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# Heat transfer coefficients in artificially roughened pipes

Mehmet Hayati Turan

# HEAT TRANSFER COEFFICIENTS
# IN ARTIFICIALLY ROUGHENED PIPES

BY

MEHMET HAYATI TURAN

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A

THESIS

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Approved by

Aaron J. Miles (Advisor) Ralph E. Schowalter

Ralph E. Lee R. F. Davidson

Robert V. Wolf

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# NOMENCLATURE

| | | |
|---|---|---|
| A | Surface area | sq. ft. |
| C | Constant | |
| $C_p$ | Specific heat | Btu/(lb)(deg. F) |
| D | Diameter of the pipe | ft. |
| e | Depth of the surface projection | ft. |
| f | Friction factor | dimensionless |
| h | Heat transfer coefficient | Btu/(hr)(sq.ft.)(deg.F) |
| $j_h$ | Heat transfer factor, $h(Pr)^{2/3}/C_pV\rho$ | dimensionless |
| k | Thermal conductivity | Btu/(hr)(sq.ft.)(deg.F/ft.) |
| L | Length | ft. |
| m | Constant | |
| n | Constant | |
| Pr | Prandtl number, $C_p\mu/k$ | dimensionless |
| q | Rate of heat transfer | Btu/hr |
| Re | Reynolds number, $DV\rho/\mu$ | dimensionless |
| T | Temperature | deg. F |
| t | Thickness of the boundary layer | ft |
| V | Velocity of the fluid | ft/hr |
| W | Mass rate of flow of fluid | lb/hr |
| x | Distance | ft |
| $\alpha$ | Constant | |
| $\mu$ | Absolute viscosity of the fluid | lb/(hr)(ft) |
| $\rho$ | Density of the fluid | lb/cu. ft. |

indices

f fluid

i inside

w wall

# I. INTRODUCTION

The purpose of this investigation is to determine experimentally the effects of three different types of artificial roughness on the convective heat transfer coefficient in a pipe and compare these values with that for a smooth pipe for turbulent flow.

In a turbulent flow, in the bulk of the fluid i.e., in the core of the flow[4], heat is exchanged by mixing movements predominately; and in the boundary layer which is in laminar motion along the wall, heat exchange is solely due to the conduction. The thickness, t, of the boundary layer is usually expressed[2] as

$$t = 32.8\ D\ /\ Re\sqrt{f} \qquad \text{.............} \quad (1)$$

If the projections which make the inner surface of the pipe rough, extend through the boundary layer, the friction factor f may be increased enormously. As will be seen from Eq. (1), this causes the thickness, t, to decrease for a given Reynolds number. A decrease in the thickness of the boundary layer results in an increase in the amount of heat transferred from the pipe to the fluid according to Fourier's equation for one dimensional heat transfer in steady state condition:

$$dQ = k\ dA\ \frac{dT}{dx} \qquad \text{...............} \quad (2)$$

On the other hand any roughening of the surface causes an increase in turbulence which helps mixing action in the core of the fluid.

(4) All references are in bibliography.

It is seen that by roughening, it is possible to increase the heat transfer coefficient in a pipe. This is especially true for flow in small pipes where a given degree of roughness is proportionately greater than in large pipes. The effect of the roughness on the heat transfer coefficient depends not only on the ratio of the mean height of the surface projections to the radius of the tube, but also on the shape and surface distribution of the projections which were also studied in this experimental work.

Intentional roughening[1] of a surface in order to increase heat transfer can have only limited practical application, however since the increase in heat transfer due to roughness is accompanied by the disadvantage of a still greater increase in frictional resistance to flow. The degree of increment in pressure loss due to increase in frictional resistance will be mentioned later in the Review of Literature. This phenomena is considered beyond the scope of this investigation.

Heat transfer engineers who are dealing with the subject of heat removal from nuclear reactors face with the problem of removal of heat rates as high as 1,000,000,000 Btu/(hr) (cu ft) coolant. A method developed to increase the heat transfer coefficient could be helpful in several ways: reduce the mass rates of coolant for a given amount of heat transfer; since the heat release is independent of the heat removal rate, remove more heat from the reactor with a given amount of coolant. This is the field where pressure loss would not be considered as a disadvantage.

The author wishes to acknowledge his special indebtedness to Dr. Aaron J. Miles, Chairman of the Mechanical Engineering Department at Missouri School of Mines and Metallurgy, for his valuable guidance. Thanks are due to A. V. Kilpatrick, Professor in Mechanical Engineering, and R. D. Smith, Mechanical Engineering Technician, for their appreciable work in preparing the testpieces and setting up the test apparatus.

## II. REVIEW OF LITERATURE

W. F. Cope[6] studied cooling of water in tubes with three degrees of artificial roughness, in which the height of projections which were in the shape of a pyramid, ranged from one forty-fifth to one-seventh of the radius of the pipe. Although in the turbulent region the friction ran as high as six times that for smooth tubes, the heat transfer ran only 20 to 100 percent greater than for smooth tubes. It was concluded that, for the same power loss or pressure drop, greater heat transfer was obtained from a smooth rather than from a rough tube.

R. C. Hastrup, R. H. Sabersky, D. R. Bartz and M. B. Noel[3] investigated friction and heat transfer in a rough tube at varying Prandtl numbers. As the working fluid they used water. By working at various temperatures, experiments were performed over a continuous range of Prandtl numbers from about 1 to 7. Roughening of the test tube was obtained by knurling from the outside. During the knurling operation the inside of the tube was filled with a low melting point alloy ("Cerro bend") which was melted out after completion of the operation. The knurl pattern used was diamond shaped and relatively coarse. The diagonals of the diamond were approximately 0.10 and 0.05 in., and the depth of impressions produced on the inside of the tube was approximately 0.008 in. The experiments covered a range of relative roughness up to 1/15, where the relative roughness was defined as the ratio of the sand grain diameter to the tube diameter. The results of the experiments showed that the heat transfer coefficient increases with roughness and the increase in the friction coefficient exceeds that of the heat transfer coefficient. From this it was concluded that friction is more strongly affected by roughness than is heat transfer.

Benjamin Pinkel[7] summarized the results of experimental and analytical studies performed at the NACA to determine the heat transfer and friction coefficients for the flow of air through tubes with large difference in temperature between tube wall and air, a) for smooth tubes of circular cross section; b) for tubes of noncircular cross-sectional shapes; and c) for tubes with various degrees of surface roughness. Three degrees of surface roughness were obtained by machining square threads into the inner surface of the tube. Investigation was made for temperature differences between the air and tube wall up to 1500 F and Reynolds numbers from 1000 to 350,000. The author attempted to achieve further generalizations by analytical study more than to bring out numerical concepts of the subject. But he showed that the heat transfer coefficient was higher for rough tubes than that for smooth tube. He used the roughness factor which was defined as the ratio of the effective thickness of boundary layer associated with roughness to tube diameter. It was shown that this factor is a function of the several dimensions that characterize the roughness. The author drew the attention to the fact that it is possible that because of the fin action of the roughness that the roughness parameters for heat transfer and friction differ. He also pointed out the necessity of further work to understand the various types of roughness and their influence on heat transfer and friction.

J. W. Smith and Norman Epstein[8] studied the effect of wall roughness on convective heat transfer in commercial pipes using air as the working fluid. Seven commercial pipes were investigated, covering roughness ratio D/e from 641 to 63.4. A Reynolds number range of 10,000 to 80,000 was covered. Some increase in heat transfer coefficients with roughness was found, but the heat transmission per unit power loss always decreased. The latter conclusion was arrived by plotting the group $j_h/Re^2 f$ which represents heat transfer coefficient per unit frictional power loss.

# III. DISCUSSION

## 1. APPARATUS USED IN EXPERIMENT

Apparatus was composed of several components. A general view of it is given by Figure (1). The major components are described below.

1. Westinghouse Multi-Purpose Single Phase Induction Heater:

This equipment is a multi-purpose induction heater operating with a 30 KW 9600 cycle motor-generator set. It is capable of supplying output voltages ranging from 12 ½ to 800 volts.

The station has two output terminals, one for high and one for low voltage output. The high voltage output terminal is fed from a high frequency auto-transformer which provides 100, 200, 300, 400, 500, 600, 700, and 800 volts output. These output voltages are easily obtained by changing the readily accessible links on the auto-transformer located just inside the rear door. Low voltage terminal can give 12 ½, 37 ½, 50, 62 ½, 75, and 87 ½ volts by changing the links on the auto-transformer from inside the rear access doors.

A copper induction heating coil heats the test piece which is placed at the longitudinal centerline of the coil.

The characteristics of the heater necessitate the use of water cooling to maintain proper operating temperature within the equipment.

Front panel of the heater is equipped with the start, stop, rheostat control push buttons, power control knob, and voltage, ampere, kilowatt etc. meters.

