RICE UNIVERSITY

Interaction of Active Sites
in
Pool Boiling Heat Transfer

by

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ABSTRACT

Heat transfer studies were performed on saturated carbon tetrachloride under natural convective and nucleate boiling conditions. A flat, polished, stainless steel plate was the heat source; and degassed, spectroscopic grade carbon tetrachloride was the heat transfer fluid. Tests were conducted at both one, and one and one-half atmospheres. Heat flux measurements were made for superheats ranging from 10 to 40°F, with corresponding heat fluxes ranging from 700 to 2500 Btu/hr-ft\(^2\).

Two runs were performed. In the first run, seven equidistant cylindrical cavities, 0.006 inches in diameter and 0.010 - 0.012 inches deep, were drilled into the central portion of the heat transfer area. In the second run, additional holes with identical geometries were drilled halfway between the cavities studied in the first run. These artificial cavities served as active sites for the boiling studies.

The purpose of the study was to investigate the nature of the interaction between these carefully positioned sites under nucleate boiling conditions. However, the drilling procedure created small protrusions around the mouths of the cavities. As a result, the bubbles obtained during nucleate boiling were smaller than would normally be expected, and no significant interaction between active sites was obtained.
for the site population densities studied. Nonetheless, the results provide useful information about the influence of cavity lip geometry on the heat transfer characteristics of an ebullating site. The boiling results obtained in this study reveal that the local heat transfer coefficient for the protruding lip case is an order of magnitude smaller than those obtained in other studies involving non-protruding lips.
INTRODUCTION

The general shape of a boiling heat transfer curve is presented in Figure 1. This curve may be divided into three characteristic regions as follows:

1. Natural convection.

The portion of Figure 1 labeled "N.C." depicts the natural convection region. Natural convection heat transfer is characterized by an almost linear relationship between heat flux and temperature driving force.

2. Nucleate boiling.

In the regions labeled "N.B.(I)" and "N.B.(II)", higher heat transfer coefficients are obtained. "N.B.(I)" is that portion of the curve corresponding to somewhat lower heat fluxes (the present study). In this region the heat transfer behavior is similar to natural convection; and the heat flux is almost linear with the temperature driving force. Note that even with low fluxes, boiling heat transfer yields larger heat transfer coefficients than a natural convection regime. Along "N.B.(II)", much higher heat transfer coefficients are obtained, and small increments in surface temperature result in large increments in net heat flux.

Physically, both "N.B.(I)" and "N.B.(II)" are
HEAT TRANSFER CHARACTERISTICS OF A BOILING LIQUID

\[ q \] (HEAT FLUX TO SURFACE)

\[ T_s - T_L \] (TEMPERATURE DRIVING FORCE)

**Figure 1**
characterized by steady ebullition from certain distinct sites on the surface. As the temperature is raised, the number of sites and the shape of the gaseous phase boundary changes. At "N.B.(I)" distinct bubbles approximately spherical in shape form at widely separated locations. Along "N.B.(II)" continuous vapor columns emanating from many closely grouped sites are observed.

3. Burnout and film boiling. (Not shown in Figure 1)

There is a point of maximum temperature difference at which any further temperature increment will not yield increases in heat flux; this point is referred to as the "burnout point". Higher temperature differences produce a continuous vapor film on the surface; and heat transfer occurs by convection and radiation across this film.

In the heat flux range of "N.B.(I)", the boiling curve is a multivalued function of the past heating history of the liquid, the surface geometry, the surface chemistry and the physical properties of the liquid. Primary interest will be focused in this region.
LITERATURE AND BACKGROUND

Recent investigations in the field of nucleate boiling have shown that ebullition occurs preferentially at deep cavities on the heat transfer surface (1) (2) (3) (5). In addition, most authors (7) (10) (11) (13) (15) (16) believe that a superheated thermal boundary layer exists adjacent to the heating surface; that a thermal-hydrodynamic interaction between this superheated layer and a bubble growing from the surface is primarily responsible for the large heat transfer coefficients obtained in nucleate boiling.

The disagreement between various workers lies in the nature of the thermal-hydrodynamic interaction between the superheated layer and the growing bubble. Basically, there are two modes of thought as to the nature of this interaction:

1. the growing bubble withdraws latent heat of vaporization from the nearby thermal boundary layer, or
2. the growth and detachment of a bubble disrupt the thermal boundary layer, and promote very high heat fluxes in the neighborhood of the site.

Rohsenow and Clark (15) made a photographic study of the number, frequency, and volume of bubbles leaving a superheated surface. They used this information, together with known values of the latent heat of vaporization and the surface tension of the heat transfer fluid to calculate the amount of heat required for bubble formation. The increase
in heat transfer from a non-boiling to a boiling regime was two-fold (350,000 Btu/hr-ft$^2$ to 721,000 Btu/hr-ft$^2$); but the heat absorbed by the bubbles amounted to only one percent of this increase. From these results, the authors concluded that the second mechanism is the more probable one.

Hsu and Graham (10) obtained schieren and shadowgraph images of nucleate boiling phenomena. Their results show a strong agitation effect on the thermal boundary layer produced by the departing bubbles. The ebullition phenomena which they observed may be summarized as follows:

1. following departure of a bubble from the surface, the thermal boundary layer was disrupted,

2. cooler fluid from the bulk liquid pool rushed into the void created by the recently departed bubble and

3. an appreciable waiting period was required to restore the thermal boundary layer.

The disruption of the thermal boundary layer occurred within a circular region approximately one bubble diameter in radius. From these observations, the authors concluded that the heat transfer coefficient should be a strong function of bubble population, size, and frequency.

Engelberg-Forster and Greif (7) proposed a vapor-liquid exchange action as the mechanism of heat transfer. In their model the authors argue that the growing bubble pushes a
quantity of superheated liquid from the thermal layer into the bulk liquid. Crude calculations based on the displacement volumes of the growing bubbles were presented to support their argument. Reasonable agreement with existing data was obtained.

Very recently, Moore and Mesler (3) performed some interesting studies which indicate that the latent heat transport mechanism may possibly be of importance. The authors used a special thermocouple in their studies. The thermocouple had an exceedingly rapid response time (one microsecond), and was mounted flush with the heating surface, enabling direct measurement of surface temperature. The results of the boiling experiments were quite surprising. Over the majority of the time span, the temperature was fluctuating slightly about some mean value; however, at the end of a given time period temperature drops of twenty or thirty degrees below the mean value were obtained. These thermal "dips" occurred at regular time intervals, with a duration of about two milliseconds; and their frequency varied directly with the average heat flux. Calculations were performed which indicated that the local heat flux increased six-fold over the average heat flux, during the rapid temperature drop. In order to determine whether or not the inrush of bulk fluid into the locale of the departed bubble was responsible for the steep temperature drop (as
would be indicated by the photographic data of Hsu and Graham), the authors analyzed a hypothetical heat transfer model. The results indicate that the bulk liquid temperature necessary to account for the observed flux would have to be \(-147^\circ F\), which is obviously not reasonable for the fluid studied (water).

As an explanation for their results, Moore and Mesler propose that the bubble grows in a mushroom fashion. The base of the mushroom is exposed to the superheated layer, while the mushroom cap protrudes into the bulk liquid. As the cap of the mushroom grows, the base maintains its small cylindrical geometry. When the mushroom cap reaches a certain critical diameter, the underlying superheated layer suddenly evaporates into the bubble, which then assumes a hemispherical form. It is this rapid vaporization that is assumed to be responsible for the severe temperature drops observed in the solid.

