CONVECTIVE HEAT TRANSFER
OF
ROTATIONAL FLOW

A Dissertation
submitted in partial fulfilment
of
the requirements for the degree
of
MASTER OF ENGINEERING
in
MECHANICAL ENGG. (App. Thermodynamics)

By
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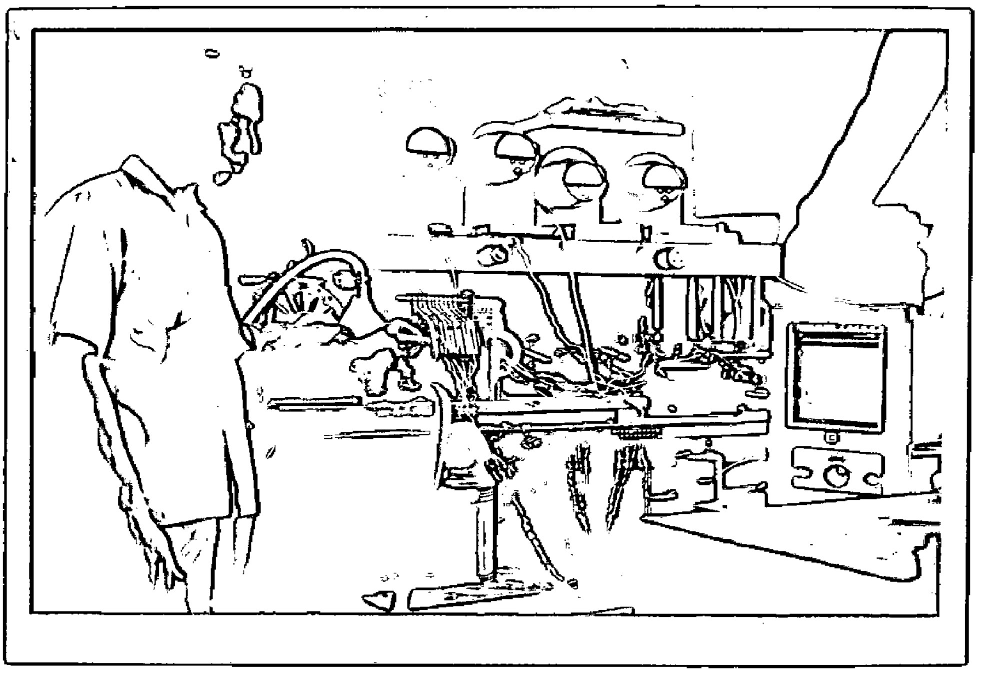

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A B S T R A C T

The heat transfer by convection in the annular gap between the rotating heated inner cylinder and the stationary outer cylinder was determined in the air, at various rotational speeds, and variable heat inputs. The results show that the characteristics of the heat transfer have two modes.

At low rotational speeds, namely when $\left(\frac{Vh}{\nu} \sqrt{\frac{h}{R}} \right) < 39$, the heat transfer is not effected by the rotational speed, which is perhaps due to the laminar flow, and the heat transfer by conduction and radiation predominates.

At higher speeds of rotation, which makes the Taylor number greater than 39, the heat transfer increases with rotational speeds. This is perhaps due to the influence of secondary vortices induced by the centrifugal force. In this range the heat transfer may be expressed by

$\frac{Uh}{K} = .152 \left(\frac{Vh}{\nu} \sqrt{\frac{h}{R}} \right)^{.52}$ for air only where

U denotes the over all heat transfer coefficient through the gap, h the width of the gap, K the thermal conductivity, ν the kinemetic viscosity, R and V are the radius and rotational peripheral velocity of the rotating cylinder. The maximum deviation was found to be 16% . The experimental results obtained were compared with those of previous workers, and found to be in reasonably good agreement.

C O N T E N T S

LIST OF FIGURES

NOMENCLATURE

A	:	Cylindrical surface area of the test section,	$ft.^2$
b	:	width of the annular gap, ...	ft.
l	:	length of the test section , ...	ft.
R	:	radius of rotating surface, ...	ft.
D	:	diameter, ...	ft.
ω	:	angular speed of rotation, ...	rad./sec.
V	:	peripheral val. of rotor surface , ...	ft./sec.
K	:	thermal conductivity of fluid,	Btu/ft.hr ^oF
ν	:	kinemetic viscosity of the fluid, ...	$ft.^2/sec.$
μ	:	absolute viscosity based on mean temp.,	lbs/ft.sec.
ρ	:	density of the fluid,	$lbs/ft.^3$
g	:	accelaration due to gravity,	$ft./sec.^2$
β	:	coefficient of volumetric expansion,	$1/^oF$
U	:	overall convective heat transfer coeff.	$Btu/ft.^2hr\ ^oF$
σ	:	stephan Boltzman constant	
ε	:	emissivity	
Δt	:	temperature difference between inner and outer cylinder, ...	^oF
T	:	absolute temperature , ...	^oK
t_1	:	temperature of cooling water at inlet,	^oF
t_2	:	temperature of cooling water at outlet,	^oF
t_m	:	mean temperature of the fluid $(t_r+t_s)/2$,	^oF
Q_t	:	rate of total heat flow by convection and radiation, ...	Btu/hr.
Q_c	:	rate of heat flow by convection	Btu/hr.
Q_{rad}	:	rate of heat flow by radiation	Btu/hr,
w	:	Rate of cooling water flowing per time	
T_a	:	Taylor number, ...	$\frac{Vb}{\nu}\sqrt{\frac{b}{R}}$

Nu	:	Nusselt number , ...	$\frac{Uh}{K}$
Pr	:	Prandtl number , ...	$C_p \mu / K$
Gr	:	Grashof number , ...	$\frac{g \beta \Delta t \; b^3}{\nu^2}$
Re	:	Reynolds number, ...	$\frac{Vh}{\nu}$

SUFFIXES

r	:	refers to the rotating cylinder
s	:	refers to the stationary cylinder
c	:	referes to the convection

The simplest case of concentric cylinder flow is that in which no axial flow occures. In this case the flow is actuated only by the rotation of one of the cylinder.

The primary problem is to investigate those variables which control the rate of heat transfer in the air gap between a rotating inner cylinder and the concentric outer cylinder. The first factor on which the heat transfer depends is the flow geometry, or in this case it can be represented as a dimensionless curvature factor, which can be expressed as the ratio of the rotor radius to gap width i.e. $\frac{R}{b}$.

At the start of this work, it appeared that for a fixed geometary the rate of heat transfer in the gap would depend on the following variables.

1. Speed of Rotation
2. Temperature gradients at walls annulus
3. Axial velocity of the fluid in the gap
4. Surface roughnes in air gap due to teeth, slots, and core laminations.
5. Entrance effects caused by development of boundary layer flow in entrance region of air gap

In the present study, the annulus is formed by rotating smooth inner cylinder and stationary outer cylinder. This is chosen in order to eliminate, as nearly as possible, the surface roughnes and entrance effects present in actual rotating machines and this simplifying the actual complex problem.

Another variable is the speed of rotation, which determines whether the gap Reynolds number based on mean speed of rotation is below or above the critical value.

Nu	:	Nusselt number , ...	$\frac{Uh}{K}$
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Re	:	Reynolds number, ...	$\frac{Vh}{\nu}$

SUFFIXES

r	:	refers to the rotating cylinder
s	:	refers to the stationary cylinder
c	:	referes to the convection

INTRODUCTION

Heat transfer by convection in an annular gap between a rotating inner cylinder and stationary outer cylinder occurs in various rotating machines such as electric motors and generators. Careful heat transfer analysis in the design of electric rotating machinery is necessary not only to prevent exceeding material temperature limitations, but also to enable size reduction and increase in power rating.

Considering the conventional electric machines, the rotor is the inner moving cylinder of the annular gap, and irreversible electrical, mechanical, and magnetic processes within the rotor and stator result in generation of heat throughout the machine An illustration of the overall thermal analysis of the relatively simple electric machine, namely, a small D.C.Motor is found in the paper by Kaye and Gouse. It is well known that the heat generated within the rotor by " Core", "Copper" and other losses results in excessive temperatures unless this heat is removed in an orderly fashion by careful design . A portion of the heat generated in the rotor is removed by the axial conduction through the shaft and some is removed also by convection from the ends of the rotor to the air within the machine housing. However, in many electric machines much of the heat generated in the rotor is transferred from the cylinderical surface of the rotor to the air in the air gap. Frequently cool air from the fan or blower is often forced through this air gap in the axial direction. Other means of colling rotors, employing ducts for internal air or liquid colling, also have been utilized.

The simplest case of concentric cylinder flow is that in which no axial flow occures. In this case the flow is actuated only by the rotation of one of the cylinder.

The primary problem is to investigate those variables which control the rate of heat transfer in the air gap between a rotating inner cylinder and the concentric outer cylinder. The first factor on which the heat transfer depends is the flow geometry, or in this case it can be represented as a dimensionless curvature factor, which can be expressed as the ratio of the rotor radius to gap width i.e. $\frac{R}{b}$.

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In the present study, the annulus is formed by rotating smooth inner cylinder and stationary outer cylinder. This is chosen in order to eliminate, as nearly as possible, the surface roughnes and entrance effects present in actual rotating machines and this simplifying the actual complex problem.

Another variable is the speed of rotation, which determines whether the gap Reynolds number based on mean speed of rotation is below or above the critical value.

$$\text{Gap Renulds Number } Re_g = \frac{(\text{Eq. Dia.})\ (\text{Mean velocity})}{\text{Kinematic Viscosity}}$$

$$= \frac{2bV/2}{\nu}$$

$$= \frac{bV}{\nu}$$

The second variable is the temperature gradient at walls annulus which effects greatly the heat transfer. The temperature gradient depends upon the heat input to the test section, which can be varied according to the requirement.

Finally the Prandtl number, which is a function of the fluid properties alone , and is a measure of the ratio of heat transmission and energy storage capacities of the molecules, also effects the heat transfer characteristics. In the present investigation experimental data on heat transfer through the air gap with rotating inner cylinder and stationary outer cylinder, without the axial flow have been reported. The inner cylinder is heated electrically at a uniform temperature. The outer cylinder is water jacketed and then finally insulated from the surroundings, in order to keep the stationary surface at uniform temperature and to measure the total heat passing through the annulus.

SURVEY OF LITERATURE

Although a considerable amount of research work has already been carried out on heat transfer and fluid flow in annuli, it has not been possible to determine the combined effect of all the factors actually encountered in practice due to the complexity of the problem . In fact, it appears that the relationship between the rate of heat transfer, speed of rotation, and axial flow, are quite complex even if the surface and enterance effects are not taken into consideration.

Fluid flow in ducts and passages of various shapes has been studied quantitatively almost for two centuries. A special type is the flow in an annular duct; one limit of annular flow corresponds to flow in a simple duct when the inner surface of the annulus shrinks to zero size for a fixed size of the outer surface, a second limit of annular flowcorresponds to flow between parallel plates when the annular gap or opening shrinks to zero size for a fixed size of either the inner or the outer surface of the annulus. If now we consider that one of the concentric surfaces forming the annulus can rotate, then it is possible to combine such rotation with axial flow of fluid through the annulus to produce different and interesting flow combinations. Perhaps Osborne Reynold in 1883 was the first person to investigate the axial flow in annular passages without any rotation. G.I. Taylor (3) in 1923 analysed mathematically the stability of incompressible Viscous flow in a narrow annulus between rotating concentric cylinders of infinite length for the case of zero axial flow.

He assumed small perturbations in the velocity components of the basic equations of laminar flow ; he then expanded the solution in a series of Bessel junctions, and finally solved the resulting infinite determinant for the lowest values of the speed of rotation for which the perturbations would grow. The speedof rotation for which the laminar flow breaks down and at which the perturbations grow, leading to the formation of secondarly Taylor vortices, is called the critical speed.

