THE DESIGN OF EQUIPMENT TO DETERMINE
TWO-PHASE, TWO-COMPONENT
HEAT TRANSFER COEFFICIENTS

By

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Thesis Approved:

Thesis Adviser

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PREFACE

The Atomic Energy Commission has given Oklahoma State University a grant to build a water heat transfer loop. The purpose of this thesis is to present the design of supplementary equipment and instrumentation to convert the basic water loop into a two-component loop.

The design of equipment needed to introduce air and extract air from the basic water loop is presented. In order for the reader to be familiar with the basic water loop a chapter describing it was included. The procedure of testing the two-component loop is also presented. The calculations to determine the heat transfer coefficients are presented in the appendix.

The design incorporates the following items: compressor, oil-water separator, air purifier, air filter, pressure, temperature, and flow rate instrumentation, mixing tee, sight glass, and air bleed-off valve.

I wish to express my thanks to Dr. J. H. Boggs for his invaluable aid and assistance in the writing of this paper. I wish to thank, also, Mr. G. E. Tanger for his helpful suggestions in the preparation of this paper. I am indebted to Professor B. S. Davenport for his constructive criticism of my thesis. Indebtedness is expressed to Mrs. Mildred Avery for efficient and capable services as my typist. Acknowledgement would be incomplete without mentioning the sacrifices made by my wife, Marlene, to whom I dedicate this paper.

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SYMBOLS AND ABBREVIATIONS

A_c = circumferential area, square inches or square feet

α_1 = ratio of actual cross-sectional area of liquid flowing to area of a circle of diameter D_L.

α_2 = temperature coefficient of electrical resistivity

β_1 = ratio of actual cross-sectional area of gas flowing to area of a circle of diameter D_g.

β_2 = temperature coefficient of thermal conductivity

C_L = experimental constant

C_g = experimental constant

C_p = specific heat at constant pressure

D_L = actual diameter of pipe filled with liquid, inches or feet

D_p = diameter of pipe, inches or feet

D_g = actual diameter of pipe filled with gas, inches or feet

E = voltage drop, volts

°F = temperature degrees Fahrenheit

f = Fanning friction factor

g = gravitational constant

G = mass velocity

h_L = heat transfer coefficient for liquid phase

h_{TP} = heat transfer coefficient of two-phase, two-component flow

h_x = heat transfer coefficient at (x) cross-section

I = electrical current, amps

J = conversion constant

K = thermal conductivity

M = experimental constant

N = experimental constant

P = power, watts

Q = power dissipated, Btu/hr

R_x = electrical resistivity at (x) cross-section, ohms

R = electrical resistivity

r_o = radius, outside of test section, inches

T_o = outside wall temperature, °F

T_b = bulk temperature of fluid, °F

T_s = inside wall temperature, °F

$\triangle T_w$ = temperature drop through wall of test section, °F

$\triangle T_f$ = temperature drop through film, °F $(T_s - T_b)$

t = turbulent flow

u = dynamic viscosity

v = viscous flow

$\triangle x$ = tube wall thickness

Z_g = experimental constant

Z_L = experimental constant

ρ_L = average density of liquid

ρ_G = average density of gas

ABBREVIATIONS

AICHE - American Institute of Chemical Engineers

ASME - American Society of Mechanical Engineers

cfm - cubic feet per minute

gpm - gallons per minute

in - inches

rpm - revolutions per minute

Trans - transactions

CHAPTER I

INTRODUCTION

The subject of two-phase, two-component flow has become increasingly important because of the need for a method to determine the pressure drop and heat transfer coefficient for gas-liquid flow. The petroleum industry has been interested in pressure drop because this type of flow exists in gas-lifts. The power industry is interested in heat transfer characteristics because this type of flow is encountered in atomic reactors.

I. Description of Two-Phase, Two-Component Flow

The phenomenon of two-phase, two-component flow is very complex to analyze mathematically. The classical laws of fluid flow and heat transfer no longer define the phenomenon taking place. The velocity and temperature profiles can no longer be predicted. These usual profiles are distorted by the introduction of two fluids at different rates of flow and different temperatures. This means that the maximum velocity and temperature of each fluid is not at the center of the pipe, which would be the case if either fluid was flowing alone in the pipe. This leaves the determination of the average velocity and temperature to experimental means. All of the results presented to date on this subject have been experimental correlations. The phenomenon called two-phase, two-component flow indicates that there are two fluids of different specific weights flowing in the same pipe; either of the two fluids may be in the liquid or gas phase, or in both phases. One example might be water and air flowing concurrently.

II. Flow Mechanisms

The descriptions are further complicated by the fact that in gas-liquid flow there are four possible combinations of viscous and turbulent flow conditions. This is to say that each fluid could be in either of the two flow conditions. These combinations are as follows:

	Liquid		Gas
A.	Viscous	————	Viscous
B.	Viscous	————	Turbulent
C.	Turbulent	————	Viscous
D.	Turbulent	————	Turbulent

III. Flow Patterns

In addition to these flow combinations there are numerous geometric flow patterns. To understand and illustrate these patterns, the case of a horizontal pipe will be described. (See Fig. 1).

A. BUBBLE FLOW - If a small amount of gas is admitted to a pipe flowing full of a liquid, the gas would flow along the top of the pipe in small bubbles. This is called bubble flow.

B. PLUG FLOW - The gas rate is increased until the bubbles collect into plugs which flow at intervals at the top of the pipe.

C. STRATIFIED FLOW - The gas rate is increased until the plugs connect and the gas occupies a definite portion of the pipe. This is called stratified flow.

D. WAVY FLOW - This flow is essentially stratified but the gas rate is increased to form waves on the liquid surface.

E. SLUG FLOW - The gas rate is increased so that the waves fill the entire pipe at intervals.

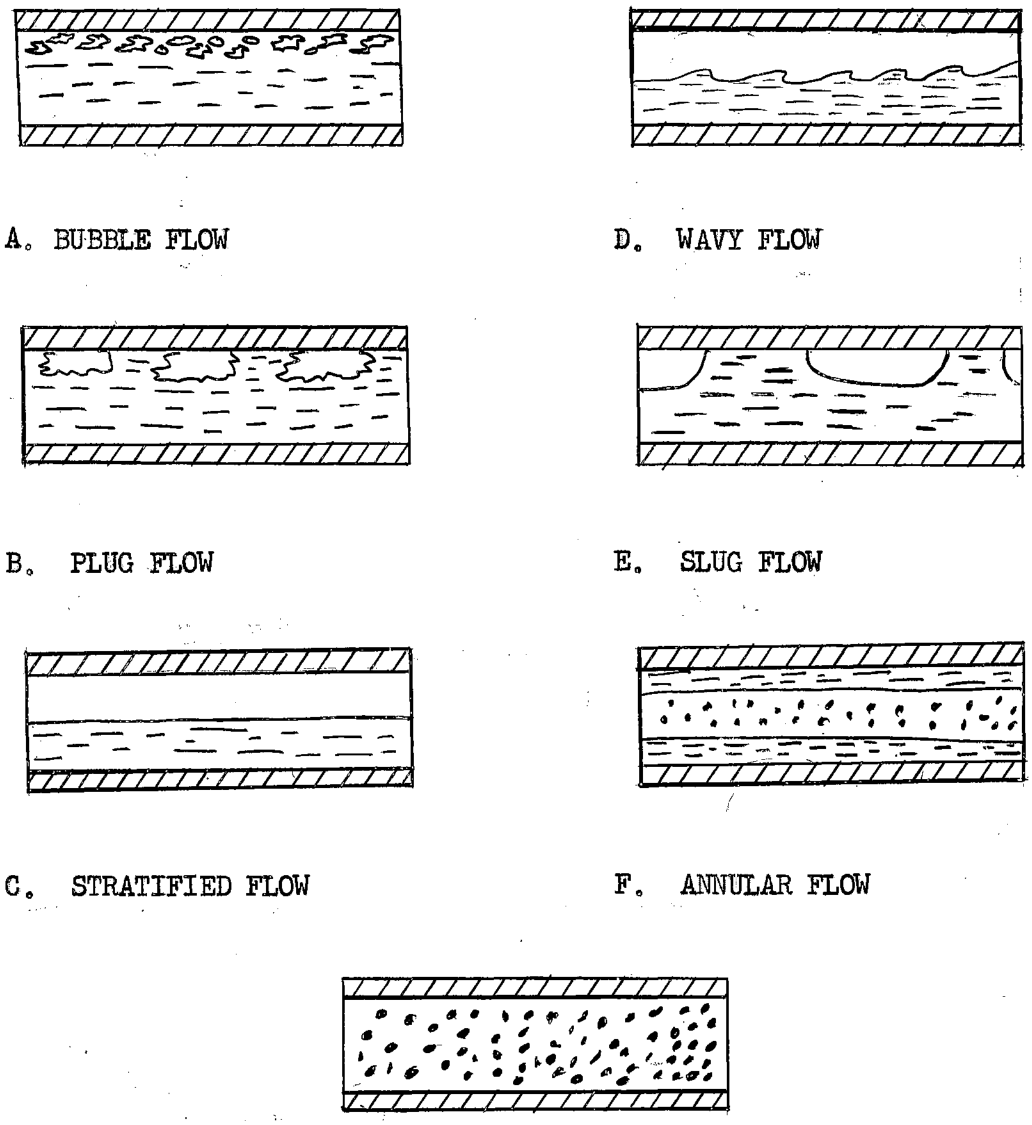


Fig. 1 Flow Patterns Occurring in a Horizontal Pipe

F. ANNULAR FLOW - At high flow rates the liquid moves to the tube walls and the gas flows in the core.

G. MIST FLOW - At very high flow rates the annular layer of liquid disintegrates into a mist or spray evenly distributed in a continuous gas flow.

