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by Y. Y. Hsu
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TECHNICAL PAPER proposed for presentation
at Cryogenic Engineering Conference
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NATIONAL AERONAUTICS AND SPACE ADMINISTRATION

A REVIEW ON FILM BOILING

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ABSTRACT

This report summarizes the major contributions and states of the art for film boiling. The topics covered are (1) boiling of drops and puddles, (2) pool boiling for various geometries, and (3) forced convective film boiling inside channels and tubes. It also points out areas where more study are needed.

I. INTRODUCTION

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Among the various modes of heat transport to liquids, film boiling is considered to be an inefficient mechanism. However, in many practical engineering applications it occurs. Consequently, film boiling must be studied and understood for design applications. Film boiling is invariably encountered in quenching of metals, in chilling of biological species, in regenerative cooling of rockets, and in cooling down a cryogenic fuel tank, and sometimes film boiling can also happen in the nuclear reactor or in the cryomagnet.

This paper will discuss what we know about film boiling and what future work would be productive or of interest. The author will not attempt to cite every article on film boiling that appears in the literature. Instead, the part "What We Know About Film Boiling" is really meant to set the stage for "What We Would Like to Know."

The discussion will be divided into three parts: film boiling of the unconstrained liquid mass (Leidenfrost phenomenon); the pool film boiling; and forced convective film boiling inside a channel or tube.

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II. FILM BOILING OF UNCONSTRAINED LIQUID MASS - "LEIDENFROST PHENOMENON"

In this area, we refer to the film boiling of droplets, flattened drops, or liquid patches on a hot surface. The heat transfer rate is usually determined by recording the evaporation time of the drop in film boiling. Work along this line is reported by Goldleski and Bell (ref. 1), Gottfried and Bell (ref. 2), Patel and Bell (ref. 3), Watcher, Bonne, and Van Nouchuis (ref. 4), Baumeister, Hamill, and Schoessow (ref. 5), and Bradfield (ref. 6). Recently, Bell (ref. 8) made a survey on this subject. The important topics in this area are the evaporation time, the Leidenfrost temperature (for definition, see next section), and the effect of relative velocity between the drop and surface on heat transfer.

1. Evaporation Time. A typical evaporation curve is shown in Fig. 1. To predict the evaporation rate, most researchers (e.g., ref. 5) postulated that the vapor formed at the interface flows laminarly along the bottom side of the drop through the thin vapor gap. The pressure gradient required to flow the vapor also causes the drop to be lifted above the plate. Due to the pressure distribution, the bottom of the drop should be curved upward in the center. However, the film boiling analysis is greatly simplified if the bottom of the drop is assumed to be flat. The drop bottom can be assumed to be rigid or circulating and the vapor flow boundary condition will be affected accordingly. Wachter (ref. 4) considered four cases covering various combinations of bottom shapes and flow boundary conditions. He found better agreement with the experimental data when the analytical model was based upon a flat bottom assumption. It would have seemed more realistic to use the curved bottom geometry but for some reason the prediction was poorer than for the flat bottom. It was speculated that oscillations of the drop bot-

tom may render an effective average surface which is flat. Baumeister, Hamill, and Schoessow (ref. 5) analyzed the Leidenfrost problem by approximating the drop with a flat cylindrical body. They found that the relation between the volume of liquid drop and its thickness can be expressed by a simple power function in three segments as shown in Fig. 2. In this way, the mathematics was greatly simplified to enable a closed form solution. They were also able to form dimensionless groupings correlating heat transfer rates with the liquid masses of various sizes, as shown in Fig. 3 (see table I also). In the analysis in Ref. 5 the subcooling effect and the vapor diffusion to the atmosphere were neglected. Experimental results of Ref. 4 show that higher heat transfer rates exist in a dry atmosphere than in a saturated one. Such a result cannot be explained by considering only the bottom of the drop; the top and edges of the drop must also contribute to the higher rates. Therefore, for drops in a dry atmosphere some modification of the model postulated in Ref. 5 is needed. Bradfield (ref. 6) and Cumo, Ferello, and Ferrari (ref. 7) showed experimentally that intermittent liquid-solid contacts exist across the vapor gap. The contact probability diminishes with increasing ΔT. All these were not taken into account in the model proposed by Baumeister, Hamill, and Schoessow (ref. 5). However, these effects may be important only in the low ΔT range, and their importance decreases with increasing ΔT.

The theory of Baumeister and Hamill (ref. 5) was recently tested by Keshock and Bell (ref. 8) against experimental results with nitrogen. They found that Baumeister and Hamill's equation predicted the evaporation time well for small drops, but overpredicted the time for large liquid masses. They proposed that the breakthrough of the vapor-domes should be considered for film boiling models with large liquid masses.

2. Leidenfrost Temperature. The so-called Leidenfrost temperature is

the temperature at which the transition from the film boiling mode to the nucleate boiling mode takes place as shown in Fig. 1. Bell (ref. 8) surveyed various contributions on this subject and showed that the data were very scattered. He even mentioned that results obtained by different investigators on the same experimental apparatus can be significantly different. It appears that some of the human factors involved are the technique of placing the drop on the plate and the judgment about the termination of Leidenfrost state. The first source of error can only be reduced by placing the drop on the plate as gently as possible. The second error can be eliminated by relying on continuous temperature recording instead of visual judgment. There are other factors that can cause error which are related to the test system:

(a) Solid-liquid contacts - Bradfield (ref. 6) reported intermittent solid-liquid contact during stable film boiling and noted that the contact increases with the roughness of the solid, wettability of the surface and the porosity of the solid. Presumably, this solid-liquid contact precedes the Leidenfrost transition. Intuitively, it seems to the writer that, holding temperature constant, the area fraction of solid-liquid contact ought to increase with increasing mass. The change of contact area with drop size should cause a change in Leidenfrost temperature. Yet, according to an unpublished work of Baumeister, Henry, and Simon (proposed NASA publication) ethanol drops of 0.0125 cm^3 and 6 cm^3 showed very little difference in Leidenfrost temperature. Roughness of surface also affect the solid-liquid contact, as reported in Ref. 7.

(b) Thermal properties of the solid - Many of the Leidenfrost experiments were carried out by placing a drop onto a preheated plate at an initial temperature T_{wo} and then observing whether transition takes place or not. If the transition takes place, the initial temperature is reported as the

Leidenfrost temperature. Since most of the plates were not equipped to maintain uniform and constant temperature, the initial temperature is certainly higher than the temperature at the location and time that Leidenfrost transition takes place. Figure 4 shows the temperature record of stainless steel and aluminum in determining the termination of Leidenfrost state. It is clear that aluminum, with greater thermal diffusivity, approaches closer to the isothermal condition than stainless steel. Certainly those Leidenfrost data involving quenching of low diffusivity materials must be examined as to how the surface temperature was determined. Apparently Cu, Brass, and Al are close to the isothermal case, stainless steel is marginal, and glass will introduce serious error if the initial temperature instead of the actual transient temperature is used.