Taylor obtained the following equations for the critical speed for zero axial flow but with rotation :

$$\omega_c^2 = \frac{\pi^4 \nu^2 (R_r + R_s)}{2 P b^3 R_r^2} \quad \ldots\ldots \qquad 1$$

where ω_c = Angular critical speed of rotation

$$P = .0571 \left(1 - .652 \frac{b}{R_r}\right) + .00056 \left(1 - .652 \frac{b}{R_r}\right)^{-1} \qquad 2$$

R_r = radius of rotating cylinder

R_s = radius of stationary cylinder

b = gap width

ν = kinemetic viscosity of fluid

lat R_m = mean gap width

$$= \frac{R_r + R_s}{2}$$

for limiting value of $\frac{b}{R_m} \to 0$

The value of P is .0577

Taylor also defined a dimensionles factor

$$Ta = \text{Taylor number} = \omega_c R_m^{\frac{1}{2}} b^{3/2} / \nu \quad \ldots\ldots \qquad 3$$

$$\text{and } (Ta)_0 = 41.2 \quad \ldots\ldots \qquad 4$$

$$\text{and } (\omega_c)_0 = 41.2\, \nu / R_m^{1/2} b^{3/2} \quad \ldots\ldots \qquad 5$$

For any finite value of $\frac{b}{r_m}$ he obtained

$$\omega_c = (\omega_c)_o \, F_g \qquad \cdots\cdots\cdots \qquad 6$$

Where Fg is the geometrical factor and is a function of b/r_m, which is given by

$$F_g = \frac{\pi^2}{41.2} (1 - b/2R_m)^{-1} \times p^{-1} \qquad \cdots\cdots\cdots \qquad 7$$

G.I.Taylor (3) predicted that for values for speed of rotation less than ω_c, the flow in the annulus would be stable and laminar, where as for speeds greater than Wc, the flow would be unstable with the formation of steady secondary flow in the form of pairs of counter rotating donghnut shaped vortices. These vortices are shown schematically in Fig. 1. He also showed that the critical value of the gap Reynolds number is given by

$$(Reg) = 41.1 (R/b)^{\frac{1}{2}} \qquad \cdots\cdots\cdots \qquad 8$$

Taylor (3) confirmed experimentally the above expressions for critical velocity (for no axial velocity but with rotation,) as well as the existence of the pairs of vortices by using the flow of water between vertical cylinders with zero axial velocity and by injecting dye at various points in the annulus. Taylor also investigated the axial flow with rotation.

J.W. Lewis in 1927 confirmed experimently Taylors expressions given in equations (1) and (8) for zero axial velocity by observing the motion of minute particles suspended in viscous fluids...

He also observed that once the Taylor vortices are formed, as the speed of rotation is then gradually reduced , the vortices persist to lower speeds than the critical speed at which they originated. This phenomenon is similar to the persistence of turbulent flow in round pipes to Reynolds numbers less than the critical value of about 2000 as the velocity is gradually reduced below the critical value.

H. Jeffreys in 1928 investigated mathematically the stability of the layer of incompressible fluid with a decreasing temperature in the vertically upward direction.

S. Gold Stein(5) in 1937 analysed mathematically the stability of the incompressible viscous fluid. Using the method of the small purturbations, and using severe approximations, he found by numerical integration the critical speed of rotation for which the laminar flow becomes unstable. For no axial flow his value of the critical speed agreed with that of Taylor(4).

Shih I.Pai(8) in 1943 studied experimentally the turbulent flow of air in an annulus between a rotating inner cylinder and an outer stationary cylinder for zero axial flow. He used a hot wire anemometer to measure the mean valocity and the root mean square fluctuations of the velocity , and also measured the static pressure distribution along the annulus. He concluded that both the steady motion of the Taylor's vortices and the random motion of the turbulance were present.

W.H. Hagerty in 1950 utilized the optical properties of solutions of glycerin and water to study the flow patterns in a short annulus between a rotating inner cylinder and stationary outer cylinder without axial flow.

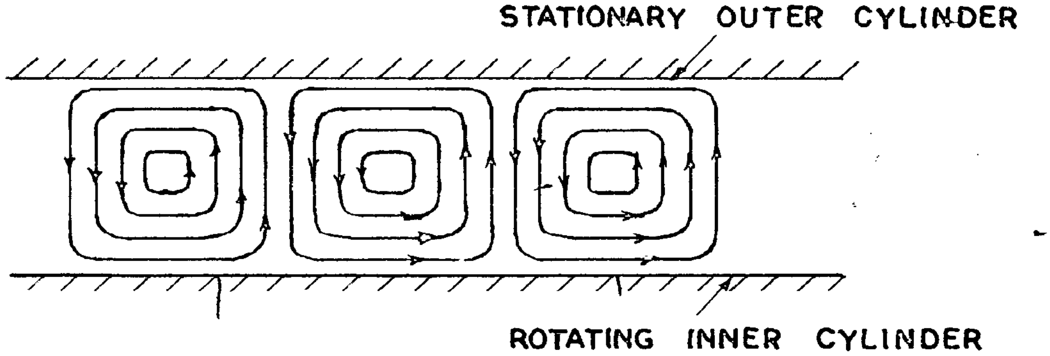


FIG.1.(A) SECONDARY FLOW OCCURING IN LAMINAR FLOW BETWEEN STATIONARY CYLINDER & ROTATING CYLINDER

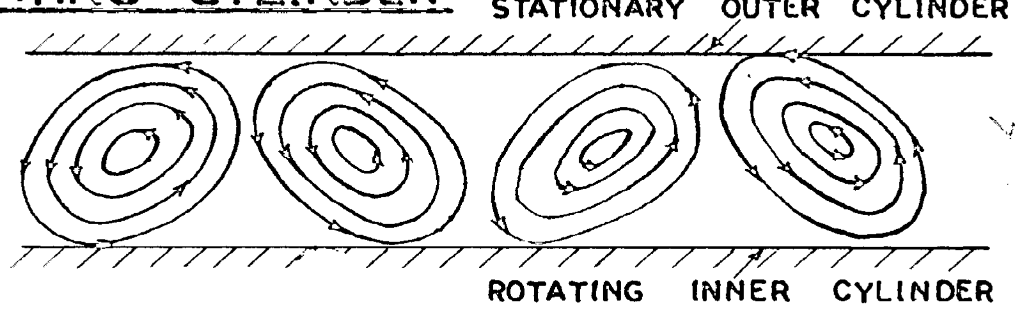


FIG.1.(B) SECONDARY FLOW OCCURING IN TURBULENT FLOW BETWEEN STATIONARY AND ROTATING CYLINDERS

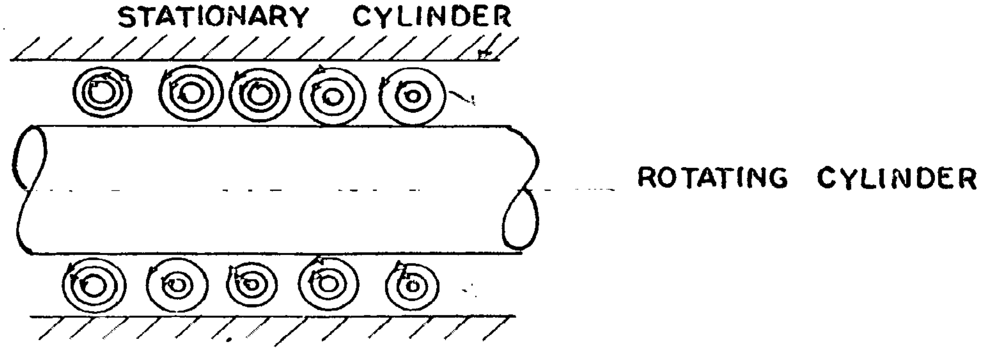


FIG. 1. (C). GAP FLOW

Other investigators also studied some of the cases without axial flow with different variables and different boundary conditions. These studies were found in papers of S.Chandra Shakar and H. Schlichting.

J.Kaye and E.C.Elgar (9) investigated the flow in an annular gap with a rotating inner cylinder by the hot wire method and photography. Their findings confirmed Taylor's predictions. Fig 1 is the schematic picture of vortices, which were generated in the gap.

Ref to their paper !!

The heat transfer charactistics were studied; however the results were presented qualitatively but not quantitatively.

F. Tachitana, S.Fukui, and H.Mitsumura(10) in 1960 investigated, as a simplified case, the heat transfer between an inner rotating cylinder and a stationary outer cylinder with out axial flow with various gap widths, rotational speeds, radii of the cylinders, and different fluids. They used five sizes of the annular gap : 088,1.97,4,6, and 10 mm in the cylinder of 58 mm diameter, and and four sizes 2,12,20,and 55mm in the cylinder of 120 mm diameter.

They used thermisters for measuring the surface temperature of the rotor surface . They evaluated the heat transfer in the air , spindle oil and motiloil for a range of Raynolds number from .84 to 4.7 x 10^4 and Taylor number from 0.49 to 4.5x10^4 with the rotational speeds of 3 to 2840 r.p.m. The radii of the inner cylinders were 29, and 60 mm.

It was predicted that when the rotational speed was low with a narrow gap in the air, the over all heat transfer coefficient was smaller. Radiation and conduction govern the major part of of the whole heat transfer.

Accordingly the heat transfer was not effected by the rotational speed.

When the rotational speed was increased and the width of gap was large, and the square of the Taylor number difined as

$$Ta = \frac{Vb}{\nu}\sqrt{\frac{b}{R}} \quad \text{------} \qquad 9$$

exceeded about 1700, a secondary vortex was generated by the centrifugal force, and the coefficeent of heat transfer was increased with velocity increase. In this region it was shown that the radiation and conduction become less prominent, and the convection of the secondary vortex was predominating. The experimental data were correlated by the Nusselt number and the Taylor number.

By this correlation the influence of the radius, gap width and rotational speed were represented by one straight line. However, because the Prandtl number of the fluid again effects the nusselt number, the data were rearranged by the Prandtl number of fluid and represented by other straight lines. In their paper the ordinate was $Nu/P_r^{1/4}$ and the abscissa was $(Ta)^2$ for $Ta^2 > 1700$, they found good agreement of data with the equation

$$Nu = .21\ (Ta^2\ Pr)^{1/4} \quad \text{-------} \qquad 10$$

This flow in the annulus was a secondary steady flow in the form of pairs of counter rotating vortices and the vortices continued regularly in the stripe pattern.

This machanism of heat transfer in the annular gap is similar to that of the air gap of the enclosed horizontal parallel planes when the heat is transferred from the lower one to the upper one. For this case the next equation was given by Jakob:

$$\frac{Ub}{K} = 0.195\ Gr^{\frac{1}{4}} \quad \ldots\ldots \qquad 11$$

$$(\ 4{*}10^5 > Gr > 10^4)$$

Where U is the overall heat transfer coefficient from the lower surface to upper one. As the motive force of convection is centrifugal force in the case of a rotating cylinder, the centrifugal force term $\frac{V^2}{R}$, was substituted for the term of the buoyancy force $g\ \beta\Delta t$ in the above equation then:

$$\frac{Ub}{K} = .195\ \left(\frac{V^2b^2}{\nu^2} \cdot \frac{b}{Rr}\right)^{1/4} \qquad 12$$

This equation is applicable only for air ,so it is rewritten by assuming that Prandtl number is .71, as follows:

$$\frac{Ub}{K} = 0.211\left(\frac{V^2b^2}{\nu^2} \cdot \frac{b}{R} \cdot Pr\right)^{\frac{1}{4}} \qquad 13$$

Which is the same as equation (10)

In this pa er it was further predicted that the above equation can be applied upto $Ta^2 = 10^{10}$ even though exceeding the Grash of limit 4×10^5 .
In reference (10) it has been shown that when the rotational speed was low at small gap widths, and when $Ta^2 > 1700$, the flow in the annulus was of the laminar character and the heat transfer by conduction and radiation was prodominating, and in addition the effect of natural convection was super imposed.

For air this effect was shown to be small, but for often liquids it was rather large as the product Gr.Pr exceeds 10^4 the experimental data agreed well with the equation

$$Nu = .11\ (Gr.Pr)^{.29} \qquad 14$$

Further for Ta > 1000, the heat transfer coefficent can not be represented by the above equation as the flows due to rotation and natural convection are super inposed on each other.

