

EFFECT OF PRESSURE
ON HEAT TRANSFER IN NUCLEATE POOL BOILING

A Thesis
Presented to
the Faculty of the Department of Mechanical Engineering
University of Houston

In Partial Fulfillment
of the Requirements for the Degree
Master of Science in Mechanical Engineering

by
Kenneth Kai-Hong Tam

August, 1968

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ABSTRACT

An experimental study of nucleate pool boiling heat transfer was carried out over a pressure range between 4.70 psia to 3 atmospheres pressure in order to understand the effect of pressure upon boiling heat transfer. Degassed, distilled water was used as the boiling fluid. Experimental data were obtained from high speed motion pictures, consisting of number of bubbles (N) on the heating surface, bubble departure diameter (D_b) and frequency of departure (f). Heat transfer rate was obtained from a wattmeter connected between power source and heater, and independently by calculation employing a measured temperature gradient within the heating rod.

A comparison of measured heat flux with that calculated by the equation of I. A. Raben, R. T. Beaubouef and G. E. Commerford (1)

$$\dot{Q} = \frac{K}{8} a_c \Delta T + \left[\frac{\pi}{6} D_b^3 \rho_v \lambda + \frac{\pi}{3} \delta^2 \left(\frac{3}{2} D_b - \delta \right) (\rho_l - \rho_v) C \frac{\Delta T}{2} \right] N f$$

was presented. It was shown that the equation gave a satisfactory approximation in calculating heat transfer rate in nucleate pool boiling under pressure.

N , D_b , f and ΔT were found to be important factors in boiling. It was found that N increased with pressure while D_b and ΔT decreased as pressure went up. As pressure increased f was found to decrease but the tendency was not pronounced.

TABLE OF CONTENTS

	PAGE
I INTRODUCTION	1
II LITERATURE SURVEY	2
III EXPERIMENTAL APPARATUS	11
IV DISCUSSION OF RESULTS	18
V CONCLUSION	34
NOTATION	41
BIBLIOGRAPHY	43
APPENDIX	48

LIST OF TABLES

TABLE	PAGE
1 Boiling Heat Transfer Data, 4.70 psia Pressure	35
2 Boiling Heat Transfer Data, 1 Atmosphere Pressure ..	36
3 Boiling Heat Transfer Data, 2 Atmospheres Pressure .	37
4 Boiling Heat Transfer Data, 3 Atmospheres Pressure .	38
5 Boiling Parameters	39
6 Calculated Boiling Parameters	40

LIST OF FIGURES

FIGURE	PAGE
1 Heat Transfer Apparatus with Auxiliary Equipment	12
2 Details of Boiling Heat Transfer Apparatus	13
3 Details of Boiling Heater	14
4 Boiling at 4.70 psia Pressure, $\dot{Q} = 258.00$ Btu/hr	19
5 Boiling Curve for 4.70 psia Pressure	21
6 Boiling Curve for 1 Atmosphere Pressure	22
7 Boiling Curve for 2 Atmospheres Pressure	23
8 Boiling Curve for 3 Atmospheres Pressure	24
9 Effect of Pressure on Heat Transfer	25
10 Boiling at 4.70 psia Pressure, $\dot{Q} = 97.40$ Btu/hr	26
11 Boiling at 1 Atmosphere Pressure, $\dot{Q} = 110.10$ Btu/hr .	26
12 Boiling at 2 Atmospheres Pressure, $\dot{Q} = 87.60$ Btu/hr .	27
13 Boiling at 3 Atmospheres Pressure, $\dot{Q} = 87.60$ Btu/hr .	27
14 Boiling at 1 Atmosphere Pressure, $\dot{Q} = 237.50$ Btu/hr .	28

I. INTRODUCTION

Heat transfer with change in phase has long been known as one of the most effective ways to remove large amount of heat from a hot surface. The phenomenon has been studied extensively in the past few decades because of the needs of modern technology; the large energy generation rate per unit volume occurring in nuclear reactors and rocket nozzles required a very high heat transfer rate.

In boiling heat transfer, the pressure, the surface characteristics, the density of the boiling medium, the latent heat of evaporation, the surface tension and possibly many other properties of the liquid and vapor play an important role. There is much experimental data (1 to 7)* concerning nucleate boiling heat transfer, and many theoretical works have been done to clarify this mechanism. Unfortunately, the phenomenon still is not clearly understood.

An experimental investigation of water pool boiling heat transfer was carried out in order to learn more about the mechanism of this phenomenon.

* Numbers in parentheses denote similarly numbered references in the Bibliography.

II. LITERATURE SURVEY

Boiling heat transfer is a convection process involving a change in phase from liquid to vapor. The phenomena are considerably more complex than those of convection without phase change because in addition to all the variables associated with convection, those associated with the change in phase are also relevant. Although the phenomena have not been completely understood, considerable progress has been made during the last two decades and many papers have been published hoping to clarify the effect of all variables involved. These papers can be conveniently classified into the following seven categories:

(1) Phenomena of Various Boiling Regimes

In 1934, S. Nukiyama (8) carried out a scientific study of boiling. He found that heat transfer to boiling water increased rapidly as the heating surface was heated above 212°F . This increase continued until the surface temperature was about 300°F . Then the metal surface temperature jumped suddenly to about 1800°F . E. R. G. Eckert (9) found that the temperature jump occurred because there were actually four boiling regimes, namely; free-convection boiling, nucleate boiling, transition boiling and film boiling. The author defined that for $0 < \Delta T < 10^{\circ}\text{F}$., the mechanism of heat transfer was mainly by free-convection, bubbles occurred only on a few selected spots.

At $10 < \Delta T < 40^{\circ}\text{F.}$, nucleate boiling occurred, a considerable stirring action caused by the rising of bubbles increased the heat transfer rate. Transition boiling occurred at $40 < \Delta T < 200^{\circ}\text{F.}$, part of the heating surface being covered by vapor film. Above $\Delta T = 200^{\circ}\text{F.}$, film boiling dominated the heat transfer process; a continuous vapor blanket was found covering the heating surface.

(2) Type of Bubbles and Their Agitating Effects

Ten years ago, the formulation theory on nucleate boiling was based upon the information derived from discrete bubbles. In 1965, N. Zuber (10) reported that there actually existed several subdivisions in nucleate boiling, namely; a discrete bubble regime, a merging bubble regime and perhaps a vapor patch regime. L. G. Hamburger (11) and Y. Y. Hsu (7) showed that bubble formation was a step-by-step process and that the transition from the discrete bubble regime to multi-bubble regime was gradual. R. F. Gaertner (12) reported more clearly that the vapor structure on a heating surface progressed through a sequence of first discrete bubbles then vapor column and vapor mushroom and finally vapor patches as ΔT increased.

W. M. Rohsenow and J. A. Clark (13) reported that an increase in heat transfer rate in boiling was accomplished by agitating due to the formation and departure of the bubbles. K. Engelberg-Forster and R. Grief (14) found the assumption that the action of the bubbles was to increase heat flux by

causing strong micro-convection in the highly superheated thermal layer was incomplete. They postulated that the action of bubble growth and collapse acted as highly efficient piston pumping hot liquid from the heating surface to the bulk and the cold liquid from the bulk to the heating surface. Y. P. Chang and N. W. Snyder (15) reported that nucleate boiling heat transfer was due principally to the agitation caused by the bubble growing instead of detaching and did not depend on the detaching size of the bubble.

(3) Thermal Boundary Layer

M. Jakob (16) reported the effect of the thin thermal boundary layer near the heating surface to the heat transfer rate in nucleate boiling. He proposed a relationship between heat transfer coefficient and the thickness of the thermal boundary layer as $h = K/\delta$. K. Engelberg-Forster and R. Grief (14) presented the vapor liquid exchange mechanism caused by the growth and collapse of bubbles which induced the circulation between the cooler bulk liquid and the superheated liquid of the thermal layer. F. D. Moore and R. B. Mesler (17) reported a rapid drop in surface temperature near a nucleation site during nucleate boiling of water. They postulated that the heating surface was cooled during initial bubble growth by evaporation of a microlayer into the bubble. S. G. Bankoff (18) argued that the thermal boundary layer thickness could not be much larger than the maximum bubble radius. T. F.

Rogers and R. B. Mesler (19) and R. C. Hendricks and R. R. Sharp (20) obtained support for the microlayer evaporation by determining the relation between bubble growth and surface temperature. B. D. Marcus and D. Dropkin (21) found the temperature profile within the thermal layer to be essentially linear very near the heating surface and within 0.57 δ , temperature distribution could be expressed as:

$$\frac{T - T_b}{T_s - T_b} = 1 - \frac{y}{\delta}$$

I. A. Raben, R. T. Beaubouef and G. E. Commerford (1) derived an equation

$$\dot{Q} = \frac{k}{\delta} a_c \Delta T + \left[\frac{\pi}{6} D_b^3 \rho_v \lambda + \frac{\pi}{3} \delta^2 \left(\frac{3}{2} D_b - \delta \right) (\rho_l - \rho_v) C \frac{\Delta T}{2} \right] N f$$

to predict the heat transfer rate in nucleate pool boiling and showed that at low pressure, vapor liquid exchange mechanism gave great contribution to the heat transfer rate, and in 1966, N. B. Hospeti and R. B. Mesler (22) confirmed that the hypothesis of thermal layer evaporation was correct.

(4) Latent Heat Transport

Latent heat transport accounts for the heat that leaves the surface by conduction and convection to form bubbles up to the point of their departure. Y. P. Chang and N. W. Snyder (15) and C. J. Rallis and H. H. Jawurek (23) showed that latent heat transport was one of the important phenomena in boiling heat transfer especially when the condition of peak heat flux

was reached although W. M. Rohsenow and J. A. Clark (13) reported that latent heat effects could not account for the heat transfer rate and K. Engelberg-Forster and R. Grief (14) found that latent heat transport accounted for only 2 per cent of the total heat flux.

In 1962, S. G. Bankoff (18) showed that latent heat transport by simultaneous evaporation and condensation at different portions of the bubble surfaces could account for the major portion of the total heat flux in the neighbourhood of the departure from subcooled nucleate boiling. C. P. Costello, C. O. Bock and C. C. Nichols (24) reported that the total heat flux at burnout was the sum of the latent heat transport and the energy removed by convection. Radiation was not included in the report. I. A. Raben, R. T. Beaubouef and G. E. Commerford (1) showed that latent heat transport increased to become dominant as heat flux went up.

(5) Bubble Departure Diameter, Shape and Frequency

In 1935, W. Fritz presented an equation

$$D_b = 0.01475 \beta \left[\frac{2\sigma}{g(\rho_l - \rho_v)} \right]^{\frac{1}{2}}$$

to predict the bubble departure diameter, but M. Jakob (16) indicated that such an equation did not always agree with experimental findings. R. Cole and H. L. Shulman (25) presented another equation for predicting departure diameter as:

$$D_b = \frac{1000}{P} \left[\frac{g_c \sigma}{g(\rho_l - \rho_v)} \right]^{\frac{1}{2}}$$

and stated that such an equation covered the widest range of experimental conditions and did not require a knowledge of the heat flux level or wall superheat.

For the shape of departing bubbles, most investigators assumed a spherical bubble shape for simplicity in calculation. M. S. Plesset and S. A. Zwick (26) assumed spherical bubbles. E. G. Keshock and R. Siegel (27) showed that the shape of departing bubbles was close to a sphere. However, M. A. Johnson Jr., Javier de la Pena and R. B. Mesler (28) reported that bubbles could actually be classified as spherical, hemispherical and oblate bubbles and found that for spherical bubbles, inertial forces were small because of the slow growth rate, for hemispherical bubbles, the fast growth rate caused a very large inertial force while for oblate bubbles, neither the inertial force nor the surface tension determined the shape.

Concerning the frequency, M. Jakob (16) reported that a bubble broke off when its volume had grown so much that the buoyancy exceeded the capillary forces which bond it to the heating surface. He assumed that the growth period and the waiting period for a bubble was about the same, and that the frequency of bubble formation depended on the size of bubble at the moment of breaking off, but P. H. Streng, A. Orell and J. W. Westwater (29) and D. B. Kirby and J. W. Westwater (30) reported that Jakob's assumption that the waiting period and the growth period was about the same was incorrect, but $f \times D_b$ was nearly constant for each liquid. T. Hara (31) showed that

the waiting period was determined by the size of cavity and the thermal layer thickness. G. I. Bobrovich and N. N. Mamontova (32) found experimentally that in nucleate boiling of water, the number of vapor formation centers increased with heat flux density but the frequency of departure did not depend on the heat flux for a wide range and was approximately equal to $1/35$ second. In 1966, V. I. Tolubinsky and J. N. Ostrovsky (33) reported that the product of f and N depended on physical properties of liquid. For a given liquid $f \times N$ was constant and did not depend on heat flux. Pressure, however, had great influence.

