THE EFFECT OF VISCOSITY ON THE LIQUID FILM HEAT TRANSFER COEFFICIENT IN AN INCLINED TUBE NATURAL CIRCULATION EVAPORATOR

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By

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PREFACE

This thesis is concerned with the effect of viscosity on the boiling film heat transfer coefficient in an inclined tube natural circulation evaporator set at an angle of 30° with respect to the horizontal.

ACKNOWLEDGMENT

The author expresses his gratitude to Dr. Charles L. Nickolls for his recommendations which aided in the performance and completion of this investigation. He thanks the Department of Chemical Engineering for several requisitions while performing the experimentation.

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INTRODUCTION

The viscosity of a liquid is one of the major factors influencing the circulation and heat transfer rate in evaporation. This factor has been mentioned by many investigators, but its specific effect has not been studied in detail. The present investigation will determine the effect of viscosity on the liquid film heat transfer coefficient in an inclined tube evaporator.

In the great majority of cases of heat transfer encountered, heat flows from some medium into and through a solid wall and out into some other medium. Fourier's Law for this case states that the instantaneous rate of heat flow, $dQ/d\theta$, is equal to the product of three factors: the area A of the section, taken at right angles to the direction of the heat flow; the temperature gradient $- dt/dx$; and a proportionality factor k, known as the thermal conductivity. Mathematically expressed, Fourier's Law is as follows:

$$\frac{dQ}{d\theta} = -KA \frac{dt}{dx} \qquad (1)$$

The temperature at any point remains constant for the conduction in the steady state, so the above equation is independent of time. Thus $\frac{dQ}{d\theta} = \frac{Q}{\theta} = q$, and equation (1) becomes:

$$q = -KA \frac{dt}{dx} \qquad (2)$$

For the steady conduction of heat through a single homogeneous body, equation (2) may be written thus:

$$q = \frac{\Delta t}{x/KmAm} = \frac{\Delta t}{R} \qquad (3)$$

where R is the resistance.

If there is a steady flow of heat through several homogenous solids in series, there will be the same amount of heat flowing through each solid.

Thus one can write

$$q = \frac{\Delta T_1}{R_1} = \frac{\Delta T_2}{R_2} = \frac{\Delta T_3}{R_3} \qquad (4)$$

where R_1, R_2, etc., are the individual resistances to heat flow.

By addition, one can obtain

$$q = \frac{\Sigma(\Delta T)}{R_T} = \frac{(\Delta T_1) + (\Delta T_2) + (\Delta T_3)}{R_1 + R_2 + R_3} \qquad (5)$$

which may be expressed as:

$$q = \frac{\Sigma(\Delta T)}{\frac{x_1}{k_1 A_1} + \frac{x_2}{k_2 A_2} + \frac{x_3}{k_3 A_3}} \qquad (6)$$

For heat transfer through a film of liquid, it is not convenient to measure the thickness of the laminar film or the temperature at the interface between the films. The individual heat transfer coefficient, h, will be defined as

$$q = h A \Delta T \qquad (7)$$

where ΔT = temperature of the liquid minus the temperature of the wall. The overall coefficient of heat transfer, U, will be defined as

$$q = U A \Delta T_T \qquad (8)$$

where ΔT = total temperature difference across all the films.

For heat transfer between fluids separated by a tube wall, the equations are:

$$q = h_1 A_1 (t_1 - t_2)$$
$$q = \frac{k_2 A_2 (t_2 - t_3)}{x_2} \qquad (9)$$
$$q = h_3 A_3 (t_3 - t_4)$$

where t_1 = temperature of fluid outside the tube

t_2 = temperature at the outside wall of the metal tube

t_3 = temperature at the inside wall of the metal tube

t_4 = temperature of liquid inside the tube.

and this leads to the equation

$$q = \frac{t_1 - t_4}{\frac{1}{h_1 A_1} + \frac{x_2}{k_2 A_2} + \frac{1}{h_3 A_3}} = U A \Delta T_T \qquad (10)$$

By assuming that U is to be based on A_1, equation (10) becomes:

$$U = \frac{1}{\frac{1}{h_1} + \frac{x_2 A_1}{k_2 A_2} + \frac{A_1}{h_3 A_3}} \qquad (11)$$

and by substitution

$$q = \frac{A_1 \Delta T_T}{\frac{1}{h_1} + \frac{x_2 A_1}{k_2 A_2} + \frac{A_1}{h_3 A_3}} \qquad (12)$$

By means of equation (12) the individual coefficients can be related to the heat flow. Knowing h_1, k_2, and h_3 together with the dimensions of the equipment, it is possible to calculate q for many cases without actually performing experiments to determine the values.

In order to determine an individual coefficient, h_1, in a tubular heat exchanger, A_1, $(t_1 - t_2)$, and q must be known. A_1 is the area of the tube surface and q can be determined by the amount of liquid evaporated while t_2 can be measured by a thermocouple buried in the tube wall. Thus, all the factors for the calculation of h_1 are known.

The present investigation considers the effect of viscosity on the liquid film heat transfer coefficient. Solutions with viscosities covering the range desired will be produced by using different concentrations of sucrose in distilled water. The three unknown quantities in equation (12); A_1, q, and t_2, will be determined experimentally. Using these data, the values of h_1 will be calculated and correlated to determine the effect of viscosity.

LITERATURE SURVEY

Heat transfer data for an inclined tube evaporator is rather meager. Different types of equipment have been used, and the methods of correlation were not similar.

Van Marle (21) used a rapid circulation type evaporator with inclined steam chest, at an angle of 45°. The heating surface consisted of seven 3 inch O.D., 14-gage copper tubes, 4 feet 10½ inches long, giving an interior tube surface of 25.3 square feet. He found that the overall heat transmission coefficient varied from 835 Btu/ft^2/hr/°F at a 50° F. temperature difference to 1140 Btu/ft^2/hr/°F. at a 81° F. temperature difference. Distilled water, colored with tannin extract to detect entrainment, was used for the limited range of temperatures of evaporation.

Linden and Montillon (13) used apparatus consisting of a small inclined tube evaporator containing one 1 inch copper pipe 4.08 feet long. They used distilled water boiling at 180° F., 195° F., and 210° F. with temperature differences ranging from 8° to 28° F. for each evaporation temperature.

By plotting the liquid film coefficients against the temperature drop across the liquid film, they derived the following approximate correlations:

$$\text{at } 180^\circ \text{ F., } h_L = 0.632\,\Delta T_L^{\,2.5}$$

$$\text{at } 195^\circ \text{ F., } h_L = 1.010\,\Delta T_L^{\,2.5}$$

$$\text{at } 210^\circ \text{ F., } h_L = 1.564\,\Delta T_L^{\,2.5}$$

By plotting the liquid-film coefficients against the log mean velocity of the steam plus water flowing up the heating tube, the relationship was correlated by the following approximate equations:

$$\text{at } 180^\circ \text{ F., } h_L = 97\, V_m^{0.89}$$

$$\text{at } 195^\circ \text{ F., } h_L = 120\, V_m^{0.89}$$

$$\text{at } 210^\circ \text{ F., } h_L = 150\, V_m^{0.89}$$

This log mean average velocity was calculated by the following equation:

$$V_m = \frac{\text{steam velocity}}{2.3 \log \dfrac{\text{steam velocity} + \text{liquid velocity}}{\text{liquid velocity}}}$$

No theoretical significance was claimed for employing the log mean velocity since it was entirely empirical.

Elimination of h_L from the two sets of equations gave the following values of T_L:

$$\text{at } 180^\circ \text{ F., } \Delta T_L = 7.49\, V_m^{0.356}$$

$$\text{at } 195^\circ \text{ F., } \Delta T_L = 6.76\, V_m^{0.356}$$

$$\text{at } 210^\circ \text{ F., } \Delta T_L = 6.20\, V_m^{0.356}$$

From the above empirical equations, they examined the possibility that an expression might be found defining h_L in terms of the dimensionless groups employed for heat transfer coefficients in turbulent flow. In this region, according to Morris and Whitman (17)

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

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 $feet^3$.

Values of hd/k and $dUm\rho/\mu$ were plotted on log-log paper, for the three temperatures resulting in straight lines. They found that the equation

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

represented the conditions with an average accuracy of $\pm$ 5 per cent, with k, c, μ, and ρ taken at the temperature of evaporation.

Aubrecht (1) compiled data on both natural and forced circulation on an evaporator inclined at 45°. The apparatus used was that of Michaelson (10) with various modifications. The heating surface consisted of a four foot copper tube, one inch inside diameter and steam heated, externally.

Some of the problems with which he was concerned were:

1. To find the effect of liquid circulation in a forced circulation inclined tube evaporator in the lower ranges of liquid velocity below five feet per second.

2. To measure the rate of natural circulation in an inclined tube evaporator.

3. To correlate the effect of liquid circulation, in both forced and natural circulation, on the heat transfer rate.

4. To determine the effect of temperature drop on heat transfer for both forced and natural circulation.

5. 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.

By his methods of correlation, Aubrecht made the following statements:

1. The data for forced circulation were correlated by means of an equation:

$$\frac{Q}{\Theta}\ \frac{1}{G^n}\left(1 - 0.79\ \frac{k^{.6}\ c^{.4}}{\mu^{.4}}\right) = \Psi \Delta T^m$$

where $\frac{Q}{\Theta}$ = Btu transferred/hr.

G = mass velocity in lbs/ft^2 sec.

n = a constant

k = thermal conductivity

c = specific heat

μ = viscosity

Ψ = a constant

ΔT = average film temperature drop

m = a constant

2. The forced circulation data were correlated in two ranges, above $\frac{Q}{\Theta}$ = 30,000 Btu/hr, and below this value. The values of the constants in the above equation were:

	n	Ψ	m
Below $\frac{Q}{\Theta}$ = 30,000 Btu/hr	+ 0.3	17.65	1.55
Above $\frac{Q}{\Theta}$ = 30,000 Btu/hr	− 0.1	184.8	1.55

3. The natural circulation data were correlated by the same equation, the constants in this case were:

$$n = +0.3 \qquad \Psi = 1.124 \qquad m = 2.55$$

4. In forced circulation evaporation in velocities up to 5 ft/sec., the Dittus-Boelter equation did not hold except at low temperature differences.

5. A heat load was reached in forced circulation where an increase of velocity caused a decrease of $\frac{Q}{\theta}$ at constant ΔT. The critical heat load in this experiment was approximately 30,000 Btu/hr.

6. The effect of the temperature differences was not as great in forced circulation as it was in natural circulation.

7. In natural circulation, the liquid circulation increased with increasing heat load to a maximum at about 30,000 Btu/hr., and then decreased.

8. The effect of a small increase in concentration of sugar was much greater at low concentrations that it was at high concentrations.

Michaelson (16) used the same evaporator as did Linden. He studied natural circulation but did not measure the velocity. He calculated the liquid film coefficients for water boiling from 144° F. to 210° F. at temperature drops varying from 0 to 85° F., but gave no correlation of the data.

Obert (18) studied the effect of scale formation on the heat transfer coefficient. The apparatus was similar to that of Aubrecht's (1) with a heating tube of standard one-inch, type 316, stainless steel pipe with an effective heating length of 69.6 inches. Obert's data were obtained at an angle of inclination of 45°, and a temperature value difference of approximately 27° F. He found that the equation of Robinson and McCabe (15) was applicable to the range of scaling in this particular evaporator.

Steffee (20), working on an evaporator similar to that of Aubrecht with certain modifications in the entrainment separator and liquid velocity measurements, studied the effect of the angle of inclination on the liquid film coefficient. He found that the natural circulation decreased as the angle of inclination approached zero.

As the angle of inclination was changed from the vertical position to the horizontal, the rate of heat transfer and liquid film coefficients increased for constant temperature drops. Values of these two calculations for the 45° angle were found to be intermediate between those for 75° and 90°, and for the 15° angle, the values were intermediate between the 30° and 60° angles. The optimum angle for an inclined tube evaporator, as a result of Steffee's investigation, was 30°.