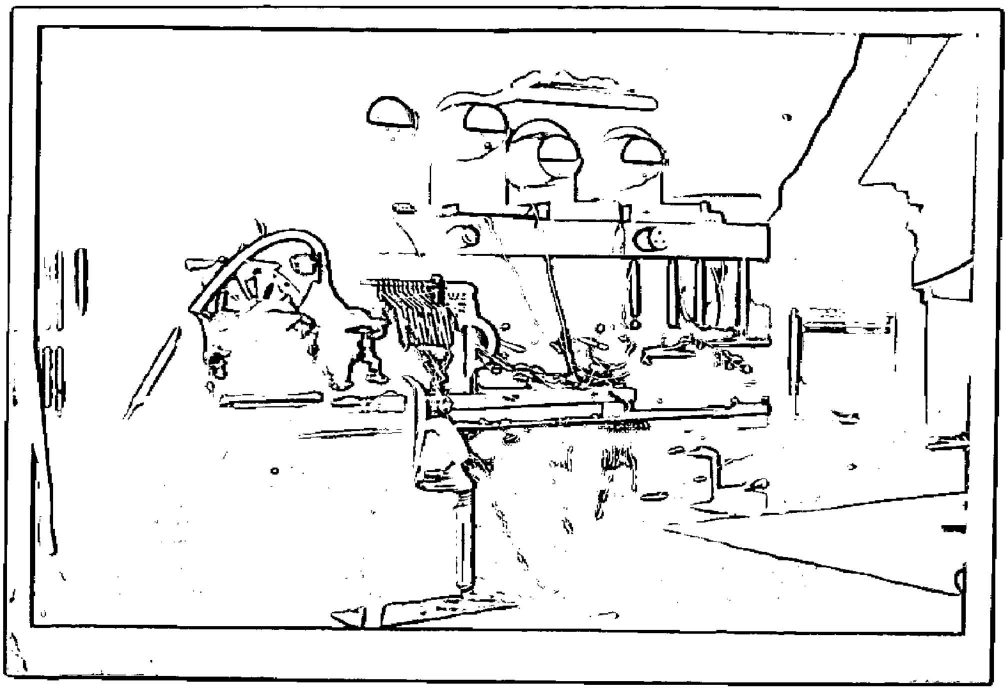
STATEMENT OF THE PROBLEM

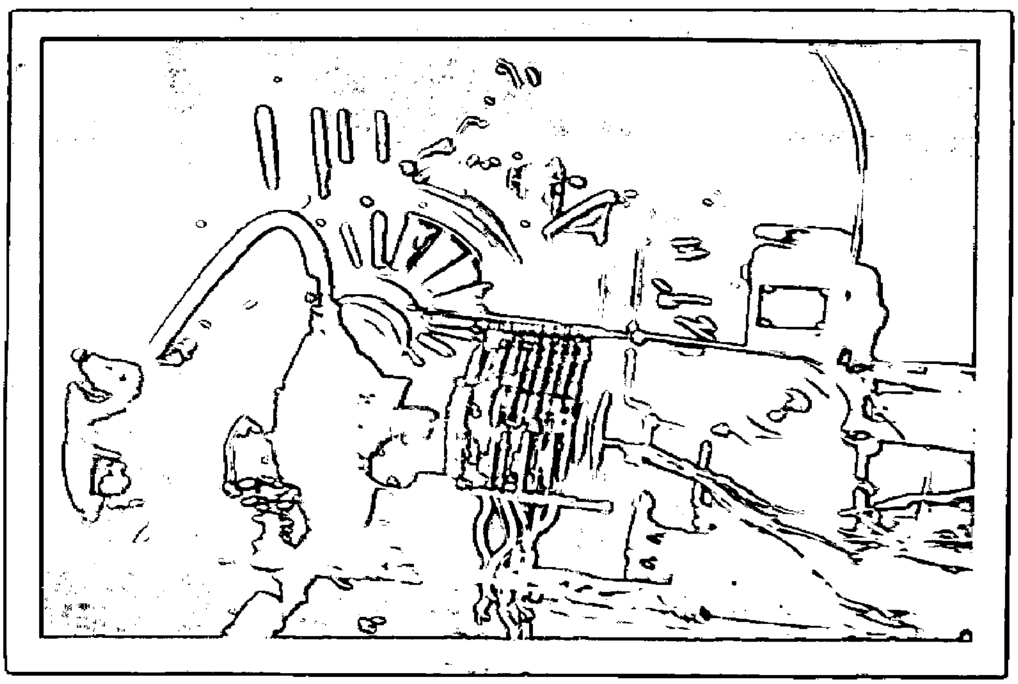
The main objectives of the present work were the following:-

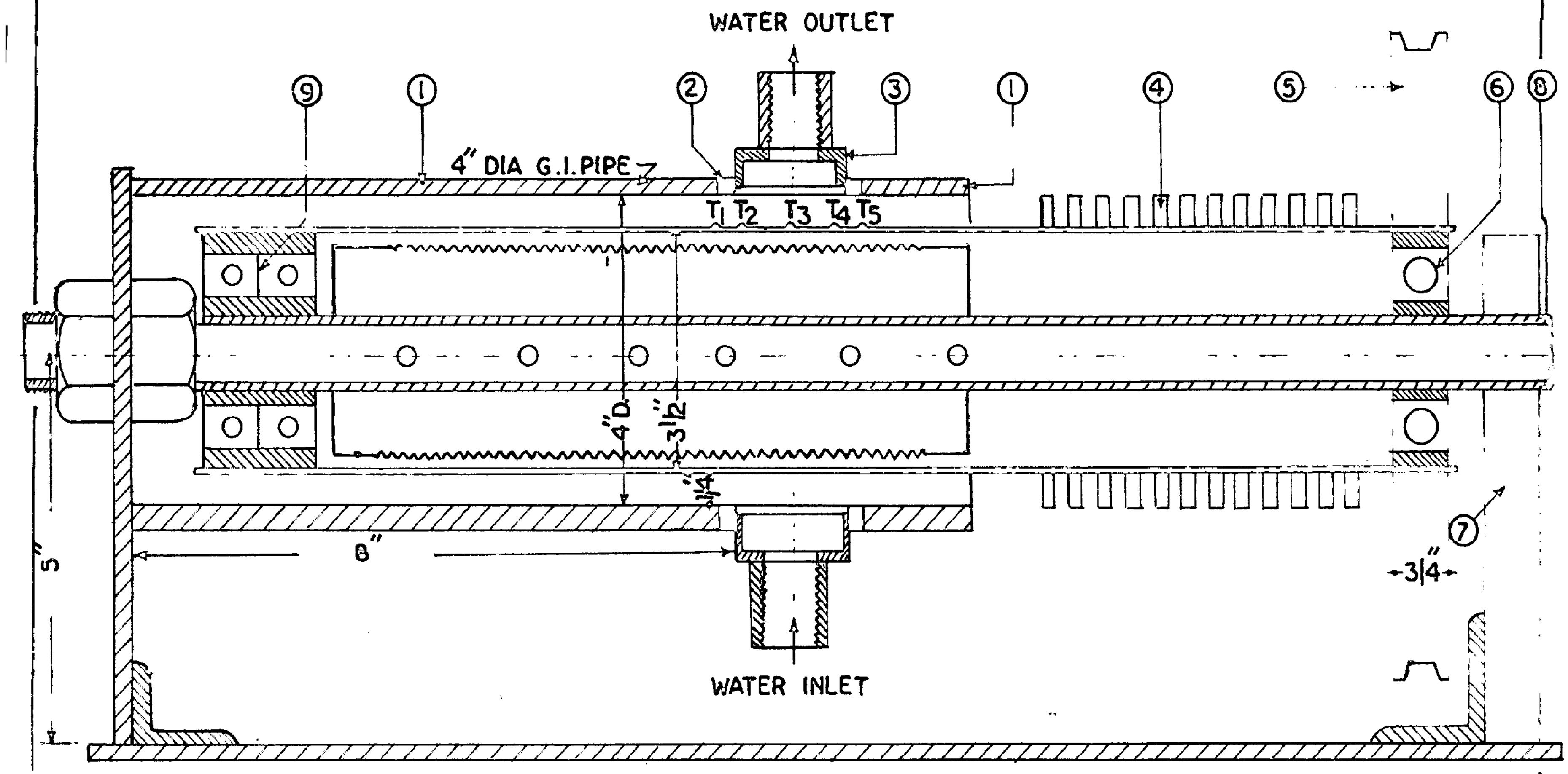
1. To obtain heat transfer data through the air gap from a rotating inner cylinder at constant temperature to a stationary outer cylinder at uniform temperature.

2. To correlate from the experimental data the non dimensional heat transfer coefficients in the form of Taylor Number and Nusselt Number

3. To compare the results with those of other investigators.







(1)	4" DIA. G.I. PIPE	(6)	BALL BEARINGS
(2)	EBONITE RING	(7)	M. S. STAND
(3)	TEST SECTION WATER JACKET	(8)	1 1/2" G. I. PIPE
(4)	COPPER SLIP RING	(9)	ALUMINIUM RING
(5)	C.I. PULLEY		

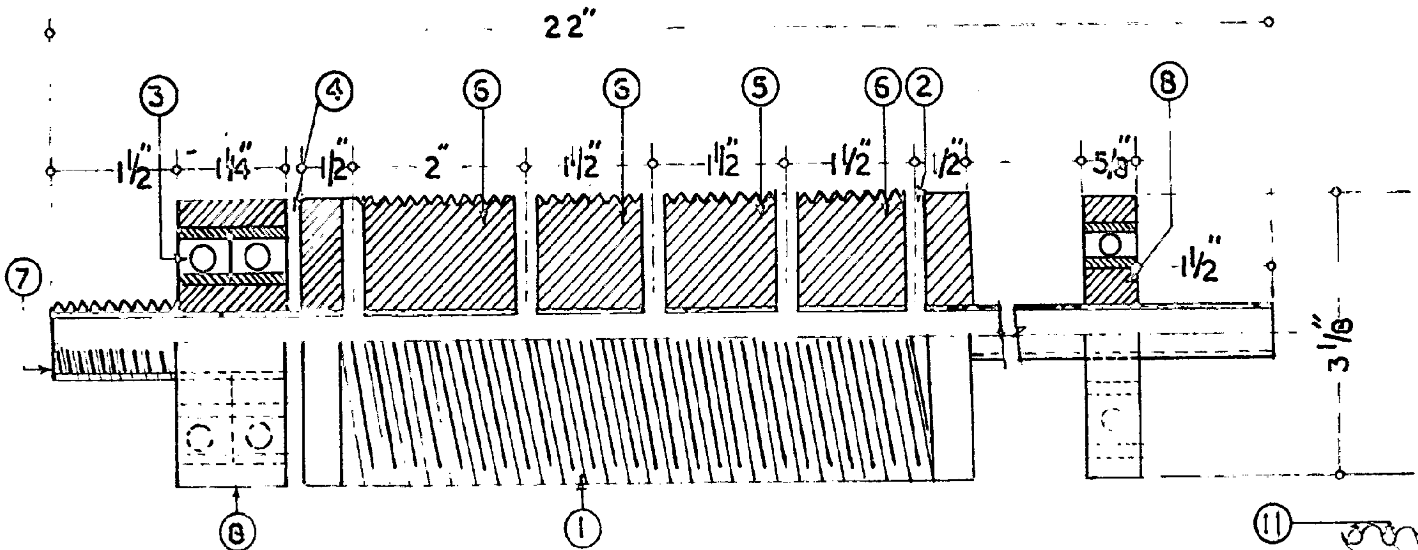
FIG.2. ASSEMBLY DETAILS

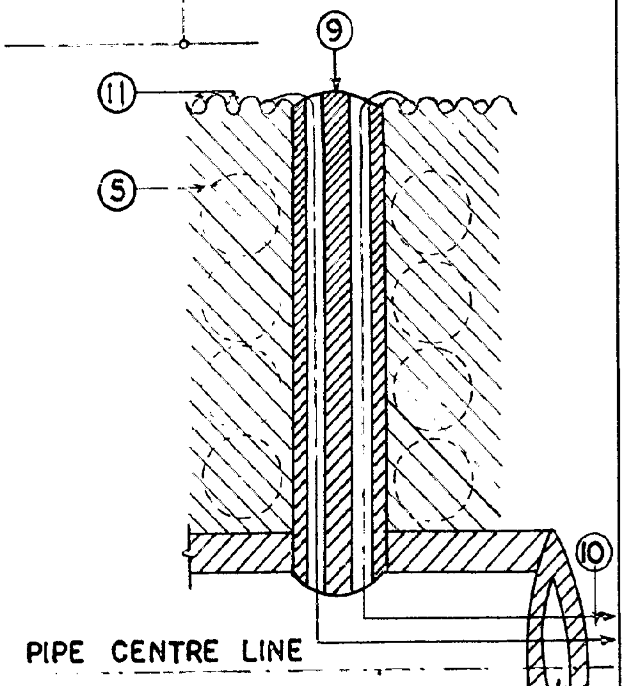
EXPERIMENTAL SET UP

3.1 Photographs on page 16 shows the general view of experimental set up . The details of the complete assembly are shown in the separate assembly in Figure 2 The equipment has an indirectly heated inner rotating cylinder, the outer and inner diamenters of which were 3.5/13 and 3.1/8 inches respectively; and was made from M.S.Pipe 18 inches long. The test section is nearly in the centre and is only 1½". The remaining portion on both sides of the test cylinder was used to serve as the guard section heater upto 4 inches from right, the cylinder had a knurling portion for fixing the 10 sliprings,each guard heater section on both sides is 1½ inches long. Each section was heated by means of nichrome wire provided in the heater assembly, the details of which are separately shown and discussed.

