AN INVESTIGATION OF THE EFFECT OF SCALE FORMATION ON THE HEAT TRANSFER COEFFICIENT IN AN INCLINED TUBE EVAPORATOR

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By

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Bachelor of Science

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1943

Submitted to the Department of Chemical Engineering

Oklahoma Agricultural and Mechanical College

In Partial Fulfillment of the Requirements

for the Degree of

MASTER OF SCIENCE

1948

ACKNOWLEDGMENT

The author wishes to take this opportunity to express appreciation to Dr. D. A. Aubrecht under whose advisorship this investigation was undertaken and to Dr. C. L. Nickolls under whom most of the work was done due to the departure of Dr. Aubrecht. Invaluable aid was also supplied by Mr. E. W. Steffee in construction of the equipment and in running the tests.

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PREFACE

The field of evaporation has been of major importance in industry for many years and remains so at the present time. To attempt an investigation of more than a very small phase of the field and of the many problems which have developed would be an undertaking of great magnitude.

This thesis is concerned with two problems; the construction and testing of an experimental inclined tube evaporator and the determination of the effect of scale formation on the liquid side heat transfer coefficient.

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Introduction and Review of Selected Literature

The basic equation for heat conduction was developed by Fourier. This law states that the instantaneous rate of heat flow is equal to the product of three factors; the area of the section taken at right angles to the direction of heat flow, the temperature gradient over the path of heat flow and a proportionality factor k, known as thermal conductivity. k will very with different substances. Expressing this equation mathematically we have:

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

where:

Q = Amount of heat flowing

θ = time interval chosen

A = area through which the heat is flowing

t = temperature

x = length of the path of heat flow

Since most heat transfer calculations are carried out using English units, Q will be in B.t.u., θ in hours, A in square feet, t in °F and x in feet. The English system will be used in this report.

In steady state conduction the temperature at any one point remains the same making $\frac{dt}{dx}$ a constant. On integrating we obtain:

$$\frac{Q}{\theta} = - kA\frac{dt}{dx} \qquad (2)$$

If k is a straight line function of t, we obtain by integration of equation (2):

$$\frac{Q}{\theta} = q = \frac{k_{Avg}A\Delta t}{x}$$

Rearrangement of the equation will give:

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

This reduces the equation to the form of the fundamental rate equation:

$$\text{Rate} = \frac{\text{Driving force}}{\text{Resistance}}$$

If heat is flowing through several homogeneous solids in series then:

$$q = \frac{\Delta t_1}{R_1} = \frac{\Delta t_2}{R_2} = \frac{\Delta t_3}{R_3} \text{......or}$$

$$q = \frac{\Delta t_{total}}{R_1 + R_2 + R_3} \qquad (4)$$

When considering heat flow across a fluid film the determination of film thickness is very difficult and in all practical cases unwarranted. Until recent years investigations of heat flow across liquid films were concerned only with determining what is known as the overall heat transfer coefficient defined by the equation:

$$q = UA\Delta t \qquad (5)$$

The disadvantage of this method lies in the fact that a separate coefficient must be determined for each set of conditions. If individual coefficients can be determined, the time spent in experimental determinations can be greatly decreased. Taking as an example the transfer of heat from one fluid to another through a metal wall, we will have three resistances in series to the flow of heat; the film of liquid number one, the metal wall and the film of liquid number two. Designating the individual film coefficient as h we obtain the following equations:

$$q = h_1A_1\Delta t_1 = \frac{\Delta t_2}{x/kA} = h_3A_3\Delta t_3$$

or:

$$q = \frac{\Delta t_1}{R_1} = \frac{\Delta t_2}{R_2} = \frac{\Delta t_3}{R_3} = \frac{\Delta t_{total}}{R_1 + R_2 + R_3} \qquad (6)$$

then:

$$q = \frac{\Delta t_{total}}{\frac{1}{h_1A_1} + \frac{x}{k_2A_2} + \frac{1}{h_3A_3}} \qquad (7)$$

and since:

$$q = UA\Delta t$$

then:

$$UA = \frac{1}{\frac{1}{h_1A_1} + \frac{x}{k_2A_2} + \frac{1}{h_3A_3}} \qquad (8)$$

From equation (8) we see the relation between individual and overall coefficients and the ease with which overall coefficients may be calculated for different combinations of the individual coefficients.

In evaporation in addition to the resistance of a metal wall we are interested in two individual film coefficients, that of the boiling liquid and also that of the condensing vapor. This investigation will be concerned primarily with the boiling liquid coefficient. The specific problem to be investigated is scale formation and its effect on the boiling liquid coefficient in an inclined tube evaporator.

Some of the early work with inclined tube evaporators was carried out by Van Marle[16], Cleve[3], and Linden and Montillon[7]. The results of these investigators are not comparable on a common basis. Different types of equipment were used and the methods of correlation were not similiar.

Van Marle working with a seven tube evaporator inclined at an angle of forty-five degrees investigated overall coefficients and found a straight line relation between the overall coefficient and the temperature of evaporation.

Cleve worked with a small electrically heated tube inclined at angles of forty-five, twenty and ten degrees. He found that the velocity increased with an increase in Δt until a maximum was reached. As Δt increased further, the velocity decreased.

Linden and Montillon correlated their data on a single copper tube inclined at an angle of forty-five degrees by use of a modified Dittus-Boelter equation. The tube was kept clean during the investigation and distilled water was used.

Aubrecht[1] in a comprehensive study of the evaporation of sucrose solutions in an apparatus similiar to that used by Linden and Montillon correlated his data by the following equation:

$$\left(\frac{Q}{\theta}\right)\left(\frac{1}{G^n}\right)\left(1 - .79\,\frac{k^{.6}c^{.4}}{\mu^{.4}}\right) = \psi \Delta t^m \qquad (9)$$

For forced circulation:

	n	ψ	m
Below Q/θ of 30,000 B.t.u./hr.	-.3	17.65	1.55
Above Q/θ of 30,000 B.t.u./hr.	-.1	184.8	1.55

For natural circulation:

$n = + .3$
$\psi = 1.124$
$m = 2.55$

Where:

G = mass velocity lb./sq.ft./sec.
k = thermal conductivity B.t.u./ft.^{2}F- hr./ft.
μ = viscosity in centipoises
c = sp. heat in B.t.u./lb./°F.

He found that with forced circulation above five feet per second the Dittus-Boelter equation will hold due to absence of boiling in the tube. Below this velocity the Dittus-Boelter equation will not hold except for very low temperature drops. With forced circulation he found that a maximum Q/θ was found for a given Δt as velocity was increased. This occurred at a heat load of approximately 30,000 B.t.u./hr. for his apparatus. In natural circulation the velocity was found to increase until a heat load maximum of approximately 30,000 B.t.u./hr. was reached. The velocity of circulation then decreased. Temperature drop was found to have less effect on Q/θ using natural circulation than with forced circulation. Changes in viscosity due to an increase in the sucrose concentration had more effect on Q/θ at low concentrations than at high concentrations.

Syverson[14] determined liquid side coefficients in a clean vertical tube evaporator of an experimental type. The tube was electrically heated which provided for greater accuracy in measuring heat input and losses.

Pridgeon and Badger[12] in an investigation on a submerged horizontal copper tube, semi-commercial evaporator found that a scale thickness of 0.0079 inches reduced the overall coefficient from 820 B.t.u./hr.-sq.ft.-°F. to 580. The scale was presumed to be $CaCO_3$ from the city water mixed with dirt which was present in the equipment.

McCabe and Robinson[9] found that in two out of three sets of data gathered from independent workers, the relation between rate of scale formation and overall coefficient would fit the equation:

$$\frac{1}{U^2} = A_1 + A_2\theta \qquad (10)$$

A_1 and A_2 are empirical constants and θ is any unit of time. In the data which failed to follow this relation it was found that on plotting, a straight line relation was obtained for the first three hours and also for the interval between the sixth and tenth hours. The investigators suggest that the discrepancy was caused by scale breaking off due to some change in evaporator conditions.

Badger[2] in an investigation concerned with the evaporation of waste sulfite liquor found no scale had formed after 150 hours of operation. Forced circulation was used for the investigation and a high circulation rate was believed to be responsible for preventing scale deposits. Numerical values for rate of circulation were not given.

Sauer, Cooper, Akin and McAdams[13] found that with water boiling outside submerged copper tubes a very thin film of scale will cause a substantial increase in the individual heat transfer coefficients. No work was conducted on thicker scale formation.