(c) Coating or scale - It was observed by Hoffman (ref. 9) that oxide scale can promote the transition from Leidenfrost condition to nucleate boiling. A similar effect was observed by Bradfield (ref. 6). Maddox and Frederking (ref. 10) and Allen (ref. 11) all reported earlier transition to nucleate boiling during cooldown when the tubes were coated with Teflon, mylar, ZnO_2, etc. Recent studies of Baumeister, Henry, and Simon showed that the Leidenfrost transition temperature for a freshly polished and cleaned surface is much lower than those for the subsequent drops on the same location. They attributed this upward shift in the Leidenfrost temperature to the formation of oxide.

(d) Wettability - Burge (ref. 12) studied the behavior of liquid metal sprays introduced on a hot molybdenum-titanium surface. He noted that the wetting sodium or lithium drops would flatten out and eventually spread over the surface, but nonwetting mercury drops would flatten and recoil from the surface.

(e) Metastable condition - In the absence of vibration, disturbance, roughness, etc., a Leidenfrost drop may persist even if the temperature of the surface is cooled down well below the common range of the Leidenfrost temperature. This phenomenon was observed by Baumeister, Hendricks, and Hamill (ref. 13). The existence of metastable mode was also observed in Ref. 4 and it was hypothesized (ref. 4) that a drop could persist in the Leidenfrost mode as long as the surface temperature is higher than the dew point of the environment.

3. Velocity Effect on Drops. The effect of velocity is different depending on whether the velocity vector is parallel or perpendicular to the surface. The effect of parallel velocity is to increase the heat transfer rate. An empirical correlation was proposed in Refs. 14 and 15 to account for the enhanced heat transfer. The velocity range studied was from 0 to 15 ft/sec (46 cm/sec). The effect of impinging velocity normal to the surface is more drastic than the translational velocity along the wall. Wachter and Westerling (ref. 16) showed that, for a saturated liquid, when the impinging velocity was high (Weber number larger than 30), the drop would deform and flatten out and vibrate; for very high velocity (Weber number larger than 80), the drop would disintegrate. Burge (ref. 12) experimented with a high velocity spray impinging on a hot plate. He reported that there was flattening of the drop at center region and that boiling took place at the edge of the flattened disc. He hypothesized that the impact pressure caused the liquid to be highly subcooled at the center of the drop. However, at the edge of the drop, the pressure was low enough to permit boiling.

4. Discussion. In the area of film boiling of drops and puddles, there are several subjects worthy of further study.

(a) The vapor velocity at the liquid-vapor interface is still an unknown.

The true boundary condition at the liquid-vapor interface is somewhere between slip and nonslip condition, depending on the presence or absence of impurities and also upon the surface properties of the fluid. In general, circulation can be observed if the liquid is clean, then the boundary condition probably tends toward the slip condition. However, more study of the surface phenomena is needed to reach a definite conclusion about the boundary condition.

(b) The effect of mass diffusion from the drop to the atmosphere should be considered and incorporated into any general correlation.

(c) The relation between vapor gap thickness surface roughness and probability of solid-liquid contact should be quantitatively determined.

(d) The problem of quenching with a high velocity impinging jet is an interesting subject requiring more study. Practically speaking, this is a very important industrial application of film boiling. Academically, it is always an intriguing question as to the state of the fluid near the wall and the mode of heat transfer involved when a highly subcooled liquid makes contact with a hot solid of several thousand degrees in temperature. On impingement, the instantaneous liquid pressure may become supercritical before reverting to its initial subcritical pressure.

III. POOL BOILING

Pool film boiling has been surveyed previously by Brentari and Smith (ref. 17) and by Clark (ref. 18). In the present study, we will mention the more recent development and discuss the areas where more work is needed. The survey will be presented basically according to the geometry of the heater; namely, vertical surface, horizontal surface, horizontal cylinder, sphere, and then a special section on boiling helium.

1. Vertical Surfaces. Early work on film boiling on a vertical surface

was done by Bromley (ref. 19) who followed the same approach used by Nusselt for film condensation. The vapor formed by evaporation was assumed to flow laminarly upward under the influence of a buoyance force. The interface was assumed to be smooth, with vapor velocity at the interface either zero or maximum (no-slip or slip boundary condition). The heat transfer coefficient equation was shown to be (ref. 20)

$$h = \begin{matrix} 0.5 \\ 0.732 \end{matrix} \left[\frac{K_v^3 \rho_v (\rho_l - \rho_v) g \lambda^*}{\mu_v L \, \Delta T} \right]^{1/4} \tag{1}$$

where the two coefficients, 0.5 and 0.732, are for the no-slip and slip boundary conditions, respectively. Equation (1) apparently predicts film boiling for a short, lower portion of the surface fairly well. Many papers pursued the same laminar flow approach but with more refinements to take into account convection effects and subcooling effects. Notable ones are those by Koh (ref. 21), Sparrow and Cess (ref. 22), Tachibana (ref. 23), Nishikawa (ref. 24), McFadden and Grosh (ref. 25), etc. All these refinements, however, only brought a limited improvement over the original crude approach, and are restricted to the idealized case of laminar flow with a smooth interface. Hsu and Westwater (refs. 26 and 27) found that Bromley's equation underpredicted experimental data when the height of the heating surface was larger than 1 in. (2 to 3 cm). They noted the presence of a wave profile at the interface which invalidated the basic assumptions of laminar flow in a smooth channel. With the onset of a wavy interface, the laminar velocity profile can no longer hold. They proposed a model assuming the onset of a turbulent film instability at a critical Reynolds number. The vapor flow is assumed to have a laminar sublayer and a turbulent core. The approximate model was later modified and extended by Dougall and

Rohsenow (ref. 28) to include the resistance at the side of the liquid-vapor interface. Dougall considered the region near the vapor-liquid interface as a turbulent layer, or as a buffer layer with a turbulent layer. The resulting predicted heat transfer coefficient is 100 percent higher than Bromley's but 50 percent lower than that by Hsu and Westwater. Morgan (ref. 29) treated the turbulent layer by integrating assumed velocity and temperature profiles. Recently, Greitzer (ref. 30) decided to examine experimentally the flow pattern in wavy channel. He built a water channel with the wall-contours modelled after the wave profile at the liquid-vapor interface observed by Hsu (ref. 27). He then observed the flow pattern in the channel using dye tracer at a proper Reynolds number. He found that in the crest of the vapor wave, a big eddy was present while laminar flow was observed in the thin layer adjacent to the wall with the thickness about the same as that of the film thickness at the wave valley. The presence of wave pockets apparently took most of the vapor flow away from the regular laminar flow channel. The net result is a thinner thermal resistance layer near the wall. Thus it appears that the waves on the interface should be an important factor in analyzing film boiling and attention is beginning to be paid to this factor. Simon and Simoneau (refs. 31 and 32) studied film boiling at constant heat flux. (Most of the previous analyses had been for constant wall temperature.) They took many high-speed motion pictures showing the detailed structure of the wavy liquid-vapor interface and measured the maximum and minimum thickness of the vapor film. They also showed that due to the presence of a wavy interface the time-averaged thermal resistance is smaller than a value based upon mean film thickness. The effective film resistance appears to be associated with the minimum film thickness of the wavy pattern (fig. 5). A similar hypothesis has been advanced by Coury (ref. 33).