(6) Effect of System Pressure

Pressure has great influence on nucleate boiling heat transfer. D. S. Cryder and A. C. Finalborgo (34) conducted experiments in boiling of liquid over a range of pressures and found that as pressure was reduced, an increase in wall superheat was required for the same amount of heat flux. M. T. Cicchelli and C. F. Bonilla (2) found that the coefficient of heat transfer continued to rise as the pressure was increased until nucleate boiling ceased to be stable. E. A. Farber and R. L. Scoriah (35) reported that at different elevated pressures, the same heating surface gave different numerical values for the boiling curve. W. H. McAdams, W. E. Kannel, C. S. Minder, R. Carl, P. M. Picornell and J. E. Dew (36) stated that increased pressure tended to reduce the ΔT necessary for a given heat

flux. F. Kreith and M. Summerfield (37) found that the surface temperature necessary for a given heat flux was a function of the system pressure. S. G. Bankoff (38) reported that the frequency of bubble formation would be increased at elevated pressure. D. A. Labuntsor, B. A. Kol'chugin, V. S. Golovin and G. V. Tsiklauri (39) and E. R. Hosler (40) showed that the size of departing bubble was inversely proportioned to the system pressure. I. A. Raben, R. T. Beaubouef and G. E. Commerford (1) analyzed theoretically and investigated experimentally the effect of system pressure and showed that the mechanism of energy transport in nucleate boiling became less effective with reduced pressure, and ΔT in the transition region between the convective and nucleate boiling appeared to be influenced by pressure, and S. A. Kovalev (41) found that $q_{\max}/q_{\min}$ decreased with an increase of pressure.

(7) Surface Condition, Contamination and Nucleation Site

In 1936, M. Jakob (16) reported that painting a smooth metal surface with a graphited mass yielded an increased density of bubble producing sites. W. H. McAdams, J. N. Addoms, P. M. Rinaldo and R. S. Day (3) pointed out that in the region of nucleate boiling, heat flux was independent of surface area, but was considerably affected by contamination. E. T. Sauer, H. B. H. Coope, G. A. Akin and W. H. McAdams (6) found that, by grooving a horizontal copper tube, an increase of overall coefficient by 30 per cent and the critical heat flux by 50 per cent were

obtained. C. Corty and A. S. Foust (42) also found that the roughest surface produced by using emery paper offered a heat transfer coefficient 60 per cent higher than the smoothest surface. H. W. Kurchare and J. E. Mayers (43) showed that a rough surface gave a greater number of bubble columns than a smooth surface at a given heat flux. Y. Y. Hsu and R. W. Graham (44) reported that the departure diameter was limited when the areas of influence of adjacent bubbles began to interfere, and that the period of bubble formation was dependent upon at least the cavity diameter and bulk temperature. R. T. Beaubouef (45) presented an equation relating surface condition and bulk temperature, cavity radius and certain properties of the boiling fluid. P. J. Berenson (5) found that surface roughness affected the ΔT required to transfer a given heat flux in nucleate boiling and that the heat transfer coefficient could be changed by 500 to 600 per cent owing to change in surface roughness, but the maximum nucleate boiling burnout heat flux was essentially independent of surface roughness. C. P. Costello and W. J. Frea (46) reported that bubbles coming from a heater covered by mineral deposits were smaller than those coming from the clean one and the population seemed much higher. The magnitude of the critical heat flux was affected by the presence of deposits on the test elements, and C. F. Bonilla, J. J. Grady and G. W. Avery (47) found that the heat transfer coefficient can be increased by applying sharp parallel scratches onto the heating surface and scratches of 2 to 2.5 bubble departure diameters apart yielded the highest heat transfer coefficient.

III. EXPERIMENTAL APPARATUS

The experimental apparatus with auxiliaries used in this study is shown in Figure 1. The boiling heat transfer apparatus is shown in Figure 2. The boiling chamber was made from a 3-1/8-inch length of 6-inch diameter, No. 40 stainless steel pipe and two pieces of 1/2-inch thick stainless steel plate were welded to each end. Two pieces of 3/8-inch thick, 2-1/2-inch diameter safety glass were installed in each end plate to provide a clear view area of 2 inches in diameter. A condenser made from coiled 3/8-inch copper tube with a jacket of 3-inch O.D. plexi-glass tube was mounted above the boiling chamber to condense the vapor evolved during boiling and to return condensate to the chamber. A pressure gauge with 1 psi. graduations up to 100 psig. was mounted between heating chamber and condenser. A loop made of 5/8-inch copper tube filled with distilled water was installed between the pressure gauge and the heating chamber to prevent superheated vapor going directly into the gauge. System pressure was generated from a pressurized nitrogen gas tank. Nitrogen gas entered the heating chamber from the top of the condenser. Two valves were installed to provide a means of regulating pressure. For the run below atmospheric pressure, a mercury manometer calibrated to 1/10 of an inch was installed instead of the pressure gauge. A jet pump connected to the top of the condenser was used to obtain the necessary vacuum conditions.