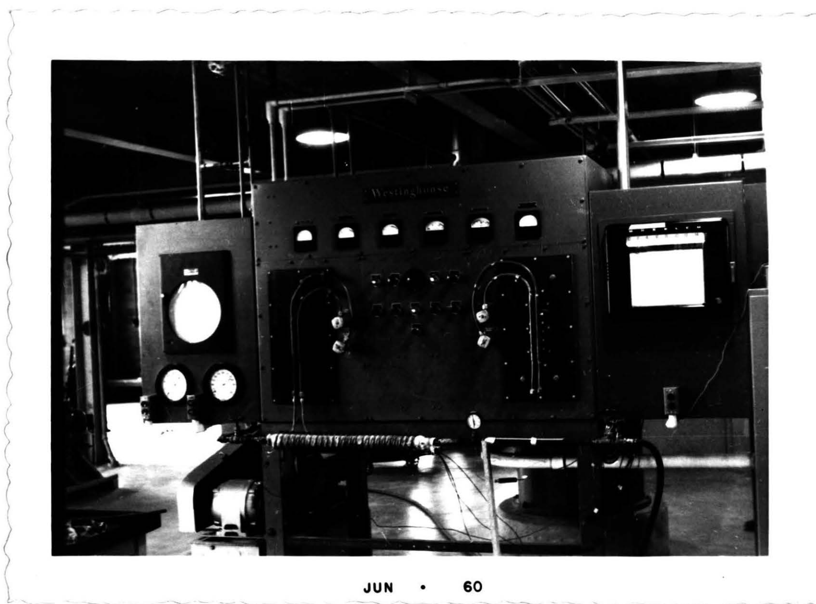



Figure 1. General View of the Apparatus

2. Wheelco Temperature Recorder:

The Series 8000 Recorder is a continuous null balance type d-c potentiometer. It records the temperatures between 0 and 1500 degrees Fahrenheit automatically. Some specifications for the recorder are given below.

| | |
|---|---|
| Chart width | : 11" calibrated width |
| Printing cycle | : 5 seconds |
| Cross chart speed | : 4 seconds |
| Chart speed | : 4" per hour |
| Accuracy | : ¼ of 1% full scale range |

3. Minneapolis-Honeywell Bellows Type Differential Pressure Flowmeter, Model 292 D 15:

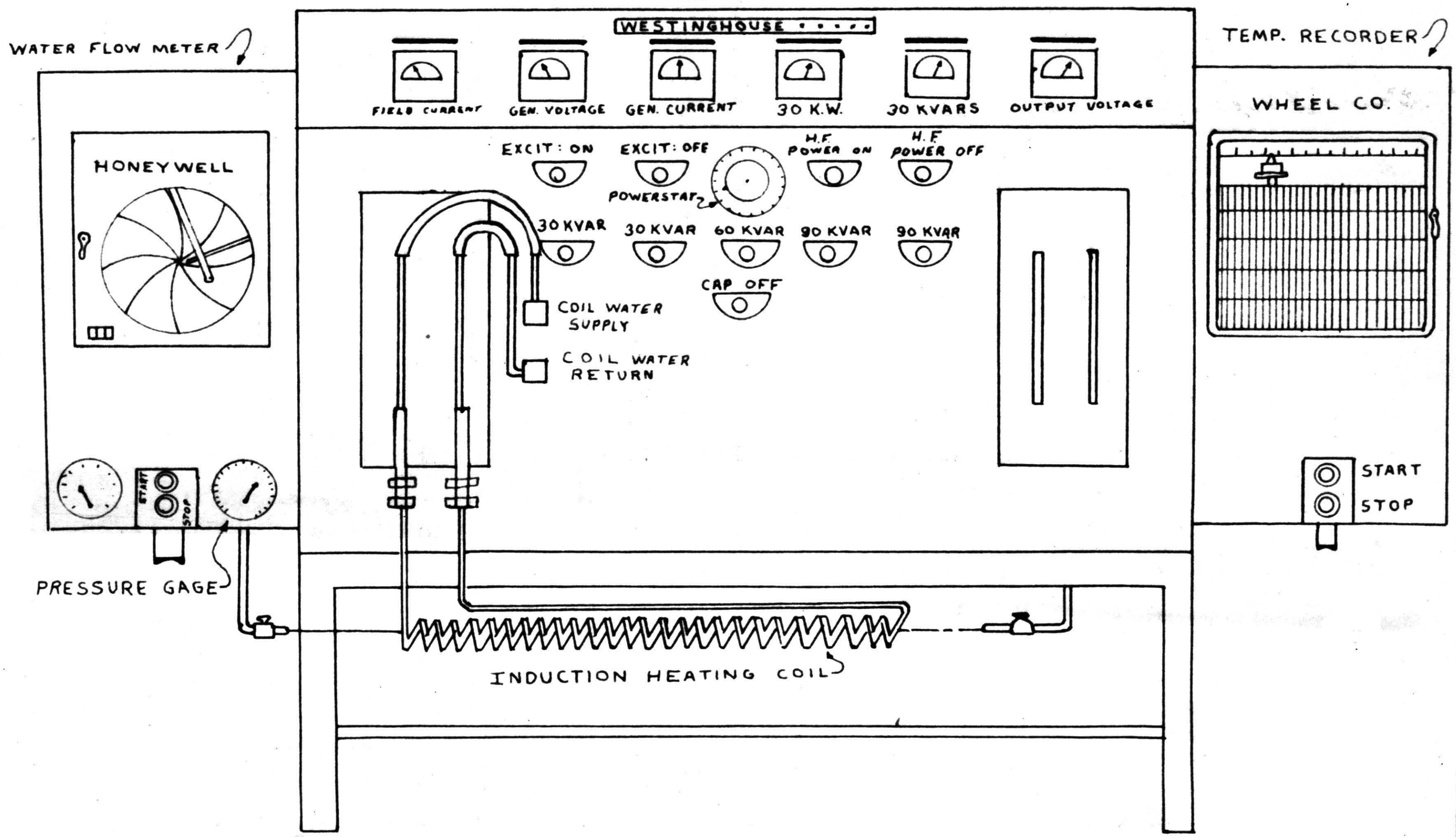
The flowmeter is calibrated to work at flow rates between 0 and 4 gallons per minute. Because its accuracy is lessened, a calibration chart is prepared. This chart corrects the values of flow rates recorded by the flowmeter and gives the flow rates on lbs/hr. The flow rates used in the experiments were taken from that chart.

Figure (2) illustrates front elevation of the high frequency induction heater with water flowmeter and temperature recorder.

4. Testpieces:

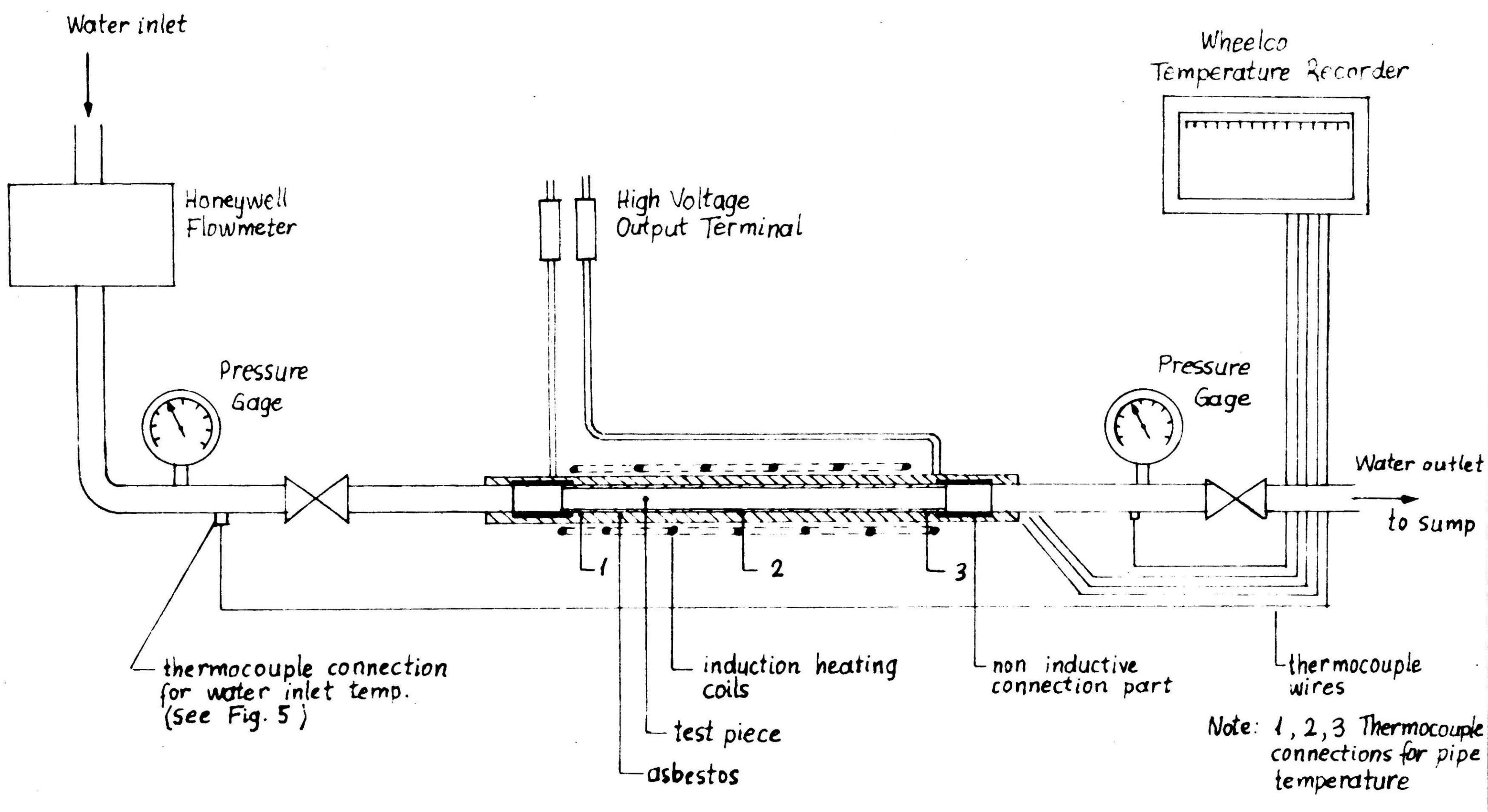
For the testpieces 3/4" 20" long pipes were used. The position of a testpiece in the apparatus is shown in Figure (3). Figure (4) illustrates four different types of testpiece with the dimensions. Testpiece No. 1 is a

