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Effect of boundary layer control, through spiral cuts, on the film coefficient of convective heat transfer

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EFFECT OF BOUNDARY LAYER CONTROL, THROUGH SPIRAL CUTS, ON THE FILM COEFFICIENT OF CONVECTIVE HEAT TRANSFER

BY

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THESIS

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ABSTRACT

This thesis is experimental investigation into the possibility of obtaining an increase in the forced convection heat transfer coefficient for a pipe by passing the fluid through a narrow spiral cut in the pipe.

Different spiral pipes with pitches of two, four, eight, ten, sixteen, and twenty inches were used; however, the width of the spiral cut remained unchanged at 3/16 of an inch. Several mass flow rates for water were used up to a maximum of 1536 lbs/hr for each run.

One of the objectives was to find which pitch would give the highest heat transfer rate for a given flow rate. The second was to find which spiral pipe would give the highest consistent values of heat transfer for the over-all range of mass flow.

It was found that a pipe with a pitch of eight inches had the highest rate of heat transfer for all cases. It will be interesting to note that there is an increase in the heat transfer rate of 77 percent over a plain pipe at the lowest mass flow rate, 798 lbs/hr, and at the highest flow rate, 1536 lbs/hr, there was a 38 percent increase in the heat transfer rate. All of the spiral pipes tested show an increase in the heat transfer coefficient compared with a plain pipe.

INTRODUCTION

Since 1904 when Prandtl (1) first proved his theory on boundary layers at the Mathematical Congress in Heidelberg, many effective ideas have been developed and are still being developed to increase the heat transfer rate in fluid flow. In this day and time the need is even greater. "A heat release of about 40,000 Btu/hr cu ft is considered good practice in modern boilers, but in a rocket or nuclear reactor it may be 1,000,000 Btu/hr cu ft." (7)

Today engineers know how to convert atomic energy to electricity, they know that there are important heating jobs which can be done with atomic energy, and they are aware of many other possible applications. But in each of these cases, this energy is competing with some other form of energy, which is initially well under the cost involved in atomic power.

The question arises how to get this high rate of heat out of the nuclear reactor at a lower cost. The answer to this question is found in three factors which make up the optimum design of a reactor: "(a) a high differential temperature is desirable across the system in order to obtain a high heat transfer rate; (b) a high differential temperature is desirable across the primary coolant-system elements; (c) a large differential temperature across the heat exchanger keeps its size and cost to a minimum." (12)

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(1) All references are in Bibliography

In all three cases if the heat transfer coefficient could be increased, the reactor would have a higher thermal transfer; thereby reducing the cost of operation. Since the greatest barrier to the flow of heat from a solid to liquid exists at the inter-face, it can be expected that any device which will alter the inter-face pattern will also alter the heat transfer coefficient.

The purpose of this thesis is to study a control boundary layer method and one which has some practical use in a heat exchanger. The method selected was to pass a fluid through a narrow spiral cut in a pipe and then to vary the pitch of the spiral to find which one has the best heat transfer rate. The effects of this method will be compared with one having no control method.

REVIEW OF LITERATURE

The study of the rate of convective heat transfer in pipes has been the subject of many investigators. Kreith (11), Kay (10), Giedt (9), and Eckert and Drake (8) have discussed in their books the involved problems in convective heat transfer. One of those problems is the boundary layer. The concept of a boundary layer was introduced by a German scientist, Prandtl (1), in 1904. At this time he proved "that the flow about a solid body can be divided into two regions or domains: a very thin layer in the neighborhood of the body (boundary layer) where the friction plays an essential part, and the remaining region outside this layer, where friction may be neglected."

One of the findings of Kreith (2) was "that when the boundary layer becomes unstable and the transition from laminar to turbulent flow begins eddies and vortexes form and destroy the laminar regularity of the boundary-layer motion. Quasi-laminar motion persists only in a thin layer in the immediate vicinity of the surface." He later stated that this portion of a generally turbulent boundary layer is called the laminar sublayer. The region between the laminar sublayer and the completely turbulent portion of the boundary layer is called the buffer layer. The structure of the flow in a turbulent boundary layer is shown here schematically in figure No. 1.

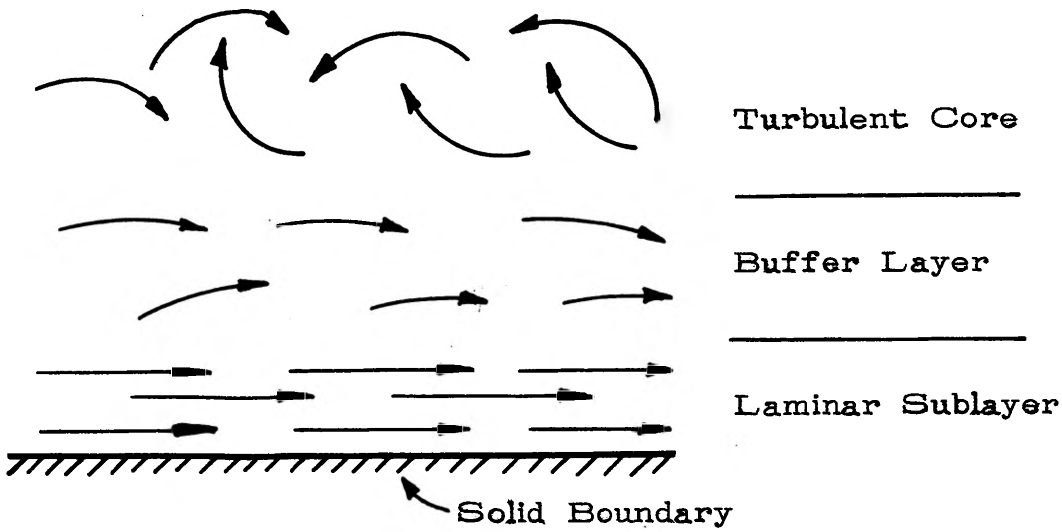


Figure No. 1

Structure of a turbulent flow field

Schlichting (3) states "several methods of controlling the boundary layer have been developed experimentally and theoretically. These can be classified as follows:

1. Motion of the solid wall
2. Acceleration of the boundary layer
3. Suction
4. Prevention of transition to turbulent flow by the provision of suitable shapes (laminar profiles)"

The last two have the greatest practical importance.

A Russian scientist, Fedynshii (4) lately has been working on the control of boundary layers and he thinks that reduction of the boundary layer depends on the shape of the body and on the distribution of temperature. Later in his discussion he says "there is a shortage of data and what there is cannot always be mutually compared."

Nevertheless if we are to move large quantities of heat rapidly, the boundary layer thickness must be reduced as much as possible. Some work has been done to reduce the thickness in pipes.

In 1960 Prasad (5) studied the effect of boundary layer control through surface holes on the film coefficient of convective heat transfer from a pipe for turbulent flow. He used 3/4 IPS pipe, 20 inches long with and without boundary layer control. This control was obtained by drilling 1/16 inch diameter holes on the pipe surface. His results show that using this method gave him a higher value of convective heat transfer from 25 percent to 140 percent compared with the water flow on both the outside and inside of the pipe surface simultaneously, depending upon the rate of flow and the number of holes. The percentage increase is greater at low rates of flow.

In the same year, 1960, George (6) studied the effect of the boundary layer control by cutting 1/16 inch wide rectangular slots on the pipe surface along the circumference. He used the same length same IPS pipe as Prasad. The length of each slot was equal to half the total circumference. He showed that the convective heat transfer coefficient was increased from 100 percent to 200 percent compared with no slots on the pipe surface and water flowing inside and outside of the pipe simultaneously, depending on the arrangement and spacing of the slots. The maximum increase of heat transfer can be obtained, by arranging the slots in a longitudinal spiral. He also noticed at lower rates of water flow, the effect of this control by slots is greater and gave a higher percentage increase of the convective heat transfer coefficient.

The author in his investigation used basically the same procedure as Prasad and George, differing only in the method of controlling the boundary layer. The method used by the author was to cut a 3/16 inch spiral in a pipe and to vary the pitch of the spiral.

DISCUSSION

FACTORS THAT EFFECT THE FILM COEFFICIENT

Many factors effect the film coefficient, h, the most important of these being the mechanism of flow, the fluid properties, geometry and velocity. These factors effect the film coefficient mainly in controlling the boundary layer at the surface of separation between the solid and fluid. The greatest resistance to heat flow occurs at the boundary layer.

The boundary layer consists actually of two layers. The first composed of particles completely without motion adhering to the surface and particles creeping along in stream-line flow with increasing velocity as the distance from the surface is increased.

The second layer being a buffer layer or transition zone composed of eddy currents moving at a higher velocity although not so swiftly as the main portion of the fluid stream also it is much thicker than the first. The boundary between the two is difficult to distinguish. We do know the boundary layer is very hard to measure and generally runs several hundredths of an inch in thickness.

The film coefficient is a hindrance to the heat flow. It adheres to the surface so closely that the first layer, laminar sublayer, transfers heat by pure conduction. A reduction in the thickness of the laminar sublayer will greatly increase the the heat transfer, also a reduction in the transition zone thickness will increase the heat transfer.

"When a fluid is allowed to enter a circular pipe from a large container, the velocity distribution in the cross-sections of the inlet length varies with the distance from the initial cross-section. In sections close to that at entrance the velocity distribution is nearly uniform. Further downstream the velocity distribution changes, owing to the influence of friction, until a fully developed velocity profile is attained at a given cross-section and remains constant downstream of it." (14)

The boundary layer thickness builds up from zero at the entrance to a maximum thickness at the section where the velocity profile is fully developed; therefore, if we cut a spiral in a pipe we would stop the build-up action of the boundary layer because every time the fluid crosses the spiral a new velocity profile is started. This is one reason why we can expect an increase in the heat transfer.

"It was observed that inside surface heat transfer coefficients in swirling flow increase as much as four-fold over the coefficients observed at the same velocity in purely axial flow. The heat transfer coefficients were found to depend on the centrifugal force component. The effect of centrifugal force can be explained in the following manner. The phenomena involved here are similar to those in free convection, but the body force is proportional to the V^2 instead of g. The effect of the body force in rotating flow is to simultaneously force heavier fluid particles outward and lighter ones inward. One consequently expects this mechanism to aid convection when the direction of heat flow is larger to smaller radii, i.e. when the fluid in the tube is heated, but to oppose the exchange mechanism when the direction of flow is reversed." (15) Here, again, we find another justification for expecting an increase in the heat transfer coefficient when using a spiral cut in a pipe because the fluid entering the pipe from outside to inside can be expected to have a spiral component of motion which should increase the heat transfer rate.

By increasing the pitch of the spiral, a breakdown of the building action in the boundary layer occurs and thereby produces an increase in the heat transfer rate. Also a change can be expected in the direction of the fluid flow at the spiral cut and the mixing with the fluid flowing axially inside the tube results in an increase of turbulence both inside and outside the pipe which tends to break down the boundary layer. This, naturally, would produce an increase in the heat transfer rate.

We can see from the discussion thus far that there should be a particular pitch where all the factors involved would give a maximum heat transfer rate for a given mass flow rate.

TEST EQUIPMENT

The apparatus constructed and used by the author was designed to permit determination of the heat transfer film coefficient. Two photographs, which show general views of the apparatus in the Mechanical Engineering Laboratory, are shown in Figures 4 and 5.

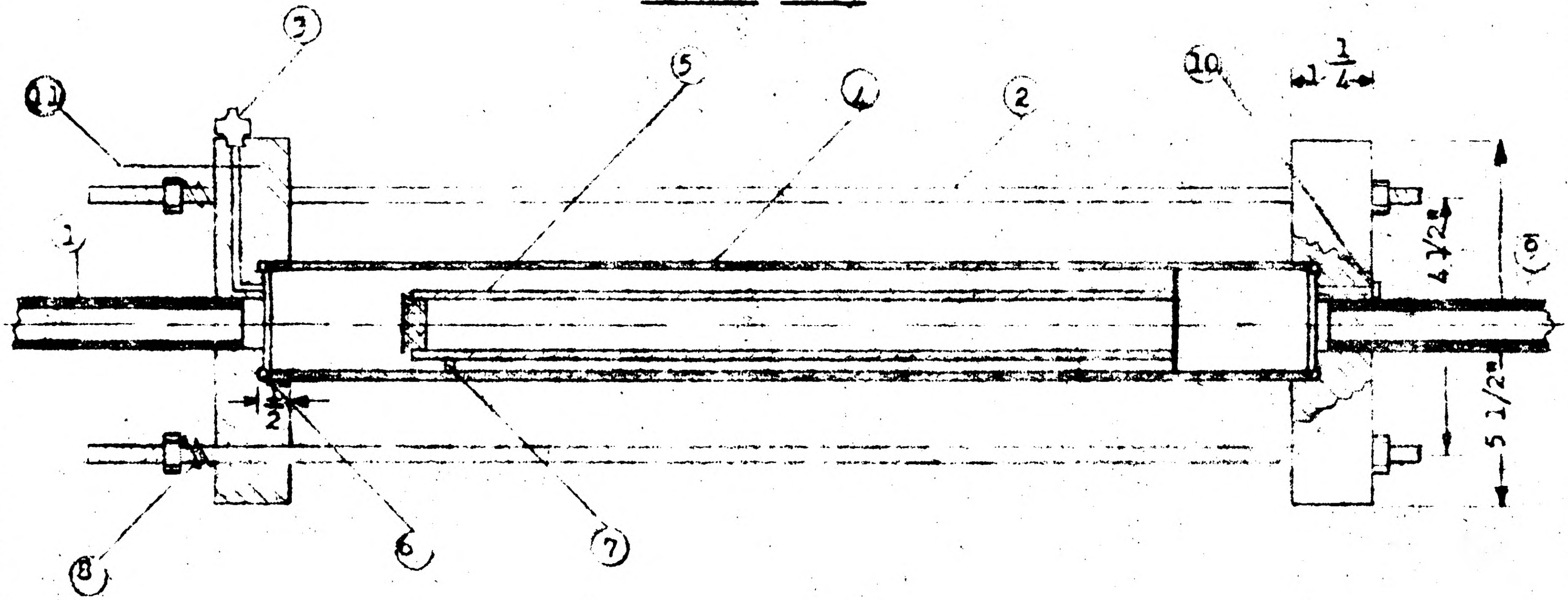
TEST APPARATUS

Details of the apparatus is shown in Figure 2. The test section used was a steel pipe (20 inches long and 3/4 inch IPS) with a 3/16 inch spiral cut in it. A typical pipe can be seen in Figure 3. The pipe was supported centrally in a glass tube (28 inches long, 1 7/8 inches I.D., and 2 1/4 inches O.D.) by 1/8 inch aluminum supporting legs and to prevent the test section from moving in the direction of the motion of the water, a plastic pipe (1 inch I.D., 1 1/4 inch O.D., and 5 inches long) was put inside the glass tube behind the test section.

