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POOL BOILING IN AN ACCELERATING SYSTEM

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TABLE OF CONTENTS

	Page
Introduction	1
Experimental Apparatus	3
Results	8
Non-Boiling Data	8
Boiling Data	8
Discussion of Results	18
Acknowledgement	23
NOMENCLATURE	25
REFERENCES	27

ABSTRACT

A study is reported of the influence of system acceleration (1 to 21 g's) on pool boiling heat transfer using distilled water at approximately atmospheric pressure. The acceleration of the system is such that the resulting force field is normal to the heating surface, thereby increasing the buoyant forces acting on the vapor bubbles.

Heat flux rate is varied from approximately, 5,000 Btu/hr-ft^2 (non-boiling) to 100,000 Btu/hr-ft^2. Data is presented for the influence of subcooling with the boiling system under acceleration at the lower values of heat flux.

A preliminary analysis is presented for a theoretical description of the process of boiling under the influence of high acceleration including the simultaneous effect of natural convection.

POOL BOILING IN AN ACCELERATING SYSTEM

Introduction

This paper presents the results of an initial study of the influence of a force field on pool boiling heat transfer. A force field was utilized which produced accelerations normal to a heated surface in the range 1 to 21 times standard gravity. Although system acceleration different from that due to gravity is not commonly encountered in boiling processes, its role is important as it is the source of the natural convective movement in fluids. With the various new problems encountered in space technology it may be expected that boiling heat transfer will occur in force fields different from that found on earth.

The high rate of heat transfer by boiling has been ascribed to the intense agitation of the liquid at the heating surface by the bubbles (1, 36). Various researchers (2-9) have described the boiling process by means of correlating equations. In some cases the action of the bubbles was considered by the use of dimensional analysis, and in others by the selection and treatment of suitable models. Owing to the inability to describe the bubble action adequately, it was necessary to conduct experiments to determine certain constants which themselves frequently include other factors. Analytical results showing agreement with certain experimental data usually are given but as yet no widely applicable theory exists.

The bubble growth rate has been described well in terms of liquid superheat by the equations of Plesset-Zwick (10), Forster-Zuber (11) and Griffith (12). Fritz (13) obtained an expression for the maximum volume of a vapor bubble at departure from a surface, based on the work of Bashforth and Adams (14) and Wark (15). It considers the equilibrium of the surface of curvature separating two phases in force field, hence is valid only for conditions of equilibrium such as might be present in boiling at low heat flux rates to a saturated liquid. Although several correlations for boiling (4,5) utilized this expression, (Equation (1) below), it has been experimentally verified only for low heat flux rates in a force field of one standard gravity (1,16).

$$V_{b_{max}} = (0.0119\phi)^3 \left[\frac{2\sigma g_c}{g(\rho_\ell - \rho_v)} \right]^{3/2} \qquad (1)$$

A relation between the frequency of bubble formation and the bubble diameter at departure from a horizontal heating surface, of the form

$$f\ D_b = \text{constant} \tag{2}$$

was observed (1,16,17) with the boiling of a saturated liquid at low values of heat flux. For methanol, Perkins and Westwater (18) determined that f, D_b and hence f D_b remained constant up to approximately 80% of the peak heat flux, after which the values increased. However, a horizontal round tube was used for the heating surface and no information was given as to where the measurements were made. Recent experimental work (19) indicates the bubble diameter at departure decreases with an increase in heat flux.

A relation for the thickness of the superheated liquid boundary layer next to a heated surface was derived by Chang (20) which predicts the thickness increases with heat flux. By using the variation in index of refraction of light in water with temperature the thickness of a "boundary layer" near a horizontal heated surface with pool boiling to a saturated water was measured (21) at low values of heat flux, and was found to decrease with an increase in heat flux. On the other hand, with forced convection boiling to saturated water, Treshchov (22) measured fluid temperatures within 0.0025 inches of the heating surface and found that at a distance of 0.006 inches the temperature of the water was uninfluenced by large increases in heat flux, up to 1.3×10^6 Btu/hr-ft^2, which suggests the presence of a "boundary layer" in boiling systems.

Jakob (23) observed a linear relationship between heat flux and the number of nucleating sites at low values of heat flux. Recent investigations (19,21), however, indicate a relationship of the form

$$\frac{q}{A} \propto \left(\frac{N}{A}\right)^{\frac{1}{2}} \tag{3}$$

The role of surface defects or crevices as providing preferential sites for the formation of vapor bubbles has been studied experimentally and theoretically (24,25).

Acceleration of a boiling system directly influences the buoyant force on the vapor bubble. The bubble growth rate also will be influenced if the thickness and temperature distribution within the superheated boundary layer at the heating surface are affected by the acceleration. In general, bubble growth rate equations have neglected buoyant forces, but it may be anticipated that at certain acceleration this factor will have to be considered. The bubble size at departure probably will decrease with an increased buoyant force. While the mechanism which causes the departure is as yet little understood a plausible cause (12,26) attributes the motion to the inertia of the surrounding liquid. A recent photographic study of boiling in the absence of gravity (27), however, seems to discount the importance of liquid inertia. The vapor remained adjacent to the heating surface, and there was no evidence of bubbles being pushed away from the surface to any appreciable extent during their formation.

The effect of a vortex flow on the peak heat flux was studied by Gambill and Green (28). Peak heat flux up to 55 x 10^6 Btu/hr-ft^2 were obtained. The high values were attributed to the increased buoyancy on the bubbles due to centrifugal action, although it was not possible in this case to separate the forced convection contribution to the heat flux associated with the high velocities.

This paper summarizes the results of a research program recently completed in which the influence of system acceleration up to 21g on pool boiling heat transfer was investigated (29). Continuation of this work is in progress to include a study of accelerations up to 100g and higher, maximum heat flux, surface orientation and the use of cryogenic fluids.

Experimental Apparatus

Centrifugal motion is used to provide a force field producing accelerations of 1 to 21 g's. To most effectively isolate the influence of the increase in buoyant force on the bubbles with nucleate boiling, the orientation of the heater was such that the acceleration is applied normal to the heated surface. The test vessel was pivoted to maintain an approximately uniform depth of liquid and a normal acceleration to all rotational speeds.

Figure 1 is a sketch of the pivoted test vessel and Figure 2 shows the overall assembly.

The heater consists of a cylindrical leaded copper block, 3 inches in diameter and 1 inch long. One end of the cylinder is chromium plated and serves as the heat transfer surface. The edge is undercut with a bevel and an undersize stainless steel mating piece 0.022 inches thick attached by a shrink fit to provide a continuous surface and to keep heat losses by conduction at a minimum. At the other end of the cylinder 32 parallel slots 0.008 inches wide by 5/16 inches deep are machined to accommodate 6 feet of chromel-A heater ribbon 1/4 inch wide by 0.002 inches thick, insulated from the block with mica strips. A guard heater minimizes heat loss from the underside.

Power to the main heater is measured with calibrated meters. The largest source of error in the determination of heat flux arises from the heat loss from the stainless steel heater skirt. This is estimated to vary from 5% at q/A = 10,000 Btu/hr-ft^2 to 1% at q/A = 100,000 Btu/hr-ft^2.

To eliminate the possible influence of the gross convection currents on the bubbles forming near the edge of the heated section a flow guide was installed consisting of a cylinder of thin sheet stainless steel open at both ends and slightly larger in diameter than the heater. The effect is that of having a system whose entire bottom surface is heated and essentially iso-thermal.

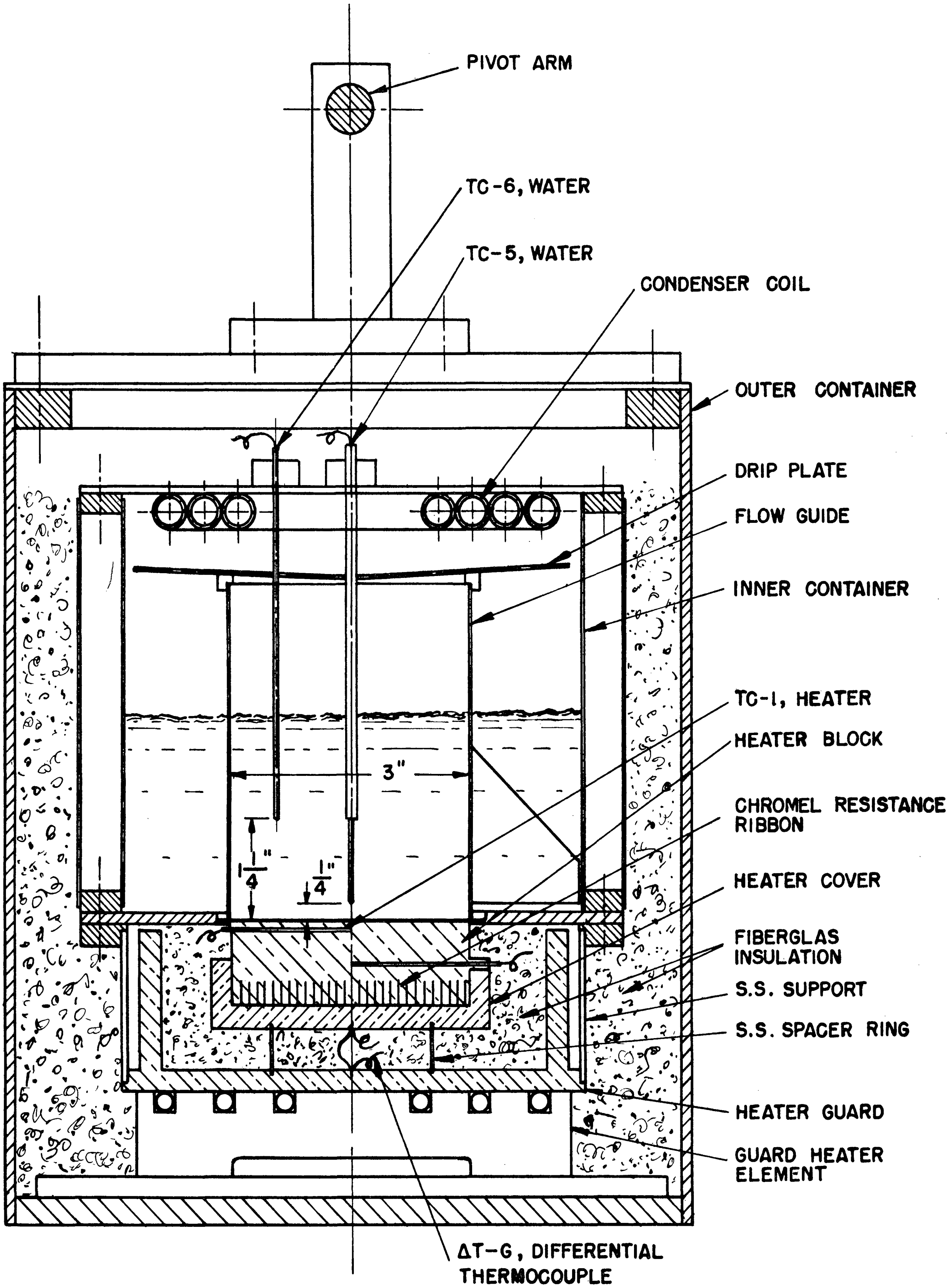

Fig. 1. Test Vessel.