Several investigators (5) (8) (14), have observed that the heat transferred per active site is a rapidly decreasing function of the number of sites. This decrease in site efficiency with increasing number of sites was primarily attributed by most authors (5) (7) (8) (9) to hydrodynamic interference between adjacent bubble columns. However, the work of Moore and Mesler suggest another possible explanation to the observed decrease in site efficiency with increasing site population, namely that there exists a competition for sensible heat in the solid surface. With increasing
site populations this competition would become severe and result in lower site efficiencies.
THEORY

The growth behavior of a bubble in contact with a heated solid surface is intimately related to the geometry of the surface defect from which the bubble is growing. Nucleation and steady ebullition have been shown experimentally (3) (5) to take place from cylindrical cavities. Theoretical arguments offered by Bankoff (1) (2) confirm these observations. However, no work has been done on the effect of cavity lip geometry on bubble growth behavior and boiling heat transfer. Many investigations have taken as their model a cylindrical cavity whose mouth is flush with a geometrically smooth, plane surface. If however, one assumes that the lip of the cavity protrudes above the surrounding surface (see Figure 2, page 10), the following relationship can be derived from geometrical considerations (6):

$$r_B = \frac{r_C + r_L - r_L \cos \theta}{\cos (\theta - \beta)} \quad (\beta > 0) \quad (1)$$

where:
- $r_B$ = radius of the growing bubble,
- $r_C$ = radius of the cavity from which the bubble is growing,
- $r_L$ = radius of the lip at the edge of the cavity,
- $\theta$ = contact angle at the gas-liquid-solid interface (measured in the liquid), and
- $\beta$ = the angle made by the lip radius vector from the horizontal to the gas-liquid-solid interface.
Ebullition from a protruding lip

Figure 2
The derivation assumes that the contact angle $\theta$ is preserved during the growth process, and that the lip radius and cavity radius may be assigned constant values.

Upon substituting various values of $\beta$ in Equation (1), the following variation of bubble radius is obtained:

$$
\frac{r_B}{\frac{r_c}{\cos \theta} + \frac{(r_c + r_L)}{\sin \theta}} = \begin{cases} 
\frac{\beta}{0} & \text{if } \beta = \frac{\pi}{2} \\
(\frac{r_c + r_L}{\sin \theta}) & \text{for } \beta \neq \frac{\pi}{2}
\end{cases}
$$

A qualitative plot of $\gamma_r$, (defined as the ratio of the radius of a bubble growing along a protruding lip to the radius of a bubble growing along a horizontal surface) is sketched in the lower right-hand corner of Figure 2. Point "0" corresponding to $\beta = \frac{\pi}{2}$ is the point at which the protruding lip geometry begins to have an effect. It is seen from this plot that as the bubble grows out along the lip, its radius of curvature becomes extremely large in comparison to a hypothetical bubble growing along the flat plane. In practice, however, the bubble will detach when the buoyancy forces (varying with $r_B^3$) overcome the surface tension forces (varying with $r_B$). Thus, for a protruding lip, a bubble will detach before it has a chance to grow to an appreciable size; its effect on the thermal boundary layer can be expected to be considerably smaller than that observed by Hsu and Graham (10).
EXPERIMENTAL

Definition of the problem

The purpose of this investigation was to study the effect of active site population, active site distribution, and system pressure on the boiling heat transfer behavior of carbon tetrachloride from pre-determined artificial cavities.

Experiments were to be performed in which site population and distribution were varied at constant pressure, and pressure varied for a given site population and distribution. An attempt was to be made to obtain insight into the nature of the interaction of active sites at low heat fluxes.

The heat transfer surface and test liquid were to constitute an isolated system whose physical characteristics could be maintained or varied in controlled fashion. The apparatus, routines, techniques and difficulties associated with the realization of these goals are presented on the following pages.
Experimental Apparatus

The experimental apparatus used in this work consisted of the following major components:

1. the heat transfer assembly,
2. the boiler system, and
3. the auxiliary glass system

The heat transfer assembly. The final design of the heat transfer assembly is depicted in Figure 3. The heat transfer assembly was designed and constructed to provide,

1. a plane metallic surface with low thermal conductivity and relatively large surface area which could be successfully polished by standard techniques, and
2. a heat source with well defined geometry which would enable precise evaluation of heat losses and the surface temperature.

The first requirement was satisfied by employing type 304 stainless steel. In addition to its low thermal conductivity, this metal resists corrosion under the conditions present in this study. The second requirement was achieved by fashioning a cylindrical heating element from pure copper round.

The stainless steel piece and the copper block were machined to a smooth finish on all exposed surfaces. The copper block had five holes 0.040 inches in diameter drilled
THE HEAT TRANSFER ASSEMBLY

Figure 3
into its curved portion as follows:

1. three holes on 120° centers (1.0, 0.75, and 0.5 inches deep) located 0.10 inches from the upper end,

2. one hole (1.0 inches deep) located 0.20 inches below the one inch hole in (1.) and

3. one hole (1.0 inches deep) located 0.20 inches below the one inch hole in (2.)

A recess was machined in one side of the stainless steel plate to receive the copper block. The copper block was then silver soldered to the stainless plate. The final assembly was similar to that depicted in Figure 3; however, the plate shown was accidently destroyed and another plate fashioned. A diagram of the plate used in this study appears in Appendix (A-3).

Three concentric grooves (not shown in Figure 3) were machined in the underside of the stainless steel plate to receive the thermocouples used to measure radial temperatures for the heat loss calculations. A hollow aluminum cylinder was wrapped with a nichrome heating element. The upper surface was polished with a buffing wheel by a local jeweler.

During a run, the entire copper block and surface heater were surrounded by a dewar flask. The annular space between the dewar flask and the heater was packed with glass wool. Heat losses across the dewar flask were
minimized by adjusting the temperature drop across the flask walls with an electrical heating tape. The tape was wound over an aluminum foil wrapping along the lateral surface of the flask. A second heating tape was wrapped on the periphery of the stainless steel plate to reduce radial heat losses.

The boiler system. The boiler system shown in Figure 4 was constructed to provide:

1. a geometric configuration at its lower end which would receive the heat transfer assembly in such fashion that the boiler system and the heat transfer assembly would form a closed system which could be evacuated to $10^{-5}$ millimeters of mercury or better,

2. a system whose internal surfaces could be heated to temperatures in excess of $300^\circ F$ under high vacuum conditions,

3. connections with the auxiliary glass apparatus,

4. automatic pressure control for pressures in the range of one atmosphere,

5. windows for observing the ebullition phenomena, and

6. a means of determining bulk liquid temperatures.

To achieve the first requirement, eight evenly spaced holes were drilled and tapped in the underside of the stainless steel housing. Also, eight holes were drilled on the
periphery of the stainless steel portion of the heat transfer assembly. Various gasketing materials were tried to seal the flanged area, but best results were obtained with a teflon O-ring.

Temperatures in excess of 300°F on the internal surfaces of the boiler system were obtained with a large aluminum cylinder wrapped with nichrome ribbon and insulated with one inch of asbestos cement (hereafter referred to as the bakeout oven). It was constructed to fit over the stainless steel housing. The bakeout oven and surface heater enabled the experimenter to obtain the desired temperatures.

The boiler system was connected to the high vacuum pumping system with a Veeco* high vacuum valve. One valve port was welded to the boiler unit (as shown in Figure 4), while the other valve port was connected to a flange. The flange in turn, was welded to a kovar-to-pyrex graded seal. A similar connection was employed from the boiler system to the liquid degassing system; a high vacuum needle valve was incorporated at this point.