EXPERIMENTAL EQUIPMENT

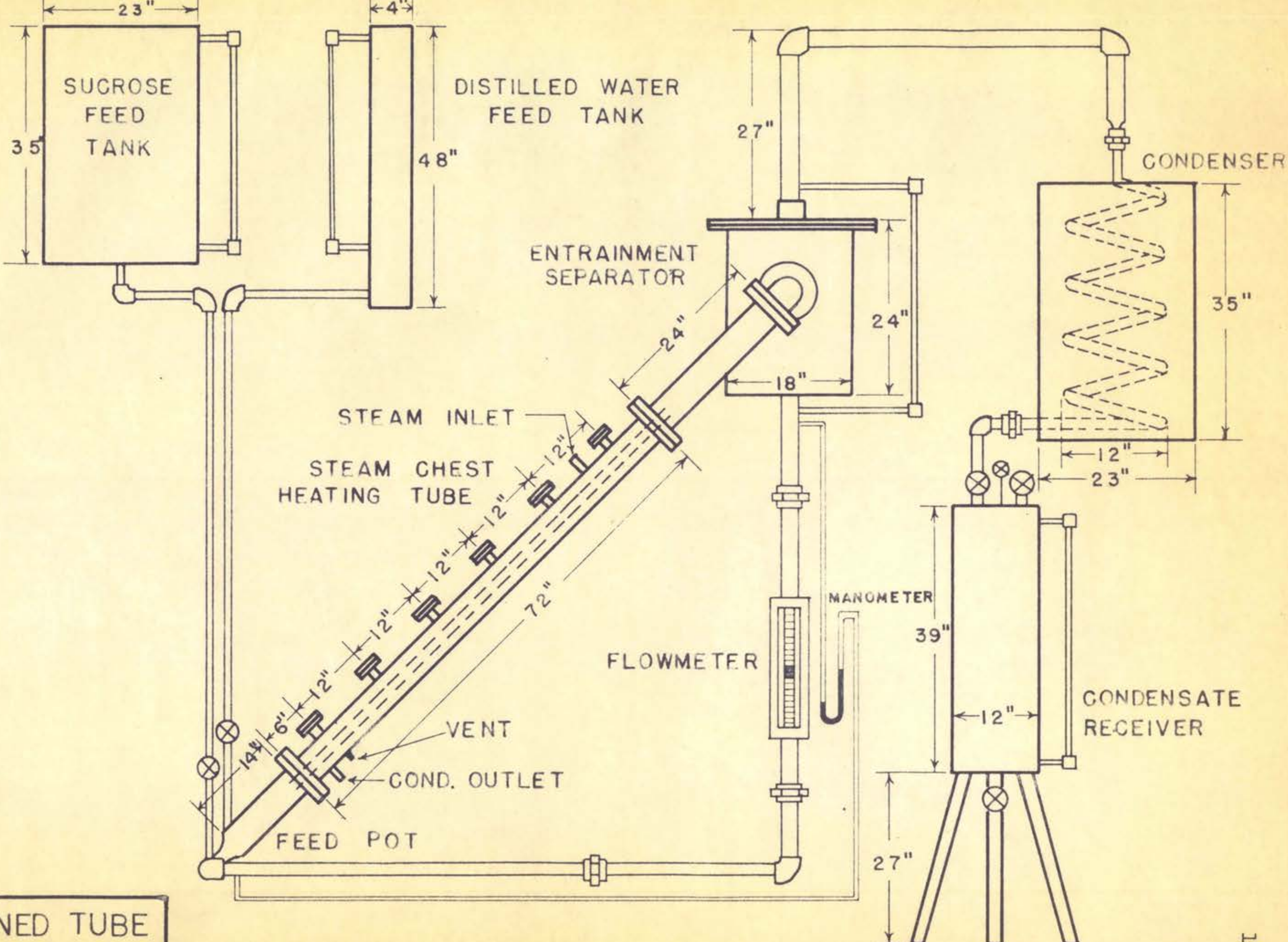
The apparatus used for this investigation was that of Aubrecht with certain modifications by Obert and Steffee. The evaporator contained a single tube, externally steam heated, with the evaporating unit inclined at an angle of 30°, with respect to the horizontal (Schematic diagram, Figure 1).

The sucrose solution feed reservoir was a fifty-gallon steel drum with a central outlet located in the bottom. The solution was admitted to the system by a one-half inch gate valve installed in a one-half inch line which connected the feed reservoir to the feed pot. The reservoir was equipped with a sight glass which indicated the liquid level of the feed at all times.

The distilled water feed reservoir was a $7\frac{1}{2}$ inch square tank, 4 feet high with the outlet located $2\frac{1}{2}$ inches from the bottom. The water was admitted to the system by a one-half inch line to the feed pot. The amount of water was regulated by a one-half inch gate valve installed in the line. A sight glass enabled the operator to observe the feed level at all times.

The feed pot for the evaporating unit was a fourteen inch section of four inch pipe flanged to the bottom of the steam chest. The feed pot was connected by a one-half inch line to the feed reservoirs and by a one-inch recirculation line to the entrainment separator. The feed pot had a one-half inch hole through which a glass tube was inserted. The junction of the thermocouple was placed in the glass tube filled with oil so as to measure the temperature of the recirculating liquid.

The steam chest was a standard four inch iron pipe with flanged ends. Steam was admitted one foot from the top of the chest and removed at the



INCLINED TUBE EVAPORATOR
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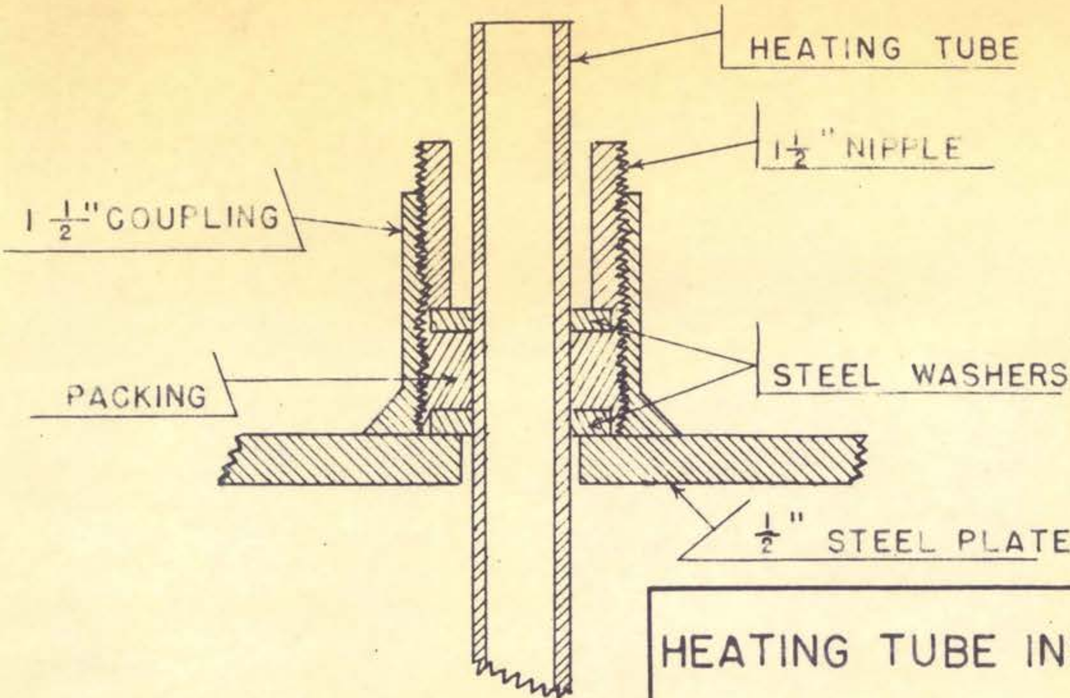
FIGURE 1

bottom by an Anderson #21 steam trap. Near the outlet for the steam trap was a one-fourth inch outlet for the continuous venting of non-condensable gases.

Six one-inch couplings were welded into the steam chest. These served as outlets for the thermocouple leads which were attached to the outside surface of the heating tube. The first couple was six inches from the end of the steam chest and the others spaced at one foot intervals. 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 2.

The heating element of the evaporator was enclosed in the steam chest by stuffing glands in the two ends of the steam chest as shown in Figure 2. The heating element was a standard one-inch, type 316, stainless steel pipe, 74 inches long. The effective heating length was 69.6 inches. 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 (8) and approved by McAdams (14). Slots about 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 perimeter of the heating tube. The insulated leads were brought away from the junction through the slot to the opposite side of the pipe.



HEATING TUBE INSTALLATION

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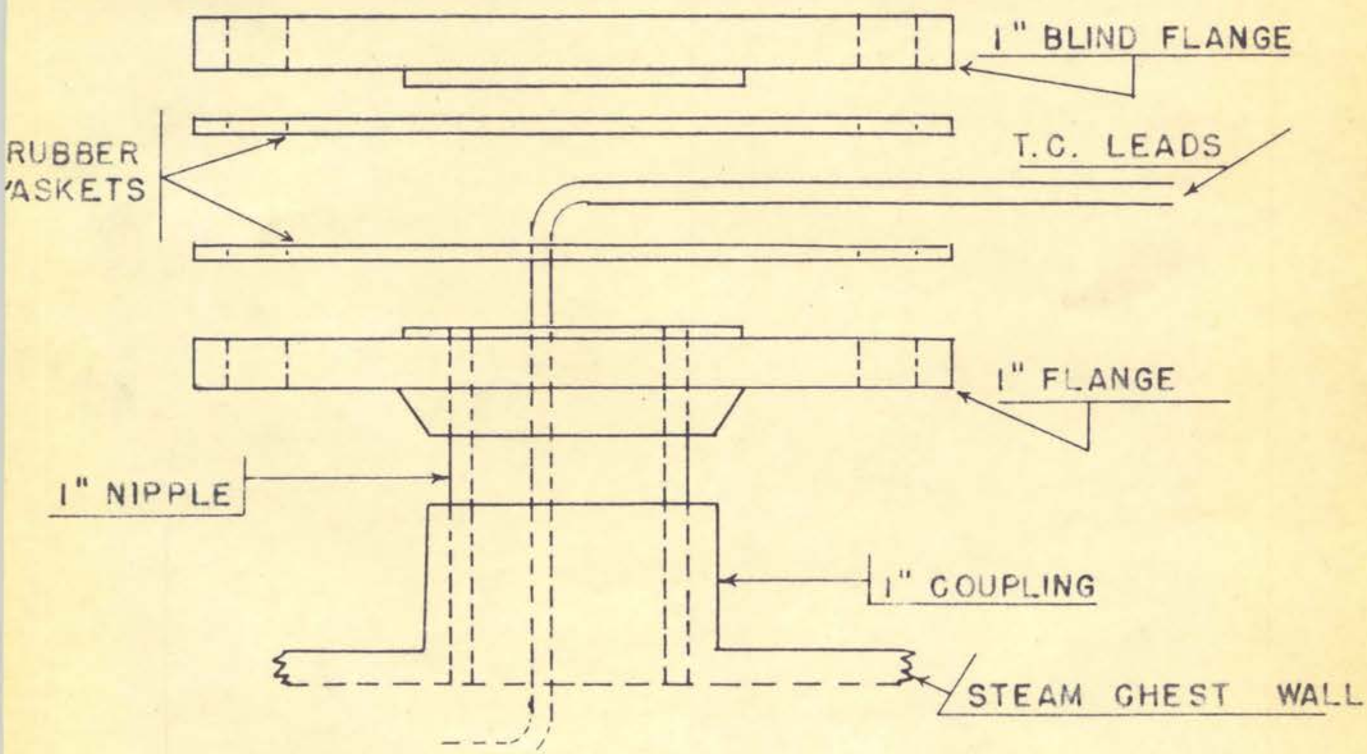

