PRESSURE DROP IN A TUBE WITH FORCED CONVECTION HEAT TRANSFER IN THE NONBOILING REGION

By

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IN THE NONBOILING REGION

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SYMBOLS

V_1	Volume rate of flow, ft^3 per sec
C	Orfice coefficient of discharge
h_1	Differential head across orifice, ft of flowing fluid
g	Acceleration of gravity, ft per sec
h	Film coefficient of heat transfer, Btu per hr ft^2 ^oF
f	Friction factor
N_{Re}	Reynolds number
N_{Pr}	Prandtl number
D	Inside diameter of test section, ft
k	Thermal conductivity, Btu per hr ft ^oF
Q_1	Total test section heating, Btu per hr
c_p	Specific heat at constant pressure, Btu per lb hr ^oF
P	Total test section I^2R loss, kilowatts
L	Test section length, inches
q"	Heat flux, Btu per ft^2 hr
G	Mass flow rate, lb per ft^2 sec
A_t	Total inside tube surface, ft^2
M	Mass rate of flow, lb per hr
ρ	Density, lb mass per ft^3
V	Velocity, ft per sec
Δt_i	Temperature rise across the film, ^oF
μ	Viscosity, lb mass per ft sec

CHAPTER I

INTRODUCTION

The need for greater understanding of the mechanisms of heat transfer and pressure drop for fluids exposed to heat fluxes in excess of those formerly encountered by industry in general, has become apparent with advent of nuclear reactors. Removal of thermal energy produced within the reactor core requires heat fluxes of the order of 500,000 Btu per hr sq ft. The purpose of this thesis is the determination of pressure drops encountered by water in flowing through a straight section of insulated horizontal tubing. The system pressure at intervals along the test section was observed with and without heat addition to the test section. Heat fluxes up to 310,000 Btu per sq ft hr were produced by using the tube itself as a resistance heater. The pressure profiles were predicted by using the Colburn relation to calculate the inside wall temperature of the test section. The friction factors along the test section were then calculated using the Blasius relation for isothermal flow, evaluating the variables of this equation at the inside tube wall temperatures. These predicted profiles were then compared with actual data to check the test results.

CHAPTER II

PREVIOUS INVESTIGATIONS

The relationships used to calculate the pressure drop of fluids flowing in pipes under isothermal flow conditions have been verified by extensive experimentation. These relationships depend on the use of a friction factor in evaluating pressure drop. The friction factor is a means of adjusting the equations for specific flow and surface roughness conditions. For smooth tubes the friction factor can be expressed by the Blasius relation:

$$f = \frac{(0.316)}{(N_{Re})^{1/4}}$$

If conditions are changed so that there is heat transfer across the inside tube wall, and hence a temperature change across the fluid, evaluation of the friction factor becomes more difficult.

In the evaluation of the inside wall temperatures, the work of S. J. Kaufman and R. W. Henderson (1) proved to be of interest. Their report dealt with forced convection heat transfer to water at high pressures and temperatures in the nonboiling region. Water was pumped under pressure through an electrically heated test section. The outside surface temperatures of the tube were measured and the data was used to compute inside wall temperatures. By assuming uniform heating throughout the test section, the Colburn relation was evaluated and the computed film coefficient was compared with the experimental value. In this way they were able to substantiate the Colburn correlation for heat transfer coefficients for heat fluxes of 175,000 to 600,000 Btu per hr sq ft.

Krieth and Summerfield (2) have presented papers on pressure drop of water flowing by forced convection through an electrically heated tube. They determined the inside wall temperature by measuring the outside wall temperature and using this information to calculate inside wall temperatures. They found that the nonisothermal friction coefficient could be expressed by the following relation:

$$\frac{f_{non}}{f_{iso.}} = \left(\frac{\mu_W}{\mu_B}\right)^{0.13}$$

Where f_{non} is the friction factor for nonisothermal flow and f_{iso} is the friction factor for flow at the same bulk temperature without any heating of the fluid. μ_w is the viscosity of the flowing fluid, evaluated at the wall temperature of the nonisothermal case, and μ_b is the viscosity evaluated at the bulk temperature. The experimental friction factors were predicted within 3 percent by the use of this relation for Reynolds numbers of 100,000 to 250,000.

CHAPTER III

EQUIPMENT

The basic design of the system is similar to a system already in operation at the Reactor Engineering Division of the Argonne National Laboratory and used by J. F. Mumm (3) and J. B. Reynolds (4). The layout of the system will be covered by starting at the storage tank in the loop and describing the parts in the order encountered as the fluid passes through the loop. The storage tank provides a supply of water at virtually constant positive head. The water then flows into the main pump and from there passes through one of two orifice plates placed in parallel in the line. From the orifice plates the water flows through the preheater and into the test section. A sight glass follows the test section followed in turn by four expansion valves in parallel. The water, then at a pressure approaching atmospheric, is directed through a shell and tube condenser or cooler depending on the state of the fluid. The condensate, or cooled liquid as the case may be, then passes to a rectangular tank, as shown in Fig. 1, which contains cooling coils for further reduction of temperature. A transfer pump then forces the water into the storage tank. This basic flow system is the same in the loops at the Argonne National Laboratory and the Oklahoma State University.

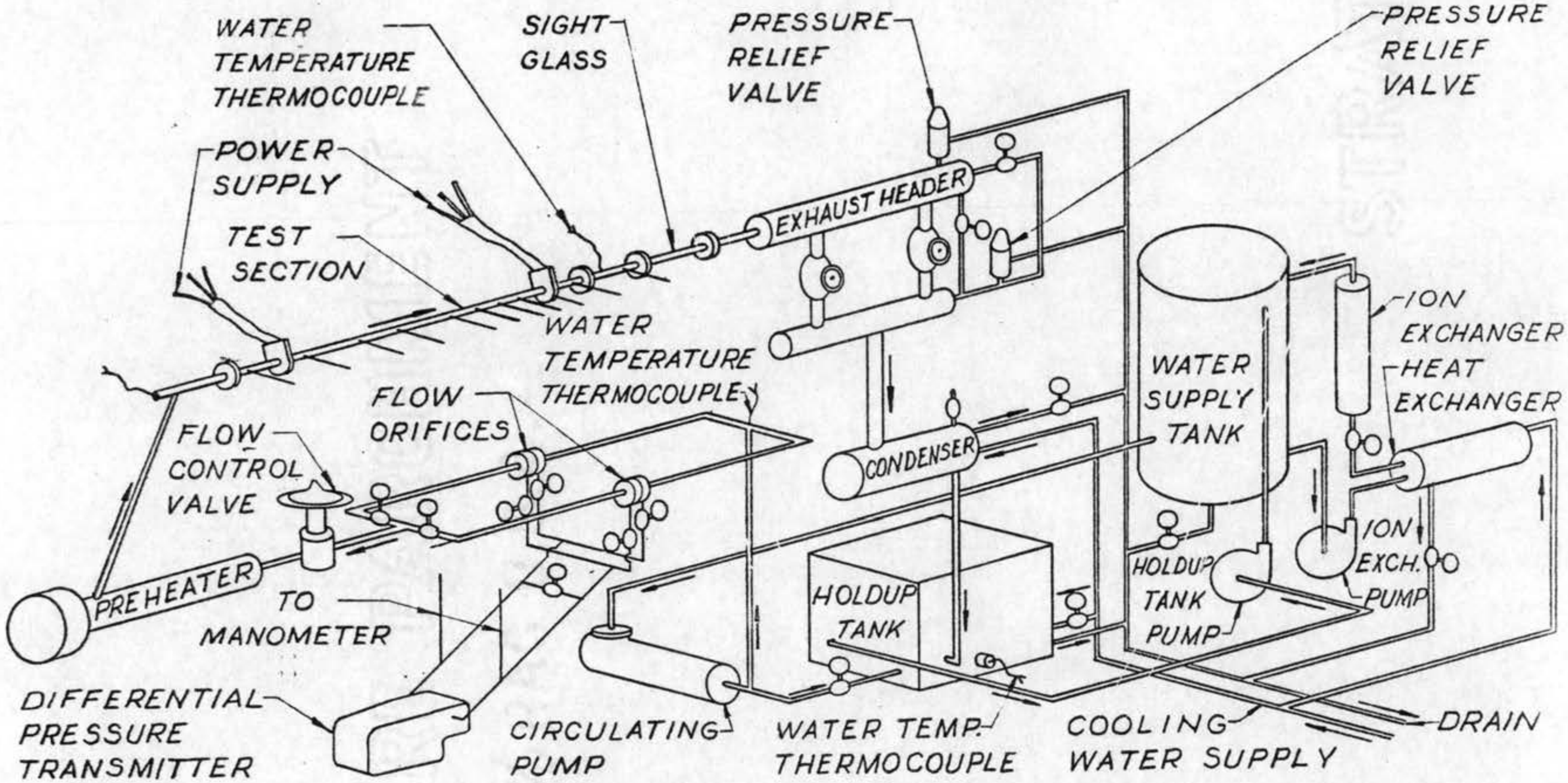


FIGURE 1
SCHEMATIC DIAGRAM
OF TEST FACILITY

Design of Components

Storage Tank. The storage tank is a twelve cubic foot cylindrical stainless steel container. The purpose of this tank is to provide a virtually constant positive head of two to three feet on the suction side of the main pump, and because of its volume, to operate as a thermal and a flow desurge tank. The tank provides a steady reference point in the system. The system fluid in the tank is maintained by a one third horsepower Moto centrifugal pump. The pump is of bronze and steel construction and is actuated by a mercury switch coupled to a float valve in the tank. This pump obtains its supply from the holdup tank.

Main Pump. One of the most important differences between this system and the Argonne facility is the main pump system. The Argonne system utilizes a pair of A P C O Turbine Pumps placed in series so as to achieve a steep pump discharge vs. capacity curve. The pumps were driven by a single speed motor, so that the choice of either flow rate or pressure dictated the setting of the other variable. The flow rate is very sensitive to any changes in discharge pressure when compared with the positive displacement pump.

The heat transfer loop at Oklahoma State University utilizes a Moyno 6M2 pump manufactured by Robbins and Myers. It is a positive displacement pump capable of continuous flow at pressures of 500 to 600 pounds. A plot of the discharge pressure vs. volume rate flow for the pump is shown in Fig. 2. The pump operates on the principle of an eccentric screw, so that fluid is trapped in pockets formed by the screw and the stator and forced through the pump. Because of the eccentric motion this particular pump has six pockets of fluid, or stages, thus providing six sealing surfaces that the

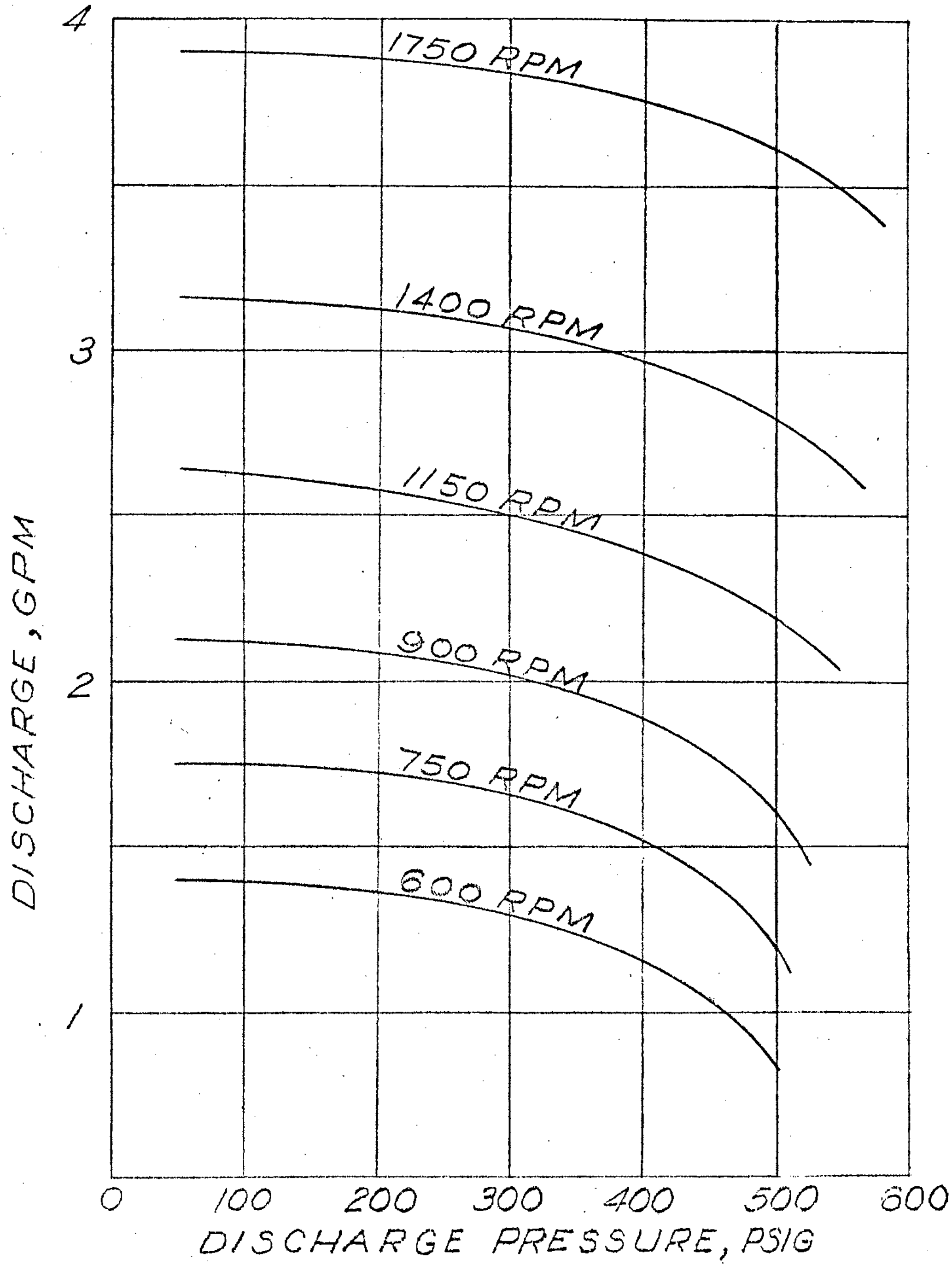


FIGURE 2
PUMP PERFORMANCE
FOR 6 STAGE MOYNO PUMP

high pressure fluid must overcome to reach the low pressure side of the pump. The pump is driven by a ten horsepower U. S. Varibelt motor capable of varying the pump speed continuously from 150 rpm to the maximum recommended by Robbins and Myers. Because of the large size of the driving motor in comparison with the requirements, the speed of the motor remains essentially unchanged when varying the system pressure. This feature was of considerable use when a specific set of conditions were being set into the system. A tachometer was installed on the control board which measured motor speed and served as a reference. The speed of the pump is controlled remotely from the instrument panel by mechanical means. Generally all metalic materials exposed to the system fluid were of stainless steel or copper.

