

AN INVESTIGATION OF CRITICAL RATES OF HEAT TRANSFER IN AN  
INCLINED TUBE NATURAL CIRCULATION EVAPORATOR

By

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## PREFACE

This thesis is concerned with liquid film heat transfer coefficients in a natural circulation evaporator having a single stainless steel heating tube inclined at an angle of thirty degrees to the horizontal, and relates specifically to the comparison of coefficients for moderate heat transfer rates with those obtained for the highest and the lowest heat transfer rates possible with the evaporator.

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TABLE OF CONTENTS

	Page
Introduction . . . . .	1
Description of Apparatus . . . . .	5
Experimental Procedure . . . . .	.13
Calculation Methods and Results . . . . .	.16
Discussion of Results . . . . .	.21
Conclusions . . . . .	.28
Recommendations for Further Study . . . . .	.29
Appendix . . . . .	.31
Nomenclature . . . . .	.32
Series Designations . . . . .	.34
Sample Calculations . . . . .	.35
Tabulated Data . . . . .	.37
Bibliography . . . . .	.45

## LIST OF FIGURES

No.	Following Page
1. Photograph of inclined tube evaporator . . . . .	5
2. Schematic drawing of inclined tube evaporator . . . . .	6
3. Photograph of flow meter arrangement . . . . .	8
4. Comparison of three methods of calculating the liquid film temperature difference . . . . .	18
5. Logarithmic plot of liquid film heat transfer coefficients versus corrected liquid film temperature differences, atmospheric data . . . . .	20
6. Logarithmic plot of liquid film heat transfer coefficients versus corrected liquid film temperature differences, data for 1/2 atmosphere . . . . .	20
7. Logarithmic plot of heat transfer rate versus corrected temperature difference between thermocouple and liquid, atmospheric data . . . . .	20
8. Rectangular plot of heat transfer rate versus corrected temperature difference between thermocouple and liquid, atmospheric data . . . . .	20
9. Logarithmic plot of heat transfer rate versus corrected temperature difference between thermocouple and liquid, data for 1/2 atmosphere . . . . .	20
10. Rectangular plot of heat transfer rate versus corrected temperature difference between thermocouple and liquid, data for 1/2 atmosphere . . . . .	20
11. Plot of circulation rate versus heat transfer rate . . . . .	20
12. Comparison of atmospheric data with the Dittus- Boelter equation . . . . .	20
13. Comparison of data for 1/2 atmosphere with the Dittus-Boelter equation . . . . .	20
14. Comparison of data with Jakob's equation for low-rate heating on flat surfaces . . . . .	20

15.	Comparison of data with Thorstenberg's correlation equation . . . . .	20
16.	Comparison of data with Aubrecht's correlation equation . . . . .	20
17.	Influence of inlet liquid temperature upon tube-wall temperature distribution . . . . .	21
A1.	Tube-wall temperature distribution for water runs . . . . .	44
A2.	Tube-wall temperature distribution for 40% sucrose runs . . . . .	44

## LIST OF TABLES

No.		Page
I	Experimental data for water and 10% sucrose solution . . . . .	37
II-IV	Calculated data for water and 10% sucrose solution . . . . .	38
V	Experimental data for 20, 30, and 40% sucrose solutions . . . . .	41
VI-VIII	Calculated data for 20, 30, and 40% sucrose solutions . . . . .	42



## INTRODUCTION

The purpose of this investigation was to determine what resulted when a steam-heated evaporator, having a 30°-inclined stainless steel heating tube through which liquid circulated naturally, was operated at very low and at moderately high steam pressures.

Information concerning liquids boiling inside of inclined tubes are extremely meager, the only published data being those of Van Marle<sup>22</sup>, Cleve<sup>4</sup>, and Linden and Montillon<sup>11</sup>. Because different types of information were obtained by them, their results are not comparable on a common basis.

Van Marle<sup>22</sup> investigated overall heat transfer coefficients in a seven-tube evaporator inclined at an angle of 45°. He found a straight-line relation between the overall coefficients and the temperature of the boiling liquid, but did not obtain sufficient data for a plot of overall coefficients versus the temperature differences.

Cleve<sup>4</sup> measured the liquid film coefficients and rates of circulation for a small electrically-heated tube inclined at angles of 10, 20, and 45°. For each angle of inclination, he found that the velocity of the liquid increased with an increase in the rate of heat input until a maximum was reached, after which a further increase in heat input rate caused a decrease in the velocity.

Linden and Montillon<sup>11</sup> obtained data with a small inclined tube evaporator so constructed that the velocity of liquid in the downtake pipe could be measured. They made runs with water boiling at 180, 195, and 210° F., and were able to correlate their data by means of a modified

Dittus-Boelter equation, in which the logarithmic mean of the liquid and vapor velocities was used for the mass flow rate. The use of logarithmic mean for this purpose was selected arbitrarily.

Aubrecht<sup>2</sup> used a one-inch externally steam-heated copper tube inclined at 45° and measured the rate of natural circulation as well as the rate of heat transfer. Like Cleve, he found that increases in the heat load caused the liquid circulation rate to increase to a maximum, then decrease. He reported that a plot of heat load versus liquid film temperature difference made a slight break at the heat load corresponding to the maximum rate of natural circulation. Using concentrations of sucrose solutions ranging from 0 to 30%, and considering the effect of liquid properties, he correlated the liquid film temperature difference, the heat load, and the circulation rate by means of an equation which described the results of his experiments with a maximum deviation of 20%.

Obert<sup>15</sup>, using a stainless steel heating tube in an evaporator similar to that of Aubrecht, investigated the effect of scale formation on the heat transfer coefficient. He used only one temperature difference, and calculated his coefficients by taking the liquid temperature as the average of the inlet and outlet liquid temperatures.

Steffee<sup>20</sup>, using a modification of the evaporator of Obert, studied the effect of the angle of inclination of the heating tube upon the rate of heat transfer. Operating at angles of 15, 30, 45, 60, 75, and 90°, he found that higher coefficients were obtained from the 30° inclination of the tube. Steam pressures used by him ranged from 5 to 30 psig., and the liquid studied was water under atmospheric pressure. By assuming that the recirculating liquid entered the heating tube at its boiling

point he calculated liquid film temperature differences of from  $6.2^{\circ}$  F. with five-pound steam pressure to  $17.4^{\circ}$  with thirty-pound steam, using a  $30^{\circ}$  tube inclination in both cases. For these runs the liquid film heat transfer coefficients were calculated to be, respectively, 381 and 1776 Btu/hr-ft<sup>2</sup>-°F.

Thorstenberg<sup>21</sup>, using the same evaporator as did Steffee, studied the effect of viscosity upon the liquid film heat transfer coefficient. He used sucrose concentrations ranging from 0 to about 55% in weight, steam pressures of 10 to 30 psig., and a  $30^{\circ}$  angle of inclination for the heating tube. His temperature differences, and consequently his coefficients, were, like Steffee's, calculated on the assumption that the liquid temperature was its boiling point. He presented a correlation which was based on an assumption that a logarithmic plot of the temperature difference versus the film coefficient yielded straight lines, which for sucrose solutions were parallel to the line for water.

None of the previous workers on inclined tubes appear to have investigated operation at steam pressures higher than 30 nor lower than 5 psig. to check for a departure from a straight-line relation between the coefficient and the temperature difference, though a considerable volume of such work has been carried out with liquids boiling outside of tubes or on vertical or horizontal plates.

Although it had been known for many years that vapor binding occurred when a liquid touched a very hot surface, it was only comparatively recently that this phenomenon was shown to occur at moderately low values for the difference between the temperatures of the liquid and the heated surface. Apparently the first to appreciate this fact was Nukiyama<sup>14</sup>, who in 1934 published a "boiling curve", obtained by heating

water with an electrically heated wire, which showed that the rate of heat transfer increased with the temperature difference at moderately low values, reached a maximum, decreased, then began to increase at higher values of the temperature difference. This type of curve has since been verified by a number of investigators, using various liquids and heating surfaces <sup>1</sup>, 3, 6, 7, 13, 17, 18.

It now seems clear that there are four distinct sections of the "boiling curve". As the temperature difference increases from zero the rate of heat transfer increases slowly and the boiling rate is so small that the flux is of the same order of magnitude as that calculated by means of equations for heat transfer by natural convection of a non-boiling liquid. As the temperature difference increases, the rate of boiling increases so much that agitation by the vapor bubbles causes a much higher rate of heat transfer, resulting in a rather large slope of the curve in this section. Eventually vapor bubbles are formed with such rapidity that the heating surface becomes surrounded by a steam film which, because of its low thermal conductivity, retards the rate of heat transfer. As a consequence of this the rate of heat transfer passes through a maximum and begins to decrease. This decreasing heat transfer rate continues until the temperature difference becomes so large that radiation becomes appreciable, at which time the rate again starts increasing.

In the present work experiments were conducted with the idea of determining whether normal operation of the natural circulation evaporator would give results similar to the boiling curve of Nukiyama.

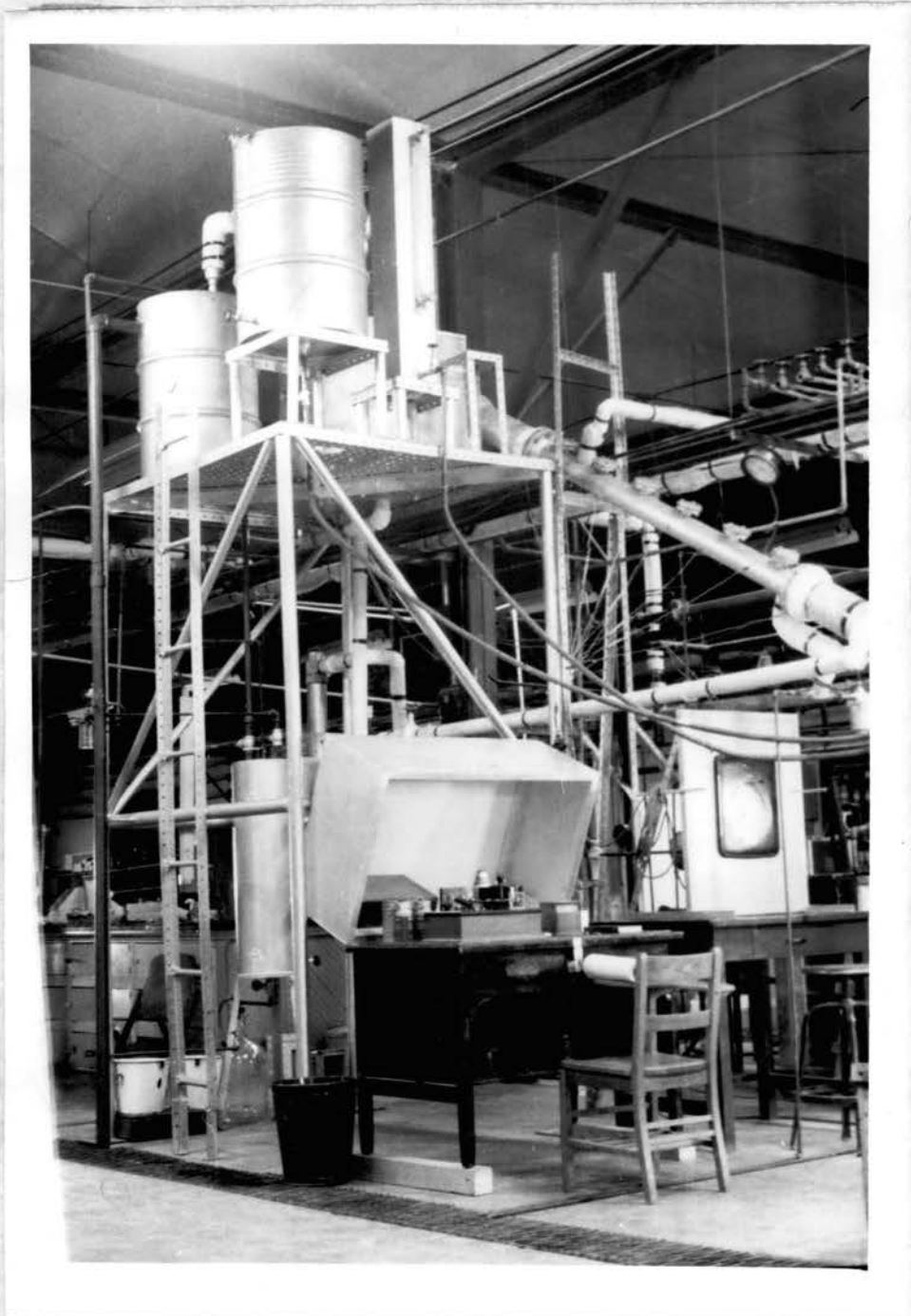


FIGURE 1. INCLINED TUBE EVAPORATOR.

the gaskets. Drawings of this assembly were shown by Obert<sup>15</sup>, Steffee<sup>20</sup>, and Thorstenberg<sup>21</sup>.

The heating tube was held in place in the steam chest by a packing gland at each end of the chest. These glands were made by combining a 1 1/2-inch coupling and nipple. A 1/2-inch steel plate with a hole in its center slightly larger than the outside diameter of the heating tube was welded to each end flange of the chest. The 1 1/2-inch coupling was then centered around this hole and welded to the plate. The heating tube was then inserted in the chest by sliding it through the holes in the end plates. Steel washers and 1/4-inch asbestos-graphite rope-type packing were placed between the couplings and the protruding ends of the heating tube, after which the 1 1/2-inch nipple was tightened in place in the coupling. Drawings of this arrangement have also been shown by Obert<sup>15</sup>, Steffee<sup>20</sup>, and Thorstenberg<sup>21</sup>. These packing glands held the heating tube in place very nicely without leakage of steam, but had the disadvantage of complicating the removal and replacement of the tube.

Steam was introduced into the chest through a one-inch pipe connection placed 12 in. from the top of the chest, midway between the two top thermocouple outlets. A Bourdon-type gage was connected to the chest by means of a 1/4-inch threaded hole which was located midway between the bottom two thermocouple outlets. The condensate outlet was a one-inch coupling welded into the bottom of the steam chest next to the bottom flange.

A twelve-inch section of two-inch pipe was attached to the condensate outlet by means of a nipple and a reducing elbow to provide a space for the temperature element of an automatic temperature control instrument. A two-inch tee on the end of the twelve-inch section permitted mounting the

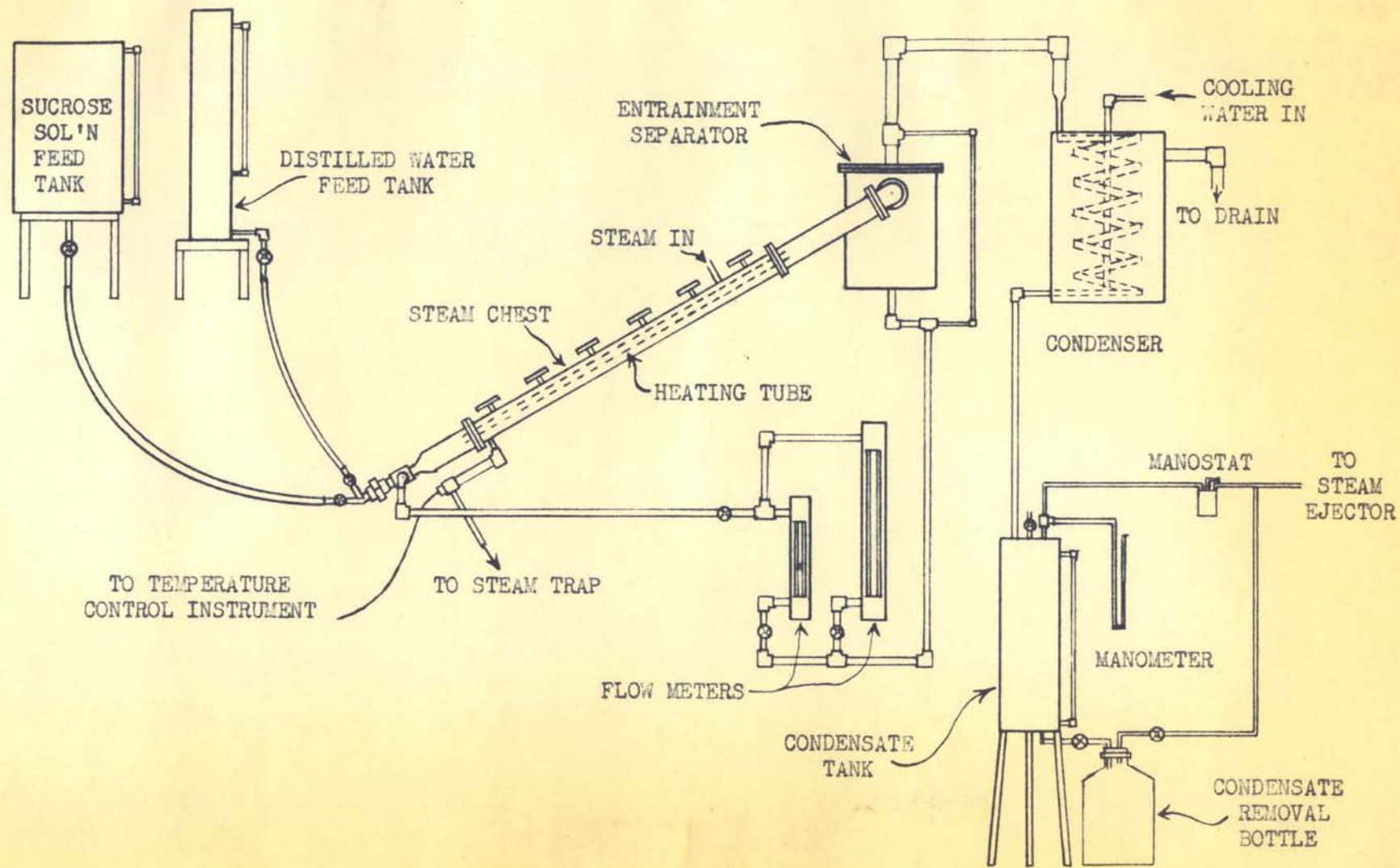


FIGURE 2. SCHEMATIC DRAWING OF INCLINED TUBE EVAPORATOR

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13-inch temperature element so that it would not touch the pipe. A 1/2-inch pipe led from the leg of the tee to an Anderson No. 21 bucket-type steam trap. This pipe contained a 1/4-inch vent valve which was left partially open during each run to bleed off non-condensable gases.

Steam was supplied by the central heating plant at pressures ranging from 120 to 150 psig. The steam pressure in the chest was controlled and its temperature was recorded by means of a Foxboro Model 40 Stabilog temperature controller having a range of 0 to 500° F., and a 3/4-inch Foxboro Stabilflo valve. Compressed air at 17 psig. was supplied to the instrument through a reducing valve, from a source which varied between 90 and 120 psig.