FIGURE 2

THERMOCOUPLE OUTLET

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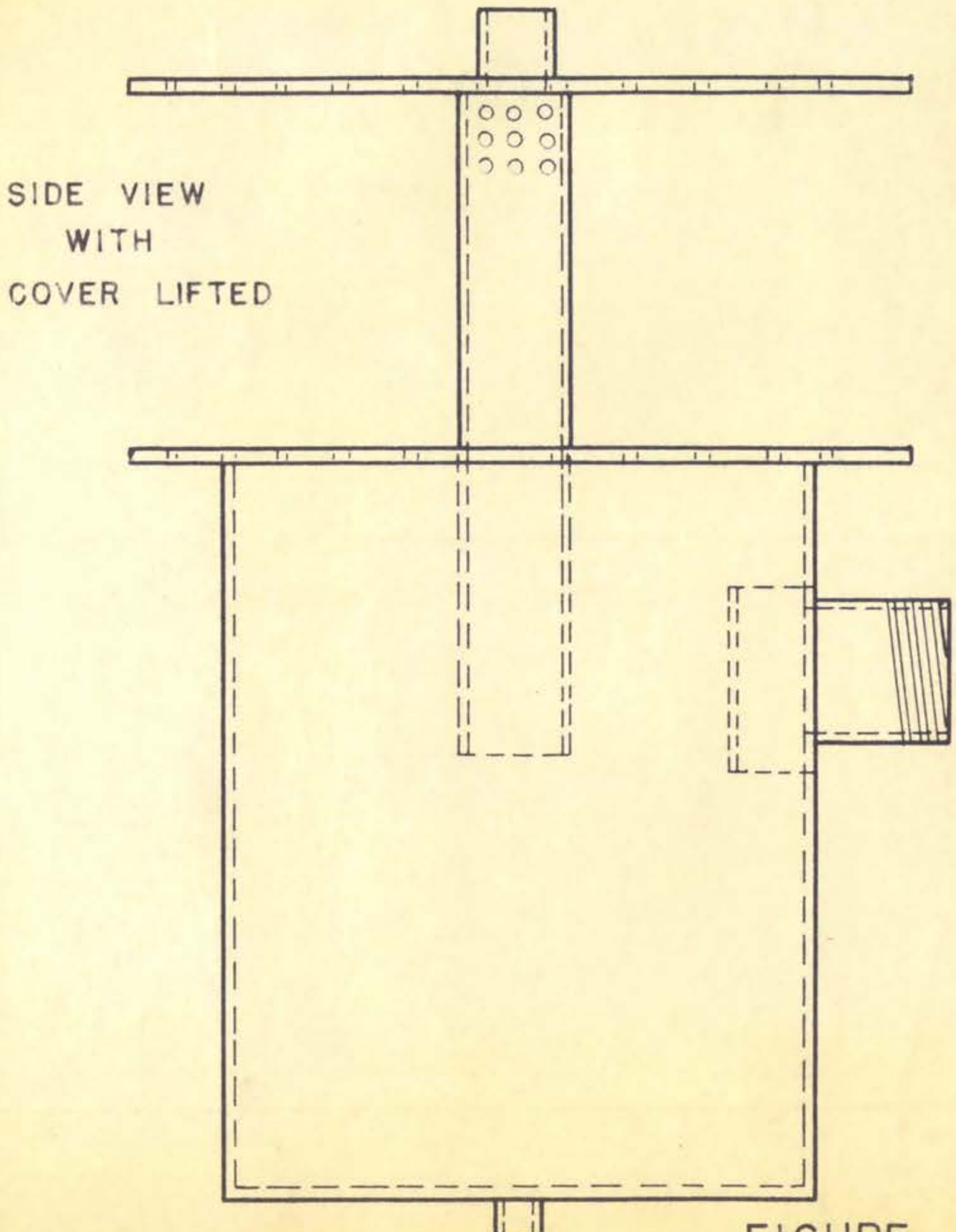
Lead was then pounded into the slot and the excess polished off plush with the surface of the tube (see Figure 4). After carefully placing the heating tube in the steam chest, the thermocouple leads were brought out of the steam chest through the six one-inch flanges as shown in Figure 2.

The disengaging section was a 24 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. The section also had a 1/2 hole through which a thermocouple junction was placed to measure the temperature of the boiling liquid. The junction was inserted in a glass tube filled with oil and placed near the end of the tube.

The entrainment separator was of the cyclone type. The body consisted of a twenty-four inch section of standard 18-inch iron pipe. The bottom was a 3/8 inch steel plate and contained a one-inch outlet 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 serves as an outlet for the vapors. The vapor inlet was a standard four-inch nipple welded into the side of the 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 bath. See Figure 3. There were twelve 1/2-inch holes in a 4-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 90°, 4 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.



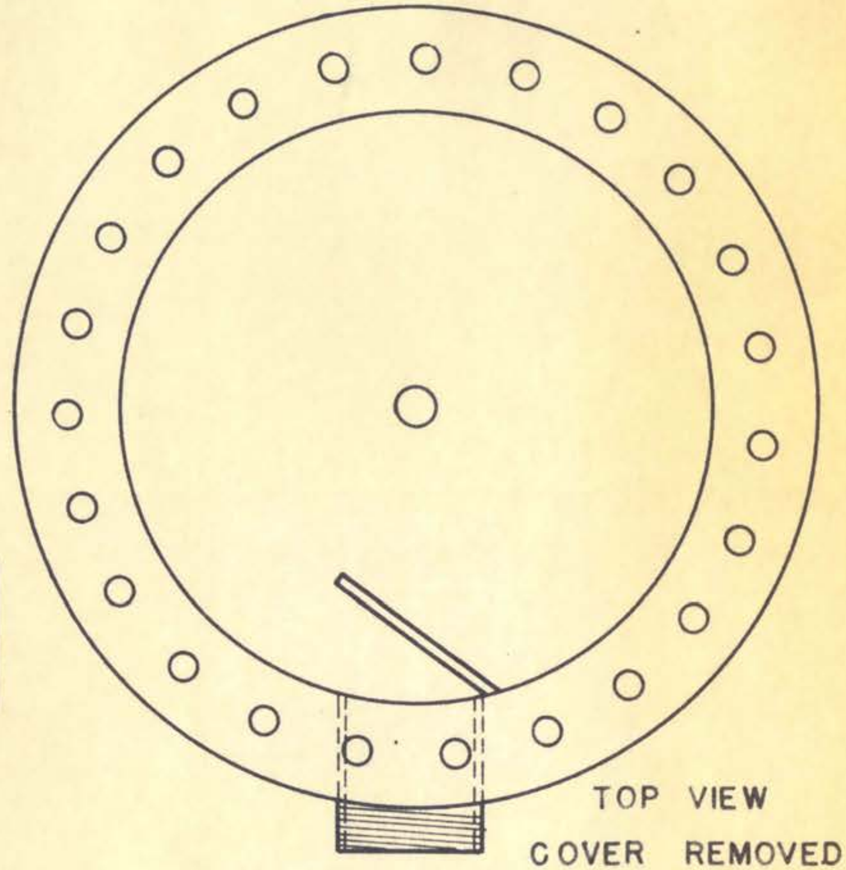


FIGURE 3

ENTRAINMENT SEPARATOR

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The liquid return line was connected to the entrainment separator and to the feed pot by standard one-inch pipe. Two Fischer-Porter flowmeters were selected for measuring the natural circulation liquid velocities. One of the meters was used for measuring fluid flow over the range of 0.1 to 2.4 G.P.M. (Figure A-2). 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 the scale and the plotting of the calibration curve (Figure A-3) readings for liquid velocity were readily obtained in terms of G.P.M. The meters were connected in parallel in the return line. 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.

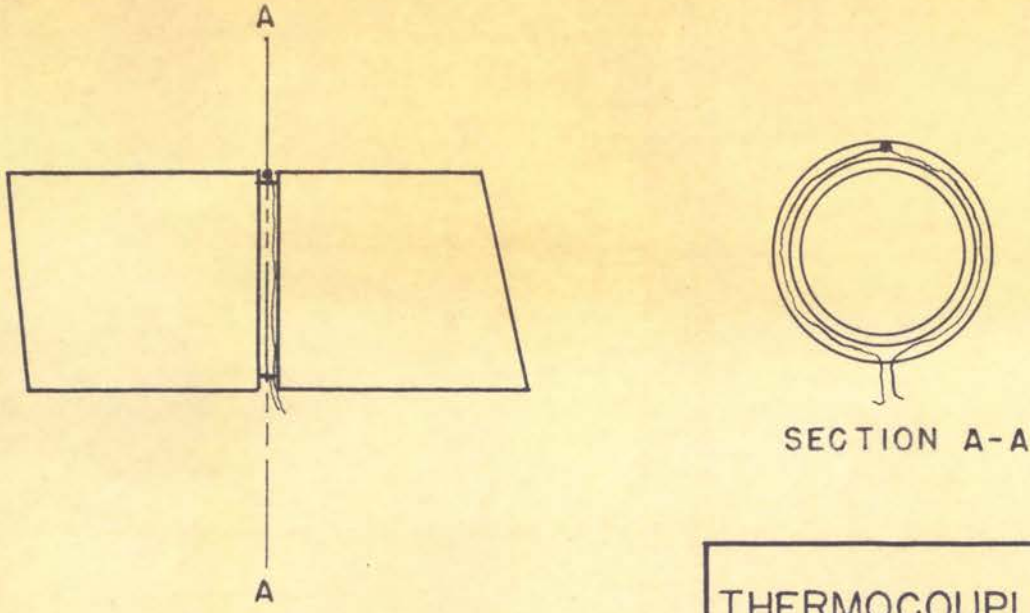
Manometer connections were located at the outlet of the entrainment separator and at the inlet of the feed pot. Connections were then made to a mercury manometer by means of rubber tubing. The purpose of the manometer was to measure the friction loss through the liquid return line and the flow meters.

The condenser was made of ten turns of one-inch copper tubing in a vertical coil 12 inches in diameter. It was encased in an open fifty gallon steel drum. Cooling water was introduced at the bottom of the drum and flowed out of a two-inch overflow line located near the top of the drum.

The condensate receiver was a 12 inch by 39 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 indications. On the bottom was a 1/2 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. 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 shown in Figure 4. The E.M.F. of the thermocouple was measured by a Leeds and Northrup Type K-2 potentiometer (12). This instrument is capable of reading hundredths of a millivolt 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.

Only the parts of the evaporator from which heat loss could not be tolerated were insulated. These parts included the feed pot, the disengaging section, the 90° 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.



THERMOCOUPLE INSTALLATION

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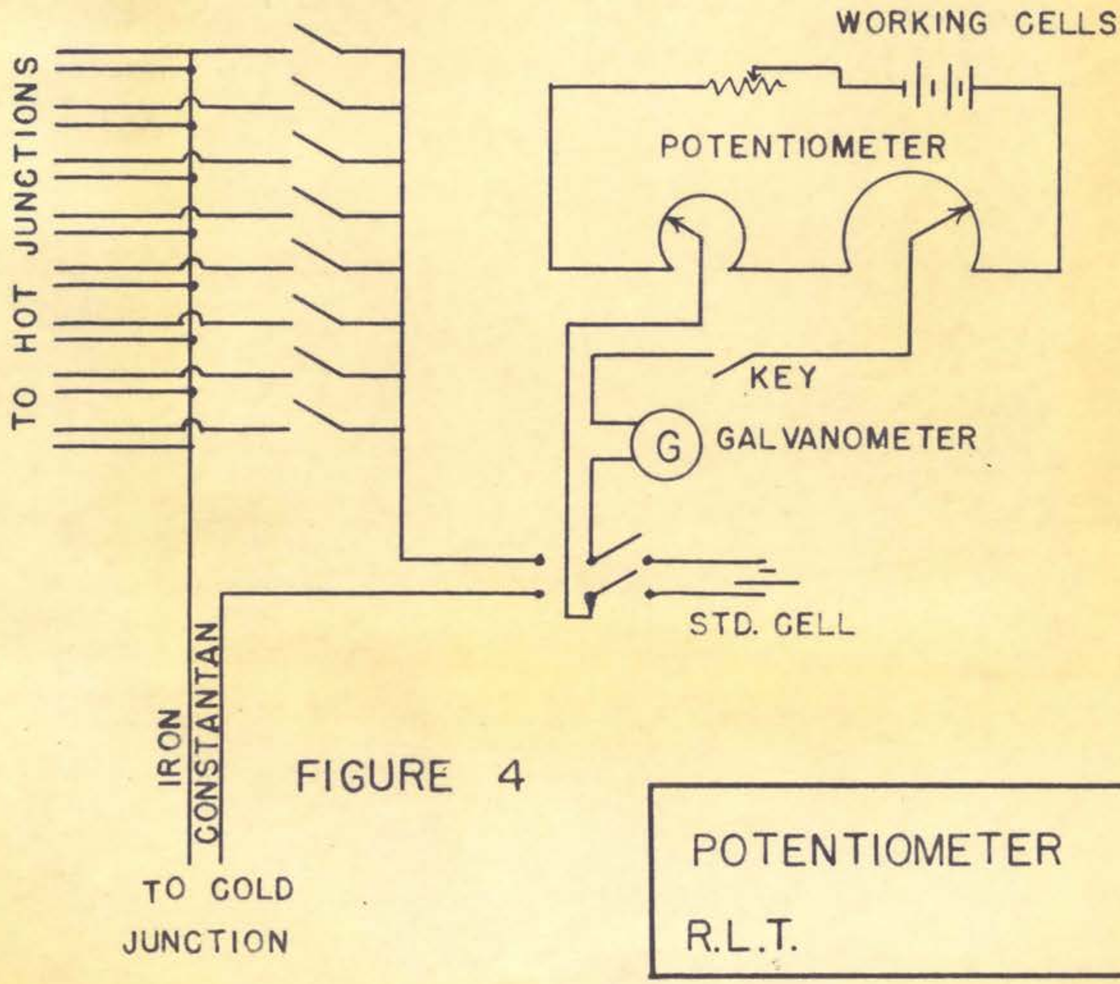