It should also be mentioned that Coury (ref. 33) observed a significant amount of fluctuation in thermocouple reading indicating liquid and vapor contact. The frequency was in 100 to 1000 cps range. This information indicates that when the wave amplitude is high, it might cause occasional contact of the liquid with the heating surface.

2. Film Boiling from Horizontal Surfaces. The problem of film boiling from horizontal surface was first studied by Berenson (ref. 34) who assumed that Taylor instability causes breakup of waves in a cell pattern with the distance between bubbles being equivalent to the most dangerous wavelength. The vapor being generated is considered to flow laminarly into the vapor domes which are spaced at a distance equivalent to the most dangerous wave length λ_d associated with Taylor's instability. The resulting equation of Berenson's is similar to Eq. (1) with the wavelength λ_d being used as the length parameter, and with the coefficient being changed. A similar equation was obtained later by Baumeister and Hamill (ref. 35) who showed by maximization principle that λ_d is the optimal spacing between domes for the maximum heat transfer coefficient (i.e., $\partial h/\partial L = 0$ at $L = \lambda_d$). Ruckenstein (ref. 36) also analyzed the problem of film boiling on a horizontal surface. He used Taylor's instability to determine the bubble spacing and used bubble departure criterion to determine vapor removal rate. In all the above analyses, the vapor flow was assumed to be laminar, and the vapor domes were assumed to be regularly arranged. The alternative assumptions would be turbulent flow of vapor and irregular distribution of domes. Frederking (ref. 37) made an interesting comparison of the four combinations of the various alternative hypothesis, as shown in table I.

Frederking, Wu, and Clement (ref. 37) compared these correlations against the data of water, freon, CCl_4 and N-pentane as well as their data of He-1, a fluid with very low σ/ρ and therefore a very low L, and found that the $(Ra)^{1/3}$ correlation represents the experimental results better. He further concluded that since the length parameter L cancels out in the $(Ra)^{1/3}$ correlation thus the model of irregular cell distribution represents the true picture better. The experimental measurement of cell distribution by Hosler and Westwater (ref. 40) tended to show that the distribution is indeed irregular (fig. 6) and that the minimum inter-bubble distance is close to the most dangerous wavelength. The correlation of hydrocarbon data by Sciance et al. (ref. 41) and by Colver and Brown (ref. 42) show that the Nu is proportional to $Ra^{0.267}$ which is between the Berensen's (ref. 34) and Chang's (ref. 38) model. All this shows that perhaps the Laplace length $L = \sqrt{\sigma/[g(\rho_l - \rho_v)]}$ offers a basis for model postulation. But due to the nature of film boiling mechanism, the parameter L is only a weak controlling factor in film boiling correlation for horizontal surface. It should be added that although helium data of Ref. 37 appear to follow $(Ra)^{1/3}$ trend, it was difficult to say whether water, CCl_4, freon, etc. follow the $(Ra)^{1/3}$ or $(Ra)^{1/4}$ trend. Since hydrocarbons follow the $(Ra)^{0.267}$ trend, it is likely that the exponent of Ra may vary with the Ra or with fluid properties. (More will be discussed in section 3.)

The heat transfer rate for film boiling horizontal surface is solely dependent upon the rate of removal of vapor. Since removal rate from above is not very easily controlled, attempts have been made to remove vapor through porous heat surface (refs. 43 to 45). The heat-transfer rate can then be improved by increasing the suction rate. Early attempts found that the vapor film could be unstable when the suction rate was too high and tongues of liquid would enter the porous media. The problem was solved later by covering the porous heating surface with another thin porous heet, such as asbestos or ceramic sheet which apparently prevented liquid from touching the heater.

All the above work was mainly for saturated fluid and neglecting radiation contribution. An interesting analysis was proposed by Hamill and Baumeister (ref. 46) who considered the effect of subcooling and radiation on film boiling from a flat plate. They concluded that for very large subcooling, film boiling cannot be sustained. It would be interesting to test their hypothesis experimentally.

3. Film Boiling from a Horizontal Cylinder. Bromley studied this problem in the early 1950's and the result was the famous Bromley equation (ref. 20)

$$h = 0.62 \left[\frac{K_v^3 \rho_v (\rho_l - \rho_v) g \lambda^*}{\mu_v D \, \Delta T}\right]^{1/4} \tag{2}$$

But this equation was derived for laminar flow with the buoyancy force balanced by the viscous force. It was later found by Breen and Westwater (ref. 47) that Bromley's equation failed to be valid for very small tubes where surface force becomes important and for large tubes where turbulence might set in. They proposed a correlation in which the length ratio of (L/D)† was plotted against the ratio

$$\frac{h}{\left[\dfrac{K_v^3 \rho_v (\rho_l - \rho_v) g \lambda^*}{\mu_v \, \Delta T D}\right]^{1/4}}$$

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†The ratio $L/D = \sqrt{\sigma \Big/ \left[g(\rho_l - \rho_v) D^2\right]}$ can be considered as $(Bo)^{-1/2}$ where Bo is the Bond number. Bond number represents the ratio between the body force to the surface force.

The correlation showed that when the length ratio approaches zero (i. e., the case of large cylinder), the heat transfer reduces down to Berensen's equation for horizontal surface and that when the length ratio is very large (i. e., the case of small wires) the h-ratio is proportional to the L/D. Later Baumeister, Hamill (ref. 48) derived an equation with similar functional groupings. The equation was based upon the assumption that vapor flows axially into the periodically-spaced vapor domes. This assumption is a departure from Bromley's circumferential-flow model. The comparison of Baumeister and Hamill's equation with experimental data of most organic fluids and water were good. However, recent nitrogen film boiling data are separated from the rest of the points as shown in Fig. 7 (ref. 49). Park, Colver, and Sliepevich (ref. 50) also noted that the film boiling data for nitrogen was segregated from the other fluids. The reason for this discrepancy is unknown, and may be worthy of investigation.

All the correlations for the horizontal cylinder are based upon the assumption of laminar flow in a smooth vapor gap. Such an assumption might be correct for the lower half of the cylinder but certainly do not reflect the true situation at the upper half, where flow pattern is wavy and chaotic, especially for larger tubes. Yet the existing correlations were able to make fairly close predictions of the overall heat transfer rate. This moderate success might be due either to fortuitous coincidence, or to some other unknown reason. However, since the error range is about ±25 percent, further refinement in modelling may not necessarily bring much improvement.

There are some other interesting studies in the film boiling from horizontal cylinders. Bromley, LeRoy, and Robbers (ref. 51) studied the effect of cross flow and found that the velocity effect is negligible when $(u/\sqrt{gD}) < 2$ but became dominant for $(u/\sqrt{gD}) > 2$. The effect of pulsating pressure on film boiling of CCl_3F was studied by DiCicco and Schoenhals (ref. 52) for GP = 90 psi. They found that when the frequency is low the heat flux was higher than the average value of heat flux without a pulse and the heat flux at peak pressure (i. e., $h_{exp} > 1/2\,(h_{no\,pulse} + h_{at\,peak\,press.})$. For high frequency the heat flux with pulse is even higher than that to be expected for a steady pressure field at peak value where subcooling is very high. The transient behavior of a wire, when given a sudden temperature rise to achieve film boiling mode, was analyzed and measured by Pitts, Yen, and Jackson (ref. 53). The experimental result indicates that the temperature took a finite time to reach the pre-set value presumably due to solid-liquid contact.

