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Linear and vortex flow heat transfer coefficients

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LINEAR AND VORTEX FLOW

HEAT TRANSFER COEFFICIENTS

BY

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A

THESIS

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INTRODUCTION

In order to make possible better, more efficient equipment, a higher and higher rate of heat transfer has been the goal of engineers for many years. The need for heat transfer rates never thought of 25 years ago has been brought about by the advent of modern high-performance devices such as nuclear reactors and rocket motors. "A heat release of about 40,000 Btu/hr.cu.ft. is considered good practice in a modern boiler, but in a rocket or a nuclear reactor it may be 1,000,000,000 Btu/hr.cu.ft." (1) This heat must be removed by transfer to a coolant or converted into work or the device will fail since in these devices the heat release is independent of the heat removal rate.

When heat is applied to a container containing a fluid such as water, the fluid adjacent to the surface is heated and then replaced by the colder, less dense fluid from the top of the container. In this manner, convection currents are created and heat is transferred to all parts of the fluid. This gravitational effect is called natural convection. If the surface temperature is greater than the saturation temperature of the fluid, bubbles of vapor form, and the convection current, and thereby, the heat transfer rate increases greatly due to the large difference between the density of the vapor and the fluid. This process, when the heating surface is hotter than the saturation temperature but the bulk fluid temperature is less than the saturation temperature, is called subcooled nucleate boiling or

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(1) All references are in bibliography.

local boiling. If the gravitational effect causing natural convection can be increased many times over and combined with the turbulent nature of forced convection, an increase in heat transfer should be realized.

The multiplication of the gravity effect in natural convection can be achieved as done in a centrifuge, by causing the fluid to move in a curved path, thereby creating centripetal acceleration many times greater than gravitational acceleration. The resulting transport of the hot fluid from the heating surface and the replacement by the colder fluid should cause an increase in the heat transfer coefficient.

The curved flow required can be generated by forcing the water to flow in a helical path down a test section from which heat is transferred to the water. Fluid flow with combined axial and tangential components of velocity is termed source vortex or simply vortex flow.

It is the purpose of this investigation (1) to design and build apparatus for the determination of the heat transfer film coefficient when using an induction heater to generate heat in the test section, (2) to check the accuracy of the measurements by determination of the film coefficients for linear flow, the values of which are quite well known and (3) to make preliminary evaluation of the effects of vortex flow on heat transfer.

LITERATURE SURVEY

Any of the various heat transfer textbooks provide background material on heat transfer to non-boiling water flowing in tubes. McAdams (2) summarizes the principle convective heat transfer correlation equations, and two recent books by Bonilla (3) and Kreith (1) give much of the new information which has been found in recent years about boiling heat transfer. Rohsenow (4) gives an excellent discussion on all phases of boiling heat transfer. The theory behind and a description of the apparatus used at Argonne National Laboratory to measure straight flow water film coefficients is given by Rohde (5).

Literature on heat transfer to water in vortex flow is meager. Burnout of tubes with water boiling in vortex flow has been studied by Gambill and Greene (6) (7). They report a maximum attained burnout heat flux of about 50 x 10^6 Btu/hr.sq.ft., which is several times larger than the usual value for linear flow. They used spiral ramp and tangential slot vortex generators to produce the swirl in small diameter short tubes.

Kreith (8) (9) has studied analytically and experimentally the effect of curvature on the film coefficient for non-boiling heat transfer. The studies were made for flow in curved channels with radii of an inch and a half and greater. He found a 25 to 60% increase in the Nussault number for a concave surface over that for a convex surface at the same fluid velocity.

Kreith and Margolis (10) have studied the heat transfer to swirling air and non-boiling water. The swirl was introduced by twisted strips and coiled wires in tubes of about one half inch and one inch in diameter. They report nearly a fourfold increase in the Nussault number for swirl flow of water over that for linear flow at the same mass flow rate.

Several studies of curved fluid flow without heat transfer have been made particularly in connection with investigations of the Ranque-Hilsch energy separation effect in compressible fluid flow. The latest include the analytical and experimental study of Lay (11) and the analytical study of Deissler and Perlmutter (12). Treatments of curved incompressible fluid flow are those by Marshall (13), Wattendorf (14), Einstein and Li (15) and Eskinazi and Yeh (16).

APPARATUS

The apparatus constructed and used by the author was designed to permit determination of the heat transfer film coefficient for linear flow as a check on the feasibility of using the induction field as a heat source and to make preliminary evaluation of the effect of vortex flow on the film coefficient. The apparatus consisted of a tangential orifice vortex generator designed with removable plug for linear flow tests situated upstream from a steel pipe test section. The water used in the tests flowed out of the generator, through the test section, into a mixing chamber for temperature measurement, and then into a weighing tank. The coil from the 9600 cycle per second induction heater was located around the center portion of the test section. Figure 1 shows a schematic diagram of the equipment and Figures 2, 3, and 4 are general views of the apparatus located in the Mechanical Engineering Laboratory.

VORTEX GENERATOR

Several different methods of generating the vortex flow were investigated. A twisted steel strip placed down the axis of the test section was considered initially. Since an induction field was to be used as a heat source and there would be heat generation in the strip proportional roughly to its volume, it was decided to investigate the tangential orifice type of vortex generator. Here the vortex is created by injecting the water perpendicular to the test section axis from several orifices tangent to the test section wall.

Several generators were initially fabricated from polystyrene for ease of machining and to enable the characteristics of flow to be