FIGURE 4

POTENTIOMETER CIRCUIT

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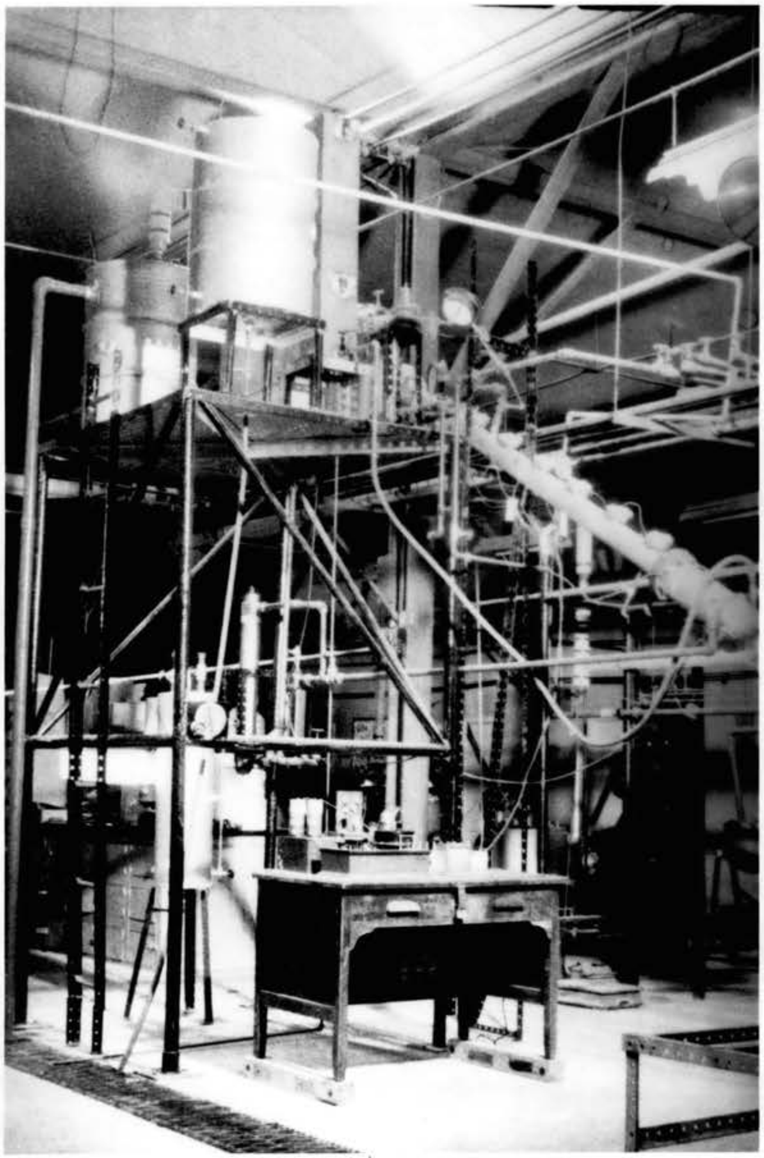
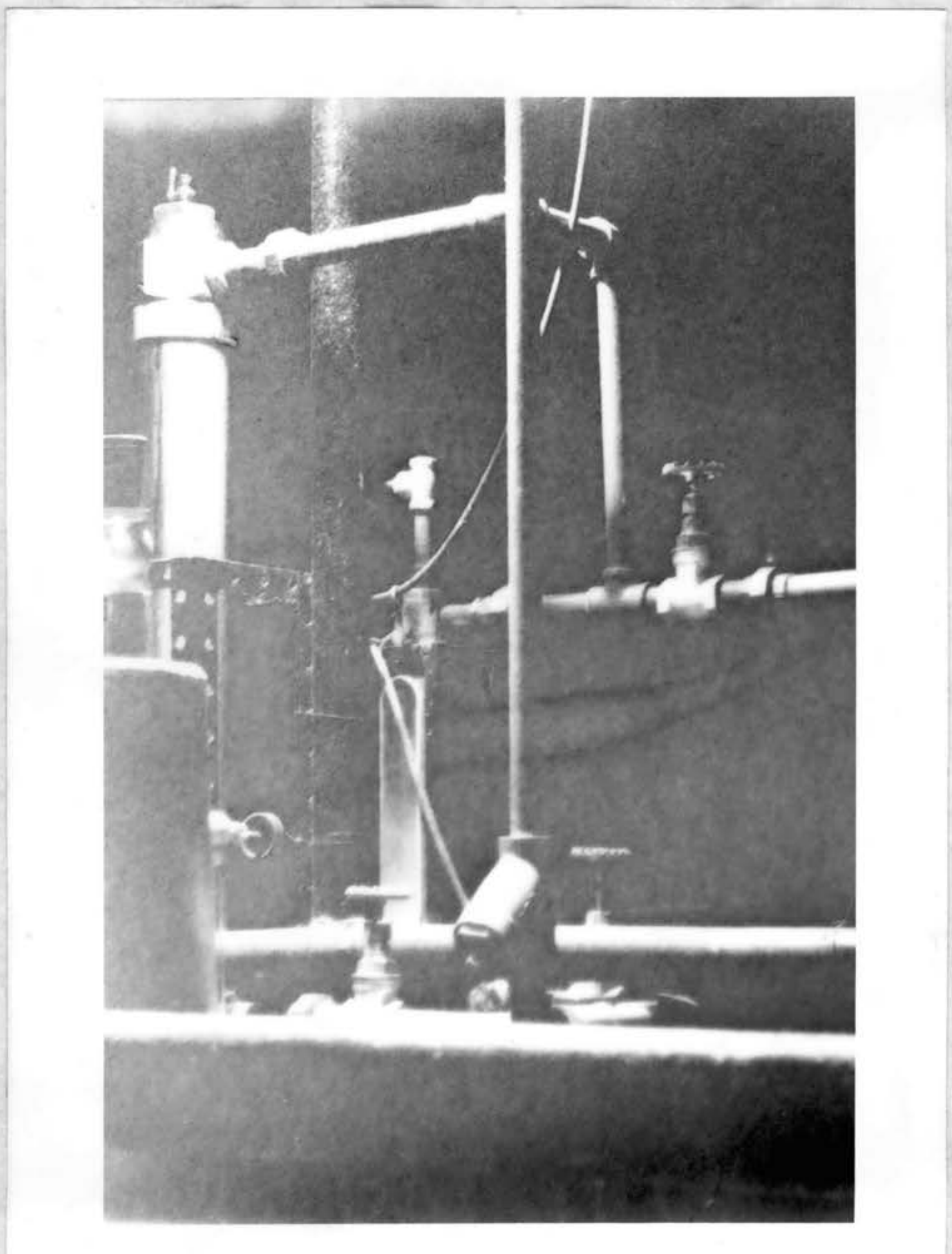
FIGURE 5



Photograph of Inclined Tube Evaporator

FIGURE 6



Photograph of Flowmeter Piping

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 Figure A-2.

The larger flowmeter was calibrated by weighing the amount of water passing through it per unit time at various float levels. The calibration data were corrected to 212° F. by means of the correction curve previously mentioned. The corrected calibration curve is presented in Figure A-3.

All thermocouples were checked by comparing them against boiling water at atmospheric pressure and were found to be accurate to $\pm$ 0.2° F. The thermometer used for obtaining the liquid temperature out of the distilled water feed reservoir was found to be accurate to $\pm$ 1.0° F. Better accuracy was not needed for any of the calculations.

Before any tests were run on the apparatus, it was cleaned by boiling a 5% Na_2CO_3 solution in it, followed by a water wash, and then a 5% HCl solution. Rust particles from the connecting piping were removed prior to a run by means of forced circulation of water.

The actual procedure in making a run was as follows:

Before starting a run the batteries for the potentiometer were connected so that they could come to a fairly steady state before they were used. Ice was crushed to approximately 1/4 inch size, washed and packed in the thermos bottle for a cold junction. The steam line was blown free of condensate.

Before the first run of the day was started, the tube was cleaned with a swab wet with 5% HCl. This removed any deposit from the tube after the tube was washed with distilled water.

If the run was to be made on a sucrose solution, the solution was made up in the sucrose feed tank using distilled water and refined beet sugar. With all the water drained from the evaporator, the cold sucrose solution was allowed to flow by gravity to the proper level.

During a run, the liquid level and the concentration of sucrose were kept constant by feeding distilled water equal to the rate of evaporation and there was little or no sugar lost by entrainment. The level in the evaporator was maintained at a height of 2-2½ inches above the vapor inlet to the entrainment separator.

The steam pressure was adjusted and the liquid level kept constant. Steady state conditions were attained after 5 to 10 pounds of water were evaporated. The runs were of 1/2 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.

The concentrations of sucrose solutions used in these experiments were 0%, 11.5%, 20.7%, 27.0%, 39.0%, 46.0%, and 55.4% at steam pressures of 10, 15, 20, 25, and 30 p.s.i.g. These concentrations changed the viscosity from 0.2838 cp. to 2.730 cp. The tabulated data are given in Table I.

CORRELATION OF DATA

McAdams (14) gave a detailed report on the methods of treating heat transfer data and recommended that Q/θ or h_L should be plotted vs. ΔT_L. This method was used for representing the data.

The plots of log Q/θ vs. log ΔT_L (Figure 7) for the runs show the values at low temperature difference to fall below the best straight line. The plots of log h_L vs. log ΔT_L (Figure 8) show the same characteristics as those of log Q/θ vs. log ΔT_L. Thus it can be seen that the liquid circulation has an effect upon the heat transfer in natural circulation. Aubrecht (1) used an exponential factor for the mass velocity G to produce a linear relation between $h_L \cdot \frac{1}{G^n}$ vs. ΔT_L. The exponent 0.2 was chosen by trial and error to represent the present data. The graph of $\frac{h_L}{G^{0.2}}$ vs. ΔT_L is shown in Figure 9. At a constant value of ΔT_L of 20° F., values of $\frac{h_L}{G^{0.2}}$ were read which were proportional to the constant of the equation of each line.

The three variable properties of the fluid, i.e., thermal conductivity, specific heat, and viscosity, were effected by using solutions of different concentrations of sucrose. Since these properties could not be changed independently, a method for finding the relative effect of each was determined. Much of the tube was non-boiling and heat transfer was governed by the Dittus-Boelter equation, so this equation provided the method of attack. By combining the exponents of k, C_p, and μ from the three dimensionless groups in the equation, the effect of these properties was calculated to be:

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

where k = thermal conductivity, Btu/ft^2/$^\circ$F/ft/hr.

C_p = specific heat, Btu/lb$^\circ$F.

μ = viscosity, ft/lb-hr.

The thermal conductivity and specific heat data were taken from the International Critical Tables (10) and the viscosity data were those of Bingham and Jackson (3).

The plot of $\frac{h_L}{G^{0.2}}$ vs. $\frac{k^{0.6}\,C_p^{0.4}}{\mu^{0.4}}$ did not result in a linear relation. The plot of $\log \frac{h_L}{G^{0.2}}$ vs. $\log \frac{k^{0.6}\,C_p^{0.4}}{\mu^{0.4}}$ did not result in a linear relation. The plot of $\frac{G^{0.2}}{h_L}$ vs. $\frac{k^{0.6}\,C_p^{0.4}}{\mu^{0.4}}$ did result in a straight line with a negative slope, as shown in Figure 10. This method was adopted because it represented the data in a satisfactory manner.

