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# FILM BOILING ON VERTICAL SURFACES

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**Abstract** – This paper presents the results of an experimental and theoretical study of heat transfer during film and transition boiling on vertical surfaces. The theoretical analysis is confined to natural convection saturated film boiling, while the experimental work shows the effect of bulk liquid velocity with saturated and with subcooled methanol. The theory is based on a new model of film boiling in which the time dependence of the phenomenon is taken into account. The predicted heat transfer coefficients show good agreement with data from different investigators covering a wide range of experimental conditions.

With the experimental apparatus the liquid velocities which can be achieved are high enough and the subcooling large enough to clearly show the influence of both of these parameters on the heat transfer coefficient. It is found that as either or both subcooling and liquid velocity are increased, the condition of the heat transfer surface becomes progressively more important. Although no quantitative analysis is made for subcooled and/or forced convection film boiling, some qualitative comments on the mechanism of heat transfer in these cases are presented.

## NOMENCLATURE

| | |
|---|---|
| $c$, | wave speed [ft/s]; |
| $C_p$, | specific heat [Btu/slug-deg F]; |
| $g$, | acceleration of gravity [ft/s$^2$]; |
| $h_{fg}$, | latent heat of vaporization [Btu/slug]; |
| $h'$, | modified latent heat, $h_{fg} + C_p\Delta T/2$ [Btu/slug]; |
| $h_e$, | effective latent heat of vaporization (see Appendix) [Btu/slug]; |
| $H$, | heat transfer coefficient [Btu/hft$^2$deg F]; |
| $k$, | thermal conductivity [Btu/hftdeg F]; |
| $L$, | heated surface vertical height [ft]; |
| $Pr$, | Prandtl number [dimensionless]; |
| $q$, | heat flux [Btu/hft$^2$]; |
| $T$, | temperature [deg F]; |
| $\Delta T$, | temperature difference across vapor film [deg F]; |
| $\Delta T_L$, | liquid subcooling [deg F]; |
| $U$, | vapor velocity in radial direction [ft/s]; |
| $W$, | vapor velocity in direction normal to heated surface [ft/s]; |
| $\delta$, | vapor film thickness [ft]; |
| $\kappa$, | thermal diffusivity [ft$^2$/s]; |
| $\lambda$, | wavelength of interfacial waves [ft]; |
| $\mu$, | viscosity [slug/fts]; |
| $\nu$, | kinematic viscosity [ft$^2$/s]; |
| $\rho$, | density [slug/ft$^3$]; |
| $\sigma$, | surface tension [lb/s]. |

Subscripts

| | |
|---|---|
| $L$, | liquid; |
| sat, | saturation; |
| $v$, | vapor; |
| $w$, | wall (heated surface). |

A bar ($^-$) over a quantity denotes an averaged value

## INTRODUCTION

FILM boiling is the regime of boiling in which the bulk liquid is separated from the heating surface by a film of vapor. The temperature difference between the heating surface and the saturation temperature of the liquid being boiled is higher than for nucleate boiling, and for common liquids at pressures near atmospheric is of the order of several hyndred deg F.

The first theoretical analysis of the film boiling phenomenon was by Bromley [1]. He described saturated, natural convection, film boiling on horizontal cylinders in terms of a laminar, steady vapor film flowing between the heating surface

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and the liquid. Since then, other investigators (e.g. [2–8]) have examined both this configuration and that of film boiling on vertical surfaces in greater detail, based on the idea of a laminar, steady vapor film. For pressures that are close to atmospheric (which is the only pressure condition of concern in this study) the heat transfer coefficients predicted by these more sophisticated analyses are quite close to those given by the simple theory of Bromley. Although the analyses based on the steady, laminar model give quite good results when applied to film boiling on horizontal cylinders with "small" diameters, it has been known for some years that they do not apply to cylinders with "large" diameters (more precisely for cylinders where the diameter is significantly greater than $\lambda_c = 2\pi\sqrt{[\sigma/(\rho_L - \rho_v)g]}$, the critical wavelength for Taylor instability) [9]. Experiments on vertical cylinders with heights greater than about 2 in. also yield heat transfer results which are not in accord with the steady theory, but can be higher than predicted by more than a factor of two [10, 11]. In addition, the appearance of the liquid–vapor configuration in saturated natural convection film boiling appears to be quite different from that postulated in other models. The interface is not smooth but is covered with large (compared to the steady-state film thickness) amplitude waves. The existence of these waves is not surprising, as the steady, laminar vapor film has been shown to be unstable under the above-mentioned conditions [11]. The waves can be seen in Fig. 1, which shows natural convection film boiling of saturated methanol on a vertical cylinder at atmospheric pressure.

In view of the discrepancies between the actual phenomenon and the laminar model, several investigators have described the process in terms of a turbulent vapor film [12–14]. These turbulent analyses also predict a significantly lower heat transfer coefficient than the experimentally determined values. In addition they do not describe those film boiling situations where interfacial waves exist but the vapor film Reynolds number is low. For example, the vapor film Reynolds numbers for experiments reported in this paper are below 200, and while the flow is time-dependent, it is almost certain that the vapor film is not turbulent.

The existing experimental investigations of film boiling have, in the main, concentrated on natural convection with a saturated liquid. In particular, there has not been a systematic investigation of the influence of forced convection and bulk liquid subcooling on the heat transfer in film boiling on vertical surfaces. For this reason, the present experimental program emphasized the elucidation of the effect of these two parameters.

## FLUID MECHANICAL MODEL

Since both the laminar and turbulent heat transfer theories proposed to date are inadequate in predicting those situations where the vapor flow path is long enough for the interfacial waves to develop, a new model is proposed taking these waves into account. The model considers the vapor flow to be time dependent, though laminar. Radiation heat transfer is not included since Bromley has shown that the small radiation contribution can be treated as an additive term in an approximate fashion [1]. The analysis is based on incompressible fluid dynamics with constant properties. The vapor properties will be evaluated at a mean temperature $T_m = T_{sat} + \Delta T/2$, a procedure which has been shown to be quantitatively quite good for film boiling [15].

The physical situation assumed to exist is shown in Fig. 2. Based on considerable photographic observation, the interfacial waves are represented as three-dimensional, with the wave number perpendicular to the plane of the paper being the same as that along the heated surface. Therefore, the model presented here is not applicable when the dimension of the heating surface in the direction perpendicular to the paper is much smaller than a wavelength, as would be the case for film boiling on a vertical wire. The wave system is taken to be traveling vertically upward with a wave velocity *c*.

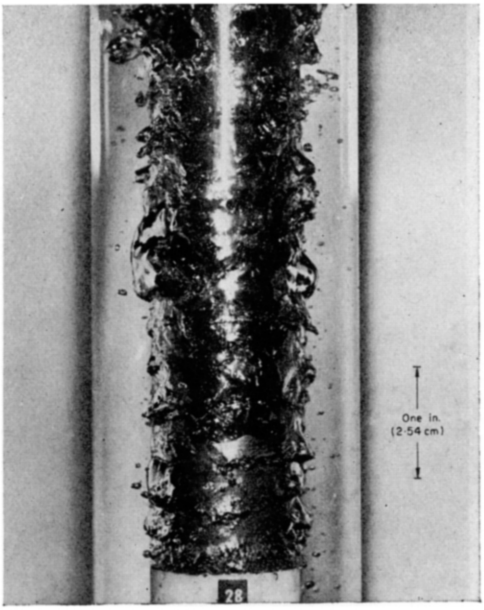

FIG. 1. Natural convection film boiling on a vertical surface: saturated liquid, $\Delta T = 289$ (degrees F).

