Direct contact liquid-liquid boiling heat transfer
by James Orvin Schagunn

A thesis submitted to the Graduate Faculty in partial fulfillment of the requirements for the degree of MASTER OF SCIENCE in Chemical Engineering
Montana State University

Abstract:
This research is a study of heat transfer by the direct contact of two immiscible liquids at contact temperatures above the boiling point of the more volatile liquid.

Mineral oil at contact temperatures above the boiling point of water was passed through tap water in a semi-batch operation. The experimental work consisted of: a) designing an evaporator which allowed prolonged boiling without boil-over and facilitated separation of oil and water; b) determining the boiling range volumetric heat transfer coefficients; c) studying the effects of the oil inlet temperature, oil flow rate, water-to-oil ratio, and size of oil droplets on the volumetric heat transfer coefficient; d) determining the conditions under which closest temperature approaches were obtained.

The results of runs indicated that: a) highest volumetric heat transfer coefficients were obtained at the lowest oil inlet temperatures; b) an optimum oil flow rate for the evaporator used existed in which maximum volumetric heat transfer coefficients were obtained; c) the volumetric heat transfer coefficients increased with increasing water-to-oil ratios; d) low oil flow rates and high water-to-oil ratios were necessary to obtain close temperature approaches; e) volumetric heat transfer coefficients comparable to those obtained with small oil droplets could not be obtained with large oil droplets.

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TABLE OF CONTENTS

LIST OF TABLES AND FIGURES

LIST OF TABLES AND FIGURES (cont)

ABSTRACT

This research is a study of heat transfer by the direct contact of two immiscible liquids at contact temperatures above the boiling point of the more volatile liquid.

Mineral oil at contact temperatures above the boiling point of water was passed through tap water in a semi-batch operation. The experimental work consisted of:

a) designing an evaporator which allowed prolonged boiling without boil-over and facilitated separation of oil and water;

b) determining the boiling range volumetric heat transfer coefficients;

c) studying the effects of the oil inlet temperature, oil flow rate, water-to-oil ratio, and size of oil droplets on the volumetric heat transfer coefficient;

d) determining the conditions under which closest temperature approaches were obtained.

The results of runs indicated that:

a) highest volumetric heat transfer coefficients were obtained at the lowest oil inlet temperatures;

b) an optimum oil flow rate for the evaporator used existed in which maximum volumetric heat transfer coefficients were obtained;

c) the volumetric heat transfer coefficients increased with increasing water-to-oil ratios;

d) low oil flow rates and high water-to-oil ratios were necessary to obtain close temperature approaches;

e) volumetric heat transfer coefficients comparable to those obtained with small oil droplets could not be obtained with large oil droplets.

INTRODUCTION

Until recently the unit operation of evaporation has been used to concentrate the solid materials of a mother liquor with little attention focused on the recovery of the vapors. Increased interest in the vapors, as in sea water conversion, has instigated new designs of evaporators and studies in heat transfer. One method being studied is the direct contact of two immiscible liquids to transfer the heat.

The advantages of the application of direct contact liquid-liquid heat transfer have led to recent studies of this method; the major advantages are as follows (2,8):

1. No surface exists upon which scale can form. Scale, which would form on the heat transfer surfaces of a conventional shell-and-tube heat exchanger, increases the resistance to heat flow and, therefore, decreases the efficiency and increases the cost of the unit.
2. About 1000 Btu are required to evaporate a pound of water from which, theoretically, 997 Btu can be recovered. The cost of an evaporator using liquid-liquid heat transfer to do this would be much less than that of a conventional shell-and-tube heat exchanger.

The disadvantages of the existing processes using direct liquid-liquid heat transfer are the high pumping costs and the necessity for a large volume of the transfer medium per unit volume of product (8).

If liquid-liquid boiling heat transfer is to be applied commercially, the liquid heat transfer medium must have the following characteristics (2): minimum solubility to allow for maximum separation; rapid and complete disengagement of any emulsion formed; chemical stability at temperatures well above the boiling point of the more volatile liquid; high heat capacity; low vapor pressure; low viscosity; low cost.

Benzene and toluene with water were used in preliminary experiments with spray towers to determine heat transfer coefficients (1,6). Volumetric heat transfer coefficients of 4 to 10 x 10^3 Btu/(hr)(ft^3)(°F) were obtained.

Wilkie, Cheng, Ledesma, and Porter used a liquid heat transfer medium more dense than the feed water (8). The medium and feed water passed through three-inch standard steel mixer tubes where the transfer of heat occurred. The volumetric heat transfer coefficients obtained had order of magnitude 10^4. Boiling and non-boiling runs were made.

FMC Corporation, under the auspices of the Office of Saline Water, used a spray column heat exchanger in which a hot petroleum base oil was introduced at the bottom through a sparger and rose counter-current to the flow of salt water (2,4). Using a three-inch diameter glass column with an effective length of 42 inches, volumetric heat transfer coefficients of 6 to 8 x 10^3 Btu/(hr)(ft^3)(°F) were obtained. Similar columns with effective lengths from 63 to 104 inches and a diameter of four inches were used to obtain coefficients up to 10 x 10^3 Btu/(hr)(ft^3)(°F). Further results showed that little salt and other impurities were transferred from the salt water to the hydrocarbon heat transfer media; hot fresh water cooled by the heat transfer media did not acquire an objectionable odor or taste (2). A later report by the Office of Saline Water explains an integrated system of a boiler and condenser using the above method of transferring heat and a multi-stage flash distillation unit (5). Preliminary results showed the salt content of the product to be as low as 12 ppm sodium chloride.

This research project is an attempt to evaporate water using an adapted spray column in which boiling takes place. A semi-batch operation is used. A petroleum based mineral oil is used as the heat transfer medium. Since a study of the volumetric heat transfer coefficient is the main objective of the project, the water vapor produced is allowed to escape to the atmosphere. As this is a preliminary investigation, low oil flow rates and small transfer volumes are used. These are adequate for the study of the effects of the flow rate, the oil-to-water ratio, and the inlet oil temperature on the heat transfer coefficient.

RESEARCH OBJECTIVES

Many heat transfer studies have been made using the direct contact of a liquid feed with a liquid heat transfer medium; however, most of this work has been done at a contact temperature below the boiling point of the liquid feed. Adaptation of this method to liquid-liquid boiling heat transfer requires heat transfer studies of oil inlet temperatures above the boiling point of the liquid feed.

The main objectives of this research project are to determine, in a mineral oil-water system, the boiling range volumetric heat transfer coefficients that can be expected and the effects of the following variables upon these coefficients:

1. oil inlet temperatures
2. flow rates of the mineral oil
3. ratios of water to mineral oil

A secondary objective is to compare the transfer coefficients obtained over a range of oil inlet temperatures using two sizes of oil droplets.

The results of this research may be used to design better equipment for further study in liquid-liquid boiling heat transfer.

EQUIPMENT AND EXPERIMENTAL CONSIDERATIONS

A. Experimental Process

A flow diagram of the process is shown in Figure 1, page 30. Oil from the feed reservoir passes through the coil heater and on through the sparger into the evaporator by gravity flow. The thermocouple below the sparger measures the incoming oil temperature. As the oil droplets pass through a measured quantity of water, the heat is dissipated to the water. The top oil and any water trapped by the oil overflow into a settling chamber. The glass thermometer measures the temperature of the oil as it overflows. The water that has been trapped by the oil settles to the bottom of the settling chamber and re-enters the evaporation chamber. The oil flows from the settling chamber into the separatory funnel. If any water has entered the separatory funnel with the oil, it settles to the bottom. The Sigmamotor pump transfers the oil from the separatory funnel back to the feed reservoir.

B. Materials

Fischer's bath oil was used as the heat transfer medium. This is a petroleum base mineral oil which was assumed to be derived from mid-continent crude petroleum. The specific gravity was determined to be 0.81 at a temperature of 200°F. The specific heat of 0.5 Btu/lb-°F was approximated from a specific heat versus oil temperature curve of mid-continent oils (3). Ordinary tap water was used as the heat recipient liquid.

C. Equipment

Before making experimental runs, it was necessary to design an evaporator which would allow prolonged boiling without boil-over and would facilitate separation of oil and water. The first evaporator used is shown in Figure 2, page 31. The evaporation and settling chambers were made of two pieces of 2-inch glass tubing, 30 and 18 inches long, respectively. The chambers were joined by the horizontal oil overflow tube and water return tube; these tubes and the oil exit tube were made of 1/2-inch glass tubing. The oil inlet sparger was located at the bottom of the evaporation chamber. An aluminum foil screen was placed within the settling chamber between the oil overflow tube and the oil exit tube to facilitate oil-water separation.

