THE RICE UNIVERSITY

A STUDY OF NUCLEATE BOILING
IN AN ACCELERATED SYSTEM

by

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ABSTRACT

A study was made of the influence of system acceleration on nucleate boiling heat transfer in saturated distilled water at approximately atmospheric pressure. Tests were made under standard gravitational acceleration and under 2.85 times the standard gravitational acceleration.

Chromel A electrical resistance wire, 0.008 inch in diameter, served as the heat transfer area.

Acceleration of the system was obtained by use of the centrifuge principle. The boiling system was pivoted from the cross arm of a vertical shaft in such a manner that the acceleration was always normal to the heating wire.

Heat flux rates were varied from approximately $10^4$ to $4 \times 10^5$ BTU's/hr Ft$^2$.

The change in the heat flux as a function of the temperature difference $\Delta T$ between the wire temperature and the saturation temperature of the water was used as a comparison for the standard one and 2.85 times the standard gravitational field.

The following equations of the form $q/A = C(\Delta T)^N$ were obtained:

Standard gravitational field

$$q/A = 2.3 \Delta T^{2.8}$$

2.85 times the standard gravitational field

$$q/A = .42 \Delta T^{3.2}. $$
I. INTRODUCTION

A. PURPOSE

The purpose of this study was to investigate the effect of an increased body force field on a system under boiling conditions. Acceleration of the system by rotation provides the equivalent of a change in the body force field acting on the system.

B. THE BOILING PHENOMENON

1. General

The process of the purposeful conversion of a liquid into a vapor is of obvious engineering importance. Engineers and designers have long been taking advantage of the boiling phenomenon in the conversion of liquids into vapors.

Experiments have shown that if a liquid such as water is distilled and completely degasified it will undergo a liquid-vapor change without the appearance of bubbles when it is evenly heated in a smooth, clean container. Under normal conditions, however, when the rate of heat input to a saturated liquid is great enough, vapor bubbles are observed to be formed next to the heating surface under the free liquid surface. This phenomenon is generally described as nucleate boiling. This thesis will be concerned primarily with that phase of boiling normally described as nucleate boiling.
2. Regimes of Boiling

The existence of several regimes of boiling was first discussed by Nukeyoma in 1934, although Leidenfrost in 1756 and Lang in 1888 discussed maximum and minimum rates of evaporation in boiling.

a. Surface or convective type of boiling

Surface or convective boiling will occur in the range AB on the boiling heat transfer curve in Figure 1. This curve shows the relation of the heat flux q/A in BTU's/ft² to the temperature difference between the heating surface Tw and the bulk water temperature Tsat. The water is being heated by natural convection and evaporation occurs only at the surface of the liquid. In this range, experiments have shown that q/A is proportional to T⁵/⁴.

b. Nucleate boiling

(1) Definition

Nucleate boiling may be defined as boiling that takes place with the formation of vapor bubbles at active nuclei along the heated surface. These bubbles form, grow, and break away from the heating surface, creating turbulence and setting up currents which greatly increase the rate of heat transfer. This phase of boiling is shown on the curve in Figure 1 as BC. Point C is known as the point of maximum heat flux with nucleate boiling. The temperature difference

*Numbers denote items in the bibliography on Page 43.
Figure 1. Boiling Heat Transfer Curve
here usually referred to as the "critical ΔT." The higher heat transfer rates associated with nucleate boiling over that of convection or surface boiling have been ascribed to the action of the bubbles as agitators in the "laminar sub-layer" rather than as a means of energy transport themselves.\(^3\)

(2) Mechanisms of boiling\(^4\)

The mechanisms of nucleate boiling have been considered in the following stages:

(a) The presence or formation of a nucleus from which the growth of a bubble may start. This nucleus formation will depend on the surface tension of the heated fluid and the degree of superheat of the fluid next to the heating surface. The location of the nuclei formation will depend on the condition of the heating surface. Any dissolved gases will also increase the rate of nucleus formation.

(b) Growth of the vapor bubble

Forrester and Zuber,\(^5\) combining Rayleigh's equation of motion of a bubble, the law of mechanics for the pressure due to surface tension, and the Clausius-Clapeyron equation, have shown the initial growth rate of a bubble is slow due to the surface tension, but after the radius of the bubble has increased to a size where the surface tension has little effect on the rate of growth the rate of growth of the bubble is very large. During this
later stage, the growth rate is dependent upon the rate of heat transfer across the bubble surface; therefore, the rate of growth is dependent upon the degree of superheat of the surrounding liquid.