Attached to the bottom of the steam chest by flange was a 14-inch section of four-inch standard iron pipe, which served as the feed pot. This section reduced at its lower end to standard two-inch pipe, to which was attached one branch of a two-inch tee. The liquid return line was connected to the leg of the tee, while the other branch was fitted with a nipple and a two-inch union, which was bushed down to a 1/2-inch tee through which feed solution and make-up water were supplied. The purpose of the union was to facilitate cleaning the heating tube.

The disengaging section was a 24-inch section of four-inch pipe attached by flange to the upper end of the steam chest. Its purpose was to separate the heating tube from the entrainment separator.

A cyclone-separator type of entrainment separator was used. The body consisted of a 24-inch section of standard 18-inch iron pipe. The bottom was made by welding on a 3/8-inch steel plate, in the center of which was a one-inch coupling for the return of liquid to the feed pot. The top of



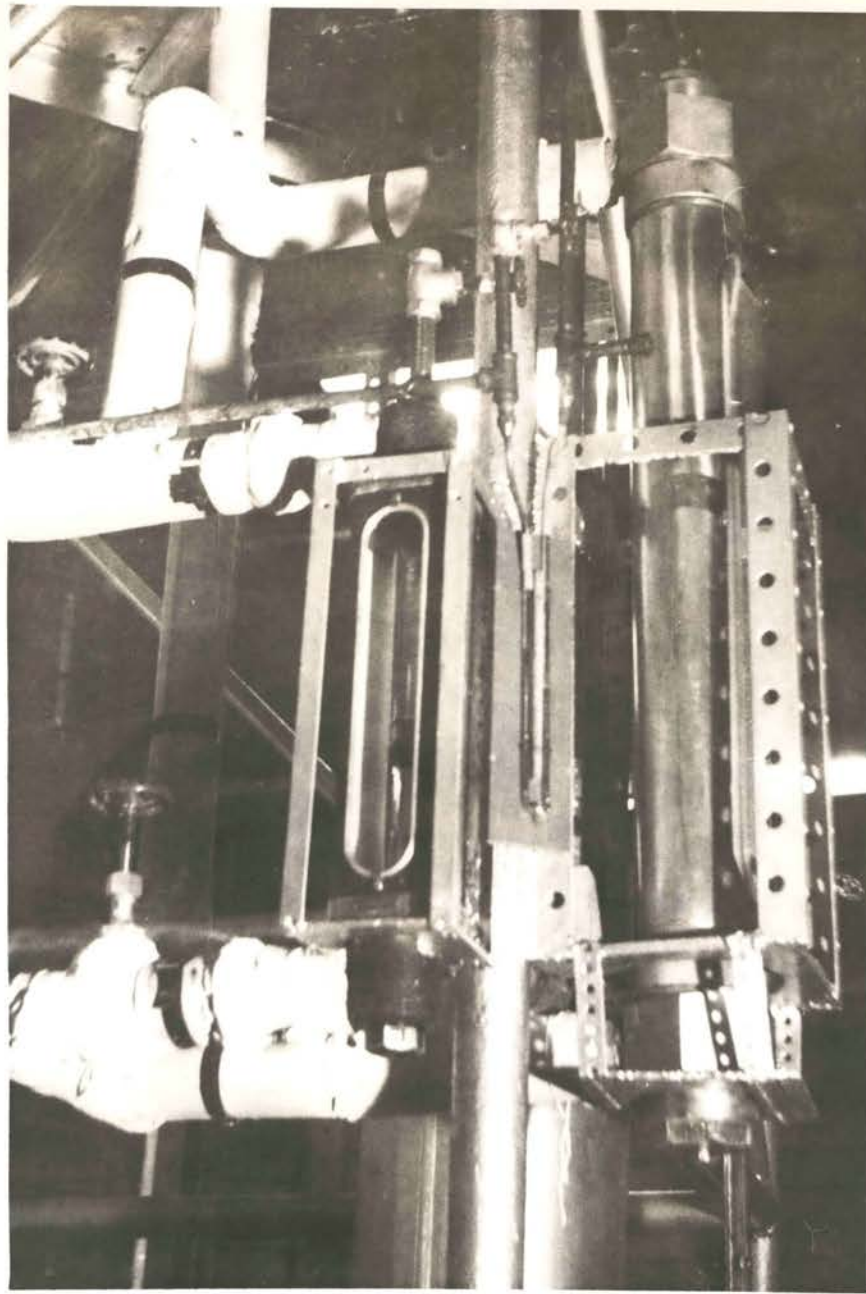


FIGURE 3. FLOWMETER ARRANGEMENT.

curve. Specific gravity correction was made by reference to the same curve used by the smaller meter. The larger meter was used for flow rates higher than approximately 1.2 G.P.M. The two flow meters were connected in parallel, with suitable valves to permit shutting off flow to the meter not needed during a run. Located at the outlet of the entrainment separator and at the inlet of the feed pot were manometer connections. A small mercury manometer, placed near the flow meters, was connected to these by means of 1/4-inch pipe.

The vapor outlet of the entrainment separator was connected to the condenser by a two-inch line which rose to a height of 27 in. above the separator. A horizontal section extended from the vertical section to the condenser and was inclined slightly downward toward the condenser to prevent any condensate from falling back into the entrainment separator.

The condenser was made of ten turns of one-inch copper tubing in a vertical coil twelve inches in diameter. This coil was encased in a 50 gal. steel drum. Cooling water was introduced at the bottom of the drum and flowed out of a two-inch overflow line near the top.

The condensate receiver was a twelve-inch diameter tank 39 in. high. The top of this tank was fitted with two standard one-inch connections and one 1/4-inch connection. One one-inch connection was used for receiving condensate from the condenser, the 1/4-inch connection was used as a vent, while the third connection led, through a Model 5 Industrial Cartesian Manostat, to the vacuum source, a steam ejector. The bottom of the tank contained a 1/2-inch line through which condensate was removed for measuring. During vacuum runs a five-gallon Pyrex glass bottle was attached to this 1/2-inch pipe by means of a two-hole No. 12

rubber stopper. A vacuum line was connected through the other hole in the stopper to permit withdrawal of condensate while the system was under a vacuum. Pressure in the system was measured during vacuum runs by means of a mercury manometer, one side of which was vented to the atmosphere. The absolute pressure was found by difference between the reading of this manometer and the reading of a barometer located nearby.

The feed reservoir for the sucrose solution was a fifty-gallon steel drum with a central outlet in the bottom. The distilled water feed reservoir was an eight-inch square tank 4 ft. high, having a 1/2-inch outlet 2 1/2 in. from the bottom. Both feed reservoirs were connected to the feed pot of the evaporator by means of 1/2-inch rubber hose and 1/2-inch gate valves, and were mounted on a stand sufficiently high to permit gravity flow from them to the evaporator. Sight glasses in the reservoirs permitted easy determination of the amount of liquid in them.

The condensate was measured by weighing it on a set of balances equipped with a weighing bucket. These balances had a capacity of eight pounds and a scale which was marked directly in thousandths of a pound.

All temperature measurements were made by the use of iron-constantan thermocouples and a Leeds and Northrup Type K potentiometer. All thermocouples were made from Leeds and Northrup #24 duplex iron-constantan wire having asbestos-woven glass insulation. Each division on the potentiometer scale represented 0.005 millivolts, making possible a close approximation to thousandths of millivolt readings. This was equivalent to a direct reading to the nearest 0.15° F. and a close estimation to the nearest 0.03° F.

The thermocouple junctions were made in the following manner: A 14-inch length of the duplex wire was cut from the spool. The two wires were separated for about an inch at each end by pushing back the woven glass insulation with the fingers. Then the asbestos on the individual wires was carefully pushed back about 1/4 inch and the exposed wires were scraped to give a shiny surface. The pair of wires was then twisted slightly to hold them together for the fusion process. The bead was made by an electric arc, which was obtained by placing the wire in an electrical circuit containing a dish of mercury covered by a film of oil. The arc resulted when the twisted wires were touched to the surface of the mercury. The beads obtained by this method seemed to be of excellent quality and had dimensions of approximately 0.04 in. in diameter and about 0.06 in. from tip to lead wires. After formation of the bead, the wire was carefully straightened and the asbestos was pushed down until it touched the bead. The completed thermocouple was then checked against a mercury thermometer which had previously been calibrated against a Bureau of Standards thermometer. Using Leeds and Northrup standard conversion tables, the couples were found to be accurate within 0.2° F.

The six thermocouples for the heating-tube wall were installed by a method approved by McAdams<sup>12</sup> and similar to the one used by Colburn and Hougen<sup>5</sup>. The thermocouple wires were placed in the groove of the tube wall in such a manner that the lead wires left the groove on the side of the pipe opposite to the junction. A small amount of asbestos fiber was placed beneath the bead to prevent shorting where the insulation had been disturbed, and to elevate the tip of the bead very slightly above

the surface of the tube wall. Thin strips of shiny lead were tapped in place around the bead to give good thermal contact between the bead and the tube wall. The excess lead was polished off flush with the tube surface, and the installation was checked to see that a small tip of the bead was visible. This accomplished, more lead was tapped into the groove to cover the lead wires, and this, too, was polished off flush with the surface. During the installation process, great care was taken to insure that the bead and the lead wire insulation were not damaged.

The heating tube was installed in the steam chest with the thermocouples on the lower side of the tube and the lead wires extending upward and through the outlet flanges as explained above.

Two more thermocouples were used to measure the liquid temperature in the feed pot and in the disengaging section. These couples were inserted through small holes in rubber stoppers and were placed directly in the liquid stream, approximately six inches from the ends of the heating tube.

Because electrical as well as thermal contact was made with the tube wall, it was necessary to use separate leads for each thermocouple. These lead wires were also Leeds and Northrup #24 iron and constantan, and were insulated with shellac and asbestos. These lead wires extended to the instrument table, where they were connected, by means of a switchbox, to a 32° F. cold junction and the potentiometer.

All heated parts of the evaporator were insulated except for the steam chest and the two flow meters. The entrainment separator was insulated by a wooden box packed with a mixture of asbestos-felt insulation and vermiculite. The feed pot, disengaging section, and 90° elbow were covered with 85% magnesia pipe insulation, and all remaining insulated parts were lagged with Johns-Mansville "Asbesto-Cel".

## EXPERIMENTAL PROCEDURE

Before any tests were made the apparatus was disassembled and cleaned with a wire brush to remove scale. After reassembly and before each series of runs the apparatus was washed free of loose rust particles by forced circulation of water. The heating tube was then swabbed thoroughly with a snugly fitting cotton cloth. Following these operations the apparatus was completely drained and rinsed with distilled water before being filled with the liquid to be investigated.

A run was made by operating the apparatus under steady conditions for a period of thirty minutes, during which time all operating variables were carefully recorded. A series of runs consisted of several runs made at different steam pressures while all other operating values were held at the same value. Nine series of runs were made, five series using water and 10, 20, 30, and 40% sucrose solutions with atmospheric system-pressure, and four series using water, 10, 20, and 40% sucrose solutions with a system pressure of  $1/2$  atmosphere.

Before starting a run the sucrose solution feed reservoir was filled with the liquid to be studied, either distilled water or a solution of sucrose in distilled water. A sucrose solution of the desired concentration was made up by weighing both the sucrose and the distilled water and mixing them in the feed reservoir. The specific gravity of the resulting solution was checked by a four-place hydrometer and compared with values in the literature<sup>19</sup> to insure that the solution had the proper concentration. After the desired concentration had been obtained, the evaporator was filled, by gravity flow, to a level in the entrainment separator approximately two inches above the vapor inlet, as indicated

by a scale on the sight glass of the separator. The distilled water feed reservoir was filled nearly to the top of its sight glass, and this level was read from a scale provided for that purpose.

When a vacuum run was to be made, the steam ejector was started. The desired manometer pressure was found by referring to the barometer, then the Cartesian Manostat was adjusted to maintain the desired pressure in the system.

Steam to the chest was next turned on, and the control index of the automatic instrument was set at the desired steam temperature. Ice was crushed and placed in a thermos bottle for the thermocouple cold junction. The working cells of the potentiometer were connected to the circuit to allow them to reach a steady voltage. Immediately preceding a series of thermocouple readings (usually requiring four minutes) these working cells were checked against a standard cell.

Usually a two hour warming-up period was required when starting up the evaporator with cold liquid, although thirty minutes or an hour usually sufficed if the liquid was still hot from a preceding run. During the warming-up period the valve in the distilled water feed line was adjusted by trial and error so that the feed rate matched the evaporation rate and the evaporator liquid level remained the same. The adjustment was checked by pouring condensed vapor back into the feed reservoir. With the proper adjustment the feed level was at its original position immediately following each addition of condensate. While this adjusting process was being carried out, periodic readings were made of the temperature of the liquid in the feed pot. A run could not be started until a satisfactory matching of the feed and evaporation rates had been effected and constant readings of the feed-pot liquid were obtained.

A run was started by collecting the vapor condensate. Each run was of thirty-minute duration and was timed by a stop watch. Immediately after starting a run and immediately after ending the run the liquid levels in the entrainment separator and the distilled water feed reservoir were read and recorded. The flow meter, steam pressure gage, system pressure manometer, and all thermocouples were read three times during each run, and the values obtained were averaged. If any of these variables changed appreciably during a run, the run was repeated.

The 3/4-inch Stabilflo valve used with the temperature control instrument was found to be much too large for optimum results in so small a system, being only 1/4 open for even the highest steam pressure runs. Nevertheless, with some practice it was found possible to obtain excellent control of the steam pressure during runs.



## METHOD OF CALCULATION AND CALCULATED RESULTS

The liquid film heat transfer coefficient is, by definition<sup>12</sup>, a proportionality constant which relates the rate of heat transfer to the product of the heat transfer area and the difference between the temperatures of the heating surface and the bulk of the liquid being heated.

Thus,

$$h_L = \frac{q}{A_S \Delta t_L}$$

where  $h_L$  is the local individual coefficient of heat transfer (or merely the liquid film coefficient),  $q$  is the amount of heat transferred per unit time,  $A_S$  is the area of the heat transfer surface, and  $\Delta t_L$  is the difference between the temperatures of the heating surface and the bulk of the liquid being heated. It should be pointed out that this local coefficient will be the same for different parts of a heat transfer surface only if each part has the same ratio of  $q$  to  $\Delta t_L$ .

In the present work, as well as in most practical problems dealing with evaporation, both  $q$  and  $\Delta t_L$  vary from point to point on the heat transfer surface, thus ruling out the calculation of a single  $h_L$  which could be applied to any part of the surface. It is desirable, however, to have a value which may be applied to the surface as a whole, and that is the type of coefficient which was calculated in this work. It must be remembered that this is not a point-to-point value, and is meaningless unless applied to the entire heating surface of the evaporator.

Even the calculation of such a coefficient presents many difficulties. The area can be easily measured, and the total heat transfer rate can be determined with reasonable accuracy from the evaporation rate and the temperature rise of the liquid. However, evaluation of the temperature

difference is much more complicated. If the liquid had entered the tube at its boiling point and the tube wall temperature distribution had been uniform, the difference could easily have been taken. Conversely, if the liquid had reached its boiling temperature just as it reached the top of the tube so that no change of phase resulted, a correct value could have been obtained by a logarithmic-mean temperature difference. Neither of these methods could be correctly applied to the data of this work, however, because the liquid conditions were intermediate between the two extremes just described. The liquid entered the tube at some temperature below its boiling point, was elevated to the boiling point while rising in the tube, and then began to boil at some point below the top of the tube.

It was reasoned, therefore, that calculation of heat transfer coefficients by both of these methods would give values which formed the upper and lower limits of any calculated value. However, even this procedure was complicated by lack of uniformity of the tube wall temperature. A length-mean tube wall temperature was finally selected. Because of the location of the thermocouples, this amounted to an arithmetic mean of the temperature readings of each of the six points.

To obtain the temperature of the inside surface of the tube it was necessary to first determine how much temperature drop occurred between the thermocouple junction and the inside surface. This was done by use of the equation

$$\Delta t_p = \frac{q x}{k_w A_{ave}}$$

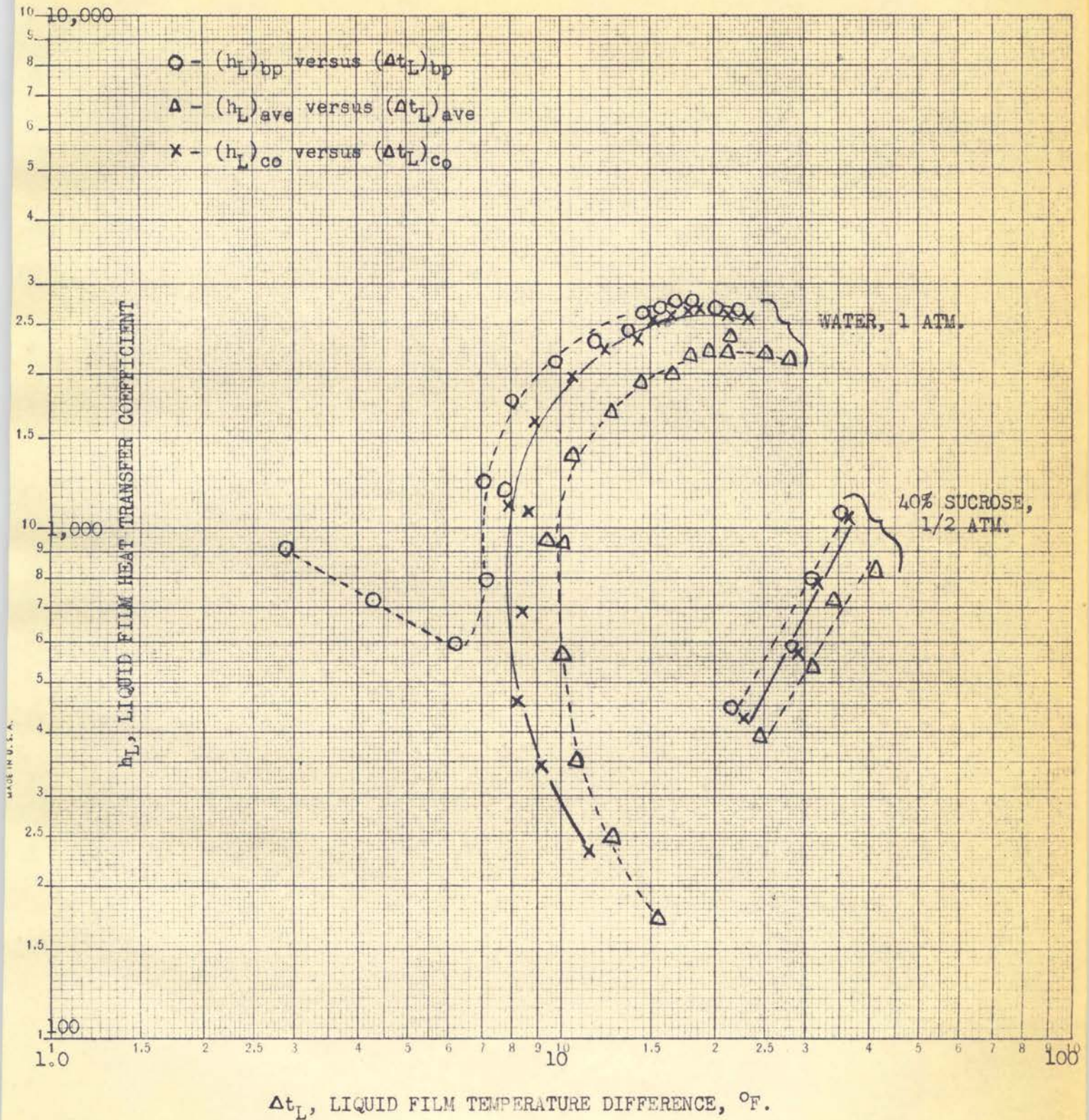
where  $\Delta t_p$  is the temperature drop through the pipe wall,  $x$  is the wall

thickness,  $k_w$  is the thermal conductivity of the wall, and  $A_{ave}$  is the log mean area through which the heat passes between the two surfaces. Here, again, incorrect assumptions are made. Since  $q$  varies from point to point on the tube, the temperature drop through the tube wall varies from point to point. However, there appeared to be no more practical alternative method of calculation.