FIGURE # 2




FRONT ELEVATION OF THE HIGH FREQUENCY INDUCTION HEATER
WITH WATER FLOW METER AND TEMPERATURE RECORDER

FIGURE # 3




SCHEMATIC DIAGRAM OF APPARATUS AND TESTPIECE

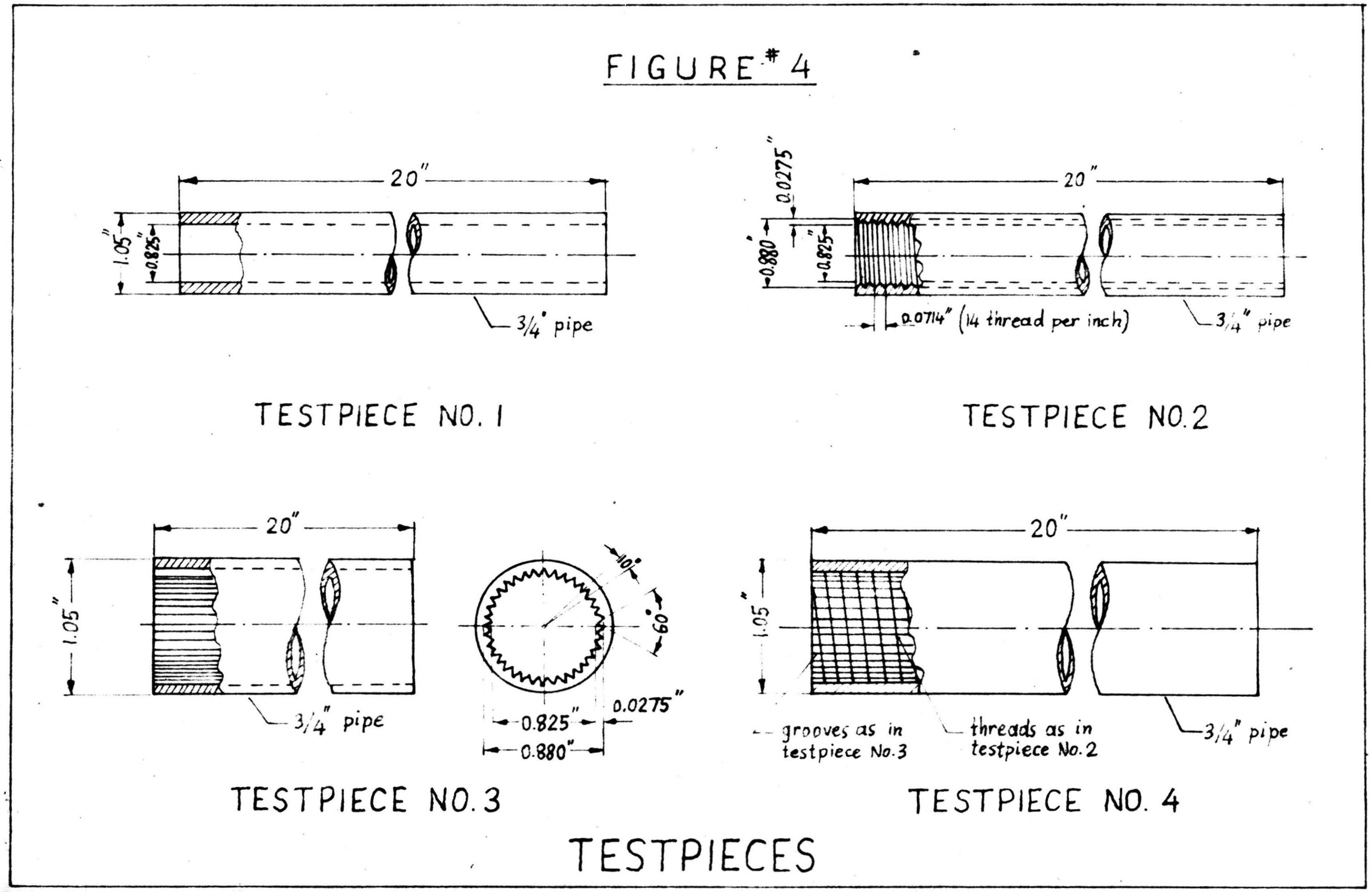


FIGURE # 4
20"
1.05"
0.825"
3/4" pipe
TESTPIECE NO. 1
20"
0.0275"
0.880"
0.825"
0.0714" (14 thread per inch)
3/4" pipe
TESTPIECE NO.2
20"
1.05"
3/4" pipe
10°
60°
0.825"
0.880"
0.0275"
TESTPIECE NO.3
20"
1.05"
grooves as in testpiece No.3
threads as in testpiece No.2
3/4" pipe
TESTPIECE NO. 4
TESTPIECES

smooth 3/4" pipe; No. 2 is inside threaded with a 7/8" tap 14 thread per inch. The depth of the threads is 0.0275". No. 3 testpiece is inside splined. The number of the grooves is 36 and the depth of them is 0.0275". No. 4 testpiece is a combination of No. 2 and No. 3. That is, it is threaded and splined with the same dimensions as No. 2 and No. 3. All the testpieces have the same roughness ratio $e/D = 0.033$.

In order to heat only the testpiece by induction, it is installed in the apparatus with non-inductive bakalite-like connection parts at both ends. To prevent heat loss from pipe to air, asbestos was used to cover the testpiece.

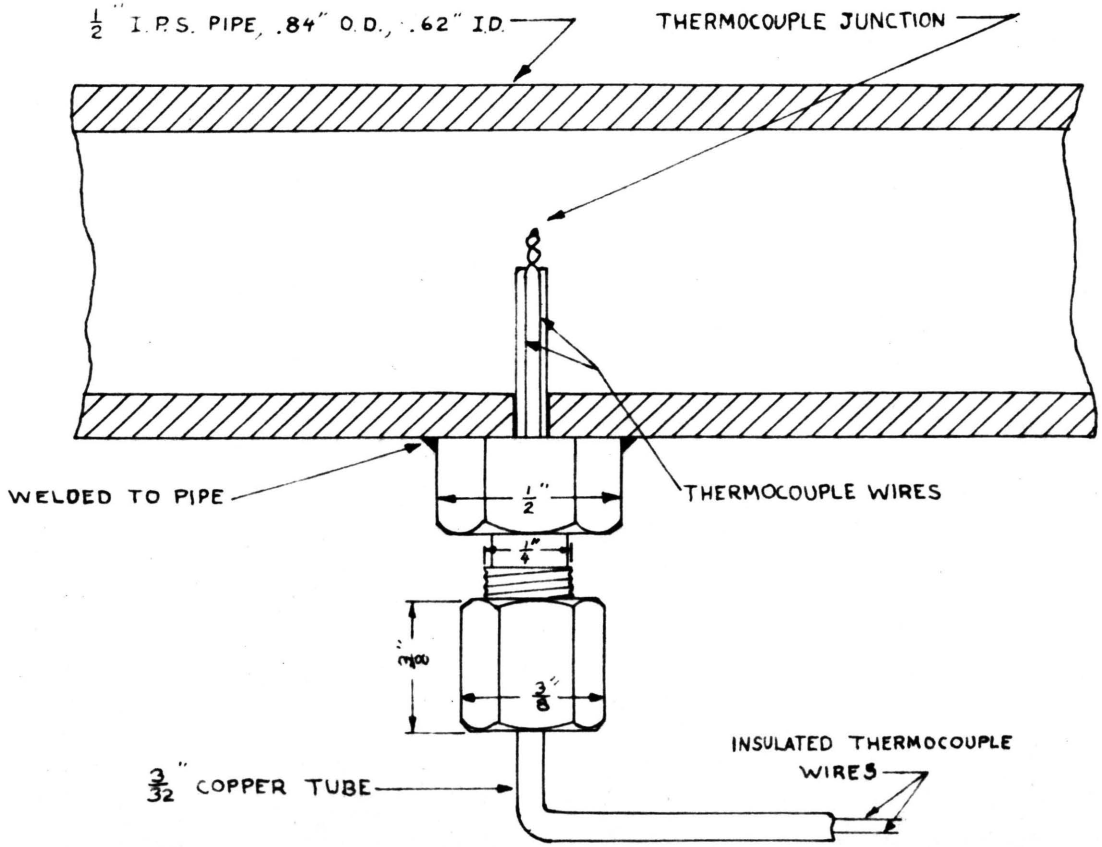
5. Thermocouples:

Iron-Constantan type thermocouples which were manufactured to work between 0 to 1400 degrees Fahrenheit were used to measure the temperatures. Two thermocouples were inserted in the pipe, before and after the testpiece, to measure the inlet and outlet water temperatures. Details of this type connection are shown in Figure (5). Three thermocouples were brazed on the testpiece, one at the middle and two at a distance approximately one inch from both ends to measure the pipe temperature. The other ends of the thermocouples were connected to the Wheelco Temperature Recorder.

2. EXPERIMENTAL PROCEDURE

A total of eight experiments were performed, two for each testpiece. One experiment contained seven runs. After starting the first run it was necessary to wait at

FIGURE # 5




THERMOCOUPLE INSTALLATION TO MEASURE WATER TEMPERATURE IN PIPE least 15 minutes to ready a steady state condition for recording the data. The inlet water was condensed water supplied from the power plant near to the laboratory. Its temperature showed considerable difference between the first and last runs at each experiment. There was no certain time to be sure to have a steady water inlet temperature. This can be seen from the data taken.

With the water running through the testpiece, induction equipment was put into operation. Its cooling system was also started. Since it was not desirable to have boiling in the pipe, testpiece temperature was taken as the controlling temperature, the highest being limited by the water pressure in the pipe. The temperatures and pressures will be given later in the section "The Range of Measurements." Controlling the power output from the heating equipment, convenient temperatures were maintained. Flow rates were controlled by a throttle valve installed after the testpiece. The reason to use this valve was to have pressure in the testpiece so that it was possible to reach pipe temperatures higher than 212 degrees Fahrenheit without any boiling in the water. Having high pipe temperatures resulted in considerable difference between the inlet and outlet temperatures which played a great role in the accuracy of the results.

After each water flow rate adjustment, data was not recorded before at least 15 minutes which was the minimum time for the testpiece to reach a steady state condition. Yet, it was almost impossible to have a steady inlet water temperature although it was changing so slowly that it was considered constant for each run.