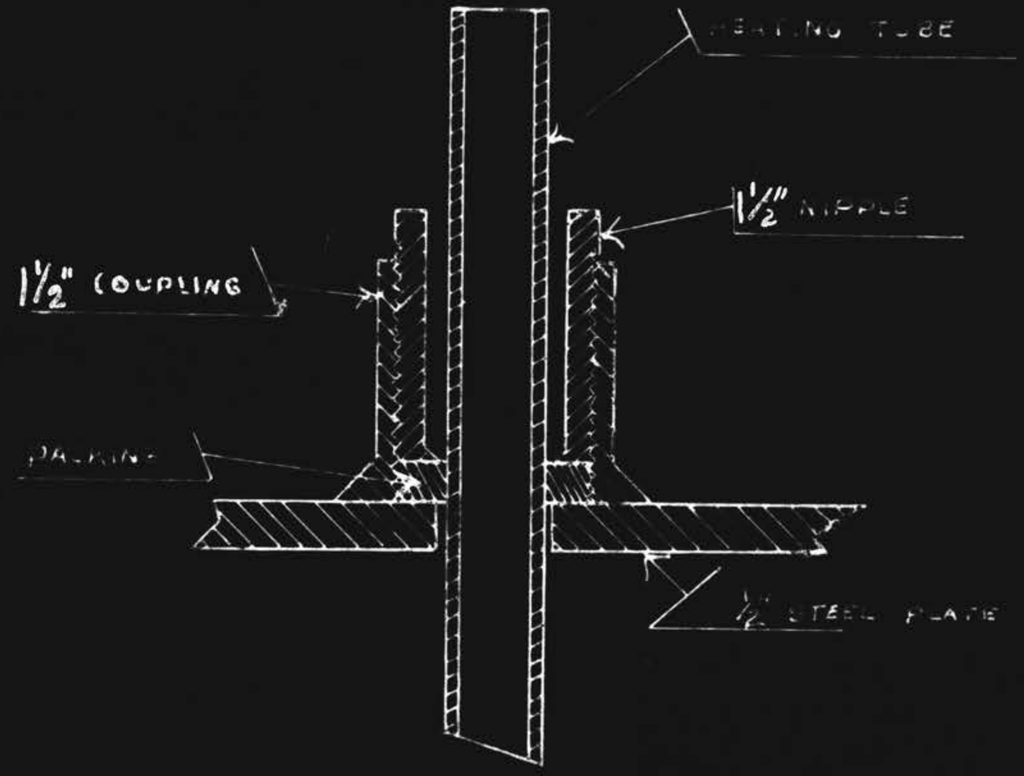
DESCRIPTION OF APPARATUS

The apparatus used for this investigation is a single tube, steam heated evaporator which is constructed in such a manner as to make it flexible enough to be used for a variety of problems. The tube and surrounding steam chest may be inclined at any angle between zero and ninety degrees with a minimum amount of effort. A flow meter is available to measure velocity of flow in the heating tube and the evaporator is easily adapted to either natural or forced circulation operation.

The heating element is a standard one inch, Type 316 stainless steel pipe. The overall length of the tube is seventy-four inches with the length enclosed in the steam chest being 69.5 inches in length. It is supported in the steam chest by two packing glands made by combining a one and one-half inch coupling and nipple. Details of these glands are shown in Figure 1.

The steam chest is a standard four inch iron pipe flanged at both ends. The length is six feet. It has six one inch couplings welded to it. These couplings are used as outlets for thermocouple leads. These thermocouples are attached to the heating tube. The first thermocouple outlet is six inches from the end of the steam chest and the others are spaced one foot apart as shown in Figures 2 and 3. These couplings are fitted with one inch flanges composed of a standard 125 pound screw flange and a blind companion flange. Two soft rubber gaskets are inserted between these flanges and the thermocouple leads are brought out between the two gaskets. The steam inlet is a one inch pipe connection placed between the two top thermocouple outlets. The condensate outlet is also a one inch connection welded into the bottom of the steam chest next to the bottom flange. To the outlet is connected an Anderson No. 21, bucket type steam trap. A one-fourth inch outlet is located at the top of the steam chest and fitted with a valve to act as a vent for non-condensable gases. This valve is cracked at all times during operation. Before the steam enters

FIG. 1



PACKING GLAND
CROSS SECTION VIEW

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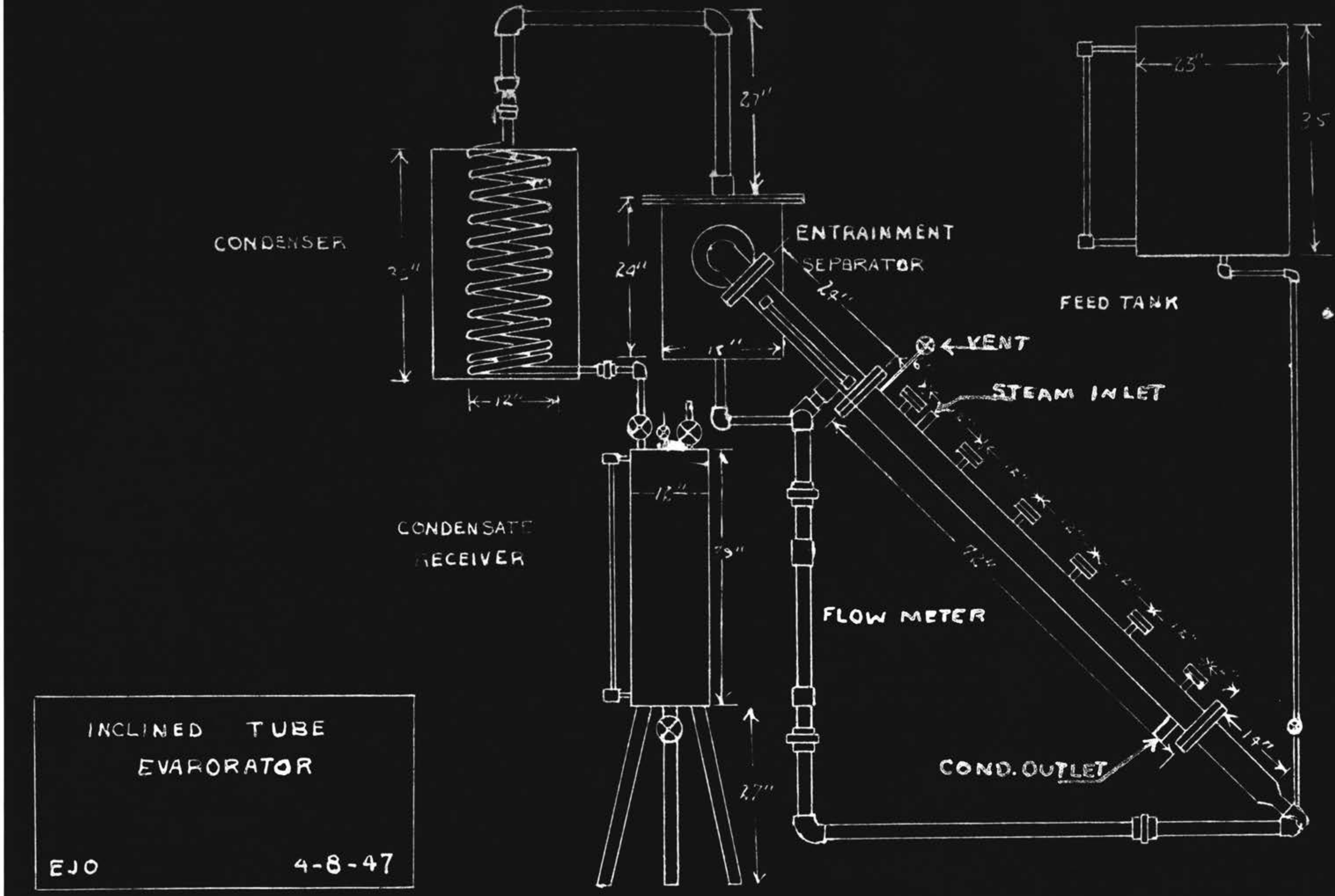
CONDENSER
32"
12"
27"
24"
15"
ENTRAINMENT
SEPBRATOR
28"
FEED TANK
23"
35"
VENT
6"
STEAM INLET
CONDENSATE
RECEIVER
12"
29"
27"
FLOW METER
72"
14"
COND. OUTLET
INCLINED TUBE
EVARORATOR
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Fig 2

FIG 3

THERMOCOUPLE OUTLET

EXPLODED VIEW

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the steam chest it passes through a Fisher Type 10-A regulator and then through a "U" shaped pipe with a trap at the bottom. Steam which may condense in the line collects in the bottom of the "U" joint and is discharged through the trap.

Attached to the bottom of the steam chest by flange is a fourteen inch section of four inch standard iron pipe which acts as the liquid reservoir. This section is reduced to standard two inch pipe size on the lower end. A two inch tee is attached to the lower end. One branch is used for the feed inlet and the other connects to the liquid return line.

A twenty-four inch section of standard four inch pipe is attached to the upper end of the steam chest by flanges. This section serves as disengaging space between the steam chest and entrainment separator. It contains a gage glass which is used to observe the liquid level in the evaporator and also contains a two inch connection on the lower side near the bottom which acts as the upper outlet for the liquid return line.

The liquid return line is composed of one vertical and one horizontal branch. The length of each branch will vary with the angle of inclination. A twenty-four inch section of two inch pyrex glass pipe which serves as the observation glass for the flow meter is included in the vertical branch. This glass section is joined to the pipe above and below the glass by two inch size radiator hose and clamps. Unions are located six inches above and below this joint in order to facilitate removal of the flow meter without danger of breaking it. The flow meter is a perforated, truncated, conical section suspended from a spring made of a metal whose elasticity is unaffected by the range of temperatures encountered. Spring elongation is a measure of flow rate.