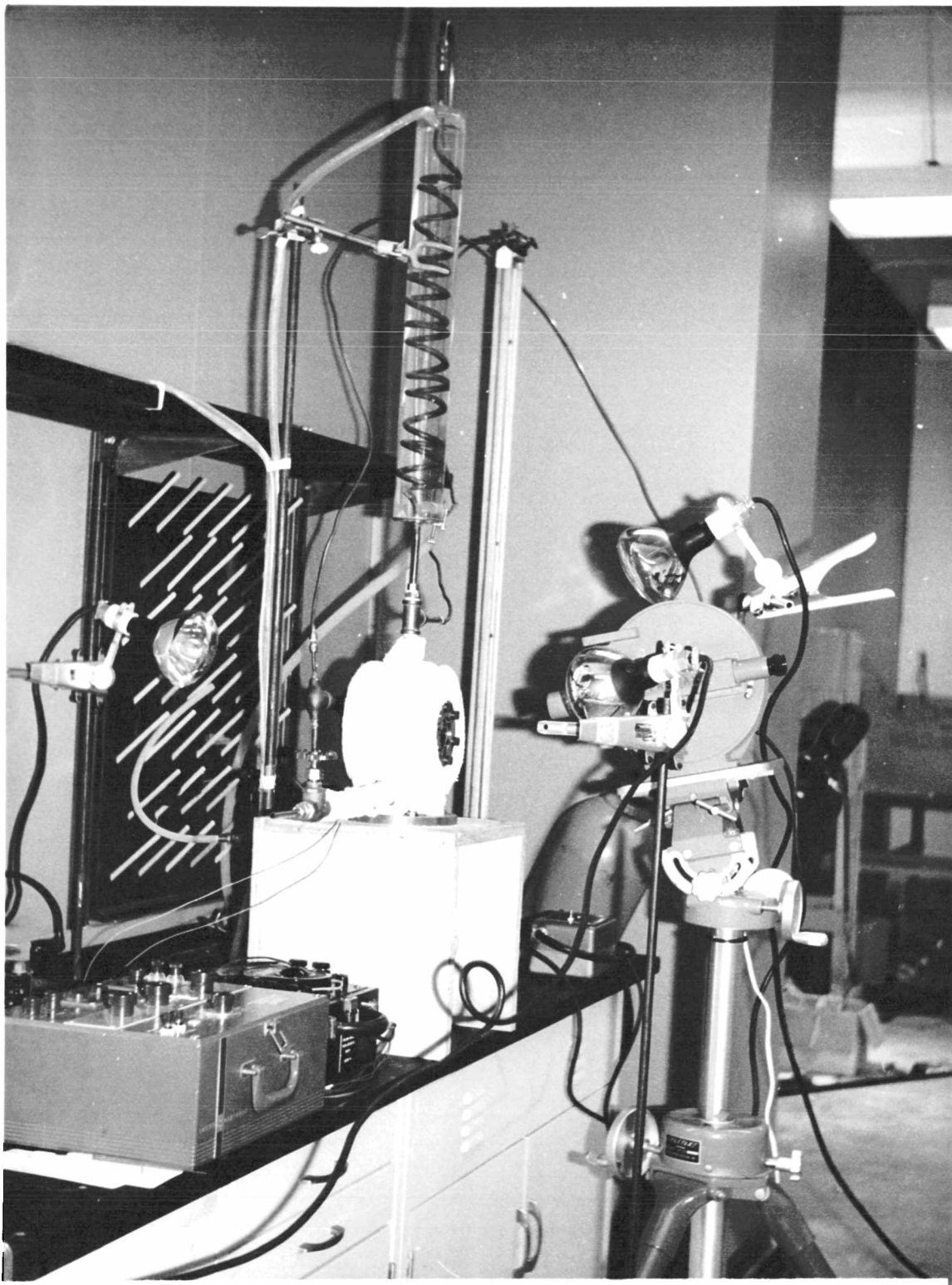


FIGURE 1 HEAT TRANSFER APPARATUS WITH AUXILIARY EQUIPMENT

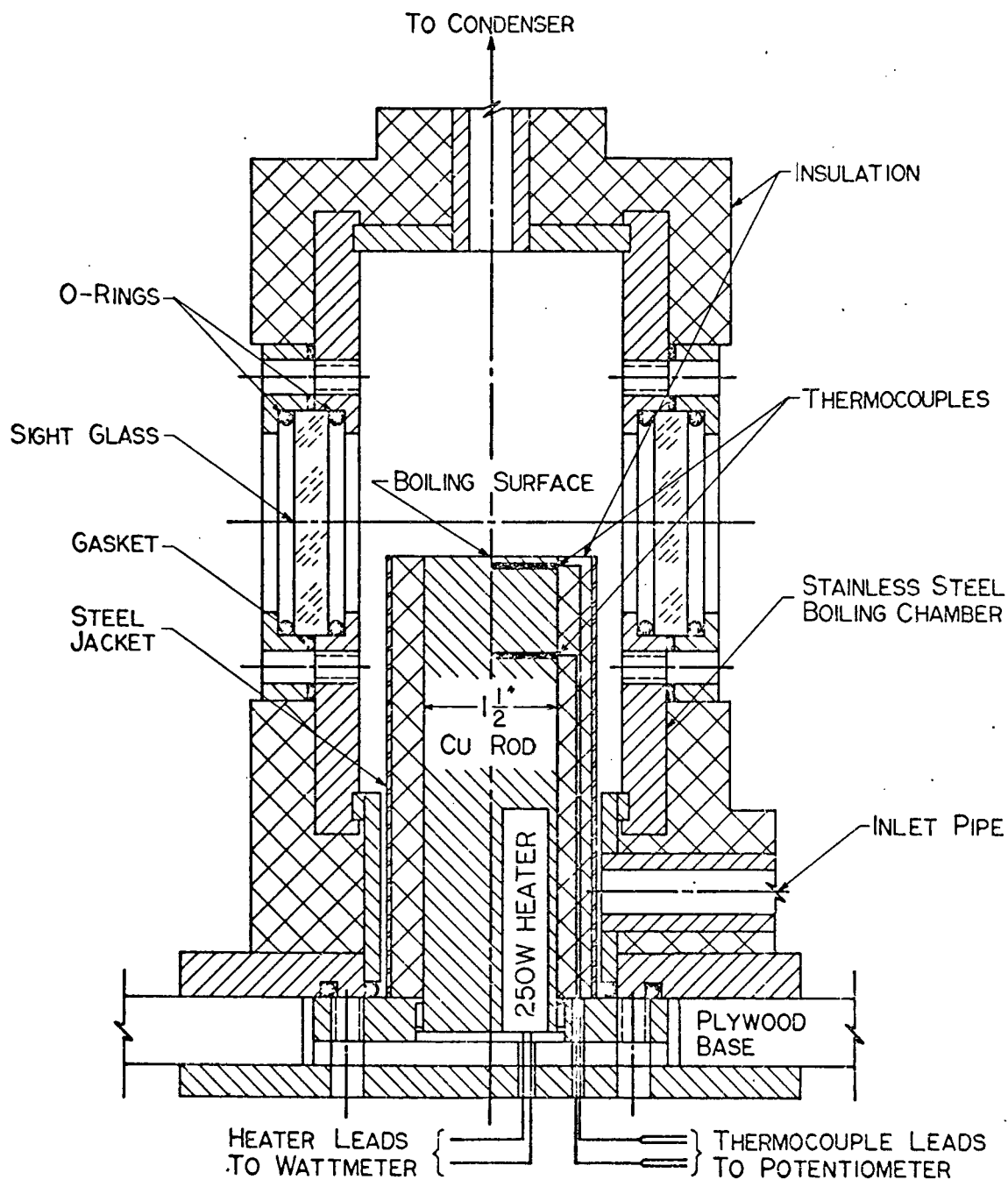


FIGURE 2 DETAILS OF BOILING HEAT TRANSFER APPARATUS

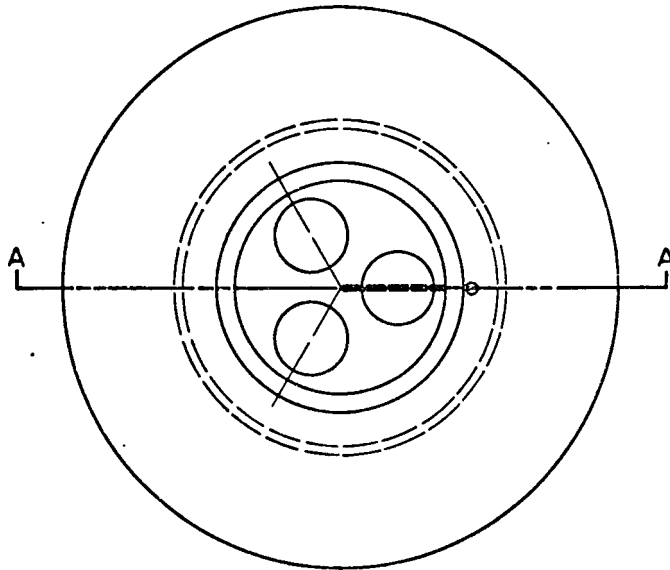
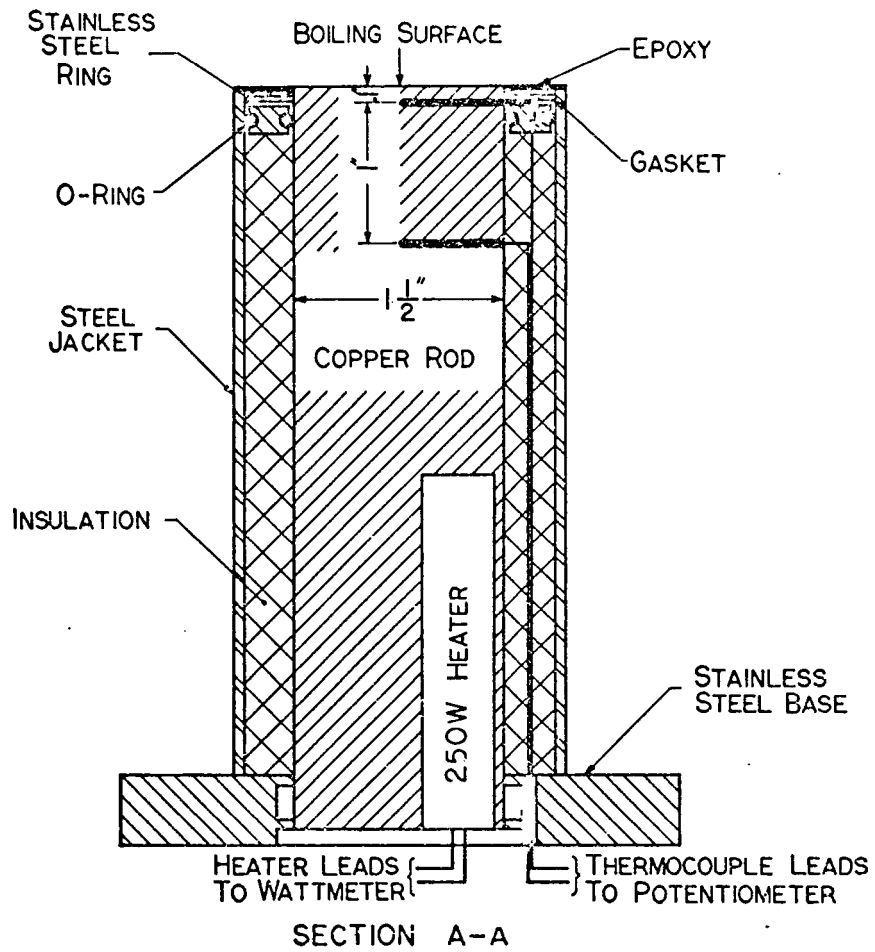


FIGURE 3 DETAILS OF BOILING HEATER

The heating surface used for study was the top of a 1-1/2-inch vertical cylindrical copper rod of high purity. The cylindrical surface was insulated with 3/8-inch thick calcium silicate compound insulation and a steel jacket was used to prevent water being absorbed by the insulation. The upper surface of the insulation was covered with a 1/4-inch thick stainless steel ring which pressed two silicone rubber O-rings, one placed against the outside cylindrical surface of the copper rod and the other placed against the inside cylindrical surface of the steel jacket. On the top of the stainless steel ring, there were two layers of asbestos rubber compound gasket fitted flush with the copper rod. The gasket to metal joint was sealed with epoxy adhesive to prevent nucleation. Figure 3 shows the details of this assembly. Calcium silicate was chosen because it was the best available material with the necessary temperature stability. The use of the stainless steel ring with two silicone rubber O-rings was to prevent vapor passing through the gasket metal joint into the insulation due to pressure difference in case leaks occurred around the epoxy-metal joint. Estimated maximum heat loss through the insulation was less than 5% of the heat conducted through the copper rod.

Two thermocouple were installed in the copper rod at distances of 0.1 inch and 1.1 inches below the copper rod surface. The thermocouple beads were placed at the center of the rod. This arrangement gave an accurate measurement of the temperature field and permitted calculation of the heating surface

temperature and heat transfer rate. Bulk water temperature was measured from a thermocouple going through one of the six screws which fixed the glass window to the end plate of the heating surface. The thermocouple bead was immersed in the water near the heating surface. Bulk temperature was controlled within $\pm 1^{\circ}\text{F}$. of the saturation temperature at all pressures. All thermocouples were calibrated at atmosphere pressure at ice point, saturated steam and boiling point of aniline of high purity. Conversion tables were calculated according to each individual thermocouple calibration. All thermocouple readings were obtained from a millivolt potentiometer which provided an accurate reading down to 0.005 millivolt. An ice bath filled with mixture of distilled water and ice made of distilled water was used as reference junction.

Heat was supplied at the lower end of the copperrod by a 'Chromalox' electrical resistance heater imbedded in the rod. The power supplied to the heater was controlled by a powerstat and measured by a wattmeter. Heat loss through the heating chamber was compensated by a 'Briskeat' flexible heating tape wrapped around the chamber. Power supplied to the heating tape was regulated by another powerstat. Several layers of 1/8-inch asbestos tape covered the heating tape and the surface of the heating chamber except the glass window in order to reduce heat loss.

The heating surface was obtained by first smoothing with 600A emery paper and then polishing with a rotating rubber buffer

and jeweller's rouge. The surface was refinished after obtaining a set of data for each pressure and, after each refinishing, the surface was subjected to vigorous boiling for several hours and heat flux and ΔT were checked with a known previous condition to ensure that the boiling characteristics of the surface had not been altered.

Degassing of the water was achieved by vigorous boiling in a 5-liter flask for a period of several hours, prior to transfer into the boiling chamber.

A 'Fastax' camera, capable of shooting 8,000 frames per second was used to photograph the boiling process through the sight ports. In this study, a film speed of 5,000 frames per second was used.

IV. DISCUSSION OF RESULTS

The principal portion of the experimental program consisted of obtaining nucleate boiling heat transfer data such as saturation temperature (T_{sat}), heating surface temperature (T_w), heat transfer rate ($\dot{Q}$), and measurements of number of bubbles (N) on the heating surface, frequency of bubble (f) leaving from the surface and the bubble departure diameter (D_b).

Data reported for N , f , D_b were obtained from high-speed motion pictures of the boiling process over the entire heating surface. A 'Recordak' Film Reader with a screen of 12" x 9" and affording an image 4.45 times actual size was used to study the films frame-by-frame. The scaling factor of 4.45 is obtained by comparing the measured diameter of the copper rod from the screen to the actual size. Diameter of the bubble at departure was measured to 1/32 of an inch from the screen of the reader, i.e., to 1/(32 x 4.45) of an inch of actual size. It was measured from a section of the films which represents the recorded bubble activity. For spherical bubbles, D_b was taken from the measurement parallel to the heating surface. For hemispherical and oblate bubbles, two major axes were measured and the average of the two was used as diameter to calculate the volume of the bubble. In this way, it is expected to obtain a more accurate result than that of measuring from one axis. Figure 4 shows one such bubble.

Number of bubbles was the number of bubbles on the heating

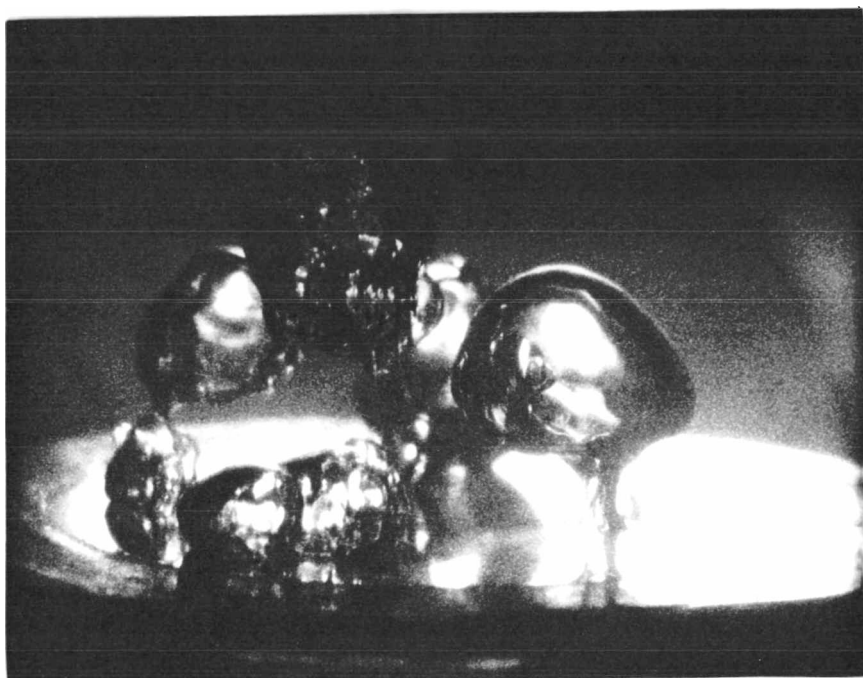


FIGURE 4 BOILING AT 4.70 psia PRESSURE, $\dot{Q} = 258.00$ Btu/hr surface during the bubble period. It was obtained by counting the number of nucleation sites on the heating surface found in the section of film used.

Frequency was taken to be the reciprocal of the period between departure of successive bubbles from a given nucleation site. The period was measured from the timing marks which the camera, through the use of a timed flashing light, placed on the film.

At high heat flux, bubbles were often found to merge with one another. It would have served no purpose to average the diameter and frequency of merged bubbles with that of discrete bubbles.

Figures 5 through 8 present heat transfer curves (Q/A vs ΔT) for distilled water at system pressure of 4.70 psia, 1 atmosphere pressure, 2 atmospheres pressure and 3 atmospheres pressure respectively. Figure 9 presents effect of pressure on heat transfer rate.

Tables 1, 2, 3, and 4 present measured parameters for each heat flux and pressure. These parameters consist of P , T_{sat} , T_w , ΔT , $\dot{Q}$ and Q/A .

Table 5 summarizes the measured parameters from the motion pictures for each heat flux and pressure. These parameters include $\dot{Q}$, ΔT , D_b , f and N , and the calculated value of δ^* .

Table 6 presents the calculated values of $\dot{Q}_c$, $q_{\lambda} Nf$, $q_{v-l} Nf$, calculated and measured values of $\dot{Q}$ and the difference between $\dot{Q}_{meas}$ and $\dot{Q}_{calc}$.

In Figures 10 through 13, it is clear that the bubble departure diameter (D_b) decreases as pressure increases for the same heat flux. This is expected from the equation of R. Cole and H. L. Shulman (25)

$$D_b = \frac{1,000}{P} \left[\frac{g_c \sigma}{g(\rho_l - \rho_v)} \right]^{\frac{1}{2}}$$

and corroborates the view point of certain investigators (1), (39) and (40).

As concerning the shape of departing bubbles, it is found that at low heat flux, departing bubbles were close to spherical

* See Appendix for sample calculation.