One curious omission is the study of film boiling on a bank of tubes. This is certainly a practical problem for heat exchanger or cooling design. One can well imagine the important effect of the vapor wake behind a tube in determining the spacing of the heating tubes. The spacing between adjacent tubes should affect the heat transfer too.

4. Film Boiling from a Sphere. Data for film boiling from a sphere is mostly obtained by quenching of a submerged sphere. Experimental data for water, nitrogen, and freon have been reported by Bradfield (ref. 54), Merte and Clark (ref. 55), Frederking, Chapman, and Wang (ref. 56), Frederking and Clark (ref. 57), Hendricks and Baumeister (ref. 58), etc. Frederking and Clark (ref. 57) derived an expression similar to Bromley's analysis with the coefficient of 0.586 (see Eq. (5)). But they found that the experimental data of nitrogen for spheres and other geometries could be all correlated by an equation

$$Nu = 0.14\left(Ra \cdot \frac{\lambda}{cp\ \Delta T}\right)^{1/3} \tag{3}$$

However, their correlation was based upon data obtained from limited range of diameters. Hendricks and Baumeister (ref. 58) observed that the above correlation did not predict the film boiling data for small spheres. Instead they found that the equation relating Nu with both Ra and Bo numbers derived by Hendricks and Baumeister (ref. 59) was able to correlate those data. However, although their equation predicts overall heat transfer coefficient, the same criticism leveled at the analyses for film boiling from cylinders can be made for the case of spheres; namely, the vapor flow conditions on the upper half of the sphere are not steady laminar, as assumed in the model. It was shown in Ref. 59 that laminar and pseudo-laminar flow exist in the lower half of the sphere but became turbulent in the top part for the spheres of diameter around 1 cm submerged in nitrogen. More turbulent region is expected if the spheres are larger.

The discrepancy between the overall macroscopic result and the microscopic measurement of film boiling on sphere was also reported by Frederking, Chapman, and Wong (ref. 56) for nitrogen. They measured local vapor film thickness fluctuated with time (Fig. 8). The peak value was much higher than that calculated from the film boiling equation. However, the overall average heat transfer rate could be successfully predicted by the same equation. It is not known whether such inconsistency is peculiar to nitrogen or to all the normal fluids. The discrepancy certainly warrants further investigation.

When a hot sphere is placed on the free surface of a liquid to setup film boiling, the sphere may be kept floating by the combination effects of the buoyancy force and a surface force even if the sphere density exceeds the liquid. Such interesting phenomenon was studied by Hendricks, Baumeister, and Ohm (refs. 60 and 61) and shown dramatically in a movie. Another puzzling thing is that floating spheres tend to attract each other. The questions to ask are: what is the effect of the curvature of the hot body on flotation; and what is the range of influence of each sphere?

The result of studies of forced cooling of a hot sphere with highly subcooled liquid are very puzzling. Bradfield (ref. 59) measured the heat flux as a function of wall superheat with various liquid subcooling by quenching copper spheres with subcooled water and he found a very strong subcooling effect. Witte (ref. 62) derived a film boiling equation for a sphere with imposed velocity U_∞. His equation was similar to Sparrow's (ref. 22) analysis of film boiling with forced convection along a plate, except that he used spherical geometry. But his equation apparently underpredicted his own experimental data for quenching tantalum (ref. 63) spheres by subcooled sodium. Considering a 1/2 inch (1.25 cm) tantalum sphere, at 3500° F (2200 K) moving through 572° F (574 K) sodium at 10 ft/sec (304.5 cm/sec), his analysis would result in a film thickness of 10^{-6} in. (2.5×10^{-6} cm), which is smaller than the surface roughness. If the experimental heat transfer rate were used, the film would be even thinner. Therefore, he concluded that there must be liquid-solid contact. Comparison with Sideman's equation (ref. 64) for non-boiling forced convective cooling of a solid sphere with a liquid show that the prediction is about 1.7 times the experimental result, if ΔT is assumed to be the wall temperature minus the liquid temperature $T_w - T_1$. Later Jacobson and Shair (ref. 65) measured the heat transfer rate of a steel sphere in flowing water (at 12.2° to 50.6° C) under steady conditions. Their results are similar to the findings of Witte et al. (ref. 63) (i.e., liquid-solid single phase forced convective cooling prevails). However, their visual observation indicated the presence of a vapor film, which is contradictory to the previous conclusion. Therefore, it seems that the underprediction by Witte's film boiling equation may either indicate the strange behavior of liquid metals or indicate that a new analysis is needed, instead of just discounting the existence of a vapor film. In Witte's (ref. 62) analysis, he neglected all the heat transfer contribution by the surface downstream of the separation locus, which was assumed at an angle of $\pi/2$ from the frontal stagnation point. The wake region may well have a much higher heat transfer rate than that in the frontal region.

5. Film Boiling of Helium. Since He - II is a superfluid, the behavior of He is somewhat different from normal fluids in many respects, including their peculiar characteristics in film boiling. Much research has been conducted in this area, notably those listed as Refs. 66 to 71. In Frederking's survey paper (ref. 63) a section was devoted to film boiling of He.

Basically, when a heating surface in a He - II bath is heated up to the Λ-temperature, a film of He - I is developed separating the superfluid He - II from the heating surface. He - I is lighter than He - II and rises in the film to form a plume above the heating surface. If the pressure is below the Λ-point, the film boiling occurs separating He - II from the surface. The unique features of film boiling of He - II are:

(a) Strong depth effect - The heat transfer rate is strongly dependent upon the depth of immersion of the heater, the heat flux is higher with increasing of depth. (Goodling and Irey (ref. 67); Irey, McFadden, and Madsen (ref. 68); Lemieax and Leonard (ref. 69).) It was hypothesized by Rivers and McFadden (ref. 70) that at a depth He - II is in an effective subcooling condition. This is due to the uniformity of the bath temperature of He - II which is equal to the saturation temperature at the free surface. Since the local saturation temperature increases with depth due to hydrostatic head, the subcooling increases with the depth. A detailed study of depth effect is needed.

(b) Noise in film boiling - Coulter, Leonard, and Pike (ref. 71) reported that film boiling can be accompanied with noise or without it. For the case without noise, very few bubbles were observed. More bubbles were observed when noise was present but no thermal current was observed in the He - II.

(c) The He - II film boiling heat transfer rates are much higher (by an order of magnitude) than that predicted for a normal fluid using an equation such as Breen-Westwater's, except when ΔT is large so that He - I is formed (ref. 72). He - I shows the same heat transfer characteristics as a normal fluid.

6. Discussion. It appears that for pool film boiling of a normal fluid, the existing equations were able to predict overall heat transfer rate fairly well. However, the models on which all these equations were based apparently cannot quite describe the physical picture, including the pattern in the upper half of the curved surfaces, the film thickness, etc.