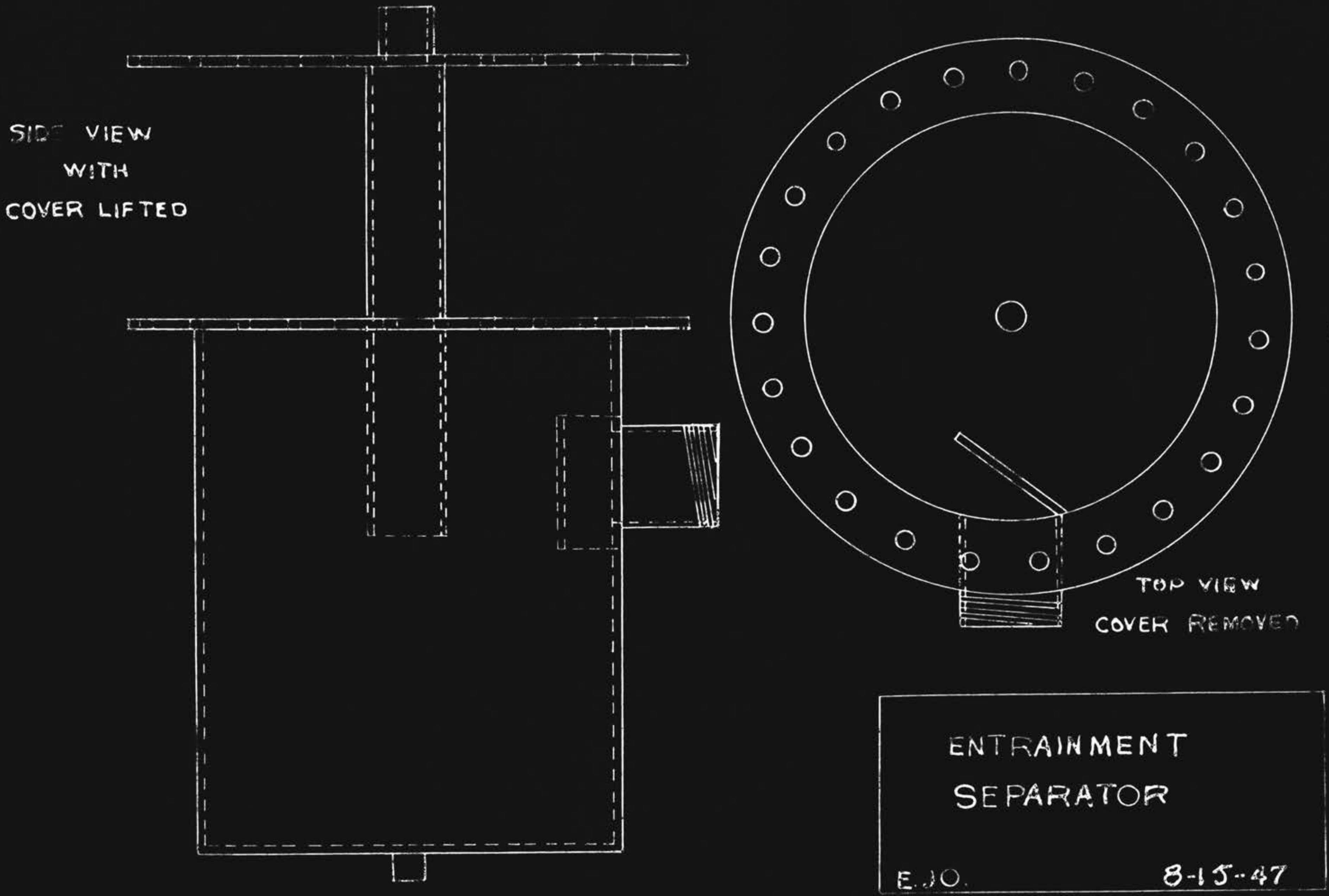
The feed tank is a fifty gallon steel drum with a central outlet located in the bottom. This connects through a one-half inch line to the bottom of the liquid reservoir. The line contains a one-half inch globe valve for regulation of feed rate. The feed tank is provided with a sight glass which indicates the liquid level in the tank at all times.

The entrainment separator is of the cyclone separator type. See Figure 4. The main body is a twenty-four inch section of standard eighteen inch iron pipe. The bottom is a slightly dished section of three-eights inch armour plate steel. The bottom is dished to allow the separator to keep completely drained at all times. The top of the separator is a flat flanged cover of the same material as the bottom. It is fastened with twenty-three, one-half inch machine bolts equally spaced around the circumference of the flange. A standard two inch pipe coupling is welded into the center of the top to act as an outlet to the condenser. A one inch coupling is centrally located in the bottom of the separator to act as a return line for the liquid which is removed from the vapor. The vapor inlet is a standard four inch nipple welded into the side of the separator, five inches below the top flange. A six inch square baffle is located inside the separator, in front of the vapor inlet to deflect the incoming vapor so as to make it follow the required circular path. A four inch screw flange is used on the vapor inlet to connect it to a ninety degree, four inch, flanged elbow which in turn connects to the section of four inch pipe at the top of the steam chest. This flanged connection allows the angle of inclination of the evaporator tube to be varied at will.

The vapor outlet of the entrainment separator is connected to the condenser through a two inch line which rises to a height of twenty-seven inches above the separator. A horizontal section is connected to this vertical pipe and runs over to the condenser where it reduces to standard one inch pipe size and is connected to the condenser coil. The horizontal section is inclined slightly downward toward the condenser to prevent any condensed vapor from falling back into the entrainment separator.

The condenser is made of one inch copper tubing. It consists of ten coils twelve inches in diameter. These coils are fastened in an open top steel drum in an upright position. Cooling water is introduced at a point one inch from the bottom of the drum through a three-fourth inch line which runs down through the axis of the coil. A two inch overflow line is located two inches below the top

FIG. 4


SIDE VIEW
WITH
COVER LIFTED
TOP VIEW
COVER REMOVED
ENTRAINMENT
SEPARATOR
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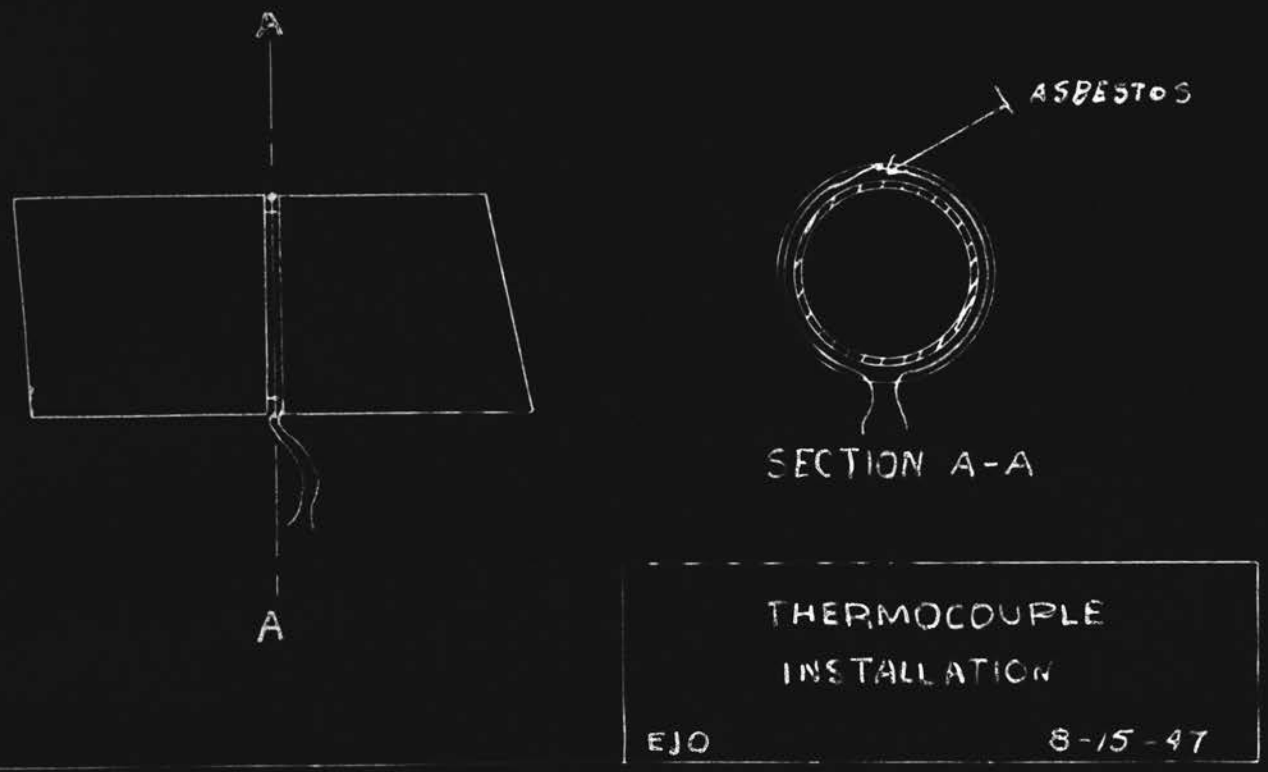
of the drum.

The condensate receiver is a twelve by thirty-nine inch cylindrical tank supported twenty-seven inches from the floor level by three legs. The top is provided with two standard one inch and one standard one-fourth inch pipe connections. One of the one inch connections is used for receiving condensate from the condenser, the other for attaching to a vacuum pump. The one-fourth inch connection is provided with a valve for releasing the vacuum when desired. The condensate receiver contains a half-inch gage glass for indication of the liquid level. The condensate is drawn from the bottom of the receiver through a one-half inch valve.

All temperatures except the feed temperature are read by means of thermocouples. The feed temperature is obtained with a mercury thermometer. Iron-constantan junctions are used for the thermocouples and the E.M.F. is measured by a Leeds and Northrup Type K potentiometer. This instrument is capable of reading hundredths of milli-volts directly with the third place being a close estimate. This reading is equivalent to a direct temperature reading to the nearest 0.3 of one degree Fahrenheit and a close estimate to the nearest 0.03 of one degree. The heating tube is provided with six thermocouple junctions which give the outside skin temperature of the tube. These are spaced twelve inches apart with the two junctions near the ends of the steam space being approximately six inches from the packing glands. The thermocouples are installed according to the method described by Colburn and Hougen[4]. See Figure 5. This method has been approved by McAdams[8]. A slot approximately three thirty-seconds of an inch in depth was cut entirely around the pipe with a hack saw. The thermocouple junction is placed in this slot with the fused junction just level with the pipe surface. The junction was soldered in place and the electrically insulated leads brought away from the junction through the slot to the opposite side of the pipe. The slot was then filled with solder. The excess solder was polished off flush with the surrounding pipe surface. This minimizes conduction of heat to or from the junction.