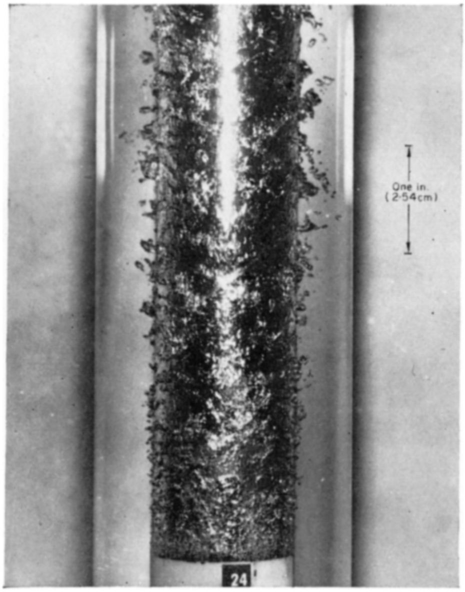

FIG. 15. Forced convection film boiling on a vertical tube: saturated liquid, liquid velocity 23·0 ft/s, $\Delta T$ = 276 (degrees F).

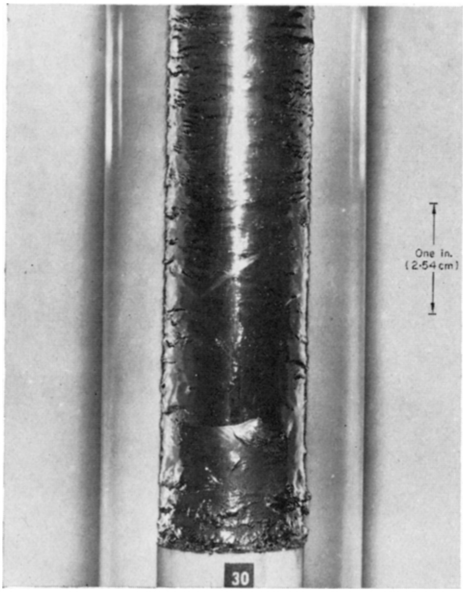

FIG. 16. Natural convection film boiling on a vertical tube: subcooled liquid, $\Delta T_L$ = 27 (degrees F), $\Delta T$ = 287 (degrees F).

Adopting coordinates fixed to the wave system, the phenomenon becomes steady and the liquid far from the heated surface moves downward with a velocity $(-c)$ as shown in Fig. 2. Each wavelength of the liquid vapor interface has a thin and a thick vapor region. Under the assumed interface configuration almost all of the heat will be transferred through the thin vapor portion of each wavelength. The thicker portion acts chiefly as a vapor reservoir or bubble for the vapor generated by the heat transferred in the narrow sections. Justification for the general features of this model was obtained from similitude experiments in a porous-walled water channel [11].

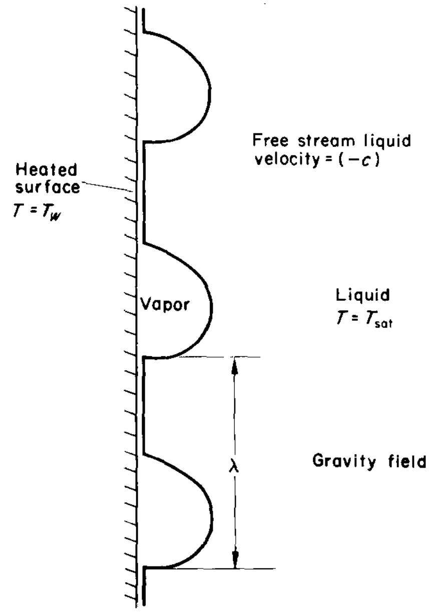

FIG. 2. Model of interfacial waves in saturated natural convection film boiling (not to scale).

The dominant fluid mechanical and heat transfer portion of each wavelength is the thin vapor region. Vapor is created at the interface and flows out radially. Vapor is also forced through the narrow gap by the hydrostatic pressure gradient and by the motion of the interface relative to the heated wall. If we say that the vapor film Reynolds number is low enough so that the vapor inertial forces may be neglected in these narrow regions, then the contributions to the pressure distribution in the vapor film are linear in the velocity. The vapor flow may thus be broken into the three "component" parts mentioned and the significance of each part examined separately. As far as a gross picture of the dynamics of the liquid adjacent to the thin film region is concerned, the only component of the vapor flow that is important is that arising from the radial outflow of the vapor created at the interface. Viscous resistance to this outflow results in a pressure gradient which creates higher pressures at the center of the thin film region than at the edges.

Regardless of what the exact shape of the interface is (plane or curved) a relation between some average film thickness, $\delta$, and the total force exerted on the mass of liquid adjacent to the thin film region can be derived. A force balance in the vapor film yields

$$\frac{\Delta P}{r_0} \sim \frac{\mu_v \bar{U}}{\delta^2}. \tag{1}$$

The length, $r_0$, defined in Fig. 3, is a representative length associated with the size of a thin film region and $\Delta P$ is the pressure difference between the center and the edge of the thin film region caused by the radial outflow of vapor. A

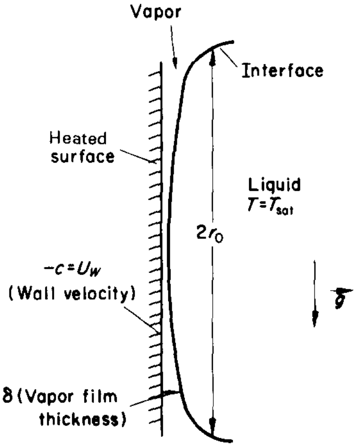

FIG. 3. Sketch of vapor–liquid configuration in regions of significant heat transfer.

mass flow balance between the vapor created at the interface and that flowing out radially, in the thin film yields

$$r_0 \bar{U} \delta \rho_v \sim \frac{k_v \Delta T r_0^2}{\delta h_e}. \qquad (2)$$

The effective latent heat $h_e$, which is defined so as to include the effect of vapor superheating, is discussed in the appendix.

Combining the force and mass flow balances yields an expression for the total force exerted by the vapor on the mass of liquid adjacent to the vapor film,

$$\text{Total force} \sim \Delta P r_0^2 \sim \frac{v_v k_v \Delta T r_0^4}{\delta^4 h_e}. \qquad (3)$$

This force is balanced by the inertial forces in the liquid resulting from streamline curvature. Since photographic observation shows that the bubble dimensions are of order $r_0$, these forces are proportional to $\rho_L c^2 r_0^3 / r_0$ and hence,

$$\Delta P r_0^2 \sim \rho_L c^2 r_0^2. \qquad (4)$$

The pressure in the bubbles can be taken as constant because vapor inertial forces are negligible, and the distances are so large that vapor viscous forces can also be neglected. The bubble dimensions are large enough to allow surface tension forces to be neglected in a gross description of the bubble. Therefore, the hydrostatic pressure increases can only be compensated by static pressure decreases caused by local increases in the velocity. For large single bubbles, this balance enabled Taylor and Davies [16] to predict the terminal rise velocity, although in view of the rotational nature of the flow impinging on any one bubble in the present case, their quantitative conclusions can not be applied here.