In determining the relationship between $\frac{G^{0.2}}{h_L}$ and $\frac{k^{0.6}\,C_p^{0.4}}{\mu^{0.4}}$, the absolute value of $\frac{G^{0.2}}{h_L}$ was not important since the value of $\frac{h_L}{G^{0.2}}$ could have been taken at any constant ΔT_L from the plot of $\frac{h_L}{G^{0.2}}$ vs. ΔT_L. If the equations of the lines are expressed by the two point analytical equation as:

$$\frac{G^{0.2}}{h_L} = \beta \left(1.0 - a\,\frac{k^{0.6}\,C_p^{0.4}}{0.4}\right)$$

where β = proportionality constant

a = constant

The values of <u>a</u> will be the same. By choosing any two points on the plot, the value of <u>a</u> was found to be 1.10.

The values of $\frac{h_L}{G^{0.2}}\left(1.0 - 1.10\ \frac{k^{0.6}\ Cp^{0.4}}{\mu^{0.4}}\right)$ were plotted vs. ΔT_L in Figure 11 to find the final correlation equation. The final correlation of the data was:

$$\frac{h_L}{G^{0.2}}\left(1.0 - 1.1\ \frac{k^{0.6}\ Cp^{0.4}}{\mu^{0.4}}\right) = 5.18\ \Delta T_L^{\ 1.075}$$

with the points $\pm$ 25% of the best linear relation.

An examination of the literature shows that the correlation of this work compares favorably with those of other workers.

Brooks and Badger (5) have a correlation in the form of a Dittus-Boelter equation which holds for liquid velocities above 5 ft. per second and has a spread of data of $\pm$ 20% from the line.

Boarts, Badger and Meisenberg (4) correlate the overall boiling coefficients vs. ΔT_L with 80% of the data within $\pm$ 20% of the best straight line.

Foust, Badger and Baker (9) correlated liquid velocities and heat transfer coefficients vs. the submergence ratio with the data being within $\pm$ 20% of the line drawn.

Cessna, Lientz, and Badger (6) gave a correlation of the non-boiling heat transfer coefficients with the experimental data being $\pm$ 30% from the calculated values.

DISCUSSION

The inclined tube section of the evaporator was set at an angle of 30° to the horizontal. This angle had previously been determined the optimum angle of operation by Steffee (20).

The main purpose of this investigation was to find the effect of viscosity on the liquid film heat transfer coefficient. Since other properties of the solution were changed, it was necessary to consider thermal conductivity and specific heat. The most feasible method of changing these properties was by using different concentrations of a solute in distilled water.

Several solutes were studied, for example, glycerol, but due to high boiling point elevations with increase of concentration, were not acceptable. The solute chosen was sucrose because the boiling point elevation was only 4° F. at the maximum. The viscosity was changed by a factor greater than 9, while the other properties, thermal conductivity and specific heat, were changed by less than 50%.

Figure 7 shows the plot of log Q/θ vs. log ΔT_L. Figure 8 shows the plot of log h_L vs. log ΔT_L. Previous investigations (1,6) found that the data was best represented by straight parallel lines. Thus, in the present work, the best straight line was drawn through the points parallel to the curve for water. Figure 7 and 8 show the values of Q/θ and h_L fall below the line at the lower temperature differences. The deviations can be explained by the effect of velocity upon the heat transfer.

At the lower temperature differences, the natural circulation rate was small. At the higher temperature differences both the natural circulation and the heat transfer were increased. These increases are

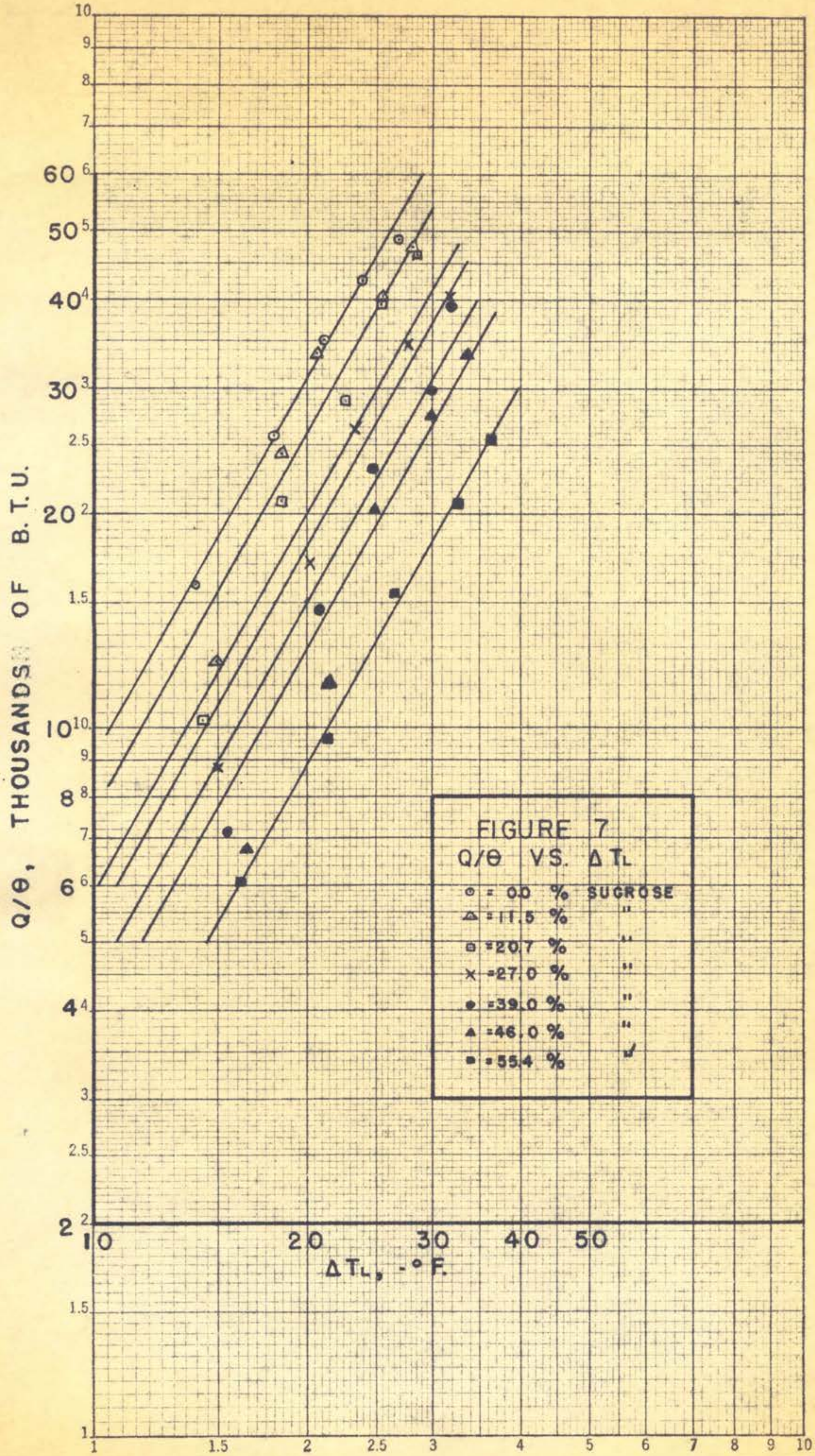


FIGURE 7
Q/Θ VS. ΔTL
⊙ = 0.0 % SUCROSE
△ = 11.5 % "
⊡ = 20.7 % "
× = 27.0 % "
● = 39.0 % "
▲ = 46.0 % "
■ = 55.4 % "
Q/Θ, THOUSANDS OF B. T. U.
ΔTL, - ° F.
10
20
30
40
50
60

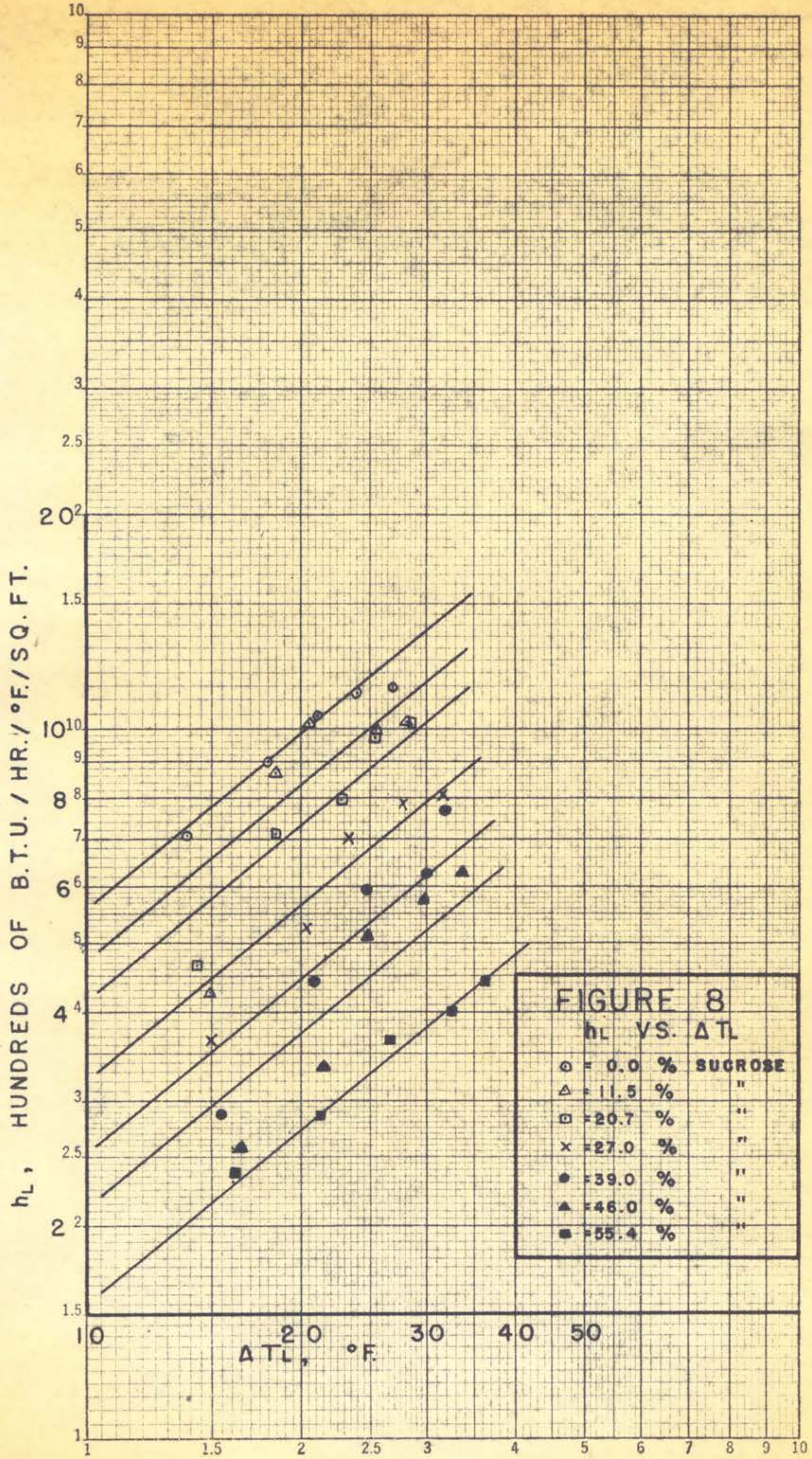


FIGURE 8
h_L VS. ΔT_L
⊙ = 0.0 % SUCROSE
△ = 11.5 % "
⊡ = 20.7 % "
× = 27.0 % "
● = 39.0 % "
▲ = 46.0 % "
■ = 55.4 % "
h_L , HUNDREDS OF B.T.U. / HR. / °F. / SQ. FT.
ΔT_L, °F.
10
20
30
40
50
2
4
6
8
10
20

generally considered to be due to an increase in agitation caused by more vigorous evolution of bubbles. The liquid circulation of the sucrose solutions did not differ much from that of water, although the viscosity of the sucrose solutions was greater than 9 times that of water at that temperature.

The second factor mentioned above was the effect of the boiling characteristics of the liquid upon the heat transfer. Badger, Boarts (4,5) et. al.(6), reported that the liquid is first warmed without evaporation as it flows through the tube until it reaches a section where boiling starts. Thus, in this evaporator, there are two zones of heat transfer: a zone where heat was transferred by a non-boiling mechanism; and a zone where heat was transferred by a boiling mechanism.

