THE PERFORMANCE OF AN INCLINED TUBE NATURAL CIRCULATION EVAPORATOR

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NATURAL CIRCULATION EVAPORATOR

By

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PREFACE

This thesis is concerned with the following subjects in the field of evaporation:

The effect of the angle of inclination on the boiling film coefficient and the rate of natural circulation at various steam pressures.

TABLE OF CONTENTS

	Page
Introduction	1
Description of Apparatus	18
Procedure	32
General Nomenclature	34
Sample Calculations	36
Discussion of Results	37
Conclusions	43
Literature Cited	44
Appendix	45

Introduction and Literature Survey

In the majority of the heat transfer problems encountered, heat is being transferred from one fluid through a solid retaining wall to another fluid. For such conditions it has been found that the rate of heat transfer q is directly proportional to the overall difference between the temperature of the warmer and colder fluids $t_w - t_c$ or ΔT, and to the heat transfer surface A.

$$q = UA\Delta T \tag{1}$$

where U is a proportionality factor, more commonly known as the overall coefficient of heat transfer.

Before the overall coefficient is established, a definite surface area A must be chosen. The selection of this area is arbitrary, and may be any one of three areas, i.e., the area on the cold side, or the area on the hot side or an average of the two. Once the surface area is chosen the coefficient is automatically based on it.

The heat flux from one fluid medium to another through a retaining wall is retarded by the presence of thermal resistances; namely, the films present on the surface areas and the solid medium through which the heat must pass. Calculations of heat flux through these resistances is quite complicated unless an indirect method is employed, and this involves the computation of film coefficients. The individual resistances after being computed are correlated with the overall heat transfer coefficient U.

$$U_i = \frac{1}{\frac{A_i}{h_1 A_1} \quad \frac{LA_i}{kA_{av}} \quad \frac{A_i}{h_2 A_2}} \tag{2}$$

where U_i = Overall heat transfer coefficient based on A_i
L = Thickness in feet of retaining wall
h = Film coefficients
A_1, A_{av}, A_2 = Surface area in square feet
k = Thermal conductivity of retaining wall
A_i = Indicates one of the following areas; A_1, A_2, or A_{av}

Substitution into equation 1 gives

$$q = \frac{\Delta T}{\frac{1}{h_1 A_1} + \frac{L}{k A_{av}} + \frac{1}{h_2 A_2}} \tag{3}$$

By assuming that U is to be based on A_1, equation 3 becomes

$$q = \frac{A_1 \Delta T}{\frac{1}{h_1} + \frac{L A_1}{k A_{av}} + \frac{A_1}{h_2 A_2}} \tag{4}$$

$$q = \frac{A_1 \Delta T}{\frac{1}{h_1} + \frac{L D_1}{k D_{av}} + \frac{D_1}{h_2 D_2}} \tag{5}$$

By use of equation 5, a study of film coefficients may be made.

The matter of treatment of these equations may be found in several texts on heat transfer study [1, 2].

There are heat transfer data available for both horizontal and vertical tube evaporation, particularly by Badger and his associates. The literature on the performance of the inclined tube evaporator is very meager.

Van Marle [3] has published data on the inclined tube evaporator showing that the overall coefficient for distilled water boiling in a range of 138° to 143° F. increased from 835 B.t.u. per sq. ft. per hr. per °F. at a 50° temperature difference to 1140 B.t.u. per sq. ft. per hr. per °F. at an 81° temperature difference. Van Marle presented data on overall coefficients only and covered a limited range of temperatures of evaporation. The evaporator operating at a 45° angle of inclination and under vacuum had for its heating surface seven 3-inch o.d., 14-gauge copper tubes, 4 feet 10.5 inches long.

Linden and Montillon [4] worked with a small inclined tube evaporator which had one heating tube of 1-inch copper pipe (of about four feet length). Their evaporator was constructed so that any velocity of fluid circulation in a down-take pipe could be measured. They made runs with distilled water boiling at three different temperatures 180°, 195°, and 210° F., and over a range of temperature differences varying from 8° to 28° F. for each evaporation temperature.

They found that the coefficients on the liquid side increased with the temperature difference at any given temperature of evaporation as indicated in Figure 1. The same effect was observed in the case of the overall coefficients as shown in Figure 2.

Linden and Montillon also plotted the logarithmic mean velocity of circulation of liquid plus vapor and found that as the rate of circulation increased so did the film coefficients on the liquid side. This showed that the liquid coefficients bear a definite relation to the velocities of circulation. See Figure 3.

The use of the log mean for the calculation and plotting of the velocity was chosen arbitrarily and was admitted to have no particular significance.

By plotting the liquid film coefficient versus the liquid film temperature drop (Figure 1), a relationship was found to exist as shown by the following approximate equations:

$$\text{At } 180^{o}\text{ F., } h_1 = 0.632\Delta T_1^{2.5} \tag{6}$$

$$\text{At } 195^{o}\text{ F., } h_1 = 1.010\Delta T_1^{2.5} \tag{7}$$

$$\text{At } 210^{o}\text{ F., } h_1 = 1.564\Delta T_1^{2.5} \tag{8}$$

A relationship was also shown between the liquid film coefficients and the log mean velocities (Figure 3).

$$\text{At } 180^{o}\text{ F., } h_1 = 97\ V_m^{0.89} \tag{9}$$

$$\text{At } 195^{o}\text{ F., } h_1 = 120\ V_m^{0.89} \tag{10}$$

$$\text{At } 210^{o}\text{ F., } h_1 = 150\ V_m^{0.89} \tag{11}$$

By eleminating h_1 from these two sets of equations, a relationship was formulated between the temperature drop across the liquid film and the log mean velocity (V_m) of the liquid inside the heating tube.

$$\text{At } 180^{o}\text{ F., } \Delta T_1 = 7.49\ V_m^{0.356} \tag{12}$$

$$\text{At } 195^{o}\text{ F., } \Delta T_1 = 6.76\ V_m^{0.356} \tag{13}$$

$$\text{At } 210^{o}\text{ F., } \Delta T_1 = 6.20\ V_m^{0.356} \tag{14}$$

These equations were designated as empirical equations, correlating only the data given over the range of temperatures investigated. However, they

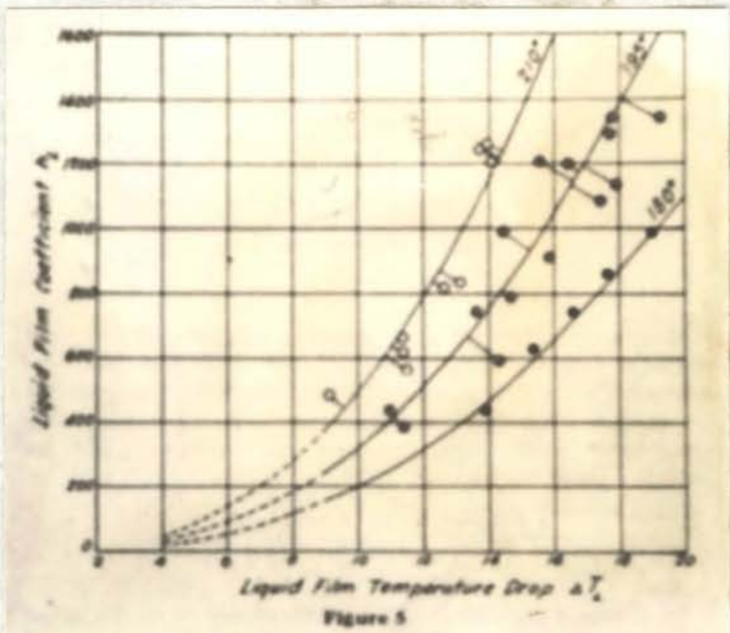

Liquid Film Coefficient h_L
Liquid Film Temperature Drop ΔT_L
210°
195°
180°
Figure 5

Fig. 1

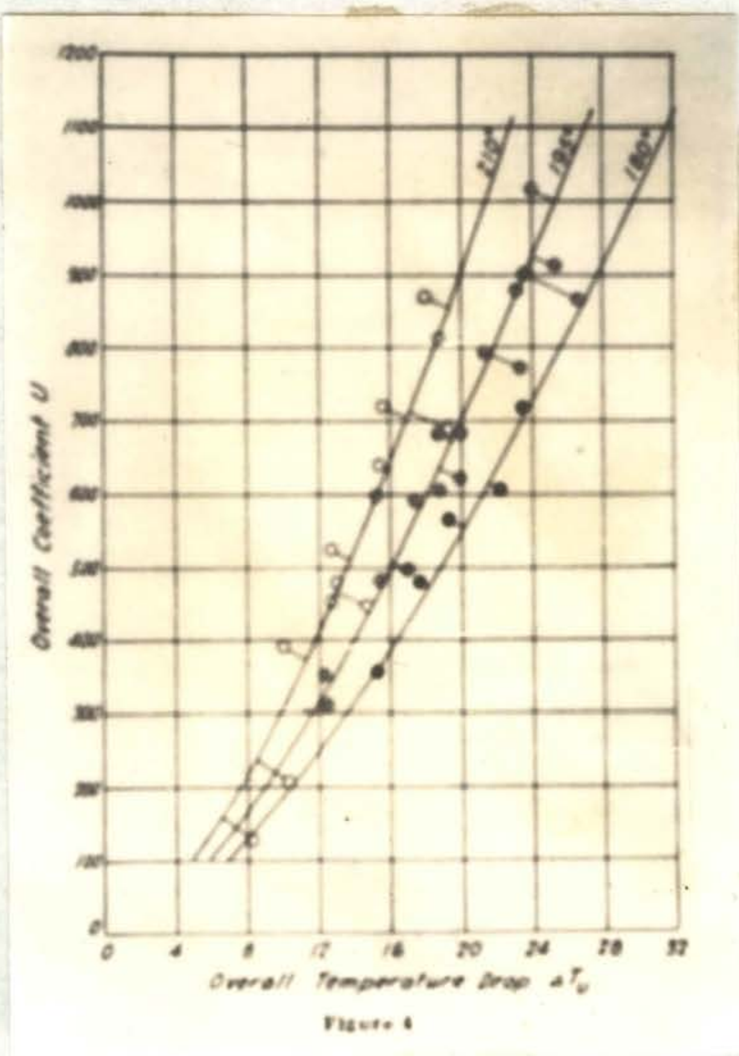

Overall Coefficient U
Overall Temperature Drop ΔT_u
Figure 4

Fig. 2

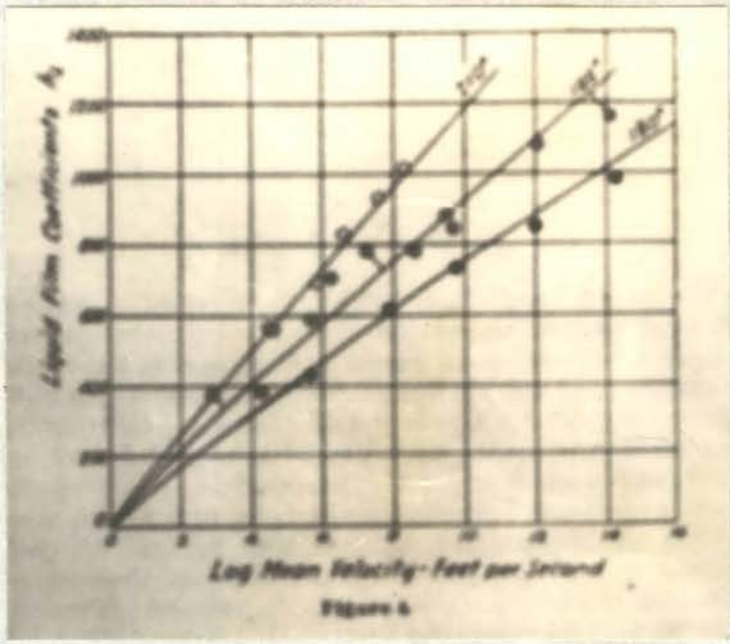

Liquid Film Coefficients
Log Mean Velocity- Feet per Second

Fig. 3

indicated to Linden and Montillon the possibility of defining h_1 in terms of the dimensionless groups employed for heat transfer coefficients as in the Morris and Whitman expression

$$\frac{hd}{k} = f_1\left(\frac{c\mu}{k}\right) f_2\left(\frac{du\rho}{\mu}\right) \qquad (15)$$

where
- c = Specific heat
- μ = Viscosity in centipoises
- k = Thermal conductivity
- d = Inside diameter of pipe in inches
- u = Velocity of liquid in feet per second
- ρ = Density of liquid in pounds per cubic foot

By calculating hd/k and $dV_m\rho/\mu$ and plotting these values on log log paper, straight lines resulted. See Figure 4. Using the reciprocal of the group $c\mu/k$ in equation (15), the following formula resulted:

$$\frac{hd}{k} = 4.15\left(\frac{k}{c\mu}\right)\left(\frac{dV_m\rho}{\mu}\right)^{0.8} \qquad (16)$$

It was found that this equation represented the conditions of operation with an average accuracy of $\pm$ 5%, where k, c, and μ were taken at the temperature of evaporation.

No claim was made that equation 16 established any definite precise relationship. It was significant, however, that their results agreed with it quite well indicating possibilities for its usefulness.

Some of the problems of evaporation with which Aubrecht (5) was concerned were:

1. To measure the rate of natural circulation in an inclined tube evaporator.
2. To correlate the effect of liquid natural circulation on heat transfer.
3. To determine the effect of temperature drop on heat transfer for natural circulation.
4. To find the effect of changing the properties of the liquid mainly the viscosity; for example by the use of sugar solutions as compared to water.