Automatic pressure control was vital. Fine automatic control of the boiler system pressure was achieved with a hollow metal leg welded at the flanged section above the condenser (as shown in Figure 3) operated in the following fashion: a small nichrome-wound heater (hereafter referred to as the pressure control heater) was placed over the metal

*Veeco is the trade name for Vacuum Electronics Corp., Plainview L.I., N.Y. The valve used was a stainless steel, bellows sealed, angle valve, type R25PSS, with a teflon seat.
FIGURE 4: BOILER ASSEMBLY
leg, and a brass cooling jacket sat over this heater. If the pressure in the boiler system was low, the mercury level in the control manometer (shown schematically in Figure 5) would break contact with a nichrome probe and an electronic relay switch then threw the pressure control heater on. If the system pressure was too high, the mercury level and nichrome probe would make contact and the pressure control heater was turned off. The pressure control heater evaporated small amounts of liquid at low system pressures, and the brass cooling jacket condensed small amounts of vapor at high system pressures.

The physical behavior of the heat transfer fluid was observed through two pyrex windows fastened to the boiler as shown in Figure 4. The one half inch thick windows were sealed to the boiler with teflon O-rings. Each window and its O-ring were held in place with a metal flange and four flange bolts on 90° centers. A flat lead gasket was placed between the window and the flange to eliminate point loading.

Bulk liquid temperatures were determined from a thermocouple inserted in a small tube passing through the boiler wall. The tube was sealed in the wall with a special teflon collar followed by a packing nut.

The auxiliary glass system. The all-glass auxiliary system was designed and constructed to:

1. produce pressures in the boiler assembly ranging from $10^{-5}$ millimeters of mercury to one and one half atmospheres,
2. measure accurately the pressures mentioned in (1.), and
3. degas a reasonably large supply of heat transfer fluid, and keep this supply in isolation for a reasonable length of time.

The essential features of the final design are shown schematically in Figure 5. Note that in the actual physical system, provision was made for an additional liquid degasser so that several test fluids could be studied with a minimum of contamination. Since carbon tetrachloride was the only fluid utilized in the present study, valve V3 was closed and the second liquid degassing unit removed from the system.

The glass system divides into two systems with valve V2 closed - the high vacuum arrangement and the liquid degassing system.

1. The high vacuum system. The high vacuum system consisted of the following components:
   a) a two stage, water cooled mercury diffusion pump (shown as pump 2 in Figure 5),
   b) a two stage vacuum pump capable of pressures to $10^{-4}$ millimeters of mercury (shown as pump 1 in Figure 5),
   c) a Veeco vacuum gauge type RG-3A to measure boiler system pressures during degassing of
FIGURE 5: SCHEMATIC DIAGRAM OF APPARATUS

HIGH VACUUM EQUIPMENT
the interior metal surfaces (see "Metal Degassing" in "Experimental Procedure"), and

d) two dewar flasks (traps T1 and T2 in Figure 5) filled with liquid nitrogen. (Trap T1 was used to condense greases, traces of CO₂ and CCl₄, and protect the ionization gauge from mercury contamination, while trap T2 protected the forepump from mercury contamination.)

2. The liquid degassing system. The liquid degassing system is shown schematically in Figure 5. It consisted of a liquid reboiler, vapor bubbler, fractionating tower packed with twisted metal ribbon, water jacketed condenser, and sonic orifice. Noncondensible gases were removed from the liquid degassing system by the auxilliary vacuum pump. Details of operation of all the above-mentioned components may be found in any standard text on distillation, and for brevity will not be presented here.

The heat transfer fluid was kept in isolation during the liquid degassing operation (see "Degassing of Liquid" under "Experimental Procedure") by closing valve V2. During this period, the control manometer measured the pressure of the liquid degassing system. While, during an experimental run, the control manometer controlled the pressure in the
boiler. To accomplish this control with the same manometer, a thick walled U-tube (trap 3 in Figure 5) was frozen by immersing the U-tube in a dewar flask containing a dry ice-methyl cellosolve mixture. In summary, with valve V2 open and trap 3 frozen, the boiler assembly pressure was controlled; while with valve V2 closed and trap 3 thawed, the liquid degassing system pressure was controlled.
Experimental Procedure

The experimental procedure used to obtain the heat transfer data may be divided into two steps:

1. preparation of the boiler assembly and the heat transfer fluid for a given run, and
2. measurement of the heat input to the surface as well as the temperatures of the surface and the liquid.

The following is an exposition of the laboratory routine. It will be assumed that a run has been completed and that preparation for a succeeding run must be made. All references to equipment components will be found in Figure 4, unless otherwise specified.

**Preparation of Boiler Assembly and Heat Transfer Fluid**

1. **Removal of existing fluid from the boiler unit**
   
   Residual bubble columns from the previous run were eliminated by cooling the heat transfer surface and reducing the power to the bake-out oven. The voltmeter used for the heat flux measurements was then removed from the heater circuit, and the ammeter switched to its highest range. Right liquid valve V2 was then closed with a torque wrench to forty five inch pounds. Immediately, the dry-ice cellosolve mixture was removed from trap 3, and the boiler unit condensor water was turned on. After a few
minutes, the fluid frozen in trap 3 melted and the right liquid valve was slowly opened. The boiler fluid was then boiled up into the liquid degassing system. The boil-up period was accelerated by turning the surface heater and bakeout oven up to voltages higher than operating values, and by turning the boiler unit condensor water off. In one hour, the liquid remaining in the boiler unit was about one quarter of an inch deep. At that time, the surface heater and bake-out oven were turned off, the right liquid valve (V2) was closed, and the boiler unit condensor water was turned on to give a rapid flow rate.

When all interior metal surfaces had attained near ambient conditions, the heat transfer assembly was removed from the boiler system, carefully placed in a safe locale, and the next heat transfer assembly was prepared.

2. **Cleansing the Interior Metal Surfaces**

Preparation for degassing of the interior metal surfaces proceeded by thoroughly flushing the boiler unit interior with reagent grade carbon tetrachloride and wiping the exposed surfaces with lint-free paper. Window sections were examined for deposits of scale. If scale deposits were found, the flange, pyrex window and teflon O-ring were removed, flushed with reagent grade carbon tetrachloride, and replaced.

Then, the new heat transfer assembly and teflon O-ring
were flooded first with reagent grade acetone and then with reagent grade carbon tetrachloride, and the assembly allowed to drain vertically dry. The new heat transfer assembly was given a final rinse with spectroscopic grade carbon tetrachloride, and the assembly and teflon O-ring were bolted to the boiler unit. The bolts were uniformly loaded with a torque wrench to forty five inch pounds.

3. **Degassing the Metal Surfaces**

Start up of the high vacuum equipment was carried out concurrently with the above cleansing routine. High vacuum valve V1 was closed at this point. Pump 1 and the mercury diffusion pump were operating and the liquid nitrogen traps (traps T1 and T2) had been filled. After the heat transfer assembly had been secured to the boiler housing, valve V1 was cracked and the boiler unit evacuated at room temperature for one hour, or until the system pressure was between 1 or 2 x 10^{-4} millimeters of mercury.

When the boiler pressure had attained the desired value, the bake-out oven was given an initial setting of forty volts, and the surface heater was set to ten volts. During the next few hours, the surface heater voltage was incremented until the desired degassing temperature was reached. The teflon O-rings were tightened uniformly with a torque wrench with each corresponding temperature increment. Note that the bake-out oven voltage setting remained
fixed. Higher voltages on the bake-out oven would have initiated degassing of the entire boiler, resulting in the emission of large volumes of gas.

While initial evacuation at room temperature was effected, the set of copper-constantan thermocouples was greased, positioned in the copper block, and secured with heat resistant electric tape. The dewar flask was positioned around the copper block and stuffed with glass wool.