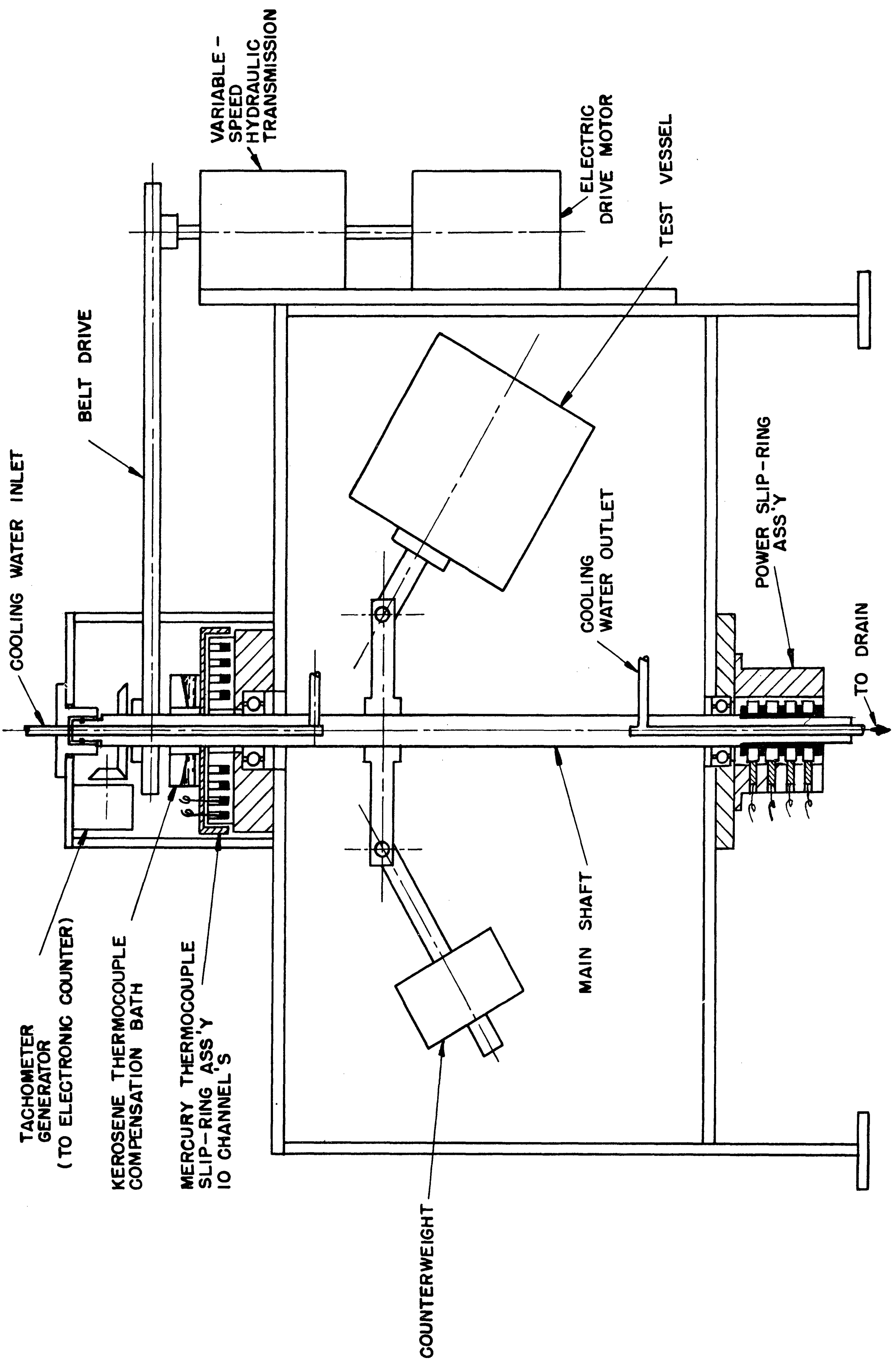

Fig. 2. Centrifuge Assembly.

Four 1/32 inch diameter holes are drilled radially in the heater block at four locations for the insertion of thermocouples. Two of the holes are drilled to the center at distances of 1/16 and 7/16 inches from the heating surface for purposes of extrapolating measured temperatures to obtain that of the heater surface.

The thermocouples in the heater consist of an insulated 30 gage constantan wire passing through a 1/32 inch O.D. copper tube with a capicator discharge welded junction at the tip. A 30-gage copper wire is welded to the other end of the tube to complete the circuit. This Cu-Cu junction produces negligible thermal emf. Two 30-gage copper-constantan thermocouples, designated TC-5 and TC-6 in Figure 1, were encased in 1/16 inch O.D. stainless steel tubing for water temperature measurements. All thermocouples were calibrated at the steam point and by comparison with Bureau of Standards calibrated mercury-in-glass thermometers. A Leeds & Northrup type K-3 potentiometer with a mirror-type galvanometer was used throughout.

The measurement of the thermocouple emf from the rotating member was accomplished through the use of liquid mercury commutators. Ten concentric mercury channels were machined in a plexiglas base which rested on a heavy aluminum block to insure isothermal conditions. To avoid amalgamation with the mercury a pure iron wire served as an intermediate metal between the rotating and stationary copper and constantan wires. Figure 3 shows the equivalent thermocouple circuit. Extensive tests were conducted to determine the extent of errors in thermal emf introduced by this circuit. Thermocouples were checked at the steam point using a hypsometer under non-rotating conditions, and the emf were identical with those obtained with a direct connection. The emf for the non-rotating and rotating conditions were measured for each of the mercury channel pairs forming a circuit for no power input to the heater. Changes in emf were observed in each pair which depended upon the rotational speed. Reversing the direction of rotation reversed the polarity of the change. Hence, it is believed this emf results from the influence of the earth's magnetic field on the rotating circuit, producing in effect the action of a unipole generator. This effect was reproducible within $\pm$ 2 μ_V and the correction at the maximum speed was 10 μ_V for TC-6 of Figure 1, 4 μ_V for TC-5 and zero μ_V for the thermocouples in the heater block. Considering other sources of error, it is estimated that the uncertainty in the level of temperature of any thermocouple on the rotating assembly is $\pm$ 0.2°F, but that changes in the temperature of the thermocouple are known with an uncertainty of $\pm$ 0.1°F.

The acceleration at the heating surface is calculated using the measured location of the center of gravity of the test vessel and the blue print dimensions. The increase in hydrostatic pressure at the heating surface owing to rotation was computed from the depth of the water and system acceleration. The uncertainty in T_{sat} at the heating surface, employed in the boiling parameter $T_W - T_{sat}$, is estimated to be $\pm$ 0.1°F. This estimated uncertainty results from the fact that the depth of the boiling fluid was not measured while under rotation. Visual observation of the boiling process at the maximum heat flux of 100,000 Btu/hr-ft^2 without rotation indicated that the change in liquid depth due to the bubbles was almost negligible. As will be noted from Figure 8 the liquid

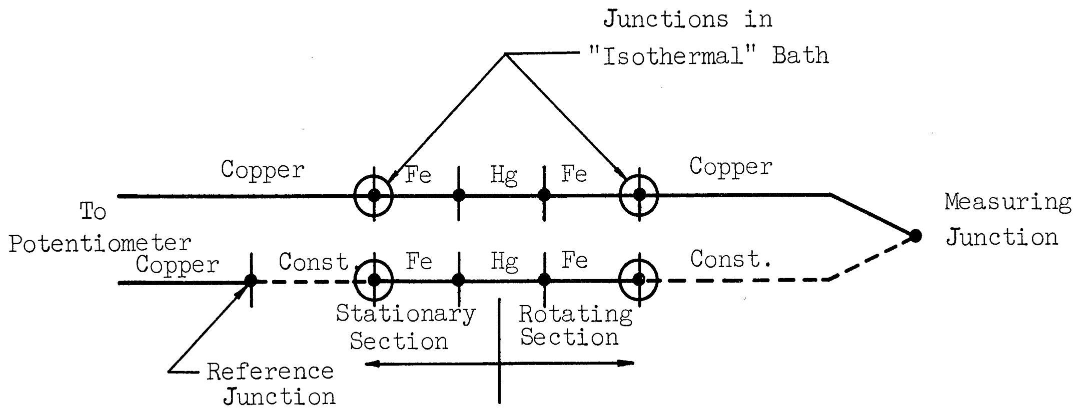

Fig. 3. Equivalent Thermocouples Circuits.

subcooling increases as the system is subjected to acceleration. Hence, under these circumstances, the vapor volume can be expected to be less than under standard gravitational acceleration (no rotation) and the change in depth correspondingly smaller. The test vessel was vented to the atmosphere. The depth of the water was $2\frac{1}{2}$ inches in all tests, and its initial resistivity was at least 1.5×10^6 Ω-cm, as measured with a conductivity cell. For the data reported here, the resistivity at the conclusion of a test varied from 0.8×10^6 to 1.1×10^6 Ω-cm.

