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Design and Fabrication of an Apparatus to Determine Heat Transfer Coefficients in an Air-Particle Mixture Over a Horizontal Flat Plate at Uniform Temperature

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DESIGN AND FABRICATION OF AN APPARATUS TO DETERMINE HEAT TRANSFER COEFFICIENTS IN AN AIR-PARTICLE MIXTURE OVER A HORIZONTAL FLAT PLATE AT UNIFORM TEMPERATURE

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Master of Science

by

Louis G. Hunter, Jr.

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This thesis, by Louis G. Hunter, Jr., is accepted in its present form by the Department of Mechanical Engineering of Brigham Young University as satisfying the thesis requirements for the degree of Master of Science.

August 1963
Date

Typed by Catherine Hunter

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NOMENCLATURE

C_{Pa}	heat capacity of air, Btu/lb ^{o}F
C_{Pw}	Heat capacity of water, Btu/lb ^{o}F
h_{Lx}	local laminar film coefficient, Btu/hr ft ^{o}F
h_{tx}	local turbulent film coefficient, Btu/hr ft ^{o}F
J	intensity of turbulence, $\frac{(U'^{2})^{\frac{1}{2}}}{U_\infty}$
K	air thermal conductivity, Btu/hr ft ^{o}F
$\dot{M}$	flow rate, lb/hr
N_{NUX}	local Nusselt number, $\frac{h_x x}{K}$
N_{PR}	local Prandtl number, $\frac{\mu C_p}{K}$
N_{REX}	local Reynolds number, $\frac{P U_\infty x}{\mu}$
N_{STX}	local Stanton number, $\frac{h_x}{P U_\infty C_p}$
P	density, lb/ft^{3}
P	pressure, lb/ft^{2}
q'	heat flux, Btu/hr ft^{2}
t_∞	free stream temperature, ^{o}F
te	evaluation temperature, ^{o}F
ti	inlet temperature of active water channel, ^{o}F
to	outlet temperature of active water channel, ^{o}F
tw	wall temperature, ^{o}F
u	velocity component in X direction

$\bar{u}'$	time average of the X component of velocity
μ	micron
U_∞	free stream velocity, ft/hr
v	velocity component in Y direction
ν	kinematic viscosity, ft^2/hr
W	weight of air, lbs
Ws	weight of particles, lbs

CHAPTER I

INTRODUCTION

The study of two phase flow, where a finely divided solid is suspended in a fluid medium, has received considerable attention in the last five years due to the increased number of systems in which this phenomena occurs. The two most important areas involving the gas-particle mechanism are nuclear reactor heat exchangers and liquid and solid rockets.

Metal particles are added to the propellants of solid and liquid rockets to increase the burning temperature which in turn increases the performance. These particles are thereby introduced into the exhaust gas stream and a study of gas particle mixture is fundamental to the design and analysis of the system.

In the field of nuclear reactors, much consideration has been given to the fact that heat transfer rates can be increased by the addition of particles into the cooling medium.

I. LITERATURE SEARCH

Experimental investigation has shown that greater heat transfer rates can be obtained with gas-solids mixtures compared with the gas alone.

Farber and Morley (7)[1] reported that an appreciable increase in heat transfer results from adding solids to gas, flowing at a constant rate. Their system consisted of drawing through a vertical isothermal tube suspended alumina silica particles (ranging in size from sub-micron to 200 micron) in air. Their results show for a solids

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[1] Numbers in parentheses refer to the bibliography at the end of the thesis.

loading ratio of 8.0, the Nusselt number was increased from 45 to 125 at a Reynolds number equal to 15×10^3.

Depew (4) performed experiments at Reynolds numbers 13500 and 27400 with solid glass spheres in an isothermal tube apparatus similar in design to the equipment used by Farber and Morley. He found that the 200 micron spheres had no effect and the 70 micron spheres had only a slight effect on the overall heat transfer coefficient compared with air alone. Depew then extended his work to 30 micron glass spheres which had a considerable effect on the heat transfer compared with air alone.

Experiments with an electrically heated tube containing a carbon suspension in nitrogen and helium gases were performed by the Babcock and Wilcox Company (1). Solids-to-gas mass ratio up to 100 was used. Their results show that the overall heat transfer coefficient can be increased up to six times by the addition of particles.

Tien and Quan (9), using apparatus like Depew's, extended his work by comparing air-glass and air-lead heat transfer characteristics. The importance of the specific heat of the particles was significant. The Nusselt number obtained with 30 micron glass particles was much greater than that with 30 micron lead particles since the specific heat of glass was about six times greater than lead.

II. SCOPE AND PURPOSE OF THIS WORK

This work consisted of designing and building a system capable of measuring heat transfer coefficients over a horizontal flat plate at uniform temperature with negligible pressure gradient. The test fluids to be used were air and an air-particle mixture.

CHAPTER II

DESCRIPTION OF APPARATUS

The experimental apparatus is shown in Figures 1 and 1a. The apparatus consists of a fan, screen mesh flow straighteners, test section (which enclosed the flat plate), heater section, and connecting duct work.

The instrumentation primarily consisted of a thermocouple network, pressure measuring devices, and flow measuring equipment (Fig 5).

The flat plate was composed of three components: the top plate over which the heat transfer coefficients were measured (active plate), the center insulator, and the bottom plate. (Figs 1b, 1c, 3).

The active plate was fabricated from 1100 alloy aluminum, which is ninety-nine per cent commercially pure. The thermal conductivity of this metal was 128 Btu/hr ft ^{o}F. This metal was chosen to minimize the temperature gradient in the axial direction making it possible to attain a more uniform surface temperature. The active plate was 23.8" long, 7.85" wide, and 0.10" thick (Fig 1b). The plate has ten water channels which were milled out 0.88" from the original 3/16" thickness. Figure 1c shows the water channel detail.

Two different techniques were used to measure the surface temperature. One technique employed was to butt weld .02" diameter iron constantan thermocouple wire and place it in a groove approximately .01" beneath the active plate surface (Fig 1c). The length of the grooves was the width of each water channel and was designed to minimise conduction loss from the thermocouples. The thermocouples were bonded in the grooves with Decon two-ton epoxy cement. After the bonding agent cured, the thermocouples were ground off flush with the surface, so that the butt weld was an integral part of the surface. A typ-

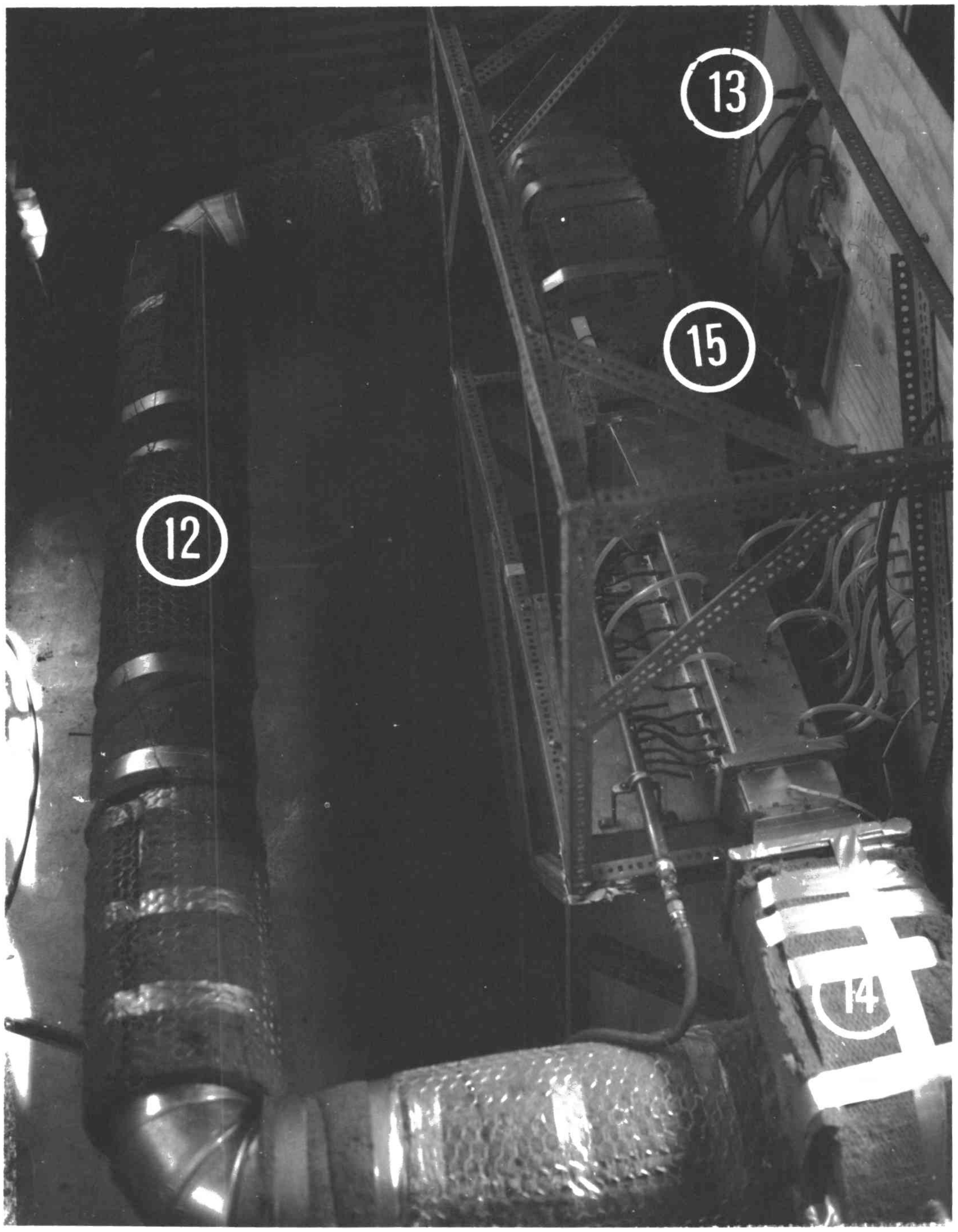
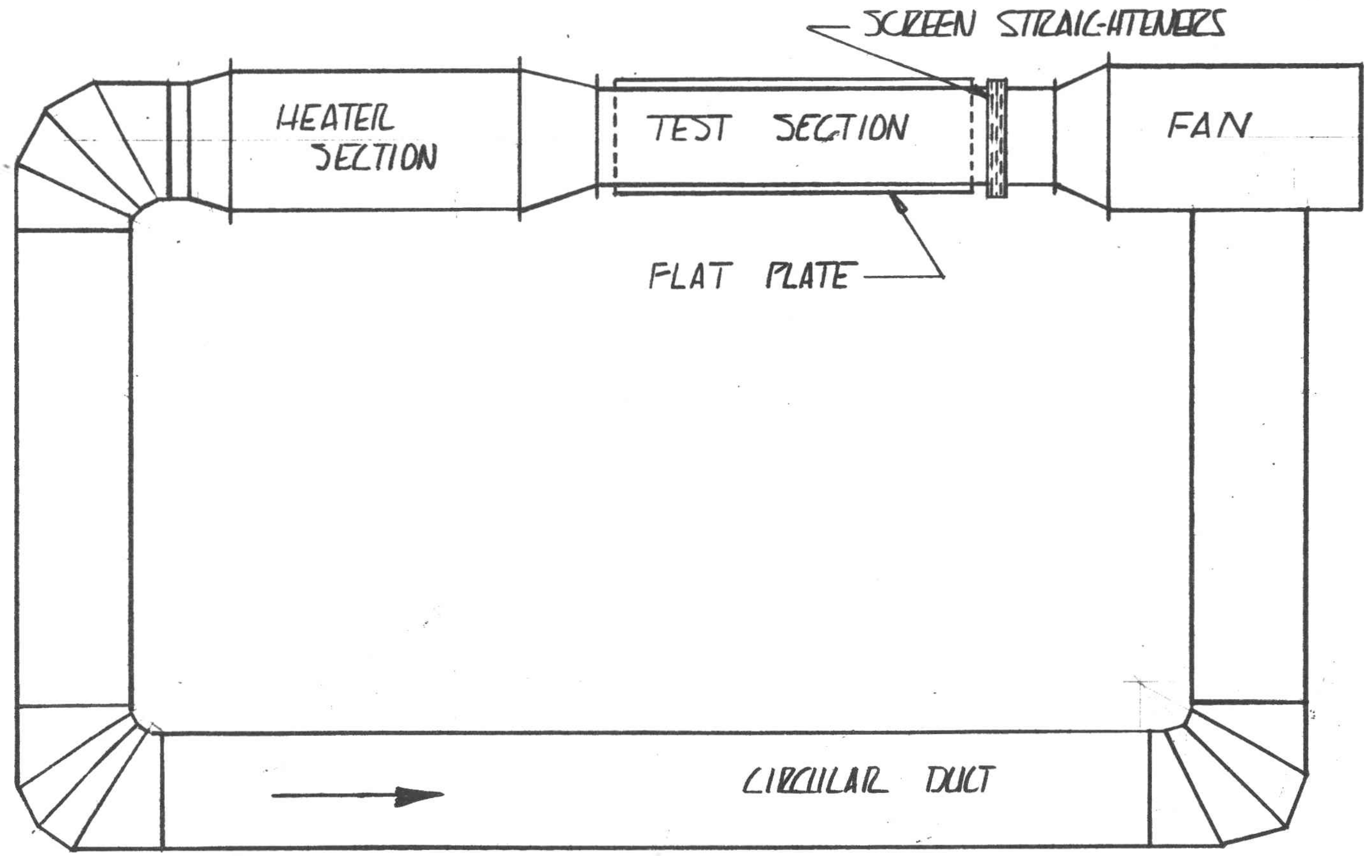