CHAPTER II

PREVIOUS RESEARCH

The study of pressure drop and heat transfer characteristics of two-phase, two-component flow has been conducted by numerous investigators. The results may be found in the articles listed in the appendix. The most widely accepted work done on pressure drop was accomplished by Martinelli and Lockhart. (4). The most widely accepted work done in the field of heat transfer was done by Johnson and Abou-Sabe. (7). These men correlated all experimental variables into dimensionless parameters.

Martinelli et al, made two basic assumptions in their correlation. These are:

1. The static pressure drop for the liquid phase equals the static pressure drop for the gaseous phase regardless of the flow pattern, so long as an appreciable radial static pressure difference does not exist.
2. The volume occupied by the liquid plus the volume occupied by the gas at any instant must equal the total volume of the pipe.

The static pressure drop, defined by the Fanning equation, for liquid and gas flow is:

$$\left(\frac{\Delta P}{\Delta L}\right)_{TP} = \left(2\ f_L\ \frac{\rho_L\ V_L^2}{D_L\ g}\right) \text{ - - - - - - - - Liquid} \qquad (1)$$

$$\left(\frac{\Delta P}{\Delta L}\right)_{TP} = \left(2\ f_g\ \frac{\rho_g\ V_g^2}{D_g\ g}\right) \text{ - - - - - - - - Gas} \qquad (2)$$

Substituting into Eq. (1), the friction factor in Blasius form and the velocity in terms of mass and area are:

$$\left(\frac{\Delta P}{\Delta L}\right)_{TP} = \left(\frac{\Delta P}{\Delta L}\right)_{L} \alpha^{N-2} \left(\frac{D_p}{D_L}\right)^{5-N} \tag{3}$$

For simplicity Martinelli defined:

$$\phi_L^2 = \frac{\Delta P}{\Delta L}_{TP} \Big/ \frac{\Delta P}{\Delta L}_{L} \tag{4}$$

Thus:

$$\phi_L = \alpha_1^{\frac{N-2}{2}} \left(\frac{D_p}{D_L}\right)^{\frac{5-N}{2}} \tag{5}$$

and similarly for equation (2):

$$\phi_g = \beta_1^{\frac{M-2}{2}} \left(\frac{D_p}{D_g}\right)^{\frac{5-M}{2}} \tag{6}$$

where (α_1) and (β_1) are the ratio of actual cross-sectional area of flow to the area of a circle of diameter (D_L) and (D_g). The values of (N), (M), (C_L), (C_g) are experimental constants.

It is assumed that all other variables not included in (ϕ) are functions of a new variable (X).

The new variable may be defined as:

$$X^2 = \frac{\left(\frac{\Delta P}{\Delta L}\right)_L}{\left(\frac{\Delta P}{\Delta L}\right)_g} \tag{7}$$

These parameters will have subscripts defining the state that each fluid is in, such as, vv, vt, tv, tt. The first letter of each subscript describes the liquid flow condition, while the second defines the gas.

Johnson and Abou-Sabe postulated that the heat transfer coefficient for two-phase, two-component flow was defined by:

$$h_{TP} = Z_g h_L + Z_L h_g \tag{8}$$

where Z_g and Z_L are, respectively, gas and liquid rate factors with values between (0) and (1), to be determined from experimental data. The h_L and h_g are, respectively, liquid and gas heat transfer coefficients. The h_L and h_g are assumed to be represented by the following equation:

$$\frac{hD}{K} = 0.023 \left(\frac{DG}{u}\right)^{0.8} \left(\frac{C_{pu}}{K}\right)^{0.4} \tag{9}$$

where D and G are based on effective hydraulic diameters.

The values of (Z_L) for the data presented by Johnson and Abou-Sabe were found to be quite small. The terms were dropped from equation, and the empirical relation for (Z_g) was obtained.

$$Z_g = \frac{1}{1 + 0.006\ (D_p G_g u_g)} \tag{10}$$

This is true if the liquid determines the volume of gas flowing in the system. The values of (Z_g) and (Z_L) are calculated from experimental data. They are calculated from Eq. (8). This equation equates the heat transfer coefficient for two-phase flow to the sum of the products of the heat transfer coefficient for liquid and gas flow, and those constants. The results then may be correlated to give a direct solution of the two-phase, two-component heat transfer coefficients.

This correlation has yet to be given absolute verification.

CHAPTER III

STATEMENT OF PROBLEM

The Atomic Energy Commission has given Oklahoma State University a grant for construction of apparatus to study single-component pressure drop and heat transfer characteristics. The regions of investigation are forced convection, local boiling, and net boiling.

Since the original grant was for the study of single-component flow, it was decided by the author to design the supplementary equipment necessary to study two-phase, two-component flow. The original loop was designed to use water as the single-component. Air is the second component for which the present alteration in the design was made. The object, therefore, becomes the design of the equipment to introduce air into the loop and the equipment to extract air from the loop. This object is to be accomplished without appreciable change in the original loop, if possible. The secondary object is to present the calculations necessary to determine the heat transfer coefficient.

CHAPTER IV

SELECTION OF DESIGN VARIABLES

Many of the design variables depend upon the type and capacity of the compressor. The compressor chosen is a two-stage, positive-displacement compressor with intercooler. The compressor is rated at 75 cfm free air at a discharge pressure of 125 psig. The compressor can be operated at a discharge pressure of 200 psig. This increase in discharge pressure decreases the flow rate to 65 cfm free air. Since the compressor will be discharging into a line, the minimum discharge pressure to have flow at the end of the line is the pressure necessary to overcome friction in the line. This value of discharge pressure was chosen to be 20 psig. This sets the maximum and minimum discharge pressures and flow rates. (See Fig. 2).

	Maximum	Minimum
Pressure	200 psig	20 psig
Flow Rate	65 cfm	87 cfm

If the maximum discharge pressure of the compressor is 200 psig, this dictates the maximum pressure of the water which must be 200 psig. This sets the maximum temperature of the water as the saturation temperature at 200 psig. This value is 381°F. The temperature of the air as it reaches the loop will be approximately ambient temperature.

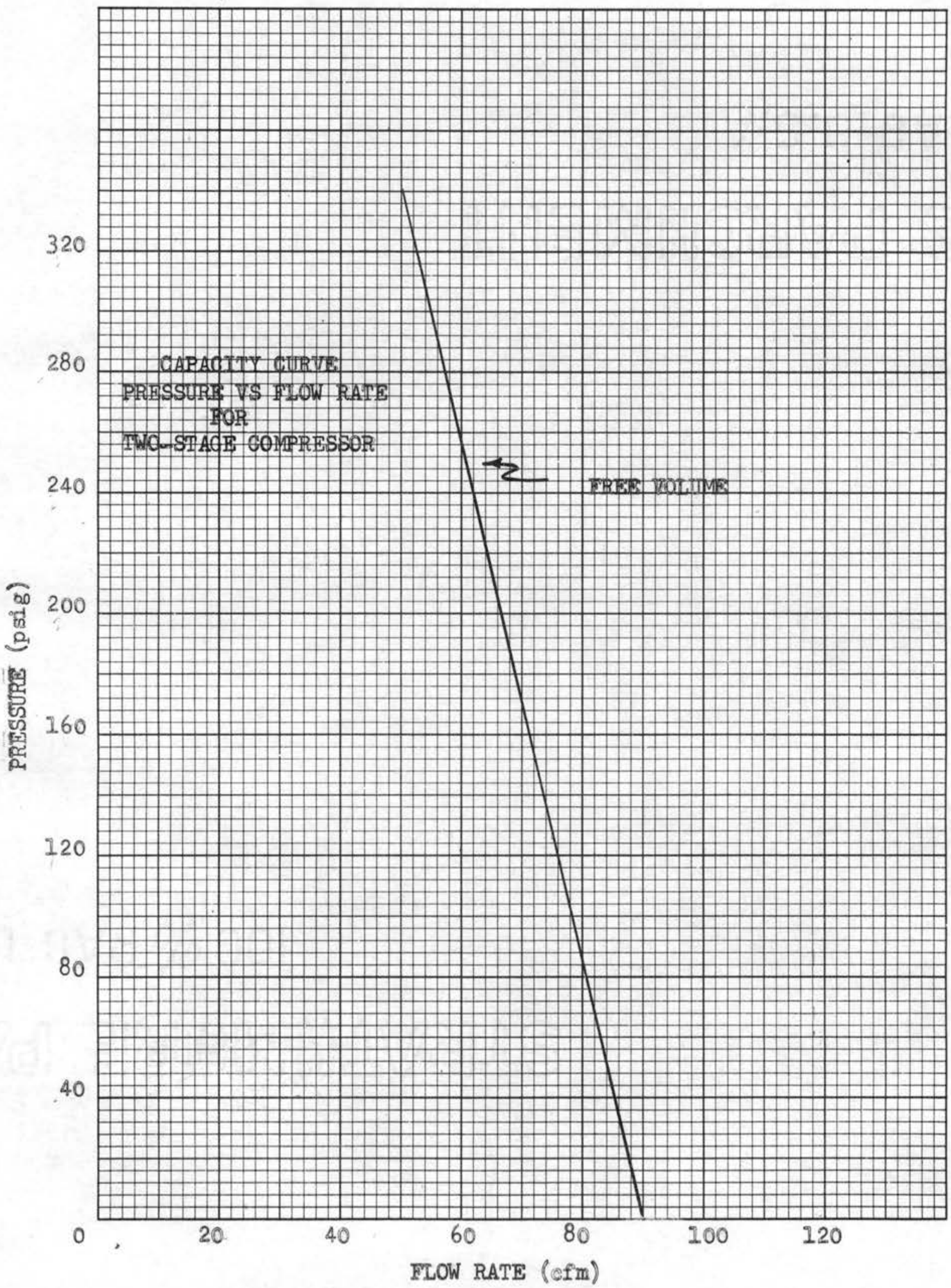


Fig. 2 Pressure vs. Flow Rate for Two-Stage Compressor

The maximum flow of water is 4.55 gpm against a head of 360 psig.