The effect of the liquid circulation, the mass velocity G, is shown in Figure 9. The exponent 0.2 was chosen for G so that the plot of log $h_L \cdot \frac{1}{G^{0.2}}$ vs. log ΔT_L would produce a linear relation with a minimum deviation of points. At a constant value of ΔT_L of 20° F., values of $\frac{h_L}{G^{0.2}}$ were read which were proportional to the constant of the equation of each line.

The relative effect of the three changing properties was found to be:

$$\frac{k^{0.6} \; C_p^{0.4}}{\mu^{0.4}}$$

with consistent units.

Figure 10 shows the plot of $\frac{G^{0.2}}{h_L}$ vs. $\frac{k^{0.6} \; C_p^{0.4}}{\mu^{0.4}}$. The absolute value of $\frac{G^{0.2}}{h_L}$ was not important since the value of $\frac{h_L}{G^{0.2}}$ could have been

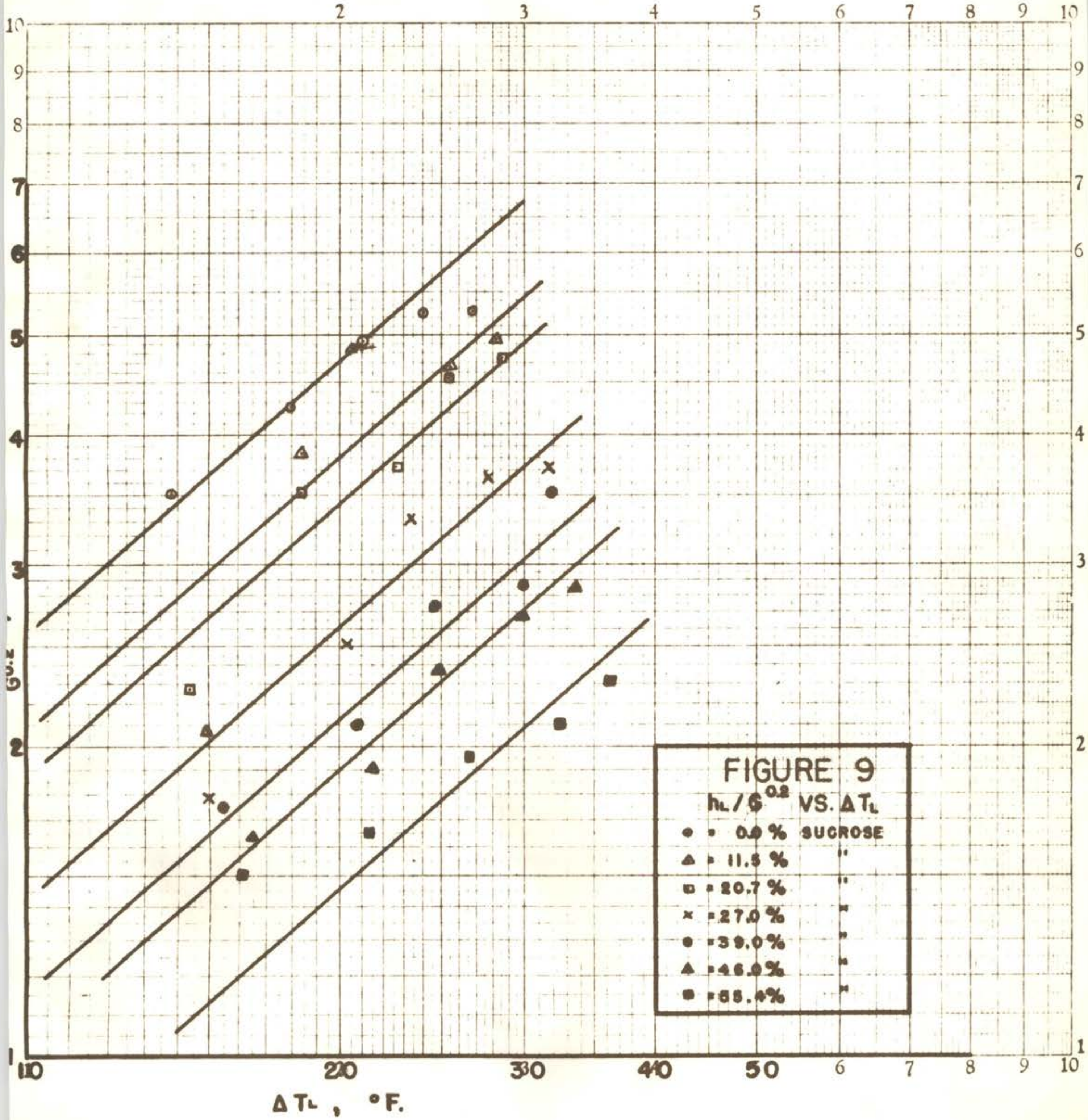


FIGURE 9
$h_L / G^{0.2}$ VS. ΔT_L
● = 0.0 % SUCROSE
△ = 11.5 % "
⊡ = 20.7 % "
× = 27.0 % "
● = 39.0 % "
▲ = 46.0 % "
■ = 55.4 % "
ΔT_L , °F.
$G^{0.2}$

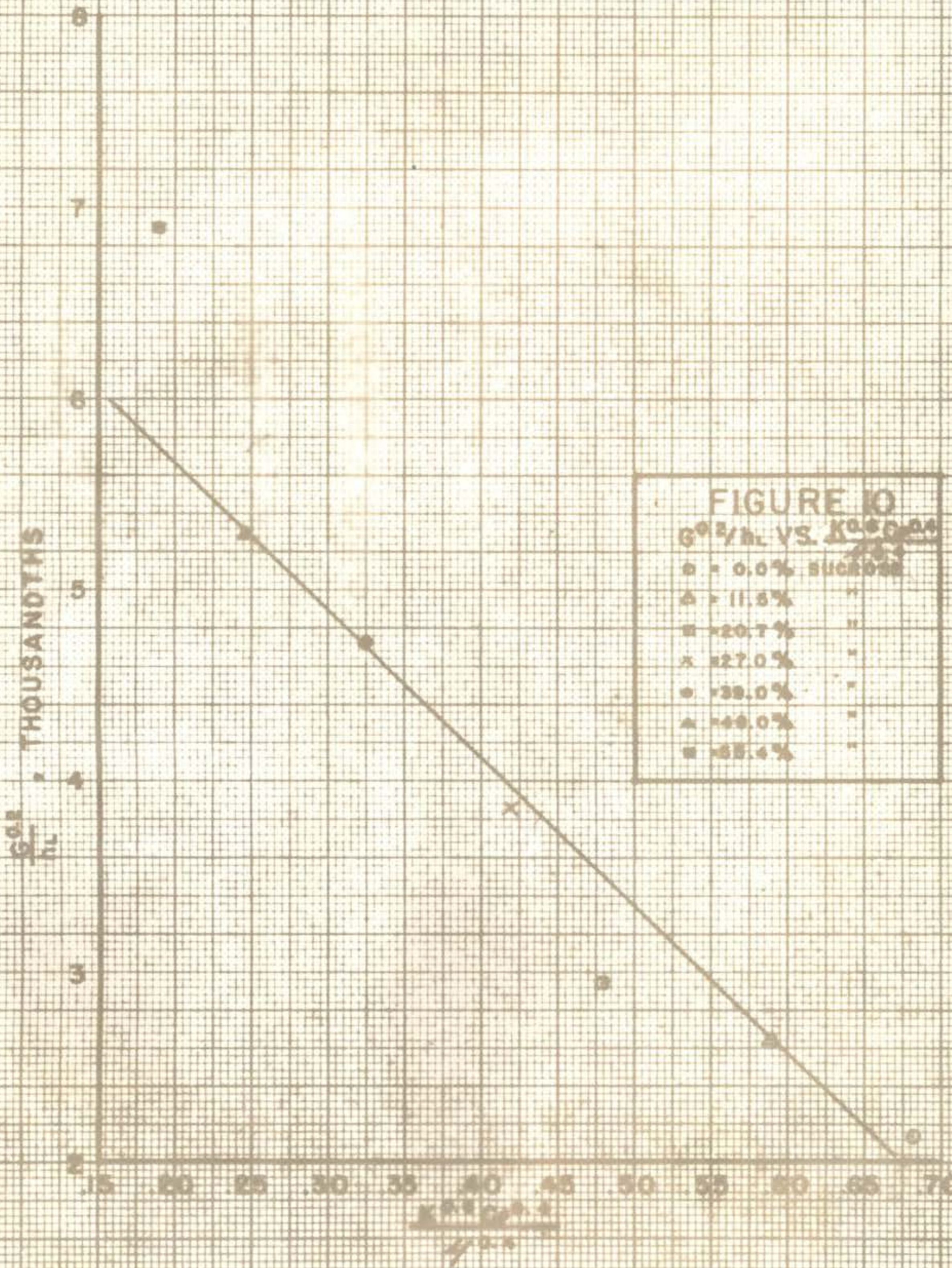


FIGURE 10
= 0.0% SUCROSE
= 11.5%
= 20.7%
= 27.0%
THOUSANDTHS
8
7
6
5
4
3
2
.15
.20
.25
.30
.35
.40
.45
.50
.55
.60
.65
.70

taken at any constant ΔT_L from Figure 9. The equation was expressed by the general equation as:

$$\frac{G^{0.2}}{h_L} = \beta \left(1.0 - a \frac{k^{0.6} c_p^{0.4}}{\mu^{0.4}}\right)$$

β = proportionality constant

a = constant

with the constant being 1.10.

The final correlation plot is shown in Figure 11. The values of $\frac{h_L}{G^{0.2}} \left(1.0 - 1.10 \frac{k^{0.6} c_p^{0.4}}{\mu^{0.4}}\right)$ were plotted vs. ΔT_L. The final correlation equation was found to be:

$$\frac{h_L}{G^{0.2}} \left(1.0 - 1.10 \frac{k^{0.6} c_p^{0.4}}{\mu^{0.4}}\right) = 5.18 \quad \Delta T_L^{1.075}$$

with the experimental data within $\pm$ 25% of the best straight line.

An examination of the literature showed that this correlation compares favorably with the correlations of other workers.