The apparatus used was that of Michaelsen (6) with various modifications. The evaporator was of the inclined type with a heating surface consisting of

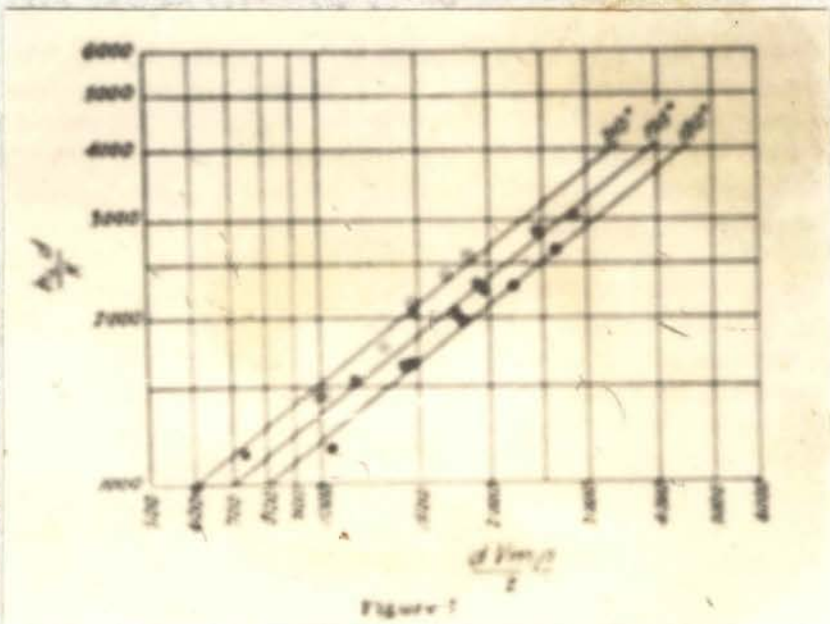


Fig. 4

a four foot copper tube, one inch inside diameter and operated at a 45° angle of inclination.

As stated by Aubrecht, the original method of correlating evaporation data was to plot the heat transfer coefficient h against the liquid film temperature drop, ΔT_{film}. This method has its disadvantages because there is difficulty in obtaining a satisfactory average value of ΔT_{film}. It has been shown by Jacob and Fritz (7) that in the case of nuclear boiling, the active spot on a flat plate at which boiling started, shifted as time passed. This condition is probably also true in a tube. Consequently, the boiling zone shifts and since the ΔT is different in the boiling and the nonboiling zones, considerable difficulty is experienced in obtaining average values. In this method of plotting, the effect of any error in ΔT is squared as the measured value of Q/θ is divided by ΔT and plotted against ΔT. So if the value of ΔT is low, the value of h will be high and if h is plotted against ΔT, the error is much greater.

There have been German investigators according to Aubrecht who correlated data by plotting h against Q/θ so that the ΔT is eliminated. This amounts to plotting $Q/\theta \times 1/\Delta T$ against Q/θ and masks some of the error of measurement.

The method of correlating data finally selected by Aubrecht was to plot the primary data of Q/θ and ΔT which would show the true error of the measurements.

Aubrecht found when Q/θ versus ΔT was plotted on log-log paper (Figure 5), that a break occurred at about the heat load corresponding to a maximum in the plot of liquid circulation versus heat load (Figure 6). From this it can be seen that the rate of natural circulation does have an effect upon heat transfer. The quantitative effect of liquid circulation was found by obtaining the value for the exponent n for G necessary to straighten out the curve of $\log Q/\theta \cdot 1/G^n$ versus $\log \Delta T$. A value of $n = 0.3$ was finally selected because it gave a minimum spread of points (see Figure 7).

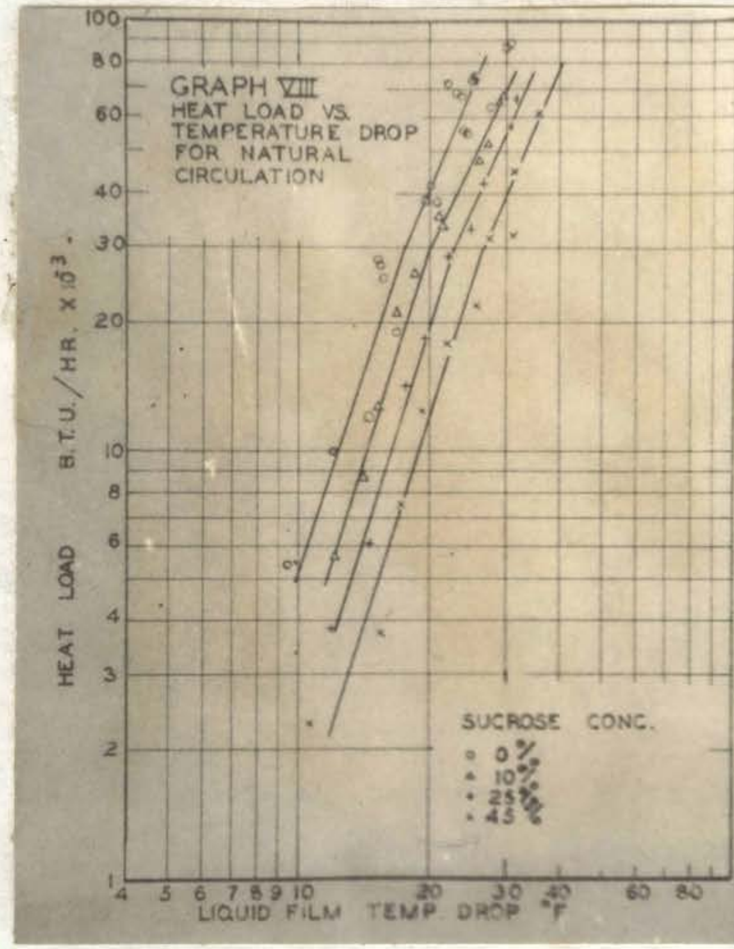

GRAPH VIII
HEAT LOAD VS.
TEMPERATURE DROP
FOR NATURAL
CIRCULATION
HEAT LOAD B.T.U./HR. X 10^3
LIQUID FILM TEMP. DROP °F
SUCROSE CONC.
0%
10%
25%
45%
100
80
60
40
30
20
10
8
6
4
3
2
1
4
5
6
7
8
9
10
20
30
40
60
80

Fig. 5

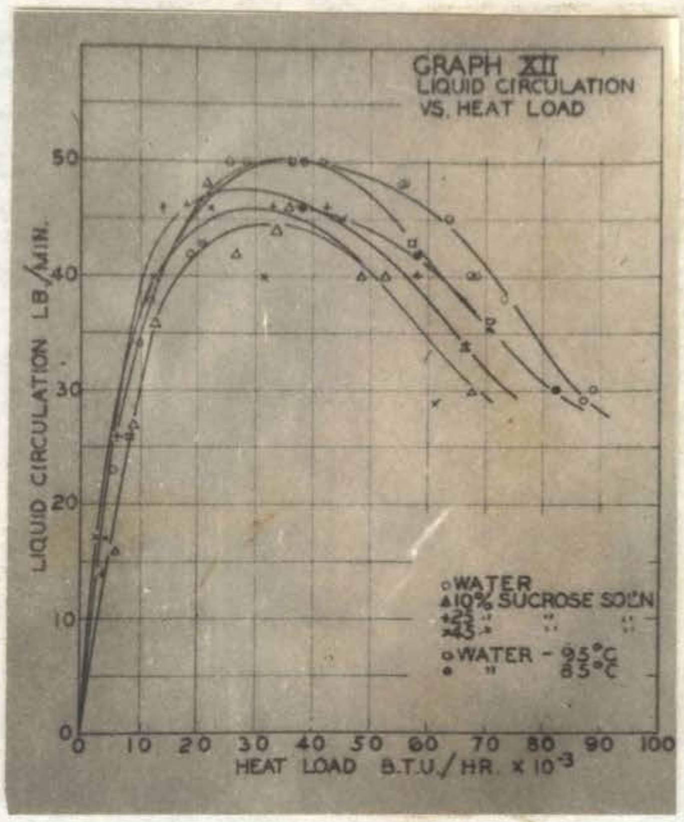

GRAPH XII
LIQUID CIRCULATION
VS. HEAT LOAD
LIQUID CIRCULATION LB./MIN.
HEAT LOAD B.T.U./HR. x 10^-3
○WATER
▲10% SUCROSE SOL'N
+25 " " "
×45 " " "
○WATER - 95°C
● " 85°C

Fig. 6

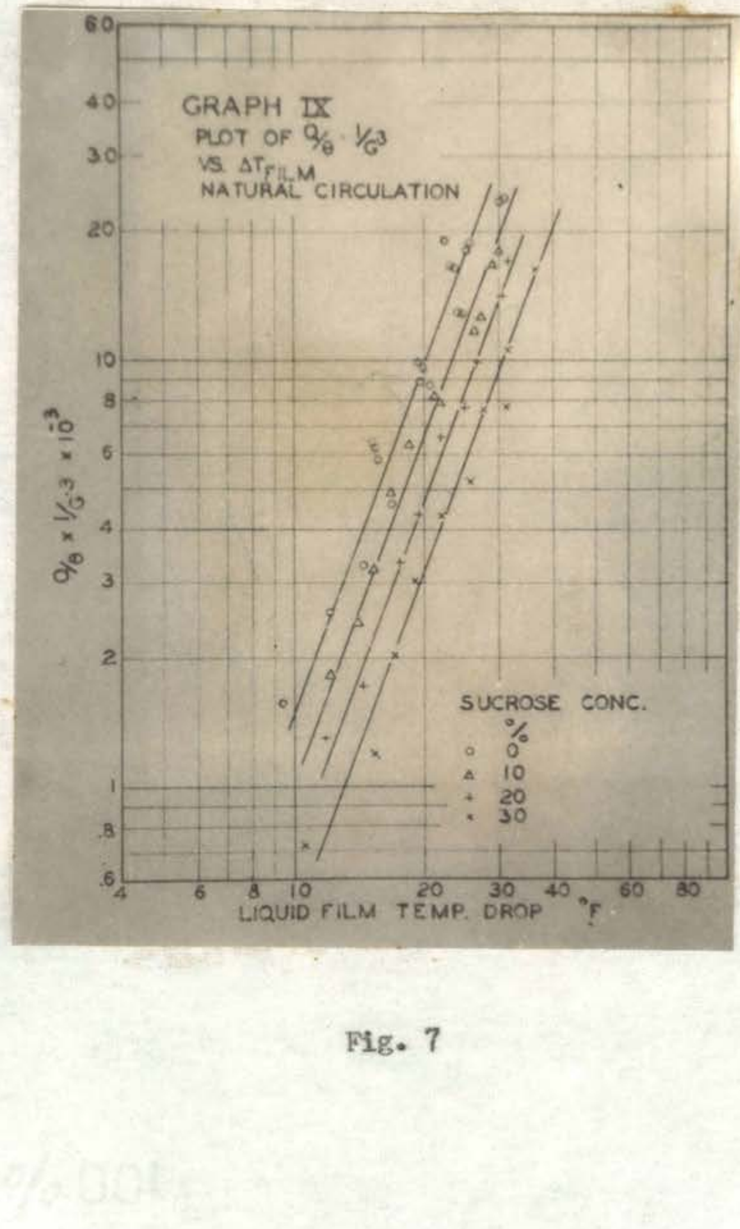

GRAPH IX
PLOT OF Q/θ · 1/G³
VS. ΔT FILM
NATURAL CIRCULATION
SUCROSE CONC.
%
0
10
20
30
Q/θ × 1/G³ × 10⁻³
LIQUID FILM TEMP. DROP °F

Fig. 7

Sugar solutions were used to obtain fluids of various viscosities, but at the same time the other properties of the liquid were also affected, namely, the thermal conductivity and the specific heat. Since each of these variables cannot be changed independently, a method of determining the relative effect of each was found through the use of the Dittus-Boelter equation which contained these factors. Accordingly, the effect of sugar solution was correlated to

$$\frac{k^{0.6}\,c^{0.4}}{\mu^{0.4}}$$

k = Thermal conductivity
c = Specific heat
μ = Viscosity

The plotting of $Q/\theta \times 1/G^n$ versus ΔT gave parallel lines, the parameter being the concentration of sucrose. The values of $Q/\theta \times 1/G^n$ at a given value of ΔT were designated as K; it was found that a correlation of K and $\log \frac{k^{.6}c^{.4}}{\mu^{.4}}$ was impossible, and that a plot of K versus $\frac{k^{.6}c^{.4}}{\mu^{.4}}$ did not result in a straight line. However, the plot of 1/K versus $\frac{k^{.6}c^{.4}}{\mu^{.4}}$ did give a straight line and from this plot the following correlation for natural circulation resulted:

$$1/K = P\left(1 - a\,\frac{k^{.6}c^{.4}}{\mu^{.4}}\right) \qquad (17)$$

P = Proportionalty constant

From data obtained by varying conditions in the evaporator a value of 0.79 was selected for a giving

$$1/K = P\left(1 - 0.79\frac{k^{.6}c^{.4}}{\mu^{.4}}\right) \qquad (18)$$

As shown in Figure 8, Aubrecht's natural circulation data were finally correlated by the following equation:

$$Q/\theta \times 1/G^{0.3}\left(1 - 0.79\,\frac{k^{.6}c^{.4}}{\mu^{.4}}\right) = 1.124\,\Delta T^{2.55}$$

with points being within ± 20% of the line.

From Figure 6, it can be seen that the rate of natural circulation shows a maximum at a heat load of 30,000 B.t.u./hr. According to Aubrecht, the

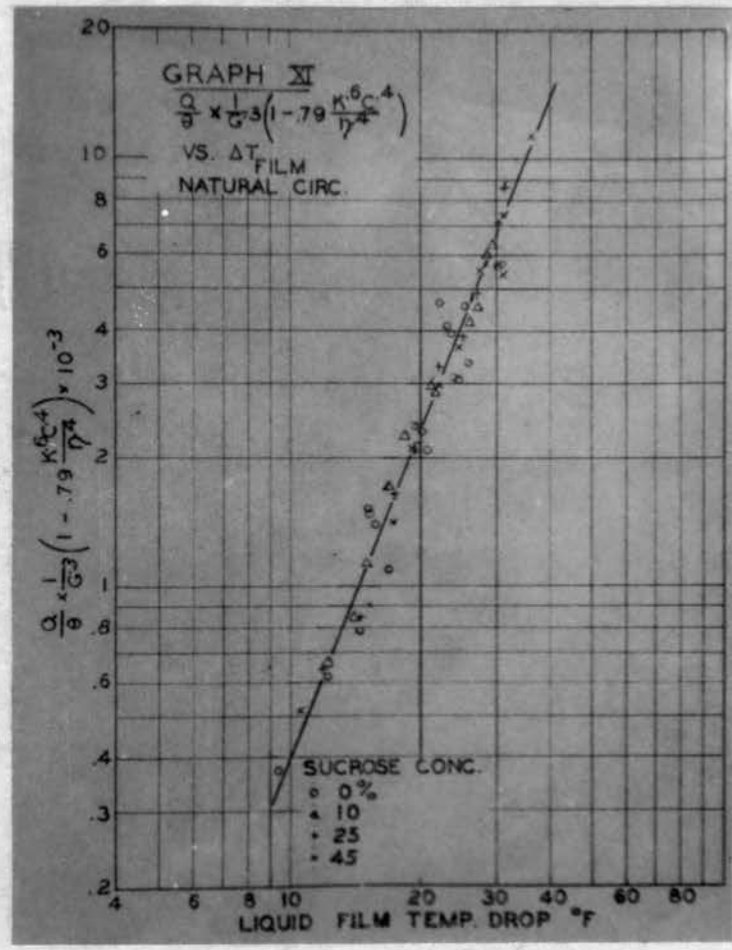

GRAPH XI
$\frac{Q}{\theta} \times \frac{1}{G^{.3}}\left(1-.79\frac{K^{.6}C^{.4}}{\eta^{.4}}\right)$
VS. ΔT_{FILM}
NATURAL CIRC.
SUCROSE CONC.
0%
10
25
45
LIQUID FILM TEMP. DROP °F
$\frac{Q}{\theta} \times \frac{1}{G^{.3}}\left(1-.79\frac{K^{.6}C^{.4}}{\eta^{.4}}\right) \times 10^{-3}$

Fig. 8

increase of velocity of circulation with increase of Q/θ is caused by the increased vapor lift effect, but as the heat load increases, the amount of vapor passing out the tube increases and causes an increase in the pressure drop through the tube. This increased pressure drop, consequently, causes the liquid circulation to decrease and the curve then shows a maximum.