Adaptations of the evaporator described above were also tried. The first change was to slope the oil overflow and water return tubes downward from the evaporation chamber to the settling chamber. Short pieces of rubber tubing connected the oil overflow tube and the water return tube to the settling chamber; this avoided breakage caused by stresses in the glass when the unit cooled. Two extra tubes were attached to the evaporation chamber for intended studies at various water levels. These were plugged with rubber stoppers when not in use.

The evaporation chamber of the above model was then shortened by 6 inches; this necessitated removal of one of the extra outlet tubes. The remaining extra outlet was not used. The water return tube sloped upward from the evaporation chamber to the settling chamber.

The water return tube of the final model of the first evaporator used sloped slightly downward from the evaporation chamber to the settling chamber.

The second evaporator, shown in Figure 3, page 32, was used for making experimental runs. This evaporator consisted of two concentric pipes of stainless steel sheet metal. The 4-inch diameter inner pipe was the evaporation chamber, and the 2-inch annular space between it and the outer wall was the settling chamber. Four 1/2-inch holes were in the walls of the evaporation chamber; the two upper holes were for oil overflow, and the two lower holes were for water return. The water return holes were baffled to prevent the boiling action from spreading into the annular settling chamber. The oil exit pipe was inserted into the outer wall of the annular space above the level of and perpendicular to the center lines of the oil overflow holes. The exit tube was effectively baffled with stainless steel plates that extended well below the level of the oil overflow holes. Figure 4, page 33, shows the positioning of the oil exit tube and baffles in relation to the holes of the evaporation chamber.

Two different spargers were used during the experiment. The first sparger was constructed from a 3/4-inch galvanized standard steel pipe cap and four 1/8-inch stainless steel tube jets. The second sparger was made from a 3/8-inch cap with a single 1/4-inch OD copper tube jet. An iron-constantan thermocouple, encased in 1/8-inch stainless steel tubing, was inserted into the feed line and positioned approximately one inch below the sparger head.

Auxiliary experimental equipment included the following: a five-gallon feed reservoir with a glass tube level indicator; a copper tube coil heater with gas burner; a 1000 ml separatory funnel; an iron-constantan thermocouple attached to a Minneapolis-Honeywell temperature indicator; a -10° to 110° Centigrade thermometer; a Model T6S Sigmamotor pump; a 1/4-inch OD copper tube oil feed line; a 3/8-inch ID copper tube oil return line; two graduated cylinders, one 500 ml and one 1000 ml; a stop watch; three gate valves.

D. Experimental Procedure

The steps taken in preparation for a run are the standardization of the temperature recorder, drainage of water and foreign matter from the separatory funnel, a check of the condition of the rubber tubing of the Sigmamotor pump, and placement of a measured quantity of water into the evaporator.

At the start of a run, both the oil rate regulator valve and the oil shut-off valve are fully opened. Later, after the oil reaches a temperature of about 200°F, the oil rate is adjusted with the oil rate regulator valve. When the oil reaches the separatory funnel, the pump is turned on and adjusted to keep the liquid level in the separatory funnel constant. The temperature of the inlet oil is controlled by adjusting the burner flame in the coil heater. The system is allowed to run until boiling occurs in the evaporation chamber. After a constant inlet oil temperature has been reached, the temperatures of the oil entering and leaving the evaporator, the barometric pressure, and the oil flow rate are recorded. The oil flow rate is measured by diverting the oil leaving the evaporator unit into a 100-ml graduated cylinder; the flow time for a volume of oil is measured. The heater is then turned off and the oil shut-off valve closed. Finally, the water volume in the evaporator is measured by draining the water in the evaporator into a 500-ml graduated cylinder. Make-up water is added before the next run.

In the above process the following factors were varied:

1. A series of runs was made in which the oil flow rate was varied from 7 to 15.5 ml/sec. A set of runs for each flow rate was made in which the inlet oil temperature was varied. The oil-to-water ratio was held constant throughout the series of runs.
2. A series of runs was made in which the water-to-oil ratio was varied. Water volumes of 525, 500, 475, 450, and 435 ml corresponding to water-to-oil ratios of 0.41:1, 0.38:1, 0.36:1, 0.33:1, and 0.32:1, respectively, were used. For each water-to-oil ratio, a set of runs was made in which the inlet oil temperature was varied. The oil flow rate was held constant throughout the series of runs.
3. The oil inlet temperature was varied while the oil flow rate and water-to-oil ratio were held to small variations from a particular value, and a sparger with a single, large diameter jet was used in place of the sparger with the four small diameter jets.

RESULTS AND DISCUSSION

A. Glass Tube Evaporator and Adaptations

The settling and evaporation chambers of the original glass evaporator model were joined by 1/2-inch glass tubes. Since no allowance was made for uneven thermal expansion and contraction in the glass, the connecting tubes either cracked or broke when the unit was subjected to heat or was allowed to cool. Other disadvantages, also inherent in the other glass evaporator models, included the following:

1) A glass evaporator has very few boiling sites. This caused the water to superheat before boiling occurred. When a vapor bubble did form, it became large and stable in the 2-inch diameter tubing; this caused boil-over at inlet oil temperatures above the boiling point of the water. An attempt was made to control the superheating by introducing artificial nuclei in the form of small air bubbles until a natural rolling boil took place. However, as soon as the source of these nuclei was removed, superheating re-occurred.
2) The diameter of the evaporator was small, so any large vapor bubble that formed remained stable and forced oil over the top.
3) The water that collected in the separation chamber did not flow back into the evaporation chamber.

Rather, it superheated or boiled in the settling chamber; this caused excessive amounts of water to overflow with the exiting oil into the separatory funnel.

4) The temperature of the oil leaving the evaporator was too near that of the boiling water to allow the calculation of a log mean temperature difference. This temperature difference is needed to calculate the overall volumetric heat transfer coefficient. The evaporation chamber was shortened in an attempt to gain a greater temperature difference between the boiling water and the outgoing oil. This attempt failed.

The glass evaporator models did have the obvious advantage of transparency which allowed observation of the action within. After boiling started in the evaporator, the action resembled a fluidized bed. The action extended from the tips of the sparger to the top of the liquid mixture. The water below the tips of the sparger was undisturbed.

A mixture of oil and water passed through the oil overflow tube into the settling chamber. The separation of the water from the oil could be observed. Water droplets clung to the oil layer surrounding a vapor bubble. Bubbles which had a buoyancy force greater than the force due to the weight of the clinging water droplet rose and were caught in the current of the exiting oil. Therefore, a baffle was necessary to keep the water from leaving the system in this way.

B. Concentric-Pipe Evaporator

The concentric-pipe evaporator did not have the disadvantages of the glass evaporator models. However, with an evaporator constructed with stainless steel, observation was restricted. The 4-inch diameter of the evaporation chamber allowed the larger vapor bubbles to break up before they caused boil-over. The annular settling chamber, much larger than the settling chamber of the previous design, allowed a longer settling time; this improved the oil-water separation. With this evaporator, the temperature measurement of the oil leaving the evaporation chamber was possible. This made possible the calculation of log mean temperature differences and, therefore, of heat transfer coefficients. This evaporator was used to make the experimental runs for this project.

C. Equations

The volumetric heat transfer coefficients were calculated using the equation

$$U_v = \frac{q}{(V)\,\Delta T_m} \qquad (1)$$

where U_v = overall volumetric heat transfer coefficient, Btu/(hr)(ft^3)(°F);

q = quantity of heat transferred from oil to water per hour, Btu/hr;

V = transfer volume, ft^3;

ΔT_m = log mean temperature difference, °F $\dfrac{(T_{in} - T_W) - (T_{out} - T_W)}{\ln\left(\dfrac{T_{in} - T_W}{T_{out} - T_W}\right)}$

T_{in} = inlet oil temperature, °F;

T_{out} = exiting oil temperature, °F;

T_W = temperature of boiling water, °F.

The quantity of heat being transferred per hour, as used in the preceding equation, was calculated using the equation

$$q = wC\Delta T \tag{2}$$

where w = mass rate of flow of oil, lbs/hr;

C = heat capacity of oil, Btu/lb-°F;

ΔT = temperature difference of the oil, entering and leaving, °F.

Equations (1) and (2) may be combined to yield

$$U = \frac{wC}{A}\left[\ln(T_{in} - T_W) - \ln(T_{out} - T_W)\right] \tag{3}$$

Sample calculations of two runs under different conditions are shown on page 24 in the Appendix.

