The new steric factor is found to be an improvement on that of Bird in that the reaction path may be calculated independently of the choice of the internal energy contribution. For a simulation in which the temperature is constant, e.g., in heat bath studies, the new model is just as efficient as Bird's formula. However, for the investigation of flows in which temperature gradients exist, it is necessary to evaluate a local temperature from the molecular data. It is estimated that such procedures incur no more than a 5% computational overhead.

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Hydrodynamics of Film Boiling from a Cylinder in Crossflow

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Nomenclature

c = wave speed
n = wave number
p = pressure
q = heat flux
t = time
ΔT_B = degree of liquid subcooling, $T_{sat} - T_B$
ΔT_w = wall superheat, $T_w - T_{sat}$
U = velocity
V = freestream velocity approaching a cylinder
W = complex potential
x,y = curvilinear coordinates shown in Fig. 4
z = complex number, $x + y$
γ = amplitude of wave
δ = vapor film thickness
ζ = liquid layer dimension
η = coordinate for vapor film
θ = angle
λ = wavelength
ρ = density
θ = surface tension
ω = frequency

Subscripts
cr = critical
d = most dangerous
i = imaginary
l = liquid
s = separation
sat = saturation
v = vapor
δ = interface

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Introduction

FILM boiling during crossflow over cylinders involves several hydrodynamic aspects that make a mathematical prediction of heat transfer difficult. A liquid boundary layer rides over a vapor layer; each layer must be properly represented mathematically and compatibility conditions must be specified at the interface. Also, a wake is formed behind the cylinder. Even though it resembles a cavitation wake in many respects, see Kaul and Witte,[1] its behavior is not very well understood. Furthermore, as the vapor film thins in response to decreasing surface temperature, the vapor film can become unstable causing liquid-solid contact and leading to transition boiling as recently measured by Chang and Witte.[2]

In this paper, we report research into the nature of vapor film crossflow over a cylindrical heater. Photographs of R-11 boiling from a 6.35-mm-diam heater reveal the details of wake formation and behavior near the minimum heat-flux condition. A stability analysis of flow boiling is presented and related to experimental observations. An explanation for the behavior of the vapor wake near the minimum heat flux condition is proposed.

Photographic Observations

Experiments involving film and transition boiling of R-11 from a 6.35-mm-diam heater were performed. The apparatus and details of liquid-solid contract and heat transfer measurements are described in detail by Chang and Witte[2] and Chang[3]; those details will not be repeated here.

Photographs were obtained to better understand forced convection boiling from cylinders. A Nikon 3F camera with a 30–105 zoom lens was used at 1/2000 s with an aperture of 5.6.

Figure 1 shows stable film boiling of R-11 for low velocity, 0.7 m/s, and low subcooling, 3.3°C. Liquid is flowing upward over the heater. The vapor film over the front of the heater is smooth. A separation line can clearly be seen that indicates where a wake is formed on the cylinder. The average separation angle for this case was measured to be 105-deg. This point is just beyond the liquid-solid probe, which can be seen at the 90-deg point. It is clear that the probe did not interfere with the basic boiling process.

Figure 1 shows film boiling just above the minimum point on the boiling curve. The bottom of the heater was covered by a smooth vapor film, while vapor patches were being swept away from the wake by the liquid flow. The effect of drag on the edge of a rough, thick vapor layer in the wake can be seen in this photograph.

Figure 2 shows film boiling at a higher velocity and slightly higher subcooling. The behavior is basically the same except that vapor is torn away from the edge of the vapor wake at more locations than for lower velocities and lower subcoolings.

Because the electrical heating produces a constant heat flux condition at the heater surface, considerable variation of temperature around the heater occurred. This influences the local vapor film thickness. In Fig. 1, 2, and 5, the local ΔT_w measured by the contract probe is used to characterize the experimental condition.

Film Boiling Wakes

A model for the behavior of the wake under these experimental conditions can be formulated based on photographs and visual observations. The wake collects the vapor produced by the smooth thin film that exists over the forward portion of the heater. The vapor wake is an almost-stagnant vapor region with condensation occurring at its liquid-vapor interface when the liquid is subcooled.

Figure 3 is a sketch that shows that vapor is torn away intermittently from the wake along the separation line. Following this, an incursion of liquid occurs. The photographs clearly show this behavior. Figures 1 and 2 show that for points below the large patches that have been torn away, the separation line is pushed significantly toward the rear of the heater. Where no vapor has been removed, few or no patches were observed.

Data from the liquid-solid contact probe tended to verify this model. When the probe was at 180 deg, random small-scale liquid-solid contacts were detected,[3] much like those detected when the probe was at 90 deg; even though the vapor film thickness was much larger in the wake than 90 deg. Apparently, during these incursions—in which liquid pushes over the liquid-vapor interface separating the liquid and vapor wakes—some significant "falling" of the liquid occurred partially wetting the surface. Liquid-solid contact produced local temperature fluctuations, averaging 1-5°C in most cases, see Ref. 2 for details.

