DETERMINATION OF THE CHARACTERISTICS OF HEAT TRANSFER FROM A HORIZONTAL SILVER SURFACE TO BOILING MIXTURES OF ETHANOL AND BENZENE

by

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A Thesis Submitted for Partial
Fulfillment of the Requirements
for the
Degree of Master of Science
in
CHEMICAL ENGINEERING

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Virginia Polytechnic Institute
Blacksburg, Virginia

1949

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I. INTRODUCTION

Heat transfer to a boiling liquid is of primary industrial concern. There are few chemical plants which do not utilize such heat transfer somewhere in their process. The transfer of heat from a metal surface to a boiling liquid employs both the principle of conduction and convection and in some extreme cases may incorporate radiation. Such exchange of heat is believed to be affected by a considerable number of variables among which are surface tension, viscosity, equilbrium wetting angle, surface condition of the metal, temperature gradient between metal and liquid, concentration and others.

The major variables currently used as criteria for evaluating the boiling characteristics of liquids are the temperature gradient existing between the metal and the liquid, termed thermal driving force or "dt", and the rate of heat transfer, termed heat flux. Heat flux and temperature are interdependent, i.e., for low values of heat flux and driving force an increase in thermal driving force results in an increase in heat flux; while for high values of heat flux and driving force an increase in driving force eventually results in a maximum value of heat flux. Once the maximum heat flux is attained any further increase in driving force results in a decrease in heat flux because of the

formation of a partial or total vapor blanket at the liquid-metal interface which increases the resistance to heat transfer. The temperature gradient existing between the heating surface and the main body of liquid at maximum heat flux is termed the critical driving force or dt_c, because with this gradient the vapor blanket begins to form.

Data concerning the critical driving force and maximum heat flux enables designers and operators to obtain maximum performance from their equipment. The literature contains limited data concerning the critical driving force and maximum heat flux for pure liquids, solutions of salts in liquids, and binary mixtures of miscible liquids but the information is by no means sufficient. The available correlations are inadequate for other than the specific cases for which they were derived. Most noticeably lacking from the available literature are relationships involving binary mixtures of miscible liquids which would enable interpolation for values of critical driving force and maximum heat flux for intermediate concentrations.

The purpose of this investigation was to relate maximum heat flux and critical driving force with the concentration of ethanol in benzene.

II. LITERATURE REVIEW

The literature reviewed in this investigation included the available technical periodicals and books pertaining to the fields of chemistry, chemical engineering, mechanical engineering, and physics. The following review is the result of a comprehensive study of the information extracted from the literature surveyed.

Types of Boiling

Boiling of liquids has been classified into four major types or categories. McAdams, et al[25] have defined these four types both in terms of the physical action which takes place at the liquid-metal interface and in terms of the relationship between the heat transfer rate and the thermal driving force (Figure 1).

Convective Boiling. Sector A-B (Figure 1) represents a state of convective boiling, i.e., the liquid, in intimate contact with the heating surface, is heated to a slightly higher temperature than the main body of liquid. Convection currents are set up in the liquid and vaporization occurs at the liquid-vapor interface.

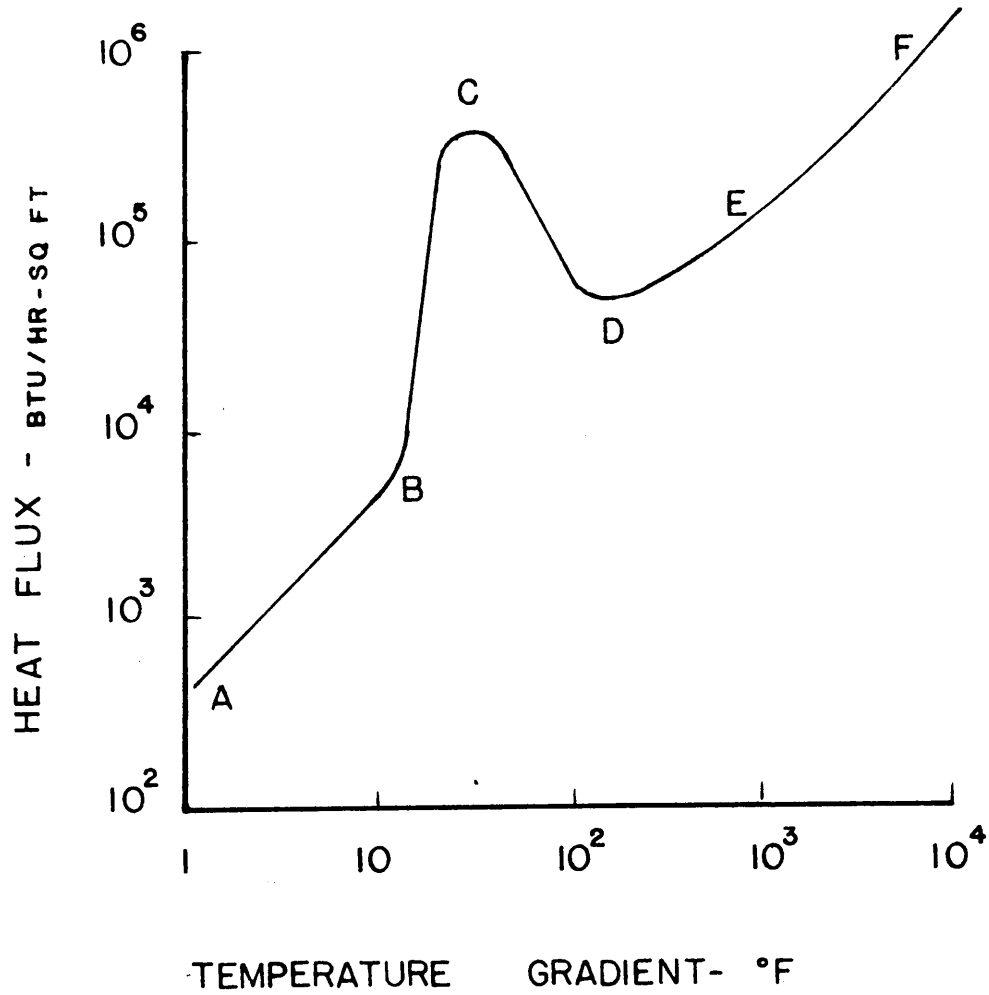


FIGURE 1

TYPICAL HEAT FLUX CURVE FOR WATER

McADAMS, W. H., ET AL, HEAT TRANSFER FROM SINGLE HORIZONTAL WIRES TO BOILING WATER. CHEM. ENG. PROGRESS 44, 639 (1948)

Nucleate Boiling. Curve segment B-C (Figure 1) represents a state of nucleate boiling, i.e., the liquid, in intimate contact with the heating surface, is superheated sufficiently to cause vaporization. The bubbles of vapor thus formed remain in contact with the heating surface and increase in size until they are large enough to overcome the physical binding forces. They then break free and rise through the liquid space and finally merge with the vapor phase. As the name implies the vaporisation occurs at small nuclei or discontinuities on the heating surface.

Transitional Boiling. Point C (Figure 1) represents a state of incipient film boiling. In other words, at point C nucleate boiling is so intense that any further increase in the thermal driving force will result in the formation of a partial vapor blanket on the heat transfer surface. Since vapor has a much lower thermal conductivity than liquid the heat transfer rate is reduced when the vapor film begins to form. Curve segment C-D (Figure 1) represents a so-called state of transitional boiling which is in effect a matter of the degree to which the heat transfer surface is blanketed with vapor.

Film Boiling. Point D (Figure 1), the point of minimum heat transfer, represents a state of total film boiling in which the entire heat transfer surface is covered with a vapor blanket. The sector of the curve between D and E represents heat transfer from the metal surface through the vapor film and finally to the liquid. In this region the radiation is not believed[25] to be appreciable. In the range of film boiling designated by point F it is believed that radiation plays an important part in the rate of heat transmission.

History

The Leidenfrost Phenomenon. The earliest known report of observed film boiling was by Leidenfrost[24] in 1756. His work consisted of a study of the behavior of small drops of liquid deposited on surfaces having sufficient temperatures to induce the state of spheroidal boiling. In such a state the drop or ball of liquid is said to be existing in a non-equilibrium condition and will move or "dance" around on the heating surface for a considerable length of time. However, once the surface cools, the drop of liquid vaporized very quickly. In as much as Leidenfrost was the first to study this phenomenon the state of

so-called spheroidal boiling was called the "Leidenfrost Phenomenon" for a considerable length of time.

Perkins[28] in 1827 attributed the explosion of a boiler to the collapse of the spheroidal state which he stated occurred as the boiler cooled. Boutigny[6] confirmed Perkins' statement in 1844.

Lang[21] of Scotland in 1880 reported a curve which related over-all coefficient of heat transfer with thermal driving force for an evaporator used for distilling sea water. Although his data was presented in one curve irrespective of boiler pressure, a trend could be seen which indicated that a state of maximum heat flux had been attained. No mention was made as to the possible relationship between the peak on the curve and the critical point for transition from nucleate to partial film boiling.

Effects Causing the Spheroidal State. Boutigny[6] and Berger[3] were the first known investigators to attempt to determine the effects causing the formation of the spheroidal state of boiling. Their approach to the problem was to determine the minimum surface temperature necessary to yield spheroidal boiling for various pairs of liquids and metals. While little is known as to the actual technique employed in their investigations, it is known that they used a dish of the metal being tested and placed small drops of the test liquid in

this dish. Despite the fact that the results of their experiments have been subject to considerable question the effect of the various liquid-metal combinations may be worthwhile. The results of the investigations conducted by Boutigny[6] and Berger[3] are presented in Table I.

Langmuir[23] did considerable fundamental work on the subject of evaporation and condensation.

Nucleate Boiling

In as much as the range of nucleate boiling is the most desirable from the industrial standpoint it has been investigated much more extensively than the other ranges of heat transfer to boiling liquids. Whether early investigators failed to realize that a state of film boiling could exist or not is unknown, at any rate, until recently the nucleate range of boiling has certainly received the greatest quantity of attention.

Bubble Formation. Foremost among early investigators were Jakob and his associates[5,15,16,17]. Jakob and Fritz[17] approached the problem from the fundamental viewpoint of bubble formation. They believed that the rate of heat transfer was largely dependent upon the rate and type of bubble formation

TABLE I

Effect of Nature of Hot Surface on Minimum Surface Temperature for Spheroidal State

(All data at Normal Atmosphere)

Investigator	Boutigny	Berger	
Test Liquid	Water	Ether	
B P of Liquid, °C	100	35	
Nature of Surface	---	Smooth	Rough
Heating Surface	Surface Temperature °C	Surface Temperature °C	Surface Temperature °C
Copper	---	86	---
Iron	---	127-8	140
Lead	260	130	134
Mercury	---	170	---
Platinum	171	96	---
Silver	142	78	---
Tin	---	117	---
Zinc	---	108	116
Porcelain	---	210	---
Glass	---	150	---

Berger, - -, Ann. Physik (2), 119, 594-637 (1863). Trans. Am. Inst. Chem. Engr. 33, 449-473 (1937).

Boutigny, P. H., Ann. chim. phys. (3) 9, 350-370 (1843). Trans. Am. Inst. Chem. Engr. 33, 449-473 (1937).

at the heating surface. Their first consideration was the law set forth by Lord Kelvin which dealt with the vapor pressure within the bubble of vapor. Jakob[16] credits Bosnjakovic[5] with the origination of the theory that slight superheating of the liquid in intimate contact with the plate and the bubble of vapor provides the impulse for heat transfer in the case of nucleate boiling. Jakob and Frits[17] carefully determined the degree of superheating which occurred in liquid bodies. Their investigation incorporated the use of a horizontal heating surface and a thermocouple which could be moved in a vertical axis in the liquid space above the heating surface. With such an apparatus they were able to obtain plots of temperature along with the vertical axis in the liquid. These plots indicated that the majority of liquid superheat was present within five millimeters of the horizontal heating surface.

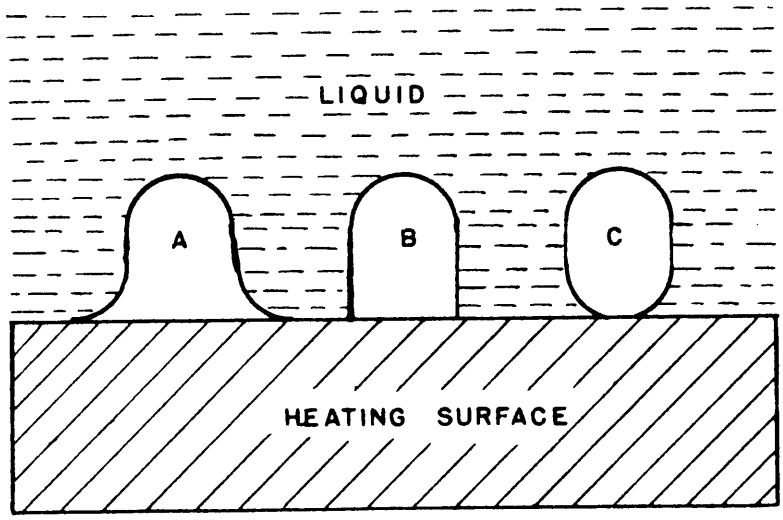
Jakob and Frits[17] also investigated nucleate boiling from the standpoint of a pictorial study of the size and shape of the bubbles formed on various types of surfaces. A camera in conjunction with stroboscopic light afforded the pictures used in the study. The results of the investigation were essentially twofold, namely, (1) a measurement of the volume occupied by the bubbles as formed on the heating surface and later as they rose through the liquid revealed that vaporization of liquid

was occurring as the bubble rose through the superheated liquid. Increases of bubble volume were obtained which ranged from 140 to 4,500 per cent; (2) a qualitative study of the effect of heating surface conditions revealed that the physical shape of the bubble as formed was controlled to a large extent by the wetability of the heating surface. Figure 2 shows diagramatically the results of the investigation.

Boiling Nuclei. Jakob[15] also conducted an investigation which revealed, to some extent, the arrangement of nuclei on a heating surface. He observed the surface of a copper plate used as a heating surface and noted fouled spots which he claimed had served as nuclei for boiling. The spots were random in size and placement on the plate and were reported to compositely cover only about one per cent of the entire heat transfer area. No mention was made of the rate of heat transfer when such spots were noticed and measured.

Bubble Diameter. Jakob[19] gives the following relationship for the diameter of bubbles that will release from a heated horizontal surface into boiling liquid:

$$d = 1.034 \sqrt{\frac{\gamma}{\rho_l}}$$



A. SURFACE COVERED WITH THIN FILM OIL — NOT WETTED

B. SURFACE POLISHED — HALF WETTED

C. SURFACE CLEANED — ENTIRELY WETTED

FIGURE 2

EFFECT OF INTERFACIAL TENSION ON SHAPE OF VAPOR BUBBLES

JAKOB, M. AND W. FRITZ, FORCH. GIEBETE INGENIEURW. 2, 434-447 (1931); MCADAMS, W.H. "HEAT TRANSMISSION". P. 301. MCGRAW-HILL BOOK CO., INC., NEW YORK. 1942. 2 ND EDITION.

where:

d = bubble diameter, ft

γ = surface tension of liquid, lbs per ft

ρ_L = liquid density, lbs per cu ft

Over-all Coefficient of Heat Transfer. Design of industrial equipment involving heat transfer to boiling liquids usually includes a factor known as the over-all coefficient of heat transfer. Badger and McCabe[2] state that the rate of heat transfer is correlated with the thermal driving force in the following expression:

$$\frac{Q}{A} = U \Delta t$$

where:

U = over-all coefficient of heat transfer, Btu per hr-sq ft-°F

Q = heat transfer rate, Btu per hr

A = heat transfer area, sq ft

Δt = over-all temperature gradient, °F

Film Coefficient. The same general principle as the one above is known to apply when the heat transfer being considered is from a heat transfer surface (usually metal) through a so-

called liquid film to the main body of liquid. In such a case the Δt is the temperature gradient existing between the heating surface and the main body of liquid. Mathematically the expression incorporating the film coefficient is as follows:

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

where:

h = film coefficient, Btu per hr-sq ft-°F

Q = heat transfer rate, Btu per hr

A = heat transfer area, sq ft

Δt = temperature gradient, °F

Correlations Involving the Film Coefficient. The film coefficient (h) is merely a proportionality constant and in the case of heat transfer to boiling liquids is subject to considerable variation over the range of heat fluxes commonly encountered. A considerable number of attempts have been made to obtain correlations which could be used to predict the value of the film coefficient for any given liquid-metal pair under specified conditions. The following relationship was preposed[9] and will to some degree afford a prediction:

$$h = a(\Delta t)^n$$

where:

h = film coefficient of heat transfer, Btu per hr-sq ft-°F

a and n = constants

Δt = temperature gradient, °F

While "a" in the above equation applies to a specific liquid under specific conditions there seems to be general agreement as to the value of the exponent "n". Several investigators[9,11,20,27] reported the value of "n" to range from 2.4 to 2.4. A more recent report by Jakob and Linke[18] who have conducted extensive investigations in the range of nucleate boiling claim 4.0 as the proper value for the exponent.

Cryder and Finalborgo[8] proposed the following equation as a possible means of predicting the film coefficients:

$$h = C(\Delta t)^n (B)^{t_L}$$

OR:

$$\log h = a + n(\log \Delta t) + b(t_L)$$

where:

h = film coefficient, Btu per hr-sq ft-°F

t_L = boiling point of liquid, °F

t = temperature gradient across film, °F

C, B, a, n, and b = constants

They point out that the value of the constant "a" is a function of the type of equipment used and the constant "b" is a function of the liquid in question.

Jakob and Linke[18] correlated data on water, solutions of setting agents, and carbon tetrachloride very well with the following equation:

$$\frac{h}{k}\sqrt{\frac{\gamma}{\rho}} = 30\left(\frac{Q\,V_g}{A\,\lambda\,W}\right)^{0.8}$$

where:

h = film coefficient, Btu per hr-sq ft-°F

k = thermal conductivity, Btu per hr-ft-°F

γ = surface tension, lbs per ft

ρ = density of liquid, lbs per cu ft

Q = rate of heat transfer, Btu per hr

V_g = specific volume of saturated vapor, cu ft per lb

A = heat transfer area, sq ft

λ = latent heat of vaporization, Btu per lb

W = constant (919), ft per hr

Film Boiling

The investigation of the phenomenon of film boiling has included a study of many variables. Most prominent among these variables are:

1. Effect of various metals on the film boiling characteristics of a given liquid
2. Effect of various liquids on the film boiling characteristics of a given metal
3. Effect of the geometrical shape and relative size of the heating surface
4. Effect of the wetability of the heating surface
5. Effect of contamination of the heating surface
6. Effect of pressure
7. Heat transfer characteristics of ethanol and benzene

Effect of Various Metals of the Film Boiling Characteristics of a Given Liquid. Sauer, et al[30] report the results of a study of heat transmission in the range of film boiling using a horizontal steam heated tube. Their results indicated that of the four metals tested, (iron, chrome plated copper, copper, and

aluminum) using water as a test liquid, iron gave the highest heat flux with chrome plated copper, copper, and aluminum following in the order listed. They further state that the iron is believed to have given higher heat transfer rate despite its lower thermal conductivity due to the fact that the surface of iron has more discontinuities and thus favored bubble formation more than did the other metals.

Moscicki and Broder[26] report that the critical temperature gradient for water boiling on various metal surfaces follows the hydrogen overvoltage series, namely, platinum black, platinum, iron, silver, nickel, copper, and lead. They advanced no theory to substantiate the experimentally determined results.

Effect of Various Liquids on the Film Boiling Characteristics of a Given Metal. Sauer, et al[30] did considerable work on the heat transmission characteristics of various liquids boiling on the same metallic heating surface. Their data afforded no apparent mathematical correlation but merely supplied the literature with information regarding these liquids. They did obtain one conclusion which applied to liquids, namely, that the highest maximum heat flux was obtained for water, intermediate values for ethanol and methanol, and lowest values for non-polar liquids such as ethyl acetate, benzene and carbon tetrachloride.

No information could be found which correlated the heat transfer characteristics of liquids in homologous series, likely enough these characteristics are inherent properties of the liquid.

Effect of Geometrical Shape and Relative Size of Heating Surface. Jakob and Linke[18] report that the results obtained using a horizontal heating surface were nearly as good as those obtained with a vertical heating surface. No further explanation of this statement could be found.

Drew and Mueller[10] report that unsatisfactory results were obtained by using an inclined plate as a heating surface. They concluded that in the range of film boiling the vapor tended to escape from the boiling surface by rising along the plate. This explanation appears to contradict the results obtained by Jakob and Linke[18] and to defy the theory of film boiling.

Abbott and Comely[1] report that the maximum heat flux and critical temperature gradient obtained with a single tube evaporator were essentially the same as those obtained with a sixty tube evaporator.

McAdams, et al[25] report that for an investigation using platinum wire as a heater the characteristics of heat transmission did not change appreciably with various sizes of wire.

Effect of Wetability of the Heating Surface on Heat Transfer Characteristics. Rhodes and Bridges[29] state that the presence of certain substances in relatively small amounts may have a great effect in altering the manner of boiling and the unit rate of heat transfer. In their investigation they showed that quantities of oleic acid and waxes could cause considerable reduction in the maximum rate of heat transfer and the critical temperature gradient. They also caused water in a state of film boiling to revert to a state of nucleate boiling by the addition of sodium carbonate. The accompanying theory was that the heating surface had been rendered wetable by the salt addition.

Jacoby and Bichmann[14] conducted a photographic study of film boiling on tubes which were smooth, coated with calcium carbonate, and covered with a thin film of oil. Their conclusions stated that carbonate afforded more nuclei for boiling than did the smooth, clean tube. They also concluded that the presence of oleic acid in the test water to the extent of five parts per million could cause considerable lowering of the critical temperature gradient.

Effect of Contamination of the Heating Surface. Cryder and Gilliland[9] state that it was necessary to clean their heating surface after each test to obtain reproducible results because tarnishing was obtained when using methanol, n-butanol, carbon

tetrachloride, and solutions of some salts. No reference was made as to the effect of fouling on the characteristics of heat transfer.

Results obtained by McAdams, et al[25] indicate that fouling decreases the rate of heat transfer for any given temperature gradient and will increase the critical temperature gradient.

Jakob and Linke[18] showed that adsorbed gases on the heat transfer surface increased the rate of heat transfer for any given temperature gradient. By continued boiling the adsorbed gas was removed and the higher rate of heat transfer dropped to constant values.

Insinger and Bliss[13] reported that for boiling water from a vertical surface for 700 hours the rate of heat transfer for the first 46 hours was higher than for the last 654 hours for any given temperature gradient.

Bonilla and Perry[4] found that by using a chromium plated surface and cleaning each three hours with a non-scratching soap powder would give reproducible results.

Effect of Pressure. McAdams, et al[25] give data relating pressure with maximum heat flux which increases almost linearly from 400,000 Btu per hr-sq ft at atmospheric pressure to 2,200,000 Btu per hr-sq ft at approximately 1,250 lbs per sq in. abs.

Cichelli and Bonilla[7] advanced a correlation which differs somewhat from that obtained by McAdams, et al[25] in that the

pressure had less effect on maximum heat flux and was not a linear relationship.

Cichelli and Bonilla[7] also determined the effect of pressure and the critical temperature gradient and found that for ethanol the critical temperature gradient decreased from 62° F at 14.7 lbs per sq in. abs to 10° F at a pressure of 765 lbs per sq in. abs.

Heat Transfer Characteristics of Ethanol and Benzene. Bonilla and Perry[4] used an evaporator having a horizontal chromium plated surface for the determination of the heat transfer characteristics of various liquids and mixtures of liquids. They report the maximum heat flux for ethanol boiling at atmospheric pressure to be 170,000, 170,000, and 160,000 Btu per hr-sq ft for three of the tests conducted. The corresponding critical temperature gradients were 73, 63, and 63° F, respectively.

Cichelli and Bonilla[7] using a similar type evaporator reported the maximum heat flux for ethanol to be 180,000 Btu per hr-sq ft and a critical temperature gradient for the same test of 65° F. They also reported the maximum heat flux for benzene to be 125,000 Btu per hr-sq ft with a critical temperature gradient of 81° F.

Heat Transfer to Binary Mixtures

Bonilla and Perry[4] and Cichelli and Bonilla[7] have conducted the greater part of the investigations concerning heat transmission to binary mixtures of miscible liquids. In as much as this investigation is devoted to this topic their work has been reviewed in much more detail than the work of other investigators.

Apparatus. The apparatus employed by Bonilla and Perry[4] in their investigation of binary mixtures of liquids consisted essentially of a horizontal heating surface and a vertical tube above the heating surface which acted both as a liquid space and as a condenser.

Two heaters were constructed with effective diameters of 3.58 and 2.62 inches each to study the effect of size. The heaters were similar in principle and construction. Each heater consisted of a copper plate 3/4 inch thick with copper fins welded on the bottom. Between these copper fins were placed electrical heating units composed of nichrome ribbon wrapped on mica centers and insulated with asbestos.

The heating surface was plated with chromium to the extent of 0.02 inches which was said to prevent pinholes and to give reproducible results. The main condenser was the same diameter

as the liquid space to eliminate any possibility of different pressures for different boiling rates.

The plate thermocouples were of the one lead variety, that is, the copper was used as one lead for the thermocouple junction. The thermocouples were made by inserting constantan wire, insulated with pyrex capillary tubing, into holes drilled radially in the copper disc. These wires were then soldered at the point of contact to the copper by pouring molten tin into the holes forming what was effectively a constantan-copper thermocouple. A copper lead from any point on the copper plate made the circuit complete. The temperature drop from the thermocouple tip or joint to the surface of the heater was calculated by the conduction equation and subtracted from the overall temperature gradient.

Operational Techniques. Electrical burnouts were elminiated by opening the heater input circuit once film boiling was attained which was indicated by a sudden rise in heater plate temperature. The high heat capacity of the copper plate also tended to eliminate any local overheating. Cleaning of the chromium plate each three hours with a non-scratching soap powder kept the surface in a condition which gave reproducible results.

The apparatus used by Cichelli and Bonilla[7] closely resembled the apparatus employed by Bonilla and Perry[4].

Heat Transfer Characteristics of Binary Mixtures. Bonilla and Perry[4] did not include in their conclusions the quantitative effect of concentration on the maximum heat flux and critical temperature gradient but stated that the characteristics of binary mixtures fell between the values obtained for the pure compounds. An examination of the graphically presented results indicated that such a statement held true from the critical temperature gradient but not for maximum heat flux. The values (obtained with a heater diameter of 2.62 inches) read from the presented graphs showed maximum heat flux for water to be 380,000 Btu per hr-sq ft while a 3.0 mole per cent solution of ethanol in water gave a maximum heat flux of 410,000 Btu per hr-sq ft. The values obtained using the heater with a diameter of 3.58 inches were in close agreement with those obtained for the smaller unit. In both cases the maximum heat flux for ethanol was around 200,000 Btu per hr-sq ft.

Correlation Using the Reduced Pressure. Cichelli and Bonilla[7] obtained good correlation of the critical temperature gradient with the reduced pressure (P_r) of liquids and mixtures of liquids tested. Liquids investigated were ethanol, benzene, propane, n-pentane, n-heptane, and mixtures of n-pentane and propane. No specific method was outlined for the calculation of reduced pressure for mixtures of liquids. The results of the correlation are presented in Figure 3.

Cichelli and Bonilla[7] also advanced a correlation relating maximum heat flux ((Q/A)max) with reduced pressure (P_r). In tests in which fouling of the heat transfer surface was known to exist the data was rectified by dividing the unit rate of heat transfer by an empirically determined factor of 1.15. The results of this correlation are presented in Figures 4 and 5.