Orifice Plates. In the original design the orifice plates were to serve two functions, as a means of sensing rate of flow and controlling rate of flow. There are two orifice plates set in parallel in the system. By means of valves the flow may be directed through either of the orifice plates. The sharp edged orifices are 0.453 and 0.353 inches in diameter, installed between flanges manufactured by the orifice manufacturer, Danial Orifice Fitting Company. The installation and manufacturing of the flanges and orifices were in accordance with the requirements set forth by the Brown Instrument Company and the American Meter Company (5) (6).

Preheater. The preheater was another piece of equipment that differed considerably from that in the Argonne loop. The preheater consists of a cylindrical stainless steel pressure vessel approximately one foot in diameter and three and three quarters feet in length. The thermal energy is supplied by twelve Chromalox five thousand watt immersion heaters ,

PLATE I. CONTROL PANEL

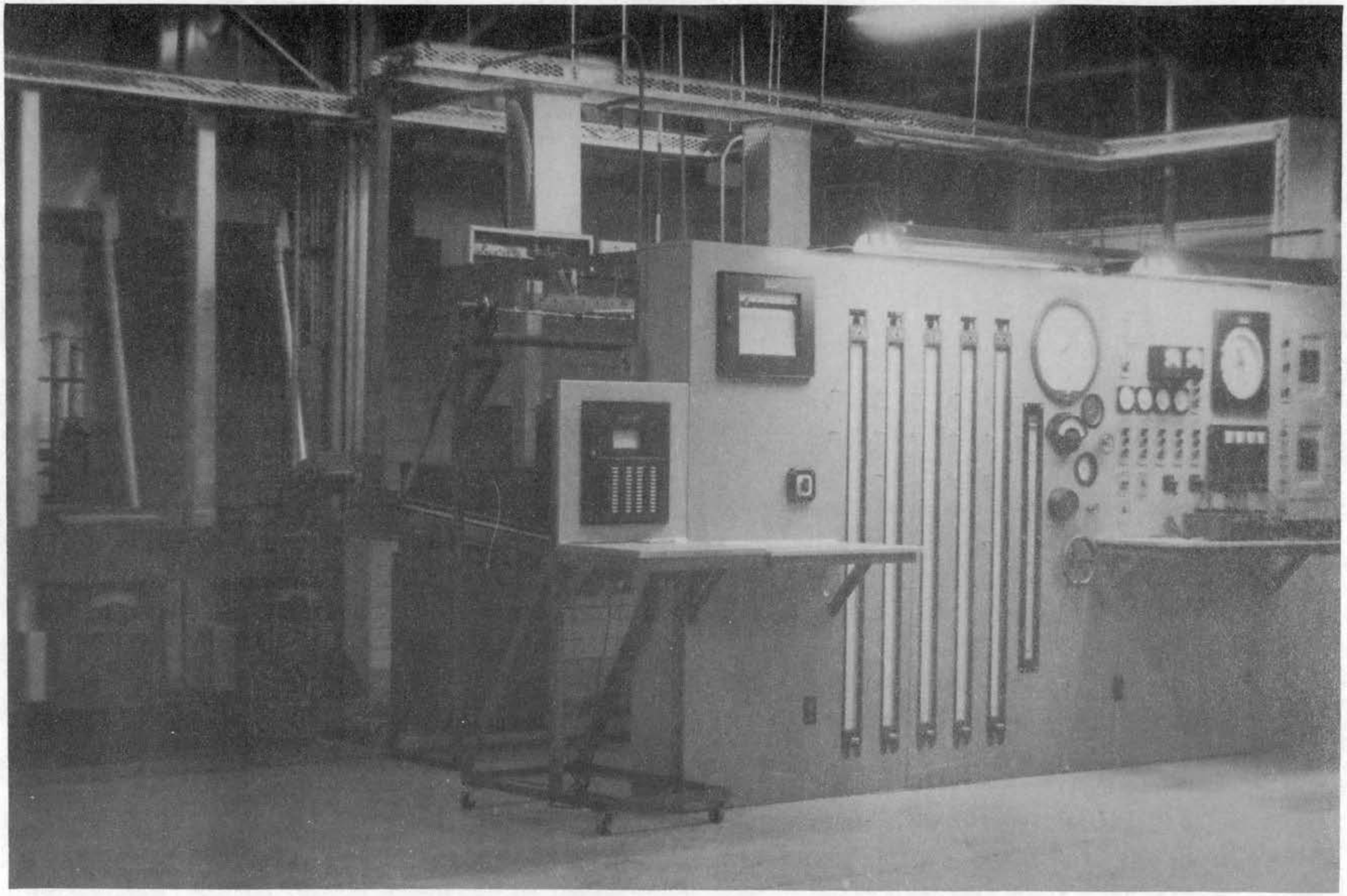


PLATE II. TEST SECTION

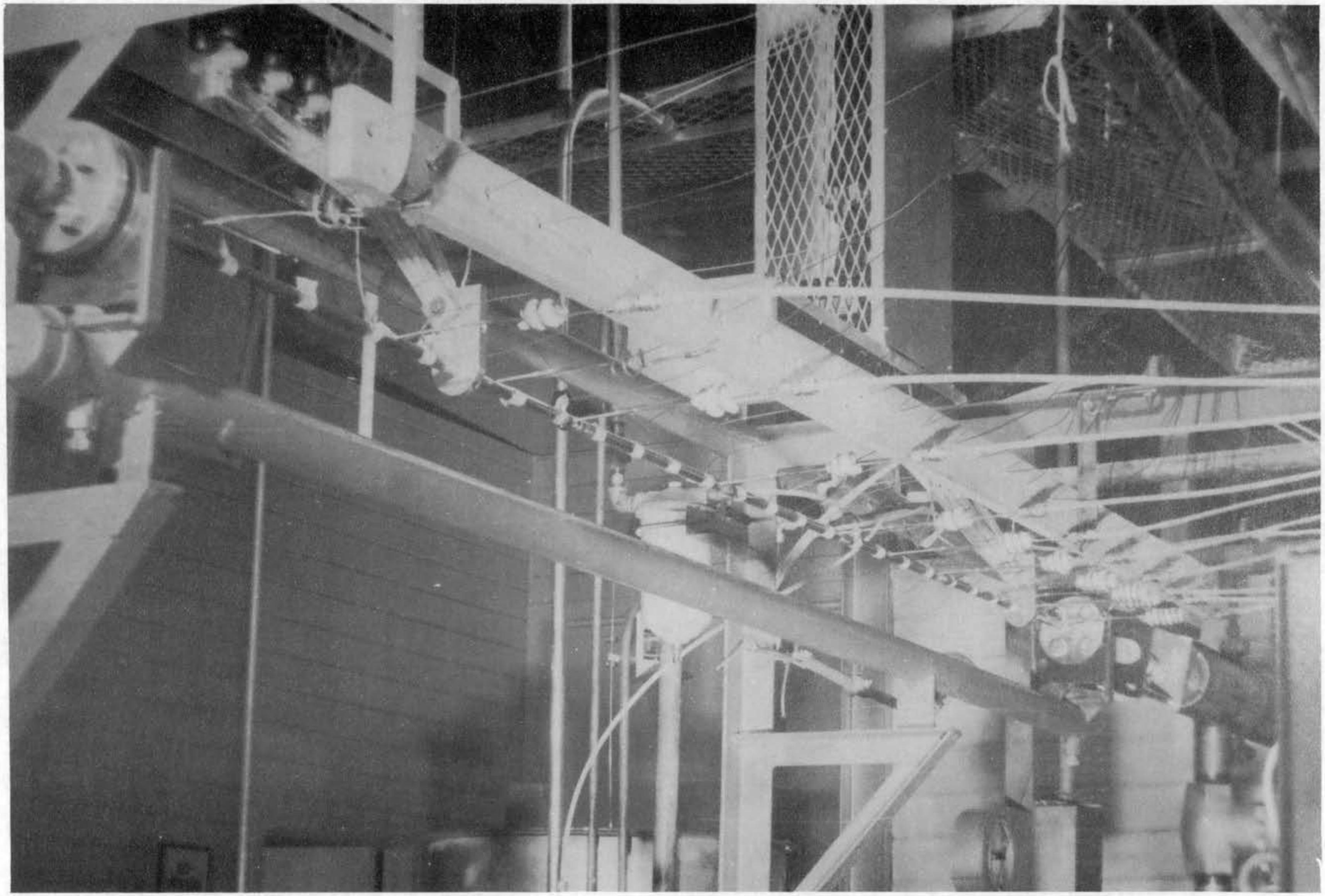
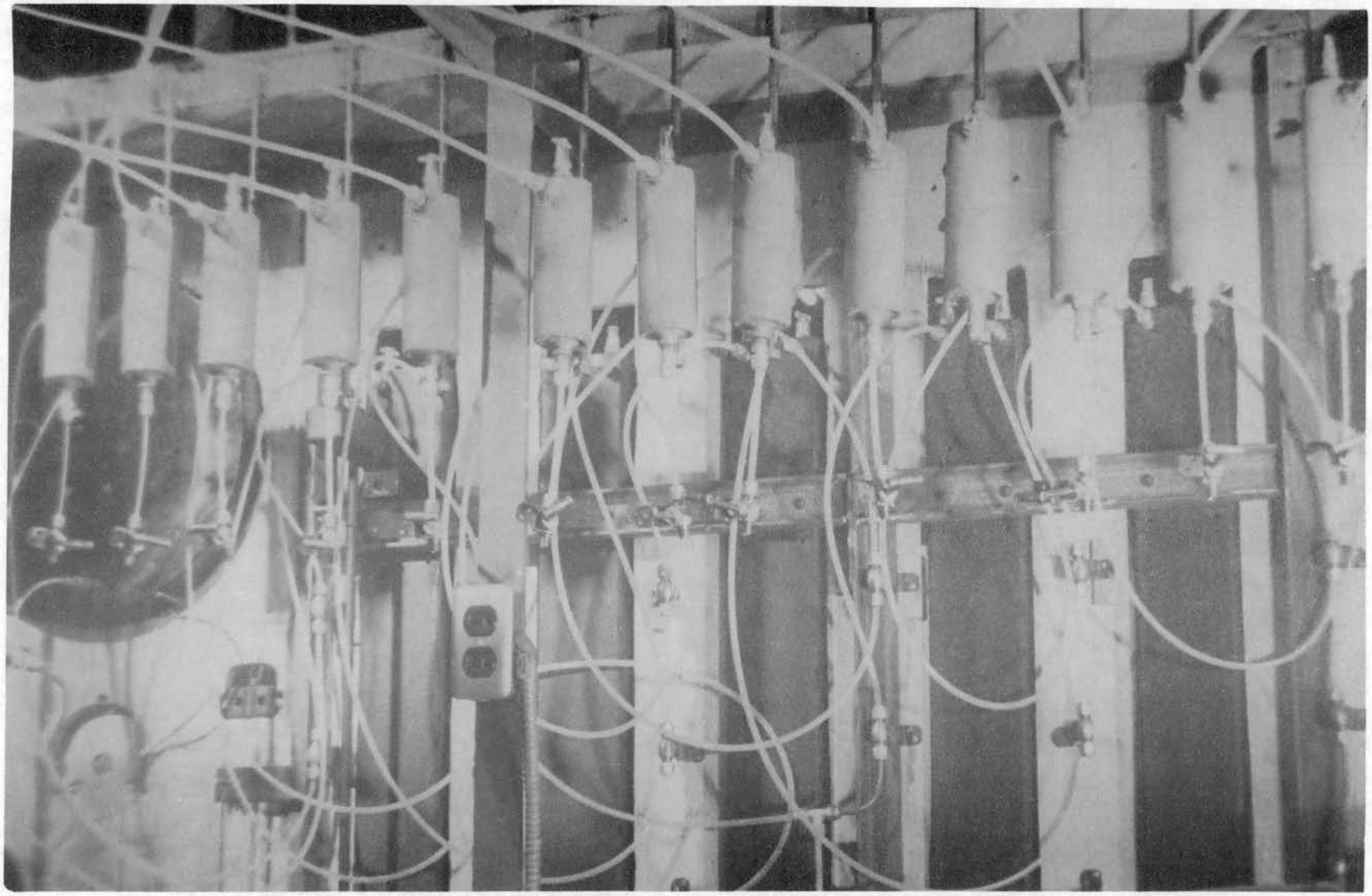


PLATE III. MANOMETER SEAL POTS



manufactured by the Edwin L. Wiegand Company. These heaters run the length of the cylinder, held in position by baffle plates which also channel the flow. Each heater consists of a pair of copper tubes with the heating element inside. With a total of twenty four tubes running the length of the condenser, the unit resembles a shell and tube heat exchanger. The preheater has a theoretical capacity of sixty kilowatts. The 220 volt preheater elements are in parallel and are controlled by a combination of manual switches and a Powerstat manufactured by the Superior Electric Company. The Powerstat varies the voltage across six of the twelve heating elements and allows an infinite number of power settings to be obtained. In addition, provisions exist for switching any number of these six heating elements in and out of the circuit. The other six preheater elements are provided with on-off control by panel switches. Although the Powerstat is connected to only six preheaters drawing 47 kilowatts, it is capable of producing 60 kilowatts maximum if more preheat becomes desirable.

Test Section. The test section proper is fabricated from AISI TP304 stainless steel tubing with an inside diameter of 0.399 inches and a wall thickness of 0.0515 inches. The test section receives power through copper lugs which are silver soldered to the test section with a spacing of 60 inches. This length was a foot shorter than the Argonne loop, due to the use of lower capacity welding transformers. The transformers used are 3 Lincoln Fleet-arc AC Industrial Type welders, connected in parallel with the test section. The transformers are each rated at 400 amps at 40 volts or 16 kilowatts, operating on a 60 percent duty cycle with 70 volts on open circuit. The welders used at Argonne produced a total of 60 kilowatts compared to 48 kilowatts for the system at Oklahoma State

University. This decrease in power available dictated the use of a shorter distance between power lugs. Output of the welding transformers is controlled by a motor and chain drive linkage, which allows each transformer to supply the same fraction of power desired, at an infinite number of settings. Thermal overload switches are also provided to prevent damage to the transformers.