The plot of log Q/θ versus logΔT for natural circulation shows a break at about the same heat load of 30,000 B.t.u./hr., where the liquid circulation shows a maximum.

Aubrecht, while working with natural circulation, also compiled data on forced circulation. A comparison of the two sets of data showed that the effect of ΔT_{film} is much greater in natural circulation than in forced circulation, the heat load Q/θ being proportional to ΔT to the 2.55 power for natural circulation and to the 1.55 power for forced circulation.

The effect of a solute as sugar on the heat transfer was found to be expressed by the following function:

$$Q/\theta \quad \frac{1}{1 - .79\,\frac{k^{.6}c^{.4}}{\mu^{.4}}}$$

The effect of lower concentrations of sugar was found to be much greater than that for higher concentrations; that is, an increase of concentration at low concentrations would cause a much greater decrease in Q/θ or h than a proportionate increase of concentration at a high concentration. It was found that the effect of the solute was due mainly to the change of viscosity because it was changed by a factor of three while the thermal conductivity and specific heat changed only by 25%.

Obert[8], working on an evaporator similar to that of Aubrecht, studied the effect of scale formation on the heat transfer coefficient. His data were obtained for only one angle of inclination (45°) and one temperature difference. They indicate the equation of Robinson and McCabe[2] to be applicable to the range of scaling in his particular evaporator.

Michaelson[6], working on the same evaporator as did Linden, studied natural circulation but did not measure the velocity. He calculated the liquid film coefficients for water under different pressures and temperature drops, but gave no correlation of the data.

Cleve[10] worked with a very small inclined single-tube electrically heated evaporator and measured the liquid film coefficients and rates of circulation for inclinations of 45°, 20° and 10° to the horizontal. For each angle of inclination he found that with increases in the rate of heat input the velocity of natural circulation increased to a maximum and then decreased. A plot of these results is shown in Figure 9. Cleve did not explain the reason for the overlapping of the curves.

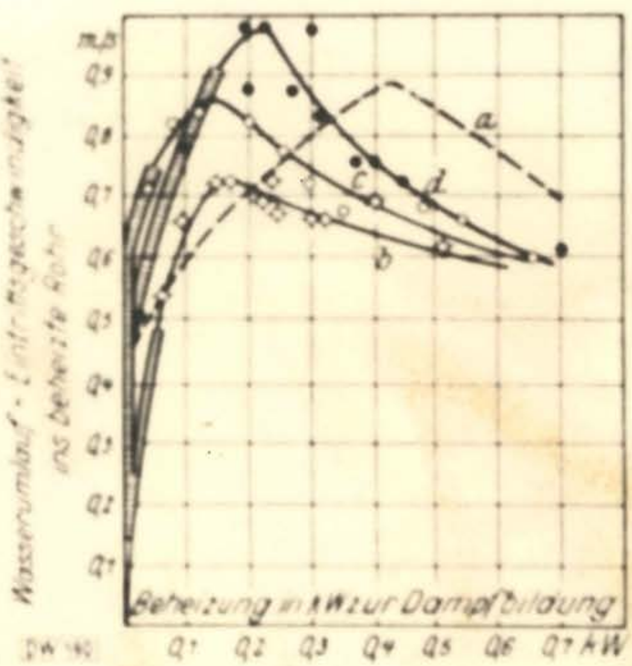


Abb. 11. Kleine Versuchsanlage. Einfluß der Neigung des Heizrohres auf den Wasserumlauf (vgl. Abb. 4 und 8)
Abstand der Mitte der beheizten Rohrlänge vom Wasserspiegel: 0,45 m
Heizrohr von 10,66 mm l. W.

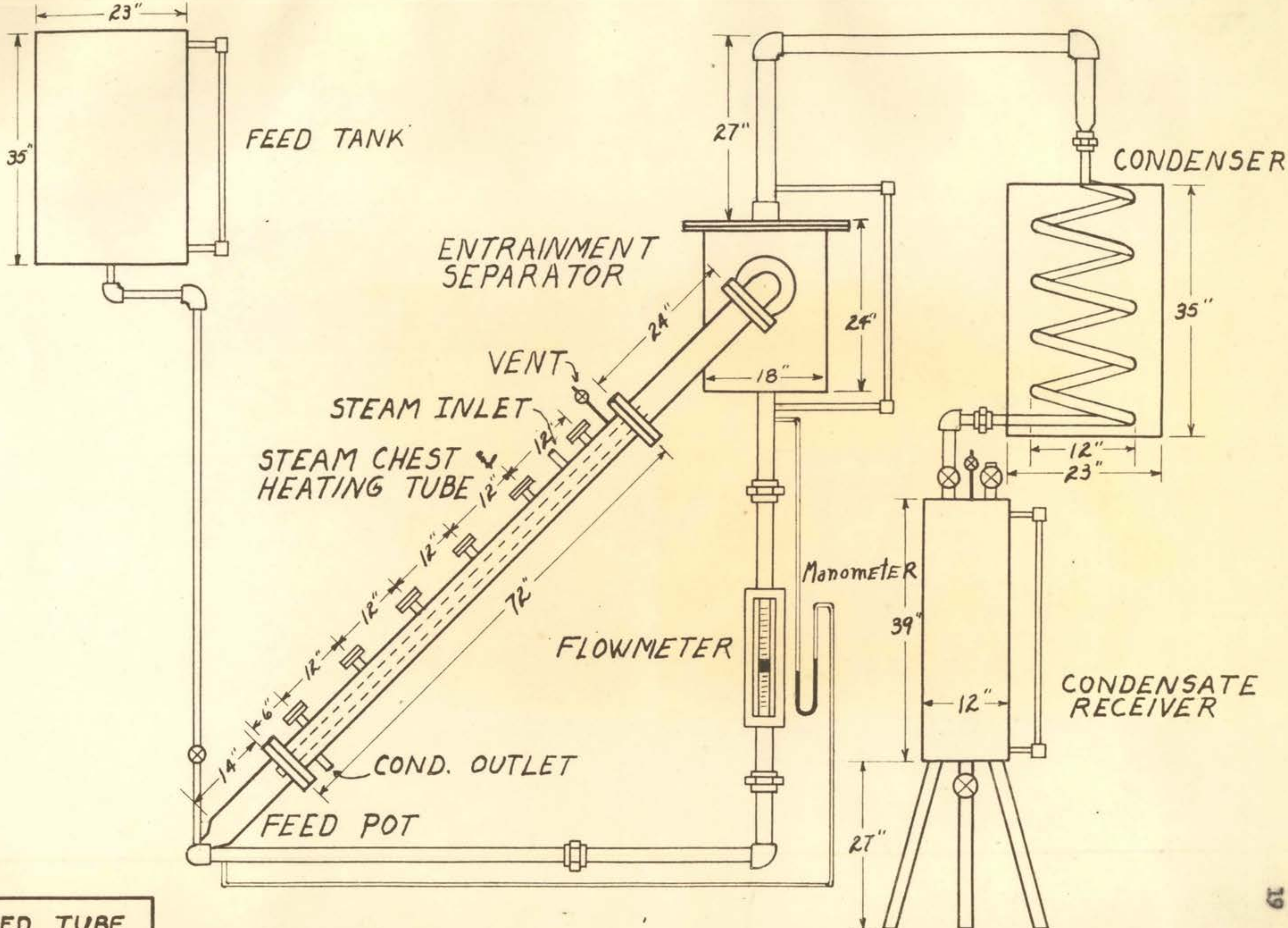
a senkrecht, *b* 45°, *c* 20°, *d* 10° gegen die Wagerechte geneigt

Fig. 9

Description of Apparatus

The apparatus which was used for this investigation was that of Aubrecht [5] with certain modifications in the entrainment separator and means for the measurement of the liquid velocity. The evaporator contained a single tube, steam heated and was so constructed that the evaporating unit could be inclined at any angle between a vertical and horizontal position (see Figure 10).

The selection of a suitable meter for measuring the liquid velocity presented quite a problem. The use of the orifice and the pitot tube were inadequate as shown by Aubrecht. The reason for the rejection of the orifice was that it would cause a considerable pressure drop and thus materially reduce the flow of natural circulation. The pitot tube was not used because of the fact that slight changes in piping gave different calibrations and there was drifting in the manometer readings due to air and water vapor in the circulating liquid. The spring flowmeter was not used because of the surging that took place during operation. Two Fischer-Porter flowmeters were selected for measuring the liquid natural circulation velocity (see Figure 10a). One of the meters was used for measuring fluid flow over the range of 0.1 to 2.4 G.P.M. It was the non-guided plumb-bob float type of meter No. J7-2177. The second meter was used for measuring larger rates of flow and was graduated in millimeters. It was the center guided stablvis float type No. B35625. By calibration of this scale and the plotting of the calibration curve (Fig. 19) readings for liquid velocity were readily obtained in terms of G.P.M. The meters were connected in parallel by a standard one-inch pipe to the entrainment separator and to the feed pot. One-inch gate valves were conveniently located in the return line in order to shut off the flow to the meter that was not needed during a run. Located at the outlet of the entrainment separator and at the inlet of the feed pot were manometer connections, for a mercury manometer, connections being made with rubber tubing. The manometer was installed in order to measure the friction loss through the liquid return line and the



INCLINED TUBE
EVAPORATOR
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Fig. 10

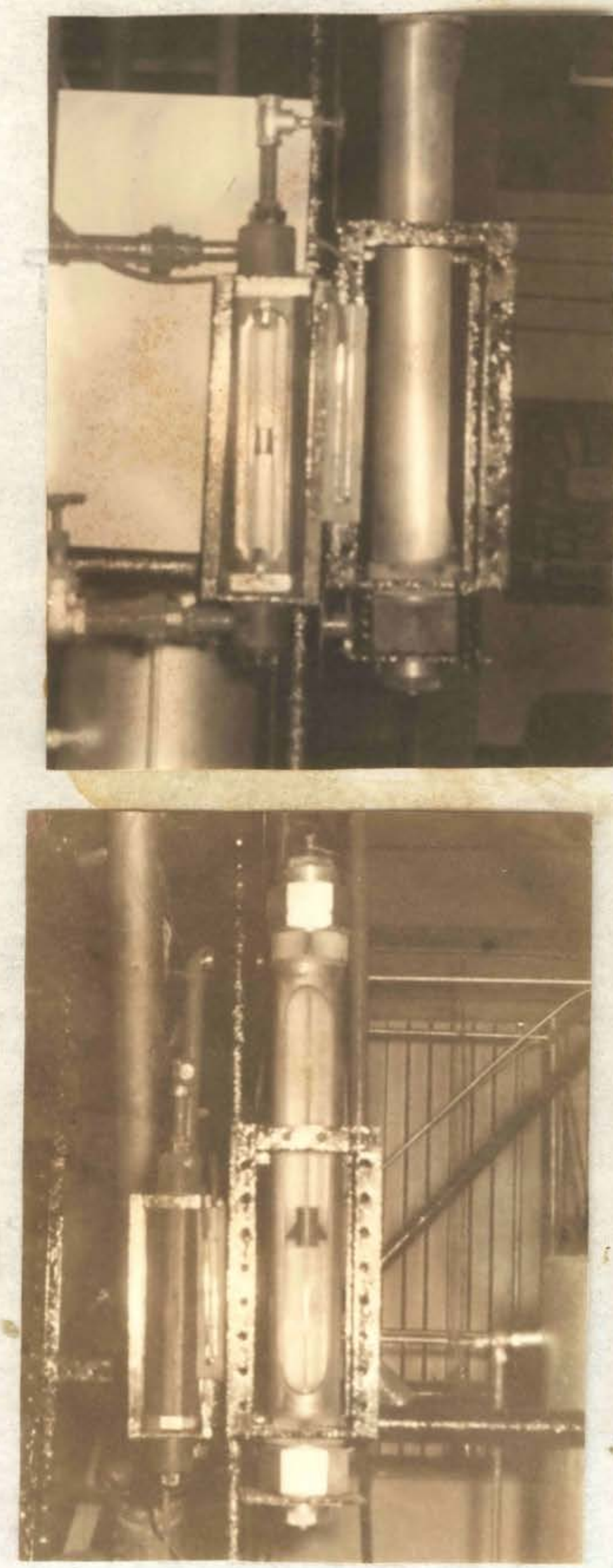


Fig. 10a

flow meters.

The heating element of the evaporator consisted of a standard one-inch, Type 316 stainless steel pipe 74 inches long. The effective heating length was 69.6 inches. The steam chest enclosing the element was a standard four-inch iron pipe with flanged ends. Steam was admitted near the top of the chest and was removed at the bottom by an Anderson #21 steam trap. Also at the bottom of the steam chest was a one-fourth inch outlet for the continuous venting of non-condensable gases.

The heating unit was fastened to the two ends of the steam chest by two stuffing glands as described by Obert[8].

Six one-inch couplings were welded into the steam chest. These served as outlets for thermocouple leads which were attached to the outside surface of the heating tube. The first thermocouple outlet was six inches from the end of the steam chest and the others were spaced one foot apart as shown in Figure 10. The coupling outlets were fitted with one-inch flanges composed of a standard screw flange and a blind companion flange. Two soft rubber gaskets were inserted between these flanges and the thermocouple leads were brought out between the two gaskets as shown in Figure 11.