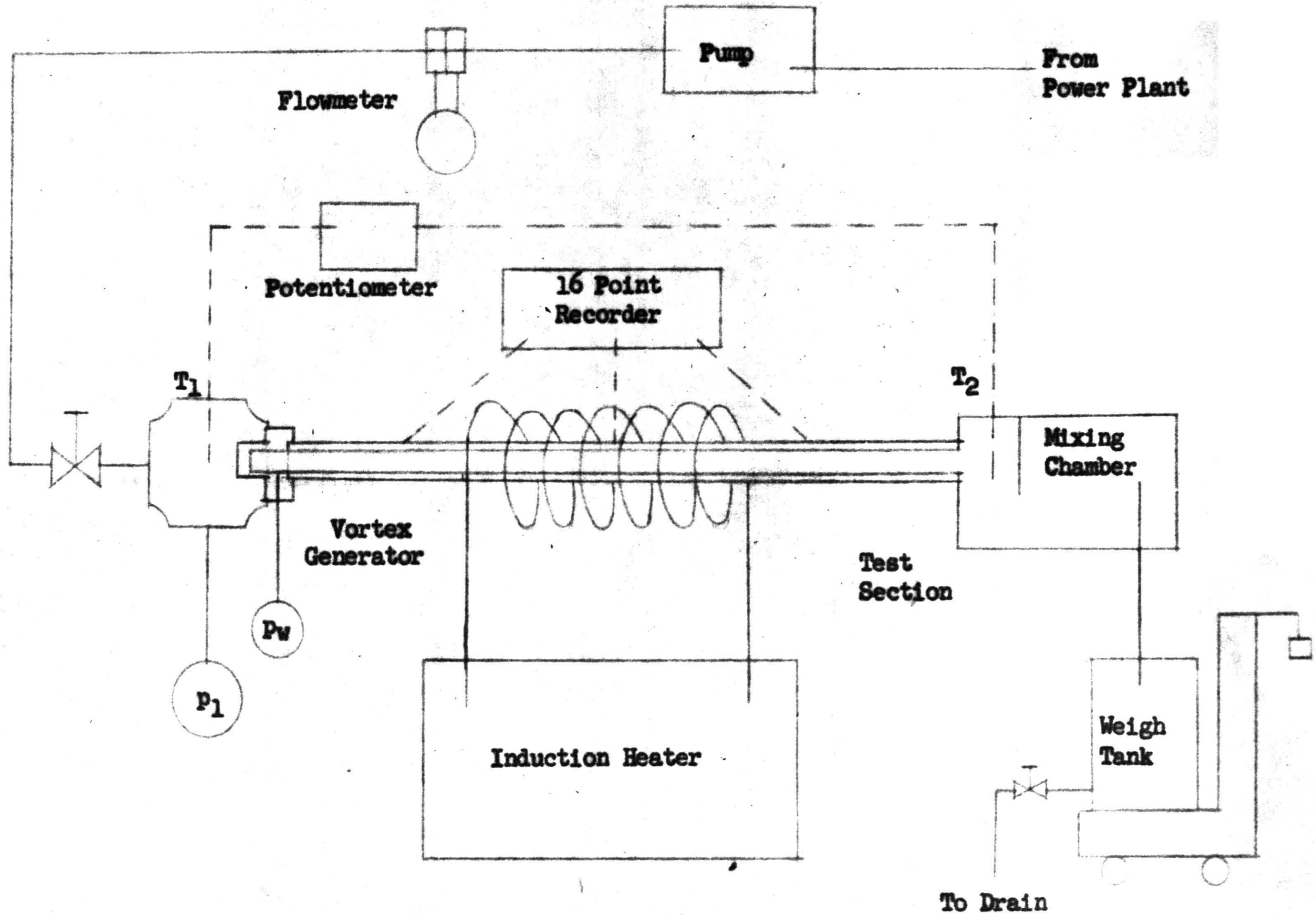

FIGURE 1

SCHEMATIC DIAGRAM OF EXPERIMENTAL EQUIPMENT

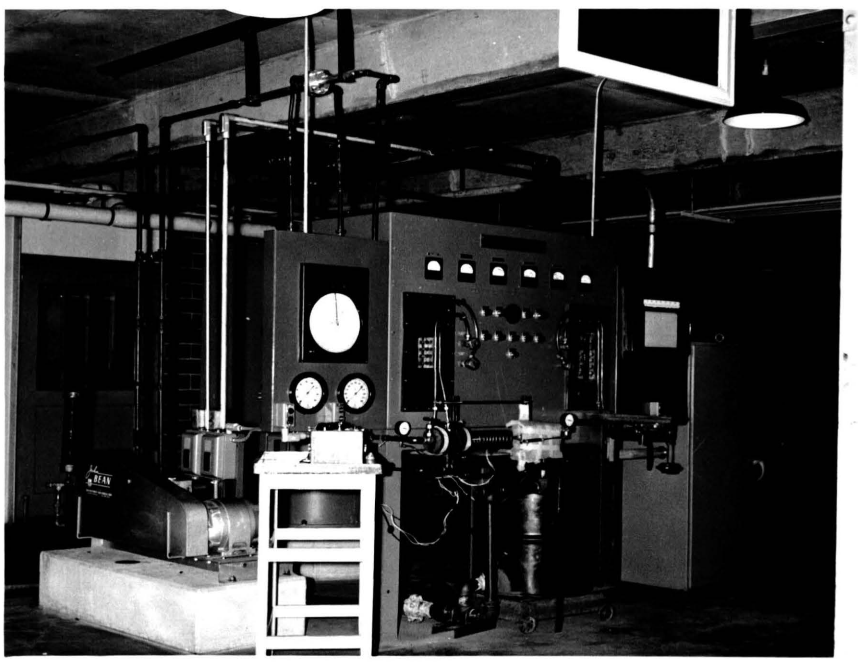

FIGURE 2

EXPERIMENTAL APPARATUS

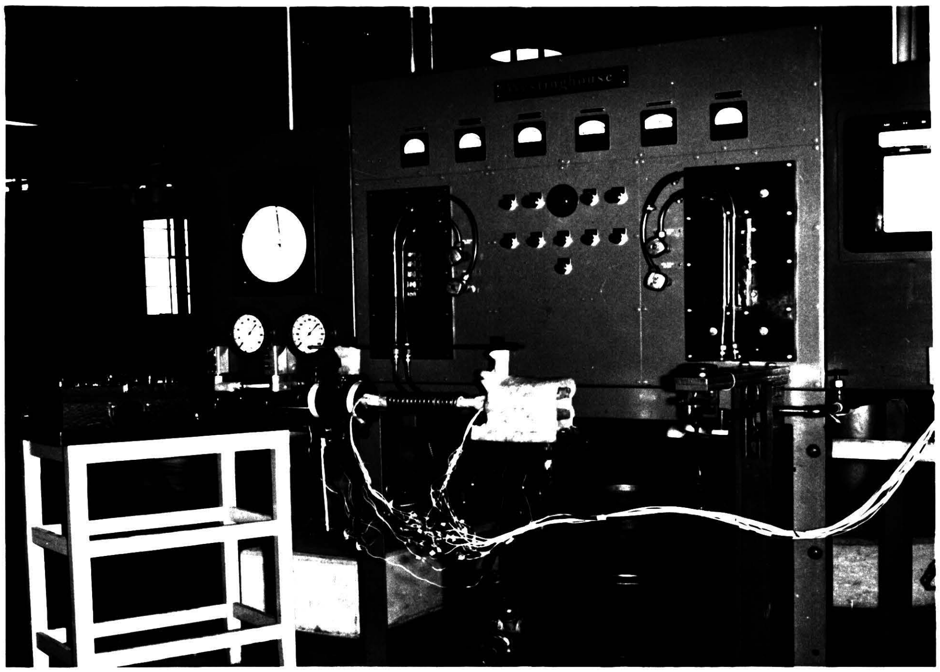

FIGURE 3

EXPERIMENTAL APPARATUS

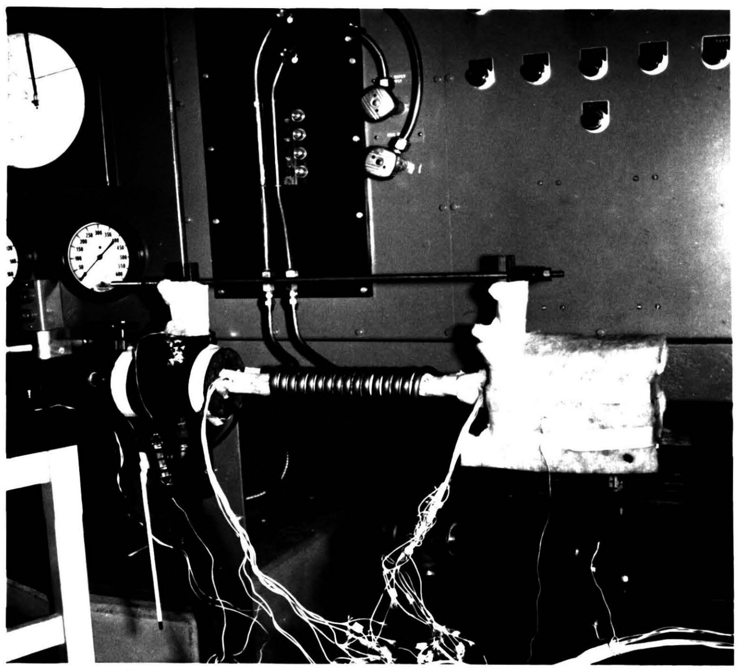

FIGURE 4

COIL AND INSULATED INLET FITTING,
TEST SECTION AND MIXING CHAMBER

observed. The stream was directed down horizontal glass tubing of lengths varying from 8 to 24 inches and then exhausted to the atmosphere. A considerable number of isothermal tests were made to ascertain the required orifice size, flow rate and tube length to produce a stable vortex of sufficient length.

It was found that at low flow rates, a core of air extending several inches downstream from the generator formed on the tube axis. This core was extended periodically by pulsations of air traveling up the axis of the tube from the discharge end. As the flow increased the frequency of pulsations increased until at a certain flow rate, determined by the length of tube and size of orifices, the flow stabilized and the air core was continuous over the entire length of the tube. The air core water interface appeared much like a screw thread and had a pitch of about 3/4 inch.

The generator used in the heat transfer tests was made from 1040 steel according to the design given in Figure 5. This design is essentially the same as that of the initial polystyrene models. The plug at the inlet end was removed for the linear flow tests.

TEST SECTION

The final test section used was a piece of 1/4 inch steel pipe 19 inches long. It was held in the vortex generator and in the mixing chamber by "O" ring seals and then the three portions drawn together with spring loaded tie rods. After installation of the test section thermocouples, it was covered with an inch of fiberglass insulation.

Initially a length of stainless steel tubing supported between two pieces of glass was used for a test section, but because of the lower magnetic permeability of the stainless steel, this had to be