The forced convective cooling of a highly-superheated surface by a subcooled liquid is still quite intriguing. Much needs to be resolved to arrive at a reasonable understanding. These include the frequency of solid-liquid contact, the time required to transit into film boiling, the character of a very highly superheat liquid, etc.

He - II remains as an enigma. Quantitative analysis is still lacking, including the depth effect.

IV. FILM BOILING IN A CHANNEL

1. Flow Pattern.

(a) Film boiling in a channel can have one of two configurations.

(i) The heating surface is separated from the core by a vapor film, with the core made up of a liquid interspersed with gas phase. This is the case when the void fraction is low. This can be called flow film boiling.

(ii) The core is made up of vapor phase with liquid droplets dispersed in the vapor matrix. This is the case where the void is high. It can be called dry-wall mist flow.

(b) The situation also varies with the condition of the heating surface. If the heating surface is of constant heat flux type, the film boiling required may be in coexistence with the wet-wall regime (where nucleate boiling or evaporation exist). Rise or fall of the heat flux would shift the intersection zone between the dry-wall and wet-wall regions up or down stream. If the heat flux is high enough so that wall temperature is higher than the Leidenfrost temperature, the wet wall zone may be pushed out of the heating section. This is usually the case for cryogenic fluids. The other type is the constant temperature case where the wall temperature is held constant by means of heating with a hot fluid or using thick heating surface of large thermal diffusivity. If the wall temperature is set above the Leidenfrost temperature, there is no wet-wall regime. Typical temperature or heat flux profiles of these two types are shown in Fig. 9. Most experimental studies on this subject were carried out with an electrically-heated tube, and thus were the constant heat flux type. Examples are given in Refs. 73 (Hendricks, Graham, Hsu, and Friedman); 75 (Lewis, Goodykoontz, and Kline); 76 (Ward); 77 (G. E.). For the constant wall temperature experiment, an example is the experiment done by Rankin (ref. 78).

2. Correlations. Empirical correlations in this regime include that proposed by Hendricks, Graham, Hsu, and Friedman (ref. 73) to treat the vapor film annular flow in the same fashion as Martinelli did with liquid film annular flow. The parameters χ_{tt} and Nu/Nu_{calc} were used to correlate the data. The term Nu_{calc} was calculated from Dittas-Boelter's equation using properties based upon a dispersed flow model. Thus it is a

parameter more suitable for dry-wall mist flow regime. The parameter χ_{tt}, on the other hand, is derived from an annular flow concept. In the high quality region, Nu_{calc} approaches that of gas correlation and thus is independent of χ_{tt}. In the middle quality range, the correlation worked fine. In the low quality range, the scattering was large. This scattering is either due to the entrance effect or due to the invalidity of Nu_{calc} for annular region. It was difficult to discern which caused the more serious trouble. Other correlations involving modifying gas-phase convention equations were proposed by G. E. (ref. 77) and Westinghouse (ref. 76).

3. Analysis. Most of the analyses for forced convection film boiling were aimed at the dry mist regime. Only Dougall's (ref. 28) analysis was aimed at the flow film boiling regime. He hypothesized the existence of thermal resistance at the liquid-vapor interface, which was contributed by a laminar zone, a buffer zone, and a turbulent zone. As the film Reynolds number was increased, the various zones of resistance were gradually removed. In dispersed flow none of them remained.

For the dry-wall mist flow regime, Hsu, Cowgill, and Hendricks (ref. 79) attempted to represent the fluid properties by using synthesized properties from mixtures of liquid and vapor weighted according to the local void fraction. The two-phase flow problem is thus reduced to a single-phase variable property problem. However, their analysis met only moderate success due to the lack of information as to the distribution profile of droplets and the behavior of those droplets and turbulence under accelerating conditions.

Two principal problems about film boiling in a channel are the effect of acceleration (the increase of volumetric flow rate due to the increase of void fraction) and the effect of thermal nonequilibrium. The effect of acceleration on turbulence boundary layers caused by thermal expansion is still a field virtually untouched and may prove to be very fertile. The acceleration of drops and the nonequilibrium aspect of the problem have been partially dealt with by a series of MIT reports (Laverty, ref. 80; Forslund, ref. 81; Hynek, ref. 82) and by Bennett, Hewitt, Kearsey, and Keeys (ref. 83), etc. By measuring the corresponding wall temperature for a given heat flux, Laverty (ref. 80) was able to calculate the vapor temperature and showed that the vapor in the core was highly superheated with drops at saturation temperature dispersed in it. From the temperature profile of vapor in the core, the heat available for evaporation can be determined. Then the size of the drops can be estimated by considering the total drop surface area available for heat transfer. Laverty's estimation of drop sizes was later verified by Forslund's experimental data (ref. 81). Forslund also developed a program by which the evaporation rate and the acceleration rate of drops can be determined. This program was later modified and improved by Hynek (ref. 82). In their program the wall is considered to be cooled both by the vapor phase forced convection and by the impingement of liquid drops. The drop size was subject to a critical Weber number $We^* < 7.5$. The resulting wall temperature profile for various fluids was found to be fairly close to the prediction value, provided that the values of empirical parameters (K_1, K_2) were varied from fluid to fluid. These parameters were included to account for the fraction of drops hitting the wall (K_1) and to account for the deceleration experienced by the drop upon impinging (K_2). Both parameters are unknown. Apparently, while some progress was made in estimating drop acceleration and drop breakup, much study is needed to determine the drop depositon rate and to determine impinging deceleration.

Bennett, Hewitt, Kearsey, and Keeys (ref. 83) took an approach similar to that taken by Forslund and Hynek with a few variations. One variation is to neglect the contribution of the liquid drops in cooling the wall. Another variation is to determine the saturation temperature of the drop by considering simultaneously the conduction of heat to the drop and the diffusion of vapor from the drop. Their calculated wall temperature was found to be very close to the experimental value. They also considered two extreme conditions: either the drops assume thermodynamic equilibrium or they are in nonequilibrium state without evaporation taking place (labelled "nonevaporative"). It was shown that the experimental result approached the equilibrium case at high mass velocity and approached the nonevaporation situation at low mass velocity (Fig. 11).

The body-force effect on film boiling in a channel has been reported by Papell (ref. 84). Wall temperature distributions along the test section are presented for 1, 2, and 3 g's. It was found that increase of g-load shift the wet-wall zone downstream. Film boiling in horizontal tubes was studied by Kruger (ref. 85). Pressure and heat transfer of He - I in a helically coiled tube were studied by de La Harpe, Lehongre, Mollard, and Johannes (ref. 86). The entrance effect to film boiling was studied by Papell and Brown (ref. 87). They found a great improvement in heat transfer coefficients over that of the fully developed flow.

Discussion

Although much progress has been made in studying convective film boiling in a channel, we still need the following information:

(1) Two-dimensional profiles of drop distribution

(2) Drop size distribution

(3) Behavior of an accelerating boundary layer under thermal expansion

(4) Kinematic relation among velocity profile, void profile, and slip

ratio similar to those developed by Zuber and Findley (ref. 88) for bubbly flow

Another important area is to determine the transition between the regime of flow film boiling with liquid core and the regime of dry-wall mist flow. The models postulated for these two regimes must be different and appropriate for the particular regime, and then the application limit of each regime must be established.