(c) Departure of the bubble

After the superheat surrounding the bubble has been consumed, the bubble will either collapse at the heating surface, depart from the heating surface and collapse due to the bulk temperature of the surrounding liquid being subcooled, or depart from the heating surface and rise to the free liquid surface.

Experiments have shown that the number $n$ of active nuclei is proportional to the net heat flux $(n \sim \frac{q}{A})$. As the surface temperature is increased, the number of active nuclei is increased. This increases the agitation next to the surface and tends to increase the heat flux to the liquid.

c. Film boiling

As the temperature difference $(T_w - T_{sat})$ is further increased, the number of vapor bubbles is increased until the entire surface becomes "vapor blanketed." This results in a process called film boiling. While the heat transfer rate associated with vigorous nucleate boiling is high due to the agitation by the bubbles near the heating surface, the heat transfer rate associated with film boiling is much lower because of the insulating effect of the
vapor blanket. On Figure 1, CD would be a regime of partial film boiling while DE would be true film boiling.

An example of film boiling is the Leidenfrost phenomenon of water droplets' "dancing" on a very hot surface. The droplets do not evaporate rapidly because of the insulating vapor film formed between the hot surface and the droplets. Point D on Figure 1 is called the Leidenfrost point.

C. RELATIONSHIPS IN THE LITERATURE SHOWING AN EFFECT OF ACCELERATION

1. Rohsenow

\[ \frac{C_{pe} \Delta T}{h_{fg}} = C_{ef} \left( \frac{q/A}{\mu_s h_{fg}} \right)^{1/2} \left( \frac{C_{pe} \mu_s}{K_e} \right)^{1.7} \]

The terms are as follows: \( C_{pe} \) is the specific heat of the liquid, \( \Delta T \) is the temperature difference between the heating surface and the fluid temperature, \( h_{fg} \) is the latent heat of vaporization, \( C_{ef} \) is a constant determined experimentally, \( q/A \) is the heat flux in BTU's/Hr Ft\(^2\), \( \mu_s \) is the liquid viscosity, \( g_0 \) is a conversion factor, \( \gamma \) is the surface tension of the liquid vapor interface, \( g \) is the acceleration of gravity, \( \rho_s \) is the density of the liquid, \( \rho_v \) is the density of the vapor, and \( K_e \) is the thermal conductivity of the liquid.

In arriving at the above equation, Rohsenow postulated that most of the heat is transferred directly from the surface to the liquid and that the bubbles act
as agitators. Using this assumption, he looked to the 
heat transfer in forced convection turbulent flow without 
boiling for a comparison. In this latter case the 
heat transfer rate is correlated by a relation

\[ N_{NU} = \varphi (N_{RE}, N_{PR}) \]  

(2)

where \( N_{NU} \) is the Nusselt number, \( N_{RE} \) the Reynolds number, 
and \( N_{PR} \) the Prandtl number.

The \( N_{RE} \) in this relation measures the magni-
tude of a kind of agitation associated with turbulent 
flow. For the boiling correlation, this \( N_{RE} \) would indi-
cate the magnitude of the stirring associated with the 
bubble motion.

Rohsenow defined a bubble Reynolds number as

\[ N_{RE_b} = \frac{G_b D_b}{\mu} \]  

(3)

where \( G_b \) is the average mass velocity of the vapor bubbles 
receding from the surface and \( D_b \) is the bubble diameter 
just before departure or collapse.

He used a relationship developed by Fritz for 
the diameter of a bubble.

\[ D_b = C_d \beta \left( \frac{2\varrho_b T}{\varrho_f (\rho_f - \rho_b)} \right)^{1/2} \]  

(4)

The average mass velocity of the bubbles leaving 
the surface was defined as

\[ G_b = \frac{\pi}{6} D_b^3 \rho_v \eta^f \]
These values were substituted in the definition of the bubble Reynolds number and then by dimensional analysis equation (1) was obtained. The constants in this equation are determined by comparison with experimental results.

Equation (1) may be rearranged to give

\[ \frac{g \Delta h}{A} = \left( \frac{C_s}{4} \right)^{1/2} \left( \frac{\rho_v - \rho_l}{A} \right)^{1/2} \frac{\kappa}{h_f} \frac{s_l}{N_{PR}} \left[ - \frac{C_{2\delta}}{C_{3\delta}} \right]^{-3} \Delta T^{-3} \]  

or

\[ N_{NU_b} = \frac{1}{C_{3f}} N_{RE_b}^{1.47} N_{PR}^{-0.7} \]  

Equation (6) may be compared to the equation for non-boiling forced convection.

\[ N_{NU} = C N_{RE}^{m} N_{PP}^{4/3} \]  

where \( m \) is an experimental constant in the range of 0.5 to 0.7.