A cooling coil mounted on the underside of the test vessel cover serves to condense the vapor formed. At the lower heat flux rates it is possible to control the temperature of the test water by varying the cooling water flow rate and thus obtain limited data for boiling of a subcooled liquid.

The rotational speed was obtained from a tachometer-generator generating 60 pulses per revolution. Its output was fed into a Model 522B Hewlett-Packard Electronic Counter.

Detailed description of the experimental apparatus is given in Reference (29).

Results

Non-Boiling Data

Several tests were conducted under non-boiling conditions, Figure 4 indicates how the temperature difference T_W-T_5 varies as a function of acceleration. For a given heat flux, the presence of the flow guide decreases the temperature difference.

The data are plotted in Figure 5 using the conventional free convection modulii, where the diameter of the heating surface is taken as the characteristic length and g in the Grashof Number is replaced with acceleration. The influence of the flow guide is noted. Included for reference is the correlation recommended by McAdams (30) for a horizontal heated plate facing upward. A correlation which best fits the data with the flow guide is

$$N_u = 0.0505\ (G_R P_R)^{0.396} \qquad (4)$$

For one particular test (C-5) the data fell above the best fit of the data. In this case the water was highly subcooled and it is believed this lack of correlation occurs because of increased heat loss through the heater skirt.

Boiling Data

After filling the test vessel with distilled water, power was turned on and the water boiled vigorously to degas both it and the heating

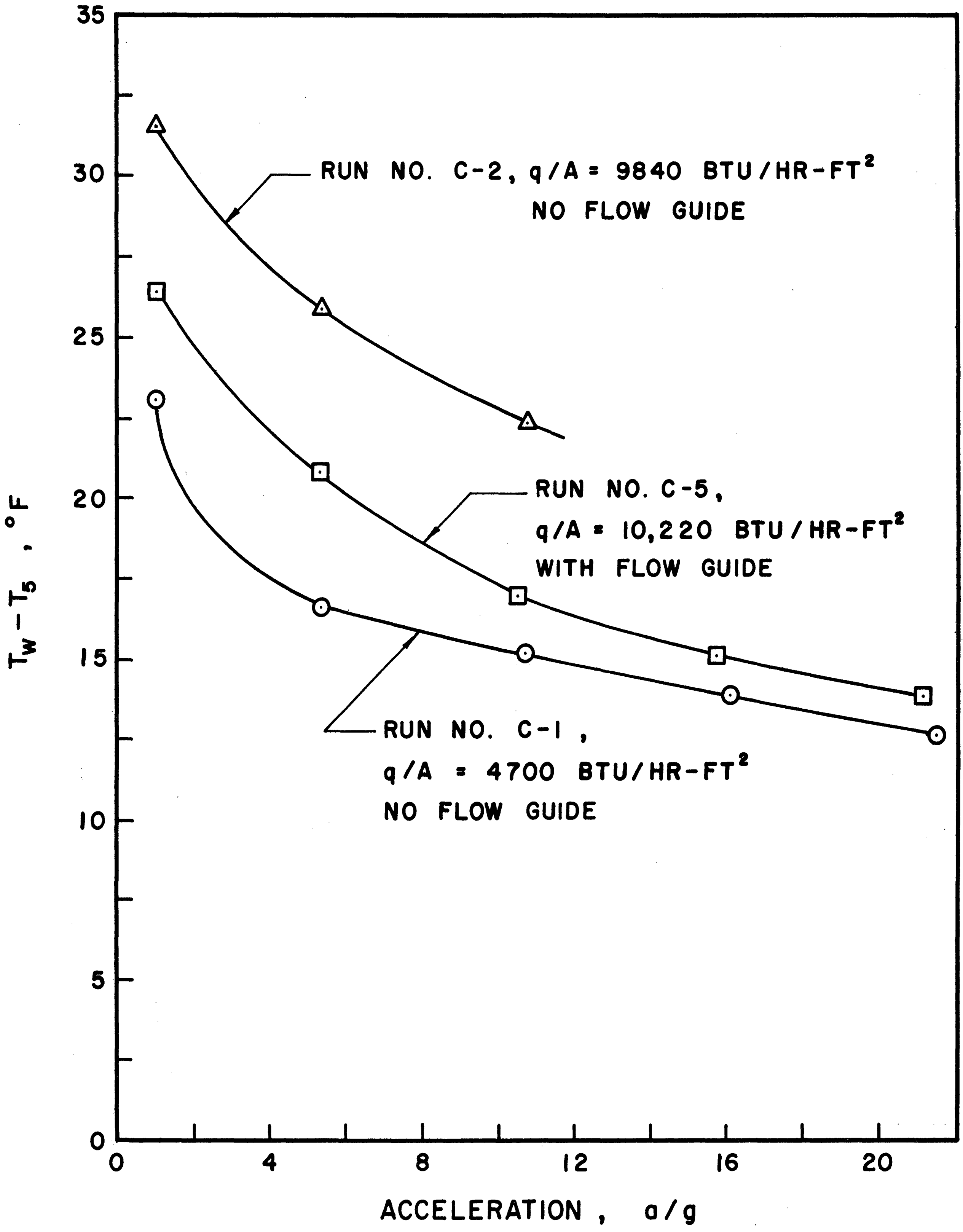

Figure 4. Plot of ΔT_c versus Acceleration for Natural Convection Indicating the Effect of the Flow Guide.

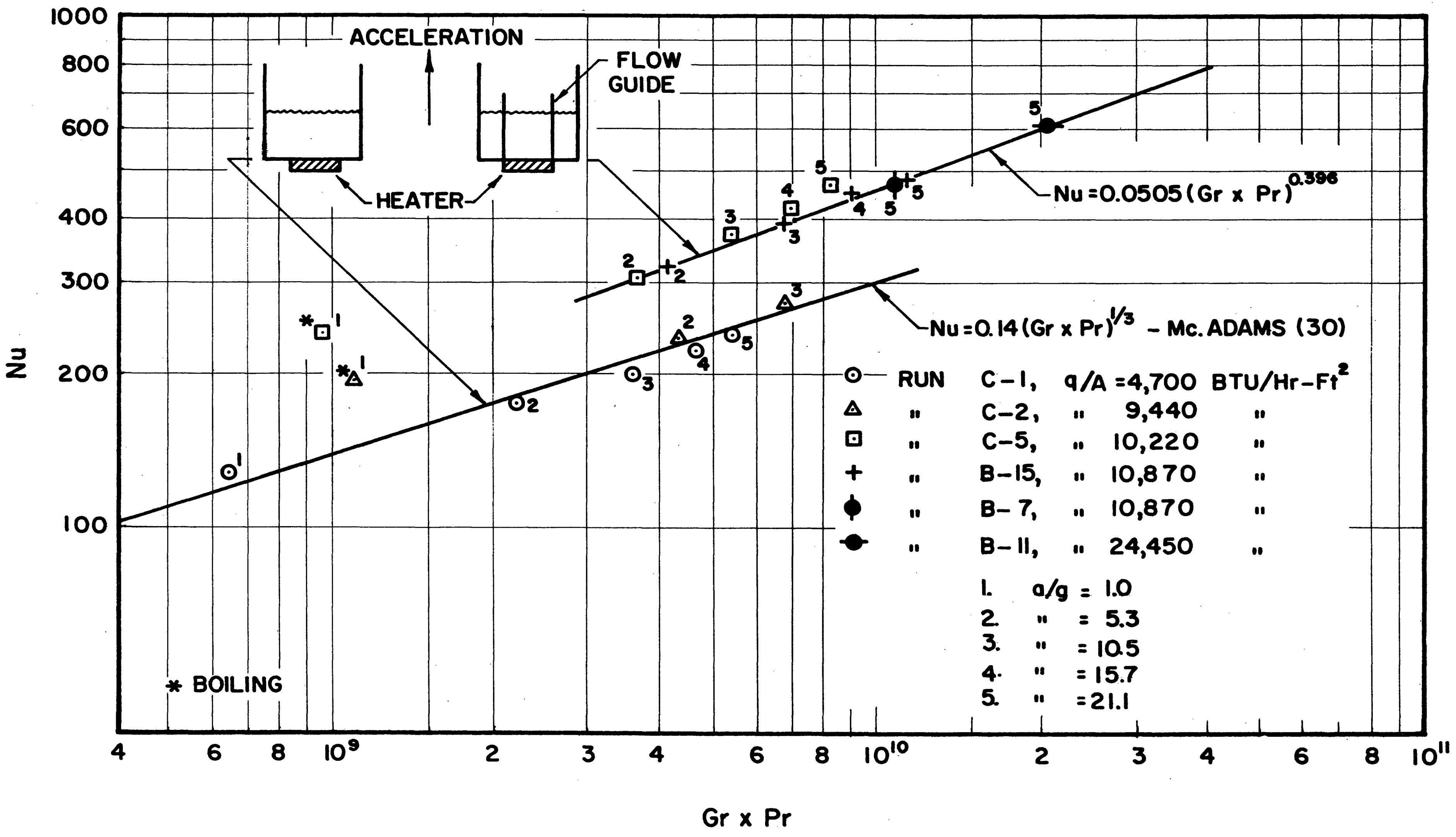

Figure 5. Correlation of Natural Convection Data with Acceleration Normal to Heating Surface.

surface. Prior to each run, this was done for a minimum of 4 hours, and in the majority of cases degassing over a period of 16 hours was used. For any particular test the heat flux was maintained constant while acceleration was varied. When conditions such as acceleration and cooling water flow rates were changed sufficient time was allowed for the attainment of steady state conditions before data were taken.