The glass tube was kept water tight by using two aluminum flanges which were tightened by four stay rods (3/8 inch in diameter and 36 inches long). To insure that the glass tube was water tight, the ends were sealed with "O" rings.

In the inlet aluminum flange, an air valve was installed to drain away any air collected inside the tube. In order to force all the water through the spiral, a rubber seal was placed at the downstream end and on the outside of the test section.

FIGURE NO. 2



(1) 1/2" Water Inlet Pipe
(2) Stay Rods (3/8" x 36")
(3) Drain Cock For Air
(4) Glass Tube 30" Long
(5) Test Section
(6) O-Ring

(7) Support For Test Section
(8) Springs On Stay Rods
(9) 1/2" Water Outlet Pipe
(10) Plug For Thermocouple Wires
(11) Aluminum Flange

NOTE: Dimensions As Shown, Not To Scale

SECTIONAL ELEVATION OF APPARATUS

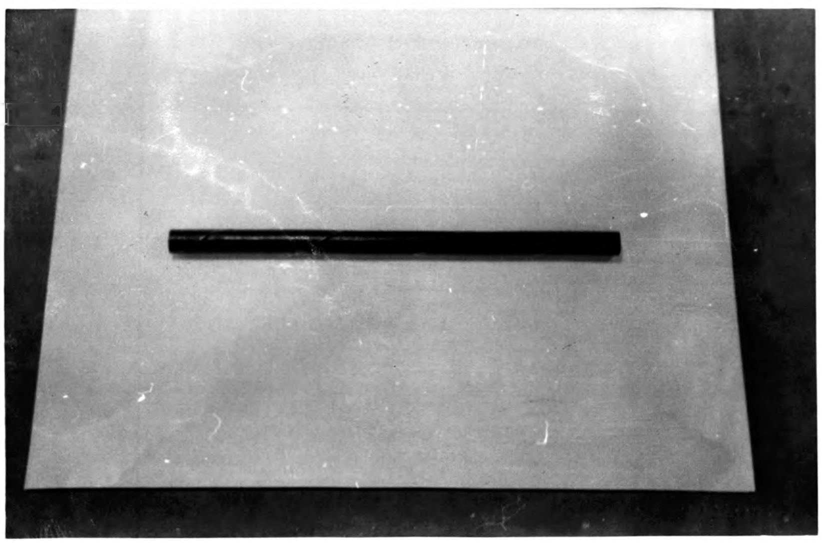


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FIGURE 3

TYPICAL TEST SECTION

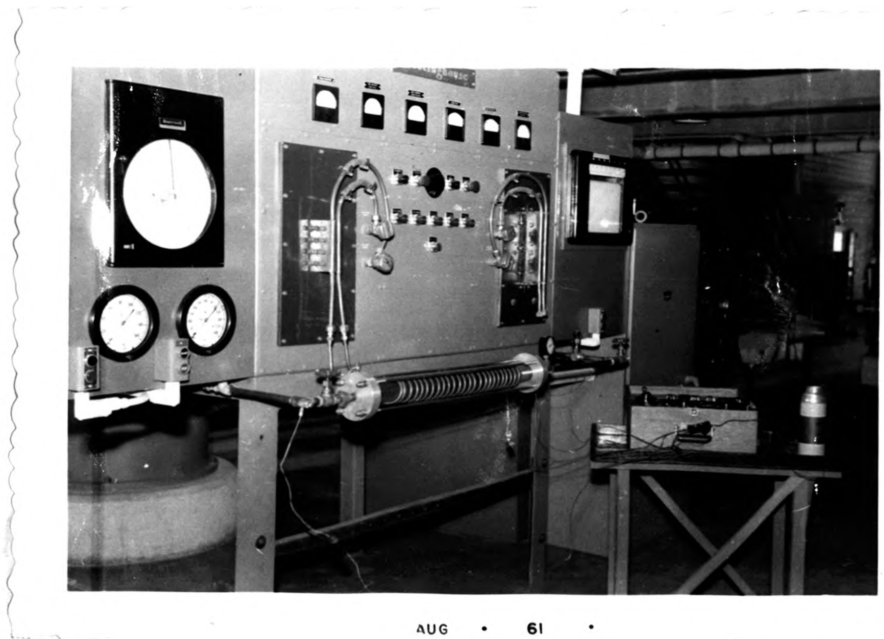

FIGURE 4

EXPERIMENTAL APPARATUS

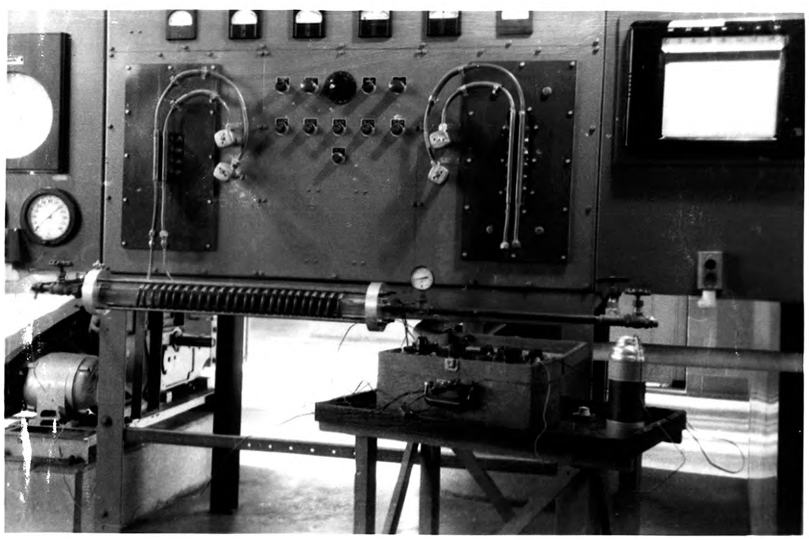

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FIGURE 5

EXPERIMENTAL APPARATUS

PROCESS WATER

The water used was supplied from the Missouri School of Mines power plant and had been processed by a lime-soda ash hot process water softener and was condensate returning from the heating system. The process water had a PH of 7.5 to 8.0. A pump was installed in the supply line to give the necessary head to circulate the water through the apparatus. A general flow diagram is illustrated in Figure 6.

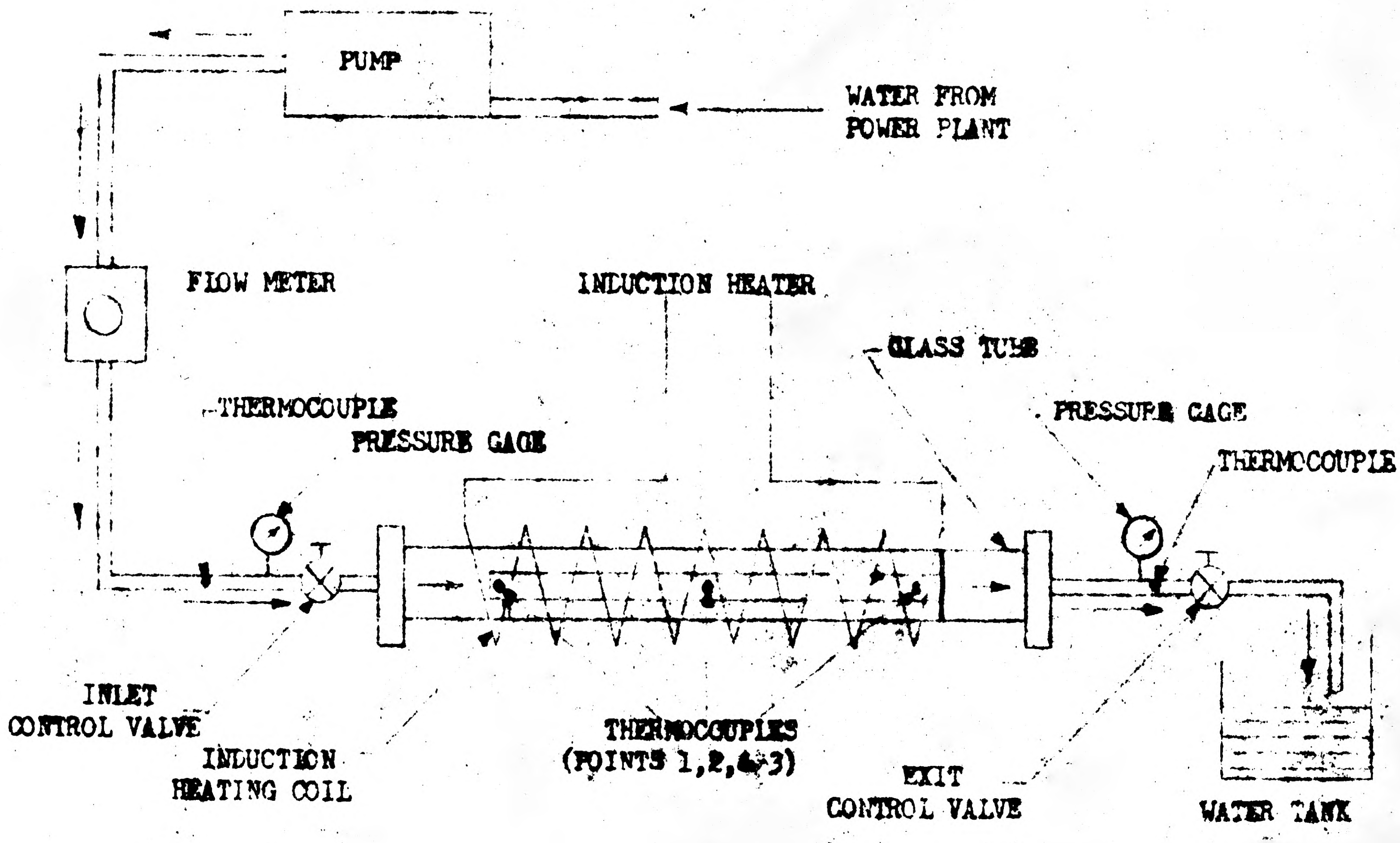
INDUCTION HEATER

A Westinghouse 30 Kilowatt multipurpose induction heater was used to heat the test section. It is a motor-generator type heater with a 9600 cps output at voltages up to 800 volts. The induction coil was made from 3/8 inch water-cooled copper tubing and was 3 inches in diameter, 24 inches long, and consisted of 25 turns (See figure 7 for a picture of the induction coil). The motor-generator set has a water temperature and pressure protective device which was connected to the motor starter so that the power to the machine will be cut off if it overheats.