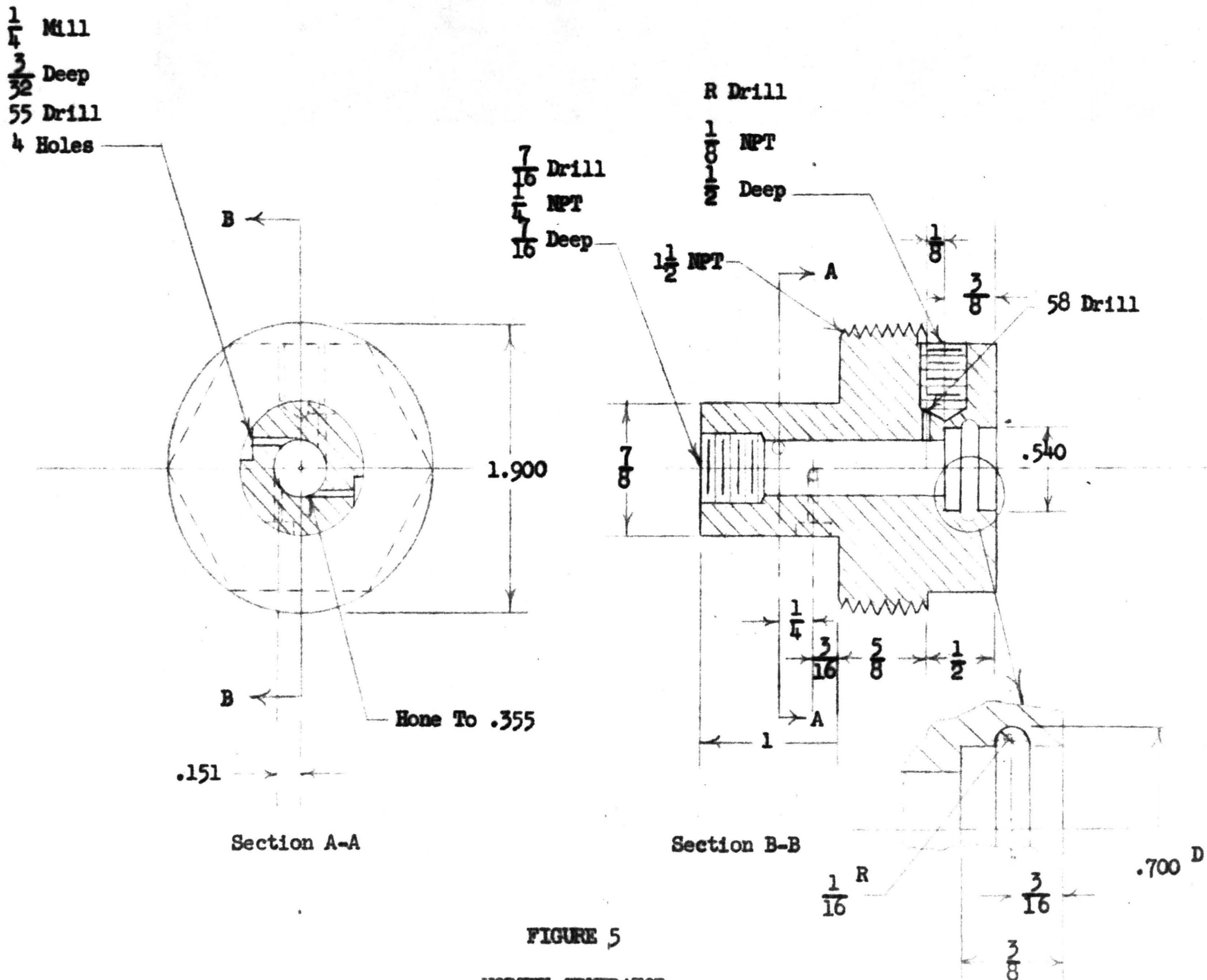

FIGURE 5

VORTEX GENERATOR

discarded. The reasons for this are given below.

SURFACE TEMPERATURE MEASUREMENT

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 independent 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 potentiometer are eliminated.

Because of the low magnetic permeability of the stainless steel test section originally used, a large magnetic flux had to be used and this caused the iron wire of the iron-constantan thermocouple wires to attain a temperature greater than the one attempting to be measured. When a mild steel test section was used, the ratio of the heating in the thermocouple wires to the heating in the test section was reduced and the couple output was a true indication of the temperature.

The outputs of the 24 gauge iron-constantan thermocouples were measured by a 16 point automatic recorder. Initially there was considerable oscillation of the recorder when heat was applied with the induction field. This effect was eliminated by connecting a 2 mfd capacitor across the thermocouple input of the recorder, by grounding the iron side of the thermocouple and by grounding the test section.

It was found that the method of application of the thermocouples to the test section surface was critical mainly because of the consistency of readings from one to another under isothermal conditions. The most reliable method found and the one used was to apply the thermocouple with a small amount of silver solder. The 16 thermocouples were located at one inch intervals as shown in Figure 6. After silver soldering, the test section was covered with a layer of high temperature glass tape; the thermocouple wires were wrapped around the circumference of the test section several times and then additional layers of glass tape applied.

WATER TEMPERATURE MEASUREMENT

The process water temperature was measured at the inlet and outlet of the test section by calibrated 24 gauge iron-constantan thermocouples imbedded in the bottom of two thermocouple wells. The construction of these wells is shown in Figure 7. The inlet temperature was measured in the fitting supplying the vortex generator, and the outlet temperature was measured in the outlet mixing chamber. Both the outlet and inlet fittings were covered with several inches of insulation to reduce heat loss. The thermocouple output was measured with a portable precision potentiometer.

PRESSURE MEASUREMENT

The inlet water pressure was measured with a new Bourdon tube type pressure gauge. A new 0 to 60 psi gauge was used on the pressure tap of the vortex generator to obtain an idea of the test section wall pressure.

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. The water was condensate returning from the heating

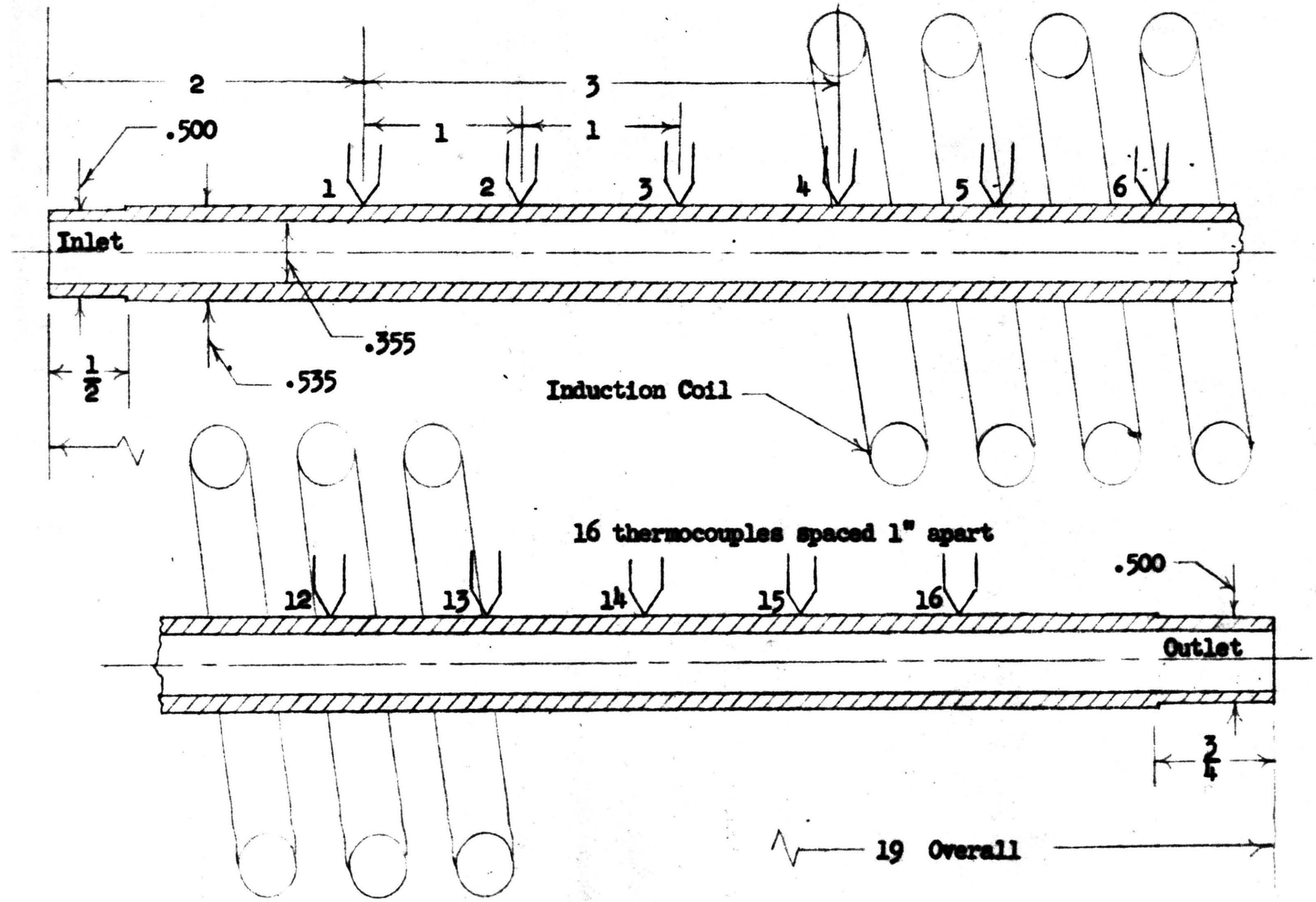

FIGURE 6

THERMOCOUPLE INSTALLATION ON TEST SECTION

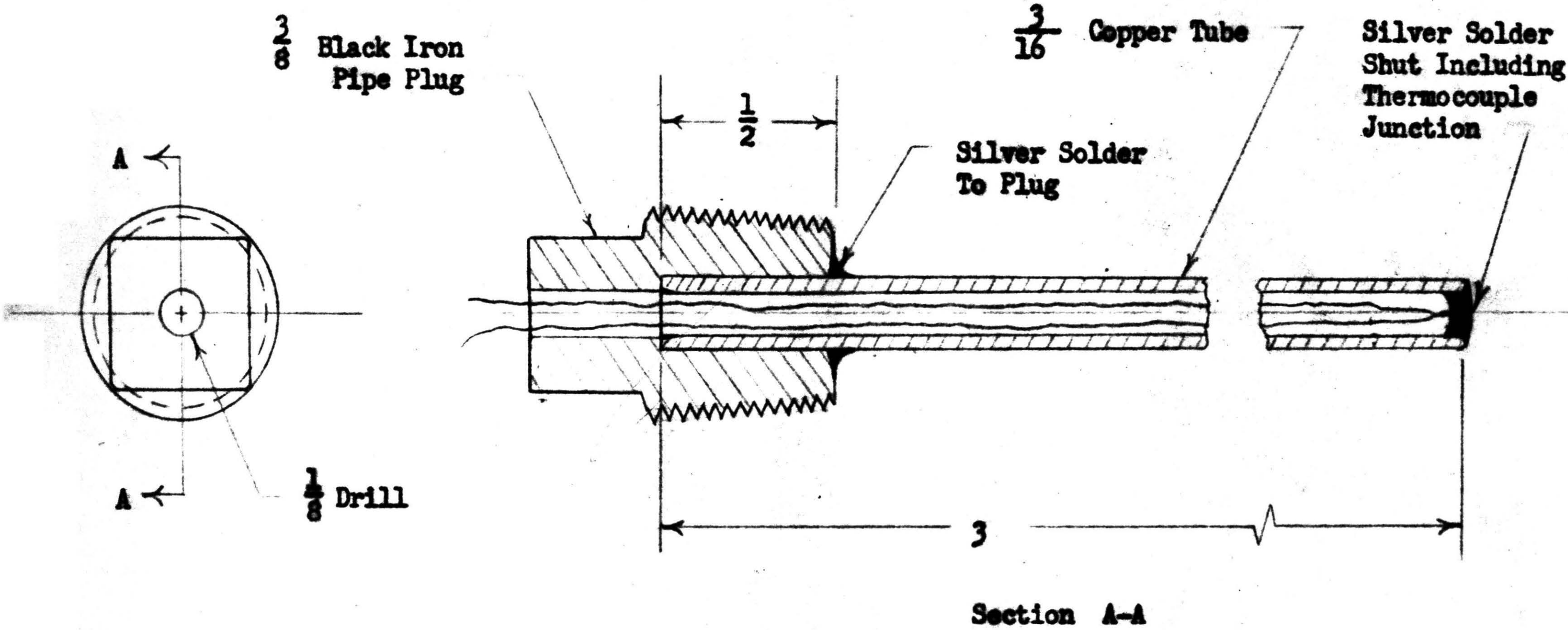

FIGURE 7

THERMOCOUPLE WELL

system and had a PH of 7.5 to 8.0.