3.2 Heater Assembly:

Fig. 3 shows the details of the Heater Assembly the Central pipe is ½" standard G.I.Pipe and 22 inches long. This is having 4 nos. ¼ inch dia. holes in half of the portion of the pipe as shown. The 1/8" asbestos rope soaked in plaster of paris was wound on it upto 2.3/4 inches diameter, leving the holes. Over the asbestos rope, the plaster of paris mixed with gum was cast in order to make the diameter of about 3½" leaving the holes. This was then turned on lathe to makethe correct diameter of 3.1/8". V threads (10 threads per inch) were cut on this as shown.





SILICA TUBE FITTING DETAILS

1. PLASTER OF PARIS
2. 1/4" DIA. HOLES
3. BALL BEARINGS
4. ALUMINIUM WASHER
5. TEST SECTION
6. GUARD SECTION
7. 1/2" G.T. PIPE
8. C.I. BUSHES
9. SILICA TWIN HOLE TUBE 1/4" DIA.
10. HEATER LEAD WIRES
11. MICHROME WIRE

FIG. 3. HEATER ASSEMBLY

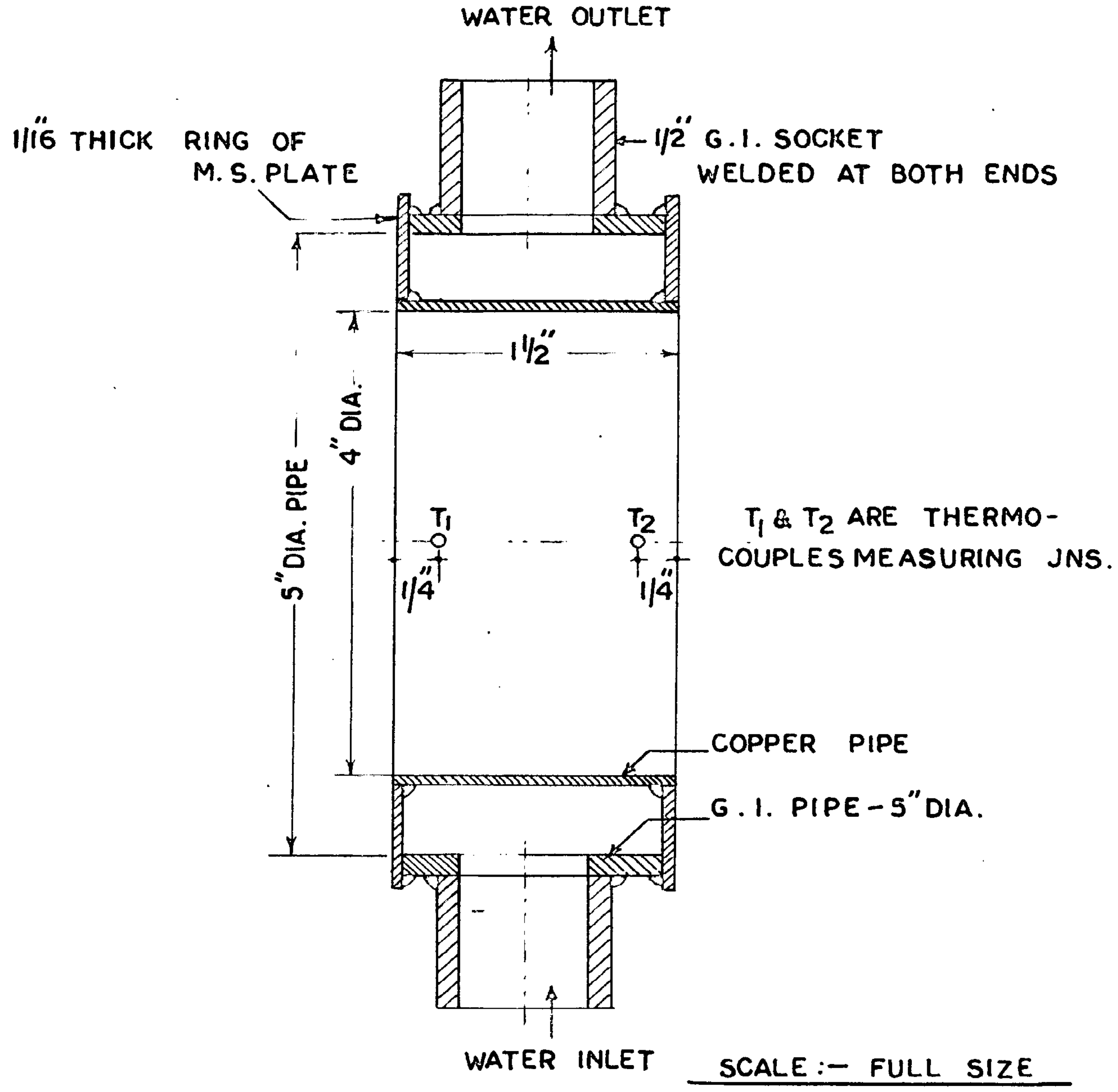


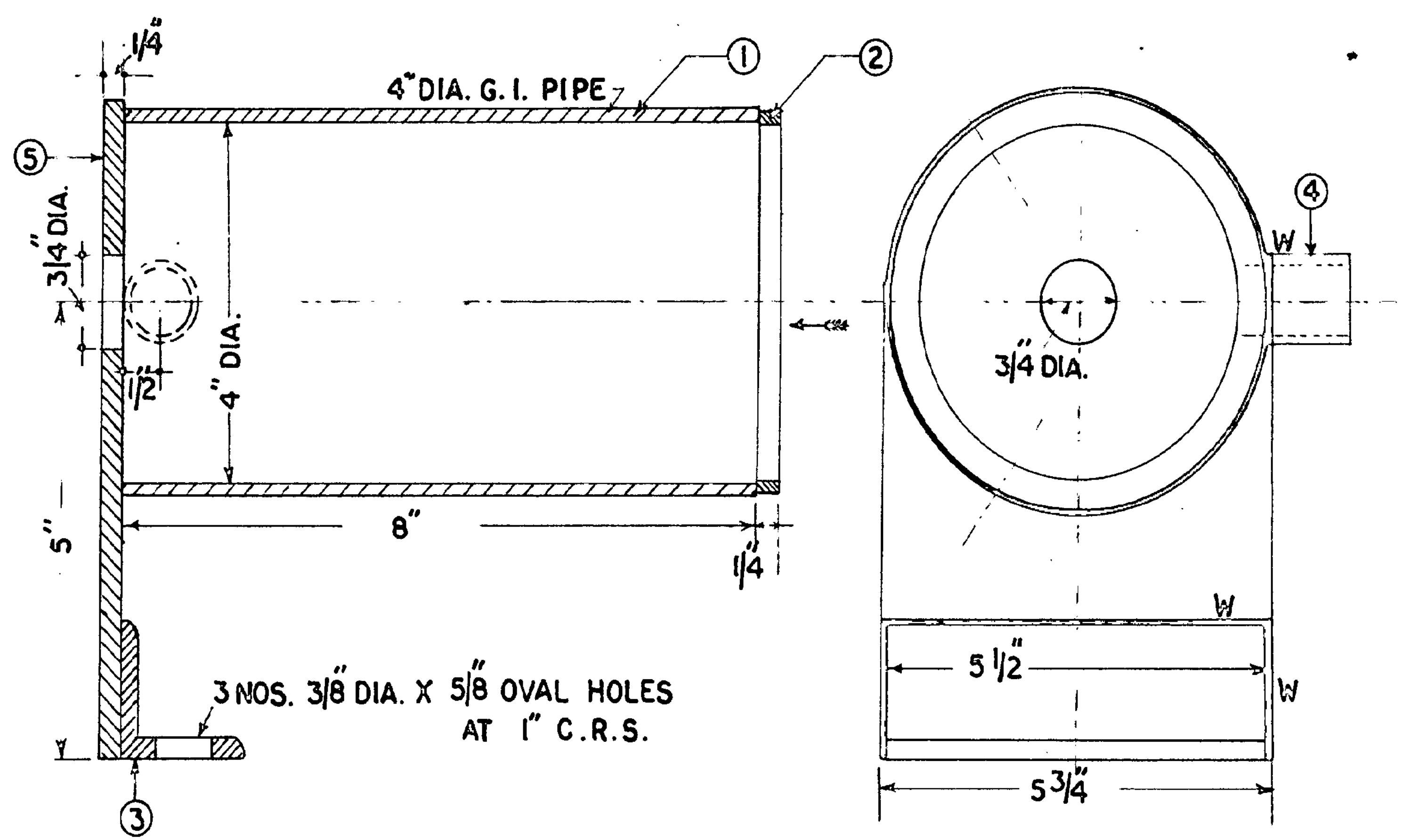
FIG. 4. TEST WATER JACKET

The 24 S.W.G. Nichrome wire was wound in the thread grooves in 4 sections the leads were then taken out through the two holes in the silica tubes, which were fitted in each $\frac{1}{4}$" hole. The enlarged view of the silica tube fitting is separately shown in the figure. To avoid the short circuiting of the heater lead wires, a 16 S.W.G. enamelled copper wire was used and on which again good quality glass sleeves were pushed on them upto the root of the silica tube.

3.3. <u>Stationary outer Cylinder:</u>

It was made up in three portions separately, the two guard sections on either side and the test section in the middle. The test section was jacketed by another cylinder of 5" dia and $1\frac{1}{2}$ inches long. The details of the test section are shown in Fig. 4 The inner cylinderical portion of the test section was made from 1/16" thick copper sheet and was of 4" inner diameter and $1\frac{1}{2}$ inches long. The outer jacket was made from 16 B.G.M.S.Sheet to form the 5 inches outer diameter and it was $1\frac{1}{2}$ inches long.

The jacketed cylinder was provided with two 1/2" G.I. sockets, which were used as inlat and outlet for the cooling water. One ebonite ring of 4" inner diameter , $4\frac{1}{2}$ inches outer diameter and $\frac{1}{4}$ inch thick, was fixedon each side of the test section by means of Araldite, Two thermocouple junctions were also fixed on the inner surface of the test section and the thermocouple wires were taken out through a 1/8" dia. hole in each ebonite ring. These holes were then closed by Araldite.



1. 4" DIA. X 8" G.I. PIPE
2. EBONITE RING 4" & 4 1/2" DIA. X 1/4" THICK
3. 1 1/2" X 1 1/2" X 3/16" M.S. ANGLE
4. 3 NOS. 1/2" G.I. SOCKET AT 120° APART
5. 1/4" M.S. PLATE WELDED TO PIPE

FIG.5. DETAILS OF FABRICATED STATIONARY CYLINDERICAL CASING

On each side of the test section the guard section stationary pipe (4" standard G.I.Pipe) was fixed as shown in the figure 2.

Thermocouple junctions were placed in the inlet and outlet water sockets by passing through 1/8" dia holes, which were finally closed by Araldite adhesive.