TEMPERATURE MEASUREMENT

The water temperature was measured at the inlet and outlet of the test section by iron-constant thermocouples and the thermocouples were inserted into the inlet and exit pipes. Figure 8 shows which

FIGURE NO. [illegible]



FLOW DIAGRAM

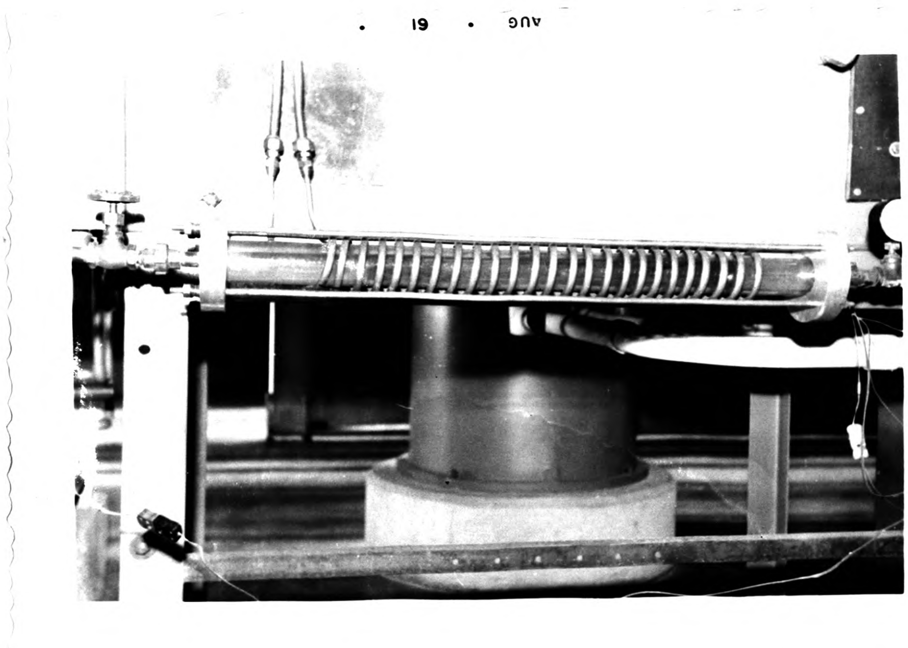


FIGURE 7

INDUCTION COIL AND TEST SECTION

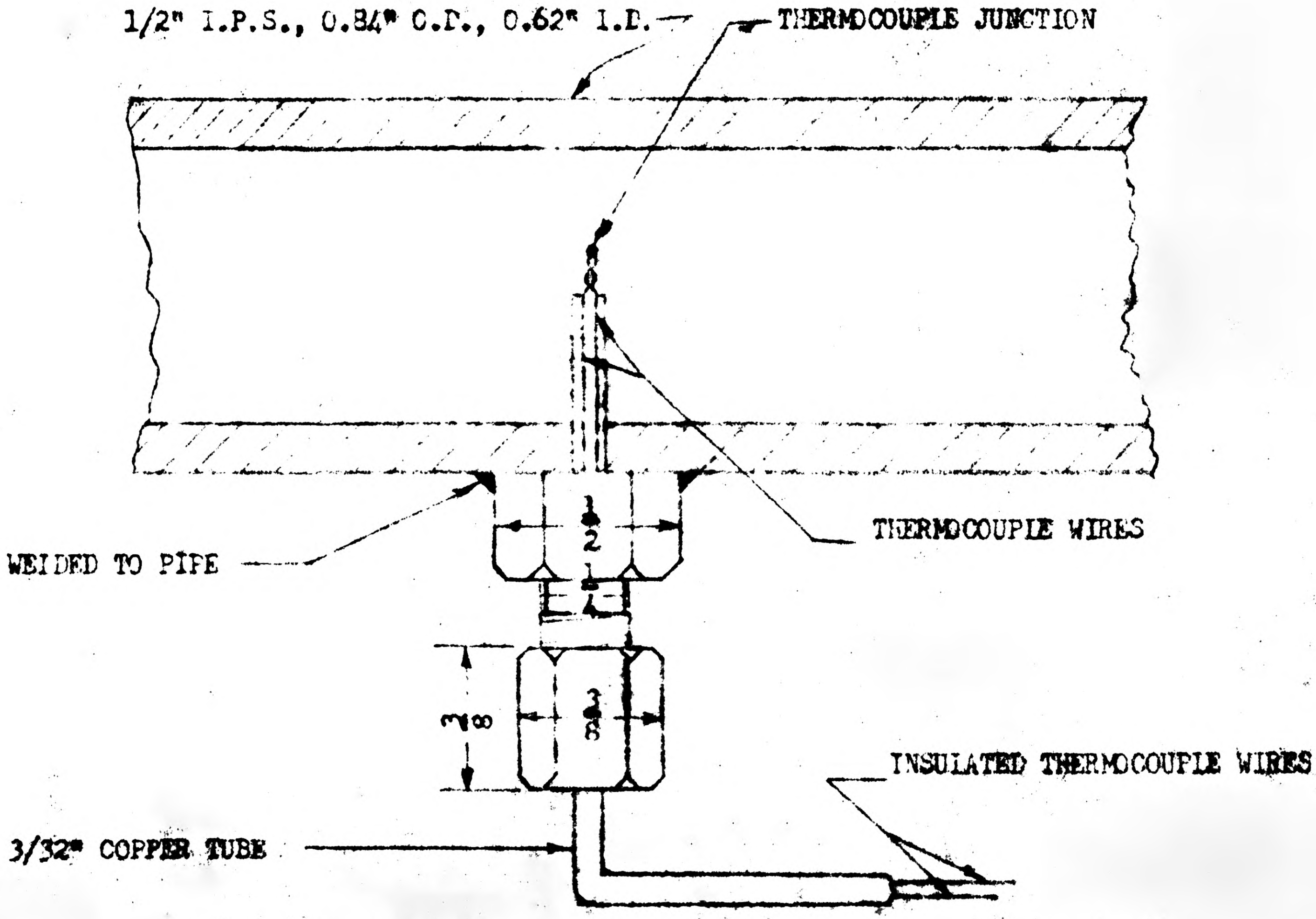
type of thermocouple was used. The measurement of the surface temperature of the test section posed an extremely complex problem due to the presence of the induction field. Thermocouple wires when placed in the induction field tend to heat up independently of the heating of the test section. If the wires attain a temperature greater than the temperature of the test section, the temperature recorded will be in error, but if the test section reaches a higher temperature than the adjacent thermocouple wires, the thermocouple output should be a true indication of the temperature provided that any stray currents which might effect the reading of the potentimeter are eliminated. In this investigation the thermocouples placed in the induction field showed that the error introduced is very small.

All the outputs of the thermocouples were measured with a portable precision potentiometer (Figure 9 shows the potentiometer). Thermocouples were soldered to the ends and the middle of the pipe and their wires were taken out through the plugs provided in the outlet flange. The thermocouples were then tested in an ice bath to insure that they were measuring correctly.

FLOW MEASUREMENT

To control the mass flow of water two control valves were fitted in the pipe line, one on each side of the test

FIGURE NO. 8



THERMOCOUPLE INSTALLATION TO MEASURE WATER TEMPERATURE IN PIPE section. A "differential pressure" type flow meter was used to measure the flow of water circulating through the apparatus. The instrument was made by Minneapolis-Honeywell Regulator Company. The flow was automatically recorded on a chart and was calibrated under actual testing conditions (Table 8 and Figure 11 in the appendix).

PRESSURE MEASUREMENT

To measure the pressure of the water entering and leaving the test section, two Bourdon gauges were used. One Bourdon gauge, a tube type of a range 0-600 psi, was connected in the pipe line ahead of the inlet throttle valve to measure the water entering test section. The other gauge with a range 0-60 psi was fitted between the test section and the exit control valve. See Figure 6 for their position.

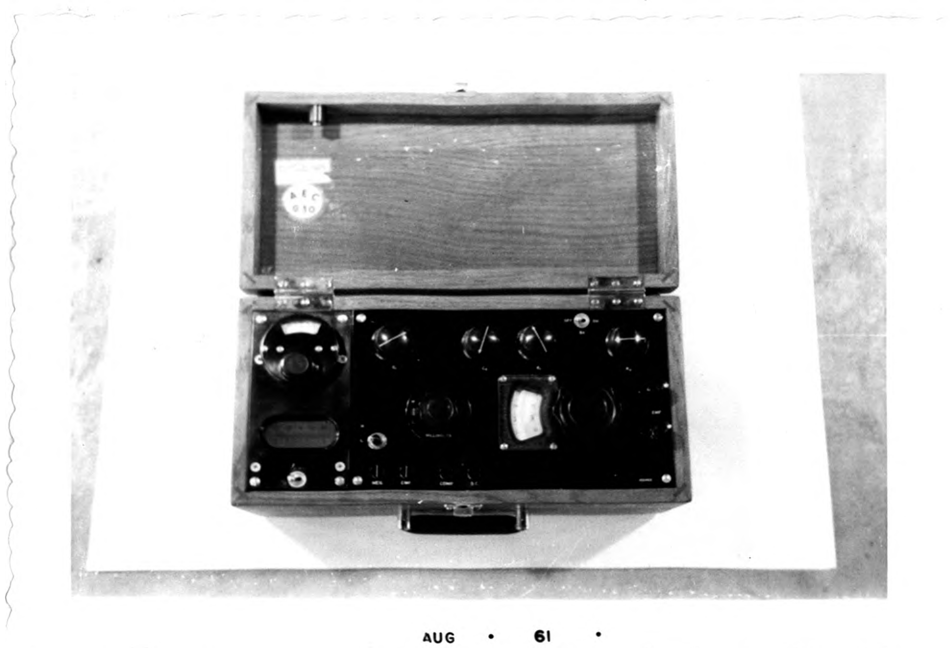


FIGURE 9

PORTABLE POTENTIOMETER

TEST PROCEDURE

The experiment was carried out in the following manner:

1. After brazing the thermocouple wires to the test section, the test section was placed centrally inside the glass tube. To allow for the expansion of the supporting aluminum legs and prevent the breaking of the glass tube, the legs were made to fit loosely inside the tube. The thermocouple wires were taken out through plugged holes in the outlet flange and connected to a portable thermocouple potentiometer. The stay bolts were tightened to make the glass tube water tight.
2. The potentiometer's galvanometer and current were adjusted to zero. Using a reference junction of 32° F, ice bath, the compensator was adjusted to zero.
3. The flowmeter switch was turned to the "on" position.
4. The inlet throttle valve was closed, the process water pump was started. The flow rate was then adjusted to the 200 mark on the indicator chart by opening and adjusting the inlet throttle valve. After the flow was established, any air collected inside the glass tube was removed by closing the outlet throttle valve and opening the air outlet valve. It was kept fully open so that the pressure inside the glass tube was constant and below atmospheric. This prevented sudden pressure fluctuation inside the glass pipe when flow rate was changed by the control valve.

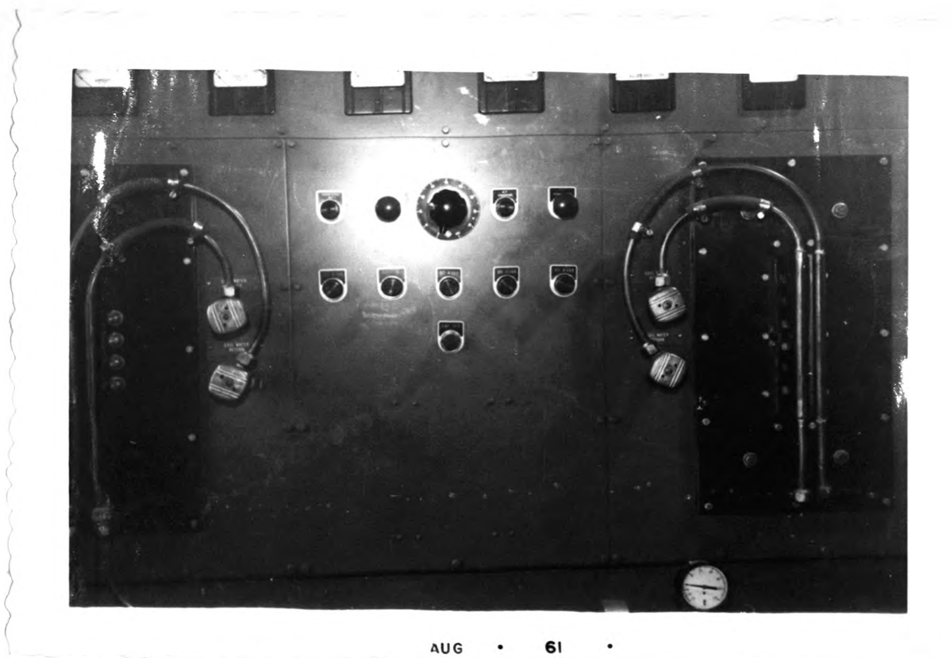


FIGURE 10

INDUCTION HEATER CONTROL PANEL

5. The cooling water pump for the induction heater coil was started.

6. When the pressure of the cooling water for the induction heater reached 60 psi, the motor-generator set was started. About 90 seconds was allowed for it to come to a steady speed.

7. The powerstat pointer was kept in the zero position and the excitation, high frequency power, and 30 KVAR switches on the panel board was turned on.

8. The powerstat was slowly turned to apply the load. The maximum load was limited by the capacity of the induction equipment and also by the maximum temperature in the test section, which should be less than 200° F. to prevent any local boiling from taking place.

9. The flow meter pointer was adjusted carefully to the 200 and about 30 minutes was allowed in order to attain steady state conditions.

10. The potentiometer current was rechecked and adjusted when necessary. The thermocouple was connected to the "EMF" binding posts and the slide wire was adjusted until the galvanometer read zero. The emf and temperature was recorded.

11. Any air collected in the glass tube was removed after turning the powerstat to the zero position.

12. The flow was changed to the 175 mark on the flow meter recorder. The powerstat was adjusted, if necessary, to keep the maximum temperature of the test section below 200° F. Ten minutes were allowed to reach steady state conditions, then the second set of emf's were read and the corresponding temperatures recorded.

13. The above procedure was repeated for flow readings of 150, 125, 100, 75, and 50 on the flow meter.

14. After taking the seven different flow rate readings the induction heater was turned off as were also the flow meter, cooling water pump, and the process water pump. The inlet throttle valve was then closed.

From the temperatures and the flow rates the heat transfer coefficient was calculated for each flow rate by the Missouri School of Mine's electronic computer, Royal McBee Corporation LGP-30. (A copy of this program and a sample run of the computer can be found in the Appendix, fig. 12 and fig. 13).