From the value calculated for the temperature of the heating surface, the two previously mentioned types of liquid film temperature differences were calculated. The one obtained by assuming the liquid to be at its boiling point was designated  $(\Delta t_L)_{bp}$ , while the one assuming no boiling to occur was referred to as  $(\Delta t_L)_{ave}$ . From these the corresponding coefficients could readily be calculated, and were designated  $(h_L)_{bp}$  and  $(h_L)_{ave}$ , respectively. Figure 4 shows a comparison of these two methods of calculation.

Assuming that the error involved in calculating the tube wall temperature drop was not too great, the two values thus obtained should give an upper and a lower limit for any calculated coefficient. Thus, if all heat transferred were in the form of latent heat,  $(h_L)_{bp}$  would be correct. On the other hand, if  $q$  were composed entirely of sensible heat  $(h_L)_{ave}$  would apply. All runs of the present work contained both latent and sensible heat; therefore, correct values would be intermediate between the calculated values. A key to the relative distance of the correct value from the two limits seems to be offered by a ratio of sensible and latent heats. Therefore, a "corrected" value of the temperature difference,  $(\Delta t_L)_{co}$ , was calculated, and from it the corresponding coefficient,  $(h_L)_{co}$ , was obtained. This corrected difference was

FIGURE 4. COMPARISON OF THREE METHODS OF CALCULATING THE LIQUID FILM TEMPERATURE DIFFERENCE.



obtained as follows:

$$(\Delta t_L)_{co} = (\Delta t_L)_{bp} + \left[ (\Delta t_L)_{ave} - (\Delta t_L)_{bp} \right] \frac{q_s}{q}$$

where  $q_s$  is the sensible heat. This correction was not at all rigorous and amounted simply to a weighted average of the two extreme values for  $\Delta t_L$ . It seems reasonable to allot temperature difference on this weighted basis, because in so doing the proper part of the temperature difference is allotted to both the boiling and the non-boiling portions of the tube. In so doing the weighted proportion of heat transfer is given to each mode of heat transfer. Values calculated by this method should, therefore, be less in error than either of the extreme values. A comparison of the corrected values with the uncorrected values for the water runs is shown in Figure 4. Logarithmic plots of the corrected liquid film temperature difference versus the liquid film heat transfer coefficient calculated from this corrected difference are shown in Figures 5 and 6.

Plots of  $q$  vs.  $\Delta t_t$ , the temperature drop from the thermocouple junction to the liquid, were constructed to eliminate the error involved in calculating the tube wall temperature drop. Logarithmic plots are shown in Figures 7 and 9, while the same data are shown on rectangular coordinates in Figures 8 and 10.

Figure 11 shows a plot of the natural circulation rate versus the total heat transfer rate.

To show a comparison of the data of this work with data for heat transfer to non-boiling liquids, values of  $\frac{DG}{\mu}$  and  $\frac{hD/k}{(c_p \mu/k)}$  0.4 were calculated and, with the Dittus-Boelter equation, were plotted on logarithmic coordinates, as shown in Figures 12 and 13.

Comparisons with correlations of Jakob<sup>8</sup>, Thorstenberg<sup>21</sup>, and Aubrecht<sup>2</sup> are shown in Figures 14, 15, and 16, respectively.

The distribution of temperatures in the tube wall is shown in Figure A1 for runs with distilled water and in Figure A2 for runs using a 40% aqueous solution of sucrose. Runs with other concentrations of sucrose solutions yielded the same type of curve as was obtained with the 40% concentration.

In Figures 5 to 11 and A1 and A2 the data points are represented by symbols, the sizes of which give an approximate indication of the precision of the data. In general, the three thermocouple readings made for a thermocouple position during a given run agreed with a precision of from  $\pm 0.1$  to  $\pm 0.6^\circ$  F., so the precision of the temperature difference may be considered to be within  $\pm 1.2^\circ$  F. The precision of the measurements of liquid circulation rate was about  $\pm 15\%$  for very low circulation rates and increased to about  $\pm 1.0\%$  for maximum circulation rates.

FIGURE 5. LIQUID FILM HEAT TRANSFER COEFFICIENTS VERSUS CORRECTED LIQUID FILM TEMPERATURE DIFFERENCES, ATMOSPHERIC DATA.

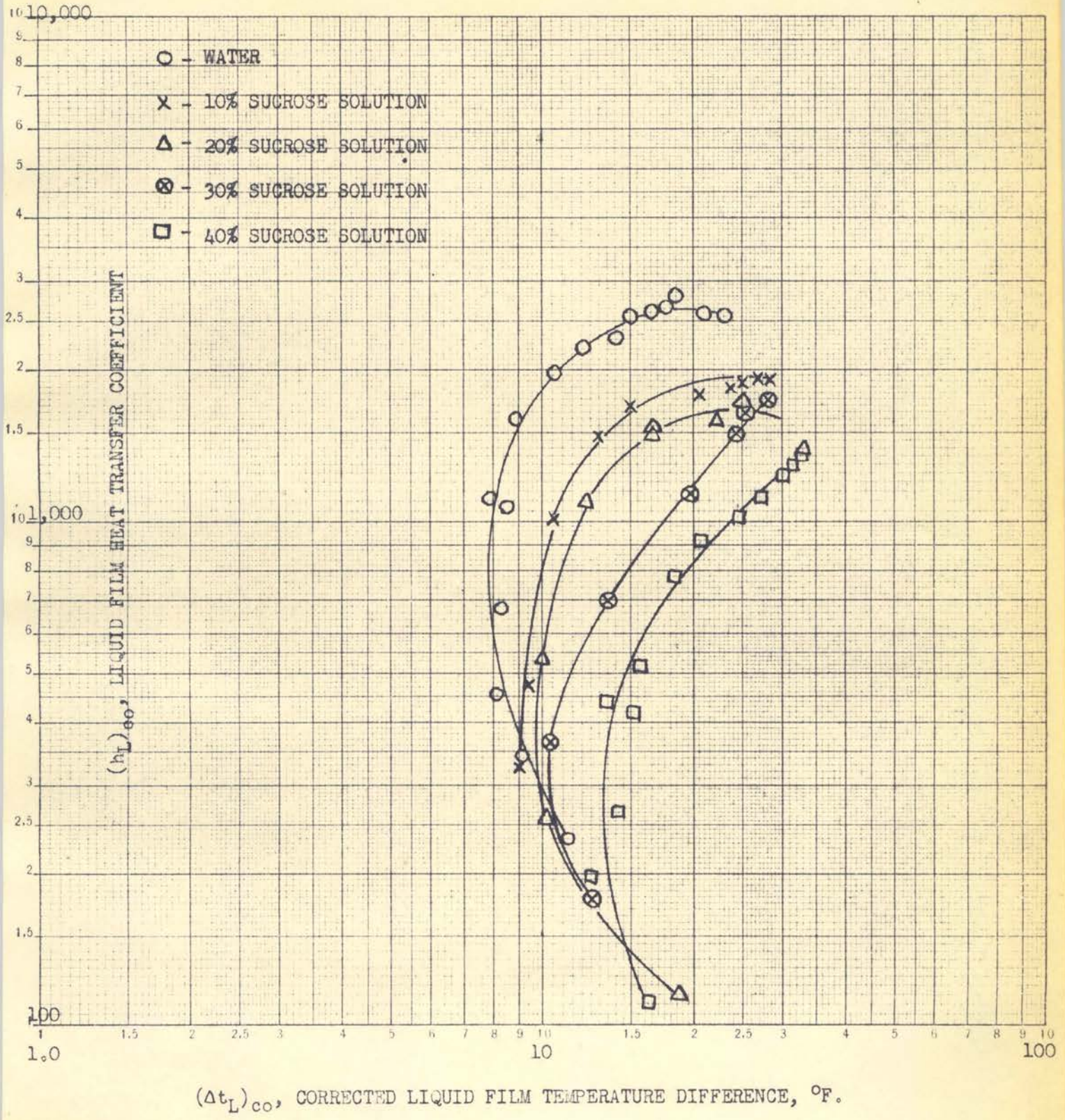


FIGURE 6. LIQUID FILM HEAT TRANSFER COEFFICIENTS VERSUS CORRECTED LIQUID FILM TEMPERATURE DIFFERENCES, DATA FOR 1/2 ATMOSPHERE.

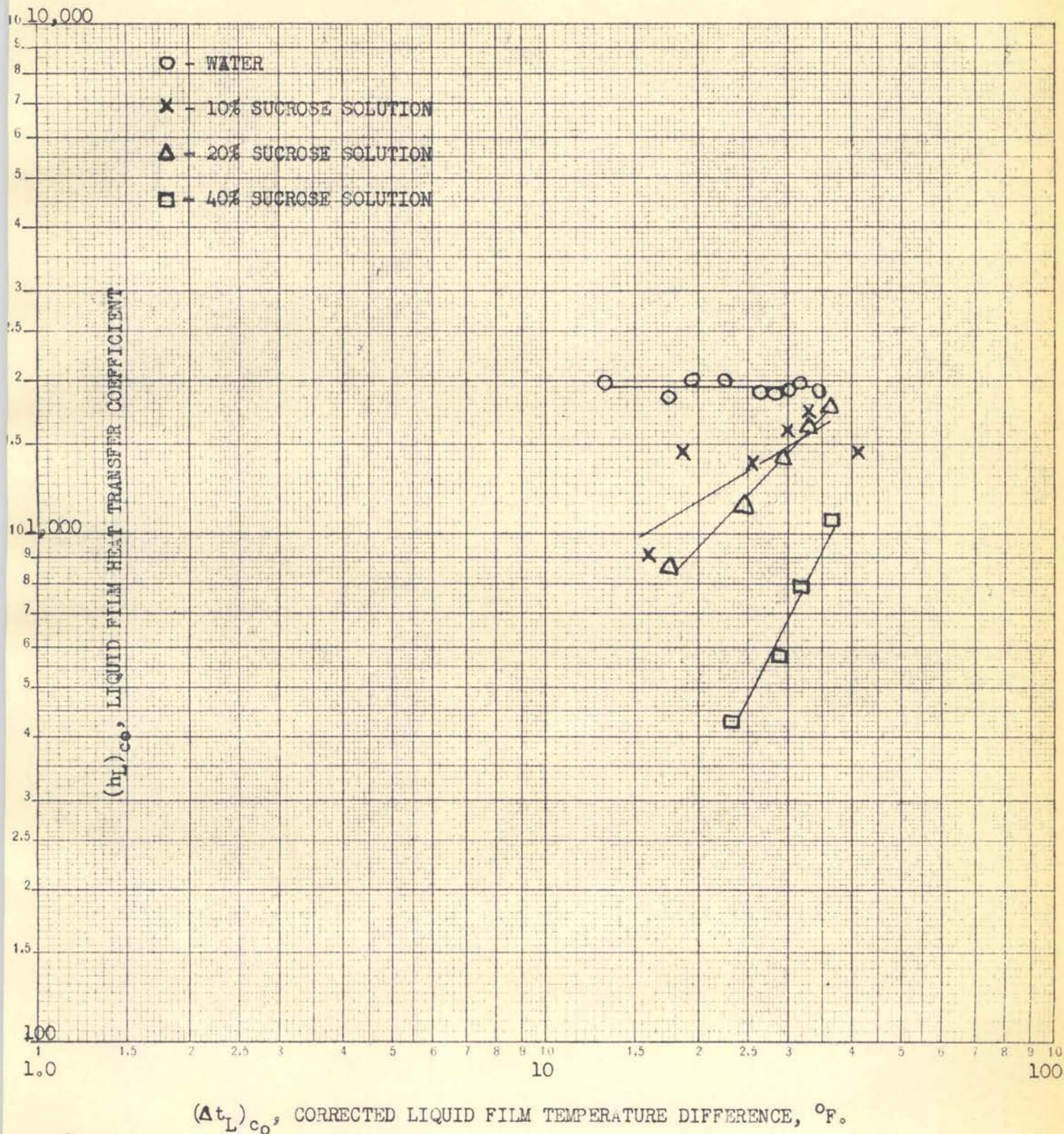
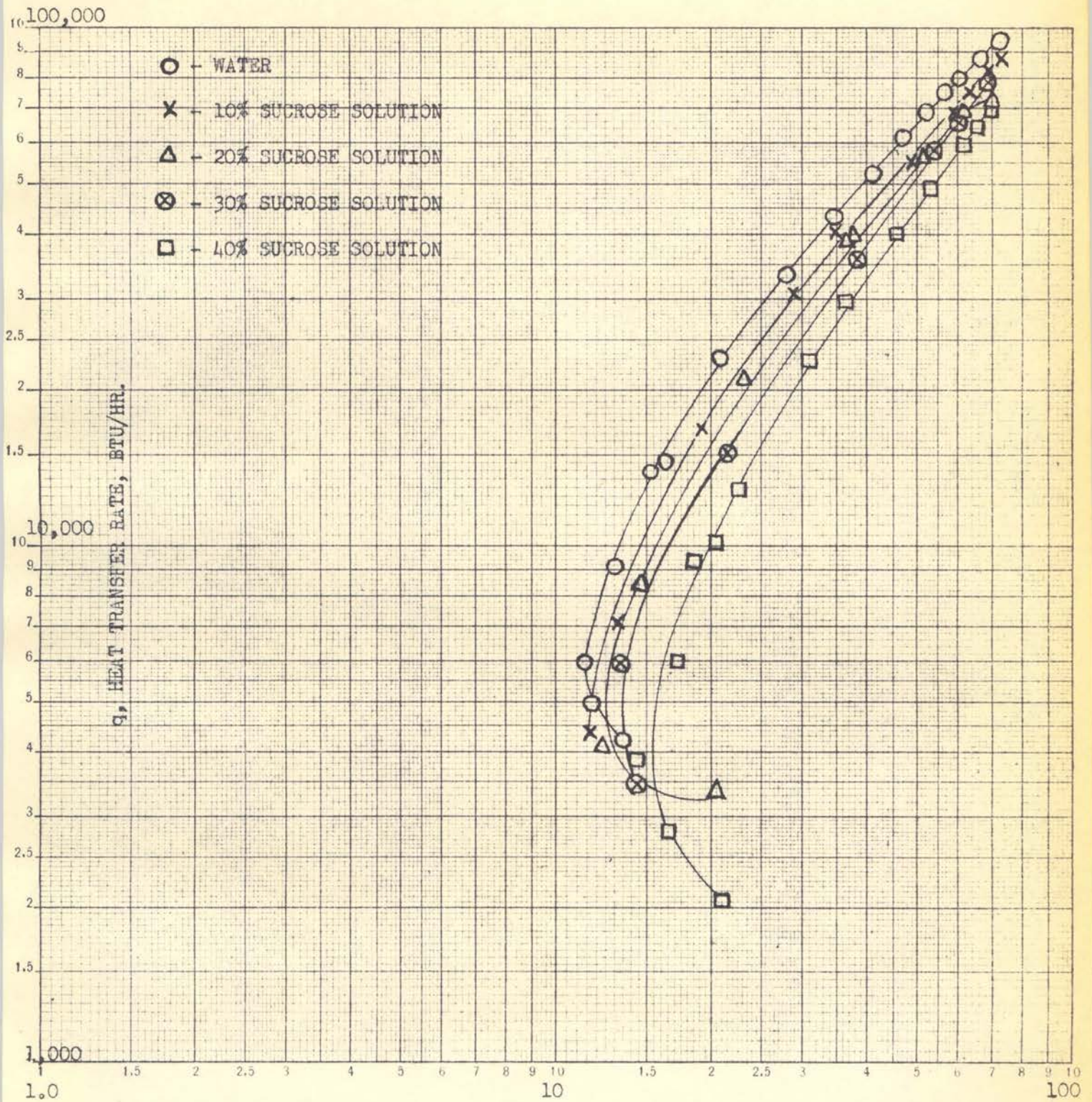




FIGURE 7. HEAT TRANSFER RATE VERSUS CORRECTED TEMPERATURE DIFFERENCE BETWEEN THERMOCOUPLE AND LIQUID, ATMOSPHERIC DATA.



$(\Delta t_t)_{co}$ , CORRECTED TEMPERATURE DIFFERENCE BETWEEN THERMOCOUPLE AND LIQUID, °F.

FIGURE 8. HEAT TRANSFER RATE VERSUS CORRECTED TEMPERATURE DIFFERENCE BETWEEN THERMOCOUPLE AND LIQUID, ATMOSPHERIC DATA.