After fixing all the thermocouples the water jacket was completely heat insulated from the surroundings by providing about one inch thick layer of asbestoss

3.4 Heating System.

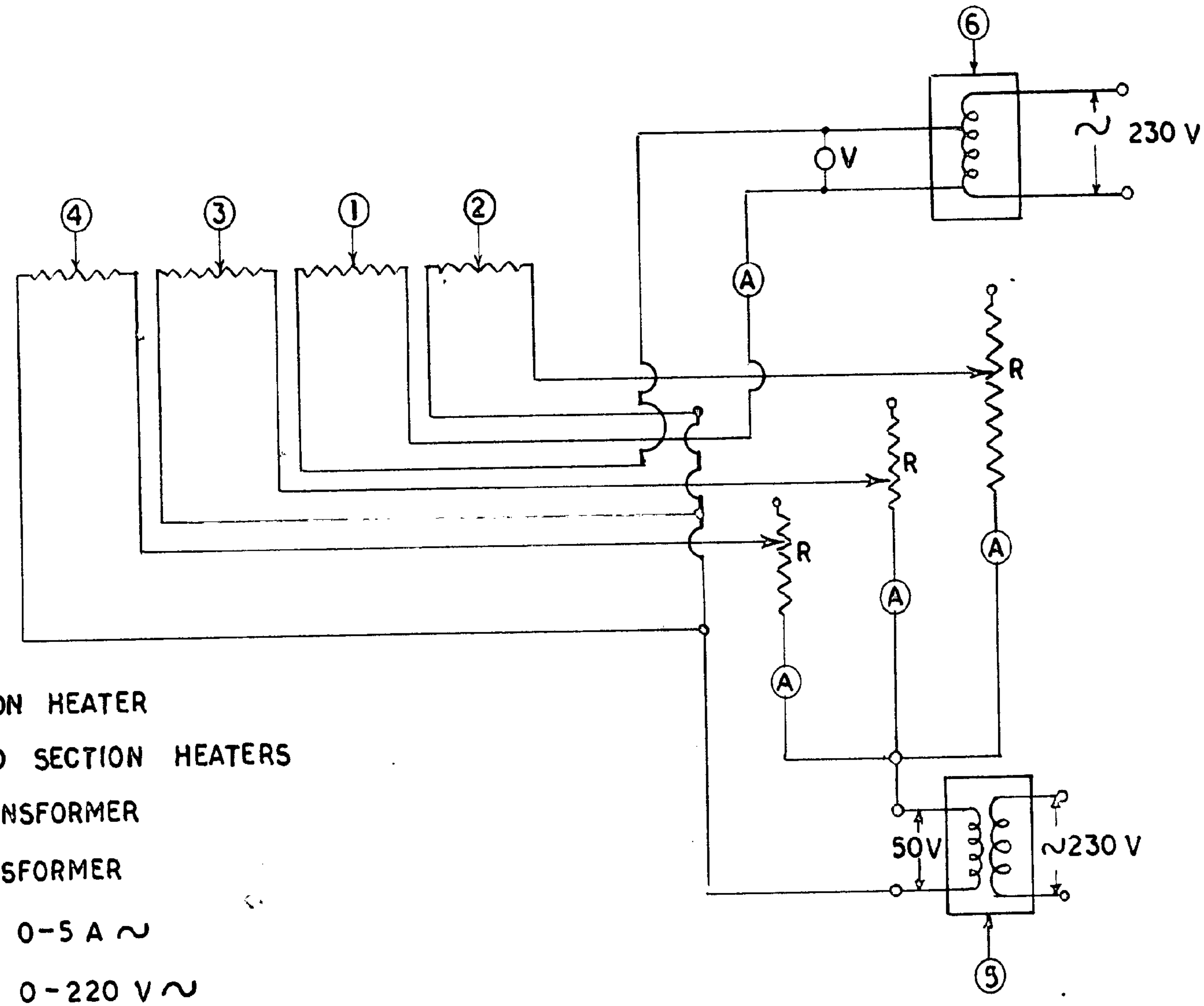
As a continuously low voltage supply is required individually to all the 4 sections of the heater, a power transformer is used to supply 55 volts from its fixed tapping of the secondary. The input is 230 volts to its primary side. One rheostat (capacity 5 A,45 Ohms) and one ammeter (O-5A), were connected in series with each guard heater, and finally each heater along with the rheostat was supplied 50 volts individually as shown in Fig. 6

The test section heater was supplied power directly from an autotransfer variac . The voltmeter and ammeter were also connected to measure the power supplied to this section.

3.5 Temperature Measurement:

Copper constantan thermocouples made from 24 S.W.G. enamelled thermocouple wires manufactured by Leads and Northrup Co., Philadalphia, U.S.A., were used to measure various temperatures.

Thermocouple junctions were made by fusing the two wires by mercury arc. Fig 7 shows the electrical circuit



1 TEST SECTION HEATER

2 3 4 GUARD SECTION HEATERS

5 POWER TRANSFORMER

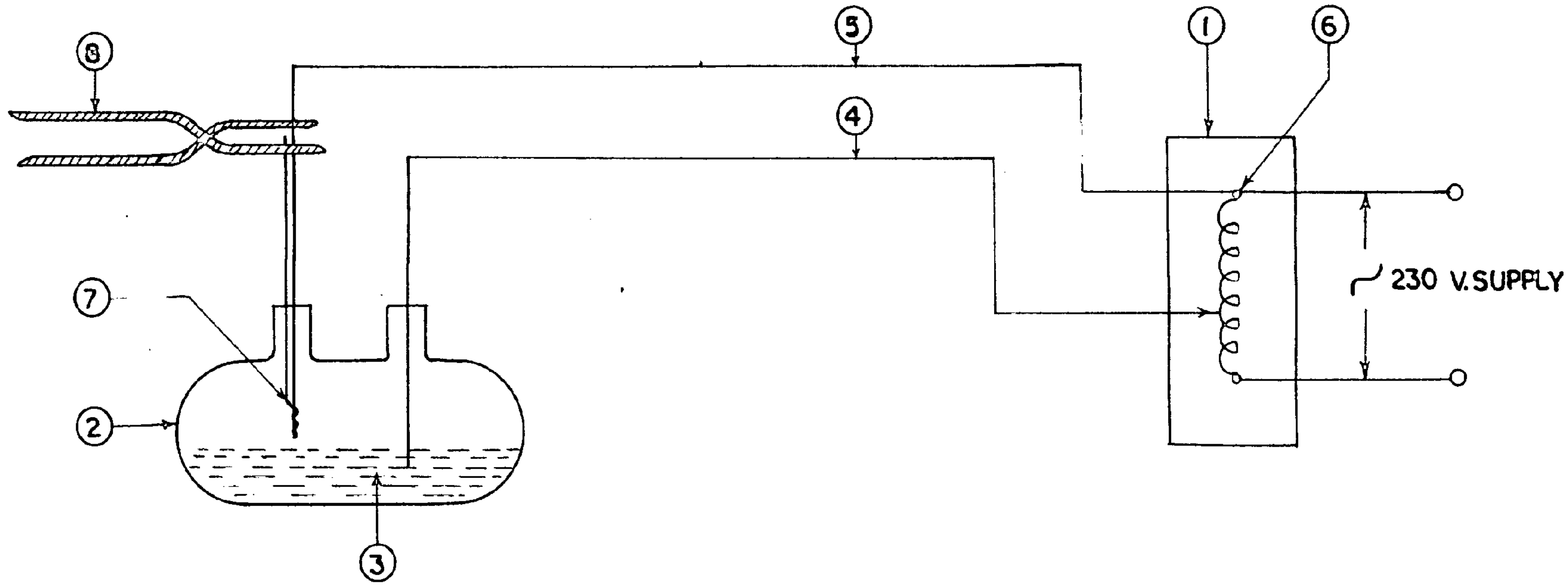
6 AUTO TRANSFORMER

A AMMETERS 0-5 A ~

V VOLTMETER 0-220 V ~

R RHEOSTALS (5A, 45 Ω)

FIG. 6. ELECTRICAL CIRCUIT OF HEATERS



1. AUTO TRANSFORMER
2. MURCURY FLASH BATH
3. MURCURY
4. LEAD FROM THE AUTO TRANSFORMER
5. COPPER THERMOCOUPLE WIRE
6. COMMON TERMINAL (NEUTRAL)
7. THERMOCOUPLE WIRE TWISTED TOGETHER
8. INSULATED GRIP PLIER

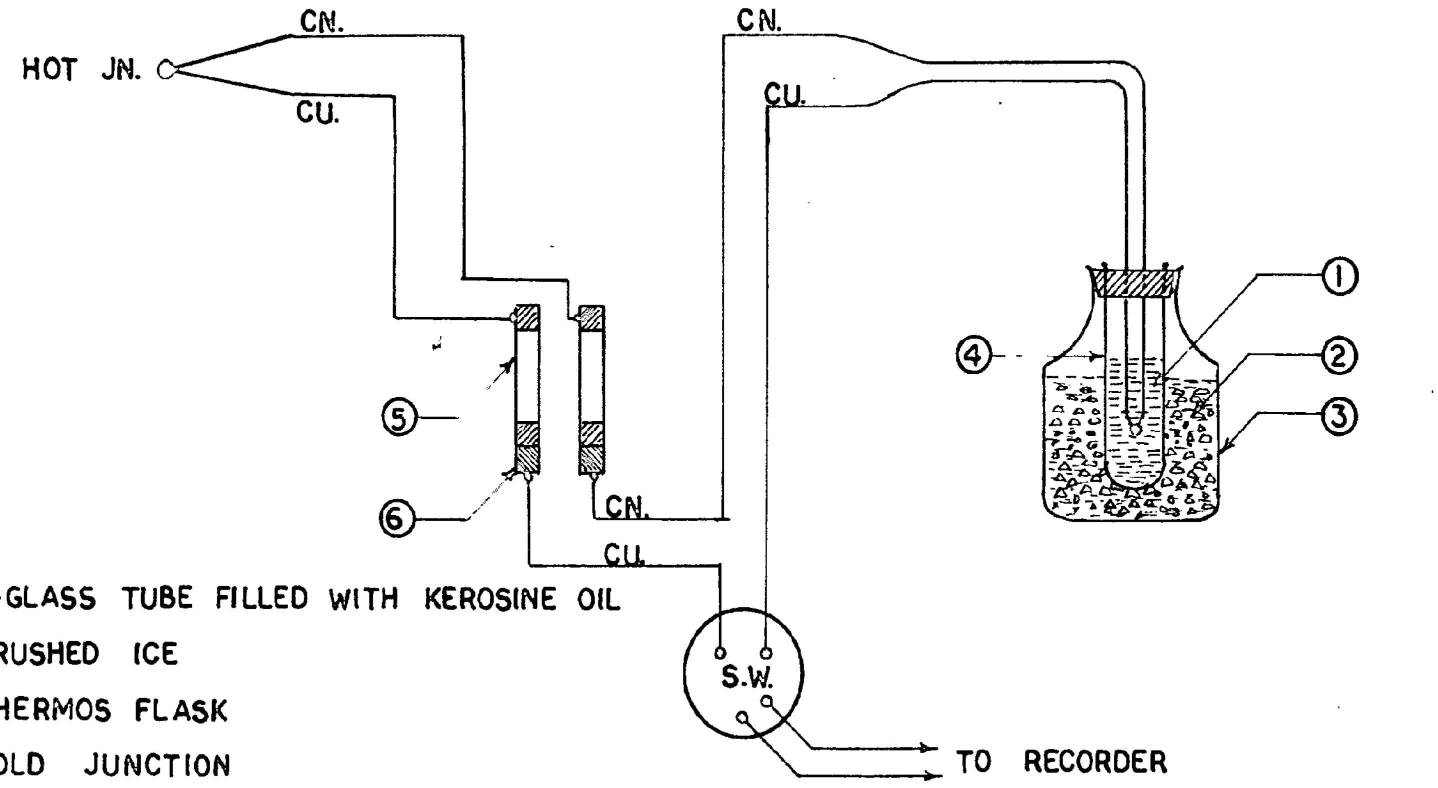
FIG. 7. CIRCUIT FOR MAKING THERMO-COUPLE JUNCTIONS

used for making the thermocouple junctions. About ½" insulation was fully removed from the two ends of the wires and than twisted. About 15 volt was tapped from the auto-transformer and the thermocouple junction and was made to just touch the mercury surface, giving rise to spark and consequent fusing of the two ends . It was necessary to get as perfect as possible truely spperical beads in order that the junctions have only a point contact when fixed to the surface.

The stationary thermocouples were connected directly to the selector switch. However in the case of the rotor thermocouples a more complicated circuit-ary was required to compensate for slipring connections.

The schematic for the rotating temperature measurement is shown in Fig 8. The thermocouple leads from the rotor x surface were brought to the slip rings separately through the key way in the rotor. Each lead of the thermocouple wires was silver soldered to the copper slip rings. One copper brush rode on each slip ring which was connected to the same material lead wire as the respective slip ring. The lead comming originally from the copper side of the thermocouple was connected directly to the selector switch However the lead comming from constantan side of the thermocouple was utilized to form the cold junction. The copper lead comming from the cold junction was then again brought to the selector switch.