4. Degassing of Liquid

Liquid degassing was the last major step in the preparation for a run. Following closure of the right liquid valve V2 and the sonic orifice valve, the liquid degasser heater was turned on to sixty five volts. About fifteen minutes later, a loud bump was heard, indicating that the liquid charge in the reboiler was refluxing. When the bump was heard, the vapor bubbler heater was set at four volts, condensor water to the fractionating tower was started, the auxilliary vacuum pump was turned on, and the sonic orifice valve was cracked open. At this stage, pressure control in the liquid reboiler was controlled with the control manometer.

Small amounts of liquid condensed in the glass lines leading to the boiler and the control manometer. The condenser liquid was boiled from these lines by gently heating with a soft flame. If the liquid had been degassed well,
these glass lines refilled quite rapidly.

5. **Filling the Boiler Unit**

The high vacuum system was isolated from the boiler and shut down by closing valve V1. The right liquid valve V2 was then opened, the boiler condenser water was turned on, and the degasser test fluid was boiled into the boiler chamber. When the desired liquid level was reached, right liquid valve V2 was closed, and the liquid degassing system was isolated from the boiler. The liquid degassing system was shut down by turning the liquid degasser heater, vapor bubbler heater, and liquid degasser condenser off, by closing the sonic orifice valve, and finally by shutting off the auxiliary vacuum pump.

Once the liquid degasser was shut down, any vapor enclosed in trap 3 would have been condensed. A dewar flask filled with methyl cellosolve was brought under the trap and powdered dry ice was slowly added. When trap 3 was successfully frozen, right liquid valve V2 was opened, the pressure control heater turned on, and the boiler system brought up to operating pressure.

6. **Final Adjustment of System Pressure**

A vital part of the measurement routine was adjustment of the system pressure to the value at which heat transfer data were to be taken. The bake-out oven was set to its operating voltage at the desired pressure, and then
the surface heater was raised to thirty five volts. The high voltage on the surface heater insured vigorous boiling of the test fluid. The millivolt potentiometer was set to read the saturation temperature at the desired operating pressure; e.g., 3.210 millivolts or 170.0°F, the saturation temperature of carbon tetrachloride at one atmosphere. The potentiometer's galvanometer was deflected at regular intervals, and when the desired temperature reading was obtained, the mercury level at the open leg of the control manometer was immediately observed. The nichrome probe was then adjusted to the observed mercury level. The pressure in the system was then lowered below the probe setting by turning on the condenser water. When the mercury level was about one inch below the probe setting, the pressure control heater circuit was plugged in, and the boiler condenser water shut off. As the boiler system pressure increased once again, the millivolt potentiometer was watched cautiously. A "click" from the pressure control relay switch indicated that the mercury level had made contact with nichrome probe. If the "click" was heard at the same instant the potentiometer was at the desired millivolt setting, then the probe was at its correct position. Correct placement of the manometer probe was usually obtained after two trials.

7. The Measurement Routine

Measurements could be made when the pressure
control system was observed to be operating successfully (constant clicking of the relay switch). A given heat input was decided upon and the surface heater was adjusted to this desired heat input by measuring voltage and current to the surface heater. The steady state temperature distribution in the copper block was anticipated and the voltage to the dewar heating tape was adjusted accordingly. After one and one half hours, ice was placed in the thermocouple reference junction and the twelve thermocouple leads were stirred vigorously in the ice water bath. The galvanometer was standardized, and all twelve thermocouples were read. About fifteen minutes later, the procedure was repeated. The process was continued at fifteen minute intervals until all twelve thermocouple readings were either no longer changing with time, or until the small changes encountered showed no particular trend. The voltage and current were read again, a new setting made, and the procedure repeated.
EXPERIMENTAL TECHNIQUES

1. **Preparation of Bridge Thermocouples**

The successful completion of the experimental objectives of this study depended in large measure on accurate evaluation of the radial heat loss. This in turn required precise knowledge of the radial temperature distribution on the thin stainless steel bridge. The desired temperature distribution was obtained by securing six copper-constantan thermocouples in three concentric grooves milled into the underside of the fin.

The thermocouples were secured in these grooves with "Foreign Cement"*. The cement was sufficiently dry in a few minutes so that manual support could temporarily be removed. Then "Scotch Brand"** Electric Tape No. 27 was pressed over the thermocouple-cement joint, small aluminum blocks were clamped onto the cemented area, and the area was heated to about 350°F. Both the electric tape and the cement had optimum adhesive strength at the operating temperatures involved.

Thermocouple calibrations were performed at a temperature close to the operating conditions. The boiling point of water at atmospheric pressure was chosen because the calibration could be performed with a minimum of apparatus at temperatures sufficiently close to those at

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** Scotch Brand is the trade name for Minnesota Mining and Manufacturing Company, St. Paul, Minn.
operating conditions. First a large copper block was placed into a water bath. Two holes were drilled into the copper piece - one hole received the calibration thermometer, and the other hole received a glass tube. The thermocouples were inserted into the glass tube, copper powder was added to insure good thermal contact, and the open end of the tube was plugged with cotton. When the water had been boiling vigorously for one-half hour, the thermocouples were read.

2. **Drilling of the Artificial Sites**

Ebullition from pre-determined locations on the heat transfer surface was achieved by drilling round cylindrical cavities in the polished surface. A watchmaker drilled the holes 0.006 inches in diameter, and 0.010 - 0.012 inches deep. The cavities had small protrusions at their mouths.
EXPERIMENTAL RESULTS

The results of this study are presented as heat flux versus temperature excess plots. Tabulated data from which these graphs were plotted are presented in appendix B-2.

Natural convection data were taken at one atmosphere for two runs, with temperature excesses ranging from 20 to 40°F. A check point was taken for the one and one-half atmosphere studies. The results are shown in Figures 6 through 8. The natural convection data for Run 1 were almost coincidental with that for Run 2. The deviations were assumed to stem from systematic errors in going from one run to the next; hence only the natural convection data for a given run were used with the boiling data for that run. During the gathering of natural convection data in Run 1, ebullition from the teflon gasket was observed, as a result, data could be obtained to determine the effect of long range ebullition on natural convection results. The results of the long range interference on natural convection are presented in Figure 9.

Nucleate boiling results are presented in Figures 6 through 8. Most of the bubble columns were from the drilled sites. Occasionally, a few stray sites would appear at the higher superheats. The positions of both artificial sites and strays were carefully recorded, and are presented in the appendix B-2.
Increasing the bubble column at a fixed temperature excess and fixed pressure resulted in an increase in the net heat flux \( q_e \). The increase in heat flux from one bubble column to six bubble columns was greater at one atmosphere, than the corresponding increase at one and one-half atmospheres.

The effect of increasing the temperature excess \( (T_S - T_L) \) for a fixed bubble column population and fixed pressure was an increase in the net heat flux. In most cases, the separation between natural convection curves and nucleate boiling curves increased slightly with an increase in temperature excess.

The boiling heat fluxes for both one and six column populations at one and one-half atmospheres were greater than boiling results at one atmosphere. This shift was not expected and will be discussed in the following section.