Because the liquid Reynolds number associated with the rising bubbles is of the order of $10^4$, the terminal velocity is set by the form drag associated with the total array of bubbles. A balance between this drag and the buoyancy force on the bubble array yields a relationship involving the terminal rise velocity, $c$,

$$r_0^2 \rho_L c^2 \sim (\rho_L - \rho_v) g r_0^3. \qquad (5)$$

The length, $r_0$, is some fraction of the wavelength, $\lambda$. Two reasonable estimates of this fraction can be made by saying either: (1) the heated surface area adjacent to the thin film regions is the same as that adjacent to the bubbles; (2) the diameter of the thin film regions is half a wavelength. The first assumption yields $r_0 = \lambda/\sqrt{2\pi}$, and the second $r_0 = \lambda/4$. For either case one can substitute $\lambda$ for $r_0$ in the mechanistic arguments which are carried out here:

$$\rho_L c^2 \lambda^2 \sim \rho_L (1 - \rho_v/\rho_L) g \lambda^3 \qquad (6a)$$

$$\rho_L (1 - \rho_v/\rho_L) g \lambda^3 \sim \frac{v_v k_v \Delta T \lambda^4}{\delta^4 h_e}. \qquad (6b)$$

A heat transfer coefficient, $\bar{H}$, averaged over time, the total vertical heating surface length, $L$, and a wavelength, $\lambda$, in width can be related to the other variables already defined. The net heat added between top and bottom of the heating surface in a region one wavelength wide is $\bar{H} \Delta T L \lambda$. This must be equal to the heat content of the vapor passing a horizontal plane at the top of the heating surface. Since the volume of vapor which passes this plane is proportional to the volume of a vapor bubble ($\lambda^3$) times the number of vapor bubbles passing per second ($c/\lambda$), the heat balance can be written as,

$$\bar{H} \Delta T L \lambda \sim \lambda^3 (c/\lambda) \rho_v h', \qquad (7)$$

where $h'$ is the latent heat modified to represent an average vapor superheating in the bubbles of $\Delta T/2$. An explicit expression for $\lambda$ can now be obtained from (6a) and (7),

$$\lambda \sim \{\bar{H} \Delta T L / [\rho_v h' g^{\frac{1}{2}} (1 - \rho_v/\rho_L)^{\frac{1}{2}}]\}^{\frac{2}{3}}. \qquad (8)$$

It has been assumed that all the heat is transferred through the thin film sections of each wavelength. Therefore, an explicit expression for $\bar{H}$ can be obtained using equations (8) and (6b),

$$\bar{H} = G\left\{\frac{k_v^2\rho_L h_e g(1-\rho_v/\rho_L)}{\nu_v\Delta T}\right\}^{\frac{5}{14}} \times \left\{\frac{h'\mu_v}{Lh_e\rho_L[k_v g(1-\rho_v/\rho_L)]^{\frac{1}{2}}}\right\}^{\frac{1}{7}} \qquad (9)$$

where $G$ is a non-dimensional constant. To check this theory or correlation, existing data for several liquids with widely varying physical properties has been used. The results are plotted in Fig. 4. If the theory is valid, the ordinate [the dimensionless heat transfer coefficient, $G$, introduced in equation (9)] should be constant. The abscissa in Fig. 4, $C_p\Delta T/Prh'$, is the obvious dimensionless temperature difference. It varies over an order of magnitude showing the widely varying fluid properties and $\Delta T$'s of the data. The length variation for the experimental results that are shown was a factor of two. In utilizing the data of Hsu and Westwater [12] only data in the film boiling regime has been used. In the absence of a more precise criterion, this was taken to mean only data points at temperature differences which were clearly higher than that corresponding to the minimum heat flux. This means that only the argon, nitrogen, and methanol points from [12] were used. It can be seen that the correlation derived above is quite good, with the limits of scatter as good or better than those shown in the paper of Bromley [1]. A least squares fit gives a value of 0·86 for $G$.*

An alternative way of comparing the present theory with experiment is to plot the predicted dimensional heat transfer coefficient vs. the experimentally determined value. This is done in Figs. 5a and 5b, in which the theoretical predictions of the laminar steady theory and Dougall's turbulent theory [14] are also shown. Because of the different heat transfer mechanisms

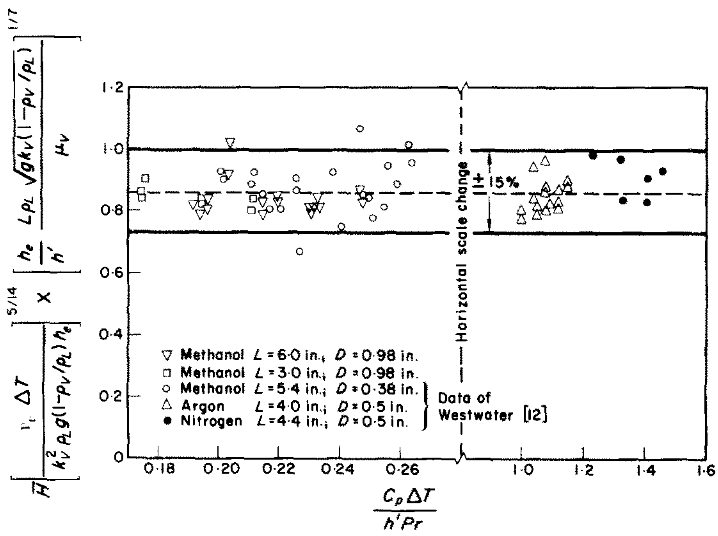

FIG. 4. Film boiling heat-transfer correlation.

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* A reviewer has noted that the quantity

$$\left\{\frac{h'\mu_v}{h_e\rho_L[k_v g(1-\rho_v/\rho_L)]^{\frac{1}{2}}}\right\}^{\frac{1}{7}}$$

does not vary appreciably for all the data shown in Fig. 4 (approximately five per cent). In addition since we are considering pressures near atmospheric, $\rho_v/\rho_L$ can be neglected compared to one. Therefore, for these fluids, at least, if one selects some average value of the above quantity and treats it as a (dimensional) constant, $A$, a simpler form of equation (9) can be written, which may be more convenient to use in engineering calculations. This is

$$\bar{H} = 0{\cdot}86\left\{\frac{k_v^2\rho_L g h_e}{\nu_v\Delta T}\right\}^{\frac{5}{14}}\cdot\left\{\frac{A}{L}\right\}^{\frac{1}{7}}.$$

proposed by the three theories it is necessary to plot the average heat transfer coefficient, $\overline{H}$, as a function of $\Delta T$ for a given fluid and a given heated surface length. Figures 5a and 5b show the results of doing this for two cases which typify the total data included in Fig. 4: natural convection film boiling of saturated methanol on a 6 in. heated surface and saturated argon on a 4 in. heated surface.

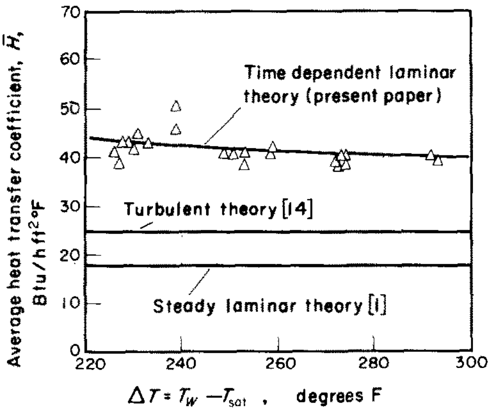

FIG. 5a. Comparison of different boiling theories for saturated methanol on a 6 in. vertical surface (Δ denotes experimental point).

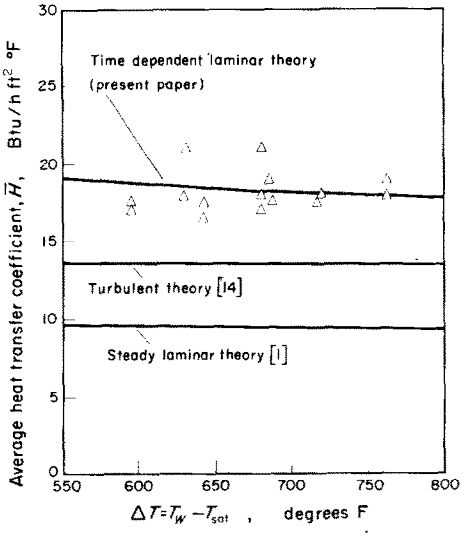

FIG. 5b. Comparison of different boiling theories for saturated argon on a 4 in. vertical surface (Δ denotes experimental point).