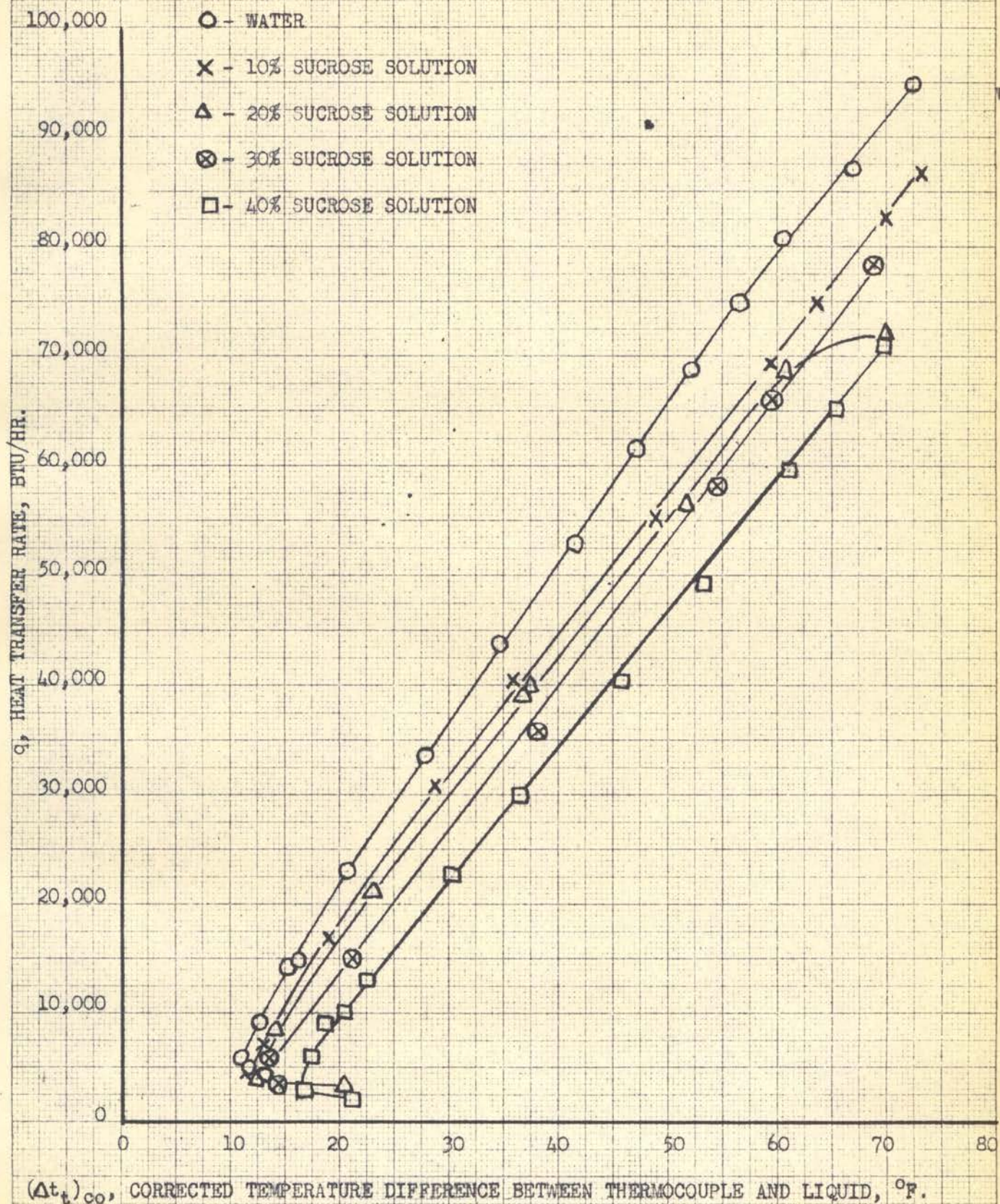
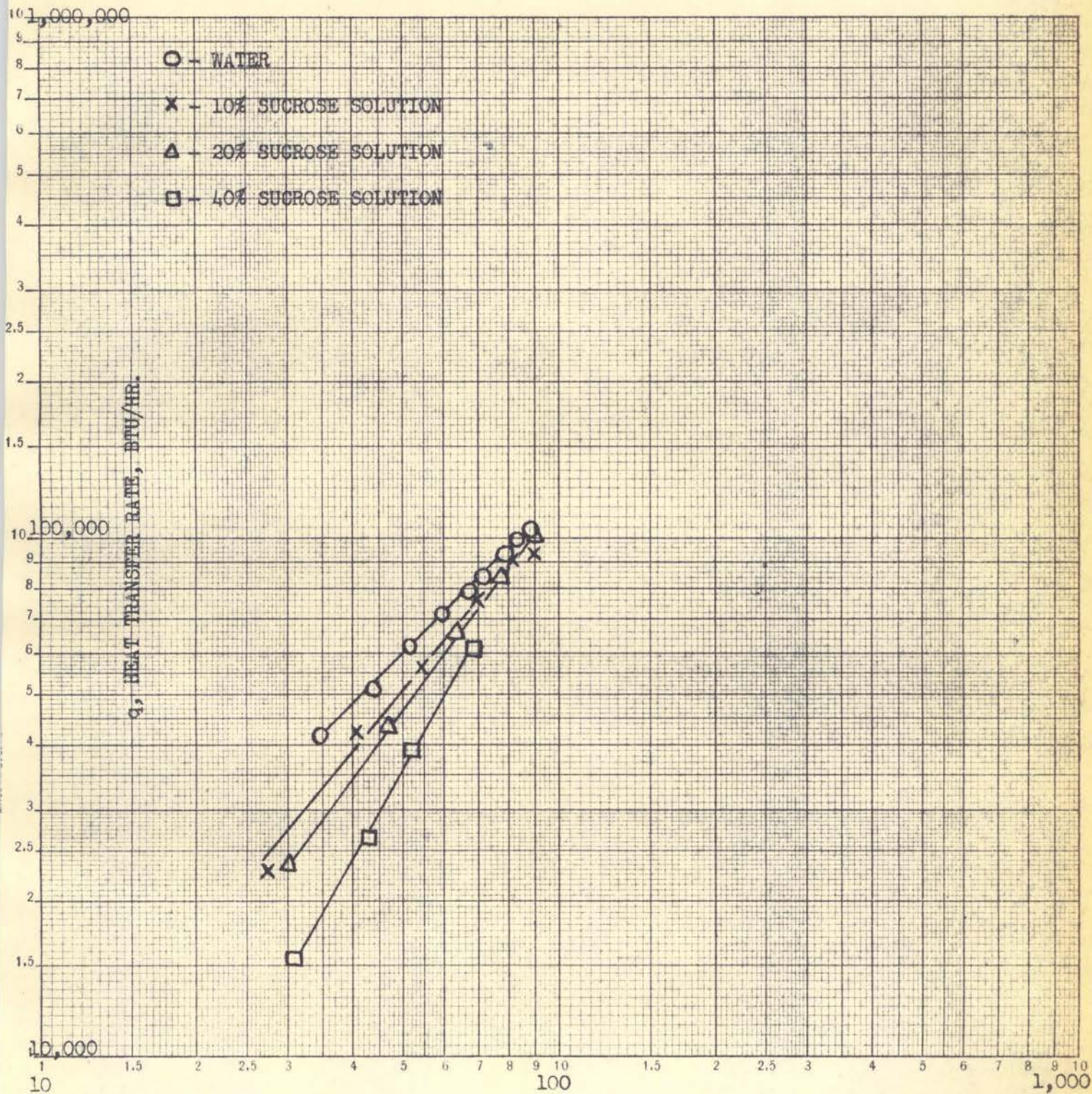
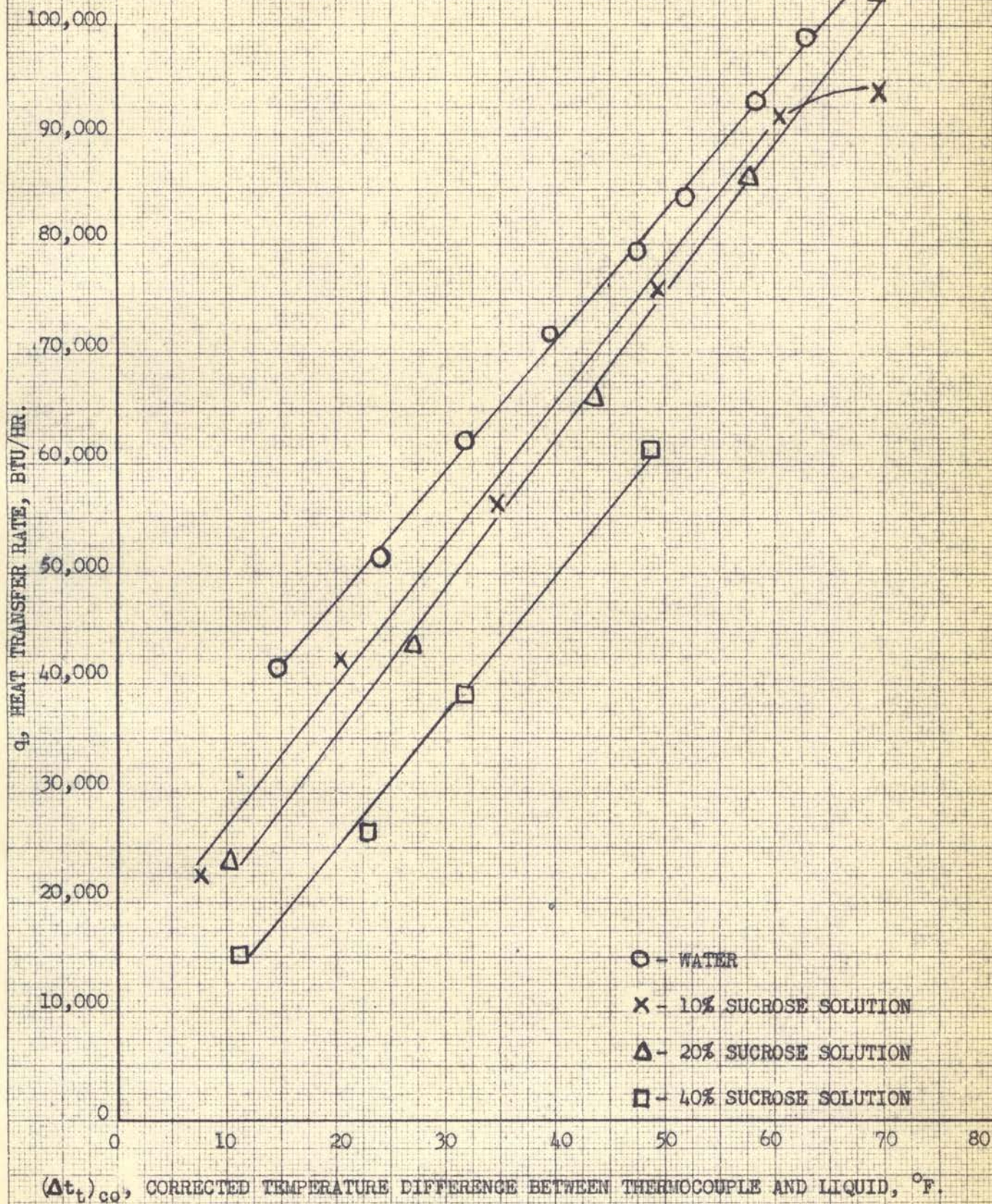


FIGURE 9. HEAT TRANSFER RATE VERSUS CORRECTED TEMPERATURE DIFFERENCE BETWEEN THERMOCOUPLE AND LIQUID, DATA FOR 1/2 ATMOSPHERE.



$(\Delta t_t)_{co}$ , CORRECTED TEMPERATURE DIFFERENCE BETWEEN THERMOCOUPLE AND LIQUID, °F.

FIGURE 10. HEAT TRANSFER RATE VERSUS CORRECTED TEMPERATURE DIFFERENCE BETWEEN THERMOCOUPLE AND LIQUID, DATA FOR 1/2 ATMOSPHERE.



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FIGURE 11. NATURAL CIRCULATION RATE VERSUS HEAT TRANSFER RATE.

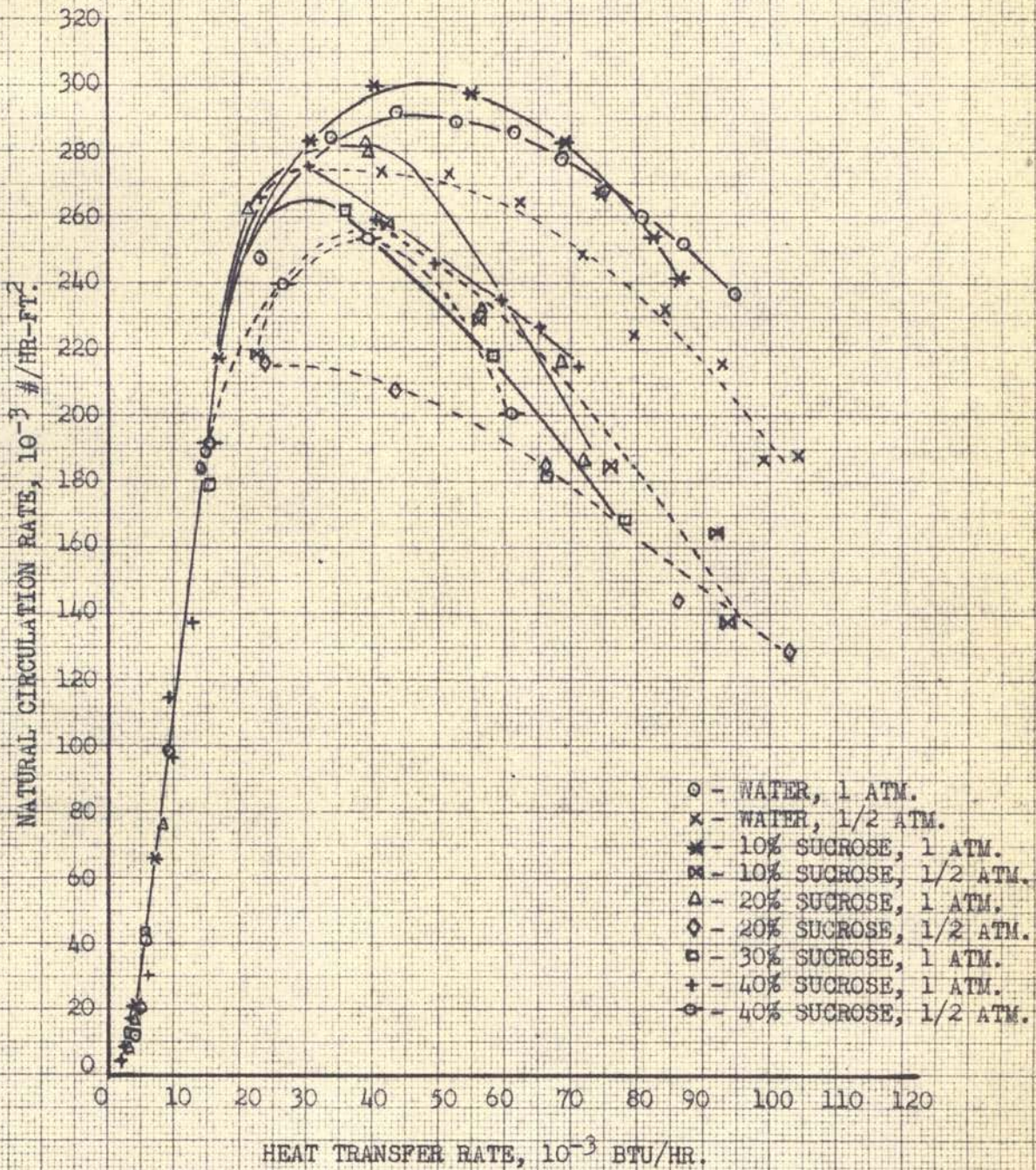


FIGURE 12. COMPARISON OF ATMOSPHERIC DATA TO DITTUS-BOWLER EQUATION.

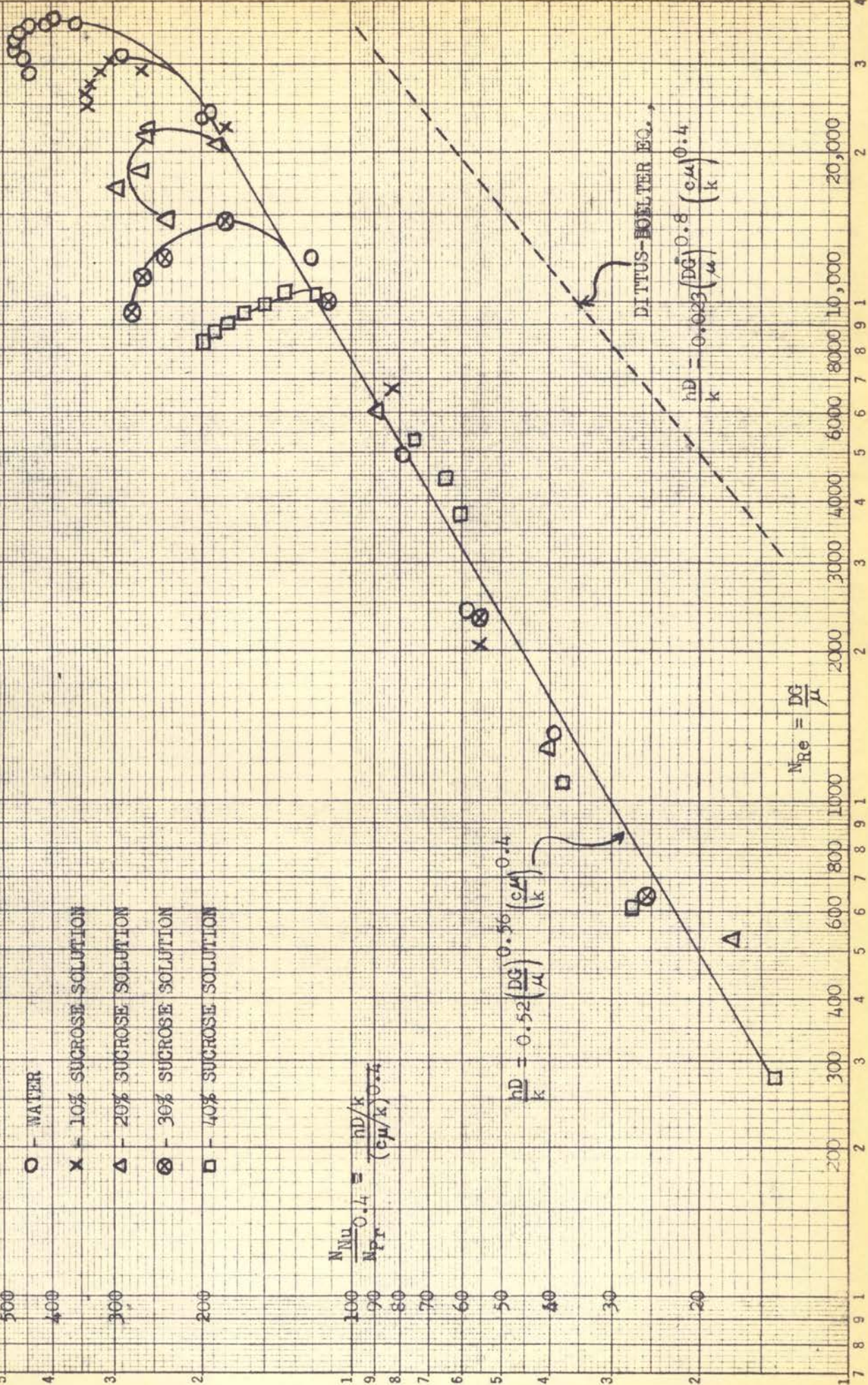


FIGURE 13. COMPARISON OF DATA FOR 1/2" ATMOSPHERE WITH THE DITTUS-BOWLER EQUATION AND WITH STRAIGHT-LINE PORTION OF FIGURE 12.