CHAPTER V

DESCRIPTION OF SINGLE-COMPONENT LOOP

The single-component loop is shown schematically in Fig. 3. The test facility consists of a closed loop which circulates distilled water. The test section is heated electrically and the water absorbs the heat generated. The pressure is maintained by throttling the flow at the discharge end of the loop. The flow of the water through the components of the loop will be discussed as the water makes a cycle through the loop.

The water flows from the bottom of the supply tank, point A, to the circulating pump. This pump, point B, is a screw type (MOYNO) pump. It provides a continuous pushing action in one direction without pulsation. It incorporates a high head which is not dependent upon speed. It is rated as follows: 0.26 gpm per 100 rpm of pump against a discharge pressure of 360 psig. The water is forced into the parallel orifice assemblies, point C, which have valves so that the flow may be directed through either of the two orifices.

The orifice pressure taps are connected to the differential pressure transmitter and to the U-tube manometer which is in parallel with the transmitter. The manometer was chosen for flow measurement because of its accuracy. A 3 to 15 psig air signal from the transmitter, together with the 3 to 15 psig control point air pressure,

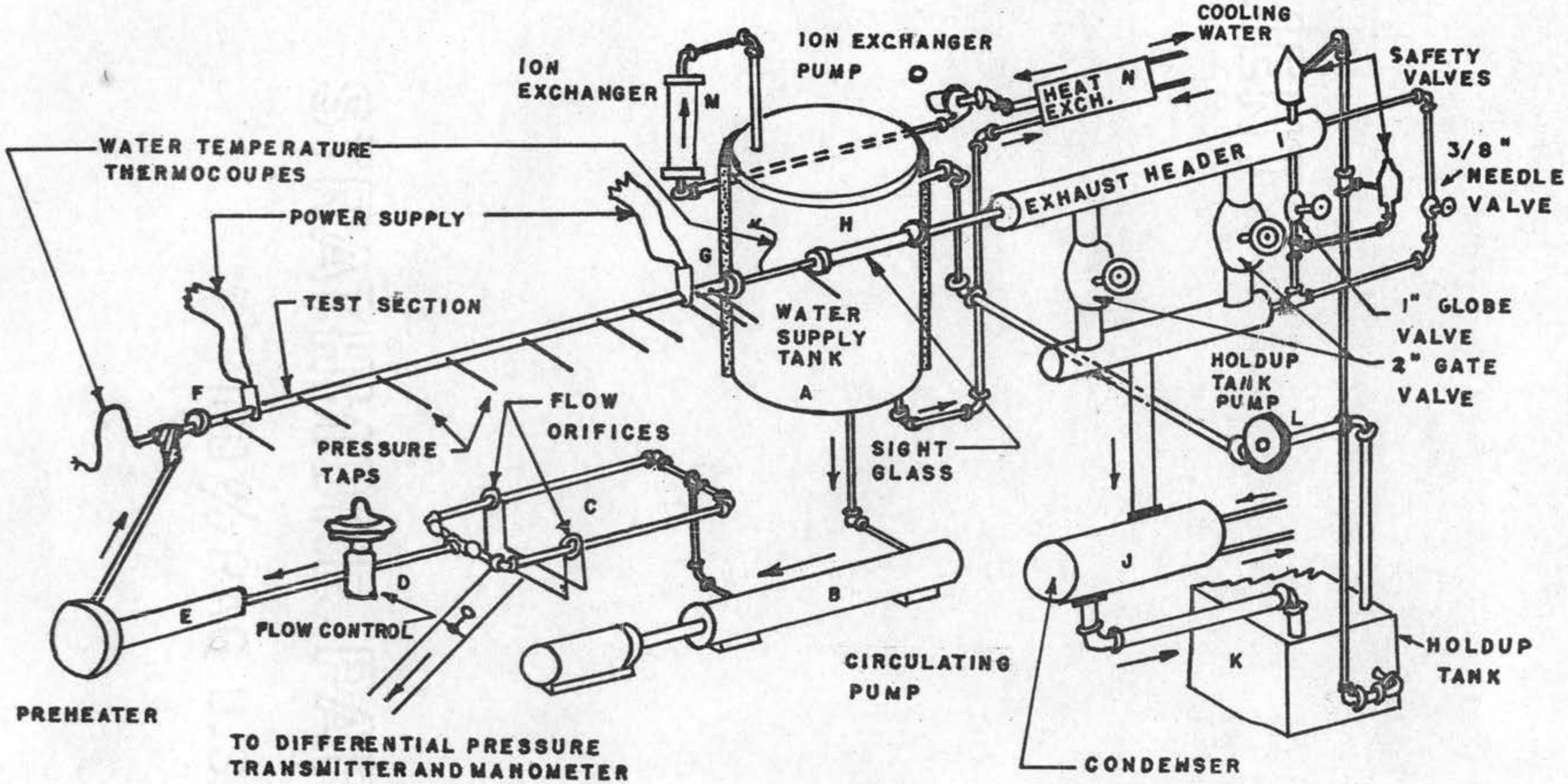


FIG. 3
SCHEMATIC DIAGRAM
OF TEST FACILITY

was balanced by the flow controller which sent a 0 to 15 psig signal to the air-actuated throttle valve, point D.

After the flow is throttled it is allowed to enter a 60 kw, 220 volt single phase Calrod immersion preheater, point E. The preheater is controlled by a powerstat. The powerstat is divided into two 30 kw units. One unit has a variable output. This unit can be varied from 0 to 30 kw in six 5 kw steps, by separate switches. The second unit cannot be varied and has an output of 30 kw. This design gives control over the entire 60 kw range. Either 30 kw unit may be turned off without affecting the system.

The heated water enters the test section through a pair of Teflon-insulated flanges, point F. The temperature and pressure are measured at the entrance to the test section. The temperature is measured with a thermocouple and the pressure with either a Bourdon-tube gage or a manometer.

The test section (points F to G) consists of a round, type 304 stainless steel tube, 0.4628 inch inside diameter, with wall thickness 0.02 inch, and 7 feet long. Thin wall tubing was used for two reasons. The first reason is that the thin wall tubing has a small temperature drop through the wall; the second is that its electrical resistivity changes very slightly with temperature.

Electric power is put into the test section through copper lugs silver-soldered to the surface. Single phase, alternating current will be brought to the test section through three 20 kw welding transformers connected in parallel. The power output may be changed in a stepless manner by using a rotary voltage induction regulator on each transformer, which is synchronized by means of a chain drive, and controlled by an electric motor; thus the transformers share the load equally at the various settings.

Pressure taps are placed along the length of the test section. They are attached by silver solder. The pressure taps give instantaneous readings of static pressure along the loop.

The temperature at the exit to the test section is measured by use of a thermocouple placed in the line, point G. The thermocouple is placed so that it measures temperature of water as it leaves the test section flange.

Visual observation of flow patterns is made possible by the sight glass, point H. It consists of a glass tube with the same inside diameter as the test section. The observation tube length is approximately 4 inches. (See Figure 4).

The exhaust header is made up of a $4\frac{1}{2}$" stainless steel pipe and a 3" stainless steel pipe separated by two 2-inch gate valves, a 1-inch globe valve, and a 3/4-inch needle valve, point I. The valves are used to build up pressure to the desired amount. The gate valves are for large adjustments, and the globe and needle valves are for small adjustments on pressure.

After being throttled in the exhaust header, the steam enters a water cooled heat exchanger, point J. The steam is condensed to saturated water.

Upon leaving the condenser the condensate enters the hold up tank, point K. Additional cooling is attained in the hold up tank, if desired. The condensate is removed from the hold up tank by the condensate