The electrical energy is converted to thermal energy in the test section when the high current travels between the power lugs by way of the relatively small cross section of the tube. Teflon insulators, Fig. 3, are used to insulate the test section electrically from the rest of the loop and prevent any current flow into other sections of the loop. Twelve pressure taps are positioned at various intervals, as shown in Fig. 3. The pressure tap holes are 1/32 inch in diameter, drilled concentric to the tubing through the wall of the test section. The tube was inspected for any burrs on the inner wall of the test section and these were removed by reaming.

Sight Glass. A glass tube with an inside diameter of 0.39 inches is in the loop just down stream of the test section for visual checks of flow conditions.

Exhaust Header. The purpose of this header is to throttle the water or two phase mixture to conditions approaching atmospheric pressure. To achieve this purpose, four valves are placed in parallel with the flow. For coarse adjustments, two 2 inch gate valves are provided. Finer adjustment is accomplished with a one inch gate valve and a 3/8 inch needle valve. The 3/8 inch needle valve is controlled from the panel board for ease of setting.

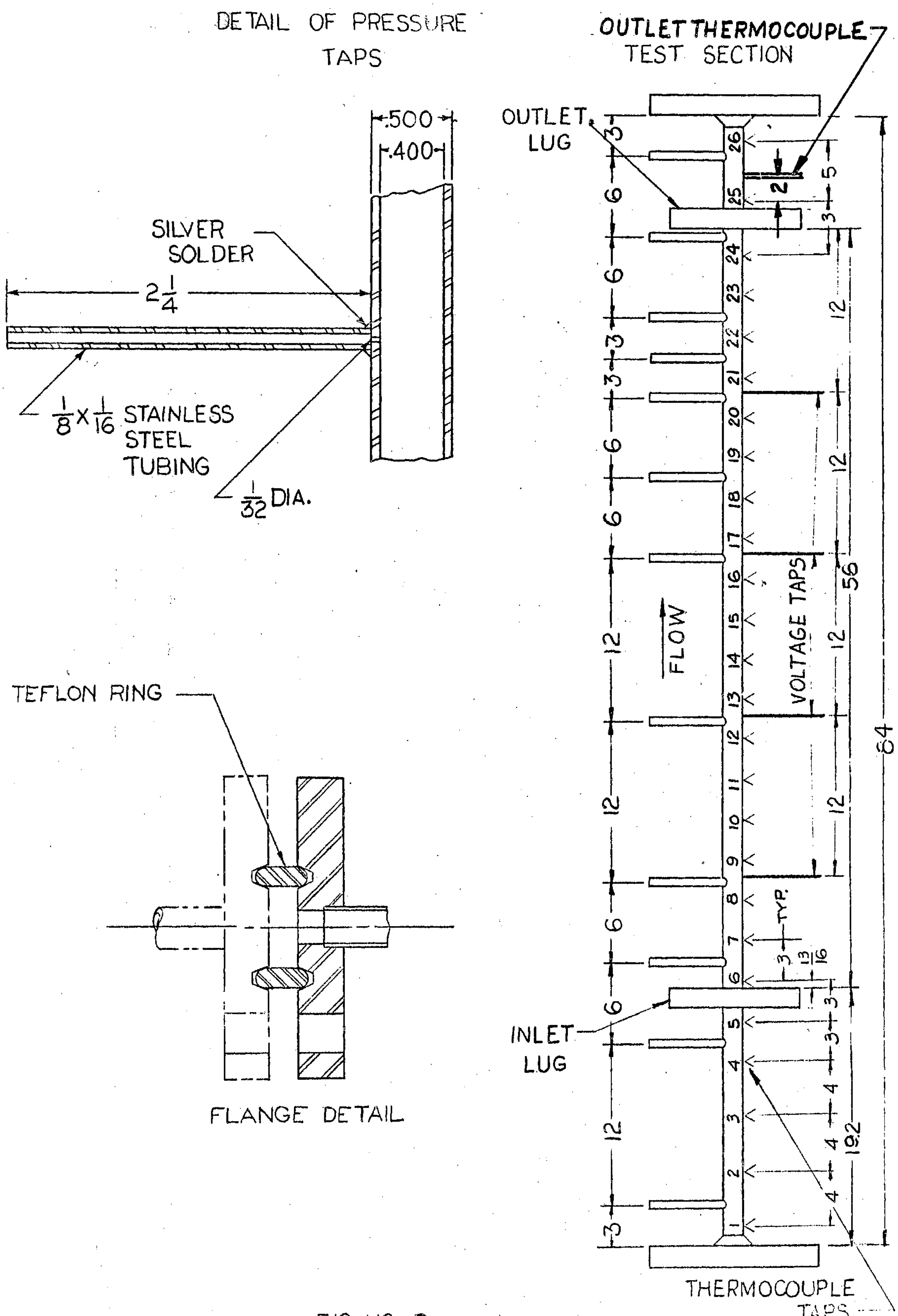
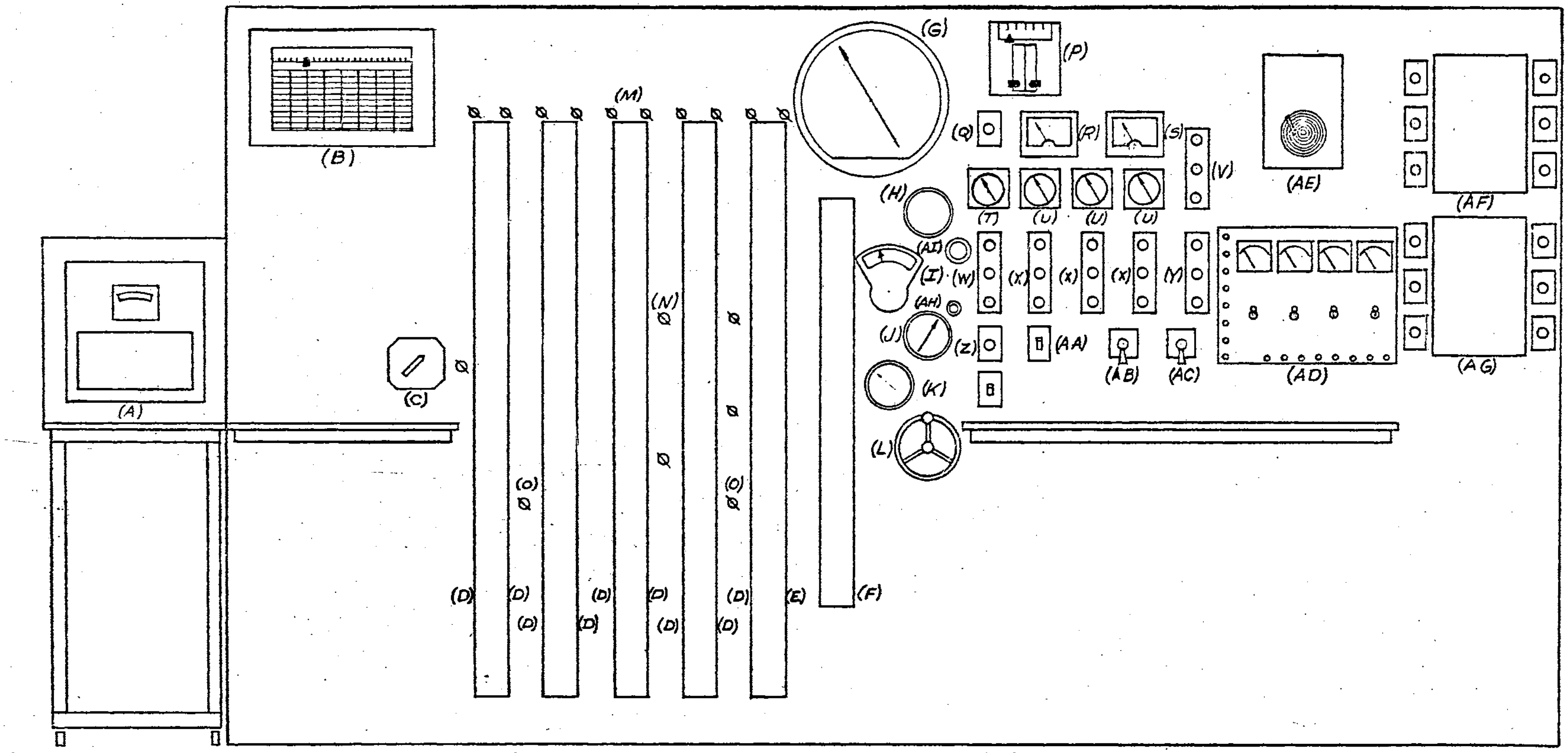


FIG. NO. 3
TEST SECTION



(A) 48 POINT TEMPERATURE INDICATOR
(B) TEMPERATURE RECORDER
(C) THERMOCOUPLE SELECTOR SWITCH
(D) PRESSURE DROP MANOMETERS
(E) FLOW MANOMETER
(F) 36 INCH MANOMETER
(G) PRECISION PRESSURE GAGE
(H) OBSERVATION PORT
(I) VARIDRIVE SPEED INDICATOR
(J) SYSTEM PRESSURE
(K) HOLDUP TANK TEMPERATURE INDICATOR
(L) VARIDRIVE SPEED ADJUSTMENT
(M) MANOMETER VALVES

(N) MANOMETER BYPASS VALVES
(O) MANOMETER REFERENCE VALVES
(P) CONTROL SYSTEM PRESSURE INDICATOR
(Q) PANIC SWITCH
(R) TEST SECTION VOLTMETER
(S) TEST SECTION WATTMETER
(T) TOTAL CURRENT AMMETER
(U) INDIVIDUAL TRANSFORMER AMMETER
(V) DIRECT CONNECTED PREHEATER
(W) MAIN PUMP SWITCH
(X) TRANSFORMER POWER SWITCH
(Y) POWERSTAT SWITCH
(Z) TRANSFER PUMP SWITCH

(AA) ION PUMP SWITCH
(AB) POWER TRANSFORMER DRIVE SWITCH
(AC) POWERSTAT POSITION SWITCH
(AD) PREHEATER INSTRUMENT PANEL
(AE) TEST SECTION CURRENT RECORDER
(AF) PREHEATER JUNCTION BOX
(AG) POWERSTAT JUNCTION BOX
(AH) BYPASS VALVE
(AI) THROTTLE VALVE