The boiling heat transfer contribution per active site \( (Q_e/n) \) was not a strong function of the number of sites. The values obtained were an order of magnitude smaller than values reported by other workers (5).
Boiling Data: Run 1 - one atmosphere

Figure 6

\[ Q_{\text{v}} \text{ (BTU/hr-ft}^2) \]

\[ T_{5} - T_{L} \text{ (} ^{\circ}\text{F)} \]

- Natural Convection
- Natural Convection Check (1/2 ATM)
- Numbered points refer to number of ebulliating sites
Boiling Data, Run 1 - one and one-half atmospheres

$Q_z$ (BTU/hr-ft$^2$)

$T_s - T_L$ (°F)

NATURAL CONVECTION

- 6 COLUMNS EMBOLATING
- 1 COLUMN EMBOLATING
Figure 8

Run 2; one atmosphere

\[ q \text{ (BTU/hr-ft}^2) \]

\[ T_s - T_L \text{ (°F)} \]

Natural Convection

Five Columns

Numbers refer to the number of Boiling Columns
DISCUSSION OF RESULTS

Errors

A complete treatment of the errors incurred in the measurements and in the calculation of the results is presented in appendix C. In summary, the estimated overall error in the heat flux is fifteen per cent; while the estimated error in temperature excess is six and one-half per cent. The former error estimate involves both random errors of measurement and systematic errors of procedure. Since the results of this study are primarily concerned with comparisons of heat flux for a given temperature excess, the random errors of measurement are the more important, and are estimated to be less than ten per cent.

A measure of the reliability of the heat transfer results can be seen by comparing the natural convection results for this study with those presented by Denny (5). The present results are approximately ten per cent higher than those obtained in the earlier work. This systematic deviation is probably due to one or both of the following factors:

1. differences in procedure for determining radial heat losses, or
2. differences in geometry of the heat transfer surface.
The method for evaluating the radial fin losses in this study incorporated experimental measurements of the temperature profile on the fin with an approximate calculation procedure for determining the radial heat flux. Denny employed a theoretical approach and solved the fin loss problem numerically. Both methods have different systematic errors associated with them.

In addition, the heat transfer unit used by Denny (5) was much smaller in diameter, with an area for heat transfer and a fin length which were approximately 1/4 and 1/2 respectively of those of the present study. The differences undoubtedly caused contrasting effects on the hydrodynamic behavior of the pools of liquid, due to the non-infinite boundary conditions.

Natural Convection

Natural convection data were taken for each of the two runs performed and were found to be substantially in agreement. The small differences were probably due to systematic errors arising from changes in instrumentation from one run to the next. A natural convection check point at one and one-half atmospheres agreed with the natural convection data taken at one atmosphere. However, the check point was taken at a large value of the temperature excess, and other effects discussed below, were
felt to be present at that range of superheat. The assumption that natural convective heat transfer results were the same for the two pressures was felt to be in error.

When the initial natural convection data was taken, ebullition was observed at four sites on the teflon gasket. Data were obtained, so that the long range effects produced by these extraneous bubble columns could be determined. The data for no ebullition on the teflon gasket, and for one and four bubble columns on the gasket are presented in Figure 9. The results indicate that greater heat fluxes were obtained with greater numbers of bubble columns. The results for data taken with one bubble column and data taken with no bubble columns are contradictory at the greater temperature excesses. The intersection of the two curves indicates greater heat fluxes were obtained with no interference. The operating conditions for the three natural convective modes were carefully investigated. The data with one bubble column on the gasket were taken with the bake-out oven setting at thirty volts, and the data with no ebullition were taken with the bake-out oven setting at fifty five volts. When the bake-out oven was set at the greater voltage, more heat transfer fluid would be evaporated and condensed than with the bake-out oven at the lower voltage.
An increase in the condensate flow rate increased the amplitude of the disturbances present at the gas-liquid interface, which in turn increased the heat flux. The effect of condensate returning to the liquid pool would be pronounced at the higher superheats, and higher pressures (higher pressures result in higher temperatures). This hypothesis was borne out by the natural convective results obtained by Denny (5). The heat fluxes he obtained were greater when the gas-liquid interface was in a turbulent mode.

**Nucleate Boiling**

Nucleate boiling data obtained in this work could only be compared with another investigator's extrapolated values (5). The results for one bubble column were about 75 Btu/hr*ft\(^2\) lower in the present study. Data for higher bubble column populations could not be compared, but the present results for six bubble columns were equivalent to results for two and three bubble columns in the earlier work (5).

The larger heat fluxes for one, two, and three columns populations that were obtained in the earlier work and not obtained presently may be attributed to differences in lip geometry. In the present study, a watchmaker drilled the artificial sites and he made slight eruptions (from 0.005 to 0.010 inches high) at the cavity mouth. During the boiling run, the bubble growth behavior could be observed at the
lower heat fluxes. The base diameter of the bubble was roughly the same as the diameter of the drilled cavity. This observation was quite different from that of the earlier work (5), where bubbles were observed to grow out onto the surface and the base diameter before detaching was several times larger than the cavity diameter.

The results of Hsu and Graham's photographic studies (10) indicate that bubbles departing from a heated surface disrupt the thermal boundary layer. The disruption and subsequent inrush of bulk liquid promote high rates of heat transfer. The thermal boundary layer thickness was estimated at one hundredth of an inch*. This value is about the same as the estimated height of the protruding lip (0.005 to 0.010 inches). The lip was shielding the bubble from the thermal boundary layer, and it was considerably reducing the base diameter of the bubble. A small base diameter would not significantly disturb the fluid in the superheated layer, and small heat transfer coefficients could be expected.

The data obtained in this and earlier work (5) are successfully interpreted on the basis of this analysis. In addition, some success in the interpretation of the one and one half atmosphere data can be achieved. The absolute values of the boiling heat fluxes at one and one-half atmospheres were greater than the one atmosphere boiling heat fluxes.

* A rough estimate was made by dividing an observed heat transfer coefficient into the thermal conductivity of carbon tetrachloride.
fluxes. This is attributed to a systematic error incurred at the higher pressure. However, the difference in heat flux at one and one-half atmospheres between one column and six columns was 25 Btu/hr-ft², while the corresponding difference at one atmosphere was 100 Btu/hr-ft². At the higher pressure, both the detachment diameter of the bubble and the frequency of bubbles leaving the surface were smaller (5). Hence, additional bubble columns at higher pressures would not be expected to increase the heat flux as much as a similar population at a lower pressure.

The heat transfer contribution per active site (Q*/n) was plotted against the number of active sites (n). The error incurred in preparing this plot is very large in comparison with the precision of the heat flux measurements. A one per cent deviation in both natural convective and boiling heat transfer results, would result in a forty per cent error in Q*/n. In view of the large errors incurred, nothing more than a qualitative discussion is permissible. The values of Q*/n for the one atmosphere data are an order of magnitude less than those obtained by Denny (5). In addition, the plots that are presented here indicate little functional dependance on Q*/n with n, which in turn implies no interaction exists. The corresponding plot at one and one-half atmospheres is in error because the one atmosphere natural convection data was used in the analysis.
Interaction of active sites

\[
\frac{Q^*}{n} \text{ (BTU/HR-ACTIVE SITE)}
\]

\[
\Delta T = 29.0, 1\text{ATM}
\]

\[
\Delta T = 31.5, 1\text{ATM}
\]

\[
\Delta T = 25.5, 1\text{ATM}
\]

\[
\text{n, (NUMBER OF ACTIVE SITES)}
\]

**Figure 10**
CONCLUSIONS

1. Nucleate boiling is preferentially maintained in deep cylindrical cavities on the solid heating surface. The dimensions necessary for a given surface defect to support steady ebullition are determined primarily by the degree of surface superheat.

2. The hydrodynamic mode of the liquid-vapor interface has a pronounced effect on the quantity of heat transferred for the range of temperature excesses studied. An interface in an oscillatory mode induces greater heat fluxes than a stationary interface at the same superheat.