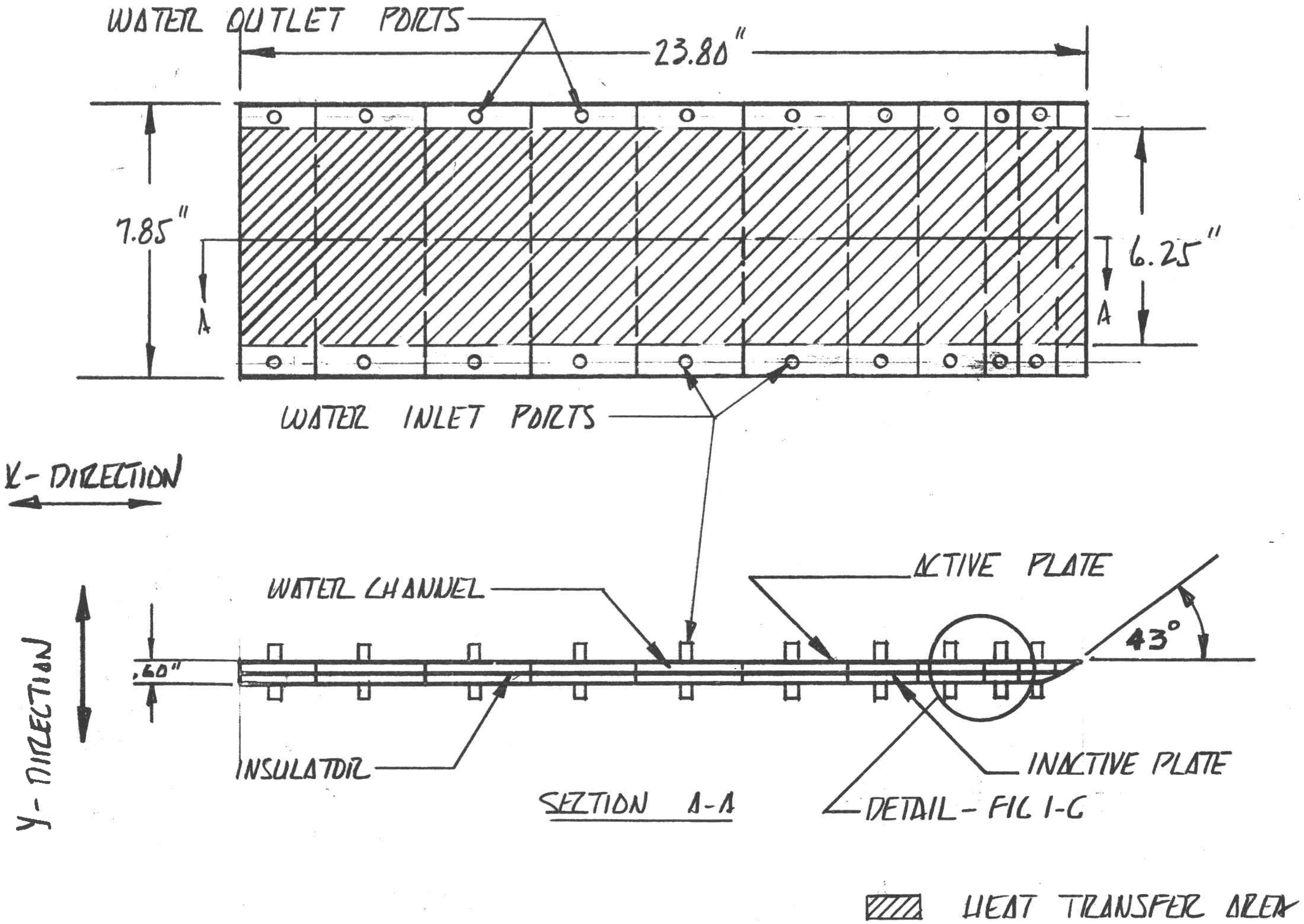
FIGURE 1

EXPERIMENTAL APPARATUS

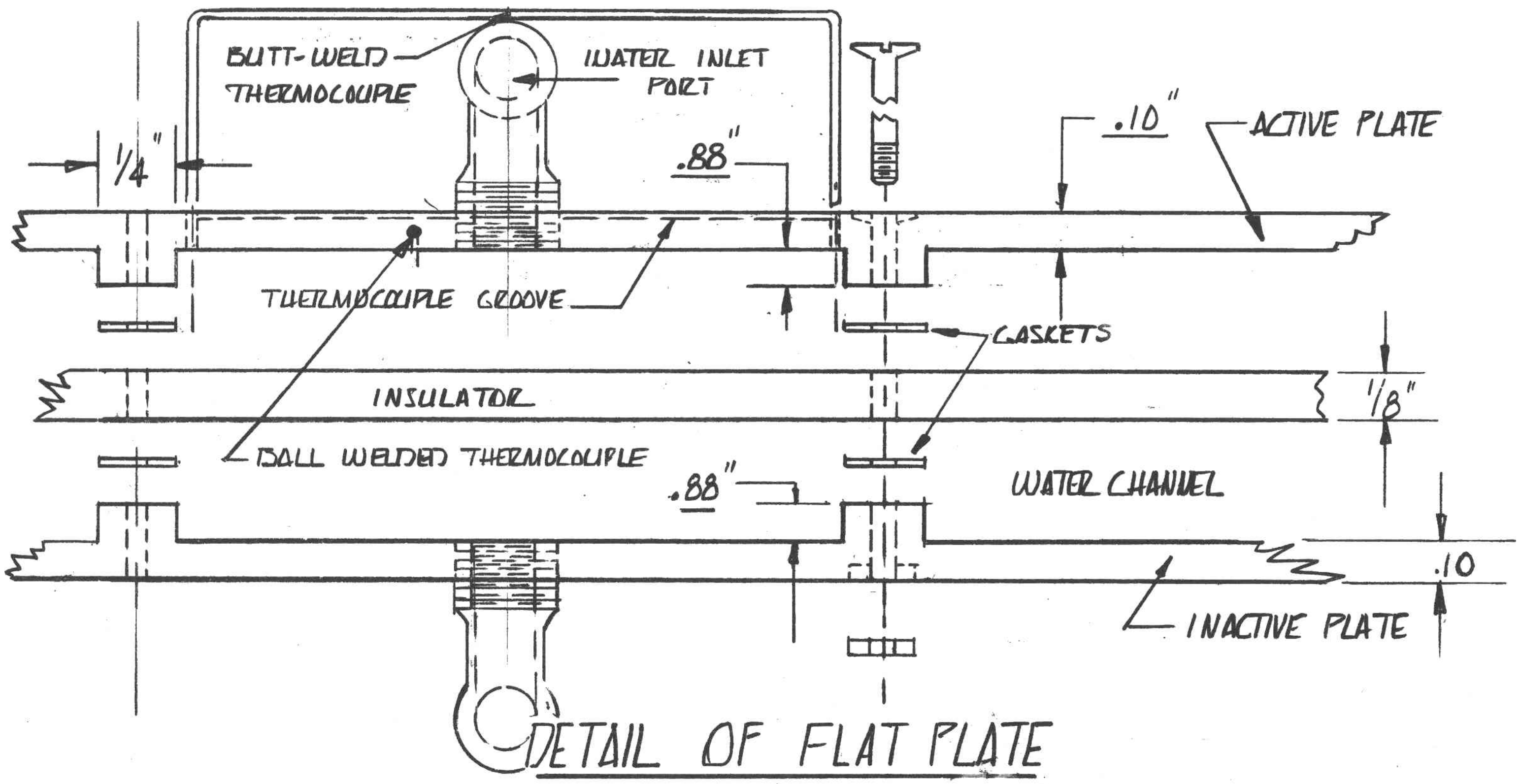


TOP VIEW OF EXPERIMENTAL APPARATUS

FIG 1a.