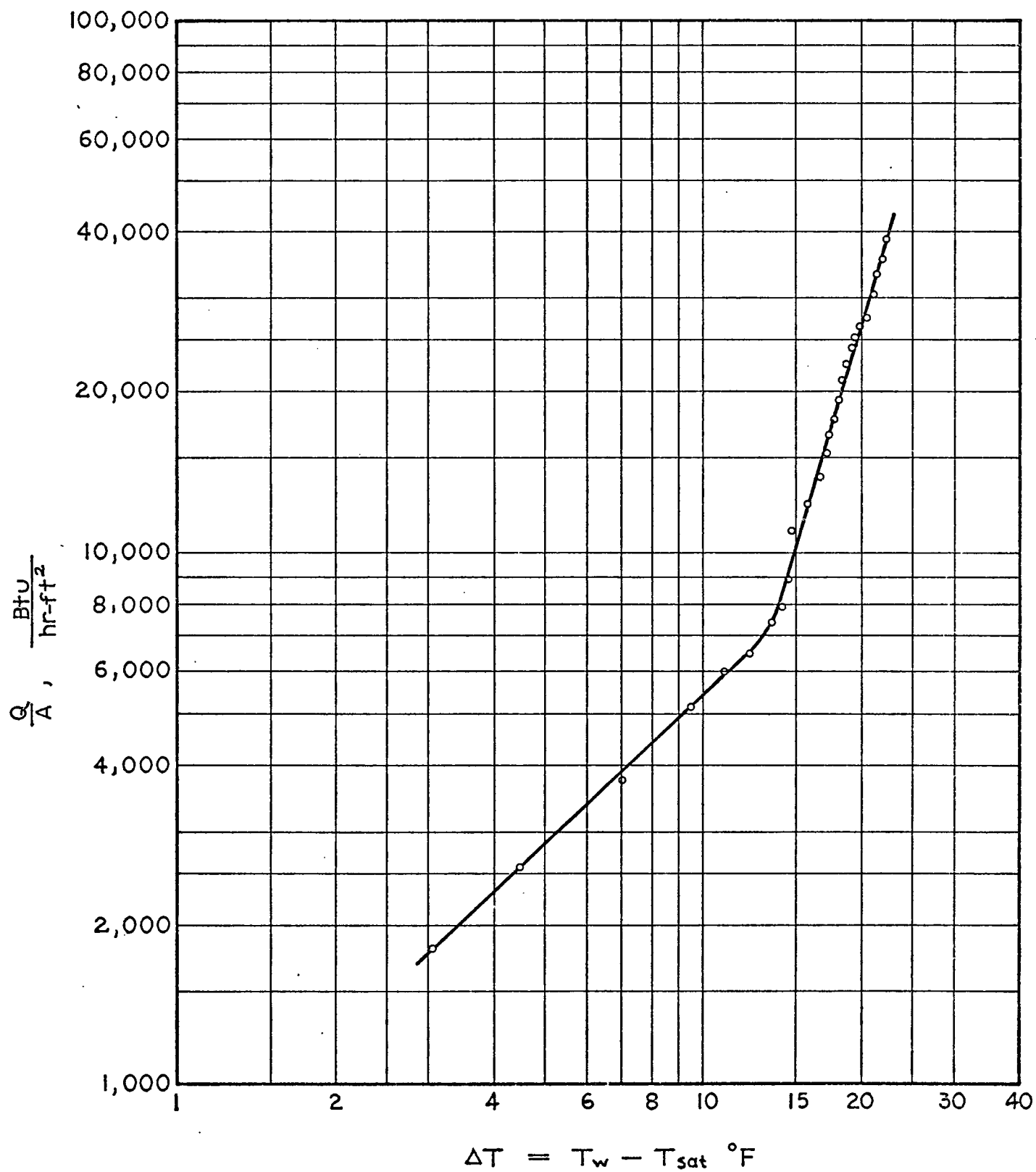
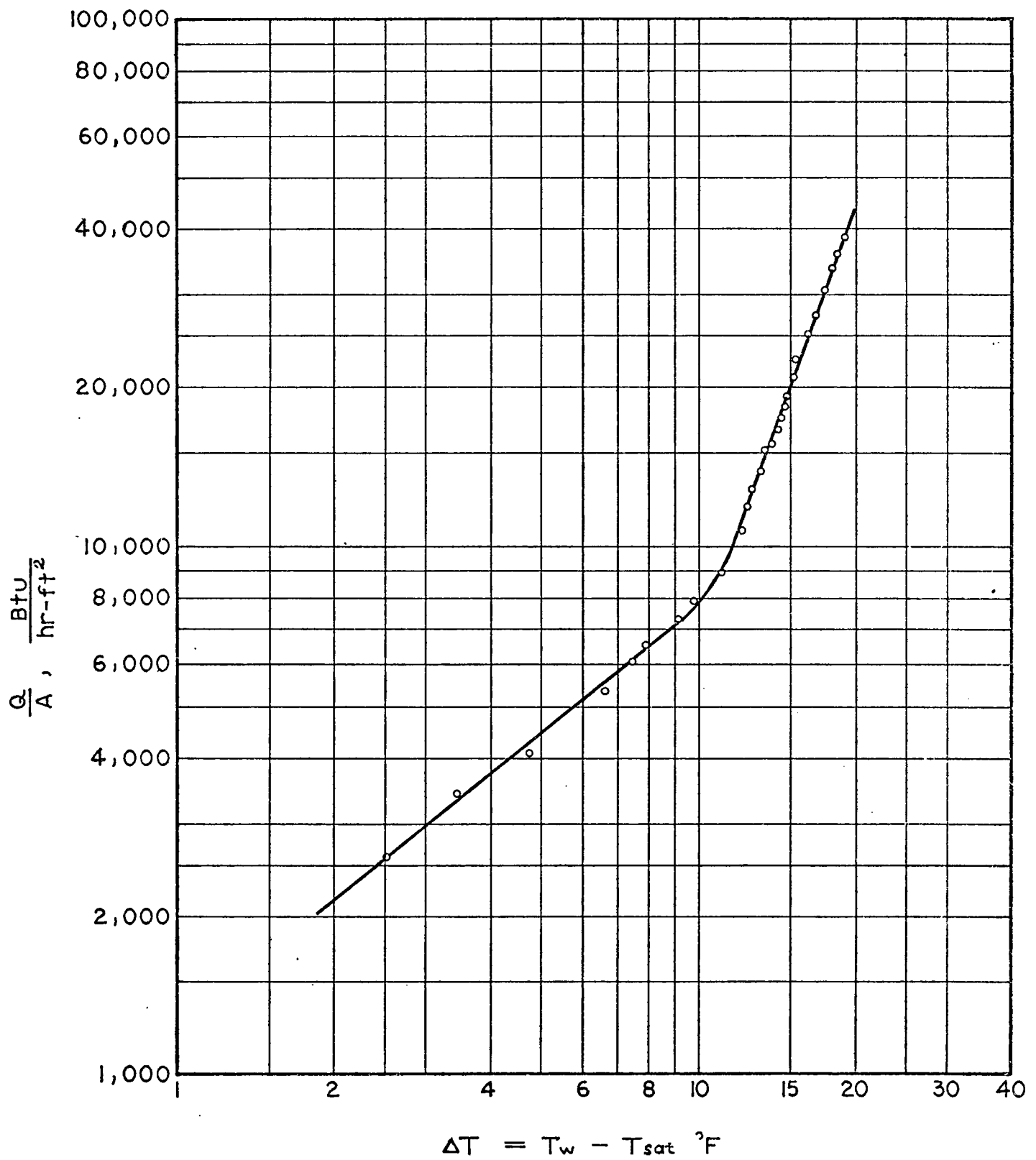