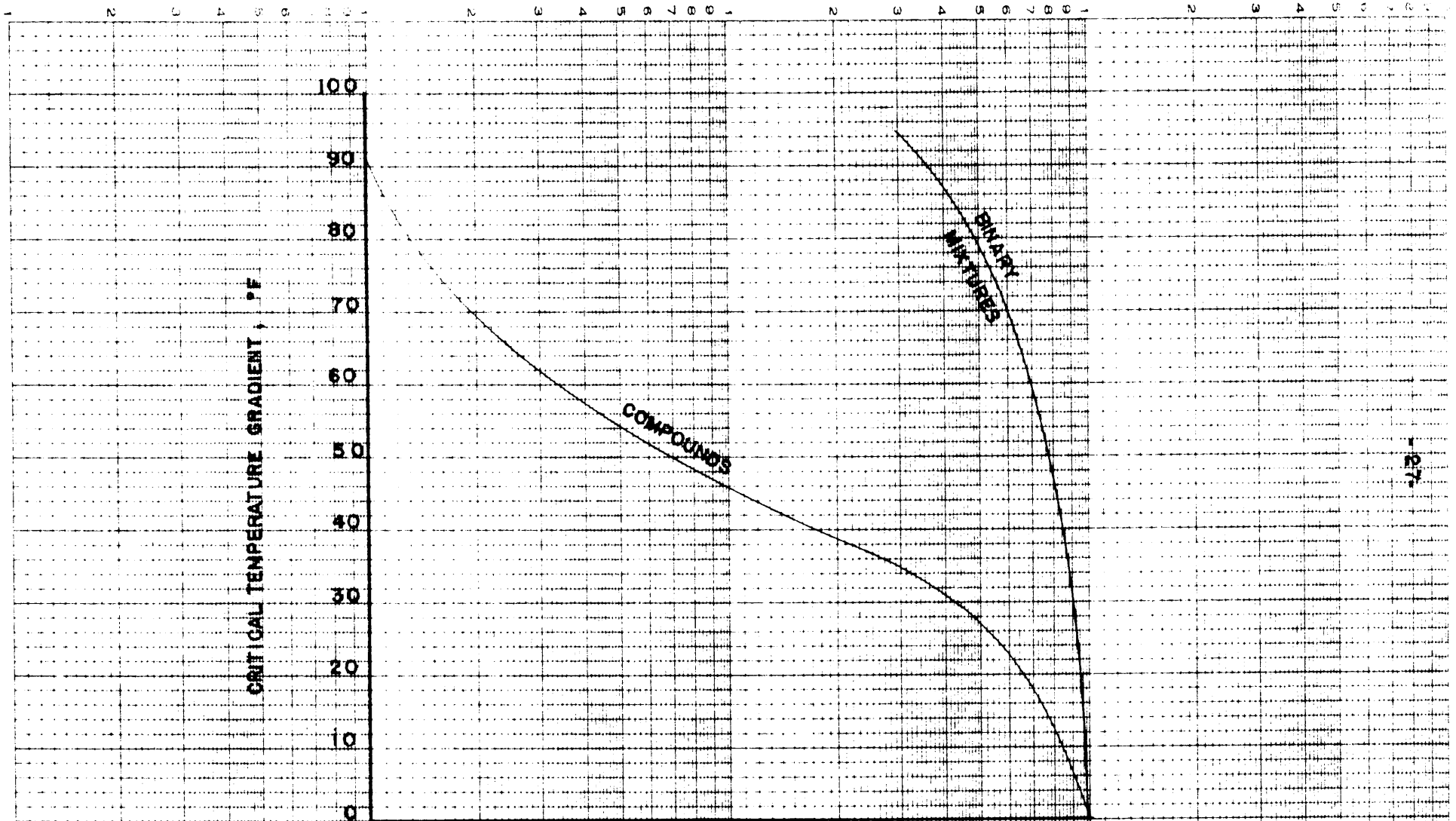


FIGURE 3

A CORRELATION OF CRITICAL TEMPERATURE GRADIENT WITH REDUCED PRESSURE FOR LIQUIDS

CICHELLI, M.T., AND C.F. BONILLA, HEAT TRANSFER TO LIQUIDS BOILING UNDER PRESSURE, AM. INST. CHEM. ENGRS. 41, 755 (1945)

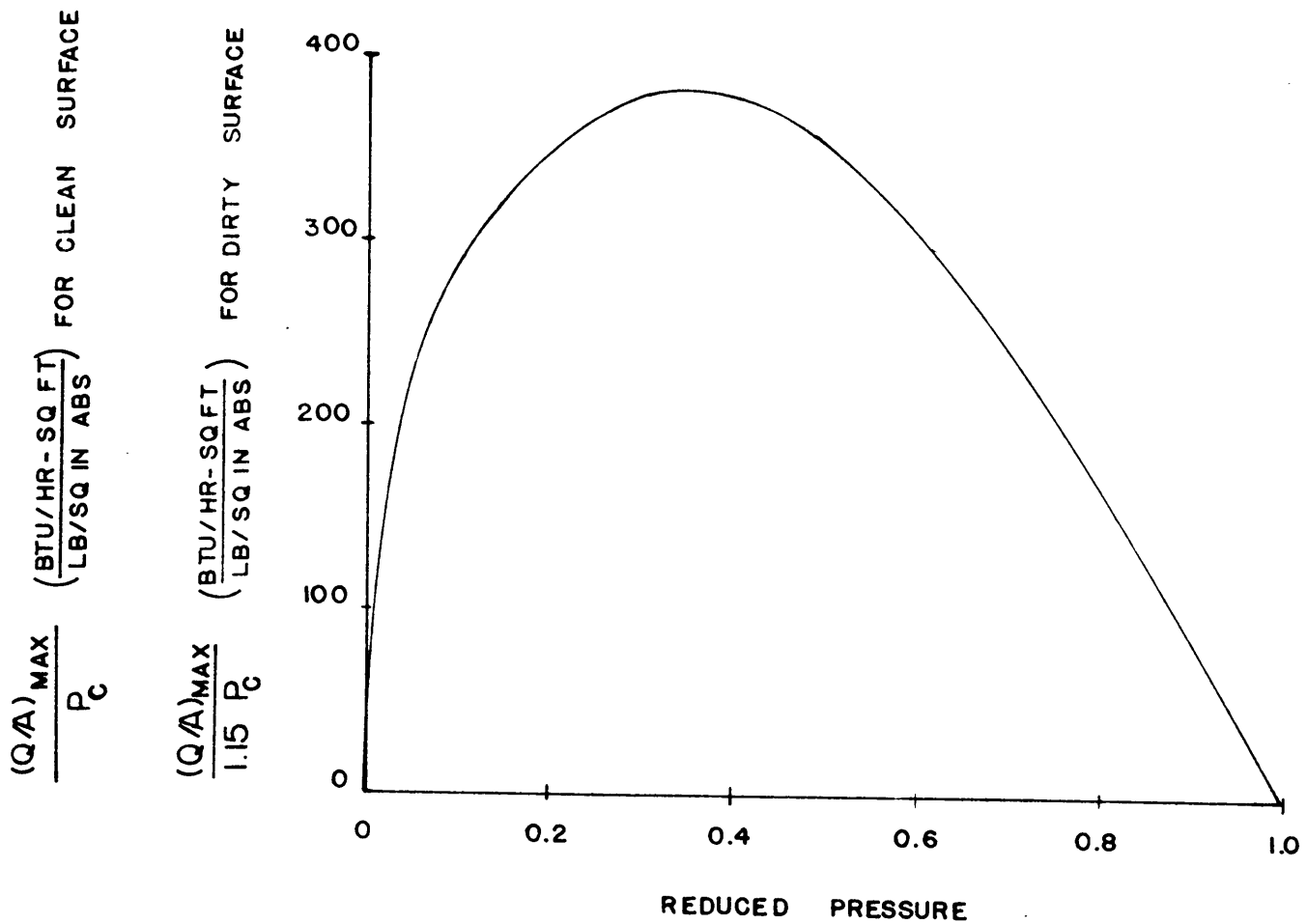


FIGURE 4

FILM BOILING CORRELATION

CICHELLI, M.T., AND C.F. BONILLA, HEAT TRANSFER TO LIQUIDS BOILING UNDER PRESSURE. TRANS. AM. INST. CHEM. ENGRS. 41, 755 (1945)

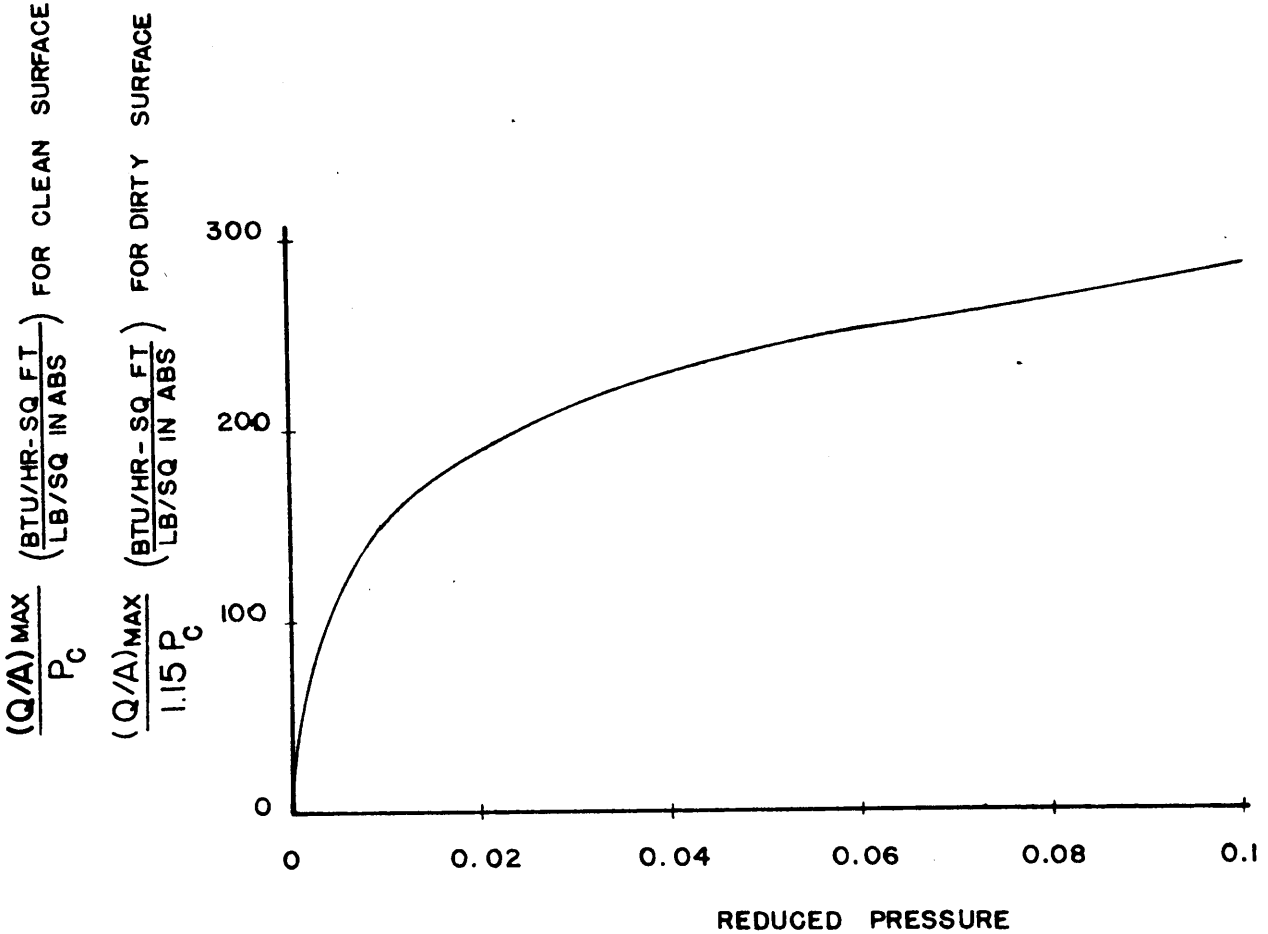


FIGURE 5

FILM BOILING CORRELATION

CICHELLI, M.T. AND C.F. BONILLA, HEAT TRANSFER TO LIQUIDS BOILING UNDER PRESSURE, TRANS. AM. INST. CHEM. ENGRS. 41, 755 (1945)

III. EXPERIMENTAL

Purpose

The purpose of this investigation was to relate maximum heat flux and critical driving force with the concentration of ethanol in benzene.

Plan

This investigation was conducted under the following general plan:

Test samples of ethanol-benzene mixture were prepared for each 10 volume per cent increment of ethanol from 0 to 100 per cent. The silver heat transfer surface was then cleaned by prescribed methods and a 250-milliliter sample of a specific concentration charged to the evaporator body. The heat input to the heater was set by means of the electrical control circuit and the liquid allowed to boil until steady state conditions were attained. Once steady state conditions were attained the emf from the evaporator plate and liquid-vapor space thermocouples were read for the ultimate determination of the temperature gradient. The wattage input to the heater was noted and recorded for the ultimate determination of the heat flux.

Conditions were than changed to a higher rate of heat flux, steady state conditions attained, and the data noted and recorded. These steps were repeated until a state of film boiling was reached. This over-all procedure was followed for all test liquids.

The investigation was conducted in the following series of steps:

Literature Review. The purpose of the literature search was essentially threefold, namely, (1) to ascertain the extent of previous work regarding the immediate problem, (2) to obtain information regarding the concepts involved and problem encountered in the general field of heat transfer to boiling liquids, and (3) to obtain data and information which would assist in the design and construction of the apparatus used in the study.

Design and Construction of Evaporator. A circular, silver-plated copper plate was used as a heat transfer surface. Electrically heated copper fins, welded on the bottom of the plate, served to supply the heat for boiling. The evaporator body was a flanged pyrex pipe, in a vertical position, sealed to the heat transfer surface by means of a compression flange in conjunction with a suitable gasket.

Design and Construction of Condenser. Since the evaporator was to operate at steady state conditions, a system of total reflux of condensed liquid was employed. An extension of the pyrex pipe

which served as the evaporator body served also as the inner pipe of the water condenser. Copper tubing was used as the outer shell of the condenser. The condenser utilized the principle of downward flow and was equipped with an air vent. An orifice with mercury manometer was installed in the outlet cooling water line to assist in regulating the cooling water rate. The test liquid was charged through the top of the inner tube of the condenser.

Design and Construction of the Electrical Heating Circuit. Three parallel heating circuits were used; one was a variable wattage circuit, the other two had fixed capacities. With such an arrangement any desired wattage from 0 to 2,700 watts could be applied to the heating fins. The wattage input was determined by an ammeter-voltmeter combination.

Design, Construction and Calibration of Temperature Measuring and Regulating Equipment. Three thermocouples in the liquid-vapor space and three thermocouples imbedded in the copper evaporator plate were used in conjunction with a potentiometer to determine the temperatures for each steady state condition. An ice-filled cold junction bottle referred the circuit to 32° F for all tests. A thermocouple actuated a Wheelco "Capacitrol" which measured and controlled the ambient heater temperature. The thermocouples were calibrated before use.

Preparation of Test Liquids. The ethanol used in the investigation was dried before use by refluxing with calcium oxide and purified by subsequent distillation. The water used was distilled. The technical grade benzene was not pretreated but used as received.

Preliminary Tests of Apparatus. The liquids, ethanol and benzene, were used for preliminary qualitative and quantitative evaluations of the apparatus.

The heat losses of the apparatus were evaluated for the various pure test liquids. The selection of a suitable gasket material for the evaporator body gasket was included among the preliminary tests.

Determination of the Characteristics of Heat Transfer for Various Concentrations of Ethanol in Benzene. The constructed apparatus was employed for this phase of the investigation. The cold liquid level was maintained constant for all tests.

Evaluation of Results. The experimentally determined data were evaluated and analyzed. Any pecularities in the calculated results were reported and discussed.

Materials

The following materials were used in this investigation:

Benzene. Technical grade, 99 to 100 per cent pure, lot No. 8240, code No. 1444. Manufactured by General Chemical Division, Allied Chemical and Dye Corporation, New York, N. Y. Used as a test liquid.

Ethanol. 95 volume per cent. Obtained from Phipps and Bird, Inc., Richmond, Va. Used as a test liquid.

Apparatus

The following apparatus was used in this investigation:

Potentiometer. Type S, range: 0 to 0.017 and 0 to 1.70 v. Manufactured by Fisher Scientific Co., Pittsburgh, Pa. Used to measure electromotive force from thermocouples.

Galvanometer. Center zero type, catalog No. 570-201. Manufactured by G. M. Laboratories, Inc., Chicago, Ill. Used in conjunction with potentiometer.

Cell, Standard. 1.019 v, No. 392006, internal resistance not over 500 ohms. Manufactured by Epperly Laboratory Inc., Newport, R. I. Used in conjunction with potentiometer.

Battery, Dry Cell. 3 v, No. VS-100, type A. Manufactured by Radio Corporation of America, New York, N. Y. Used in conjunction with potentiometer.

Glassware. Miscellaneous sizes of beakers, graduate cylinders, pipets, flasks and burets. Obtained from Chemistry Department Stockroom, Virginia Polytechnic Institute, Blacksburg, Virginia. Used for preparation of ethanol-benzene mixtures and storage of test liquids.

Horizontal Plate Evaporator and Accessory Equipment. All materials required for the construction of the evaporator and accessory equipment are included in Table II.

TABLE II

Bill of Materials for Horizontal Plate Evaporator and Accessory Equipment

Number Required	Unit	Name	Description	Supplier
HEATER				
1	ea	Copper Plate	Commercial grade, 6 x 6 x 3/4 inch.	Seaboard Brass and Copper Co., Baltimore, Md.
4	ft	Copper Strip	Commercial grade, 3 x 3/16 inch.	Seaboard Brass and Copper Co., Baltimore, Md.
200	ft	Resistance Wire	Chromel A, B & S gage 22, asbestos covered.	Fisher Scientific Co., Pittsburgh, Pa.
3	lbs	Refractory	Alumina, T-16, minus 325 mesh.	Alumina Ore Co., E. St. Louis, Ill.
1	ea	Silver Surface	Electroplated 0.003 inch on heat transfer surface.	Magic City Plating, Roanoke, Va.
THERMOCOUPLES				
8	ft	Thermocouple	Constantan, B & S gage 20, uninsulated, date: 4-23-47, spool No. 885-66-55.	Leeds and Northrup Co., Philadelphia, Pa.
8	ft	Thermocouple Wire	Iron, B & S gage 20, uninsulated, date: 4-23-47, spool No. 899-37-14.	Leeds and Northrup Co., Philadelphia, Pa.
50	ft	Thermocouple Wire	Constantan, B & S gage 24, insulation EN, order No. 128001.	Leeds and Northrup Co., Philadelphia, Pa.
40	ft	Thermocouple Wire	Copper, B & S gage 24, insulation EN, order No. 178001.	Leeds and Northrup Co., Philadelphia, Pa.
2	ea	Switches	Radio type, 4 positions	Chemical Engineering Department Stock Room, Virginia Polytechnic Institute, Blacksburg, Va.
PIPE AND TUBING				
5	ft	Pyrex Pipe	Nominal diameter 2 in., flanged.	Corning Glass Works, Corning, N. Y.
4	ft	Copper Tubing	Outside diameter 3-1/8 in., wall thickness 1/8 in.	Noland Co., Inc., Roanoke, Va.
20	ft	Copper Tubing	Outside diameter 1/2 in., wall thickness 1/16 in.	Noland Co., Inc., Roanoke, Va.
12	ft	Copper Tubing	Nominal diameter 3/8 in., galvanized.	Noland Co., Inc., Roanoke, Va.
2	ea	Street Ells	Nominal diameter 3/8 in., galvanized.	Noland Co., Inc., Roanoke, Va.
2	ea	Needle Valves	Nominal diameter 3/8 in., bronze.	Noland Co., Inc., Roanoke, Va.
10	ft	Glass Tubing	Nominal diameter 1/4 in., soft glass.	Eimer and Amend, New York, N. Y.
20	ft	Glass Tubing	Nominal diameter 1/8 in., pyrex glass.	Eimer and Amend, New York, N. Y.
EQUIPMENT SUPPORTS				
20	ft	Angle Iron	1-1/2 x 1-1/2 in.	Noland Co., Inc., Roanoke, Va.
2	ft	Angle Iron	1 x 1 in.	Noland Co., Inc., Roanoke, Va.
2	ft	Strip Stock	2 x 1/4 in., iron.	Noland Co., Inc., Roanoke, Va.
2	ea	Strip Stock	2-1/2 x 2-1/2 x 1/8 in., iron.	Noland Co., Inc., Roanoke, Va.
12	ft	Lumber	8 x 3/4 in., pine.	Chemical Engineering Department Stock Room, Virginia Polytechnic Institute, Blacksburg, Va.
8	ft	Lumber	6 x 1/2 in., pine.	Chemical Engineering Department Stock Room, Virginia Polytechnic Institute, Blacksburg, Va.
2	ft	Plastic	2 x 1/8 in., Lucite	E. I. duPont de Nemours Co., Arlington, N. J.
ELECTRICAL				
1	ea	Voltage Regulator	"Transtat", No. 32322, rating 9 amp.	American Transformer Co., Newark, N. J.
1	ea	Ammeter	Alternating current, U.S.N. type CAY 22026 A, range 0 to 5 amp.	Westinghouse Electric and Manufacturing Co., East Pittsburgh, Pa.
1	ea	Voltmeter	Alternating current, range 0 to 300 v.	Chemical Engineering Department Stock Room, Virginia Polytechnic Institute, Blacksburg, Va.
12	ft	Copper Wire	Double strand, motor type, double rubber insulated, rating 5 amp.	Chemical Engineering Department Stock Room, Virginia Polytechnic Institute, Blacksburg, Va.
100	ft	Copper Wire	Single strand, B & S gage 14, rubber and cambric insulated.	Chemical Engineering Department Stock Room, Virginia Polytechnic Institute, Blacksburg, Va.
7	ea	Switches	Toggle type, single pole, rating 10 amp at 120 v, 5 amp at 250 v.	Brown Stores Co., Blacksburg, Va.
1	ea	Switch	Toggle type, double pole, rating 250 v.	Brown Stores Co., Blacksburg, Va.
1	ea	Switch Box	Rating 30 amp at 230 v, catalog No. 88351, series No. 1.	Square "D" Electrical Co., Detroit, Mich.
4	ea	Fuses	Type NON, rating 15 amp.	Bussmann Manufacturing Co., St. Louis, Mo.
1	ea	Temperature	"Capacitrol", model 224, range 0 to 1,000° C, rating 110 to 220 v, 35 amp.	Wheelso Instrument Co., Chicago 7, Ill.
MISCELLANEOUS				
15	lb	Insulation	85 per cent magnesia.	Chemical Engineering Department Stock Room, Virginia Polytechnic Institute, Blacksburg, Va.
3	ea	Stoppers	Natural rubber, size 1.	Chemical Engineering Department Stock Room, Virginia Polytechnic Institute, Blacksburg, Va.
2	ea	Stoppers	Natural rubber, size 14.	Chemistry Department Stock Room, Virginia Polytechnic Institute, Blacksburg, Va.
2	ea	Clamps	Laboratory ring stand type.	Chemistry Department Stock Room, Virginia Polytechnic Institute, Blacksburg, Va.
1	ea	Manometer	Mercury filled, range	Chemical Engineering Department Stock Room, Virginia Polytechnic Institute, Blacksburg, Va.
4	ea	Bolts	Diameter 1/4 in., length 2-1/2 in.	Chemical Engineering Department Stock Room, Virginia Polytechnic Institute, Blacksburg, Va.
1	ea	Pulley	Diameter 3 in., aluminum.	Chemical Engineering Department Stock Room, Virginia Polytechnic Institute, Blacksburg, Va.
1	pt	Shellac	Pure white, undiluted.	Springdale Shellac Co., Inc., Springdale, Conn.
1	tube	Rubber Cement	Automotive weather stripping type.	E. I. duPont de Nemours Co., Wilmington, Del.

Methods of Procedure

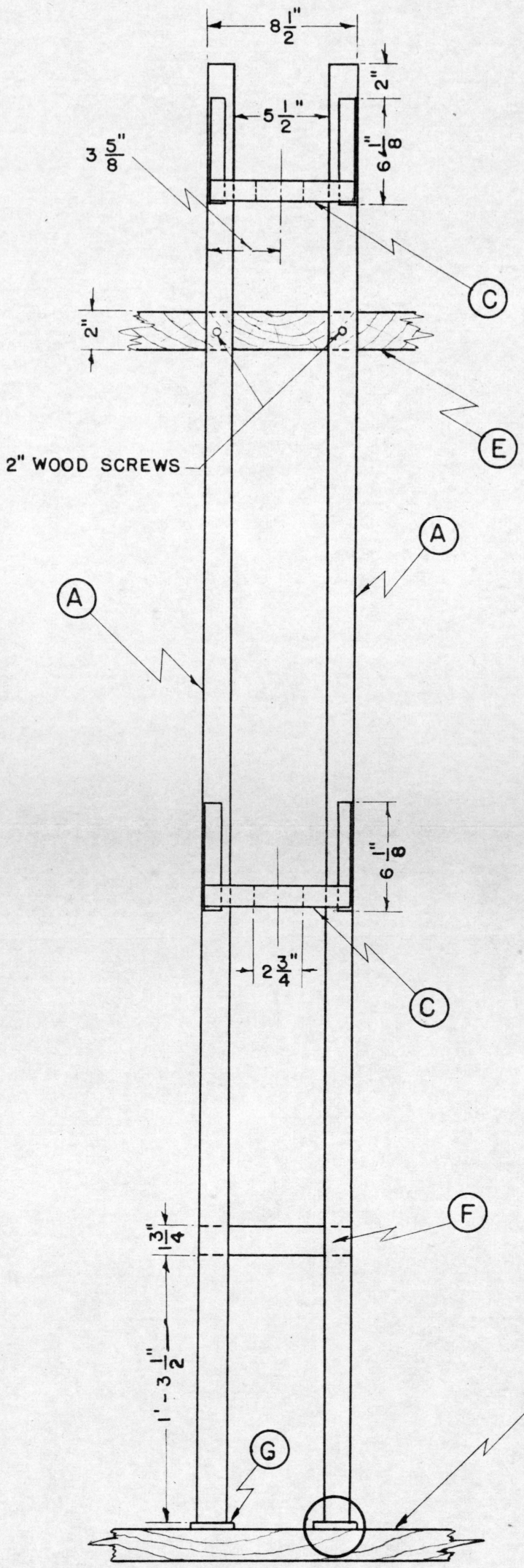
The methods of procedure used in this investigation were as follows:

Design and Construction of Evaporator and Condenser Supports. The horizontal evaporator and vertical water cooled condenser were independently supported by two horizontal shelves, reinforced with angle iron, and two vertical angle iron columns.

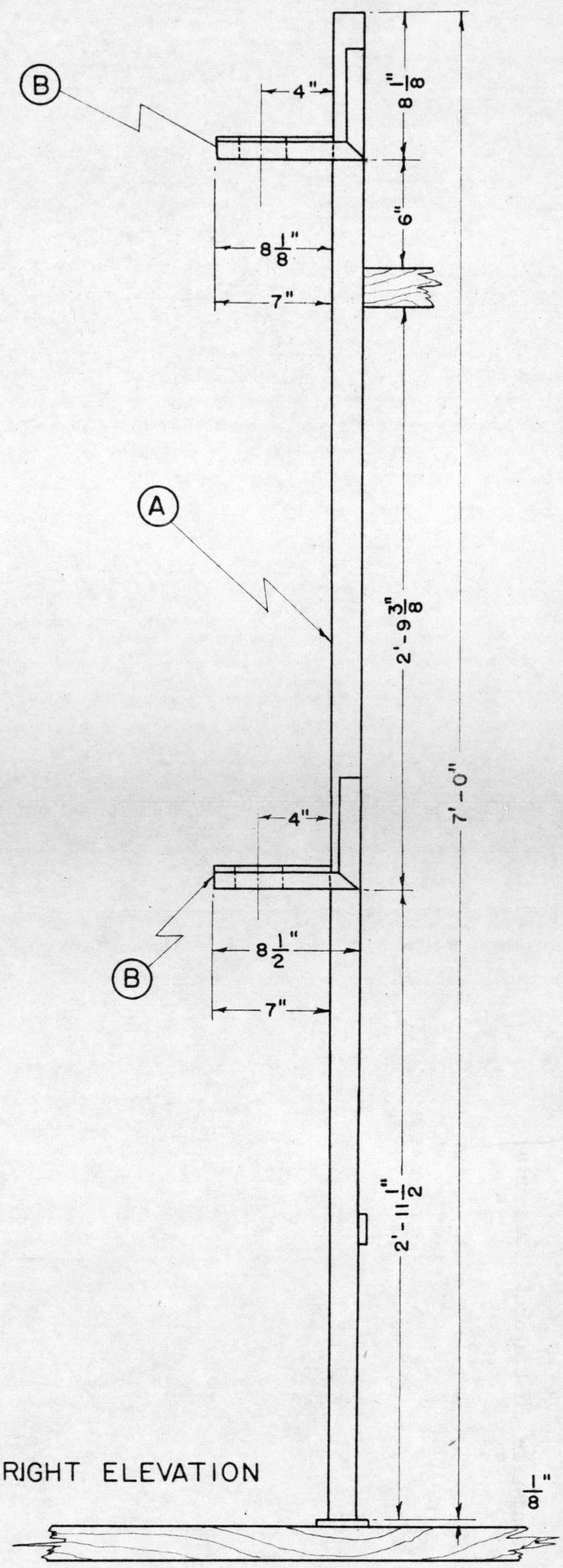
The main supports for the assembly were of 1-1/2 x 1-1/2-inch angle iron placed in a vertical position 5-1/2 inches apart (A Figure 6). These angle iron columns were fastened to the table top, which served as the base for the entire equipment, by means of two column supports and two 2-inch wood screws. The column supports were 2-1/2 x 2-1/2 x 1/8-inch pieces of iron strip stock welded to the bottom of the angle iron columns.

The tops of the 1-1/2 x 1-1/2-inch angle iron columns were held in place by means of two 2-inch wood screws which secured the angle iron to an overhead 2-inch pine shelf.

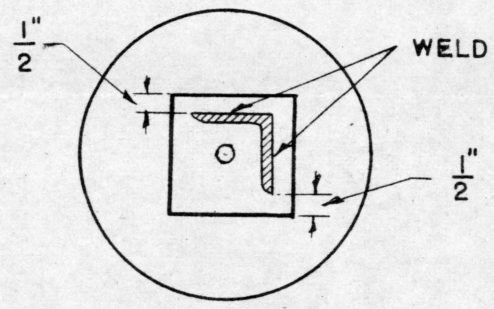
In as much as the water condenser (B Figure 7) contributed the greater percentage of the total weight of the apparatus, it was decided to support the pyrex pipe and copper condenser separately. Such a means of support was designed to eliminate stress both in the pyrex pipe and in the seal between the copper condenser shell and the pyrex pipe.



FRONT ELEVATION



RIGHT ELEVATION



COLUMN FOOTING
PLAN VIEW

LEGEND

(A) MAIN SUPPORTS - $1\frac{1}{2}$" X $1\frac{1}{2}$" ANGLE IRON

(B) CONDENSER SUPPORTS - 1" X 1" ANGLE IRON

(C) CONDENSER SUPPORTS - $1\frac{1}{4}$" PINE

(D) TABLE TOP

(E) HORIZONTAL SHELF

(F) THERMOCOUPLE TERMINAL BOARD- LUCITE

(G) COLUMN FOOTING - $2\frac{1}{2}$" X $2\frac{1}{2}$" IRON STRIP

NOTE: $\frac{3}{16}$" CARRIAGE BOLTS USED ON ALL JOINTS UNLESS OTHERWISE SPECIFIED.