The acceleration rates were varied from 1 to 21.15 times normal gravitational acceleration in 5 to 8 steps. In earlier tests it was noted that the wall temperatures at $a/g = 1$ shifted gradually during a long period of time. To isolate the effect of the acceleration a set of measurements were taken at $a/g = 1$ before and after each higher acceleration. Thus any shift in the surface temperature during boiling at $a = g$ could be compensated by considering the change only. The maximum shift which occurred was 0.8°F over a test period of 12 hours.

To determine the reproducibility of the data, several accelerations were duplicated during the course of a specific test. No significant change was detected.

The heater surface was highly polished prior to its chromium plating. No roughness measurements were made, but the heater surface was given a similar treatment prior to each run and it is believed that the roughness was the same for all tests.

Except for one case the data presented are representative of at least two reproducible test runs. Because of the sensitivity of the boiling process at lower values of heat flux to even small degrees of subcooling, it was necessary to vary the water temperature in order to determine when the liquid was saturated, or as nearly saturated as possible. The water temperature was controlled by varying the cooling water flow rate. Figures 6 and 7 indicate the influence of a limited degree of subcooling at several values of heat flux and acceleration. The asterisks on the data points indicate where the coolant flow was so low that a net quantity of vapor escaped from the atmospheric vent, with an attendent possible increase in pressure.

In Figure 6 for $q/A = 10{,}870$ Btu/hr-ft^2 and $a/g = 5.29$, the heater surface temperature exhibits a pronounced maximum as the subcooling is decreased, and becomes constant as the saturation temperature is reached. At $a/g = 10.47$, no maximum in $T_w - T_{sat}$ is observed, but a long plateau is noted. In Figure 7 for $q/A = 24{,}450$ Btu/hr-ft^2, subcooling was not increased sufficiently to prevent boiling and a maximum could not be observed. The decrease of the heater surface temperature with decreasing subcooling is still present, as is the leveling off of $T_w - T_{sat}$ except for the highest acceleration. For values of heat flux up to 50,000 Btu/hr-ft^2 it was thus possible to determine when $T_w - T_{sat}$ became independent of the degree of subcooling. This was used as a criterion for defining the existence of "saturated" pool boiling.

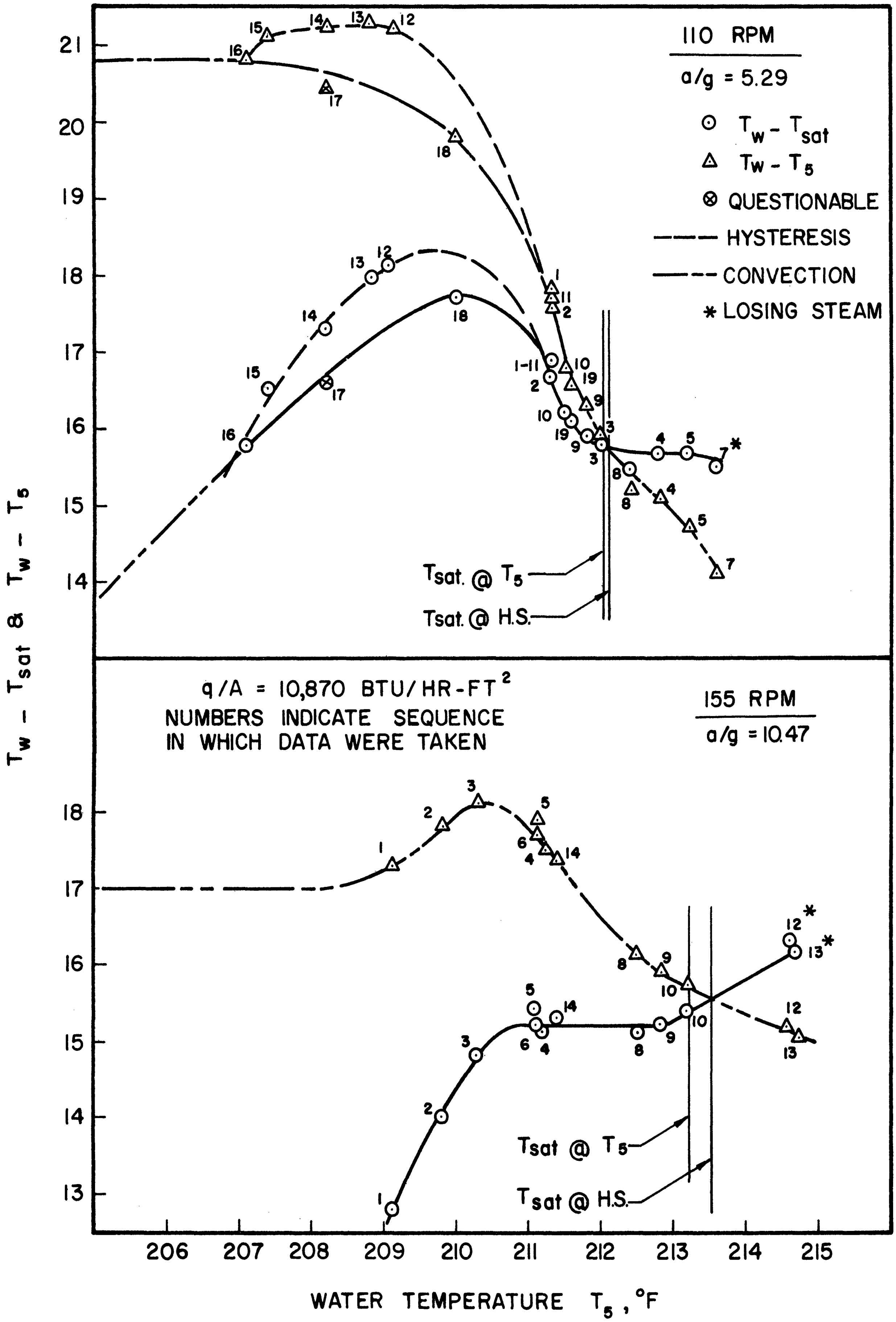

Figure 6. Run No. B-15. Plot of T_W-T_{sat} and T_W-T_5 vs. Water Temperature T_5 at 110 and 155 RPM.

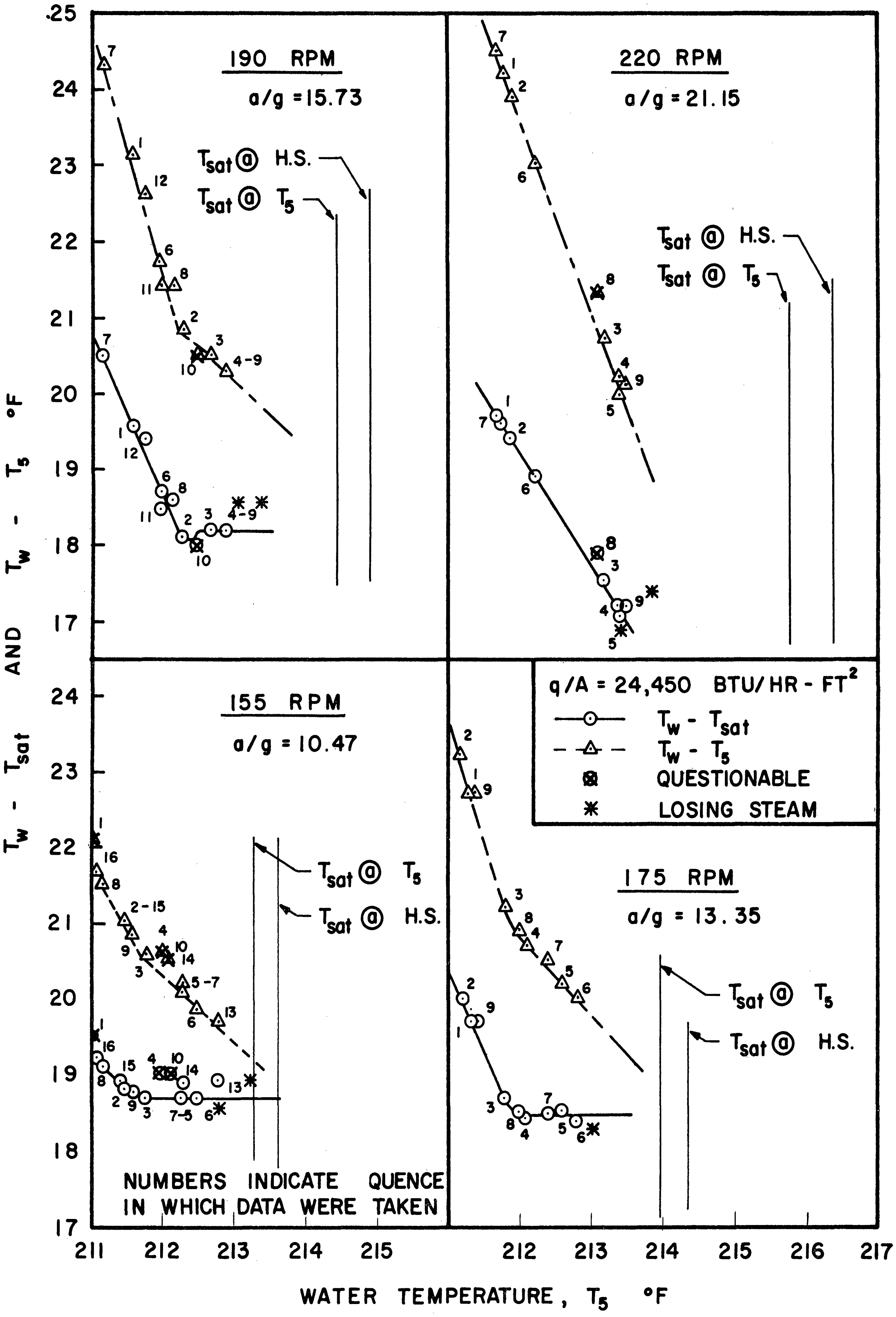

Figure 7. Run No. B-9. T_W-T_{sat} and T_W-T_5 vs. Water Temperature T_5 at Various Accelerations.