At the high water flow rates, power output from the heating equipment was increased to maintain the inlet and outlet water temperature difference as high as possible but being careful in not exceeding the overall average of the water bulk temperature.

Some discontinuities of the data are due to the fact that the heating equipment is shut off automatically by the relays in the middle of the experiment when the temperature within the equipment exceeds a predetermined value. After a cooling off period of approximately 30 minutes second half of the experiments were performed. Sometimes new power adjustments resulted in slight discontinuities of data, but these did not affect the results very much.

3. RANGE OF MEASUREMENTS

1. Temperatures:

Inlet water temperatures changed between 100 and 127.5 degrees Fahrenheit. Outlet water temperatures were between 112.5 and 135 degrees Fahrenheit. The bulk temperatures of water changed from 114 to 126.5 degrees Fahrenheit. The difference gives Prandtl numbers not differing more than 10 percent which was considered within the accuracy of the results obtained in this investigation. Pipe temperatures ranged between 160 to 233 degrees Fahrenheit.

2. Pressures:

20.5 psi for the maximum and 17 psi for the minimum water pressures were indicated on the pressure gage located between the testpiece and the throttling valve. The boiling temperature of water at 17 psi (gage pressure) is approximately 254 degrees Fahrenheit.

3. Water Flow Rates:

Water flow rates changed from 550 to 1285 lb/hr. Taking the properties of water at the average bulk temperature which is 120 degrees Fahrenheit the Reynolds numbers are calculated and given below:

| W lb/hr | Re |
|---|---|
| 550 | 7500 |
| 719 | 9800 |
| 878 | 11950 |
| 1005 | 13700 |
| 1112 | 15150 |
| 1195 | 16300 |
| 1285 | 17500 |

4. CALCULATIONS

The flow of heat[1] through the fluid film that is assumed to adhere to the surface of any solid in contact with a fluid may be expressed as

$$q = hA\,(T_w - T_f) \quad \text{Btu/hr} \qquad (3)$$

In this investigation h was called "heat transfer coefficient." A is given by

$$A = \pi D_i L \qquad \text{sq. ft.} \tag{4}$$

When the values of $D_i$ and L are substituted in Eq. (4) A is found to be 0.36 sq. ft. $T_w$ is wall temperature and is defined as the mean temperature along the testpiece

$$T_w = \frac{T_3 + 2T_4 + T_5}{4} \qquad \text{deg. F} \tag{5}$$

where $T_3$, $T_4$, and $T_5$ are the pipe temperatures at three different points. Fluid bulk temperature $T_f$ is found from the inlet ($T_1$) and outlet ($T_2$) water temperatures

$$T_f = \frac{T_1 + T_2}{2} \qquad \text{deg. F} \tag{6}$$

On the other hand, the amount of heat transferred by the water is

$$q = WC_p\ (T_2 - T_1) \qquad \text{Btu/hr} \tag{7}$$

From Eq. (3) and (7)

$$h = \frac{WC_p\ (T_2 - T_1)}{A\ (T_w - T_f)} \qquad \text{Btu/hr sq. ft. deg. F} \tag{8}$$

is found. Assuming $C_p$ is equal to 1 and is constant throughout the experiments and substituting the value of A in Eq. (8) the final expression for h is obtained

$$h = \frac{W\ (T_2 - T_1)}{0.36(T_w - T_f)} \quad \text{Btu/hr sq. ft. deg. F} \qquad (9)$$

This value of h is an average heat transfer coefficient of the testpiece under investigation. $(T_w - T_f)$ was shown as $\Delta T$ in the Tables I (a) to IV (b).

## 5. EXPERIMENTAL RESULTS

Heat transfer data taken from the experiments is given in the Tables I (a) to IV (b). Heat transfer coefficients calculated by Eq. (9) using the heat transfer data for each testpiece are plotted on logarithmic graph papers and presented in Figures (6), (7), (8), and (9).

The conventional expression of heat transfer coefficient in convection is

$$h = \alpha \frac{k}{D} \left(\frac{VD\rho}{\mu}\right)^m \left(\frac{C_p\mu}{k}\right)^n \quad \text{Btu/hr sq. ft. deg. F} \qquad (10)$$

The constants $\alpha$, m, and n have been specified by various investigators for different conditions. If all the terms except V in the right side of the Eq. (10) are kept constant then

$$h = C_1 V^m \quad \text{Btu/hr sq. ft. deg. F} \qquad (11)$$

would be obtained. Since V is directly proportional to the fluid flow rate W, Eq. (11) can be written as

$$h = C_2 W^m \quad \text{Btu/hr sq. ft. deg. F} \qquad (12)$$

TABLE NO. I(a)

HEAT TRANSFER DATA

TESTPIECE NO. 1: Smooth Steel Pipe 0.825" I. D., 1.05" O. D., 20" long, A = 0.36 sq. ft.

| Run No. | Flowmeter Reading | W lbs/hr | Water Temp. °F. $T_1$ | $T_2$ | $T_2$-$T_1$ | Pipe Temp. °F. $T_3$ | $T_4$ | $T_5$ | $T_w$ | $T_f$ °F | $\Delta T$ °F | h $\frac{Btu}{hr\ ft^2 °F}$ |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 1 | 20 | 550 | 113 | 123 | 10 | 170 | 212 | 227 | 205.25 | 118 | 87.25 | 175 |
| 2 | 40 | 719 | 119 | 126 | 7 | 172 | 201 | 213 | 196.75 | 122.5 | 74.25 | 188 |
| 3 | 60 | 878 | 119 | 127 | 8 | 181 | 210 | 225 | 206.5 | 123 | 83.5 | 234 |
| 4 | 80 | 1005 | 119.5 | 127.5 | 8 | 180 | 205 | 219 | 202.25 | 123.5 | 78.75 | 284 |
| 5 | 100 | 1112 | 120 | 128 | 8 | 187 | 210 | 228 | 208.75 | 124 | 84.75 | 292 |
| 6 | 120 | 1195 | 117 | 125 | 8 | 188 | 209 | 226 | 208 | 121 | 87 | 306 |
| 7 | 140 | 1285 | 122.5 | 130 | 7.5 | 191 | 209 | 227 | 209 | 126.25 | 82.75 | 324 |

TABLE NO. I(b)

HEAT TRANSFER DATA

TESTPIECE NO. 1: Smooth Steel Pipe 0.825" I. D., 1.05" O. D., 20" long, A = 0.36 sq. ft.

| Run No. | Flowmeter Reading | W lbs/hr | Water Temp. °F. $T_1$ | $T_2$ | $T_2-T_1$ | Pipe Temp. °F. $T_3$ | $T_4$ | $T_5$ | $T_w$ | $T_f$ °F. | $\Delta T$ °F. | h $\frac{\text{Btu}}{\text{hr ft}^2\,°\text{F.}}$ |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 1 | 20 | 550 | 102 | 112.5 | 10.5 | 166 | 210 | 227 | 203.25 | 107.25 | 96 | 167 |
| 2 | 40 | 719 | 118 | 126 | 8 | 175 | 205 | 219 | 201 | 122 | 79 | 202 |
| 3 | 60 | 878 | 115.5 | 122.5 | 7 | 175 | 202 | 216 | 198.75 | 119 | 79.75 | 214 |
| 4 | 80 | 1005 | 112 | 119 | 7 | 173 | 198 | 213 | 195.5 | 115.5 | 80 | 245 |
| 5 | 100 | 1112 | 113 | 121 | 8 | 182 | 206 | 225 | 204.75 | 117 | 87.75 | 282 |
| 6 | 120 | 1195 | 115 | 122.5 | 7.5 | 182 | 202 | 220 | 201.5 | 118.75 | 82.75 | 301 |
| 7 | 140 | 1285 | 117 | 124 | 7 | 182 | 201 | 219 | 200.75 | 120.5 | 80.25 | 312 |

TABLE NO. II(a)

HEAT TRANSFER DATA

TESTPIECE NO 2: Steel Pipe 0.825" I. D.; 1.05" O. D.; 20" long; inside threaded with a 7/8" 14 thread per inch tap, A = 0.36 sq. ft.

| Run No. | Flowmeter Reading | W lb/hr | Water Temp. °F. $T_1$ | $T_2$ | $T_2-T_1$ | Pipe Temp., °F. $T_3$ | $T_4$ | $T_5$ | $T_w$ | $T_f$ | ΔT °F. | h $\frac{Btu}{hr\ ft^2\ °F.}$ |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 1 | 20 | 550 | 119.5 | 130 | 10.5 | 183 | 212 | 233 | 210 | 124.75 | 85.25 | 188 |
| 2 | 40 | 719 | 123 | 132.5 | 9.5 | 179 | 194 | 211 | 194.5 | 127.75 | 66.75 | 284 |
| 3 | 60 | 878 | 124 | 132.5 | 8.5 | 177 | 187 | 202 | 188.25 | 128.25 | 60 | 346 |
| 4 | 80 | 1005 | 124.5 | 132.5 | 8 | 177 | 187 | 202 | 188.25 | 128.5 | 59.75 | 374 |
| 5 | 100 | 1112 | 125 | 132.5 | 7.5 | 178 | 185 | 200 | 187 | 128.75 | 58.25 | 398 |
| 6 | 120 | 1195 | 125 | 133 | 8 | 183 | 188 | 206 | 191.25 | 129 | 62.25 | 427 |
| 7 | 140 | 1285 | 125.5 | 133 | 7.5 | 183 | 188 | 206 | 191.25 | 129.25 | 62 | 432 |

# TABLE NO. II(b)

## HEAT TRANSFER DATA

TESTPIECE NO. 2: Steel Pipe 0.825" I. D., 1.05" O. D., 20" long, inside threaded with a 7/8" 14 thread per inch tap, A = 0.36 sq. ft.