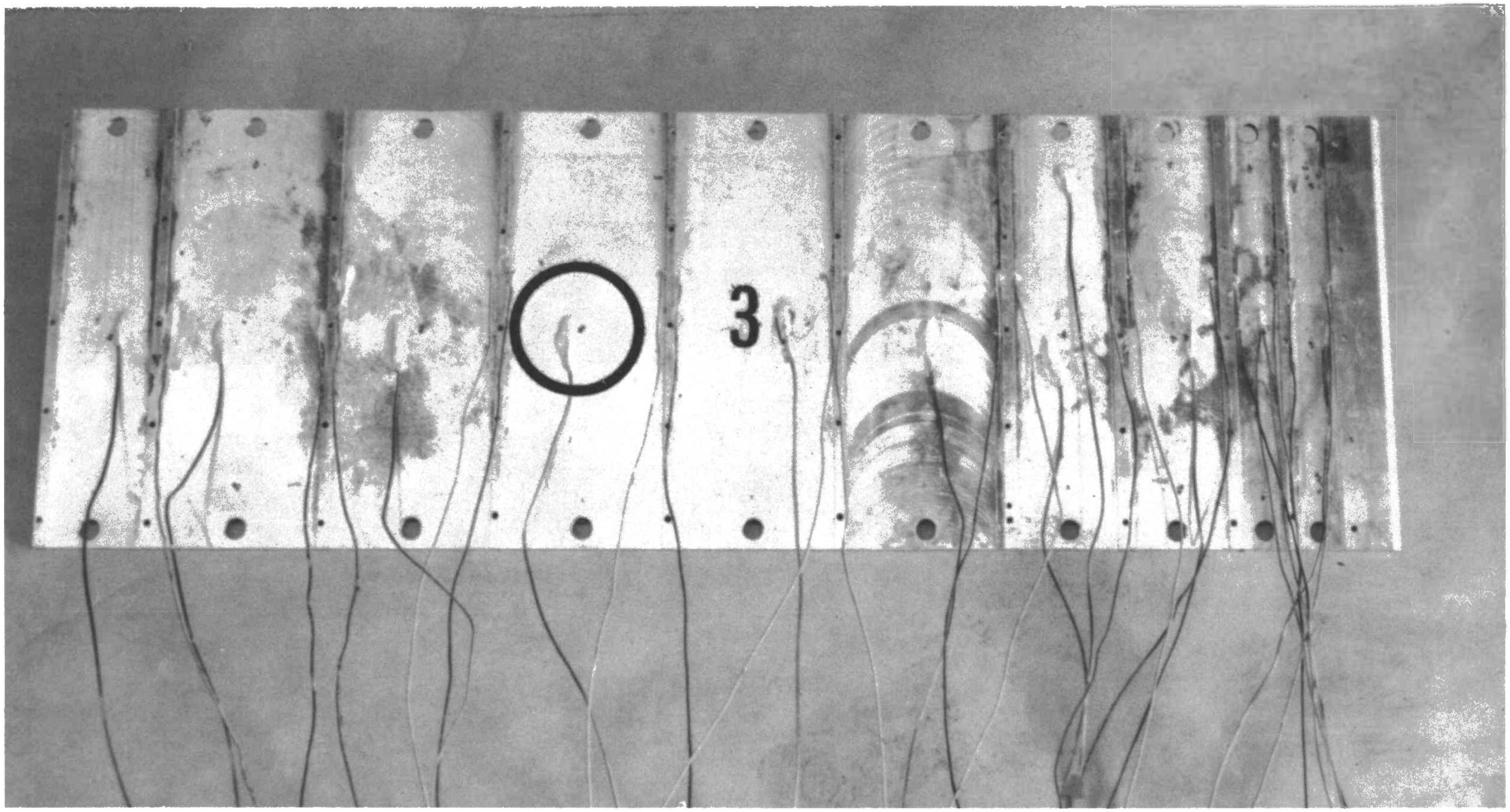


TOP VIEW OF FLAT PLATE - FIG 1-B



DETAIL OF FLAT PLATE

FIG 1-2



ACTIVE PLATE BEFORE MOUNTING
Figure 2

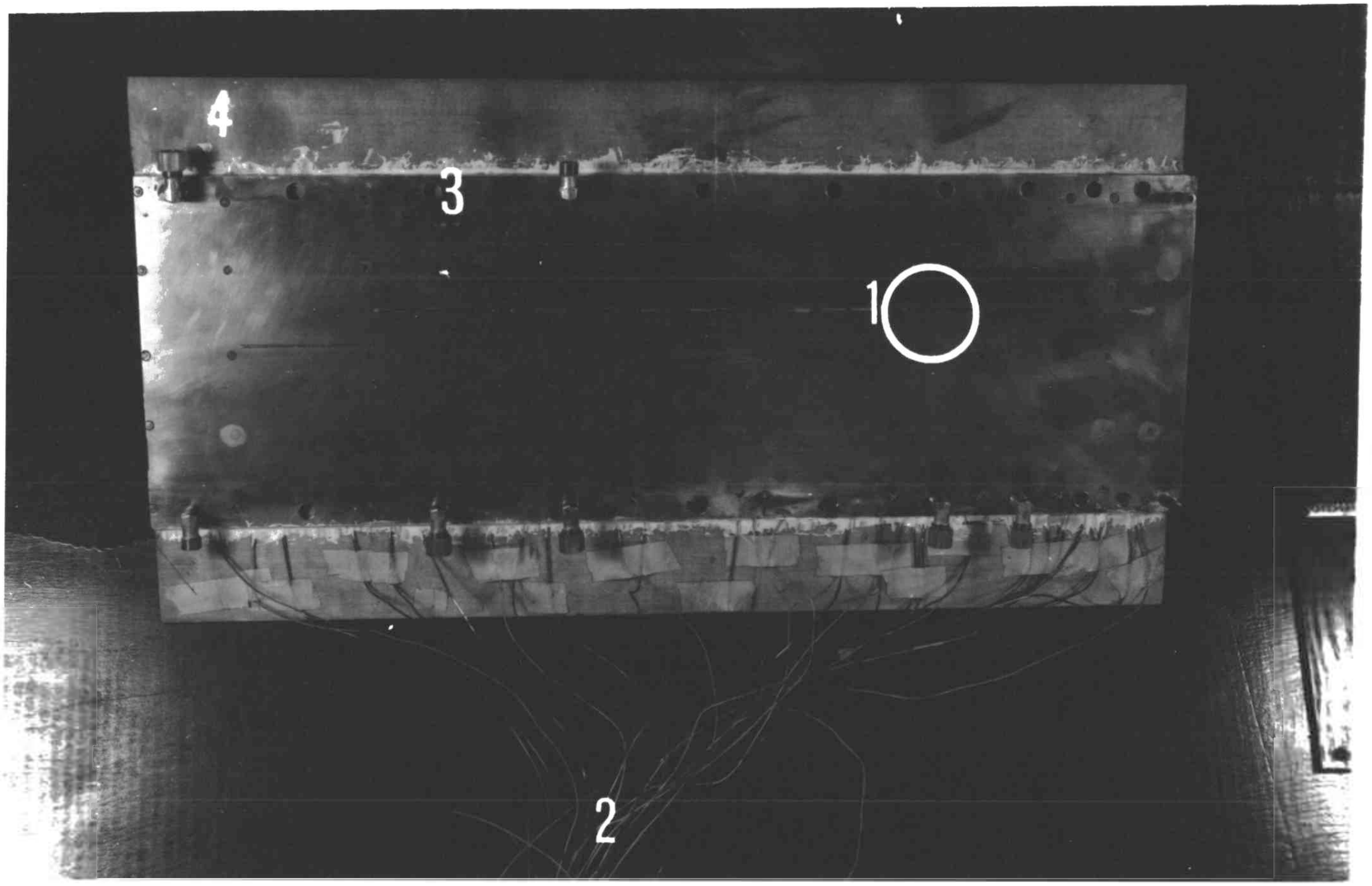
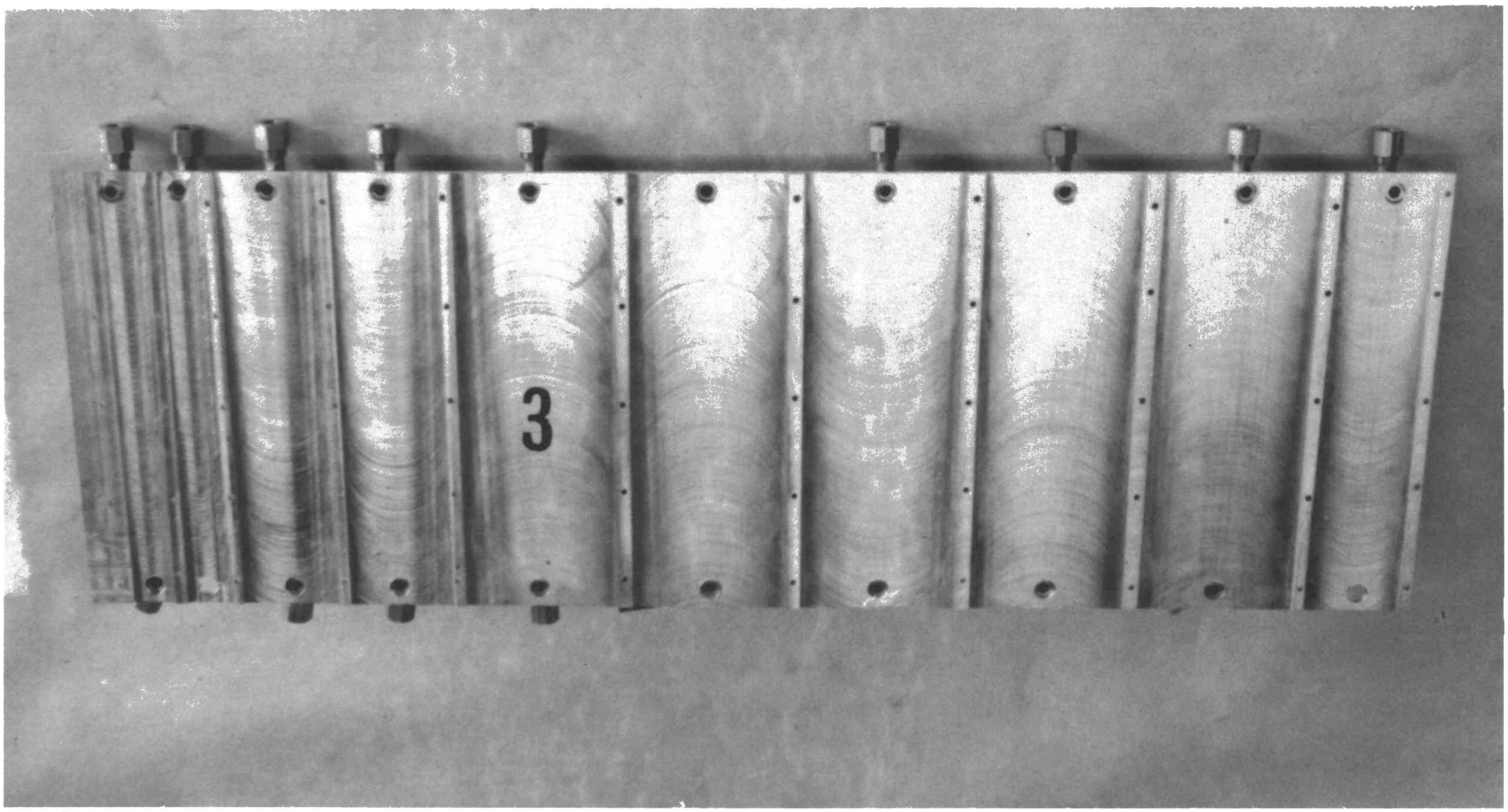