Sufficient data are not available at this time to make generalized conclusions as to the role of subcooling or liquid depth with the system under acceleration. It is believed possible that the decrease in T_w* with decreasing subcooling is an indication of increased fluid agitation or turbulence within the superheated region near the heated surface. With a reduction in heater surface temperature it seems unlikely that additional nucleating sites are being formed. If so, then the existing ones must be more effective in providing increased agitation either from more rapid bubble growth rates, increased bubble sizes, increased frequency of formation or a combination of these.

At nominal heat flux values of 75,000 and 100,000 Btu/hr-ft^2 it was not possible to vary the water temperature with the present system. The water temperature near the heated surface remained subcooled with the system under acceleration. The subcooling occurs because of the large saturation temperature changes in the water resulting from the larger acceleration. Similar phenomena of lesser magnitude have been observed with water (1) and mercury (31) in a gravity field. Figure 8 shows the profile of the water and saturation temperatures between the heated surface and the vapor interface. The water depth at filling in all cases was $2\frac{1}{2}$ inches, and the variation in T_{sat} results solely from the change in hydrostatic pressure with depth. The change in depth under boiling at the various accelerations was not considered great, as was discussed earlier. The system accelerations specified are at the heating surface. While this is located approximately 15.4 inches from the centerline of rotation the hydrostatic pressure within the liquid above this surface was computed taking into consideration the small local variation in acceleration.

The relation between the measured water and saturation temperature in Figure 8 are typical for heat flux between 50,000 and 100,000 Btu/hr-ft^2. A minimum subcooling was obtained at heat flux above 50,000 Btu/hr-ft^2 which was a function of acceleration and liquid depth. It is probable that this minimum subcooling (or maximum water temperature) results from the liquid temperature at the vapor interface reaching a maximum, i.e., its saturation value. Data reported at heat flux of 75,000 and 100,000 Btu/hr-ft^2 are thus considered to correspond to the same conditions as those at the lower heat flux where subcooling did not influence $T_w - T_{sat}$.

Figure 9 is a composite plot of $T_w - T_{sat}$ versus acceleration for the various values of heat flux with pool boiling of essentially saturated water. Some deviation of the data with a gravity field did exist, and several of the curves were displaced to bring $T_w - T_{sat}$ at $a/g = 1$ in agreement with the best fit of data from a number of tests conducted at $a/g = 1$, shown in Figure 10.

The difference between the heater surface temperature T_w and the water saturation temperature T_{sat} at the heating surface decreases with an increase in acceleration at lower values of heat flux. This is attributed to the increasing contribution of natural convection (non-boiling) with

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*Occasioned by the decrease in $T_w - T_{sat}$, as T_{sat} at the wall is constant for a given acceleration.

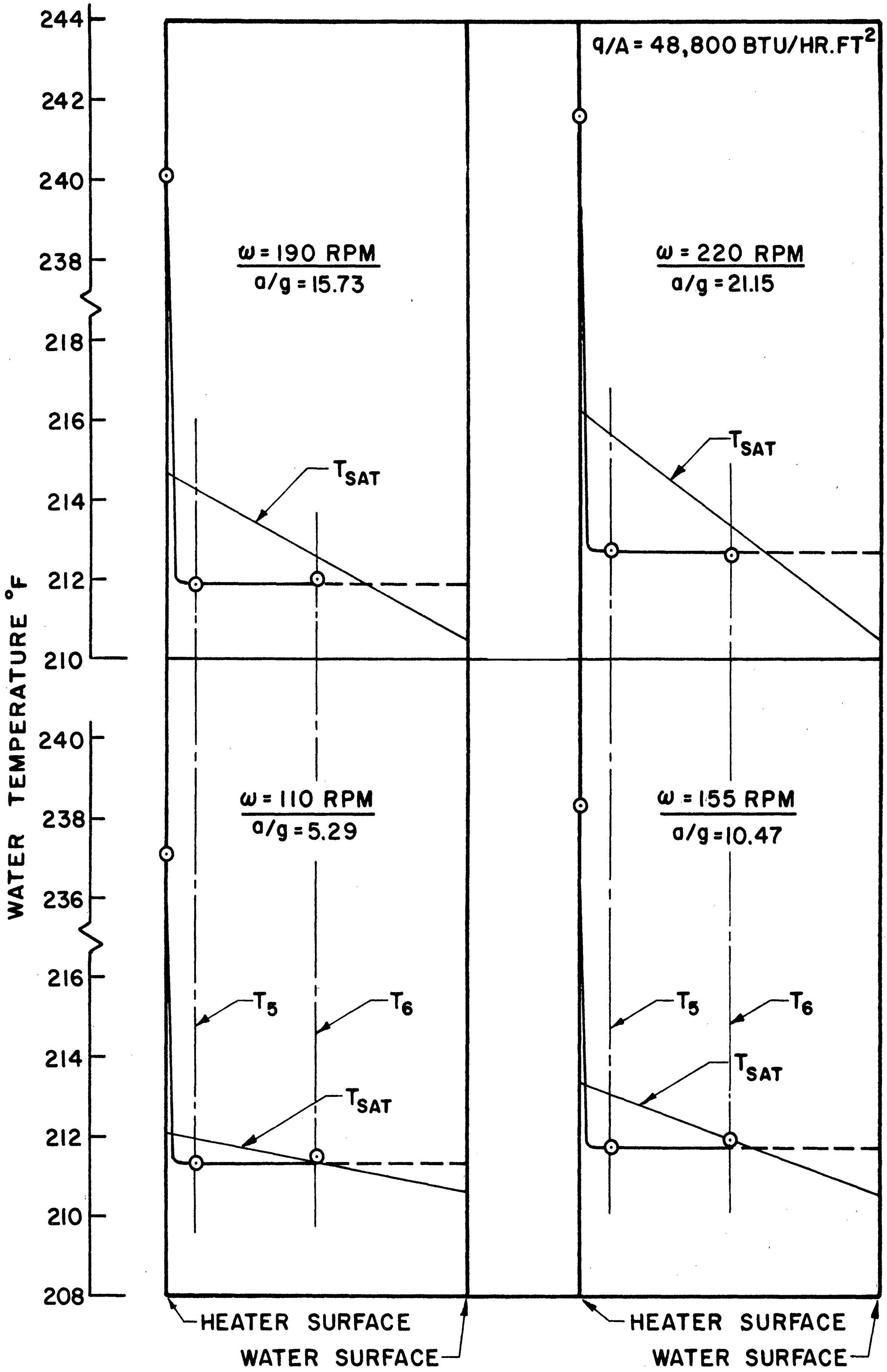

Figure 8. Run No. B-14. Temperature Profile between Heater and Water Surface for Various Accelerations.

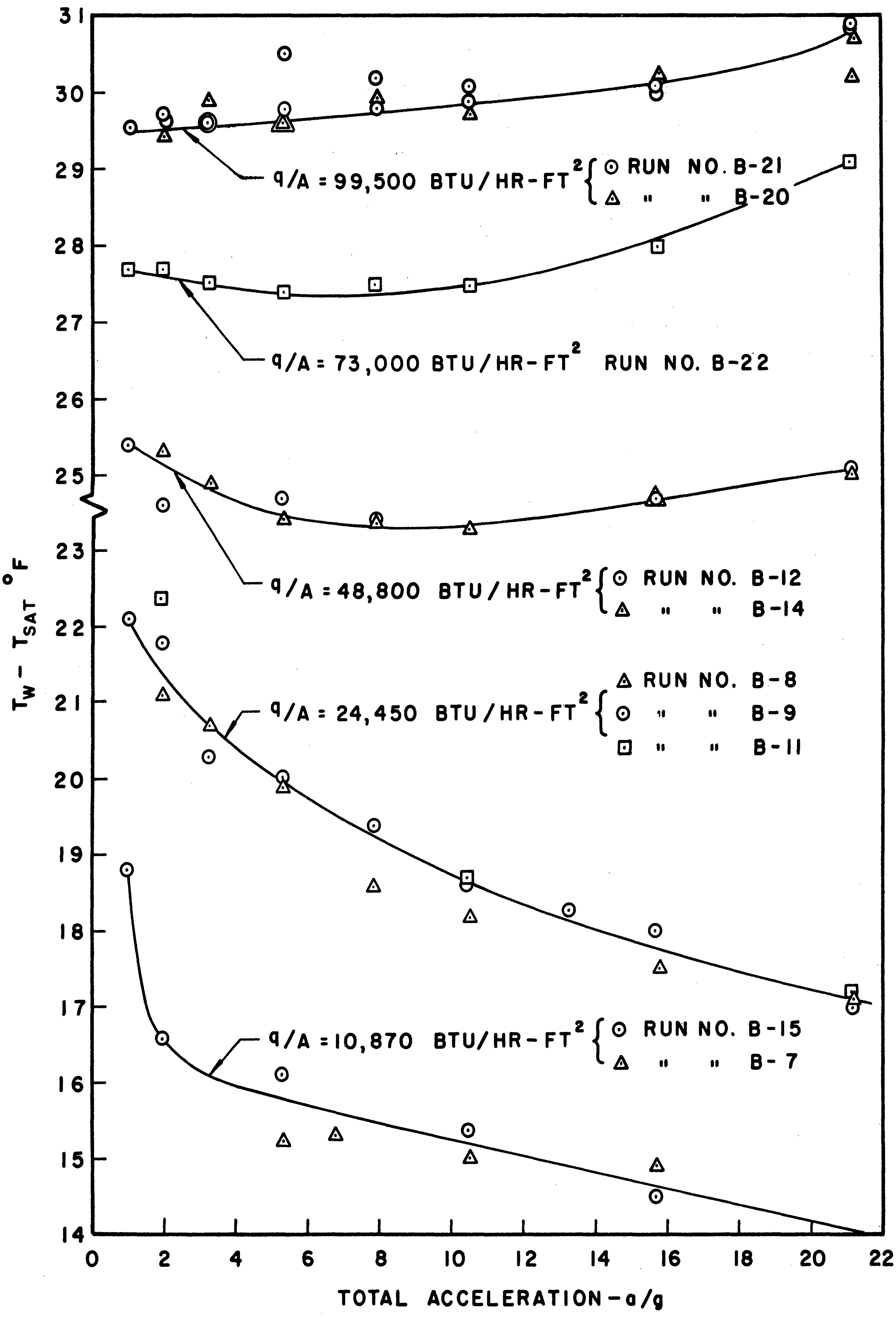

Figure 9. Influence of Acceleration on T_W-T_{sat} with Pool Boiling to Saturated Water.