Fig. 5



THERMOCOUPLE INSTALLATION

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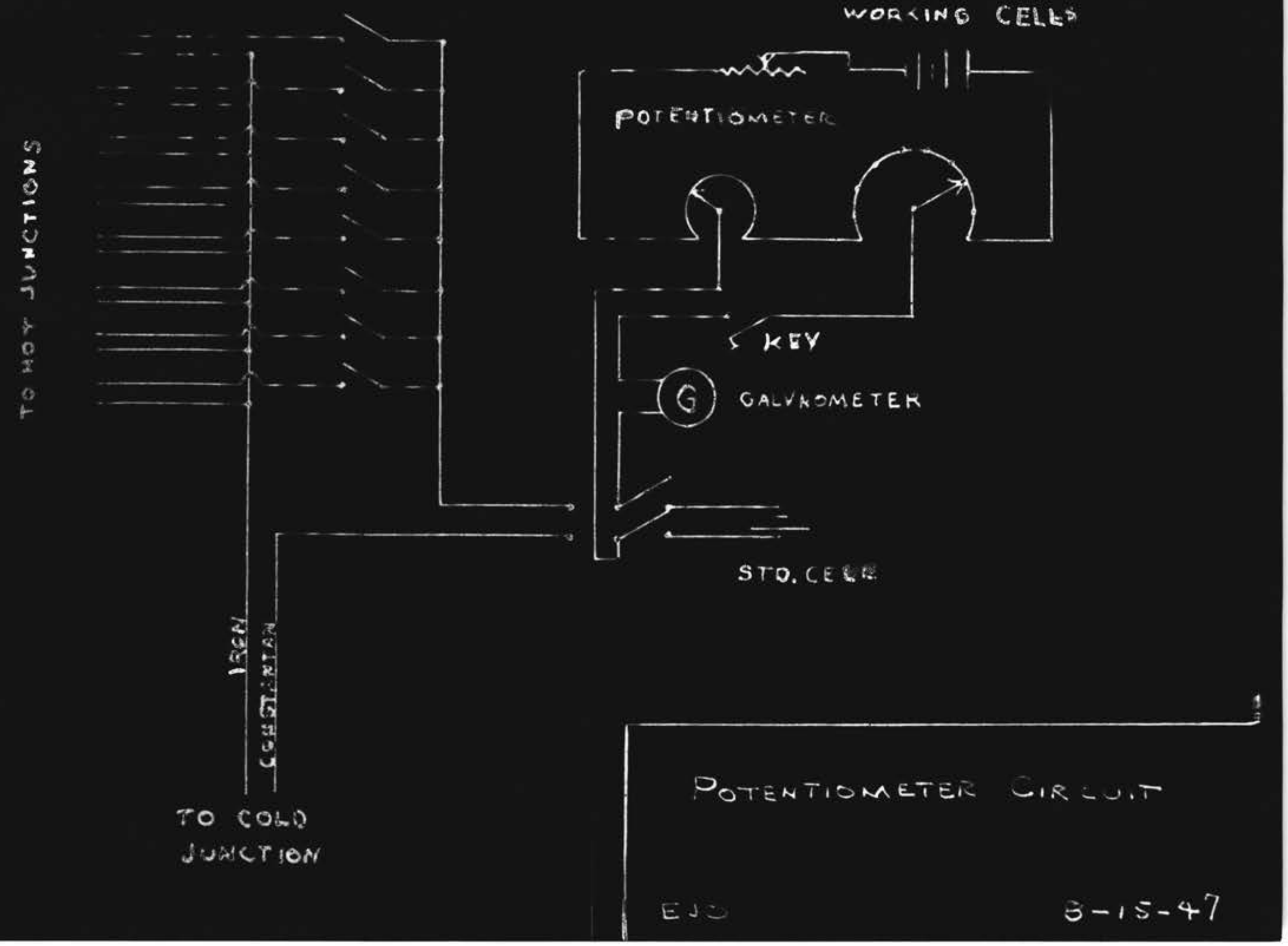
Fig. 6



POTENTIOMETER CIRCUIT

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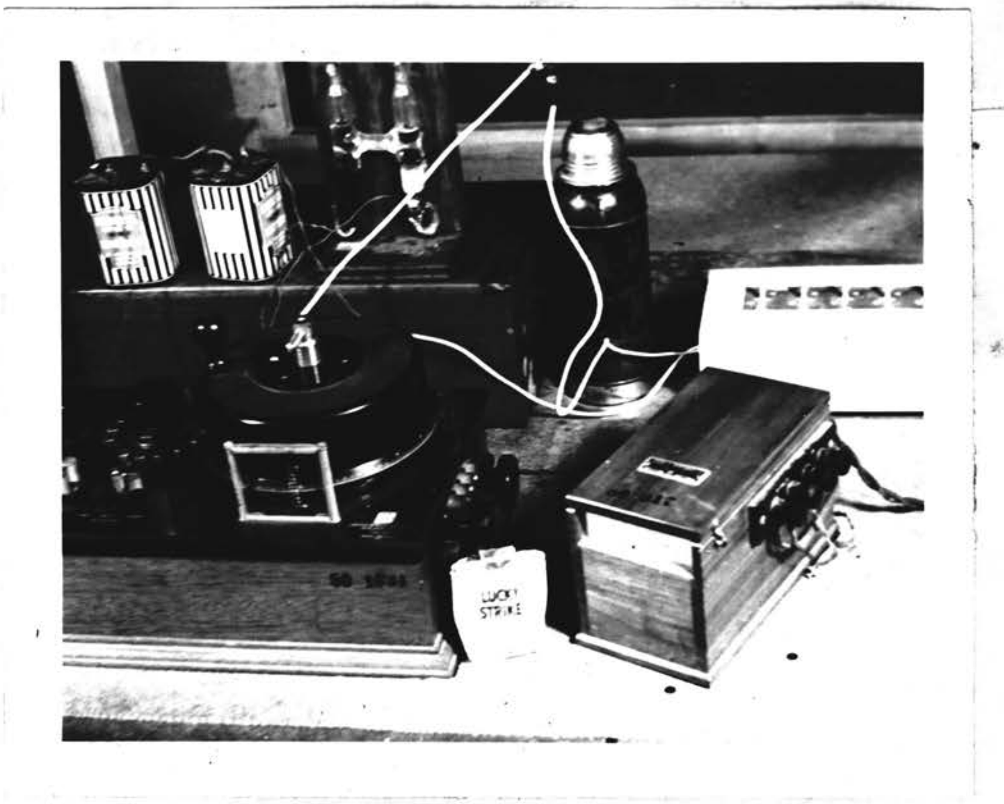
through the lead wires. This would occur if the leads were brought out directly through a medium differing an appreciable amount in temperature, such as the steam chest. Subsequent tests proved the installation to be satisfactory when checked against a mercury thermometer capable of reading to the nearest tenth of one degree Fahrenheit. The thermocouple leads are brought out of the steam chest by the method previously described. See Figure 3. Two more thermocouple junctions are used to obtain the average liquid temperature as it passed through the tube. These junctions are inserted in a glass tube filled with oil and placed in the liquid line approximately two-thirds of the distance down from the top on a cross section view of the liquid line. The indicated temperature was found to be independent of the thermocouple position in the pipe. All thermocouple leads are brought to the instrument table where a common cold junction is provided. The wire used for the thermocouple leads and junctions is No. 24 gage iron and constantan with woven glass insulation. A smaller size would be preferred but none could be obtained.

After erection and testing of the apparatus, insulation was applied to the steam chest, entrainment separator, vapor lines and the liquid return line. In order to facilitate removal of the insulation for any reason, boxes are placed around the steam chest and entrainment separator. This space is filled with Eagle-Pitcher "Super 66" asbestos felt insulation. The vapor lines and liquid return line are insulated with Johns-Mansville "Abestocel" pipe lagging. The glass enclosing the flow meter is left bare.



Instrument Table

Evaporator Before Insulating Steam Chest

Evaporator After Addition of Insulation

EXPERIMENTAL PROCEDURE

Since it was desired to use a feed solution which would readily deposit a scale, a substance exhibiting an inverse solubility curve would be most desirable. The feed solution used in the investigation was a saturated solution of calcium sulfate. This was prepared by dissolving powdered calcium sulfate in distilled water until a noticeable amount settled out in the bottom of the feed tank. After filling the feed tank, the apparatus was filled until the liquid level was high enough to just cover the outlet end of the heating tube. Steam was then admitted to the steam chest and kept at a pressure of approximately ten pounds per square inch gage. This pressure was kept constant throughout the entire run. The time required for the evaporator to start circulation and reach a steady state condition was approximately one and one half hours. During this time the feed rate was adjusted to keep the liquid in the evaporator at the required level. The working voltage and standard cell were connected to the potentiometer circuit at the beginning of the warm-up period and the potentiometer was standardized against the Weston cell immediately before the first reading was taken. This was repeated before each reading throughout the run in order to assure maximum accuracy in obtaining thermocouple potential. Temperature readings were taken at thirty minute intervals and the condensate from the evaporated feed was weighed at the end of each hour.

Nine temperatures were recorded at the end of each half hour interval. Six of these were temperatures along the heating tube. Two temperatures of the circulating liquid were obtained, one at the bottom of the heating tube and the other at the top. The temperature of the feed solution was also recorded.