FIGURE 4
CONTROL PANEL

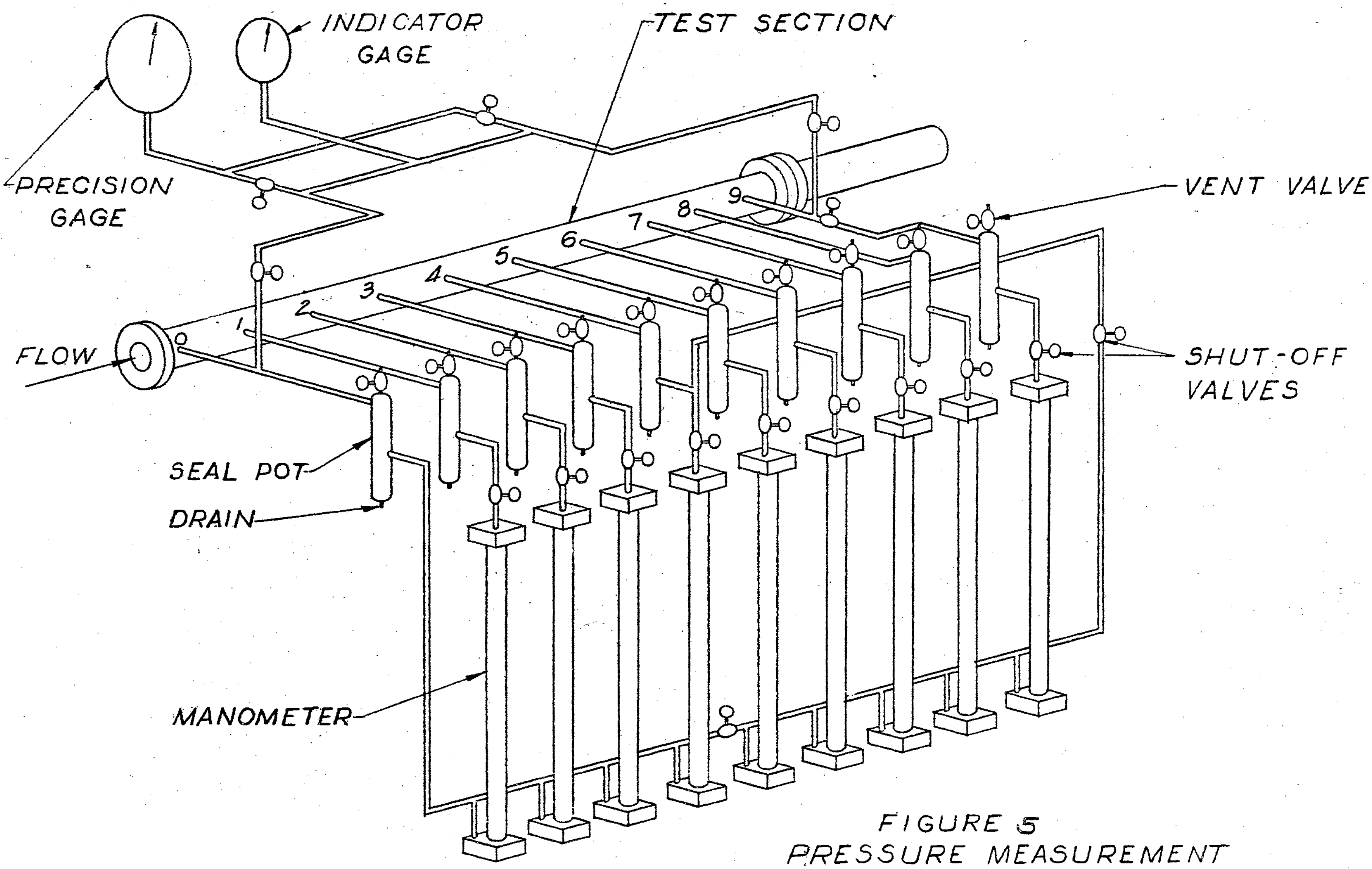


FIGURE 5
PRESSURE MEASUREMENT SYSTEM

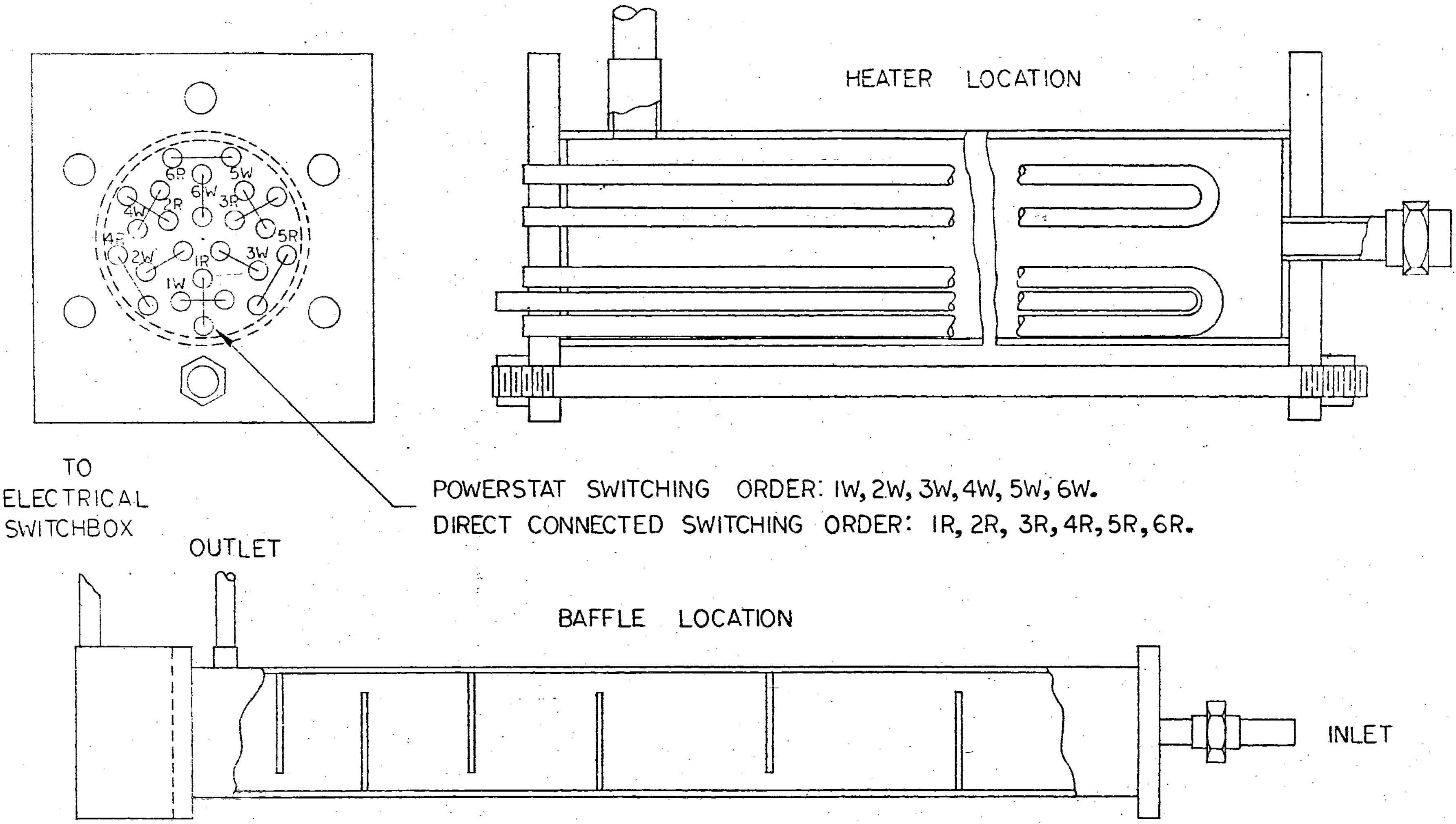


FIGURE 6
PREHEATER CONSTRUCTION

Condenser. The condenser is a Ross type BCP 2 pass shell and tube heat exchanger, with the cooling water flowing in tubes serviced by 1 1/2 inch cooling water entrance and exit lines. The condenser shell is brass and the tubes copper with steel completely eliminated from the construction. The degree of cooling is controlled by the adjustment of the flow of tap water which is used as the cooling medium. An air bleed is provided at the high point of the condenser for removal of entrained air.

Holdup Tank. This tank is the rectangular 1 1/2 x 2 x 3 foot stainless steel container shown in Fig. 1. Four layers of copper tubing with cooling water circulating through them are placed in the bottom half of the tank. Either two or all four layers of tubing can be used, as each set of two layers has separate water inlets and outlets. The tank serves two purposes, to act as a hotwell for the condensate and to further cool the system fluid.

Ion Exchange System. The entire loop is with one exception, completely free of steel in contact with the system fluid. All piping, valves, fittings, pump casings, and tanks are of stainless steel, bronze, brass, or copper. Due to the presence of steel impellers in two transfer pumps and some question as to the purity of the distilled water used, an auxiliary ion exchange system is used. The system fluid is pumped from the supply tank through a small heat exchanger consisting of coiled copper tubing in the form of a helix inside a 4 inch diameter by 2 foot stainless steel pipe. The cooling water flows through the copper tubing. The purpose of this exchanger is to lower the temperature to below 100°F, which is necessary for proper action between resin and system fluid. The fluid then flows into the ion exchanger proper. Essentially the exchanger consists of a vertical stainless steel cylinder 4 inches in diameter, with the flow entering at the bottom. The ion exchanger media is a resin in a capsule design which permits easy recharging. The fluid flows through the resin and out the column at a point above the level of the resin and is then pumped back into the storage tank.

Instrumentation

Pressure Measurements. The basic instrument for determining a reference pressure in the system is a Heise Model I7293 Bourdon tube gage of 0 to 750 psi range, checked by the manufacturer for errors exceeding 1/4 psi over the full range. The gage is connected to both the first pressure tap on the test section and the tap next to the downstream power lug, so that either pressure can be used as a reference in calculations of pressure at any of the other taps on the test section. The pressure at the pump outlet is also measured by a Bourbon-tube type pressure gage, placed there essentially for the operation of an over-pressure cut-off switch which is integral with the gage.

Nine 60 inch Meriam Series 30 well type manometers are used to obtain a pressure profile in the test section. The manometers are filled with a Meriam Red Fluid with a specific gravity of 2.95. The manometers are installed with the wells linked together and connected to the number zero pressure tap on the test section, the pressure at the number pero tap thus serving as a reference pressure and shown in Fig. 3. Check valves are connected to each of the high pressure sides of the manometers, allowing flow of water only and closing upon entrance of red fluid into the valves. The low pressure sides of the nine manometers are connected to the series of pressure taps downstream of the number zero tap. Because of the temperatures involved, it was found necessary to install a one foot section of coiled 1/8 inch copper tubing between the pressure tap and the nylon tubing used for all manometer lines. This section of copper tubing also acts as a condenser, keeping any steam from entering the manometer lines and altering the common effective head of water acting on the low pressure sides of each manometer. A seal pot is also located in each low pressure manometer line for the putpose of preventing red fluid from entering the system proper and because the pots are the high points in the manometer system, to bleed any air in the lines. A Meriam Series 30 well type 60 inch manometer is also used to determine the pressure differential across the calibrated orifice plates.

Temperature Measurement. Thermocouples, fabricated from Honeywell Type J Model 9B3C5 Iron Constantan thermocouple wire, are used for all temperature measurements with the exception of the holdup tank temperature. The bulk temperature of the system fluid is measured at the orifice inlet, the test section inlet, and the test section outlet, by means of temperature probes. The thermocouple wires were joined by a resistance weld and in the case of wall temperature measurements, spot welded to the test section thermocouples as shown in Fig. 3. Insulation temperatures on the test section are also measured for a check of thermal losses from the test section. Measurements of thermocouple electromotive force is done by a Brown Electronik 48 point, 0 - 1200°F range, temperature indicator and a Brown Electronik 12 point, 0 - 600°F range strip chart recorder, both operating by achieving a continuous balance of the thermocouple emf.

Power Measurement. In order to measure the thermal energy added to the system fluid it becomes necessary to measure the current and voltage drops across the preheater and test section. Knowledge of voltage drops along various intervals of the test section is desirable for the purpose of checking power dissipated per unit length of test section. The total current from the three welding transformers is indicated by a 0 - 5 amp General Electric Type AB Switch Board Ammeter which receives current from a General Electric instrument transformer with a ratio of 300 to 1, placed in the line. The voltage drops along the test section are measured by six evenly spaced taps. For purposes of current and voltage measurements the preheater is divided into Powerstat and switch controlled sections. Lines connected to the appropriate points are brought to the control panel for connection with precision portable ammeters and voltmeters. The precision meters used for the measurements were Weston Model 155 AC instruments. The ammeters were read through a 40 to 1 step down instrument transformer. A General Electric P-3 Wattmeter was used to read test section power. Because of the 300 to 1 instrument transformer previously mentioned in connection with the test section current, it was necessary to multiply the wattmeter reading by 300 or, by 600 when the instrument's own built-in range doubler was used. A General Electric Type P-3 voltmeter was used to read voltage drops directly across any of the six voltage taps connected to the test section. A Brown Electronik Circular Current Recorder recorded the current to the test section in terms of percent of full scale deflection - full scale deflection representing 1500 amps.

CHAPTER IV

CALIBRATIONS

Orifice Plate

A calibration was made with a 0.353 inch sharp edged orifice installed in the loop. The basic equation relating volume rate of flow and manometer level as used in this work is:

$$V_1 = C A_c \sqrt{2 g h_1} \quad ,$$

where, V_1 = volume rate of flow, in cubic feet per second.

C = orifice coefficient of discharge.

A_c = cross-sectional area of flow on the basis of internal pipe diameter, in square feet, = 0.006005 square feet.

h_1 = differential head across the orifice, in feet of fluid flowing.

g = acceleration of gravity = 32.17 feet per second squared.

The determination of the orifice coefficient was by the weighing method. The manometer system was purged of any air in the lines by bleeding at appropriate points. Some difficulty was encountered in getting consistent results due in part to the difficulty in removing all the air from the lines. The air trapped in the lines was believed to be the cause of the manometer failing to return to the zero position after stopping the pump. However all the data used in orifice coefficient calculation was taken with the manometer consistently returning to the initial zero setting plus or minus 0.02 inch.

The measurement of water flow rate was by the following method. The fluid after having passed through the orifice, was diverted from the loop at a point just downstream from the condenser. The fluid flowing through the orifice was directed into a barrel and the system was allowed to stabilize. The steam of water was suddenly diverted into the weighing tank, at the same time the stop watch was started. When a minimum of three minutes had passed the water was diverted out of the weighing tank, at the same time stopping the watch. The weight of the water was then measured on previously calibrated scales. The maximum error of the platform scales was 0.1 pound in the region of flow rates measured, 20 to 120 pounds, so that no corrections were made to the scale readings. As a check on the consistency of the method of weighing, a number of measurements were made at constant flow rates. The variation in readings was approximately 1/3 percent which was considered acceptable. All the data taken in determination of the orifice coefficient was under conditions of 50 pounds per square inch gage pressure at the orifice plate, and temperatures from 70°F to 83°F. The pressure was held constant to minimize the effect of any bubbles in the manometer sensing lines. A plot was made of log Reynolds Number vs. orifice coefficient. A straight line was found to represent the locus of these points with the maximum deviation from the line of 0.0004 measured along the C axis and the average deviation from the line about 0.0002. These deviations were 4/7 and 2/7 percent respectively. A plot of orifice coefficient vs. log Reynolds number is shown in Fig. 7.

The Reynolds number was found using the diameter of the orifice plate as a characteristic diameter D in the equation:

$$N_{Re} = \frac{\rho D V}{\mu}$$

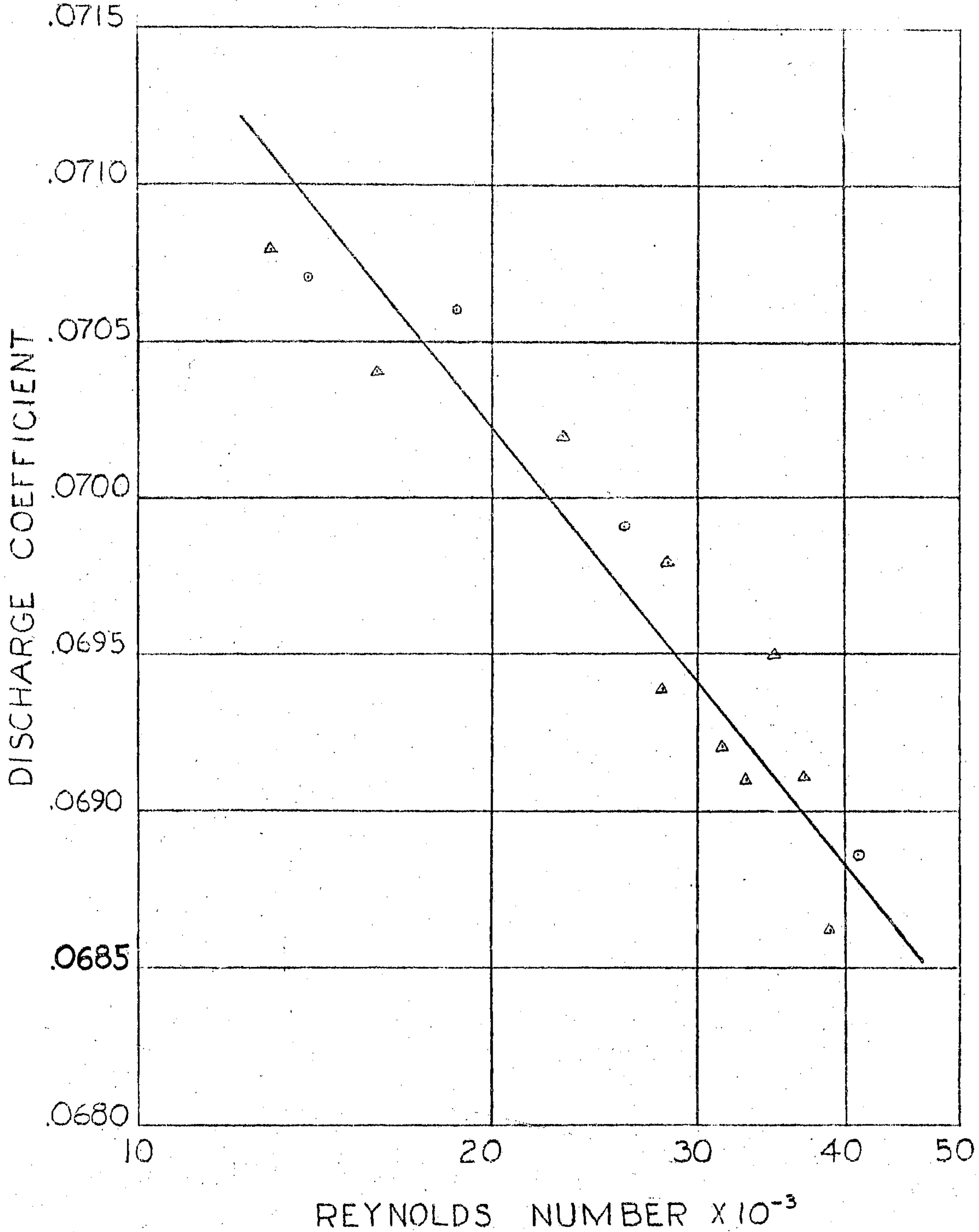
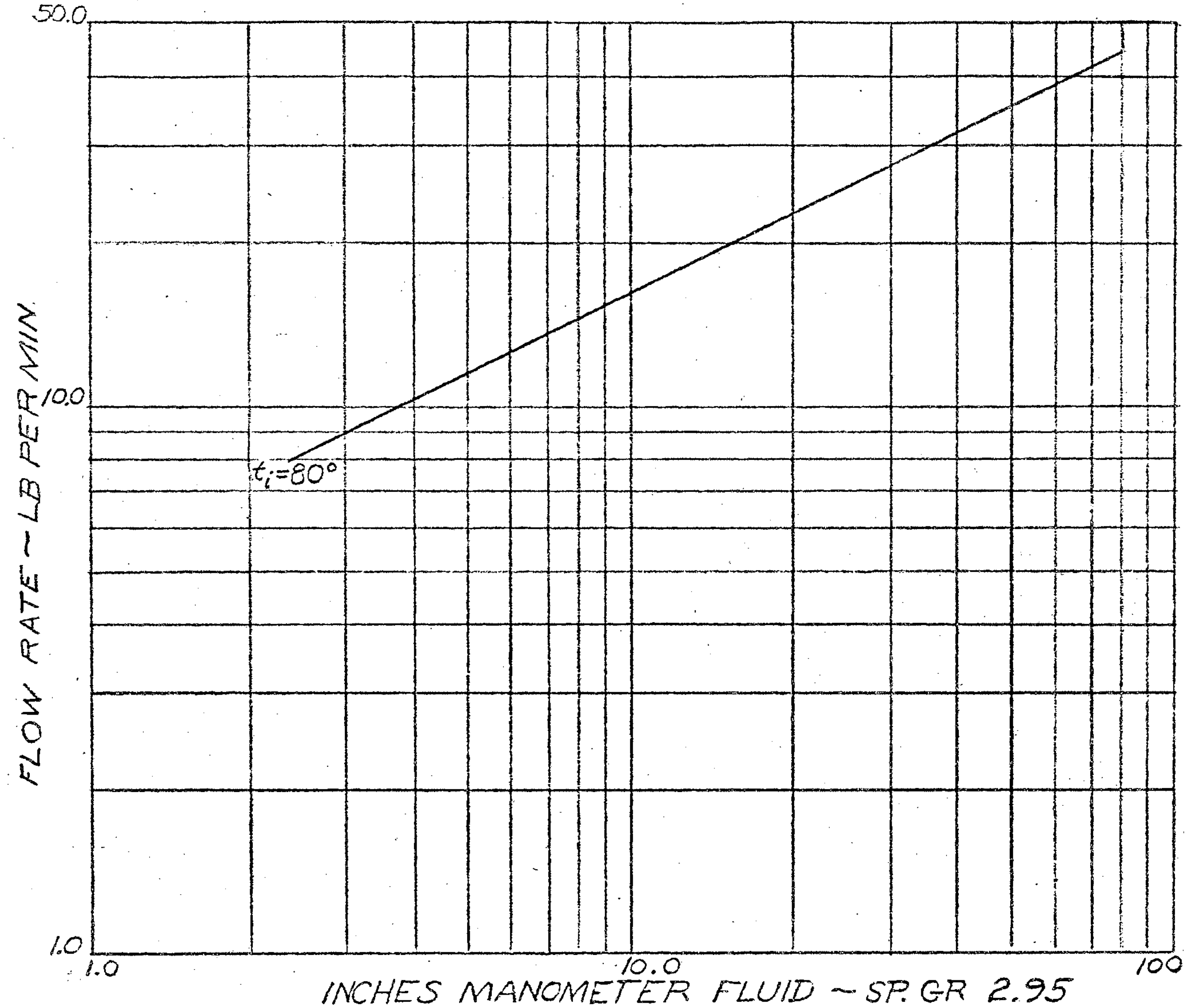


FIGURE-7
ORIFICE CALIBRATION CURVE



FLOW RATE VS. IN. MANOMETER FLUID

FIGURE 8

The temperature of the water upstream from the orifice was measured during each reading and the Reynolds number was adjusted accordingly. The value of C was calculated by using the basic flow equation:

$$V_1 = CA_c\sqrt{2g\,h_1}$$

with the known values of V_1 and h_1 substituted.

In plotting Reynolds number vs. orifice coefficients the degree of scattering of data was essentially uniform over the range of Reynolds numbers plotted. It was felt one of the prime reasons for the scattering was the instability of the manometer fluid meniscus. The level of the indicating fluid in the manometer would not hold a fixed position for a given flow setting. The level varied by as much as 0.3 inch at high rates of flow and varied with an erratic motion. The fluid level seemed to vary between two extremes of excursion for a given flow setting, holding either extreme for relatively long periods of time. The higher of the two extremes of excursion was read as it was held the greater length of time.

The mass rate of flow vs. the manometer reading was plotted on log log paper. The mass rate of flow used for this plot was calculated using values read from the plot of orifice coefficient vs. Reynolds number. This plot was compared with a theoretical graph constructed from tables of Reynolds number vs. C in, "The Flow Meter Engineering Handbook" Table 31 for a standard sharp edged orifice. The two plots differed by a maximum of 1 percent in h for any value of mass rate of flow.

Thermocouple

A sample of the thermocouple wire stock used throughout this work was selected and a thermocouple fabricated in the same manner as previously described in Chapter III. The 48 point Brown Electronik Temperature Indicator and thermocouple were calibrated as a unit. The plot of the required corrections is shown in Fig. 9.

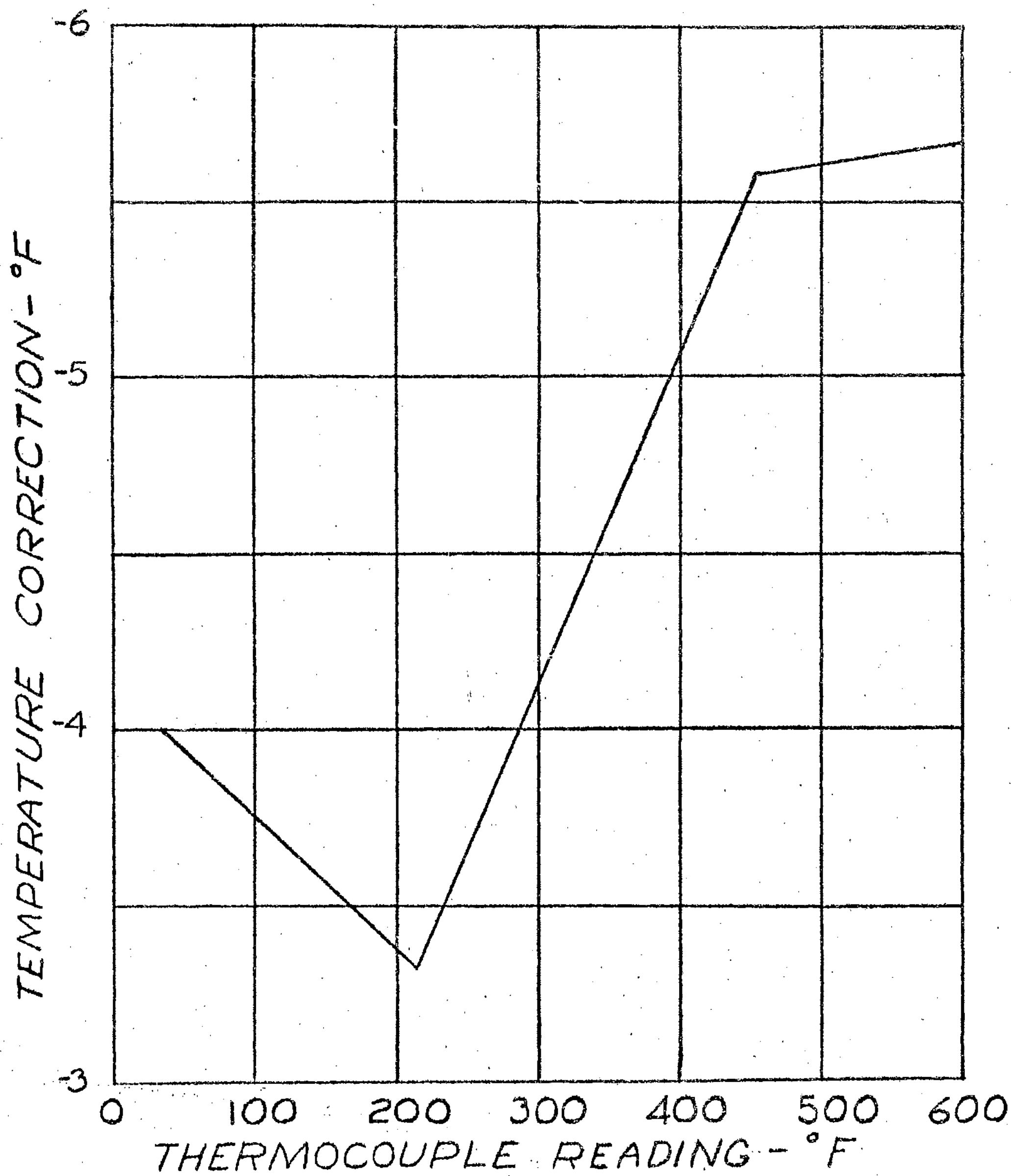


FIGURE-9
TEMPERATURE CORRECTION VS.
THERMOCOUPLE READING

CHAPTER V

EXPERIMENTAL PROCEDURE

Start Up

A certain amount of routine adjustment was necessary in setting the test equipment into operation before any data could be taken. Due to a system of safety interlocks it was necessary to have the lines into the condenser and hold up tank open to the cooling line pressure before any power for heating could be applied. The maximum allowable pressure was also selected and set so that the entire system would shut off if the line pressure at the pump outlet exceeded the predetermined set point. The maximum allowable pressure in the system of approximately 490 psig was controlled by a pressure relief valve placed in the manifold section.

The main pump was first started and the fine adjustment needle valve set to produce a pressure of from 10 to 30 psig in the test section without any heat being added to the loop. The manometer systems were then bled to remove any air trapped in the lines. This was done by opening the valves located on the tops of the seal pots, since these were the highest points in the systems. The system fluid was allowed to bleed off until a visual check of the lines showed no air bubbles to be present. After the test section pressure profile manometers had stabilized, which required up to 5 minutes after setting the system variables, the manometers were read and the results plotted vs. distance along the test section. A straight line was drawn through the locus of these plotted points and individual adjustments were then made to the scale position to move the plotted points into the straight line previously plotted. This procedure was followed because the meniscus of the manometer fluid was so poorly defined under conditions of zero flow. This was due to deposits on the portion of the glass exposed to the manometer fluid when there was no flow. The procedure had the disadvantage of masking over any variations in a linear pressure profile which would be produced by inconsistencies in pressure tap position or test section inner surface smoothness. These deviations in pressure drop would become evident again as soon as the flow rate was changed from the setting at which the corrections were made.

Procedure

The pump speed was held essentially constant throughout all the variations in the other parameters. The system pressure was controlled by the needle valve in the manifold and this pressure set an upper limit on the flow rate through the test section. Any flow rate less than the maximum could be achieved by opening the bypass valve. This permitted a portion of the flow from the pump to be diverted to the holdup tank and thus bypass the test section. The throttle and bypass valves were interdependent so that a change in one affected the other. To achieve a desired flow rate and test section entrance pressure it became necessary to adjust the valve settings.

A series of test section inlet pressures and flow rates were chosen and data was obtained with varying degrees of preheat and test section heating. To start a series of runs at a given test section inlet pressure and flow rate, maximum preheat was first used and the test section power was increased until the first trace of bubbles was noted in the sight glass indicating nucleate boiling in the test section.

Upon stablization of the system, the outside test section temperature and the pressure profile were recorded. There were some fluctuations in pressure and flow rate under hands-off operation which necessitated one person constantly observing the flow manometer and test section inlet pressure and making correcting adjustments while the pressure drop profile was being recorded. The fluctuations, if allowed to remain unchecked, would reach 0.3 inches of manometer fluid under conditions of maximum flow rate and at the number nine manometer. The test section voltage drops were measured and recorded across five equal intervals. Test section power, voltage drop, and current, as well as preheater current and voltage were also recorded for each setting.

After the reading with maximum preheat was recorded the preheat power was reduced so as to reduce the inlet temperature to a predetermined point. Test section power was then increased to the point where bubbles began to be observed again in the test section. This procedure was repeated with still lower inlet temperatures until a point of maximum test section temperature was reached. A new test section inlet pressure was then chosen and a series of runs starting with maximum preheat was made. When the five pressures had been covered a new flow rate was selected and the same five pressures used with varying inlet temperatures. Throughout this series of runs the holdup tank temperature was kept at 80°F to 90°F and was very susceptible to any changes in temperature due to changes in test section or preheater power. All the cooling needed was provided by the condenser; the holdup tank coils were not used.

A series of runs was also made under conditions of constant test section inlet temperature and pressure and variable rates of flow. There was no power added in the test section during these runs.

CHAPTER VI

TEST RESULTS AND CORRELATIONS

Flow at Constant Temperature

The pressure drop between the reference pressure tap and each downstream pressure tap was measured in inches of manometer fluid and plotted against position along the test section. In this manner a profile of pressure drop was developed. The location of the pressure taps is shown in Fig. 3. Two principal conditions of flow were observed, first with and second without heating of the test section. The pressure profiles for a series of runs with flow rates ranging from 10.15 to 18.16 pounds per minute and a constant fluid bulk temperature of 80°F is shown in Fig. 11.

The zero position on the test section axis represents the reference pressure. Since all the differential pressure manometers used this reference pressure all pressure profiles should pass through the zero point. However, the plotted curves intersected the zero position at points varying from 0.5 to 0.7-inches of manometer fluid. This was due to the manometers reading 0.5 to 0.7-inches of manometer fluid under static conditions. All the profiles resulted in straight lines indicating a linear pressure drop over the length of the tube under isothermal flow conditions.

In order to compare the test results with predictions by empirical equations, the theoretical pressures at points along the test section were computed and plotted as dash lines in Fig. 11. The equation used for these correlations is:

$$\frac{\Delta P}{\Delta L} = \frac{f}{D} \frac{\rho V^2}{2}$$

Since the friction factor is dependent on the surface conditions and the Reynolds number, and both are held constant throughout the entire length of the test section, and also the kinetic energy remains constant, then $\frac{\Delta P}{\Delta L}$ equals a constant.