FIGURE 5 BOILING CURVE FOR 4.70 PSIA PRESSURE



FIGUER 6 BOILING CURVE FOR 1 ATMOSPHERE PRESSURE

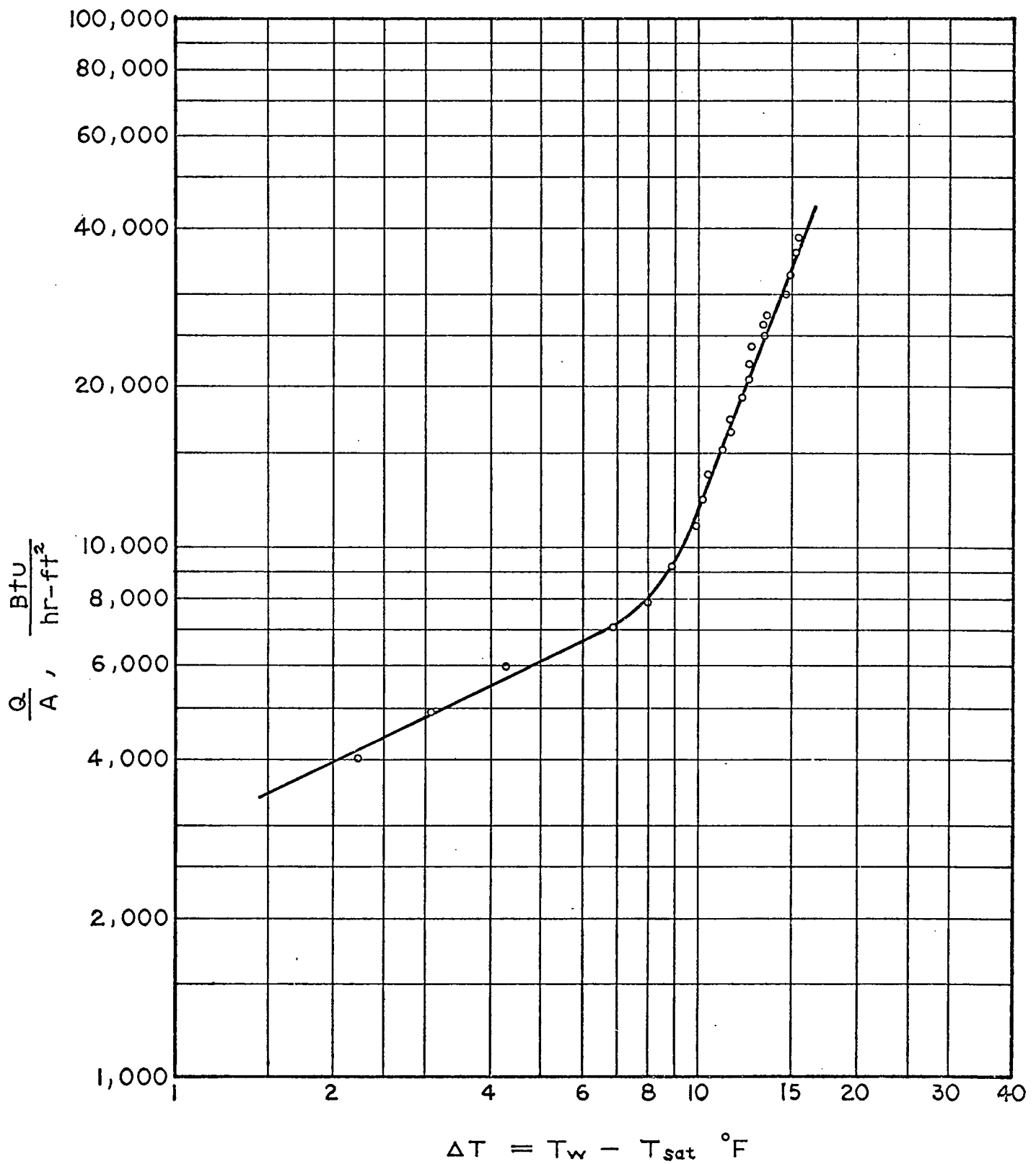


FIGURE 7 BOILING CURVE FOR 2 ATMOSPHERES PRESSURE

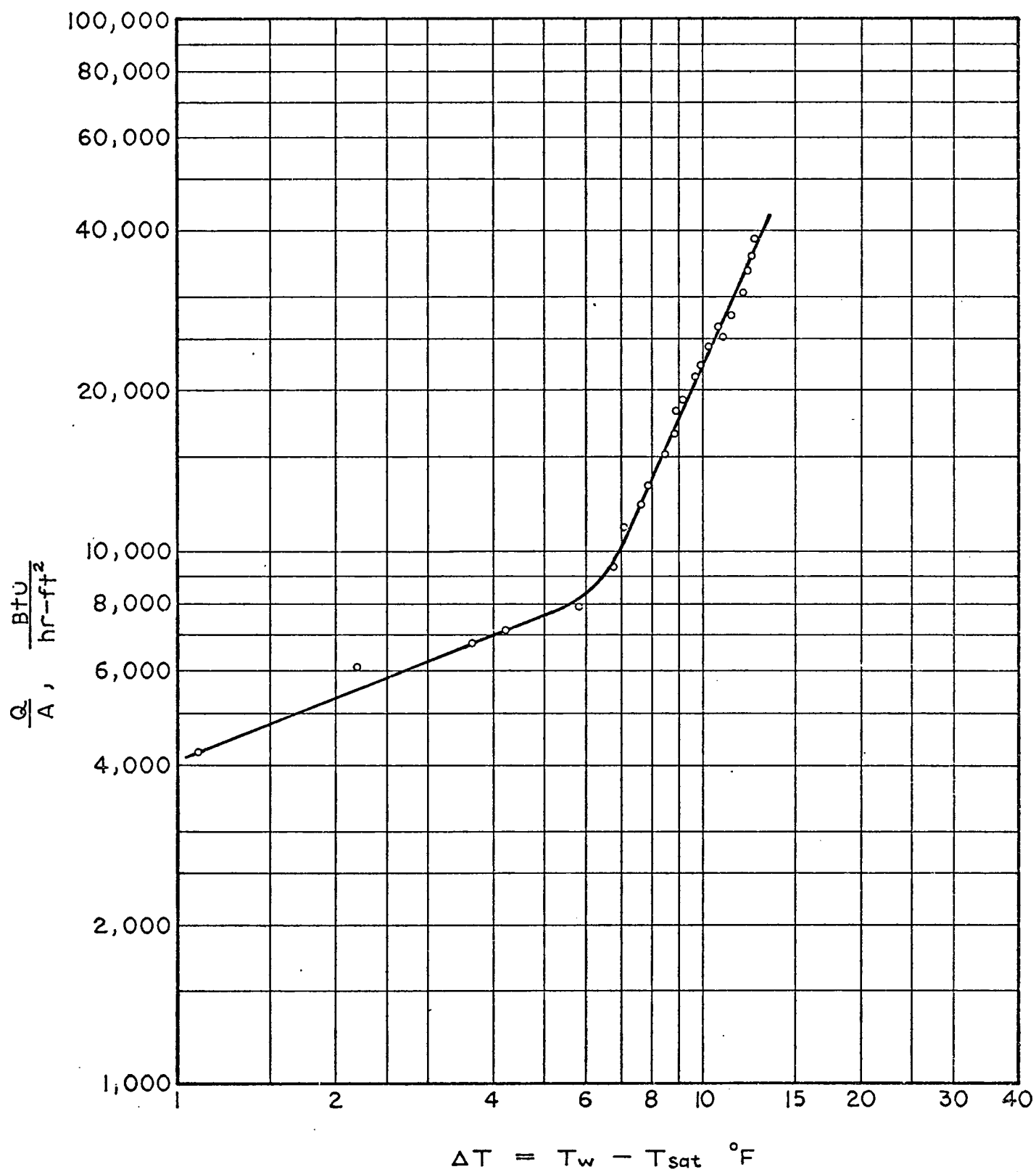


FIGURE 8 BOILING CURVE FOR 3 ATMOSPHERES PRESSURE

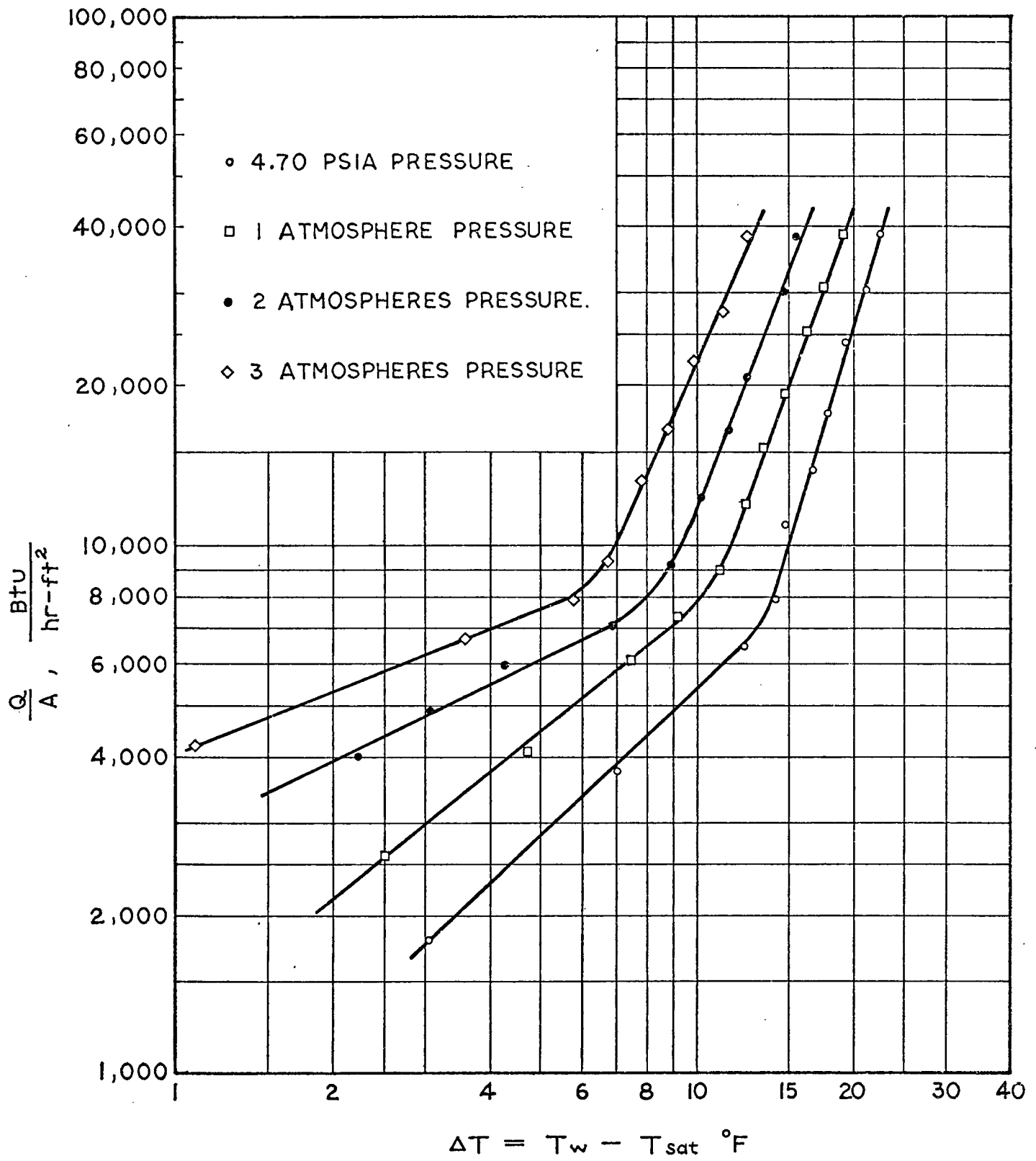


FIGURE 9 EFFECT OF PRESSURE ON HEAT TRANSFER

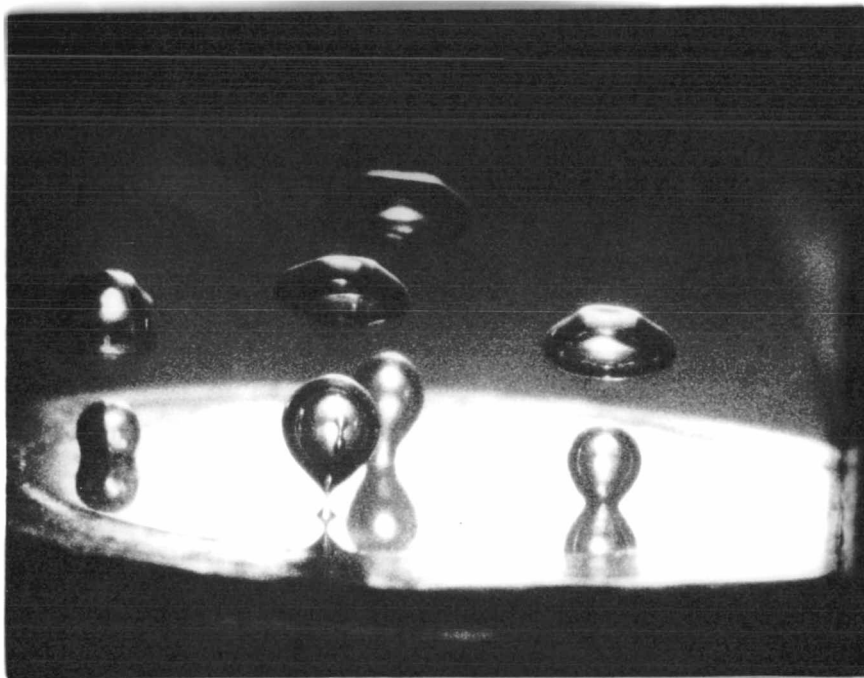


FIGURE 10 BOILING AT 4.70 psia PRESSURE, $\dot{Q} = 97.40$ Btu/hr

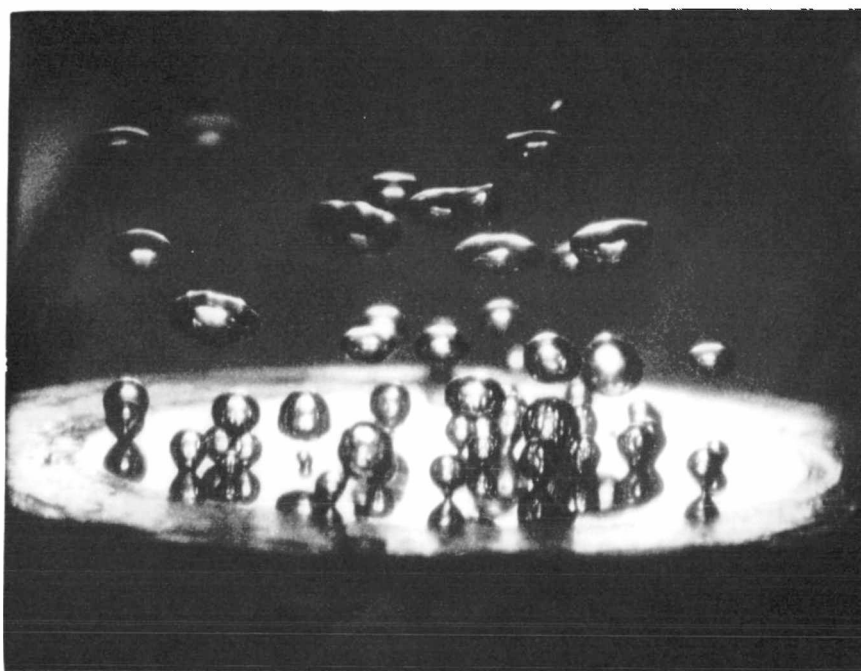


FIGURE 11 BOILING AT 1 ATMOSPHERE PRESSURE, $\dot{Q} = 110.10$ Btu/hr

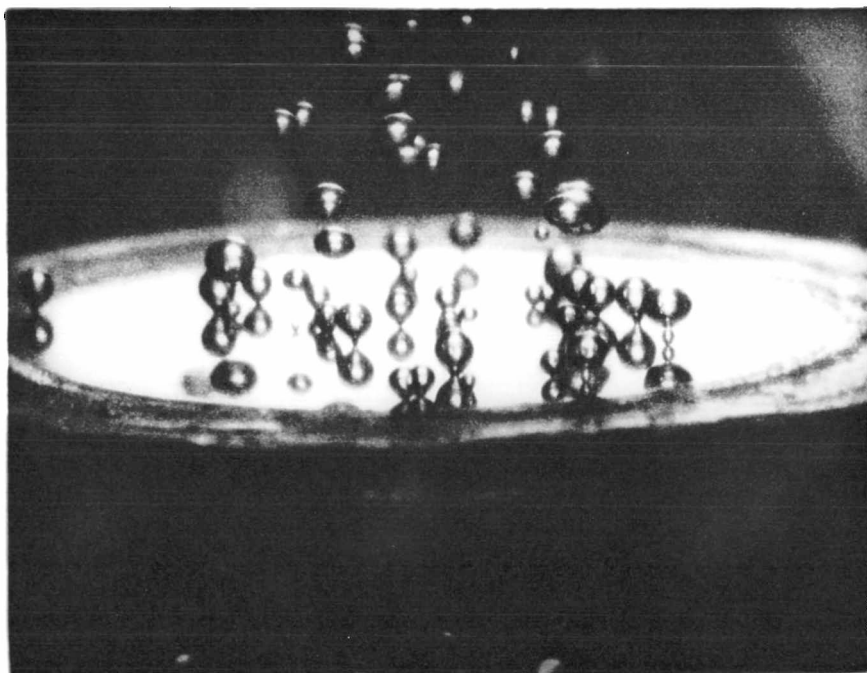


FIGURE 12 BOILING AT 2 ATMOSPHERES PRESSURE, $\dot{Q} = 87.60$ Btu/hr

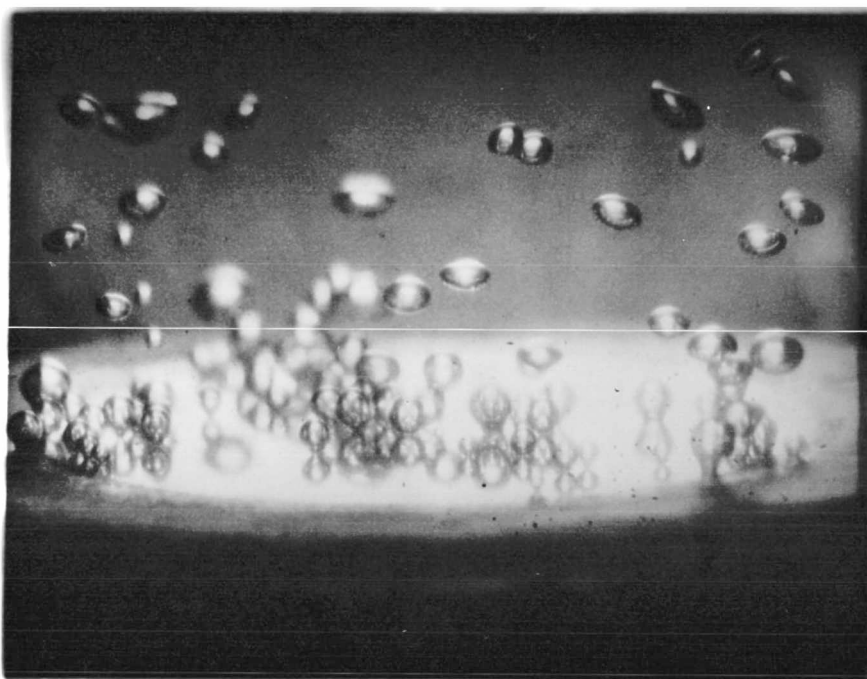


FIGURE 13 BOILING AT 3 ATMOSPHERES PRESSURE, $\dot{Q} = 87.60$ Btu/hr

but at high heat flux, the departing bubbles were not all spherical in shape. They could be classified as spherical, hemispherical, and oblate bubbles as reported by (10), (11) and (28). The possible explanation of this phenomenon is that under high heat flux, the heating surface is crowded with bubbles. Y. Y. Hsu and R. W. Graham (44) reported that in order to maintain the production of individual bubbles in nucleate pool boiling, it is necessary that the spacing (S) between neighbouring active sites should be $S \geq 1.85 \times D_b$ and N. Zuber (10) found that $S = 1.44 \times D_b$. When a bubble grows within the influence area of its neighbouring bubble, the two will merge (7) and form a single bubble which is not in point contact with the heating surface. Such a bubble is