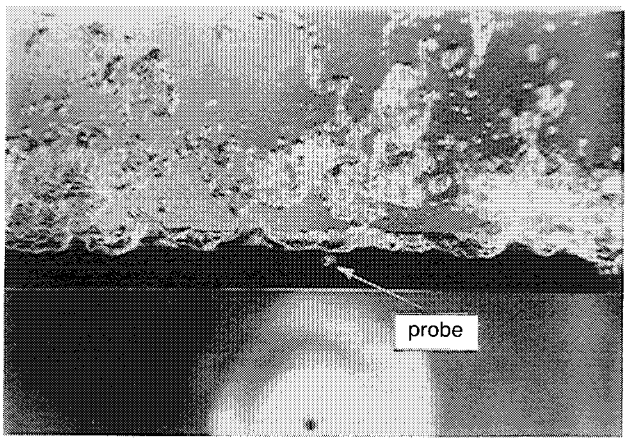

Fig. 1 Photograph of flow film boiling from a 6.35-mm cylinder in Fr-11: $V = 0.7$ m/s, $\Delta T_B = 3.3$ C, probe at 90 deg position, $q = 79.1$ kW/m^2, $\Delta T_w = 208$ C. (The flow direction is upward.)

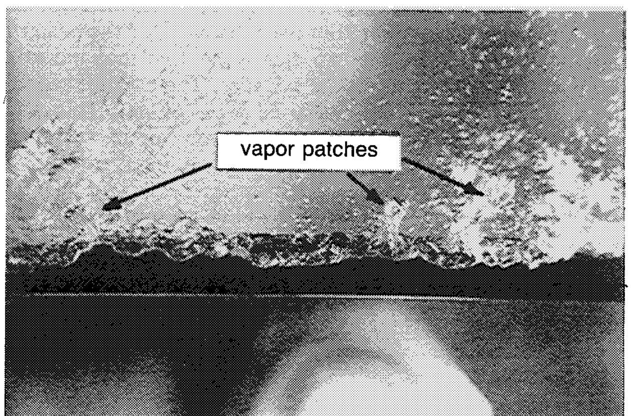

Fig. 2 Photograph of flow film boiling from a 6.35-mm cylinder in Fr-11, $V = 1.5$ m/s, $\Delta T_B = 5$ C, $q = 114.9$ kW/m^2, $\Delta T_w = 193$ C.

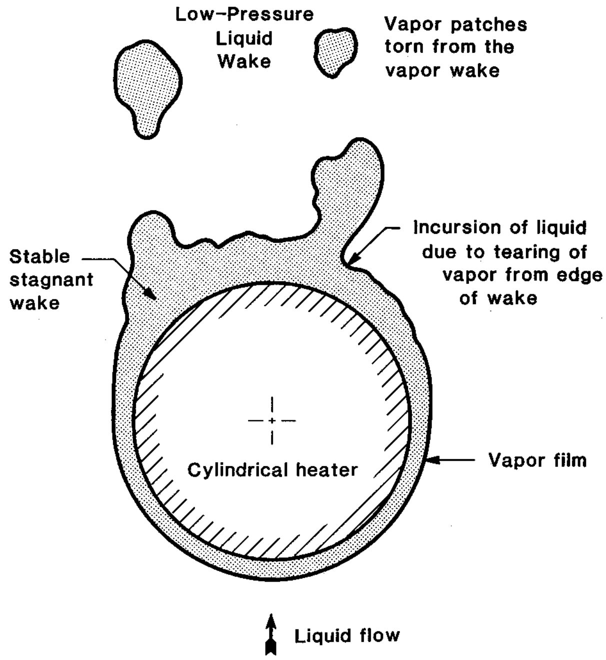

Fig. 3 A physical model for the behavior of the vapor wake with liquid incursions.

When the flow velocity was increased along with the liquid subcooling, the interaction of the vapor wake and the liquid flow intensified. Under those conditions, large vapor patches no longer existed; only scattered small bubbles could be seen in the wake instead. This is consistent with the proposed model because the incursions are smaller in spatial extent but more of them exist at any time.

Stability of Vapor Film Boiling

Transition boiling is characterized by increasing heat flux with decreasing surface temperature. This is caused by liquid contacting the heated surface as surface temperature decreases. It is generally conceded that such contacts begin when the vapor film can no longer prevent the liquid from touching the surface. For pool boiling, this simply means that the film becomes so thin that undulations of the liquid-vapor (l-v) interface caused by vapor bubble departure cause touching of the surface. However, in flow boiling, the film is much smoother owing to the fact that vapor is swept around the body to be collected in a vapor wake. Thus, a mode(s) of vapor film instability other than Rayleigh-Taylor instability is required to account for the breakdown of the vapor film.