FIGURE 3

ACTIVE SIDE OF PLATE AFTER MOUNTING



INACTIVE PLATE BEFORE MOUNTING

Figure 4

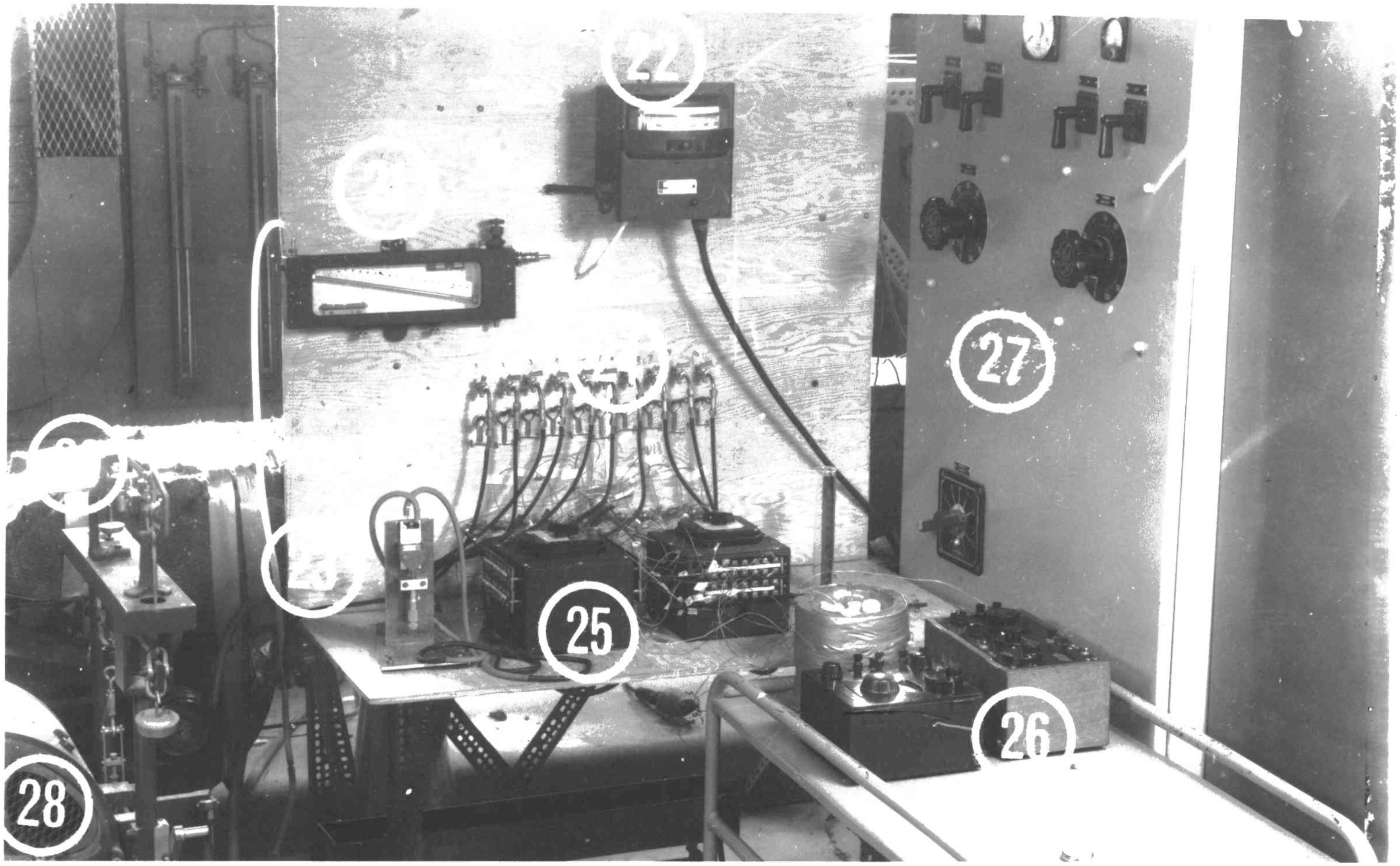


FIGURE 5

INSTRUMENTATION AND CONTROLS

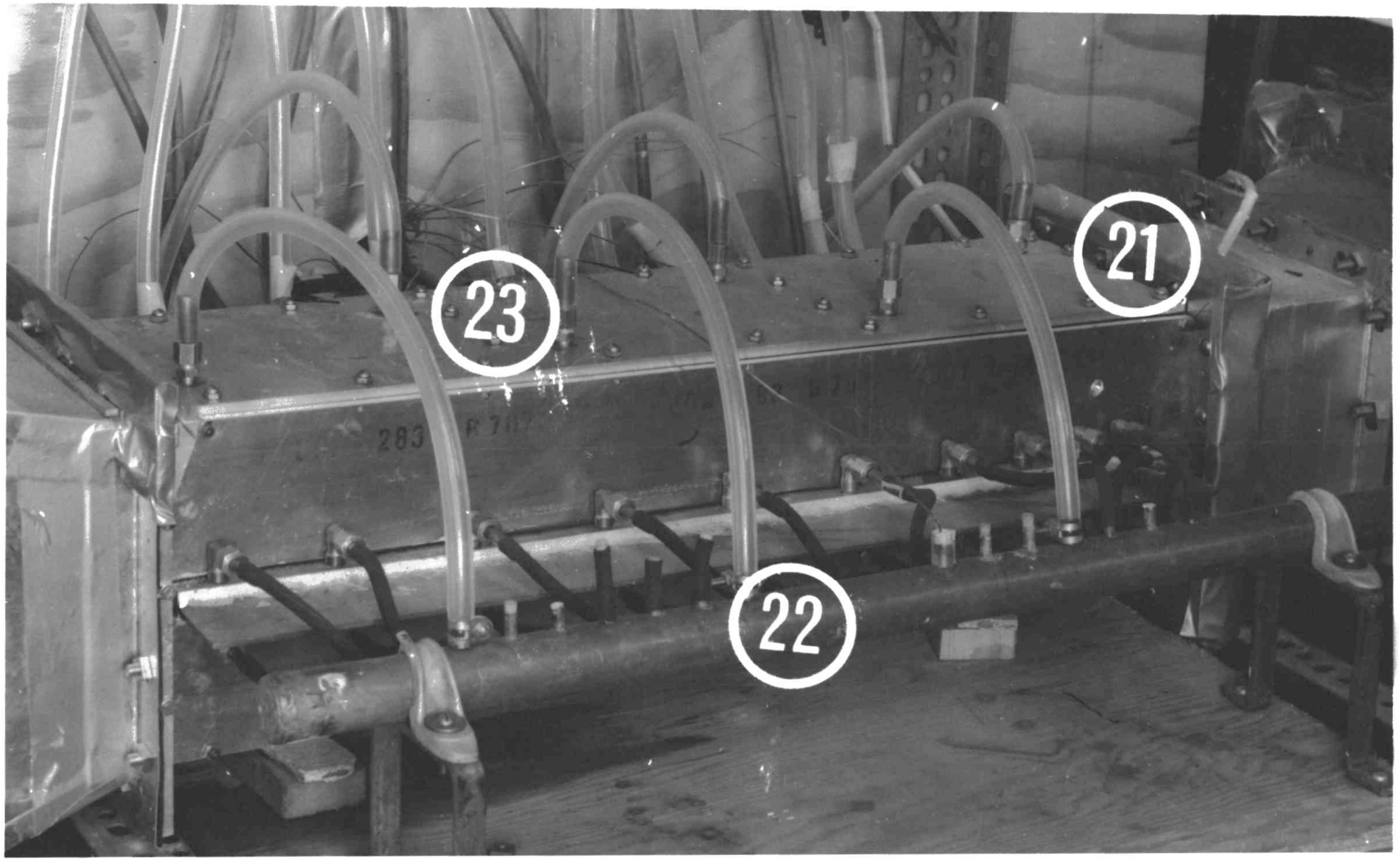


FIGURE 6

TEST SECTION HOUSING FLAT PLATE

ical thermocouple installation is circled in Figure 3.

The other technique used to determine the surface temperature was to imbed approximately .05" from the surface ball welded copper constantan (.01" diameter) thermocouples into the active plate (Fig 1c). The accuracy of these thermocouples depends on the negligible temperature gradient from the active surface to the thermocouple location. In view of the high thermal conductivity of the plate and the distance of the thermocouples from the surface, the thermocouples will measure within 0.3 ^{o}F of the surface temperature.

The center insulator was 1/8" thick glass phenolic.

The inactive plate was identical to the active plate with the following exceptions: the material used to fabricate the plate was 7075 alloy aluminum (K = 70 Btu/hr ft ^{o}F); no thermocouples were installed in the plate; and the plate was shorter due to the beveled leading edge (Fig 1c, 4). It was designed to prevent any heat transfer of the free stream from coming in the bottom.

Rubber gaskets 0.05" thick were used to seal the water channels (Fig 1c).

After the plates were bolted together, both sides were polished with a buffer and rouge.

The ends of all the water channels were sealed with Decon two-ton epoxy cement. The water channels were fed from a manifold (22 Fig 6)[2]. The water was then routed through a valve and into an identical manifold to drain. The flow for both inactive and active channels was controlled by stop cocks (24 Fig 5).

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[2] Numbers before the figure number refer to specific items in the picture.

Thermocouples, connected to selector boxes (25 Fig 5), were installed in the inlet manifold and the active and inactive water channel outlet tubes. Voltage output of the thermocouple system was measured by a Rubicon potentiometer, model 2732, and a Brown portable potentiometer, model 126W.

The test section (Fig 6), fabricated from ¼" 7075 alloy aluminum, housed the flat plate and was 6¼" x 5 7/8" x 24½". It was designed to obtain velocities in the range of 50 to 200 ft/second. The top of the test section pivoted through small angles (23 Fig 6) so that there was a negligible change in pressure gradient in the longitudinal direction over the plate. This top side had three water jackets with baffles to insure that no radiation heat transfer occurred between the top of the test section and the flat plate (23 Fig 7).

At five positions in the test section, static pressure taps were installed (.03" diameter). In approximately the center of the test section a pitot static tube and a radiation shielded total temperature probe were installed.

A number 25 Norblo high speed exhaust fan was used to circulate the air mixture (14 Fig 1). The fan was driven by a DC electric dynamometer which delivered a maximum of six horsepower and has a speed range from 1400 to 4000 RPM (28 Fig 5).

The heater section (15 Fig 1) was composed of thirteen 1.5 ohm General Electric nichrome resistors. Five internal and one external resistors were connected in series to a 250 volt DC power source (13 Fig 1). Four resistors were connected to a 110 volt AC temperature controller and the other four were not used. Three 18" x 14" mesh screens (21 Fig 6) were installed before the test section to straighten the flow.

CHAPTER III

EXPERIMENTAL PROCEDURE

The initial phase of this work was carried out using air alone to determine if correlation existed between the theory and experimentation.

Three correlation runs using air alone were made at 52 ft/second, 80 ft/second, and 107 ft/second. Two runs were made with particles at 52 ft/second. In each case the steady state free stream temperature was approximately 215 ^{o}F.

At the beginning of each run, the thermocouples were checked for proper operation using the building's cold tap water. If all thermocouples read the same value, they were working. Checking was accomplished after steady state conditions were reached between the water and the thermocouple surroundings. Thermocouples were located in the active side of the plate, in the outlet water tubes of both the active and inactive water channels, and in the common manifold inlet which supplied water to both active and inactive channels.

After thermocouple checking was accomplished, the free stream temperature control (22 Fig 5) was turned on and set at 215 ^{o}F. The temperature control regulated one module (mod 2) of heaters, one module (mod 1) was on continuously, and another module (mod 3) of heaters was left inactive for later use in higher temperature ranges. After modules 1 and 2 were energised, steady state was reached in about one hour.

The top side of the test section (21 Fig 6) was adjusted so the pressure gradient in the longitudinal direction could be varied. Static pressure readings were taken on the inclined manometer (21 Fig 5). The velocity pressure readings were taken on the precision U-tube manometer (23 Fig 5). The free stream temperature was taken from the total temperature probe and read out on the potentiometer.

Before steady state was reached, the water flowing in the active and inactive water channels was adjusted until the temperature of all the active side channels of the flat plate were at a constant temperature plus or minus 2.5 ^{o}F. The inactive flow valves were simultaneously adjusted so that corresponding inactive and active outlet temperatures were within plus or minus 5 ^{o}F. This feature was designed into the system so that a minimum of heat transfer would occur between active and inactive water channels.

The flow rate of the active channels was measured by valving the flow into a graduated cylinder and measuring the change in volume per unit time measured by a stop watch.

The same procedure was followed for the runs with particles entrained in the air.

CHAPTER IV

SUMMARY OF APPLICABLE THEORY

I. FILM COEFFICIENTS IN TURBULENT FLOW

Over a flat plate at constant temperatures for a Prandtl number greater than 0.6, Colburn (3) gave the following equations for heat transfer in turbulent flow over a flat plate at constant temperature in the range of a Reynolds number between 5×10^5 and 10^7.

$$N_{STx} \, N_{PR}^{2/3} = 0.0292 \, N_{REx}^{-0.2} \tag{1}$$

and the local Nusselt number to be

$$N_{Nux} = 0.0292 \, N_{REx}^{4/5} N_{PR}^{1/3} \tag{2}$$

Eckert (6) suggested that the physical properties be evaluated at the temperature

$$t_e = \frac{0.1 \, N_{PR} + 40}{N_{PR} + 72} \left(t_s - t_\infty \right) + t_\infty \tag{3}$$

as given in Hoffman's paper (8).