Attached to the bottom of the steam chest by flanges was a fourteen-inch section of four-inch pipe which served as a feed pot for the evaporating unit. The feed pot was connected by a one-half inch line to the feed reservoir and by a one-inch recirculation line to the entrainment separator.

The disengaging section was a twenty-four inch section of four-inch pipe attached to the upper end of the steam chest. Its purpose was to separate the heating tube from the entrainment separator.

Located in the liquid return line (recirculation) were the fluid meters used for measuring the rate of natural circulation in the evaporator.

The feed reservoir was a fifty-gallon steel drum with a central outlet located in the bottom. It was equipped with a sight glass which indicated the

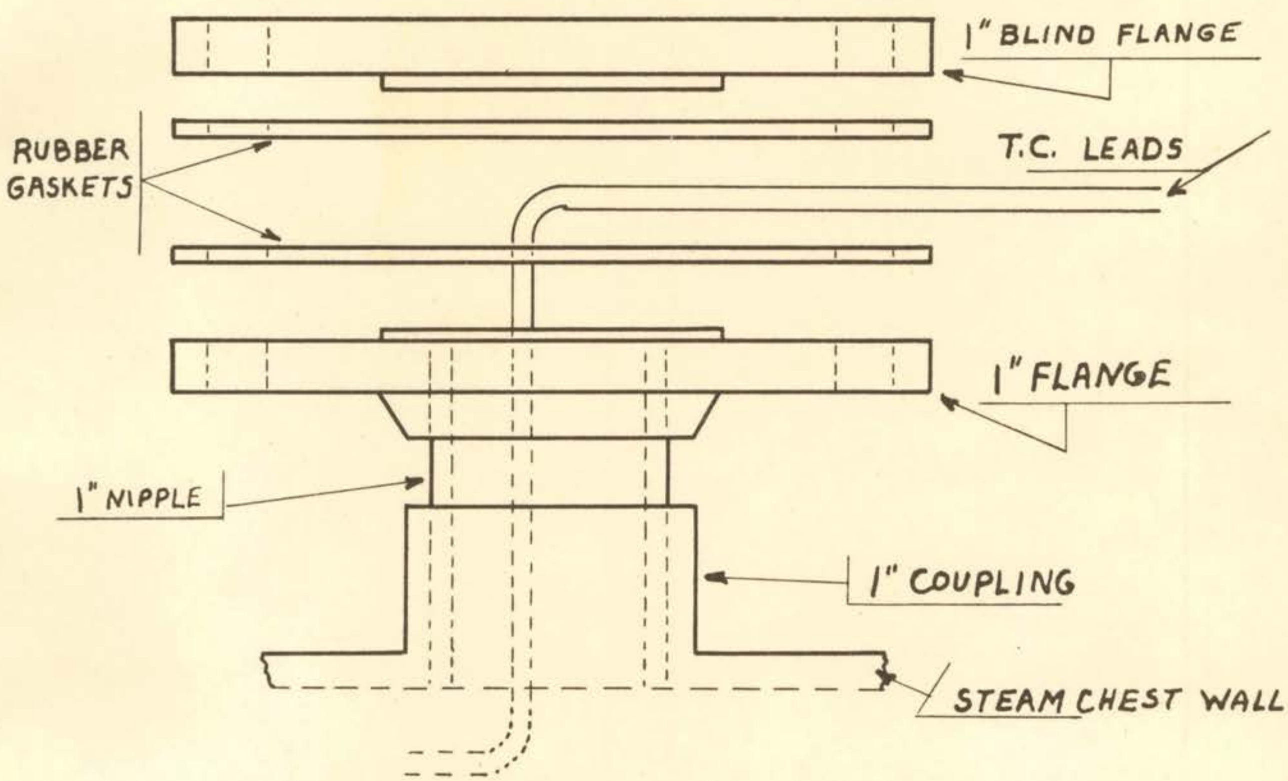


Fig. 11

THERMOCOUPLE OUTLET

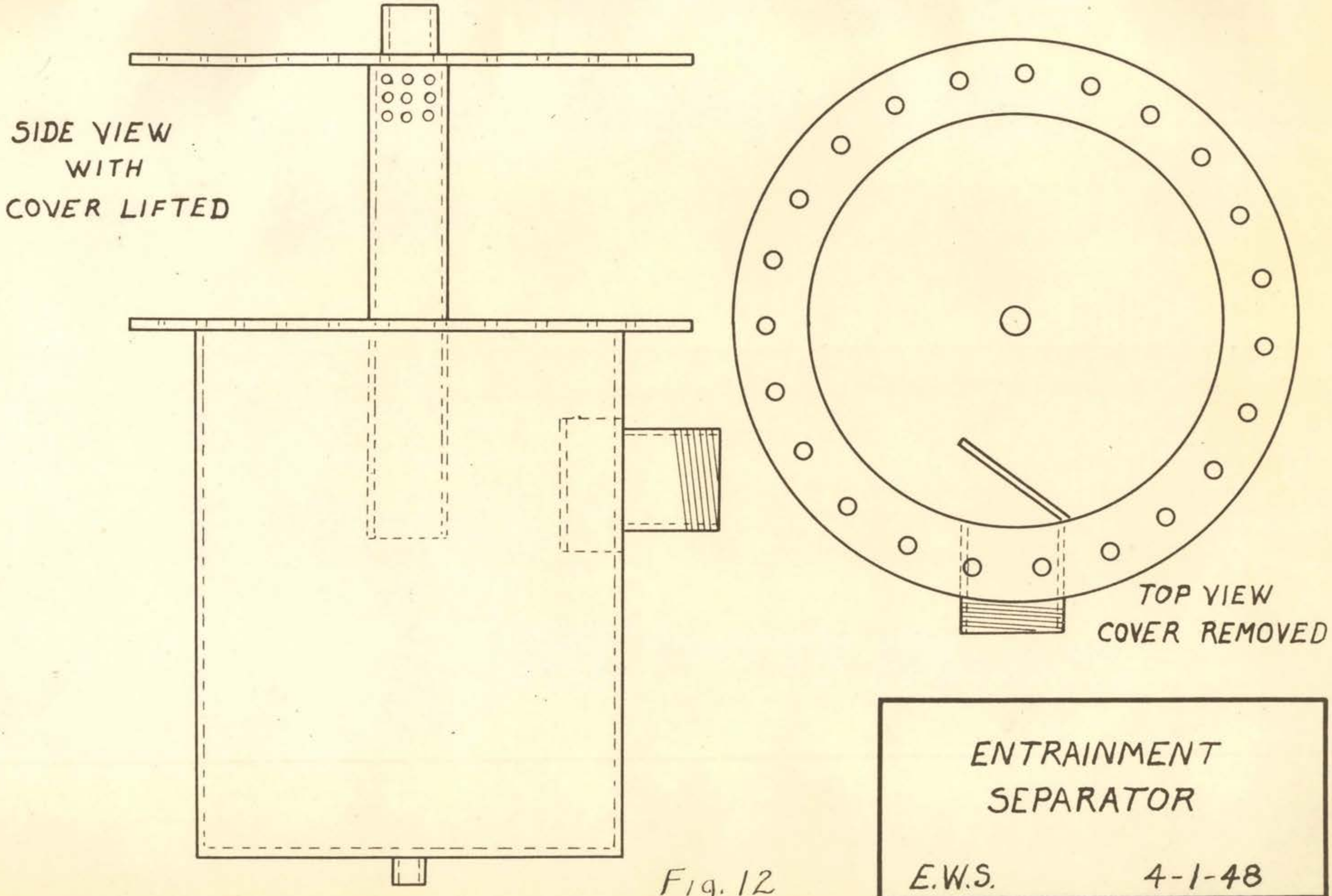
E.W.S. 4-1-48

liquid level of the feed at all times. The feed to the system was admitted by a one-half inch gate valve installed in a one half-inch line which connected the feed reservoir to the feed pot.

The entrainment separator was of the cyclone type. The body consisted of a twenty-four inch section of standard eighteen-inch iron pipe. The bottom was a three-eights inch steel plate and contained a one-inch outlet coupling centrally located for the return of liquid to the feed pot. The top of the separator was a flat-flanged cover. A standard two-inch coupling was centrally located in the flanged plate. This served as an outlet for the vapors. The vapor inlet was a standard four-inch nipple welded into the side of the SEPARATOR evaporator five inches below the top. A six-inch square baffle was located inside the separator in front of the vapor inlet to deflect the incoming vapor and cause it to follow a circular path. See Figure 12. There were 12 one-half inch holes in a four-inch vertical pipe welded centrally into the bottom of the cover plate. The vapor from the entrainment separator left through these holes. The separator was connected to a ninety degree, four-inch flanged elbow by a threaded four-inch nipple welded to the side of the separator to the disengaging section of the evaporator. This threaded connection permitted the angle of inclination of the evaporator tube to be varied at will.

The condenser was made of ten turns of one-inch copper tubing in a vertical coil twelve inches in diameter. It was encased in a fifty gallon steel drum. Cooling water was introduced at the bottom of the steel drum and flowed out of a two-inch overflow line located near the top of the drum.

The condensate receiver was a twelve inch by thirty-nine inch cylindrical tank. The top was provided with a standard one-inch connection and a standard one-fourth inch pipe connection. The one-inch connection was used for receiving the condensate from the condenser. The other connection served as a vent. On the side of the receiver was a gauge glass for liquid level indication. On


SIDE VIEW
WITH
COVER LIFTED
TOP VIEW
COVER REMOVED
ENTRAINMENT
SEPARATOR
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4-1-48

Fig. 12

the bottom was a one-half inch pipe for the removal of condensate.

All temperatures except those of the feed were measured by means of thermocouples. The feed temperature was measured with a thermometer. The thermocouples were made of #24 gauge iron and constantan wire having a woven glass-varnish insulation. The E.M.F. of the thermocouple was measured by a Leeds and Northrup Type K potentiometer. This instrument is capable of reading hundredths of a milli-volt directly. The third place can be closely approximated. This permitted a direct temperature reading to the nearest 0.3° F. and an estimation to the nearest 0.03° F. The heating tube was provided with six thermocouple junctions which gave the outside temperature of the tube. The junctions were spaced twelve inches apart with the two end thermocouples six inches from the packing glands.

The thermocouples were installed according to the method described by Colburn and Hougen [10] and McAdams [1]. See Figure 13. Slots approximately three thirty-seconds of an inch were cut around the outside perimeter of the heating tube. The thermocouple junctions were placed in these slots with the fused junctions just level with the tube surface. The temperature read was estimated to be that at a point 1/32" below the outside surface of the heating tube. After the junctions were soldered in place, the insulated leads were brought away from the junction through the slot to the opposite side of the pipe. The slot was then filled with solder and the excess polished off flush with the surface of the tube. After placing the heating tube in the steam chest, the thermocouple leads were brought out of the chest through the six one-inch flanges as shown in Figure 11. Two more thermocouple junctions were installed to obtain the inlet and outlet temperatures of the liquid in the heating tube. These junctions were inserted in glass tubes filled with oil and placed as close as possible to the inlet and outlet of the heating tube. All thermocouple leads were brought to an instrument table where a common cold junction (32° F.) was provided. A diagram of the potentiometer circuit is