3. The lip geometry of a given active site is an important factor in determining interaction characteristics between adjacent ebulliating cavities. Active sites with protruding lips are not appreciably influenced by adjacent sites with similar geometric characteristics.

4. A protruding lip at the edge of an active site is a less efficient heat transfer promoter than a similar cavity with a non-protruding lip.

5. Disruption of the thermal boundary layer by a detaching bubble, as proposed by Hsu and Graham, adequately describes the difference observed in this work and the results of other investigators.
SUGGESTIONS FOR FURTHER WORK

The author feels that the following topics could be studied with the existing apparatus or by making minor modifications on the existing apparatus:

1. a statistical study of the nucleation characteristics of various geometrically defined artificial sites,

2. an analysis of the types of energy perturbations capable of initiating nucleation,

3. an investigation of the effect of lip geometry on boiling heat transfer, and

4. a heat transfer experiment similar to the one performed, but with varying thicknesses of the stainless steel heat transfer area.

The last study would yield useful information as to the dependance of the interaction between adjacent ebulliating sites and the solid heating surface temperature distribution.


NOMENCLATURE

**Letters**

E  potential (volts)
H  enthalpy (caloried/gm-mole)
h  bridge thickness (inches)
I  current (amperes)
k  thermal conductivity (Btu/hr-ft^2-°F/ft)
n  number of active sites (pure number)
P  pressure (millimeters of mercury)
Q  heat transferred (Btu/hr)
q  heat flux (Btu/hr-ft^2)
R  gas constant (1.987 cal/°K-gm mole)
r  radius (inches)
T  temperature (°F), T_S - T_L - overall temperature
driving force
W  power (watts)
z  direction normal to heat transfer area

**Subscripts**

B  bubble
C  cavity
CB  copper block
L  lip, liquid
R  radial
S  surface
T  total
NOMENCLATURE (Continued)

$Z$  direction normal to heat transfer area

$1,2,3$  first, second and third radial groove positions on bridge

$\text{Net}$  difference between total heat delivered and heat lost in electrical leads

$a$  current (amperes)

$v$  potential (volts)

$\ast$  overall boiling contribution

Greek letters

$\beta$  angle made by lip radius vector from horizontal to gas-liquid-solid interface (radians)

$\phi_r$  ratio of bubble radius growing from protruding lip to radius of hypothetical bubble growing on flat surface (dimensionless)

$\Delta$  difference operator

$\theta$  contact angle, as measured in the liquid phase (radians)
APPENDIX A - Fixed Data for Runs 1 and 2

1. Thermocouple placement on bridge:
   \[ r_1 : 1.000 \text{ inches} \]
   \[ r_2 : 1.250 \text{ inches} \]
   \[ r_3 : 1.500 \text{ inches} \]

2. Thermal conductivity of stainless steel 304:
   \[ k = 9.4 \text{ BTU/hr-ft}^2\text{-°F/ft} \]

3. Thickness of stainless steel heat transfer area:
   \[ = 0.0300 \text{ inches} \]

4. Thickness of bridge:
   \[ h = 0.0844 \text{ inches} \]

5. Resistance of ammeter lead wires:
   \[ 0.003955 \text{ ohms/ft} \]

6. Length of ammeter lead wires:
   \[ 8 \frac{5}{8} \text{ feet} \]

7. Liquid degassing time and pressure:
   A) Run 1: 5\frac{1}{2} \text{ hours at one atmosphere}
   B) Run 2: 4 \text{ hours at one atmosphere}

8. Surface degassing time and temperature:
   A) Run 1: 7 \text{ hours, temperature raised from ambient to } 330^\circ\text{F}
   B) Run 2: 10 \text{ hours at } 357^\circ\text{F}

9. Ammeter scale used: 5^a
10. Voltmeter scale used: 15\textsuperscript{V}

11. Artificial site geometry:
   A) Run 1: 7 cylindrical holes, 0.006" in diameter and about 0.010" to 0.012" deep (distribution pattern shown on p. B-1).
   B) Run 2: 19 cylindrical holes, 0.006" in diameter and about 0.010" to 0.012" deep (distribution pattern shown on p. B-1).
The Heat Transfer System

FIGURE II
Artificial Site Patterns

Run 1:

Run 2:

Figure 12
## Tabulated Data, Run 1 (1 Atmosphere)

<table>
<thead>
<tr>
<th></th>
<th>( T_0 )</th>
<th>( Q_{ro} )</th>
<th>( Q_z )</th>
<th>( T_s - T_l )</th>
<th>( q_z )</th>
<th>Description</th>
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<td>6.00</td>
<td>9.95</td>
<td>14.31</td>
<td>10.1</td>
<td>655</td>
<td>Natural convection, 4 ebullating sites on gasket</td>
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<td>2.</td>
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<td>25.00</td>
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<td>1145</td>
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</tr>
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<td>3.</td>
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<td>31.00</td>
<td>19.7</td>
<td>1420</td>
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</tr>
<tr>
<td>4.</td>
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<td>28.20</td>
<td>41.30</td>
<td>28.2</td>
<td>1890</td>
<td>Same as 1</td>
</tr>
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<td>5.</td>
<td>20.00</td>
<td>33.20</td>
<td>49.70</td>
<td>37.3</td>
<td>2280</td>
<td>Natural convection, 1 ebullating site on gasket</td>
</tr>
<tr>
<td>6.</td>
<td>17.10</td>
<td>28.40</td>
<td>42.60</td>
<td>32.2</td>
<td>1960</td>
<td>Same as 5</td>
</tr>
<tr>
<td>7.</td>
<td>11.40</td>
<td>18.90</td>
<td>24.55</td>
<td>20.0</td>
<td>1125</td>
<td>Same as 5</td>
</tr>
<tr>
<td>8.</td>
<td>11.70</td>
<td>19.40</td>
<td>24.80</td>
<td>22.0</td>
<td>1140</td>
<td>Natural convection, no extraneous ebullition</td>
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<tr>
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<td>23.40</td>
<td>33.00</td>
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<td>1512</td>
<td>Same as 8</td>
</tr>
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<td>20.20</td>
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<td>55.50</td>
<td>38.2</td>
<td>2545</td>
<td>Same as 8</td>
</tr>
<tr>
<td>11.</td>
<td>17.20</td>
<td>28.60</td>
<td>43.10</td>
<td>32.1</td>
<td>1975</td>
<td>Same as 8</td>
</tr>
<tr>
<td>12.</td>
<td>16.80</td>
<td>27.35</td>
<td>44.75</td>
<td>31.5</td>
<td>2060</td>
<td>Sites A, B, C, D, E, F, G, and stray, 1( \frac{1}{2} ) inches to right of E (8 sites)</td>
</tr>
<tr>
<td>13.</td>
<td>15.70</td>
<td>26.00</td>
<td>39.00</td>
<td>29.1</td>
<td>1790</td>
<td>Sites A, B, C, D, E, F, G, (7 sites)</td>
</tr>
<tr>
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<td>24.90</td>
<td>36.80</td>
<td>28.2</td>
<td>1690</td>
<td>Same as 13</td>
</tr>
<tr>
<td>15.</td>
<td>13.80</td>
<td>22.90</td>
<td>33.10</td>
<td>25.9</td>
<td>1520</td>
<td>Sites A, B, D, E, F, G (6 sites)</td>
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<td>25.90</td>
<td>39.10</td>
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Tabulated Data, Run 1 (1 Atmosphere) (Continued)