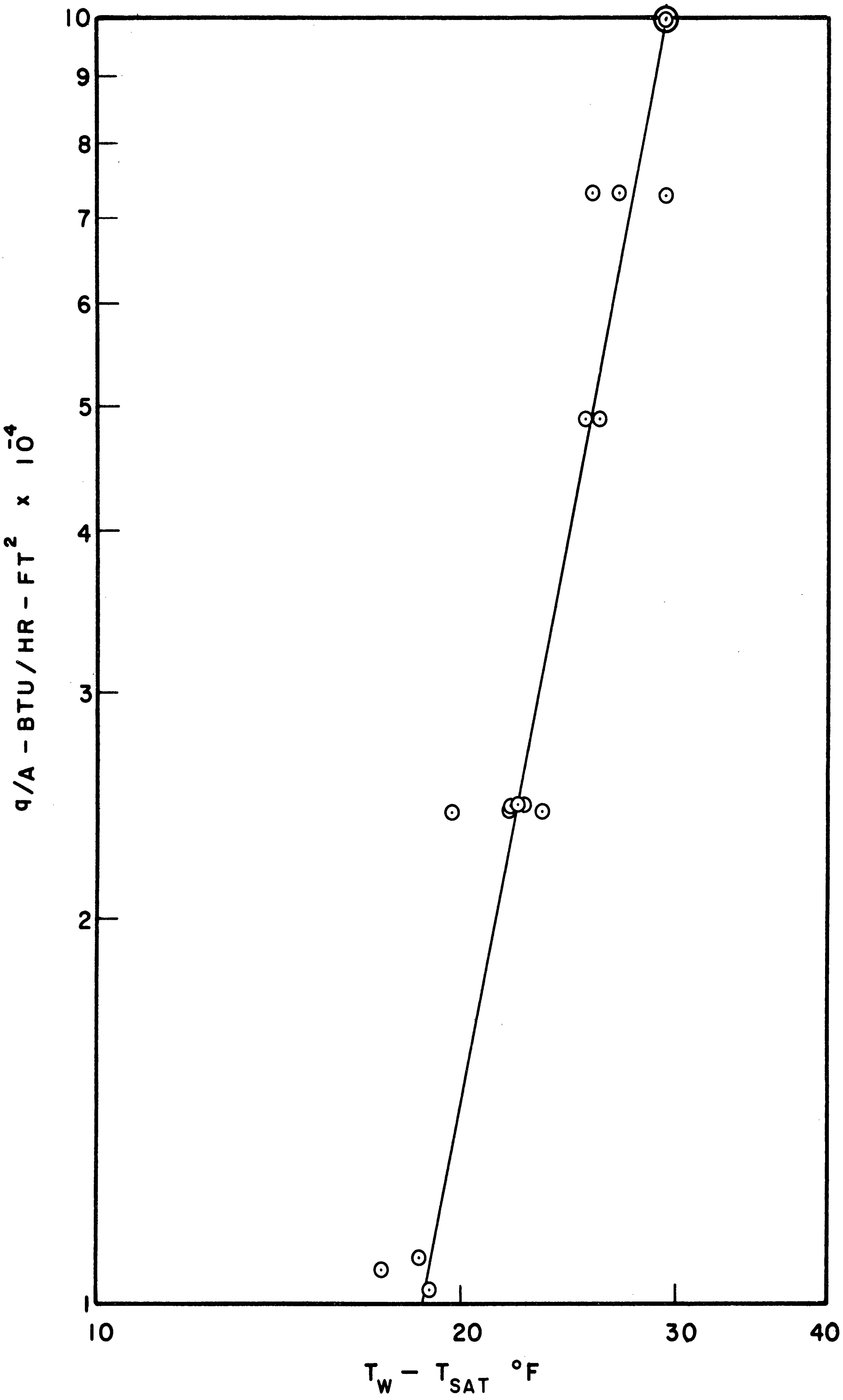

Figure 10. Plot of q/A vs. T_w-T_{sat} for Boiling in Standard Gravitational Field.

acceleration. As the heat flux level increases, the decrease in $T_W - T_{sat}$ becomes smaller, and at the higher values of heat flux $T_W - T_{sat}$ increases with an increase in acceleration. The contribution of natural convection is relatively smaller at the higher heat flux. It is believed that increased system acceleration results in smaller bubble sizes at departure, with an attendent decrease in agitation. To provide for a fixed total heat flux more nucleating sites then are required, which in turn requires the increased wall temperatures, as observed.

Discussion of Results

Because the quantity of data obtained is not large and is for one Prandtl number over a relatively small range of conditions it is believed that to attempt a direct correlation of the data either with existing relationships or with ones which might be worked out would be premature, although a start is made below. Further, none of the correlations in the literature which have been examined (e.g., 4,5,6,7,9,20) appear capable of predicting the trends shown in Figure 9 resulting from acceleration of the boiling system. This may be due in part to the inadequacy of the models used to describe heat transfer by boiling or to the neglecting of the non-boiling contribution.

Under acceleration, the non-boiling free convection contribution is significant. This is more readily seen in Figure 11, which is a replotting of the data from Figure 9. It places the data in the more familiar form of q/A versus $T_W - T_{sat}$. In the lower portion the curves corresponding to natural convection for the same accelerations are shown using the correlation given in Equation (4). The data do not permit determination of the intersection between the non-boiling and boiling curves except for the higher values of heat flux. It is of interest to note however, that if the boiling curves of constant acceleration are extrapolated, the intersection with the non-boiling curves occurs for $T_W - T_{sat}$ between 14 and 15°F, indicating the possibility that the point of incipient boiling of a saturated liquid is not greatly influenced by a change in the force field. The data also suggest that one result of system acceleration is to widen or delay the transition from non-boiling to fully established boiling, the retention of non-boiling influences being more persistent at the higher accelerations.

The influence of acceleration is greatest in the heat flux range up 50,000 Btu/hr-ft^2, where it appears that a transition in the convective mechanism is taking place. Within this range acceleration decreases the temperature difference $T_W - T_{sat}$ required for a given flux. Above this heat flux range a reversal of the effect of acceleration is observed on the temperature difference, although the total influence is small. The upper range of the curve is extended to $(q/A) = 200{,}000$ Btu/hr-ft^2 to include some recent data of Costello (32) for water on a similar configuration for $a/g = 1$ and $a/g = 20$. The general agreement between the two sets of data is quite satisfactory.

With fully established pool boiling in a standard gravitational field, the contribution of non-boiling convection to the total heat transfer is so small that generally it is not separated from the total heat

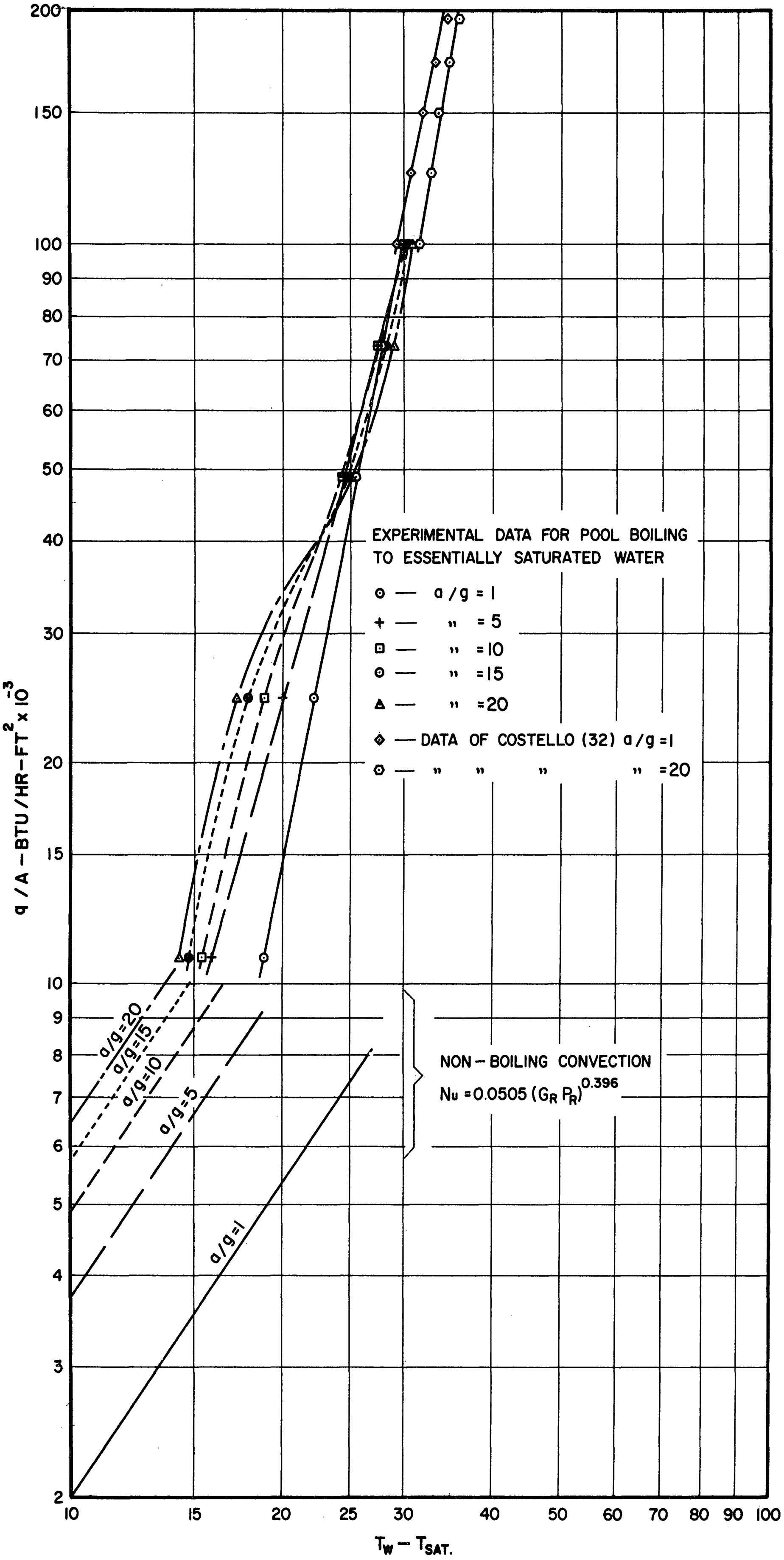

Fig. 11. Influence of System Acceleration Normal to Heating Surface on Convection and Pool Boiling.