The method of least squares was used to determine the equations of the straight line relationships of experimental results, Figures 5-9, pages 34-39 , and Figure 18, page 47. In determinant form the equation of a straight line "through" the experimental points may be written as (6):

$$\begin{vmatrix} x & y & 1 \\ \sum_{i=1}^{n} x_i & \sum_{i=1}^{n} y_i & \sum_{i=1}^{n} 1 \\ \sum_{i=1}^{n} x_i & \sum_{i=1}^{n} y_i x_i & \sum_{i=1}^{n} x_i \end{vmatrix} = 0 \tag{4}$$

where in this case the x_i's are the oil inlet temperatures, the y_i's are the volumetric heat transfer coefficients calculated from (1), and n is the number of experimental points.

D. Experimental Results for the Concentric-Pipe Evaporator

Experimental data for the concentric-pipe evaporator are listed in Table I, page 27. Tabulated data for each experimental run are the oil flow rate, water-to-oil ratio, inlet and outlet oil temperatures, temperature approach, and the volumetric heat transfer coefficient.

Since the temperature of boiling water does not change, the temperature differential between the boiling water and the incoming oil can be varied by varying the inlet oil temperature. Figures 5-9, pages 34-38, show the curves of the heat transfer coefficient versus the inlet oil temperature at various ranges of oil flow rates. The experimental points seem to form straight lines with negative slopes; the volumetric heat transfer coefficient decreases as the oil inlet temperature increases. The experimental points of Figures 8 and 9 are more scattered than those of Figures 5 and 7. This can be explained by the logarithmic terms in equation (3). At the high oil flow rates, high inlet oil temperatures could not be reached; therefore, the temperature difference in the first logarithmic term was low. However, the temperature difference of the second logarithmic term is increased by the high oil flow rate. This temperature difference is smaller than that in the first logarithmic term. Therefore, the second logarithmic term is more sensitive to small changes than is the first.

Since the magnitude of both terms is approximately the same, the difference is quite sensitive to the outgoing oil temperature. The volumetric heat transfer coefficient is, therefore, quite sensitive to the outgoing oil temperature. Figure 10, page 39, shows all of the curves of volumetric heat transfer coefficient versus the oil inlet temperature on a common graph. This figure indicates the existence of a maximum flow rate range, since the slopes of the curves determined by points obtained in the flow rate ranges of 7.0 to 7.6 ml/sec, 8.5 to 10.5 ml/sec, 13.6 to 14.2 ml/sec, and 15.0 to 15.7 ml/sec are less than the slope of the curve determined by points obtained in the flow rate range 12.6 to 13.0 ml/sec.

From data given by the above curves, a series of curves of the heat transfer coefficient versus oil flow rate were determined, Figures 11-13, pages 40-42. These curves show maximum heat transfer coefficients of approximately 1300 Btu/(hr)(ft^3)(°F) at an oil inlet temperature range of 214 to 226°F, 1000 Btu/(hr)(ft^3)(°F) at a temperature range of 229 to 238°F, and 750 Btu/(hr)(ft^3)(°F) at a temperature range of 241 to 251°F, all at a common oil flow rate range of 11.0 to 12.5 ml/sec. These curves thus show a decrease in the maximum value of the heat transfer coefficient as the oil inlet temperature increases. This can be better observed in Figure 14, page 43, which shows the above curves on a common graph.

Both sets of curves discussed above show one common factor. The highest heat transfer coefficients are obtained at the lowest temperatures where the least amount of heat is being transferred. This is understandable when comparing the calculations of the coefficients at two condition extremes in the same flow rate range. (Refer to the sample calculations on page 24 in the Appendix.) The amount of heat being transferred per hour in the second condition is about twice that being transferred in the first condition but the log mean temperature difference in the second condition is close to three times that of the first condition. This, again, may be explained by the logarithmic terms of equation (3). As the inlet oil temperature increases, the temperature differential between the boiling water and the outgoing oil increases. Since this temperature difference is smaller than that in the first logarithmic term, the value of the second logarithmic term increases faster than the value of the first logarithmic term. Where the temperature differential of the outgoing oil and boiling water was relatively small, as in condition I, the second logarithmic term was close to zero and insignificant. Where the temperature differential of the outgoing oil and boiling water was relatively high, as in condition II, the second logarithmic term increased in value more than the first logarithmic term. Therefore, as the incoming oil temperature increased, the second logarithmic term increased in significance.

It can also be seen from equation (3) that if the temperature of the exiting oil could be held to a low value as the inlet oil temperature increases, the negative logarithmic term would remain insignificant. Thus the volumetric heat transfer coefficient could increase with an increase of the inlet oil temperature.

The series of runs to determine the effects of oil-to-water ratio variation on the heat transfer coefficient was made using a constant oil flow rate range of 9 to 10 ml/sec. Since the level of the oil and water together was fixed by the location of the exit spout, the ratio could be controlled by varying the amount of water in the evaporator. The volume of 525 ml was the maximum allowable without incurring boiling in the annular settling chamber. The minimum volume allowable without incurring excessive superheating was 435 ml. The results of this series of runs are shown graphically in Figure 15, page 44. Since the same oil flow rate range was used for each run, the curves have approximately the same slopes. In considering the three middle curves, it is noticed that the slopes are very close to being equal and that the water volumes differ by equal increments. Also, considering the same curves, the distance between each of the outer curves and the center curve is approximately equal. This indicates that if a similar apparatus having a deeper evaporation chamber were used, facilitating a greater water-to-oil ratio, higher heat transfer coefficients could be obtained.

A close temperature approach, i.e., a small temperature differential between the boiling water and the exiting oil, is desirable for the design for evaporators with many effects. Figure 16, page 45, shows the curves of the oil inlet temperature versus the temperature approach at various oil flow rates. This figure shows that the temperature approach increases as the oil flow rate increases at a constant incoming oil temperature. Figure 17, page 46, contains curves of the temperature approach versus the oil inlet temperature at various water-to-oil ratios. This figure indicates the temperature approach decreases with increasing water-to-oil ratios at a constant oil inlet temperature. Therefore, a close temperature approach would be obtained at high water-to-oil ratios and low oil flow rates.

The final series of runs was attempted using a sparger with one jet of large diameter in place of a sparger with four jets of smaller diameter. It was observed from the glass evaporator that a rolling action resembling a fluidized bed began at full boil. If the rolling action had enough force to break up the larger oil droplets of the single-jet sparger, heat transfer coefficients similar to those obtained with the four-jet sparger could be expected.

The single-jet sparger did not extend as far into the evaporator and did not displace as much water. This allowed water-to-oil ratios greater than those obtained with the four-jet sparger.

Several runs were made but sporadic results were obtained. The primary cause of the unreliability of these results was an uncorrectable fluctuation of the temperature of the outgoing oil. This temperature was quite critical in determining the log mean temperature difference used for calculating the heat transfer coefficient. Another cause was the uneven boiling that took place.

This series of runs did show, however, that the heat transfer coefficients would be much lower; this indicated that the action in the evaporator could not break up the larger droplets. Figure 18, page 47, compares the points obtained from this series of runs with one of the curves from Figure 15. This figure shows that comparable heat transfer coefficients were not reached even though greater water-to-oil ratios were used.

SUMMARY

The highest boiling range volumetric heat transfer coefficients, obtained by semi-batch evaporation in the concentric-pipe evaporator, occur at the lowest oil inlet temperatures. As the oil inlet temperature increases at a constant oil flow rate, the heat transfer coefficient decreases linearly.

An optimum oil flow rate exists for the concentric-pipe batch evaporator.

The boiling range volumetric heat transfer coefficient increases as the water-to-oil ratio increases. This indicates higher coefficients could be obtained if the water capacity of the evaporator were increased.

The volumetric heat transfer coefficients obtained were about one order of magnitude smaller than those reported in the literature, therefore compare unfavorably. However, the coefficients reported were obtained from counter-current flows of oil and water and most were obtained from data determined using non-boiling liquid-liquid heat transfer.

The boiling action in the evaporation chamber does not have sufficient force to completely break up the larger oil droplets emitted from the single, large diameter jet.

Close temperature approaches may be obtained at high water-to-oil ratios and low oil flow rates.