- - WATER
- × - 10% SUCROSE SOLUTION
- △ - 20% SUCROSE SOLUTION
- - 40% SUCROSE SOLUTION

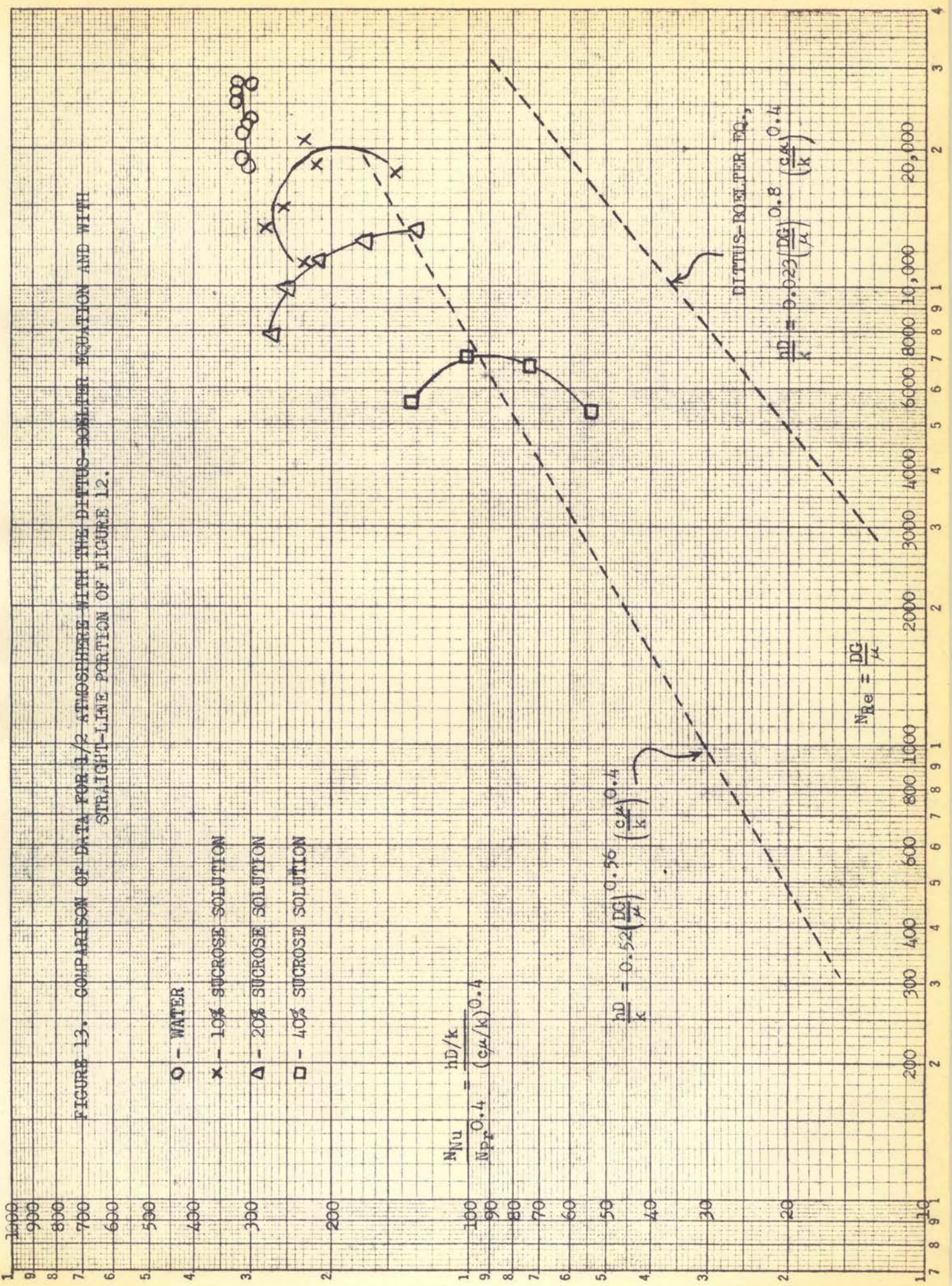
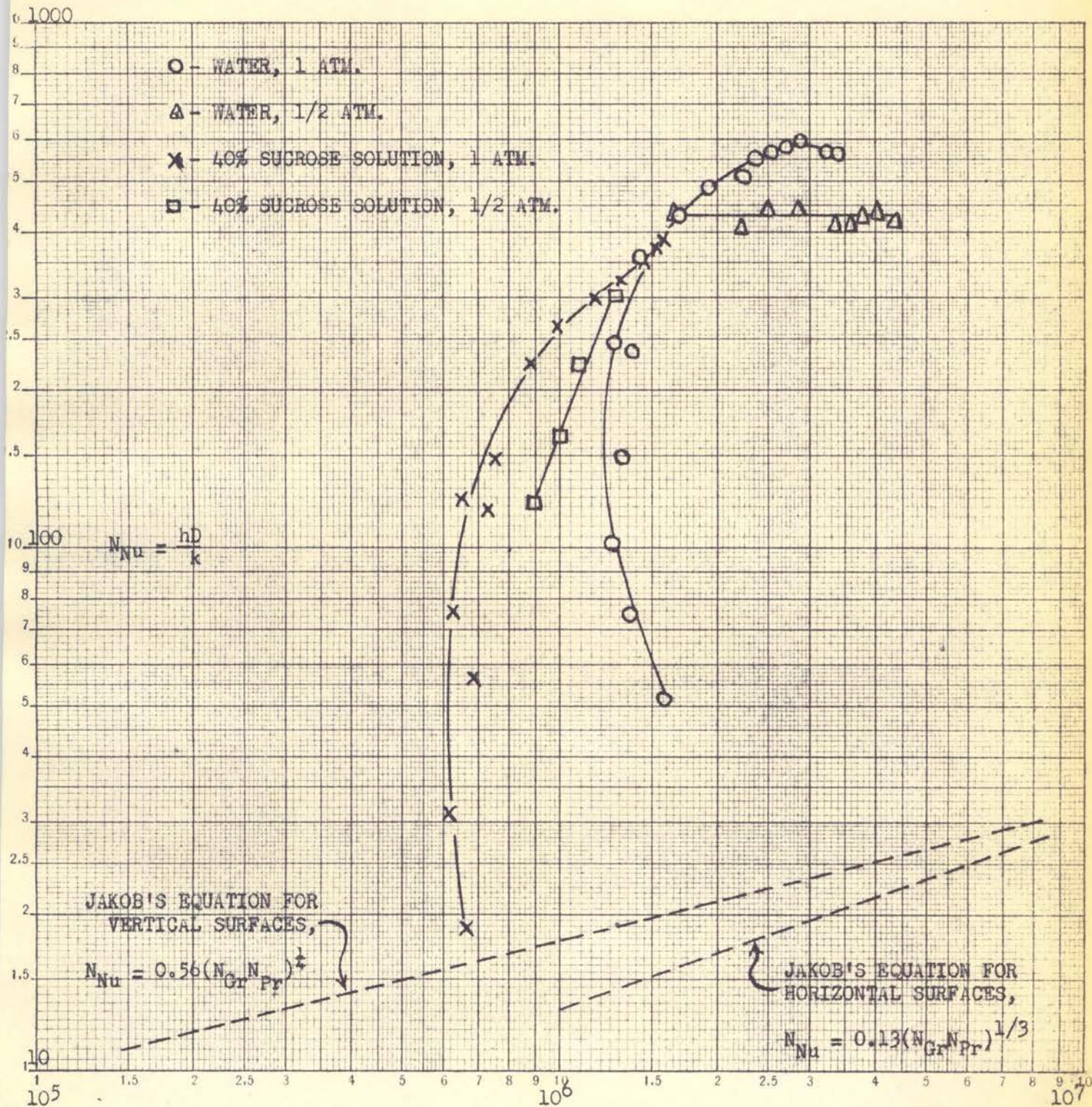


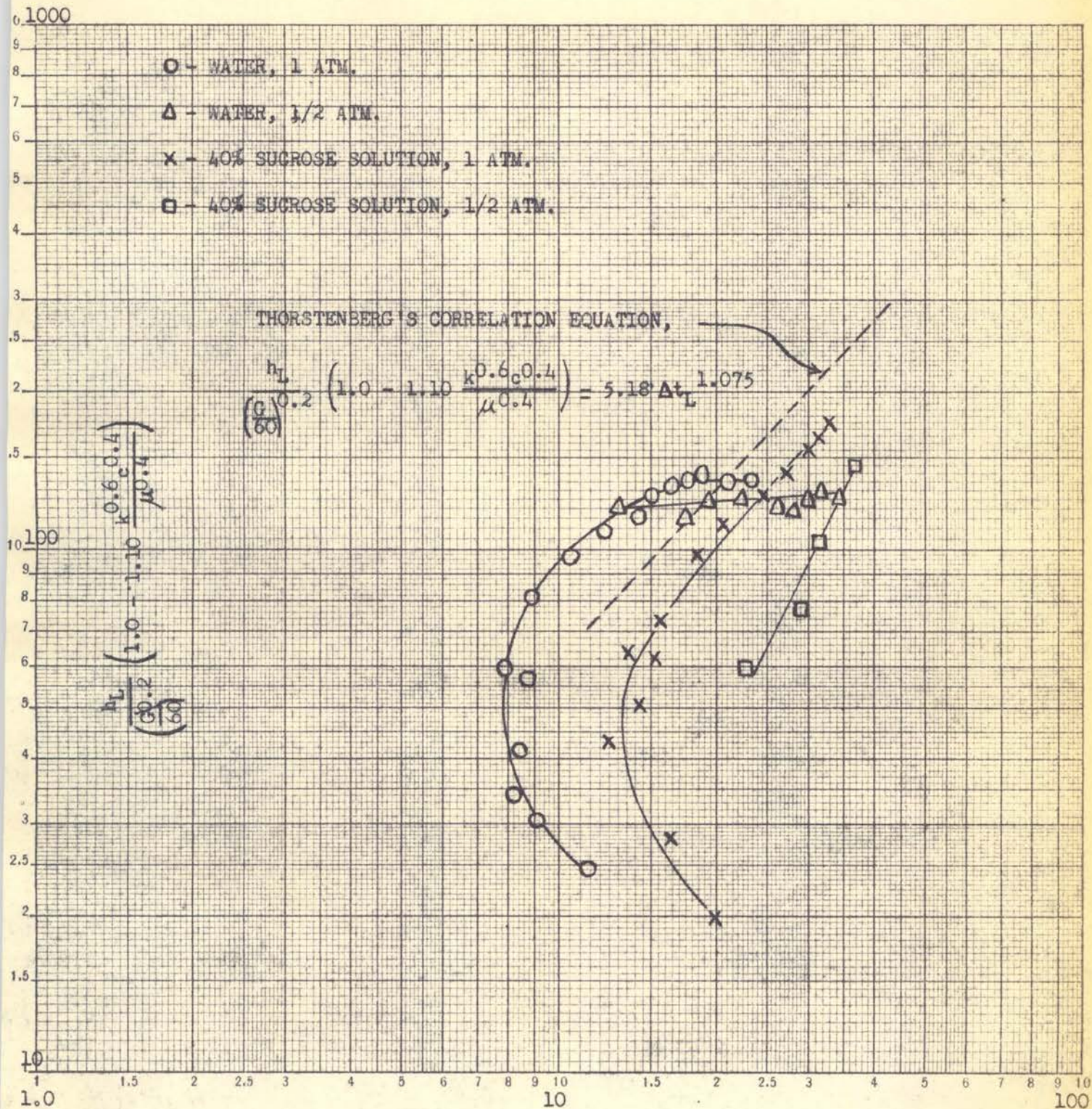
FIGURE 14. COMPARISON OF DATA WITH JAKOB'S EQUATION FOR LOW-RATE HEATING ON FLAT SURFACES.



$$(N_{Gr})(N_{Pr}) = \frac{(D^3 \rho^2 g \beta \Delta t)(c \mu)}{(\mu^2)(k)}$$

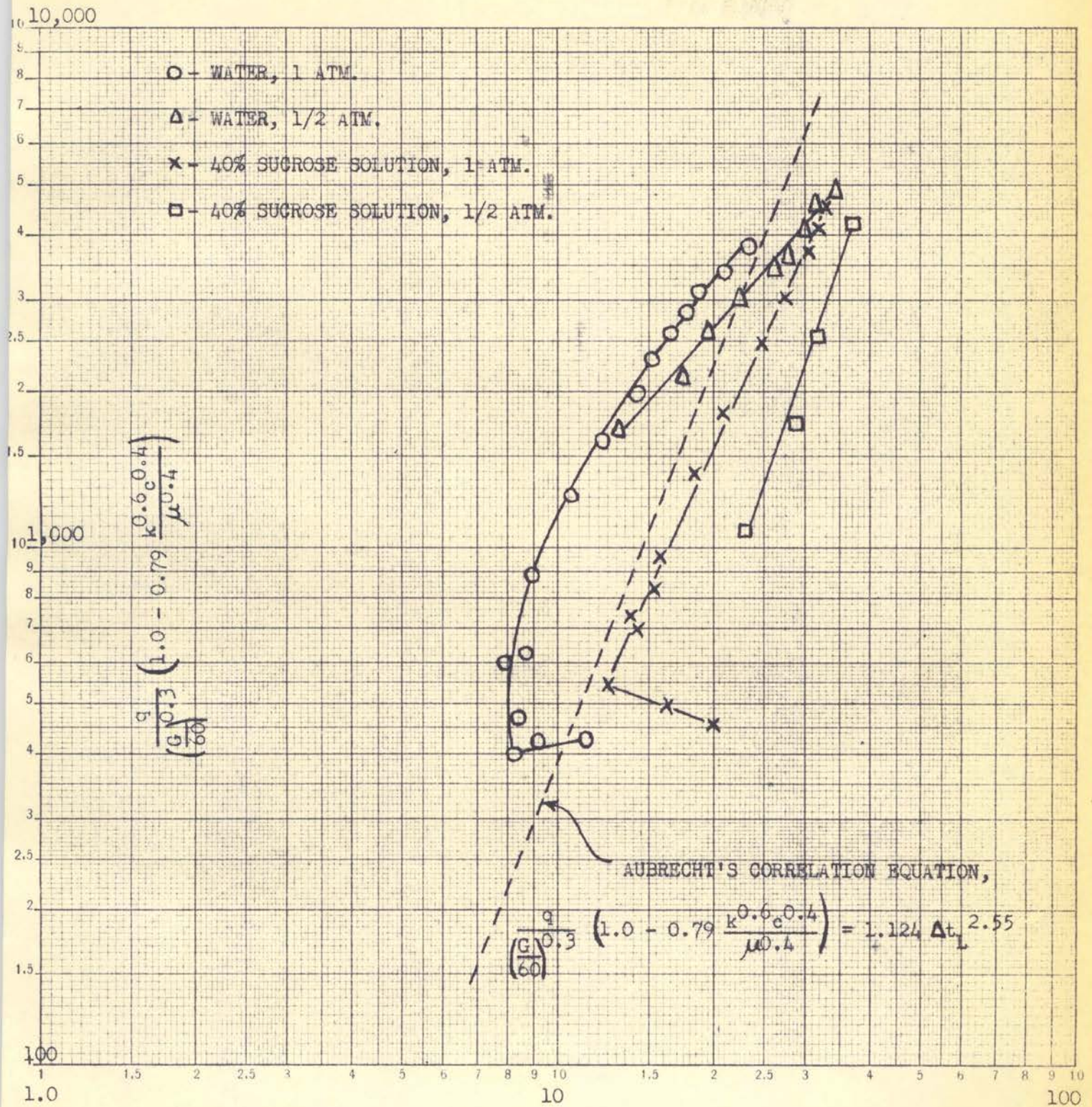


FIGURE 15. COMPARISON OF DATA WITH THORSTENBERG'S CORRELATION EQ.



$(\Delta t_L)_{Co}$ , CORRECTED LIQUID FILM TEMPERATURE DIFFERENCE, °F.

FIGURE 16. COMPARISON OF DATA WITH AUBRECHT'S CORRELATION EQ.



$(\Delta t_L)_{co}$ , CORRECTED LIQUID FILM TEMPERATURE DIFFERENCE, °F.

## DISCUSSION OF RESULTS

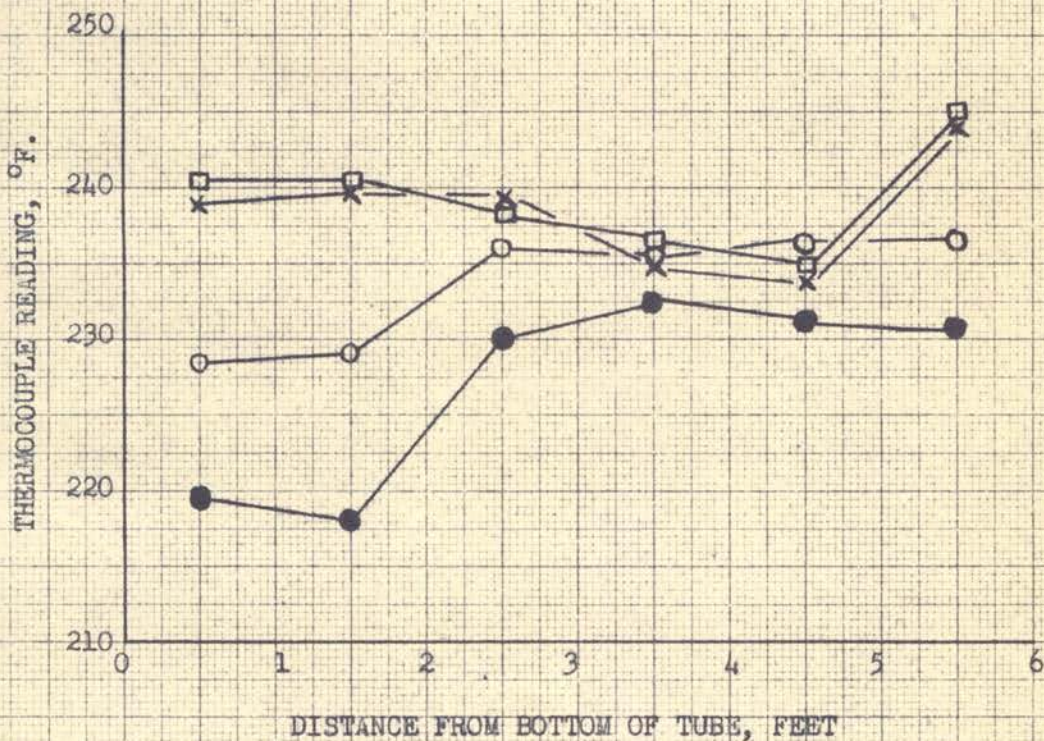
It was expected that the temperature of the tube wall would not be precisely the same at different points along its length. However, the sharp variation of the temperature at the top two measuring points was greater than had been anticipated. The large difference and the consistency with which it occurred rule out the possibility of incorrect readings of the potentiometer. Nor does it seem likely that the thermocouples were at fault, because the distribution pattern was not the same for all conditions of feed temperature, as shown by Figure 17. This difference could not logically be attributed to the thickness of the steam film outside of the tube, because the film, running down the tube, would be thickest at the lower sections, and therefore would not account for the low reading at the second position from the top. It must be concluded, therefore, that the conditions of the liquid inside of the tube were the principal factors contributing to this distribution of temperature.