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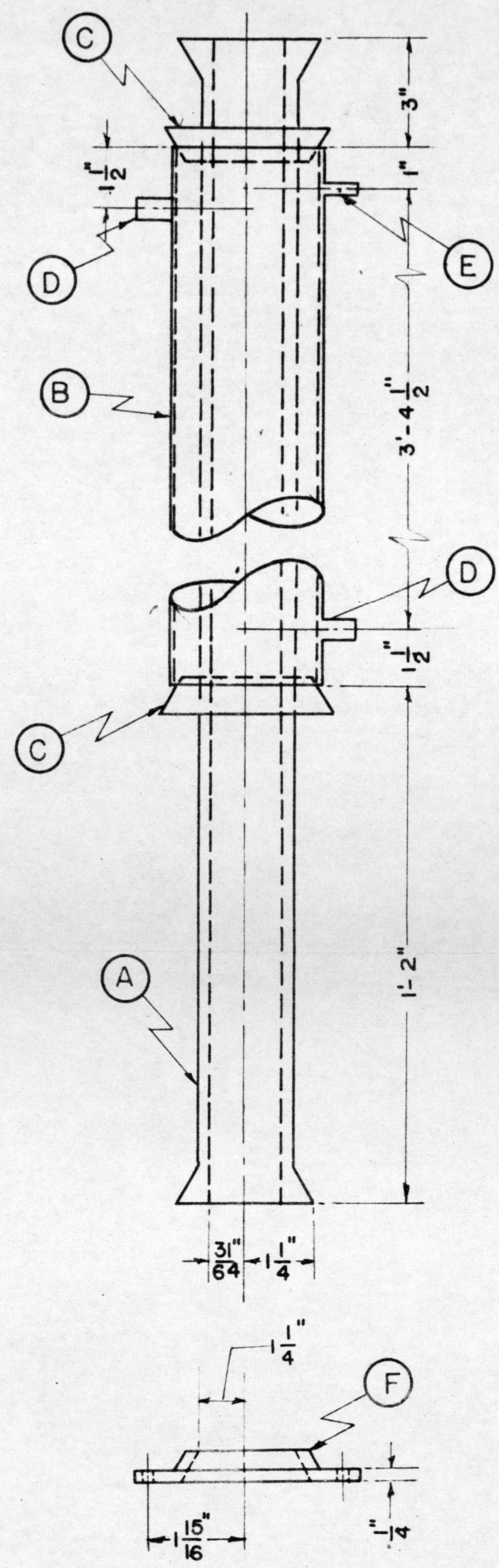
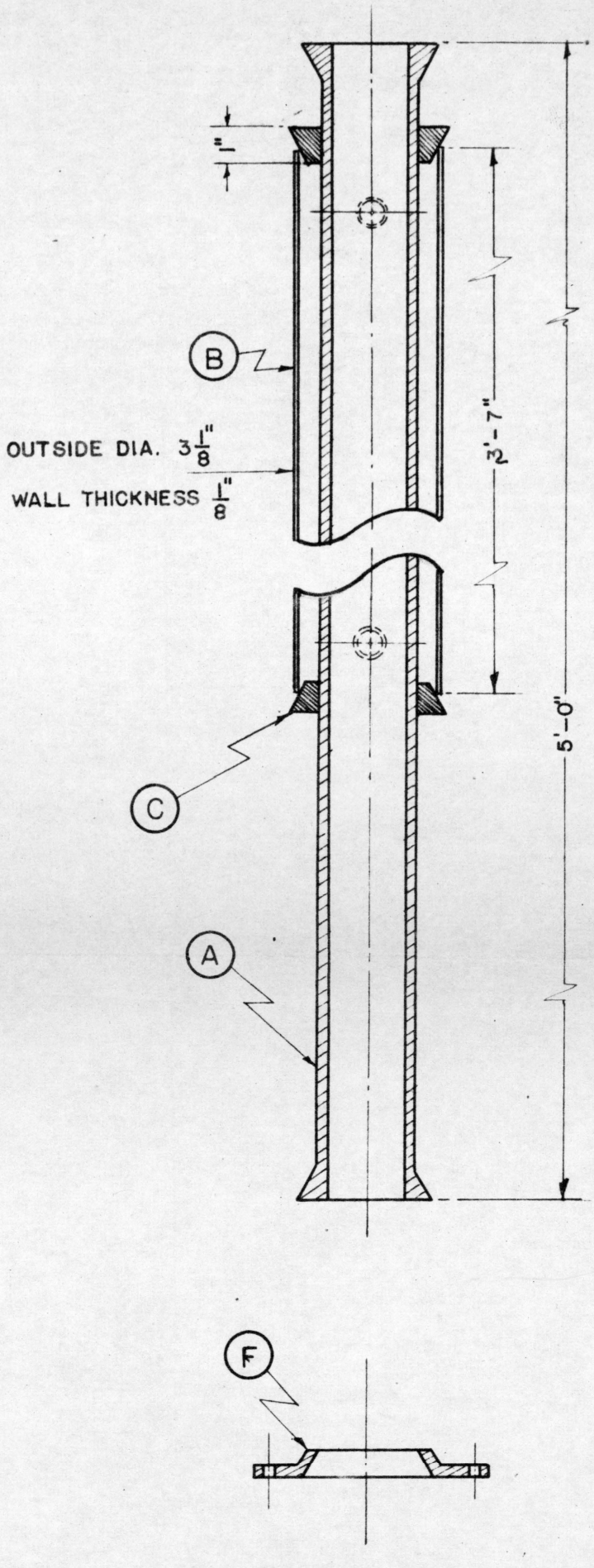
VIRGINIA POLYTECHNIC INSTITUTE
BLACKSBURG, VIRGINIA

CONDENSER AND EVAPORATOR SUPPORTS

DRAWN BY: MBH 7/9/49	SCALE: $1\frac{1}{2}$" = 1'-0"
TRACED BY: R.C.S 7/12/49	JULY 9, 1949
APPROVED BY: [illegible] 7/12/49	FIGURE NO. 6



FRONT ELEVATION



RIGHT ELEVATION

(A) EVAPORATOR SECTION. PYREX PIPE. NOMINAL DIA. 2"

(B) CONDENSER SECTION. COPPER TUBING.

(C) RUBBER STOPPERS. NO. 14 NATURAL.

(D) 1" SHORT NIPPLES. BLACK IRON. NOMINAL DIA. 3/8"

(E) 1" SHORT NIPPLES. BLACK IRON. NOMINAL DIA. 1/4"

(F) COMPRESSION TYPE FLANGE. STEEL. FOR 2" PYREX PIPE.

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BLACKSBURG, VIRGINIA

EVAPORATOR BODY
AND
CONDENSER

DRAWN BY: DWB 7/20/49	SCALE · 3" = 1'-0"
TRACED BY: R.C.S. 7/22/49	JULY 20, 1949
APPROVED BY: FWB 7/24/49	FIGURE NO. 7

The condenser was supported by a shelf constructed of 1 x 1-inch angle iron and 1-1/2-inch pine board (B and C Figure 6). Two lengths of 1 x 1-inch angle iron were notched, bent, and welded to form supports for the shelf. These supports were bolted to the main supports with two 1/4-inch carriage bolts. The pine board was furnished with a hole 2-3/4 inches in diameter and placed so that it rested on the 1 x 1-inch angle iron supports. The pine shelf was attached to the angle iron by four 1-inch wood screws. Such a shelf permitted the pyrex pipe to extend through the board unrestricted but supported the full weight of the copper condenser shell. The shelf also tended to compress the rubber stopper, which sealed the copper condenser to the pyrex pipe, and thus formed a more durable seal.

The pyrex pipe was supported by a shelf, identical to the one used for the condenser shell support (C Figure 6), but in conjunction with a compression type flange (F Figure 7). The flange was inverted with the sleeve resting on the edges of the 2-3/4-inch hole in the pine board and held in position with 1-inch wood screws. The pyrex pipe was raised through the flange, a rubber gasket inserted between the pipe and the flange, and the pipe allowed to rest in position.

The distance between the shelves (D Figure 6) was calculated to be such that the pyrex pipe and copper condenser shell rested independently on their respective foundations and thus minimized any stress on the rubber condenser seal.

Design and Construction of Evaporator Body and Condenser. The evaporator body was designed to have an internal diameter of approximately two inches. A flanged pyrex pipe having an internal diameter of 1-31/32 inches was selected to serve the purpose (A Figure 7). In order to eliminate the possibility of varying pressure due to different vapor velocities at various boiling rates the inner condenser tube was designed to have the same internal diameter as the evaporator body. To attain this a five-foot pyrex pipe was selected which permitted the single tube to act as both evaporator body and condenser inner tube.

Approximately one foot of the pipe was allotted to serve as the evaporator body. The effective length of the condenser section was calculated to be 3 feet, 6 inches with a maximum cooling water rate of 4 pounds per minute. The evaporator and condenser were constructed with due consideration to these preliminary design figures.

A piece of copper tubing, having an outside diameter of 3-1/8 inches and a wall thickness of 1/8 inch, was selected to serve as the outer shell of the condenser (B Figure 7). The

condenser therefore had an effective internal diameter of 3 inches. The tubing was cut to a length of 3 feet, 7 inches and the burrs, formed in cutting, were removed with a manual reamer.

The condenser shell was then equipped with provisions for cooling water flow. The copper tube was fitted with a nominal 3/8-inch short nipple 1-1/2 inches from each end of the shell (D Figure 7). These nipples provided the inlet and outlet cooling water ports. Since a downward flow of cooling water was desired for closer regulation of the reflux liquid temperature in the evaporator it was necessary to equip the condenser with an air vent (E Figure 7). This vent was to provide a means of removing entrained air from the condenser and thus eliminate the possibility of an air block. Such an air block would reduce the effective condenser area.

The vent was a nominal 1/4-inch short nipple placed 1 inch from the top of the condenser tube (E Figure 7). This meant that the air vent was 1/2 inch above the inlet cooling water port and allowed a maximum cooling water height of 3 feet, 6 inches.

Two number 14 natural rubber stoppers were then prepared for use as gaskets between the condenser shell and the pyrex pipe serving as the inner condenser tube. Two holes, approximately 2-3/8 inches in diameter, were drilled in the center of the stoppers by means of an extension-type wood bit. The diameter

of the holes in the rubber stoppers closely approximated the outside diameter of the pyrex pipe. The copper tube was slipped over the pyrex pipe and adjusted to its predetermined position. Several coats of rubber cement were then applied to the areas on the pyrex pipe and the copper tube where the rubber gaskets would contact (C Figure 7). The rubber gaskets were then stretched over the flanged ends of the tubing and forced into position between the copper tubing and the pyrex pipe. Several attempts were made before a satisfactory seal was obtained on both ends of the condenser. When the seal was found to be satisfactory the column was placed in its proper position on the equipment supports.

The condenser was then equipped with inlet, drain, and air vent lines. The inlet and outlet cooling water pipes included a needle valve for regulation of the flow. It was necessary to include a needle valve in the outlet line in order to maintain the prescribed liquid height of 3 feet, 6 inches in the condenser. The outlet cooling water line was also equipped with an orifice-manometer combination to facilitate ease of regulating flow rates. This orifice-manometer combination was calibrated over a range of flow rates from 0 to 3-3/4 pounds per minute.

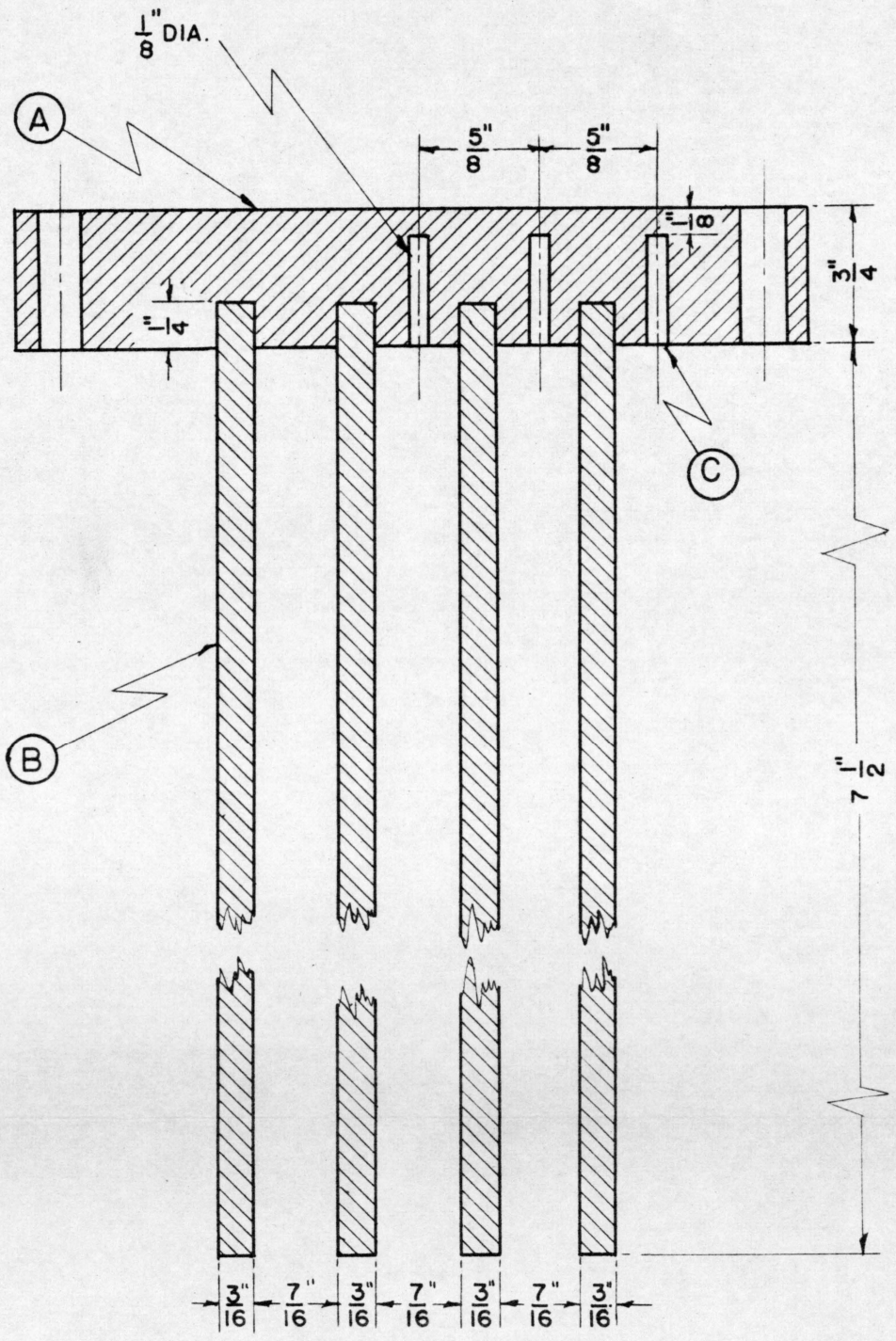
The purpose of the particular design of equipment support employed was to eliminate to as great a degree as possible all stresses in the pyrex pipe which served as an evaporator body and

condenser shell. The two principal aims were (1) to afford a support as free as possible of vibration and movement, and (2) to eliminate any factors which would cause tensile stress in the pyrex pipe.

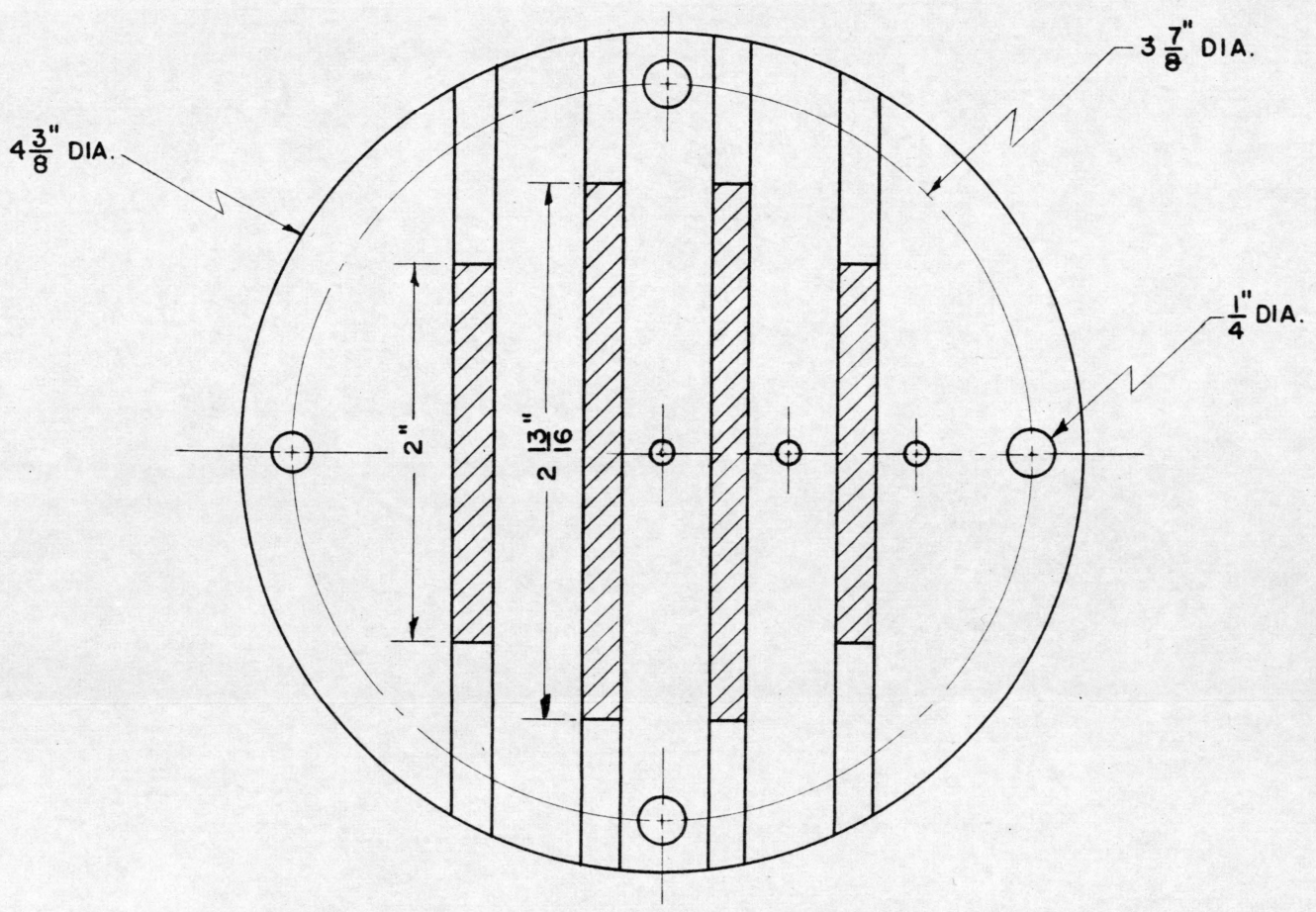
Design and Construction of Evaporator Plate and Heating Fins. The design of the evaporator plate and heating fins was affected not only by the prescribed conditions for heat transfer but also by the geometrical arrangement of the heater electrical windings and the number and placement of the plate thermocouples.

A commercial grade copper block 6 x 6 x 3/4 inch was obtained to form the evaporator plate. This block was reduced in size to 4-1/2 x 4-1/2 x 3/4 inch. The block was then placed in a lathe and machined to form a circular plate 4-3/8 inches in diameter and 3/4 inch thick (A Figure 8). The purpose in using a block of this thickness was twofold, namely, (1) the large block would tend to eliminate temporary fluctuations in plate temperature, and (2) the plate would tend to absorb the surplus heat once a state of film boiling was attained during the operation of the evaporator.

Since a compression type flange (F Figure 7) was to be used to hold the evaporator body to the heating surface, the plate was equipped with four 1/2-inch diameter holes corresponding in position with the holes in the flange.



FRONT ELEVATION



BOTTOM ELEVATION

LEGEND

(A) EVAPORATOR PLATE

(B) HEATING FINS

(C) THERMOCOUPLE WELLS

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VIRGINIA POLYTECHNIC INSTITUTE
BLACKSBURG, VIRGINIA

EVAPORATOR PLATE
AND
HEATING FINS

DRAWN BY: WBH 7/13/49
TRACED BY: R.C.S. 7/15/49
APPROVED BY: [illegible] 7/16/49

SCALE: FULL SIZE
JULY 13, 1949
FIGURE NO. 8

Copper heating fins were designed to transfer the heat from the electrical heating coils to the evaporator plate (B Figure 8). These fins were machined to the proper dimensions and press fitted into 1/4-inch deep milled slots in the copper disc. The contact area of the fins and the disc could be inscribed in a 3-inch circle. This meant that the area in which the heat was transferred to the plate was over twice as large as the area from which the heat was transferred to the boiling liquid. This feature was an undesirable point in the design of this equipment but was necessitated by the geometrical pattern required for the heater electrical windings. Three 1/8-inch diameter holes were then drilled into the plate for the placement of the plate thermocouples (C Figure 8). This feature will be discussed in detail under the description of the construction of the thermocouples.

The evaporator surface of the copper plate was then plated with a coat of metallic silver 0.001 inch thick. Silver was the actual heat transfer surface in the operation of the equipment. Silver was selected for the heat transfer surface because (1) it resists oxidation and corrosion better than the un-noble metals, and (2) its relative softness would allow the heat transfer surface to be buffered free of pinholes.

The press fitted heating fins were silver soldered in position to the evaporator plate. In order to silver solder the fins it was necessary to heat the entire assembly to a dull red heat. Such heating partially destroyed the silver surface. The marred silver surface was cleaned and a new coat of silver applied making the total average thickness of the silver surface 0.003 inch. The thickness of the silver surface was determined in both cases by measuring the thickness of the plate at ten randomly chosen points before and after plating and averaging the values.

Design and Construction of Thermocouples. Seven thermocouples were employed, three in the evaporator plate, three in the liquid-vapor space, and one in the electrical heater (Figure 9). All seven of the thermocouples were required to meet the specifications of (1) mechanical strength, (2) large change in voltage produced per unit change in temerature, and (3) age resistant. For ease of discussion the design and construction of the thermocouples has been sub-divided into the following headings:

Design and Construction of Evaporator Plate Thermocouples. The evaporator plate was designed to include three thermocouples rather than one in order to determine the possibility of a temperature gradient existing laterally across the plate during operation of the equipment. The thermocouples to measure the plate temperature were designed to be placed within

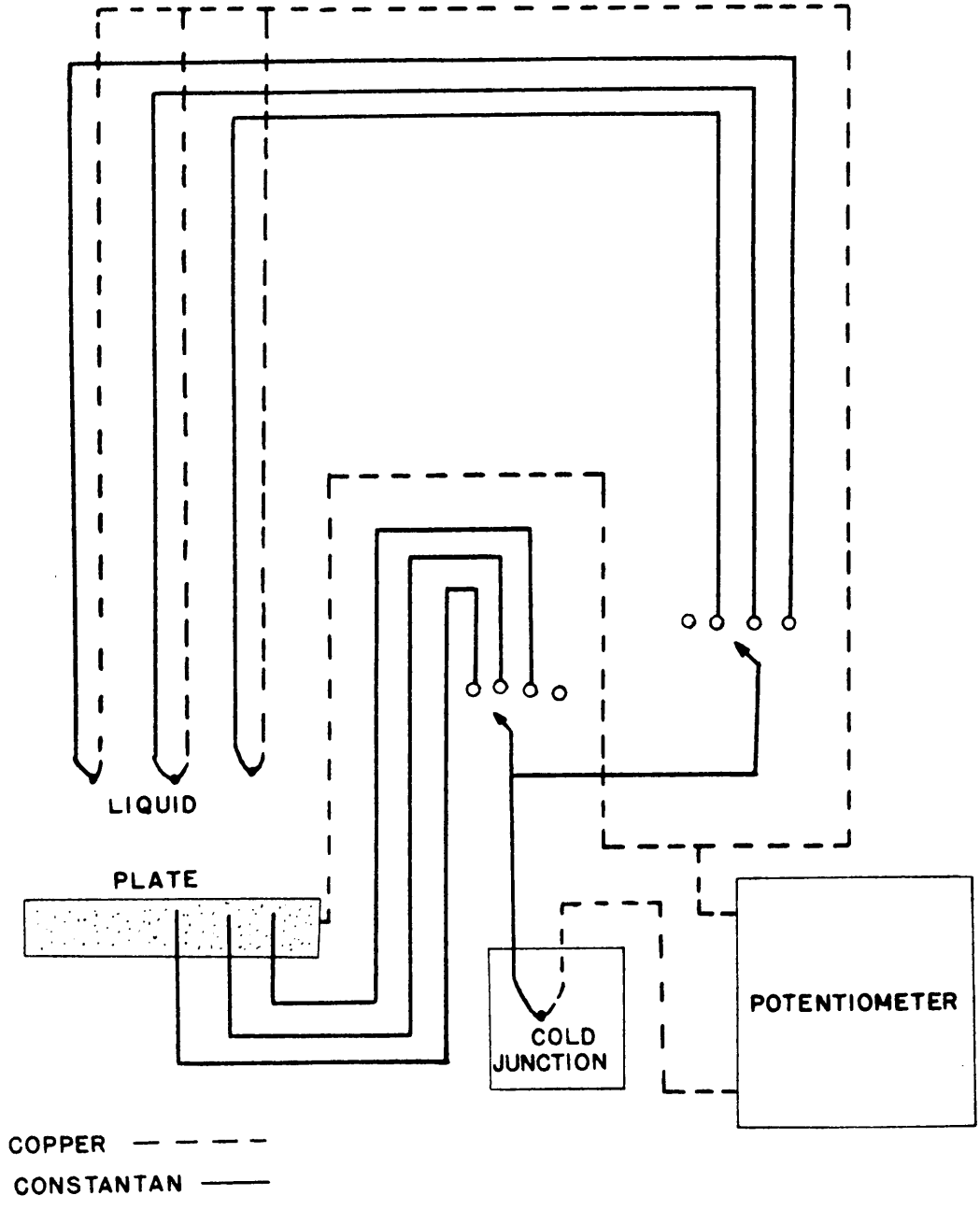

FIGURE 9

SCHEMATIC DIAGRAM OF THERMOCOUPLE CIRCUITS

1/8 inch of the copper heating surface or 0.128 inch of the silver heating surface (Figure 10). The actual temperature of the metal surface at the liquid-metal interface was to be calculated by means of the following conductivity equation:

$$A = \frac{k\,A\,dt}{L}$$

where:

Q = rate of heat transfer, Btu per hr

k = thermal conductivity of metal, Btu per hr-ft-°F

A = area through which heat is transferred, sq ft

L = distance through which heat is transferred, ft

dt = temperature gradient across the distance L, °F.

The actual design of the plate thermocouples is a modification of the type used by Bonilla and Perry[4]. Holes 1/8 inch in diameter were drilled from the underside of the copper evaporator plate to within 1/8 inch of the copper surface. A single lead type of thermocouple was used, i.e., the copper block served as one terminal for the thermocouples. Since copper-constantan fulfilled the requirements set forth for thermocouples, constantan was chosen to serve as the other terminal.

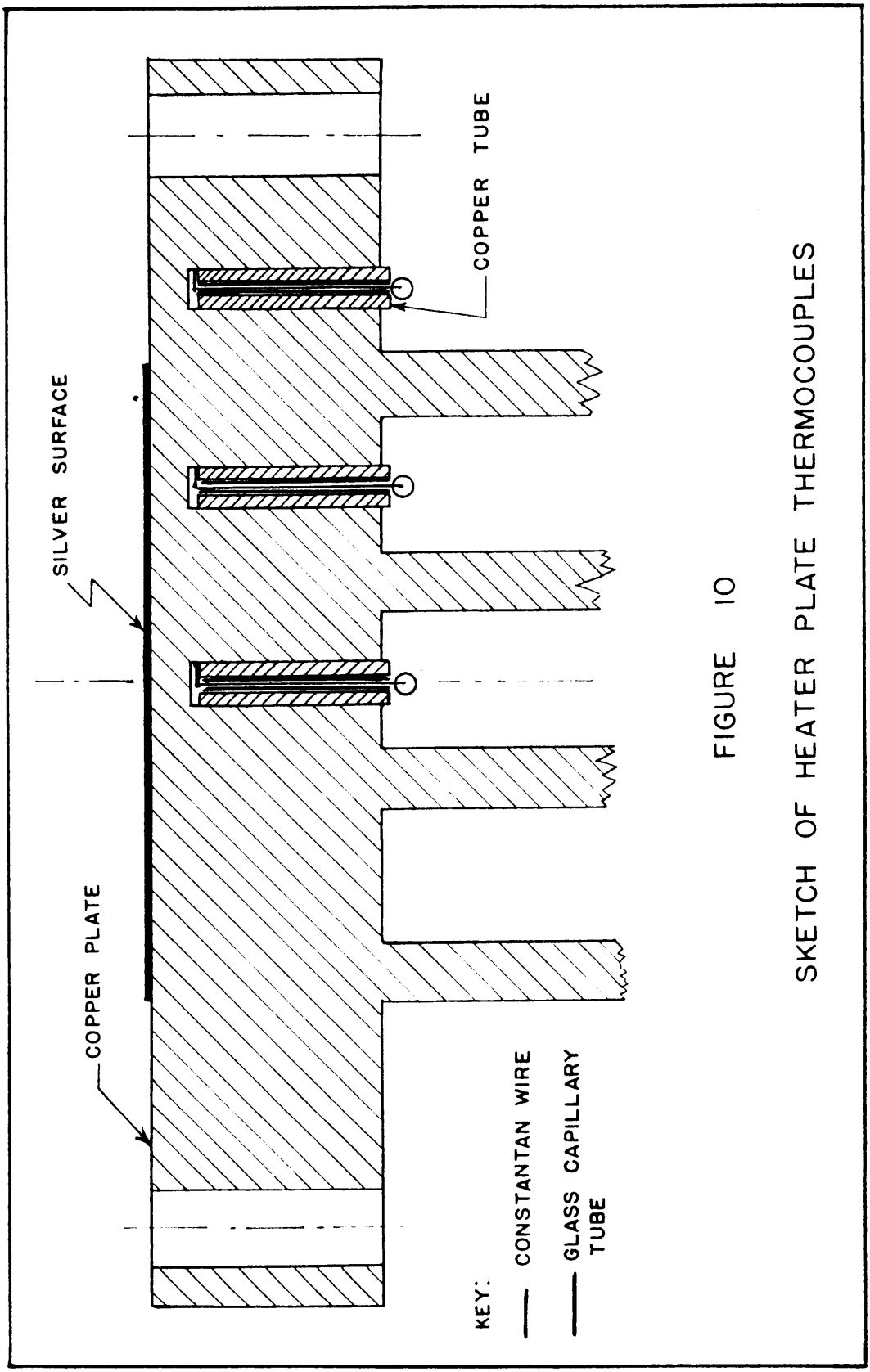

FIGURE 10

SKETCH OF HEATER PLATE THERMOCOUPLES

The problem of installing the constantan lead such that the only contact with the copper plate was at the bottom of the blind hole was the principal problem. This problem was solved mainly by trial and error. Attempted methods included arc welding with the constantan wire as a welding rod, molten lead, and peening or compression. The latter method proved to be the only one which was satisfactory.