SUGGESTIONS FOR FUTURE STUDY

The results of this project indicated that higher boiling range heat transfer coefficients could be obtained if greater oil-to-water ratios were used. The concentric-pipe evaporator used in this study could be given a deeper evaporation chamber. Higher flow rates and a larger heater would be necessary if the change was made. The water-to-oil ratio could be increased without enlarging the transfer volume by elevating the water return holes. This would also decrease the oil hold-up and allow more even boiling.

Before boiling occurred in the evaporator, the oil droplets emitted from the spargers were observed rising in the middle of the evaporation chamber. A sparger with many radiating jets might allow more efficient heat dispersion.

APPENDIX

SAMPLE CALCULATIONS

Condition I

Temperature of incoming oil = 221°F

Temperature of outgoing oil = 204.1 °F

Oil Flow Rate = 8.98 ml/sec

Volume of Water = 510 ml

Temperature of boiling water = 203.1°F

Equation (2) with unit equation of conversion factors:

$$q = wC\Delta T$$

$$q = (cc/sec)(sec/hr)(Btu/lb\text{-}°F)(g/cc)(lbs/g)(°F) = Btu/hr.$$

$$q = (8.98)(3600)(0.5)(0.81)(1/454)(221 - 204.1) = 488 \text{ Btu/hr.}$$

$$\Delta T_m = (\Delta T_1 - \Delta T_2)/\ln(\Delta T_1/\Delta T_2) = 16.9/\ln(17.9/1) = 5.96°F.$$

Equation (1):

$$U = q/V\Delta T_m$$

$$U = 488/(5.96)(0.0636) = \underline{1290 \text{ Btu/(hr)(ft}^3\text{)(°F)}}.$$

Condition II

Temperature of incoming oil = 240°F

Temperature of outgoing oil = 209.5°F

Temperature of boiling water = 203.1°F

Oil Flow Rate = 9.36 ml/sec

Volume of Water = 500 ml

SAMPLE CALCULATIONS (continued)

Equation (2), calculations include combined conversion factors:

$$q = wC\Delta T$$

$$q = (9.36)(3.21)(30.5) = 916 \text{ Btu/hr.}$$

$$\Delta T_m = 30.5/\ln 36.9/6.4 = 17.4°F$$

Equation (1):

$$U = q/V\Delta T_m$$

$$U = 916/(17.4)(0.0636) = \underline{827 \text{ Btu/(hr)(ft}^3\text{)(°F)}}.$$

TABLE I -- Experimental Data, Four-Jet Sparger

Oil Flow Rate (ml/sec)	Water-to-Oil Ratio	Oil Temperature (°F)		Temperature Approach (°F)	Volumetric Heat-Transfer Coefficient (Btu/(hr)(ft^3)(°F)
		Inlet	Outlet		
7.0	0.44:1	251.5	212.3	9.1	558
7.3	"	245.5	210.0	6.8	674
7.5	"	237.5	208.3	5.1	722
7.5	"	242.5	208.8	5.6	735
7.6	"	233.5	207.0	3.8	785
7.6	"	249.0	214.6	11.3	544
8.5	"	226.0	205.5	2.2	1000
8.8	"	232.0	207.4	4.1	865
8.8	"	241.0	210.5	7.2	738
9.1	"	247.0	212.5	8.5	745
9.3	"	234.0	208.3	5.0	848
10.0	"	222.0	206.5	3.0	1000
10.0	"	248.0	214.5	11.0	708
10.3	"	244.0	213.0	9.5	727
10.5	"	232.0	208.5	5.0	922
12.6	"	222.0	206.5	2.5	1260
12.6	"	238.0	212.0	8.0	922
12.7	"	230.0	208.3	4.3	1140
12.8	"	225.0	206.5	2.5	1370
12.9	"	248.0	217.5	13.5	747

TABLE I -- (continued)

Oil Flow Rate (ml/sec)	Water-to-Oil Ratio	Oil Temperature (°F) Inlet	Outlet	Temperature Approach (°F)	Volumetric Heat-Transfer Coefficient (Btu/(hr)(ft^3)(°F)
13.0	0.44:1	222.0	206.5	2.5	1300
13.6	"	217.0	206.5	3.2	995
13.6	"	221.5	208.3	5.0	889
13.8	"	225.0	210.0	6.7	820
14.0	"	231.0	211.0	7.4	922
14.0	"	237.5	214.0	10.4	834
14.1	"	233.0	212.0	8.7	873
14.1	"	214.0	206.0	2.6	1000
14.2	"	229.0	211.0	7.7	868
15.0	"	218.0	210.0	6.4	614
15.2	"	226.0	212.8	9.2	682
15.2	"	222.0	211.0	7.4	701
15.2	"	230.0	213.7	10.1	738
15.2	"	223.0	211.0	7.6	735
15.6	"	226.0	214.5	10.9	567
15.7	"	231.0	216.4	12.8	595
9.7	0.41:1	225.0	204.7	1.2	1410
9.7	"	231.0	205.5	2.0	1250
9.2	"	235.0	206.5	3.0	1060
9.6	"	241.0	209.3	5.8	891
9.0	0.38:1	221.0	204.1	1.0	1290

TABLE I -- (continued)

Oil Flow Rate (ml/sec)	Water-to-Oil Ratio	Oil Temperature (°F) Inlet	Outlet	Temperature Approach (°F)	Volumetric Heat-Transfer Coefficient (Btu/(hr)(ft^3)(°F)
9.2	0.38:1	229.0	205.0	1.9	1200
9.2	"	235.0	207.4	4.3	927
9.4	"	240.0	209.5	6.4	827
9.9	0.36:1	227.0	206.0	2.6	1110
9.1	"	230.0	206.5	3.1	987
9.1	"	233.0	207.5	4.1	911
9.4	"	242.0	212.0	8.6	715
8.7	0.33:1	220.0	204.7	1.3	1150
9.4	"	226.0	206.0	2.6	1030
9.3	"	233.0	208.3	4.9	845
9.3	"	236.0	209.2	6.1	791
8.4	"	238.0	210.0	7.0	683
9.4	0.32:1	224.0	206.0	2.6	981
9.4	"	227.0	207.0	3.6	896
9.9	"	233.0	209.3	5.9	806
9.4	"	241.0	214.4	11.0	586

TABLE I -- (continued)

Single Jet Sparger

Oil Flow Rate (ml/sec)	Water-to-Oil Ratio	Oil Temperature (°F) Inlet	Oil Temperature (°F) Outlet	Temperature Approach (°F)	Volumetric Heat-Transfer Coefficient (Btu/(hr)(ft^3)(°F)
9.4	0.63:1	231.0	209.2	6.5	695
9.8	0.61:1	248.0	212.4	9.7	759
8.9	"	218.0	204.8	2.1	894
9.1	"	227.0	207.7	5.0	722
9.7	"	235.0	210.1	7.0	743
9.1	0.57:1	219.0	206.0	2.9	781
9.1	"	216.0	204.7	1.6	955
9.7	0.53:1	242.0	215.6	12.9	548
9.6	0.49:1	229.0	212.6	9.5	486
9.6	"	240.0	214.6	11.5	563