In spite of the lack of detailed knowledge of such things as the exact interface configuration and the flow field between the bubbles, it is felt that the simple arguments presented above do contain the essentials of a correct description of the mechanics of heat transfer in non-steady film boiling on vertical surfaces.

## EXPERIMENTAL APPARATUS

The experimental apparatus was designed to permit measurements of the rates of heat transfer in all three regimes of boiling. In transition boiling an increase in surface temperature decreases the heat transferred. Therefore, for stable operation, the surface temperature must be the independent variable rather than the heat flux as is commonly the case in heat transfer experiments. This consideration led to the use of condensing steam as the heating agent, since the pressure and therefore the temperature of saturated steam can be easily and accurately controlled.

The general layout of the system is presented in Fig. 6. The facility consists of an electrically heated water boiler which supplies saturated steam to the inside of an approximately one in. dia. nickel plated Admiralty brass tube, the heated surface for the experiments. The heat transfer liquid (methanol in the present case) flows vertically upwards past the outside of the tube. As shown in Fig. 7, saturated steam enters the testing section from above after passing through a steam separator. It arrives at the inside of the heated surface by passing through a small diameter ($\frac{3}{8}$ in.) brass tube. This tube is insulated from the heat transfer liquid by a teflon bushing of the same outside diameter as the heated surface. The steam which condenses on the inside of the heated surface passes vertically downwards through another small diameter insulated tube into a collection tube inside the steam line. The collecting tube is connected to a sight gage so the rate of condensate can be metered. Care was taken in the mechanical design to insure that only condensate from the inlet and outlet tubes, the heated

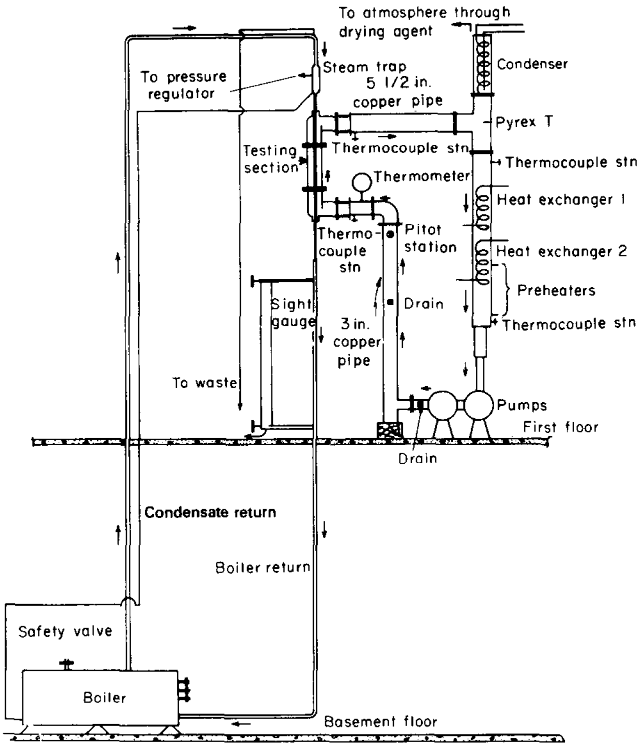

FIG. 6. Flow diagram (not to scale).

surface, and the sight gage line reaches the measuring station. The amount of condensate extraneous to the heat transfer surface can be calibrated and the corrected rate of condensate thus used to measure the heat transfer across the heated surface.

With the teflon bushings covering the $\frac{3}{8}$ in. dia. steam inlet and outlet tubes, the heat transfer fluid in the testing section flows vertically upwards in an annulus of constant dimensions although the "effective flow-through area" changes slightly with distance along the heating surface because of boundary layer growth and the presence of the vapor film. The effect of this relatively small change can be calculated fairly accurately.

The temperature along the inner surface of the annulus varies in the direction of flow. On the teflon it is constant and near the boiling point of the alcohol. It then rises abruptly to a new higher value on the heated metal surface. Because of the excellent insulating properties of teflon, the temperature changes at the ends of the heated surface take place over very thin thermal boundary regions, and the temperature boundary condition along the surface is close to being a step change at the junctions of the teflon bushings and the heated surface.

Since it was expected that under high heat flux conditions there could be a substantial difference between the steam temperature and the temperature of the outside of the heated surface, four small (36 gauge) iron–constantan thermocouples were embedded in each end of the large brass tubes. The holes for these had their centers 0·036 in. from the outer surface of the nickel plated brass tube, and extended axially $\frac{1}{2}$ in. into the wall of the tube, with the

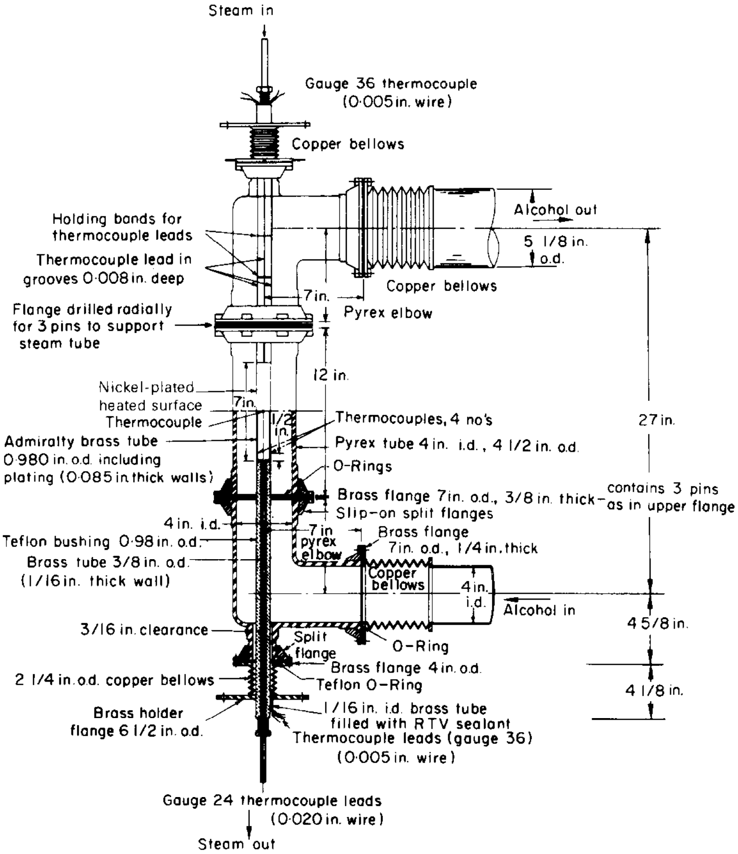

FIG. 7. Sectional elevation of testing section (shown with 4 in. i.d. Pyrex tube and 7 in. heating surface.

thermocouple wires coming out of the holes and running along 0·008 in. grooves cut in the teflon.

The piping and fittings used in the alcohol loop were glass, copper, and brass while the two positive displacement pumps were made of cast iron. The temperature of the test liquid was controlled by means of the strip heaters and heat exchangers shown in Fig. 6. Thermocouples placed in the liquid (also shown in Fig. 6) monitored its temperature. To obtain more accurate readings of the liquid temperature when it was subcooled, especially at low or zero liquid velocity, additional thermocouples were installed within the testing section; one about 2 in. below and the other approximately 8 in. above the top end of the heated surface.