PUMP AND FLOW MEASUREMENT

A three cylinder reciprocating pump with a capacity of 4.2 gallon per minute at 1000 psi was used to supply the required pressure for the vortex generator. A surge tank fabricated from a section of 2 1/2 inch pipe was used to reduce pressure variations. The normal system pressure was sufficient for the linear flow tests. Flow was measured by the use of a weigh tank and stop watch. A recording flowmeter was also available for estimates of the flow.

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 copper tubing, and was two inches in diameter, 9 inches long and consisted of 15 turns. A close-up of the coil and the test section is shown in Figure 8.

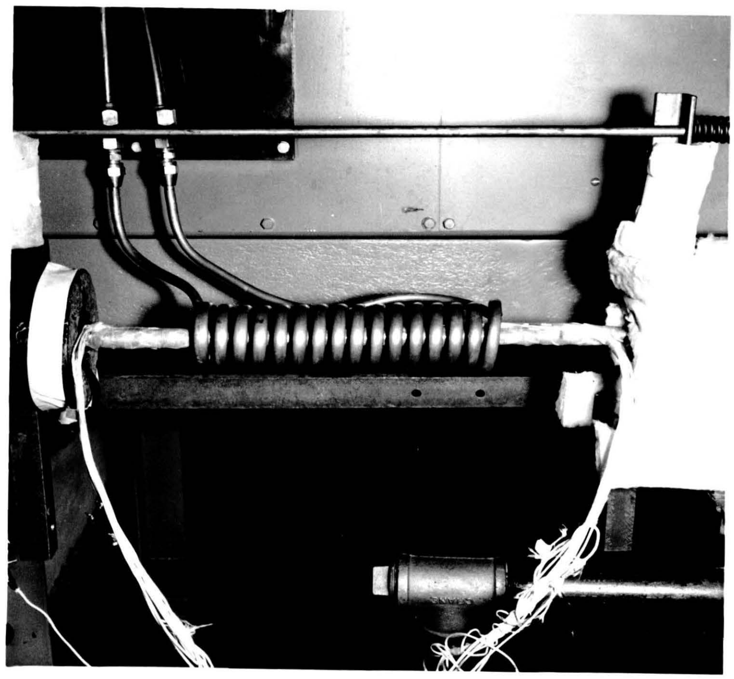

FIGURE 8

INDUCTION COIL AND TEST SECTION

NOTATION

Symbol	Description	Units
A	Surface area of heat flow	sq ft
A_f	Water flow area	sq ft
C	Specific heat of water	Btu/lb °F
D	Diameter	ft
g	Acceleration of gravity	ft/sec^2
G	Mass velocity	lb/hr sq ft
h	Film coefficient of heat transfer	Btu/hr sq ft °F
k	Thermal conductivity	Btu/hr ft °F
L	Effective length of test section	ft
m	Mass	slug
p	Pressure	lb/sq ft
q	Heat transfer rate	Btu/hr
q/A	Heat flux	Btu/hr sq ft
r	Radius	ft
t	Test section wall temperature	°F
Δt	Water film temperature drop	°F
T	Water temperature	°F
ΔT	Temperature rise across test section	°F
V	Water velocity	ft/sec
V_h	Heated volume of test section	cu ft
W	Mass flow rate of water	lb/hr
Ω	Circulation constant	sq ft/sec
ρ	Fluid weight density	lb/cu ft
μ	Fluid viscosity	lb/ft hr

<u>Subscripts</u>

a	water - aircore interface
ax	axial
b	bulk fluid
e	effective
L	linear flow
ow	test section outside wall
t	tangential
v	vortex
w	inside wall of test section
1	inlet water
2	outlet water

CALCULATIONS

The water inlet temperature was determined by the measured millivolt output of the inlet thermocouple taking into account the reference junction temperature as measured with a mercury thermometer. Conversion tables put out by Leeds and Northrup (17) were used for this. The temperature rise across the test section was found from the difference between the inlet and outlet thermocouple readings in millivolts divided by 0.0295 mv/°F the conversion factor for iron-constantan thermocouples. The bulk fluid temperature is then

$$T_b = T_1 + \Delta T/2 \tag{1}$$

The rate of fluid flow was found simply from the weight of fluid collected divided by the time required for collection.

The heat transfer rate is given by

$$q = C \dot{W} \Delta T \tag{2}$$

where the specific heat was taken as 1.0 Btu/lb °F.

To determine the area across which the heat is transferred, it was assumed that the entire heat input was generated uniformly over an effective length of test section. Since the temperature of the outside wall of the test section is proportional to the heat generated in the wall, the effective length for heat transfer was determined from a plot of this temperature as measured by the 16 surface temperature thermocouples. These temperatures were plotted versus the thermocouple position as shown in Figure 9. The area between this curve and a straight line indicating the process water temperature was found with a planimeter and then this divided by the temperature difference between

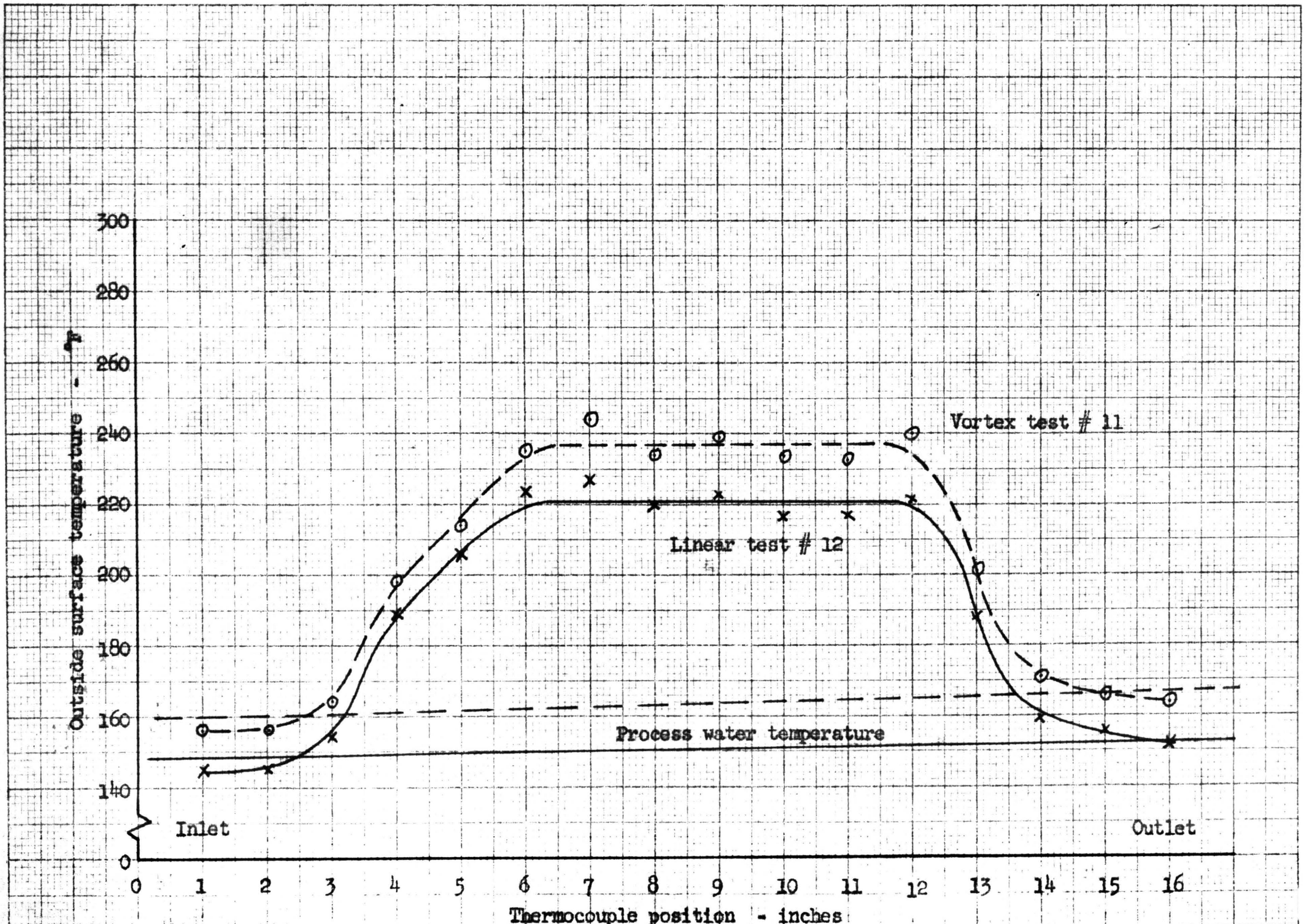

FIGURE 9

TEST SECTION OUTSIDE SURFACE TEMPERATURE VERSUS THERMOCOUPLE POSITION the outside wall and the fluid at the midpoint of the test section to give the effective, uniformly heated length. Since for each of the two types of flow, linear and vortex, the effective lengths calculated were all grouped quite closely, a single mean effective length was used for each flow type. The lengths used were 9.05 inches for linear flow and 8.71 inches for vortex flow. The maximum deviation from these averages was in each case less than 3%. The difference between the lengths for the two types of flow will be discussed later.