The gravitational term enters the above equations as a result of using Fritz's relationship for the bubble diameter.

2. Chang

Chang uses a new and novel approach to explain the boiling phenomenon. He uses an extension of a wave analysis for natural convection and applies this to the
boiling state. The following equation is obtained.

\[ h = 0.146 K_b \left( \frac{P_x}{k_b} - \frac{P_x}{T} \Delta T \right)^{1/3} \]  

where \( h \) is the film coefficient at the heating surface and \( \Delta T \) is the coefficient of expansion of the vapor. Then

\[ K_b = K + C \rho_x \varepsilon \]  

and is called the equivalent thermal conductivity of boiling liquid where

\[ \varepsilon = C \left( \frac{\rho_x}{h f_b} \right) \left( \frac{1}{\rho_u} \frac{1}{\beta^2} \right)^N \]  

\[ C \text{ and } N \text{ are experimentally determined constants, } \beta \text{ is the} \]  

bubble contact angle in radians and \( \beta \) is the liquid viscosity. Substituting a value of \( \varepsilon \) arrived at by dimensional analysis into equation (8) gives

\[ h = 0.146 K \left[ 1 + N_{PR} \left\{ C_1 \left( \frac{h f_b}{\rho_u} \right)^N - 1 \right\} \right] \]  

If

\[ N_{PR} \left\{ C_1 \left( \frac{h f_b}{\rho_u} \right)^N - 1 \right\} > 1 - N_{PR} \]

one can solve for \( q/A \) and obtain the following relation,

\[ \frac{q_{1A}}{T} = K, \frac{q_{1A}}{T} \Delta T \Delta T \]  

where

\[ K, = \left[ 0.056 C \rho_x \left( \frac{\mu x}{T} \right)^{1/2} \left( \frac{\mu x}{h f_b P_b \beta^2} \right)^N \right]^{2/3 - N} \]
The terms $C_1$ and $N$ are constants and are obtained from experimental data.

In attempting to apply equation (11) to an accelerating system, not only will it be necessary to determine the constants $C_1$ and $N$ but also $K_1$, since the function of the properties and characteristics expressed in the equation for $K_1$ will be directly affected by the accelerated system. For example, these properties affected will be the density, $\rho_s$ and $\rho_v$, and also the bubble contact angle $\beta$.

3. Addoms\textsuperscript{1,3}

Addoms, whose work is cited in MoAdams and in the report by Merte and Clark, correlated experimental peak flux of several liquids and plotted

$$\frac{(\text{g} / \text{A})_d}{h_f \rho_v} \left( \frac{\rho_s}{\rho_v \epsilon \epsilon_p} \right)^{\frac{4}{3}} \cdot \frac{\rho_s - \rho_v}{\rho_v}.$$

The dimensionless abscissa contains the buoyancy term $\frac{\rho_s - \rho_v}{\rho_v}$ which is a relation of liquid and vapor densities. The dimensionless ordinate contains the average volumetric vapor disengaging rate per unit surface area, $\frac{\text{g} / \text{A}}{h_f \rho_v}$, divided by the cube root of the product of the thermal diffusivity in the liquid and the acceleration due to gravity, $(\frac{\rho_s}{\rho_v \epsilon \epsilon_p})^{\frac{4}{3}}$.

Merte and Clark have reviewed the plotted information of Addoms and have found the best line of data yields
an expression in the form

\[
\left( \frac{q}{h} \right)_p = C \text{h}_{fg} \rho_v \left( \frac{q}{\text{h}_{fg}} \right)^{1/5} \left( \frac{\rho_v - \rho_l}{\rho_v} \right)^N
\]  

(13)

where \( C = 2 \) and \( N = 1/2 \).

4. Zuber

Zuber formulated several equations regarding heat transfer in boiling. In one of the equations developed by Forrester and Zuber, the gravitational term is neglected. This equation is developed as the equation above by Rohsenow. The only real exception to the derivation is in the definition of the Reynolds number. Zuber used the rate of bubble growth in defining the Reynolds number which is developed with the use of Rayleigh's equation for bubble growth.

\[
\rho_v \left[ R \left( \frac{d^2 R}{d t^2} \right) + \frac{3}{2} \left( \frac{d R}{d t} \right)^2 \right] = \Delta P
\]  

(14)