| Run No. | Flowmeter Reading | W lb/hr | Water Temp. °F. $T_1$ | $T_2$ | $T_2-T_1$ | Pipe Temp., °F. $T_3$ | $T_4$ | $T_5$ | $T_w$ | $T_f$ | $\Delta T$ °F. | h $\frac{Btu}{hr\ ft^2\ °F.}$ |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 1 | 20 | 550 | 122.5 | 133 | 10.5 | 180 | 202 | 220 | 201 | 127.75 | 73.25 | 219 |
| 2 | 40 | 719 | 127 | 135 | 8 | 178 | 192 | 206 | 192 | 131 | 61 | 262 |
| 3 | 60 | 878 | 127 | 135 | 8 | 181 | 192 | 208 | 193.25 | 131 | 62.25 | 314 |
| 4 | 80 | 1005 | 127 | 135 | 8 | 180 | 190 | 205 | 191.25 | 131 | 60.25 | 371 |
| 5 | 100 | 1112 | 127.5 | 135 | 7.5 | 180 | 187 | 202 | 189.25 | 131.25 | 57.75 | 401 |
| 6 | 120 | 1195 | 127.5 | 135 | 7.5 | 182 | 190 | 205 | 191.75 | 131.25 | 60.5 | 412 |
| 7 | 140 | 1285 | 127.5 | 135 | 7.5 | 182 | 188 | 204 | 190.50 | 131.25 | 59.25 | 452 |

TABLE NO. III(a)

HEAT TRANSFER DATA

TESTPIECE NO. 3: Steel Pipe 0.825″ I. D., 1.05″ O. D., 20″ long, inside splined

$A = 0.36$ sq. ft.

| Run No. | Flowmeter Reading | W lb/hr | Water Temp. °F. $T_1$ | Water Temp. °F. $T_2$ | $T_2-T_1$ | Pipe Temp. °F. $T_3$ | Pipe Temp. °F. $T_4$ | Pipe Temp. °F. $T_5$ | Pipe Temp. °F. $T_w$ | $T_f$ °F | $\Delta T$ °F. | $\bar{h}$ $\frac{Btu}{hr\ ft^2\ °F.}$ |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 1 | 20 | 550 | 113 | 126 | 13 | 170 | 208 | 196 | 195.5 | 119.5 | 76 | 261 |
| 2 | 40 | 719 | 118 | 130 | 12 | 168 | 200 | 190 | 189.5 | 124 | 65.5 | 366 |
| 3 | 60 | 878 | 120 | 130.5 | 10.5 | 170 | 202 | 193 | 191.75 | 125.25 | 66.5 | 385 |
| 4 | 80 | 1005 | 119 | 128 | 9 | 167 | 195 | 187 | 186 | 123.5 | 62.5 | 402 |
| 5 | 100 | 1112 | 117 | 126 | 9 | 167 | 188 | 182 | 181.25 | 121.5 | 59.75 | 465 |
| 6 | 120 | 1195 | 117.5 | 126.5 | 9 | 165 | 192 | 183 | 183 | 122 | 61 | 490 |
| 7 | 140 | 1285 | 119.5 | 129.5 | 10 | 168 | 197 | 188 | 187.5 | 124.5 | 63 | 566 |

TABLE NO. III(b)

HEAT TRANSFER DATA

TESTPIECE NO. 3: Steel Pipe 0.825" I. D., 1.05" O. D., 20" long, inside splined

A = 0.36 sq. ft.

| Run No. | Flowmeter Reading | W lb/hr | Water Temp. °F. $T_1$ | $T_2$ | $T_2-T_1$ | Pipe Temp. °F. $T_3$ | $T_4$ | $T_5$ | $T_w$ | $T_f$ °F | ΔT °F. | h $\frac{\text{Btu}}{\text{hr ft}^2\text{°F.}}$ |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 1 | 20 | 550 | 100 | 113 | 13 | 160 | 200 | 191 | 187.75 | 106.5 | 81.25 | 244 |
| 2 | 40 | 719 | 120 | 132 | 12 | 173 | 207 | 199 | 196.5 | 126 | 70.5 | 340 |
| 3 | 60 | 878 | 121 | 132 | 11 | 172 | 203 | 195 | 193.25 | 126.5 | 66.75 | 402 |
| 4 | 80 | 1005 | 121.5 | 131.5 | 10 | 170 | 198 | 190 | 189 | 126.5 | 62.5 | 447 |
| 5 | 100 | 1112 | 121.5 | 130.5 | 9 | 168 | 195 | 188 | 186.5 | 126 | 60.5 | 459 |
| 6 | 120 | 1195 | 121.5 | 130 | 8.5 | 168 | 195 | 187 | 186.25 | 125.75 | 60.5 | 466 |
| 7 | 140 | 1285 | 120 | 129.5 | 9.5 | 169 | 196 | 188 | 187.25 | 124.75 | 62.5 | 543 |

TABLE IV(a)

HEAT TRANSFER DATA

TESTPIECE NO. 4: Steel Pipe, 0.825" I. D., 1.05" O. D., 20" long, inside threaded and splined as testpiece no. 2 and 3, A = 0.36 sq. ft.

| Run No. | Flowmeter Reading | W lb/hr | Water Temp. °F. $T_1$ | $T_2$ | $T_2-T_1$ | Pipe Temp. °F. $T_3$ | $T_4$ | $T_5$ | $T_w$ | $T_f$ °F | $\Delta T$ | h $\frac{Btu}{hr\ ft^2\ °F.}$ |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 1 | 20 | 550 | 116.5 | 127 | 10.5 | 197 | 219 | 233 | 217 | 122.25 | 94.75 | 170 |
| 2 | 40 | 719 | 120 | 130.5 | 10.5 | 195 | 206 | 221 | 207 | 125.25 | 81.75 | 257 |
| 3 | 60 | 878 | 120 | 131 | 11 | 196 | 204 | 221 | 206.25 | 125.5 | 80.75 | 332 |
| 4 | 80 | 1005 | 120 | 130 | 10 | 196 | 200 | 215 | 202.75 | 125 | 77.75 | 359 |
| 5 | 100 | 1112 | 119.5 | 128 | 8.5 | 196 | 197 | 213 | 200.75 | 123.75 | 77 | 343 |
| 6 | 120 | 1195 | 122 | 130 | 8 | 199 | 199 | 214 | 202.75 | 126 | 76.75 | 346 |
| 7 | 140 | 1285 | 122 | 130 | 8 | 200 | 200 | 213 | 203.25 | 126 | 77.25 | 370 |

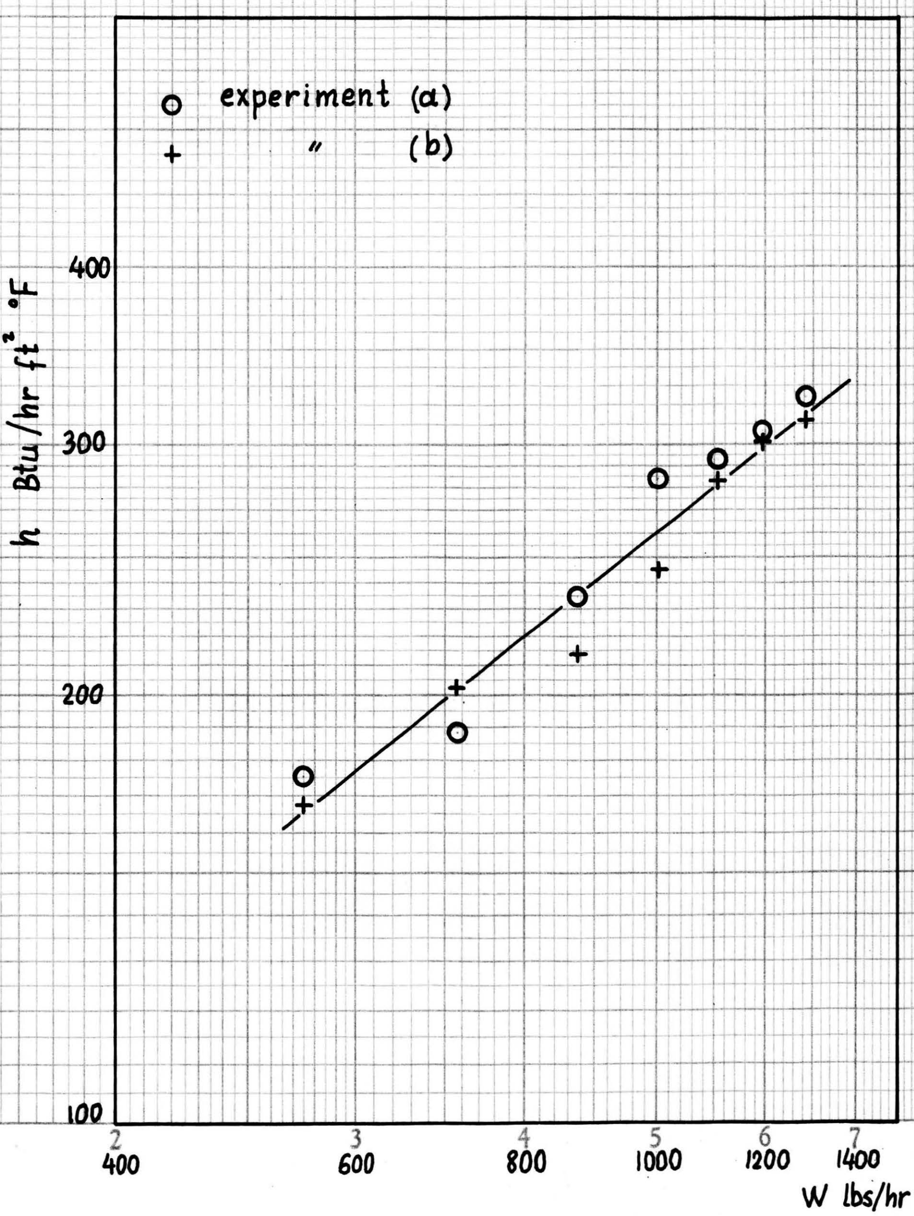
TABLE IV(b)

HEAT TRANSFER DATA

TESTPIECE NO. 4: Steel Pipe, 0.825",I. D., 1.05" O. D., 20" long, inside threaded and splined as testpiece no. 2 and 3, A = 0.36 sq. ft.