Figure 17 is used as a key to this analysis. In the lowest curve the feed entered quite cold. The liquid film temperature difference, therefore, was rather high, resulting in a large value of  $q$ , and consequently a high temperature drop across the steam film between the steam and the thermocouple. The thermocouple reading for this section of the tube, as a result, was considerably lower than the steam temperature. As the liquid rose in the tube its temperature became higher, thus reducing  $\Delta t_f$  and consequently  $q$ , and resulting in a thermocouple reading which was nearer to the steam temperature and therefore higher than that

FIGURE 17. INFLUENCE OF INLET LIQUID TEMPERATURE UPON TUBE WALL TEMPERATURE DISTRIBUTION.

- - INLET TEMP. 92.9° F., OUTLET TEMP., 211.6° F.
- - INLET TEMP. 137.2° F., OUTLET TEMP., 211.7° F.
- × - INLET TEMP. 169.5° F., OUTLET TEMP., 211.8° F.
- - INLET TEMP. 208.5° F., OUTLET TEMP., 211.9° F.

NOTE: Data is for 10% sucrose solution at 1 atm.,  
Steam pressure 14 psig.



of the preceding section. After the boiling point had been reached, vapor bubbles agitated the liquid, resulting in an increased rate of heat transfer which caused a higher steam film temperature drop and a correspondingly lower thermocouple reading. The boiling rate here apparently was very low, as shown by the very slight drop from the temperature of the preceding section.

In the second curve from the bottom, the feed temperature was much higher. Consequently  $\Delta t_L$  was smaller,  $q$  was smaller, the steam film temperature drop was smaller, and the thermocouple reading was higher than was the case with colder feed. The boiling temperature seems to have been reached by the time the third measuring point was reached. At about four feet from the bottom of the tube the thermocouple temperature began rising slightly, indicating that the vapor being formed inside the tube was causing some insulation of the heating surface, thus reducing the heat transfer rate and causing a higher reading of the thermocouple. In the top two curves this effect became even more pronounced, and vapor binding appeared to become serious at the top point.

An investigation of Figure A1 will show that vapor binding did not occur with low values of  $\Delta t_L$  but became more and more appreciable as the  $\Delta t_L$  value increased. An exception to this is curve No. 12 for atmospheric water, for which the vapor binding appeared to cover a smaller portion of the tube than was the case for the curves on either side of it. This overlapping of curves also was seen to occur with run No. 6 in the vacuum series for water. This occurrence very likely was caused by some freakish condition of flow inside the tube, such as slugging, although no explanation is readily apparent.

The plots of the liquid film coefficient versus the liquid film temperature difference in Figure 5 show a curving toward the right for lower values of the coefficient, which is contrary to what is expected on the basis of the Nukiyama boiling curve. This apparent contradiction probably is the result of incorrect measurement of the evaporation rate at these low rates. In order for the vapor condensate actually collected to give a true indication of the evaporation rate it is essential that the liquid in the entrainment separator be at the boiling temperature. If the circulation rate became so low that the liquid in the entrainment separator cooled slightly from radiation loss, then some of the vapor would have been condensed by the liquid, and the condensate collected would have been less than the evaporation rate. As the circulation rate decreased still further the error in measuring the evaporation rate became even greater. Therefore, the quantity of heat transfer calculated from the experimental data may differ from that actually transferred through the walls of the tube. These lower points, therefore, cannot be relied upon for a true picture of what occurred in the heating tube. Because they include the effect of the entrainment separator, they would apply only to this particular piece of equipment.

As shown by Figure 11, the rate of circulation decreased at higher values of the heat transfer rate. Because the circulation rate was still quite large even at the highest steam pressures used, it is not likely that much error was involved in the measurement of the evaporation rate for these higher runs, and these upper points would, therefore, be more reliable. The leveling off of these curves for water and for 10 and 20% sucrose solutions show that a plateau has been reached,

above which a further increase in the temperature difference would result in a decrease in the liquid film coefficient. That such a critical point has actually been reached is not certain, however, in view of the uncertainty of the value which should be used for the liquid film temperature difference.

Figures 7 to 10 make use of a semi-overall temperature difference and eliminate the error involved in calculating the temperature drop through the tube wall. However, even this value is subject to errors, because it assumes a length mean average for the tube wall and only an approximate value for the liquid temperature. These plots show the same departure for the runs having low circulation rates as was obtained in Figures 5 and 6, and therefore seem to substantiate the suggestion of error in the measurement of the evaporation rate. These curves do not show any evidence of a plateau at the higher steam pressures as did the plots involving only the liquid film. This leads to speculation that the leveling off in the liquid film plots resulted from errors in the calculation of liquid film temperature differences, which of course would be magnified by plotting the temperature difference versus the coefficient.

Figure 11 is quite similar to plots obtained by Cleve<sup>4</sup> and Aubrecht<sup>2</sup>. However, one would expect the curve for water to lie above those for sucrose solutions. The reason for the reversal of water with 10% sucrose is not readily apparent.

A comparison of the data of this evaporator with the Dittus-Boelter equation may be seen in Figures 12 and 13. Data for steam pressures up to about 10 psig. fell on a line above but somewhat similar in appearance to the equation which holds for heat transfer to non-boiling liquids in

turbulent flow. If, as mentioned above, the value used for the evaporation rate is too small and the value for the temperature difference too large, then this portion of the data should be somewhat higher. However, there would be little change in the Reynold's number, so the appearance of the curve would be somewhat similar. It may be said, then, that conditions of heat transfer in the evaporator bear a resemblance to conditions of non-boiling heat transfer when steam pressures of lower than about 10 psig. are used, but depart from such a resemblance when higher steam pressures are used. The equation shown is intended as a comparison only and not as a correlation.

Jakob<sup>8</sup> plotted  $\log N_{Nu}$  versus  $\log (N_{Gr}) (N_{Pr})$  and obtained two equations in dimensionless variables to represent low-rate heating from horizontal and vertical surfaces. These equations held well for his data so long as the formation and motion of vapor bubbles had a negligible influence upon heat transfer. At higher rates of heat flow agitation of the liquid by vapor bubbles caused a great increase in the coefficient and  $N_{Nu}$  increased accordingly. Consequently, the curves made a sharp turn upward from the lines for low-rate boiling. Data for water and 40% sucrose solutions were calculated and plotted in this manner to show a comparison with these natural convection equations, as illustrated in Figure 14. All points are seen to lie above Jakob's equations, showing that there is considerable influence from agitation by vapor bubbles. Because of the importance of  $h$  in the Nusselt number and of  $\Delta t$  in the Grashof number, the curves have a shape similar to those in the plot of  $h$  versus  $\Delta t$ . However, the other properties in the dimensionless groups, principally viscosity, have changed the relative positions of the curves



so that the curve for 40% sucrose now lies to the left of the water curve. The tendency of the four curves to pass through one mutual point is probably mere chance and does not appear to have theoretical basis. As was expected, this method of plotting does not provide a correlation of the data.

The data were calculated in terms of the correlation equation of Thorstenberg and plotted in Figure 15. Because this correlation was obtained for data from the same apparatus, close agreement is to be expected. Such, however, is not the case, as may be seen in the plot. As has been explained previously, the points for low circulation rates are not reliable. Disregarding these points, it is seen that the correlation does not satisfactorily compensate for the differences in viscosity and boiling temperature of the liquids. It should be pointed out that Thorstenberg's method of calculation of data assumed the liquid to enter the heating tube at its boiling point. Therefore, his value of  $q$  was smaller, which gave a lower calculated tube wall temperature drop and consequently a higher liquid film temperature difference. It appears, then, that his correlation was based upon data which were even less accurate than the present data. From this point of view the disagreement is not surprising.

Of the several methods of plotting, perhaps the nearest to a correlation is the one shown in Figure 16. For this plot, data for water and 40% sucrose were calculated in terms of the equation presented by Aubrecht for natural circulation of liquids in a  $45^\circ$  inclined heating tube. The curves still are somewhat separated but are straightened and have more nearly the same slope, giving promise that this method is on the right track.

Because of the magnitude of the error involved in the calculation of temperature drop through the stainless steel tube, it does not seem worthwhile to obtain an exact correlation for this data if, indeed, one were possible. The purpose of these plots has been to show how the data compare with existing correlation methods, and to suggest methods of approach for use with data having fewer inherent inaccuracies.

## CONCLUSIONS

1. An accurate calculation of liquid film heat transfer coefficients for the inclined tube natural circulation evaporator is greatly complicated by the existence of different rates of heat transfer for different sections of the heating tube.
2. No definite maximum was reached in a plot of liquid film coefficients versus liquid film temperature differences, though a leveling off occurred with water and with 10 and 20% sucrose solutions. Plots for 30 and 40% sucrose solutions showed no tendency to level off.
3. As the rate of heat transfer increased, the rate of natural circulation increased to a maximum and began decreasing.
4. For very low rates of heat transfer, radiation losses from the entrainment separator complicated the determination of the heat transfer rate.
5. The stainless steel heating tube caused a temperature drop so great that its influence overshadowed the influence of the liquid film.

## RECOMMENDATIONS FOR FURTHER STUDY

1. Further work of this nature should be carried out using a heating tube of high thermal conductivity, such as copper. This should give better data for two reasons: (1) the tube wall temperature difference would be small, so that errors in calculating it would be much less appreciable, and (2) the reduced temperature drop across the pipe would permit a much higher liquid film temperature difference.

2. The entrainment separator should be modified by lowering the vapor inlet to give a larger vapor space. The increased vapor space would reduce the possibility of entrainment, which gave some difficulty at high rates of heat transfer. This change in the separator would reduce the amount of liquid needed in the separator and thereby would decrease the time required to warm-up the apparatus for a run.

3. The steam chest and condensate lines should be insulated and the steam condensate during runs should be collected so that an overall heat balance could be made. This should give more reliable values of the heat transfer rate, particularly for runs having a low rate of natural circulation.

4. The method of balancing the make-up water rate with the evaporation rate should be improved. Possibly a flow meter in the make-up water line would be the answer.

5. It may be desirable to use this type of apparatus with liquid systems other than aqueous ones. However, for the present it is suggested that work be confined primarily to water and aqueous solutions because of their importance in evaporation problems and also because of the availability of physical constants, which may not be so easily found for other liquid systems.

6. It is suggested that methods be devised to heat the recirculating liquid to compensate for the heat loss it experiences in passing through the return line, and also to heat the make-up water. The use of feed liquid at near its boiling point would practically eliminate the non-boiling section of the heating tube, thereby permitting a more intensive study of heat transfer to boiling liquids.

APPENDIX

## NOMENCLATURE

- $A_{ave}$  = log mean area of the tube wall,  $ft^2$ .  
 $A_S$  = area of heating surface,  $ft^2$ .  
 $A_t$  = transverse area of the heating tube where the tube temperature is measured,  $^{\circ}F$ .  
 $c$  = specific heat of the liquid,  $Btu/\#-^{\circ}F$ .  
 $D$  = inside diameter of the heating tube,  $ft$ .  
 $G$  = mass velocity,  $\#/hr-ft^2$ .  
 $g$  = acceleration due to gravity,  $4.17 \times 10^8 \text{ ft/hr}^2$ .  
 $h_L$  = coefficient of heat transfer between the surface and the liquid,  $Btu/hr-ft^2-^{\circ}F$ ;  
 $(h_L)_{ave}$  based upon an average temperature of the liquid;  
 $(h_L)_{bp}$  based upon the liquid's boiling temperature;  
 $(h_L)_{CG}$  based upon a liquid temperature difference obtained from a relation between the sensible and latent heats.  
 $k$  = thermal conductivity of tube wall,  $Btu/(hr)(ft^2)(^{\circ}F/ft)$ .  
 $k_w$  = thermal conductivity of tube wall,  $Btu/(hr)(ft^2)(^{\circ}F/ft)$ .  
 $N_{Gr}$  = Grashof number, dimensionless,  $(D^3 \rho^2 g / \mu^2) (\beta \Delta t)$ .  
 $N_{Nu}$  = Nusselt number, dimensionless,  $hD/k$ .  
 $N_{Pr}$  = Prandtl number, dimensionless,  $c\mu/k$ .  
 $N_{Re}$  = Reynolds number, dimensionless,  $DG/\mu$ .  
 $q$  = total rate of heat transfer between the surface and the liquid,  $Btu/hr$ .  
 $q_L$  = latent heat of vaporization of liquid,  $Btu/hr$ .  
 $q_S$  = sensible heat of the liquid,  $Btu/hr$ .  
 $t_S$  = temperature of the heating surface,  $^{\circ}F$ .  
 $\Delta t_L$  = difference between temperature of heating surface and the bulk temperature of the liquid,  $^{\circ}F$ .

$(\Delta t_L)_{ave}$  uses average of outlet and inlet liquid temperatures;

$(\Delta t_L)_{bp}$  assumes liquid to be at its boiling temperature;

$(\Delta t_L)_{co}$ , a corrected value obtained from a relation between the sensible and latent heats;

$$\Delta(\Delta t_L) = (\Delta t_L)_{ave} - (\Delta t_L)_{bp}$$

$\Delta t_p$  = temperature drop from the thermocouple to the heating surface, °F.

$\Delta t_t$  = temperature drop from the thermocouple to the liquid, °F.

$x$  = distance from the center of the thermocouple junction to the heating surface, °F.

$\beta$  = volumetric coefficient of thermal expansion, °F<sup>-1</sup>.

$\mu$  = viscosity of the liquid, #/hr-ft.

$\rho$  = density of the liquid, #/ft<sup>3</sup>.



## SERIES DESIGNATIONS

S10A = a 10% sucrose solution at atmospheric pressure.

S10V = a 10% sucrose solution at pressure of 1/2 an atmosphere.

S20A = a 20% sucrose solution at atmospheric pressure.

S20V = a 20% sucrose solution at pressure of 1/2 atmosphere.

S30A = a 30% sucrose solution at atmospheric pressure.

S40A = a 40% sucrose solution at atmospheric pressure.

S40V = a 40% sucrose solution at pressure of 1/2 atmosphere.

WA = distilled water at atmospheric pressure.

WV = distilled water at 1/2 an atmosphere of pressure.

## SAMPLE CALCULATIONS

$$A_s = 3.14 \times \frac{1.049 \times 69.5}{144} = 1.59 \text{ ft}^2.$$

$$A_t = 3.14 \times \frac{1.315 - 0.06}{12} \times \frac{69.5}{12} = 1.903 \text{ ft}^2.$$

$$A_{ave} = \frac{1.903 - 1.59}{\ln \frac{1.903}{1.59}} = 1.745 \text{ ft}^2.$$

$$x = (0.133 - \frac{0.06}{2}) \frac{1}{12} = 0.00858 \text{ ft.}$$

$$D = \frac{1.049}{12} = 0.0873 \text{ ft.}$$

$$q = q_s - q_L$$

Using 9.4 for the thermal conductivity of type 316 stainless steel 16,

$$\Delta t_p = \frac{0.00858 q}{9.4 \times 1.745} = 0.00052 q$$

Calculation of the data of run S40AR4 is shown as being typical of the method used for all runs.

Feed rate	9.25	#/hr.
Recirculation rate	816.	#/hr.
Evaporation rate	9.258	#/hr.
Temperature of entering liquid	208.3	°F.
Temperature of leaving liquid	215.2	°F.
Average tube wall temperature:		
	238.5	
	236.8	
	236.0	
	235.2	
	234.6	
	238.7	
	<u>1419.8/6 =</u>	236.7 °F.

$$q_L = 9.258 \times 968 = 8,980 \text{ Btu/hr. (enthalpy from Keenan and Keyes}^9).$$

$$q_S = 0.72 (816 + 9.25) (215.2 - 208.3) = 4,050 \text{ Btu/hr.}$$

(specific heat from Spencer and Meade<sup>19</sup>).

$$q = 8980 + 4050 = 13,030 \text{ Btu/hr.}$$

$$\Delta t_p = 13,030 \times 0.00052 = 6.8^\circ \text{ F.}$$

$$t_s = 236.7 - 6.8 = 229.9^\circ \text{ F.}$$

$$(\Delta t_L)_{bp} = 229.9 - 215.2 = 14.7$$

$$(h_L)_{bp} = \frac{13,030}{14.7 \times 1.59} = 558 \text{ Btu/hr-ft}^2\text{-}^\circ\text{F.}$$

$$(\Delta t_L)_{ave} = \frac{(229.9 - 208.3) - (229.9 - 215.2)}{\ln \frac{229.9 - 208.3}{229.9 - 215.2}} = 18.1^\circ \text{ F.}$$

$$(h_L)_{ave} = \frac{13,030}{18.1 \times 1.59} = 453 \text{ Btu/hr-ft}^2\text{-}^\circ\text{F.}$$

$$(\Delta t_L)_{co} = 14.7 + (18.1 - 14.7) \frac{4,050}{13,030} = 15.7^\circ \text{ F.}$$

$$(h_L)_{co} = \frac{13,030}{15.7 \times 1.59} = 522 \text{ Btu/hr-ft}^2\text{-}^\circ\text{F.}$$

$$(\Delta t_t)_{co} = 15.7 + 6.8 = 22.5^\circ \text{ F.}$$

$$G = \frac{816 + 9.25}{3.14 (1.049)^2} \times 4 \times 144 = 137,500 \text{ \#/hr-ft}^2.$$