The friction factor is dependent on the Reynolds number and relative roughness. The relative roughness is the ratio of $\frac{\epsilon}{D}$, where D is the inside diameter of the tube and ϵ a term dependent on the roughness of the inside surface of the test section. The friction factors used in the isothermal flows were read from Moody's plot using a relative roughness of 0.00014, equivalent to an ϵ of 0.000005 for drawn tubing. This value of ϵ is based on clean tubing. The relative roughness for smooth tubes such as drawn tubing is very sensitive to changes in surface smoothness. It is possible that the relative roughness of the tube wall might have had an actual value as high as 0.0005 changing the friction factor by 2 to 3 percent at the higher flow rates. This could partially explain the 2 percent maximum difference between the experimental and computed pressure profiles. As shown in Fig. 11 the correlation is best at low flow rates which would strengthen the preceding theory.

Flow With Test Section Heating

The pressure drops were plotted for a series of runs with heating in a manner identical to the isothermal runs. The flow rates were the same as the isothermal runs with the heat flux varying from 64,000 to 312,000 Btu per sq ft hr. The thermal energy was added to the flowing fluid under conditions of constant heat flux throughout the test section. This fact was substantiated by observing the voltage drop across a series of segments of test section.

As shown in Fig. 10 the test section voltage drop vs. test section length resulted in a straight line. Since power dissipated equals voltage drop times current the power dissipated per unit length was constant. The full length of the test section was not used during these runs since the power added was always adjusted so that nucleate boiling would occur at a position from four to five feet from the entrance to the test section. The pressure profiles shown in Fig. 12 to 18 were drawn by plotting the pressure drops from points zero to four and extrapolating the remaining two feet. In all cases the pressure profile proved to be best represented by a straight line within the accuracy of plotting. In roughly 20 percent of the runs a very slight curvature would have produced a better fit. Because the curvature was so slight and much smaller than scattering due to other effects, straight lines were used in all cases.

A series of pressure drop profiles were computed using a set of heat fluxes and inlet conditions encountered in 25 runs. The runs selected covered the complete range of heat fluxes and mass rates of flow.

In order to compute the pressure profiles a number of assumptions were made. As previously mentioned the power dissipated per unit length was assumed constant over the entire test section for a given power setting. The specific heat c_p was assumed constant and equal to one over the temperature range encountered. The bulk temperature of the fluid was assumed to increase linearly over the length of test section. It was assumed that the test section lost no heat to the surroundings.

The major problem in calculating the pressure drop in the test section was the determination of the friction factor along any increment of test section. The friction factor can be expressed as a function of Reynolds number for flow without heating in smooth pipes, i. e. the film temperature and bulk temperature are essentially the same. It was assumed that from a pressure drop standpoint two flow conditions, one with no heating and another with heating, would have the same friction factor at a given point if the film temperatures were the same. The friction factor was then computed using the Blasius equation for turbulent flow:

$$f = \frac{0.316}{(N_{Re})^{1/4}}$$

computing the Reynolds number of a fluid flowing through the test section without heating and at a bulk temperature equal to the film temperature of the point in question.

The film temperature was determined by adding the temperature difference across the film to the bulk temperature of the fluid. The bulk temperature was determined by observing the test section inlet and outlet temperature and assuming a linear bulk temperature profile through those points. The inside wall temperature was determined by computing the film coefficient of heat transfer h, from the relation:

$$\frac{hD}{k} = 0.023\,(N_{Re})^{0.8}\,(N_{Pr})^{0.4}$$

With h calculated, the basic heat transfer equation:

$$Q = hA\,\Delta t$$

was used to determine Δt since, assuming uniform heating, the quantity Q/A was constant and known. The sum of Δt and the bulk temperature was then used as inside wall temperature.

The test section was then divided into one foot intervals starting at the number 1 pressure tap and an average friction factor and fluid density computed at each interval. The pressure drop across each interval was then computed by the formula:

$$\Delta P_{1-2} = \frac{f_{avg.}}{\rho_{avg}}\,\frac{l_{1-2}}{D}\,\frac{\rho V^2}{2}$$

From the preceding discussion it can be seen that the calculations if done by hand for 25 runs would prove lengthy. The solutions were obtained by the use of an IBM 650 computer after developing the system of equations in terms of the input variables: inlet temperature, heat flux, and mass flow rate.

The following is a development of the equations used in the computer solution.

Mixing Cup Temperature Distribution:

$$\frac{Q_1}{L} = \frac{W(3412)}{L},$$

$$\Delta t_i = \frac{W(3412)}{L} l_i \frac{1}{M},$$

Δt_i is the rise in bulk temperature above the inlet temperature at a point i, a point l_i inches from the upstream power lug.

In terms of the input variables:

$$\Delta t_i = \frac{3412 q'' A_t l_i}{L G A_c 3600}$$

Evaluating the constants:

$$\Delta t_i = \frac{3412(.4940)}{56(8.73 \times 10^{-3})(3600)} \cdot \frac{l_i q''}{G}$$

$$t_i = t_{ent.} + \Delta t_i$$

Inside Wall Temperature Distribution:

$$Q = h A_t \Delta t_i'$$

$$\frac{Q}{A_t} = const. = h \Delta t_i' = q''$$

$$\Delta t_i' = \frac{1}{h} \frac{Q}{A_t}$$

For fluid flow in pipes with forced convection, the correlation:

$$\frac{hD}{K} = 0.023 (N_{Re})^{0.8} (N_{Pr})^{0.4}$$

$$N_{Re} > 2300$$

is representative with the variables evaulated at the mixing cup temperature.

$$\frac{1}{h} = \frac{D}{k(0.023)(N_{Re})^{0.8}(N_{Pr})^{0.4}}$$

$$\Delta t_i' = \frac{Q D}{A_t k(0.023)(N_{Re})^{0.8}(N_{Pr})^{0.4}}$$

$$\Delta t_i' = \frac{q'' D}{k(0.023)(N_{Re})^{0.8}(N_{Pr})^{0.4}}$$

$$t_{i(WALL)} = t_i + \Delta t_i$$

Pressure Drop:

$$f = \frac{0.316(\mu)^{1/4}}{(GD)^{1/4}}$$

$$\Delta P_{1-2} = \frac{f l_{1-2}}{D} \frac{\rho V^2}{2} + \frac{G^2}{g_c}\left(\frac{1}{\rho_2} - \frac{1}{\rho_1}\right)$$

$$\Delta P_{1-2} = \frac{f_{avg.}}{\rho_{avg.}} \frac{L}{2D} \frac{G^2}{g_c} + \frac{G^2}{g_c}\left(\frac{1}{\rho_2} - \frac{1}{\rho_1}\right)$$

The computer solutions were plotted as pressure profiles along with the observed pressure profiles as shown in Figs. 12 through 17.

After studying the profiles the following observations were made. The degree of correlation followed a definite pattern at the higher flow rates. Pressure profiles for the middle and high ranges of mass flow approached maximum correlation as the heat flux approached 250,000 to 260,000 Btu per hr ft^2. The lower flow rates had no definite pattern in variation of degree of correlation. As the flow rates decreased from the maximum the computations at first predicted pressure profiles that were lower than the experimental results and then changed so that at lower flow rates the computed profiles were higher than the experimental results. The correlation was able to predict pressure profiles usually within 3 percent with a few predictions off by 5 percent.

As previously mentioned in Chapter II, Kreith and Summerfield obtained the following equation to predict friction factors:

$$\frac{f_{non.}}{f_{iso.}} = \left(\frac{\mu_W}{\mu_B}\right)^{0.13}.$$

The Kreith Summerfield paper stated the above equation predicted the friction factor to ± 3 percent with Reynolds numbers of 100,000 to 250,000. Other sources suggest the value of $\left(\frac{\mu_W}{\mu_B}\right)^{.25}$ for flow in the laminar region.

The Kreith Summerfield relation was evaluated for four sets of experimental inlet conditions with four mass rates of flow. The linearity of the experimental pressure profiles suggested that the ratio of $\frac{f_{non.}}{\rho}$ was constant over the entire length of the test section for any given run. The value of $\frac{f_{non.}}{\rho}$ was calculated from the experimental pressure profiles. The Kreith Summerfield relation was used to calculate another value of $\frac{f_{non.}}{\rho}$. The friction factor $f_{iso.}$ was evaluated by the Blasius relation. With the Blasius substitution the Kreith Summerfield relation becomes:

$$\frac{f_{non.}}{\rho} = \frac{0.316}{(0.427)G^{1/4}} \frac{(\mu_B \cdot \mu_W)^{0.13}}{\rho}$$

The viscosity μ_W was evaluated at the inside wall temperature at a point five feet from the zero pressure tap. The inside wall temperature was calculated by using the Colburn relation in a manner previously discussed.

The Kreith Summerfield relation predicted the experimental $\frac{f_{non}}{\rho}$ values and hence the pressure profiles within from 3 to 7 percent. The correlation was best at high flow rates with Reynolds numbers of 40,000 to 50,000. This was to be expected since the Kreith Summerfield relation is stated to hold for Reynolds numbers of 100,000 to 250,000.

CHAPTER VII

ANALYSIS OF ERRORS

It is estimated that the measurement of total power into the test section was accurate to within ± .5 percent. Heat losses through the test section insulation were of the order of 1 percent were ignored in the calculations. The measurement of mass rate of flow of system fluid was estimated to be accurate to within ± 1.5 percent. System pressure was measureable to within ± 2 psi. The only temperature required was the test section inlet temperature which was estimated accurate to within ± 1.5°F. Pressure drop along the test section was the most critical measurement and the accuracy of measurement varied with flow rate. At high flow rates the manometer was less readable due to movements of the manometer fluid level. The average error is estimated at ± 0.1-inches of manometer fluid.

In computing the pressure drop in the test section, with test section heating, the most sensitive factor is the mass rate of flow, G, to the 1/2 power. It was therefore estimated that the pressure drop calculations were within 2 percent of the results that would be obtained if the true values of all measured quantities required in the solutions were known.

CHAPTER VIII

CONCLUSIONS AND RECOMMENDATIONS

To pressure drop over the test section was observed for a series of isothermal flow conditions. For turbulent flow in smooth tubes under isothermal conditions, the Blasius relation, was found to predict friction factors in agreement with Moody's results for drawn tubing. The Blasius friction factors were then used to predict the isothermal pressure profiles. The resulting pressure profiles were within 2 percent of the measured profiles. These results served as a check on the methods used in sensing pressures, temperatures, and mass rates of flow.

The pressure profiles were next plotted for a series of monisothermal flow conditions. The Colburn relation for predicting film coefficients was used to calculate the inside wall temperatures. The inside wall temperatures were used in evaluating the Blasius relation, thus the friction factors were obtained for any point along the test section. The pressure profiles were then calculated and compared with the experimental results. The correlation was best at Reynolds numbers of 20,000 to 40,000, ranging from 2 to 3 percent. Over the complete range of Reynolds numbers encountered the correlation was successful within ± 5 percent. The Kreith Summerfield relation previously mentioned was also used to predict the pressure profiles. It predicted the profiles most successfully at the higher Reynolds numbers ranging from 4 percent at the higher flow rates to 7 percent at the lowest.

From these results it was concluded that the Blasius relation, when used in conjunction with the Colburn equation for Reynolds numbers of 10,000 to 50,000 was slightly more successful in predicting friction factors than the Kreith Summerfield relation when evaluated using the same set of inside wall temperatures.

It is recommended that further studies be made with Reynolds numbers considerably higher than those encountered in this thesis. This would provide information as to the suitability of the methods used in this thesis for predicting pressure drop at higher Reynolds numbers.

For continued testing in the range of Reynolds numbers from 10,000 to 30,000 it is recommended that the manometer fluid be replaced with fluid with a specific gravity of approximately 2.

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APPENDIX

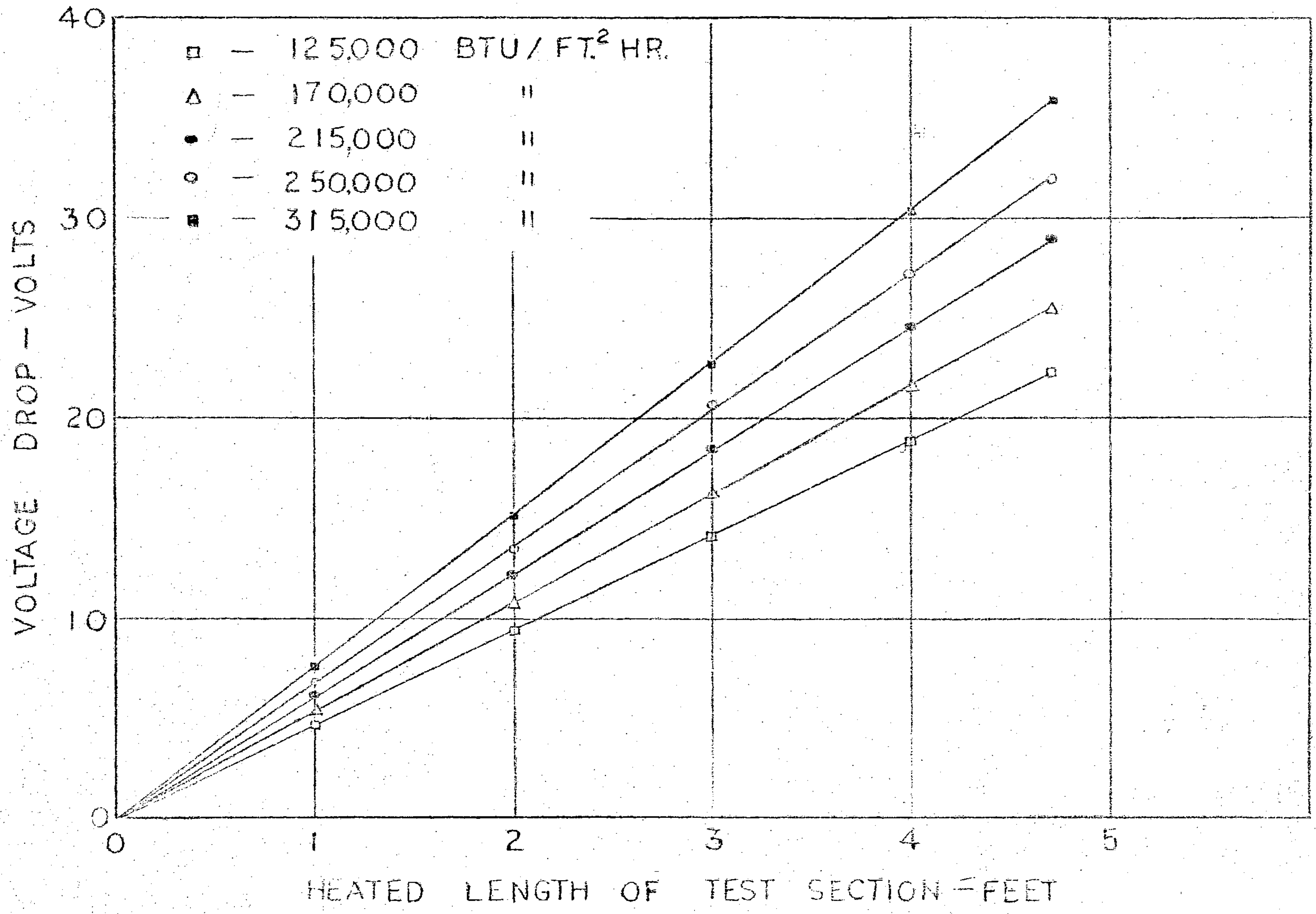

FIGURE — 10 TEST SECTION VOLTAGE DROP

TABLE I

TEST SECTION VARIABLES

Run Number	Inlet Temperature °F	Inlet Pressure psi	q" Btu/ft^2 hr	G lb/ft^2 sec
8	177.5	84	131,600	307.9
10	216.4	84	161,700	307.9
19	202.5	185	244,600	307.9
24	234.3	35	62,100	346.7
25	127.8	35	221,900	346.7
28	177.4	85	215,600	346.7
39	197.5	85	149,200	270.3
48	126.8	35	142,900	232.9
52	220.4	85	103,500	232.9
59	164.0	135	217,300	434.4
62	220.9	135	132,700	232.9
72	207.1	185	174,100	232.1
77	196.0	35	64,300	194.3
79	145.3	35	101,600	194.3
87	176.9	85	118,100	194.3
95	184.9	135	146,800	194.3
102	235.3	185	125,600	194.3
111	145.8	35	151,700	270.3
113	83.7	35	234,200	307.9
120	146.3	85	257,000	346.7
121	163.4	135	306,800	346.7
126	193.0	185	311,800	346.7
150	266.1	235	110,300	193.8
162	206.6	235	190,700	232.1
170	196.5	235	282,000	307.9