Five thermocouples were fixed on the rotor as shwon in Fig. 2 . Two thermocouples were fixed on the stator inner surface by passing through a 1/8" dia hole in the ebonite ring.



① 1/4" GLASS TUBE FILLED WITH KEROSINE OIL
② CRUSHED ICE
③ THERMOS FLASK
④ COLD JUNCTION
⑤ COPPER SLIP RING
⑥ COPPER BRUSH
S.W. SELECTOR SWITCH

FIG. 8. SCHEMATIC DIAGRAM FOR ROTATING TEMPERATURE MEASUREMENT BY THERMOCOUPLE

Two thermocouples were used to measure the inlet and outlet temperature of the colling water. The thermocouples were fixed on the surface so that only a point contact as far as possible is obtained between the surface and the junction. This was achieved by providing good quality insulation sleeves to the lead wires individually.

Cold junction of all the thermocouples were insulated from each other by placing them in separate gla-ss tubes filled with kerosine oil . This was necessary to avoid the short circuiting of the thermocouples through the rotor or stator surfaces the thermocouple junctions were then dipped in these glass tubes (1/8" dia)upto a depth of about 4" i.e. nearly upto the bottom. All these tubes containing thermocouple junctions were placed in a thermosflask,filled with crushed ice. The crushed ice is than again pressed in from the top so that it has a good contact with the tube. A small amount of water is also poured in, to ensure the uniform temperature of 32° F though out.

A twelve point selector swithc was used to connect the desired thermocouple to the temperature recorder. The E.M.Fs. were measured with a single point speedomax H, continuously adjustable Azar Recorder, manufactured by M/s Leeds and Northrup . For measuring the temperatures of the stationary surfaces (i.e.,low temperatures) the 0-2 range was used , while for measuring the higher temperatures of the rotor the range of 0-10 was used.

3.6 SPEED MEASUREMENT

The surface speed of rotating cylinder was calculated by measuring the surface speed fof slip rings($4\frac{1}{4}$"dia) with the help of a speedometer . Knowing the surface speed of slip rings, the surface speed of the rotating cylinder of 3.5/16" diameter was computed.

The currents in each part of the heaters was adjusted to attain a steady value and the same temperature of the rotor surface (as indicated by the constant reading of five rotor thermocouples with the help of the temperature recorder). The flow of water was adjusted to get a temperature difference of about 2°F. During this interval the water flow rate was measured several times . At the steady state, out puts of all the thermocouples were noted with the help of the selector switch and recorder. The voltage supplied to the main test section heater was also noted.

The electrical input to all the heaters was increased in steps for the same rotational speed and the readings were taken at the steady state for each set. Similary readings were taken for other speeds of rotation of the inner rotating cylinder.

All the observations and the results of calculations were tabulated in a convinient form as shown in tables I & II various columns in these tables represent the variables as indicated below:

Column 1	:	S.No.,	it represents the Run Number
Column 2	:	Vs,	Surface speed of slip rings, ft/min.
Column 3	:	V	Peripheral velocity of the rotating test section ft/sec.
Column 4	:	tr,	Temperature of rotating surface, °F
Column 5	:	ts,	Temp. of stationary surface, °F
Column 6	:	t_1 ,	Temp. of inlet water, °F
Column 7	:	t_2 ,	Temp. of outlet water, °F
Column 8	:	w, oz/min,	Rate of cooling water
Column 9	:	E,	Input volts to test section, volts

EXPERIMENTAL PROCEDURE AND TABULATION OF DATA

Before starting the experimentation, it was necessary to check that all the thermocouple reference junctions which were kept in the individual glass tubes filled with kerosine were at 32°F. For this all the tubes were taken out from the thermosflask and the thermocouples were pushed in the glass tubes if necessary. The thermosflask was cleaned and filled with crushed ice upto three fourth of its height, the tubes were placed in it and then again the crushed ice was filled from the top. Small amount of water was added to ensure uniformity of temperature. The selector switch was rotated around for about 4 times to ensure good contacts at the switch. The temperature recorder was first switched on, and after allowing for the warming time, the outputs of all the thermocouples were noted. It was observed that all the thermocouples were at the same temperature, which confirmed that all the reference junctions were at the required 32°F temperature.

The speed variator motor was started and the speed brought to the required speed; by measuring the surface speed of rotation at the slip ring with the help of a speedometer. Again the output of the thermocouples were noted and it was ensured that all the thermocouple measuring junctions were at the same temperature and the reference junctions at another same temperature.

The cooling water was allowed to flow arround the jacket . The Heaters were also switched on. Heating was continued for about half an hour till the steady state was reached.

The currents in each part of the heaters was adjusted to attain a steady value and the same temperature of the rotor surface (as indicated by the constant reading of five rotor thermocouples with the help of the temperature recorder). The flow of water was adjusted to get a temperature difference of about 2°F. During this interval the water flow rate was measured several times . At the steady state, out puts of all the thermocouples were noted with the help of the selector switch and recorder. The voltage supplied to the main test section heater was also noted.

The electrical input to all the heaters was increased in steps for the same rotational speed and the readings were taken at the steady state for each set. Similary readings were taken for other speeds of rotation of the inner rotating cylinder.

All the observations and the results of calculations were tabulated in a convinient form as shown in tables I & II various columns in these tables represent the variables as indicated below:

Column 1 :	S.No.,	it represents the Run Number
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Column 4 :	tr,	Temperature of rotating surface,°F
Column 5 :	ts,	Temp. of stationary surface, °F
Column 6 :	t_1 ,	Temp. of inlet water,°F
Column 7 :	t_2 ,	Temp. of outlet water, °F
Column 8 :	w,oz/min,	Rate of cooling water
Column 9 :	E,	Input volts to test section, volts

Column	10 :	Q_c	Rate of total heat passing through the annular gap (per hr.) by convection and radiation
Column	11 :	Q_{rad}	Rate of heat passing to stationary Cylinder by radiation, Btu/hr.
Column	12 :	Q_c	Rate of heat passing to stationary cylinder by convection $(Q_c - Q_{rad})$ Btu/hr.
Column	13 :	U ,	Over all heat transfer coeff. by convection, Btu/ft^{2}hr $^\circ$F
Column	14 :	t_m ,	Mean temp. of the fluid $\frac{(t_r + t_s)}{2}$, $^\circ$F.
Column	15 :	ν	Kinematic viscosity of fluid at t_m, ft^2/sec.
Column	16 :	K ,	Conductivity of fluid at mean temp., Btu/ft hr $^\circ$F.
Column	17 :	Nu,	Nusselt number $\frac{Ub}{K}$
Column	18 :	Ta,	Taylor number $\frac{Vb}{\nu}\sqrt{\frac{b}{r}}$
Column	19 :	Ta^2	Square of Taylor number
Column	20 :	$\log(Ta)^2$	log. of the square of Taylor number
Column	21 :	log(Nu)	log. of the nusselt number.

TEST RESULTS AND CORRELATION OF DATA

The experimental test data was obtained by rotating the inner cylinder at different speeds, and for each speed the heat input was varried for a wide range. The data thus obtained is presented in table I & II . The experimental range covered is given below:

1. Rotational speed : 15 R.P.M. to 400 R.P.M.
2. Surface temperatures 100°F to 300 °F
3. Heat input 0-70 Btu/ hr.
4. Taylor number 15 to 300

Most of the heat input of the test section is dissipated by convection and radiation from the test surface to the stationary surface; this was computed by measuring the temperature rise of cooling water and its flow rate. The cooling water was made to flow around the test stationary surface only. While calculating the convective heat transfer coefficient the radiation loss from the surface was taken into account to avoid the heat flow by conduction axially, the guard heaters were provided which maintain the same temperature as the temperature of the test section.

The radiation heat transfer can be computed from the equation

$$Q_{rad} = FA\,\varepsilon\sigma\,(T_r^{\,4} - T_s^{\,4}) \qquad \ldots\ldots\ldots \qquad 1$$

The value of emmisivity ε for mild steel was taken as .8 and that for copper as 0.78 and then the combined emmisivity and shape factor F_{1-2} was calculated as 0.787 using the equation

$$F_{1-2} = \frac{1}{F + \left(\frac{1}{\varepsilon_1} - 1\right) + \frac{A}{A_2}\left(\frac{1}{\varepsilon_2} - 1\right)} \qquad \ldots\ldots \qquad 2$$

The radiant heat transfer was then calculated by using the following equation

$$Q_{rad} = F_{1-2} A_1 \sigma (T_r^4 - T_s^4) \quad \ldots\ldots\ldots \quad 3$$

where σ = Boltzman constant

Heat transfered by convection is then obtained by subtracting the heat transfered by radiation from the total heat transfered $Q_c = Q_e - Q_{rad}$. The over all convection heat transfer coefficient is obtained by

$$U = \frac{\text{Heat transfered by convection}}{A(t_r - t_s)}$$

Where A is the area of rotating surface under test and t_r, and t_s are the temperatures of the rotating and stationary surface respectively.

In defining the Raynold number $(\frac{VD}{\nu})$ for rotation, characteristic diameter D, has to be specified. For flows through a section other than a circular one, usually an equivalent diameter is used for D. Most of the investigators in the field of annulus heat transfer have used equivalent diameter as four times the hydraulic radius as

$$D_e = 4 \cdot \frac{\frac{\pi}{4} \cdot (D_2^2 - D_1^2)}{(D_2 + D_1)}$$

$$= (D_2 - D_1)$$

$$= 2\,b$$

where b is the gap width.

Vr is mean speed of rotation of the outer stationary and inner rotating cylinder (i.e. $V_r = \frac{V}{2}$)

Thus the mean reynolds number $(\frac{V_r D}{\nu}) = \frac{V/2 \; .2b}{\nu}$

$$= \frac{Vb}{\nu}$$

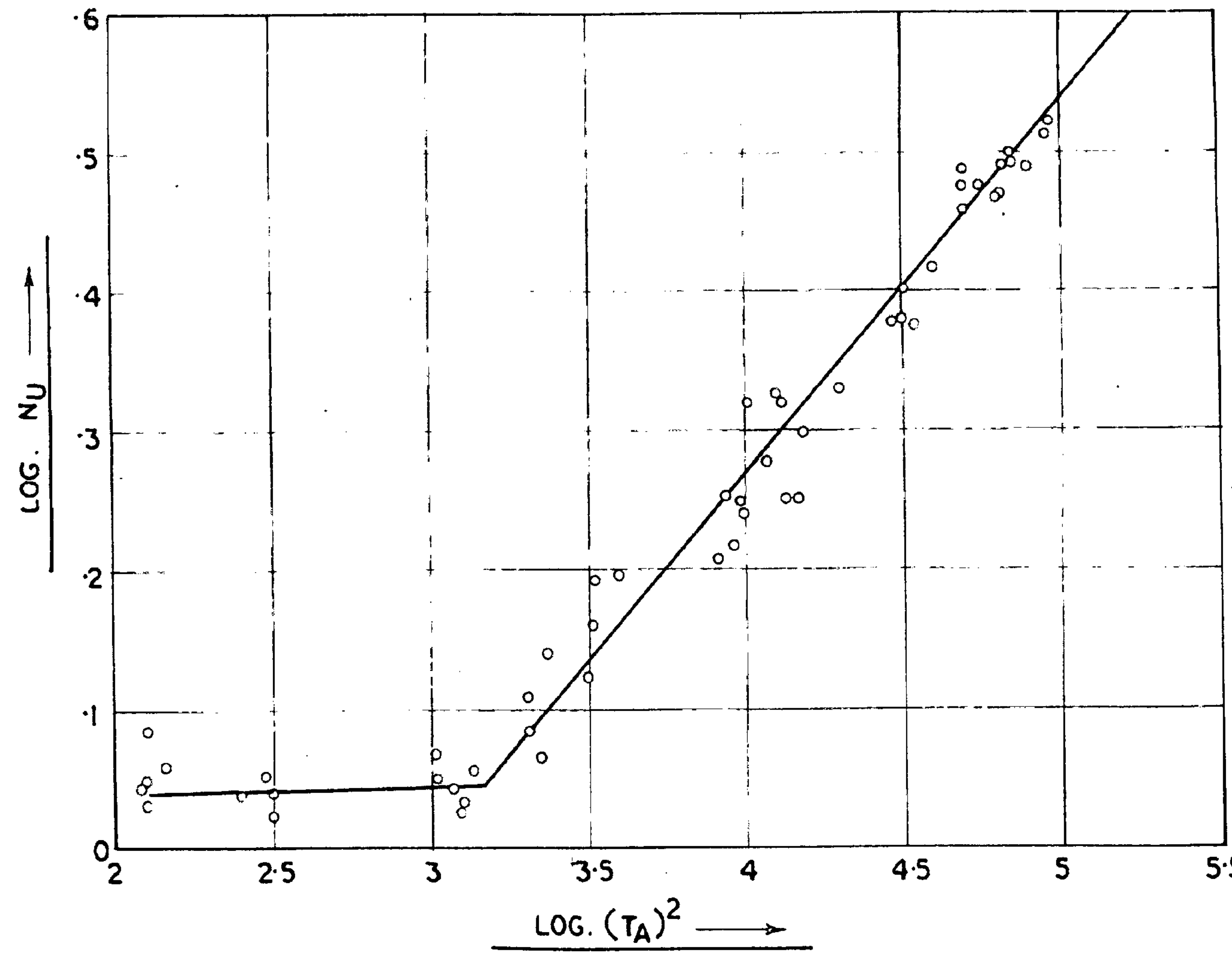

FIG.9. A PLOT BETWEEN $LOG_{10}\ (T_A)^2$ AND LOG OF N_U NUMBER

Similarly the Taylor number is defined as the product of mean Reynolds number and the dimensionless cuwature factor i.e. ($T_a = \frac{Vh}{\nu} \sqrt{\frac{h}{R_r}}$) for the purpose of present investigations, these difinitions have been used

As the data on coefficient of heat transfer by convection is frequently represented in non dimensional form, involving characteristic quantities fof the system the data has been corralated by the equation of the form

$$Nu = C(T_a)^n \ (Pr)^n \qquad \ldots\ldots \quad 4$$

All the property values have been evaluated at the mean temperature of the rotating and stationary surface.