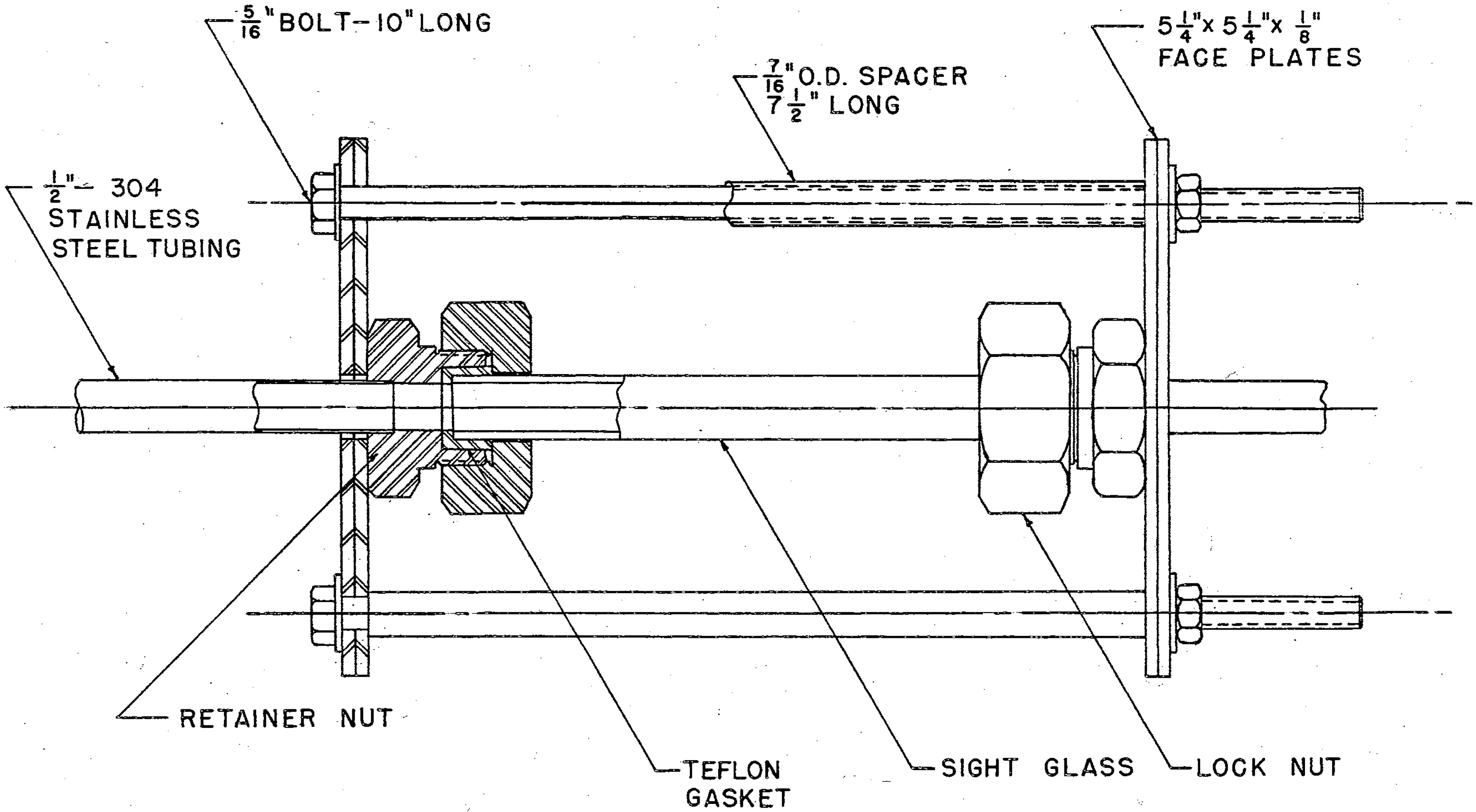


FIG. 4
SIGHT GLASS

pump, point L. The condensate pump operates intermittently, controlled by a float in the supply tank. The condensate pump returns the water to the supply tank.

Since all the piping, valves, and components of the loop are not stainless steel a small ion exchanger is provided, point M. It takes a portion of the water from the supply tank, cools it in a heat exchanger to 100°F, point N. Then the ion-exchanger pump, point O, forces the water through the vertical ion exchanger and back to the supply tank.

The cycle described is carried out continuously while the loop is in operation.

With the data available, the pressure drop and heat transfer characteristics for forced convection, local boiling, and net boiling can be investigated for a single component.

CHAPTER VI

DESIGN

The design of equipment to introduce two-component flow separates into two parts. The first part is the design of the equipment necessary to introduce air into the loop, while the second part deals with the extraction of air from the loop.

I. Introduction of Air

The system which introduces air into the loop is composed of a compressor, flow measurement apparatus, oil-water separator, air purifier, air filter, temperature and pressure measurement apparatus, mixing tee, and sight glass assembly. Each of these components will be discussed in the order in which they are used.

A. Compressor

The compressor, discussed previously, is a two-stage, positive-displacement compressor with intercooler. The compressor was manufactured by the Pennsylvania Pump and Compressor Company. The compressor ratings are as follows: (See Fig. 1).

Maximum R. P. M. --	400
Working Pressure --	125 psig
Flow Rate at Working Pressure --------------------	75 cfm free air
Compressor Driven by 25 hp Electric Motor.	

It is estimated that the compressor can furnish a discharge pressure of 200 psig. This higher discharge pressure will reduce the flow to

67 cfm of free air. The design was therefore made on the basis of 200 psig and 67 cfm of free air.

B. Oil-water Separator, Surge Tanks, and Piping

An oil-water separator is installed on the entrance to the surge tanks. It serves to trap out some of the oil and water vapor in the air.

The surge tanks consist of two cylindrical steel tanks 2 feet in diameter by 5 feet high. The tanks smooth out the surges set up by the compressor. The air is cooled in the tanks by convection to atmosphere, and some of the water vapor will condense out.

The air is carried by a 2-inch standard weight pipe to the test area. The pressure drop through piping was considered negligible. After the air purifier the pipe size is reduced to 1 inch. (See Fig. 5).

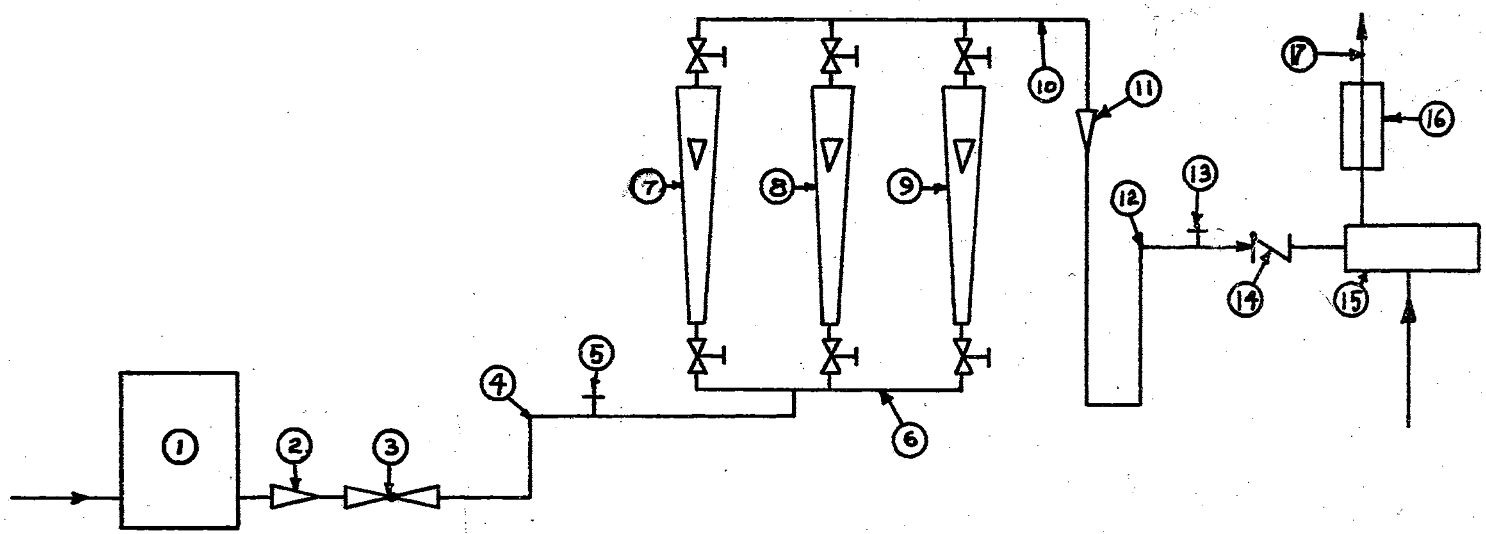
C. Air Purifier

The air purifier is composed of a cylindrical steel tank with a 2-inch pipe mounted in the center. (See Fig. 6). The annular space between the tank and stand pipe is filled with cotton batting. The cotton batting serves to collect the oil carried in suspension in the air. The flow of air is into the bottom and through the stand pipe. The air leaves out of the bottom, after flowing through the batting.

D. Air Filter

The air filter consists of a reinforced, 200-mesh (74 micron), Monel wire screen. Scale deposits and particles of cotton batting carried in suspension by the air will be trapped in the filter.

The specifications on the filter are as follows:



1 AIR PURIFIER
2 2" TO 1" REDUCER
3 THROTTLE VALVE
4 THERMOMETER WELL
5 PRESSURE TAP
6 2" HEADER
7 3/4" ROTAMETER
8 1" ROTAMETER
9 1 & 1/2" ROTAMETER
10. 2" HEADER
11. 2" TO 1" REDUCER
12. THERMOMETER WELL
13. PRESSURE TAP
14. CHECK VALVE
15. MIXING TEE
16. SIGHT GLASS
17. MIXTURE TO TEST SECTION