Conclusion

It appears that for Leidenfrost boiling and pool film boiling, we are now able to predict the overall heat transfer coefficient for various geometries with moderate success. A few notable exceptions are the strange behavior of nitrogen and helium. However, all the models giving overall correlations still fail to describe closely the details of the physical phenomena involved.

For two-phase flow film boiling there are still some uncertainties in predicting the overall heat transfer coefficient. More study is needed to understand the effect flow acceleration, drop distribution, drop impinging, etc.

SYMBOLS

A	area
C	constants
cp	specific heat of constant pressure
D	diameter
G	mass velocity
g	gravitational acceleration
h	heat transfer coefficient
K	thermal conductivity
L	Laplace length
l	drop thickness
Nu	Nusselt number
P	pressure
P_c	critical pressure
Pr	Prandtl number
Q	heating rate
q	heat flux
Ra	Raleigh number
T	temperature
ΔT	temperature difference
t	time
u	velocity
V	volume
X	distance from leading edge
δ	vapor gap thickness
ϵ	ratio of vapor gap thickness at valley to the mean thickness
λ	latent heat

λ^* modified latent heat to include sensible heat effect

λ_d dangerous wavelength in Taylor's instability

ρ density

σ surface tension

μ viscosity

χ_{tt} Martinelli parameter

Subscripts:

calc calculated

exp experimental

l liquid

sat saturation

v vapor

w wall

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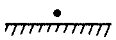
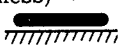
TABLE I. - SUMMARY TABLE FOR LEIDENFROST DROP RESULTS FROM REFERENCE 5

[Effect of vapor density on drop shape and drop buoyancy has been included in this table. It was not included in reference 5.]

Dimensionless volume range, $V^* = \frac{V}{L^3}$	Drop shape	Dimensionless average drop thickness, $l^* = \frac{l}{L}$	Dimensionless area, $A^* = \frac{A}{L^2}$	Dimensionless vapor gap thickness beneath drop, $\delta^* = \frac{\delta}{\left(\frac{k\,\Delta T\,\mu L}{\lambda^+(\rho_L - \rho_V)\rho_V g}\right)^{1/4}}$	Dimensionless heat transfer coefficient, $h^* = \frac{h}{\left(\frac{k^3\lambda^*(\rho_L - \rho_V)\rho_V g}{\Delta T\,\mu L}\right)^{1/4}}$	Dimensionless vaporization time $t^* = \frac{t}{\lambda\rho_L f\left(\frac{\mu L^5}{k^3\lambda^* g(\rho_L - \rho_V)\rho_V\,\Delta T^3}\right)^{1/4}}$
$V^* \leq 0.8$	Small spheroid 	$l^* = 0.83V^{*1/3}$	$A^* = 1.81V^{*2/3}$	$\delta^* = 0.91V^{*1/12}$	$h^* = 1.1V^{*-1/12}$	$t^* = 1.21V^{*5/12}$
$0.8 < V^* \leq 155$	Large drop 	$l^* = 0.8V^{*1/6}$	$A^* = 1.25V^{*5/6}$	$\delta^* = 0.93V^{*1/6}$	$h^* = 1.075V^{*-1/6}$	$t^* = 2.23V^{*1/3} - 0.97$
$V^* > 155$	Extended drop (constant thickness) 	$l^* = 1.85$	$A^* = 0.54V^*$	$\delta^* = 0.61V^{*1/4}$	$h^* = 1.64V^{*-1/4}$	$t^* = 4.52V^{*1/4} - 5$

where

$$L = \left(\frac{\sigma g_c}{(\rho_L - \rho_V)g}\right)^{1/2}$$

TABLE II

	Laminar Flow where $V \propto \frac{g(\rho - \rho_v)\delta^2}{\mu}$	Turbulent Flow where $V^2 \propto \frac{g(\rho_l - \rho_v)L}{P_v}$
Regular Cell Distribution where $L = \sqrt{\frac{\sigma}{(\rho_l - \rho_v)g}}$ is a parameter.	$Nu = C_1\left(Ra\,\frac{\lambda^*}{cp\,\Delta T}\right)^{1/4}$ (2) Berenson, Ref. 34	$Nu = C_2\left[Ra \cdot Pr\left(\frac{\lambda^*}{cp\,\Delta T}\right)^2\right]^{1/3}$ (3) Kistemaker, Ref. 39
Irregular Cell Distribution where cell-spacing distance varies over wide range. In this case L should be cancelled out in the equation. Since L is no longer uniquely determining the cell spacing.	$Nu = C_3\left(Ra\,\frac{\lambda^*}{cp\,\Delta T}\right)^{1/4}$ (4) Chang, Ref. 38	$Nu = Cu\left[Ra \cdot Pr\left(\frac{\lambda^*}{cp\,\Delta T}\right)^2\right]^{1/3}$ (5) Frederking, Ref. 37

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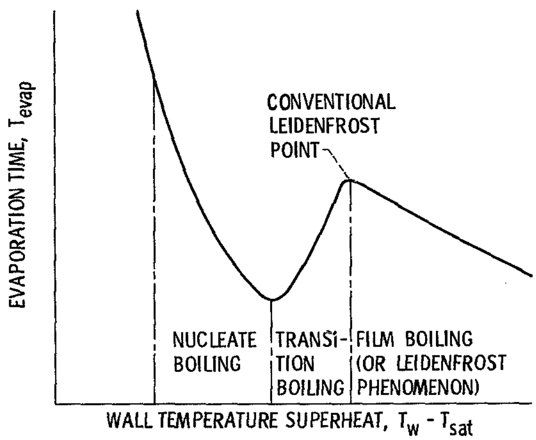


Figure 1. - Typical evaporation time curve for a liquid drop.

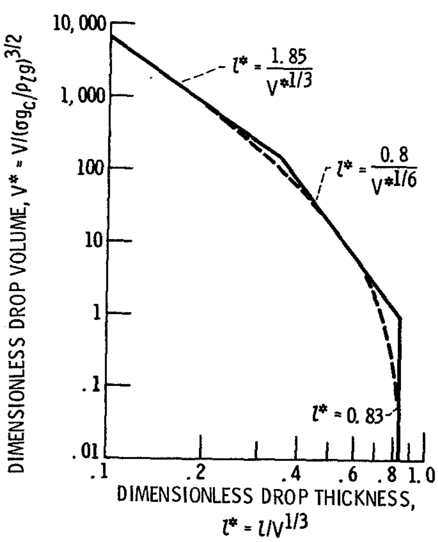


Figure 2. - Universal average drop thickness curve (ref. 5).