The plate thermocouples were constructed by insulating a length of Number 24 constantan wire with pyrex capillary tubing (Figure 10). The wire and tubing were then slipped into a piece of copper tubing having 1/8-inch outside diameter and the constantan wire bent over the end of the copper tubing. The assembly was then pressed into the blind hole as shown in Figure 10 and peened securely into position with a special peening tool. The constantan wire was contacted with the copper plate 1/8 inch from the copper heating surface or 0.128 inch from the silver heating surface. The constantan wires, insulated with pyrex tubing, were brought along the underside of the block between the heating fins and free space of the evaporator plate (Figure 10). The entire underside of the evaporator plate was then covered to a depth of 1/4 inch with alundum cement and dried thoroughly. This

cement held the thermocouple leads in a fixed position and thus eliminated the possibility of changes or breakages in the thermocouples themselves.

The thermocouple circuit was completed to the potentiometer by extending the constantan leads from the block with Number 24 constantan wire and by peening a Number 24 copper wire into the evaporator plate.

Design and Construction of the Liquid and Vapor Space Thermocouples. In as much as copper-constantan thermocouples were used in the evaporator plate it was decided to use the same type of thermocouples for the liquid, vapor, and reflux liquid thermocouples. Since temperature gradients are known to exist in static columns of liquid receiving heat, the thermocouple used to measure the liquid phase temperature was designed such that it could be raised or lowered.

The thermocouples were constructed by twisting the ends of a Number 24 constantan wire and Number 24 copper wire together; the twisted ends of the wires were fused into a bead of metal by means of an electric arc; the thermocouples with leads were then threaded through glass tubing approximately six feet in length depending on the specific thermocouple being constructed.

The liquid and vapor space thermocouples were suspended in the pyrex column by means of laboratory clamps. These clamps were attached to iron supports, mounted on the wooden shelf (C Figure 6). The bead on the thermocouple, used to measure the temperature of the reflux liquid, was bent perpendicular to the insulating glass tubing such that it rested on the inner wall of the evaporator and would be in the stream of refluxing liquid. The vapor space thermocouple was suspended about 1/2 inch from the inner wall of the evaporator body. Both reflux liquid and vapor space thermocouples could be adjusted for height by adjusting the laboratory clamps supporting them. This provision was included to allow for changing conditions of the liquid in the evaporator body.

The liquid space thermocouple was attached to a calibrated pulley such that it could be raised and lowered in the liquid space along a vertical axis in the center of the evaporator body.

The tips of the glass tubes containing the thermocouples were sealed with litharge and glycerine cement.

Design and Construction on the Heater Thermocouple. The Wheelco thermo-regulator was equipped with an iron-constantan thermocouple as specified by the manufacturer. The thermo-

couple was protected with a section of 1/8-inch pyrex tubing and was placed in the heater such that the temperature adjacent to the heating coils was measured and controlled. The purpose of this couple was to eliminate excessive heater temperatures and consequently electrical failures once a state of film boiling was attained during the operation of the evaporator.

Calibration of the Thermocouples. The copper-constantan thermocouples were calibrated at two points, the steam point and the 100° F point. The calibration at the steam point was performed by suspending both plate and liquid-vapor space thermocouples in a large metal drum which was vented to the atmosphere. Steam was throttled in from the steam line until the thermocouples gave a constant reading. Barometric pressure was noted and the temperature calculated from handbook values.

The three liquid-vapor space thermocouples were calibrated at the 100° F point by suspending each thermocouple individually in a constant temperature water bath which was maintained at 100 ± 0.025° F. Since the evaporator plate had been coated on the underside with alundum cement, it could not be immersed in the water bath for calibration at the 100° F point. The evaporator plate was calibrated by comparison with a thermocouple made in a copper block having the same thickness as the evaporator plate. The thermocouple

in the test block was made by the same method as the three thermocouples in the evaporator plate. The test block and the evaporator plate thermocouples were both calibrated at the steam point and the deviation between the evaporator plate thermocouples and the test block thermocouples called a fixed deviation over a range of temperatures. The test block thermocouple was then calibrated at the 100° F point by immersing it in the constant temperature bath used for the liquid-vapor space thermocouples. The results of this calibration were applied to the evaporator thermocouples by means of the fixed deviation.

The calibration data was applied to Holman's equation[32] which is known to apply to copper-constantan thermocouples. From this equation the temperature corresponding to any thermocouple voltage could be calculated.

The iron-constantan was calibrated at the steam point only since the voltage-temperature curve for this metal pair is known to follow a straight line. Since the iron-constantan thermocouple was only used to actuate the thermo-regulator the item of accuracy was not of primary importance.

The cold junction bottle used in this investigation is shown in Figure 11. Cracked ice was used as a reference junction in both the calibration and operation of the copper-constantan thermocouples.

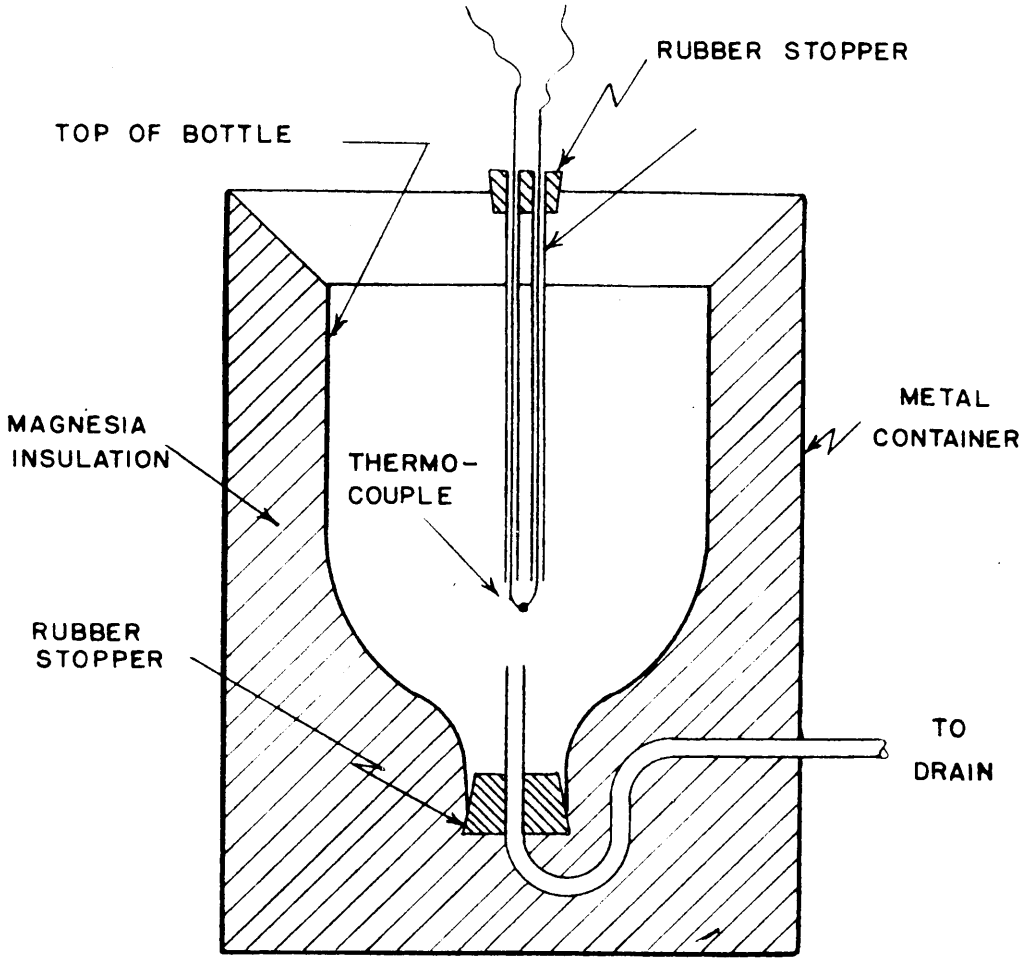

FIGURE 11

SKETCH OF COLD JUNCTION BOTTLE

Design and Construction of Electrical Heater Windings. The electrical heater windings were designed to supply a regulated and metered quantity of heat to the heater fins. The geometrical arrangement of the heater windings was calculated and considered both in the design of the heater fins and the design of the heater circuit.

The heating wire was Number 22 asbestos covered Chromel "A" which was used in three parallel circuits. The purpose of the three parallel circuits was essentially that of convenience of regulating. Two of the circuits were fixed at 900 watts, while the other was variable from 0 to 900 watts.

The windings were constructed by wrapping the nichrome wire into a helical coil using a 1/8-inch welding rod as a mandrel. The three helical windings were approximately five feet in length. It was necessary to heat the coils to a dull red heat before installing in the heater to remove the wax employed in the asbestos wrapping process. After the coils were burned free of wax, they were wrapped in parallel around the heating fins. The coils were insulated from each other and from the copper heating fins with a paste made from alumina and water. The coils were wound uniformly around each fin in the heater and then the entire heater section was packed with alumina mud. Extreme cau-

tion was taken in the wrapping of the coils to avoid any possibility of electrical short circuits.

The heater packed with alumina was then covered with a layer of magnesia approximately one inch thick. The completed heater was then placed in an infra-red drier and dried until a test with an ohmmeter revealed that the heating coils were insulated from each other and from the copper fins. The overall diameter of the finished heater was approximately six inches.

Design and Construction of Electrical Supply and Control Circuit. The electrical supply and control system was based on the principle of the three parallel circuits designed for the heater. The principal factor considered in the design and construction of the electrical system was that of safety.

A panelboard was constructed for the mounting of the electrical control circuit shown in Figure 12. All connections were made behind the panel board with only switches and the variable transformer available to the operator. Number 14 copper wire, insulated with both rubber and cambric, was employed for all connections. Copper blocks approximately one inch square and one-fourth of an inch thick were employed as bus bar terminals at junction points of more than two wires. The entire panel board and all wiring were given two coats of undiluted clear shellac as an added safety precaution.

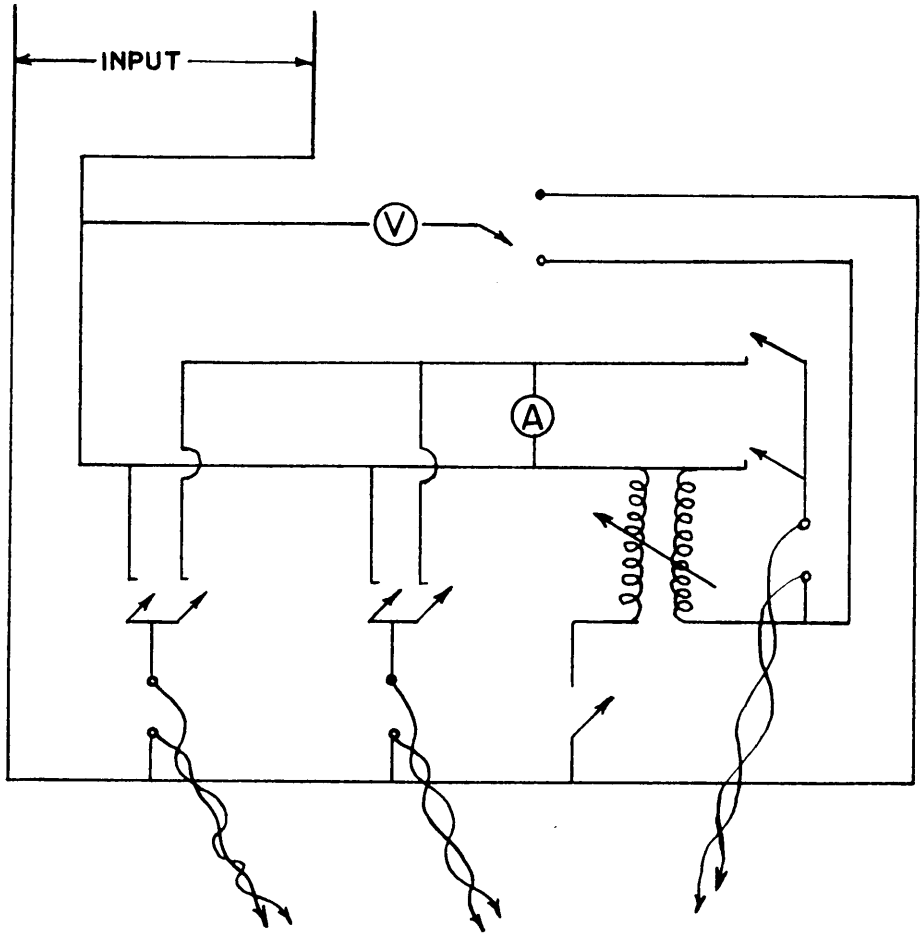

FIGURE 12

SCHEMATIC DIAGRAM

FOR

ELECTRICAL CONTROL CIRCUIT

In as much as the heater circuit was predominately resistive in nature it was assumed that the power factor in the heating circuit was for all practical purposes unity.

With this assumption in mind, the voltmeter-ammeter combination which was used to obtain the wattage were arranged so that they could be switched into the secondary of the variable transformer circuit.

Assembly of Equipment. The principal work in this phase of the investigation included completion of the electrical and thermocouple circuits to include the evaporator plate and heating. The thermocouple leads from the lucite panel (Figure 6) were covered with short lengths of 1/8-inch glass tubing which insulated the wires and still allowed flexibility. The assembled apparatus is shown in Figure 13 and represented by a diagram in Figure 14.

Preparation of Test Liquids. The two test liquids employed in this investigation were ethanol and benzene.

The ethanol was dried by refluxing for one-half hour in the presence of excess calcium oxide and purification by subsequent distillation. The benzene was not pretreated.

The test mixtures of ethanol and benzene were prepared by combining the proper volumes of ethanol and benzene to yield a

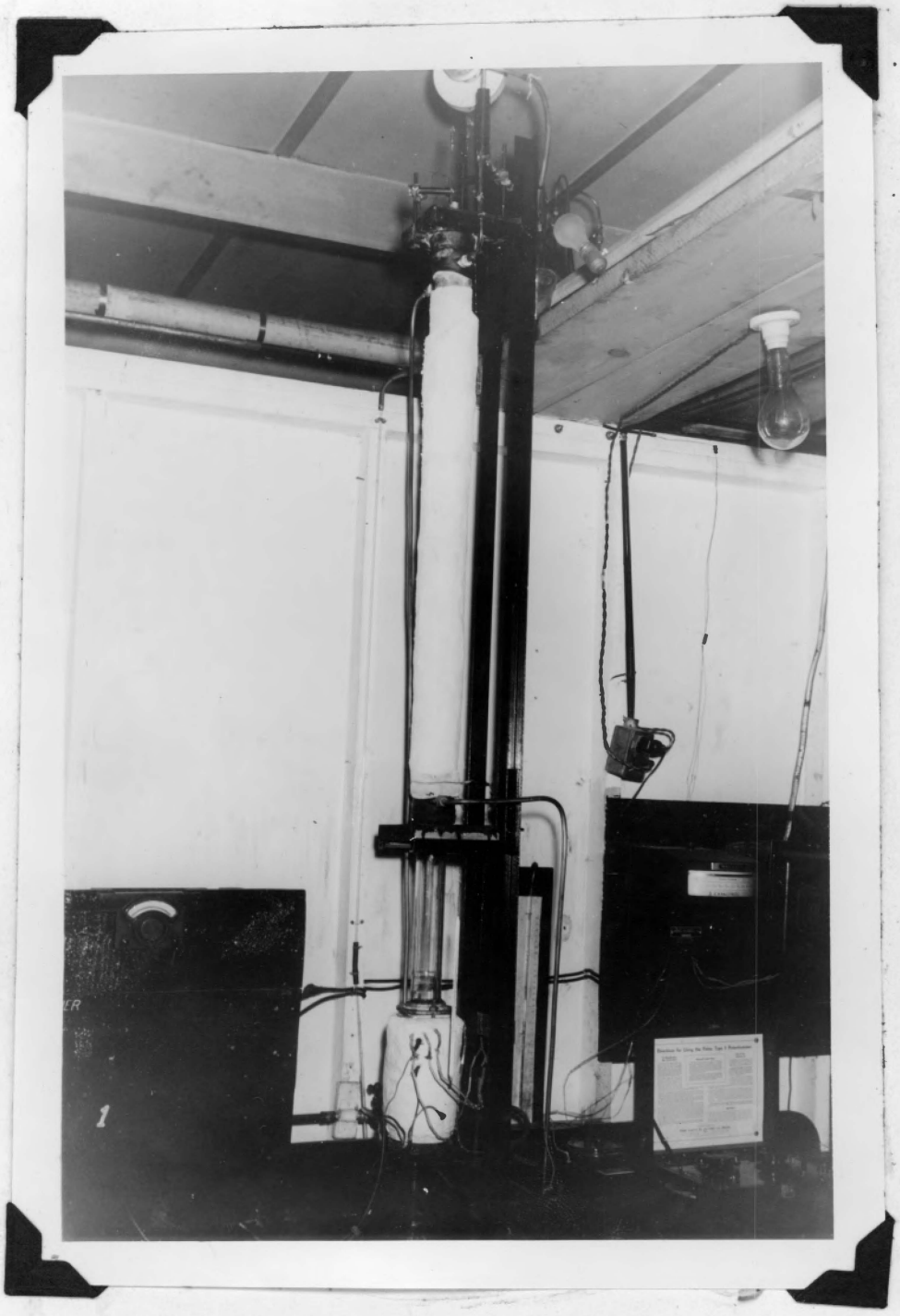

FIGURE 13

PHOTOGRAPH OF ASSEMBLED APPARATUS

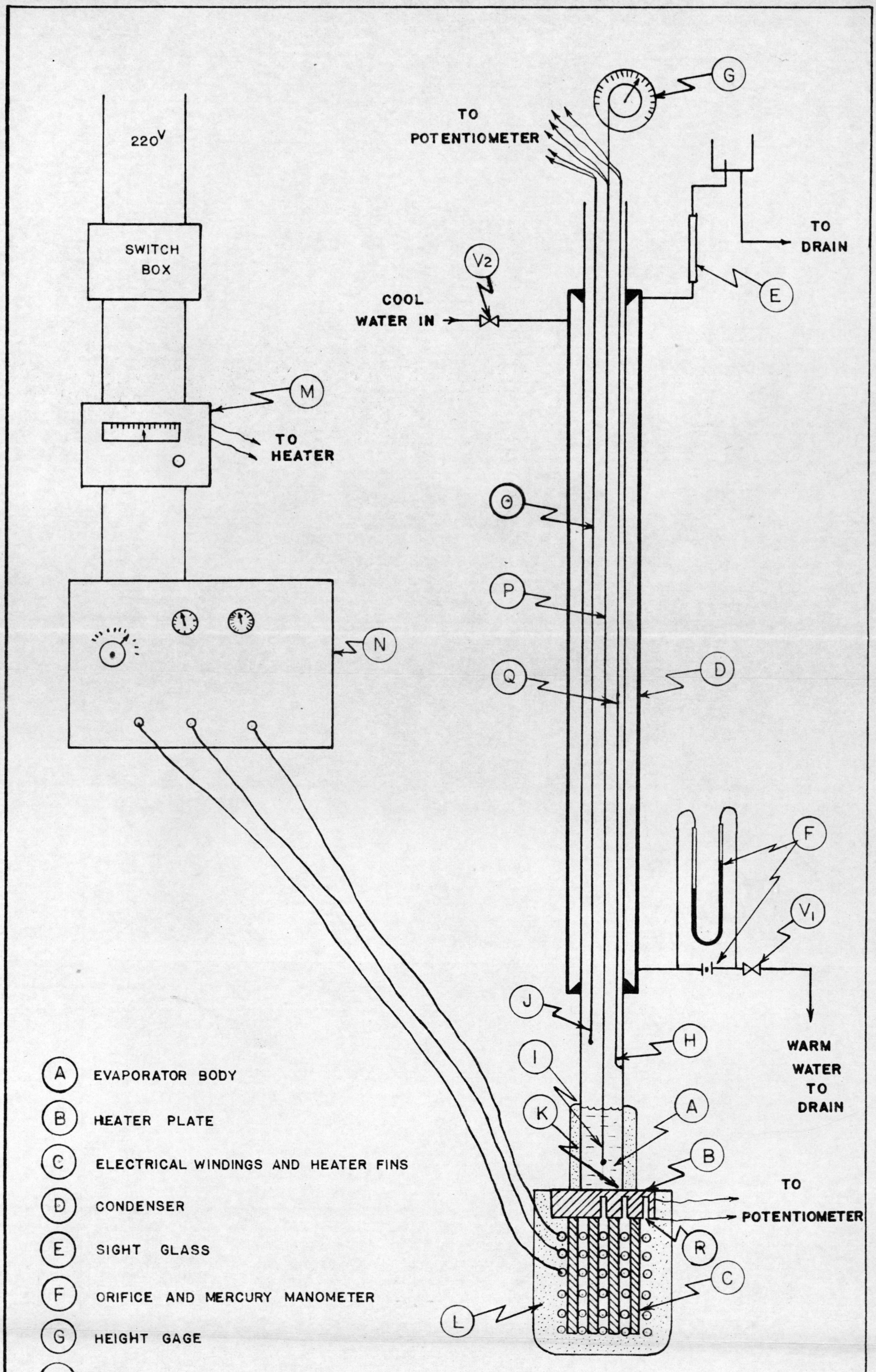

FIGURE 14

ASSEMBLY SKETCH OF APPARATUS

total volume of 250 milliliters. These samples were prepared by using either a pipet or buret or both to measure the correct volumes of ethanol and benzene.

Experimental Operation. The apparatus was operated in the following manner:

The silver surface (K Figure 14) was cleaned first with silver polish and finally with Ivory soap and water. The surface was then dried with a freshly laundered towel. A suitable gasket was placed in its proper position on the heater plate (B Figure 14) and the entire heater unit raised into position. The heater was then bolted to the evaporator by means of a compression type flange (F Figure 7). Considerable care was taken to tighten the flange bolts uniformly in order to avoid breaking the pyrex pipe. Once the evaporator was in place the position of the gasket was checked by observations through the pyrex pipe.

When the evaporator was correctly assembled, the test liquid was charged to the evaporator body from the top of the pyrex pipe. It was customary to initially charge only 50 milliliters of liquid and then check to see if the gasket was leaking before charging the remainder of the liquid. In the event that the gasket seal was imperfect, it was then necessary to siphon the liquid from the evaporator and adjust the gasket.

The cooling water rate in the condenser was then set by first opening value V_1 (Figure 14) as wide as possible and slowly opening V_2 until the manometer F indicated the desired flow rate. This initial flow rate was usually three and one-half pounds per minute. Valve V_1 was then closed slowly until the water rose into the sight glass E. Valve V_1 was manipulated to maintain a fixed head of water in the sight glass. This indicated that the condenser was free of air blocks.

The next step was to place cracked ice in the cold junction bottle and to install the cold junction thermocouple in its proper position in the ice. Caution was taken to keep the thermocouple out of the water in the bottom of the bottle in order to avoid an erroneous reference temperature. The potentionmeter was then standardized by balancing the bridge against a standard cell. Once the potentiometer was balanced against the standard cell it switched to position preparatory for use with the thermocouple. An electric light was used when operating the potentiometer to supply light for noting the galvanometer deflections.

The temperature regulator (M Figure 14) was set by adjusting the knob on top of the instrument. The initial setting was arbitrarily 200° C but was always increased as the heater temperature rose during the normal course of operation. It was customary to

keep the controller set within 50° C of the actual operating temperature to protect the heating coils in the event film boiling was attained.

The next step was to cut on the main switch and set the wattage input to the heater. In adjusting the wattage input the universal practice was to use the variable wattage circuit first (Figure 12). This practice was important for two reasons, namely, (1) in the event of a short circuit in the heater winding the coils would not be exposed to full line voltage, and (2) the heater temperature could be raised slowly to avoid localized overheating in the coils. The ammeter and voltmeter (Figure 12) were switched to the variable wattage circuit and the variable transformer set in the zero output position. The input and ammeter output switches were cut on and the transformer adjusted to give the desired current and voltage. This setting was held until steady state conditions were attained in the evaporator and all data noted and recorded.

The test for steady state conditions was the voltage reading obtained from the plate thermocouples. The voltages from these thermocouples were read every five minutes after a new setting of wattage input was made. If the values of voltage were constant for

three consecutive readings the system was assumed operating under steady state conditions and all data taken.

The use of the selector switches (Figure 9) enabled the determination of the various thermocouple voltages. For each steady state condition a temperature traverse was obtained for the liquid in the evaporator body. This was done by reading the voltage for the thermocouple for a series of positions along a vertical axis in the liquid. The height of the thermocouple bead was read by means of a calibrated pulley which controlled the position of the thermocouple. It was customary to maintain the vapor and reflux liquid thermocouples about two inches above the surface of the liquid during operation.

The ammeter and voltmeter readings were noted and recorded. Room temperature, heater temperature, and cooling water manometer readings were observed and recorded. The latter data was not believed to have been pertinent to the investigation but nevertheless was taken.

Once all data for any given steady state condition was recorded the conditions were changed by adjusting the wattage input to the heater and the foregoing procedure repeated. It was customary to re-standardize the potentiometer and check the cold junction between steady state conditions.

When a state of film boiling was attained the transition from nucleate boiling was detected quickly by the physical action taking place in the evaporator. The liquid would settle from a state of violent ebullition to a state of slow boiling. The transition could also be detected by a rapid rise in plate and heater temperatures. As soon as the state of film boiling was detected the main switch was thrown to the off position to avoid overheating of the electric windings. All data was noted and recorded for steady state conditions at film boiling.

On the completion of a series of tests the evaporator was allowed to cool to room temperature and the cooling water cut off by closing valve V_2 (Figure 14). The test liquid was then siphoned from the evaporator and the heater unit removed from the evaporator body. The silver plate was inspected and its condition noted and recorded. The silver surface was then cleaned, first with silver polish and then with Ivory soap and water. All switches were turned to the off position and the potentiometer disconnected from the circuit.

Preliminary Tests of Apparatus. The first problem included in the preliminary tests was the selection of a gasket to seal the pyrex evaporatory body to the heater plate. The prime requirements of the gasket were (1) must provide a leakproof seal, (2) must be readily removable when equipment is disassembled, (3) must not contaminate the test liquid or silver surface, and (4) must retain its properties over a range of temperatures from 60 to 400° F.

Gaskets tested included metallic lead in single and multiple sheets, permatex gasket cement, tygon paint, butyl rubber, and teflon (polytetrafloroethylene). The lead proved satisfactory when used in multiple sheets but required frequent replacement. Teflon proved completely satisfactory and was used throughout the investigation.

The second item included in preliminary tests was the evaluation of the heat losses of the apparatus. It was planned to evaluate the heat transferred from the silver surface to the boiling liquid by subtracting the heat losses from the total input. The heat losses were evaluated by determining the wattage input required to bring a liquid within 4° F of its boiling point and maintain it there. It was assumed that losses did not change appreciably with heater temperature. The techniques employed were described under Experimental Operation.

Determination of the Characteristics of Heat Transfer of Various Concentrations of Ethanol in Benzene. The operational techniques employed in this phase of the investigation were identical to those outlined under Experimental Operation. The test liquid in this case was a specific concentration of ethanol in benzene. The only modification of these tests was that the top of the pyrex column was fitted with a cork stopper after the liquid was charged. This was done in an attempt to prevent the alcohol from absorbing atmospheric moisture.

Data and Results

The following paragraphs describe the tabulations and graphs of data and results obtained while boiling various mixtures of ethanol and benzene in a horizontal plate evaporator. The general conditions under which the test were made were: heat transfer surface, silver; cold liquid height, 4.5 inches; evaporator diameter, 1-31/32 inches. The test liquids were miscible mixtures of ethanol and benzene varying in composition from 0 to 100 per cent ethanol. Steady state conditions were established before the data were taken. The tests were made at normal atmospheric pressure which varied from 710.0 to 718.0 millimeter of mercury. The condenser cooling water rate was varied from 0.0 to 3.75 pounds per minute depending on the temperature of the reflux liquid. The temperatures in the liquid space were measured at points from 0.0 to 4.0 inches from the silver heat transfer surface.

Calibration of Condenser Cooling Water Orifice and Manometer. Table III contains the data taken during the calibration of the manometer across the orifice in the cooling water line. The manometer readings are reported as read, that is, both of the mercury meniscuses were read and are included in the data. The manometer readings were quoted in graduations where each graduation is equal to one-half inch of mercury.