TABLE I

Experimental Data For Water And 10% Sucrose Solution

Run	Steam Press. Psig	Tube Temp. °F.	Inlet Temp. °F.	Outlet Temp. °F.	Feed Rate #/hr.	Circ. Rate #/hr.	Evap. Rate #/hr.
WA1	3.0	217.1	169.3	212.0	0.0	74	1.104
WA2	4.0	218.9	188.4	212.0	0.0	123	2.142
WA3a	4.8	221.4	201.5	212.0	0.0	246	3.488
WA3b	5.5	223.7	205.4	211.8	0.0	591	5.478
WA4b	7.5	226.6	207.6	212.2	0.0	1107	9.262
WA4a	7.8	227.4	207.0	211.9	0.0	1132	9.574
WA5	10.8	231.7	206.5	211.7	12.7	1477	15.810
WA6	15.3	238.9	206.0	211.7	27.1	1675	24.836
WA7	20.5	246.0	206.5	211.7	27.1	1725	35.670
WA8	25.3	252.7	205.5	211.8	35.2	1700	43.258
WA9	30.3	258.2	205.0	211.8	40.9	1675	51.638
WA10	35.3	263.3	204.5	211.8	44.4	1625	58.534
WA11	40.0	267.7	203.1	212.0	57.1	1550	62.988
WA12	45.0	271.5	203.9	212.5	60.5	1500	69.828
WA13	50.5	279.2	203.2	212.7	61.1	1450	75.450
WA14	57.5	284.2	201.5	212.7	67.5	1353	81.974
WV1	5.8	213.8	174.9	179.6	12.1	1630	32.916
WV2	10.0	223.5	174.3	180.0	28.3	1615	42.610
WV3	15.0	231.2	172.1	180.3	56.5	1530	50.242
WV4	20.5	238.8	172.4	180.0	49.0	1445	61.388
WV5	25.0	247.8	172.5	180.9	53.6	1295	69.334
WV6	29.0	252.0	171.7	181.0	63.4	1330	72.530
WV7	35.0	258.4	171.5	180.9	57.7	1235	81.918
WV8	40.0	263.1	170.5	181.0	62.2	1060	88.782
WV9	45.5	268.6	168.1	181.3	85.8	1020	91.416
S10A10	4.0	220.3	193.6	212.0	0.0	125	2.656
S10A3	6.0	224.1	205.2	212.0	0.0	399	4.779
S10A2	9.0	230.2	206.9	211.8	9.8	1298	11.228
S10A1	14.0	239.9	206.5	211.8	22.5	1672	23.184
S10A4	19.5	247.2	205.9	212.0	28.9	1772	31.312
S10A5	30.0	260.8	205.7	212.3	39.2	1747	45.670
S10A6	40.5	271.2	204.0	212.2	49.6	1646	58.424
S10A7	45.0	275.3	203.7	212.4	55.4	1547	64.164
S10A8	52.0	281.4	203.1	212.5	55.4	1472	71.802
S10A9	56.3	284.6	201.2	212.4	66.9	1378	74.334
S10V1	2.0	207.3	174.9	180.6	15.3	1302	16.108
S10V2	7.0	220.0	174.7	180.2	23.0	1528	30.484
S10V3	15	234.4	173.4	180.4	35.7	1342	48.136
S10V4	25.5	249.2	170.4	180.6	57.7	1052	66.740
S10V5	36.0	260.6	168.5	181.0	61.7	926	81.884
S10V6	46.0	270.0	166.3	181.7	66.9	762	83.676

TABLE II

Calculated Data For water and 10% Sucrose Solution

Run	Latent Heat BTU/hr.	Sensible Heat BTU/hr.	Total Heat BTU/hr.	$\Delta t_p$ °F.	$(\Delta t_L)_{ave}$ °F.	$(h_L)_{ave}$	$(\Delta t_L)_{bp}$ °F.	$(h_L)_{bp}$
WA1	1,072	3,160	4,232	2.2	15.5	172	2.9	916
WA2	2,080	2,902	4,982	2.6	12.6	249	4.3	729
WA3a	3,385	2,585	5,970	3.1	10.7	350	6.3	596
WA3b	5,310	3,790	9,100	4.7	10.1	567	7.2	795
WA4b	8,990	5,080	14,070	7.3	9.4	940	7.1	1245
WA4a	9,290	5,550	14,840	7.7	10.1	925	7.8	1196
WA5	15,350	7,670	23,020	12.0	10.6	1380	8.0	1810
WA6	24,100	9,550	33,650	17.4	12.6	1680	9.8	2160
WA7	34,600	8,970	43,570	22.6	14.3	1920	11.7	2340
WA8	42,000	10,720	52,720	27.3	16.7	1985	13.6	2440
WA9	50,100	11,400	61,500	31.9	17.9	2160	14.5	2670
WA10	56,800	11,860	68,660	35.6	19.5	2220	15.9	2710
WA11	61,100	13,800	74,900	38.8	21.3	2210	16.9	2790
WA12	67,700	12,900	80,600	41.8	21.5	2360	18.2	2785
WA13	73,200	13,780	86,980	46.1	25.1	2180	20.4	2685
WA14	79,500	15,160	94,660	49.1	28.0	2125	22.4	2660
WV1	32,550	8,630	41,180	21.7	14.8	1775	12.5	2070
WV2	42,200	9,200	51,400	26.6	19.7	1640	16.9	1910
WV3	49,700	12,540	62,240	32.3	22.7	1728	18.6	2110
WV4	60,700	10,980	71,680	37.1	25.5	1770	21.7	2075
WV5	68,600	10,880	79,480	41.2	29.9	1670	25.7	1945
WV6	71,800	12,560	84,360	43.7	31.9	1660	27.3	1940
WV7	81,100	11,620	92,720	48.1	34.1	1710	29.4	1984
WV8	87,800	11,130	98,930	51.2	36.1	1725	30.9	2010
WV9	90,500	13,460	103,960	53.9	40.0	1633	33.4	1955
S10A10	2,575	2,140	4,715	2.5	12.9	230	5.8	512
S10A3	4,630	2,525	7,155	3.7	11.5	391	8.4	536
S10A2	10,900	5,910	16,810	8.7	12.1	873	9.7	1090
S10A1	22,500	8,240	30,740	15.9	14.8	1307	12.1	1585
S10A4	30,350	10,050	40,400	20.9	17.3	1470	14.3	1775
S10A5	44,300	10,720	55,020	28.5	23.3	1485	20.0	1730
S10A6	56,600	12,560	69,160	35.9	27.2	1600	23.1	1884
S10A7	62,620	12,070	74,690	38.7	28.5	1648	24.2	1942
S10A8	69,700	12,880	82,580	42.8	30.8	1686	26.1	1990
S10A9	72,100	14,350	86,450	44.8	33.0	1646	27.4	1983
S10V1	15,950	6,900	22,850	11.7	17.8	808	15.0	958
S10V2	34,500	7,810	42,310	21.9	20.1	1324	17.9	1486
S10V3	47,600	8,740	56,340	29.2	28.3	1250	24.8	1428
S10V4	66,100	9,980	76,080	39.4	34.3	1396	29.2	1635
S10V5	81,000	10,780	91,780	47.6	38.2	1510	32.0	1802
S10V6	82,800	10,910	93,710	48.6	47.4	1245	39.7	1485

TABLE III

Calculated Data For Water And 10% Sucrose Solution (Continued)

Run	$\Delta(\Delta t_L)$	Correc- tion	$(\Delta t_L)_{co}$	$(h_L)_{co}$	$(\Delta t_t)_{co}$	$\frac{G}{\text{hr-ft}^2}$	$\frac{DG}{\mu}$	$\frac{hD/k}{\left(\frac{c_p \mu}{k}\right)^{0.4}}$
WA1	12.6	9.4	11.3	235	13.5	12,340	1,375	39.2
WA2	8.3	4.8	9.1	343	11.7	20,500	2,425	59.0
WA3a	4.4	1.9	8.2	457	11.3	41,000	5,000	79.2
WA3b	2.9	1.2	8.4	681	13.1	98,500	12,300	119.2
WA4b	2.3	0.8	7.9	1118	15.2	184,200	23,400	197
WA4a	2.3	0.9	8.7	1072	16.4	189,000	24,000	190
WA5	2.6	0.9	8.9	1626	20.9	248,000	31,500	287
WA6	2.8	0.8	10.6	1995	28.0	284,000	36,100	352
WA7	2.6	0.5	12.2	2245	34.8	292,000	37,100	396
WA8	3.1	0.6	14.2	2330	41.5	289,000	36,100	408
WA9	3.4	0.6	15.1	2560	47.0	286,000	35,700	448
WA10	3.6	0.6	16.5	2620	52.1	278,000	34,400	458
WA11	4.4	0.8	17.7	2660	56.5	268,000	33,000	462
WA12	3.3	0.5	18.7	2710	60.5	260,000	32,000	471
WA13	4.7	0.7	21.1	2590	67.2	252,000	31,000	451
WA14	5.6	0.9	23.3	2550	72.4	237,000	28,900	442
WV1	2.3	0.5	13.0	1990	34.7	274,000	27,500	321
WV2	2.8	0.5	17.4	1855	44.0	274,000	27,500	298
WV3	4.1	0.8	19.4	2015	51.7	264,500	26,700	324
WV4	3.8	0.6	22.3	2020	59.4	249,000	25,200	325
WV5	4.2	0.6	26.3	1900	67.5	224,500	22,700	306
WV6	4.6	0.7	28.0	1890	71.7	232,500	23,000	304
WV7	4.7	0.6	30.0	1944	78.1	216,000	21,850	312
WV8	5.2	0.6	31.5	1975	82.7	187,000	18,900	317
WV9	6.6	0.9	34.3	1905	88.2	184,000	18,600	307
S10A10	7.1	3.2	9.0	330	11.5	20,800	2,030	56.3
S10A3	3.1	1.1	9.5	474	13.2	66,400	6,650	82
S10A2	2.4	0.8	10.5	1005	19.2	217,500	22,400	175
S10A1	2.6	0.7	12.9	1498	28.8	282,500	29,100	262
S10A4	3.0	0.7	15.0	1692	35.9	300,000	30,900	296
S10A5	3.3	0.6	20.6	1680	49.1	297,500	30,600	293
S10A6	4.1	0.7	23.8	1830	59.7	282,500	29,100	319
S10A7	4.3	0.7	24.9	1887	63.6	267,000	27,500	329
S10A8	4.7	0.7	26.8	1940	69.8	254,000	26,200	338
S10A9	5.6	0.9	28.3	1920	73.1	241,000	24,850	335
S10V1	2.8	0.8	15.8	910	27.5	219,000	17,620	144
S10V2	2.2	0.4	18.3	1455	40.2	258,500	20,800	230
S10V3	3.5	0.5	25.3	1373	54.5	229,000	18,430	217
S10V4	5.1	0.7	29.9	1600	69.3	185,000	14,900	253
S10V5	6.2	0.7	32.7	1765	80.3	165,000	13,280	280
S10V6	7.7	1.1	40.8	1442	89.4	138,000	11,100	228

TABLE IV

Calculated Data For Water And 10% Sucrose Solution (Concluded)

Run	N <sub>Nu</sub>	N <sub>Pr</sub>	Aubrecht Correl*	Thors. Correl**	N <sub>Gr</sub> x 10 <sup>-3</sup>	(N <sub>Gr</sub> )(N <sub>Pr</sub> )10 <sup>-3</sup>
WAL	51.5	1.97	426	24.6	804	1,582
2	75.3	1.85	421	30.4	734	1,358
3a	100.2	1.80	404	34.2	700	1,260
3b	149.5	1.755	472	41.6	752	1,320
4b	236.5	1.725	625	56.7	805	1,388
4a	245	1.725	598	59.2	731	1,260
5	357	1.725	895	81.5	823	1,420
6	438	1.725	1,252	97.3	980	1,690
7	493	1.725	1,610	108.9	1,129	1,945
8	512	1.725	1,970	115	1,270	2,230
9	562	1.725	2,305	126.7	1,351	2,370
10	575	1.77	2,600	131	1,451	2,570
11	584	1.785	2,870	135.2	1,535	2,740
12	595	1.785	3,130	139	1,621	2,890
13	569	1.785	3,410	133.5	1,830	3,265
14	560	1.80	3,790	134	1,858	3,345
WV1	437	2.17	1,696	121.0	763	1,658
2	407	2.17	2,115	113.0	1,021	2,215
3	442	2.17	2,590	123.5	1,139	2,470
4	443	2.17	3,040	125.4	1,310	2,845
5	417	2.17	3,485	120.5	1,545	3,350
6	415	2.17	3,655	119.1	1,645	3,570
7	426	2.17	4,100	124.2	1,760	3,820
8	433	2.17	4,570	129.8	1,850	4,010
9	418	2.17	4,850	125.8	2,010	4,360

$$* \text{ Aubrecht Correlation data} = \frac{g}{(60)^{0.3}} \left( 1 - 0.79 \frac{k^{0.6} c^{0.4}}{\mu^{0.4}} \right)$$

\*\* Thorstenberg Correlation data

$$= \frac{k_L}{(60)^{0.2}} \left( 1.0 - 1.10 \frac{k^{0.6} c^{0.4}}{\mu^{0.4}} \right)$$

TABLE V

## Experimental Data For 20, 30, And 40% Sucrose Solutions

Run	Steam Press Psig	Tube Temp. °F.	Inlet Temp. °F.	Outlet Temp. °F.	Feed Rate #/hr.	Circ. Rate #/hr.	Evap. Rate #/hr.
S20A2	2.8	216.5	142.1	212.5	0.0	51	0.294
S20A1	4.0	220.1	188.3	212.6	0.0	102	2.004
S20A3	6.5	226.1	205.9	212.6	0.0	457	6.022
S20A4	11.5	235.2	208.2	212.7	13.3	1560	15.704
S20A5b	20.0	249.0	206.0	212.7	23.1	1650	31.124
S20A5a	20.3	248.7	207.1	212.5	21.4	1675	32.250
S20A6	32.0	263.9	204.4	213.1	42.9	1345	47.630
S20A7	41.0	273.2	203.0	213.5	48.5	1243	59.434
S20A8	52.0	282.3	201.1	213.4	51.5	1066	72.796
S20V1	2.0	210.0	174.5	181.2	12.4	1283	16.580
S20V2	9.0	227.6	174.1	181.3	25.4	1222	36.322
S20V3	20.3	243.8	172.0	181.0	39.3	1070	58.404
S20V4	33.0	258.4	168.7	181.6	51.5	815	77.960
S20V5	45.5	269.5	165.2	181.0	58.9	713	93.998
S30A1	4.0	220.4	177.2	214.0	0.0	78	1.260
S30A8	6.0	225.9	202.6	214.7	0.0	259	3.578
S30A2	10.0	234.6	208.2	214.1	12.7	1060	10.500
S30A3	20.5	251.7	207.8	214.0	27.7	1542	29.068
S30A4	35.0	268.1	205.5	214.0	35.8	1273	51.170
S30A5	40.3	273.7	202.6	215.2	47.4	1045	57.554
S30A6	50.0	282.4	199.2	215.0	59.4	957	68.234
S40AR3	4.0	220.7	162.3	214.6	0.0	53	0.862
S40A1	4.0	221.0	136.3	216.0	0.87	26	0.602
S40AR2	5.0	223.8	188.0	214.9	0.0	105	1.936
S40A2	6.5	228.3	191.6	216.3	0.0	184	2.862
S40A3	9.0	233.9	209.5	216.6	4.62	685	6.212
S40AR1	9.5	233.8	206.1	214.9	8.1	579	6.650
S40AR4	10.5	236.7	208.3	215.2	9.25	816	9.258
S40AR5	15.0	244.8	209.9	215.3	12.7	1580	17.328
S40AR6	19.5	251.0	209.9	215.3	16.2	1632	24.660
S40AR7	26.8	260.9	208.9	215.7	24.3	1527	34.154
S40AR8	33.3	268.2	208.8	215.6	24.8	1448	43.666
S40AR9	42.0	276.2	207.5	215.7	34.7	1370	53.134
S40AR10	46.5	280.5	206.0	216.0	44.2	1316	57.354
S40AR11	52.0	284.6	204.2	216.0	52.0	1238	62.328
S40V1	2.5	213.6	177.9	183.9	10.4	1135	10.626
S40V2	7.0	225.6	177.9	183.6	24.8	1415	20.896
S40V3	12.0	234.5	176.6	183.7	33.5	1490	31.946
S40V4	24.0	251.2	174.3	183.9	45.1	1161	53.740



TABLE VI

Calculated Data For 20, 30, And 40% Sucrose Solutions

Run	Latent Heat BTU/hr.	Sensible Heat BTU/hr.	Total Heat BTU/hr.	$\Delta t_p$ °F.	$(\Delta t_L)_{ave}$ °F.	$(h_L)_{ave}$	$(\Delta t_L)_{bp}$ °F.	$(h_L)_{bp}$
S20A2	285	3,090	3,375	1.8	20.1	105	2.2	965
S20A1	1,940	2,130	4,070	2.1	14.3	179	5.4	474
S20A3	5,840	2,635	8,475	4.4	12.1	441	9.1	585
S20A4	15,230	6,040	21,270	11.0	13.7	976	11.5	1164
S20A5b	30,200	9,510	39,710	20.6	19.0	1313	15.7	1590
S20A5a	31,300	7,770	39,070	20.3	18.6	1322	15.9	1545
S20A6	46,200	10,070	56,270	29.2	25.9	1366	21.6	1640
S20A7	57,600	11,230	68,830	35.7	29.2	1480	24.0	1801
S20A8	70,600	11,280	71,880	37.3	37.7	1200	31.6	1430
S20V1	16,410	7,400	23,810	12.4	19.7	760	16.4	913
S20V2	35,950	7,570	43,520	22.5	27.4	997	23.8	1150
S20V3	57,800	8,280	66,080	34.2	33.1	1252	28.6	1451
S20V4	77,100	9,050	86,150	44.7	38.5	1390	32.1	1688
S20V5	93,100	9,700	102,800	53.3	43.1	1500	35.2	1833
S30A1	1,222	2,270	3,492	1.8	16.8	131	4.6	477
S30A8	3,470	2,480	5,950	3.1	13.3	281	8.1	462
S30A2	10,200	4,940	15,140	7.8	15.5	615	12.7	750
S30A3	28,250	7,560	35,810	18.6	22.2	1015	19.1	1180
S30A4	49,700	8,550	58,250	30.2	28.1	1305	23.9	1532
S30A5	55,800	10,400	66,200	34.3	30.5	1365	24.2	1720
S30A6	66,200	11,950	78,150	40.5	34.8	1413	26.9	1826
S40AR3	837	1,994	2,831	1.5	20.9	85	4.6	387
S40A1	584	1,492	2,076	1.1	26.1	50	3.9	335
S40AR2	1,879	2,020	3,899	2.0	17.1	143	7.1	346
S40A2	2,780	3,275	6,055	3.1	18.6	205	8.9	429
S40A3	6,020	3,500	9,520	4.9	15.7	381	12.4	482
S40AR1	6,450	3,670	10,120	5.2	17.8	358	13.7	465
S40AR4	8,980	4,050	13,030	6.8	18.1	453	14.7	558
S40AR5	16,820	6,150	22,970	11.9	20.3	711	17.6	821
S40AR6	23,900	6,350	30,250	15.7	22.7	838	20.0	952
S40AR7	33,100	7,470	40,570	21.0	27.6	925	24.2	1055
S40AR8	42,300	7,080	49,380	25.6	30.4	1022	27.0	1150
S40AR9	51,600	8,090	59,690	30.9	33.7	1114	29.6	1268
S40AR10	55,600	9,470	65,070	33.7	35.8	1142	30.8	1329
S40AR11	60,500	10,510	71,010	36.8	37.7	1185	31.8	1405
S40V1	10,520	4,900	15,420	8.0	24.7	393	21.7	446
S40V2	20,700	5,810	26,510	13.7	31.1	536	28.3	589
S40V3	31,600	7,610	39,210	20.3	34.0	725	30.5	809
S40V4	53,200	8,030	61,230	31.7	41.4	930	35.6	1082

TABLE VII

Calculated Data For 20, 30, And 40% Sucrose Solutions (Continued)