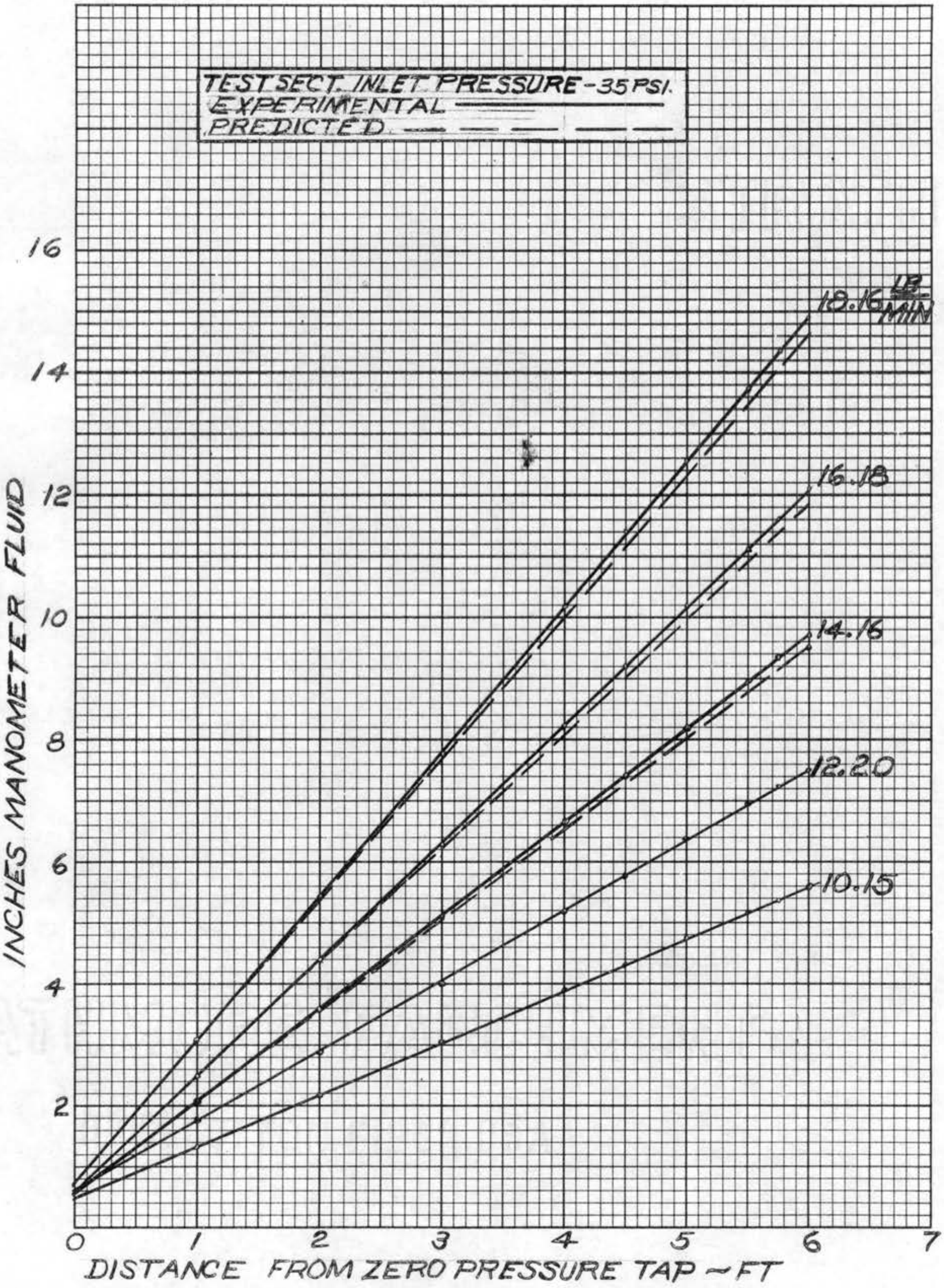

FIG. 11 ISOTHERMAL PRESSURE PROFILE

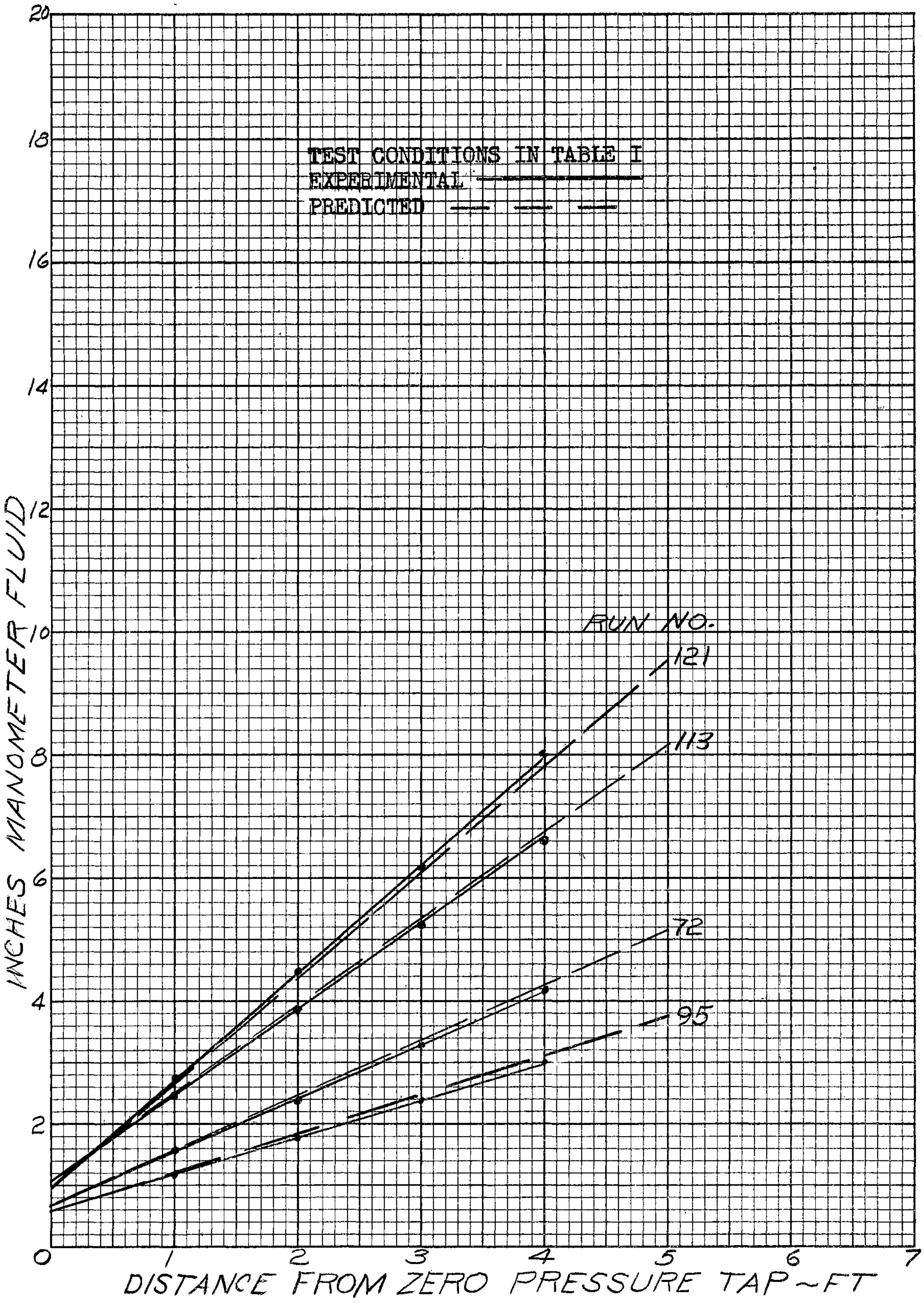

FIG. 12 NONISOTHERMAL PRESSURE PROFILES

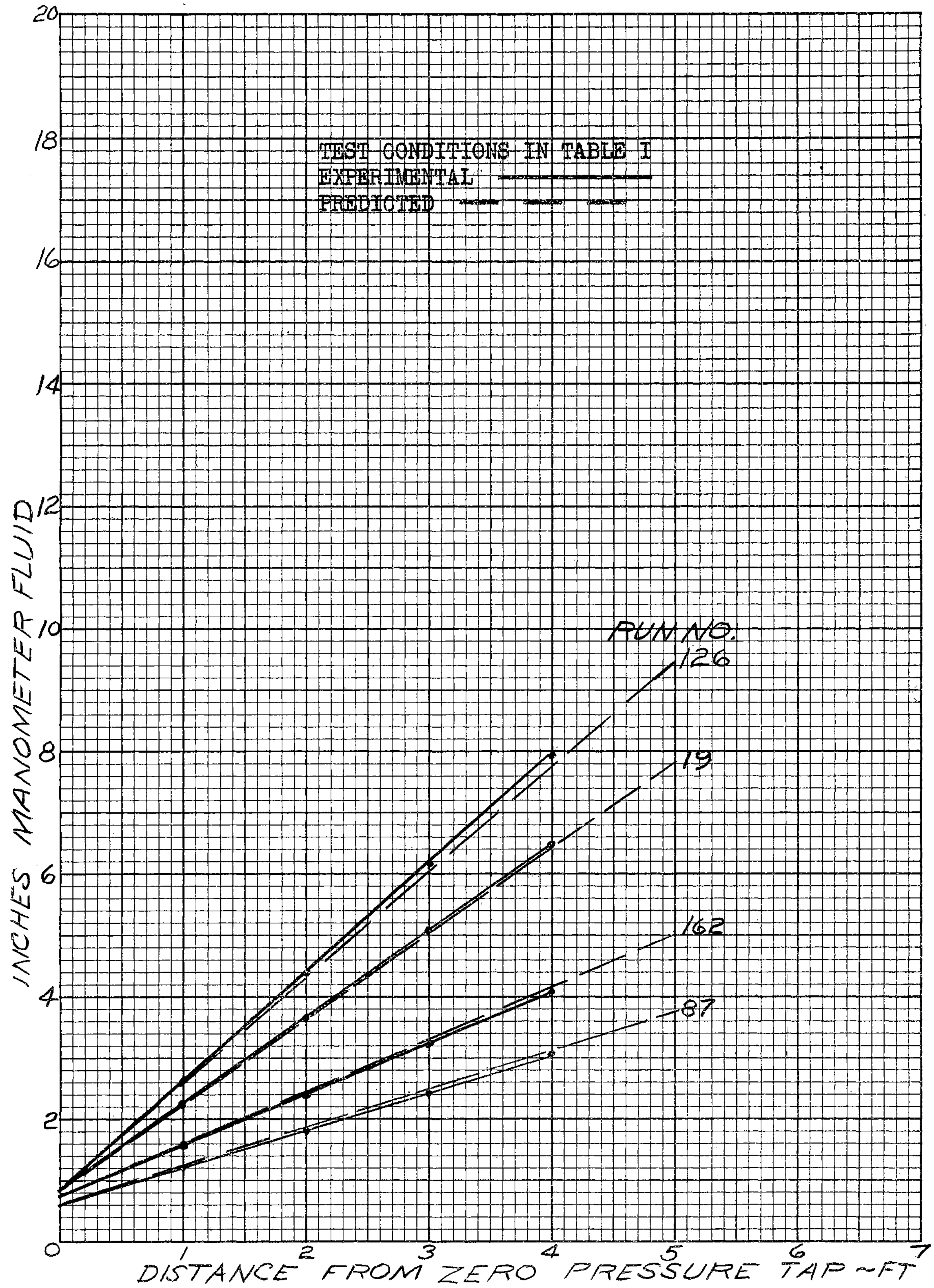

FIG. 13 NONISOTHERMAL PRESSURE PROFILES

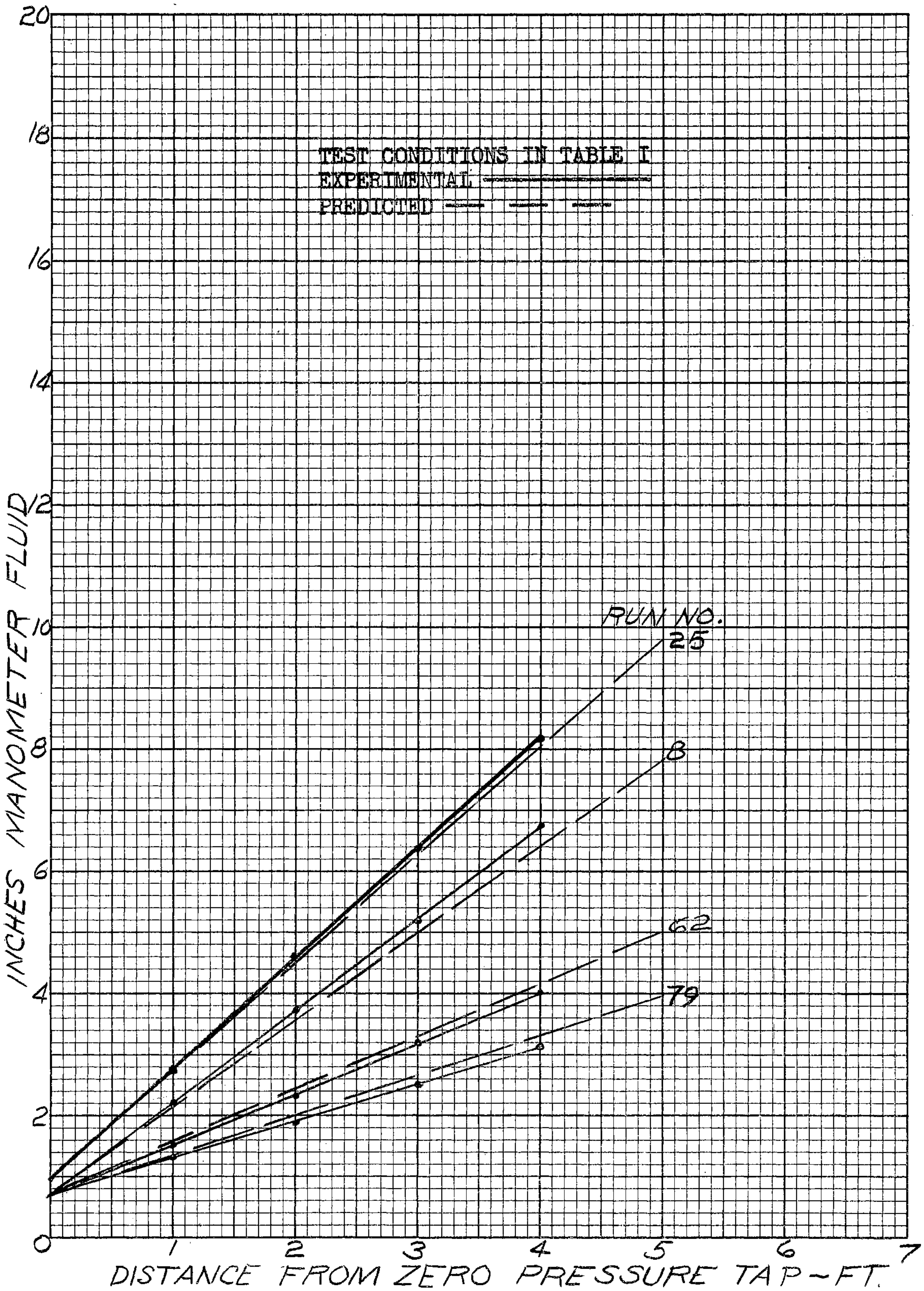

FIG. 14 NONISOTHERMAL PRESSURE PROFILES