Even at high heat inputs and low rotational speeds, the difference between surface temperature is 60 ^oF, the variation of the Prandtl number for air is very small, of the order of 1.0% its influence has been neglected, for the air only. Thus the correlation is represented for air in the form of

$$Nu = C(Ta)^n \qquad \ldots\ldots \quad 5$$

A graph was plotted as shown in fig 9 with logarithm of the square of Taylor number as abscissa and logarithm of Nusselt number as ordinate.

It was found that the following correlation represents most of the data (above the critical value of Taylor number 39) fairly well.

$$Nu = .152(T_a)^{.54} \qquad \ldots\ldots. \quad 6$$

In the range of $39 < Ta < 304$, the maximum deviation of the data was ± 16%. other investigators have also reported the maximum deviation of the similar order of magnitude in their correlations.

It would be of interest to compare equation with similar equations given by other investigators. Substituting the value of Prandtl number as 0.72 in the equation

$$Nu = .21 (T_a^{2} Pr)^{\frac{1}{4}}$$

Suggested by Tachibana and Fukui (10) for Ta > 41, we get

$$\begin{aligned} Nu &= .21 (Ta^{2} (.72))^{\frac{1}{4}} \\ &= .21 \times .92 (Ta)^{0.5} \\ &= .193 \ (Ta)^{.5} \qquad \ldots\ldots\ldots \ 7 \end{aligned}$$

The value of the constant in the suggested coorelation is 0.193, where as in the present investigation is 0.152, the difference can be accounted by considering the index of Taylor number which is 0.52 in this investigation inplace of the index 0.5 as suggested by Tachibana and Fukui in their paper.
Further the value of the critical Taylor number is found to be 39 while the above investigators reported the value of the critical Taylor number as 41. This difference of the critical Taylor number is mainly due to the turbulance occured due to the thermocouple lead wires which were fixed on the rotating cylinder. Further the difference in constant and index may be accounted for by the consideration of the following points:

1. The formation of true annulus is difficult to produce experimentally by means of fixing two separate cylinders.

2. The use of combinations of inner surface and outer surface with different finished can produce the discrepancy.

3. In the present investigation the presence of the thermocouple junctions on the rotating cylinder causes about 5% of the increases in diameter,which in turn increases the local Taylor number, and on the other hand reduces width also, which mayresult to increase in heat transfer.

4. The local heat transfer may be different due to the non uniform thicknesses of the inner and outer cylinder. Different temperature gradients would be established which inturn may effect the over all heat transfer coefficient.

5. The temperatures obtained through the rotating junctions may be different than the actual temperatures at the surfaces.

CONCLUSION

The heat transfer by convection in the annular gap between the rotating inner cylinder and stationary outer cylinder was experimented for the air at various rotational speeds. It was found that the heat transfer can be divided into two regions:

1. When the speed of rotation is greater than the speed which makes the Taylor number greater than 39, the heat transfer increases with the rotational speed probably due to the formation of secondary vertices. This relation for air can be expressed in the form of

$$Nu = .152 \; (Ta)^{.52} \ldots\ldots \; Ta > 39$$

2. At low rotational speeds i.e. when the Taylor number is less 39, the heat transfer is not effected by the speed of rotation . In this region the flow is laminar and the heat transfer by conduction and convection prodominates.

TABLE NO. I

OBSERVATIONS

For notations see page. 31...

1	2	3	4	5	6	7	8	9	10
S.No.	V_s, ft/min.	V_2 ft/sec.	t_r, °F	t_{sT} °F	t_1, °F	t_2 °F	w oz/min.	E.volts	Q_T
1.	360	4.51	158	92	87	89	2.75	10	20.6
2.	"	"	170	92	87	89.5	"	13	25.8
3.	"	"	203	93	87	90.5	"	15	36.1
4.	"	"	270	95	87	93.5	"	18	67
5.	315	3.933	165	90	86.5	88.5	3.125	10	23.4
6.	"	"	178	92	87	89.5	3.125	12	29.1
7.	"	"	228	93	87	90.	4.0	15	45
8.	"	"	280	94	87	91.5	4.0	18	67.5
9.	270	3.4	162	95	89.5	92	2.125	10	20
10.	"	3.4	186	96	90	93.5	2	12	26.3
11.	"	3.4	247	100	91.5	98	2	15	48.7
12.	"	3.4	307	103	91.5	101	2	18	71

1	2	3	4	5	6	7	8	9	10
S.No.	V_s,	V_2	t_r, °F	t_s °F	t_1, °F	t_2 °F	W oz/min	E,volts	η_T
13	215	2.833	166	94	89.5	92	2.125	10	20
14	"	"	209	95	91.0	94.5	2.5	12	32.8
15	"	"	236	95	90	94.5	2.5	15	42.2
16	"	"	327	96	91	95	5.25	18	79
17	150	1.95	178	97	94	96	2.625	10	16.7
18	"	"	196	98.5	94	96.5	"	12	24.6
19	"	"	217	99	94	97	"	13	29.5
20	"	"	240	100	94	98	"	14	39.4
21	"	"	266	101	94	99	"	15	49.22
22	"	"	290	102	94.5	100	"		54.14
23	125	1.32	182	97	92.0	95	1.75	10	19.7
24	"	"	198	97.5	92.0	93	"	12	26.3
25	"	"	221	99	92.0	93.5	"	13	26.6
26	"	"	243	100	92.0	97.5	"	14	36.1
27	"	"	269	100.5	92.0	99	"	15	46

1	2	3	4	5	6	7	8	9	10
S.No.	V_s,	V_2,	t_r, °F	t_s, °F	t_1, °F	t_2, °F	W oz/min.	E,volts	Q_T
28	80	1	205	95	89	92	2.25	10	25.3
29	"	"	228	97	89	92.5	"	12	29.5
30	"	"	251	97.5	89.5	94	"	13.5	38
31	"	"	269	98	89.5	95	"	15	46.2
32	60	.78	209	94	88.5	91.5	2.25	10	25.3
33	"	.78	232	96	89	92.5	"	12	29.5
34	"	.78	258	97.5	89	93.5	"	13.5	38
35	"	.78	284	98	89	94.5	"	15	46.2
36	50	.58	213	98.5	93.6	96	2.5	10	23.4
37	"	"	232	99	94	97.0	"	12	28.1
38	"	"	247	99	94	97.5	"	13	32.7
39	"	"	262	100	94.5	98.5	"	14	37.7
40	"	"	280	102	95	99.5	"	15	42

1	2	3	4	5	6	7	8	9	10
S.No.	V_s,	V_2,	t_r, °F	t_s, °F	t_1, °F	t_2, °F	W oz/min	E, volts	Q_T
41	25	.29	202	98	95	97	2.75	10	20.7
42	"	"	221	99	96	98.5	"	12	25.9
43	"	"	240	101	96	99	"	13	30.6
44	"	"	255	102	97	100.5	"	14	36.3
45	"	"	277	104	97	101	"	15	41.2
46	16	.195	197	97	94	96	2.75	10	20.7
47	"	"	221	98	95	97.5	"	12	25.9
48	"	"	240	100	95	98	"	13	30.6
49	"	"	252	101	95	98.5	"	14	36.3
50	"	"	277	103	96	100	"	15	41.2

TABLE NO. II

CALCULATIONS

For notations see page..31...

1	11	12	13	14	15	16	17	18	19	20	21
S.No.	Q_{rad}	Q_c	U	t_m	10^{-3}	K	N_u	T_a	$T_a^2 \times 10^{-4}$	$\log T_a^2$	$\log N_u$
1	7.71	12.9	1.807	125	.195	.016	3.24	304.1	9.25	4.966	.511
2	9.412	16.4	1.845	131	.198	.016	3.29	299.5	8.97	4.952	.518
3	14.4	21.7	1.842	148	.21	.017	3.16	282.4	7.97	4.902	.5
4	30.3	36.7	1.92	182	.226	.017	3.12	262.4	6.83	4.838	.494
5	8.9	14.5	1.8	127	.196	.016	3.2	263.9	6.96	4.842	.506
6	10.6	18.5	1.8	135	.201	.016	3.1	257.3	6.62	4.821	.492
7	18.99	26.07	1.78	160.5	.214	.017	3.0	241.7	5.84	4.766	.478
8	29.92	37.57	1.87	186	.233	.076	3.04	223.0	4.97	4.696	.483
9	7.9	12.03	1.73	128	.196	.016	2.92	228.1	5.2	4.716	.466
10.	10.5	15.8	1.74	141	.204	.016	3.00	219.2	4.8	4.682	.492
11	22.04	26.65	1.68	173	.223	.017	2.7	200.5	4.02	4.606	.432
12	35.73	35.26	1.6	205	.242	.018	2.5	184.75	3.41	4.532	.398

1	11	12	13	14	15	16	17	18	19	20	21
S.No.	Q_{rad}	Q_c	U	t_n	10^{-3}	K	N_u	T_a	$T_a^2\ 10^{-4}$	$\log T_a^2$	$\log N_u$
13	8.63	11.36	1.46	130	.198	.016	2.57	188.2	3.54	4.55	.41
14	15.34	17.46	1.418	152	.210	.016	2.417	173.6	3.117	4.493	.384
15	20.34	21.86	1.435	164	.216	.017	2.39	172.5	2.974	4.473	.379
16	42.01	36.98	1.476	211	.245	.018	2.31	152.0	2.312	4.364	.364
17	10.1	6.598	.75	137	.202	.016	1.34	126.9	1.611	4.207	.127
18	12.62	11.97	1.15	147	.208	.016	2.01	123.3	1.52	4.182	.302
19	16.36	13.14	1.03	158	.214	.016	1.78	119.8	1.436	4.158	.25
20	20.33	18.77	1.24	170	.221	.016	2.12	116.03	1.346	4.13	.322
21	26.0	23.2	1.30	184	.23	.017	2.18	111.5	1.243	4.094	.328
22	31.5	22.6	1.11	196	.236	.017	1.84	108.2	1.171	4.068	.27
23	10.7	8.99	.979	140	.204	.016	1.73	104.4	1.09	4.038	.238
24	13.22	13.1	1.2	148	.208	.016	2.11	102.4	1.049	4.02	.322
25	17.1	12.5	.95	160	.215	.016	1.34	99.1	.98	3.99	.213
26	21.2	14.9	.96	171	.222	.016	1.34	95.5	.913	3.96	.213
27	26.7	19.3	1.05	185	.23	.017	1.77	92.6	.857	3.93	.248