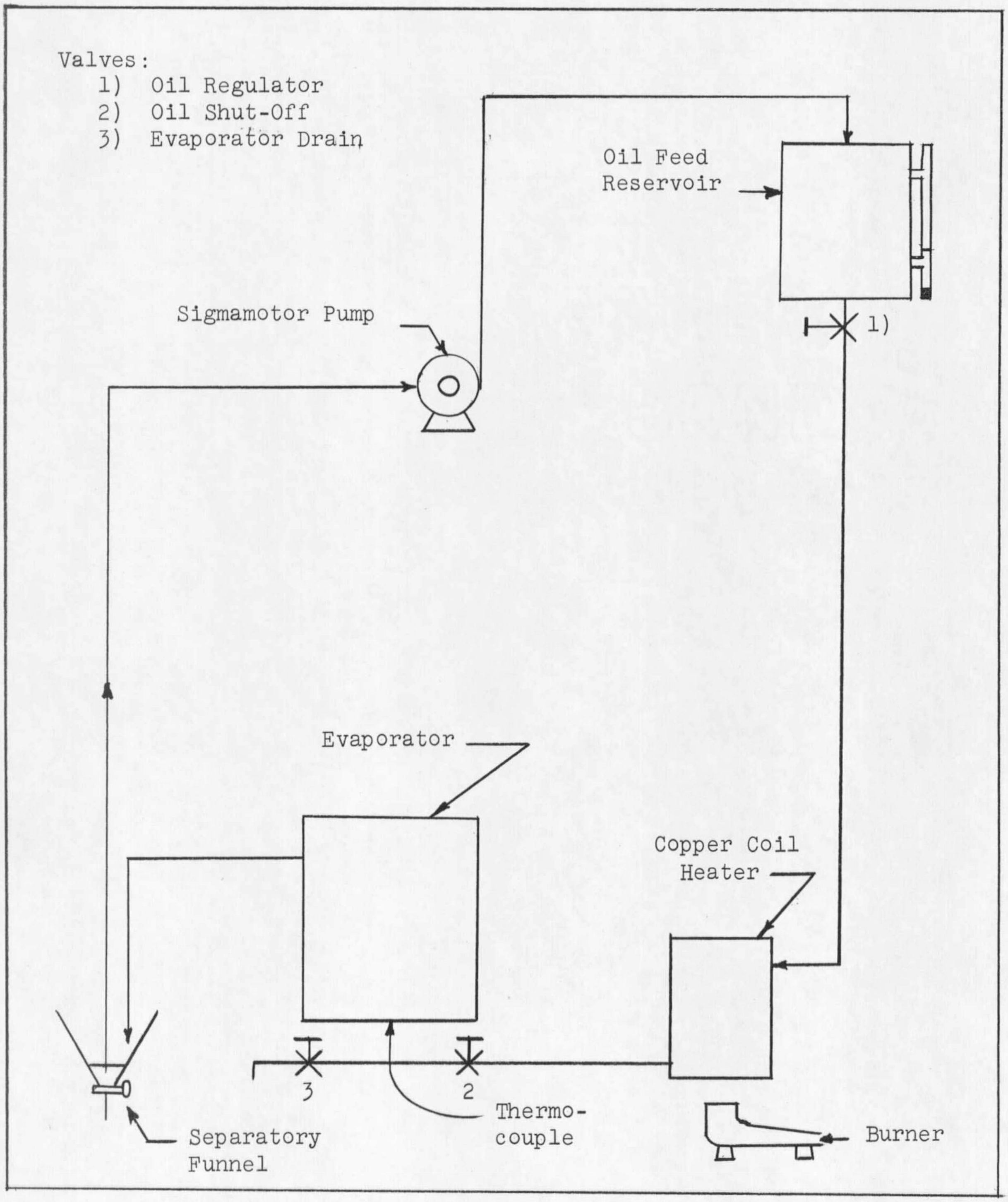

Figure 1. Flow Diagram of Experimental Process.

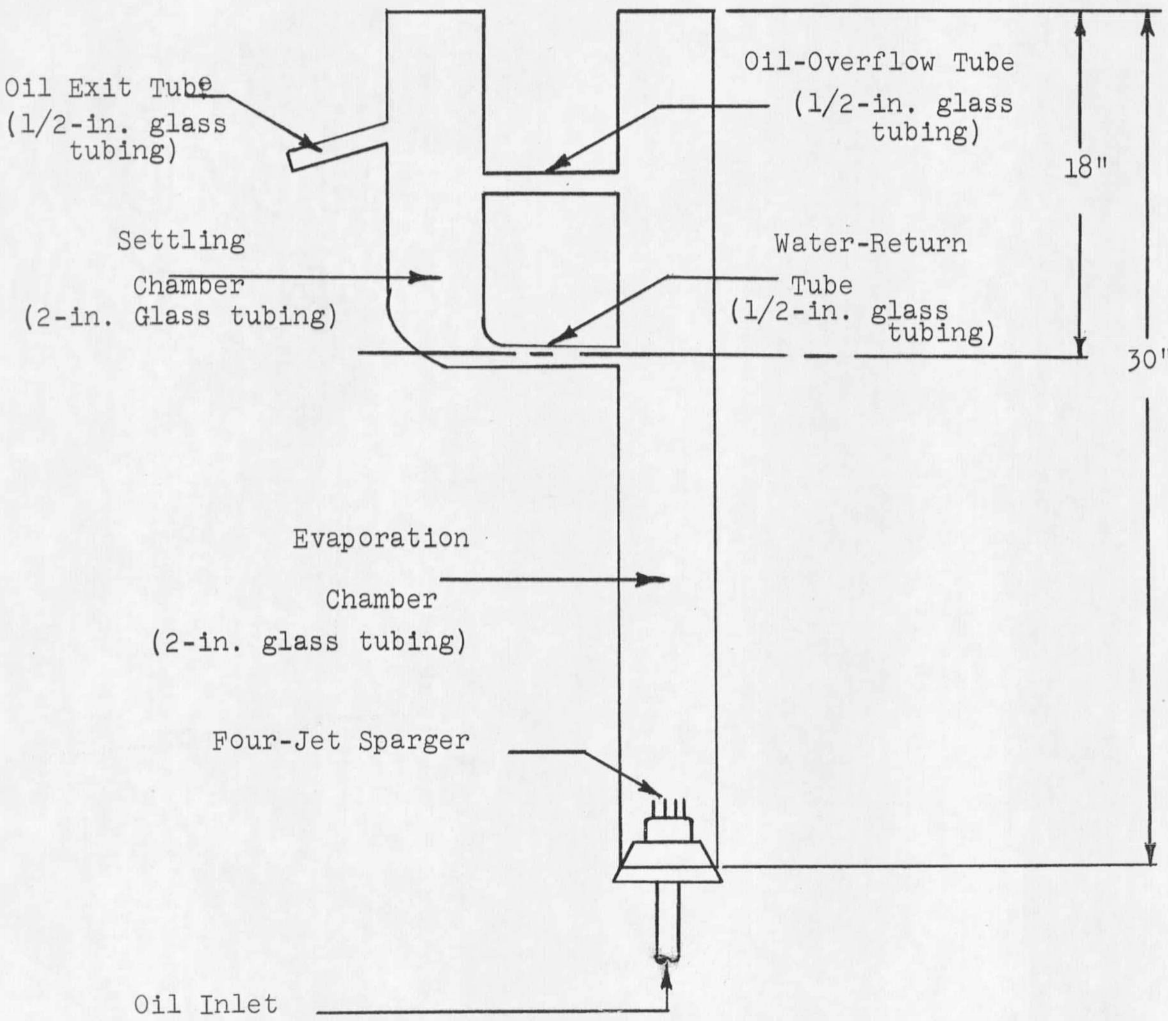

Figure 2. Original Glass Tube Evaporator Design.