Figure 7 shows the testing section with a 4 in. i.d. glass tube. Three in. and 2 in. i.d. sections, which were used with a nozzle, were also available. The close agreement of the results obtained for free convection with the 2 in. section with those of [12], which were obtained in essentially laterally unbounded volumes of liquid, made it unnecessary to run many trials with the larger diameter sections and the 2 in. i.d. section was used for most of the runs. To facilitate cleaning the heated surface, a brass tube, which could be pushed up into the elbow, was adopted in place of the straight glass tube and used in the majority of runs. Although not shown in Fig. 7, the testing section was provided with turning vanes, which were designed from information in [17], as well as flow straightener in the elbow before the heated section.

In view of the higher heat transfer coefficients associated with the dropwise condensation of steam a "promoter" was introduced into the steam system. A higher heat transfer coefficient was desirable in order to ensure a reasonably isothermal surface. The promoter used was heptyl mercaptan, recommended by Emmons [18].

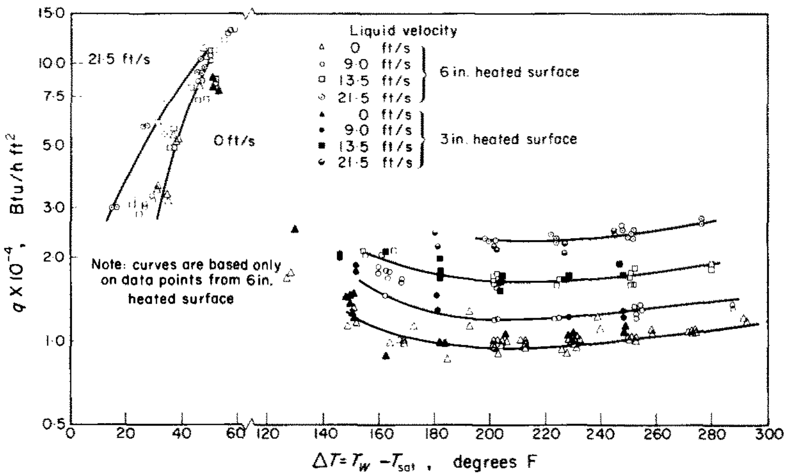

FIG. 8. Boiling heat transfer to saturated methanol on a vertical surface.

The experiments were carried out with the alcohol level approximately one-half inch below the top of the upper horizontal leg of the alcohol loop. Since it was experimentally determined that variations of ± one in. in level made no difference in the heat transfer results, all of the runs can be considered to have been carried out with a constant alcohol level.

## EXPERIMENTAL RESULTS AND DISCUSSION

Before presenting the experimental results, we should briefly consider the question of boiling heat transfer nomenclature. More specifically, an examination of the literature shows that there are divers viewpoints taken concerning the temperature difference which is considered to mark the initiation of film boiling. In addition, for any given liquid the temperature at which film boiling occurs is a strong function of parameters such as liquid subcooling and the liquid velocity. Therefore, solely to try to eliminate ambiguity in the following discussion, we will adopt the simple operational convention that "film boiling" refers to a boiling regime in which the slope of the curve of heat flux ($q$) vs. difference between heated surface and saturation

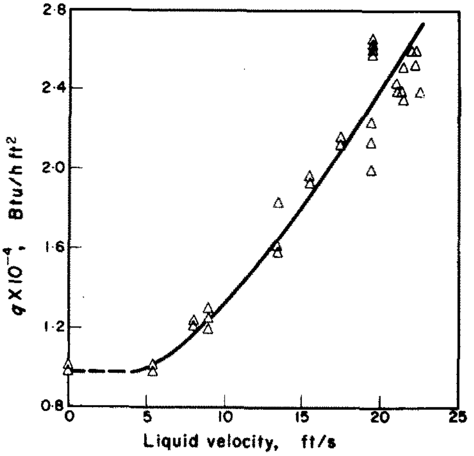

FIG. 9. Effect of liquid velocity on film boiling heat transfer, $T_{steam} - T_{sat}$ = 260 degrees F.

temperature ($\Delta T$) is positive, and the term "transition boiling" refers to a regime where the slope of the $q$ vs. $\Delta T$ curve is negative.

The boiling heat transfer curve for saturated methanol on vertical surfaces is shown in Fig. 8. Heat flux ($q$) is plotted vs. the temperature difference ($\Delta T$) between the average wall temperature and the saturation temperature for different values of the bulk liquid velocity. A more detailed picture of the velocity dependence is shown in Fig. 9, which gives heat flux as a function of liquid velocity at a constant *steam* temperature. Since the temperature drop between steam and wall temperatures is dependent on the heat flux, the wall temperature will change slightly as the velocity is increased. However for the data shown, neglecting this change and assuming constant wall temperature introduces an error in $\Delta T$ of less than one per cent, with the resulting error in the value of $q$ being much smaller because of the flatness of the $q$ vs. $\Delta T$ curve.

In order to obtain reproducible data, it was necessary to clean the heated surface before essentially all of the individual runs. The few exceptions occurred during natural convection, saturated boiling at higher values of $\Delta T$(roughly $\Delta T \geqslant 250$°F) where the rate of change of heat transfer with time (see below) was found to be imperceptible over the times involved in carrying out the measurements. If the cleaning were not done, a brownish coating gradually built up on the surface and the heat transferred increased even though the steam temperature remained constant. The larger the temperature difference and the lower the liquid velocity, the slower was this rate of increase, as shown in Fig. 10. Some alleviation of this problem was accomplished by preheating the alcohol and only introducing it into the system when it was at the desired temperature so as to lessen the total amount of time in contact with the heated surface, but the frequent cleaning was still necessary.

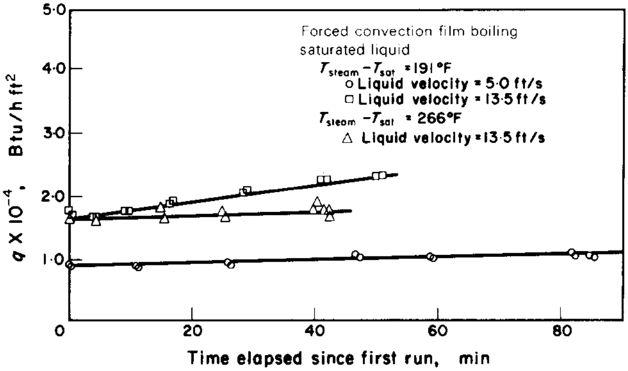

FIG. 10. Time dependence of heat flux in film boiling of methanol on a 6 in. vertical surface.

Only sparse data could be obtained in the transition regime for two reasons. First, the nature of the heat transfer surface becomes of greater importance, and the time over which surface conditions can be considered to be constant reduces to the order of a few minutes. Second, and perhaps more important, is the fact that the slope of the boiling curve $dq/d\Delta T$ is negative, and in fact in places rather steeply negative. The negative slope makes operation of a heating system with $q$ as the independent variable (an electrically heated system, for example) unstable in this region. Although the catastrophic consequences which can result from such an instability cannot occur in the present apparatus, the use of saturated steam as a heating agent does entail certain operating difficulties in the transition regime.