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<th>Q₁₀</th>
<th>Tₚ - T₁</th>
<th>q₂</th>
<th>Description</th>
</tr>
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<tr>
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<td>20.15</td>
<td>17.4</td>
<td>925 Sites E, D, F, site D intermittent (2-3 sites)</td>
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<tr>
<td>19.</td>
<td>20.10</td>
<td>33.40</td>
<td>55.20</td>
<td>38.2</td>
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</tr>
<tr>
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<td>46.35</td>
<td>34.0</td>
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<tr>
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<td>39.00</td>
<td>30.1</td>
<td>1790 Same as 19</td>
</tr>
<tr>
<td>22.</td>
<td>16.60</td>
<td>27.60</td>
<td>42.30</td>
<td>30.4</td>
<td>1940 Sites A, B, C, D, E, F, (6 sites)</td>
</tr>
<tr>
<td>23.</td>
<td>12.80</td>
<td>21.20</td>
<td>31.30</td>
<td>24.4</td>
<td>1435 Sites A, D, F, E, G, and one stray ½ inch to right of G (6 sites)</td>
</tr>
</tbody>
</table>

Tabulated Data, Run 1 (1.5 Atmospheres)

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<th>Q₀</th>
<th>Q₁₀</th>
<th>Tₚ - T₁</th>
<th>q₂</th>
<th>Description</th>
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<td>23.90</td>
<td>35.90</td>
<td>26.4</td>
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<td>26.20</td>
<td>42.10</td>
<td>29.5</td>
<td>1930 Same as 24</td>
</tr>
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<td>1560 1 site bubbling</td>
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<td>28.</td>
<td>16.30</td>
<td>27.05</td>
<td>43.75</td>
<td>30.9</td>
<td>2050 Same as 27</td>
</tr>
<tr>
<td>29.</td>
<td>20.50</td>
<td>34.00</td>
<td>53.10</td>
<td>37.3</td>
<td>2430 Natural convection check point</td>
</tr>
<tr>
<td>T₀</td>
<td>Q₀</td>
<td>Q₂</td>
<td>T₈ - T₁</td>
<td>q₂</td>
<td>Description</td>
</tr>
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<td>----------</td>
<td>------</td>
<td>-----------------------------------------------------------------------------</td>
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<tr>
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<td>22.20</td>
<td>33.50</td>
<td>24.8</td>
<td>1535 Sites A, L, B, M, E, N, Q, K, 2 strays 0.05&quot; from Q, 1 stray 0.05&quot; from K, 1 stray 0.05&quot; from B, and 1 stray 0.2&quot; from A (13 sites)</td>
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<td>24.20</td>
<td>40.80</td>
<td>27.4</td>
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<tr>
<td>32</td>
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<td>22.80</td>
<td>34.20</td>
<td>26.1</td>
<td>1570 Sites A, B, K, E, N, G, stray 0.2&quot; from A, stray 0.05&quot; from G (8 sites)</td>
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<tr>
<td>33</td>
<td>16.2</td>
<td>26.90</td>
<td>44.80</td>
<td>29.6</td>
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</tr>
<tr>
<td>34</td>
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<td>22.40</td>
<td>35.00</td>
<td>26.1</td>
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<td>19.30</td>
<td>23.60</td>
<td>22.7</td>
<td>1310 Sites C, A, M, E, G (5 sites)</td>
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<td>12.6</td>
<td>20.90</td>
<td>30.50</td>
<td>24.2</td>
<td>1400 Same as 35</td>
</tr>
<tr>
<td>37</td>
<td>10.4</td>
<td>17.25</td>
<td>26.25</td>
<td>21.2</td>
<td>1210 Same as 35</td>
</tr>
<tr>
<td>38</td>
<td>10.6</td>
<td>17.60</td>
<td>22.70</td>
<td>19.7</td>
<td>1040 Same as 35</td>
</tr>
</tbody>
</table>
APPENDIX C - Discussion of Errors

There were several sources of error involved in the results obtained. These may be classified in two categories:

1. errors in measurement, and
2. errors of computation

The general procedural analysis for each of the above errors along with an illustrative example is given below:

1. Errors in measurement

a. voltage: the voltmeter used was rated to read correctly within 0.5% of full scale deflection.

scale used: 15 volts at full scale
error: 0.075 volts
error in reading instrument: 0.05 volts
(one-half the least count of the instrument)
total error: 0.125 volts
For \( E = 9.10 \) volts, error = \( \pm 1.37\% \)

b. current: the ammeter used was also nominally rated as accurate within 0.5% of full scale deflection.

scale used: 5 amps at full scale
error: 0.025 amps
error in reading instrument: 0.025 amps
total error: 0.05 amps
For \( I = 1.87 \) amps, error = \( \pm 2.69\% \)
c. **pressure:** The greatest possible source of error is that due to erroneous temperature readings.

The Clausius-Clapeyron equation in approximate difference form will be used to determine pressure error:

\[
\frac{\Delta \ln P}{\Delta T} = \frac{\Delta H}{RT^2}
\]

For CCl₄ boiling at 760 millimeters of mercury.

\( \Delta H : \) 7150 calories/mole

\( \Delta T : \) 0.2°C (K°)

\( R : \) 1.987 cal/°K-gm mole

\( T : \) 349.7 °K

therefore,

\( \Delta \ln P = 0.006 \) and

\( P/760 = 1.006, \ P = 765 \)

or 0.7%

d. **bridge temperature:** The thermocouples on the bridge were calibrated in the approximate operating range of the system (steam point), their error was therefore estimated at 0.5°F. However, two thermocouples were placed at each radial position, and their readings averaged, hence the error actually incurred was probably much smaller.
2. **Errors of computation**

The following approximation is valid for small changes of a calculated variable *(1)*:

\[ \Delta M = \sum_{i=1}^{n} \left( \frac{2M}{\partial b_i} \Delta b_i + \sum_{j>i} \frac{2M}{\partial b_j \partial b_i} \Delta b_j \Delta b_i + \cdots + \frac{2M}{\partial b_n} \Delta b_n \right) \]

where \( M \) is the calculated value, \( b_i \) the measured or previously determined calculated values, and the partial derivatives are evaluated from the calculational formula. Equation *(1)* gives the maximum error associated with a calculated quantity.

a. **Power and heat input measurement:**

\[ W = EI \] from *(1)* we have

\[ \Delta W = \frac{2W}{2E} \Delta E + \frac{2W}{2I} \Delta I \]

and,

\[ \frac{2W}{2E} = I, \quad \frac{2W}{2I} = E \] from *(2)* hence

\[ W = (I) (\Delta E) + (E) (\Delta I) \]

or dividing by *(2)*,

\[ \frac{\Delta W}{W} = \frac{\Delta E}{E} + \frac{\Delta I}{I} \]

For the point in question,

\[ \frac{\Delta E}{E} = 0.0137 \]

\[ \frac{\Delta I}{I} = 0.0269 \] therefore,

\[ \frac{\Delta W}{W} = 0.0406 \text{ or } 4.06\% \]

the same percentage error will be valid for \( \Delta Q_T/Q_T \) hence:

\[ Q_T = 57.4 \pm 2.3 \text{ Btu/hr} \]
b. **Error in the radial flux.** The radial flux was determined by a graphical technique, which in essence, was a parabola fit to the three measured temperatures on the bridge (see p. C-6, "Radial Heat Loss"). Three possible errors are encountered utilizing this technique:

1. error in temperature measurement,
2. error in radius measurement, and
3. error in assumed parabolic temperature distribution.

The effect of errors in temperature measurement on the error in the radial heat flux was determined by first evaluating constants A, B, and C such that:

\[ T = Ar^2 + Br + C \]

was satisfied for the three temperature measurements at their corresponding radial positions. Then values of the temperatures were varied to the limit of their experimental uncertainty, and the constants A, B, and C were re-evaluated.