flux in analysis. With the application of an acceleration, however, the non-boiling convection can become quite appreciable, as was indicated by these results. In order to describe the total heat transfer, it is necessary to determine the relative contribution of the bubbles and that not resulting from the bubbles. This was accomplished in effect by Chang (20) by considering the rate of momentum exchange in the liquid near the heating surface to consist of the sum of the molecular diffusivity and the eddy diffusivity. Another method, which is introduced here, is to consider the non-boiling and boiling contributions separable and to weight them in some manner.

The total heat transfer rate q_t is

$$q_t = q_c + q_B \tag{5}$$

where q_c is that portion due to non-boiling and q_B is that portion due to boiling and includes the influence of agitation induced by the bubble action. Dividing Equation (5) by the total area, A_t, gives,

$$\frac{q_t}{A_t} = \frac{q_c}{A_t} + \frac{q_B}{A_t} \tag{6}$$

If A_B represents a time-averaged area projected on the heater surface in which the influence (agitation) of the bubbles is concentrated and A_c (A_c = A_t-A_B) is that area in which natural convection is effective; then

$$\left(\frac{q}{A}\right)_t = \frac{q_c}{A_c}\frac{A_c}{A_t} + \frac{q_B}{A_B}\frac{A_B}{A_t}$$

$$= \left(\frac{q}{A}\right)_c (1 - \gamma) + (q/A)_B\,\gamma \tag{7}$$

A_B is not the heater surface area covered by the bubbles, but is considered differently. It might be designated as an "area of influence" of the bubbles. Figure 12 is an illustration of this concept.

The fractional effective boiling area γ is found from Equation (7) as

$$\gamma = \frac{(q/A)_t - (q/A)_c}{(q/A)_B - (q/A)_c} \tag{8}$$

To evaluate γ as a function of a/g and $(q/A)_t$ it is necessary to make certain assumptions regarding $(q/A)_c$ and $(q/A)_B$. For $(q/A)_c$ it is assumed that the correlation given in Equation (4) would be appropriate for its

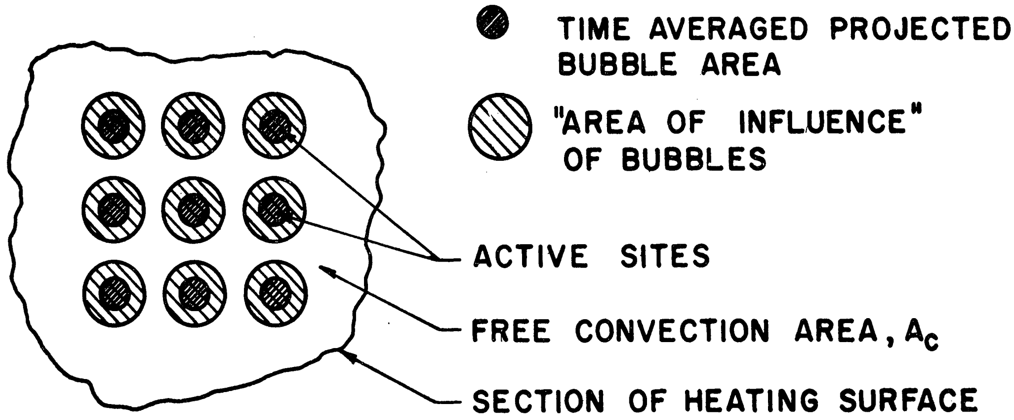

Figure 12. Illustration of Area of Influence of Bubble with Boiling Heat Transfer.

calculation using the experimentally observed value of $T_W - T_5$. Equation (4) is used as it is felt that the convective pattern between bubbles would more closely be represented by this correlation.

The quantity $(q/A)_B$ is that heat flux which may be directly associated with the micro convection of the bubbles. For fixed acceleration it is assumed that this quantity is a constant. The total heat flux $(q/A)_t$ will vary with $T_W - T_{sat}$, of course, but this results because the number of active sites, hence also γ, varies with $T_W - T_{sat}$. It will be further assumed that not only will $(q/A)_B$ be constant for a given acceleration but that it will be equal to the peak or burnout heat flux at that acceleration. This requires some explanation which follows.

Nishikaws and Urakawa (34) in a study of nucleate boiling in saturated liquids made the following independent observations from their experimental data

$$h \propto \left(\frac{N}{A} D_b^3 f\right)^{1/3} \tag{9}$$

$$\Delta\theta \propto \left(\frac{N}{A}\right)^{-(1/6)} \left(\frac{q}{A}\right)_t^{2/3} \tag{10}$$

$$\left(\frac{q}{A}\right)_t \propto \left(\frac{N}{A}\right)^{1/2} \tag{11}$$

From these relations it may then be deduced that the quantity $D_b{}^3 f$, which is proportional to the rate of volume flow of vapor away from the heated surface, is a constant and is independent of $(q/A)_t$, N/A and $\Delta\theta$. If true this is a significant result as it has been observed that $D_b f$ is constant (1), Equation (2). As mentioned earlier, recent data on methanol (18) indicated that f, D_b and thus both $f\,D_b$ and $D_b{}^3 f$ remained constant up to approximately 80% of the peak heat flux, but other observations (19) suggest that D_b decreases with increase in heat flux.

If the hydromechanics or micro convection of the bubbles is thought to be proportional in its influence ("agitation") to the volume flow of liquid into the surface associated with the rate of volume flow of vapor away from the surface or any exponential power of this flow and that this directly affects the heat flux $(q/A)_B$, then it follows that

$$\left(\frac{q}{A}\right)_B \sim (D_b^3\ f)^n = \text{constant} \tag{12}$$

Because of this result it was assumed above that $(q/A)_B$ is a constant for a given acceleration. This point remains unresolved but the assumption is necessary to this discussion.

The next problem is to find an expression for $(q/A)_B$. It seems reasonable that it should be related in some way to the maximum heat flux. In the light of recent studies by Debortoli and Green (33) it would appear that $(q/A)_B$ might be more closely related to that heat flux at which a significant departure is made from the processes of nucleate boiling, indicating a change in the mechanism of the bubbles. However, this condition is not well understood at present and for the purposes of this paper $(q/A)_B$ will be taken equal to $(q/A)_p$.

Relationships for $(q/A)_p$ have been developed which show an influence of acceleration (3,4,35) although none have been confirmed by experiment. However, for application here the following expression of Zuber (35) will be used

$$\left(\frac{q}{A}\right)_p = K_2 \lambda \rho_v^{\frac{1}{2}} \left[\sigma g (\rho_e - \rho_v)\right]^{\frac{1}{4}} \tag{13}$$

The constant K_2 was determined for these calculations from values of $(q/A)_p$ in saturated water at 1g and found to be 0.146.

Using the above data the corresponding values of γ were computed from Equation (8) and are shown in Figure 13 as a function of dimensionless acceleration a/g and total heat flux. Whether this result is generally applicable for the fractional area of effective boiling has yet to be established although the trend appears reasonable. That is, according to Equation (1) D_b is reduced as acceleration is increased which would also reduce the area influenced by the bubble. As seen from the data an increase in acceleration causes a decrease in heated surface temperature (at the lower heat flux) which would reduce the number of nucleating sites for these conditions. At the higher heat flux the decrease in γ results principally from the larger relative increase in $(q/A)_p$ with increased acceleration. It is possible the parameter on the curves should be $(q/A)_t/(q/A)_p$ rather than $(q/A)_t$. This could easily be done but a somewhat different set of curves would result.