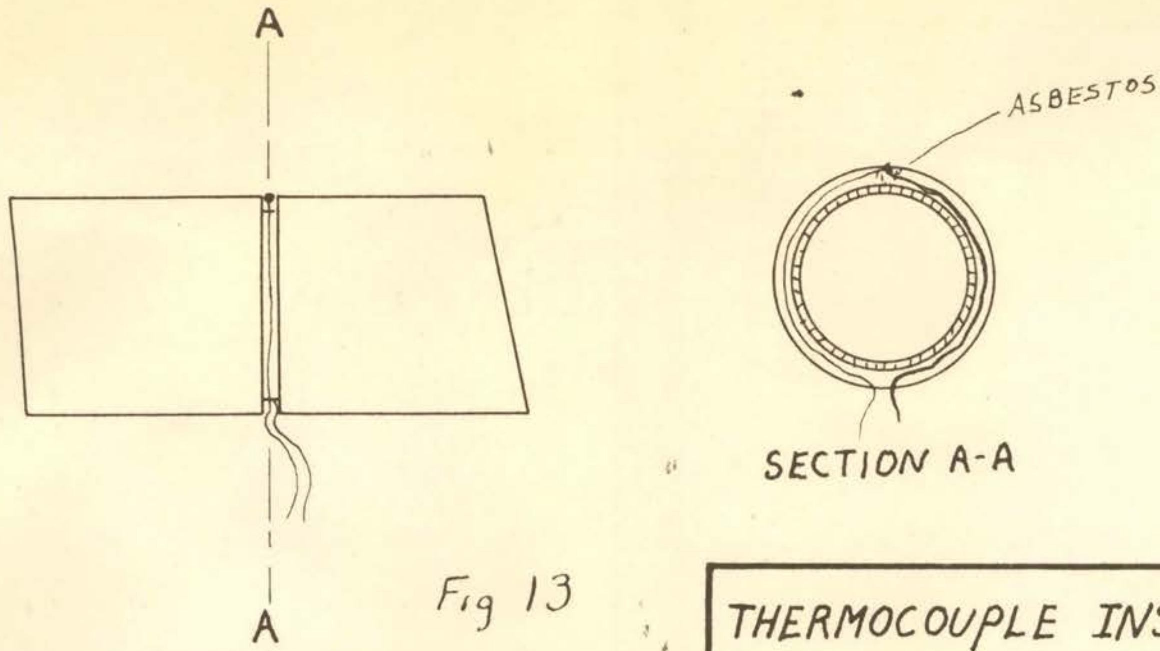


Fig 13

THERMOCOUPLE INSTALLATION

E.W.S. 4-1-48

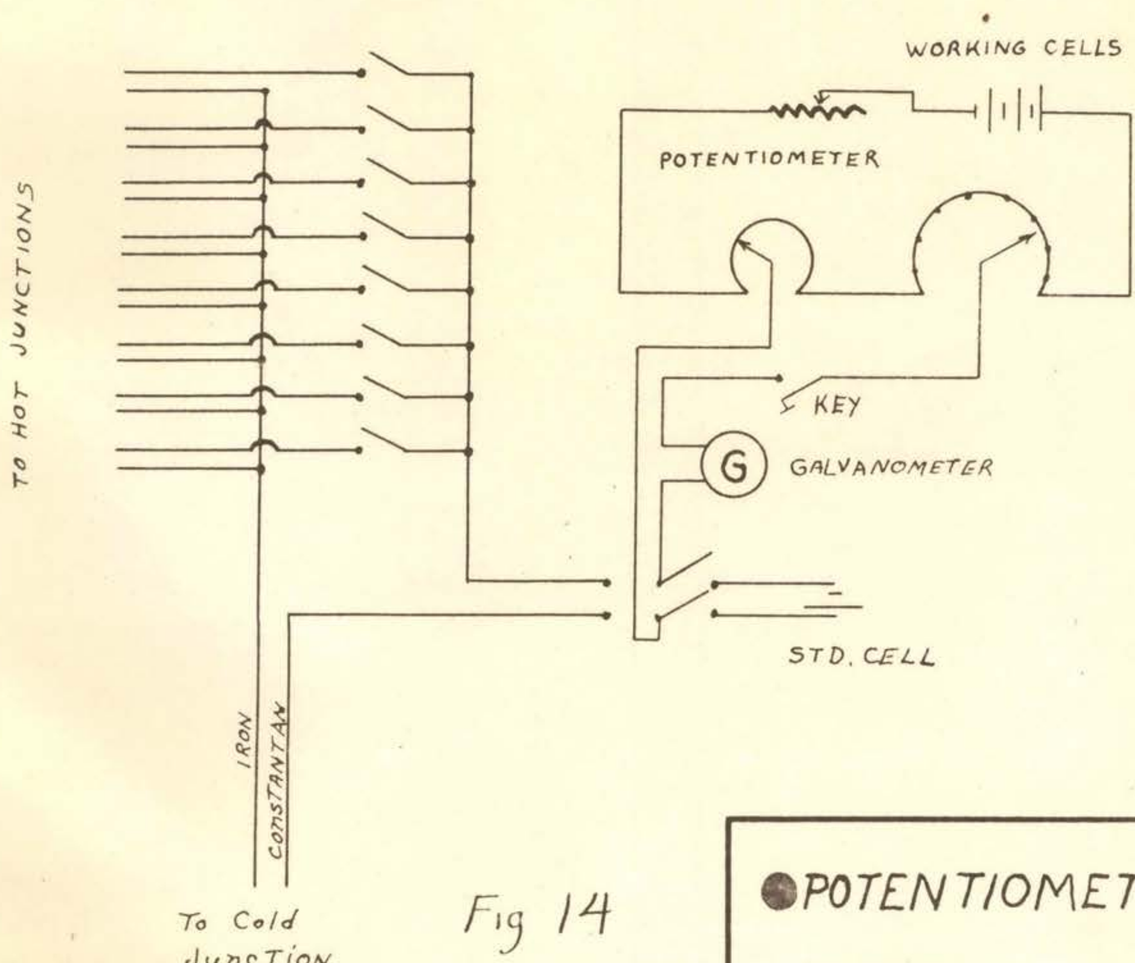


Fig 14

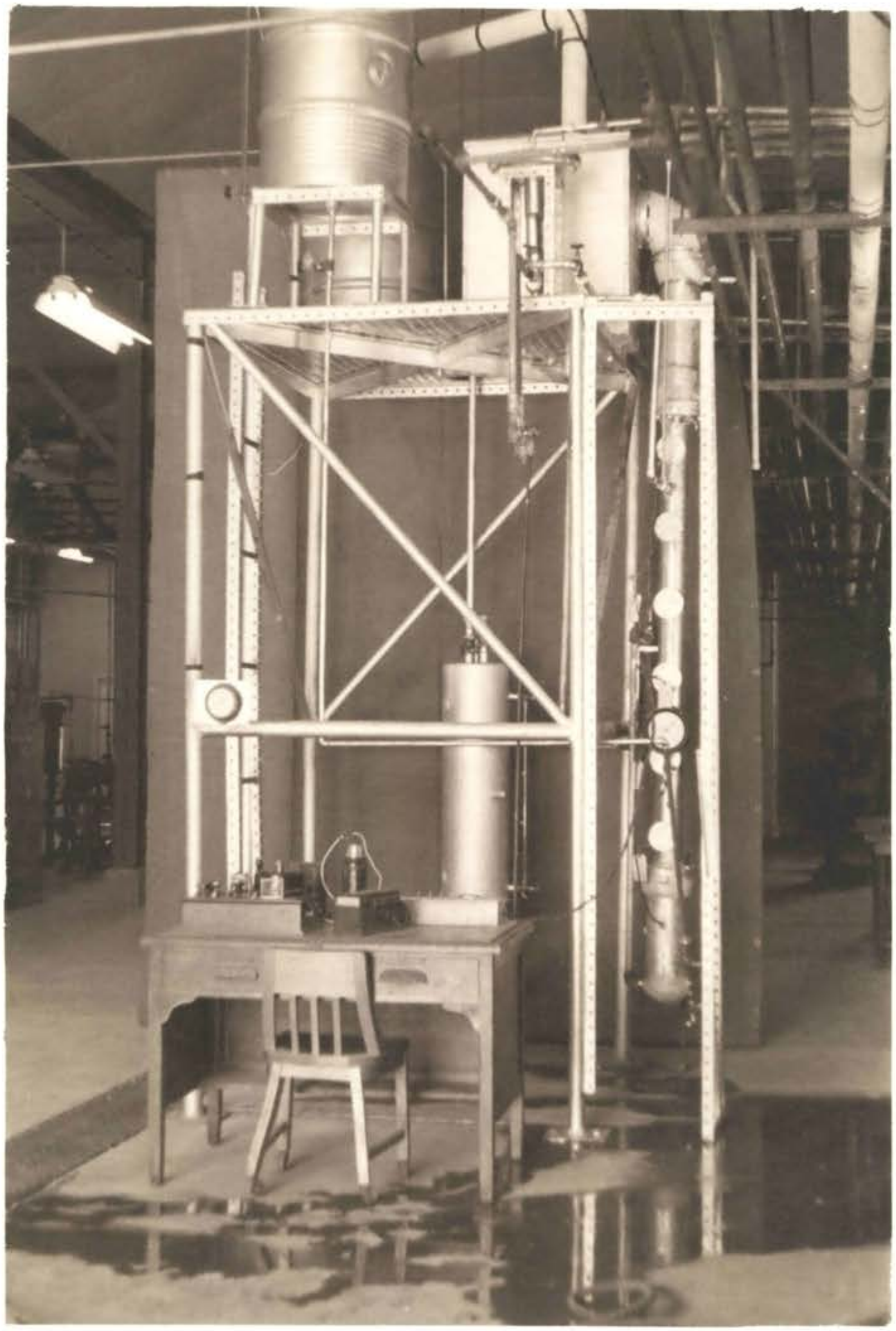
POTENTIOMETER CIRCUIT

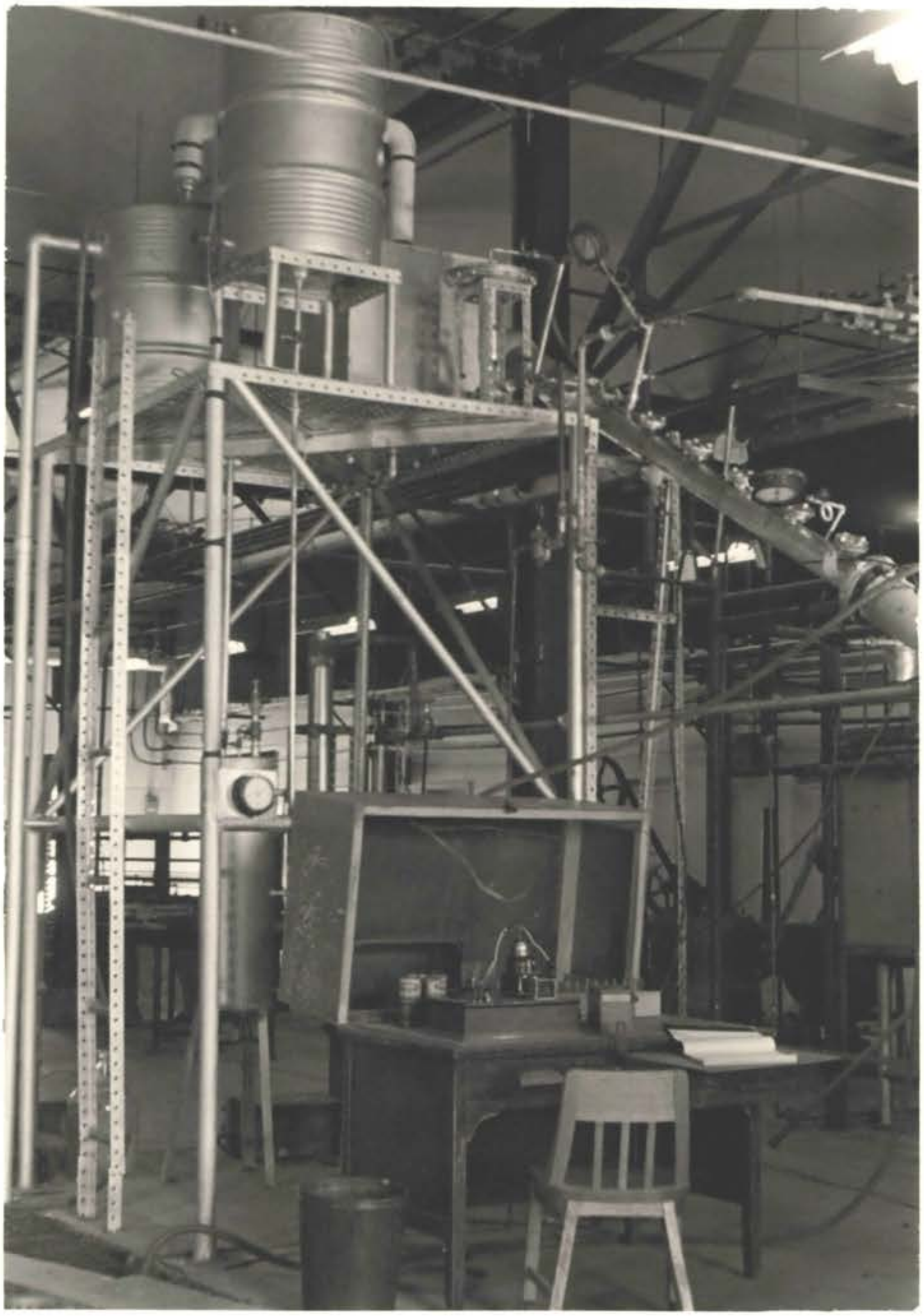
E.W.S. 4-1-48

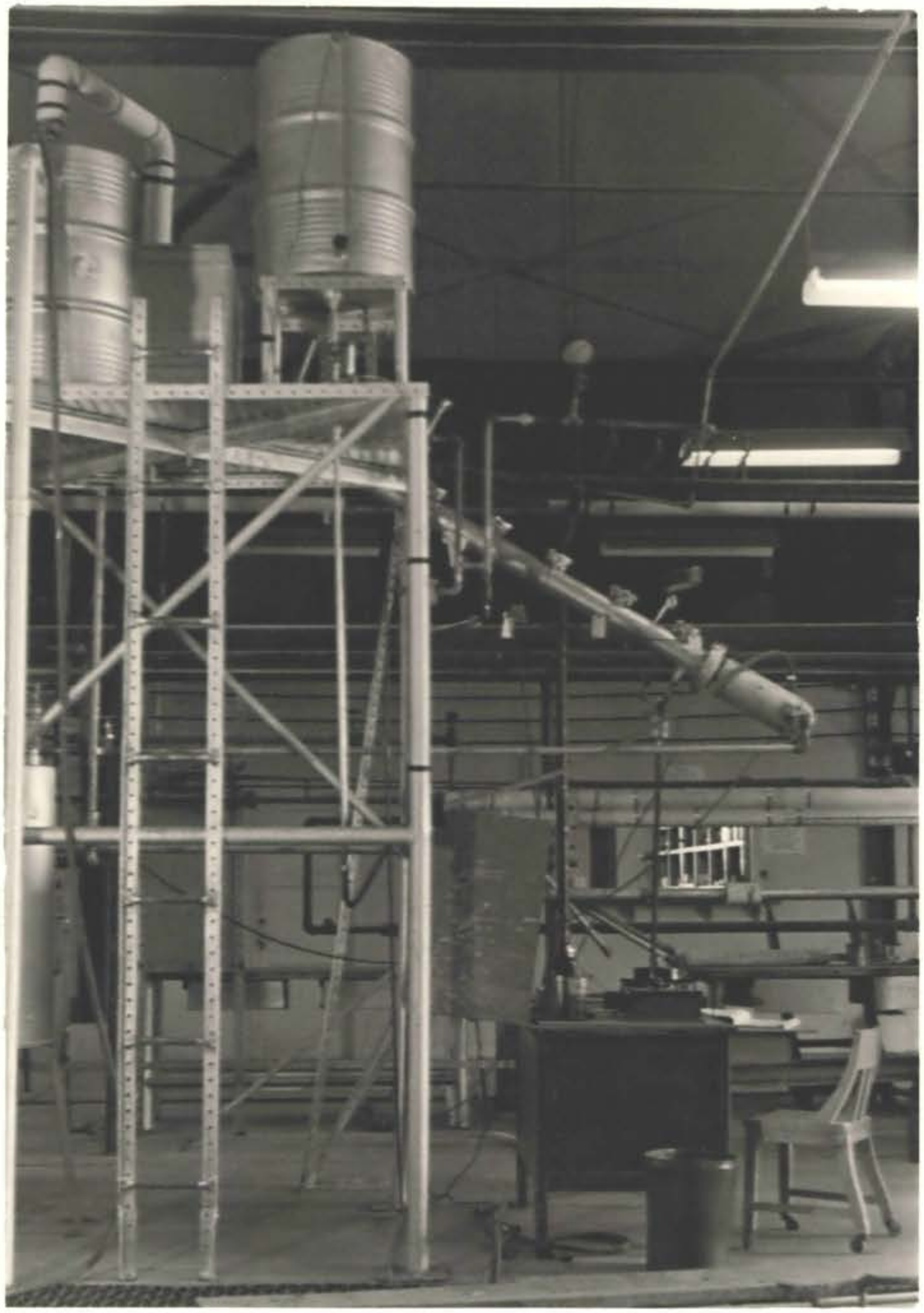
shown in Figure 14.

The evaporator was not completely insulated. Only the parts from which heat loss could not be tolerated were insulated, viz., the feed pot, the disengaging section, the ninety-degree elbow, the entrainment separator, and the line connecting the entrainment separator to the condenser. The entrainment separator was insulated by a wooden box packed with Eagle-Pitcher "Super 66" asbestos-felt insulation. The remaining parts were insulated with Johns-Mansville 85% magnesia pipe insulation.