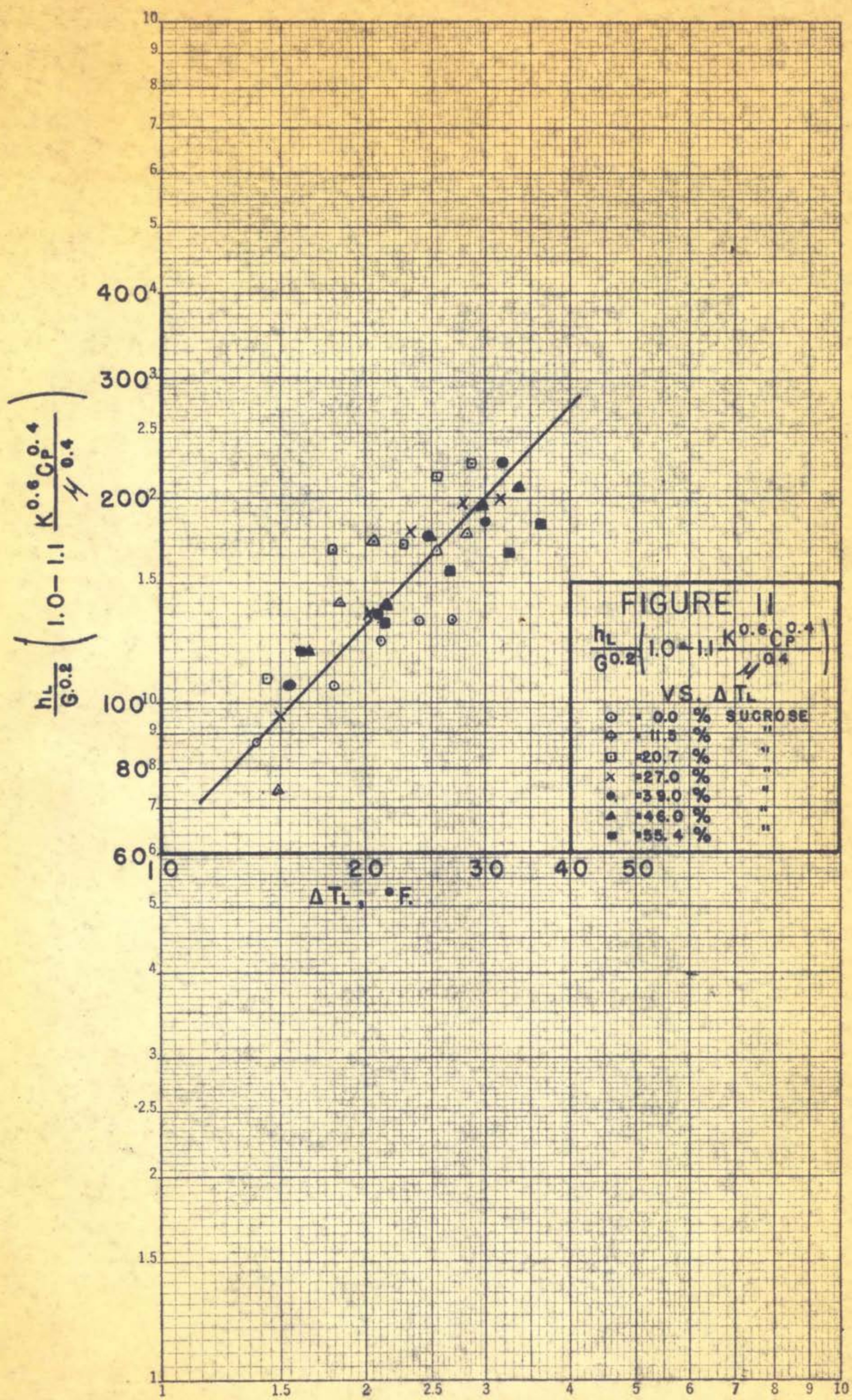


FIGURE II
$\frac{h_L}{G^{0.2}}\left(1.0 - 1.1\frac{K^{0.6}C_P^{0.4}}{\mu^{0.4}}\right)$
VS. ΔT_L
○ = 0.0 % SUCROSE
= 11.5 % "
= 20.7 % "
× = 27.0 % "
● = 39.0 % "
▲ = 46.0 % "
■ = 55.4 % "
ΔT_L, °F.
$\frac{h_L}{G^{0.2}}\left(1.0 - 1.1\frac{K^{0.6}C_P^{0.4}}{\mu^{0.4}}\right)$
10
20
30
40
50
60
80
100
200
300
400

CONCLUSIONS

1. The liquid film heat transfer coefficients increase with increase of temperature difference.
2. When the viscosity of a liquid is increased, the liquid film coefficients decrease.
3. The data for a natural circulation inclined tube evaporator were correlated by the following equation:

$$\frac{h_L}{G^{0.2}}\left(1.0 - 1.10\,\frac{k^{0.6}\,c_p^{0.4}}{\mu^{0.4}}\right) = 5.18 \quad \Delta T_L^{1.075}$$

SUGGESTIONS FOR FURTHER WORK

1. Continue the study of the effect of viscosity on the liquid film heat transfer coefficient at different angles of inclination.
2. Study the effect of tube length with varying properties of the liquid.
3. Investigate effect of surface tension on the liquid film coefficient.
4. Study effect of tube diameter using the above recommendations.
5. Consider the above at a greater change of ΔT_L and at different boiling temperatures.

APPENDIX

NOMENCLATURE

A_1 = inside area of tube, ft.^{2}.

A_2 = outside area of tube, ft.^{2}.

A_3 = cross-sectional area of tube, ft.^{2}.

A_4 = log mean average area of tube wall, ft.^{2}

C_p = specific heat of liquid, Btu/lb/^oF.

G = mass velocity of liquid in lb/ft^2/sec.

h_L = liquid film coefficient, Btu/hr/^oF/ft^2

k = thermal conductivity, Btu/ft^2/^oF/hr/ft.

l = heat necessary to heat feed water to the boiling temperature, Btu/lb.

Q = Total heat transmitted, Btu

Q/Θ = Total heat transmitted, Btu/hr.

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

ΔT_1 = total temperature difference, ^oF.

ΔT_L = liquid film temperature difference, ^oF.

ΔT_p = tube wall temperature difference, ^oF.

v = natural circulation velocity in G.P.M.

Θ = time of experiment, hours.

μ = viscosity of liquid, lb/ft-hr.

λ = latent heat of steam, Btu/lb.

l_1 = λ times pounds water evaporated per hour, Btu/hr.

TABLE I

Experimental Data

Run No.	Conc. Soln. % Sucrose	Steam Press. Psig	Tube Temp. °F.	Liquid Temp. °F.	Feed Water Temp. °F.	Water Evap/hr. Lbs.	Circulation Rate G.P.M.
1	0.0	10	233.3	210.8	83	14.50	1.75
2	0.0	15	242.9	210.8	84	23.75	1.92
3	0.0	20	251.2	210.8	84	32.00	1.95
4	0.0	25	258.1	210.8	84	39.00	1.98
5	0.0	30	264.1	210.8	83	44.25	1.98
6	11.5	10	231.6	210.9	95	11.50	1.60
7	11.5	15	242.6	210.9	96	22.50	1.85
8	11.5	20	244.9	210.9	94	31.00	1.87
9	11.5	25	258.5	210.9	90	37.00	1.91
10	11.5	30	265.1	210.9	88	43.50	1.95
11	20.7	10	231.1	211.1	83	9.50	1.50
12	20.7	15	240.9	211.1	85	19.00	1.75
13	20.7	20	249.5	211.1	85	26.25	1.85
14	20.7	25	258.0	211.1	85	36.00	1.92
15	20.7	30	265.1	211.1	85	42.50	1.95
16	27.0	10	231.3	211.5	81	8.00	1.45
17	27.0	15	241.1	211.5	82	15.50	1.65
18	27.0	20	249.3	211.5	84	24.00	1.79
19	27.0	25	258.1	211.5	83	31.50	1.90
20	27.0	30	265.3	211.5	83	37.00	1.95
21	39.0	10	231.6	212.2	79	6.50	.463
22	39.0	15	241.0	212.2	77	13.25	1.55
23	39.0	20	249.5	212.2	80	21.00	1.75
24	39.0	25	258.9	212.2	78	27.00	1.89
25	39.0	30	265.5	212.2	78	35.50	1.92
26	46.0	10	233.9	213.7	78	6.10	.366
27	46.0	15	241.5	213.7	80	10.50	.640
28	46.0	20	249.7	213.7	82	18.50	1.73
29	46.0	25	258.6	213.7	83	25.00	1.79
30	46.0	30	265.7	213.7	83	30.50	1.89
31	55.4	10	234.5	215.1	76	5.50	.356
32	55.4	15	241.7	215.1	77	8.75	.537
33	55.4	20	250.1	215.1	79	14.00	.805
34	55.4	25	258.9	215.1	80	18.75	.895
35	55.4	30	265.1	215.1	80	23.00	.900

TABLE II

Results

Run No.	Conc. Soln. % Sucrose	Q/θ Btu/hr.	ΔT_1 °F	ΔT_p °F	ΔT_L °F	h_L	G	X*	$\frac{h_L}{G^{0.2}}$ X
1	0.0	15,910	22.5	8.7	13.8	726	39.0	.250	87.7
2	0.0	25,800	32.1	14.1	18.0	902	42.2	.250	106.8
3	0.0	35,100	40.4	19.3	21.1	1047	43.4	.250	123.3
4	0.0	42,750	47.3	23.3	24.0	1121	44.1	.250	131.4
5	0.0	48,600	53.3	26.5	26.8	1142	44.1	.250	132.0
6	11.5	12,490	20.7	5.8	14.9	427	37.3	.358	74.1
7	11.5	24,410	31.7	13.3	18.4	834	43.1	.358	140.5
8	11.5	33,700	39.0	18.4	20.6	1029	43.6	.358	173.0
9	11.5	40,900	47.6	22.1	25.5	998	44.5	.358	167.2
10	11.5	47,550	54.2	26.0	28.2	1061	45.5	.358	177.5
11	20.7	10,390	20.1	5.8	14.3	467	36.3	.475	108.5
12	20.7	20,850	29.8	11.4	18.4	712	42.4	.475	168.0
13	20.7	28,780	38.4	15.7	22.7	794	44.8	.475	171.5
14	20.7	39,450	46.9	21.5	25.4	977	46.5	.475	215.0
15	20.7	46,550	54.0	25.4	28.6	1023	47.2	.475	225.0
16	27.0	8,808	19.8	4.8	15.0	364	35.9	.538	95.9
17	27.0	17,050	29.6	9.3	20.3	528	40.8	.538	135.0
18	27.0	26,350	37.8	14.4	23.4	707	44.3	.538	178.0
19	27.0	34,600	46.6	18.9	27.7	786	47.0	.538	196.0
20	27.0	40,600	53.8	22.1	31.7	808	48.2	.538	200.0
21	39.0	7,170	19.4	3.9	15.5	288	12.15	.643	112.7
22	39.0	14,640	28.8	8.0	20.8	442	41.2	.643	135.2
23	39.0	23,150	37.3	12.6	24.7	590	46.4	.643	176.4
24	39.0	29,810	46.2	16.2	30.0	626	50.2	.643	184.0
25	39.0	39,180	53.3	21.4	31.9	771	50.9	.643	226.0
26	46.0	6,740	20.2	3.7	16.5	257	9.84	.729	119.0
27	46.0	11,580	27.8	6.3	21.5	338	17.2	.729	139.2
28	46.0	20,350	36.0	11.1	24.9	513	46.5	.729	173.6
29	46.0	27,500	44.9	15.0	29.9	578	48.1	.729	195.0
30	46.0	33,550	52.0	18.3	33.7	626	50.8	.729	208.0
31	55.4	6,080	19.4	3.3	16.1	238	9.95	.793	119
32	55.4	9,675	26.6	5.3	21.3	286	15.05	.793	131
33	55.4	15,480	35.0	8.4	26.6	367	22.50	.793	156
34	55.4	20,700	43.8	11.3	32.5	400	25.00	.793	167
35	55.4	25,400	50.0	13.8	36.2	441	25.00	.793	184

$$^{*}X = 1 - 1.10\,\frac{k^{.6}c^{.4}}{\mu^{.4}}$$

SAMPLE CALCULATIONS

Calculations of areas A_1, A_2, A_3, A_4

Tube Dimensions:

I. D. - 1.049 inches

O. D. - 1.315 inches

Wall thickness - 0.133 inches

Effective length - 69.5 inches

$$A_1 = \frac{1.049\,\pi}{12} \times \frac{69.5}{12} = 1.59 \text{ ft}^2$$

$$A_2 = \frac{(1.315 - \frac{1}{32})\pi}{12} \times \frac{69.5}{12} = 1.90 \text{ ft}^2$$

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

$$A_4 = \frac{1.90 - 1.59}{2.3 \log \frac{1.90}{1.59}} = 1.73 \text{ ft}^2$$

Observed Data - Run #14

Sugar concentration = 20.7 %Sucrose

Steam pressure = 25 psig.

Feed Water temperature = 85.0° F.

Average liquid temperature = 211.1° F.

Average tube temperature = 258.0° F.

Circulation rate = 1.92 G.P.M.

Condensate collected = 18.0 lbs./half-hour

$$Q/\theta = \lambda_1 + \ell$$

$$\lambda_1 = 970.9 \times 36.00 = 34{,}900$$

$$\ell = 36.0 \times (211.1 - 85.0) = \underline{4{,}550}$$

39,450 Btu/hr.

ΔT_1 = Tube temperature - boiling temperature

$\Delta T_1 = 258.0 - 211.1 = 46.9° F.$

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

k = 9.0 at 212° F.[21]

$$\Delta T_p = \frac{39{,}450 \times .0085}{9.00 \times 1.73} = 21.5° F.$$

$$\Delta T_L = \Delta T_1 - \Delta T_p$$

$$\Delta T_L = 46.9 - 21.5 = 25.4° F.$$

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

$$h_L = \frac{39{,}450}{25.4 \times 1.59} = 977 \text{ Btu/hr/°F/ft}^2$$

$$G = \frac{G.P.M.}{748} \times \frac{\rho}{A_3 \times 60}$$

$$G = \frac{1.92}{7.48} \times \frac{65.1}{.006 \times 60} = 46.5 \text{ lb/ft}^2 \text{ sec.}$$

The tabulated results for all the runs are given in Table II.