The effective heat transfer surface area is then given by

$$A = \pi D_w L \tag{3}$$

The heat flux was found from the heat transferred, given by equation (2) divided by this effective surface area.

The test section outside wall temperature was taken as the average of the temperatures indicated by thermocouples 6 through 12 which were located in the center portion of the test section and all gave fairly consistent readings for a particular test run. The deviation from the average is shown by the surface temperatures plotted in Figure 9.

Since the reference junctions for the test section thermocouples were located outside of the recording instrument, the indicated average had to be corrected for the difference between the temperature of the actual reference junction and the somewhat higher temperature of the compensating junction in the recorder. The correction was determined by comparing the water temperature and the wall temperature under isothermal conditions and by actually comparing the temperatures of the true reference junction and the compensating junction in the recorder.

Both methods gave corrections which agreed within 2°F of each other and whose values varied from 3 to 9 degrees depending on the length of time the recorder had been operating.

The temperature drop through the test section wall was calculated from a simplified form of the heat conduction equation through a cylindrical wall with uniform internal heat generation and no heat loss through the outside wall. The equation,

$$t_{ow} - t_w = \frac{q/V_h}{2\ k} \left[r_{ow}^2 \ \ln \frac{r_{ow}}{r_w} - \frac{r_{ow}^2 - r_w^2}{2} \right] \tag{4}$$

is the same as that used by Gambill and Greene (8). The volume of heated material was based on the effective length of heat transfer found above. The thermal conductivity used was for the average of the inside and outside wall temperatures and was taken from Tebo (18).

The temperature drop across the water film

$$\Delta t = t_w - T_b \tag{5}$$

was then used in

$$h = \frac{q/A}{\Delta t} \tag{6}$$

to find the film coefficient.

The mass velocity was found from

$$G = W/A_f \tag{7}$$

For linear flow the area of flow was simply the cross sectional area of the inside of the test section. For vortex flow the flow area was the area of the test section less the area of the central air column, and is given by

$$A_{fv} = \quad (D_w^2 - D_a^2)\ /4 = A_{fL} \quad (1 - \frac{D_a^2}{D_w^2}) \tag{8}$$

The ratio D_a/D_w was measured during the preliminary tests using glass tubing and the value of 0.5 was assumed to hold for the steel test section. This value is somewhat less than that quoted by Gambill and Greene (8) and Marshall (14) of 0.7, but considering the length of tube used in the present tests the value of 0.5 seems to be in order.

The velocity of fluid flow down the tube in feet per second was found from

$$V = \frac{G}{3600\,\rho} \tag{9}$$

The results of these various calculations for the linear flow tests and for the vortex flow tests are given in Tables 1 and 2.

An estimate of the tangential component of velocity in the vortex flow can be obtained from the measured pressure at the test section wall. The following analysis is similar to that given by Marshall (14).

A force balance in the radial direction for fluid flowing in a curved path gives

$$dp = \frac{\rho\, V_t^2}{g\ r}\ dr \tag{10}$$

In frictionless flow the torque applied to the fluid is zero, or

$$\text{Torque} = \frac{d(m\ V_t\ r)}{d\ t} = 0 \tag{11}$$

therefore,

$$r = \Omega\,/V_t \tag{12}$$

where Ω is the circulation constant.

Substituting equation (12) in equation (10) and integrating between the test section wall and the water - air core interface gives

$$p_w - p_a = \frac{\rho}{2\ g}\ (V_{ta}^2 - V_{tw}^2) \tag{13}$$

Since from Equation (12)

$$\frac{D_a}{D_w} = \frac{V_{tw}}{V_{ta}} \tag{14}$$

it can be shown that

$$V_{tw} = \left[\frac{P_w - P_a}{\frac{D_w^2}{D_a^2} - 1} \quad \frac{2\,g}{\rho} \right]^{1/2} \tag{15}$$

The true velocity in vortex flow then is the vector sum of the axial and tangential components,

$$V_v = (V_{ax}^2 + V_{tw}^2)^{1/2} \tag{16}$$

The ratio of the centripetal acceleration to gravitational acceleration or the number of gee's is

$$\text{gee} = \frac{V_{tw}^2}{g\ r_w} \tag{17}$$

Using for the pressure at the test section wall half of that measured at the vortex generator, and using the value of 0.5 given above for the ratio D_a/D_w, values of the tangential velocity, the true vortex velocity and the number of gee's were calculated for several of the vortex test runs. These results are listed in Table 3.

TABLE 1 - LINEAR FLOW

Run	T_1 °F	T °F	T_b °F	t_{ow} °F	t_w °F
1	140.0	6.9	143	203	192
2	139.5	12.6	146	248	228
3	139.5	19.9	150	296	265
4	143.5	15.5	151	297	259
5	145.5	20.4	146	335	285
6	145.5	9.8	150	247	223
7	146.0	5.0	148	203	191
8	147.5	3.7	150	200	187
9	148.0	6.8	151	240	217
10	148.0	14.0	155	315	266
11	148.5	17.8	157	349	286
12	148.0	6.0	151	221	203
13	148.0	10.1	153	265	234
14	148.0	13.7	155	305	263
15	148.5	17.8	157	338	282
16	148.5	4.8	151	207	194
17	148.5	9.2	153	251	226
18	143.5	6.1	147	216	197
19	144.5	3.4	146	188	178
20	144.5	3.5	146	185	177
21	141.5	5.8	144	193	187
22	142.0	11.3	148	237	225
23	141.5	21.4	152	295	283

TABLE 1 - LINEAR FLOW (Continued)

t °F	W lb/hr	q/A Btu/hr.sq.ft.	h Btu/hr.sq.ft.°F	V ft/sec
49	833	82,100	1680	5.50
82	833	150,000	1830	5.50
115	833	237,000	2060	5.52
108	1300	288,000	2660	8.61
139	1300	380,000	2730	8.58
73	1300	181,000	2480	8.61
43	1300	92,800	2140	8.59
37	1850	97,500	2640	12.25
66	1850	180,000	2730	12.25
111	1850	369,500	3320	12.26
129	1850	470,000	3640	12.27
52	1640	140,100	2690	10.85
81	1640	236,700	2920	10.85
108	1640	321,000	2970	10.86
125	1640	416,000	3330	10.87
43	1470	101,000	2350	9.72
73	1470	192,600	2640	9.72
50	1660	144,000	2880	10.97
32	1660	80,450	2510	10.97
31	1230	61,500	1980	8.12
43	590	48,800	1140	3.89
77	590	95,200	1240	3.90
131	590	180,000	1370	3.90

TABLE 2 - VORTEX FLOW

Run	T_1 °F	T °F	T_b °F	t_{ow} °F	t_w °F
1	142.5	4.7	145	178	168
2	146.0	8.4	150	213	194
3	144.5	13.3	151	253	223
4	144.0	21.3	155	307	258
5	144.5	4.5	146	183	171
6	145.0	7.2	149	207	188
7	144.5	11.1	150	241	211
8	145.0	17.6	153	302	254
9	148.0	4.6	150	196	183
10	148.5	8.3	153	232	207
11	159.5	8.6	164	237	211
12	157.0	11.9	163	266	230
13	155.0	15.8	163	305	257
14	154.5	4.3	157	197	182
15	152.0	7.6	156	228	201
16	149.5	10.9	155	260	224
17	148.5	16.3	157	323	264
18	149.0	5.4	152	200	180
19	148.5	7.9	152	236	206
20	144.5	4.3	147	192	178
21	146.0	9.6	151	248	217
22	149.0	6.9	152	228	203
23	151.0	13.4	158	301	252
24	152.5	24.0	164	372	280

TABLE 2 - VORTEX FLOW (Continued)

t °F	W lb/hr	q/A Btu/hr.sq.ft.	h Btu/hr.sq.ft.°F	V_{ax} ft/sec
23	1170	81,700	3560	10.3
44	1170	145,800	3310	10.3
72	1170	231,000	3210	10.3
103	1170	371,000	3600	10.3
25	1390	92,900	3710	12.2
39	1390	148,800	3810	12.2
61	1390	229,500	3760	12.2
101	1390	364,000	3600	12.2
33	1550	105,800	3210	13.7
54	1550	191,000	3540	13.7
47	1550	197,500	4200	13.7
67	1550	275,000	4110	13.7
94	1550	364,000	3880	13.7
25	1850	118,000	4720	16.3
45	1850	208,500	4650	16.3
69	1850	299,000	4330	16.3
107	1850	447,000	4170	16.3
28	1940	155,500	5530	17.0
54	1940	227,500	4210	17.0
31	1650	105,400	3500	14.5
66	1650	235,000	3560	14.5
51	1860	190,000	3720	16.4
94	1860	370,000	3940	16.4
116	1860	664,000	5710	16.4

TABLE 3 - VORTEX FLOW VELOCITIES

Run	V_{ax} ft/sec	P_w lb/sq in	V_{tw} ft/sec	V_v ft/sec	Gee's	h Btu/hr sq ft °F
2	10.3	10	15.9	19.0	530	3310
7	12.2	14	18.8	22.4	740	3760
10	13.7	17	20.7	24.8	900	3540
11	13.7	17	20.7	24.8	900	4200
15	16.3	21	23.0	28.2	1110	4650
18	17.0	23	24.1	29.5	1220	5530
19	17.0	23	24.1	29.5	1220	4210
22	16.4	22	23.5	28.7	1160	3720

DISCUSSION

The discussion of the results of this investigation is in three sections. First, comparison of the linear flow results with standard correlations, second, analysis of the vortex flow results and third, a general evaluation of the investigation in view of the results.