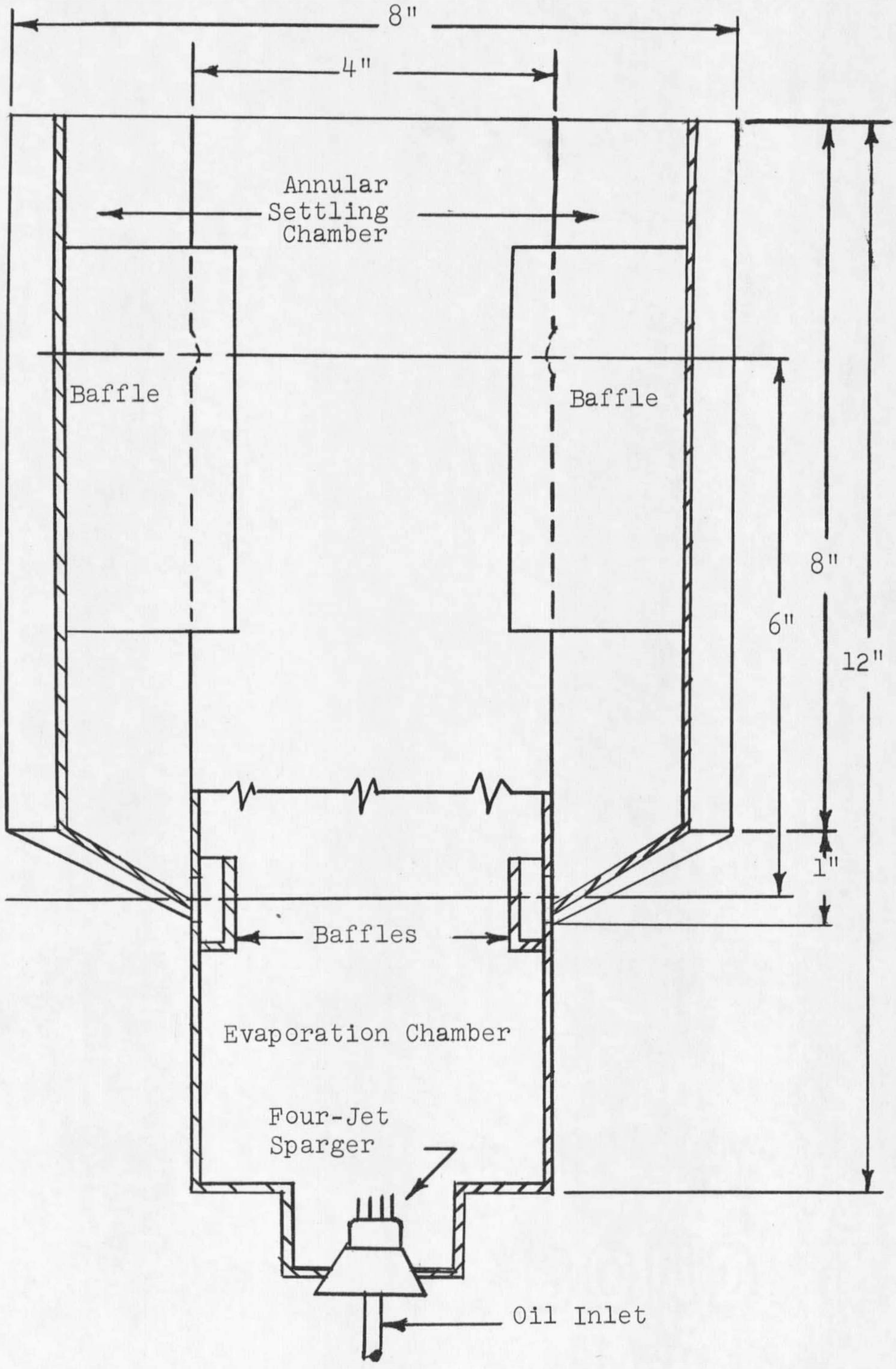

Figure 3. Cutaway Diagram of the Concentric-Pipe Evaporator.

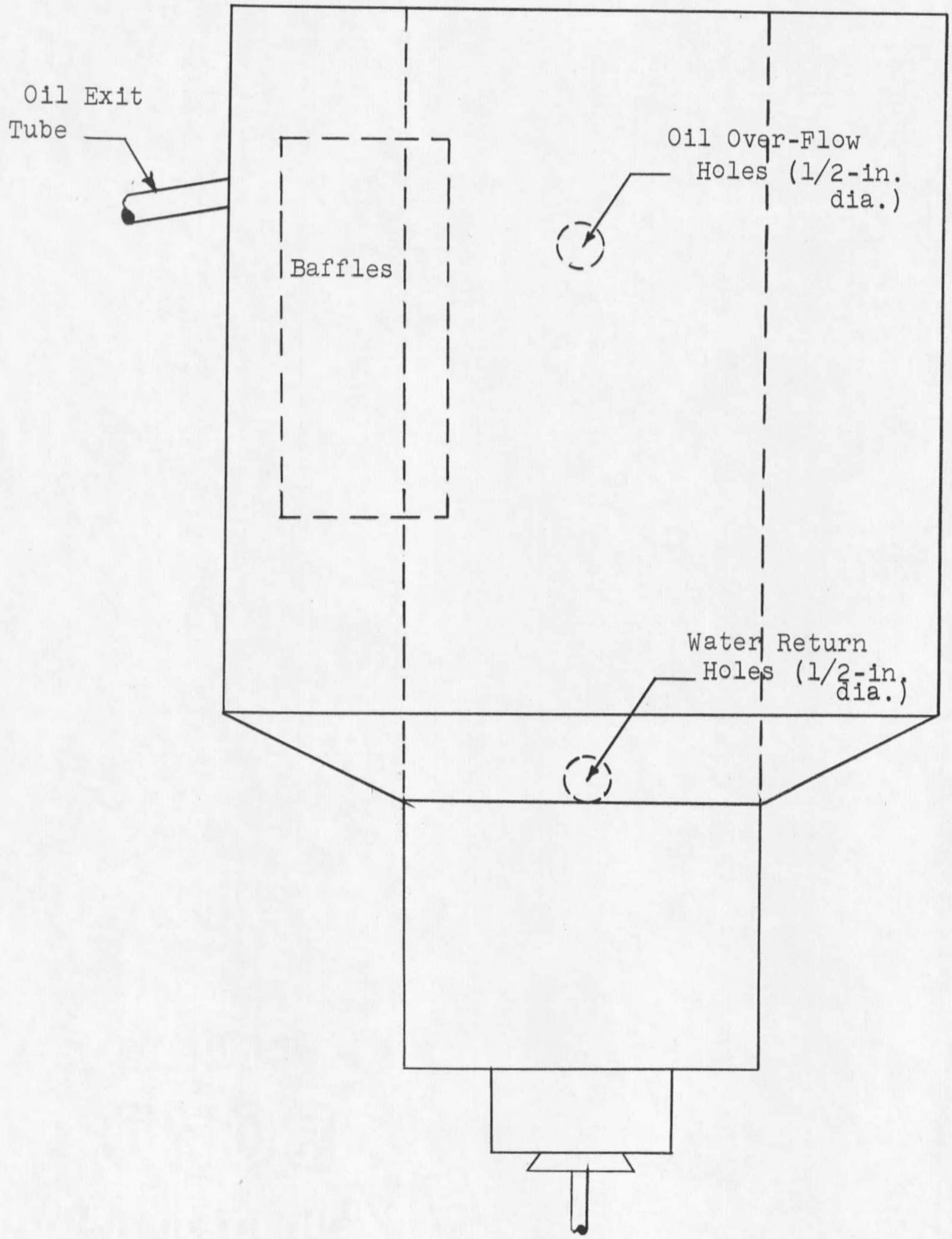

Figure 4. Exit Tube and Settling Chamber Baffle Positions in the Concentric-Pipe Evaporator.

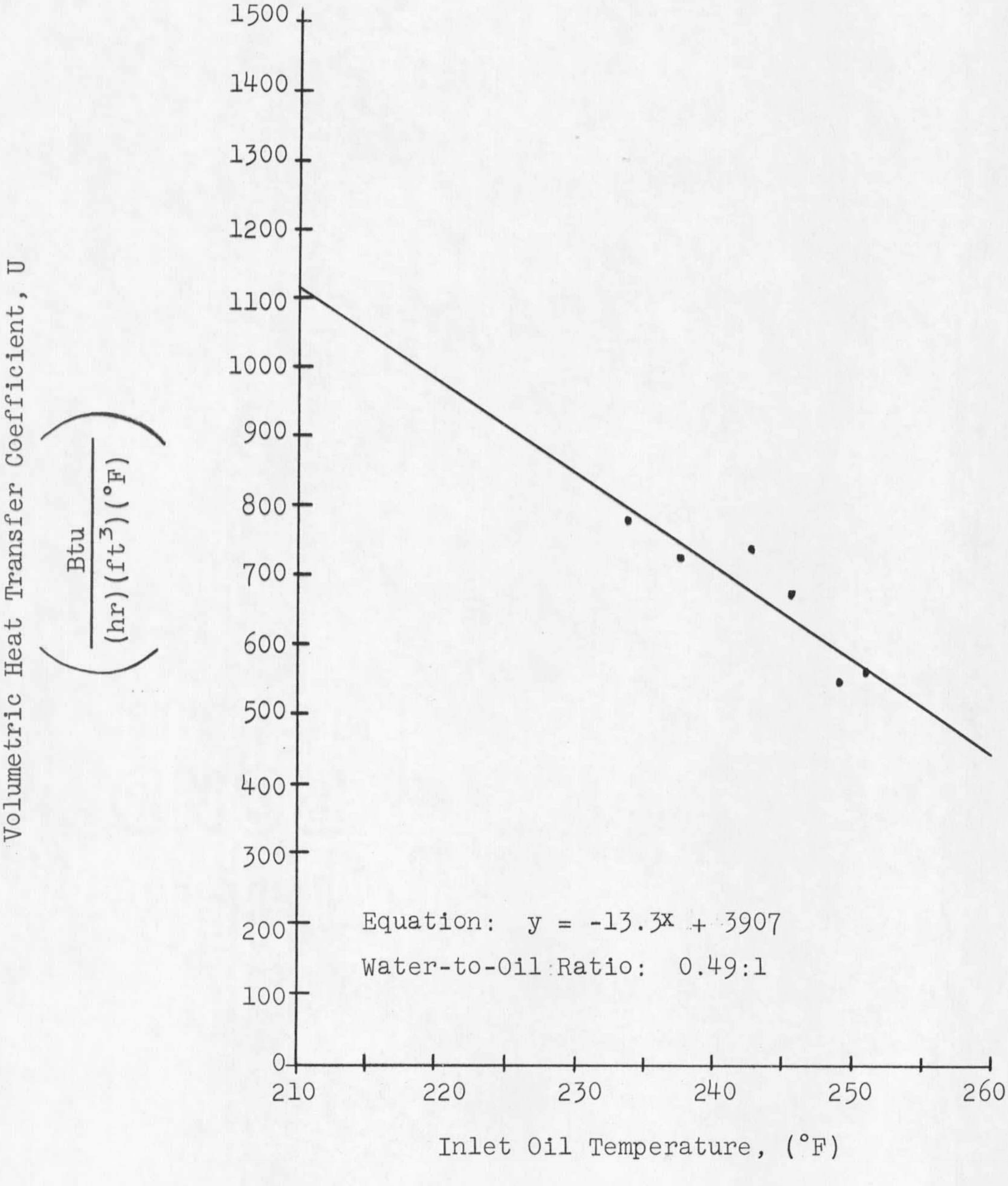

Figure 5. Volumetric Heat Transfer Coefficients at Oil Flow Rate Range 7.0 - 7.6 ml/sec.