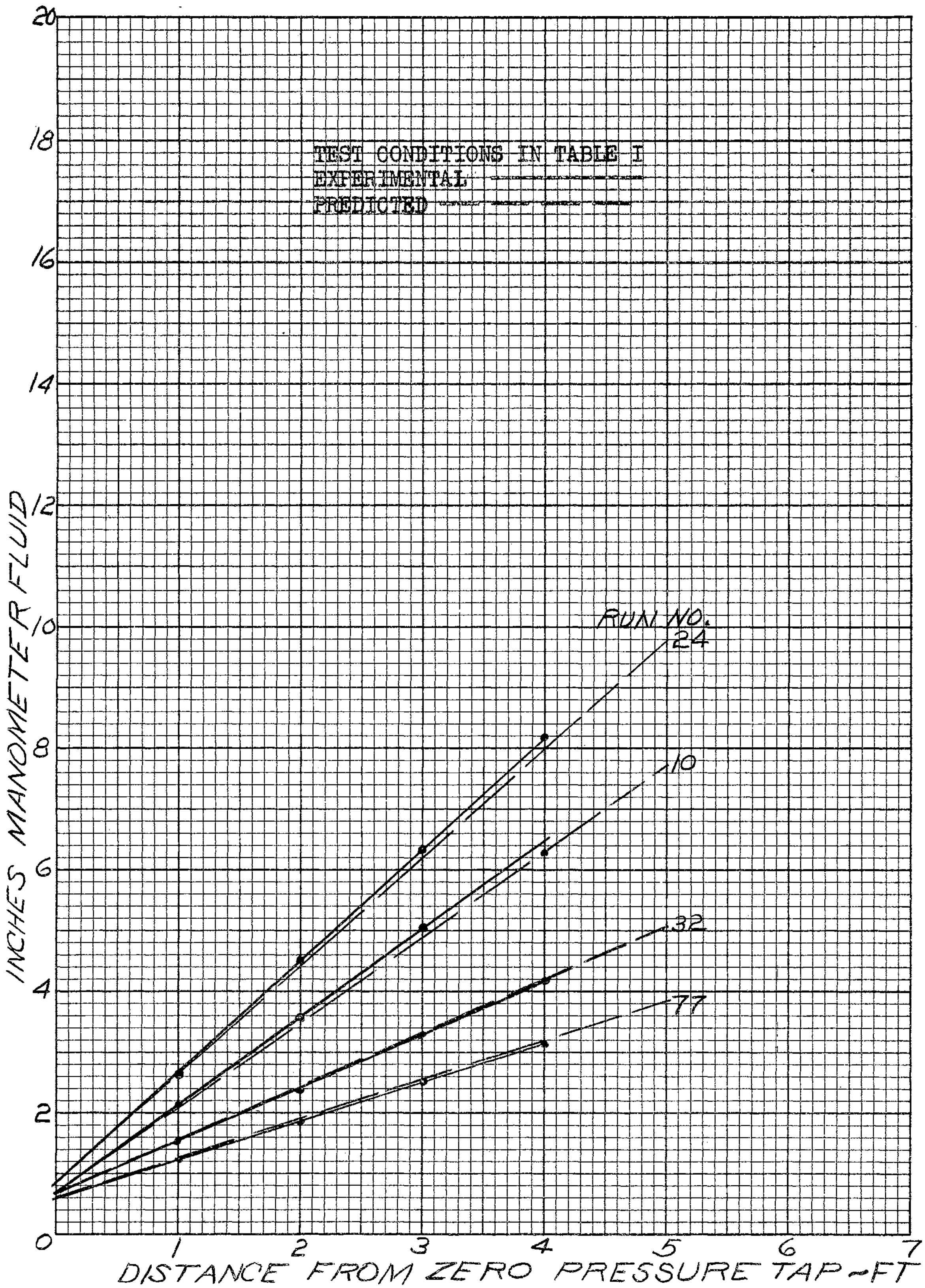

FIG. 15 NONISOTHERMAL PRESSURE PROFILES

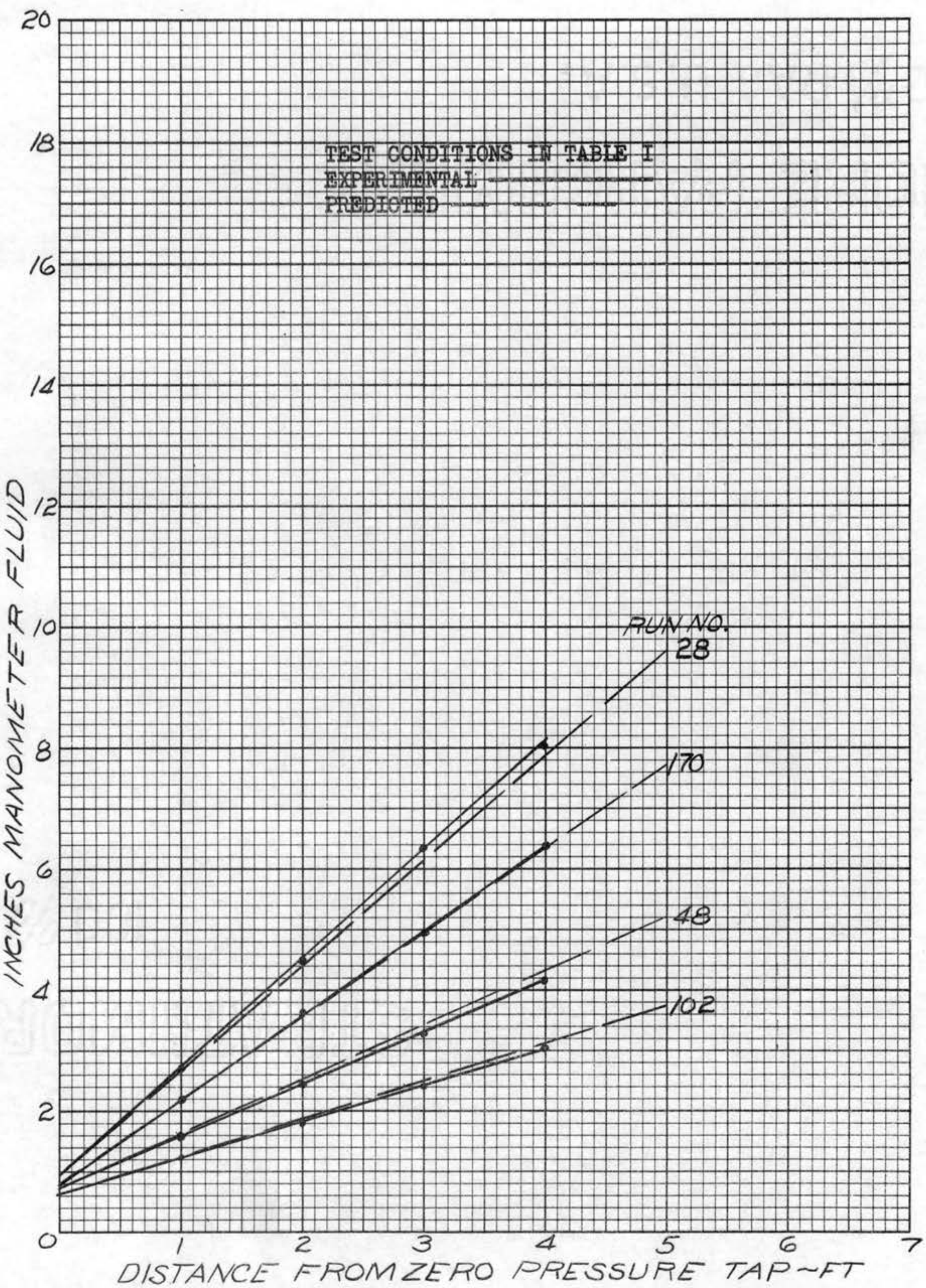

FIG. 16 NONISOTHERMAL PRESSURE PROFILES

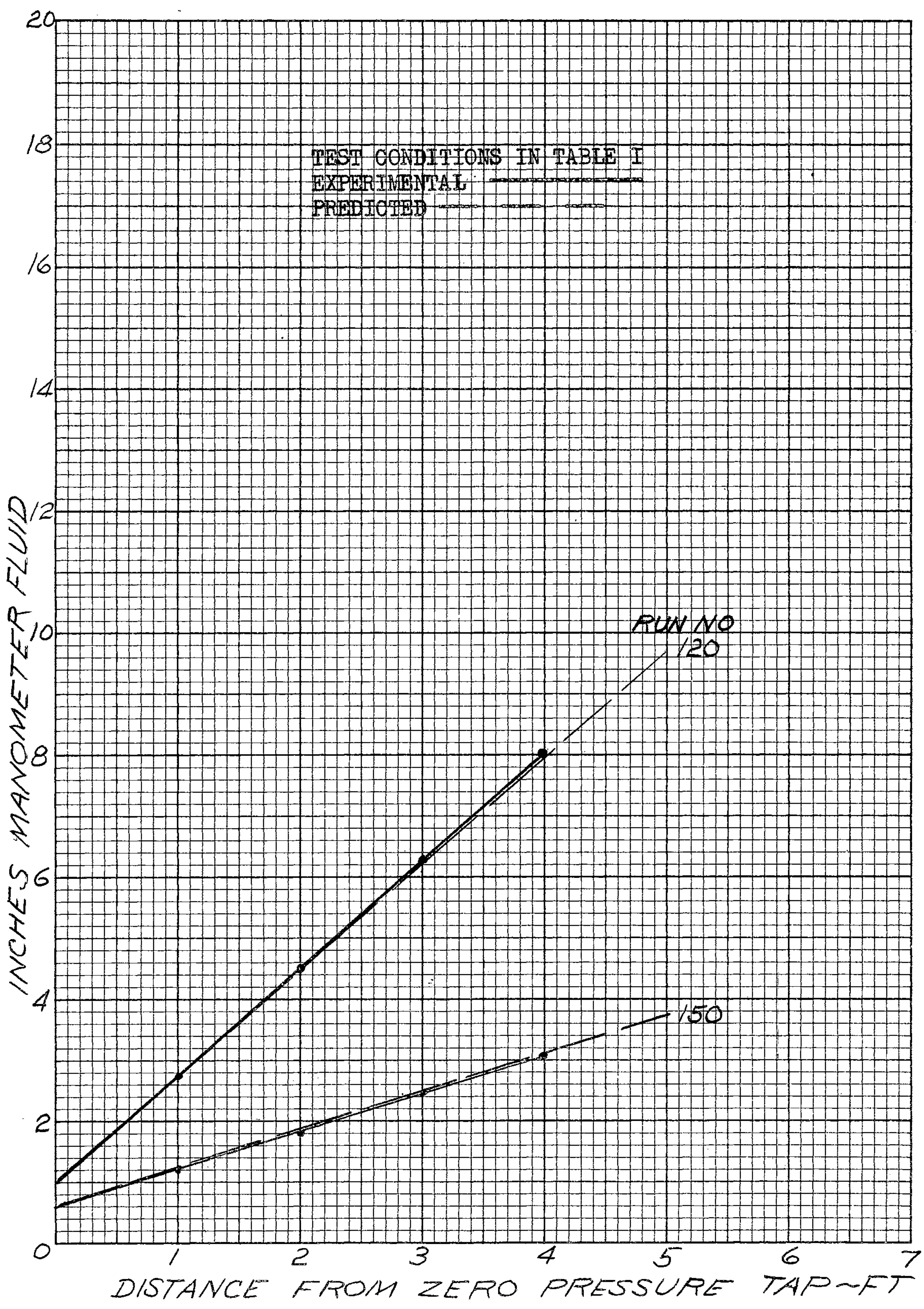

FIG 17 NONISOTHERMAL PRESSURE PROFILES

VITA

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