The experiment was repeated as a check to see that the data could be duplicated from day to day. If the second set of values were not comparable with the first, the reason or reasons were determined why the second experiment didn't compare. After the investigation, the experiment was repeated until two sets of readings could be duplicated.

The above experiment was performed by varying the pitch of a spiral cut in a 20 inch long 3/4 inch IPS steel pipe. Six different pitches (2, 4, 8, 10, 16, and 20 inches) were used.

The lower limit of mass flow was fixed at about 800 lbs/hr to insure the condition of turbulent flow in the test section as determined by Reynolds Number. The upper limit was also fixed at a value of about 1540 lbs/hr to give a reasonable value for the temperature rise of the water. The temperature rise should be 10 degrees or higher before readings are made in order to give accurate data for the determination of the film coefficient (h). If it were lower than 10 degrees, the pipe temperature was increased until there was a temperature rise of 10 degrees or more in the water.

CALCULATION

ASSUMPTIONS USED IN THE CALCULATION

1. The properties of water were assumed to be constant in the range of temperature employed in the experiment.
2. The temperature of the test section was assumed to be uniform over its thickness.
3. The temperature variation from the ends to the middle of the test section was assumed to be linear.
4. The effect of the induction field on the thermocouple wires was neglected.
5. Steady state conditions.
6. Bulk temperature was considered to be thoroughly mixed at any cross section.
7. The average bulk temperature within and around the pipe was taken as the mean of the inlet and outlet water temperature.
8. Fluid flow is negligible.
9. The heat transfer coefficient was assumed to be uniform over the surface of the test section.
10. Heat transfer from two end thicknesses of the pipe was neglected.

11. Heat transfer from 1/8 inch diameter aluminum center rods fixed at the test-section ends was not taken into account.

12. The specific heat of water was taken as constant within this temperature range equal to 1 Btu per lb.° F.

13. The change of heat transfer area due to 3/16 inch spiral cut through the pipe was taken into account.

14. The temperature drop at the inlet throttle valve was neglected.

15. The heat loss to the surroundings was assumed to be very small and in the calculations was neglected.

16. The heat transfer area lost at the base of the rubber plug used to close the annulus was neglected.

EQUATION USED IN THE CALCULATION

The flow of heat through the fluid solid inter-face may be expressed as:

$$q = h A (T - t)$$

where q= rate of heat flow, Btu/hr

h= film coefficient for convective heat transfer, Btu/(hr) (sq ft)(°F)

A = area of the surface, sq ft
$T - t$ = difference in temperature across the film, F.

The mean surface temperature of the test section is given by:

$$T = \frac{t_a + 2\,t_b + t_c}{4},$$

where t_a = temperature at point 1,
t_b = temperature at point 2,
t_c = temperature at point 3.

Subscripts 1 and 3 are the temperatures at the end of the test section and point 2 is the temperature at the center of the test section.

The bulk temperature is given by:

$$t = \frac{t_1 + t_2}{2},$$

where t_1 = inlet temperature of water,
t_2 = outlet temperature of water.

Heat transfer to water in unit time is given by:

$$q = W\,c_p\,(t_2 - t_1)$$

where q = rate of heat flow, Btu/hr.

W = mass flow rate of water, lbs/hr.

c_p = specific heat of water, Btu/(lb)(°F).

$t_2 - t_1$ = temperature rise of water, °F.

Equating the heat removed from the pipe to the heat gained by the water, we find that:

$$h = \frac{W\ (t_2 - t_1)}{A\ (T - t)}\ .$$

The value of the coefficient of heat transfer h, was calculated from the above equation. A graph was drawn for each test section taking coefficient of heat transfer as the ordinate and the mass flow of water as the abscissa.

ACCURACY OF CALCULATION

The accuracy of the calculation depends on the accuracy of the data. There are several factors which limited the measurement of the heat transfer coefficient:

(1) Error induced by iron-constantan thermo-couples

(2) The assumption of linear variation of the pipe temperature

(3) Error induced by the potentiometer

(4) Calibration accuracy of the flowmeter

(5) The assumption of no heat loss to the surroundings.

The error in the thermal electromotive forces of the iron-constantan thermocouples is about $\pm$ 1°F and the accuracy of the portable potentiometer is $\pm$ 0.25°F. As the water temperature rise was quite low because of the limitation in the maximum surface temperature, the error of $\pm$ 1.25°F in water temperature may introduce an error of 5 to 9 percent.

Since the other factors have such a small percent of the error, it is safe to say that the accuracy of the convective heat transfer coefficient calculated was $\pm$ 10 percent. The object of this investigation was mainly a qualitative study of the heat transfer coefficient in the different test sections rather than a precise quantitative study, due to the limitations imposed by the apparatus and the measuring devices.

DISCUSSION OF RESULTS

The results of this investigation on the different test sections are given in tables 1 to 6 and plates 1 to 6.

Prasad (5) and George (6) investigated convective heat transfer coefficient for a plain pipe. The author used the same procedure and equipment in this investigation as Prasad and George, differing only in the test section; however, in using the plain pipe the water was allowed to flow both inside and outside the pipe. Their results are found in table 7 and plate 7. We can see that the values of h plotted against W on the logarithmic graph gave a straight line, because the flow throughout was turbulent and the bulk temperature of water was kept almost constant in all runs. It should be noted that when the flow rate was doubled the convective heat transfer coefficient was more than doubled; therefore, higher flow rates are advisable with plain pipes for increased heat transfer per pound of water used.

Since a plain pipe has no boundary control method, we will use this as a comparison from time to time to see the effect of using the spiral cut as a boundary control method.

TABLE NO. 1

HEAT TRANSFER EXPERIMENT DATA

TEST SECTION: Steel pipe 0.825", I.D.; 1.05", O.D.; 20" long; Area = 0.8356 sq. ft. one end closed, 3/16" spiral cut in the pipe with a pitch of 2; Plate No. 1.

RUN NO.	FLOWMETER READING	W lbs/hr	WATER TEMP. °F T_1	T_2	SURFACE TEMP. °F POINT 1	POINT 2	POINT 3
1-a	50	798	135	147	152	185	148
2-a	75	969	135	147	153	187	149
3-a	100	1110	136	147	153	186	148
4-a	125	1230	137	148	154	187	149
5-a	150	1359	137.5	148	154	187	149
6-a	175	1455	138	149	156	189	150
7-a	200	1536	138	149	156	188	149
1-b	50	798	137	149	155	186	150
2-b	75	969	137	148	154	185	150
3-b	100	1110	138	148.6	154	187	149
4-b	125	1230	139	149	154	185	150
5-b	150	1359	138	148	153	185	150
6-b	175	1455	138	148	155	184	149
7-b	200	1536	138	149	156	188	150

TABLE NO. 1

HEAT TRANSFER EXPERIMENT DATA (Continued)

RUN NO.	t °F	T °F	(T_2-T_1) °F	$-\Delta t$ °F	h Btu/hr ft^2°F
1-a	167.50	141.00	12.0	26.50	432.45
2-a	169.00	141.00	12.0	28.00	496.99
3-a	168.25	141.50	11.0	26.75	546.25
4-a	169.25	142.50	11.0	26.75	605.30
5-a	169.25	142.75	10.5	26.50	644.41
6-a	171.00	143.50	11.0	27.50	696.50
7-a	170.50	143.50	11.0	27.00	748.89
1-b	169.25	143.00	12.0	26.25	436.41
2-b	168.50	142.50	11.0	26.00	490.44
3-b	169.25	143.30	10.6	25.95	542.42
4-b	168.50	144.00	10.0	24.50	600.59
5-b	168.25	143.00	10.0	25.25	643.87
6-b	168.00	143.00	10.0	25.00	696.25
7-b	170.50	143.50	11.0	27.00	748.62

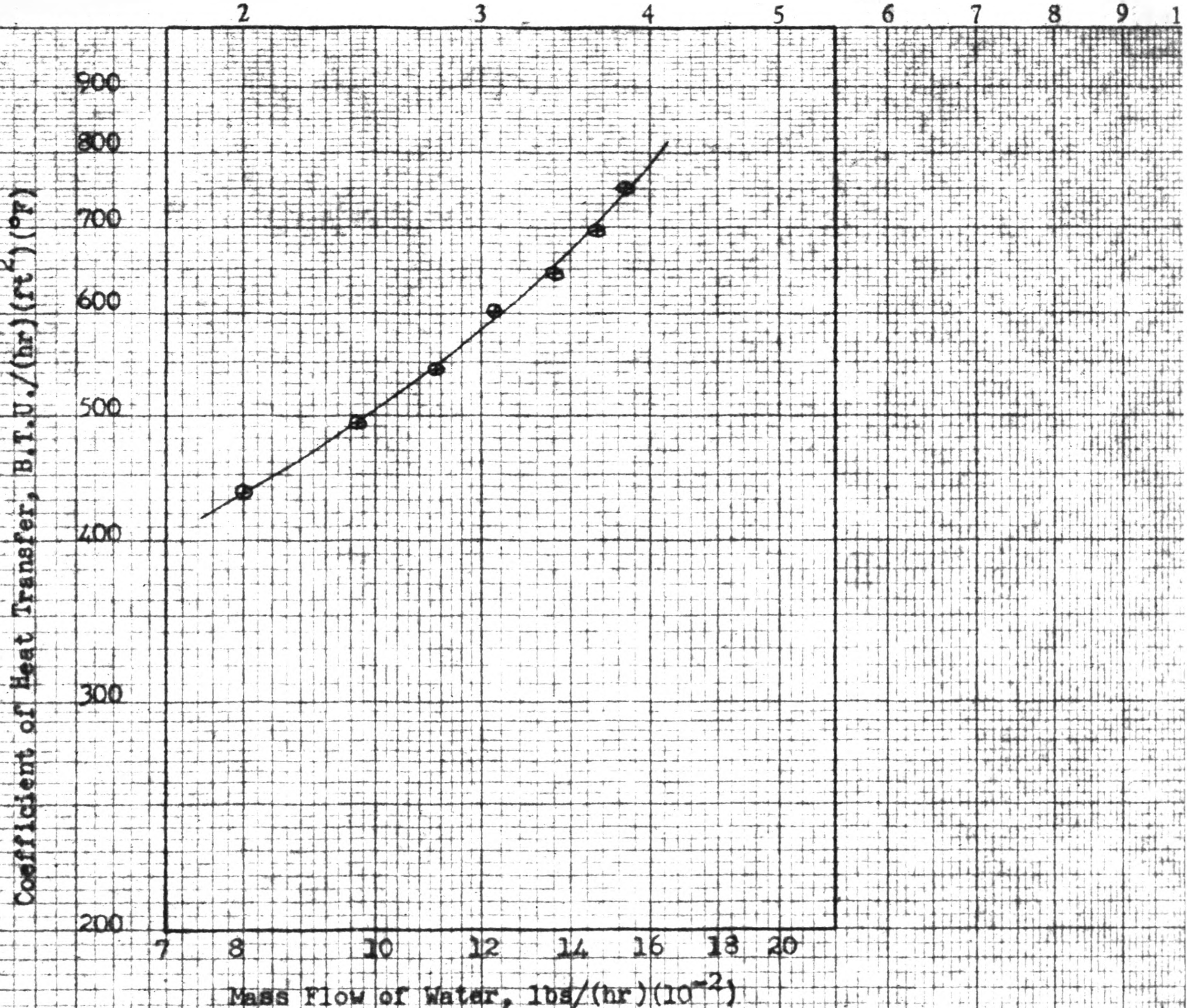


PLATE NO. 1

COEFFICIENT OF HEAT TRANSFER VS MASS FLOW FOR A PITCH OF 2

TABLE NO. 2

HEAT TRANSFER EXPERIMENT DATA

TEST SECTION: Steel pipe 0.825", I.D.; 1.05", O.D.; 20" long; Area = 0.8306 sq. ft. one end closed, 3/16" spiral cut in the pipe with a pitch of 4; Plate No. 2.