To look at these difficulties in a simple quasi-steady fashion, let us suppose that one wished to obtain data at a value of $\Delta T$ such that $dq/d\Delta T$ is strongly negative, and that by some process the metal heat transfer surface has been brought to an isothermal state. If a small temperature perturbation occurs locally on the surface so that one spot becomes cooler than the rest, the heat transfer from this spot will rise. The spot (and the area around it) will keep cooling off until a point is reached where the rate of heat transfer to the surface (from the condensing steam) is equal to the rate of heat transferred to the boiling liquid and also where the sum of the (positive) slope of heat transfer to the metal surface vs. $T_{\text{steam}} - T_{\text{surface}}$,

$$\frac{dq}{d(T_{\text{steam}} - T_{\text{surface}})},$$

and the slope $dq/d\Delta T$ is equal to or greater than zero. Commonly this means that the eventual surface temperature is equal to or less than the temperature associated with the peak heat flux. On the basis of these arguments, it may be remarked here that this effect should be felt most, and the experimental difficulties consequently greatest, at the place where $(-dq/d\Delta T)$ has a local maximum, at the point of inflection of the boiling curve.

To describe the present experiment accurately, one must emphasize that the wall temperature is not the independent variable; the steam temperature is. In practice, well into the film boiling regime the wall temperatures are quite close to the steam temperature and the relation between wall and steam temperature is relatively constant. For this reason, one *can* think of the wall temperature as being the experimentally varied quantity during film boiling. This concept is not valid in the transition range. One can set the *steam* temperature, but the *wall* temperature must then be measured, as a small change in the former does not necessarily produce a correspondingly small change in the latter.

The wall temperature which occurs at a given steam temperature can be influenced strongly by the heat transfer surface conditions. This factor is quite important in forced convection where the rate at which the surface is contaminated is faster than in natural convection for a given surface temperature. Since the lower part of the heated surface tended to have heavier deposits than the upper part, the heat flux was probably higher on the former. The surface temperature measured by a group of thermocouples in the lower end was lower than that measured by the upper group, supporting this view. A substantial temperature difference could be obtained between the top and bottom with only a moderate liquid velocity. These large differences in wall temperature did not occur in the measurements of nucleate boiling heat transfer where the heat fluxes were as high as those in the transition regime. The difference of the thermal resistance of the condensate at the top or bottom of the inside of the vertical surface of the tube used as the heated surface is not enough in itself to cause serious experimental difficulties in the obtaining of boiling heat transfer data, and the steam heated system could be used even in the high heat flux nucleate regime. The difficulties arise from the nature of the boiling curve in the transition region, which leads to a lack of control over the wall temperature in such a system.

## SUBCOOLED BOILING

When the liquid was subcooled, the influence of the buildup of surface deposits became more pronounced at a given temperature difference and extended over a wider range of temperature differences than in the saturated case. Figure 11 shows the effect of changes in the nature of the surface on the heat transfer for a representative set of experimental conditions.

In view of the influence of surface conditions in subcooled boiling, a test was carried out to see if the liquid used for cleaning affected the heat transfer results. Three solvents, trichloroethane, tetrachloroethane, and tetrachloroethylene, were tried, the last two because of their relatively high boiling points. The heat transfer after cleaning was found to be independent of the solvent used; however, it was strongly dependent on the surface temperature at the time it was cleaned.

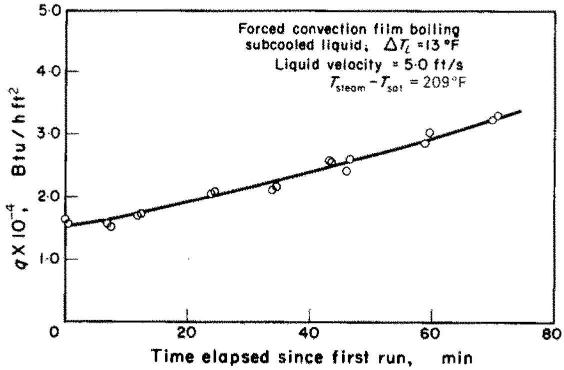

FIG. 11. Time dependence of heat flux in film boiling of methanol on a 6 in. vertical surface.

Cleaning the surface at room temperature, "cold clean", resulted in significantly lower (up to 40 per cent) initial heat transfer rates than if the surface were at temperature when it was cleaned, "hot clean". The "hot clean" values of heat flux were reproducible regardless of the amount of deposits before cleaning.

The velocity range in which film boiling can be encountered is strongly restricted by a small amount of subcooling, and most of the forced convection data had to be taken with low velocities. Although the very low velocities ($U \leqslant 5$ ft/s) had little effect on the heat transfer in saturated boiling (see Fig. 9), they significantly increased the heat flux during subcooling. At higher velocities than shown, the temperature differences between the upper and lower ends of the heated surface became too severe to permit meaningful data on heat flux as a function of wall temperature to be taken.

Figures 12–14 show results for subcooled film and transition boiling. For a subcooling of 13 deg, the only subcooling at which data could be obtained covering any substantial velocity range, the heat flux is shown as a function of velocity. For other cases the data is presented with heat transfer as a function of subcooling, with liquid velocity constant.

There have been very few systematic studies of the regimes of boiling in which the deposit problems are greatest; namely, the transition regime with saturated liquid, and the transition and low film boiling regime with subcooled liquid. Westwater and Santangelo [19] noticed

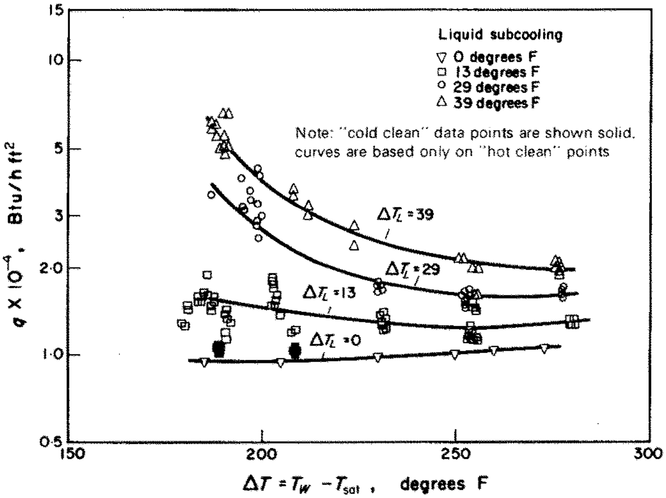

FIG. 12. Natural convection subcooled boiling on a 6 in. vertical surface—effect of liquid subcooling on heat transfer.

the need for aging the heat transfer surface in a standard manner in order to obtain reproducible results, although no detailed variations in heat flux from a given surface as a function of time were reported. The duration of their data taking runs was less than in the present case and the change of the heat transfer with time may not have been significant.

The cause of the deposits is not clear. A semi-quantitative analysis by examination of atomic emission spectra showed that the non-organic content consisted almost entirely of the following metals (listed in order of decreasing abundance): tin, copper, lead, zinc, bismuth and antimony. Tin makes up roughly 10 per cent of the deposits, copper from 5 to 10, lead somewhat less and the last three approximately one per cent each. Other metals (e.g. silver and nickel) are present, but only in trace amounts, with the remainder being organic.

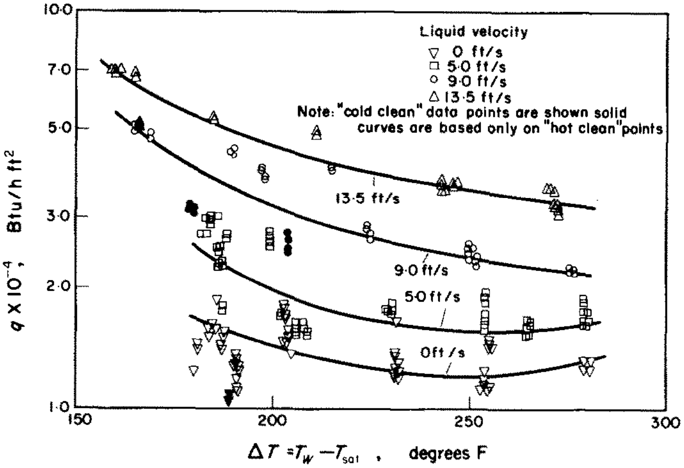

FIG. 13. Forced convection subcooled boiling on a 6 in. vertical surface—effect of liquid velocity on heat transfer. $\Delta T_L = 13$ degrees F.