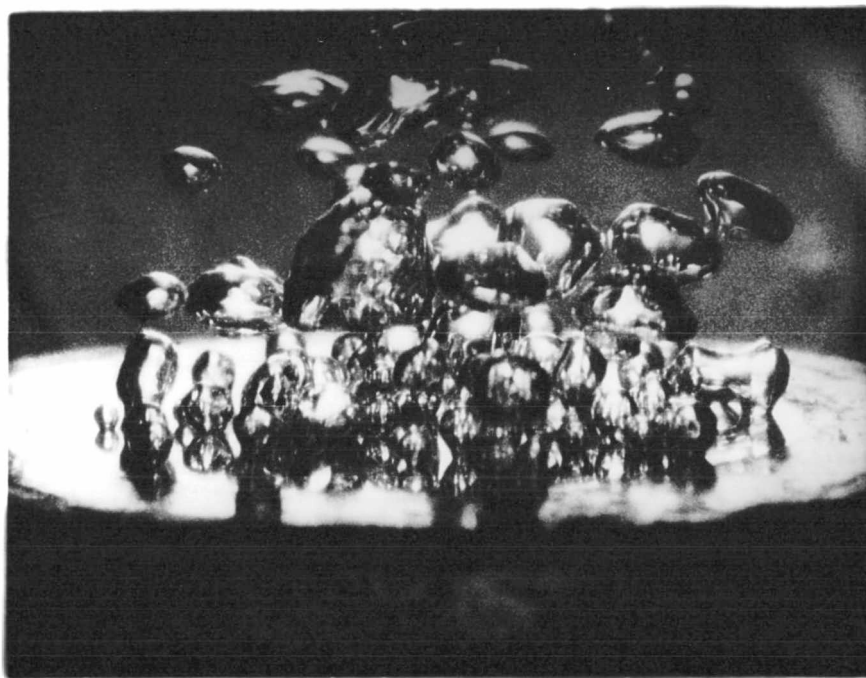


FIGURE 14 BOILING AT 1 ATMOSPHERE PRESSURE, $\dot{Q} = 237.50$ Btu/hr

often found not to be spherical in shape at break-off and the volume is found to be greater than that of a single bubble. Besides, the turbulence generated by the departure of the adjacent bubbles will affect the shape of the bubble. Figure 14 shows these phenomena.

The number (N) of bubbles on the heating surface is found to increase with pressure. The equation of P. Griffith and J. D. Wallis (48)

$$r = \frac{2\sigma T_w V_{fg}}{\lambda(\Delta T)}$$

shows that the radius of the cavity mouth (r) is determined by the minimum wall superheat necessary to begin nucleation at a particular cavity. As shown in Figure 9, for a given heat flux, ΔT decreases as pressure increases. For constant ΔT , by Clausius-Clapeyron equation:

$$\frac{dP_v}{dT} = \frac{\lambda}{T_{sat} V_{fg}}$$

r will decrease as pressure increases because, as dP_v increases, $T_{sat} V_{fg}/\lambda$ becomes smaller, since $\Delta T = T_w - T_{sat}$ is kept constant, T_w will increase by the same amount as does T_{sat} , so $T_w V_{fg}/\lambda$ will also decrease as dP_v increase, and σ is found to be inversely proportional to pressure as shown by J. H. Keenan (49). Therefore, the total effect then, is an decrease in $2\sigma T_w V_{fg}/\lambda(\Delta T)$. This suggests that for a given surface and ΔT , N will increase with pressure. Figure 10 through 13 show this phenomenon.

The frequency of bubble departure (f) showed a decrease with an increase in pressure as reported by (33), but the tendency was not very pronounced as shown in Table 5.

The discontinuity reported by (1) in the region between convective and nucleate boiling was not observed in this study due to higher system pressures studied here.

The thermal boundary layer thickness was calculated from

$$\delta = \frac{K a_c \Delta T}{\dot{Q}_c}$$

which shows that as pressure increases, the thickness decreases because ΔT decreases for a given heat flux as illustrated in Table 5. It is found that $\delta/D_b \approx 1/20$ instead of $\delta/D_b \approx 1/6$ as reported by (11).

Latent heat transport was found to be one of the important mechanisms in boiling heat transfer as shown in Table 6. It increased and became dominant as heat flux went up for all pressures studied.

The liquid vapor exchange mechanism became less effective as pressure increased. It is expected by the equation given by (1), because as pressure increases, D_b , ΔT and δ tend to decrease for the same heat flux. Although N tends to increase, f is found to decrease. The total effect is a decrease in the term

$$\frac{\pi}{3} \delta^2 \left(\frac{3}{2} D_b - \delta \right) (\rho_l - \rho_v) C \frac{\Delta T}{2} N f$$

Table 6 illustrates this result.

According to the reports of (13), (14) and (15), the investigators found an increase in heat transfer rate in boiling

was accomplished by the action of bubble growth and collapse which acted as piston pumping hot liquid from the thermal boundary layer and circulating cold liquid from the bulk to the heating surface. At high system pressure, this mechanism does not dominate the heat transfer rate, as calculated from the equation derived by (1).

A study of Table 6 shows that the equation of (1) gives a satisfactory approximation to predict heat transfer mechanism in nucleate pool boiling especially at low heat flux. The equation gives a lower value in predicting boiling heat transfer at high heat flux because it is based upon the model of a discrete bubble. The liquid vapor exchange mechanism is based upon the exchange of superheated liquid in the thermal boundary layer and the volume of displacement is that part of the thermal boundary layer occupied by the bubble, i.e., the volume of the spherical segment of the bubble immersed by the thermal boundary layer of the heating surface. As reported by (11), when a bubble grows, it forms a thermal boundary layer around its spherical surface, the superheated liquid being partly drawn from the thermal boundary layer of the heating surface and partly from heating up the adjacent bulk liquid. When the bubble departs from the heating surface, it carries with it its own boundary layer and dissolves in the bulk as it collapses. It is expected that the amount of superheated liquid displaced by a bubble is more than that calculated from the equation of (1), but L. G. Hamburger (11) did not show the relationship

between the amount of superheated liquid drawn from the heating surface to the volume of the thermal boundary layer of the bubble. Instead, he assumed that the thermal boundary layer thickness of the bubble is approximately equal to the thickness of the thermal boundary layer of the heating surface, and that the temperature of the liquid vapor interface of the bubble is equal to T_w . According to this model, it was found that it predicted too high a value of liquid vapor exchange mechanism. The possible explanation is that $D_b \gg \delta$, the temperature of liquid vapor interface is not constant, the farther away from the heating surface, the smaller the temperature difference between the bulk and the interface. This, in term, implies that the thermal boundary layer of the bubble does not have a uniform thickness. The farther away from the heating surface, the thinner of δ should be expected, but there is no literature giving an adequate calculation of the thermal boundary thickness of the bubble.

There is another effect that the equation of (1) does not account for. J. L. McGrew, F. L. Bamford and T. R. Rehm (50) reported that extensive liquid circulation had been found near a bubble when it was attached to the heating surface. This effect, called Marangoni flow, provides a contribution to heat transfer which is inversely proportional to the departure frequency (f). From Table 5, it is found that for the same heat flux, N increases as pressure increases, but f tends to decrease. So neglecting the effect of Marangoni flow and the formation of

the thermal boundary layer around the bubble surface may account for the increase of " % difference" as pressure increases as shown in Table 6.

In addition, F. D. Moore and R. B. Mesler (17) reported that when a bubble started to grow, it evaporated part of the adjacent superheated liquid at the heating surface and transferred heat to the bulk liquid due to condensation on the top of the bubble. This effect provides a contribution to heat transfer in nucleate boiling.

There is also a possible enhancement of the convective mechanism once boiling is established.

Since no report exists giving an accurate prediction of the thickness of the thermal boundary layer of the bubble and the amount of heat transfer due to Marangoni flow and condensation of the superheated liquid was found in the literature, the equation given by (1) can be used to obtain a satisfactory approximation of heat transfer rate in nucleate pool boiling.

V. CONCLUSION

1. The equation derived by I. A. Raben, R. T. Beaubouef and G. E. Commerford (1) gave a satisfactory approximation to measured heat transfer rates in nucleate pool boiling under pressure, as studied here.
2. If the effects of condensation-evaporation over the bubble surface, the Marangoni flow and displacement of the thermal boundary layer over the bubble were included in the equation given by (1), it would give a more accurate result.
3. Latent heat transport was found to be an important mechanism in heat transfer in pool boiling. Increase of pressure did not enhance this mechanism.
4. Liquid vapor exchange offered a small fraction of the total heat transfer rate and reduced its contribution as pressure went up.
5. For a constant ΔT , the heat transfer rate was found to increase with pressure.
6. Bubble departure diameter was found to be inversely proportional to pressure.
7. Number of bubbles on the heating surface increased with pressure.
8. Frequency had a tendency to decrease as pressure increased.

TABLE 1 BOILING HEAT TRANSFER DATA

System Pressure P=psia	Water Temp. $T_{\text{sat}} = ^\circ\text{F}$	Surface Temp. $T_w = ^\circ\text{F}$	Differential Temp. $\Delta T = ^\circ\text{F}$	Heat transfer Rate $\dot{Q} = \text{Btu/hr}$	Heat Flux $\frac{\dot{Q}}{A} = \frac{\text{Btu}}{\text{hr-ft}^2}$
4.700	159.62	181.99	22.37	476.00	38,800
4.700	159.62	181.44	21.82	439.50	35,600
4.700	159.62	180.75	21.13	373.60	30,450
4.700	159.62	180.13	20.51	349.50	27,650
4.700	159.62	179.00	19.38	312.00	25,400
4.700	159.62	178.38	18.76	276.00	22,500
4.700	159.62	177.91	18.39	258.00	21,050
4.692	159.54	177.36	17.82	219.20	17,860
4.700	159.62	176.93	17.31	187.50	15,300
4.700	159.62	176.20	16.58	171.40	13,960
4.682	159.45	175.26	15.81	151.80	12,370
4.700	159.62	174.38	14.76	135.00	11,010
4.676	159.40	173.96	14.56	110.10	8,960
4.676	159.40	173.60	14.20	97.40	7,930
4.682	159.45	173.02	13.57	91.50	7,460
4.676	159.40	171.53	12.13	80.30	6,540
4.676	159.40	170.40	11.00	73.70	6,010
4.676	159.40	168.84	9.44	65.50	5,340
4.676	159.40	166.51	7.11	46.20	3,762
4.676	159.40	163.90	4.50	31.65	2,580
4.676	159.40	162.48	3.08	22.20	1,810

TABLE 2 BOILING HEAT TRANSFER DATA

System Pressure P=psia	Water Temp. $T_{\text{sat}} = ^\circ\text{F}$	Surface Temp. $T_w = ^\circ\text{F}$	Differential Temp. $\Delta T = ^\circ\text{F}$	Heat Transfer Rate $\dot{Q} = \text{Btu/hr}$	Heat Flux $\frac{\dot{Q}}{A} = \frac{\text{Btu}}{\text{hr-ft}^2}$
14.593	211.64	230.69	19.05	472.00	38,500
14.604	211.68	229.59	17.91	409.00	33,300
14.593	211.64	228.31	16.67	337.50	27,450
14.568	211.55	226.90	15.35	276.00	22,500
14.568	211.55	226.08	14.53	237.50	19,360
14.506	211.33	225.55	14.22	202.80	16,520
14.516	211.37	225.18	13.81	192.20	15,670
14.541	211.46	224.94	13.48	184.40	15,020
14.506	211.33	224.51	13.18	171.20	13,960
14.506	211.33	223.72	12.39	146.30	11,920
14.506	211.33	223.36	12.03	131.00	10,680
14.506	211.33	222.42	11.09	110.10	8,970
14.506	211.33	222.12	9.79	96.90	7,900
14.506	211.33	220.40	9.10	89.80	7,310
14.506	211.33	219.29	7.96	80.80	6,590
14.506	211.33	218.83	7.50	73.60	6,010
14.506	211.33	217.96	6.63	65.60	5,350
14.506	211.33	216.06	4.73	50.10	4,090
14.506	211.33	214.82	3.49	41.90	3,410
14.506	211.33	213.86	2.53	31.65	2,580
14.506	211.33	213.32	1.99	28.30	2,308