FIG. 5
PIPING SCHEMATIC

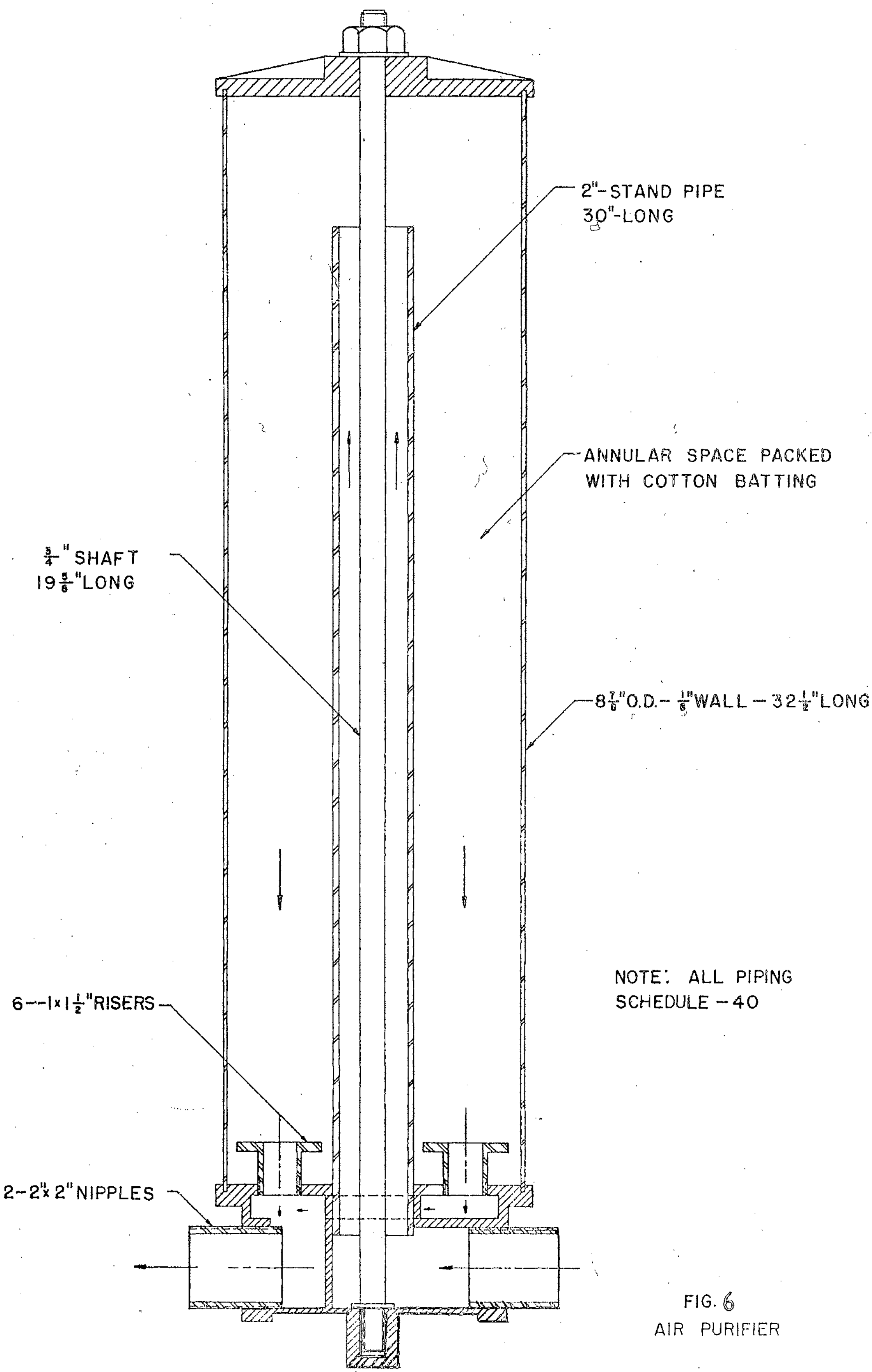


FIG. 6
AIR PURIFIER

Manufactured by	C. A. Norgren Company 3400 South Elati St. Englewood, Colorado
Model Number	560
Temperature Range	40 to 300°F
Maximum Air Pressure	250 psig
Pipe Connection Size	1 inch

E. Air Pressure Regulator

A spring tension, diaphragm operated needle valve was selected as the air pressure regulator or throttle valve. The valve has a Bourdon-tube pressure gage mounted on the valve body. The pressure gage will give throttled pressure. The specifications are as follows:

Manufactured by	C. A. Norgren Company Same address as above
Model Number	2AXB-GG
Temperature Range	40 to 200°F
Maximum Air Pressure	250 psig
Primary Air Pressure Range	20 to 200 psig
Secondary Air Pressure Range	20 to 200 psig
Pipe Connection Size	1 inch

F. Air Temperature

Liquid-in-glass thermometers are used to measure temperature of the air. The thermometers are immersed in a well inserted in the line. The thermometer well consists of a standard pipe tee with a thermometer well inserted in one end. (See Fig. 5). The thermometer well shall be filled with oil (S.A.E. 20). The range of thermometers is 0 to 100°F with 1°F increments. The temperature is measured in two places, after the throttle valve, and before the mixing tee. (See Fig. 5).

G. Pressure Measurement

Pressure is measured with Bourdon-tube gages or mercury manometers.

The pressure taps are connected to a valve which selects the gage or manometer. The pressure taps are fabricated directly to the line, so static pressure will be recorded. The gages have a range of 0 to 300 psig with 5 psig increments. The pressure is measured in two places, after each of the thermometer wells. (See Fig. 5).

H. Air Flow Measurement

Air flow measurement is accomplished by the use of three rotameters in parallel. Three rotameters are used in order to increase the accuracy of flow measurement. The maximum flow to be measured is 87 cfm of free air at 20 psig and 70°F. At the location of the rotameters the pressure will be 20 psig and 60 to 80°F. At these conditions the volume of air will decrease to 56.5 cfm. The specifications for the rotameters are as follows:

Manufactured by ---------------- Fisher and Porter Co.
Hatboro, Pennsylvania

Tube Size	Flow Measurement (cfm free air)	Pipe Size
5	8.50	3/4 inches
6	24.00	1 "
7	33.00	$1\frac{1}{2}$ "

Length of Tube ---------------- 250 millimeters
Type of Float ---------------- Free Stainless Steel

(See Fig. 5 for Piping Details)

I. Mixing Tee

The mixing tee's purpose is to mix the air and water. The design is strictly empirical. The mixing tee consists of two concentric pipes. The smaller pipe, which carries the water and is 3/4 inches in diameter, is welded to the mixing header. The mixing header is 3 inches in diameter. The larger pipe, which carries the air is welded inside the mixing header and extends to within 1 inch of the end of the header. The air enters in a 1 inch line. The two fluids leave the mixing header in a $\frac{1}{2}$ inch line welded to the header. The piping is standard weight stainless steel pipe. (See Fig. 7).

J. Sight Glass

The sight glass gives a visual observation of flow patterns before the mixture reaches the loop. Since the correlations presented to date have been dependent upon the flow pattern occurring, this component is necessary. The sight glass is constructed exactly like the one on the existing loop. The length of the observation is approximately 4 inches. (See Fig. 4).

II. Extraction of Air

Extraction of air from the loop is necessary because of the detrimental effects caused by it. Some of these effects might include air lock of condenser, air block of pumps, and inefficient operation of ion-heat exchanger. The only piece of equipment needed is a heat exchanger and bleed-off valve.

A. Heat Exchanger

As the two-component flow leaves the exhaust header some of the water will flash into steam. Therefore, the flow will now contain saturated water, wet steam, and air. Since the water is air-saturated, it will not accept any more air. Thus, as the mixture flows through the existing heat exchanger the wet steam will be condensed into saturated water. Then the air will be left free. The heat exchanger will be tapped by a bleed-off valve and thus the air can be extracted from the loop. It is possible to lose some of the wet steam but this loss

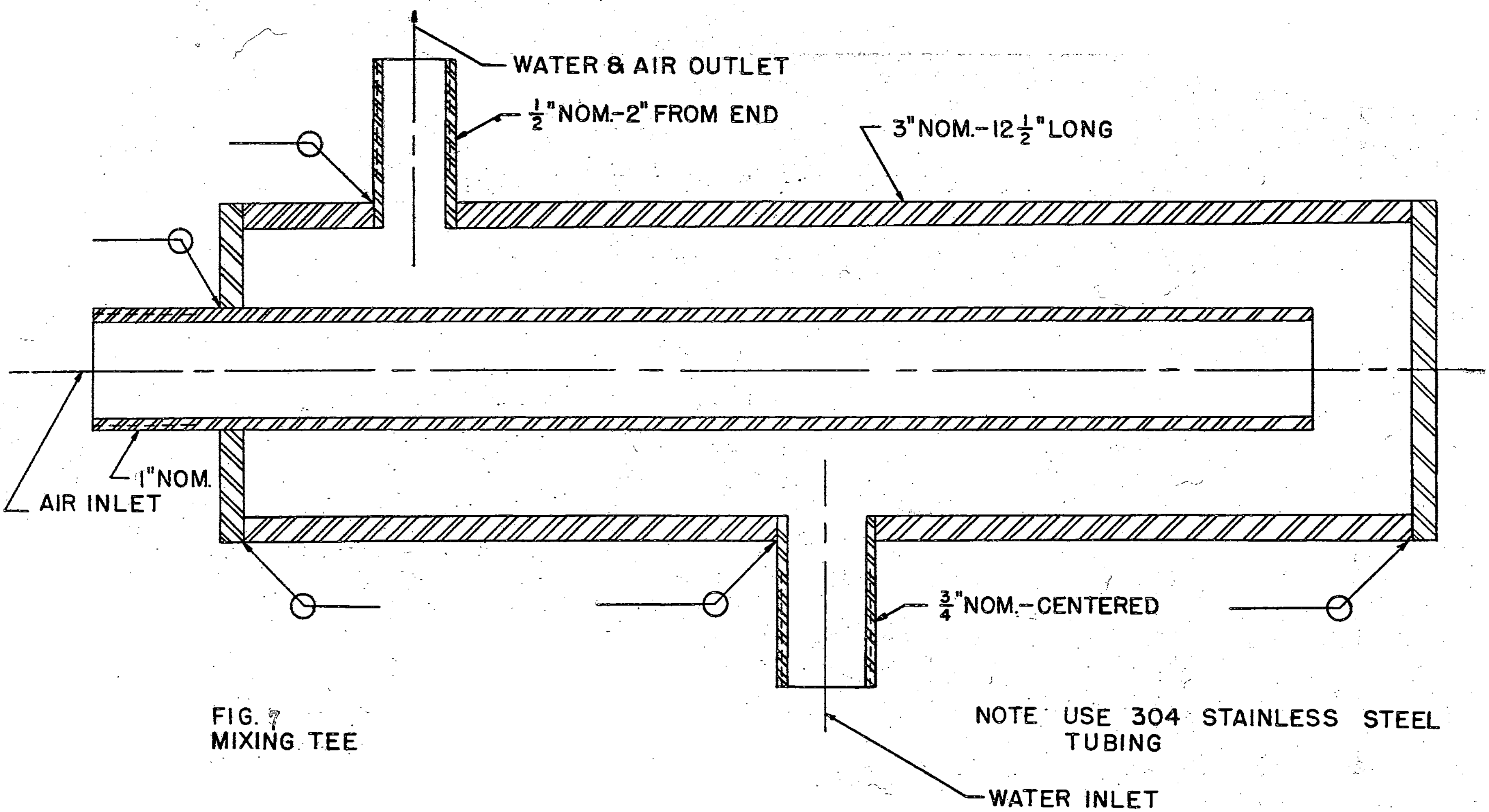


FIG. 7
MIXING TEE

can be considered negligible. The specifications on the existing heat exchanger are as follows:

Manufactured by ————	American Standard Corp. Ross Heat Exchanger Division
Mixture Properties	
Pressure ————	0 psig
Temperature inlet ————	212°F
Temperature outlet ————	212°F
Coolant Properties	
Water Pressure ————	40 psig
Water Temperature Inlet ————	80°F
Water Temperature Outlet ————	105°F

B. Bleed-Off Valve

The bleed-off valve is a 1-inch float type automatic air valve, mounted on the top of the condenser. The bleed-off valve will pass air, closes tight against steam and water. The specifications on the bleed-off valve are as follows:

Manufactured by ————	McAlear Company
Model number ————	15
Size ————	1 inch
Material ————	Stainless Steel or Brass
Maximum Pressure ————	15 psig
Maximum Temperature ————	300°F

CHAPTER VII

TEST PROCEDURE

I. Operation of Loop

A. Safety Features

The air-water heat transfer loop has several safety features which should be discussed before any operating instructions are given. These safety devices are required because of the danger of supplying conditions of temperature, pressure, and power to the system that it may not be capable of handling.

The air system has a safety valve located on the surge tanks to prevent "over-pressure" of components of the system.

The water system pump must be in operation and developing at least 20 psig before power may be applied to the preheater or test section. The maximum steam pressure switch, which automatically shuts off the preheater and test section power, prevents "over-pressure". A master switch which shuts off all power is provided.

B. Preliminary Check List

1. Clean and drain air separator
2. Drain surge tanks
3. Check air purifier and replace cotton batting if dirty
4. Clean air filter bulb
5. Fill thermometer wells with oil (S.A.E. 20)

6. Insert thermometers in wells

7. Close air throttle valve

8. Start cooling water to intercooler of compressor

9. Start compressor

10. Check water supply in the holdup tank and supply tank

11. Check which orifice valve is open

12. Check which rotameter valves are open

13. Fill ice junction with crushed ice

14. Turn on auxiliary air supply to water flow control throttle valve

15. Open all throttle valves at the discharge end of the loop

16. Open water control throttle valve for maximum flow

17. Shut off all water pressure gages except those over 400 psig

18. Close all valves on manometers except by-pass lines

19. Turn on coolant water for condenser

20. Slightly open air bleed-off valve on condenser

21. Move preheater control to minimum position

22. Open ion heat exchanger coolant valves.

C. Operation Check List

1. Turn all power switches to on position

2. Move transformer settings to minimum position

3. Start ion exchange

4. Start circulating pump

5. Adjust ion coolant water so water in ion exchanger is less than 100°F

6. Allow flow to circulate

7. Control pressure to desired value

8. Bleed off air from loop and manometers

9. Open air throttle valve

10. Turn on preheater (if necessary)

11. Put power to the test section

12. Cool discharge water in the holdup tank to correspond with conditions desired but keep below 180 F

13. Readjust water temperature out of ion exchange cooler

14. Set flow through orifice

15. Control inlet temperature to test section

16. Let loop settle out to equilibrium

17. Adjust inlet temperature and control discharge temperature

18. Check air bleed-off valve (open more if necessary)

19. Settle conditions and make adjustments with flow rates of air, water, and power

II. Data To Be Taken

The following data is to be recorded when equilibrium conditions have been reached:

1. Date

2. Run number

3. Dry-bulb ambient temperature

4. Wet-bulb ambient temperature

5. Ambient pressure

6. Flow pattern inlet

7. Flow pattern outlet

8. Power to the test section

9. Current and Voltage to test section

10. Voltage drops along the test section

11. Air flow -- (Rotameter)

12. Water flow -- (Manometer)

13. Surface temperature versus length of test section

14. Static pressure at position number seven

15. Insulation temperatures

16. Rotameters - inlet and exit temperatures

17. Orifice - inlet and exit temperatures

18. Pressure drop at ten positions on test section

19. Orifice dimensions

With the above information, calculations can be made to determine two-component heat transfer coefficients for forced convection, local boiling, and net boiling. An outline of the calculations necessary is presented in the appendix.

CHAPTER VIII

CONCLUSIONS

There are several pieces of equipment in the design of the system that might not function as expected. These are listed as follows:

1. The compressor may not be capable of supplying enough air to produce the design pressure.
2. The air purifier, packed with cotton batting, may not efficiently separate the oil from the air.
3. The mixing tee may not give sufficient turbulence to insure proper mixing.
4. The air bleed-off valve may not exhaust enough air from the condenser in order to insure proper operation of the condenser.
5. There may be surges set up in the line by the compressor.
6. There is a possibility that water may collect in the upper section of the exhaust header. If this occurs the 1-inch globe valve should be opened in order to drain the header.

These factors, which can affect the operation of the system, should be checked before any data is taken. Any part of the design may be changed without affecting the system, so long as an equivalent part to accomplish the same object is inserted in its place.

SELECTED BIBLIOGRAPHY

1. Jakob, M. and Hawkins, G. A., "Elements of Heat Transfer and Insulation". New York: John Wiley and Sons, 1950. Volume I, Second Edition.

2. McAdams, W. H., "Heat Transmission". McGraw-Hill Book Company, Inc., 1954. Third Edition.

3. Martinelli, R. C., Boelter, L. M. K., Taylor, T. H. M., Thomsen, E. G., and Morrin, E. H., "Isothermal Pressure Drop for Two-Phase, Two-Component Flow in a Horizontal Pipe". Trans. ASME, February, 1944, pp. 139-51.

4. Lockhart, R. W., and Martinelli, R. C., "Proposed Correlation of Data for Isothermal Two-Phase, Two-Component Flow in Pipes". Chem. Engr. Progress, Vol. 45, No. 1, January, 1949.

5. Bergelin, O. P., "Flow of Gas-Liquid Mixtures", Chem. Engr., Vol. 56, May, 1949, pp. 104-106.

6. McAdams, W. H., Woods, W. K., and Bryan, R. L., "Vaporization Inside Horizontal Tubes". Trans. ASME, Vol. 63, 1941, page 545.

7. Johnson, H. A., and Abou-Sabe, A. H., "Heat Transfer and Pressure Drop for Turbulent Flow of Air-Water Mixtures in a Horizontal Pipe". ASME Trans., Vol. 74, No. 6, August, 1952, page 977.

8. Freid, Lawrence, "Heat Transfer and Pressure Drop for Air-Water Mixtures Flowing in a Horizontal Tube". University of California, M. S. Thesis, January, 1953.

9. Martinelli, R. C., Lockhart, R. W., Putnam, J. A., "Two-Phase, Two-Component Flow in the Viscous Region". Trans. AICHE, Vol. 42, No. 4, August, 1946, page 681.

10. Mumm, J. F., "Heat Transfer to Boiling Water Found Through a Uniformly Heated Tube". M. S. Thesis, Illinois Institute of Technology, ANL-5276, 1954.

11. Kreith, F., and Summerfield, M., "Heat Transfer to Water at High Flux Densities with and without Surface Boiling". California Institute of Technology Jet Propulsion Laboratory, Pasadena, California, April 2, 1948.

12. Rohde, R. R., "The Methods and Apparatus Used in the Experimental Determination of Water Film Coefficients". Argonne National Laboratory, Lemont, Illinois.

13. Eshbach, O. W., "Handbook of Engineering Fundamentals". New York: John Wiley and Sons, Second Edition, 1952.

APPENDIX A

CALCULATIONS

An outline is presented of the calculations to be performed in order to determine heat transfer coefficients at any cross-section for forced convection, local boiling, and two-phase flow. The air-water mixture is flowing in a horizontal test section.

The following assumptions were made in this analysis:

1. Steady state

2. The bulk temperatures are considered to be thoroughly mixed fluid temperatures at a cross-section

3. Unity power factor

I. A curve of the variation of resistance with temperature should be plotted for the material of the test section (304 Stainless Steel). (See Fig. 8).