\( R \) is the bubble radius, \( t \) is time and \( \Delta P \) is the pressure difference on the inside and outside of the vapor bubble. Zuber's equation for boiling heat transfer is

\[
\frac{(q/h)_p}{\Delta T K_a} = 0.015 \left( \frac{R \frac{d R}{d t} \rho_v}{\mu} \right)^{1/2} \left( \frac{\mu \rho_v C_{pe}}{K_a} \right)^{33}
\]  

(15)

Zuber also states that if transition boiling is analysed instead of nucleate boiling, it is possible then with the use of a hydrodynamic stability analysis to derive an expression for the peak nucleate heat flux in pool boiling.
of sub-cooled liquid. This is

\[ g = \frac{h_{fg}}{24} \rho_v \frac{\lambda}{L} + \frac{\pi}{24} \rho_v \frac{\lambda}{L} c_p \Delta T \]

\[ + \sqrt{2\pi} \frac{K_x}{(Q_x \lambda)^{1/2}} \Delta T \]  \hspace{1cm} (16)

where

\[ \frac{\lambda}{L} = \left[ \frac{T_g (\rho_a - \rho_v)}{\rho_v^2} \right]^{1/4} \left[ \frac{\rho_a}{\rho_a + \rho_v} \right]^{1/2} \]

\[ \lambda^* = 2\pi \left[ \frac{T_g (\rho_a - \rho_v)}{\rho_v^2} \right]^{1/4} \left[ \frac{\rho_v^2}{T_g (\rho_a - \rho_v)} \right] \left[ \frac{\rho_a + \rho_v}{\rho_a} \right]^{1/2} \]  \hspace{1cm} (17)

and

The first term on the right-hand side of the equation represents the energy required to generate the critical vapor mass flow which causes stability. The second term represents the energy required to increase the enthalpy of the liquid from bulk liquid temperature to saturation temperature. The third term represents the energy transferred from the liquid-vapor interface which is at saturation temperature to bulk liquid temperature which is sub-cooled.

Since we are concerned primarily with the first term, this equation may be reduced to the following:

\[ (\Phi\lambda)_p = \frac{\pi}{24} \frac{h_{fg}}{\rho_v} \rho_v^{1/2} \left[ \frac{T_g (\rho_a - \rho_v)}{\rho_v^2} \right]^{1/4} \left[ \frac{\rho_a + \rho_v}{\rho_a} \right]^{1/2} \]  \hspace{1cm} (19)
5. Kulateladze\(^8\)

Kulateladze, starting with the non-linear Euler equation of motion for two phases and the heat transfer equation based on dimensional analysis, arrived at the following equation:

\[
\frac{q}{A} = K \frac{g}{\lambda} \left( \frac{\rho_2}{\rho_1} \right)^{1/2} \left( \frac{\rho_2 - \rho_1}{\gamma} \right)^{1/4}.
\]  

(20)

\(K\) is a constant arrived at by experiment and is found to be approximately 0.16.

Bouahanski, using Kulateladze's work, extended the analysis using a viscosity term \(N\) to obtain

\[
N = \frac{\rho_2 \left( \frac{\rho_2}{\rho_1} \right)^{1/2}}{\mu_2 \left( \frac{\rho_2 - \rho_1}{\gamma} \right)^{1/2}}.
\]  

(21)

Plotting \(K\) versus \(N\) was found to yield an equation of the form

\[
K = 0.13 + 4N^{-0.4}.
\]  

(22)

It should be noted that, except for the equation by Rohsenow, the heat flux is given as proportional to the gravitational acceleration with an exponent of either \(1/3\) or \(1/4\). Rohsenow's equation for the heat flux is proportional to the gravitational acceleration to the \(1/2\) power.
D. PREVIOUS WORK IN BOILING HEAT TRANSFER ENVOVLING AN 
ALTERATION OF THE GRAVITATIONAL FIELD

1. Merté and Clark\(^3\)

Merté and Clark of the University of Michigan performed an experiment with a heater at the bottom of a pool. The heater was a chrome-plated copper plate. The system was placed in a centrifuge and rotated so that the resultant acceleration field was always normal to the heated surface. Their tests included a range of from 1 to 21 times the standard gravitational force. (See Figure 2). There seems to be very little effect on the position of the \(q/A\) versus \((T_w - T_{sat})\) curves at the higher heat fluxes. The displacement at the lower heat fluxes is probably due to the effect of superimposed convection effects.

2. Siegel and Usiskin\(^10\)

At the NASA Laboratories, a ribbon heater immersed in a beaker of water was photographed during free fall and the results were reported by Siegel and Usiskin. In each case the heat fluxes were in the nucleate range under a normal gravity field. In the free fall condition with the lower heat fluxes the bubbles grew while remaining attached to the heating surface. At the large heat fluxes a very large vapor blanket was formed around the heater.