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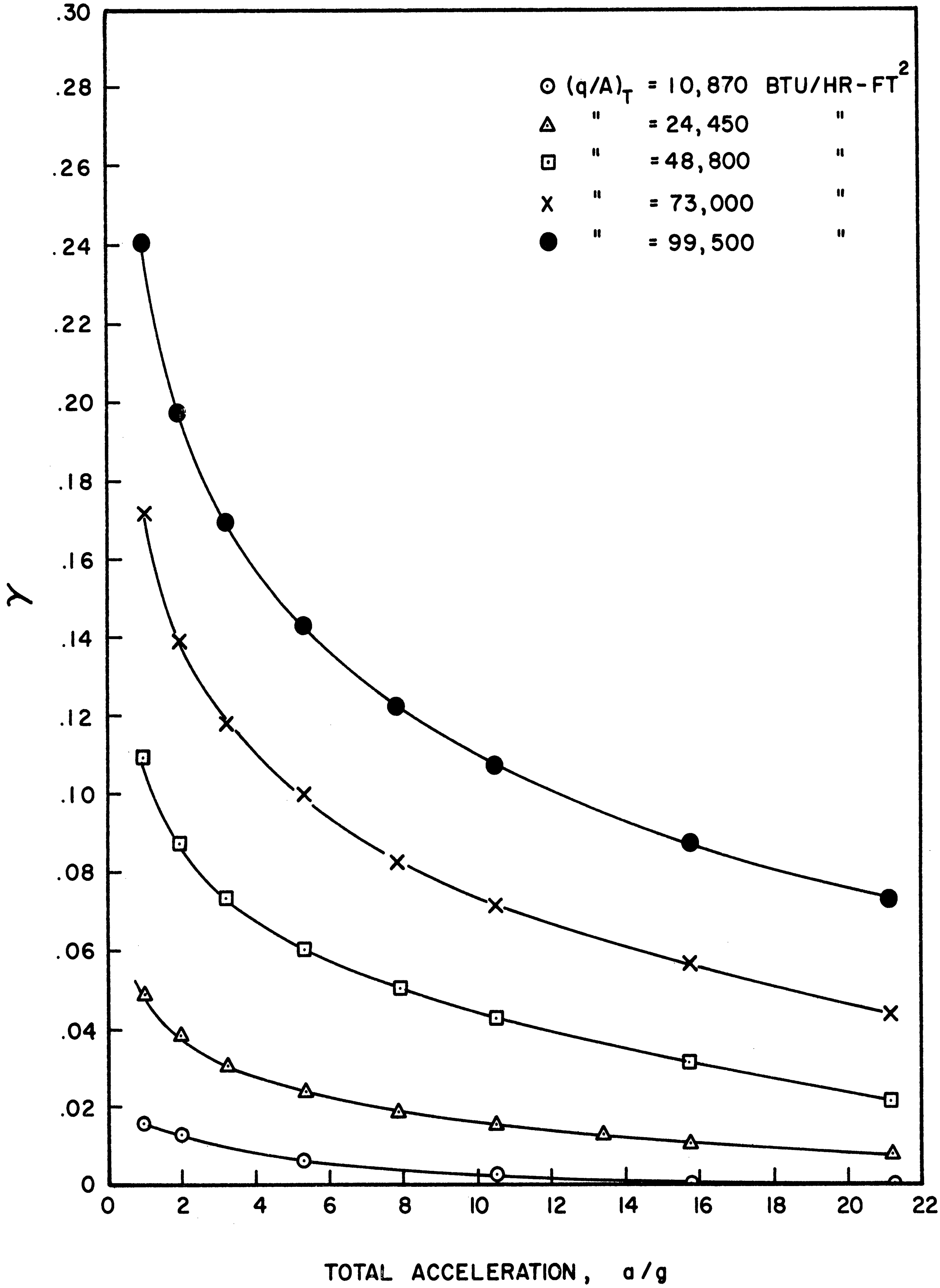

Figure 13. Calculated Values of γ as a Function of Total Heat Flux and Acceleration.

NOMENCLATURE

A	Area
a	Acceleration
a/g	Dimensionless acceleration
f	Frequency of bubble formation
g	Local gravitational acceleration
g_c	Mass-force conversion constant = $32.174\ lb_m\text{-}ft/lb_f\text{-}sec^2$
N/A	Number of active nucleating sites per unit area
n	Unknown constant
q	Heat transfer rate
q/A	Heat flux rate
T	Temperature
$T_{5,6}$	Temperatures at thermocouple TC-5 and TC-6, see Figure 1.
$T_w - T_{sat}$	Difference between heating surface temperature and fluid saturation temperature at heating surface.
V	Volume
N_u	Nusselt number
G_r	Grashof number
P_r	Prandtl number
γ	$= A_B/A_t$ (Equation (7))
σ	Surface tension
$\Delta\theta$	$T_W - T_{sat}$
ϕ	Contact angle
ρ	Density

Subscripts

b	Bubble
B	Boiling
c	Non-boiling convection
ℓ	Liquid
P	Peak or maximum
sat	Saturation
t	Total
v	Vapor
w	Wall

REFERENCES

1. Jacob, M. Heat Transfer. I. New York: John Wiley & Sons, Inc., 1949.
2. Kutateladze, S.S., "Hydrodynamic Model of the Crisis of Heat Transfer in a Boiling Liquid with Free Convection," J. Tech. Phys. Vol. 20, 1950.
3. Borishanskii, V.M., "An Equation Generalizing Experimental Data on the Cessation of Bubble Boiling in a Large Volume of Liquid." Zhurn. Tekh. Fiz., 26, (1956), 452. Translated in Soviet Physics-Technical Physics, 1. No. 2, 438, Am. Inst. of Phys., N.Y.
4. Chang, Y. and Snyder, N.W. "Heat Transfer in Saturated Boiling." Preprint 104 presented at Third National Heat Transfer Conference, ASME-AIChE, August 1959.
5. Rohsenow, W.M. "A Method of Correlating Heat Transfer Data for Surface Boiling of Liquids." Trans. ASME, 74, (1952), 969.
6. Levy, S. "Generalized Correlation of Boiling Heat Transfer." Trans. ASME, 81 Series C, No. 1, (February 1959), 37.
7. Forster, K. and Greif, R. "Heat Transfer to a Boiling Liquid-Mechanism and Correlations." Ibid, 43.
8. Hirano, F. and Nishikawa, K. "The Phenomena of Boiling Part 4," Machinery Research, Japan, Vol. 8, No. 6, 1959, pp. 625.
9. Forster, H.K. and Zuber, N. "Dynamics of Vapor Bubbles and Boiling Heat Transfer." AIChE Journal, 1, No. 4, (December, 1955), 531.
10. Plesset, M.S. and Zwick, S.A. "The Growth of Vapor Bubbles and Superheated Liquids." J. App. Phys., 25, No. 4, (1954), 493.
11. Forster, H.K., and Zuber, N. "Growth of a Vapor Bubble in a Superheated Liquid." J. App. Phys., 25, No. 4, (1954). 474.
12. Griffity, P. "Bubble Growth Rates in Boiling." Trans. ASME, 80, No. 3 (April, 1958), 721.
13. Fritz, W. "Calculation of the Maximum Volumes of Vapor Bubbles." Physik, Zeitschr., 36, (1935), 379.
14. Bashforth, F., and Adams, J. Capillary Action, Cambridge, 1883.
15. Wark, J.W. "The Physical Chemistry of Flotation." Journal of Physical Chem., 37, (1933), 623.
16. Hirano, F. and Nishikawa, K. "The Phenomena of Boiling--Part 2." Machinery Research, Japan, 8, No. 4, (1956), 431.

17. Hirano F. and Nishikawa, K. "Boiling of Hard Water." Trans. Soc. Mech. Eng., Japan 19, No. 88, (1953), 33.

18. Perkins, A.S. and Westwater, J.W. "Measurements of Bubbles Formed in Boiling Methanol" AIChE Journal, 2, No. 4, December (1956), 471.

19. Gaertner, R.F. and Westwater, J.W. "Populations of Active Sites in Nucleate Boiling Heat Transfer." Preprint 105 presented at Third National Heat Transfer Conference, ASME-AIChE, August, 1959.

20. Chang, Y.P. "Theoretical Analysis of Heat Transfer in Natural Convection and in Boiling." Trans. ASME, 79, (October, 1957), 1501.

21. Hirano, F. and Nishikawa, K. "The Phenomena of Boiling -- Part 3." Machinery Research, Japan, 8, No. 5, (1956), 521.

22. Treshchov. G.G. "An Experimental Investigation of Heat Exchange Mechanism with Surface Boiling Water." Teploenergetika, 4, No. 5, (May, 1957), 44.

23. Jakob, M. "Local Temperature Differences Occurring in Evaporation, Condensation and in Catalytic Reactions." Temp. its Measurement and Control in Science and Industry, Reinhold, N.Y., 1941, pp. 834.

24. Corty, C. and Foust, A.S. "Surface Variables in Nucleate Boiling." Chem. Eng. Prog. Symp., Series No. 17, 51, 1955.

25. Bankoff, S.G. "Ebullition from Solid Surfaces in the Absence of a Pre-existing Gaseous Phase." Trans. ASME, 79, (May, 1957), 735.

26. Ellion, M. "A Study of the Mechanism of Boiling Heat Transfer." J.P.L., Memo 20-88, Pasadena, Calif., March, 1954.

27. Siegel. R. and Usiskin, C. "A Photographic Study of Boiling in the Absence of Gravity." Trans. AMSE, 81, Series C, No. 3, (August, 1959), 230.

28. Gambill, W.R. and Green, N.D. "A Study of Burnout Heat Fluxes Associated with Forced-Convection, Subcooled, and Bulk Nucleate Boiling of Water in Source-Vortex Flow." Chem. Eng. Prog., 54. No. 10, (October, 1958), 68.

29. Merte, H. Jr., Clark, J.A. "A Study of Pool Boiling in an Accelerating System," University of Michigan Research Institute Proj. No. 2646, November, 1959.

30. McAdams, W.H. Heat Transmission. 3rd Edition. New York: McGraw-Hill Book Co. Inc., 1954.

31. Bonilla, C.F., Busch, J.S., Stadler, A., Shaikhmahmud, N.S., and Ramachandran, A. "Pool Boiling Heat Transfer with Mercury." Reactor Heat Transfer Conference of 1956.

32. Costello, C.P., Tuthill, W.E. "Effects of Acceleration on Nucleate Pool Boiling," Personal Communication to J.A. Clark, December, 1959.

33. DeBortoli, R.A., Green, S.J., *et al.* "Forced-Convection Heat Transfer Burnout Studies for Water in Rectangular Channels and Round Tubes at Pressures above 500 psia." *AEC Research and Development Report WAPD-188*, Westinghouse Atomic Power Div., Bettis, June, 1958.

34. Nishikawa, K. and Urakawa, K. "Experiment of Nucleate Boiling under Reduced Pressure." *Trans. Soc. Mech. Eng.*, Japan, 23, No. 136, (December, 1957), 935.

35. Zuber, N. "On the Stability of Boiling Heat Transfer." Trans. ASME, 80, No. 3, (April, 1958), 711.

36. Rohsenow, W.M. and Clark, J.A. "A Study of the Mechanism of Boiling Heat Transfer." *Trans. ASME, 73*, No. 5, (July, 1951), 609.