LINEAR FLOW

The linear flow tests can be broken down into two groups, non-boiling in which the inside wall temperature of the test section is below the saturation temperature of the liquid, and nucleate boiling tests where the wall temperature is greater than the saturation temperature of the fluid.

One of the standard correlation equations for relating the non-boiling film coefficient to the velocity of flow and the various fluid parameters is

$$\frac{h\ D}{k} = .023 \left(\frac{D\ G}{\mu}\right)^{.8} \left(\frac{C}{k}\right)^{.4} \tag{18}$$

where all the fluid properties are evaluated at the bulk fluid temperature. This may be simplified for water at moderate pressures and temperatures to give (2)

$$h = 150\ (1 + .011\ T_b)(V)^{.8} / (D')^{.2} \tag{19}$$

where D′ is the tube diameter in inches. This equation was evaluated at 147°F, the approximate bulk fluid temperature for the non-boiling runs, and the results plotted on the graph with the experimental data in Figure 10. The experimental values all agree quite closely with the correlation equation.

The results of boiling heat transfer tests are conventionally shown in a plot of the heat flux versus the film temperature difference.

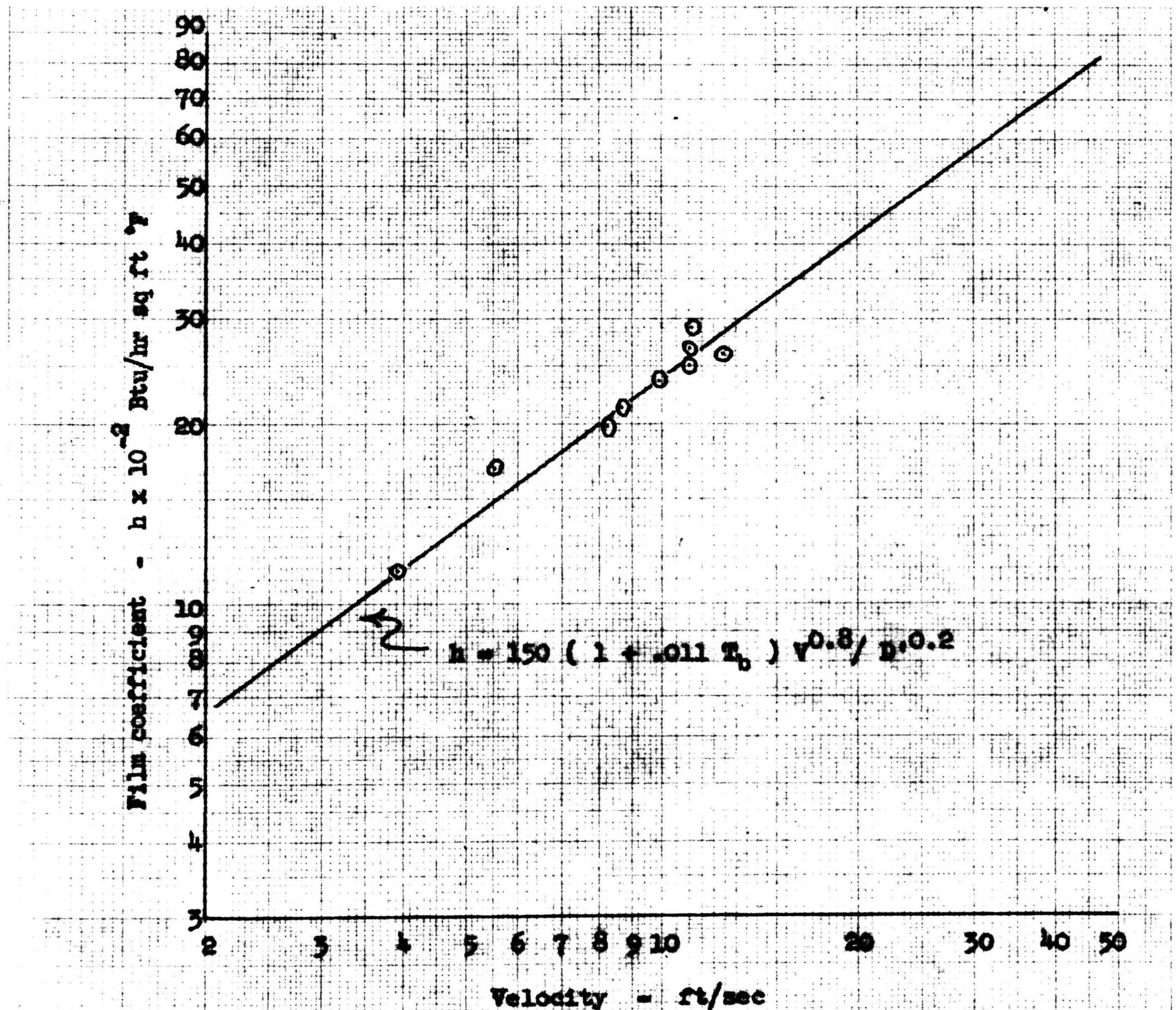

FIGURE 10

NON-BOILING LINEAR FLOW RESULTS

The 12.3, 8.6 and 3.9 ft/sec runs are shown in this manner in Figure 11. A curve for each velocity is shown with a slope of one in the non-boiling region and an increased slope in the boiling region. A typical curve as found by McAdams, et al (19) is also shown in the figure. This curve is for heat transfer to water at 60 psia, the bulk fluid temperature 50°F less than the saturation temperature, and a velocity of 12 ft/sec. Since in the present investigation the conditions of subcooling and pressure were different, this curve can only be used as an indication of the general trend of the results.

It is observed that the value of the heat flux in the boiling region for this investigation does not increase with film temperature at a rate as large as that indicated by other investigations. This apparent discrepancy might partly be due to the insufficient data available in this region from which to draw an accurate conclusion. Also erroneous values for the surface temperature of the test section might be to blame. Since in the boiling region the heat input increases with respect to the temperature difference at a much greater rate than that in the non-boiling region, the increase in the induction field required to produce a given increase in surface temperature is greater in the boiling region than in the non-boiling region. This might then cause the temperature of the thermocouple wires to exceed the surface temperature of the test section and thereby causing the thermocouple reading to be larger than the actual temperature. This would cause the observed discrepancy between the results of this investigation and the results of others.