As the run progressed the feed rate had to be varied. This was necessary because of scale forming on the heating tube and subsequent lowering of the amount of liquid evaporated per unit of time.

The individual runs were approximately twenty hours in duration. This period was sufficient to demonstrate the effect of scale formation. Runs of greater length would serve to make the task of cleaning the tube more difficult at the completion of the run.

After each run the scale had to be completely removed from the heating tube before starting the next run. This offered some difficulty. The method which seemed to be most effective was to fill the apparatus with a weak solution of hydrochloric acid, approximately one percent, and heat it in the evaporator for a period of five or six hours. This loosened most of the scale deposit and made it possible to finish cleaning the tube with a steel wool swab. This swab was made by attaching a roll of steel wool slightly larger than the inside diameter of the tube to the end of a section of one-fourth inch pipe. The evaporator was thoroughly flushed with water after the acid was drained to stop any serious corrosive effects. The heating tube was, of course, unaffected by the acid.

METHOD OF CALCULATION AND CALCULATED RESULTS

Starting with the equation defining the individual heat transfer coefficient which was mentioned in the first section of the report:

$$q = hA\Delta t$$

and rearranging we get:

$$h = \frac{q}{A\Delta t}$$

Thus if we are able to determine the three quantities on the right side of the equation, the individual heat transfer coefficient is easily calculated.

In the case of the evaporator used for this problem the area is that portion of the inside of the heating tube which is exposed to the steam on the opposite side of the tube. The tube used is standard weight one inch pipe size and the length exposed to the steam is 69.5 inches. The length of pipe required for one square foot of inside area is 3.641 feet per square foot. The area for a 69.5 inch length is then:

$$A_{\text{sq. ft.}} = \frac{1 \times 69.5}{3.641 \times 12} = 1.59 \text{ sq. ft.}$$

The amount of heat flowing q is obtained by knowing the weight of water evaporated per unit of time and from knowing the amount of heat required to raise the water to saturation temperature and to vaporize it at this temperature. Heats of vaporization are easily obtained from the steam tables.

Determining the temperature drop across the liquid film is done in the following manner. The ball junctions on the thermocouples were approximately one-sixteenth inch in diameter. In view of this fact it was assumed that the temperatures read by them were at a point one thirty-second of an inch below the outer surface of the tube rather than at the surface. Knowing the thickness of the tube, the average area, quantity of heat flowing, and the thermal conductivity of the tube wall the temperature drop across the tube wall is calculated from the equation:

$$q = \frac{\Delta t}{x/kA}$$

Since the total drop across the tube wall and film is measured directly by thermocouple junctions, the drop across the film alone can now be obtained by difference.

A complete set of calculations on data taken between the fifth and sixth hour of operation on Run I follows:

Reading	Liq. Temp.		Tube Wall Temperature					
Time in Hours	Bot.	Top						
	T_1	T_2	T_3	T_4	T_5	T_6	T_7	T_8
5.5	5.23	5.24	6.02	6.07	6.05	6.06	6.04	6.01
6.0	5.23	5.24	5.98	6.01	5.97	6.00	6.00	5.98

Feed Temp.—92.0 °F
Wt. of Cond.—12.0 lbs.

All temperatures in the table are given in millivolts.

Average Liquid Temperature

$\frac{5.23 + 5.24}{2} = 5.24$ Mv. $\qquad \frac{5.24 + 5.24}{2} = 5.24$ Mv.

5.24 millivolts is equivalent to 210.5 °F.

Average Tube Temperature:

$$\begin{array}{r} 6.02 \\ 6.07 \\ 6.05 \\ 6.06 \\ 6.04 \\ \underline{6.01} \\ 36.25/6 = 6.04 \text{ Mv.} \end{array} \qquad \begin{array}{r} 5.98 \\ 6.01 \\ 5.97 \\ 6.00 \\ 6.00 \\ \underline{5.98} \\ 35.94/6 = 5.99 \text{ Mv.} \end{array}$$

$\frac{6.04 + 5.99}{2} = 6.005$ Mv which is equivalent to 236°F.

Determining q:

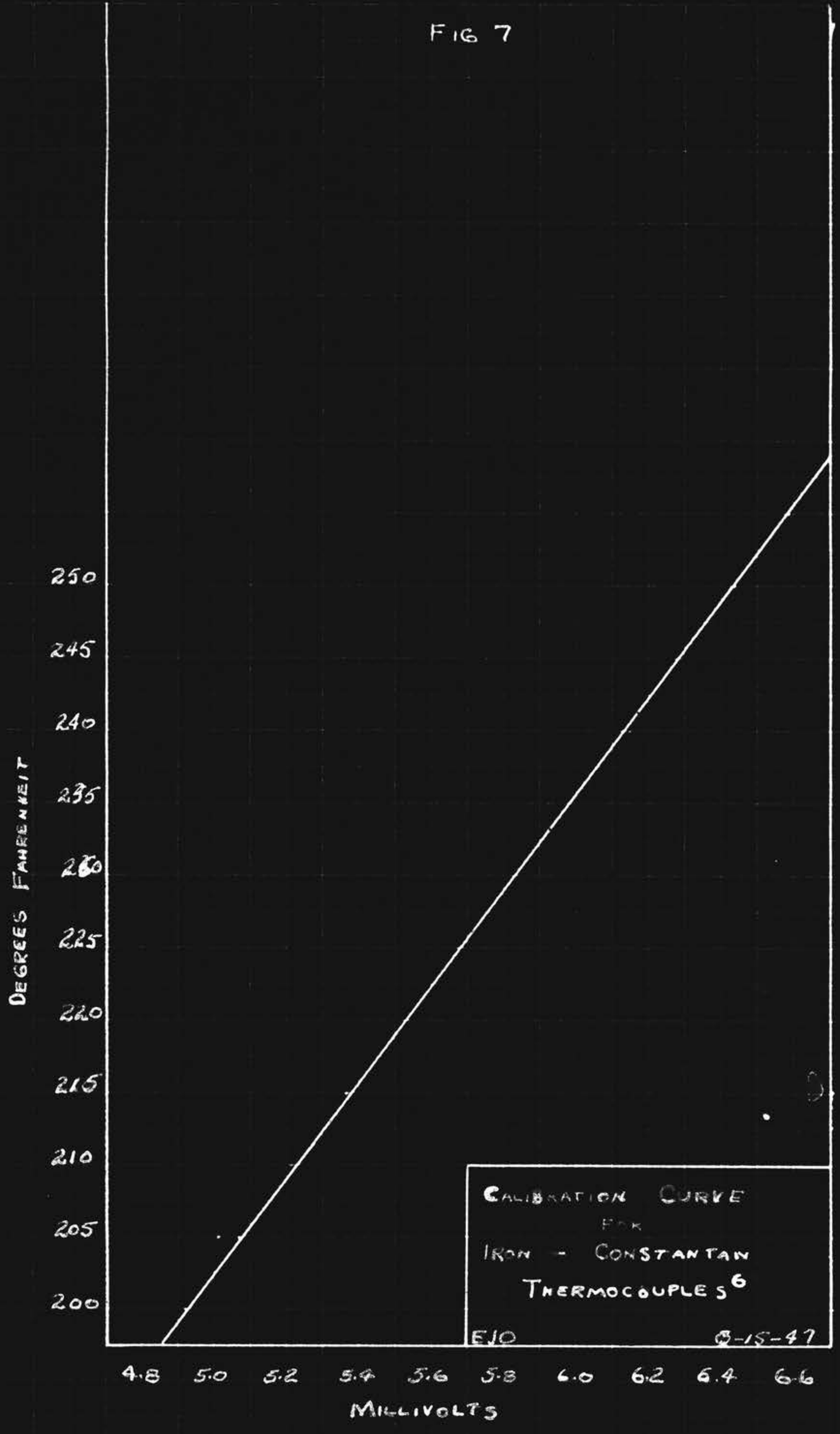
Heat added per lb. of water evaporated:

210.5°F. - 92.0°F. = 118.5°F.

or 118.5 B.t.u.'s required to raise feed to the temperature of evaporation.

Heat of vaporization @ 210.5°F. = 971.4 B.t.u./lb.

FIG 7


DEGREES FAHRENHEIT
250
245
240
235
230
225
220
215
210
205
200
MILLIVOLTS
4.8
5.0
5.2
5.4
5.6
5.8
6.0
6.2
6.4
6.6
CALIBRATION CURVE
FOR
IRON - CONSTANTAN
THERMOCOUPLES[6]
EJO
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$971.4 + 118.5 = 1089.9$ B.t.u./lb. = total heat required per pound

Heat required for 12 lbs.:

$12 \times 1089.9 = 13090$ B.t.u. $= q$

Determination of Δt Across Tube Wall:

Dia. of tube at point where temperature is read:

1.315 inches − 1/16 inch = 1.253 inches

Area per linear foot:

$$\frac{\pi \times 1.253 \times 12}{144} = 0.328 \text{ ft.}^2/\text{ft.}$$

Area of 69.5 inches of tube:

$$\frac{69.5 \times 0.328}{12} = 1.90 \text{ ft.}^2$$

Inside area = 1.59 ft.^2/69.5 inches of length

$$A_m = \frac{1.90 - 1.59}{\ln \frac{1.90}{1.59}} = 1.73 \text{ ft.}^2$$

Thickness of path:

$$x = \frac{.133 \text{ inches} - .031 \text{ inches}}{12} = 0.0085 \text{ ft.}$$

For Type 316 Stainless Steel $k = 9.0$ B.t.u./hr.-ft.^{2}-°F./ft. @ 212°F.[15]

Then:

$$\Delta t = \frac{qx}{kA} = \frac{13090 \times 0.0085}{9 \times 1.73} = 7.14^\circ\text{F.}$$

Temperature Drop Across Liquid Film:

Total $\Delta t = 236 - 210.5 = 25.5^\circ$F.