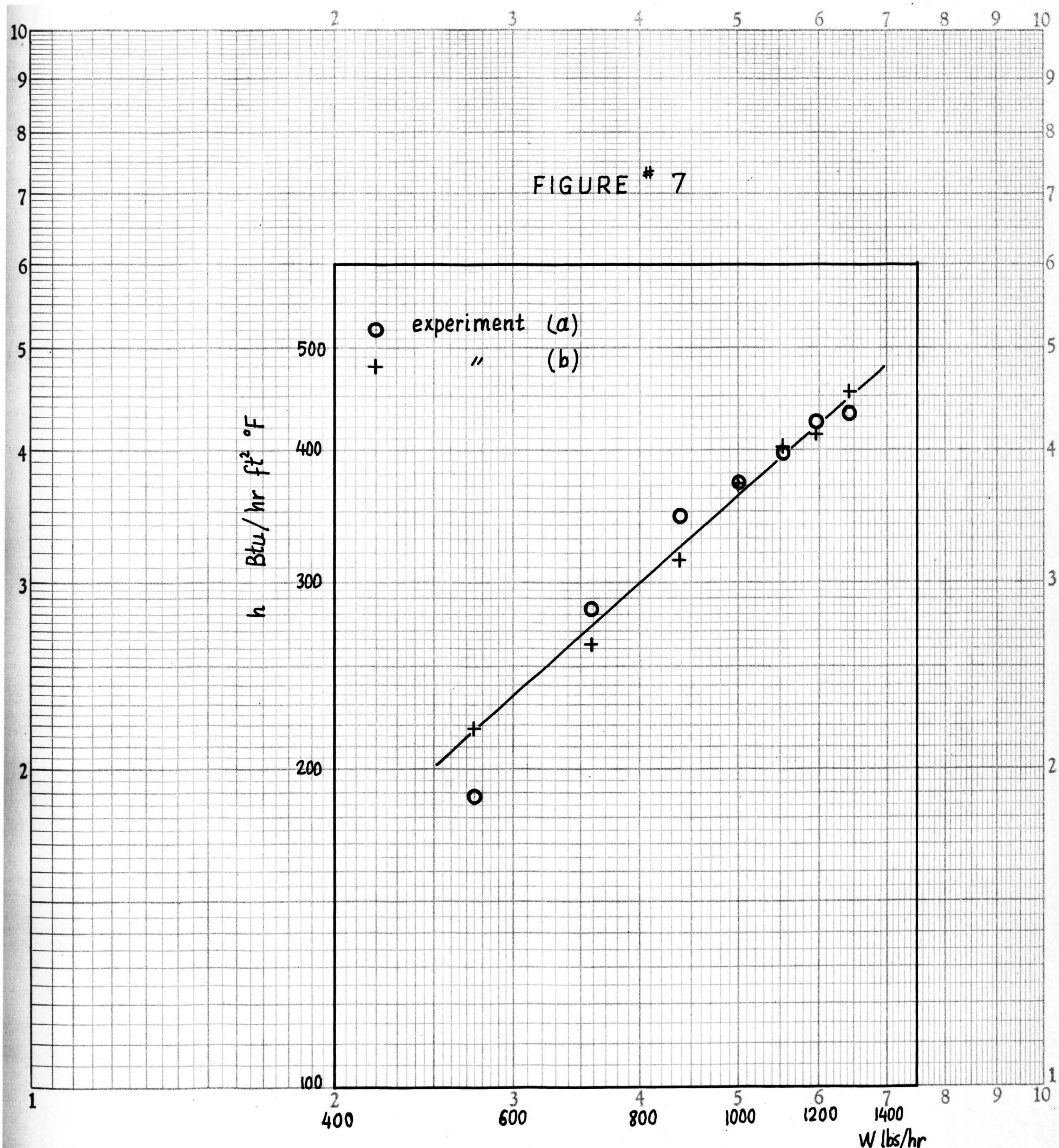
| Run No. | Flowmeter Reading | W lb/hr | Water Temp. °F. $T_1$ | $T_2$ | $T_2$-$T_1$ | Pipe Temp. °F. $T_3$ | $T_4$ | $T_5$ | $T_w$ | $T_f$ °F | ΔT | h Btu/hr ft$^2$°F. |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 1 | 20 | 550 | 108 | 120 | 12 | 178 | 193 | 203 | 191.75 | 114 | 77.75 | 234 |
| 2 | 40 | 719 | 121 | 132 | 11 | 191 | 198 | 210 | 199.25 | 126.5 | 72.75 | 302 |
| 3 | 60 | 878 | 122.5 | 131.5 | 9 | 188 | 190 | 203 | 192.75 | 127 | 65.75 | 334 |
| 4 | 80 | 1005 | 122.5 | 130 | 7.5 | 185 | 182 | 197 | 186.5 | 126.25 | 60.25 | 347 |
| 5 | 100 | 1112 | 118 | 125 | 7 | 185 | 175 | 193 | 182 | 121.5 | 60.5 | 357 |
| 6 | 120 | 1195 | 121 | 127.5 | 6.5 | 179 | 178 | 194 | 182.25 | 124.25 | 58 | 372 |
| 7 | 140 | 1285 | 121 | 128 | 7 | 187 | 183 | 200 | 188.25 | 124.5 | 63.75 | 395 |