Subsequently, Siegel and Usiskin added a small amount of friction to the free fall system raising the force field to approximately 0.09 times the normal gravitational
Figure 2. Results of Experiments by Merte and Clark
force. With this small force field it is reported\textsuperscript{4} that nucleate boiling continued during the fall.

The free fall distance was approximately 8 feet and Siegel and Usiskin states that the data is insufficient to make any quantitative statement of the above hypothesis.

3. Other

Tests have been run with flowing fluids in a tube. The heat transfer rates have been increased by having tighter twists in the tubes. This increase has been attributed to the increased force field, however, the contribution of forced convection could not be isolated.

Several experiments with condensation on rotating discs have been made.
II. TEST APPARATUS

A. CENTRIFUGE

The force field was increased by the use of the centrifuge principle. (See Figure 3.) The centrifuge was constructed of the following major parts:

1. Motor

The centrifuge was rotated by means of a 1/3-horsepower, 115-volt, AC, gear motor. The rotational speed was 35 rpm. The speed of the centrifuge was varied by changing the sheave sizes on the motor and the centrifuge shafts.

2. Rotating Shaft

The rotating shaft was a one-inch steel shaft supported by two block bearings on the apparatus frame. The length of the shaft was approximately 35 inches long.

3. Rotating Pivot Arm

A rotating arm was attached to the top of the shaft. The arm was constructed of a 3/8-inch by 1 1/4-inch steel strip and 5/8-inch diameter steel rods. The arm as shown in Figure 3 was constructed so that as the shaft rotated the arm would pivot out in order to place the container and heating wire always normal to the force field.

4. Counter Weight

The test container was counter-balanced, as shown in Figure 3, with a container holding small brass blocks. This container was constructed of four-inch pipe.
Figure 3
Schematic Drawing of Test Equipment

- Slip Rings
- Heating Coils
- Test Heating Wire
- Insulation
- Motor
- Counter-weights
5. Slip Ring Assembly

The power to the rotating test container was conducted through slip rings located on the rotating shaft. The slip rings are shown in Figure 4. The rings were made of ABCO copper alloy No. 103. The contact width of the slip rings was 1/2-inch. The slip rings were isolated from each other and the rotating shaft by the use of Micarta material.

The slip ring brushes were constructed of ABCO copper alloy No. 100. Brush contact area was 1/4 by 1/2 inch. Each brush was pushed against the slip rings by 1 by 5/16-inch-diameter springs. The brush holder was made of Micarta material. No measurable variation of the resistance of the slip rings was observed for different rotational speeds of the shaft.

6. Wiring

All electrical wiring was 12-gauge, Anaconda Densheath 900, type TW, 600V, oil and moisture-resistant wire. This large size of wire was used to obtain a high current-carrying capacity and a low resistance of the circuits.

7. Test Water Container

The water container was constructed of 16-gauge Inconel sheets. The container was 12 inches long by 6 inches wide by 6-1/4 inches deep. The outside of the container was insulated with 1-1/2 inches of insulating material. The container and insulation were saddled in a swing which was connected to the pivot arm.
Figure 4. Slip Ring Assembly
3. Heaters

The water in the test container was maintained at the saturation temperature by the use of two Precision electrical heaters. The total wattage of the heaters was 950 watts. The heater coils were placed around the inside edge of the water container so that the convection currents caused by these heaters would have a minimum effect on the test heating wire.

9. Thermometer

The water temperature was taken with a calibrated mercury thermometer. The heater size was such that the water was maintained at the saturation temperature and the size of the container was such that the temperature was measured to be uniform throughout. The accuracy of the thermometer was within 1/2 degree.

B. TEST HEATING WIRE

The heating wire used for the test was 0.008-inch diameter, Chromel-A resistance wire. The initial plans were to use 0.005-inch-diameter, Chromel-A resistance wire, but with this smaller wire local film boiling was experienced and the wire burned out frequently. This problem was eliminated with the use of the larger diameter wire.

The wire was calibrated for resistance change with an increase in temperature. This calibration was made by placing the wire in an electric furnace and measuring the resistance of the wire at different elevated temperatures.
Figure 5. Wire Temperature vs. Increase in Resistance Curve
The percentage increase in resistance is shown in Figure 5.