Distilled water was used for all runs. It was fed by gravity from the feed reservoir. It had originally been planned to maintain a liquid level in the evaporator about two inches above the outlet of the evaporating tube and have the liquid return line come out of the bottom of the disengaging section. Surging in the evaporator was so extreme that it was decided to place the return line in the bottom of the entrainment separator and to keep the evaporator filled maintaining liquid level in the entrainment separator about two inches above the bottom of the vapor inlet. In this way, surging in the return line was greatly reduced.







Procedure

The small flowmeter read directly in G.P.M. and required no calibration except for a specific gravity correction. The correction factor curve is presented in the appendix.

The larger flowmeter was calibrated by weighing the amount of water passing through it per unit time at various float levels. The temperatures during calibration varied between 61° and 66° F. The calibration data were corrected to 212° F. by means of the correction curve previously mentioned. The corrected calibration curve is in the appendix (Figure 19).

All thermocouples were checked by comparing them against a Bureau of Standards certified thermometer and were found to be accurate to about 0.3° F. The thermometer used for obtaining the liquid temperature out of the feed reservoir was found to be accurate to about 1.0° F. Better accuracy was not necessary for any of the calculations.

Before any tests were made, the apparatus was thoroughly cleaned by boiling out with a weak Na_2CO_3 solution, followed by a water wash, and then with a 5.0% HCl solution. Rust particles from the connecting piping were removed prior to a run by means of a strainer in the return line. The strainer was removed during a run.

The distilled water used was tested for iron before and after all runs. A sample was checked for clarity and for the appearance of color when treated with potassium thiocyanate, potassium ferricyanide and potassium ferrocyanide. All tests were negative.

The actual procedure in making a run was as follows:

Potentiometer batteries were connected into the circuit and allowed to come to a fairly steady voltage. The cold junction was then prepared (crushed ice in a thermos bottle). The apparatus was filled with water to a height of about 2-2½ inches above the vapor inlet to the entrainment separator. No feed was introduced after the run was started.

The steam was turned on at the beginning of a run and adjusted to pressure. Steady state conditions were considered to have been reached after the water in the system had been brought up to the boiling point, after a constant reading on the flowmeter was obtained and after 5 to 10 pounds of water had been evaporated. A run was started by collecting the vapor condensate. Each run was of one hour duration during which time readings of the thermocouples, the flowmeter and the manometer were taken. After the run was completed, the vapor condensate was weighed to the nearest 0.1 pound.

During all runs, considerable difficulty was encountered in controlling the steam pressure in the steam chest, even with a pressure controller in the line. The pressure in the steam main was found at times to fluctuate as much as 20 pounds. Partial manual control was ultimately resorted to.

The angles of inclination of the evaporating tube considered were 90°, 75°, 60°, 45°, 30° and 15°.

General Nomenclature

Q = B.t.u. transmitted

θ = time in hours

λ = latent heat of steam in B.t.u. per pound

l = heat of liquid--(temperature of boiling - temperature of inlet water) x 1 x pounds water evaporated per hour

λ_1 = λ x pounds water evaporated per hour

Q/θ = total heat transmitted per hour-- $\lambda_1 + l$

ΔT_1 = temperature drop through pipe and liquid film, °F.

ΔT_L = liquid film temperature drop, °F.

ΔT_p = tube temperature drop, °F.

u = natural circulation velocity--G.P.M.

ψ = angle of inclination with horizontal

Tube dimensions:

I.D. = 1.049"

O.D. = 1.315"

Wall thickness = 0.133"

Effective length = 69.5"

A_1 = inside area of tube

A_2 = outside area of tube at point where temperature is read, thermocouple buried in pipe to a depth of 1/32"

A_3 = cross-sectional area of tube

A_4 = log mean average area of tube wall

h_L = liquid film coefficient

$$\Delta T_p = \frac{Q/\theta \text{ x thickness of tube}}{k \text{ x } A_4}$$

k = 7.4--thermal conductivity of metal wall[11]

ΔT_L = (tube temperature - boiling temperature) $-\Delta T_p$

v = natural circulation velocity in ft./sec.

cu.ft./gal. = 0.1337

ΔP_m = pressure drop through liquid return line and flowmeter, mm Hg

Sample Calculations

Observed data (Run No. 6)

$$A_1 = \frac{1.049\ \pi}{12} \times \frac{69.5}{12} = 1.59 \text{ sq. ft.}$$

$$A_2 = \frac{(1.315 - 1/16)\ \pi}{12} \times \frac{69.5}{12} = 1.90 \text{ sq. ft.}$$

$$A_3 = \frac{(1.049)^2\ \pi}{4 \times 144} = 0.006 \text{ sq. ft.}$$

$$A_4 = \frac{1.90 - 1.59}{2.3 \log \frac{1.90}{1.59}} = 1.73 \text{ sq. ft.}$$

$$h_L = \frac{Q/\theta}{\Delta T_L \times A_1}$$

$$\Delta T_p = \frac{Q/\theta \times 0.0085}{7.4 \times 1.73} = 0.000664\ Q/\theta$$

$$\text{Thickness of path} = \frac{.133 - 1/32}{12} = 0.0085 \text{ ft.}$$

$$v = \frac{u \times .1337}{60 \times A_3}$$

$$= 0.371\ u \text{ ft./sec.}$$

Tube temperature = 263.5°F.

Boiling temperature = 212.5°F.

Inlet water temperature = 210.0°F.

Water evaporated per hour = 42.6 pounds

Meter reading = 5.98 G.P.M.

$$1 = (212.5 - 210.0) \times 1 \times 42.6 = 106.5 \text{ B.t.u./hr.}$$

$$\lambda_1 = 970.0 \times 42.6 = 41{,}322 \text{ B.t.u./hr.}$$

$$Q/\theta = 41{,}322 + 106.5 = 41{,}429 \text{ B.t.u./hr.}$$

$$\Delta T_1 = 263.5 - 212.5 = 51.0^\circ\text{F.}$$

$$\Delta T_p = 0.000664 \times 41{,}429 = 27.5^\circ\text{F.}$$

$$\Delta T_L = 51.0 - 27.5 = 23.5^\circ\text{F.}$$

$$h_L = \frac{41{,}429}{23.5 \times 1.59} = 1109 \text{ B.t.u./(hr.)(sq.ft.)(}^\circ\text{F.)}$$

$$v = 0.371 \times 5.98 = 2.22 \text{ ft./sec.}$$

Discussion of Results

The data for the heat transmitted through the tube wall and liquid film are plotted against liquid film temperature drop in Figure 15. In general, for each angle of inclination of the heating tube the rate of heat transfer increases with the temperature drop. As the angle of inclination is decreased from 90° to 60° the rate of heat transmission is increased at constant ΔT_L. At an angle of 45° the rate of heat transmission is found to drop to values intermediate between those for 75° and 90°. There is a reversal at 30°, at which angle the highest heat transmission rates are obtained. The values of Q/θ for an angle of 15° are the highest of any group except those for the 30° angle.

The reason for the overlapping and different slopes of the curves in this plot is not quite clear. It is suspected, however, that the characteristic behavior of these curves is due to three factors: first, change in properties of the liquid film; second, change in natural convection in the boiling tube; and third, change in boiling characteristics. All of these factors will vary as the angle of the heating surface is changed.

Taking the data for a ΔT_1 of 40°, values of h for the boiling film were calculated by the Dittus-Boelter equation [1]. It was found that as the position of the evaporator is changed from the vertical toward the horizontal, the value for the liquid film coefficient decreases.

The Grashof-Prandtl [1] correlation was also used to give an indication of the rate of change of h as a result of natural convection. The values for D in this correlation were taken as the vertical projection of the diameter of the tube in its actual position. At 90° this was taken as the length of the tube. The calculations showed that the liquid film coefficient increases as the angle of inclination approaches zero.

The values calculated as described above are as follows:

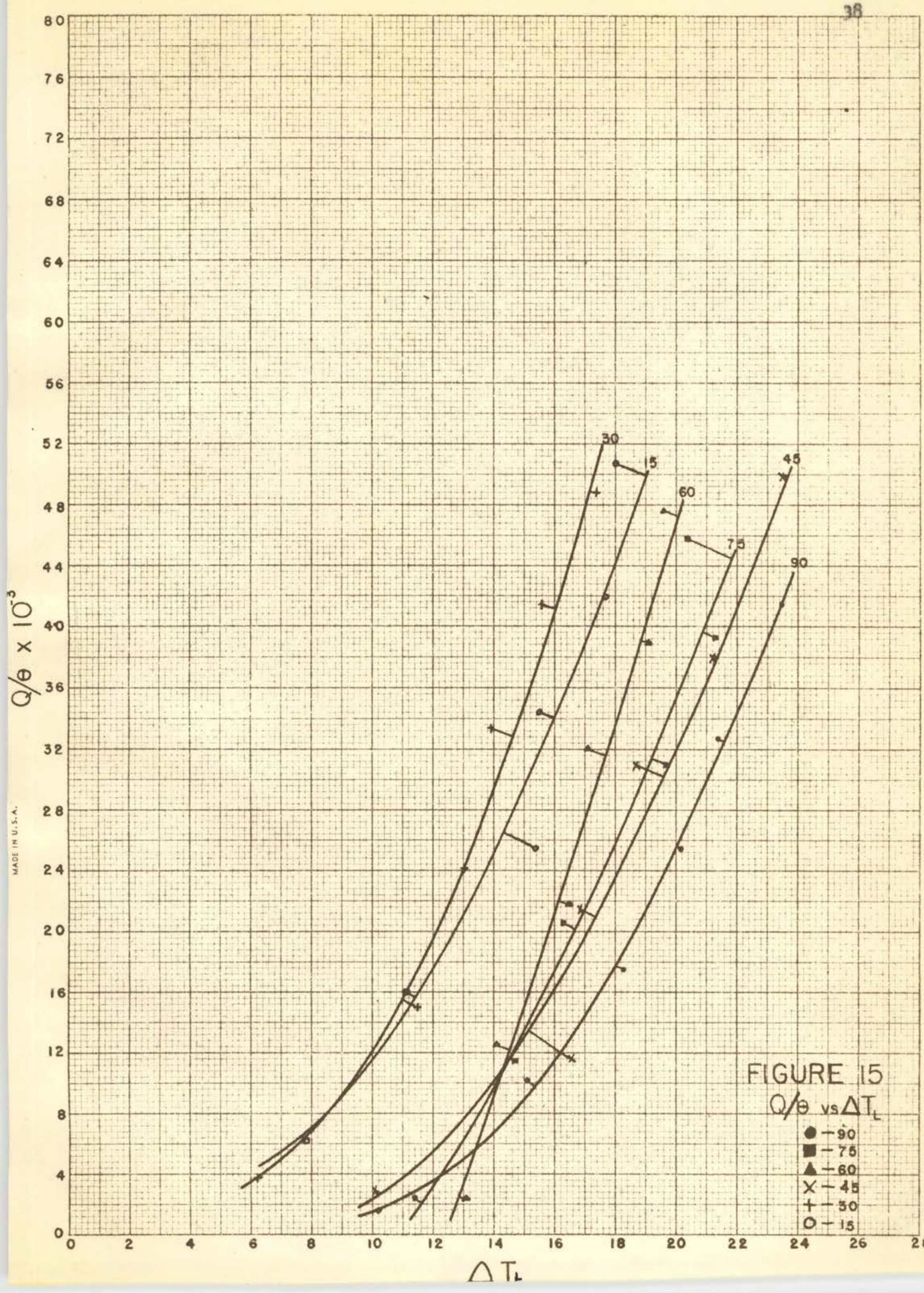


FIGURE 15

Q/θ vs ΔT_L

Angle	Values of h_L by the Dittus-Boelter Equation	Values of h_L by the Grashof-Prandtl Correlation
15	427	510
30	577	690
45	635	768
60	704	850
75	713	845
90	713	787

These values are hypothetical. They, however, indicate the way in which the film characteristics vary with changes in the angle of inclination.

The third factor mentioned above was change in boiling characteristics of the fluid itself, i.e., points of incipient boiling causing disturbances in the liquid film, the formation and growth of steam bubbles in the tube and turbulence of the vapor bubbles as they pass up through the tube.

Figure 16 is a plot of the liquid film coefficient against ΔT_1. In general, it shows that as ΔT_1 for each angle of inclination increases, so does the liquid film coefficient. Again there is present the same overlapping and change of slope of the curves as discussed for Figure 15.

The data for the velocity of natural circulation as a function of the liquid film coefficient are plotted in Figure 17. For each angle of inclination it is found that as h_L increases as a result of increased ΔT_1, the velocity of natural circulation also increases. The reason for this is that as the ΔT_1 increases, the density of the liquid decreases proportionally causing an increase in circulation velocities. As the angle of inclination approaches the horizontal, the natural circulation velocity decreases. This is caused by the decrease in the effective head available for natural circulation.

Figure 18 shows the pressure drop through the liquid return line and the flowmeter as a function of the velocity of natural circulation. From this it is seen that regardless of the angle of inclination the pressure drop remains the same for a given flow.