From equation 2, the local turbulent film coefficient is

$$h_{tx} = 0.0292 \, k \left(\frac{1}{x} \right) \left(\frac{U_\infty x}{\nu} \right) N_{PR}^{1/3} \tag{4}$$

combining the parameter X

$$h_{tx} = 0.0292 \, k \left(\frac{1}{x^{1/5}} \right) \left(\frac{U_\infty}{\nu} \right) N_{PR}^{1/3} \tag{5}$$

II. FILM COEFFICIENTS IN LAMINAR FLOW

For laminar flow past a flat plate at constant temperature, the local Nusselt number is

$$N_{Nux} = 0.332 \, (N_{REx})^{1/2} (N_{PR})^{1/3} \tag{6}$$

where the physical properties are evaluated at the temperature

$$t_a = t_\infty + 0.58(t_w - t_\infty) \tag{7}$$

III. LAMINAR BOUNDARY LAYER APPROXIMATIONS

The laminar boundary layer equations are only approximations to the governing equations that describe the flow. After an order of magnitude analysis is accomplished in the boundary layer, the steady state X-momentum equation in two dimensions (ρ, μ constant) is

$$u\frac{\partial u}{\partial x} + v\frac{\partial u}{\partial y} = -\frac{1}{\rho}\frac{\partial P}{\partial x} + \nu\left(\frac{\partial^2 u}{\partial x^2} + \frac{\partial^2 u}{\partial y^2}\right) \tag{8}$$

and is reduced to

$$u\frac{\partial u}{\partial x} + v\frac{\partial u}{\partial y} = -\frac{1}{\rho}\frac{\partial P}{\partial x} + \nu\frac{\partial^2 u}{\partial y^2} \tag{9}$$

The steady state Y-momentum equation (ρ, μ constant) is

$$\frac{\partial P}{\partial y} = 0 \tag{10}$$

The important result of the above equations is that the normal pressure gradient is zero. This implies that the pressure gradient in the boundary layer dp/dx is equal to the pressure gradient outside the boundary layer or in the potential flow region. For flow around a thin flat plate, the velocity in the potential flow region remains constant. Bernouilli's equation neglecting friction (potential flow) is

$$U_\infty\frac{\partial U_\infty}{\partial x} + \frac{1}{\rho}\frac{\partial P}{\partial x} = 0 \tag{11}$$

If U_∞ is constant, then dp/dx is zero. The test section which housed the flat plate was designed around this fact. The top of the test sections was designed to pivot through small angles to adjust the longitudinal pressure gradient to zero.

CHAPTER V

EQUATIONS USED TO REDUCE DATA

The local heat transfer coefficient is defined by the following equation

$$h_x = \frac{q_x'}{t_{\infty} - t_w} \tag{1}$$

The local wall heat flux was obtained by

$$q' = \dot{m} c_p (t_o - t_{in}) / A_x \tag{2}$$

where $\dot{M}$ is the flow rate of water through each active plate channel.

CHAPTER VI

EXPERIMENTAL RESULTS

I. SYSTEM PERFORMANCE WITH AIR ALONE

Three experimental runs were made using air alone at free stream velocities of 52 ft/second, 80 ft/second, and 107 ft/second. Referring to Figure 7a, it can be seen that there was a wide scatter in the heat transfer coefficients in each run. The majority of the heat transfer coefficients fell between the turbulent and laminar theory values. Figure 8 shows the Nusselt number variation versus the local Reynolds number.

Of the three runs, run 4 shows the closest correlation with the turbulent theory in the Reynolds number range of 2.5×10^5 to 7.5×10^5. The scatter in the data may have resulted from one or more of the following reasons:

1) The flow may have been primarily in the transition region.

2) The stop cocks could not vary the flow rate in the active and inactive water channels to the fine adjustment required to maintain a uniform surface temperature.

3) The turbulence level varied from four per cent to 0.4 per cent over the flat plate. See results of Baines and Peterson, Figure 9 (in this work, the bar width was .009").

Dryden (5) reported that transition occurs at a Reynolds number of 10^5 when the turbulence level is three per cent. This result can only give us an indication of where the transition region would occur because Dryden's results were based on a uniform turbulence level while the turbulence level in this work varies along the plate.

Because of the variation in the turbulence level and the wide scatter in the data points, no definite conclusions could be reached about the flow regime.

II. SYSTEM PERFORMANCE WITH PARTICLES

Two runs were made with an air-particle mixture at a free stream velocity of 52 ft/second and a loading ratio of 2 ($W_s/W = 2.0$). The heat transfer coefficients versus the distance from the leading edge are plotted in Figure 7.

It should be noted that there is less scatter in the data points for the runs with particles as compared with the run without particles at the same free stream velocity (Fig 7).

Again, no conclusions are justified regarding the comparison of the air and air-particle runs because of the lack of valid runs.

The pressure variation in the longitudinal direction was negligible for all runs (Fig 10).

The maximum longitudinal temperature variation on the plate surface was 5^{o} and could have been reduced considerably by a finer adjustment of the water channel flow in the active side. The variations are plotted in Figures 11 and 12.

A transverse velocity profile was taken 1/8" downstream from the screen straighteners. The profile was uniformly developed at 3/8" from both walls (Fig 13).