The procedure for determining the temperature of the wire during the test was as follows:

The total resistance of the system including the heating wire was determined by dividing the amperes into the voltage:

\[ R_T = \frac{V}{A} \]

The total resistance is equal to the resistance of the wire plus the resistance of the system:

\[ R_T = R_W + R_S \]

The resistance of the system was constant throughout the tests.

The percent increase in wire resistance is equal to the measured wire resistance minus the measured wire resistance at ambient temperature, divided by this measured wire resistance at ambient temperatures:

\[ \% \text{ Increase in Resistance} = \left( \frac{R_W - R_{Wk}}{R_{Wk}} \right) \times 100 \]

From this calculated percent increase in wire resistance, the temperature of the wire was determined from the calibration curve shown in Figure 5.

The length of the test wire was 2 5/16 inches.
C. WATER
The water used in making the tests was distilled water supplied by Silver Seal Distilled Water Company.

D. POWER SUPPLY
The power supply for the test was rectified A.C. The rectifier used was the property of the Rice University Mechanical Engineering Department and was denoted as No. 1375. The heat flux was controlled by a variable resistor located on the rectifier. During a test, the heat flux to the heating wire was varied. A schematic diagram of the electrical system for the test wire is shown in Figure 6.

E. INSTRUMENTS
1. Voltmeter
The voltmeter was a Weston D.C. voltmeter, Model 45. The manufacturer claims accuracy within ± 1/2 of 1% of full scale reading. Full scale reading was 15 volts, which gives an accuracy of ± 0.075 volts.

2. Ammeter
The ammeter was a General Electric D.C. ammeter No. 406705. The manufacturer claims accuracy of ± .2% of full scale reading. The full scale reading was 5 amperes, which gives an accuracy of ± 0.010 amperes.

3. Ohmmeter
The ohmmeter used was a General Electric portable Double Bridge. Accuracy range is as follows:
0 - .22 ohms range ± 0.0002 ohms
0 - 4.4 ohms range ± 0.004 ohms.
III. TEST PROCEDURES

A. WATER

The water was distilled and was completely degasified by vigorous boiling after it was placed in the test container. The resistance was measured with a Simpson meter by placing the leads directly into the water container. The leads were one inch apart, and the resistance varied from 55,000 to 60,000 ohms per inch.

B. HEAT FLUX

The heat flux was controlled by a variable resistor in the circuit, which was set at different positions to vary the heat flux to the wire during a test. The heat flux was determined in the following manner:

$$\text{Watts} = (\text{Amperes})^2 \times R_w$$

$$\frac{\text{q/A}}{\text{Watts x Conversion Factor}} = \frac{\text{Watts x Conversion Factor}}{\text{Area of Wire}}$$

$$\frac{\text{q/A}}{\text{Area of Wire}} = 3.434 \times 10^3 \times \text{Watts} = \text{BTU's/Hr Ft}^2$$

The heat flux was varied in the tests from approximately $1.8 \times 10^4$ BTU's/Hr Ft$^2$ to approximately $4.0 \times 10^5$ BTU's/Hr Ft$^2$.

A certain degree of sub-cooling was experienced in the lower nucleate region. This degree of sub-cooling was more noticeable in the standard gravitational field, since this sub-cooling was noticed up to a heat flux of approximately $3.5 \times 10^4$ BTU's/Hr Ft$^2$. In the acceleration range
of 2.85 times the standard gravitational acceleration, this sub-cooling effect was not noticed after heat fluxes of approximately $1.8 \times 10^4$ BTU's/hr ft$^2$. This reduction in the sub-cooling was probably due to convective currents which assisted in the nucleation process.

C. ACCELERATION OF SYSTEM

The increased acceleration of the system was determined in the following manner:

As the test container was rotated, the revolutions were counted and timed by the use of a stop watch. The procedure was to count 100 revolutions at the beginning of the tests and 100 revolutions near the end of the tests; also, during the tests periodic checks were made by counting 50 revolutions. In all cases these checks were identical as to the revolutions per minute.

The length of the lever arm was determined in the following manner:

As the unit was rotated, the height of the water container above the floor was measured. After the test was completed the container was raised to this measured height and the angle with the rotating arm was measured. The length of the lever arm was determined from the relation

$$L = L_1 + L_2 \cos \alpha$$

which can be derived by reference to Figure 7.
Figure 7: Test Vessel Dimensions Used for Calculation of the Acceleration at the Heating Surface
With the length of this lever arm, the outward acceleration was determined:

\[ a_T = L \omega^2 \]

The acceleration normal to the heating wire is equal to

\[ a_N = \sqrt{a_T^2 + g^2} \]

D. STEADY-STATE CONDITIONS

When conditions such as acceleration change or water was added to the system, sufficient time was allowed for the attainment of steady state conditions before data was taken. This ranged from 8 to 20 minutes, depending primarily on the amount of water added to the system.