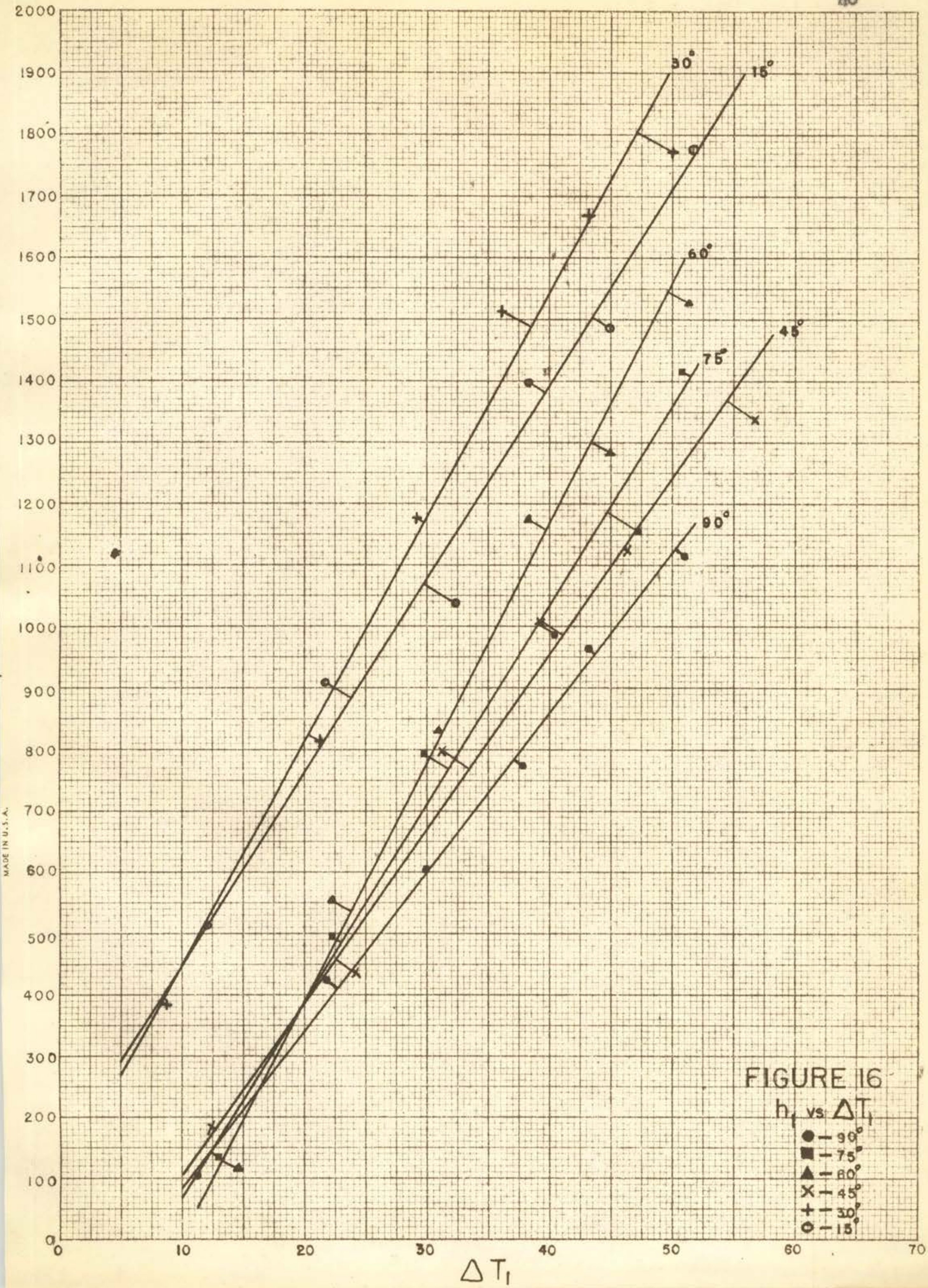


FIGURE 16

h_f vs ΔT_1

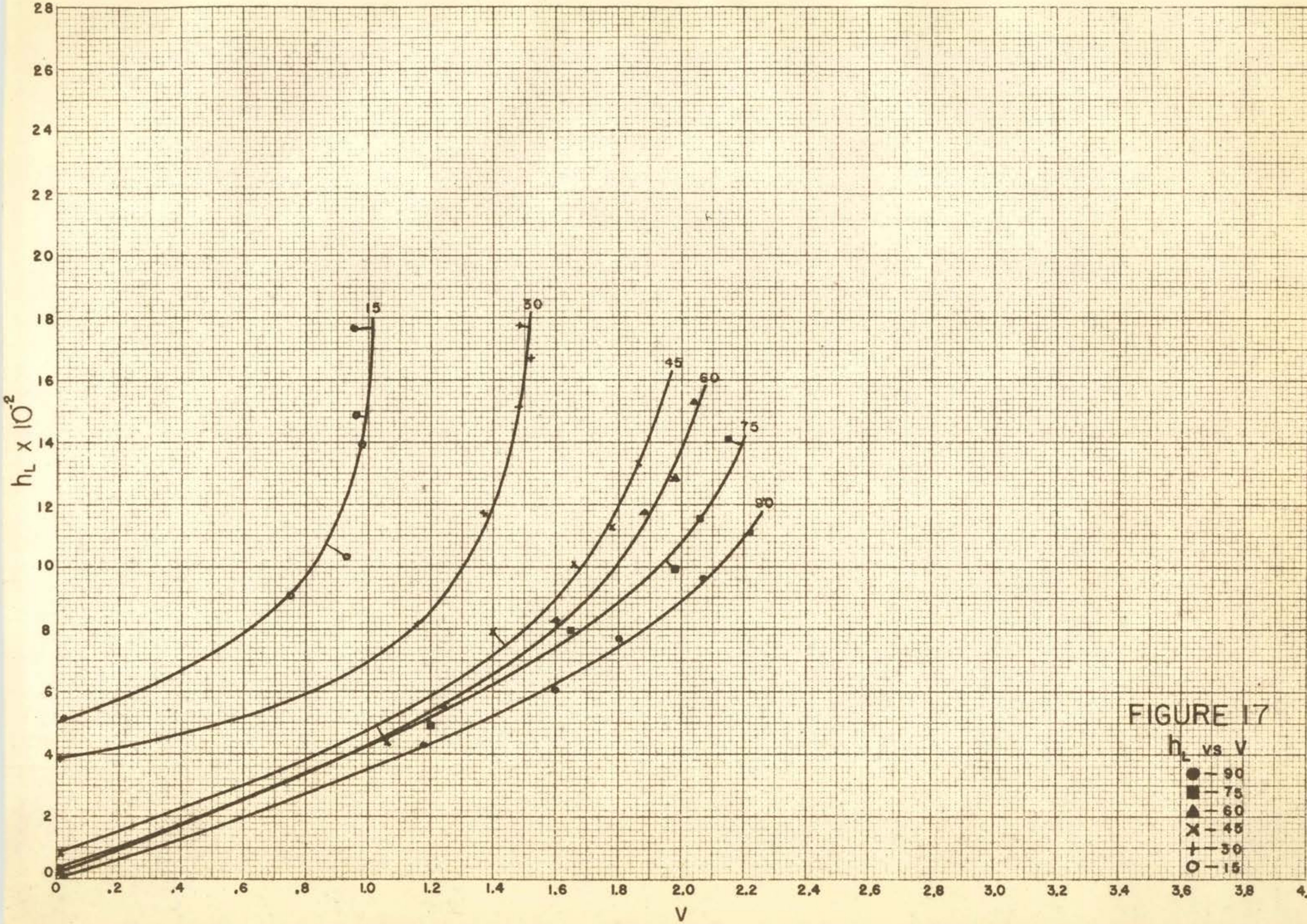


FIGURE 17

h_L vs V

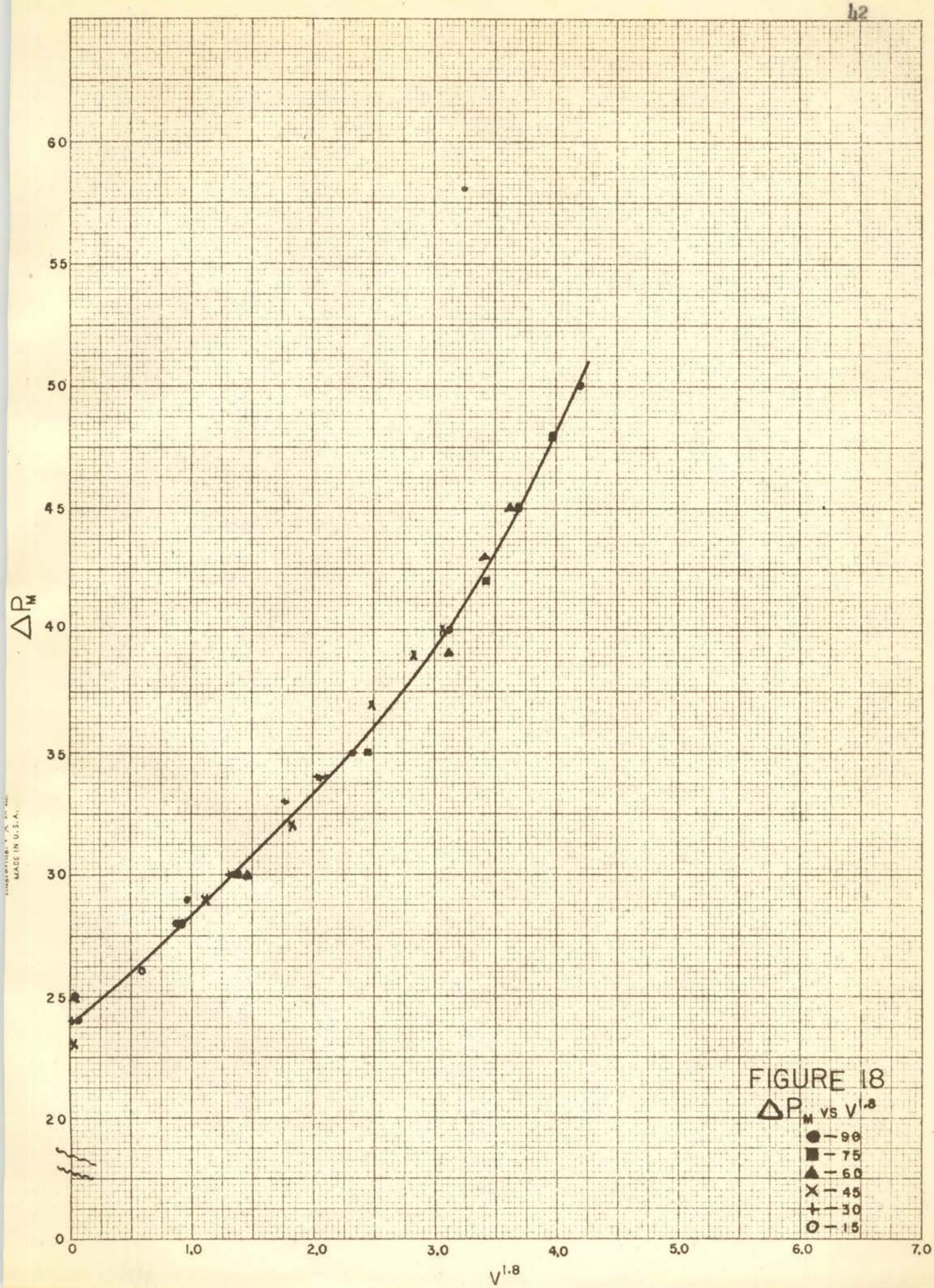


FIGURE 18
ΔP_M VS $V^{1.8}$

Conclusions

1. For each angle of inclination of the heating tube the rate of heat transfer and the liquid film coefficient increases with the temperature drop.

2. As the angle of inclination is changed from the vertical position to the horizontal, the rate of heat transfer and liquid film coefficients increase at constant temperature drops. Values of these two calculations for the 45° angle are found to be intermediate between those for 75° and 90°, and for the 15° angle the values are intermediate between the 30° and 60° angles.

3. The liquid film coefficient increases with increases in natural circulation velocities for any angle of inclination.

4. The natural circulation velocity decreases as the angle of inclination approaches zero.

5. The optimum angle for an inclined tube evaporator as a result of this investigation is 30°.

Appendix

Experimental Data

Run No.	Angle of Inclination ψ	Approx. Steam Pressure psi	Tube Temp. °F.	Boiling Temp. °F.	Feed Water Temp. °F.	Water Evap. Per Hr.	Ave. Pressure Drop thru Meters mm Hg	Natural Circulation G.P.M.
1	90	5	223.0	211.7	195.7	1.7	24.0	0.63
2	90	10	234.4	212.5	209.5	10.5	30.0	3.17
3	90	15	242.5	212.5	210.0	18.1	35.0	4.32
4	90	20	250.5	212.6	210.0	26.3	40.0	5.08
5	90	25	255.7	212.5	210.0	33.7	45.0	5.57
6	90	30	263.7	212.5	210.0	42.6	50.0	5.98
7	75	5	224.3	211.3	198.3	2.5	25.0	0.46
8	75	10	235.0	212.7	208.5	11.8	30.0	3.23
9	75	15	242.4	212.5	209.2	21.1	35.0	4.45
10	75	20	252.8	212.5	209.2	31.8	42.0	5.33
11	75	25	260.0	212.7	209.2	40.3	45.0	5.56
12	75	30	263.3	212.7	209.2	47.1	48.0	5.80
13	60	5	226.0	211.3	195.0	2.5	25.0	0.44
14	60	10	234.4	212.0	207.7	12.8	30.0	3.34
15	60	15	243.0	212.0	209.0	22.4	35.0	4.33
16	60	20	251.0	212.6	209.3	32.9	39.0	5.08
17	60	25	256.9	212.0	209.3	40.0	43.0	5.34
18	60	30	263.3	212.0	209.3	49.0	45.0	5.50
19	45	5	224.7	212.6	192.3	3.0	23.0	0.36
20	45	10	236.7	212.5	209.3	11.9	29.0	2.87
21	45	15	243.7	212.6	210.0	22.0	32.0	3.78
22	45	20	252.5	213.3	210.3	31.8	37.0	4.48
23	45	25	259.7	213.3	210.3	39.0	39.0	4.80
24	45	30	270.0	213.3	210.6	51.4	40.0	5.01
25	30	5	220.0	211.3	192.7	3.8	24.0	0.34
26	30	10	233.7	212.3	208.7	15.3	30.0	3.13
27	30	15	241.7	212.6	208.7	24.9	32.0	3.70
28	30	20	248.7	212.6	209.3	34.3	34.0	4.00
29	30	25	255.7	212.6	209.6	42.6	34.0	4.07
30	30	30	263.0	213.0	210.0	50.5	34.0	4.03
31	15	5	223.0	211.0	202.0	6.5	24.0	0.56
32	15	10	233.0	211.3	207.7	16.4	25.0	2.03
33	15	15	244.6	212.3	208.0	26.1	28.0	2.51
34	15	20	250.0	211.7	208.3	35.3	29.0	2.64
35	15	25	257.5	212.0	208.0	43.0	28.0	2.60
36	15	30	264.0	212.3	208.7	52.1	28.0	2.55