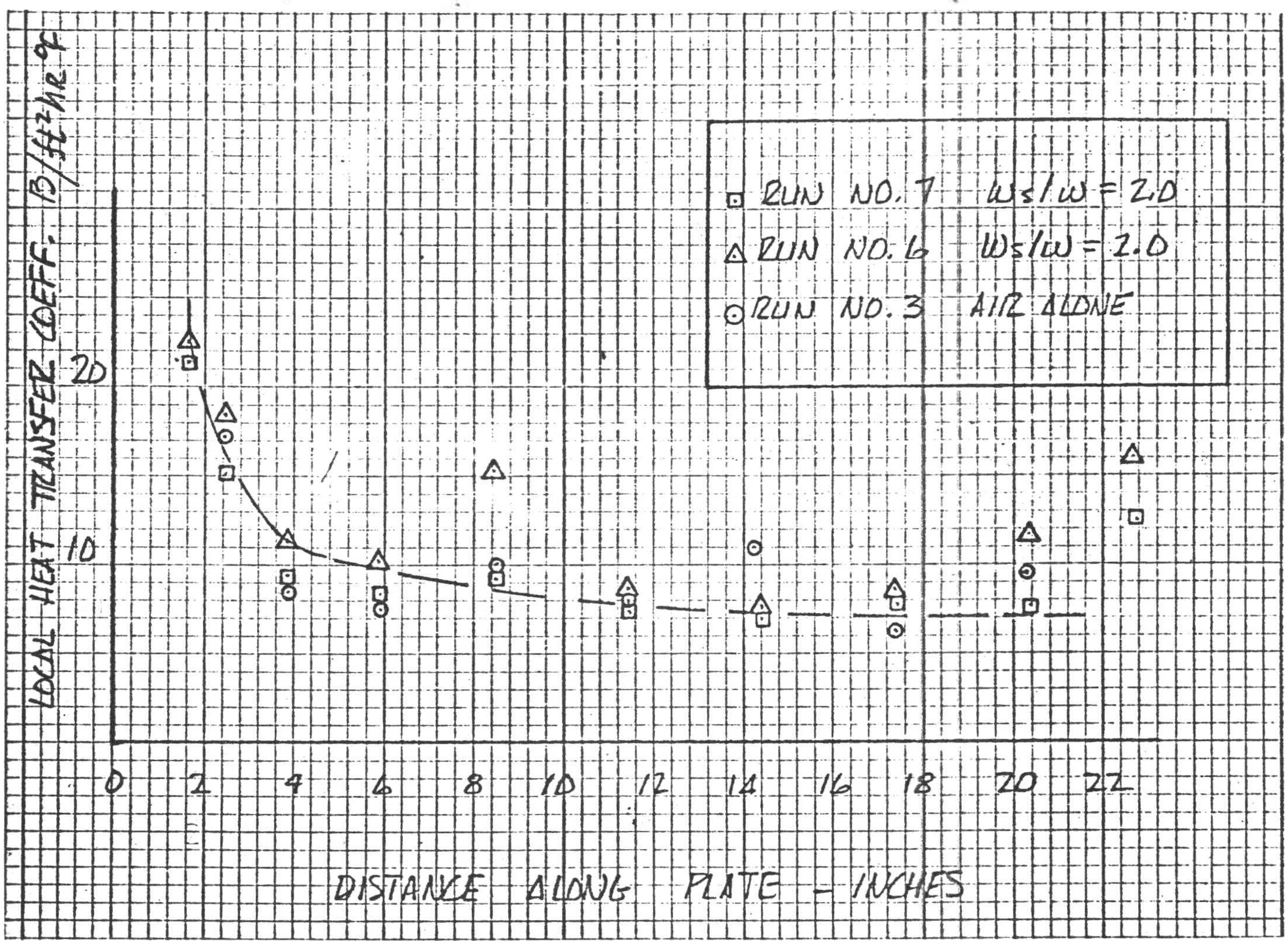

FIGURE 7

LOCAL HEAT TRANSFER COEFFICIENTS VS. DISTANCE FROM LEADING EDGE

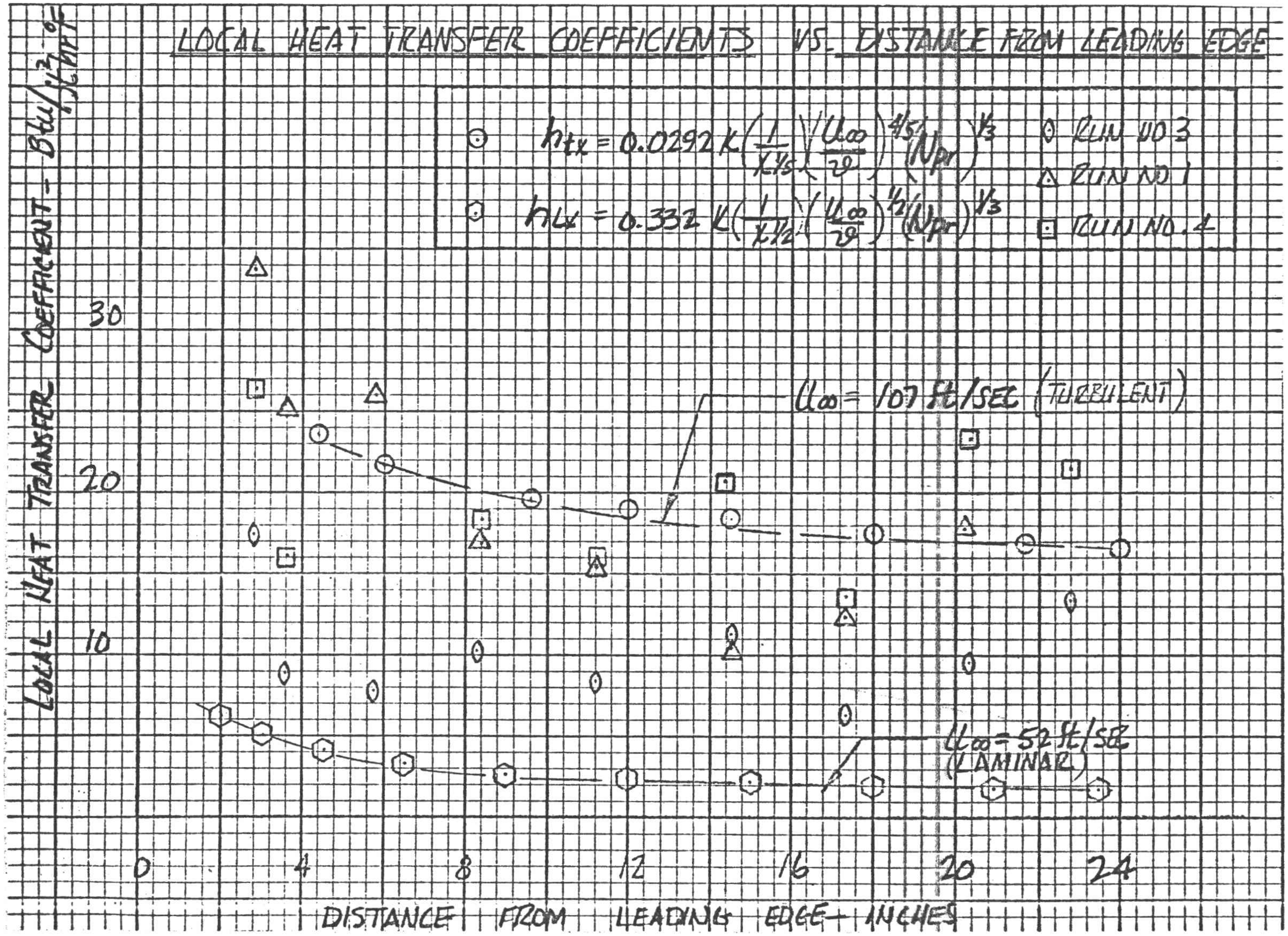

FIGURE 7a

LOCAL HEAT TRANSFER COEFFICIENTS VS. DISTANCE FROM LEADING EDGE

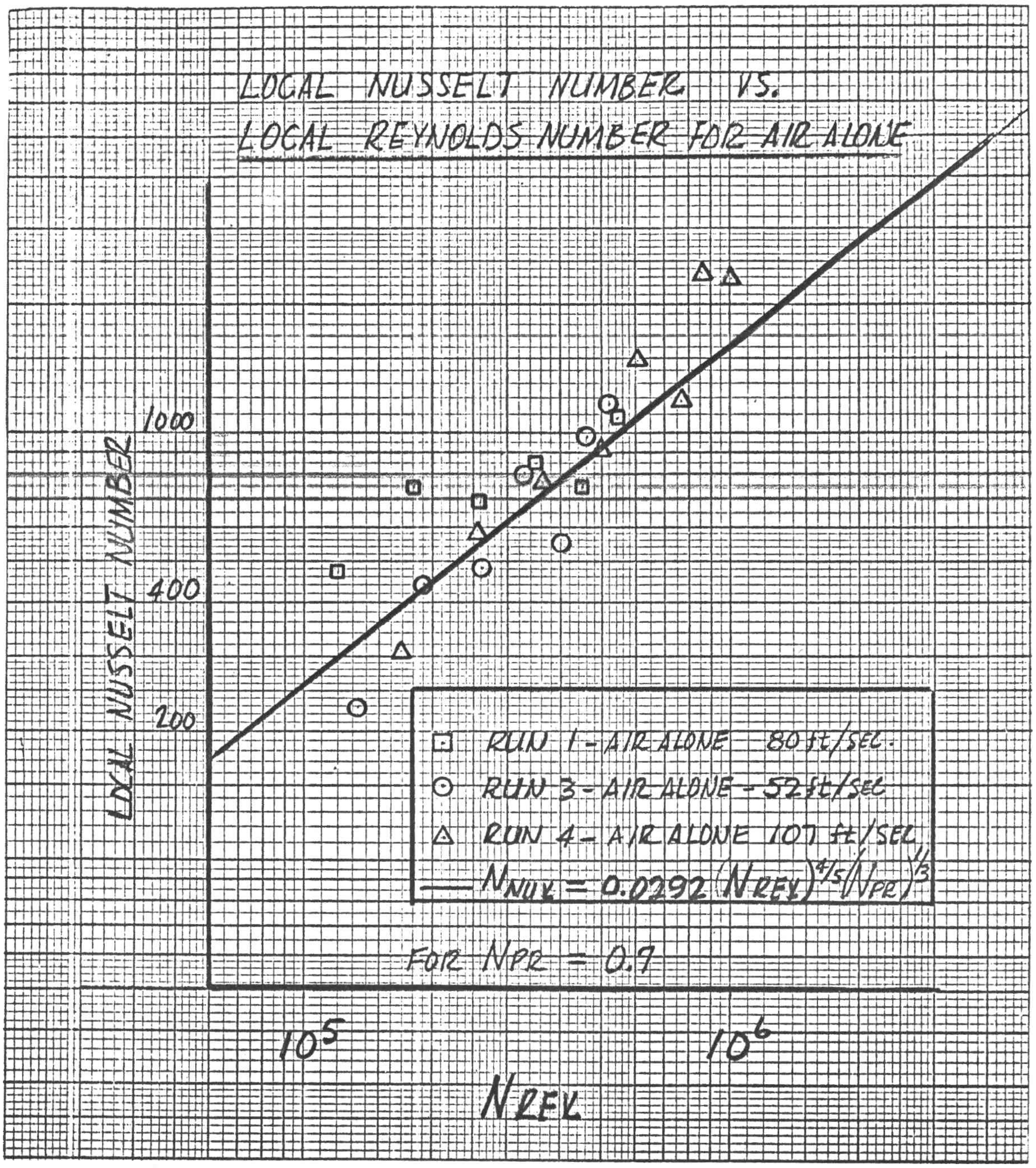

FIGURE 8

LOCAL NUSSELT NUMBER VS. LOCAL REYNOLDS NUMBER ALONE

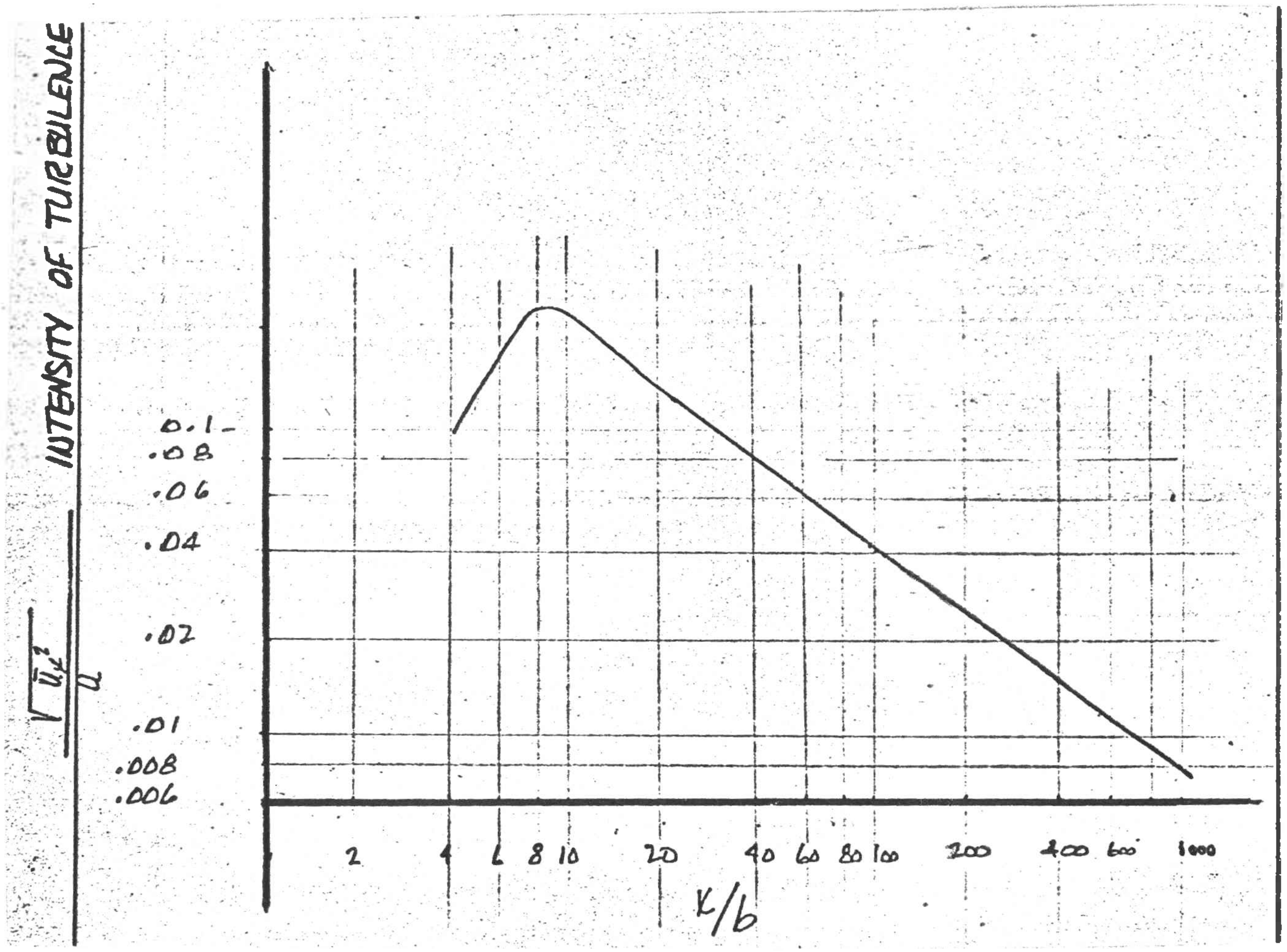

FIGURE 9

INTENSITY OF TURBULENCE VS. DISTANCE FROM SCREEN/BAR WIDTH OF SCREEN

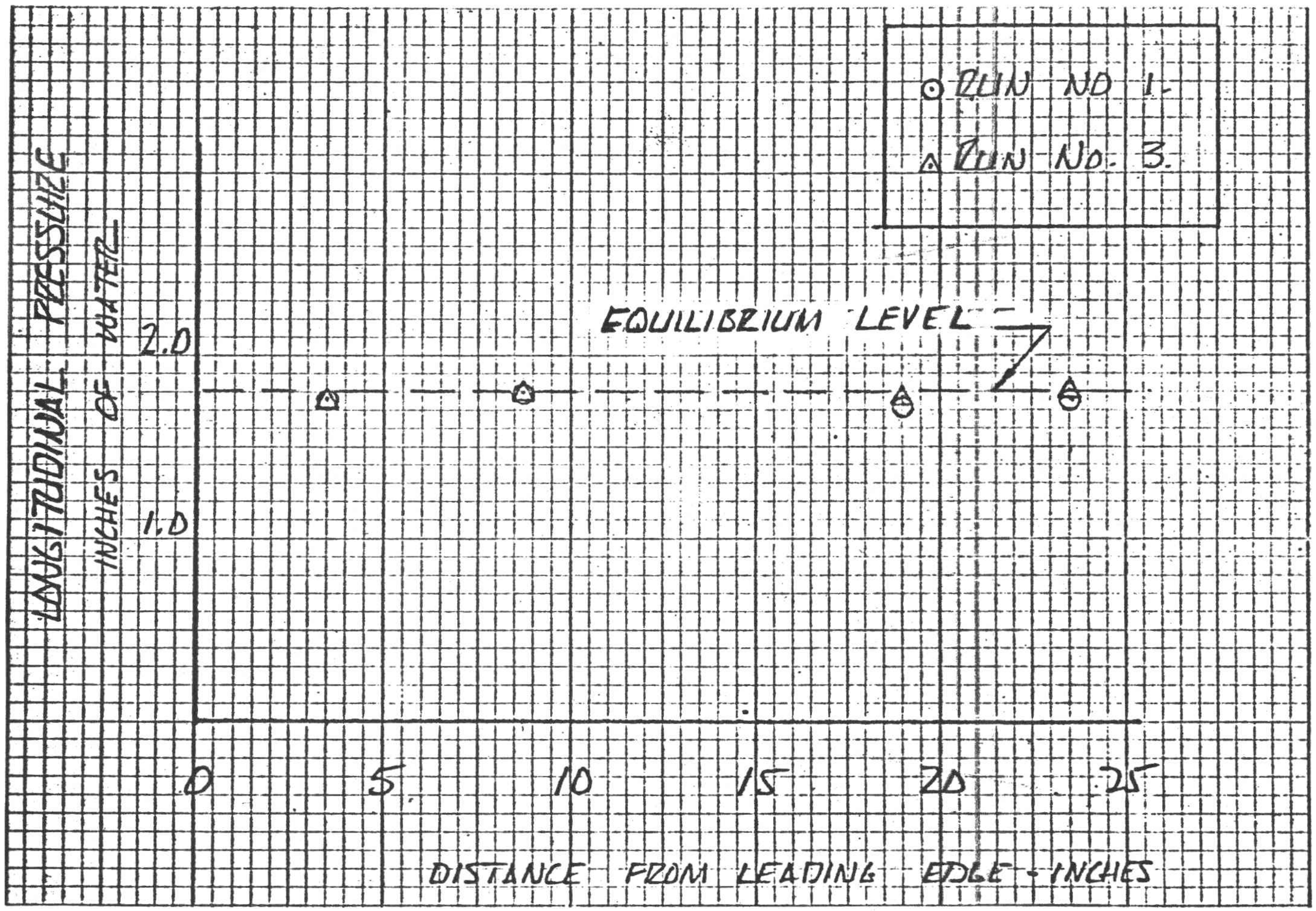

FIGURE 10a

LONGITUDINAL PRESSURE VARIATION

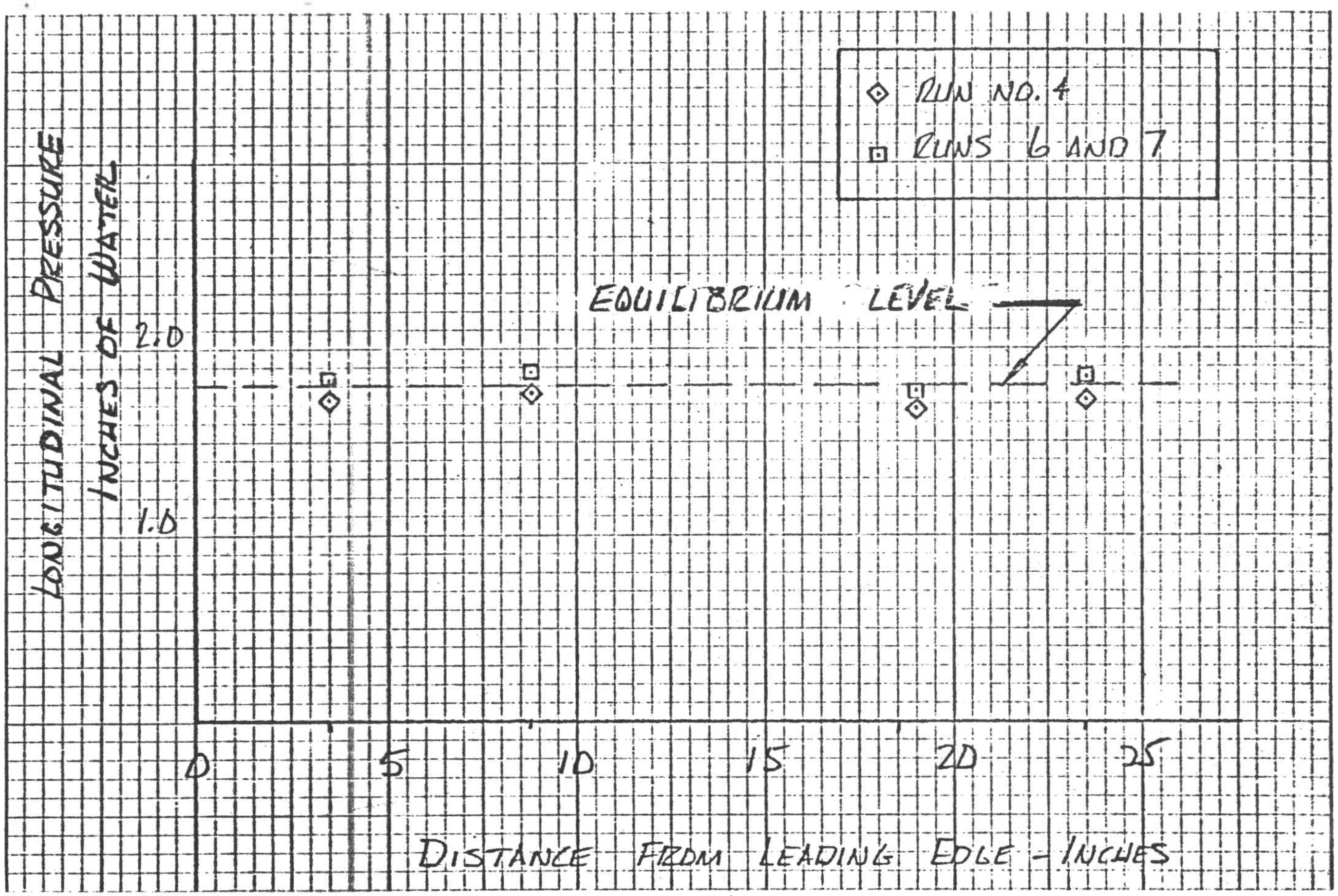

FIGURE 10b

LONGITUDINAL PRESSURE VARIATION

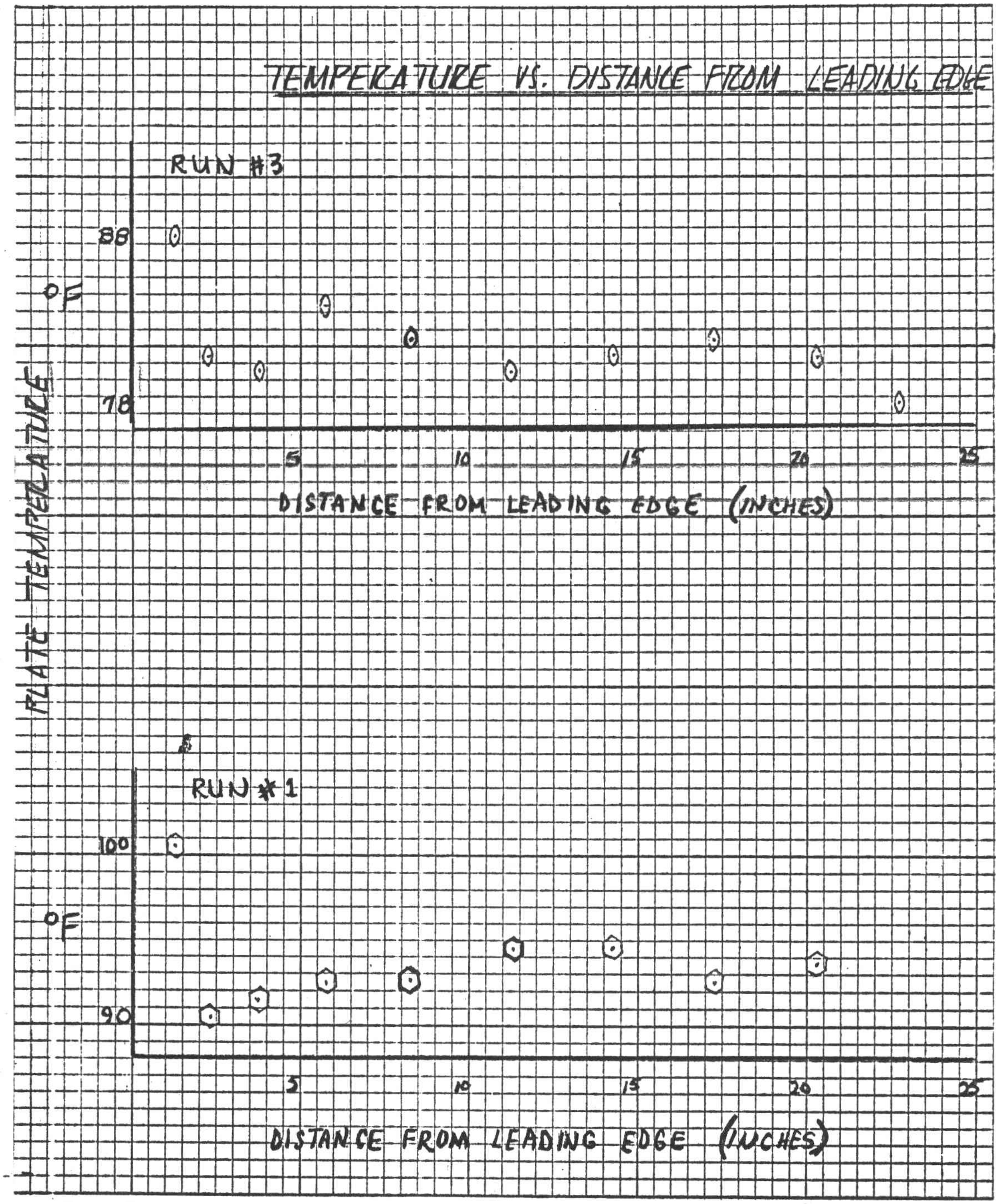

FIGURE 11

SURFACE TEMPERATURE VS. DISTANCE FROM LEADING EDGE

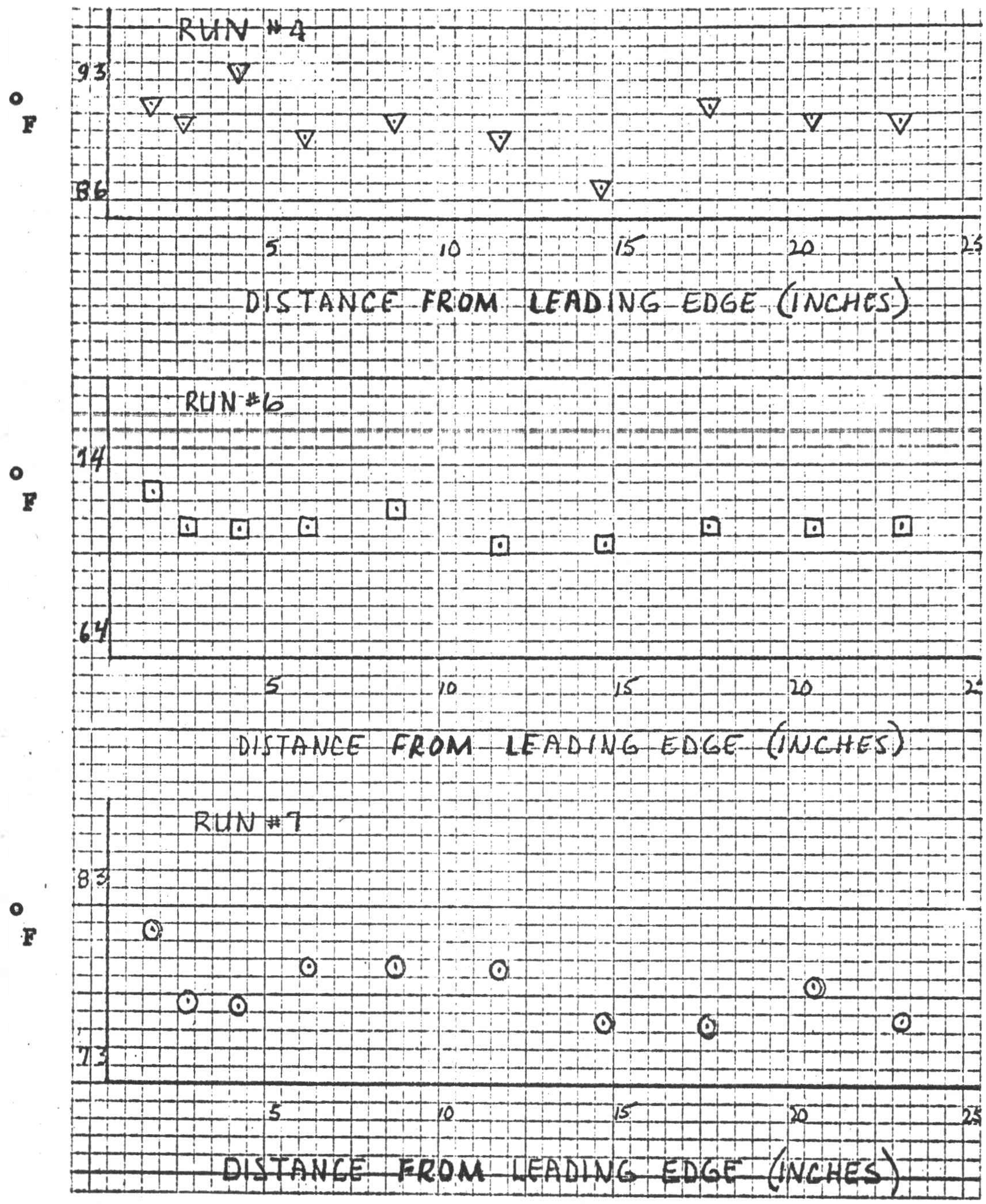

FIGURE 12

SURFACE TEMPERATURE VS. DISTANCE FROM LEADING EDGE

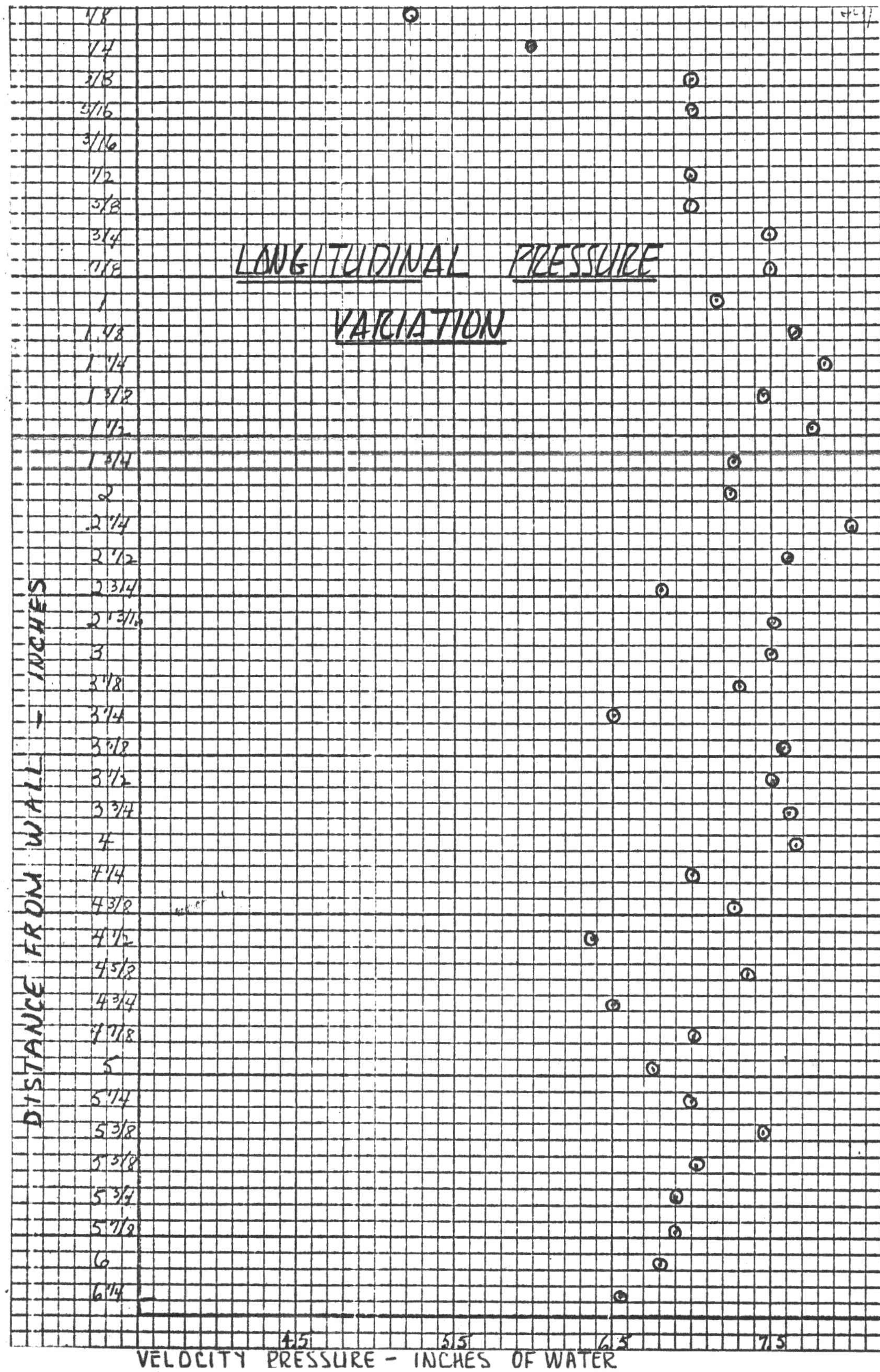

FIGURE 13

LONGITUDINAL PRESSURE VARIATION

CHAPTER VII

RECOMMENDATIONS

1. The plate thickness could be reduced by approximately one-half using the thermocouple technique of butt welding the wires and imbedding them in a shallow groove on the surface which enables a plate of much thinner material to be fabricated.

2. Reduce the center insulator thickness.

3. Eliminate the rubber gaskets by sealing the water channels with Decon two-ton epoxy.

4. The leading edge should be designed with at most a 30^{o} bevel.

5. Instead of using Decon two-ton epoxy glue which can only withstand 200^{o} F, General Electric RTV-102 silicone rubber adhesive sealent could be used as it withstands temperatures to 500^{o} F.

6. The present flat plate can be improved by activating the first water channel and using RTV-102 sealent. Sauereisen would not work to seal the water channel on the leading edge because water is a solvent.

7. Relocate the temperature probe in the test section so it does not disturb the flow.

8. Extend Depew's work by using 600 mesh carborundum grinding compound (sub-micron to 25 micron).

9. For further studies in the laminar region, it will be necessary to install a smooth duct following the screens to reduce the intensity of turbulence.

10. Replace all stop cocks with needle valves for finer adjustment.

CHAPTER VIII

CONCLUSIONS