In the preliminary work, only three uncalibrated thermocouples were used. With these thermocouples, the error of a given reading was estimated at 1°F and the corresponding error in heat flux was about 20%. By calibrating the thermocouples, and having two thermocouples at
each radial position, the error in temperature is probably less than 0.5°F and the corresponding heat flux error is no more than 10% and probably of the order of 5%.

Errors in radial groove measurement and hence thermocouple placement are small. The grooves were machined to very close tolerances. Therefore the error in fin loss as affected by misjudging the radii of the grooves on the bridge would be small.

To evaluate the error incurred by an assumed parabolic temperature profile, it was necessary to solve a one-dimensional heat conduction equation. The integrated result was a linear combination of Modified Bessel functions of the first and second kind. The solution of the heat conduction equation had three parameters, namely the heat transfer coefficient from surface to fluid, and two arbitrary integration constants. In principle, the three measured temperatures fix these parameters at some particular value for a given heat input. Since the heat transfer coefficient appeared in the argument of the Bessel function, an iterative procedure was necessary. One
assumes a value for the heat transfer coefficient, calculates the argument, evaluates both Bessel functions at two radii positions, and with the corresponding temperatures at these two radii positions, solves for the two integration constants. The assumed heat transfer coefficient and the two known integration constants provided sufficient information to calculate the third temperature. If the calculated temperature agreed with the measured value, the correct heat transfer coefficient had been chosen, if not, a new value was assumed and the procedure repeated.

The iterative procedure is simple in theory. However Bessel functions are very sensitive to small changes in their arguments, and convergence was almost impossible. For a difference of about $2^\circ \text{F}$ between measured and calculated temperatures, the corresponding heat flux was about $15\%$ lower for analytical solution than the corresponding parabola fit.

The assumed parabola fit introduces a systematic error only, and consequently affects the absolute magnitude of the results obtained. The difference between natural convection and
boiling heat transfer results would remain unaffected.

c. **Longitudinal flux error:**

From a heat balance, one gets:

\[ Q_z = Q_{\text{Net}} - Q_R \]  and from equation (1)

\[ \Delta Q_z = \Delta Q_{\text{Net}} + \Delta Q_R \]

For the point under consideration:

\[ \Delta Q_z = 2.28 + 0.15 \times 22.6 = 5.68 \text{ or } 16.7\% \]

the flux, i.e., \( Q_z/A_{CB} \) will be in error by the same percentage, if the error in area measurement is small, and this was the case.

Note, that this seemingly large error is essentially the sum of random and systematic errors. If only random errors are considered (that is the errors in voltmeter, ammeter and thermocouple readings then the corresponding random error in the longitudinal heat flux is

\[ \Delta Q_z = 2.28 + 0.05 \times 22.6 = 3.42 \text{ or } 10.2\% \]

Where 0.05 represents the fractional error introduced by erroneous thermocouple readings.

d. **Errors in temperature driving force.** \( (T_s - T_L) \)

There are two errors in estimation of the temperature driving force - the error in the estimation of the surface temperature and the error in measuring the bulk liquid temperature.
The surface temperature was estimated from the following formula.

\[ q_x = -k \frac{(T_{cb} - T_s)}{\Delta x} \]

There is a small heat loss in the radial direction on the heat transfer area, but this loss produced negligible changes in the surface temperature estimation.

The temperature driving force error is essentially that due to errors in thermocouple reading. Both thermocouples are quoted by the manufacturer* to read correct with 0.5% of the temperature given in their conversion tables. The error in temperature difference is therefore 6.5%.

Sample Calculations

1. Total heat input:
   \[ Q_T \text{ (Btu/hr)} = 3.413 \text{ (Btu/hr-watt)} \times 13.3 \text{ (Btu/hr-watt)} = 45.3 \text{ Btu/hr} \]

2. Heat loss through ammeter leads:
   With ammeter current reading and knowledge of lead resistance it is possible to obtain \( I^2R \) through the lead wires. After repeated calculations, it was noted that the lead loss amounted to \( 1/2 \) of total power input, hence:
   \[ Q_{\text{Net}} = 0.99 Q_T \]
   \[ Q_{\text{Net}} = 0.99 \times 45.3 = 44.9 \]

3. Radial heat loss:
   The following manual technique was developed so that data points could be obtained as the investigation proceeded:

   The total heat flowing radially around the bridge is given by:
   \[ Q_r = -\kappa \left( \frac{2T}{\Delta r} \right) (2\pi r h) \]
   for any radial position on the bridge, now at \( r = r_1 \) we have:
   \[ Q_r \bigg|_{r_1} = -\kappa (2\pi r_1 h) \left( \frac{2T}{\Delta r} \right) \bigg|_{r=r_1} \]
   approximating the derivative in the following fashion:
   \[ \left( \frac{2T}{\Delta r} \right)_{r_1} = \lim_{\epsilon \to 0} \left( \frac{\Delta T}{\Delta r} \right) \bigg|_{r_1, \epsilon} \]
substituting values for \( r_1, \Delta r, h, k \), we get

\[
\eta_r \bigg|_{r_1} = 1.66 \lim_{\Delta r \to 0} \frac{(\Delta T)_{r_1+\Delta r}}{\Delta r^2}
\]

Six thermocouples were placed on the bridge, two at each radial position. Each pair of readings was first corrected with the calibration data and averaged. The difference between average temperature at \( r_1 \) and average temperature at \( r_2 \) was calculated, and similarly for temperatures between \( r_2 \) and \( r_3 \). The calculated temperature differences were plotted against radial position, and a linear extrapolation to \( r_1 \) yielded the desired value for \( \Delta T \) at the point in question.

<table>
<thead>
<tr>
<th>Thermocouple number</th>
</tr>
</thead>
<tbody>
<tr>
<td>Reading</td>
</tr>
<tr>
<td>Corrected average 1-10:</td>
</tr>
<tr>
<td>3-11:</td>
</tr>
<tr>
<td>2-12:</td>
</tr>
<tr>
<td>Temperature difference: at ( r = 1.125 )</td>
</tr>
<tr>
<td>( r = 1.375 )</td>
</tr>
</tbody>
</table>
Temperature difference extrapolated to $r = 1.000$: $T = 12.0$

Fin loss: $1.66 \times 12.0 = 19.9$ Btu/hr

4. Normal heat transferred:

$$Q_z = Q_{Net} - Q_r$$
$$Q_z = 44.9 - 19.9 = 25.0 \text{ Btu/hr}$$

5. Surface temperature:

A linear extrapolation of the measured copper block temperature, through the thin stainless steel-silver solder joint enabled an estimation of the surface temperature to be made:

$$Q_z = -\frac{\partial T}{\partial z} (A_{cb}) \approx -\kappa (A_{cb})(T_s - T_{cb})/\Delta z$$

or

$$T_s - T_{cb} = -Q_z \Delta z / \kappa A_{cb}$$

substituting the known values of the parameters $\Delta z$, $k$, $A_{CB}$, we get:

$$T_{CB} - T_s = 0.0122 \text{ Q}_z$$

$$T_{CB} - T_s = 0.0122 \times 25.0 = 0.304^\circ F$$

6. Normal Heat Flux:

$$q_z = Q_z/A_{CB} = Q_z/3.14(1.000)^2 \text{ (Btu/hr-in}^2)$$

$$= Q_z/3.14 \text{ (144 in}^2/\text{ft}^2)$$

$$= 45.9 \text{ Q}_z \text{ (Btu/hr-ft}^2)$$

$$q_z = 45.9 \times 25.0 = 1145 \text{ Btu/hr-ft}^2$$