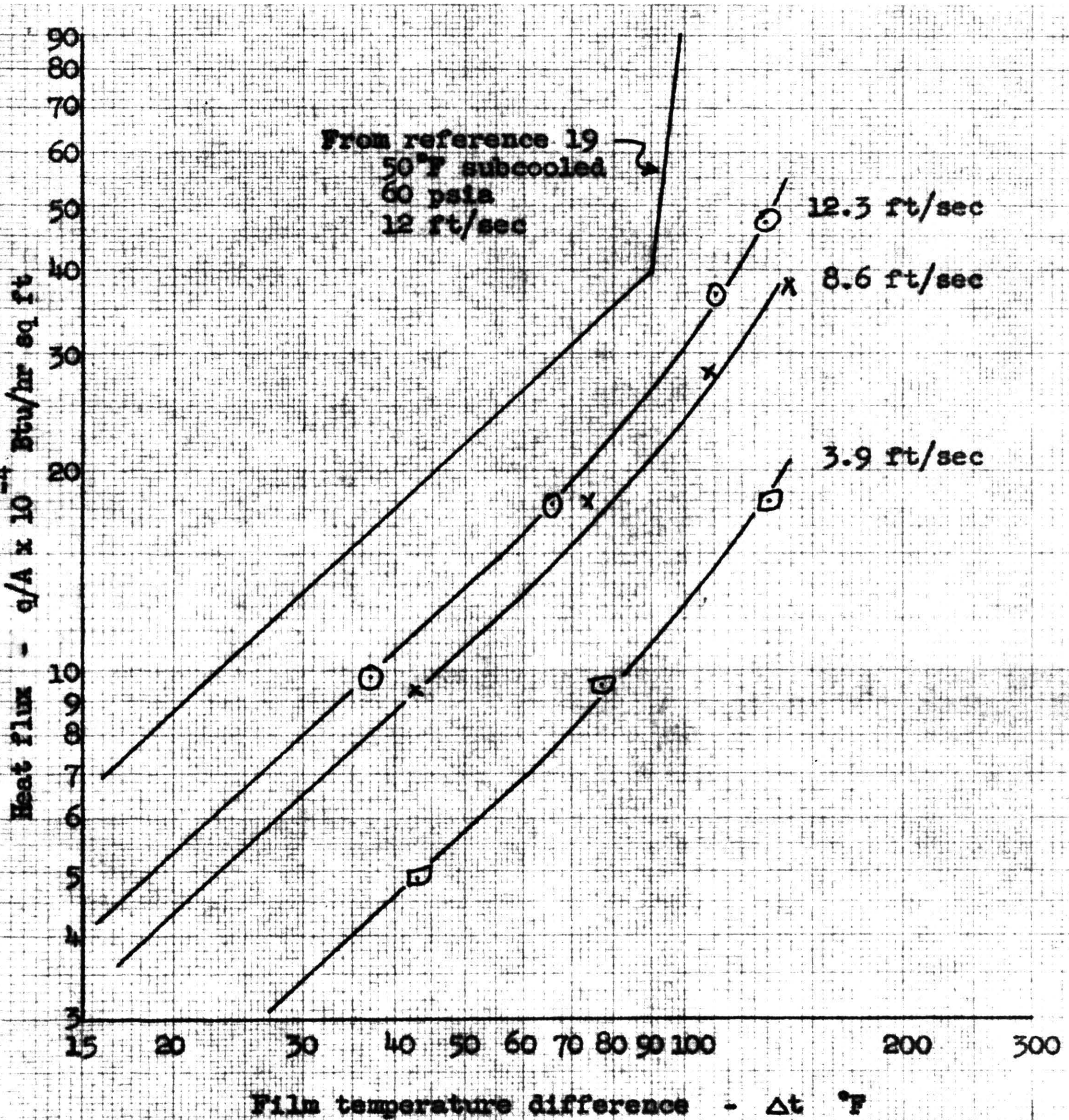

FIGURE 11

BOILING LINEAR FLOW RESULTS

VORTEX FLOW

As in linear flow the vortex data can be broken down into non-boiling and boiling tests. For vortex flow the saturation temperature of the fluid is higher than that for linear flow because the vortex causes the test section wall pressure to increase over that for linear flow.

The non-boiling film coefficients are plotted versus the axial velocity in Figure 12. The straight line shown indicates the values of the film coefficient to be expected for linear flow from equation (19) where D′ is taken as the effective diameter given by four times the cross sectional flow area divided by the wetted perimeter of the test section. The majority of the experimental values found lie 20 to 30% above the linear flow values at the same axial velocity. This is in the range of increase found by Kreith (8) (9) but is considerably less than that found by Kreith and Margolis (10). There is more scatter in the vortex results than in the linear flow results and is probably due to periodic variations in the nature of the vortex.

The variation of the heat flux with film temperature difference is shown in Figure 13. For comparison with linear flow the results of the 12.3 ft/sec linear flow tests are also shown. There is a definite increase in the heat flux for vortex flow over that for linear flow at the same temperature difference and velocity.

The film coefficient is shown versus the vector velocity of the vortex in Figure 14 along with the values expected from Equation (19). Here it appears that there is a decrease in the film coefficient for vortex flow when it is compared with the linear flow values at velocities corresponding to the vector velocity rather than the axial