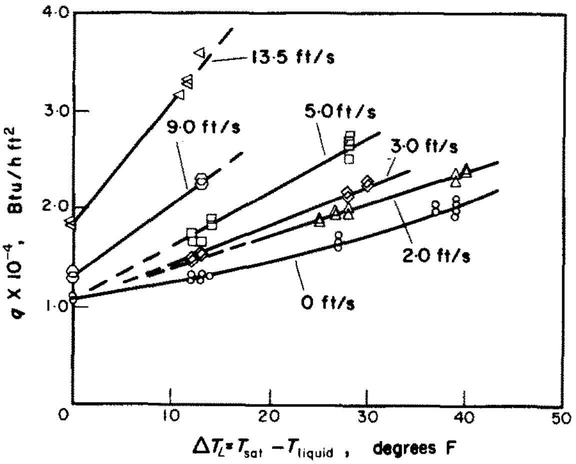

FIG. 14. Effect of liquid velocity and subcooling on boiling heat transfer. $\Delta T \simeq 275$ degrees F.

Two mechanisms can be proposed for the manner in which the heat transfer is enhanced by the presence of the deposits. The most obvious is that the surface becomes progressively rougher as the deposit becomes thicker, and has a larger rms roughness height. If the roughness elements become sufficiently large that they are comparable with the vapor film thickness, heat transfer can be increased either through a locally thinned film or even through actual liquid solid contact. The observations of Berenson [20] on saturated pool boiling are also pertinent. He found that artificially roughened surfaces tended to have higher values of the minimum heat flux than did mirror finish surfaces, although the value of $\Delta T$ was the same in both cases, even though the roughness elements were smaller than the film thickness. (The deposits which form, in the present experiment, probably also function to increase the amount of useable nucleation sites at a particular $\Delta T$ compared to the clean heated surface which is mirror-like.) Lumping the comparison of mirror-like vs. other surfaces into "smooth" and "rough", one can say generally that in the film boiling region near the minimum heat flux the rough surfaces tend to result in higher heat transfer than the smooth.

An alternative explanation can be advanced in terms of the surface conditions with regard to whether or not the heated surface is wettable by the heat transfer fluid. Far enough into the film boiling regime, the presence of a wetting agent appears to make no difference (see Bromley, [1]) since the liquid does not contact the surface. In the low transition regime, Berenson discovered that the addition of a wetting agent to the heat transfer fluid could result in an order of magnitude increase in the heat transferred. Berenson also presents data which shows the effect of surface cleanliness. On two successive runs, using the same surface finish, the value of the minimum heat flux changed by a factor of two and its position changed from a $\Delta T$ of about 100°F to roughly double that. This effect was caused by the surface not being cleaned immediately before running and becoming more wettable by exposure to the air. Room temperature tests on the nickel heat transfer surface used in the present experiment showed that methanol partially wetted the cleaned surface, but did wet those portions of the surface which had deposits on them. To a first approximation the contact angle between methanol and the cleaned nickel surface was independent of the cleaning agent. These simple observations suggest that the wettability explanation may be the more correct description; however, the true situation may be some combination of both of these causes. The structure of the surface deposits could not be measured. Various crude attempts using a micrometer placed the maximum deposit thickness at much less than one thousandth of an inch. This very small thickness of the deposit would also tend to suggest that the wettability of the surface is more important in increasing the heat transfer than is the obstruction of the vapor film by the roughness elements. The exact nature of the effect of the deposits on the heat transfer, though, must be considered as yet unresolved.

## QUALITATIVE REMARKS ON THE MECHANISM OF FILM BOILING HEAT TRANSFER

The ideas which were developed for saturated free convection film boiling earlier in this paper can be extended, though only in a qualitative way, to the cases of forced convection film boiling of saturated and subcooled liquids. Figures 1 and 15 illustrate the important differences between natural and forced convection film boiling of methanol at the saturation temperature. In the forced convection situation the rapidly moving bulk liquid tends to rip away the large masses of vapor which were present during natural convection. As a consequence, there are essentially no large areas over which significant heat transfer does not take place.

There are at present several theories of steady laminar forced convection film boiling on plane surfaces. The discrepancy between these theories and the experimental measurements is an order of magnitude and it must be assumed the true situation is not laminar and steady. Instead of the viscous drag force exerted by the liquid on the vapor, it is probable that the pressure drag associated with the interface waves is the prime force moving the vapor. In addition to the above effects of liquid velocity, the dependence of the heat flux at a given $\Delta T$ on the presence of deposits on the surface increases as the velocity increases. This may indicate that the film is thinner than in the natural convection case, implying that the liquid inertial forces (which are responsible for the motion of the liquid towards the heated surface) are greater than those in natural convection, or in other words, that they push on the vapor film more strongly.

While the basic mechanism of heat transfer in saturated forced convection film boiling may be similar to that in natural convection, i.e.

mostly quasi-steady conduction through a thin vapor film, even the fundamental fluid mechanics which determines such things as the thickness of this film is at present unclear.

The heat transfer situation for natural convection film boiling for subcooled methanol appears (see Fig. 16) to be fluid mechanically the simplest situation of them all. The large amplitude interfacial waves which occurred when the liquid was saturated have been suppressed because of the liquid subcooling, although the interface is still covered with smaller amplitude waves. Calculations of the heat transfer for a subcooled liquid case, based on a steady laminar model, predict a heat transfer which is too low by roughly 50 per cent [11]. This suggests that although the liquid inertial forces associated with the interfacial wave motion may be smaller than in the saturated case, they cannot be neglected in an attempt to describe the phenomenon correctly.

With the combination of high velocity and large subcooling, extremely high heat transfer rates have been reported in the film boiling region [21]. This interesting phenomenon could not be studied in any detail with the present apparatus because of the exceedingly rapid surface contamination and the strong dependence of the heat flux on the surface conditions. Some data points were recorded to show the order of magnitude of the heat fluxes which could be expected, and these were several times as high as the peak heat flux observed during saturated boiling. These high heat transfer rates may indicate that the dip in heat flux which occurs in the saturated boiling curve between the peak heat flux and the region in which radiation is important is not very deep in the subcooled forced convection case. In fact the curve may be rather flat to the right of the nucleate peak heat flux or even continue to rise.

## SUMMARY

A time dependent heat transfer model of film boiling appears to be adequate to describe natural convection of saturated and subcooled liquids. Forced convection on the other hand appears to introduce new facets into the problem which will require other explanations. It further appears that the nature of the heat transfer surface can become of considerable importance as the liquid subcooling and/or velocity are increased.