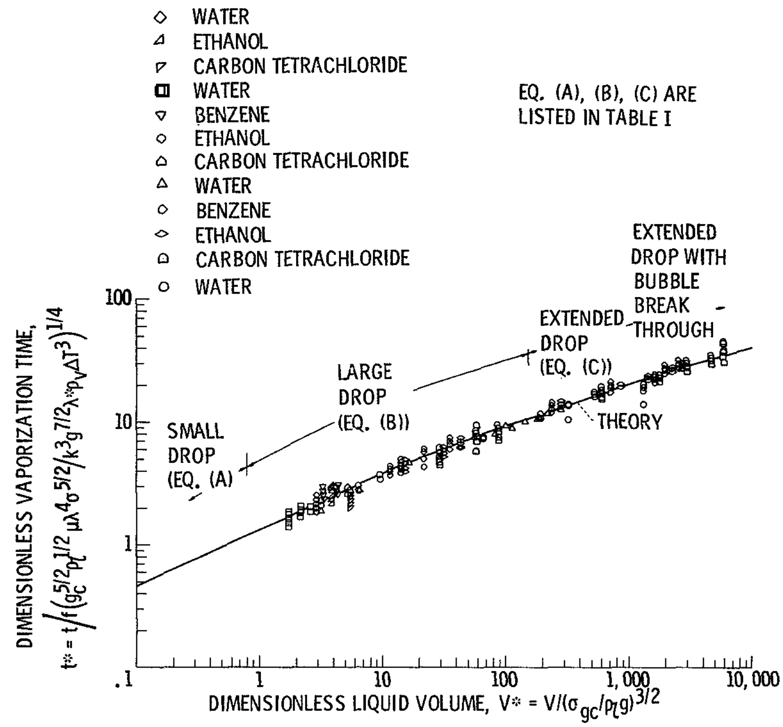


Figure 3. - Universal total vaporization time curve (ref. 5).

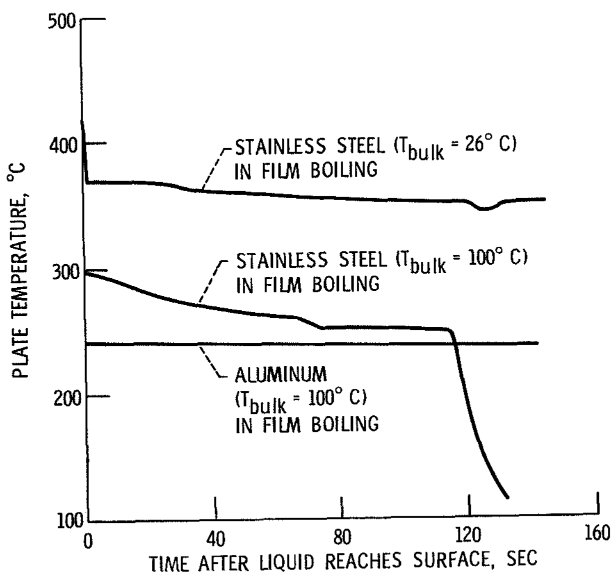


Figure 4. - Temperature of surface after 6 milliliter drop of water reaches the heating surface.

E-5750

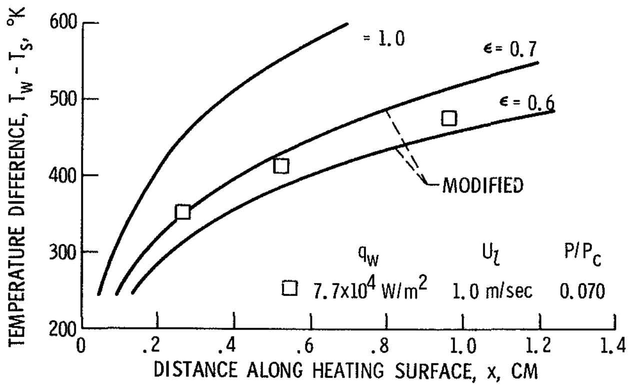


Figure 5. - Modification of laminar film boiling equation by considering the wave profile (ref. 32).

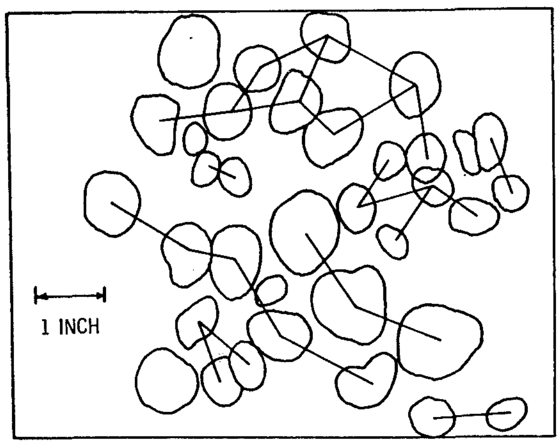


Figure 6. - Distribution pattern of bubble cells from a horizontal surface during film boiling (ref. 40).

E-5750

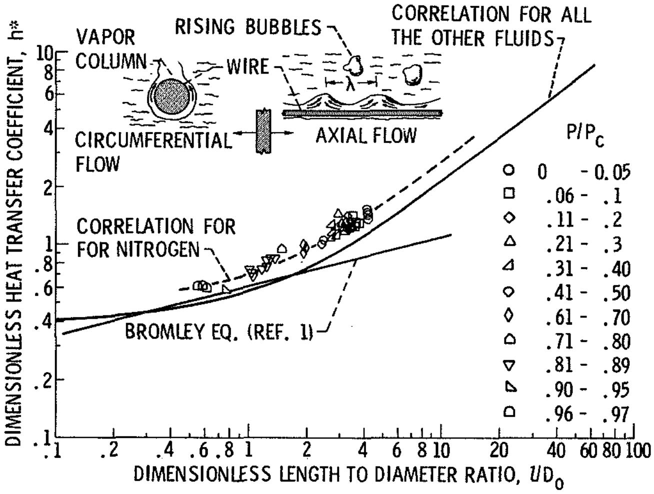


Figure 7. - Saturated film boiling heat transfer data of nitrogen from atmospheric to the critical pressure (ref. 49).

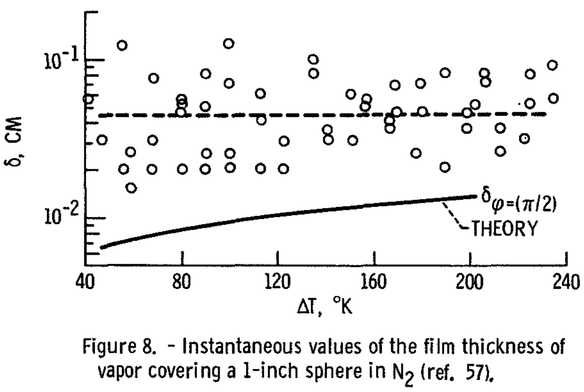
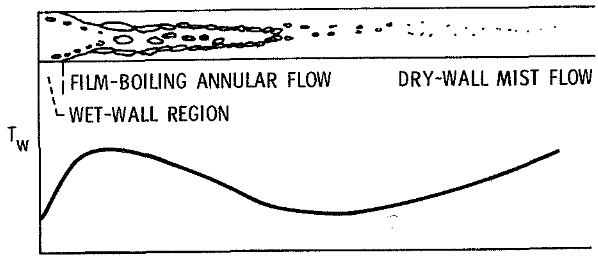
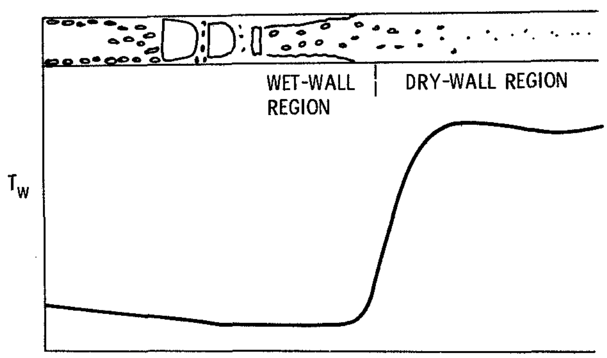


Figure 8. - Instantaneous values of the film thickness of vapor covering a 1-inch sphere in N_2 (ref. 57).