$\Delta t_{\text{film}} = 25.5 - 7.1 = 18.4^\circ$F.

Liquid Side Heat Transfer Coefficient:

$$h = \frac{q}{A\,\Delta t} = \frac{13090}{1.59 \times 18.4} = 447 \text{ B.t.u./hr.-ft.}^2\text{-}^\circ\text{F.}$$

The calculated results for the three runs are given in the three tables which follow:

RUN I

Time Hours	Ave. Tube Temp.	Ave. Liquid Temp.	Total Δ t	Temp. Feed	Wt. Cond.	q	Δ t Pipe Wall	Δ t Film	h	U
1	235.7	210.5	25.2	87.9	16.87	18450	10.07	15.1	769	402
2	237.0	210.2	26.8	88.4	15.50	16930	9.24	17.6	605	365
3	234.5	210.2	24.3	90.0	13.00	14210	7.76	16.5	542	304
4	236.0	210.0	26.0	91.0	14.37	15680	8.56	17.4	567	335
5	236.0	210.2	25.8	92.0	13.00	14150	7.73	18.1	492	305
6	236.0	210.5	25.5	92.0	12.00	13090	7.14	18.4	447	284
7	235.7	210.1	25.6	92.4	13.00	14150	7.73	17.9	496	302
8	239.5	210.5	29.0	92.2	12.56	13680	7.47	21.5	400	298
9	235.0	210.2	24.8	91.6	10.75	11710	6.74	18.1	407	255
10	235.7	210.5	25.2	91.3	10.44	11400	6.34	18.9	379	248
11	237.0	210.2	26.8	91.0	10.50	11450	6.24	20.6	350	247
12	236.5	210.2	26.3	91.0	9.94	10820	5.92	20.4	333	233
13	236.0	210.2	25.8	91.5	9.50	10360	5.64	20.2	322	223
14	238.0	210.2	27.8	92.0	9.99	10890	5.94	21.9	312	235
15	237.5	210.2	27.3	92.0	9.56	10410	5.68	21.6	303	225
16	238.0	210.2	27.8	92.0	9.06	9875	5.39	22.4	277	213
17	236.5	210.0	26.5	94.5	9.25	10040	5.47	21.0	301	217
18	237.0	210.0	27.0	95.0	8.00	8700	4.75	22.2	246	188
19	238.0	210.2	27.8	95.0	8.56	9320	5.08	22.7	256	201
20	237.5	210.2	25.3	87.5	7.67	8390	4.58	20.7	255	181
21	236.5	210.0	26.5	88.0	8.00	8750	4.77	21.7	254	188
22	237.0	210.0	27.0	88.5	6.19	6770	3.69	23.3	183	146

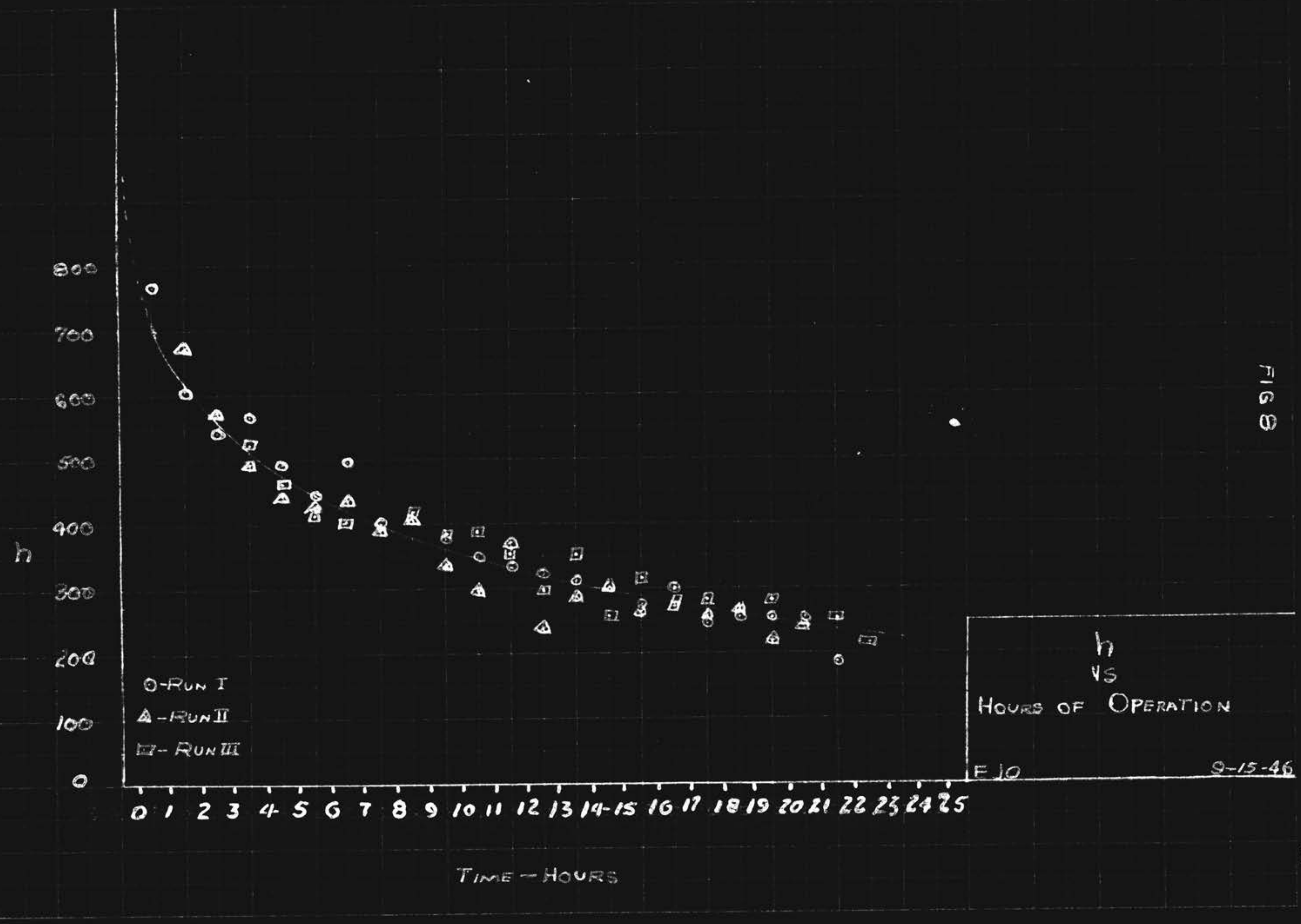
RUN II

Time: Hours	Ave. Tube Temp.	Ave. Liquid Temp.	Total Δ t	Temp. Feed	Wt. Cond.	q	Δt Pipe Wall	Δt Film	h	U
1	236.0	210.5	25.5	86.0	15.75	17250	9.42	16.1	673	375
2	237.5	210.5	27.0	87.0	14.98	16410	8.96	18.0	574	358
3	235.2	210.5	24.7	87.5	12.37	13530	7.39	17.3	492	295
4	234.0	210.2	23.8	88.5	11.06	12100	6.61	17.2	442	261
5	237.5	210.2	27.3	89.0	12.31	13480	7.36	19.9	426	291
6	236.5	210.0	26.5	89.0	12.25	13400	7.32	19.2	438	287
7	236.0	210.0	26.0	89.0	11.00	12020	6.56	19.4	390	258
8	236.0	210.0	26.0	92.0	11.38	12400	6.77	19.2	408	266
9	235.5	210.0	25.5	92.0	9.62	10500	5.73	19.8	334	227
10	237.0	210.0	27.0	92.0	9.31	10140	5.53	21.5	297	219
11	238.5	210.5	28.0	85.0	11.44	12540	6.84	21.2	372	273
12	236.0	210.0	26.0	84.5	7.43	8130	4.44	21.6	237	175
13	238.5	210.5	28.0	84.0	9.38	10280	5.62	22.4	288	224
14	235.2	210.2	25.0	85.5	8.75	9580	5.23	19.8	304	206
15	238.0	210.0	28.0	86.0	8.81	9650	5.27	22.7	267	208
16	237.0	210.2	26.8	86.5	8.56	9380	5.13	21.7	272	202
17	238.5	210.0	28.5	88.0	8.56	9370	5.12	23.4	252	202
18	236.5	210.0	26.5	88.0	8.31	9090	4.96	21.5	266	196
19	237.0	210.5	27.5	88.0	7.31	8000	4.37	23.1	218	174
20	237.2	210.0	27.2	91.0	8.01	8730	4.76	22.4	245	188