Calculated Values

Run No.	Angle of Inclination ψ	Q/θ	ΔT_1	ΔT_p	ΔT_L	h_L	v	$v^{1.8}$
1	90	1,677	11.3	1.1	10.2	103	0.23	0.072
2	90	10,217	21.9	6.8	15.1	426	1.18	1.35
3	90	17,602	30.0	11.7	18.3	605	1.60	2.33
4	90	25,579	37.9	17.0	20.9	770	1.88	3.12
5	90	32,773	43.2	21.8	21.4	963	2.07	3.70
6	90	41,429	51.0	27.5	23.5	1109	2.22	4.20
7	75	2,460	13.0	1.6	11.4	136	0.17	0.04
8	75	11,494	22.3	7.6	14.7	492	1.20	1.39
9	75	20,537	29.9	13.6	16.3	792	1.65	2.46
10	75	30,951	40.3	20.6	19.7	988	1.98	3.42
11	75	39,224	47.3	26.0	21.3	1158	2.06	3.68
12	75	45,843	50.8	30.4	20.4	1413	2.15	3.97
13	60	2,468	14.7	1.6	13.1	119	0.16	0.04
14	60	12,475	22.4	8.3	14.1	556	1.24	1.47
15	60	21,802	31.0	14.5	16.5	831	1.60	2.33
16	60	32,022	38.4	21.3	17.1	1178	1.88	3.12
17	60	38,920	44.9	25.8	19.1	1281	1.98	3.42
18	60	47,677	51.3	31.7	19.6	1529	2.04	3.62
19	45	2,971	12.1	2.00	10.1	185	0.13	0.03
20	45	11,581	24.2	7.7	16.6	439	1.06	1.11
21	45	21,397	31.1	14.2	16.9	796	1.40	1.83
22	45	30,926	39.2	20.5	18.7	1040	1.66	2.49
23	45	37,928	46.4	25.2	21.2	1125	1.78	2.82
24	45	49,971	56.7	33.2	23.5	1337	1.86	3.06
25	30	3,760	8.7	2.5	6.2	381	0.13	0.03
26	30	14,898	21.4	9.9	11.5	815	1.16	1.31
27	30	24,248	29.1	16.1	13.0	1173	1.37	1.76
28	30	33,384	36.1	22.0	13.9	1511	1.48	2.03
29	30	41,450	43.1	27.5	15.6	1671	1.51	2.10
30	30	49,121	50.0	32.6	17.4	1776	1.49	2.05
31	15	6,370	12.0	4.2	7.8	514	0.21	0.06
32	15	15,980	21.7	10.6	11.1	905	0.75	0.60
33	15	25,431	32.3	16.9	15.4	1039	0.93	0.88
34	15	34,382	38.3	22.8	15.5	1395	0.98	0.96
35	15	41,895	45.5	27.8	17.7	1489	0.96	0.93
36	15	50,725	51.7	33.7	18.0	1772	0.95	0.91

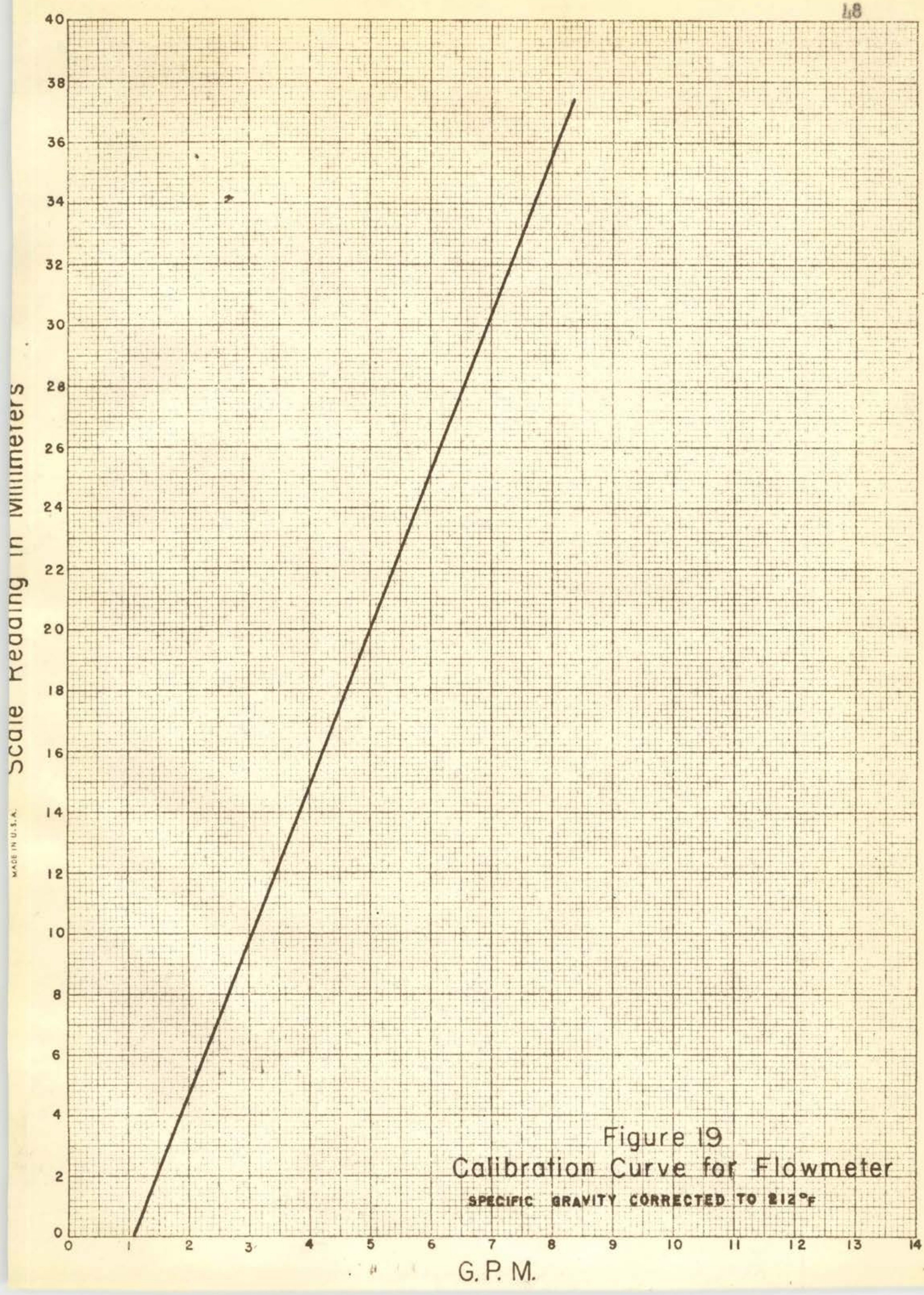


Figure 19
Calibration Curve for Flowmeter
SPECIFIC GRAVITY CORRECTED TO 212°F

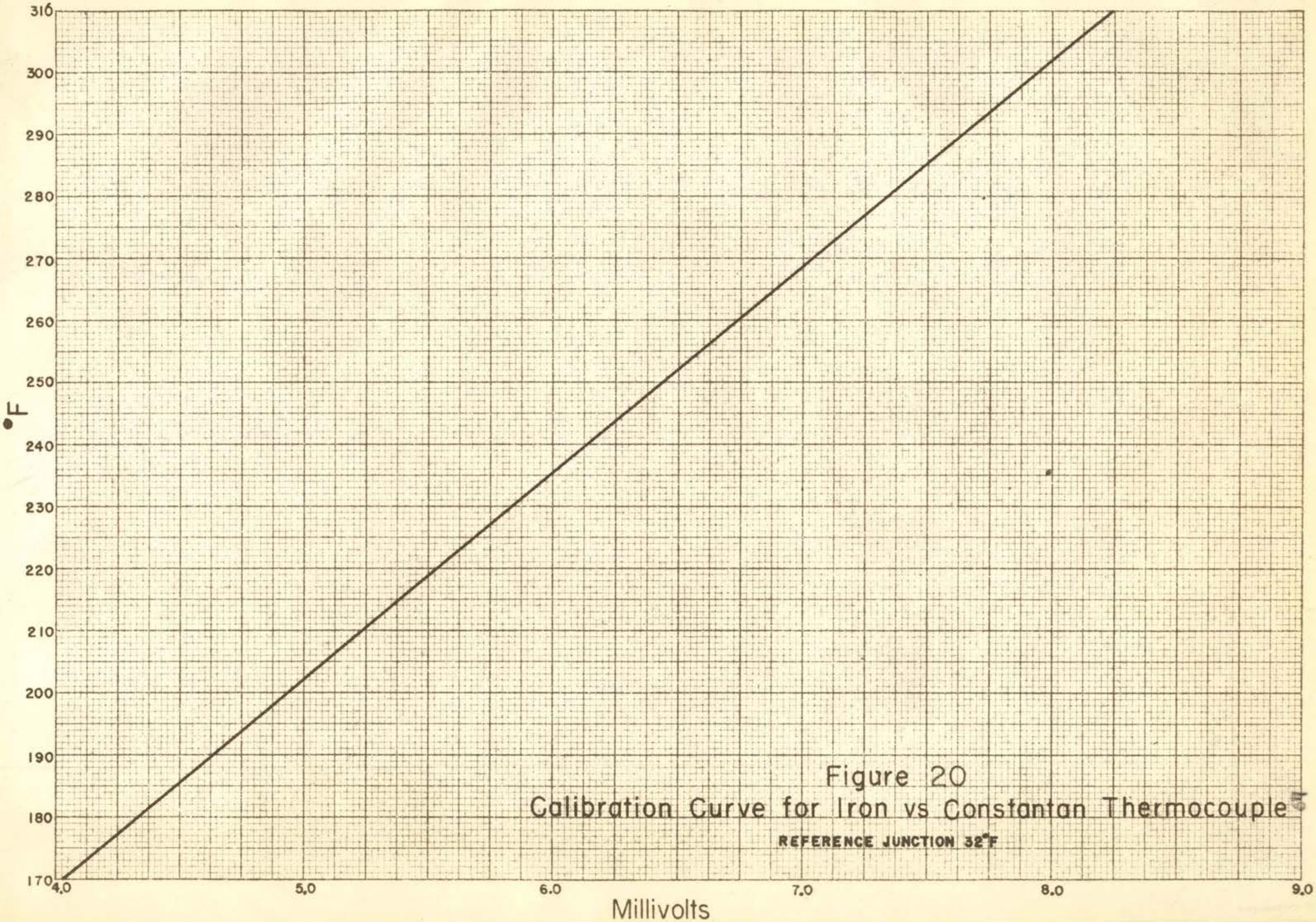
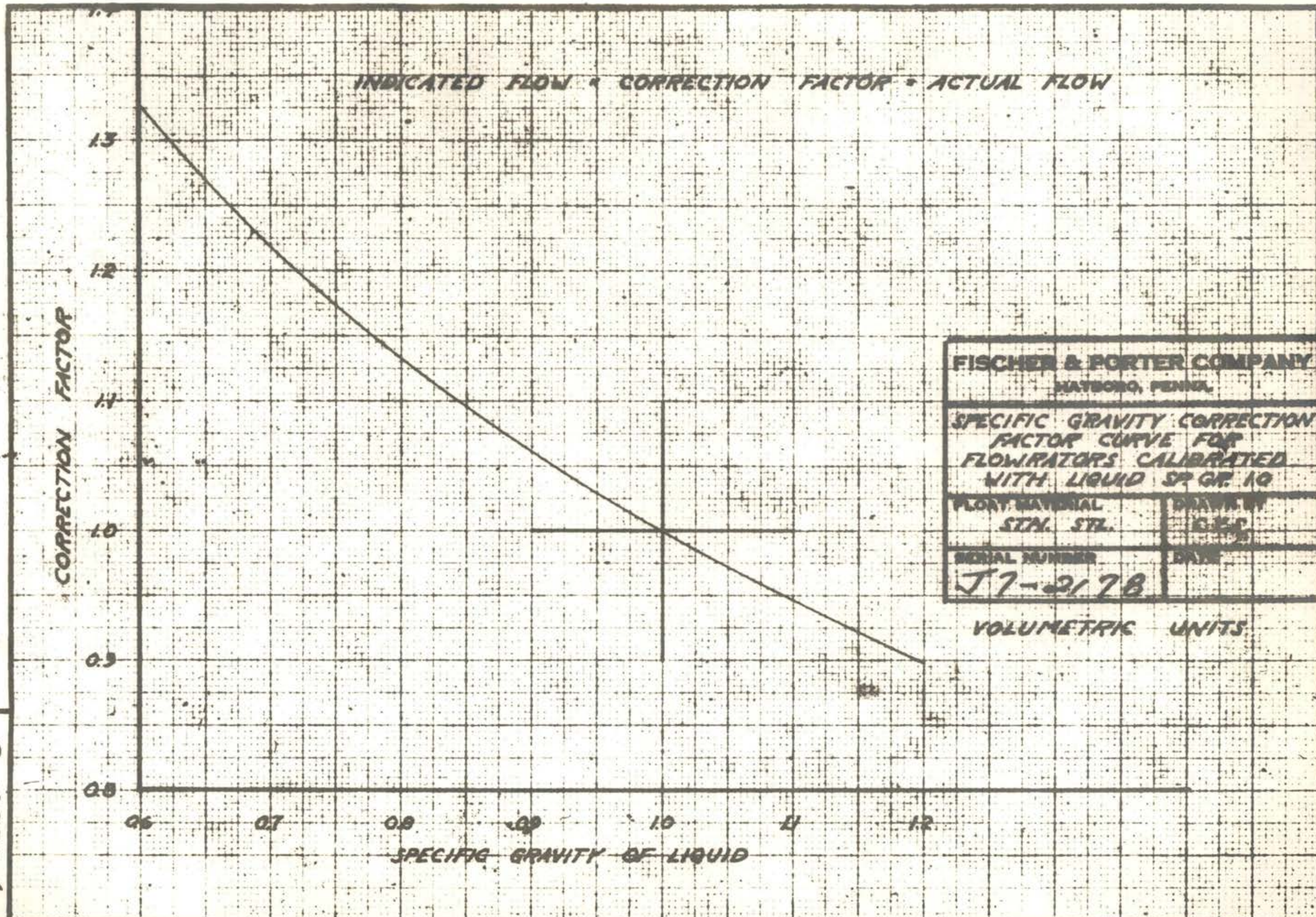


Figure 20
Calibration Curve for Iron vs Constantan Thermocouple

REFERENCE JUNCTION 32°F



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Recommendations for Future Work

1. Continue the study of the effect of the angle of inclination on the liquid film coefficient and on natural circulation at various boiling temperatures and temperature drops.

2. From a theoretical and experimental standpoint, study the effect of tube length on the liquid film coefficient and natural circulation in an inclined tube evaporator.

3. From a theoretical and experimental standpoint, study the effect of tube diameter on the liquid film coefficient and natural circulation in an inclined tube evaporator.

4. Consider the above recommendations using other boiling media.

Katheryn G. Steffee, typist