The objectives of this thesis were met; i.e. an apparatus was designed and built which measured heat transfer coefficients over a horizontal flat plate with nearly uniform surface temperature and negligible pressure gradient.

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APPENDIX

RUN # 1   AIR ALONE $T_\infty = 217°F$ $T_s = 93°F$ $U_\infty = 80$ ft/sec $(5) = 10^5$

Station	q_x B/hr	A_x ft²	T_{OUT}	T_{IN}	h_x	x(ft)	x/K	N_{NUX}	N_{REX}
1	568	.0827	53.2	47	55.4	.1125	6.63	368	$4.2 \cdot 10^{(4)}$
2	181	.0433	66.8	47	33.8	.196	11.50	388	$7.3 \cdot 10^{(4)}$
3	258	.0833	67.8	47	25.0	.321	18.90	475	$1.19^{(5)}$
4	280	.087	56.9	47	26.0	.487	28.60	745	$1.81^{(5)}$
5	246	.13	84	47	16.9	.696	41.00	695	$2.60^{(5)}$
6	246	.13	84	47	15.2	.948	55.80	848	$3.54^{(5)}$
7	171	.13	68.6	47	10.5	1.20	70.50	740	$4.47^{(5)}$
8	200	.13	74	47	12.3	1.45	85.00	1040	$5.40^{(5)}$
9	290	.13	72.6	47	17.9	1.69	99.5	1770	$6.29^{(5)}$
10		.0807	74	47					

RUN #3 AIR ALONE $T_\infty = 218°F$ $T_s = 73°F$ $U_\infty = 52$ ft/sec											
Station	q_x B/hr	A_x ft²	T_{OUT}	T_{IN}	h_x	Corrected h_x	x (ft)	x/k	N_{NUX}	N_{REX}
1	248	.0827	55.4	48	21.9	22.0	.1125	6.63	146	$3.09^{(4)}$
2	88	.0433	66.0	48	14.8	8.90	.196	11.5	388	$7.3 \cdot 10^{(4)}$
3	86.5	.0833	60.5	48	7.55	8.90	.321	18.9	172	$8.82^{(4)}$
4	78.6	.087	59.6	48	6.6	7.8	.487	28.6	230	$1.33^{(5)}$
5	152	.13	71.5	48	8.53	10.06	.696	41.0	421	$1.91^{(5)}$
6	127	.13	71	48	7.13	8.41	.948	55.8	480	$2.69^{(5)}$
7	170	.13	56.7	48	9.54	11.25	1.20	70.5	810	$3.30^{(5)}$
8	95	.13	69	48	5.31	6.26	1.45	85.0	550	$4.00^{(5)}$
9	145	.13	68.5	48	8.14	9.60	1.69	99.5	980	$4.65^{(5)}$
10	135	.0807	63	48	12.2	14.4			1164	$5.20^{(5)}$

RUN #4 AIR ALONE $T_\infty = 215°F$ $T_s = 91°F$ $U_\infty = 107$ ft/sec

Station	q_x B/hr	A_x ft²	ACTIVE WATER T_{OUT}	CHANNEL T_{IN}	h_x	Corrected h_x	(ft) X	x/K	N_{NUX}	N_{REX}
1	425	.0827	56.0	47	41.2	45.3	.1125	6.78	307	$5.81^{(4)}$
2	121	.0433	73.0	47	22.3	26.3	.196	11.8	310	$10.1^{(4)}$
3	142	.0833	65.5	47	13.7	16.1	.321	19.3	311	$1.66^{(5)}$
4	186	.087	61.5	47	17.2	20.3	.487	29.4	593	$2.52^{(5)}$
5	246	.13	73.0	47	15.5	18.3	.696	42.0	770	$3.62^{(5)}$
6	219	.13	71.1	47	13.5	15.9	.948	57.1	910	$4.93^{(5)}$
7	285	.13	62.1	47	17.5	20.6	1.20	72.3	1480	$6.20^{(5)}$
8	181	.13	64.2	47	11.3	13.3	1.45	87.5	1170	$7.50^{(5)}$