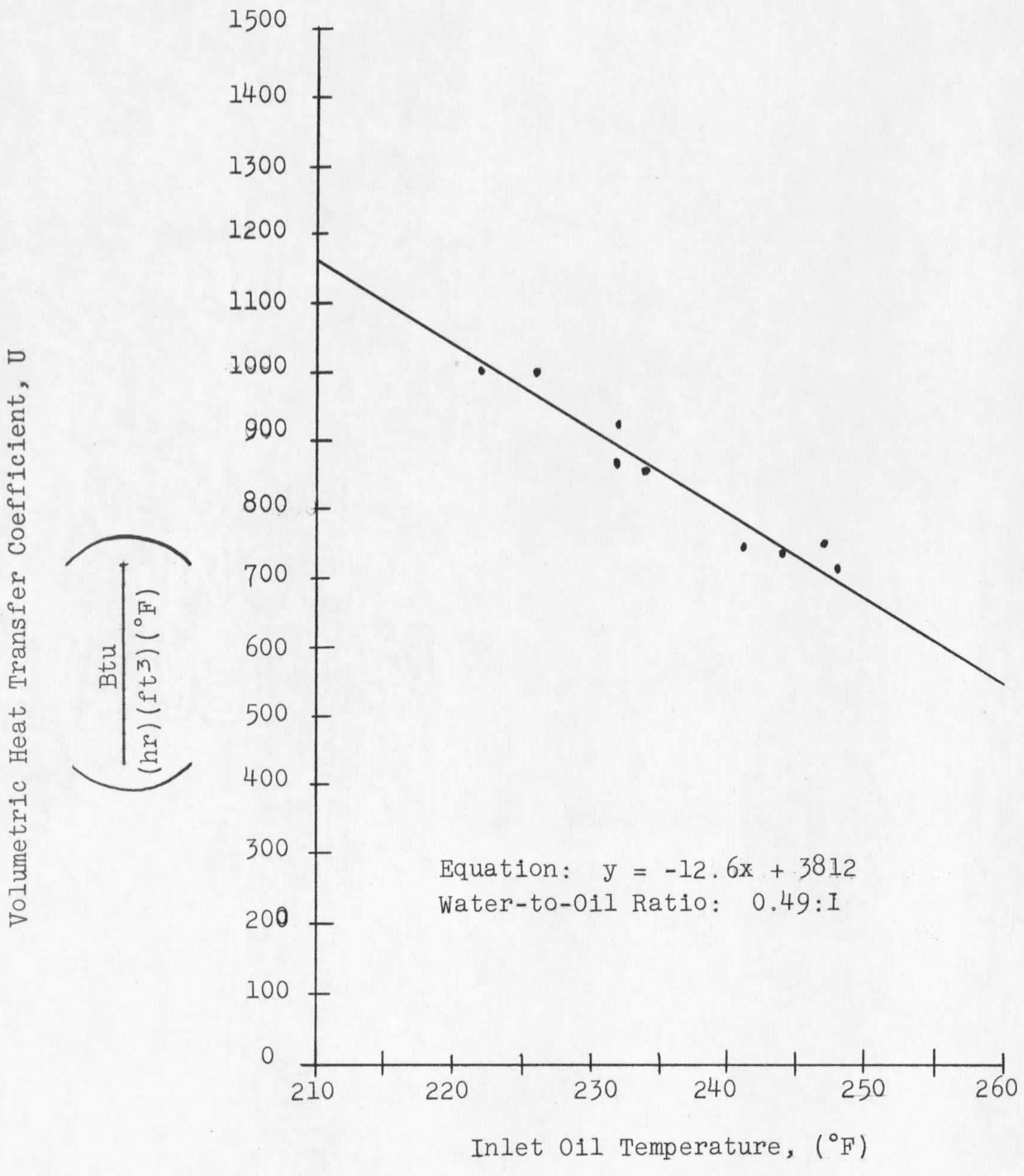

Figure 6. Volumetric Heat Transfer Coefficients at Oil Flow Rate Range 8.5-10.5 ml/sec.

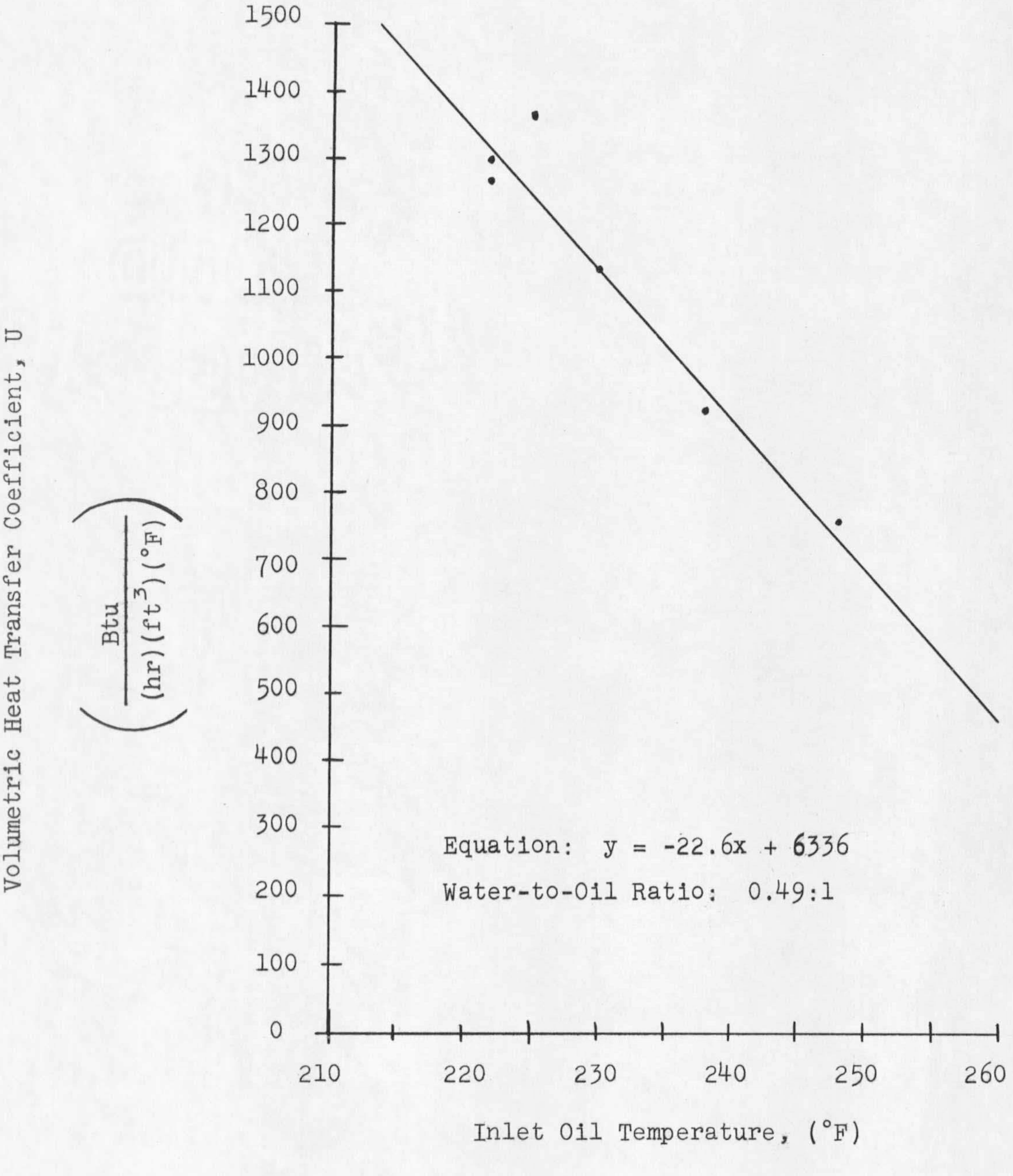

Figure 7. Volumetric Heat Transfer Coefficients at Oil Flow Rate Range 12.6-13.0 ml/sec.