FIGURE # 6



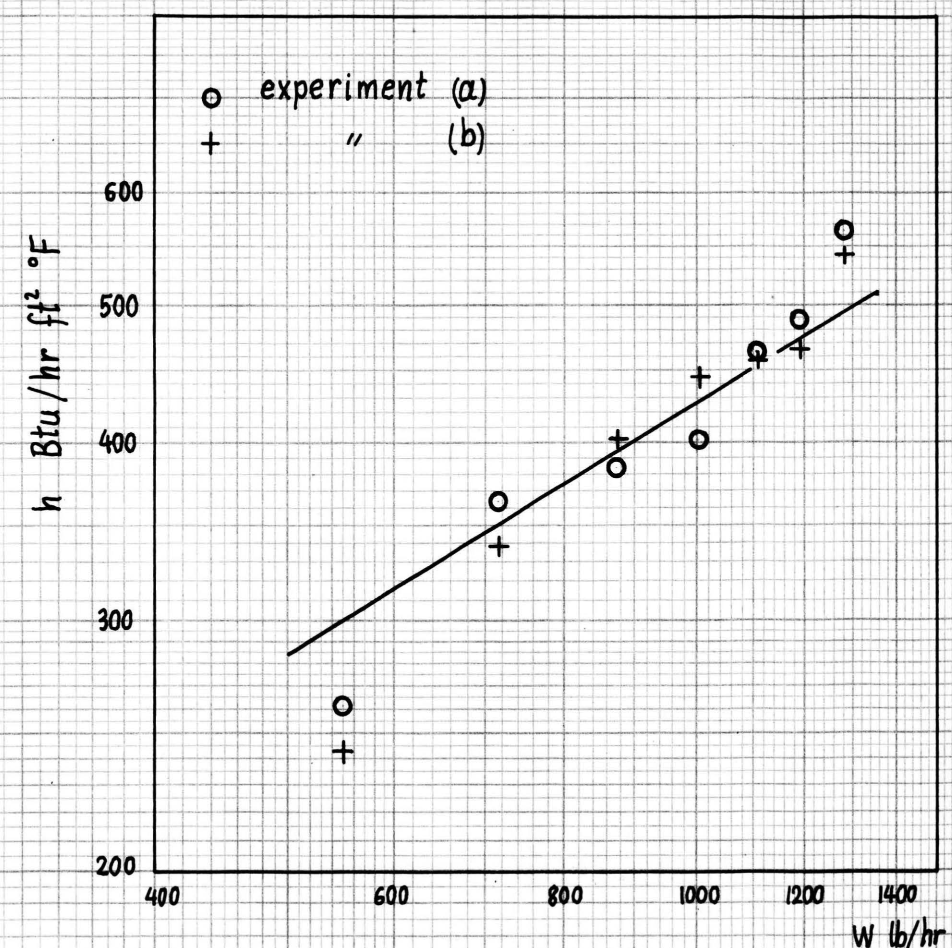

HEAT TRANSFER COEFFICIENTS IN TESTPIECE # 1

FIGURE # 7



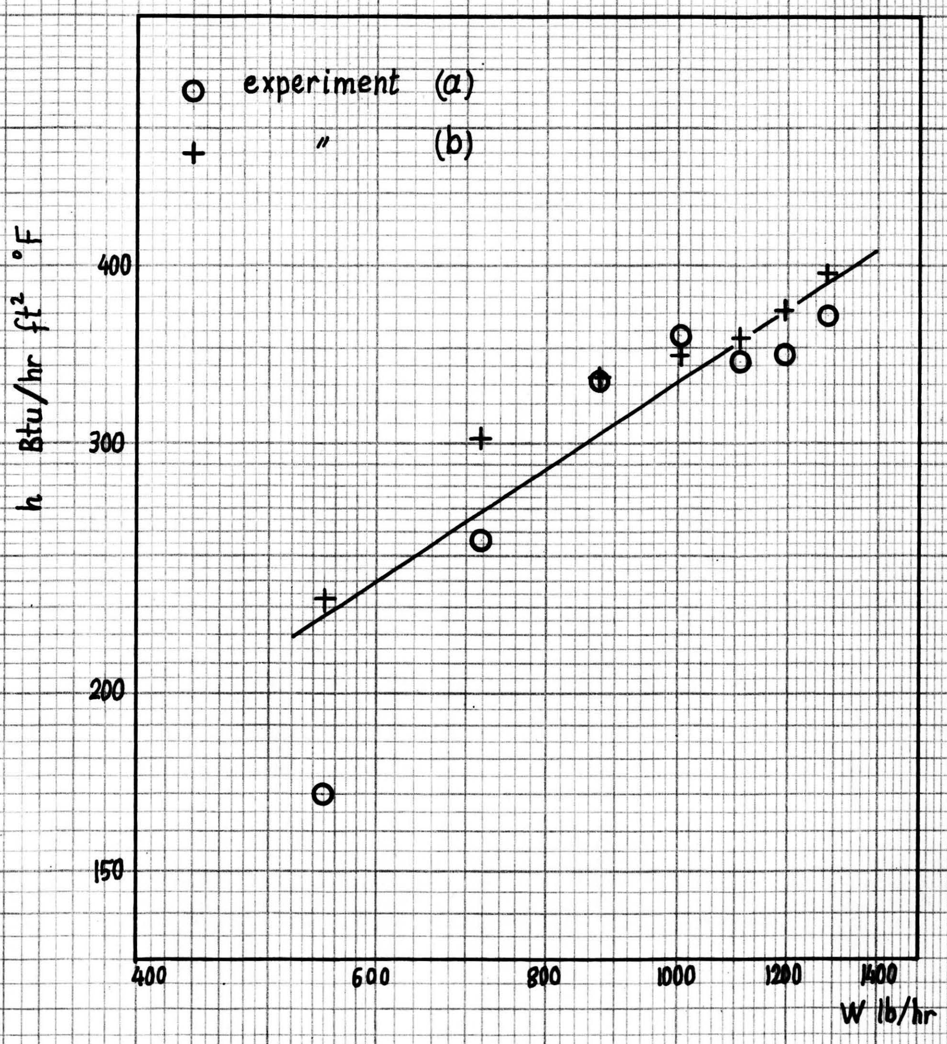

HEAT TRANSFER COEFFICIENTS IN TESTPIECE #2

FIGURE # 8




HEAT TRANSFER COEFFICIENTS IN TESTPIECE # 3

FIGURE # 9




HEAT TRANSFER COEFFICIENTS IN TESTPIECE # 4

Taking the logarithm of each side

$$\log h = \log C_2 + m\log W \qquad (13)$$

is obtained. On logarithmic coordinates[5] this reduces to an equation of the form

$$h = C_2 + mW \qquad (14)$$

which is the expression of a straight line.

It was mentioned before in the section "Range of Measurements" that bulk temperatures of water changed between 114 and 126.5 degrees Fahrenheit and the corresponding Prandtl numbers differed only about 10%. Absolute viscosity $\mu$ showed a variation of about 10% and thermal conductivity about 1.3% for the above mentioned temperature difference. $C_p$ and $\rho$ can be assumed constant between 114 and 126.5 degrees Fahrenheit. From these considerations it is seen that heat transfer coefficients can be plotted versus water flow rates on a logarithmic coordinate for the temperature ranges dealt in this thesis.

To make it easier to compare the heat transfer coefficients for all the testpieces they are tabulated in Table V and plotted in Figure (10).

6. ACCURACY OF RESULTS

Throughout the experiments and calculations the assumptions listed below are made:

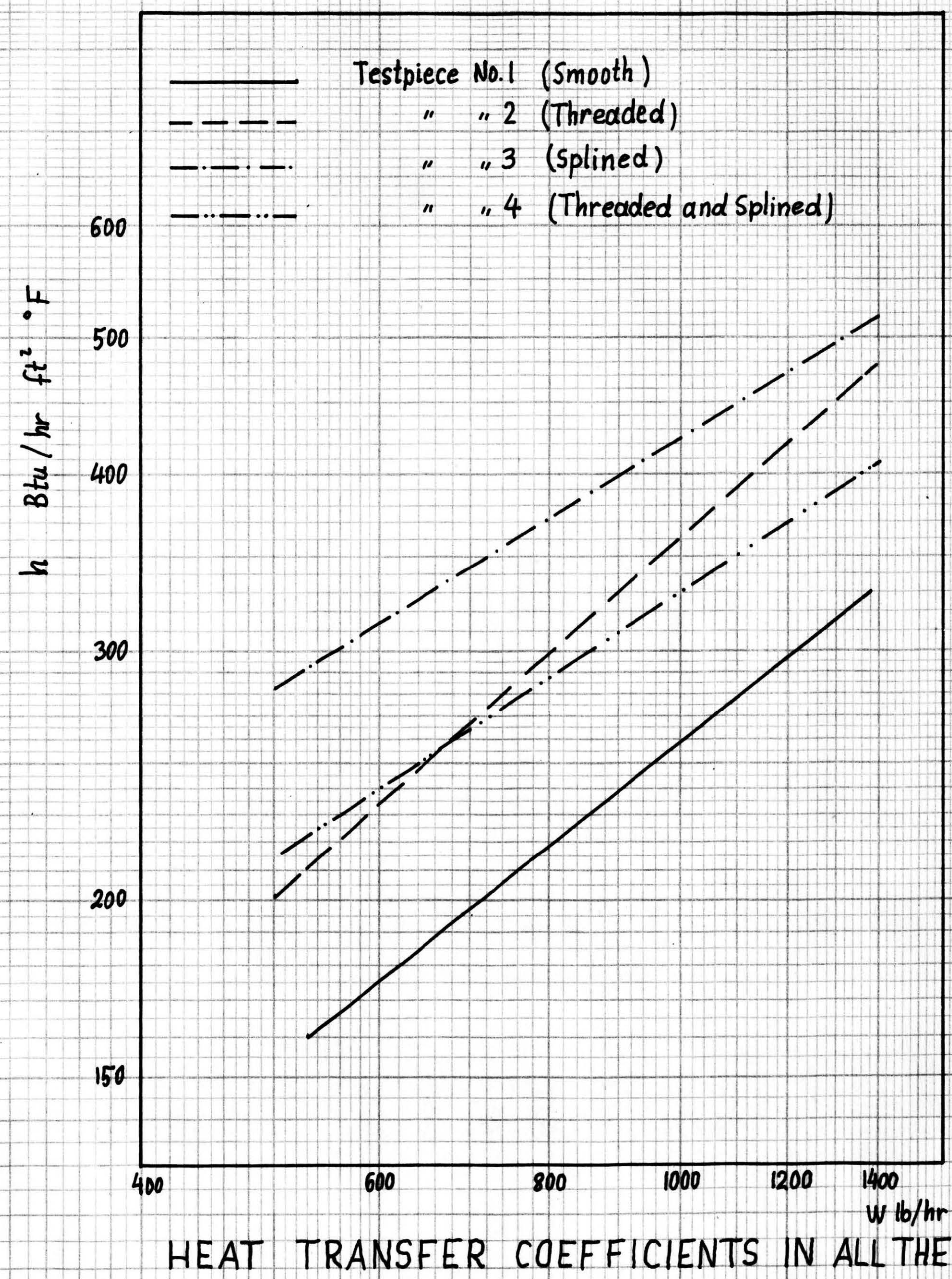
1. The temperature distribution along the testpiece is uniform.

TABLE V

HEAT TRANSFER COEFFICIENTS FOR ALL THE TESTPIECES AND RUNS

| Run No. | TESTPIECE No. 1 Smooth Pipe | | TESTPIECE No. 2 Inside Threaded Pipe | | TESTPIECE No. 3 Inside Splined Pipe | | TESTPIECE No. 4 Inside Threaded and Splined Pipe | |
|---|---|---|---|---|---|---|---|---|
| | Experiment a | Experiment b | Experiment a | Experiment b | Experiment a | Experiment b | Experiment a | Experiment b |
| 1 | 175 | 167 | 188 | 219 | 261 | 244 | 170 | 234 |
| 2 | 188 | 202 | 284 | 262 | 366 | 340 | 257 | 302 |
| 3 | 234 | 214 | 346 | 314 | 385 | 402 | 332 | 334 |
| 4 | 284 | 245 | 374 | 371 | 402 | 447 | 359 | 347 |
| 5 | 292 | 282 | 398 | 401 | 465 | 459 | 343 | 357 |
| 6 | 306 | 301 | 427 | 412 | 490 | 466 | 346 | 372 |
| 7 | 324 | 312 | 432 | 452 | 566 | 543 | 370 | 395 |

FIGURE # 10




HEAT TRANSFER COEFFICIENTS IN ALL THE TESTPIECES

2. The temperature difference between the outside and inside surface of the testpiece is negligible because heating is achieved by internal heat generation.

3. Heat loss from the testpiece to air by conduction, convection, and radiation: to the inlet and outlet water pipes by conduction and from the outlet pipe to air by conduction, convection and radiation along the length from testpiece to the point where the outlet water temperature thermocouple is located are negligible.

4. The measured inlet and outlet water temperatures represent the bulk temperatures at inlet and outlet.

5. Within the range of temperatures dealt, the absolute viscosity, the thermal conductivity, and the density of water are constant; the specific heat of water is constant and is equal to 1 Btu/lb deg. F.

6. There are no partial phase changes along the path.

7. The heat transfer coefficient h is constant over the entire length of the testpiece.

8. Steady state condition is reached throughout the experiments and water flow rate is constant for each run.

9. The heat transfer surface A is equal to 0.36 sq. ft. for all the testpieces.

10. Regardless of the end portions of the testpiece used for connection, the length of the testpiece is 20".

The accuracy of the thermocouples used is given by the manufacturer as $\pm$ 4 degrees Fahrenheit between 0 and 530 degrees Fahrenheit. The accuracy of the Wheelco Temperature Recorder is $\frac{1}{4}$ of 1% full scale range. Since the full scale range is 1500 degrees Fahrenheit, the accuracy becomes ± 3.75 degrees Fahrenheit. Considering these and the assumptions made it is estimated that the accuracy of the results is within ± 15%. The character of this thesis is more qualitative than quantitive. Therefore ± 15% accuracy can be assumed satisfactory from this point of view.