9	318	.13	65.5	47	19.6	23.1	1.69	102	2360	$8.73^{(5)}$
10	184	.0807	66.0	47	18.4	21.7	1.90	114	2480	$9.80^{(5)}$

RUN #6	$W_s/W = 2.0$	$T_\infty = 215°F$	$T_s = 70°F$	$U_\infty = 52$ ft/sec					
Station	q_x B/hr	A_x ft²	T_{OUT}	T_{IN}	h_x	X(ft)	x/K	N_{NUX}	N_{REX}
1	273	.0827	52.5	49.8	22.8	.1125	6.63		3.09 (4)
2	116	.0433	63.5	49.8	18.5	.196	11.50		5.38 (4)
3	139	.0833	57.5	49.8	11.5	.321	18.90		8.82 (4)
4	131	.087	55.5	49.8	10.4	.487	28.60		1.33 (5)
5	226	.13	62.5	49.8	20.3	.696	41.00		1.91 (5)
6	166	.13	63.2	49.8	8.8	.948	55.80		2.69 (5)
7	143	.13	52.5	49.8	7.59	1.20	70.50		3.30 (5)
8	163	.13	57.8	49.8	8.64	1.45	85.0		4.00 (5)
9	226	.13	61.8	49.8	11.9	1.69	99.5		4.65 (5)
10	189	.0807	56.9	49.8	16.0				5.20 (5)

RUN #9	$W_s/W = 2.0$		$T_\infty = 218°F$		$T_s = 77°F$		$U_\infty = 52$ ft/sec		
Station	q_x B/hr	A_x (ft²)	$\Delta T = (T_O - T_I)$	T_{IN}	h_x	X(ft)	X/K	N_{NUX}	N_{REX}
1	254	.0827	5.1	49.8	21.8	.1125	6.63		3.09 (4)
2	94.6	.0433	20.0	49.8	15.5	.196	11.50		5.38 (4)
3	114	.0833	11.5	49.8	9.66	.321	18.90		8.82 (4)
4	107	.087	9.2	49.8	8.69	.487	28.60		1.33 (5)
5	173	.13	18.3	49.8	9.43	.696	41.00		1.91 (5)
6	134	.13	20.0	49.8	7.30	.948	55.80		2.69 (5)
7	132	.13	9.2	49.8	7.19	1.20	70.50		3.30 (5)
8	145	.13	15.9	49.8	7.90	1.45	85.0		4.00 (5)
9	145	.13	22.6	49.8	7.90	1.69	99.5		4.65 (5)
10	141	.0807	15.5	49.8	12.4				5.20 (5)

ABSTRACT

DESIGN AND FABRICATION OF AN APPARATUS TO DETERMINE HEAT TRANSFER COEFFICIENTS IN AN AIR-PARTICLE MIXTURE OVER A HORIZONTAL FLAT PLATE AT UNIFORM TEMPERATURE

An Abstract

of A Thesis

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Master of Science

by

Louis G. Hunter, Jr.

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An apparatus was designed and fabricated which would measure the heat transfer from an air stream containing entrained particles to a horizontal flat plate at nearly uniform temperature and negligible pressure gradient. The particles used in this work were glass powder (about 200 mesh, 74 micron screen opening) and the loading ratio (ws/w) was 2.

Correlation between the theory and data was not achieved for the runs with air alone. Therefore, no conclusions were justified regarding the comparison of the data from the air and air particle runs.

APPROVED.

August 1963
Date