RUN NO.	FLOWMETER READING	W lbs/hr	WATER TEMP. °F T_1	T_2	SURFACE TEMP. °F POINT 1	POINT 2	POINT 3
1-a	50	798	136	147	152	179	148
2-a	75	969	136	147	153	181	149
3-a	100	1110	136	147	153	182	149
4-a	125	1230	137	147	151	180	149
5-a	150	1359	137	147	152	180	149
6-a	175	1455	137	147	152	180	149
7-a	200	1536	137	147	151	180	149
1-b	50	798	134	153	172	200	160
2-b	75	969	135	153	174	200	164
3-b	100	1110	135	153	174	200	165
4-b	125	1230	137	154	170	202	165
5-b	150	1359	136	154	171	205	167
6-b	175	1455	137	154	168	204	165
7-b	200	1536	137	154	169	202	165

TABLE NO. 2

HEAT TRANSFER EXPERIMENT DATA (Continued)

RUN NO.	t °F	T °F	(T_2-T_1) °F	$-\Delta t$ °F	h Btu/hr ft^2°F
1-a	164.50	141.50	11.0	23.00	459.48
2-a	166.00	141.50	11.0	24.50	523.79
3-a	166.50	141.50	11.0	25.00	588.00
4-a	165.00	142.00	10.0	23.00	643.85
5-a	165.25	142.00	10.0	23.25	703.72
6-a	165.25	142.00	10.0	23.25	753.43
7-a	165.00	142.00	10.0	23.00	804.02
1-b	183.00	143.50	19.0	39.50	462.13
2-b	184.50	144.00	18.0	40.50	518.50
3-b	184.75	144.00	18.0	40.75	590.30
4-b	184.75	145.50	17.0	39.25	641.39
5-b	187.00	145.00	18.0	42.00	701.21
6-b	185.25	145.50	17.0	39.75	749.17
7-b	184.50	145.50	17.0	39.00	806.09

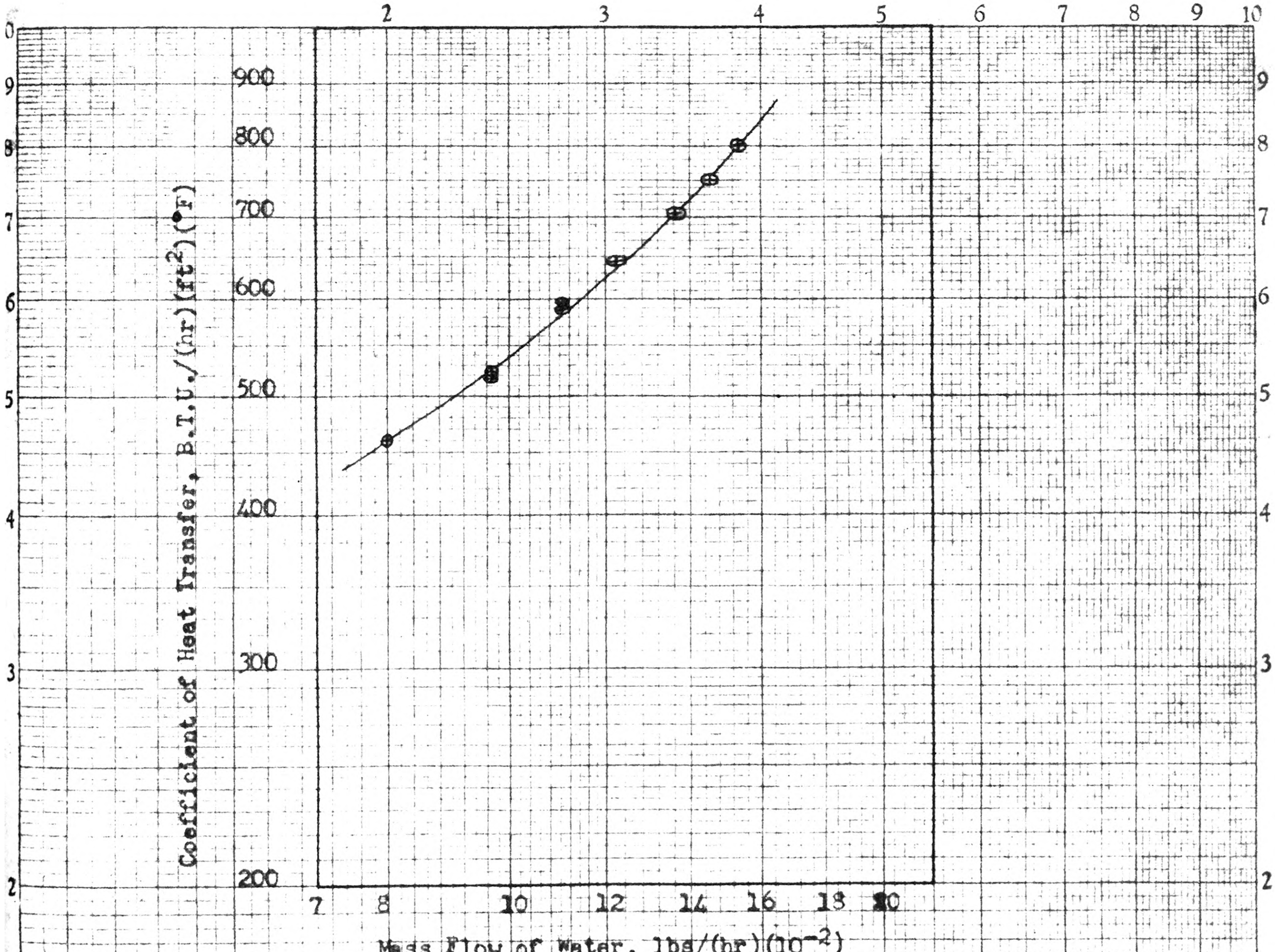


PLATE NO. 2

COEFFICIENT OF HEAT TRANSFER VS MASS FLOW FOR A PITCH OF 4

TABLE NO. 3

HEAT TRANSFER EXPERIMENT DATA

TEST SECTION: Steel pipe 0.825", I.D.; 1.05", O.D.; 20" long; Area = 0.829 sq. ft. one end closed, 3/16 spiral cut in the pipe with a pitch of 8; Plate No. 3.

RUN NO.	FLOWMETER READING	W lbs/hr	WATER TEMP. °F T_1	T_2	SURFACE TEMP. °F POINT 1	POINT 2	POINT 3
1-a	50	798	135	150	163	182	164
2-a	75	969	136	150	163	183	164
3-a	100	1110	137	150	162	182	162
4-a	125	1230	137	150	163	181	164
5-a	150	1359	138	150	163	179	163
6-a	175	1455	138	150	163	179	163
7-a	200	1536	138	151	162	183	164
1-b	50	798	134	149	163	180	163
2-b	75	969	135	149	163	181	164
3-b	100	1110	135	149	162	178	162
4-b	125	1230	136	149	162	180	163
5-b	150	1359	137	149	162	178	162
6-b	175	1455	137	149	162	178	162
7-b	200	1536	136	150	164	183	164

TABLE NO. 3

HEAT TRANSFER EXPERIMENT DATA (Continued)

RUN NO.	t °F	T °F	(T_2-T_1) °F	$-\Delta t$ °F	h Btu/hr ft^2°F
1-a	172.75	142.50	15.0	30.25	477.32
2-a	173.25	143.00	14.0	30.25	540.96
3-a	172.00	142.50	13.0	28.50	610.75
4-a	172.25	143.50	13.0	28.75	670.89
5-a	171.00	144.00	12.0	27.00	728.58
6-a	171.00	144.00	12.0	27.00	780.05
7-a	173.00	144.50	13.0	28.50	845.15
1-b	171.50	141.50	15.0	30.00	481.30
2-b	172.25	142.00	14.0	30.25	549.96
3-b	170.00	141.50	13.0	28.50	610.75
4-b	171.25	142.50	13.0	28.75	670.89
5-b	170.00	143.00	12.0	27.00	728.58
6-b	170.00	143.00	12.0	27.00	780.05
7-b	173.50	143.00	14.0	30.50	850.48

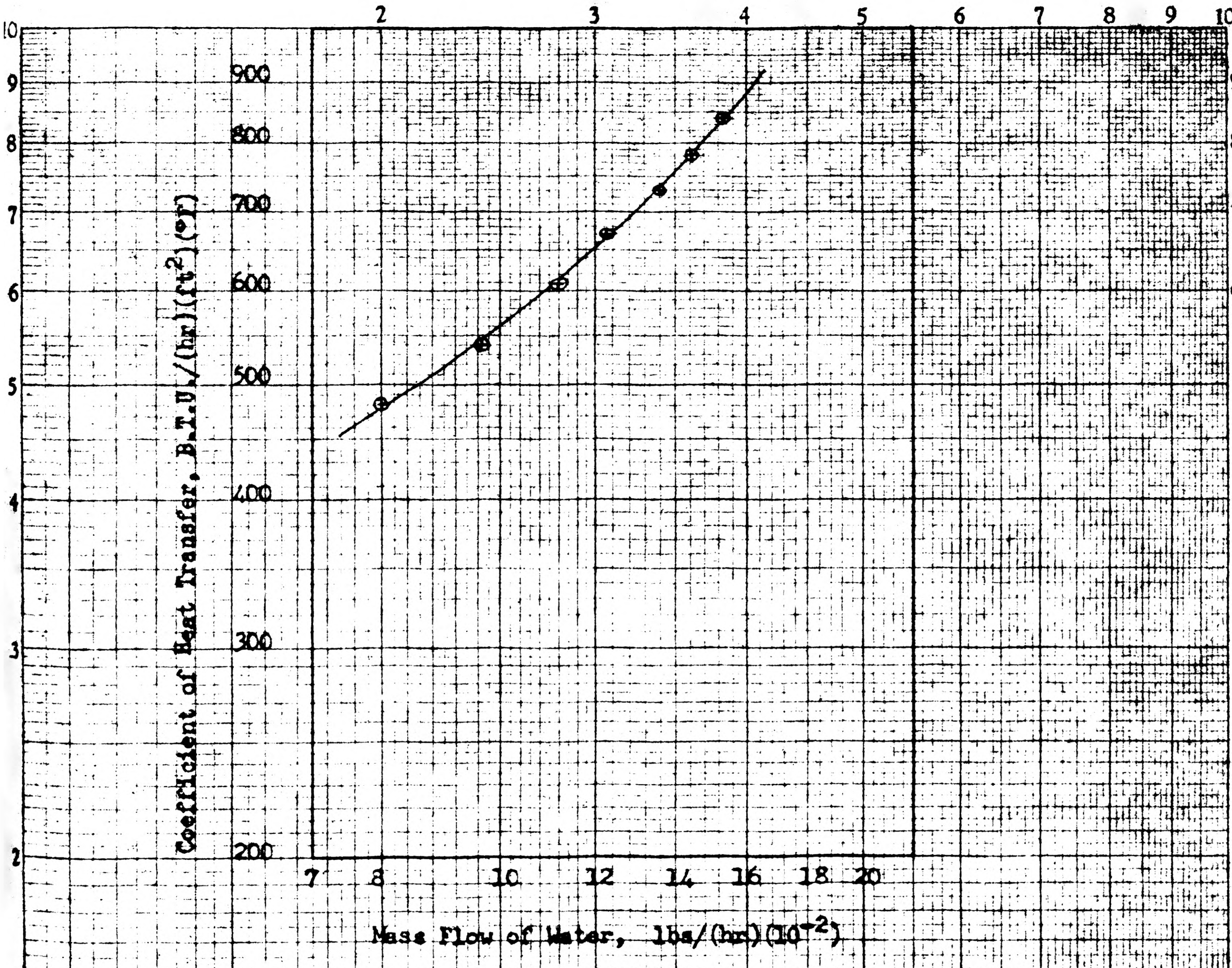


PLATE NO. 3

COEFFICIENT OF HEAT TRANSFER VS MASS FLOW FOR A PITCH OF 8

TABLE NO. 4

HEAT TRANSFER EXPERIMENT DATA

TEST SECTION: Steel pipe 0.825", I.D.; 1.05", O.D.; 20" long; Area = 0.8288 sq. ft. one end closed, 3/16" spiral cut in the pipe with a pitch of 10; Plate No. 4.

RUN NO.	FLOWMETER READING	W lbs/hr	WATER TEMP. °F T_1	T_2	SURFACE TEMP. °F POINT 1	POINT 2	POINT 3
1-a	50	798	149	159	164	187	167
2-a	75	969	152	162	167	191	173
3-a	100	1110	152	162	167	191	173
4-a	125	1230	152	162	167	191	173
5-a	150	1359	152	162	168	191	173
6-a	175	1455	152	162	167	190	173
7-a	200	1536	150	161	168	191	173
1-b	50	798	146	157	163	187	167
2-b	75	969	147	157	163	186	167
3-b	100	1110	147	158	166	189	171
4-b	125	1230	147	158	166	188	171
5-b	150	1359	149	159	165	188	170
6-b	175	1455	150	161	170	190	173
7-b	200	1536	150	161	170	190	173

TABLE NO. 4

HEAT TRANSFER EXPERIMENT DATA (Continued)

RUN NO.	t °F	T °F	(T_2-T_1) °F	$-\Delta t$ °F	h Btu/hr ft^2°F
1-a	176.25	154.00	10.0	22.25	432.73
2-a	180.50	157.00	10.0	23.50	497.51
3-a	180.50	157.00	10.0	23.50	569.90
4-a	180.50	157.00	10.0	23.50	631.52
5-a	180.75	157.00	10.0	23.75	690.40
6-a	180.00	157.00	10.0	23.00	763.28
7-a	180.75	155.50	11.0	25.25	807.37
1-b	176.00	151.50	11.0	24.50	432.29
2-b	175.50	152.00	10.0	23.50	497.00
3-b	178.75	152.50	11.0	26.25	561.22
4-b	178.25	152.50	11.0	25.75	633.97
5-b	177.75	154.00	10.0	23.75	690.40
6-b	180.75	155.50	11.0	25.25	764.79
7-b	180.75	155.50	11.0	25.25	807.37