TABLE III

Calibration Data for Orifice in Condenser Cooling Water Line

Test No.	Manometer Reading graduations [a]	Gross Weight of Beaker grams	Tare Weight of Beaker grams	Time min
1	17.3-13.1	1991	456	2
2	18.0-12.4	1438	456	1
3	19.1-11.3	1654	456	1
4	20.3-10.1	1881	456	1
5	21.8- 8.5	2162	456	1
6	21.3- 9.0	2031	456	1
7	20.3-10.1	1888	456	1
8	18.3-12.2	1490	456	1
9	16.6-13.9	948	456	1
10	16.1-14.4	456	456	-

a One graduation on manometer scale equals 0.5 inch of mercury.

Table IV presents the evaluated results of the condenser cooling water calibration data which appears in Table III. The manometer readings, in graduations, were converted from absolute to differential values. The flow in grams and time in minutes were converted to rate of flow in pounds per minute.

Figure 15 presents the calibration curve for the manometer across the orifice in the cooling water line. The chart was used by first locating the proper value of differential manometer reading on the horizontal axis, then projecting vertically to the calibration curve and across to the rate of flow axis. This chart was used during the operation of the equipment to assist in the regulation of the condenser cooling water rate. The water rate was adjusted to maintain the reflux liquid at its boiling point.

Calibration of Heater Plate and Liquid-Vapor Space Thermocouples. Table V presents the calibration data obtained for the heater plate and liquid-vapor space thermocouples. The true temperatures are presented either as the observed temperature or in terms of the conditions which controlled the temperature, i.e., condensing steam. The thermocouple readings are in millivolts.

Table VI presents an evaluation of the thermocouple calibration data contained in Table V. All thermocouples were evaluated alike since the variations in readings were beyond the sensitivity of the measuring circuit. The two calibrated points

TABLE IV

Results of Calibration of Orifice in Cooling Water Line

Test No.	Manometer Reading graduations [a]	Rate of Cooling Water Flow lbs per min
1	4.2	1.69
2	5.6	2.16
3	7.8	2.64
4	10.2	3.14
5	13.3	3.75
6	12.3	3.46
7	10.2	3.15
8	6.1	2.28
9	2.7	1.09
10	1.7	0.00

a One graduation on manometer scale equals 0.5 inch of mercury.

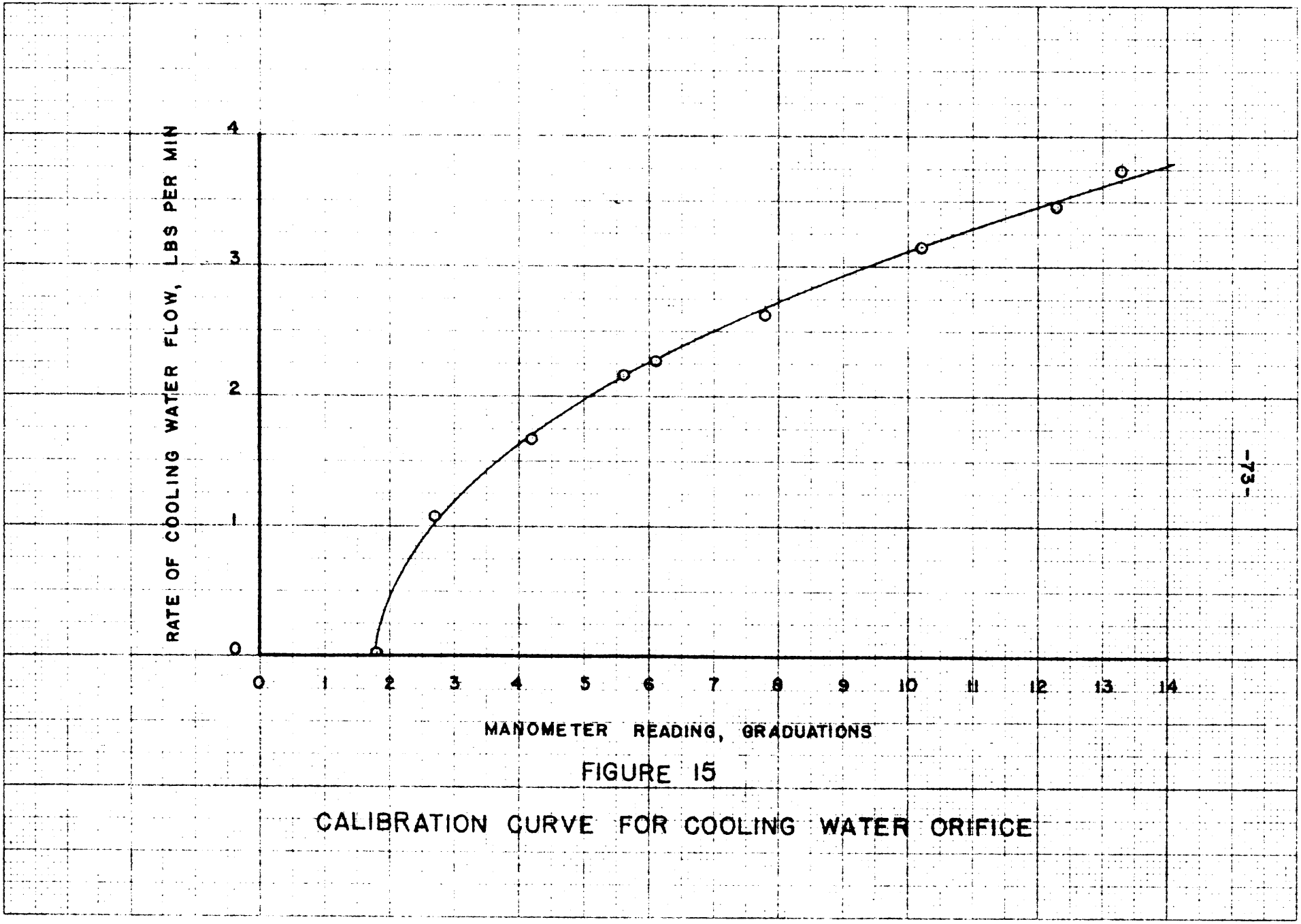

FIGURE 15

CALIBRATION CURVE FOR COOLING WATER ORIFICE

TABLE V

Calibration Data for Heater Plate and Liquid-Vapor Space Thermocouples

Standard Temperature	Steam Point [a]			100° F		
Test Number	1	2	3	1	2	3
	Thermocouple Reading mv					
Thermocouple Position						
Plate						
center	4.20	4.20	--	--	--	--
middle	4.19	4.19	--	--	--	--
outside	4.20	4.20	--	--	--	--
Test Block	4.20	4.19	--	1.49	1.49	1.49
Liquid-Vapor Space						
reflux liquid	4.20	4.21	4.20	1.50	1.50	1.50
vapor	4.20	4.20	4.20	1.51	1.50	1.50
liquid	4.20	4.20	4.20	1.50	1.50	1.50

a Barometric pressure 713.0 mm mercury.

TABLE VI

Results of Calibration of Heater Plate and Liquid-Vapor Space Thermocouples as Calculated by Holman's Equation

Temperature °F	Thermocouple EMF mv
50	0.36
75	0.92
100 [a]	1.50
125	2.10
150	2.71
175	3.33
200	4.06
209.23 [a]	4.20
225	4.49
250	5.13
275	5.91
300	6.55
350	7.87
400	9.21
425	9.89

a Experimentally determined values.

were applied to Holman's Equation[32] and the thermocouples evaluated over a temperature range from 75 to 425° F.

Figure 16 presents the evaluated results contained in Table VI. This chart was used by first locating the proper value of thermocouple electromotive force on the ordinate, then projecting horizontally to the calibration curve and down to the temperature axis. This chart was used for the evaluation of heater plate and liquid-vapor space temperatures from thermocouple readings.

Calibration of Liquid Thermocouple Height Indicator. Table VII contains the data obtained in the calibration of the liquid thermocouple height indicator. The reading of the indicator is given in terms of graduations which were arbitrarily chosen. The height of the thermocouple bead above the silver heating surface is included in inches.

Figure 17 is a graphical presentation of the liquid thermocouple height indicator calibration data contained in Table VII. This chart was used by first, locating the proper value of the height indicator reading on the horizontal axis, then projecting vertically to the calibration curve and across horizontally to the height scale. This chart was used to determine the position of the liquid thermocouple.

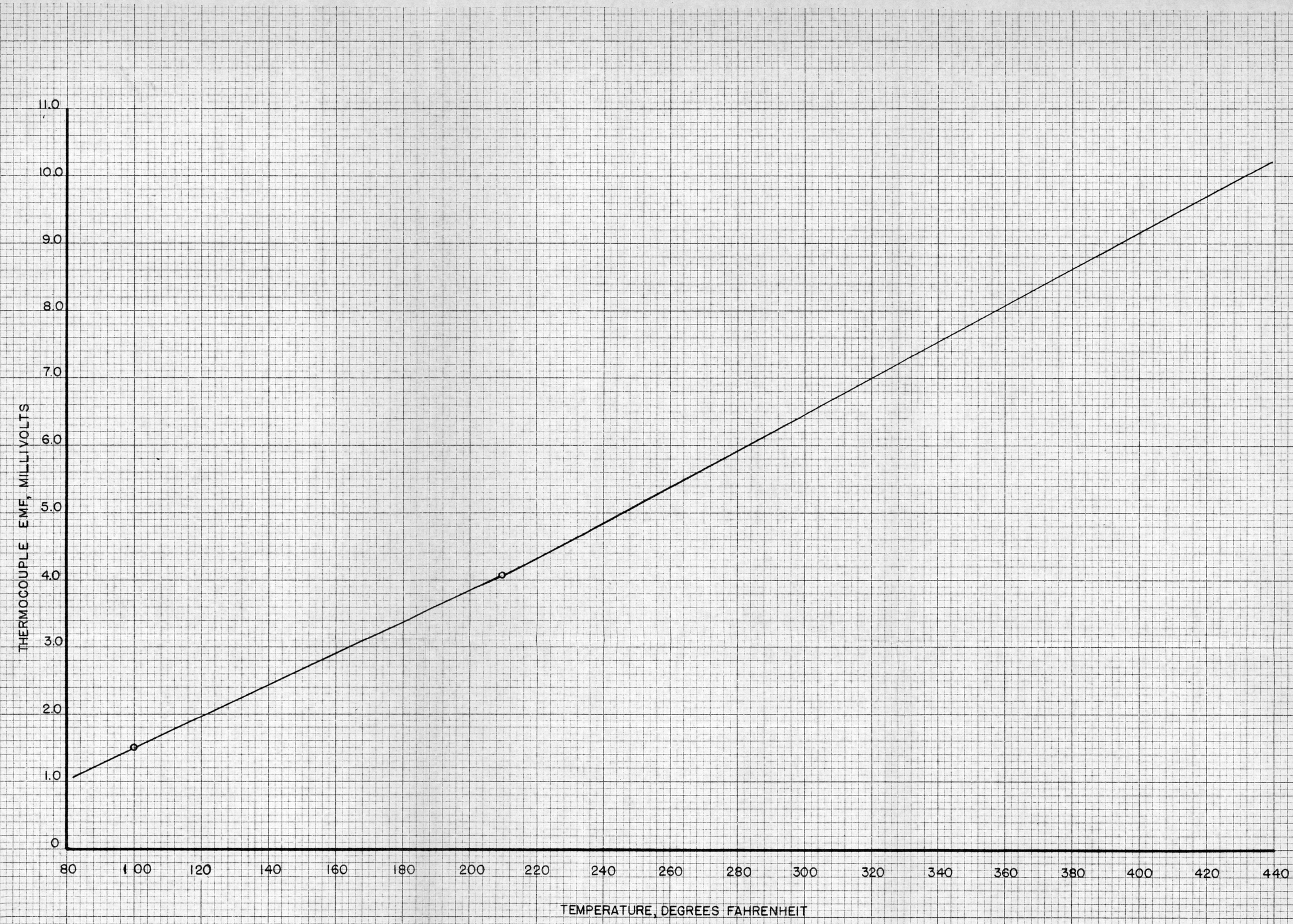

FIGURE 16

CALIBRATION CURVE FOR HEATER PLATE AND LIQUID-VAPOR SPACE THERMOCOUPLES

TABLE VII

Calibration Data for Liquid Space Thermocouple

Height Indicator

Reading of Height Indicator	Height of Thermocouple Bead Above Silver Surface
graduations	inches
12	00
15	9/16
18	1-1/8
21	1-3/4
24	2-1/4
27	2-13/16
30	3-3/8

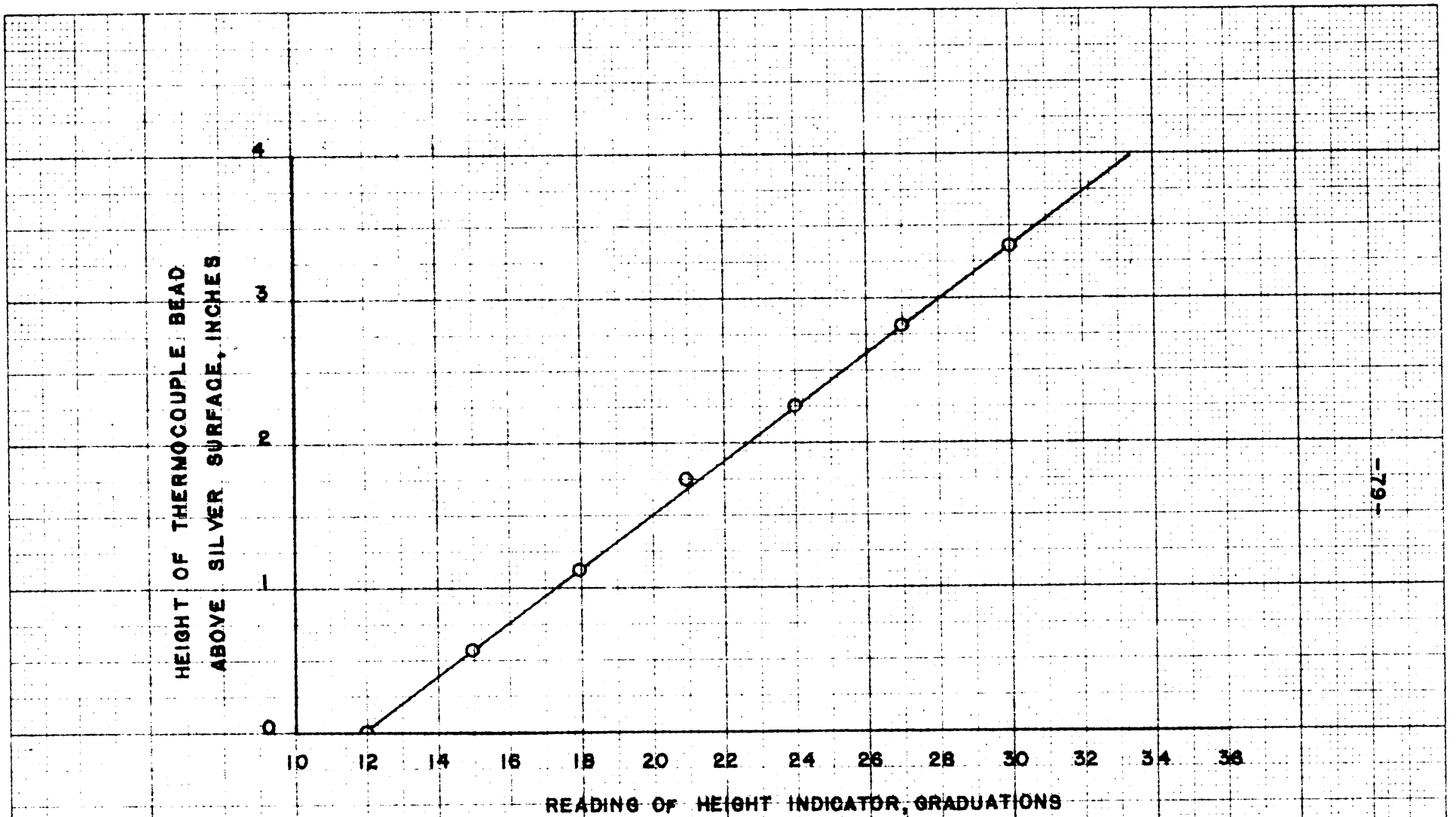

FIGURE 17

CALIBRATION CURVE FOR LIQUID THERMOCOUPLE HEIGHT INDICATOR

Summary of Operating Conditions for Horizontal Plate Evaporator Using Mixtures of Ethanol and Benzene as Test Liquids. Table VIII presents the data obtained from the operation of the horizontal plate evaporator using mixtures of ethanol and benzene as test liquids. The table shows the energy input to the evaporator in watts. The composition of the liquid as charged to the evaporator is reported in volume per cent. The cooling water manometer readings are in graduations where each graduation is equivalent to one-half inch of mercury. Room and heater temperatures are reported as read, in °F and °C, respectively. The heater plate and liquid-vapor space thermocouple readings are reported in millivolts. In the case of the liquid space thermocouple, a traverse was made in the liquid and height indicator reading with the accompanying thermocouple reading reported.

Notations in the "Remarks" column indicate that these tests represent a state of film boiling. The plate thermocouple readings were included for conditions of film boiling. The values of wattage for the tests which gave film boiling are not the wattage inputs during film boiling but represent the wattage inputs which brought the system to a state of film boiling.

Results Obtained from the Operation of the Horizontal Plate Evaporator Using Ethanol-Benzene Mixtures as Test Liquids. Table IX contains the results obtained from the operation of the horizontal

TABLE VIII

Summary of Operating Conditions for Horizontal Plate Evaporator Using Mixtures of Ethanol and Benzene as Test Liquids

Test No.	Composition of Test Liquid		Barometric Pressure	Steady State No.	Cooling Water Manometer Reading	Heater Current			Heater Voltage			Temperatures					Thermocouple Readings mv						Remarks
	Volume per cent Ethanol	Volume per cent Benzene	mm Hg		graduations	amperes			volts			Heater	Room	Plate			Liquid		Traverse in Liquid Height Indicator Reading graduations				
						1[h]	2[g]	3[f]	1[h]	2[g]	3[f]	°C	°F	1[a]	2[b]	3[c]	1[d]	2[e]	12	14	24	34	
I	0	100	710.0	1	18.8-11.2	-	-	2.00	-	-	95	120	81	4.17	4.17	4.23	3.24	3.27	3.29	3.27	3.27	3.27	
				2	18.8-11.2	-	-	2.97	-	-	142	155	81	4.47	4.53	4.59	3.24	3.28	3.27	3.27	3.27	3.27	
				3	18.8-11.2	-	-	4.00	-	-	187	210	81	4.79	8.89	5.08	3.24	3.26	3.26	3.26	3.27	3.27	
				4	18.8-11.2	-	-	4.10	-	-	189		81	8.87	8.87	8.87	3.25	3.27	3.27	3.27	3.27	3.27	Film Boiling
II	10	90	711.5	1	18.8-11.2	-	-	2.00	-	-	92	110	91	3.65	3.66	3.67	2.73	2.73	2.73	2.73	2.73	2.73	
				2	18.8-11.2	-	-	3.00	-	-	144	157	89	3.95	4.01	4.11	2.73	2.73	2.73	2.73	2.73	2.73	
				3	18.8-11.2	-	-	3.50	-	-	164	177	90	4.07	4.17	4.28	2.73	2.72	2.73	2.73	2.73	2.73	
				4	18.8-11.2	-	-	4.00	-	-	184	210	90	4.20	4.33	4.52	2.72	2.73	2.75	2.73	2.73	2.73	
				5	18.8-11.2	-	-	4.10	-	-	189	250	90	8.35	8.35	8.35	2.70	2.73	2.76	2.74	2.73	2.73	Film Boiling
III	20	80	713.0	1	18.8-11.2	-	-	2.00	-	-	96	112	81	3.59	3.61	3.63	2.73	2.73	2.77	2.75	2.73	2.73	
				2	18.8-11.2	-	-	3.00	-	-	141	148	86	3.61	3.70	3.81	2.72	2.72	2.72	2.72	2.72	2.72	
				3	18.8-11.2	-	-	3.25	-	-	152	165	86	3.73	3.82	3.95	2.73	2.73	2.73	2.73	2.73	2.73	
				4	18.8-11.2	-	-	3.50	-	-	163	181	84	3.79	3.91	4.06	2.70	2.73	2.73	2.73	2.73	2.73	
				5	18.8-11.2	-	-	3.72	-	-	173	195	85	3.88	4.00	4.17	2.69	2.72	2.73	2.73	2.73	2.73	
				6	18.8-11.2	-	-	4.00	-	-	184	210	82	3.99	4.13	4.31	2.70	2.73	2.73	2.73	2.73	2.73	
				7	18.8-11.1	-	-	4.25	-	-	192	225	83	4.10	4.24	4.47	2.73	2.73	2.73	2.73	2.73	2.73	
				8	18.8-11.1	4.35	-	-	205	-	-	225	83	7.50	7.50	7.50	2.72	2.72	2.73	2.73	2.73	2.73	Film Boiling
				9	18.8-11.1	-	-	4.30	199	-	-	227	82	4.14	4.29	4.53	2.73	2.73	2.73	2.73	2.73	2.73	
IV	30	70	715.0	1	18.8-11.2	-	-	2.00	-	-	93	110	86	3.76	3.79	3.81	2.71	2.71	2.74	2.75	2.74	2.74	
				2	18.8-11.2	-	-	3.00	-	-	142	148	86	3.88	3.96	4.05	2.73	2.73	2.73	2.73	2.73	2.73	
				3	18.8-11.2	-	-	3.50	-	-	164	182	86	3.97	4.10	4.24	2.73	2.73	2.73	2.73	2.73	2.73	
				4	18.8-11.2	-	-	4.00	-	-	184		86	4.07	4.24	4.43	2.73	2.73	2.73	2.73	2.73	2.73	
				5	18.8-11.2	-	-	4.30	-	-	197	230	87	4.17	4.35	4.56	2.71	2.73	2.73	2.73	2.73	2.73	
				6	18.8-11.2	-	-	4.40	-	-	202	240	86	4.30	4.45	4.63	2.73	2.73	2.73	2.73	2.73	2.73	
				7		-	-	4.50	-	-	206	250	86	8.01	8.01	8.01	2.73	2.73	2.73	2.73	2.73	2.73	Film Boiling
V	40	60	718.0	1	19.2-12.2	-	-	2.00	-	-	95	130	80	3.55	3.58	3.61	2.73	2.73	2.75	2.74	2.73	2.73	
				2	19.2-12.2	-	-	3.00	-	-	143	140	82	3.64	3.72	3.82	2.73	2.73	2.74	2.73	2.73	2.73	
				3	19.2-12.2	-	-	3.50	-	-	167	172	82	3.66	3.78	3.93	2.73	2.73	2.73	2.73	2.73	2.73	
				4	19.2-12.2	-	-	4.00	-	-	185	198	82	3.72	3.84	4.01	2.73	2.73	2.73	2.73	2.73	2.73	
				5	19.2-12.2	4.30	-	-	202	-	-	340	84	3.74	3.94	4.17	2.73	2.73	2.73	2.73	2.73	2.73	
				6	19.2-12.2	4.30	-	1.50	202	-	70		85	7.87	7.87	7.87	2.70	2.73	2.76	2.74	2.73	2.73	Film Boiling
				7	19.2-12.2	4.30	-	1.25	202	-	59	345	85	4.16	4.39	4.64	2.73	2.73	2.73	2.73	2.73	2.73	
VI	50	50	718.0	1	19.2-12.2	-	-	2.00	-	-	96	105	79	3.70	3.73	3.75	2.76	2.76	2.76	2.76	2.76	2.76	
				2	19.2-12.2	-	-	3.00	-	-	144	150	79	3.82	3.90	3.99	2.73	2.73	2.73	2.73	2.73	2.73	
				3		-	-	3.50	-	-	164	175	80	3.85	3.97	4.10	2.73	2.73	2.73	2.73	2.73	2.73	
				4	19.2-12.2	-	-	4.00	-	-	186	205	80	3.92	4.04	4.24	2.73	2.73	2.73	2.73	2.73	2.73	
				5	19.2-12.2	4.35	-	-	204	-	-	345	80	4.00	4.17	4.42	2.73	2.73	2.73	2.73	2.73	2.73	
				6	19.2-12.2	4.35	-	1.50	204	-	69	370	80	4.09	4.28	4.59	2.73	2.73	2.73	2.73	2.73	2.73	
				7		4.35	-	1.75	204	-	82			8.01	8.01	8.01	2.73	2.73	2.73	2.73	2.73	2.73	Film Boiling
				8	19.2-12.2	4.35	-	1.65	204	-	77	375	80	4.12	4.35	4.59	2.73	2.73	2.73	2.73	2.73	2.73	
VII	60	40	717.0	1	18.8-13.5	-	-	2.00	-	-	95	110	76	3.77	3.82	3.86	2.73	2.73	2.75	2.73	2.73	2.73	
				2	18.8-13.5	-	-	3.00	-	-	145	142	76	3.95	4.00	4.13	2.70	2.73	2.73	2.73	2.73	2.73	
				3	18.8-13.5	-	-	3.50	-	-	167	170	76	4.03	4.13	4.26	2.73	2.73	2.73	2.73	2.73	2.73	
				4	18.8-13.5	-	-	4.00	-	-	188	205	76	4.11	4.25	4.45	2.73	2.73	2.73	2.73	2.73	2.73	
				5	18.8-13.5	4.35	-	-	204	-	-	355	77	4.12	4.28	4.52	2.73	2.73	2.73	2.73	2.73	2.73	
				6	18.8-13.5	4.35	-	1.50	204	-	72	378	77	4.21	4.34	4.65	2.73	2.73	2.73	2.73	2.73	2.73	
				7	18.8-13.5	4.35	-	2.00	204	-	96	390	77	4.26	4.52	4.85	2.73	2.73	2.73	2.73	2.73	2.73	
				8		4.35	-	2.20	204	-	106			8.35	8.35	8.35	2.73	2.73	2.75	2.73	2.73	2.73	Film Boiling
VIII	70	30	717.0	1	19.0-12.8	-	-	2.00	-	-	96	100	82	3.76	3.79	3.86	2.70	2.76	2.79	2.76	2.77	2.77	
				2	19.0-13.0	-	-	3.00	-	-	142	140	83	3.94	4.00	4.11	2.76	2.76	2.76	2.76	2.76	2.76	
				3	19.0-13.0	-	-	4.00	-	-	187	197	84	4.11	4.26	4.37	2.76	2.76	2.76	2.76	2.76	2.76	
				4	19.0-13.0	4.25	-	-	201	-	-	350	82	4.15	4.30	4.53	2.76	2.76	2.76	2.76	2.76	2.76	
				5	19.0-13.0	4.25	-	2.00	201	-	96	398	83	4.28	4.51	4.87	2.76	2.76	2.76	2.76	2.76	2.76	
				6	19.0-13.0	4.25	-	2.40	201	-	117		84	8.78	8.78	8.78	2.76	2.76	2.76	2.76	2.76	2.76	Film Boiling
				7	19.0-13.0	4.25	-	2.30	201	-	113	405	84	4.30	4.64	4.99	2.75	2.76	2.76	2.76	2.76	2.76	
IX	80	20	718.0	1	19.0-13.0	-	-	2.00	-	-	95	105	80	3.85	3.89	3.95	2.71	2.82	2.83	2.83	2.83	2.83	
				2	19.0-13.0	-	-	3.00	-	-	147	150	81	4.02	4.11	4.22	2.83	2.83	2.83	2.83	2.83	2.83	
				3	18.8-12.9	-	-	4.00	-	-	186	200	81	4.18	4.36	4.56	2.83	2.83	2.83	2.83	2.83	2.83	
				4	18.8-13.3	4.22	-	-	202	-	-	348	81	4.25	4.39	4.65	2.83	2.83	2.83	2.83	2.83	2.83	
				5	18.8-13.2	4.22	-	2.00	202	-	96	390	82	4.35	4.60	4.92	2.83	2.83	2.83	2.83	2.83	2.83	
				6	18.8-13.2	4.22	-	2.50	202	-	119	400	83	9.93	9.93	9.93	2.80	2.83	2.83	2.83	2.83	2.83	Film Boiling
				7	18.8-13.2	4.22	-	2.40	202	-	114	400	82	4.41	4.69	5.01	2.83	2.83	2.83	2.83	2.83	2.83	
X	90	10	717.0	1	19.2-12.6	-	-	2.00	-	-	98	110	80	3.79	3.82	3.90	2.94	2.94	2.95	2.94	2.94	2.94	
				2	19.0-13.0	-	-	3.00	-	-	142	140	80	3.89	4.11	4.19	2.94	2.94	2.94	2.94	2.94	2.94	
				3	19.0-13.0	-	-	4.00	-	-	186	190	80	4.19	4.39	4.57	2.94	2.94	2.94	2.94	2.94	2.94	
				4	19.0-13.0	4.15	-	2.00	198	-	96	340	80	4.47	4.67	4.92	2.94	2.94	2.94	2.94	2.94	2.94	
				5	19.0-13.0	4.15	-	3.00	198	-	147	350	79	9.40	9.40	9.40	2.94	2.94	2.94	2.94	2.94	2.94	Film Boiling
				6	19.0-13.0	4.15	-	2.90	198	-	142	360	79	4.54	4.77	5.07	2.94	2.94	2.94	2.94	2.94	2.94	
XI	100	0	713.0	1	20.0-10.0	-	-	2.00	-	-	98	105	80	3.68	3.68	3.69	3.17	3.17	3.17	3.17	3.17		
				2	20.0-10.0	-	-	3.50	-	-	164	165	80	4.07	4.13	4.25	3.20	3.20		3.19			
				3	20.0-10.0	-	-	4.00	-	-	182	195	80	4.19	4.38	4.41	3.20	3.20		3.20			
				4	20.0-10.0	-	4.28	1.00	-	205	50	265	80	4.29	4.43	4.65	3.20	3.20		3.21			
				5	20.0-10.0	-	4.28	2.00	-	206	89	290	80	4.40	4.57	4.82	3.20	3.20		3.20			
				6	22.7- 7.3	-	4.28	3.00	-	204	144	330	81	4.57	4.82	5.11	3.20	3.20		3.21			
				7		-	4.28	3.25	-	204	155		81	9.23	9.23	9.23	3.19	3.19		3.19			Film Boiling
XII	100	0		1	20.0-10.0	-	-	1.00	-	-	47	90	80	3.35	3.35	3.35	3.01	2.91	3.17	3.14	3.12	3.12	
XIII	0	100		1	20.0-10.0	-	-	1.00	-	-	47	90	80	3.38	3.38	3.38	3.11	2.86	3.25	3.23	3.21	3.21	

a Thermocouple in center of heater plate
b Thermocouple between center and outside of heating area
c Thermocouple at edge of heating area
d Reflux Liquid thermocouple
e Vapor space thermocouple
f Variable capacity circuit
g Fixed capacity circuit
h Fixed capacity circuit