## ACKNOWLEDGEMENT

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## APPENDIX

### *The Effective Latent Heat*

It is well known that not all the heat which is transferred from the heating surface during saturated film boiling is used to vaporize the liquid. Some is utilized to superheat vapor which is already present in the film. Since the simple model presented above assumes that the mass flux into the vapor film is proportional to $\Delta T$, the effect of superheating can be "built in" by replacing the actual latent heat of vaporization, $h_{fg}$, by an effective heat, $h_e$, which has been defined to take this superheating into account. Methods of formulating an explicit expression for $h_e$ have been given by several authors, for example [22] and [23]. The value used in the present paper is based on a modification of the procedure used in [22]. For this calculation, the thin film region of each wavelength is modeled by a circular area of radius $r_0$ with the interface planar and parallel to the heating surface. The geometry is shown in Fig. 17. The film thickness is taken constant and equal to $d$. With these assumptions, it can be argued (see [11]) that the relative velocity of the heating surface does not affect the heat transfer, and that for the purpose of a heat transfer calculation one need only consider the radial velocity distribution, which is axisymmetric outflow given by the solution of the creeping flow equations. This known velocity distribution can be used in the energy equation to find the temperature distribution in the vapor film. The energy equation is

$$W\frac{\mathrm{d}T}{\mathrm{d}z} = \kappa_v\frac{\mathrm{d}^2T}{\mathrm{d}z^2} \qquad \text{(A.1)}$$

with boundary conditions $T(0) = \mathrm{T}_w$ and $T(d) = T_{\mathrm{sat}}$. The vapor velocity in the $z$ direction, $W$, can be found from the known radial velocity, $(U)$, by integration of the equation of continuity,

$$\frac{1}{r}\frac{\partial}{\partial r}(rU) + \frac{\partial W}{\partial z} = 0 \qquad \text{(A.2)}$$

Once the form of $w$ is known, the energy equation may be rewritten as

$$W/\kappa_v = \frac{\mathrm{d}^2T}{\mathrm{d}z^2}\,\frac{\mathrm{d}T}{\mathrm{d}z} = \frac{\mathrm{d}}{\mathrm{d}z}\ln\frac{\mathrm{d}T}{\mathrm{d}z} \qquad \text{(A.3)}$$

which can be integrated to find $T(z)$. What is pertinent is the calculated heat flux at the heating surface, since the expression for $h_e$ is derived from a comparison of this quantity with the heat flux one would calculate if no superheating were considered. The computations involved in making this comparison are straightforward and details are given in [11]. It is found that although there is no simple relation between $h_e$ and $h_{fg}$, the actual latent heat, an approximate expression for $h_e$ which is within two per cent of the computed value for $0 \leqslant C_p\Delta T/h_{fg} \leqslant 6$ is

$$h_e/h_{fg} = 1 + 0{\cdot}98\,C_p\Delta T/h_{fg} + 0{\cdot}045\,(C_p\Delta T/h_{fg})^2. \qquad \text{(A.4)}$$

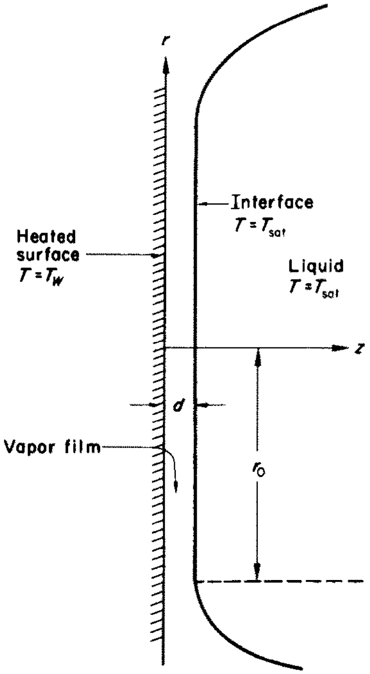

FIG. 17. Sketch of vapor–liquid configuration used in calculation of $h_e$ (not to scale).

As a final note, it may be remarked that, as the quantity $h_e$ appears in the film boiling heat transfer coefficient to the $\frac{3}{14}$ power, over the range of physical properties covered by the data, the maximum difference between the heat transfer coefficients computed using the above expression or those found from the methods given in the other references which are cited is only several per cent.

## EBULLITION EN FILM SUR DES SURFACES VERTICALES

**Résumé**—Cet article présente les résultats d'une étude expérimentale et théorique du transfert thermique lors de l'ébullition en film et de l'ébullition de transition sur des surfaces verticales. L'étude théorique est limitée au cas de l'ébullition en film saturé avec convection naturelle, tandis que le travail expérimental montre l'effet de la vitesse de la masse liquide pour du méthanol saturé et pour du méthanol sous-refroidi. La théorie est basée sur un nouveau modèle d'ébullition en film dans lequel on tient compte de la dépendance du phénomène vis-à-vis du temps. Les coefficients de transfert thermique calculés sont en bon accord avec les résultats obtenus par différents chercheurs, résultats qui couvrent un large domaine de conditions expérimentales.

Avec le montage expérimental, la vitesse du liquide et le sous-refroidissement qui peuvent être réalisés sont assez importants pour montrer clairement l'influence de ces deux paramètres sur le coefficient de transfert thermique. On montre que lorsque le sous-refroidissement ou la vitesse du liquide augmente, le transfert thermique pariétal devient progressivement plus important. Bien qu'aucune analyse quantitative n'ait été faite sur l'ébullition en film sous-refroidi et à convection forcée, on présente quelques commentaires qualitatifs sur le mécanisme du transfert thermique dans ces cas.

## FILMSIEDEN AN SENKRECHTEN FLÄCHEN

**Zusammenfassung**—Die Arbeit bringt die Ergebnisse einer experimentellen und theoretischen Untersuchung des Wärmeübergangs beim Filmsieden und Übergangs-Sieden an senkrechten Oberflächen. Die theoretische Analyse ist beschränkt auf gesättigtes Filmsieden bei natürlicher Konvektion, während der experimentelle Teil den Einfluss der Flüssigkeitsgeschwindigkeit im Behälter bei gesättigtem und unterkühltem Methanol zeigt.

Die Theorie basiert auf einem neuen Filmsiedemodell, in dem die Zeitabhängigkeit berücksichtigt wird. Die vorherbestimmten Wärmeübergangszahlen zeigten gute Übereinstimmung mit Werten verschiedener Autoren über einen weiten Bereich experimenteller Bedingungen. Die mit der Apparatur erreichbaren Flüssigkeitsgeschwindigkeiten sind hoch genug und die Unterkühlung gross genug, um klar den Einfluss dieser beiden Parameter auf die Wärmeübergangszahl zu zeigen. Es stellte sich heraus, dass mit Anwachsen des Unterkühlungsgrades oder der Flüssigkeitsgeschwindigkeit oder beider Grössen die Heizflächenbeschaffenheit in zunehmendem Masse eine Rolle spielt. Obwohl keine quantitative Analyse über unterkühltes Filmsieden und/oder bei erzwungener Konvektion gemacht wurde, sind einige qualitative Aussagen über den Wärmeübergangsmechanismus in diesen Fällen gegeben.

## ПЛЕНОЧНОЕ КИПЕНИЕ НА ВЕРТИКАЛЬНЫХ ПОВЕРХНОСТЯХ

**Аннотация**—В данной статье представлены результаты экспериментального и теоретического исследования переноса массы при пленочном и переходном кипении на вертикальных поверхностях. Теоретический анализ ограничивается рассмотрением насыщенного пленочного кипения с естественной конвекцией, в то время как экспериментальная работа показывает влияние жидкости с насыщенным и недогретым метанолом. Теория основывается на новой модели пленочного кипения, в которой учитывается временная зависимость явления. Рассчитанные коэффициенты переноса тепла показывают хорошее согласие с данными различных исследователей, полученными в широком диапазоне экспериментальных условий.

В экспериментальном аппарате скорости жидкости, которых можно достичь, достаточно высоки, а недогрев довольно велик для того, чтобы ясно продемонстрировать влияние обоих этих параметров на коэффициент переноса тепла. Найдено, что при возрастании недогрева или скорости жидкости, а также обоих факторов вместе, условия на теплообменной поверхности приобретают все большую важность. Хотя и не выполнен количественный анализ для пленочного кипения при недогреве и / или пленочного кипения при вынужденной конвекции, сделаны некоторые качественные замечания по механизму переноса тепла в этих случаях.