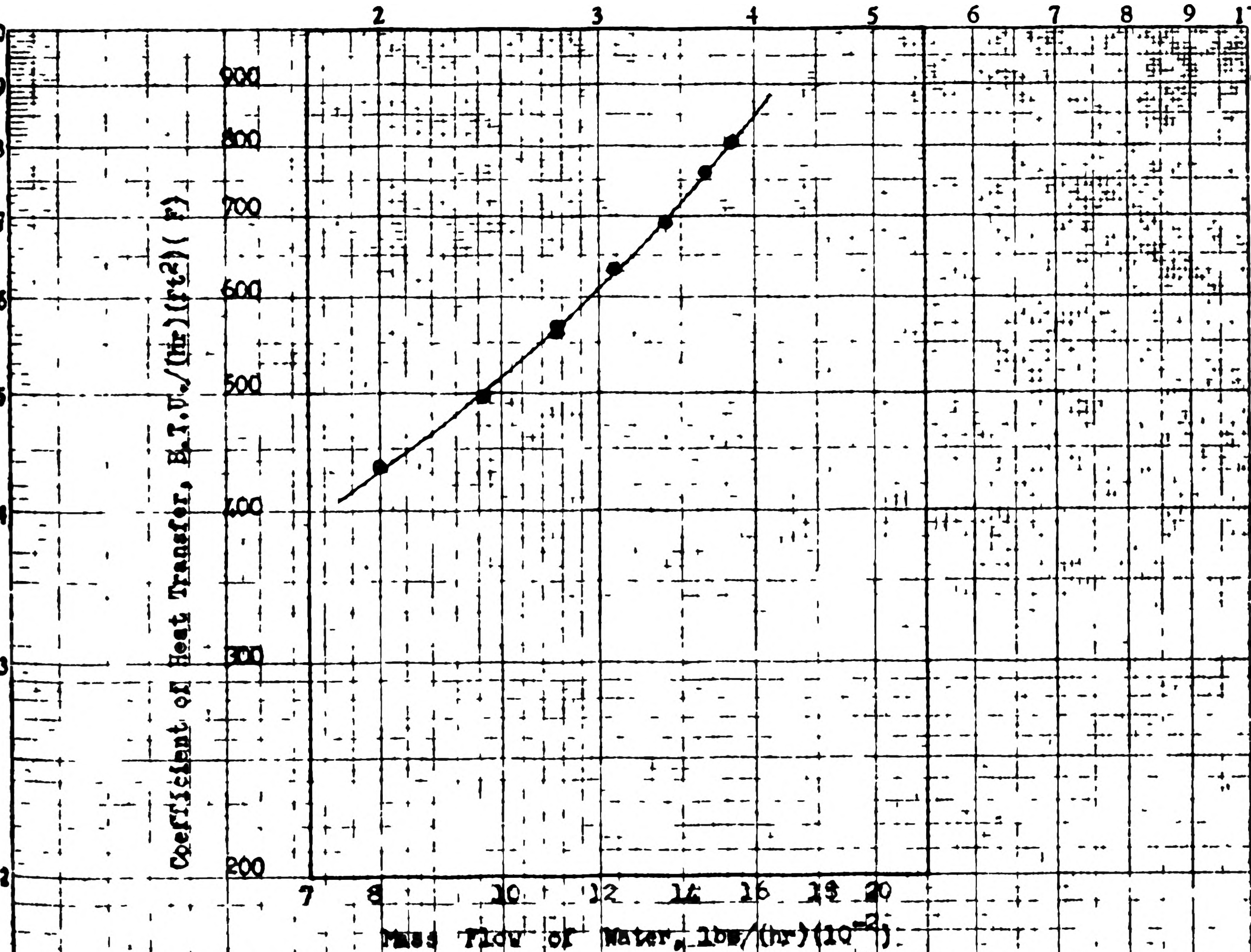


PLATE NO. 4

COEFFICIENT OF HEAT TRANSFER VS MASS FLOW FOR A PITCH OF 10

TABLE NO. 5

HEAT TRANSFER EXPERIMENT DATA

TEST SECTION: Steel pipe 0.825", I.D.; 1.05", O.D.; 20" long; Area = 0.8286 sq. ft. one end closed, 3/16" spiral cut in the pipe with a pitch of 16; Plate No. 5.

RUN NO.	FLOWMETER READING	W lbs/hr	WATER TEMP. °F T_1	WATER TEMP. °F T_2	SURFACE TEMP. °F POINT 1	SURFACE TEMP. °F POINT 2	SURFACE TEMP. °F POINT 3
1-a	50	798	135	147	167	172	167
2-a	75	969	135	147	167	173	168
3-a	100	1110	136	147	165	172	166
4-a	125	1230	136	147	165	172	165
5-a	150	1359	136	146	163	168	163
6-a	175	1455	136	146	162	168	162
7-a	200	1536	136	146	162	168	163
1-b	50	798	135	147	166	172	167
2-b	75	969	135	146	165	170	165
3-b	100	1110	135	146	165	170	165
4-b	125	1230	136	146	163	168	164
5-b	150	1359	135	145.5	163	168	164
6-b	175	1455	135	145	161	167	161
7-b	200	1536	135	145	161	167	162

TABLE NO. 5

HEAT TRANSFER EXPERIMENT DATA (Continued)

RUN NO.	t °F	T °F	(T_2-T_1) °F	$-\Delta t$ °F	Btu/hr ft^2°F
1-a	169.50	141.00	12.0	28.50	405.50
2-a	170.25	141.00	12.0	29.25	479.77
3-a	168.75	141.50	11.0	27.25	540.75
4-a	168.50	141.50	11.0	27.00	604.76
5-a	165.50	141.00	10.0	24.50	669.43
6-a	165.00	141.00	10.0	24.00	731.65
7-a	165.25	141.00	10.0	24.25	764.42
1-b	169.25	141.00	12.0	28.25	409.09
2-b	167.50	140.50	11.0	27.00	476.43
3-b	167.50	140.50	11.0	27.00	545.76
4-b	165.75	141.00	10.0	24.75	599.77
5-b	165.75	140.25	10.5	25.50	675.34
6-b	164.00	140.00	10.0	24.00	731.65
7-b	164.25	140.00	10.0	24.25	764.42

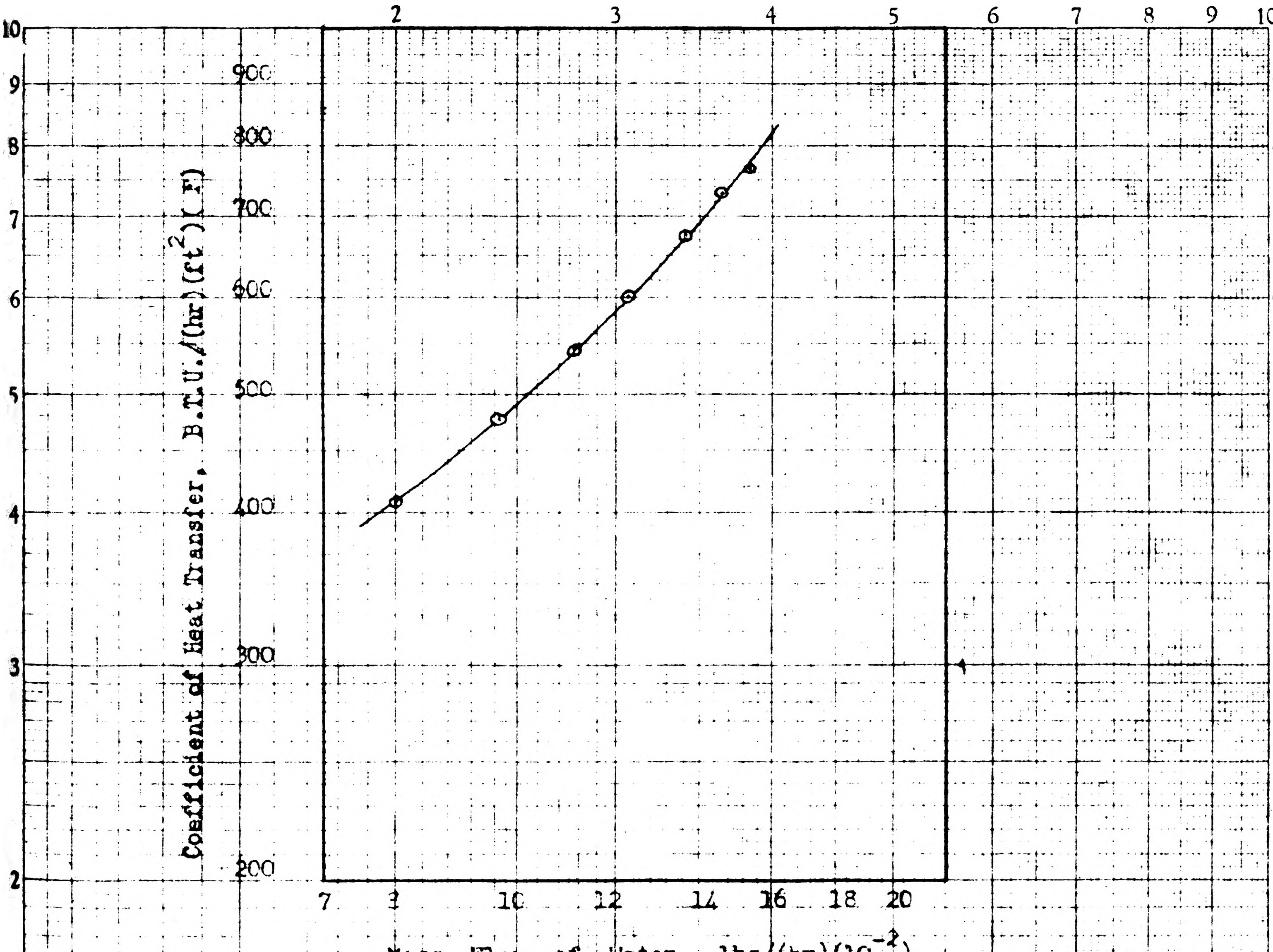


PLATE NO. 5

COEFFICIENT OF HEAT TRANSFER VS MASS FLOW FOR A PITCH OF 16

TABLE NO. 6

HEAT TRANSFER EXPERIMENT DATA

TEST SECTION: Steel pipe 0.825", I.D.; 1.05", O.D.; 20" long; Area = 0.8185 sq. ft. one end closed, 3/16" spiral cut in the pipe with a pitch of 20; Plate No. 6.

RUN NO.	FLOWMETER READING	W lbs/hr	WATER TEMP. °F T_1	T_2	SURFACE TEMP. °F POINT 1	POINT 2	POINT 3
1-a	50	798	144	155	165	191	166
2-a	75	969	145	157	170	199	172
3-a	100	1110	146	158	174	202	171
4-a	125	1230	146	158	174	202	172
5-a	150	1359	145	157	173	201	171
6-a	175	1455	145	157	172	201	170
7-a	200	1536	141	152	165	192	166
1-b	50	798	146	158	172	196	169
2-b	75	969	147	158	172	196	170
3-b	100	1110	147	157	171	193	169
4-b	125	1230	148	158	173	194	170
5-b	150	1359	148	159	174	199	173
6-b	175	1455	148	159	174	199	171
7-b	200	1536	148	159	173	199	172

TABLE NO. 6

HEAT TRANSFER EXPERIMENT DATA (Continued)

RUN NO.	t °F	T °F	(T_2-T_1) °F	$-\Delta t$ °F	h Btu/hr ft^2°F
1-a	178.25	149.50	11.0	28.75	373.02
2-a	185.00	151.00	12.0	34.00	417.83
3-a	187.25	152.00	12.0	35.25	461.66
4-a	187.50	152.00	12.0	35.50	507.97
5-a	186.50	151.00	12.0	35.50	561.25
6-a	186.00	151.00	12.0	35.00	609.47
7-a	178.75	146.50	11.0	32.25	640.08
1-b	183.25	152.00	12.0	31.25	374.38
2-b	183.50	152.50	11.0	31.00	420.08
3-b	181.50	152.00	10.0	29.50	459.70
4-b	182.75	153.00	10.0	29.75	505.12
5-b	186.25	153.50	11.0	32.75	557.67
6-b	185.75	153.50	11.0	32.25	606.32
7-b	185.75	153.50	11.0	32.25	640.08