TABLE IX

Results from Operation of Horizontal Plate Evaporator Using Mixtures of Ethanol and Benzene as Test Liquids

Test No.	Steady State No.	Composition of Test Liquid				Corrected Plate Temperature	Liquid Temperature	Corrected Temperature Gradient	Heat Flux
		Ethanol Volume per cent	Benzene Volume per cent	Ethanol Mole per cent	Benzene Mole per cent	°F	°F	°F	Btu per hr-sq ft-°F
I	1	0	100	0	100.00	210.4	174.0	36.4	23,428
	2					222.6	174.0	48.6	61,394
	3					234.0	174.0	60.0	114,845
	4					386.0x	174.0	212.0	119,252
II	1	10	90	14.42	85.58	189.4	151.0	39.4	22,445
	2					201.8	151.0	50.8	63,075
	3					206.8	151.0	55.8	92,154
	4					211.8	151.0	60.8	112,879
	5					367.0x	151.0	216.0	119,252
III	1	20	80	27.51	72.49	187.5	151.0	37.4	23,755
	2					188.2	151.0	37.2	61,600
	3					192.8	151.0	41.8	73,232
	4					195.7	151.0	44.7	85,765
	5					199.3	151.0	48.3	97,754
	6					203.6	151.0	52.6	112,879
	7					207.0	151.0	56.0	125,985
	8					335.5x	151.0	184.5	138,395
	9					208.4	151.0	57.4	132,489
IV	1	30	70	39.42	60.57	194.9	151.0	43.9	22,772
	2					199.5	151.0	48.5	62,092
	3					203.4	151.0	52.4	86,338
	4					208.2	151.0	57.2	99,772
	5					210.7	151.0	59.7	131,080
	6					215.3	151.0	64.3	137,912
	7					354.5x	151.0	203.5	144,170
V	1	40	60	50.30	49.70	186.1	151.0	35.1	23,428
	2					189.5	151.0	38.5	62,585
	3					190.8	151.0	39.8	87,977
	4					191.6	151.0	40.6	115,534
	5					195.1	151.0	44.1	134,603
	6					349.0x	151.0	198.0	151,805
	7					211.3	151.0	60.3	140,787
VI	1	50	50	60.29	39.71	192.2	151.0	41.2	23,755
	2					196.8	151.0	45.8	63,075
	3					198.4	151.0	47.3	86,338
	4					199.9	151.0	48.9	114,189
	5					203.9	151.0	52.9	137,683
	6					207.0	151.0	56.0	154,639
	7					354.5x	151.0	203.5	161,192
	8					209.4	151.0	58.4	158,497
VII	1	60	40	69.49	30.51	195.4	151.0	44.7	23,428
	2					201.5	151.0	50.5	63,566
	3					204.9	151.0	53.9	88,058
	4					208.1	151.0	57.1	115,500
	5					208.0	151.0	57.0	137,683
	6					209.2	151.0	58.2	155,376
	7					214.3	151.0	63.3	169,138
	8					367.0x	151.0	216.0	172,611
VIII	1	70	30	77.98	22.02	194.4	153.0	41.4	23,755
	2					201.3	153.0	48.3	62,092
	3					208.5	153.0	55.5	114,845
	4					209.5	153.0	56.5	132,252
	5					215.5	153.0	62.5	163,707
	6					383.0	153.0	230.0	178,255
	7					219.8	153.0	66.8	174,851
IX	1	80	20	85.86	14.14	198.4	155.5	38.9	23,428
	2					205.9	155.5	50.4	64,549
	3					212.5	155.5	58.0	114,190
	4					212.7	155.5	57.2	131,955
	5					218.7	155.5	63.2	163,065
	6					426.0x	155.5	270.5	180,695
	7					221.0	155.5	65.5	176,779
X	1	90	10	93.26	6.74	195.9	160.0	35.9	22,772
	2					205.0	160.0	45.0	62,092
	3					215.0	160.0	55.0	114,190
	4					222.2	160.0	62.2	158,574
	5					406.0x	160.0	246.0	199,168
	6					224.7	160.0	64.7	194,384
XI	1	100	0	100.00	0	189.8	170.0	19.8	24,411
	2					205.6	171.0	34.6	86,339
	3					209.9	171.0	38.9	111,568
	4					214.5	171.0	43.5	136,044
	5					218.4	171.0	47.4	165,907
	6					225.1	171.0	54.1	206,118
	7					400.0	171.0	229.0	217,873
XII	1	100	0	100.00	0	177.0	167.0x	10.0	7,708
XIII	1	0	100	0	100.00	179.0	172.0x	7.0	7,708

x Not corrected.

plate evaporator using ethanol-benzene mixtures as test liquids. These results were evaluated from the data contained in Table VIII. The values of wattage input reported in Table VIII were corrected and converted to unit rate of heat transfer, in Btu per hour-square foot. The plate and liquid temperatures were evaluated and the temperature gradient, in °F, calculated therefrom. In the tests where the temperature gradient exceeded 69° F the system was at a state of film boiling. The values of heat flux which accompanied the state of film boiling do not represent the heat transferred with the high temperature gradient but are the values which caused the system to shift from nucleate to film boiling.

The compositions of the test liquids are presented in Table IX for purposes of clarity. Compositions are presented on mole and volume per cent basis.

Tests XII and XIII represent the evaluation of heat losses of the heater using pure ethanol and benzene as test liquids.

Heat Flux Curves for Mixtures of Ethanol and Benzene. Figure 18 presents a chart relating the values of heat and temperature gradient presented in Table IX, Tests I to XI, inclusive. Each curve represents the heat flux-temperature gradient relationship for a given mixture of ethanol and benzene from a point in the nucleate range to a state of film boiling. The dotted horizontal lines represent the heat transfer rates which caused or induced a state of film boiling.

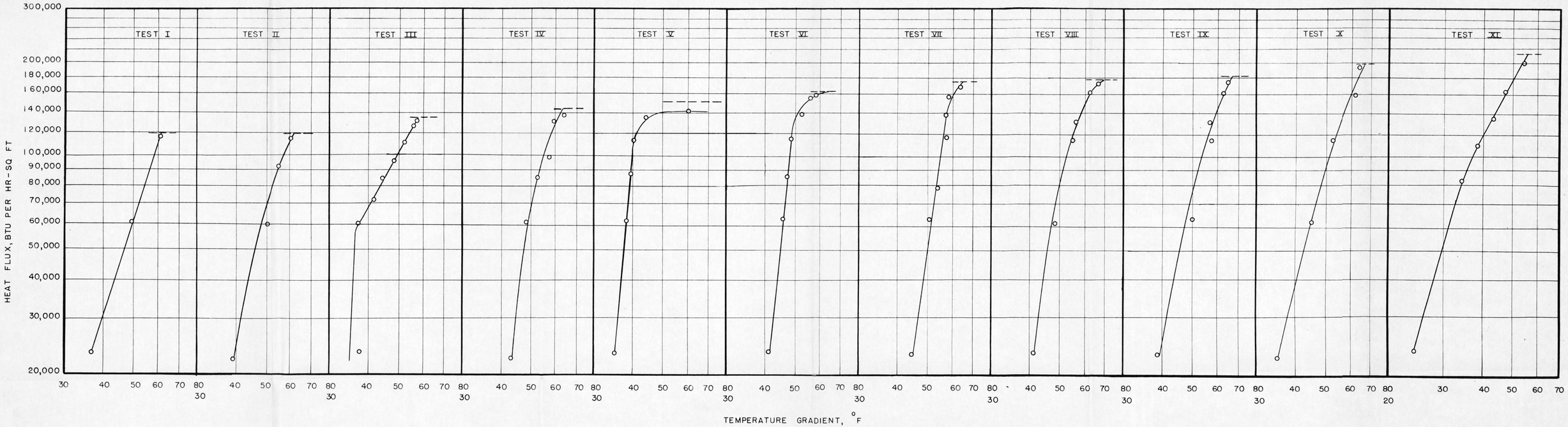

FIGURE 18

HEAT FLUX CURVES FOR ETHANOL AND BENZENE MIXTURES

The point on the chart indicated by small circles, represents steady state conditions of heat transfer which did not include film boiling. This chart was used to determine the critical temperature gradient, in °F, by extrapolation of the heat flux curve to the dotted line which represents the maximum heat flux, in Btu per hour-square foot.

Calculated Boiling Points for Mixtures of Ethanol and Benzene Boiling at 715.0 Millimeters Mercury Pressure. Table X presents the boiling points of ethanol-benzene mixtures which were calculated by the van Laar method outlined by Hougen and Watson[12]. The boiling points were calculated for comparison with the experimentally determined values presented in Table IX.

Boiling Point Diagram for Ethanol-Benzene Mixtures. Figure 19 presents the calculated boiling points of ethanol-benzene mixtures boiling at 715.0 millimeters mercury pressure. Figure 19 also presents the experimentally determined values as contained in Table IX. The solid line in Figure 19 represents the experimentally determined boiling points while the dotted line represents those obtained by calculation.

Summary of Results from Operation of Horizontal Plate Evaporator. Table XI presents the summary of results from the operation of the horizontal plate evaporator using ethanol-benzene mixtures as test liquids. The critical temperature gradients which were determined by graphical means in Figure 18 are presented, in °F.

TABLE X

Calculated Boiling Points for Mixtures of Ethanol and Benzene Boiling at 715 Millimeters of Mercury Pressure

Composition of Liquid		Calculated Boiling Point
Ethanol Mole per cent	Benzene mole per cent	°F
0	100	173.5
10	90	158.0
30	70	151.0
50	50	149.0
70	30	152.0
90	10	161.5
100	0	170.0

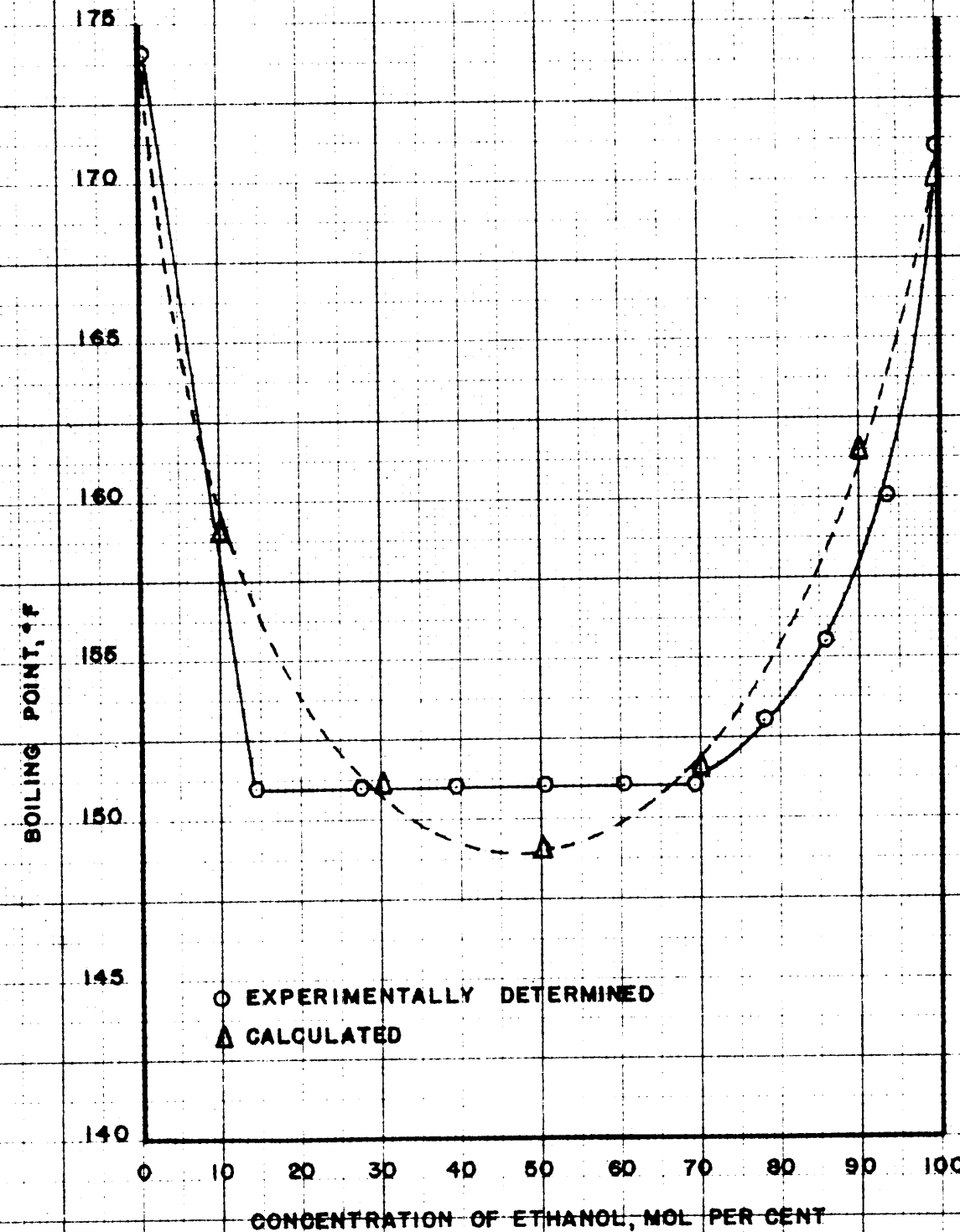

FIGURE 19

BOILING POINT DIAGRAM FOR ETHANOL-BENZENE MIXTURES*

*PRESSURE 715 MM MERCURY

TABLE XI

Summary of Results from the Operation of Horizontal Plate Evaporator

Using Ethanol-Benzene Mixtures as Test Liquids

Test No.	Critical Temperature Gradient °F	Maximum Heat Flux Btu per hr-sq ft	Maximum Film Coefficient Btu per hr-sq ft-°F
I	61.0	119,252	1,955
II	63.0	119,252	1,893
III	--	138,395	--
IV	64.0	144,170	2,253
V	--	151,805	--
VI	65.0	161,192	2,479
VII	65.0	172,611	2,656
VIII	69.0	178,831	2,592
IX	67.0	180,695	2,697
X	67.0	199,168	2,973
XI	56.5	217,873	3,856

The maximum heat flux is expressed in terms of the unit rate of heat transfer, in Btu per hour-square foot, which caused the boiling liquid to shift from a state of nucleate to film boiling. The maximum film coefficient of heat transfer, Btu per hour-square foot-°F, was evaluated from the maximum heat flux and critical temperature gradient.

Effect of Concentration on Maximum Heat Flux of Ethanol-Benzene Mixtures. Figures 20 and 21 present the effect of increasing ethanol concentration on the maximum heat flux of ethanol-benzene mixtures. Figure 20 presents the effect of concentration as a plot of ethanol concentration, in volume per cent, versus the maximum heat flux, in Btu per hour-square foot. Figure 21 presents the effect of concentration as a plot of ethanol concentration, in mole per cent, versus maximum heat flux, in Btu per hour-square foot-°F. In both cases the ethanol concentration ranged from 0 to 100 per cent.

Effect of Concentration on Critical Temperature Gradient of Ethanol-Benzene Mixtures. Figures 22 and 23 present the effect of increasing ethanol concentration on the critical temperature gradient of ethanol-benzene mixtures. Figure 22 presents the effect of concentration as a plot of ethanol concentration, in volume per cent, versus the critical temperature gradient, in °F. Figure 23 presents the effect of concentration as a plot of ethanol concentration, in mole per cent, versus the critical temperature gradient,

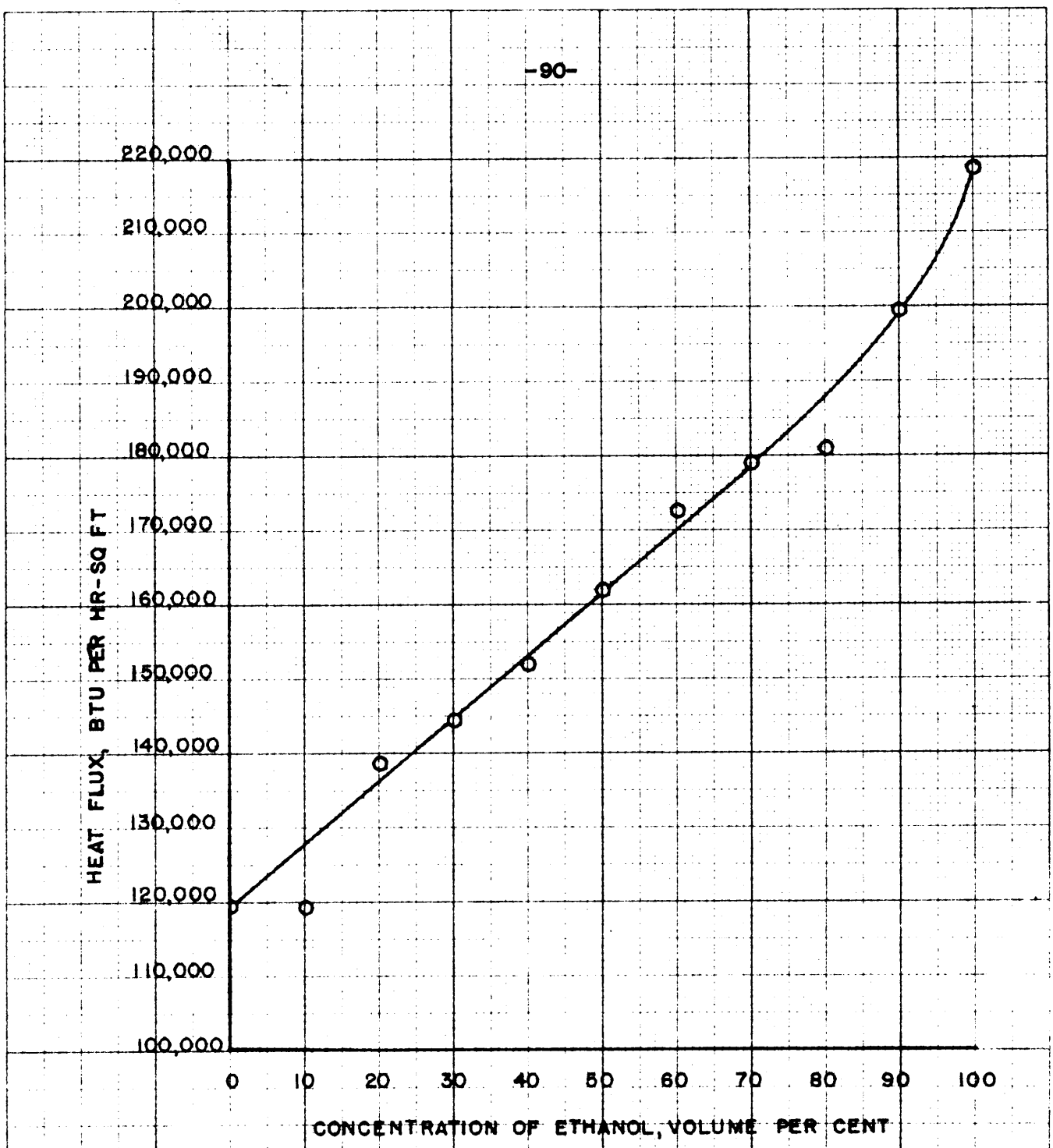

FIGURE 20

EFFECT OF CONCENTRATION ON MAXIMUM HEAT FLUX
OF
ETHANOL-BENZENE MIXTURES

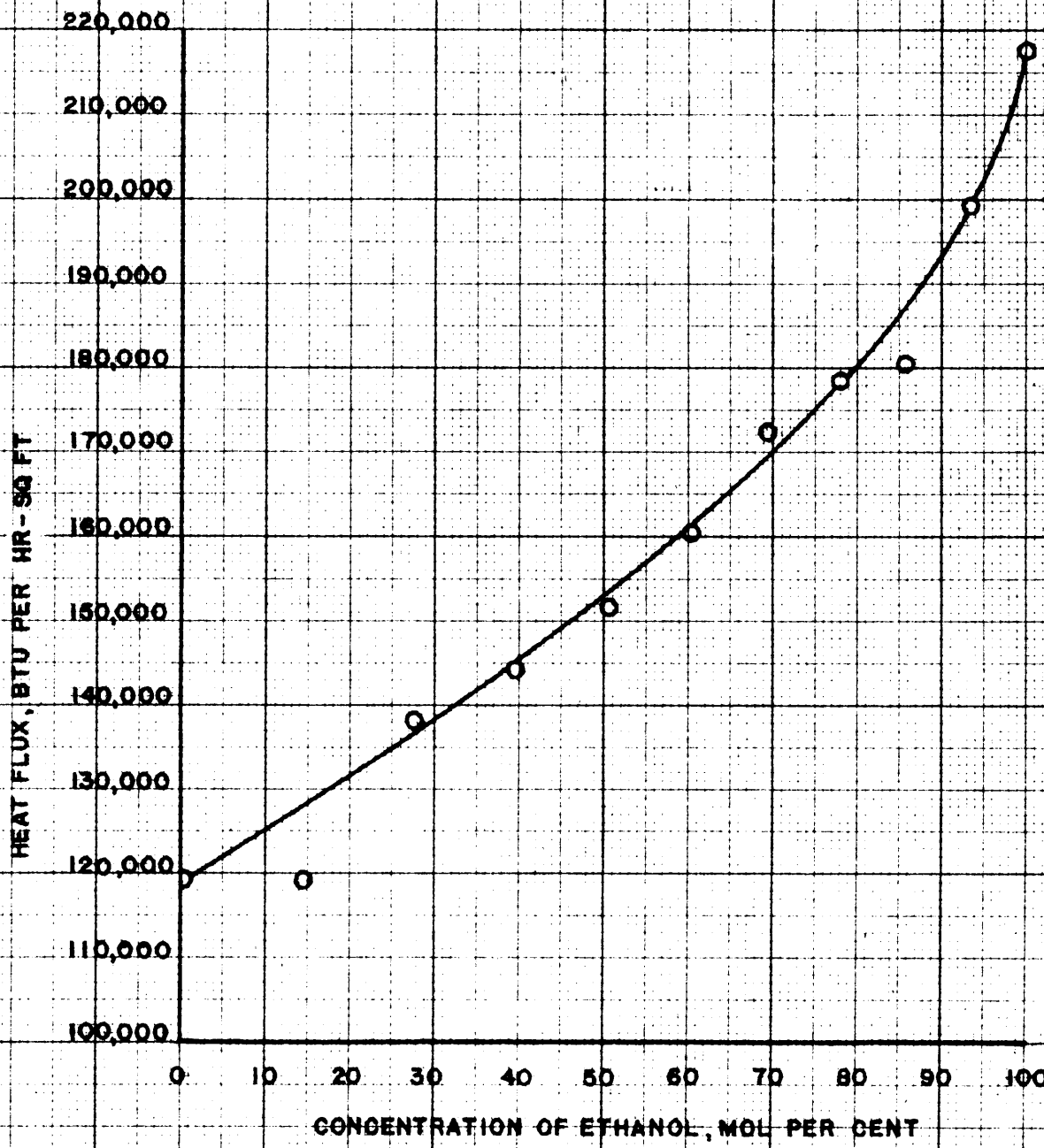

FIGURE 21

EFFECT OF CONCENTRATION ON MAXIMUM HEAT FLUX

OF

ETHANOL-BENZENE MIXTURES

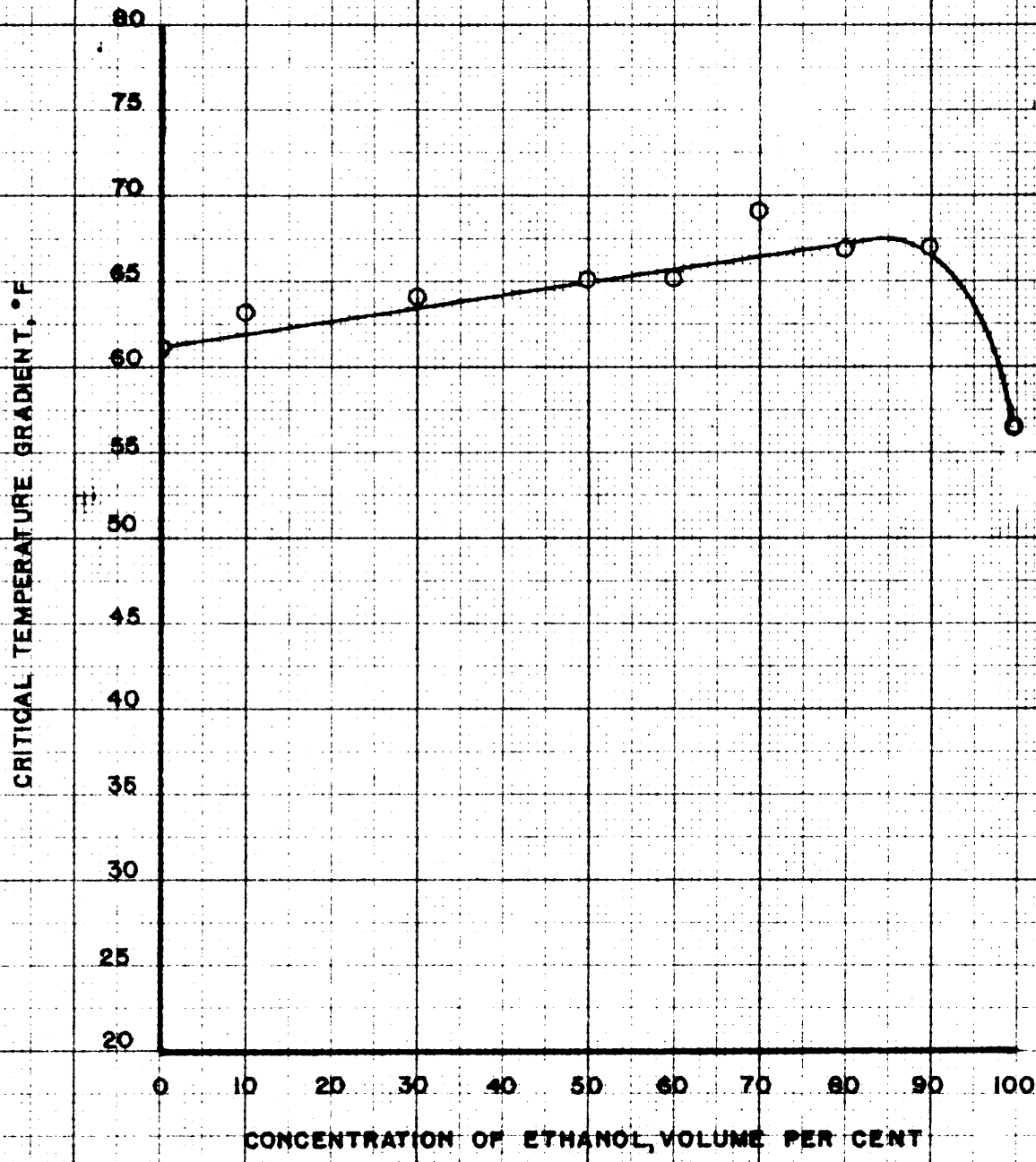

FIGURE 22

EFFECT OF CONCENTRATION ON CRITICAL TEMPERATURE GRADIENT OF ETHANOL—BENZENE MIXTURES

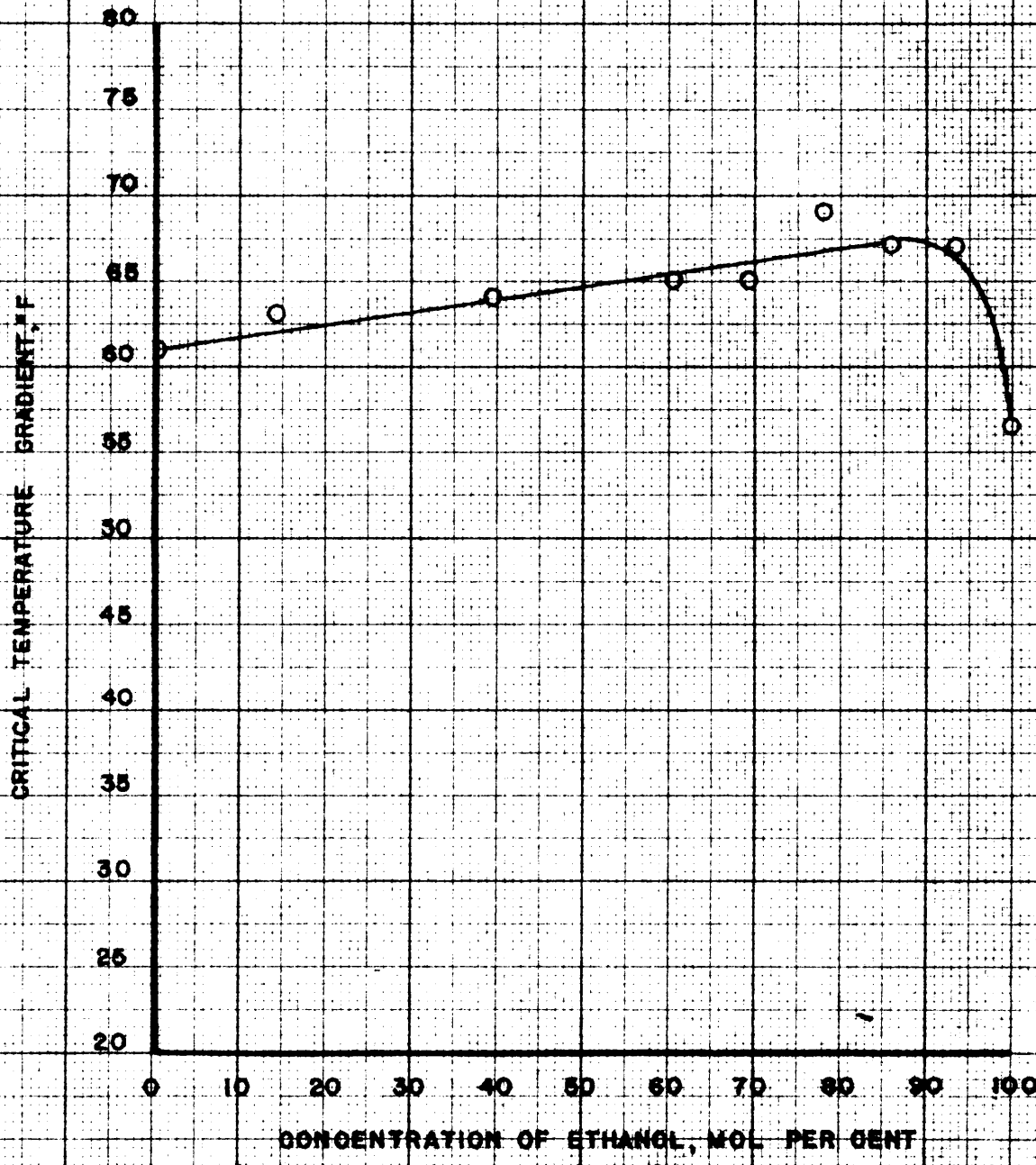

FIGURE 23

EFFECT OF CONCENTRATION ON CRITICAL TEMPERATURE GRADIENT OF ETHANOL—BENZENE MIXTURES

in °F. In both cases the ethanol concentration ranged from 0 to 100 per cent.