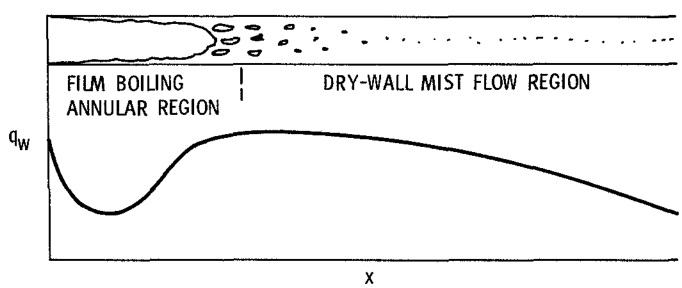
E-5750



(a) Wall-temperature profile for the case of constant heat flux (high heat flux).



(b) Wall-temperature profile for the case of constant heat flux (low heat flux).



(c) Wall heat flux profile for the case of constant wall temperature.

Figure 9

E-5750

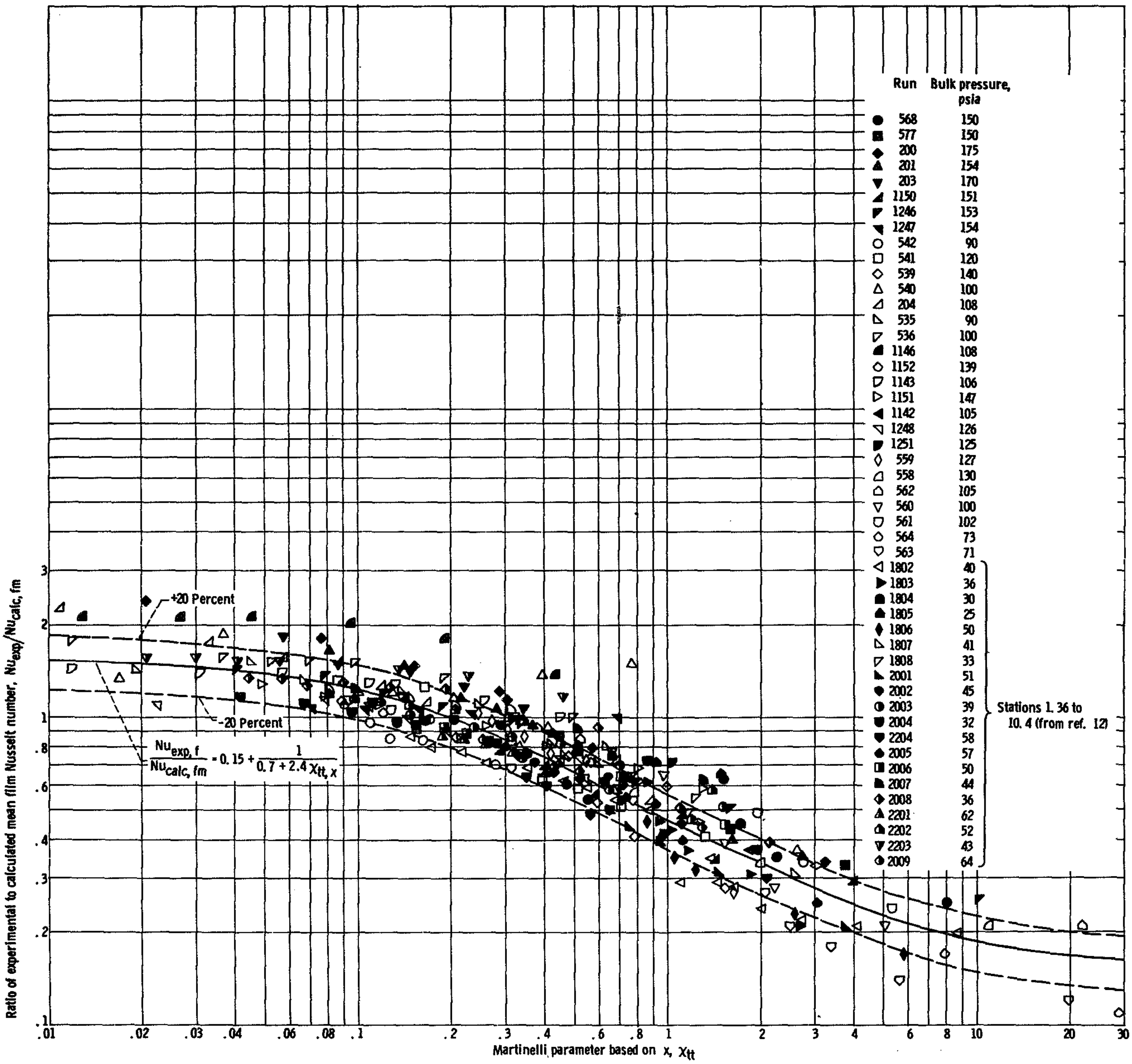


Figure 10. - Subcritical hydrogen heat-transfer data in a two-phase flow film boiling channel (ref. 74).

E-5750

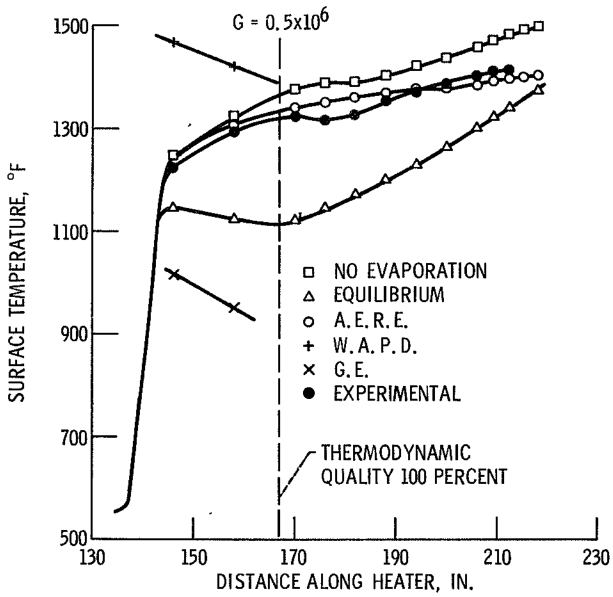


Figure 11. - Comparison of experimental temperature profiles with calculations based upon various models. (ref. 83).