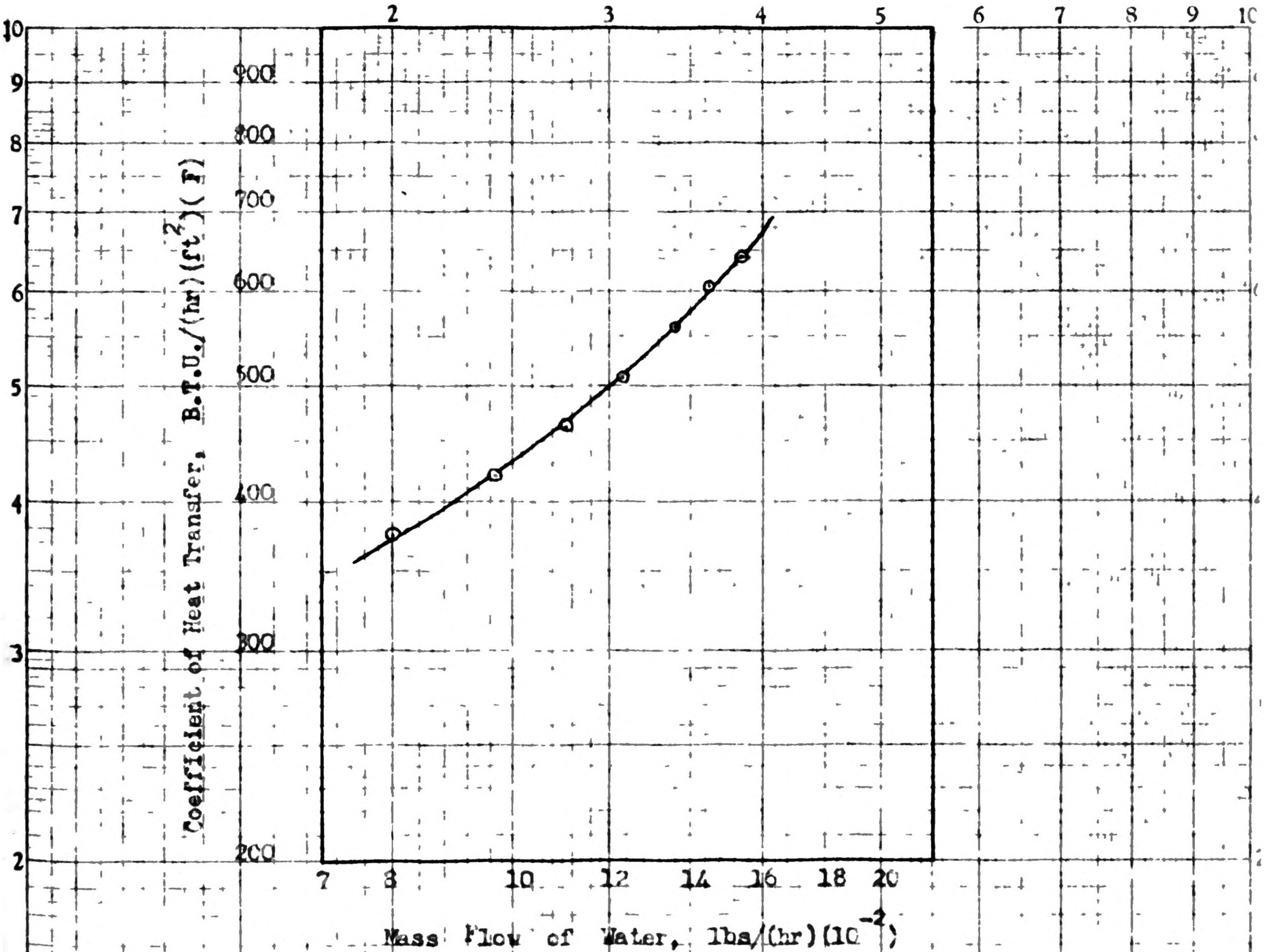


PLATE NO. 6

COEFFICIENT OF HEAT TRANSFER VS MASS FLOW FOR A PITCH OF 20

TABLE NO. 7

HEAT TRANSFER EXPERIMENT DATA

TEST SECTION: Steel pipe 0.825", I.D.; 1.05", O.D.; 20" long; Area = 0.817 sq. ft.; both ends open, without a spiral.

RUN NO.	FLOWMETER READING	W lbs/hr	WATER TEMP. °F T_1	WATER TEMP. °F T_2	SURFACE TEMP. °F POINT 1	SURFACE TEMP. °F POINT 2	SURFACE TEMP. °F POINT 3
1-a	50	805	120	132	135	186	172
2-a	75	968	120	131.5	131	185	165
3-a	100	1108	120	132	132	185	165
4-a	125	1220	120	132.5	133	185	164
5-a	150	1330	121	133.5	133	185	164
6-a	175	1435	121.5	133.5	132	183	162
7-a	200	1540	121	133.5	134	186	166
1-b	50	805	116	130	140	190	175
2-b	75	968	116	130	135	187	169
3-b	100	1108	116	129	134	183	162
4-b	125	1220	116	129	132	180	160
5-b	150	1330	116	130	134	185	163
6-b	175	1435	116	130.5	135	189	166
7-b	200	1540	116	129.5	131	185	164

TABLE NO. 7

HEAT TRANSFER EXPERIMENT DATA (Continued)

RUN NO.	t °F	T °F	(T_2-T_1) °F	$-\Delta t$ °F	h Btu/hr ft^2°F
1-a	169.75	126.00	12.0	43.75	270
2-a	166.50	125.75	11.5	40.75	335
3-a	166.75	126.00	12.0	40.75	398
4-a	166.75	126.25	12.5	40.50	460
5-a	166.75	127.25	12.5	39.50	515
6-a	165.00	127.50	12.0	37.50	562
7-a	165.50	127.25	12.5	38.25	615
1-b	173.75	123.00	14.0	50.75	272
2-b	169.50	123.00	14.0	46.50	357
3-b	165.50	122.50	13.0	43.00	410
4-b	163.00	122.50	13.0	40.50	480
5-b	166.75	123.00	14.0	43.75	520
6-b	169.75	123.25	14.5	46.50	550
7-b	166.25	122.25	14.5	44.00	620

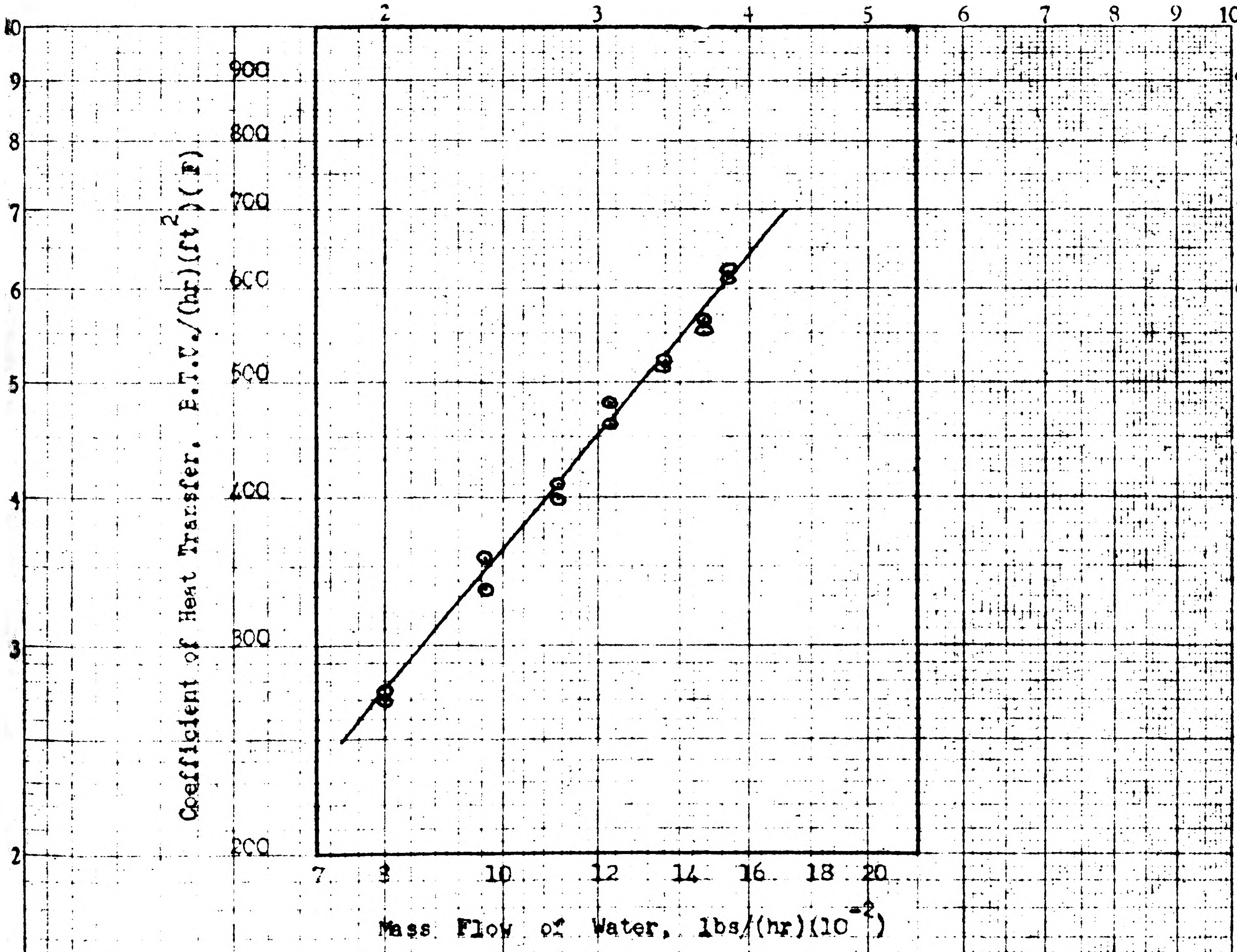


PLATE NO. 7

COEFFICIENT OF HEAT TRANSFER VS MASS FLOW FOR A PLAIN PIPE

All the test sections showed an increase in the heat transfer coefficient when compared with a plain pipe and each specimen also showed that the lower flow rates produced a higher efficiency than those at a high flow rate. The decrease in the efficiency with the increase in the flow rate indicates that the reduction in the boundary layer thickness and the increase in turbulence are higher at the lower flow rates. This does not mean that at the higher flow rates the boundary layer was thicker and there was less turbulence; on the contrary, this means that in comparing a plain pipe with a test section, this control method, was less effective at the higher flow rates and there was definitely a decrease in the boundary layer thickness and an increase in turbulence.

The test section with a pitch of 2 had an increase in the heat transfer coefficient of 60 percent at the minimum flow rate (798 lbs/hr) and only 22 percent increase at the maximum flow rate when compared with a plain pipe. Second high in efficiency at the minimum flow rate and third high in efficiency at the maximum flow rate was a test section with a pitch of 4. This specimen showed an increase of 71 percent at minimum flow rate and an increase of 31 percent at the maximum flow rate. The test section which had the highest over-all efficiency was one with a pitch of 8. It had an increase of 77 percent at the lowest flow rate and 38 percent increase at the highest flow rate. A pitch of 10 proved to have the second highest increase at the maximum flow rate with an increase of 31.5 percent and had a 60 percent increase at the minimum flow rate. A less impressive specimen (pitch of 16) showed that it only produced an increase of 50 percent at the minimum flow rate and 23.5 percent at the maximum flow rate. Using a test section with a pitch of 20 proved the least effective, having only an increase of 38 percent at the minimum flow rate and merely 4 percent increase at the maximum flow rate.

If the pitch of the spiral was decreased beyond a pitch of 20 this control method would help very little, and especially at the higher flow rates. Also, if the spiral's pitch was increased past a pitch of 2, we would expect the efficiency to decrease and that it would never have an efficiency higher than 60 percent at the minimum flow rate and 22 percent at the maximum flow rate.

When the convective heat transfer coefficient vs mass flow of water was plotted on logarithmic paper, all the test section appeared slightly curved. Plate Nos. 1, 2, 3, 4, 5, and 6 show this plot.

Plate No. 8 shows a comparison of test sections with a pitch of 2, 4, and 8 to one which has the increase of the flow rate decreases the effectiveness of the control method also we notice that the test section used seems to have developed the same type curve.

On the other hand, Plate No. 9 compares four test sections (Pitches of 8, 10, 16, and 20) to a plain pipe. This graph displays the variation of the slopes of the particular specimen and shows the effect of varying the pitch of the spiral. They all have a positive slope, but the one with a pitch of 20 has the greatest variation and shows that it may with a larger increase in the flow rate prove to have no effect in increasing the heat transfer rate. Test sections with a pitch of 10 and 16 have the sharpest increase in their slopes which could indicate that at higher flow rates (higher than the author used) these specimens could produce the highest heat transfer.