Effect of Concentration on Maximum Film Coefficient of Heat Transfer for Ethanol-Benzene Mixtures. Figures 24 and 25 present the effect of increasing ethanol concentration on the maximum film coefficient of heat transfer for ethanol-benzene mixtures. Figure 24 presents the effect of concentration as a plot of ethanol concentration, in volume per cent, versus maximum film coefficient of heat transfer, in Btu per hour-square foot-°F. Figure 25 presents the effect of concentration, in mole per cent, versus maximum film coefficient of heat transfer in Btu per hour-square foot-°F.

Observations on Experimental Tests. The silver heat transfer surface was clean and shiny prior to conducting any set of tests. At the end of each series of tests that included a state of film boiling the surface was still shiny but slightly brown in color. In preliminary tests, including the evaluation of the heat losses of the apparatus in which film boiling was not attained no fouling or discoloration was noted.

In tests V and VI the rate of ebullition in the evaporator body was far above normal. In the nucleate range the boiling action was so intense that foam or highly "fluidized" liquid filled the entire evaporator body and rose into the condenser space at

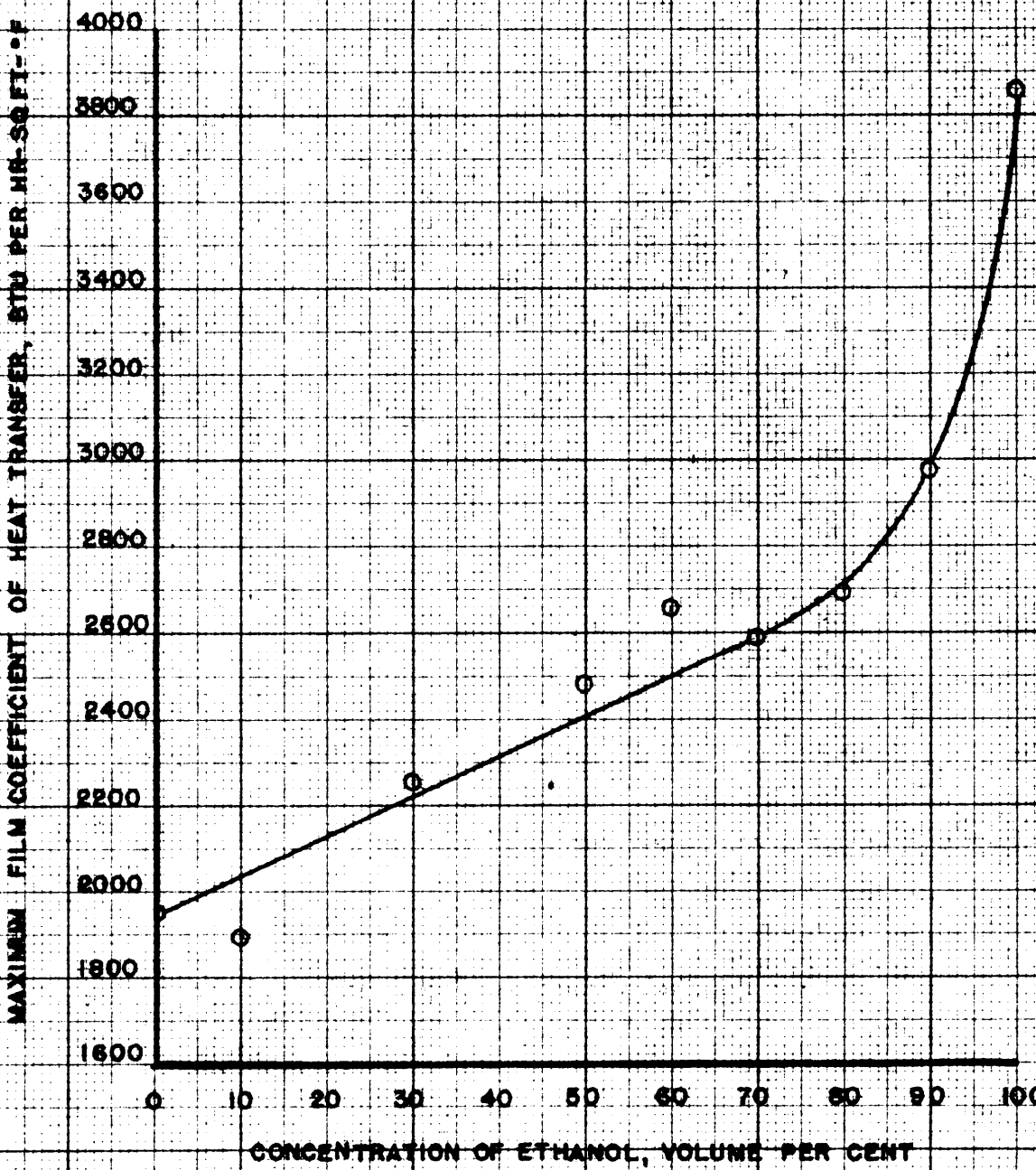

FIGURE 24

EFFECT OF CONCENTRATION ON MAXIMUM FILM COEFFICIENT OF HEAT TRANSFER FOR ETHANOL-BENZENE MIXTURES

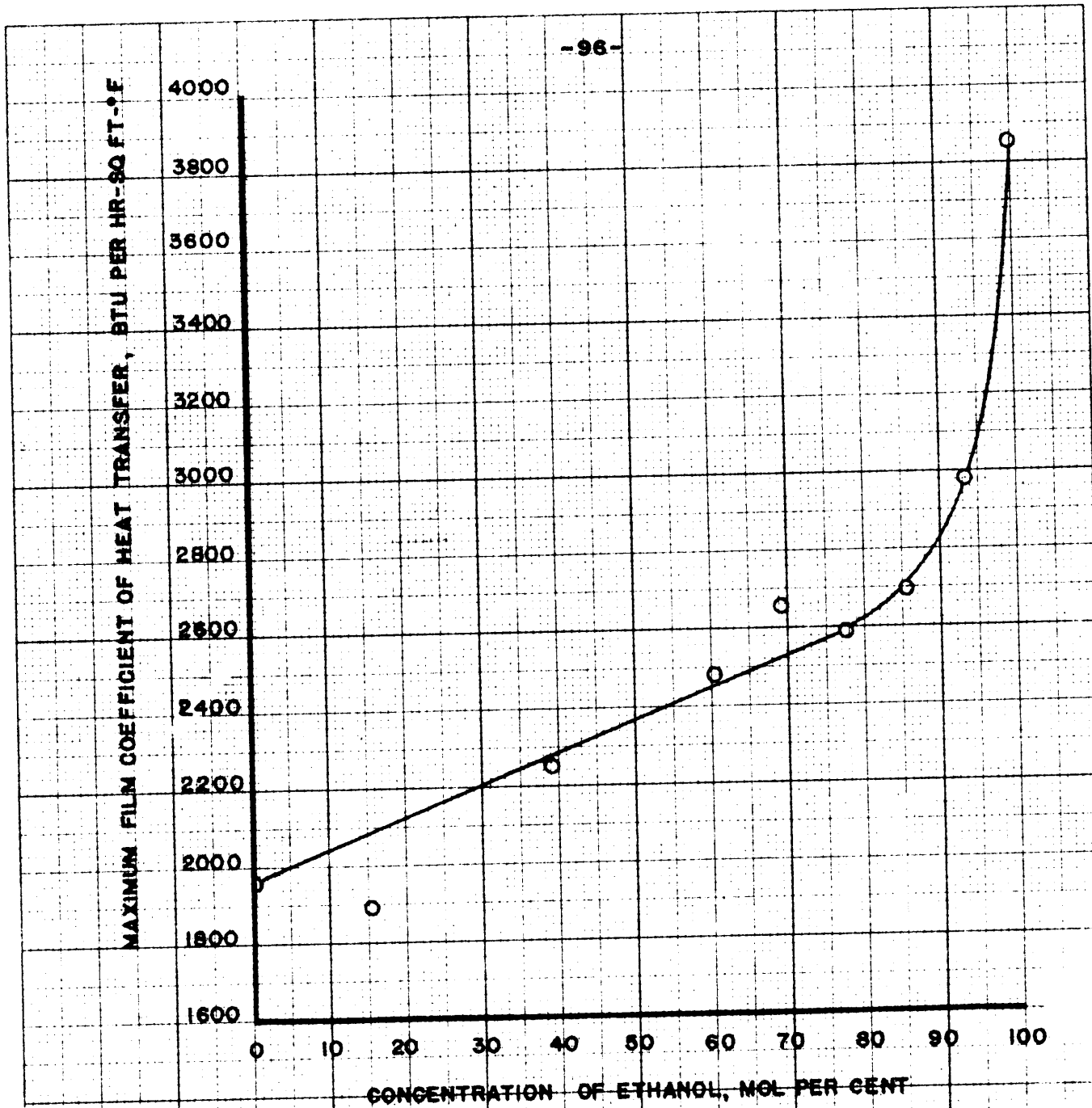

FIGURE 25

EFFECT OF CONCENTRATION ON MAXIMUM FILM COEFFICIENT OF HEAT TRANSFER FOR ETHANOL–BENZENE MIXTURES

incipient film boiling. The transition from nucleate boiling to film boiling, for tests V and VI, was gradual and required about two minutes as compared to the normal rapid transition of about fifteen seconds.

IV. DISCUSSION

The discussion of the results obtained in this investigation has been subdivided into several sections for clarity. The subdivision entitled Discussion of Results accentuates many points which are thought to have affected the results. Emphasis is placed on the techniques employed in the operation of the equipment and in the analysis of the data. The section entitled Limitations outlines, in concise form, the restrictions of the study. The subdivision entitled Recommendations includes suggestions regarding proposed equipment modification and future work. The section entitled Sample Calculations presents a representative sample of the calculations performed in the evaluation of the results.

Discussion of Results

The following paragraphs present and discuss factors which are believed to have affected the results. Also included are several comparisons of the results of this investigation with those reported by previous investigators.

Equipment. The equipment designed and constructed for this investigation fulfilled its purpose satisfactorily. The equipment was operated over one hundred hours without serious failures. How-

ever, there were several features in the design of the apparatus which were found to be undesirable. These features have been considered individually as follows:

Evaporator Body and Condenser. The pyrex pipe which served as the evaporator body and inner tube of the water cooled condenser caused considerable trouble. The difficulties which were encountered in the use of the pipe were twofold, namely, (1) to obtain a liquid tight seal between the pyrex pipe and the heating surface without cracking the pipe, and (2) to maintain a liquid tight seal between the pyrex pipe and the copper condenser shell.

Heater. The electrical heater, which supplied the heat for transfer to the boiling liquid, incorporated several undesirable features which may be traced directly to the original design of the equipment. The principal difficulty lay in the electrical insulation of the resistance wire. While the heater was operated approximately one hundred hours without an overhauling, one electrical failure was experienced. One of the fixed wattage heating circuits short circuited to the heating fins which necessitated the isolation of this unit from the control board. Fortunately the heat requirements for the remaining tests could be supplied with the remaining two heating circuits and an overhaul of the unit was not required.

The second principal objection to the heater unit was that an appreciable temperature gradient existed across the heater plate for values of heat flux over 20,000 Btu per hour-square foot-°F. Figure 27 in the Appendix shows the uncorrected temperature distribution curves for the eleven tests. The plate temperature was lower at the center of the plate than at the edge of the heater transfer area. The thermocouple placed equidistant between the center of the plate and the periphery of the circle inscribing the intersection of the heater fins with the plate registered temperatures which were between the other two.

The gradient was believed to have been caused by the fact that the circular area in which the heat was transferred to the heater plate was over twice as large as the area available for heat transfer from the plate to the boiling liquid. Since copper has an approximate thermal conductivity of 224 Btu per hour-square foot-°F, it is reasonable to assume that the film between the silver surface and the liquid controlled the rate of heat transfer from the heater plate to the liquid. If such an assumption is justifiable the sector of the heater plate not in contact with the boiling liquid would tend to equal the temperature of the heater. Such a theory would also explain the fact that the

temperature gradient across the plate increased with heat flux because the heater temperature is necessarily increased to induce greater heat flux.

It is believed that the problem of lateral temperature gradient in the heating plate could have been eliminated by constructing the heater fins such that their contact area with the heater plates would have been equal to the area available for heat transfer to the boiling liquid. Such a proposed construction was impossible due to the type and geometrical arrangement employed in the resistance windings.

Accuracy of Thermocouple Circuit. The accuracy of the thermocouple electromotive force measuring circuit was determined to be approximately 1/2° F. This accuracy was determined by the balancing of the potentiometer bridge and then moving the millivolt indicator until a state of unbalance was attained. No disturbance of the balanced potentiometer bridge could be detected for changes less than 0.02 millivolt which is roughly equal to 1/2° F. The accuracy of the instrument was not improved by approaching the balanced condition from two directions.

Boiling Point Diagrams for Mixtures of Ethanol and Benzene Boiling at 715.0 Millimeters of Mercury Pressure. Figure 17 presents the boiling point curves for mixtures of ethanol and benzene as determined experimentally and as calculated by the method of van Laar constants[12]. An examination of the two curves shows that they differ at some points as much as 2° F. The calculated curve shows a single minimum boiling point of 148° F between 40 and 50 mole per cent ethanol; while the experimental curve indicates that all mixtures between 14.4 and 69.5 mole per cent ethanol boil at the minimum temperature of 151° F. The differences in the two curves cannot logically be attributed to the technique employed in temperature measurement since the temperature measuring circuit was shown to be accurate within 1/2° F. It is believed that the shape of the experimental curve was altered due to the presence of water in the test liquids.

The water is believed to have been present in test liquids on preparation and not introduced into the liquid during the boiling operation. The possibility of air, laden with water vapor, diffusing into the vapor space is improbable since only a small air vent was permitted in the cork seal at the top of the column.

Gonzalez[31] analyzed samples of benzene and ethanol from the same lots as the samples used in this investigation. He reported that the benzene contained approximately 0.8 volume per cent water

and that 95 per cent ethanol though dried with quicklime could contain as high as two per cent water. The effect of the third component, water, on the boiling point diagram of ethanol and benzene is not known but it is reasonable that the deviations between the true and theoretical curves presented in Figure 17 may have been due to the presence of water.

Correction of Temperature Gradient. The temperature drop correction applied to the total temperature drop from the plate thermocouples to the main body of liquid is subject to considerable question. These corrections varied from less than two per cent at heat fluxes of 23,000 Btu per hour-square foot to approximately ten per cent in the range of incipient film boiling. While the plate thermocouples were placed within 0.128 inch of the heating surface they should have been placed within 1/16 inch of the surface. Such a placement would have reduced the correction factor to a maximum of five per cent.

The correction factors are believed to have introduced errors in the evaluation of the temperature gradients and consequently in the evaluation of the critical temperature gradients. The uncorrected temperature gradients invariably increased with an increase in the heat flux. However, after the correction factors were applied several exceptions to this generality were found to exist. The plots of the heat flux curves

for tests III, VII, and IX, in Table IX, contain examples of these irregularities, that is, the corrected temperature gradient decreased from one to two degrees with increases in heat flux from 20,000 to 40,000 Btu per hour-square foot. The error is believed to have been caused by the basic assumption that the area for which the temperature correction was applied was equal to the area available for heat transfer to the boiling liquid. While this assumption is not strictly true it becomes applicable for all practical purposes if the thermocouples are close enough to the heat transfer surface. It is believed that errors in the temperature gradient evaluations could have been eliminated by placing the plate thermocouples within 1/16 inch of the silver heat transfer surface.

Heat Flux Curves for Ethanol and Benzene Mixtures. Figure 18 presents the heat flux curves for the eleven ethanol-benzene mixtures studied. The concentration was varied, in 10 mole per cent increments, from 100 per cent benzene in test I to 100 per cent ethanol in test XI. The heat flux curves were plotted using logarithmic coordinates for two principal reasons, namely, (1) the relationships between heat flux and thermal driving force is made linear on the range of nucleate boiling, and (2) the extrapolation of the curve to maximum heat flux for the determination of the

critical temperature gradient is believed to be more accurate than on a regular scale.

The values of the heat flux in Figure 18 are the results of evaluated energy inputs to the electrical heater. The temperature gradients were evaluated by subtracting the temperature of the main body of liquid from the corrected plate temperature.

Temperature Gradients in the Liquid Space. The so-called temperature of the main body of liquid was evaluated from the temperature traverse which was made in the liquid space for each steady state condition. Superheating of the liquid within 3/8 inch of the silver surface was found to be common for values of heat flux below 30,000 Btu per hour-square foot. The degree of superheating was found to be as high as 2° F for some steady state conditions. When superheating was found to exist, the temperature of the main body of liquid was considered to be the temperature of the non-superheated liquid. While this consideration may be subject to criticism, it was the only satisfactory standard method which could be devised for evaluating the liquid temperature. Superheating of the liquid was uncommon for values of heat flux above 30,000 Btu per hour-square foot. This is believed due to the fact that above this heat flux the liquid was usually in a state of

sufficient agitation to eliminate temperature gradients. It is also quite likely that the liquid was uniformly superheated less than 1/2° F since this was considered the limit of accuracy of the temperature measuring circuit.

Determination of the Critical Temperature Gradient.

The heat flux curves in Figure 18 represent the relationship which existed between the rate of heat transfer and the temperature gradient for increasing values of heat flux. The dotted lines shown in the chart represent the maximum rate of heat transfer or the heat flux which caused the system to shift from a state of nucleate to film boiling. When this value of maximum heat flux was attained during the operation of the equipment, conditions shifted so rapidly that it was impossible to determine the critical temperature gradient. This gradient was determined graphically by extrapolating the heat flux curve to the value of maximum heat flux and calling this point of intersection the critical temperature gradient. Undoubtedly this method incorporates error but it was the only method which could be devised for estimating the critical thermal driving force. The principal disadvantage is the fact that the critical temperature gradient is largely dependent on the way the heat flux curve is drawn.

The heat flux curves shown in Figure 18 were drawn as an average or a locus through the points rather than as a point-to-point curve. This was done in order to attempt to isolate the effect of concentration on the critical driving force.

Effect of Concentration on the Shape of Heat Flux Curves. The heat flux curves for the two pure liquids, ethanol and benzene were essentially straight lines between values of heat flux of 23,000 Btu per hour-square foot and the critical point. However, for the mixtures of ethanol and benzene the heat flux plots tended to curve near the critical point, that is, the approach to film boiling was not linear as in the case of the pure liquids. This change in the heat flux curves can be explained to some extent by the physical action of the boiling liquid.

The pure liquids, ethanol and benzene, boiled in the higher ranges of heat flux rather vigorously. The original cold liquid height of 4-1/2 inches was expanded to 6 or 7 inches. When the proper value of heat flux was applied to the system the transition to film boiling was practically instantaneous. Once film boiling was attained, the ebullition was slight and bubbles of vapor large in size compared to those evolved in nucleate boiling.

The boiling characteristics of the ethanol-benzene mixtures were entirely different from those of the pure compounds. Nucleate boiling, in the range of high flux, was accompanied by extremely vigorous boiling. The original cold liquid height of 4-1/2 inches was expanded to over 12 inches. The liquid was in a highly "fluidized" state with slugs of vapor surging up through the mixture of bubbles and liquid. When the proper value of heat flux was applied to the system, the transition to film boiling was gradual and required several minutes for completion. Once the state of film boiling was attained, the liquid was not boiling in a quiet manner as was characteristic of the pure liquids but boiled with considerable ebullition. The liquid height was approximately six inches during the film boiling period for the mixtures.

Irregularities in the Heat Flux Curves. Test V in Figure 18 shows a heat flux curve which could not be extrapolated to the critical temperature gradient. It was in this test that the most pronounced transition from nucleate to film boiling was observed. The plot of the heat flux-critical temperature gradient relationship as drawn shows the curve becoming parallel to the value of maximum heat flux. Such a condition could not exist but the available data did

warrant drawing the curve any other way. It is believed that the heat flux plot should have intersected the dotted line representing the maximum heat flux. It is also believed that it would have been extremely difficult to determine the true nature of the plot in the range of incipient film boiling because a small change in heat flux resulted in such a large change in the temperature.

The critical temperature gradient obtained for test III was not considered valid. The plot representing the results from the test was not straight in the range of nucleate boiling as were the other ten heat flux curves. Two conditions of heat transfer seemed to have existed during the nucleate range.

No explanation can be given for the irregularity encountered in test III. The data and observations substantiate no theories as to the break in the heat flux curve.

The Characteristics of Heat Transfer for Ethanol and Benzene.

The following characteristics of heat transfer were determined for benzene: maximum heat flux 119,252 Btu per hour-square foot, critical temperature gradient 61.0° F, maximum film coefficient of heat transfer 1,955 Btu per hour-square foot-°F. Cichelli and Bonilla[7] conducted an investigation employing an

apparatus similar to the one used in this investigation. They reported the heat transfer characteristics of benzene to be as follows: maximum heat flux 125,000 Btu per hour-square foot, critical temperature gradient 81° F, maximum film coefficient of heat transfer 1,545 Btu per hour-square foot-°F.

The values of maximum heat flux for benzene agree within five per cent with the value obtained by Cichelli and Bonilla[7]. The values of critical temperature gradient and maximum film coefficient of heat transfer deviate by 25 and 21 per cent, respectively. These deviations may be attributed to the fact that Cichelli and Bonilla used a chromium heat transfer surface while a silver surface was employed in this investigation. It is probable that silver furnishes a better heat transfer surface than chromium. If such is true less thermal driving force would be required for the same heat flux.

The following characteristics of heat transfer were determined for ethanol: maximum heat flux 217,873 Btu per hour-square foot, critical temperature gradient 56.6° F, maximum film coefficient of heat transfer 3,856 Btu per hour-square foot-°F. The following characteristics of heat transfer to ethanol are an average of three values reported by Bonilla and Perry[4]: maximum heat flux 163,333 Btu per hour-square foot, critical temperature

gradient 66.3° F, maximum film coefficient of heat transfer 2,462 Btu per hour-square foot-°F. The values of maximum heat flux obtained in this study deviate from those reported by Bonilla and Perry by approximately 25 per cent. It is probable that this deviation was caused by the presence of from one to two per cent water in the test liquid employed in this investigation. The deviation of 16 per cent between the critical temperature gradients obtained in the two investigations may be attributable to one of two things. The deviation may have been caused by the presence of water in the test liquid, the difference in heat transfer surfaces, or a combination of the two. It is interesting to note that the critical temperature gradient for benzene was greater than the gradient for ethanol in both investigations.

Correlation of the Characteristics of Heat Transfer with Reduced Pressure. The reduced pressures of benzene and ethanol at 715 millimeters of mercury pressure were calculated to be 0.01975 and 0.0149, respectively. These values were then applied to the correlations proposed by Cichelli and Bonilla[7] which appear in Figures 3 and 7. The correlation predicted the maximum heat flux of benzene to be 131,000 Btu per hour-square foot and the critical temperature gradient to be 70° F. A maximum heat flux of 159,000 Btu per hour-square foot and a critical temperature gradient of 76.2° F were predicted for

ethanol. A comparison of these values with the values obtained in this investigation reveals that the correlation applies well to benzene but deviates excessively for ethanol.

Effect of Concentration on Maximum Heat Flux of Ethanol-Benzene Mixtures. Figures 20 and 21 contain plots which show the effect of concentration on the maximum heat flux for mixtures of ethanol and benzene ranging from 0 to 100 per cent ethanol. Figure 20 shows the effect of concentration as expressed in volume per cent ethanol in the ethanol-benzene mixture. The maximum deviation of the data from the curve as drawn was found to be approximately six per cent. Figure 21 shows the effect of concentration as expressed in mole per cent ethanol in the ethanol-benzene mixture. The maximum deviation of the data was found to be approximately seven per cent.

Effect of Concentration on Critical Temperature Gradient of Ethanol-Benzene Mixtures. Figures 22 and 23 show the effect of concentration on the critical temperature gradient of mixtures of ethanol and benzene ranging from 0 to 100 per cent ethanol. Figure 22 shows the effect of concentration in terms of volume per cent. The maximum deviation of the data was found to be approximately five per cent. The curve is linear between ethanol concentrations of 0 and 85 volume per cent. At this point the plot

drops off sharply to the value of 56.5° F which is the critical temperature gradient for ethanol. This plot indicates that the presence of benzene, even in small quantities, in ethanol alters the critical temperature gradient considerably. No significant difference can be detected between Figures 22 and 23 which indicates that the correlation of critical temperature gradient with concentration is not improved by expressing the concentration in mole per cent.

Effect of Concentration on the Maximum Film Coefficient of Heat Transfer. Figures 24 and 25 show the effect of concentration on the maximum film coefficient of heat transfer of ethanol-benzene mixtures. These plots were derived from the plots of maximum heat flux and critical temperature gradient shown in Figures 20, 21, 22, 23, 24, and 25. The only significant point is that the maximum film coefficient for pure ethanol is seriously lowered by addition of a small amount of benzene. According to Figure 25 the addition of five mole per cent benzene to ethanol will lower its maximum film coefficient from 3,856 to 3,100 Btu per hour-square foot-°F.

Fouling of the Silver Heat Transfer Surface. It is believed that the silver surface was fouled due to the presence of impurities in the technical grade benzene. In as much as no fouling was noticed unless film boiling was attained it is proposed that

the temperature of the silver surface was too low during nucleate boiling to cause noticeable reaction between the impurities and the silver. However, during a state of film boiling plate temperatures around 400° F were sufficient to cause the impurities to foul the silver surface.

Operation of the Equipment. The operation of the apparatus was not difficult but required the complete attention of the operator. The principal difficulty encountered was in obtaining a liquid tight seal between the pyrex evaporator body and silver heating surface. Another difficulty was maintaining the thermocouples in the proper relative position in the liquid and vapor space. In instances involving high boiling rates the thermocouples tended to sway and lie against the inner wall of the evaporator.

Source of Error. The principal source of error is believed to have been in the method of determining the maximum heat flux. It was very difficult to estimate the proper increments of heat flux to avoid overstepping the maximum heat flux. Unless each determination was repeated there is no reliable method for estimating the proper increments of heat flux. During the preliminary tests the reproducibility of the heat flux curve was to check using methanol as a test liquid. It

was found that the heat flux curve could be reproduced within five per cent. No checks were made on the reproducibility of the maximum heat flux.

Recommendations

On the basis of the results obtained in this investigation the following recommendations are offered:

Equipment Modification. The replacement of the pyrex pipe, which served as the inner tube of the condenser, with a copper tube which could be welded to the copper condenser shell would eliminate several difficulties. Such an arrangement would eliminate the possibility of leaks, provide a more efficient condenser, and eliminate the possibility of breakage due to tensile stress in the column. The evaporator body could be made of pyrex pipe and flanged to the bottom end of the copper condenser tube. This would permit observation into the liquid space of the evaporator.

The movement of the plate thermocouples to within 1/16 inch of the silver heat transfer surface would decrease the possibility of error in the determination of the temperature gradient.

The construction of an electrical heater which would be free of danger from failures would eliminate considerable difficulty.

Such a heater could well be constructed from calarod heating units. Such units would provide a heater which would be both durable and safe.

Further Investigations. A further study of the effect of concentration on the maximum heat flux and critical temperature would be of interest. The purpose of such a study should be to attempt to formulate a method for predicting the characteristics of heat transfer for all binary mixtures of liquids.

It is further recommended that the effect of the surface tension of the liquid on the maximum heat flux and critical temperature gradient of various compounds and mixtures be studied.