E. CRITERIA FOR ACCEPTABLE DATA

For the tests reported, the following conditions had to be fulfilled for acceptability of data:

1. The resistivity of the test water at the beginning and conclusion of the run had to be at least 55,000 ohms per inch.

2. The depth of the water above the test heating wire at the start and at the end of each test period must be 2-1/4 inches plus or minus 1/16 inch.

3. The value of the voltage supplied to the system for each individual test must not vary over 0.02 volts.

4. The value of the amperage supplied to the system for each individual test must not vary over 0.02 amperes.
IV. TEST RESULTS

In the original design of the centrifuge, it was planned to run tests in increased force fields from 1 to 10 times the standard gravitational force. However, as the unit was rotated more rapidly, the amount of water evaporating caused such a degree of unbalance that steady-state conditions could not take place. The major problem was vibration caused by this slight unbalance transferred to the slip rings which in turn caused the slip rings to vary slightly in resistance. This resistance variation was unsteady and, therefore, could not be compensated.

It was decided to concentrate on reproducing the boiling curve as shown in Figure 1 in the nucleate boiling region for force fields of 1 and 2.85 times the standard gravitational force. These results are shown in Figures 8 and 9.

The rotational speed to produce a force field of 2.85 times the standard gravitational force was approximately 57 rpm.

For further tests it is felt that some sort of condensing system must be developed so that there would be no loss of water from the test container. With the present system, only 2 to 3 individual tests could be made before the unit had to be stopped and water added. Also, if the author has the opportunity to run future tests of this sort, it is felt that the test container should be immersed in a bath heater and the water in the test container should be maintained
Figure 8

\[ a_n/g = 1.00 \]
Figure 9

\[ \frac{q}{A} \text{ BTU's/hr ft}^2 \]

\[ T_w - T_{sat} \]

\[ a_n/g = 2.85 \]
at the saturation temperature in this manner. This should eliminate any convective currents set up in the test container by the heating coils.
V. ANALYSIS OF TEST RESULTS

I. GENERAL

Because of the limited quantity of data obtained and the very small range of conditions covered, it is believed that it would be premature and incorrect to attempt any direct correlation of the data.

The literature indicates that for a given system the following relationship was found to hold true over a large part of the nucleate boiling range:

\[ \frac{q}{A} = C(\Delta T)^N \]

where \( C \) and \( N \) are constants experimentally determined.

As a general rule, most of the data indicates \( N \) to be between 3 and 4; however, cases have been found for \( N \) to vary from 3 to 24.

The data obtained by the author for the standard gravitational field when put into the form of \( \frac{q}{A} = C(\Delta T)^N \) gives the following equation:

\[ \frac{q}{A} = 2.34 \Delta T^{2.8} \]

The data for 2.85 times the gravitational field gives the following equation:

\[ \frac{q}{A} = 0.42 \Delta T^{3.2} \]

The above equations were derived using the method of least squares.

The data by Merte and Clark indicates in the lower heat flux region of nucleate boiling a greater \( \frac{q}{A} \) is obtained with a smaller temperature difference (refer to Figure 2);
however, their tests indicate that the boiling curve for increased force fields crosses the curve for the stationery case. The tests run here indicate the boiling curve for an increased force field lies to the right of the boiling curve for standard gravity. This agrees with the data of Nette and Clark. The slope of this increased force field curve, however, indicates that the curves will intersect; and perhaps at a very high heat flux rate, an increased force field would increase the heat transfer coefficient. (Refer to Figure 10). Figure 10 also includes as a comparison the predicted increase in the heat flux with an increased force field using the relationship by Rohsenow. In Figure 10, the relationship by Rohsenow is assumed to follow the standard boiling curve obtained by our experiment. The increase to 2.85 times the gravitational force using Rohsenow's relationship would be an increase in the predicted heat flux to $(2.85)^{1/2}$ times the heat flux at standard conditions.

Experiments have indicated that the bubble size decreases as the heat flux rate is increased. The data on the boiling phenomenon in increased force fields is not sufficient to formulate a theory as to the exact effect of the increased force field. However, from the data available, one might visualize a decrease in the bubble diameter in an increased force field. This smaller bubble diameter could decrease the amount of turbulence in the boundary layer next to the heating surface. This decrease in
turbulence could result in a reduction in the heat transfer rate. As the number of bubbles is increased for higher heat fluxes, the increase in turbulence plus the increased buoyant forces could increase the flux rates. The intersection of the standard boiling curve by the increased force field curve at high heat transfer rates could, therefore, be a result of the increased buoyant forces.