RUN III

Time Hours	Ave. Tube Temp.	Ave. Liquid Temp.	Total t	Temp. Feed	Wt. Cond.	q	t Pipe Wall	t Film	h	U
1	236.0	210.0	26.0	87.0	13.56	14850	8.1	17.9	522	318
2	237.0	210.0	27.0	88.0	12.94	14180	7.7	19.3	462	304
3	238.0	210.0	28.0	87.0	12.31	13500	7.4	20.6	412	299
4	236.3	210.0	26.3	87.0	11.44	12540	6.9	19.4	406	268
5	239.0	210.0	29.0	87.0	12.13	13300	7.3	21.7	396	285
6	236.3	210.0	26.3	88.0	10.74	12870	7.0	19.3	418	275
7	236.7	210.0	26.7	90.0	11.26	12300	6.7	20.0	386	264
8	234.0	210.0	24.0	91.0	10.19	11110	6.1	17.9	390	238
9	235.2	210.0	25.2	92.0	9.94	10830	5.9	19.3	354	232
10	237.2	210.0	27.2	96.0	9.43	10230	5.6	21.6	298	219
11	235.7	210.0	25.7	97.0	10.19	11070	6.0	19.7	353	218
12	236.7	210.0	26.7	99.0	8.25	8940	4.9	21.8	258	191
13	237.2	210.0	27.2	100.0	10.00	10816	5.9	21.3	319	232
14	236.0	210.0	26.0	100.0	9.25	10000	5.5	20.5	307	214
15	236.7	210.0	26.7	100.0	8.86	9590	5.2	21.5	281	205
16	236.3	210.0	26.3	100.0	8.50	9200	5.0	21.3	272	197
17	235.5	210.0	25.5	100.0	8.54	9230	5.0	20.5	283	197
18	234.0	210.0	24.0	100.0	6.90	7470	4.1	19.9	239	160
19	236.7	210.0	26.7	101.0	8.13	8790	4.8	21.9	252	188
20	236.5	210.0	26.5	102.0	7.31	7880	4.3	22.2	223	169

FIG 8



h
VS
HOURS OF OPERATION

E.J.O. 9-15-46

DISCUSSION AND CONCLUSIONS

The individual coefficient as calculated for Run I showed a very noticeable decrease in value. It dropped from a value of 769 B.t.u./hr.-ft.^{2}-$^\circ$F. during the first hour of observation to 183 B.t.u./hr.-ft.^{2}-$^\circ$F. during the twenty-second hour. In Run II which was of twenty hours duration the coefficient decreased from 673 to 245 B.t.u./hr.-ft.^{2}-$^\circ$F. For Run III also of twenty hours duration the corresponding values were 522 and 223 B.t.u./hr.-ft.^{2}-$^\circ$F. These results on first observation seem to be inconsistent but the deviations can be accounted for to a large extent by the fact that warm-up periods of different time lengths were used for the three runs. The coefficient decreases much more rapidly at the beginning of the run when the scale first begins to form. For this reason an additional hour used in allowing the evaporator to warm up and reach equilibrium will make a substantial difference in the value of the coefficient calculated for the first hour of observation. When plotting the data the abscissa is adjusted for Runs II and III so that the coefficients of the first hour of observation for these runs will fall approximately on corresponding values as determined in Run I. When this is done, it is found that succeeding values of coefficients for the three runs will fall close to a common curve. This indicates that the rate of scale formation was approximately the same for all three runs.

A correlation of the data was made by use of the McCabe-Robinson[9] equation:

$$\frac{1}{U^2} = A_1 + A_2\theta \qquad (11)$$

Values for U were obtained in the following manner. An average steam pressure of 10 p.s.i. gage was maintained in the steam chest which from the steam tables will have a temperature of 239.4 °F.[5] The liquid temperatures were measured by the thermocouple junctions as previously described. From these temperatures an overall Δt is obtained for use in equation (5). Values of q for each hour of operation had been calculated in determining values for h. The overall coefficient was based on the inside area of the tube since this would serve to illustrate the relative values of the coefficient as the run progressed.

After calculating values of U these were plotted against hours of operation and a smooth curve obtained from the data for the three runs. See Figure 9. Values of U were obtained from this curve and used to determine values of $1/U^2$. On plotting $1/U^2$ versus hours of operation (Figure 10) a good correlation with equation (11) was evident since the points fell very close to a straight line.

The next problem undertaken was to find the resistance which the scale deposit alone was offering to the flow of heat. There have been two approaches to this problem. One is an attempt to determine a thermal conductivity coefficient for the scale. This necessitates finding the thickness of the scale formed and is at best a rough approximation. Partridge[11] has listed values for the thermal conductivity of $CaSO_4$ scale obtained from other workers along with his own experimental values. These values range in magnitude from 0.043 to 2.12 B.t.u./ft.^{2}-°F./ft. The wide variation is due to differences in porosity of the scale as formed on the heating surface. For a dense, relatively non-porous scale Partridge suggests an average value of 1.3 B.t.u./hr.-ft.^{2}-°F./ft. Highly porous scale will have a much lower value.

The second approach to the problem of scale resistance is to state the resistance in terms of a fouling factor. This fouling factor has the same units as the heat transfer coefficient for fluids. Neither the thickness of the scale deposit nor its porosity has to be determined. Nelson[10] has given several values for fouling factors as found in petroleum refining equipment. Using h_d as a designation for the fouling factor, its numerical value may be found from the following relation[8].

$$\frac{1}{h_d} = \frac{1}{U_d} + \frac{1}{U_c} \qquad (12)$$

U_d is the overall heat transfer coefficient with scale present and U_c is the coefficient for the apparatus when clean.

Values of U were taken from Figure 9 and fouling factors determined using

FIG 9.

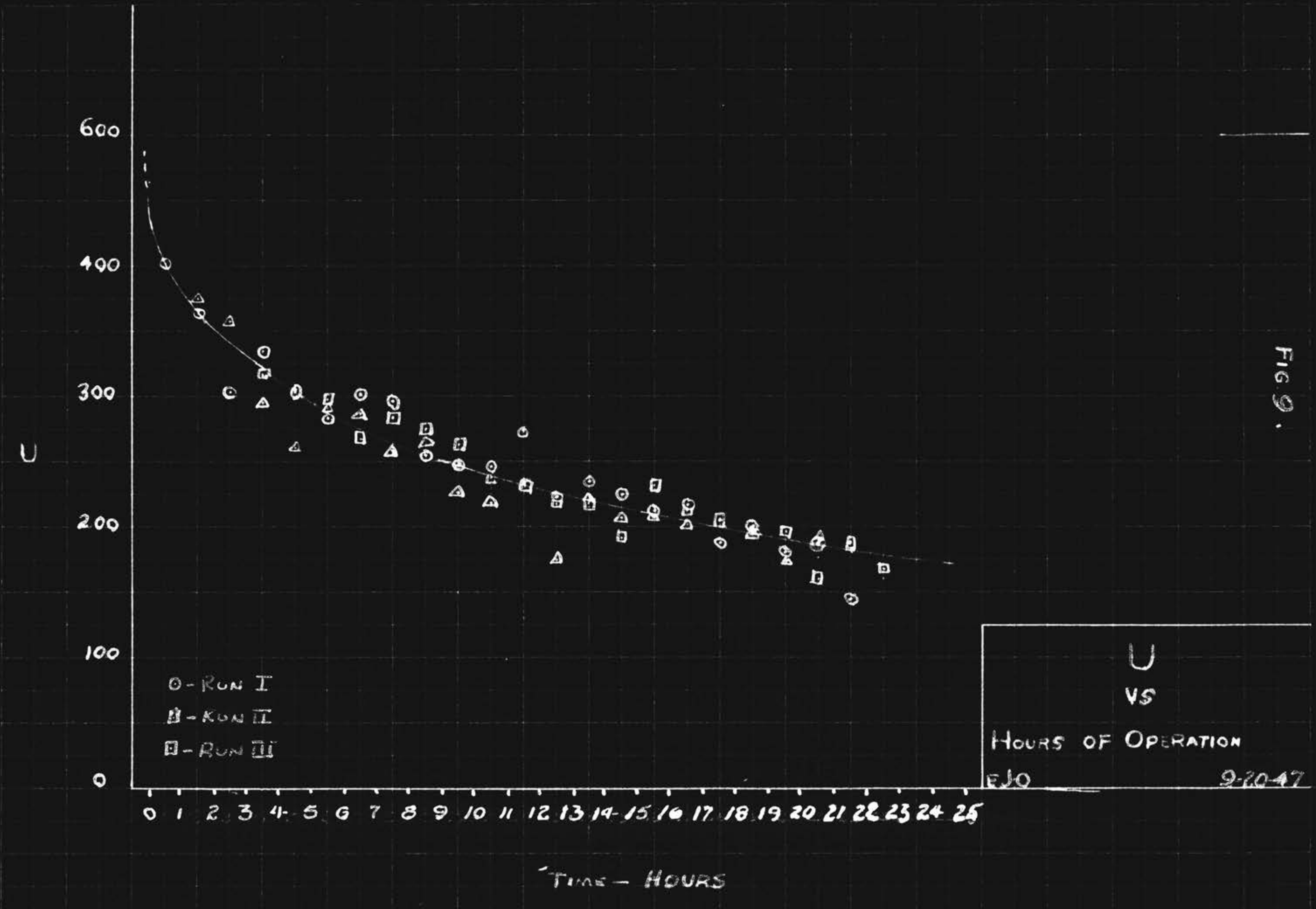
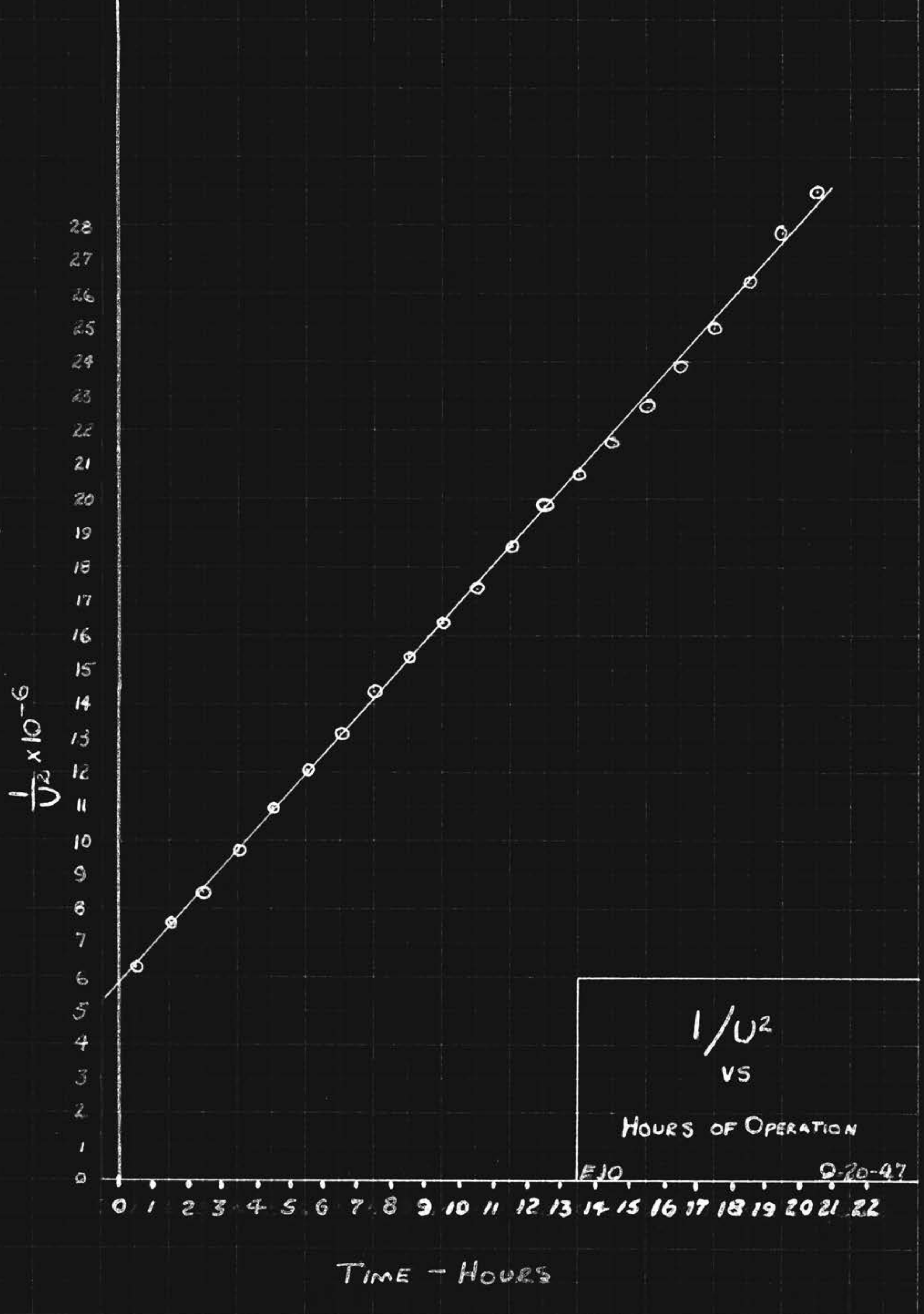