Limitations

This investigation was conducted under the following limitating conditions:

Test Liquids. The test liquids were ethanol, benzene, and mixtures of ethanol and benzene. Samples of the test liquids were prepared for all concentrations of ethanol in benzene from 0 to 100 per cent in 10 volume per cent increments.

Evaporator. The evaporator used had an effective internal diameter of 1-31/32 inches. The cold liquid height of the test liquids in the evaporator was consistently 4-1/2 inches. The

heat transfer surface was a 0.003 inch electroplated coat of silver on copper.

Temperature Measurement. The heat plate temperature was measured at three points equally spaced across the radius of the heater plate. These thermocouples were imbedded in the copper plate to within 0.128 inch of the silver heat transfer surface. The liquid temperature was determined from a temperature traverse 0 to 4-1/2 inches made in the liquid space.

Pressure. The investigation was conducted at normal atmospheric pressure which ranged from 710.0 to 718.0 millimeters of mercury pressure.

Cooling Water Rate. The condenser cooling water rate was varied from 0 to 3.75 pounds per minute depending on the temperature of the reflux liquid.

Sample Calculations

The following sample calculations are representative of those used in this investigation:

Derivation and Application of Holman's Equation for the Heater Plate and Liquid-Vapor Space Thermocouples. The results of the calibration of the heater plate and liquid-vapor space thermocouples were applied to Holman's Equation[32]. This equation is known to apply to copper-constantan thermocouples. The general form of the equation is as follows:

$$e = m\,t^{n}$$

OR

$$\log e = \log m + n\,(\log t)$$

where:

e = thermocouple electromotive force, mv

t = temperature of thermocouple junction, °C

m and n are constants.

The constants were derived by solving simultaneous equations for conditions at the steam point and the 100° F point. The derivation was as follows:

(A) at 100° F point $\log e_A = \log m + n \log t_A$

(B) at steam point $\log e_B = \log m + n \log t_B$

Or

(A) $\log 1.50 = \log m + n \log 37.78$

(B) $\log 4.20 = \log m + n \log 98.46$

Solving simultaneously gives:

$$n = 1.075$$

$$m = 0.030235$$

Holman's equation as applied to the heater plate and liquid-vapor space thermocouples is as follows:

$$e = 0.030235\, t^{1.075}$$

The calibration curve for the heater plate and liquid-vapor space thermocouples presented in Figure 16 was then calculated from this equation.

Calculation of the Unit Heat Flux. The heat flux was calculated as follows:

Calculation of the Total Input Wattage. The total input wattage was calculated as follows:

$$W = E_1 I_1 + E_2 I_2 + E_3 I_3$$

where:

W = input energy to heater, watts

E = voltage applied to resistance wire, volts

I = current through resistance wire, amperes

1, 2, designate fixed capacity circuits

3 designates variable capacity circuit.

For test I, steady state 3:

$$W = (0)(0)+(0)(0)+(187)(4.0)$$

$$W = 748 \text{ watts}$$

Evaluation of Heat Losses. The heat losses were evaluated by the following expression:

$$W_L = EI$$

where:

W_L = input energy to heater, watts

E = voltage applied to resistance wire, volts

I = current through resistance wire, amperes

$$W_L = (47)(1.0)$$

$$W_L = 47 \text{ watts}$$

Calculation of Energy Transferred to Liquid. The energy transferred to the liquid was calculated as follows:

$$W_T = W_I - W_L$$

where:

W_T = energy transferred to liquid, watts

W_I = total energy into heater, watts

W_L = energy lost, watts.

$$W_T = 748 - 47$$

$$W_T = 701 \text{ watts}$$

Calculation of Heat Transfer Area. The area available from heat transfer from the silver surface to the boiling liquid was calculated as follows:

$$A = \frac{\pi d^2}{4}$$

where:

A = heat transfer area, sq in.

d = diameter of heat transfer surface, in.

$$A = \frac{3.1416 (1.953)^2}{4}$$

$$A = 2.996 \text{ or } 3.00 \text{ sq in.}$$

Determination of Conversion Factor for Converting from Watts to Btu Per Hour-Square Foot. The conversion factor for converting from watts to Btu per hour-square foot was derived as follows:

$$\frac{\text{Btu}}{\text{hr-sq ft}} = \frac{\text{Btu/hr}}{\text{watt}} \times \frac{\frac{\text{sq in.}}{\text{sq ft}}}{A} \times E_T$$

where:

A = heat transfer area, sq in.

E_T = energy transferred, watts.

$$\frac{\text{Btu}}{\text{hr-sq ft}} = 3.413 \left(\frac{144}{3.0}\right) E_T$$

$$\frac{\text{Btu}}{\text{hr-sq ft}} = 163.83\ E_T$$

Evaluation of Heat Flux. The unit rate of heat transfer from heating surface to the boiling liquid was calculated as follows:

$$\frac{Q}{A} = 163.83\ E_T$$

where:

Q = rate of heat transfer, Btu/hr

A = heat transfer area, sq ft

E_T = energy transferred, watts.

$$\frac{Q}{A} = 163.83\ (701)$$

$$\frac{Q}{A} = 114,845\ \text{Btu/hr-sq ft}$$

Determination of the Temperature Gradient. The temperature gradient from the silver surface to the main body of liquid was determined as follows:

Determination of Mean Uncorrected Plate Temperature. The values of electromotive force indicated by the plate thermocouples for test 1, steady state 3, were applied to the thermocouples calibration curve (Figure 16) and the corresponding temperatures obtained. These values of temperature were plotted as a function of their position in the heater plate as shown in Figure 26.

The first step toward determining the mean uncorrected plate temperature was to measure the area under the temperature curve between the limits of radius equal to 0 and radius equal to 63/64 inch. The method for measuring the area was by counting the blocks. The mean temperature was then calculated as follows:

$$t_{p_m} = \frac{A_T}{r_{hs}}$$

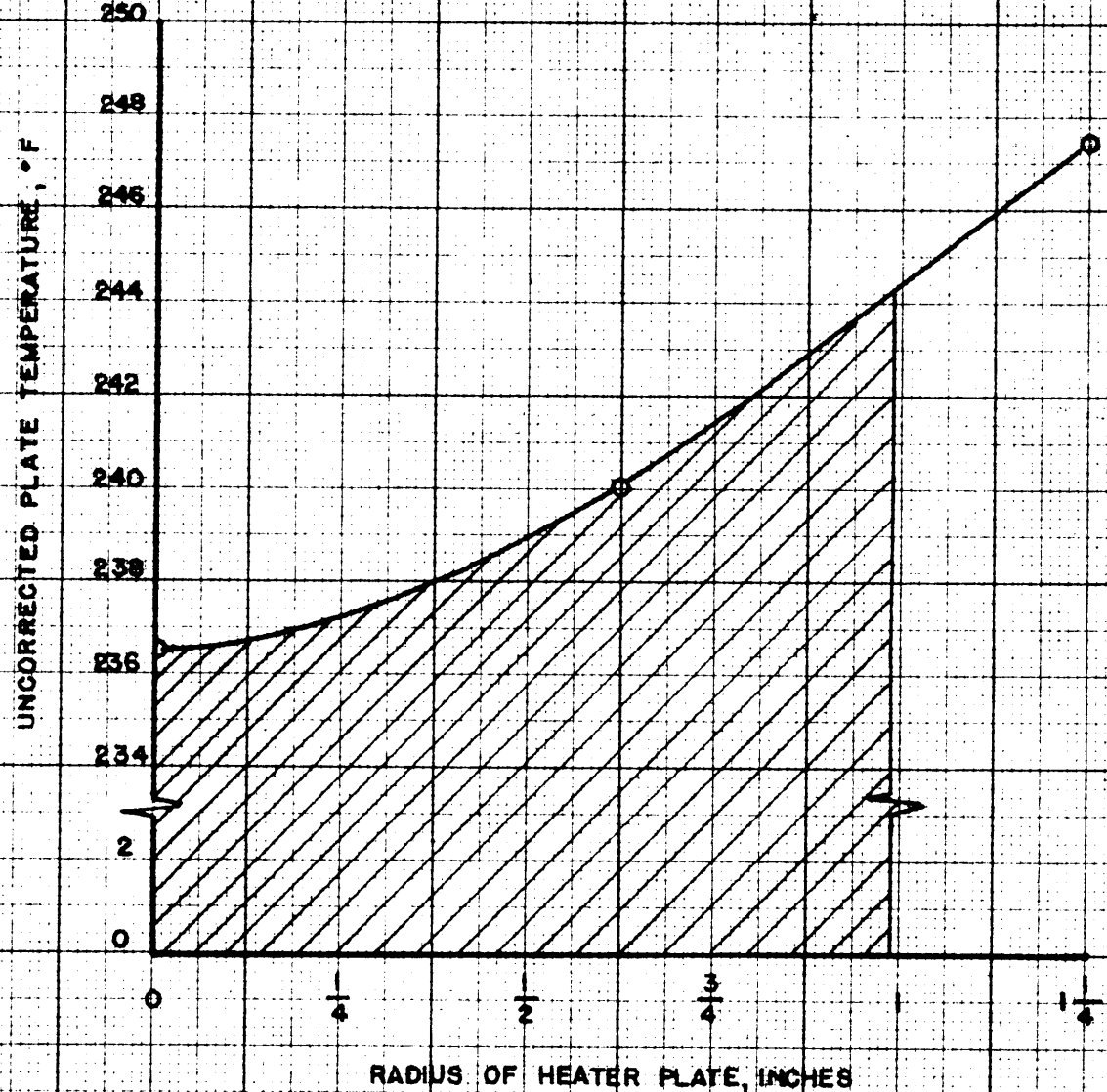

FIGURE 26

GRAPHICAL DETERMINATION OF MEAN UNCORRECTED PLATE TEMPERATURE

where:

t_{p_m} = mean uncorrected plate temperature, °F

A_T = area under curve, in.-°F

r_{hs} = radius of heating surface, in.

$$t_{p_m} = \frac{243.0}{0.985}$$

$$t_{p_m} = 239.5 \text{ °F}$$

Determination of Temperature Gradient Correction. The overall temperature gradient was corrected for the temperature drop from the thermocouple junctions to the heat transfer surface as follows:

$$\frac{Q}{A} = \frac{k(t_x - t_s)}{L}$$

where:

$\frac{Q}{A}$ = heat flux, Btu/hr-sq ft

k = thermal conductivity of material between thermocouple junction and heat transfer surface, Btu/hr-sq ft-°F

t_x = temperature of plate at thermocouple junction, °F

t_s = temperature of heat transfer surface, °F

L = distance from thermocouple junction to heat transfer surface, ft.

$$t_x - t_p = \frac{(114,845)(0.128)}{(222)\frac{12}{63/64}}$$

$$t_x - t_p = 5.5° F$$

Determination of Corrected Plate Temperature. The corrected plate temperature was determined by subtracting the correction factor from the uncorrected plate temperature as follows:

$$t_p = t_{uc} - (t_x - t_s)$$

where:

t_p = corrected plate temperature, °F

t_{uc} = uncorrected plate temperature, °F

$t_x - t_s$ = correction factor, °F.

$$t_p = 239.5 - 5.5$$

$$t_p = 234.0$$

Determination of Temperature Gradient. The temperature gradient between the heating surface and the main body of liquid was determined as follows:

$$t = t_p - t_l$$

where:

t = temperature gradient, °F

t_p = corrected plate temperature, °F

t_l = temperature of main body of liquid, °F.

$$t = 234.0 - 174.0$$

$$t = 60.0$$

Determination of the Maximum Film Coefficient of Heat Transfer. The maximum film coefficient of heat transfer was determined as follows:

$$h_{max} = \frac{\left(\frac{Q}{A}\right)_{max}}{t_{max}}$$

where:

h_{max} = maximum film coefficient of heat transfer, Btu/hr-sq ft-°F

$\left(\frac{Q}{A}\right)_{max}$ = maximum heat flux, Btu/hr-sq ft

$(t)_{max}$ = critical temperature gradient, °F

$$h_{max} = \frac{119,252}{61.0}$$

$$h_{max} = 1,955 \text{ Btu/hr-sq ft-°F}$$

V. CONCLUSIONS

An evaluation of the results from the determination of the characteristics of heat transfer from a horizontal silver surface to boiling mixtures of ethanol and benzene produced the following conclusions:

1. A state of film boiling was attained for ethanol, benzene and all mixtures of ethanol and benzene, varying from 0 to 100 per cent ethanol in increments of 10 volume per cent ethanol. The attainment of film boiling was evidenced by two facts, namely, (1) the rapid rise in the temperature gradient between the heat transfer surface and the main body of liquid, and (2) the decrease in the rate of boiling. The temperature gradients during film boiling ranged between 184.5 and 270.5° F. These temperatures have no significance other than evidence of film boiling.

2. The characteristics of heat transfer from the horizontal silver surface to boiling ethanol were: maximum heat flux, 217,873 Btu per hour-square foot, critical temperature gradient, 56.5° F, maximum film coefficient of heat transfer 1,955 Btu per hour-square foot-°F.

3. The characteristics of heat transfer from the horizontal silver surface to boiling benzene were: maximum heat flux, 119,252 Btu per hour-square foot, critical temperature gradient, 61.0° F,

maximum film coefficient of heat transfer, 1,955 Btu per hour-square foot-°F.

4. The values obtained for the maximum heat flux of mixtures of ethanol and benzene were between those obtained for the pure liquids. The relationship between maximum heat flux and ethanol concentration was found to be linear from 0 to 70 volume per cent ethanol. The terminal values of the straight line were 119,252 Btu per hour-square foot at 0 volume per cent ethanol and 178,831 Btu per hour-square foot at 70 volume per cent ethanol. The relationship between maximum heat flux and ethanol concentration was found to be a smooth curve from 70 to 100 volume per cent ethanol. The terminal values for this curve were 178,831 Btu per hour-square foot at 70 volume per cent and 217,873 Btu per hour-square foot at 100 volume per cent ethanol.

5. The values obtained for the critical driving force of temperature gradient for mixtures of ethanol and benzene were not consistently between the values obtained for the pure liquids. The maximum value for the critical temperature gradient was 67.5° F at an ethanol concentration of 85 volume per cent. The relationship between the critical driving force and ethanol concentration was found to be linear from 0 to 85 volume per cent ethanol. From the 85 volume per cent point the relationship followed a curve which fell sharply to 56.5° F at 100 volume per cent ethanol. A study of

the plot revealed that the presence of 5 to 15 volume per cent benzene in ethanol caused an appreciable change of 7 to 11° F in the critical driving force.

6. The values obtained for the maximum film coefficient of heat transfer for binary mixtures of ethanol and benzene fell between those obtained for the pure liquids. The relationship between the maximum film coefficient and ethanol concentration was found to be linear from 0 to 70 volume per cent ethanol. The terminal values of this straight line were 1,955 Btu per hour-square foot-°F at 0 volume per cent ethanol to 2,592 Btu per hour-square foot-°F at 70 volume per cent ethanol. The relationship between the maximum film coefficient and ethanol concentration from 70 to 100 volume per cent ethanol was a smooth curve rising sharply to 3,856 Btu per hour-square foot-°F at 100 per cent ethanol. A study of the plot revealed that the presence of 5 to 10 volume per cent benzene in ethanol caused an appreciable reduction of 15 to 24 per cent in the maximum film coefficient of heat transfer.

VI. SUMMARY

Heat transfer to boiling liquids is of primary industrial importance. Surprisingly enough organized study of the variables which affect heat transfer to boiling liquids has been far less than proportional to the industrial usage of this type of heat transmission. Perhaps the least investigated phenomenon involved in heat transfer to boiling liquids is the so-called critical state or maximum in the rate of heat transfer-thermal driving force relationship. This maximum or peak is believed to be caused by a change in the type or method of heat transfer from the heating surface to the boiling liquid. The system is said to pass from a state of nucleate boiling through the maximum to a state of film boiling. The critical point is known to vary for various liquids yet no adequate correlations were found which would provide a prediction of the characteristics of heat transfer for binary mixtures of liquids. In as much as ethanol and benzene are used extensively in industry they were selected for use in this investigation. The purpose of this investigation was to determine the characteristics of heat transfer from a horizontal silver surface to boiling mixtures of ethanol and benzene.

A horizontal plate evaporator, with the necessary accessory equipment for measurement and control, was designed and constructed.

The test liquids were prepared for concentrations of ethanol in benzene from 0 to 100 per cent in 10 volume per cent increments. These test liquids were charged to the evaporator and the characteristics of heat transfer determined by a series of steady state conditions of heat transfer. The rate of heat transfer was evaluated from the wattage input to the electrical heating unit. The temperature gradient between the heating surface and the main body of liquids was determined by evaluating the readings of thermocouples placed in the liquid space and the heater plate.

The maximum rate of heat transfer was considered equivalent to the heat flux which caused the boiling system to shift through the maximum in the heat flux-temperature gradient relationship. The critical temperature gradient was obtained by an extrapolation of the heat flux-temperature gradient curve to maximum heat flux.

The general conditions for the tests were: heat transfer surface, silver; cold liquid height in evaporator, 4-1/2 inches; evaporator diameter, 1-31/32 inches. The tests were made at normal atmospheric pressure which varied from 710.0 to 718.0 millimeters of mercury pressure. Steady state conditions of heat transfer were established before data was taken.

Maximum heat transfer rates were determined for all mixtures of ethanol and benzene tested. The characteristics of heat transfer for benzene were found to be: maximum heat flux, 119,252 Btu per hour-square foot, critical temperature gradient 61.0° F, maximum film coefficient of heat transfer, 1,955 Btu per hour-square foot-°F. The characteristics of heat transfer for ethanol were found to be: maximum heat flux, 217,873 Btu per hour-square foot, critical temperature gradient, 56.5° F, maximum film coefficient of heat transfer, 3,856 Btu per hour-square foot-°F. The values obtained for the maximum heat flux and maximum film coefficient of heat transfer were between those obtained for the pure liquids. The values of critical driving force for mixtures of ethanol and benzene were above the values obtained for the pure liquids and reached a maximum of 67.5° F at an ethanol concentration of 80 volume per cent. The relationships between maximum heat flux, critical temperature gradient and ethanol concentration were linear from 0 to 70 volume per cent ethanol. The relationship between the critical temperature gradient and ethanol concentration was linear from 0 to 85 volume per cent ethanol. The attempts to correlate the characteristics of heat transfer for mixtures of ethanol and benzene with ethanol concentration by means of empirical equations were unsuccessful.

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VIII. ACKNOWLEDGEMENTS

The author wishes to express his sincere appreciation to Professor F. W. Bull of the Department of Chemical Engineering, Virginia Polytechnic Institute, for his invaluable encouragement and guidance during the entire course of this investigation.

Appreciation is expressed to Dr. F. C. Vilbrandt, Head of the Department of Chemical Engineering, Virginia Polytechnic Institute, and the members of the staff of the Department of Chemical Engineering, for their suggestions and helpful criticism.

The author also wishes to express his appreciation to Professor J. F. Ryman of the Department of Physics, Virginia Polytechnic Institute, for his suggestions relating to the design and construction of the temperature measurement and control circuits used in this study.

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X. APPENDIX

This section has been included for the purpose of presenting material which was not believed pertinent in the previous sections. This material was included in hopes of presenting a clearer picture of the results to any subsequent investigator.

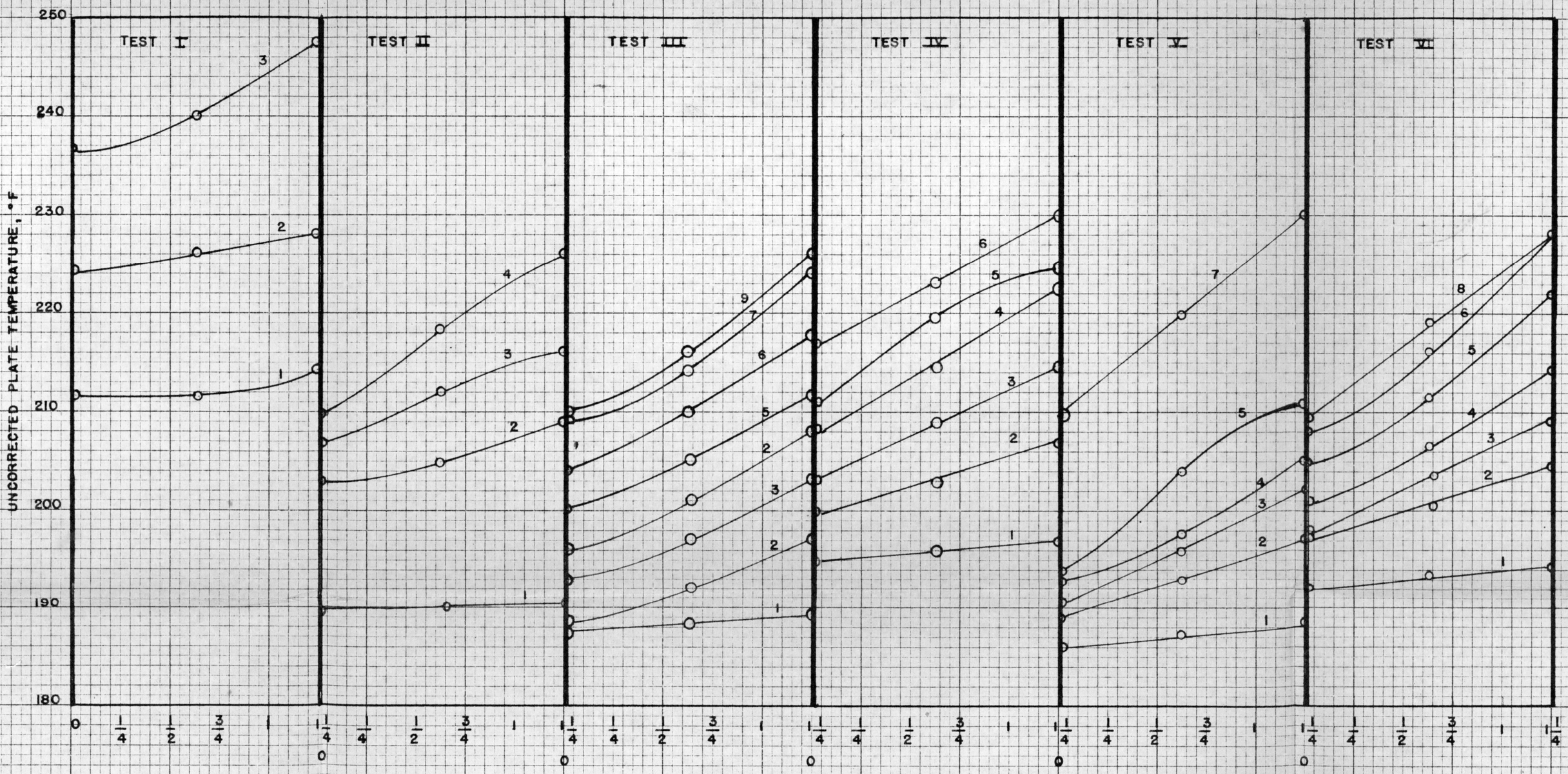
Table XII includes the temperatures obtained from the evaluation of the plate thermocouple readings and are consequently uncorrected for the temperature drop from the thermocouple junction to the silver surface. These values do indicate the manner of temperature distribution across the heater plate.

Figure 27 is a graphical presentation of the values presented in Table XII. The plate temperature, in °F, is plotted as a function of the radius of the plate, in inches.

TABLE XII

Distribution of Uncorrected Temperatures Across Heater Plate

Test No.	Steady State No.	Temperature at Center of Plate °F	Temperature Between Center and Outside of Plate °F	Temperature at Outside of Plate °F
I	1	211.5	211.5	214.0
	2	224.0	226.0	228.0
	3	230.5	234.0	247.5
II	1	189.5	190.0	190.5
	2	203.0	205.0	209.0
	3	207.0	212.0	216.0
	4	210.0	218.5	226.0
III	1	188.0	188.5	190.0
	2	188.5	192.0	197.0
	3	193.0	197.0	203.0
	4	196.0	201.0	208.0
	5	200.0	205.0	211.5
	6	204.0	210.0	217.5
	7	209.0	214.0	224.0
	9	210.0	216.0	226.0
IV	1	195.0	196.0	197.0
	2	200.0	203.0	207.0
	3	203.5	209.0	214.5
	4	208.0	214.5	222.5
	5	211.0	219.5	227.0
	6	217.0	223.0	230.0
V	1	186.0	187.5	188.5
	2	189.5	193.0	197.0
	3	190.5	196.0	202.0
	4	193.0	198.0	205.0
	5	194.0	204.0	212.0
	7	210.0	220.0	230.0
VI	1	192.0	193.5	194.5
	2	197.5	200.5	204.5
	3	198.0	203.5	209.0
	4	201.0	206.5	214.5
	5	205.0	211.5	222.0
	6	208.5	216.5	228.0
	8	209.5	219.0	228.0
VII	1	195.0	197.0	199.0
	2	202.5	205.0	210.0
	3	206.0	210.0	215.5
	4	208.5	215.0	223.0
	5	208.5	216.0	226.0
	6	213.0	218.5	230.5
	7	214.5	225.5	238.5
VIII	1	195.0	196.0	199.0
	2	202.0	205.0	209.5
	3	209.0	215.5	220.0
	4	210.5	217.0	226.0
	5	216.0	225.0	239.0
	7	217.0	230.0	244.0
IX	1	198.5	200.0	203.0
	2	206.0	209.5	213.5
	3	212.0	219.0	227.0
	4	215.5	220.0	230.5
	5	219.5	228.5	241.0
	7	221.5	232.0	245.0
X	1	196.0	197.5	200.5
	2	200.0	209.5	212.0
	3	212.0	220.5	228.0
	4	224.0	231.5	241.0
	6	226.0	236.0	247.0
XI	1	191.0	191.0	191.5
	2	208.0	210.0	215.0
	3	212.0	216.0	222.0
	4	216.5	222.0	230.5
	5	221.0	227.5	237.5
	6	227.5	237.5	249.0



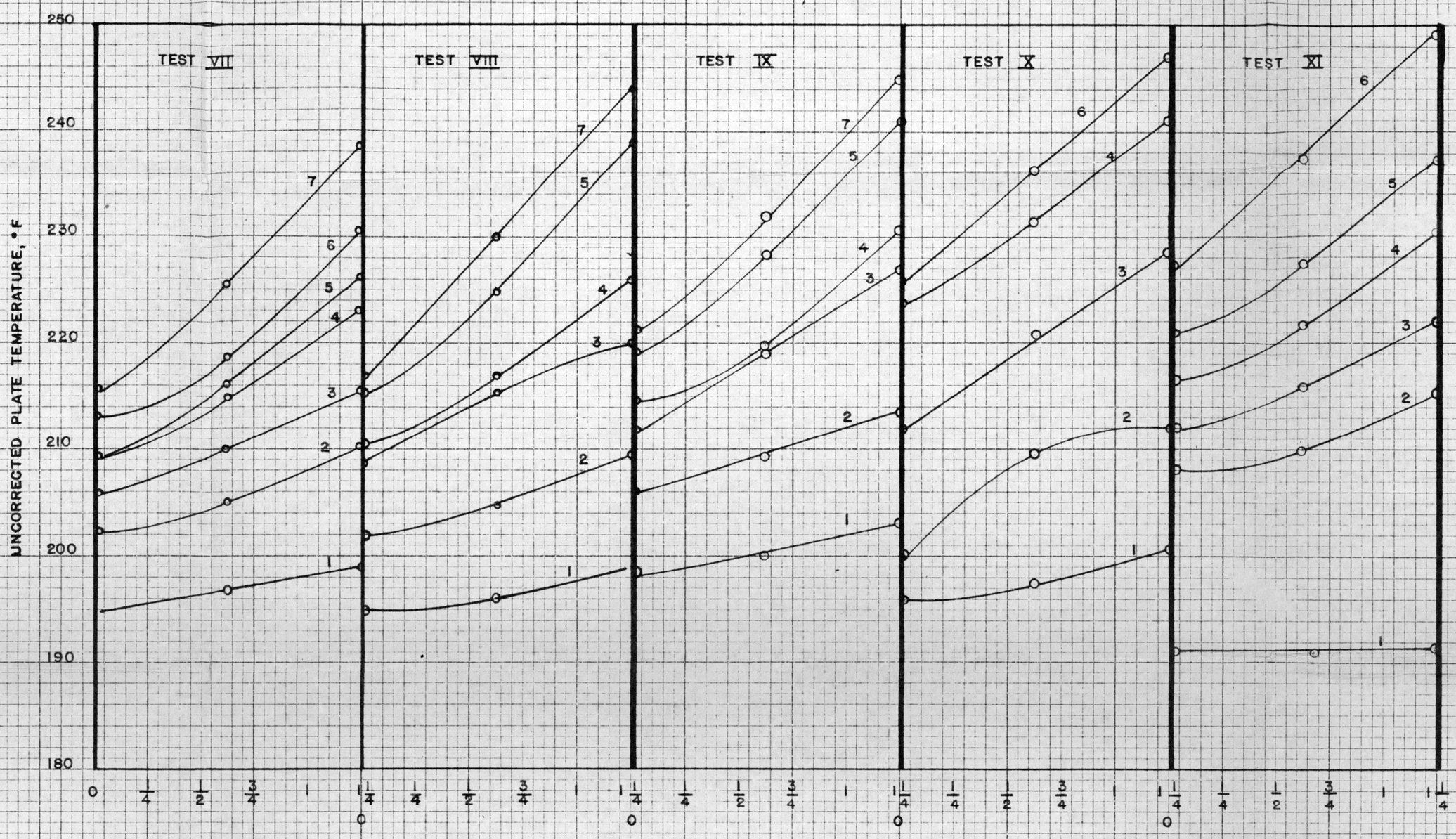


FIGURE 27

CURVES OF TEMPERATURE DISTRIBUTION ACROSS HEATER PLATE

PLATE 3 - CROSS SECTION O. P. R. & R. E. STANDARD
KEUFFEL & ESSER CO., NEW YORK.