B. CONCLUDING REMARKS

These experiments have only touched on the boiling phenomena. The interest of the experimenter has increased to the point where he would like to attempt to correlate a photographic study and a more detailed study as above. Experiments by Addoms using various size heating wires indicate a similar curve to the one by热水 and Clark. These tests, plus the tests by Jakob and others which indicate a reduction in the bubble size at higher heat flux rates, could indicate that an increased force field would introduce a parameter determined by the bubble growth rate and bubble size.
Figure 10. Composite Data Curves
APPENDIX 1: Notation

A, ......................... Area ft$^2$

$s_n$ ....................... Acceleration normal to heating surface

$s_T$ ....................... Acceleration outward

$s_n/g$ ..................... Dimensionless acceleration

b ......................... Bubbles

C, C$_1$, C$_d$, C$_{sf}$ .... Constants

C$_{P1}$ ..................... Specific heat BTU's/lb - °F

D$_b$ ....................... Bubble diameter ft

f ........................ Frequency of bubble formation

G$_b$ ....................... Average mass velocity of the vapor bubbles leaving the heating surface lb$_m$/Hr

$g$ ................. Local acceleration of gravity ft/sec$^2$

$g_0$ ..................... Conversion factor

h ........................ Film coefficient of heat transfer by convection BTU's/Hr Ft$^2$ °F

$h_{fg}$ .................... Latent heat of vaporization BTU/lb

I .......................... Amperes

K, K$_1$ ..................... Constants

K$_l$ ....................... Thermal conductivity BTU's/Hr Ft$^2$ °F/Ft

L ......................... Length of lever arm ft.

m ........................ Constant

N ........................ Constant

n ......................... Number of Nuclei formation locations

P ........................ Pressure psfa
\( \Delta P \) \hspace{1cm} \text{Pressure difference between inside and outside of bubble} \\
\( q \) \hspace{1cm} \text{Heat flow BTU's/HR} \\
\( q/A \) \hspace{1cm} \text{Heat flux rate BTU's/HR Ft}^2 \\
\( (q/A)_p \) \hspace{1cm} \text{Peak heat flux BTU's/HR Ft}^2 \\
\( R \) \hspace{1cm} \text{Bubble radius ft} \\
\( R_s \) \hspace{1cm} \text{Resistance of system ohms} \\
\( R_T \) \hspace{1cm} \text{Total Resistance ohms} \\
\( R_w \) \hspace{1cm} \text{Resistance of test wire ohms} \\
\( R_{W_k} \) \hspace{1cm} \text{Resistance of test wire at ambient conditions ohms} \\
\( T \) \hspace{1cm} \text{Temperature °F} \\
\( T_w \) \hspace{1cm} \text{Temperature of wire °F} \\
\( T_{sat} \) \hspace{1cm} \text{Temperature of water °F} \\
\( \Delta T \) \hspace{1cm} \text{T}_w - \text{T}_{sat} \\
\( t \) \hspace{1cm} \text{Time} \\
\( V \) \hspace{1cm} \text{Volts} \\

\( N_{NU} \) \hspace{1cm} \text{Nusselt number} \\
\( N_{NU_B} \) \hspace{1cm} \text{Bubble Nusselt number} \\
\( N_{PR} \) \hspace{1cm} \text{Prandtl number} \\
\( N_{RE} \) \hspace{1cm} \text{Reynolds number} \\
\( N_{RE_B} \) \hspace{1cm} \text{Bubble Reynolds number} \\

\( \alpha \) \hspace{1cm} \text{Thermal diffusivity ft}^2/hr \\
\( \beta \) \hspace{1cm} \text{Bubble contact angle radians} \\
\( \beta_r \) \hspace{1cm} \text{Thermal coefficient of expansion} \\
\( \mu \) \hspace{1cm} \text{Liquid viscosity lb/ft hr}
\( \nu \) ........................ Kinematic viscosity \( \text{ft}^2/\text{hr} \)

\( \rho \) ........................ Liquid density \( \text{lb/ft}^3 \)

\( \rho_v \) ........................ Vapor density \( \text{lb/ft}^3 \)

\( \sigma \) ........................ Surface tension \( \text{lb/ft} \)

\( \epsilon \) ........................ Eddy diffusivity \( \text{ft}^2/\text{hr} \)

\( \psi \) ........................ Function
APPENDIX 2: Bibliography


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