TABLE 3 BOILING HEAT TRANSFER DATA

System Pressure P=psia	Water Temp. T _{sat} =°F	Surface Temp. T _w = °F	Differential Temp. ΔT = °F	Heat Transfer Rate Q̇ = Btu/hr	Heat Flux Q̇/A = $\frac{\text{Btu}}{\text{hr-ft}^2}$
29.786	249.92	265.33	15.41	476.00	38,800
29.786	249.92	265.11	15.19	444.50	36,250
29.635	249.61	264.59	14.98	405.00	32,950
29.573	249.48	264.16	14.68	370.50	30,200
29.699	249.74	263.23	13.49	342.00	27,880
29.786	249.92	263.10	13.18	311.50	25,400
29.786	249.92	262.39	12.47	294.50	24,050
29.786	249.92	262.40	12.48	276.00	22,450
29.893	250.14	262.45	12.31	256.00	20,850
29.893	250.14	262.24	12.09	237.20	19,340
29.780	249.92	261.39	11.47	217.80	17,750
29.786	249.92	261.13	11.21	187.50	15,300
29.786	249.92	260.28	10.36	171.40	13,960
29.786	249.92	259.80	9.88	135.00	11,010
29.786	249.92	258.89	8.97	114.00	9,290
29.679	249.72	257.74	8.02	97.40	7,930
29.679	249.72	256.60	6.88	87.60	7,140
29.573	249.48	253.75	4.27	73.70	6,010
29.786	249.92	252.96	3.04	60.70	4,960
29.573	249.92	251.74	2.26	50.10	4,080
29.635	249.61	250.42	0.81	31.65	2,580

TABLE 4 BOILING HEAT TRANSFER DATA

System Pressure P=psia	Water Temp. T _{sat} =°F	Surface Temp. T _w = °F	Differential Temp. ΔT = °F	Heat Transfer Rate Q̇ = Bru/hr	Heat Flux $\frac{Q}{A} = \frac{\text{Btu}}{\text{ht-ft}^2}$
44.908	274.16	286.72	12.56	476.00	38,800
44.908	274.16	286.45	12.29	401.50	33,720
44.758	273.95	285.89	11.94	373.60	30,450
44.640	273.80	285.18	11.38	342.00	27,880
44.758	273.95	284.61	10.66	324.00	26,400
44.758	273.95	284.86	10.91	311.50	25,400
44.758	273.95	284.06	10.11	294.50	24,050
44.908	274.16	283.74	9.58	260.80	21,250
44.758	273.95	283.11	9.16	237.20	19,340
44.758	273.95	282.80	8.85	203.20	16,550
44.758	273.95	282.46	8.51	184.50	15,020
44.908	274.16	282.06	7.90	164.20	13,370
44.908	274.16	281.80	7.64	150.10	12,210
44.758	273.95	280.96	7.01	135.00	11,010
44.908	274.16	281.02	6.86	114.00	9,290
44.758	273.95	279.84	5.89	97.40	7,930
44.758	273.95	278.21	4.26	87.60	7,140
44.908	274.16	277.79	3.63	83.10	6,770
44.908	274.16	276.36	2.20	73.70	6,010
44.758	273.95	275.05	1.10	50.10	4,080
44.758	273.95	274.70	0.76	41.90	3,410

TABLE 5 BOILING PARAMETERS

$\dot{Q}$ Measured Btu/hr	$\Delta T, ^\circ F$	$D_b, \text{ ft}$	N	f, 1/hr	$\delta, \text{ ft}$
<u>4.70 psia PRESSURE</u>					
97.40	14.20	1.6667×10^{-2}	4	8.4488×10^4	7.255×10^{-4}
171.40	16.58	2.7260×10^{-2}	4	11.8400×10^4	7.322×10^{-4}
		1.2860×10^{-2}	1	4.8600×10^4	
258.00	18.39	3.3940×10^{-2}	4	10.0600×10^4	7.395×10^{-4}
		1.8730×10^{-2}	2	10.3500×10^4	
<u>1 ATMOSPHERE PRESSURE</u>					
110.10	11.09	8.9560×10^{-3}	25	5.5700×10^4	5.325×10^{-4}
184.40	13.48	10.7900×10^{-3}	25	5.9110×10^4	5.550×10^{-4}
		18.7030×10^{-3}	4	6.9340×10^4	
237.50	14.53	11.1100×10^{-3}	12	7.4760×10^4	5.616×10^{-4}
		20.8500×10^{-3}	8	6.2940×10^4	
<u>2 ATMOSPHERES PRESSURE</u>					
87.60	6.88	5.8830×10^{-3}	17	3.0845×10^4	3.805×10^{-4}
114.00	8.97	5.7300×10^{-3}	26	5.2340×10^4	4.355×10^{-4}
171.40	10.36	8.9200×10^{-3}	34	5.9630×10^4	4.682×10^{-4}
<u>3 ATMOSPHERES PRESSURE</u>					
87.60	4.26	4.6130×10^{-3}	32	2.0980×10^4	2.368×10^{-4}
		8.3320×10^{-3}	9	1.0610×10^4	
114.00	6.86	6.0900×10^{-3}	45	2.5370×10^4	3.165×10^{-4}
		8.3060×10^{-3}	5	1.6630×10^4	
164.20	7.90	7.0840×10^{-3}	55	2.5680×10^4	3.445×10^{-4}
		9.3600×10^{-3}	14	2.5770×10^4	

TABLE 6 CALCULATED BOILING PARAMETERS

$\dot{Q}_c$ Btu/hr	q_{λ}^{Nf} Btu/hr	q_{v-l}^{Nf} Btu/hr	$\dot{Q}_{calc}$ Btu/hr	$\dot{Q}_{meas}$ Btu/hr	% Difference
<u>4.70 psia PRESSURE</u>					
85.150	10.490	1.9460	97.5860	97.40	+ 0.1920
84.950	66.592	5.6950	157.2370	171.40	- 8.2600
76.850	115.050	8.3570	200.2570	258.00	-22.3900
<u>1 ATMOSPHERE PRESSURE</u>					
86.700	19.430	1.7150	107.9250	110.10	- 1.9750
84.050	68.840	3.9270	156.8770	184.40	-14.9300
84.750	108.980	4.2810	198.0110	237.50	-16.6100
<u>2 ATMOSPHERES PRESSURE</u>					
84.510	3.808	0.1354	88.4534	87.60	+ 0.9730
94.450	9.125	0.5950	104.1700	114.00	- 8.6250
88.780	51.650	1.8340	142.2640	171.40	-16.9700
<u>3 ATMOSPHERES PRESSURE</u>					
80.180	6.235	0.0405	86.4555	87.60	- 1.3080
91.950	15.710	0.2310	107.8910	114.00	- 5.3550
83.050	40.980	0.5535	124.5835	164.20	-24.0800

NOTATION

A	Heat surface area, ft^2
a_c	Convective heat transfer area, ft^2
C	Specific heat of liquid, $\text{Btu}/\text{lb}_m\text{-}^\circ\text{F}$
D_b	Bubble diameter at departure, ft
f	Frequency of bubble departure from surface, 1/hr
g	Gravitation acceleration, ft/sec^2
g_c	Gravitational conversion factor, $\text{lb}_m\text{-ft}/\text{lb}_f\text{-sec}^2$
h	Heat transfer coefficient, $\text{Btu}/\text{hr-ft}^2\text{-}^\circ\text{F}$
h_c	Convective heat transfer coefficient, $\text{Btu}/\text{hr-ft}^2\text{-}^\circ\text{F}$
K	Thermal conductivity of liquid, $\text{Btu}/\text{hr-ft-}^\circ\text{F}$
N	Number of bubbles on surface
P	System pressure, psi
ΔP	$P_i - P_o$, psi
Q	Heat transfer rate, Btu/hr
q	Heat transfer per bubble, Btu
$\frac{Q}{A}$	Heat flux, $\text{Btu}/\text{hr-ft}^2$
r	Radius of cavity, ft
S	Spacing between two nucleation sites, ft
T	Average temperature at height y in liquid, $^\circ\text{F}$
T_b	Average bulk temperature, $^\circ\text{F}$
T_w	Average heating surface temperature, $^\circ\text{F}$
ΔT	Surface temperature minus saturation temperature of liquid, $^\circ\text{F}$
V_{fg}	Specific volume change in vaporization, ft^3/lb_m
y	Height above heating surface, ft

Subscripts

l	Liquid
sat	Saturation
v	Vapor
v-l	Vapor-liquid transport mechanism
w	Surface

Greek Symbols

β	Bubble contact angle
δ	Thermal boundary layer thickness, ft
λ	Latent heat of vaporization, Btu/lb _m
ρ_l	Mass density of liquid, lb _m /ft ³
ρ_v	Mass density of vapor, lb _m /ft ³
σ	Surface tension, lb _f /ft

BIBLIOGRAPHY

1. I. A. Raben, R. T. Beaubouef and G. E. Commerford, "A Study of Heat Transfer in Nucleate Pool Boiling of Water at Low Pressure." Chem. Engr. Prog. Sym. Ser. Vol. 65, No. 57, pp 249-258, 1965.
2. M. T. Cichelli and C. F. Bonilla, "Heat Transfer to Liquids Boiling under Pressure." Trans. of A.I.Ch.E. Vol. 41, pp 755-787, 1945.
3. W. H. McAdams, J. N. Addoms, P. M. Rinaldo and R.S. Day, "Heat Transfer from Single Horizontal Wire to Boiling Water." Chem. Engr. Prog. Vol. 44, pp 639-646, 1948.
4. K. Nishikawa and K. Yamagata, "On the Correlation of Nucleate Boiling Heat Transfer." Int. J. of Heat and Mass Transfer, Vol. 1, No. 2/3, pp 219-235, 1960.
5. P. J. Berenson, "Experiments on Pool Boiling Heat Transfer." Int. J. of Heat and Mass Transfer, Vol. 5, No. 10, pp 985-999, 1962.
6. E. T. Sauer, H. B. H. Coope, G. A. Akin and W. H. McAdams, "Heat Transfer to Boiling Liquids." Mech. Engr. Vol. 60, pp 669-676, 1938.
7. Y. Y. Hsu, "Gradual Transition of Nucleate Boiling from Discrete Bubble Regime to Multi-bubble Regime." Chem. Engr. Prog. Sym. Ser. Vol. 61, No. 59, pp 290-298, 1965.
8. S. Nukiyama, "Maximum and Minimum Values of Heat Transmitted from a Metal to Boiling Water under Atmospheric Pressure." J. Soc. Mech. Engr. Japan. Vol. 37, No. 206, pp 367-394, 1934.
9. E. R. G. Eckert, "Introduction to Heat and Mass Transfer." McGraw Hill Book Co., Inc., 1959.
10. N. Zuber, "Nucleate Boiling, the Region of Isolated Bubbles and the Similarity with Natural Convection." Int. J. of Heat and Mass Transfer, Vol. 6, No. 1, pp 53-78, 1963.
11. L. G. Hamburger, "On the Growth and Rise Individual Vapor Bubbles in Nucleate Pool Boiling." Int. J. of Heat and Mass Transfer, Vol. 8, No. 11, pp 1369-1386, 1965.
12. R. F. Gaertner, "Photographic Study of Nucleate Pool Boiling on a Horizontal Surface." Trans. A.S.M.E. J. of Heat Transfer, Vol. 87, pp 17-29, 1965.

13. W. M. Rohsenow and J. A. Clark, "A Study of the Mechanism of Boiling Heat Transfer." Trans. A.S.M.E. Vol. 73, pp 609-620, 1951.
14. K. Engelberg-Forster and R. Grief, "Heat Transfer to the Boiling Liquid-mechanism and Correlations." Trans. A.S.M.E. J. of Heat Transfer, Vol. 81, pp 43-53, 1959.
15. Y. P. Chang and N. W. Snyder, "Heat Transfer in Saturated Boiling." Chem. Engr. Prog. Sym. Ser. Vol. 56, No. 30, pp 25-38, 1960.
16. M. Jakob, "Heat Transfer in Evaporation and Condensation I." Mech. Engr. Vol. 58, pp 643-660, 1936.
17. F. D. Moore and R. B. Mesler, "The Measurement of Rapid Surface Temperature Fluctuations During Nucleate Boiling of Water." A.I.Ch.E.J. Vol. 7, No. 4, pp 620-624, 1961.
18. S. G. Bankoff, "A Note on Latent Heat Transfer in Nucleate Boiling." A.I.Ch.E.J. Vol. 8, No. 1, pp 63-65, 1962.
19. T. F. Rogers and R. B. Mesler, "An Experimental Study of Surface Cooling by Bubble during Nucleate Boiling of Water." A.I.Ch.E.J. Vol. 10, No. 5, pp 656-660, 1964.
20. R. C. Hendricks and R. R. Sharp, "Initiation of Cooling due to Bubble Growth on a Heating Surface." NASA TND-2290, April, 1964.
21. B. D. Marcus and D. Dropkin, "Measured Temperature Profiles within the Superheated Boundary Layer above a Horizontal Surface in Saturated Nucleate Pool Boiling of Water." Trans. A.S.M.E. J. of Heat Transfer, Vol. 87, pp 333-341, 1965.
22. N. B. Hospeti and R. B. Mesler, "Deposits Formed Beneath Bubbles During Nucleate Boiling of Radioactive Calcium Sulfate Solutions." Chem. Engr. Prog. Sym. Ser. Vol. 62, No. 64, pp 72-76, 1966.
23. C. J. Rallis and H. H. Jawurek, "Latent Heat Transport in Saturated Nucleate Boiling," Int. J. of Heat and Mass Transfer, Vol. 7, No. 10, pp 1051-1068, 1964.
24. C. P. Costello, C. O. Bock and C. C. Nichols, "A Study of Induced Convective Effects on Saturated Pool Boiling Burnout." Chem. Engr. Prog. Sym. Ser. Vol. 61, No. 59, pp 271-280, 1965.