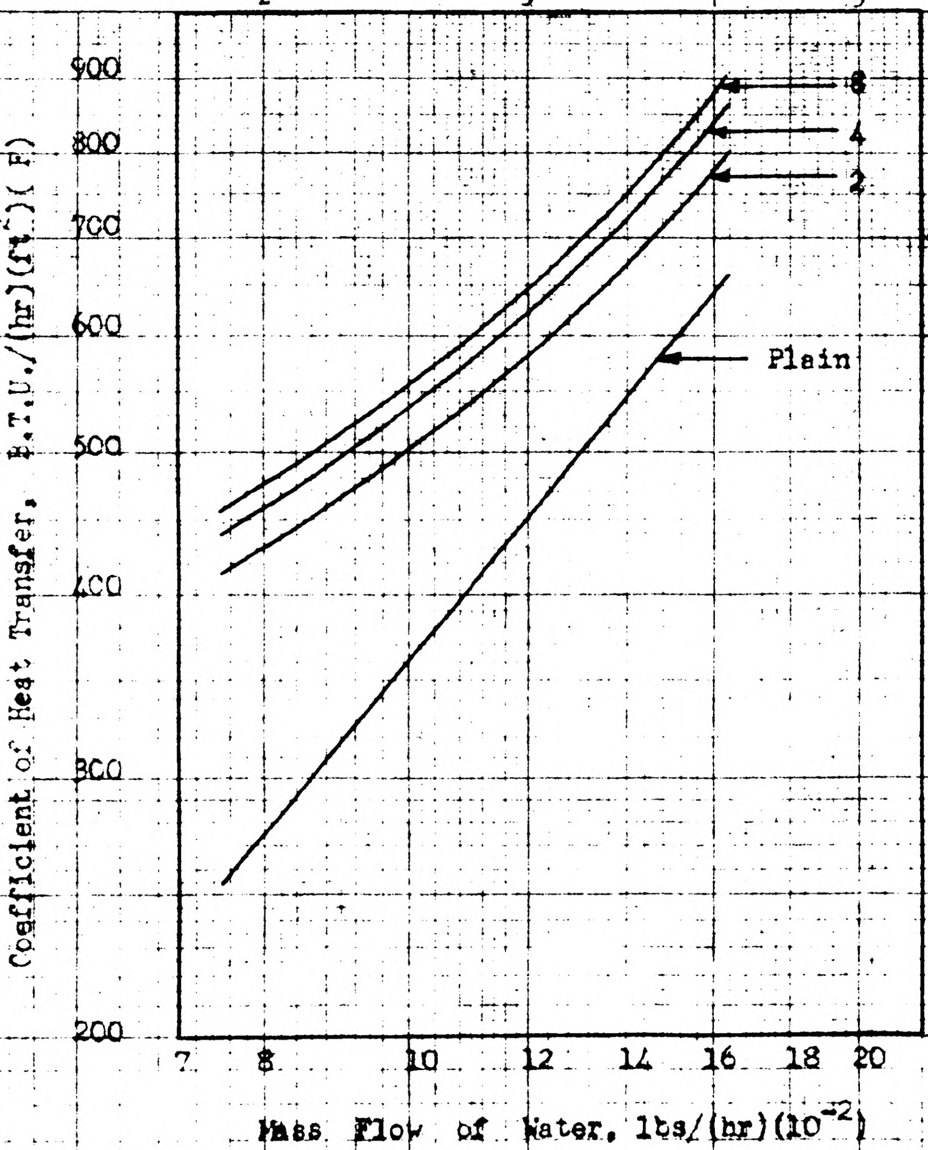


PLATE NO. 8

COEFFICIENT OF HEAT TRANSFER

VS

MASS FLOW FOR A PLAIN PIPE AND FOR PITCHES 8, 4, AND 2

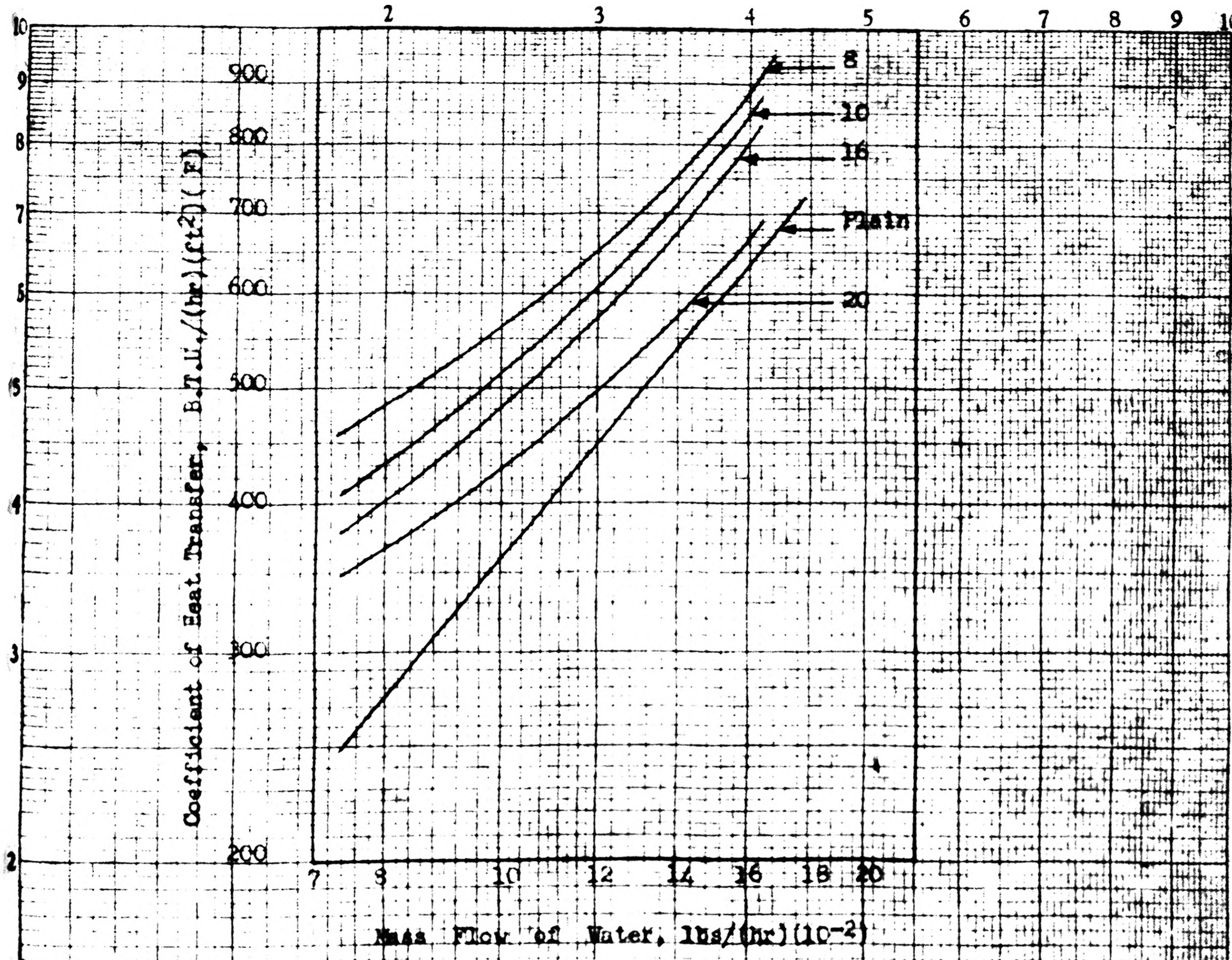


PLATE NO. 9
COEFFICIENT OF HEAT TRANSFER
VS
MASS FLOW FOR A PLAIN PIPE AND FOR PITCHES 8, 10, 16, AND 20

From the above discussion it is clear that for the test sections used and for the range of flow employed, the effect of the spiral as a boundary layer control method has definitely increased the heat transfer rate about two-fold in the minimum flow range and only a little more than one-third in the maximum flow range.

CONCLUSIONS

The following general conclusions can be drawn from the results of this investigation:

1. The effect of the boundary-layer control method 3/16 inch spiral cut) definitely increased the heat transfer coefficient for a given flow rate.
2. The total heat transfer area increased slightl y due to cutting the spiral in the pipe, which is also a factor for increasing the heat transfer from the same pipe.
3. The highest increase in the heat transfer coefficient was found when a sp'ral with a pitch of 8 was used. It produced an increase of 38 percent at the highest flow rate (1536 lbs/hr) and 77 percent at the l owest flow rate (798 lbs/hr) compared with the values obtained with flow on both the outside and the inside surface of the pipe simultaneously.
4. The percentage increase in heat transfer was more in all cases at the lower flow ranges than at the higher ranges.

5. The effect of the boundary-layer control method not only increases the rate of heat transfer but saves a considerable amount of power that normally would be required in pumping the fluid at the same rate of heat transfer in comparison with the one with no control method.

The results of this investigation show some definite directions in which further work in this fiel d can be carried out. One is to study the effect of using more than one spiral cut in a pipe this could increase the heat transfer rate and also produce a better efficiency at the higher flow rates.

This research was mainly a qualitative study of the convective heat transfer coefficients in the different test sections rather than a precise quantitative study. This was due to the limitations imposed by the apparatus and the measuring devices. The accuracy of the results are within ± 10 percent.

APPENDIX

TABLE NO. 8

FLOW-METER CALIBRATION DATA

RUN NO.	FLOW METER READING	TIME IN SECONDS	WEIGHT OF WATER (lbs)	FLOW IN LBS/HR
1	50	360	798	798
2	75	360	969	969
3	100	360	1110	1110
4	125	360	1230	1230
5	150	360	1359	1359
6	175	360	1455	1455
7	200	360	1536	1536

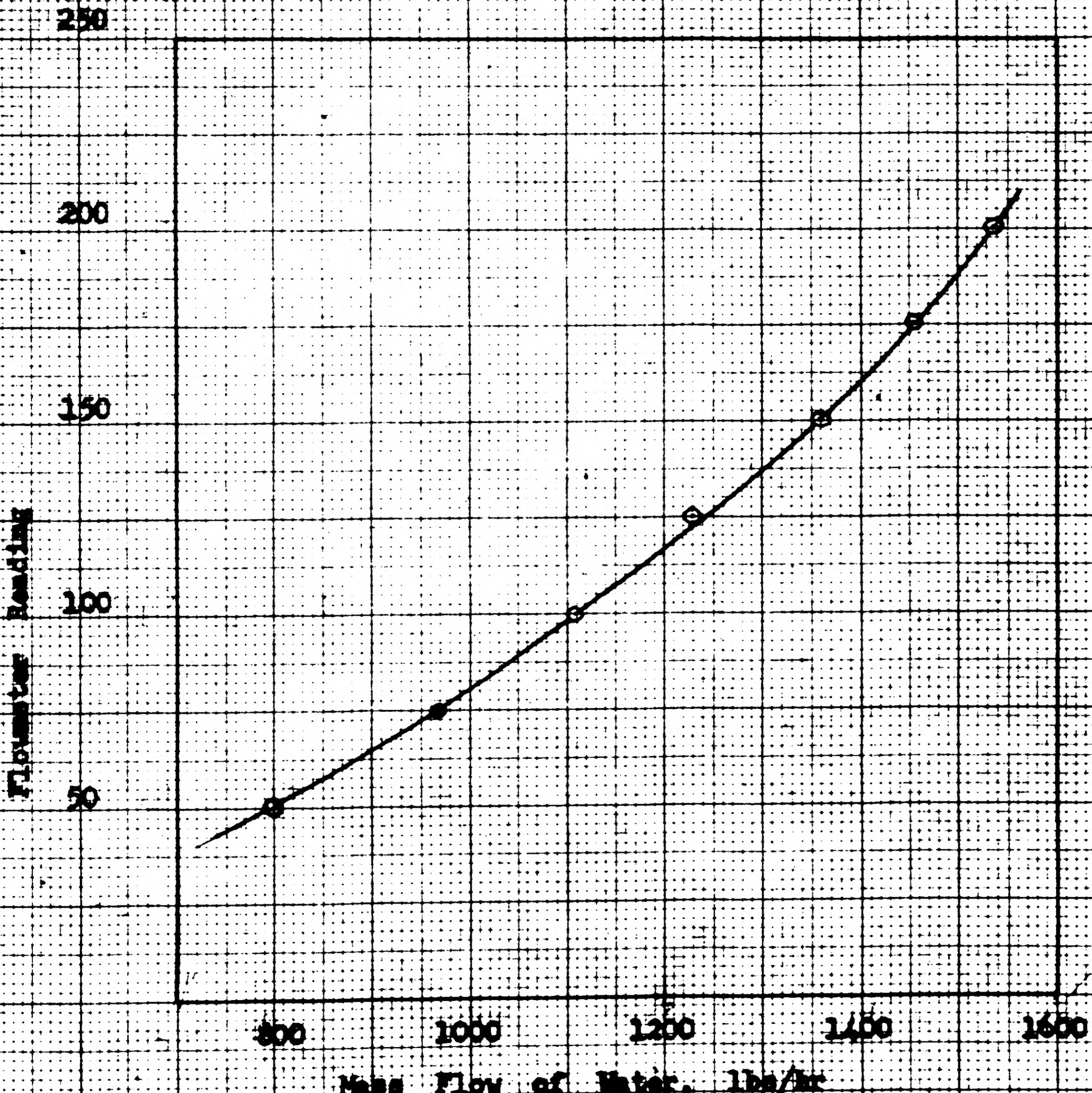


FIGURE NO. 11

CALIBRATION CURVE FOR THE HONEYWELL FLOWMETER

```
;0004800' /0000000'

r6300' u0400' m0000' m0000' u0000',0000009' 20p4i4t4'c4h4zj=j' zj4040f8'
zjm46010't46020t4' 60Δ8t460' 10-j20Δ8' 10t460h4' 00000000'
m0000' m0000' i5000' b5000' a5008' d5016' h5200' b5004' m5016'a5002'
a5006' d5014' h5202' s5200' h5204' m5012' h5206' b5008'
s5000' h5208' m5010' d5206' h5210'b5010' z0002'd0000'
b5202' z0002' d0000' b5200' z0002' d0000' b5208' z0002'
d0000'b5204' z0002' d0000' b5210' z0002' u4814'

.0004800'
```

FIGURE NO. 12

PROGRAM USED IN THE CALCULATIONS OF THE CONVECTIVE HEAT TRANSFER COEFFICIENTS

PITCH - 20

FM	t	T	ΔT	$-\Delta t$	h
144' ' 165' ' 191' '166' ' 155' ' 798' '8185' +04 ' 4' '2' ' f'					
798.00	178.25	149.50	11.00	28.74	373.02
145' ' 170' '199' '172' '157' '969' 'f'					
969.00	185.00	151.00	12.00	33.99	417.83
146' '174' ' 202' ' 171' '158' ' 1110' 'f'					
1110.00	187.25	152.00	12.00	35.24	461.66
146' '174' ' 202' ' 172" 158' ' 1230' 'f'					
1230.00	p87.50	152.00	12.00	35.49	507.97
145' ' 173' '201' ' 181' ' 157' ' 1359"f'					
1359.00	186.50	151.00	12.00	35.49	561.24
145' '172' ' 201' ' 170' '157' ' 1455' 'f'					
1455.00	186.00	151.00	12.00	34.99	609.47
141' ' 165' ' 192' '166' ' 152' ' 1536' ' f'					
1536.00	178.75	146.50	11.00	32.24	640.08

FIGURE NO. 13 SAMPLE RUN

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VITA

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