II. A curve of temperature variation with length of the test section should be plotted. (See Figures 9, 10, and 11).

III. From these curves the resistance of the test section at any cross-section may be determined.

IV. Since the current flow is constant through all cross-sections, the power dissipated at any cross-section may be calculated.

$$P = I^2 R_x$$

V. The power dissipated should be converted into BTU/HR.

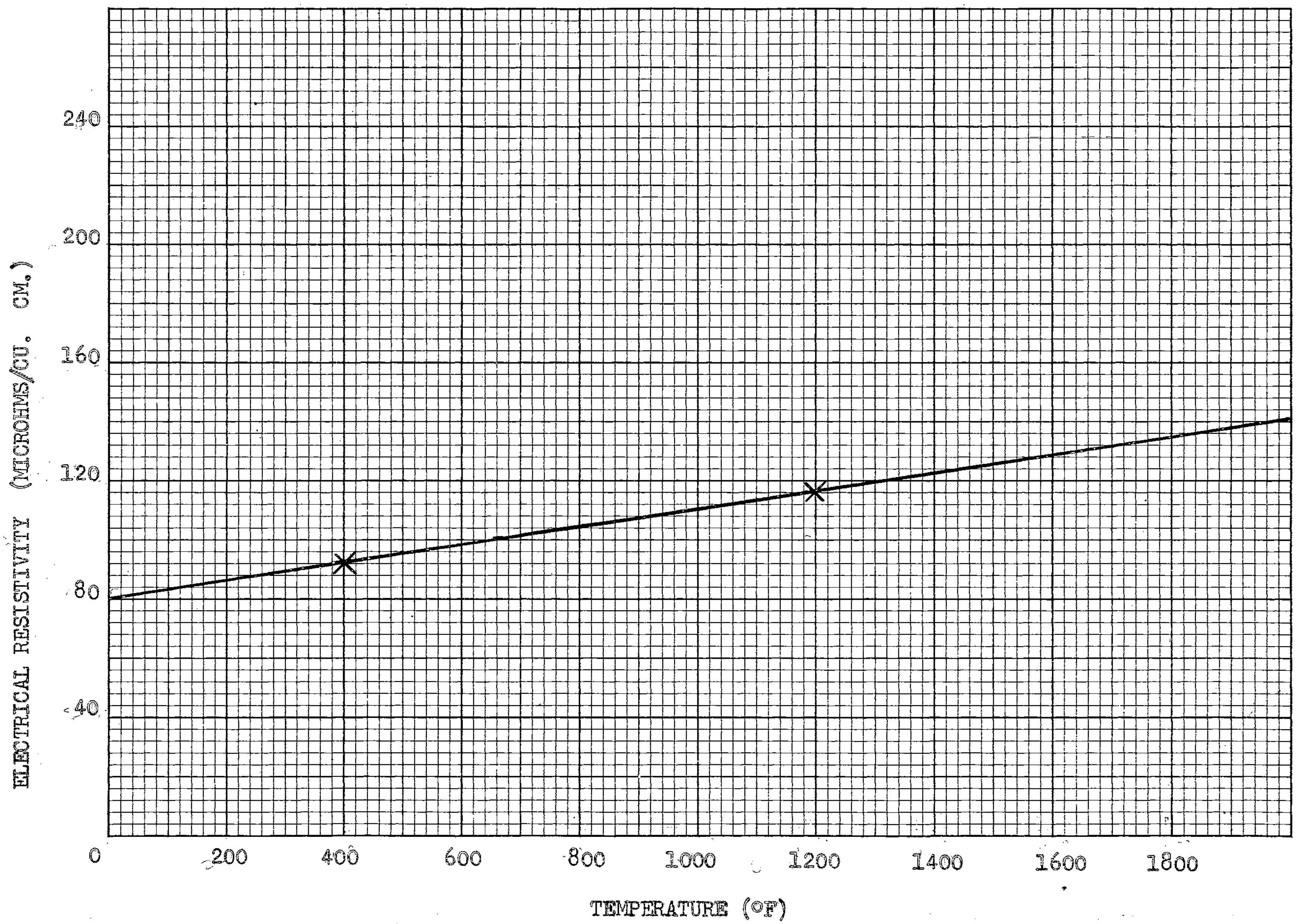


Fig. 8 Electrical Resistivity versus Temperature for 304 Stainless Steel

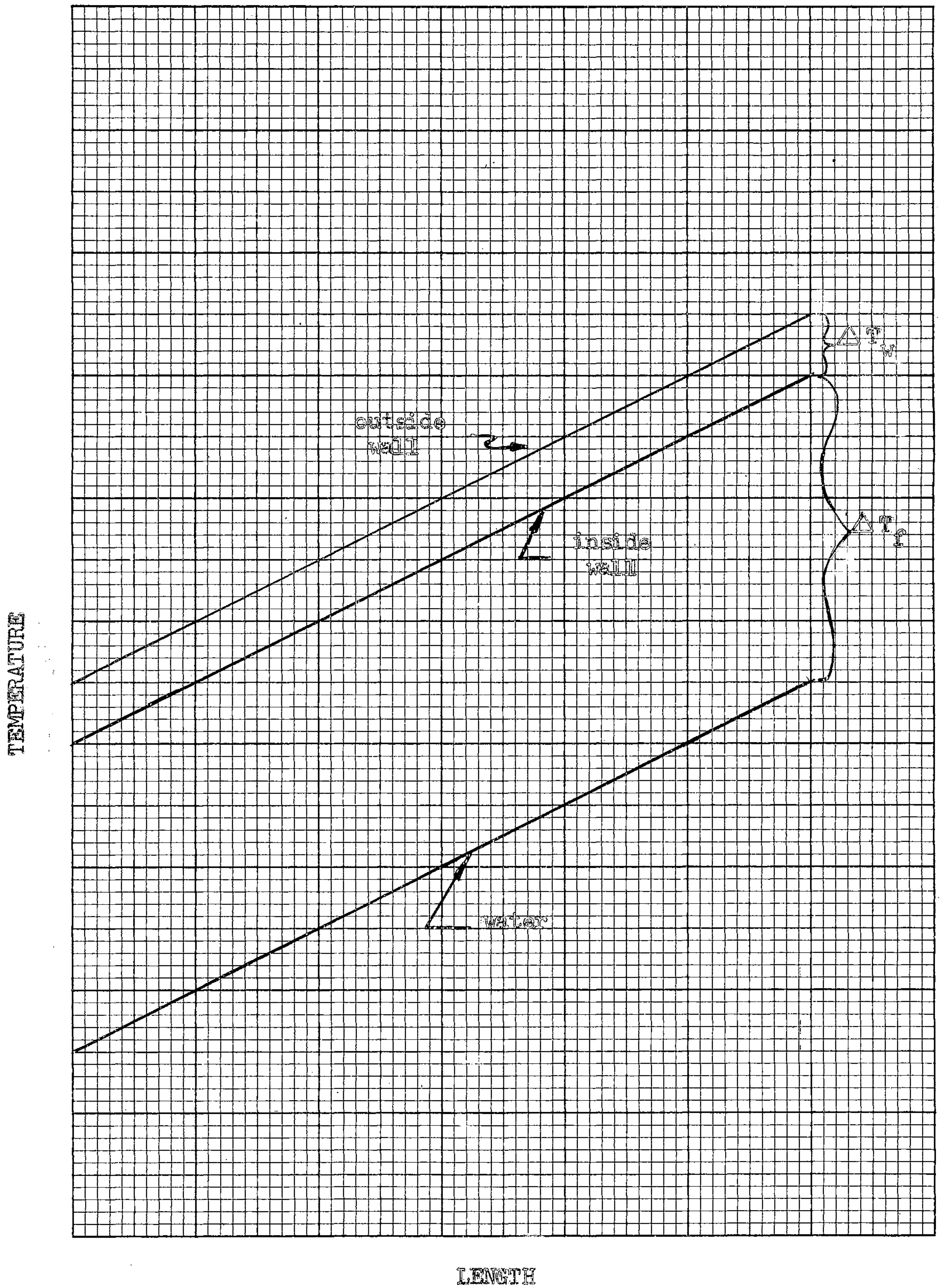


Fig. 9 Temperature Versus Length of Test Section
During Forced Convection

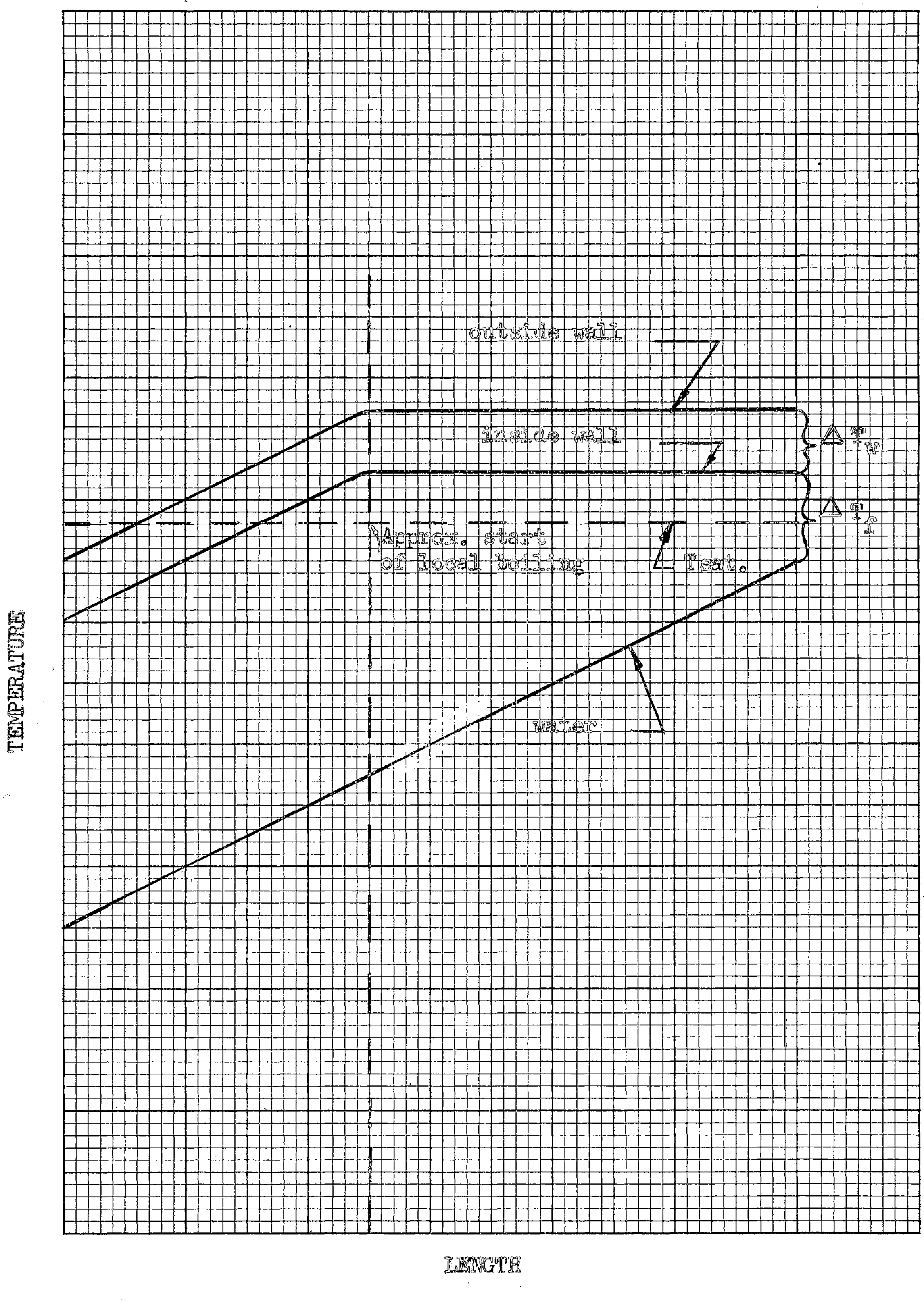


Fig. 10 Temperature Versus Length of Test Section for Local Boiling

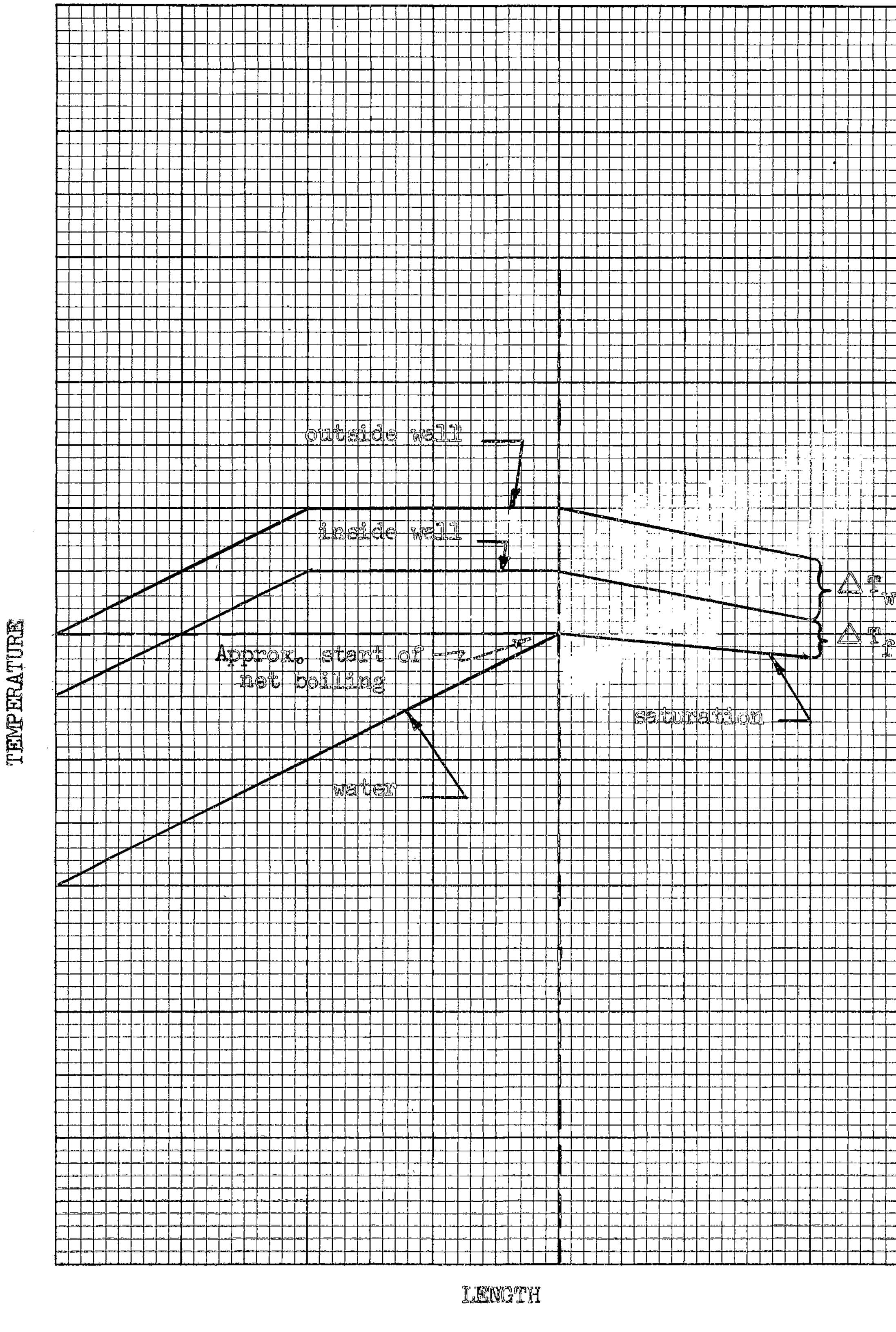


Fig. 11 Temperature Versus Length of Test Section for Net Boiling

VI. The heat flux (q'') can be calculated at any cross-section by dividing the heat flow by the cross-sectional area of the test section.

VII. The inside surface temperature can be determined from: (11)

$$T_s = T_o - \frac{M\Delta x^2}{(1 + \beta_2 T_o)(1 + \alpha_2 T_o)} - \frac{M\Delta x^3}{3r_o(1 + \beta_2 T_o)(1 + \alpha_2 T_o)}$$

$$+ \frac{M^2(3\alpha_2 + 4\alpha_2\beta_2 T_o + \beta_2)}{6(1 + \beta_2 T_o)^3(1 + \alpha_2 T_o)^3} + \frac{M\Delta x^4}{4r_o^2(1 + \beta_2 T_o)(1 + \alpha_2 T_o)}$$

where

$M = \frac{E^2}{2JK_1R_1}$

E = voltage drop

r_o = outside radius of test section (0.2514 inches)

J = conversion constant

K_1 = thermal conductivity at a point

R_1 = resistivity at a point

ΔX = tube wall thickness

α_2 = temperature coefficient of electrical resistivity

β_2 = temperature coefficient of thermal conductivity

T_o = outside surface temperature

T_s = inside surface temperature

These constants used in the equation are available in handbooks.

VIII. Determine from plot of bulk temperature versus length what the bulk temperature is at the specified cross-section.

IX. Determine from Newton's Law of Cooling the heat transfer coefficient at the cross section.

$$h_x = \frac{q}{A_s \Delta T_f}$$

The heat transfer coefficients can be determined at all cross-sections of the test section and a plot of heat transfer coefficient versus length may be plotted. This curve may be integrated to give an average heat transfer coefficient.

APPENDIX B

ANALYSIS OF RESULTS

This analysis is presented to discuss some of the calculations stated in Appendix A, and to show the analysis necessary to determine the types of heat transfer. The three types of heat transfer are forced convection, local boiling, and net boiling.