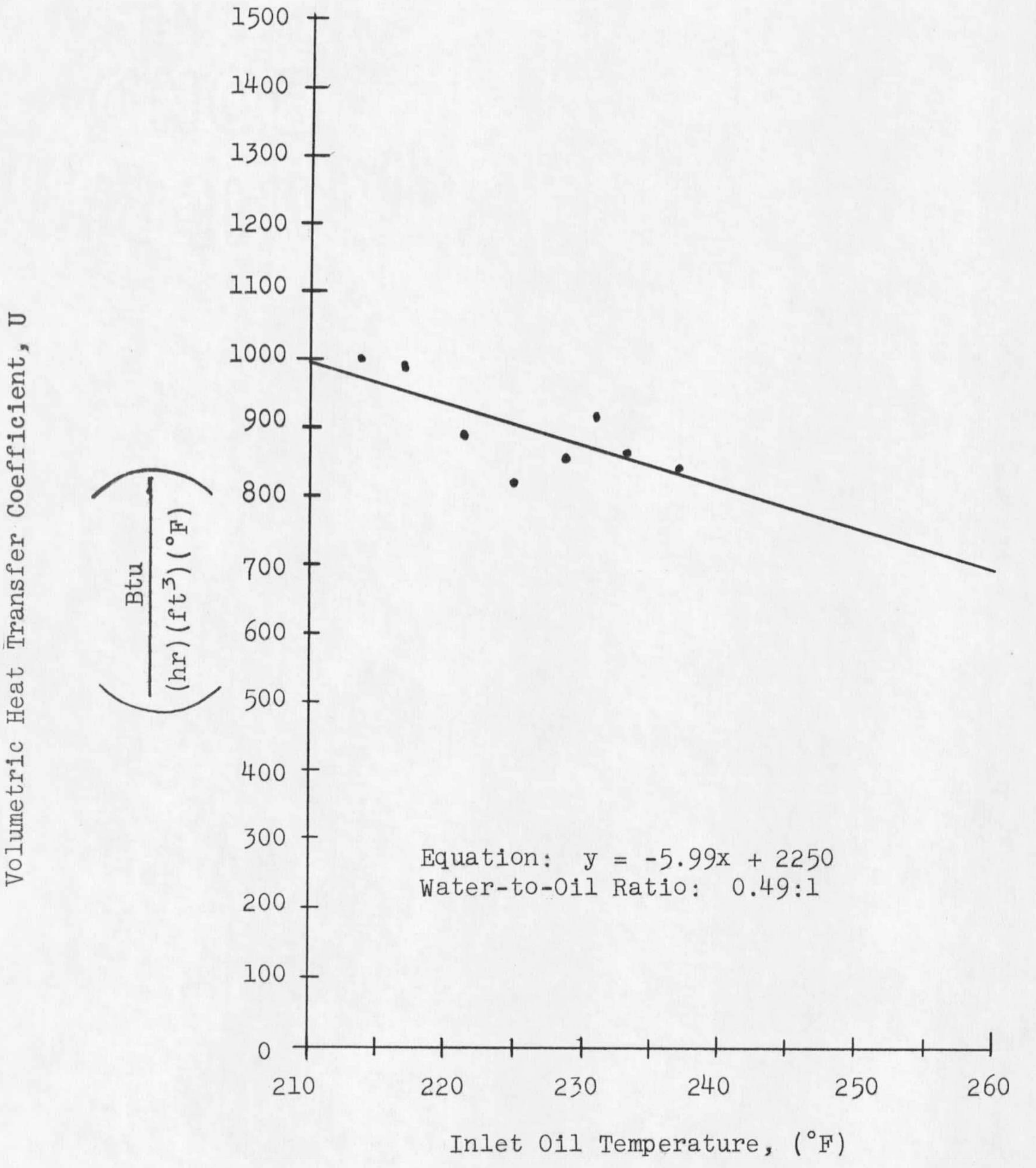

Figure 8. Volumetric Heat Transfer Coefficients at Oil Flow Rate Range 13.6-14.2 ml/sec.

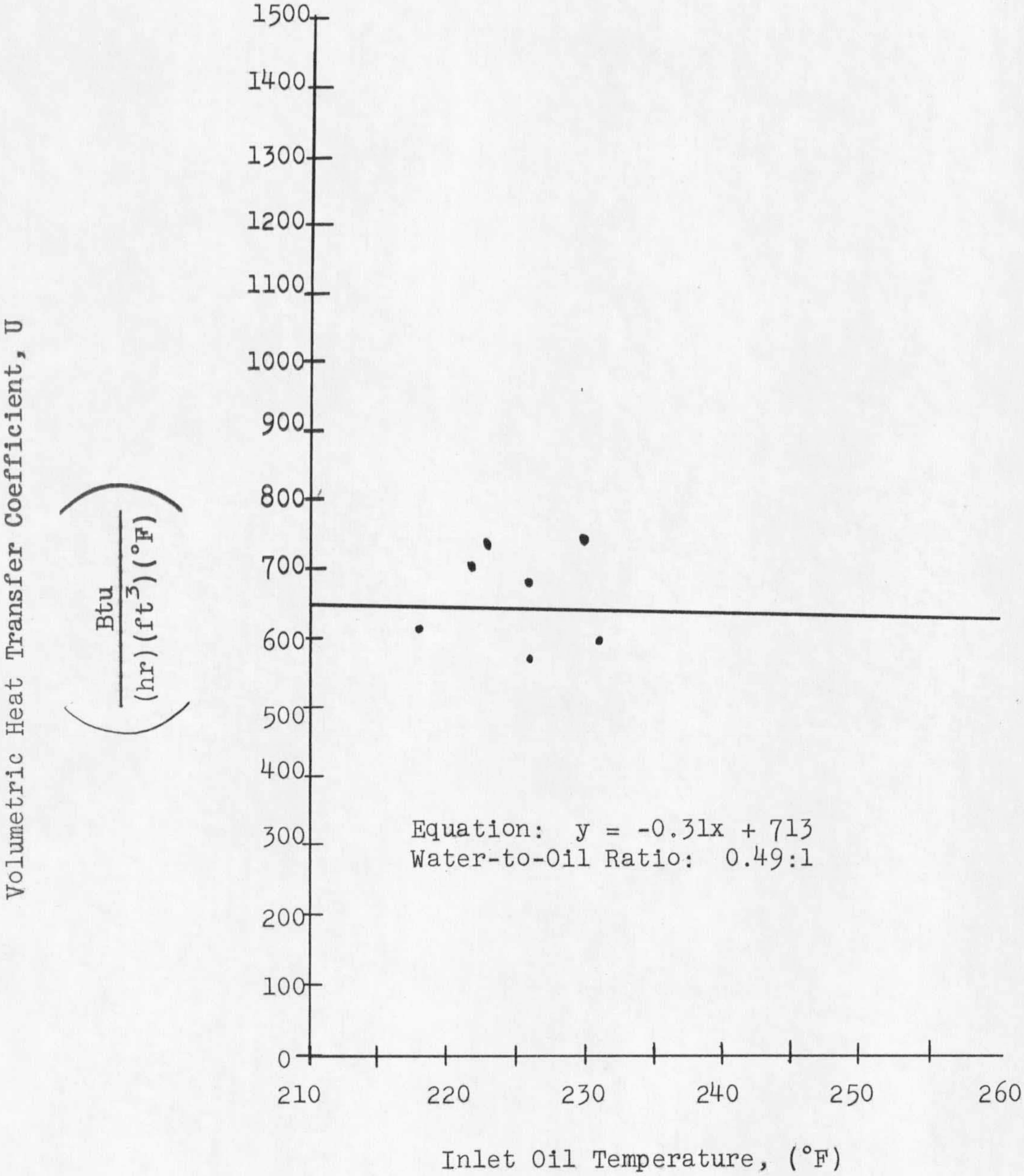

Figure 9. Volumetric Heat Transfer Coefficients at Oil Flow Rate Range 15.0-15.7 ml/sec.