1	11	12	13	14	15	16	17	18	19	20	21
S.No.	Q_{rad}	Q_c	U	t_n	10^{-3}	K	N_u	T_a	$T_a^2\ 10^{-4}$	$\log T_a^2$	$\log N_u$
28	14.65	10.65	.896	151	.21	.016	1.56	62.6	.392	3.594	.193
29	18.6	10.9	.77	162	.217	.016	1.32	60.6	..367	3.564	.12
30	23.13	14.9	.897	174	.224	.016	1.52	58.7	.345	3.538	.18
31	26.98	19.2	1.04	183	.229	.171	1.74	57.4	.33	3.518	.16
32	15.4	9.8	.794	152	.211	.016	1.38	48.6	.236	3.374	.14
33	19.46	10.0	.683	165	.218	.016	1.17	47.3	.221	3.344	.D68
34	24.6	13.4	.772	178	.226	.017	1.3	45.4	.206	3.314	.114
35	30.5	15.7	.783	191	.234	.017	1.3	43.8	.192	3.284	.114
36	15.7	7.7	.623	156	.213	.016	1.08	35.8	.128	3.108	.034
37	19.2	8.9	.622	166	.219	.016	1.06	34.8	.121	3.08	.026
38	22.1	10.5	.66	173	.223	.016	1.11	34.2	.117	3.06	.043
39	25.2	12.1	.69	181	.228	.017	1.17	33.4	.112	3.05	.07
40	29.1	12.9	.67	191	.234	.017	1.12	32.5	.106	3.02	.05

1	11	12	13	14	15	16	17	18	19	20	21
S.No.	Q_{rad}	Q_c	U	t_m	10^{+3}	K	N_u	T_a	$T_a^2\ 10^{-4}$	$\log T_a^2$	$\log N_u$
41	13.8	6.9	.61	150	.21	.0164	1.06	18.16	.033	2.51	.022
42	17.1	8.8	.67	160	.215	.017	1.1	17.7	.032	2.5	.042
43	20.5	10.8	.67	170	.221	.017	1.14	17.2	.03	2.47	.057
44	23.5	12.8	.75	180	.227	.017	1.3	16.8	.028	2.45	.114
45	28.2	13.0	.69	192	.235	.0172	1.1	16.2	.026	2.42	.042
46	13.2	7.5	.687	147	.207	.0164	1.2	12.4	.0153	2.2	.08
47	17.2	8.7	.656	160	.215	.0166	1.13	11.93	.0142	2.15	.052
48	20.6	9.97	.66	170	.221	.0168	1.12	11.6	.0135	2.13	.05
49	25.1	11.1	.642	182	.226	.017	1.08	11.3	.0129	2.11	.034
50	28.3	12.9	.68	190	.234	.0172	1.1	10.9	.012	2.08	.042

APPENDIX B

SAMPLE CALCULATION AND COMPUTOR PROGRAMMING

Reading No. 14

1. Annulus Data :- Dia of stator D_s = 4"

Dia of rotor D_r = 3.5/16"

Gap width b $= \frac{1}{2}(D_s - D_r)$

$= \frac{1}{2}(4 - 3.5/16)$ inches

$= \frac{1}{2} \times 11/16 \times 1/12$ ft.

= 1/35 ft.

Radius of rotor R_r = 53/16x2x12

= .138 ft.

$\sqrt{\frac{b}{R_r}} = \sqrt{\frac{1}{35 \times .138}}$

$= \sqrt{.210}$

= .46

$b \times \sqrt{\frac{b}{R_r}} = \frac{.46}{35}$

Test length L = 1.5 Inches

Heat transfer area from rotor $= \pi D_r L$

$= \pi \frac{53}{16} \times \frac{3}{2} \times \frac{1}{144}$

= .108 Sq.ft.

2. Surface speed at slip rings (of radius $4\frac{1}{4}$") V_s = 215 ft./min.

Surface speed of rotor V $= \frac{215 \times 4 \times 53}{17 \times 16 \times 60}$ ft./sec.

= 2.833 ft/sec. ... Column 3

3. Temp. of rotor surface = 200 °F

= 660 °R

4. Temp. of stator surface = 95 °F

= 555 °R

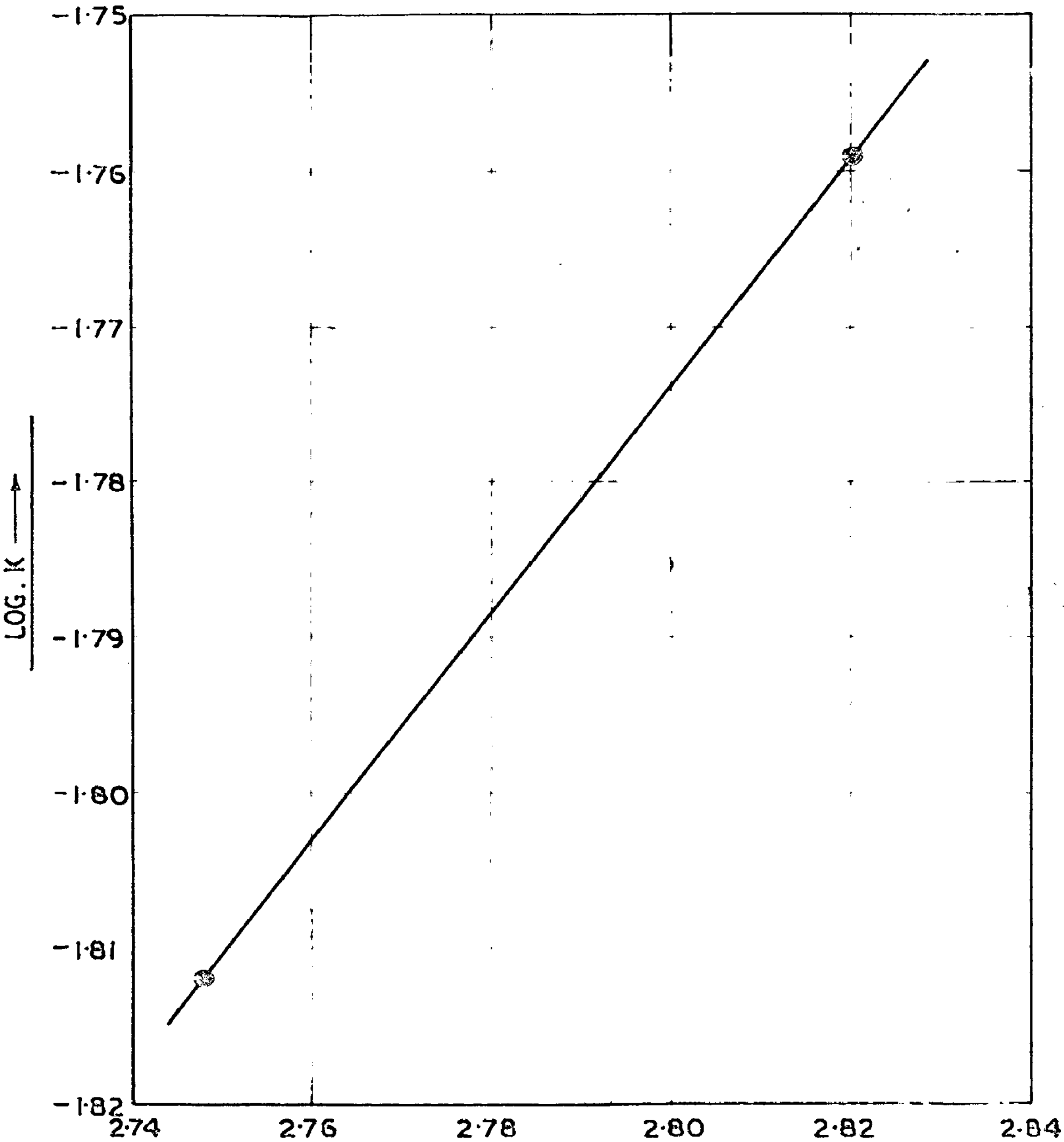

FIG. 10. A PLOT OF LOG_{10} (T) V/S LOG_{10} (K) FOR

5. Mean temp. t_m $= \frac{209 + 95}{2} = 304/2$

$= 152\,^\circ F$

6. Inlet water temp. t_1 $= 91\,^\circ F$

7. outlet water temp. t_2 $= 94.5\,^\circ F$ ... Column 7

8. quantity of water w $= 2.5 oz/min$ ... " 8

$= \frac{2.5 \times 60}{16}$

$= 9.375$ lbs/hr

Rate of total heat passing Q_t $= 9.375(94.5-91)$

$= 32.8$ Btu/hr.

Properties of air at mean temperature 152 F

$\nu = .2106$ sq.ft/sec. .... Column 15

K $= .0168$ Btu/hr.ft. °F .... Column 16

$P_r = .72$

For computing the values of conductivity of air K the graph was plotted between log T V/S log K as suggested by Kreith(14), as shown in Fig. 10

$$T_a = \frac{Vb}{\nu}\sqrt{\frac{b}{R_r}}$$

$$= \frac{2.83 \times .01315}{.2106 \times 10^{-3}}$$

$$= 176.5$$ Column 18

$$T_a^2 = 3.12 \times 10^4$$ Column 19

$$Q_{rad} = .173 \times F \times \epsilon \times A \left[\left(\frac{T_r}{100}\right)^4 - \left(\frac{T_s}{100}\right)^4\right]$$

$$= .173 \times 1 \times .787 \times .108 \left[\left(\frac{669}{100}\right)^4 - \left(\frac{555}{100}\right)^4\right]$$

$$= .0145\ [2000-950]$$

$$= .0145 \times 1050$$

$= 15.3$ Btu/hr. .... Column 11

$$Q_c = Q_1 - Q_{rad}$$

$= 32.8 - 15.3$

$= 17.5$ Btu/hr. ..... Column 12

$$U = \frac{Q_c}{A\,\Delta t}$$

$$= \frac{17.5}{.108(209-96)}$$

$$= \frac{17.5}{.108 \times 114}$$

$= 1.415$

$$N_u = \frac{Uh}{K}$$ Column 13

$$= \frac{1.415}{35 x .0168}$$

$= 2.42$ Column 17

Due to the repeated calculations the following programming was made to calculate all the results by the Digital IBM computor, the facility of which was available at the Centeral Building Research Institute, Roorkee.

PROGRAMME FOR CALCULATIONS ON ELECTRONIC DIGITAL IBM COMPUTER

```
CC  HEAT  TRANSFER  PROBLEM    C.P.SHARMA

100  FORMAT ( 6F 10.6)
101  FORMAT (4E 16.8)
  1. READ  100, V,RMU,T1,T2 ,CON, Q

          SIGMA   =  .1713
           EPS    =  .787
            A     =  .108
          DELTA   =  .02864
            TA    =  V * .01315/RMU
           TAI    =  TA * TA
            QR    =  .01455 *( (T1/100) **4-(T2/100) **4)
            QC    =  Q-QR
            U     =  QC / (A * (T1-T2))
           SNU    =  U * DELTA/COND

PUNCH  101,TA,TAI,QR,QC,U,SNU,
          GO TO 1
```

-: END :-

APPENDIX C

LIST OF INSTRUMENTS USED

1.	Speed Variator :	(1)	General Electric Triclad Motors Model 5K 180M ,H.P.1.5,phase,3, Cycles 50 1430 r.p.m.Induction Motor Made in Fort Wayne,Indians, U.S.A.
		(2)	Clevland Speed Variator, H.P. 1.5, input speed 1350, out speed at constant H.P. 485-4000 r.p.m.
2.	Reduction Gear :		Seco Reduction Gears, Made in India, Ratio 1/30, H.P. 0.75
3.	Temperature Recorder :		Speedomax H, Azar Recorder, manufactured by M/s Leeds and Northrup Company, Philadalphia U.S.A.
4.	Tachometer :		Smiths hand Tachometer, multirange 0-10000 r.p.m. made in England.
5.	Vocumtube Volt-Ohmmeter :		Simpson Model 311 made in U.S.A. by Sempson Electric Co., Chicago
6.	Ammeters :		0-5 A, Kaycee,Moving Coil type
7.	Rheostats :		OSAW,India, 45 ohms 5 Amps.
8.	Autotransformer		General Radio Company concord man U.S.A. type W 20 F Variac.
9.	Power transformer		Type 1 PHDW,input volts 230, out put volts, 5,10,15,20,55,110 manufactured by Automatic Electric Private Limited,Bombay-31
10.	Thermosflask		Eagle ,Capacity 2 lbs.

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THE END