25. R. Cole and H. L. Shulman. "Bubble Departure Diameter at Subatmospheric Pressure." Chem. Engr. Prog. Sym. Ser. Vol. 62, No. 64, pp 6-16, 1966.
26. M. S. Plesset and S. A. Zwick, "The Growth of Vapor Bubbles in Superheated Liquids." J. of Appl. Phy., Vol. 25, No. 4, pp 493-500, 1954.
27. E. G. Keshock and R. Siegel, "Forces Acting on Bubbles in Nucleate Boiling under Normal and Reduced Gravity Conditions." NASA TND-2299, August, 1964.
28. M. A. Johnson Jr., Javia de la Pena and R. B. Mesler, "Bubble Shapes in Nucleate Boiling." Chem. Engr. Prog. Sym. Ser., Vol. 62, No. 64, pp 1-5, 1966.
29. P. H. Streng, A. Orell and J. W. Westwater, "Microscopic Study on Bubble Growth during Nucleate Boiling." A.I.Ch.E.J., Vol. 7, No. 4, pp 578-586, 1961.
30. D. B. Kirby and J. W. Westwater, "Bubble and Vapor Behaviour on a Heated Horizontal Plate during Pool Boiling Near Burnout." Chem. Engr. Prog. Sym. Ser., Vol. 61, No. 57, pp 238-248, 1965.
31. Y. Y. Hsu, "On the Size of Active Nucleate Cavities on a Heating Surface." Trans. A.S.M.E. J. of Heat Transfer, Vol. 84, pp 207-216, 1962.
32. G. I. Bobrovich and N. N. Mamontova, "A Study of the Mechanism of Nucleate Boiling at High Heat Flux." Int. J. of Heat and Mass Transfer, Vol. 8, No. 11 pp 1421-1424, 1965.
33. V. I. Tolubinsky and J. N. Ostrovsky, "On the Mechanism of Boiling Heat Transfer - Vapour Bubble Growth Rate in Process of Boiling of Liquids Solution and Binary Mixture." Int. J. of Heat and Mass Transfer, Vol. 9, No. 12, pp 1463-1470, 1966.
34. D. S. Cryder and A. C. Finalborgo, "Heat Transmission from Metal Surfaces to Boiling Liquids, Effect of Temperature of the Liquid on Film Coefficient." Trans. A. I.Ch.E., Vol. 33, pp 346-362, 1937.
35. E. A. Farber and R. L. Scorah. "Heat Transfer to Water Boiling under Pressure." Trans. A.S.M.E., Vol. 70, pp 369-380, 1948.

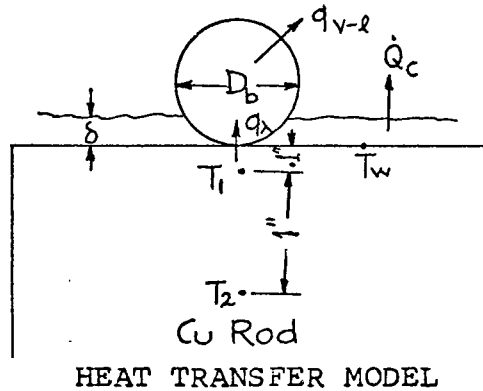
36. W. H. McAdams, W. E. Kennel, C. S. Minder, R. Carl, P. M. Picornell and J. E. Dew, "Heat Transfer at High Rate to Water with Surface Boiling." Ind. and Engr. Chem., Vol. 41, pp 1945-1953, 1949.
37. F. Kreith and M. Summerfield, "Heat Transfer to Water at High Flux Densities, with and without Surface Boiling." Trans. A.S.M.E., Vol. 71, pp 805-815, 1949.
38. S. G. Bankoff, "Surface Temperature at Incipient Boiling." Chem. Engr. Prog. Sym. Ser., Vol. 55, No. 29, pp 87-94, 1959.
39. D. A. Labunstsor, B. A. Kol'chugin, V. S. Golovin and G. V. Tsiklauri, "Study of the Growth of Bubble during Boiling of Saturated Water within a Wide Range of Pressure by Means of High-speed Moving Picture." High Temp., Vol. 2, No. 3, pp 404-409, 1964.
40. E. R. Hosler, "Visual Study of Boiling at High Pressure." Chem. Engr. Prog. Sym. Ser., Vol. 61, No. 57, pp 269-279, 1965.
41. S. A. Kovalev, "An Investigation of Minimum Heat Flux in Pool Boiling of Water." Int. J. of Heat and Mass Transfer, Vol. 9, No. 11, pp 1219-1226, 1966.
42. C. Corty and A. S. Foust, "Surface Variables in Nucleate Boiling." Chem. Engr. Prog. Sym. Ser., Vol. 51, No. 17, pp 1-12, 1955.
43. H. W. Kurchare and J. E. Mayers, "The Effects of Supper Heat and Surface Roughness on Boiling Coefficients," A.I.Ch.E.J., Vol. 6, No. 1, pp 83-91, 1960.
44. Y. Y. Hsu and R. W. Graham, "An Analytical and Experimental Study of the Thermal Boundary Layer and Ebullition Cycle in Nucleate Boiling." NASA TND-594, 1961.
45. R. T. Beaubouef, "An Analysis of Nucleation in Surface Boiling." Master Thesis, University of Houston, 1960.
46. C. P. Costello and W. J. Frea, "A Salient Nonhydrodynamic Effect on Pool Boiling Burnout of Small Semi-cylindrical Heaters." Chem. Engr. Prog. Sym. Ser., Vol. 61, No. 57, pp 258-268, 1965.
47. C. F. Bonilla, J. J. Grady and G. W. Avery, "Pool Boiling Heat Transfer from Scored Surfaces." Chem. Engr. Prog. Sym. Ser., Vol. 61, No. 57, pp 280-288, 1965.

48. P. Griffith and J. D. Wallis, "The Role of Surface Conditions in Nucleate Boiling." Chem. Engr. Prog. Sym. Ser. Vol. 56, No. 30, pp 49-63, 1960.
49. J. H. Keenan, Thermodynamics, John Wiley and Sons, New York, p 436, 1957.
50. J. L. McGrew, F. L. Bamford and T. R. Rehm, "Marangoni Flow: An Additional Mechanism in Boiling Heat Transfer." Science, Vol. 153, pp 1106-1107, 1966.

APPENDIX

Sample Calculation

The following sample calculation illustrates the use of equation derived by (1) in the calculation of heat transfer rate Q



For the calculation, data taken at 1 atmosphere pressure and $\dot{Q}_{meas} = 237.50$ Btu/hr heat flux will be used.

The measured parameters are:

$$T_{sat} = 211.55^{\circ}\text{F}$$

$$T_1 = 226.75^{\circ}\text{F}$$

$$T_2 = 233.44^{\circ}\text{F}$$

For discrete bubbles

$$D_b = 11.11 \times 10^{-3} \text{ ft}$$

$$f = 7.476 \times 10^4 \text{ 1/hr}$$

$$N = 12$$

For merged bubbles

$$D_b = 20.83 \times 10^{-3} \text{ ft}$$

$$f = 6.294 \times 10^4 \text{ 1/hr}$$

$$N = 8$$

Assume linear temperature profile, obtaining

$$\begin{aligned} T_w &= T_1 - (T_2 - T_1) \times 0.1/1.0 \\ &= 226.75 - (233.44 - 226.75)/10. \\ &= 226.08^\circ\text{F} \end{aligned}$$

$$\begin{aligned} \Delta T &= T_w - T_{\text{sat}} = 226.08 - 211.55 \\ &= 14.53^\circ\text{F} \end{aligned}$$

The physical properties were evaluated at the average boundary layer temperature. In this case

$$\begin{aligned} T_{\text{ave}} &= T_{\text{sat}} + \Delta T/2 = 211.55 + 14.53/2 \\ &= 218.815^\circ\text{F} \end{aligned}$$

At this T_{ave}

$$K = 0.38954 \text{ Btu/hr-ft-}^\circ\text{F}$$

$$\rho_l = 59.60 \text{ lb}_m/\text{ft}^3$$

$$C = 1.00 \text{ Btu/lb}_m\text{-}^\circ\text{F}$$

At $\Delta T = 14.53^\circ\text{F}$, the convective heat flux was 10,100 Btu/hr-ft² as obtained from Figure 6 by extrapolating the convective curve. Thus

$$h_c = 10,100/14.53 = 695.12 \text{ Btu/hr-ft}^2\text{-}^\circ\text{F}$$

$$\begin{aligned} \delta &= K/h_c = 0.38954/695.12 \\ &= 5.616 \times 10^{-4} \text{ ft} \end{aligned}$$

$$\begin{aligned} a_c &= A - \frac{\pi}{4} D_b^2 \times N \\ &= (0.7854)(1.5/12)^2 - ((0.7854)(11.11 \times 10^{-2})^2(12) \\ &\quad + (0.7854)(20.83 \times 10^{-3})^2(8)) \\ &= 0.00838 \text{ ft}^2 \end{aligned}$$

$$\begin{aligned} \dot{Q}_c &= h_c a_c \Delta T = (695.12)(0.00838)(14.53) \\ &= 84.75 \text{ Btu/hr} \end{aligned}$$

For discrete bubbles, the Gibbs equation is

$$P_i - P_o = 4\sigma/D_b = 1.11 \times 10^{-4} / 11.11 \times 10^{-3} \\ = 0.01 \text{ psi}$$

$$P_i = P_o + 0.01 = 14.568 + 0.01 \\ = 14.578 \text{ psia}$$

At this saturation pressure,

$$\rho_v = 37.05 \times 10^{-3} \text{ lb}_m/\text{ft}^3 \\ \lambda = 970.56 \text{ Btu/lb}_m \\ q_\lambda = \frac{\pi}{6} D_b^3 \rho_v \lambda = (0.523)(11.11 \times 10^{-3})^3 (3.705 \times 10^{-2})(970.56) \\ = 2.585 \times 10^{-5} \text{ Btu} \\ q_{v-l} = \frac{\pi}{3} \delta^2 \left(\frac{3}{2} D_b - \delta \right) (\rho_l - \rho_v) c \frac{\Delta T}{2} \\ = (1.047)(5.616 \times 10^{-4})^2 (1.5 \times 11.11 \times 10^{-3} - \\ 5.616 \times 10^{-4})(59.6 - 3.705 \times 10^{-2})(1)(14.53/2) \\ = 0.2305 \times 10^{-5} \text{ Btu}$$

For merged bubbles, the Gibbs equation yields

$$P_i - P_o = 4\sigma/D_b = 1.11 \times 10^{-4} / 20.83 \times 10^{-3} \\ = 0.00532 \text{ psi}$$

$$P_i = P_o + 0.00532 = 14.568 + 0.00532 \\ = 14.57332 \text{ psia}$$

At this saturation pressure,

$$\rho_v = 37.035 \times 10^{-2} \text{ lb}_m/\text{ft}^3 \\ \lambda = 970.68 \text{ Btu/lb}_m \\ q_\lambda = \frac{\pi}{6} D_b^3 \rho_v \lambda = (0.523)(20.83 \times 10^{-3})^3 (3.7035 \times 10^{-2})(970.68) \\ = 17.06 \times 10^{-5} \text{ Btu}$$

$$\begin{aligned}
 q_{v-l} &= \frac{\pi}{3} \delta^2 \left(\frac{3}{2} D_b - \delta \right) (\rho_l - \rho_v) c \frac{\Delta T}{2} \\
 &= (1.047)(5.616 \times 10^{-4})^2 (1.5 \times 20.83 \times 10^{-3} - \\
 &\quad 5.616 \times 10^{-4})(59.6 - 3.705 \times 10^{-2})(1)(14.53/2) \\
 &= 0.4405 \times 10^{-5} \text{ Btu}
 \end{aligned}$$

Then

$$\begin{aligned}
 q_{\lambda} \text{ Nf} &= (2.585 \times 10^{-5})(12)(7.476 \times 10^4) \\
 &\quad + (17.06 \times 10^{-5})(8)(6.294 \times 10^4) \\
 &= 108.98 \text{ Btu/hr}
 \end{aligned}$$

$$\begin{aligned}
 q_{v-l} \text{ Nf} &= (0.2305 \times 10^{-5})(12)(7.476 \times 10^4) \\
 &\quad + (0.4405 \times 10^{-5})(8)(6.294 \times 10^4) \\
 &= 4.281 \text{ Btu/hr}
 \end{aligned}$$

$$\begin{aligned}
 \dot{Q}_{\text{calc}} &= 84.75 + 108.98 + 4.281 \\
 &= 198.011 \text{ Btu/hr}
 \end{aligned}$$

$$\dot{Q}_{\text{meas}} = 237.50 \text{ Btu/hr}$$

$$\begin{aligned}
 \% \text{ Difference} &= \frac{198.011 - 237.50}{237.50} \times 100\% \\
 &= -16.61 \%
 \end{aligned}$$