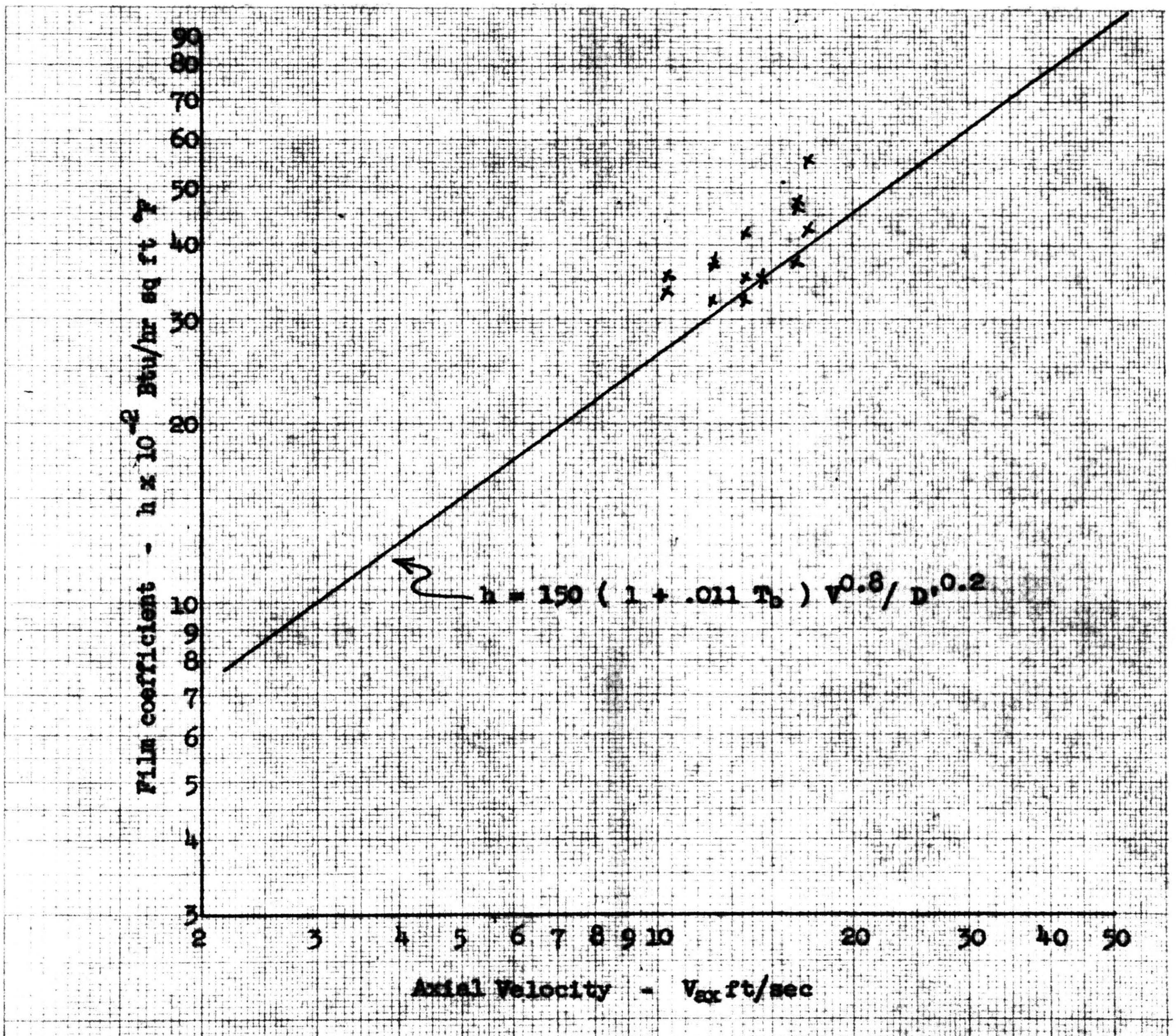

FIGURE 12

NON-BOILING VORTEX FLOW RESULTS

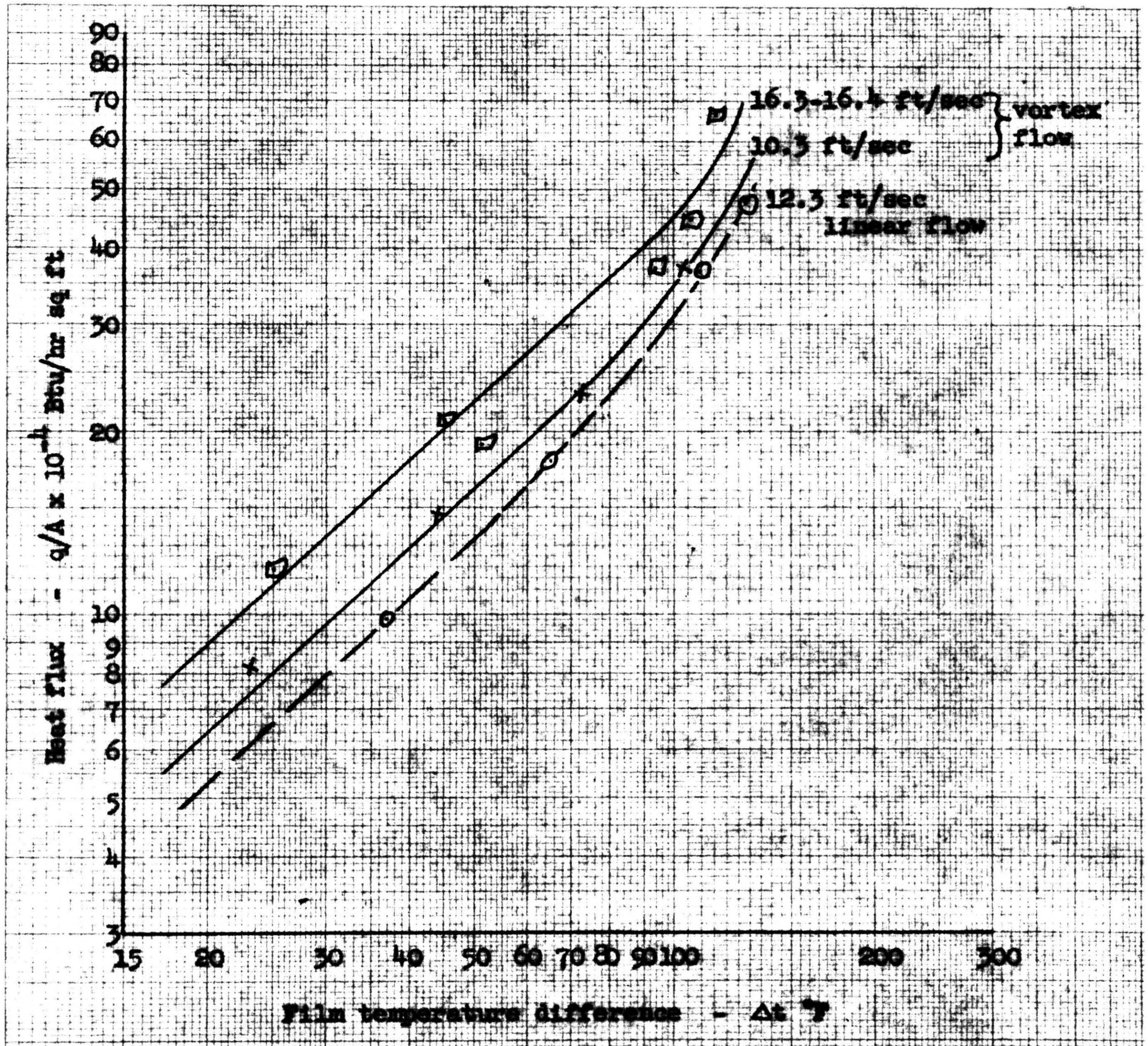

FIGURE 13

BOILING VORTEX FLOW RESULTS

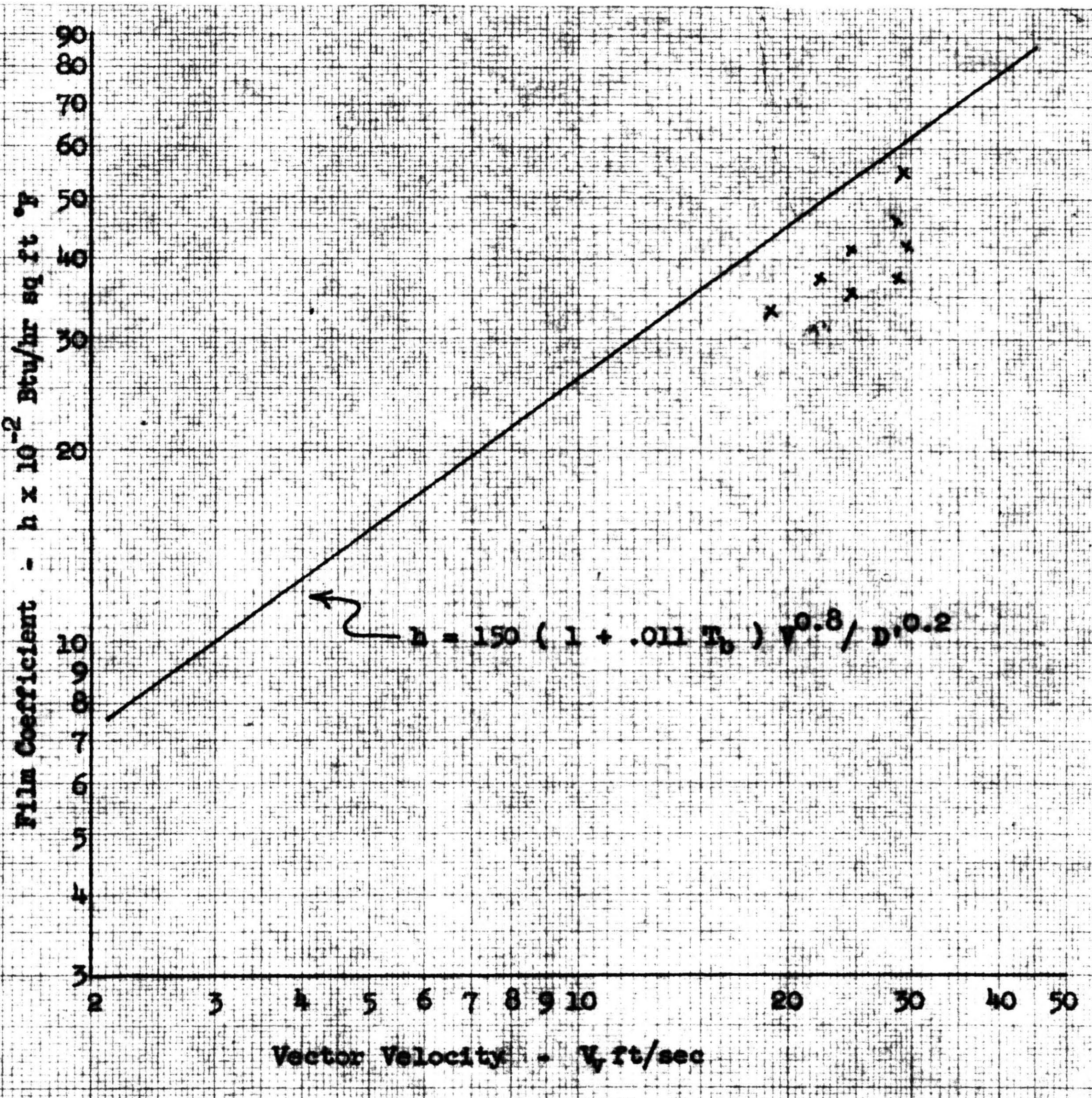

FIGURE 14

NON-BOILING VORTEX FLOW RESULTS USING VECTOR VELOCITY

velocity. This result does not seem reasonable and can be explained as follows. The vortex created by the generator used in the tests was observed initially in glass tubing. It showed a decaying tendency in the last several inches of the test section. It is reasonable to assume that with the increased friction of the rough steel test section, the decay of the vortex would be even more pronounced. If this is true, then the tangential component of the vortex velocity is considerably less than that given in Table 3. The vector velocity would be closer to the axial velocity and the vortex film coefficients do show an increase over the linear flow values when compared on the basis of the axial velocity.

The decrease in the effective heated length of the test section from linear flow to vortex flow mentioned earlier could be caused by the decay of the vortex. The vortex entering the heated portion of the test section would cause the temperatures on the inlet side to be decreased more than the temperatures further downstream and this would cause this apparent shortening of the effective length.

Even if the decay of the vortex is not sufficient to account for the apparent lack of increase in the film coefficient due to curved flow, it must be realized that with the number of gee's attained the increase predicted from the dependence of the natural convection coefficient on the acceleration of gravity is quite small. The natural convection correlation equations depend on only the 1/4 to 1/3 power of g. For the maximum number of gee's obtained in the present investigation this amounts to a 6 to 11 fold increase in the natural convection coefficient to be added on the normal forced convection coefficient. Since the natural convection coefficients are only on the order of 50 to 100 Btu/hr sq ft °F, the maximum increase expected would only be about 1000 Btu/hr sq ft °F. If the decay of the vortex is considered with this increase in the film coefficient then the results of the vortex tests seem reasonable.

GENERAL EVALUATION

The excellent agreement of the non-boiling linear flow results with the standard correlation equations indicates that the methods and apparatus used gave accurate values for the film coefficient. This shows that at least for the lower heating rates the problem of the measurement of the surface temperature with thermocouples in an induction field is not insurmountable. The possibly erroneous results obtained at high heating rates due to the heating of the thermocouple wires can probably be eliminated by further study and the use of thermocouple materials which are not ferromagnetic.

The decay of the vortex can be solved by varying the design of the generator and by using a shorter test section. Also by mounting a test section between glass tubing, as originally attempted in this investigation, would eliminate the need for assuming that the heat was generated uniformily over an average length of test section.

In many cases the tests did not cover a wide enough range of variables to permit a sufficiently accurate interpretation of the results. Changes in the apparatus to enable coverage of a wider range of velocity and heat flux would be desirable.

CONCLUSIONS

1. The agreement of the non-boiling linear flow film coefficients with standard correlation equations indicates the feasibility of using apparatus similar to that designed.
2. The boiling results differ slightly from the expected results probably because of surface temperature measurement problems at large heat flux.
3. The vortex flow indicates that there is 20 to 30% increase in the film coefficient over that for linear flow of water at the same axial velocity. This area of the investigation requires further study.

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VITA

E. Robert Schmidt, Jr. was born in Richmond Heights, Saint Louis County, Missouri, on December 7, 1936, to Edward R. and Ann D. Schmidt. From birth to the present, the author's permanent address has been #6 H Hawbrook Lane, Kirkwood, Missouri. He attended schools in the Kirkwood School District attending Henry Hough Grade School, Nipher Junior High School and graduating from Kirkwood Senior High School on June 9, 1954.

In September of the same year, he entered Missouri School of Mines and Metallurgy. As an undergraduate, his basic field of study was Mechanical Engineering with special interest in the field of Nuclear Engineering. On June 1, 1958, he received a Bachelor of Science in Mechanical Engineering, and also a commission as a Second Lieutenant in the Ordnance Corps of the United States Army Reserve.

The author was granted an Atomic Energy Commission Special Fellowship in Nuclear Science and Engineering and enrolled in the Nuclear Engineering program at Missouri School of Mines in September, 1958. From then until the present, he has been working toward the degree of Master of Science in Nuclear Engineering.

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