U
600
400
300
200
100
0
0 1 2 3 4 5 6 7 8 9 10 11 12 13 14 15 16 17 18 19 20 21 22 23 24 25
TIME – HOURS
⊙ - RUN I
△ - RUN II
⊡ - RUN III
U
VS
HOURS OF OPERATION
EJO
9-20-47

Fig. 10


$\frac{1}{U^2} \times 10^{-6}$
0
1
2
3
4
5
6
7
8
9
10
11
12
13
14
15
16
17
18
19
20
21
22
23
24
25
26
27
28
0 1 2 3 4 5 6 7 8 9 10 11 12 13 14 15 16 17 18 19 20 21 22
Time - Hours
$1/U^2$
vs
Hours of Operation
EJO
9-20-47

U at the end of the first hour of operation as U_c. The values of h_d which were obtained were plotted in Figure 11. The trend of the rate at which scale resistance increased is vividly illustrated by this plot. There is a very rapid decrease in h_d for the first six hours of observation after which the curve tends to level out with much smaller changes in h_d between succeeding hours of operation.

RECOMMENDATIONS

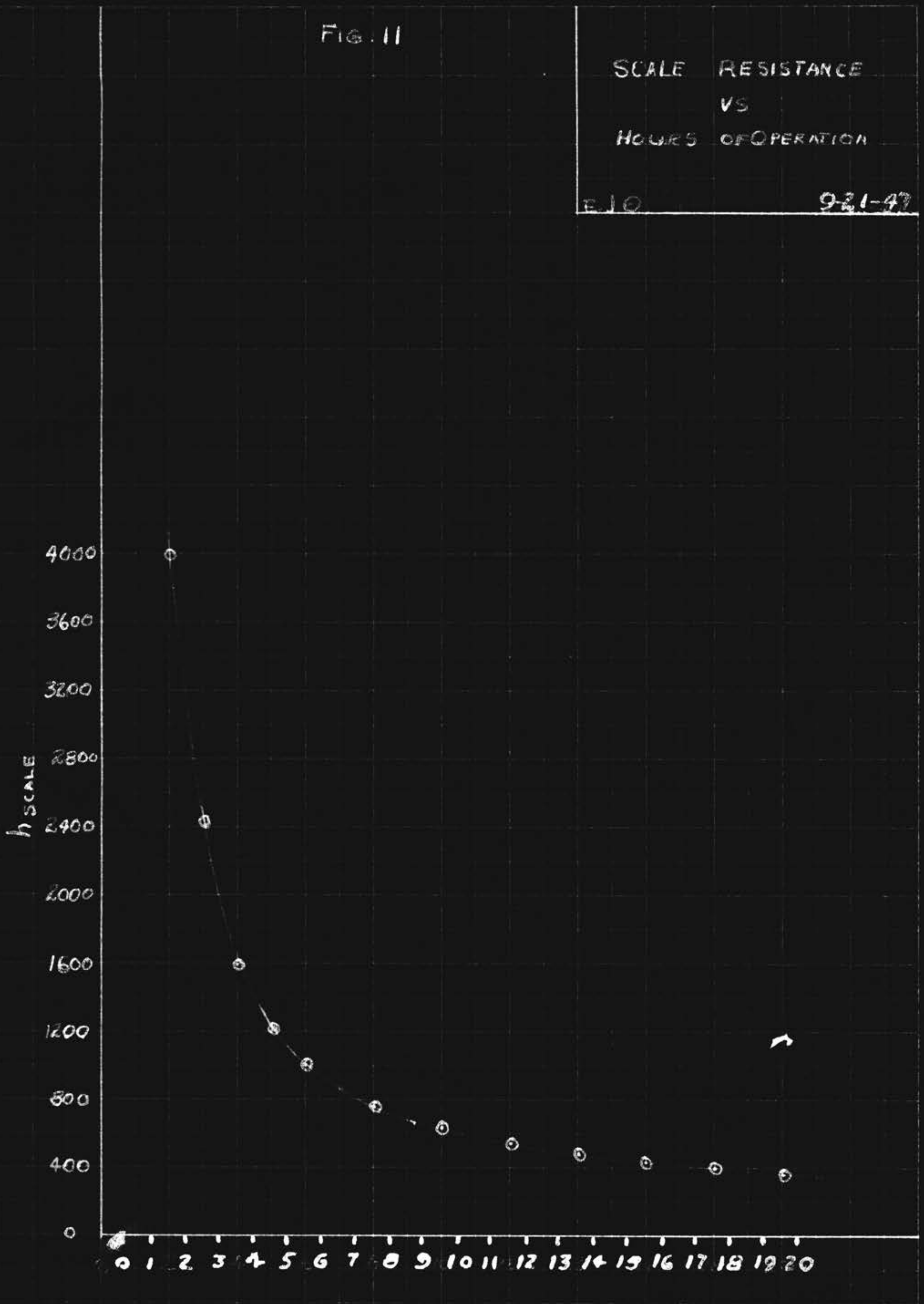
The equipment used in the investigation performed quite well. Some suggested changes to improve its performance will be presented.

It was difficult to maintain a constant steam pressure in the steam chest without constant attention and adjustment. This was due either to an inherent lag in the regulator or to the fact that the regulator was located about twenty feet from the evaporator with a bare pipe connecting. The first trial at remedying the trouble should be directed towards placing the regulator closer to the steam chest of the evaporator.

The section of four inch pipe connecting the entrainment separator to the upper end of the steam chest should be increased from twenty-four inches in length to approximately three feet. This will allow more flexibility in changing the liquid level in the evaporator. As it now stands, a level of two inches above the upper end of the heating tube is the maximum obtainable without having liquid standing in the bottom of the entrainment separator. This trouble will cause more difficulty as the tube is inclined at angles approaching a horizontal position and a different connection as indicated in the sketch below would perhaps be the best solution.

The packing glands used for supporting the heating tube in the steam chest could be improved. They should be changed to allow the heating tube

FIG. 11


SCALE RESISTANCE
VS
HOURS OF OPERATION
EJO
9-21-47
h_{SCALE}
4000
3600
3200
2800
2400
2000
1600
1200
800
400
0
0 1 2 3 4 5 6 7 8 9 10 11 12 13 14 15 16 17 18 19 20
TIME — HOURS

to be removed with less difficulty. Periodic removal may be necessary to repair the thermocouple junctions and to clean the steam side of the tube.

There are many related problems which are worthy of further study.

Since the scale found in most equipment which is used commercially is a combination of several minerals, it would be advantageous to run tests using other substances and mixtures. This would tend to duplicate actual conditions more closely.

In the past where forced circulation was used it has been found that an increase in liquid velocity tends to inhibit scale formation. This equipment is readily adaptable to forced circulation and could be used to substantiate previous findings and add to information known at the present time.

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