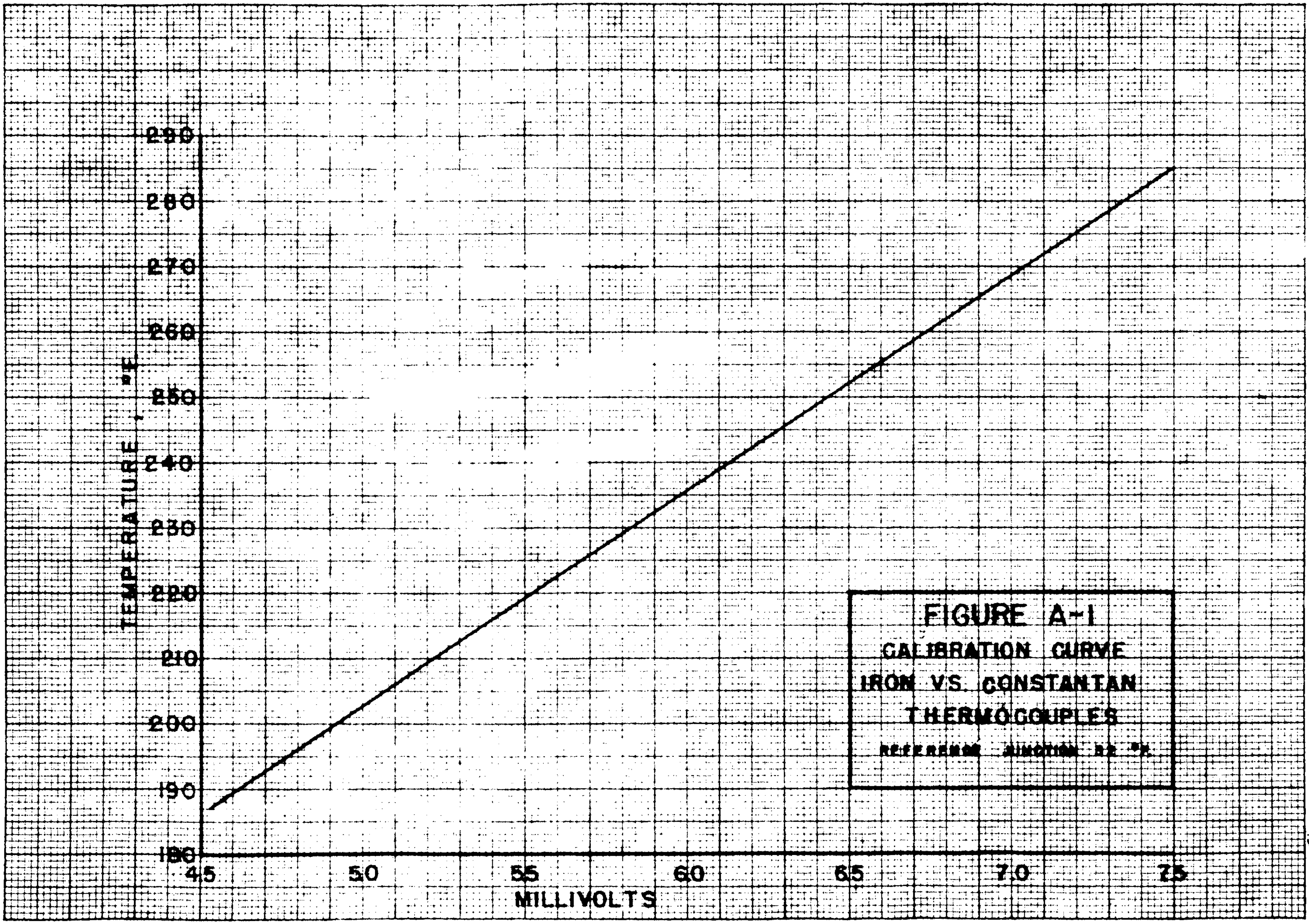


FIGURE A-1
CALIBRATION CURVE
IRON VS. CONSTANTAN
THERMOCOUPLES
REFERENCE JUNCTION 32 °F.

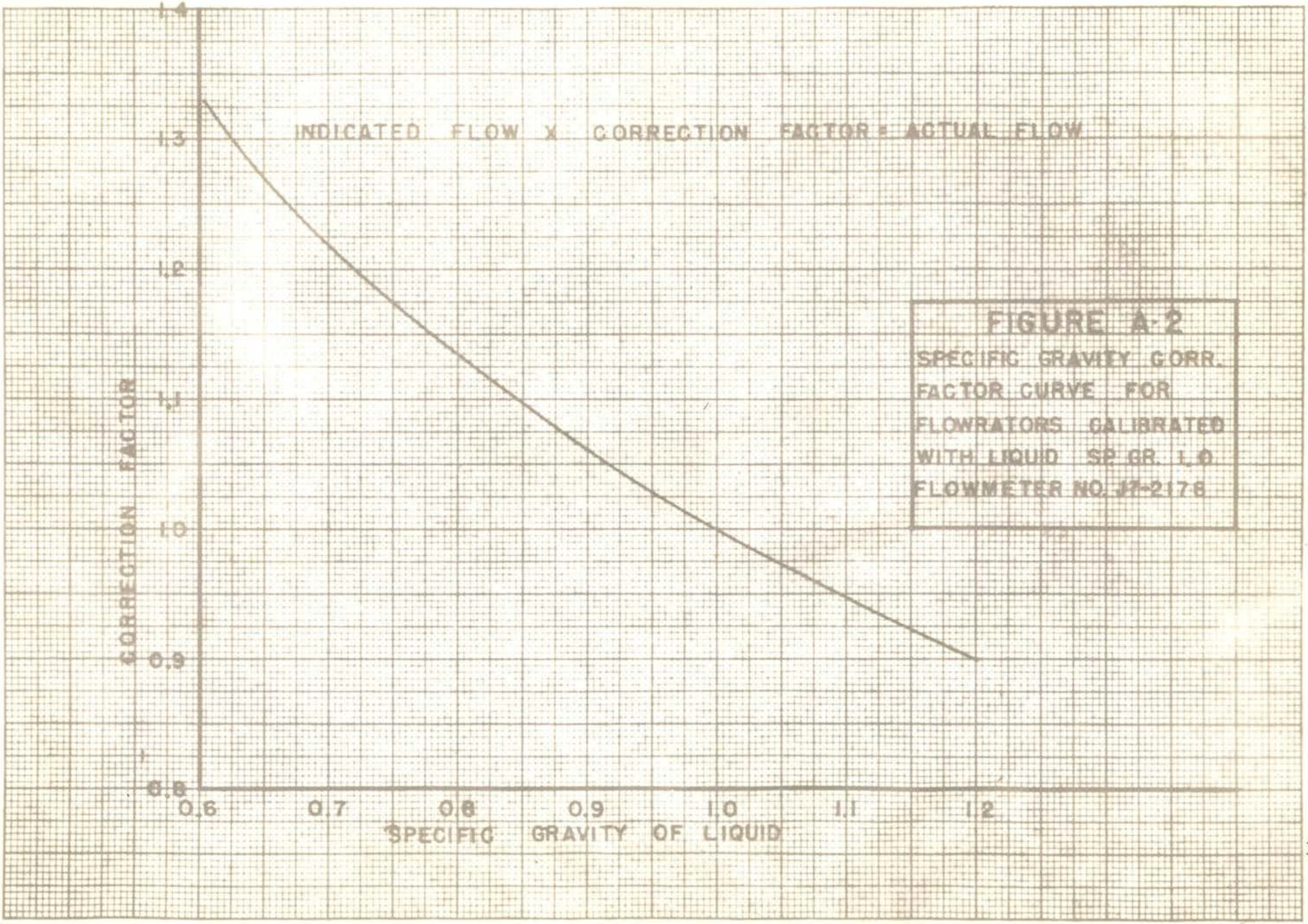
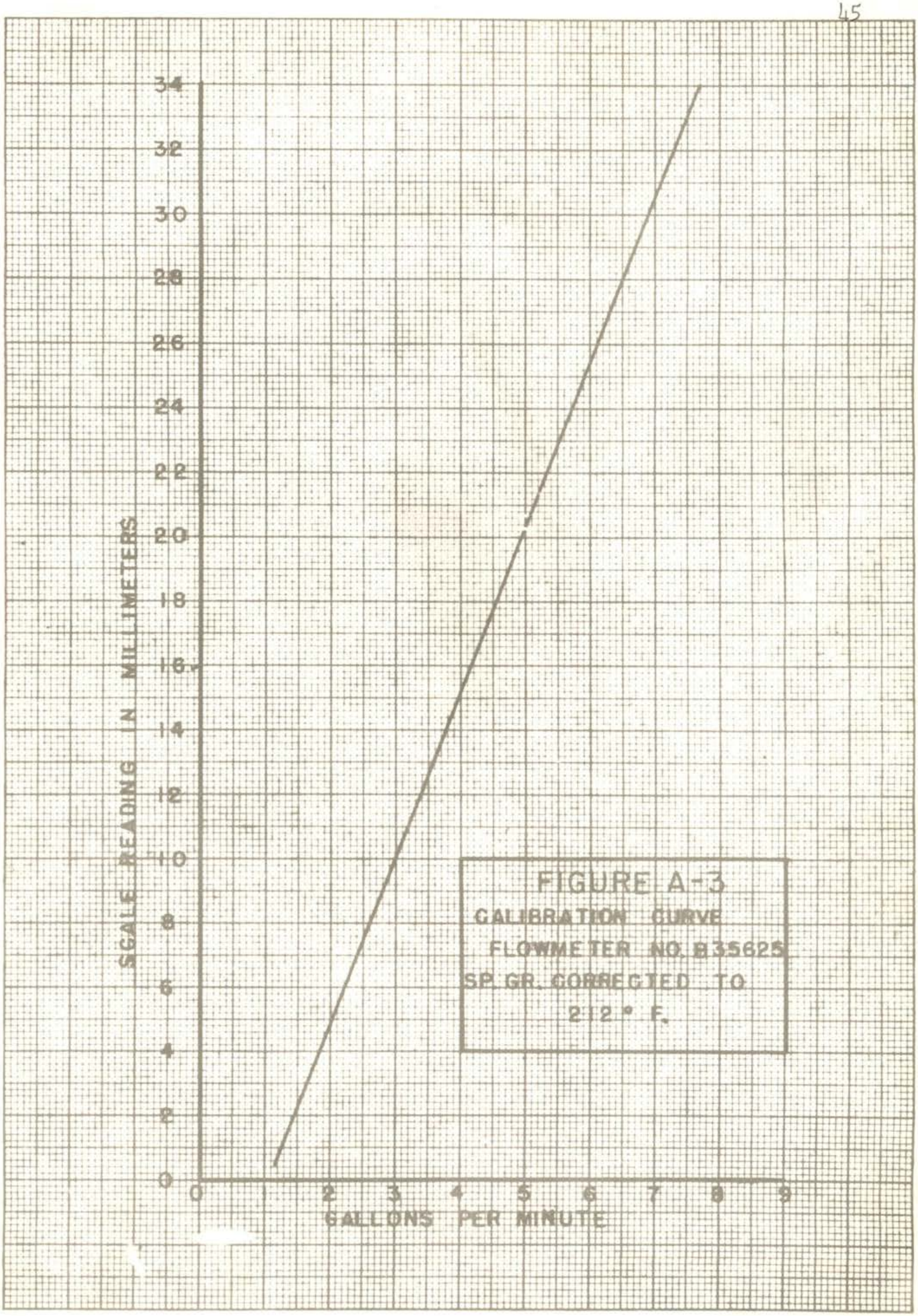


FIGURE A-2
SPECIFIC GRAVITY CORR. FACTOR CURVE FOR FLOWRATORS CALIBRATED WITH LIQUID SP. GR. 1.0
FLOWMETER NO. J7-2176


34
32
30
28
26
24
22
20
18
16
14
12
10
8
6
4
2
0
SCALE READING IN MILLIMETERS
0
2
3
4
5
6
7
8
9
GALLONS PER MINUTE
FIGURE A-3
CALIBRATION CURVE
FLOWMETER NO. B35625
SP. GR. CORRECTED TO
212 ° F.

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