Run	$(\Delta t_L)$	Correc- tion	$(\Delta t_L)_{co}$	$(h_L)_{co}$	$(\Delta t_t)_{co}$	$\frac{G}{\text{hr-ft}^2}$	$\frac{DG}{\mu}$	$\frac{hD/k}{\left(\frac{c_p \mu}{k}\right)^{0.4}}$
S20A2	17.9	16.4	18.6	114	20.4	8,500	528	17.4
S20A1	8.9	4.7	10.1	253	12.2	17,000	1,228	40.8
S20A3	3.0	0.9	10.0	533	14.4	76,200	6,000	89.3
S20A4	2.2	0.6	12.1	1105	23.1	262,000	20,600	185.4
S20A5b	3.3	0.8	16.5	1513	37.1	279,000	21,950	253
S20A5a	2.7	0.5	16.4	1500	36.7	282,000	22,200	251
S20A6	4.3	0.8	22.4	1580	51.6	231,000	18,500	264
S20A7	5.2	0.9	24.9	1740	60.6	216,000	17,000	292
S20A8	6.1	1.0	32.6	1385	69.9	186,000	14,630	232
S20V1	3.3	1.0	17.4	860	29.8	216,000	13,200	130
S20V2	3.6	0.6	24.4	1120	46.9	208,000	12,720	169
S20V3	4.5	0.6	29.2	1422	63.4	185,000	11,300	215
S20V4	6.4	0.7	32.8	1650	77.5	144,700	8,850	250
S20V5	7.9	0.7	35.9	1800	89.2	128,800	7,820	272
S30A1	12.2	7.9	12.5	176	14.3	13,000	650	26.3
S30A8	5.2	2.2	10.3	364	13.4	43,200	2,360	56.5
S30A2	2.8	0.9	13.6	700	21.4	179,000	10,100	110
S30A3	3.1	0.7	19.8	1135	38.4	262,000	14,780	178
S30A4	4.2	0.6	24.5	1495	54.7	218,000	12,300	235
S30A5	6.3	1.0	25.2	1650	59.5	182,000	11,250	260
S30A6	7.9	1.2	28.1	1745	68.6	169,500	9,550	274
S40AR3	16.3	11.5	16.1	111	16.6	8,850	271	14.8
S40A1	22.2	16.0	19.9	66	21.0	4,340	117	8.4
S40AR2	10.0	5.2	12.3	199	14.3	17,500	602	27.9
S40A2	9.7	5.3	14.2	268	17.3	30,650	1,082	37.9
S40A3	3.3	1.2	13.6	440	18.5	115,000	4,420	64.2
S40AR1	4.1	1.5	15.2	418	20.4	96,500	3,710	61.0
S40AR4	3.4	1.0	15.7	522	22.5	137,500	5,290	76.3
S40AR5	2.7	0.7	18.3	790	30.2	266,000	10,230	115.5
S40AR6	2.7	0.6	20.6	924	36.3	274,500	10,580	135.0
S40AR7	3.4	0.6	24.8	1030	45.8	258,500	9,920	150.0
S40AR8	3.4	0.5	27.5	1128	53.1	245,500	9,450	164.5
S40AR9	4.1	0.6	30.2	1244	61.1	234,500	9,020	181.4
S40AR10	5.0	0.7	31.5	1300	65.2	226,500	8,720	189.5
S40AR11	5.9	0.9	32.7	1365	69.5	215,000	8,270	199.2
S40V1	3.0	1.0	22.7	428	30.7	191,000	5,300	54.4
S40V2	2.8	0.6	28.9	576	42.6	240,000	6,660	73.8
S40V3	3.5	0.7	31.2	790	51.5	254,000	7,050	101.4
S40V4	5.8	0.8	36.4	1058	68.1	201,000	5,580	135.2

TABLE VIII

Calculated Data For 20, 30, And 40% Sucrose Solutions (Concluded)

Run	NNu	NPr	Aubrecht Correl.*	Thors. Correl.**	NGr x 10 <sup>-3</sup>	(NGr)(NPr)10 <sup>-3</sup>
S40AR3	31.7	6.71	491	28.1	91.1	611
S40A1	18.9	7.63	451	19.8	87.3	666
S40AR2	56.9	5.97	542	43.0	114.8	685
S40A2	76.5	5.82	688	50.3	107.0	623
S40A3	125.9	5.35	740	63.8	121.3	649
S40AR1	119.5	5.35	835	62.7	135.7	725
S40AR4	149.2	5.35	963	73.1	140	749
5	226	5.35	1,393	96.7	163	873
6	264	5.35	1,826	112.5	184	985
7	294	5.35	2,490	126.8	221	1,182
8	322	5.35	3,065	140.4	246	1,313
9	355	5.35	3,770	156.2	269	1,440
10	371	5.35	4,150	164.2	281	1,505
11	390	5.35	4,590	174.7	292	1,563
S40V1	122	7.40	1,076	59.5	105.8	782
2	165	7.40	1,725	76.7	134.5	991
3	226	7.40	2,510	104.0	145	1,072
4	302	7.40	4,210	145.8	169.3	1,254

\* Aubrecht Correlation data =  $\frac{g}{\left(\frac{G}{60}\right)^{0.3}} \left(1 - 0.79 \frac{k^{0.6} c^{0.4}}{\mu^{0.4}}\right)$

\*\* Thorstenberg Correlation data

$$= \frac{h_l}{\left(\frac{G}{60}\right)^{0.2}} \left(1.0 - 1.10 \frac{k^{0.6} c^{0.4}}{\mu^{0.4}}\right)$$

FIGURE A1. TUBE WALL TEMPERATURE DISTRIBUTION FOR WATER RUNS.

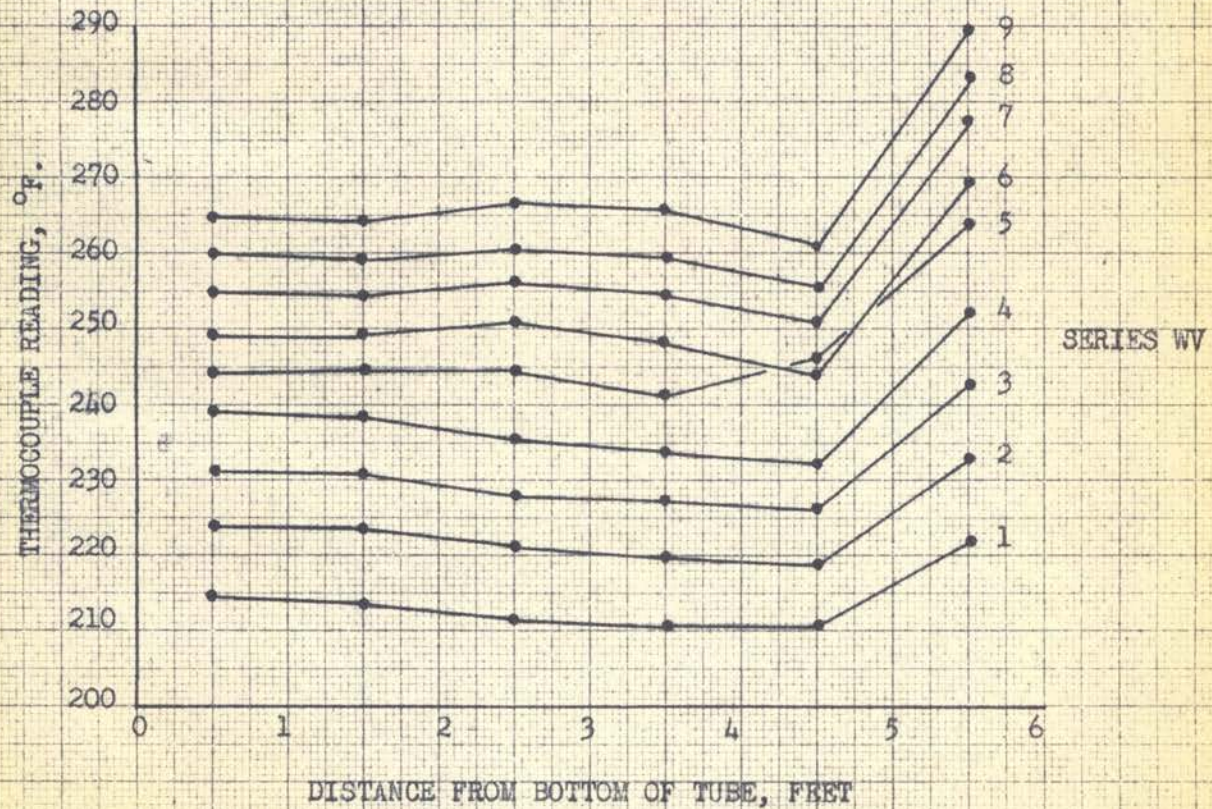
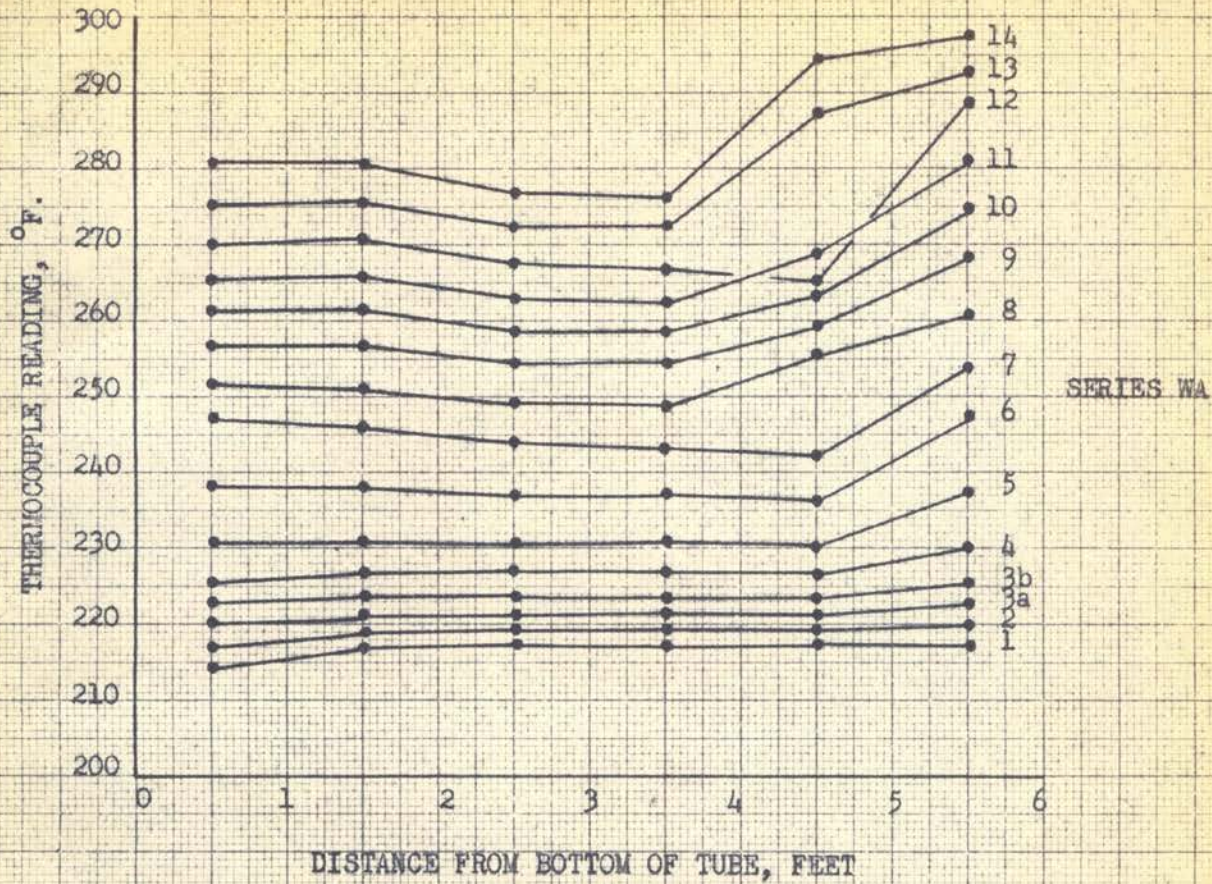
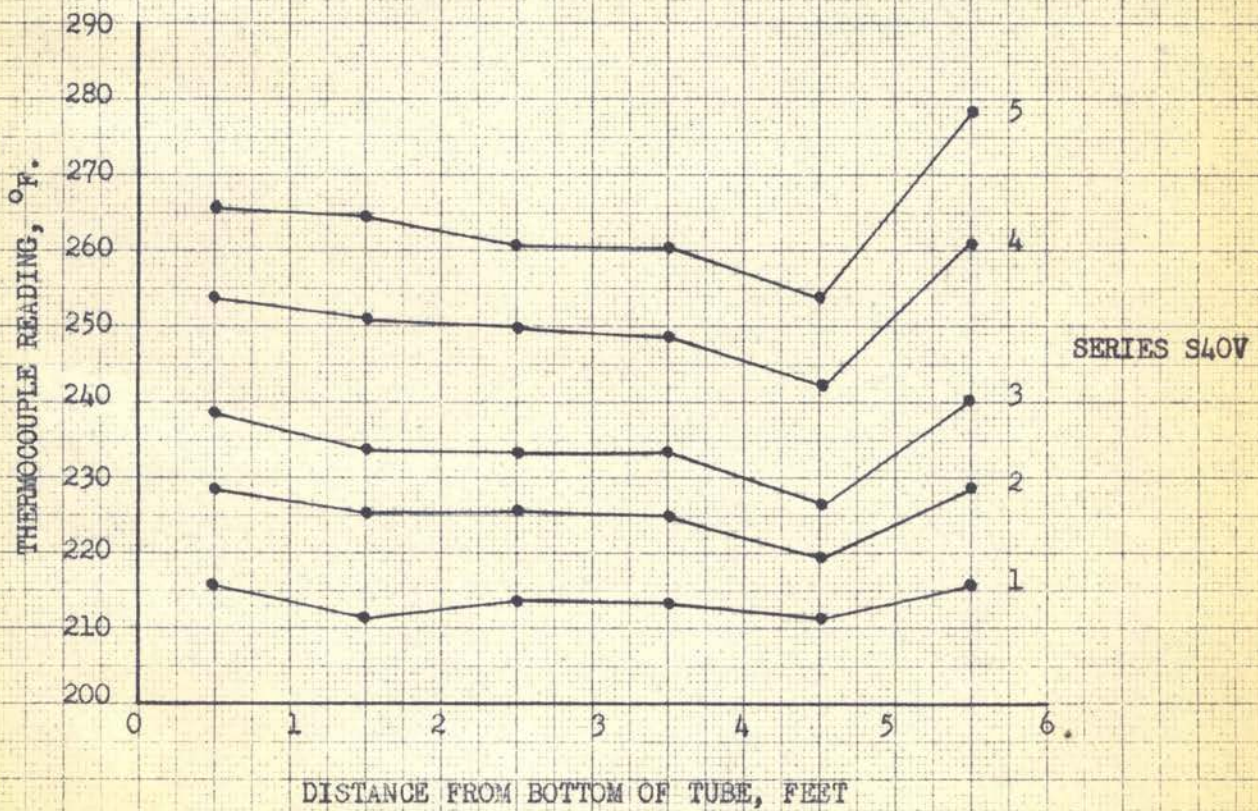
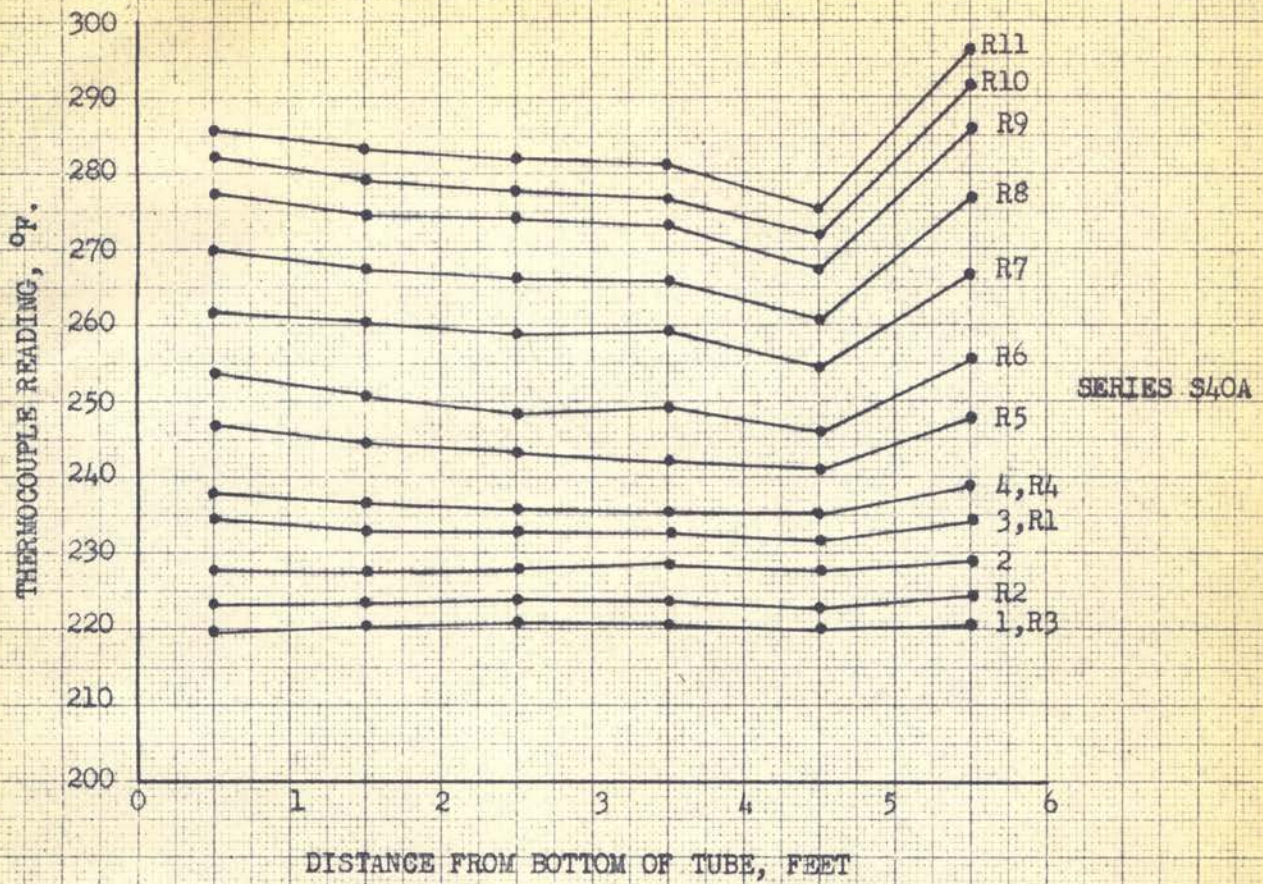


FIGURE A2. TUBE WALL TEMPERATURE DISTRIBUTION FOR 40% SUCROSE RUNS.



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