## 7. DISCUSSION OF RESULTS

As it was expected, from the Figure (10) it is seen that all the artificially roughened testpieces give higher heat transfer coefficients than the smooth pipe in the range of measurements of this investigation. The coefficient values for the splined testpiece are the highest. They are higher than the smooth pipe by 82% at the low and 57% at the high water flow rates. The difference is more pronounced at low flow rates. The second highest heat transfer coefficient line belongs to the threaded pipe. It gives h values greater than the smooth pipe by 32% at low and 43% at high flow rates. The difference is more pronounced at high water flow rates, i.e., at high Reynolds numbers. Threaded and splined pipe (Testpiece No. 4) gives heat transfer coefficients lower than the threaded pipe in a greater portion of water flow range, but gives higher values than the smooth pipe. The difference of the heat transfer coefficient values between the Testpiece No. 4 and smooth pipe is 38% at low and 23% at high flow rates. From this it can be predicted that a certain Reynolds number the value of heat transfer coefficient will be the same for the smooth pipe and the threaded-splined pipe.

The explanation of the fact that the splined pipe gives higher heat transfer coefficients than the threaded pipe can be made in this way; although in both cases the roughness ratio e/D is the same, the shape of roughness is entirely different. In the splined pipe, the form of the laminar boundary layer is quite uniform. Its thickness is almost constant and not considerably thicker than that it is in a smooth pipe at any point in the inner surface of the splined pipe. The longitudinal grooves essentially does not disturb the form of the boundary layer, although it is possible that at the roots of the grooves it may be slightly thicker than that at any other location. By this uniformity all the inner surface of the pipe being extended by the grooves is a uniform heat transferring surface. At every point heat transfer takes place almost uniformly. The increased turbulence helps to transfer heat from boundary and transition layers into the core of the flow by mixing action. In the case of threaded pipe, the roughness, namely, the threads are perpendicular to the flow of water, so that in the V shaped grooves there is almost stagnant water. It is mostly probable that small air bubbles can be trapped at the roots of these grooves. With considerably thick stagnant water layer, the thickness of which being equal to the depth of the thread at the center of the groove, and with air bubbles the inner surface of the pipe does not have so much chance as the splined pipe has to transfer heat to the water flow. In this case also increased turbulence helps to transfer heat from laminar and transition layers into the core of the flow by mixing action.

In the threaded-splined pipe the effect of longitudinal grooves predominates at low water flow rates, so that heat transfer coefficient is higher than the threaded pipe, but as the Reynolds number is increased the combined effect of two types of roughness is such that the value of h becomes lowest among the roughened testpieces. One interesting point observed is that the heat transfer coefficient line for the splined pipe is almost parallel to that for the threaded-splined one, the former being higher about 30%. This leads to the possibility that the effect of the threads in reducing the heat transfer coefficient in a splined-threaded pipe is a constant.

# IV. CONCLUSIONS

Artificially roughened pipes give higher heat transfer coefficients than the smooth pipes of same diameter under the same heat transfer conditions. Although the roughness ratio e/D is important in defining the degree of roughness, the type, shape, and distribution of the roughness are as important as e/D as far as the heat transfer coefficient is concerned. For the same roughness ratio different types of roughness give different heat transfer coefficients. A pipe with longitudinal grooves at the inner surface gives higher values of h than a pipe with the threads which have the same depth as the grooves. Heat transfer coefficient in the former can be as high as 182% or more of the smooth pipe of the same diameter at a Reynolds number of 7500. The latter gives h values 43% higher than the smooth pipes at a Reynolds number of 17500. The combination of threads and longitudinal grooves give higher heat transfer coefficients than smooth pipes, but less than splined and threaded pipes. The type of roughness which does not disturb the form of boundary layer gives highest heat transfer coefficient. The types which create stagnant fluid or air pockets and disturb the form of the laminar boundary layer give lesser values of h. The best type of roughness to obtain the highest heat transfer coefficient for a given degree of roughness appears to be one which serves to increase the heat transfer surface, yet it does not disturb the boundary layer or does not increase the thickness of stagnant layer.

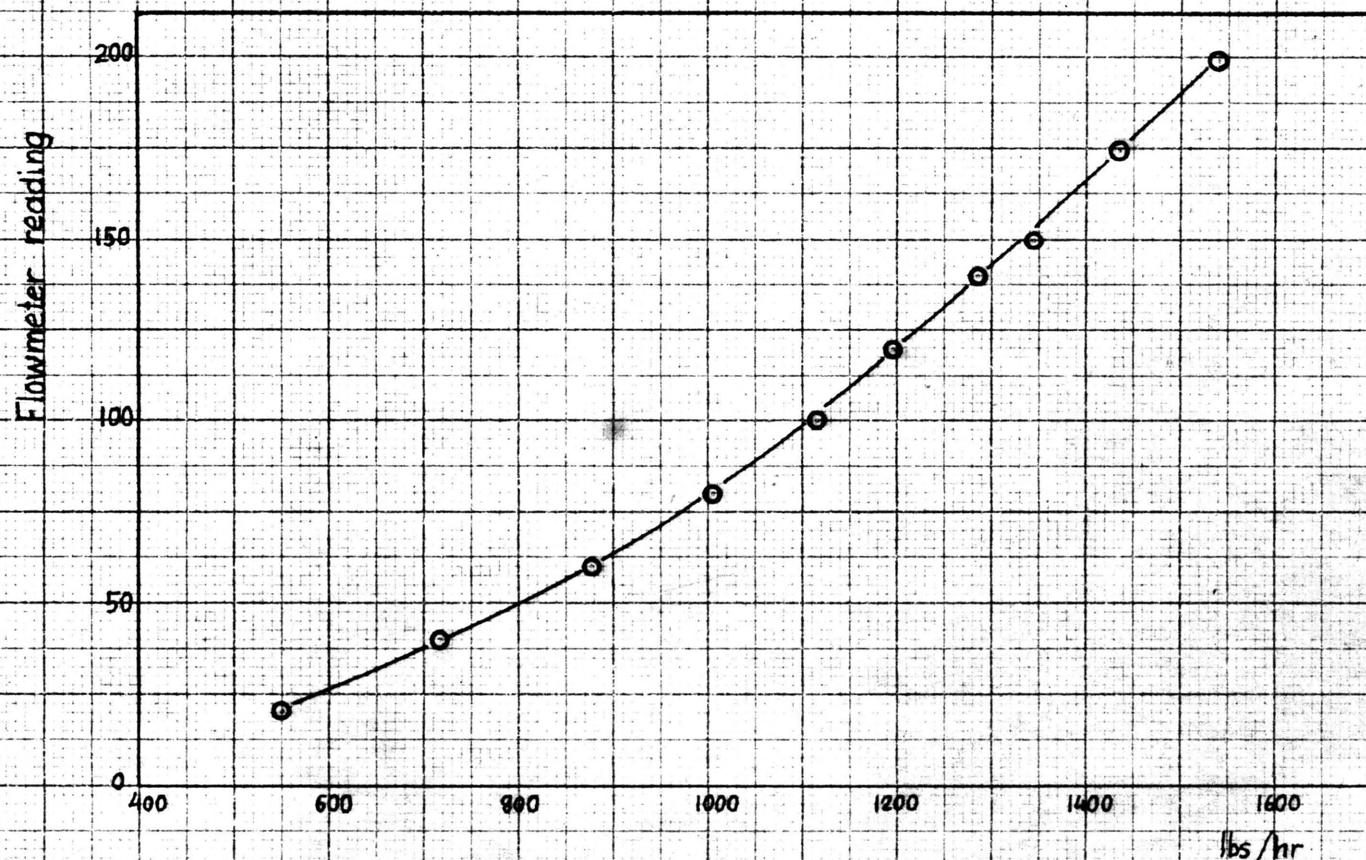
It is believed that to understand more fully the effect of roughness ratio and type of roughness on the heat transfer coefficient further investigations with different roughness ratios and types of roughness are necessary.

## V. SUMMARY

To determine the effect of different types of roughness on heat transfer coefficient in a pipe, experiments were performed on 4 testpieces. Testpiece No. 1 was a smooth 3/4" 20" long pipe, No. 2 a 3/4" 20" long inside threaded with a 7/8" 14 thread per inch tap, No. 3 the same pipe splined, 36 grooves having the same depth as the threads (0.0275"), and No. 4 a combination of No. 2 and No. 3. While the testpiece was being heated by induction, water was passed through it. Water flow rates were ranged from 550 lb/hr to 1285 lb/hr. The bulk temperature of the water was maintained between 114 and 126.5 degrees Fahrenheit throughout the experiments. Heat transfer coefficients for the splined pipe were the highest. The threaded pipe showed lesser heat transfer coefficients, but they were considerably higher than that of the smooth pipe. The threaded-splined pipe had heat transfer coefficients less than that of threaded pipe, but higher than that of the smooth pipe. It appears that the type of roughness which does not disturb the form of laminar boundary layer and does not increase the thickness of it, without causing stagnant water and air pockets provides the highest heat transfer coefficient.

It is recommended that further investigations be made to understand more fully the effect of different roughness ratios and different types of roughness on the heat transfer coefficient in a pipe.

APPENDIX I




CALIBRATION CURVE FOR HONEYWELL FLOWMETER

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# VITA

Mehmet Hayati Turan, son of Mr. and Mrs. Vuslat Turan, was born in Samsum, Turkey on November 2, 1932. Following graduation from Samsun Technical High School in 1949, he entered Technical School in Istanbul, Turkey. In June, 1953, he received his Bachelor of Science degree in Mechanical Engineering from this school. Upon the completion of his active military service of one and one half years in the Turkish Army he was honorably discharged in April, 1955. He worked in the constructions of two hydroelectric projects in Turkey until he entered Missouri School of Mines and Metallurgy in October, 1959 to study for the degree of Master of Science in Mechanical Engineering.