The variation of resistance with temperature as shown on the graph in Fig. 8 was assumed linear. This is not absolutely true; however the error is very slight. The maximum difference in temperature between inlet and outlet positions on the test section is considered to be approximately 100° F. This difference in temperature will only produce a slight difference in the electrical resistivity throughout the test section. Therefore, the heat flux will be approximately uniform over the test section. The assumption of uniform heat flux was not made so that this fact could be recognized.

A graph of average heat transfer coefficient versus air flow rate can be made in order to observe the variation in heat transfer coefficient due to the increase in air flow rate.

A plot of temperature versus length, Figures 9, 10, and 11, show the variation of temperature of outside surface, inside surface, and water bulk temperatures. From these graphs it is possible to note the types of heat transfer taking place in the test section.

In Fig. 9, all the temperatures are linear functions of length of the test section. This indicates that the heat transfer taking place is forced convection.

In Fig. 10, only the water bulk temperature is a linear function of length of the test section. The surface temperatures increase to the approximate point of local boiling; then these temperatures level off in a horizontal line. Since the water bulk temperature at outlet does not reach the saturation temperature at the outlet pressure of the test section, the type of heat transfer is local boiling.

In Fig. 11, none of the temperatures are linear functions of length of test section. If the water bulk temperature at any point in the test section is greater than the saturation temperature at that point, steam will be formed. The formation of steam will increase as the flow progresses through the test section because the pressure will decrease. This type of heat transfer is called net boiling.

APPENDIX C

ERRORS AND ACCURACY

The errors involved in the determination of the heat transfer coefficient are discussed. These errors will be discussed in relation to the equation of the heat transfer coefficient. The equation is:

$$h = \frac{q}{A_c \, \Delta T_f}$$

The heat flow (q) can be measured very accurately, because it is measured directly, as power dissipated to the loop. The heat loss through the insulation can be determined by the temperatures of the insulation. This calculation points up any loss of heat that might affect the results of the test.

The factor ΔT_f is the primary source of error. The measurement of the fluid bulk temperature is extremely difficult. This is true because there is not a uniform temperature distribution throughout the fluid. The average temperature of the fluid can be measured within $\pm$ 1° F.

The inside diameter of the test section is determined by actual measurement and is subject only to errors in the micrometers.

The estimate of overall accuracy of measurement of heat transfer coefficient is postulated to be within $\pm$ 10% of the actual value.

VITA

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