintain both the flows at a moderate level.
- 3) Wait for steady state to be attained.
- 4) Note down the reading in the observation table as given below.

Observations:

1. Shell
 - a. Inner Diameter = 208 mm
 - b. Thickness = 6 mm
 - c. Material = M. S.
2. Tubes
 - a. O.D. = 19 mm
 - b. I.D. = 17 mm
 - c. Material = Copper
 - d. Pitch = 30 mm
 - e. No. of tubes = 32
 - f. Length = 500 mm

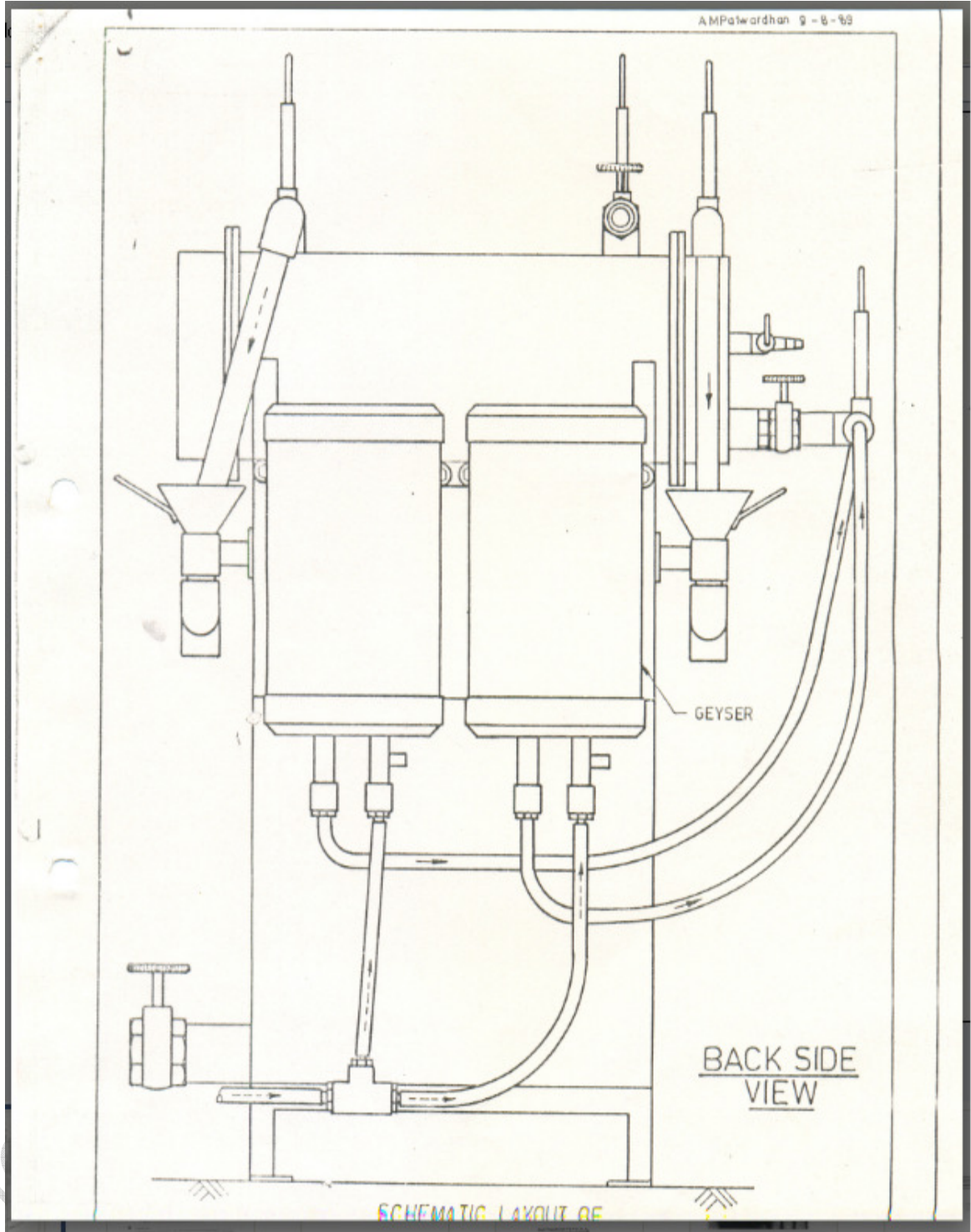
HEAT TRANSFER LABORATORY MANUAL

Observation Table:

Sr. No.	Shell Side			Tube Side		
	Flow Rate	Inlet Temperature	Outlet Temperature	Flow Rate	Inlet Temperature	Outlet Temperature
1						
2						
3						
4						
5						
6						
7						
8						
9						

Calculations:

MGM'S JNEC MECH. ENGG. DEPT.



Result:

- 1) LMTD= °C
- 2) U_i actual= W/m²K
- 3) U_i theoretical= W/m²K
- 4) U_o actual= W/m²K
- 5) U_o theoretical= W/m²K
- 6) ϵ =

Conclusion:

Also expected in this Journal:

- a. Concept: Shell & Tube Heat Exchanger, Correlations
- b. Graph (s): None
- c. Diagrams:
- d. Set-up

EXPERIMENT NO: XII

Aim: To study film wise and drop wise condensation and to determine the overall heat transfer coefficient.

Apparatus: Condenser main unit, heating element, steam generator, temperature indicator, rotameter.

Theory:

Condensation Heat Transfer:

Condensation heat transfer is a vital process in Process and Power Generation industries and as a result, has been an area of research for over a hundred years. Over this period our understanding of the condensation process has gradually improved. Theories and models have become more accurate and are now applicable to a wider range of conditions.

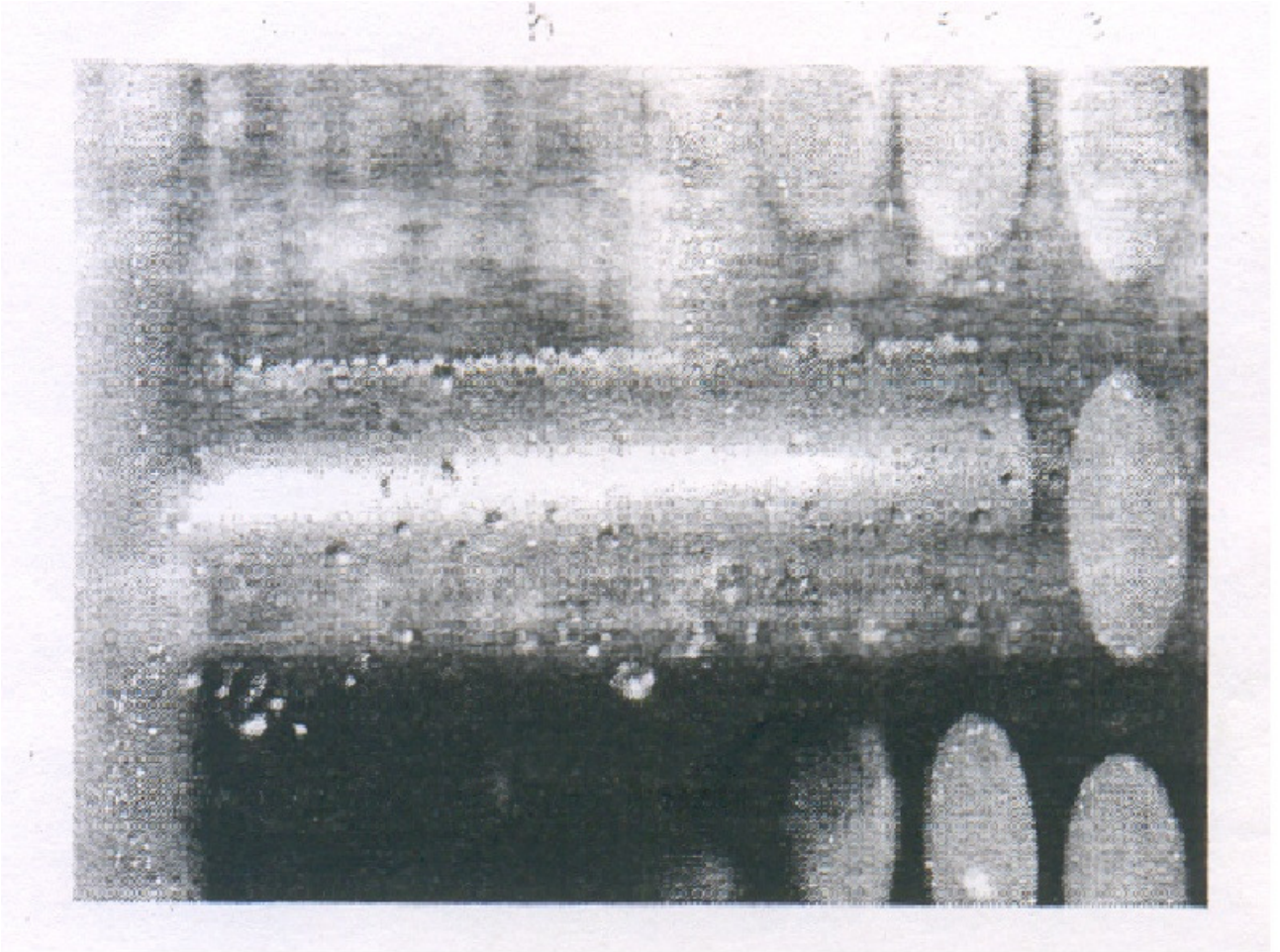
Condensation:

There are two modes of condensation, viz., Filmwise and Dropwise. Filmwise is currently used by industry, while Dropwise is an alternative which is under development because it offers attractive higher rates of heat transfer by preventing the build up of the insulating liquid layer found in filmwise condensation.

Filmwise And Dropwise Condensation:

Dropwise and filmwise are the two major forms of condensation. Dropwise condensation occurs when liquid droplets form on a cold surface and then falls from it, leaving the cold surface dry. Filmwise condensation occurs when a film of condensate coats the cold surface. This is the most common form of condensation. The only pure vapour known to condense in a dropwise manner is steam. Dropwise condensation is more difficult to create and maintain. It is important to note that there are significant differences between these two types of condensation.

Due to the resistance of heat passing through condensate film, the heat-transfer coefficients for dropwise condensation are 4-8 times greater than those for filmwise condensation. All but a few precious metals will in an untreated state tend to condensate filmwise; this is why industrial condensers currently operate filmwise. This type of condensation behaviour which a metal displays, is related to its surface energy. Materials with a high surface energy condensate *filmwise*, while those with a low surface energy condensate *dropwise*. With suitable promoters or surface treatments, most metals, including those with high surface energies, can promote dropwise condensation.



Procedure:

1. Fill the steam generator with water.
2. Start the water flow through one of the condensers which is to be tested and note down the water flow rate in the rotameter. Ensure that during the measurement of flow rate water is flowing through only the condenser under test and second valve is closed. Note the flow rate of the water.
3. Switch ON the heater.
4. Slowly steam generation will begin in the steam generator. As the steam rises to the last section, it gets condensed & flows down in the cylinder.
5. Depending upon the type of condenser under test, dropwise or filmwise condensation can be visualized.
6. Note down the temperatures of the water, steam and the surface of the condenser.

Observations:

Inner diameter of the tube= 17 mm.
Outer diameter of the tube= 20 mm.
Length of tube= 140 mm.

Observation Table:

Type	Filmwise Condensation	Dropwise Condensation
Flow Rate(LPH)		
T ₁		
T ₂		
T ₃		
T ₄		
T ₅		
T ₆		

Calculations:

MGM'S JNEC MECH. ENGG. DEPT.

Result:

Type	U_i (w/m ² K)	U_o (w/m ² K)
Filmwise Condensation		
Dropwise Condensation		

Conclusion:**Also expected in this Journal:**

- a. Concept: Phenomena governing dropwise & filmwise condensation.
- b. Graph (s): None
- c. Diagrams:
 - i. Set-up
 - ii. Condensation on Drop & Film type tubes