In most film boiling analyses, the vapor film is assumed smooth for ease of solution. However, perturbations of the vapor film have been observed in the form of axisymmetric waves and "dimples," see Stevens and Witte.[4]

Flow film boiling exhibits many of the characteristics of boundary-layer flows. But in the boiling process, there is significant vapor flux across the liquid-vapor interface. However, Greitzer[5] showed that the "vapor repulsive" force due to mass transfer across the l-v interface during film boiling is much smaller than the surface tension forces except for very small wave-number waves and for cases when the amplitude of waves is relatively large compared to the thickness of the vapor film. Observations of film boiling show that neither of those conditions is met for most flow film boiling cases. Thus in the analysis that follow we neglect this effect. This is appropriate for a linear analysis as contemplated here.

There are several modes of instability that might be operative in film boiling. In this section, we give a brief development of the Kelvin-Helmholtz instability mode.

When two flows are traveling next to each other at different speeds, their interface can display unstable behavior under certain conditions. For flow around a cylinder, the liquid and vapor velocities are maximum at the $\pi/2$ point, and the two flows approximate two parallel flows quite nicely there. For convenience, we visualize the flows as inviscid, and take the liquid velocity as $2V$ compared to the average vapor velocity of V. The $\pi/2$ point is thought to be the most likely location for Kelvin-Helmholtz instability to set in because the velocities are maximum at that point; thus we confine our analysis to that point on the cylinder. Thus, our analysis applies specifically to the 90-deg point on the heater.

Consider the parallel flow situation of Fig. 4. The liquid and vapor are confined to layers ζ and δ, respectively. Coordinates are taken as shown in Fig. 4. A wave of small amplitude $\eta = \gamma \sin(nx - \omega t)$ propagates at the interface with velocity $c = \omega/n$.

If a velocity c is imposed opposite to the direction of wave propagation, the wave becomes stationary and the liquid and vapor velocities change to $2V - c$ and $V - c$, respectively. Milne-Thompson[6] gives the complex potential for the vapor as

$$W_v = -(V-c)z - \frac{\gamma\,(V-c)}{\sinh(n\delta)}\cos[n(z+i\delta)] \tag{1}$$

where $z = x + iy$. Similarly, for the liquid

$$W_l = -(2V-c)z - \frac{\gamma\,(2V-c)}{\sinh(n\zeta)}\cos[n(z-i\zeta)] \tag{2}$$

Bernoulli's equation can be used to express continuity of pressure across the interface. Surface tension requires that the pressure be balanced according to the relation

$$P_v - P_l = -\sigma\frac{\partial^2\eta}{\partial x^2} \tag{3}$$

Milne-Thompson[6] shows that at the interface $y = \eta$, the vapor velocity, neglecting the term of order γ^2, is

$$U_v^2 = (V-c)^2[1 - 2n\eta\coth(n\delta)] \tag{4}$$

and for the liquid

$$U_l^2 = (2V-c)^2[1 - 2n\,\eta\coth(n\zeta)] \tag{5}$$

By combining Eq. (3) and Bernoulli's equation, the pressure condition may be written as

$$\frac{1}{2}\rho_v U_v^2 = \frac{1}{2}\rho_l U_l^2 + \sigma\frac{\partial^2\eta}{\partial x^2} + \text{constant} \tag{6}$$

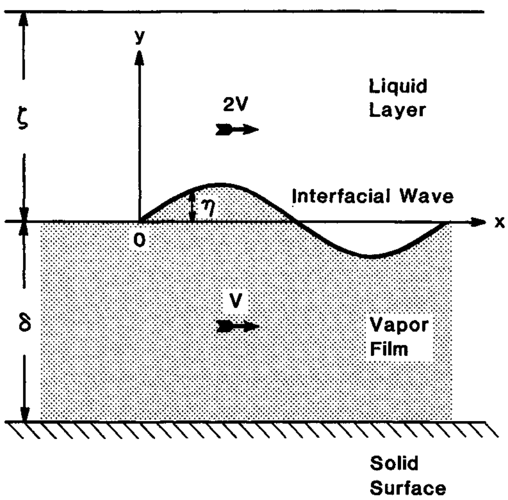

Fig. 4 Inviscid parallel flow of liquid and vapor at $\theta = \pi/2$ from the cylinder stagnation point.

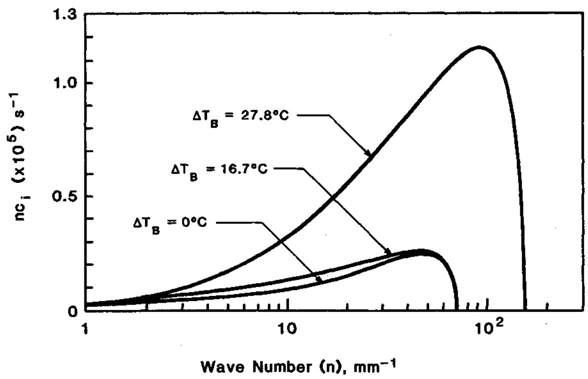

Fig. 5 Liquid subcooling effect on the growth rate of an interfacial wave in inviscid parallel flow: 6.35-mm cylinder in Freon-113; $V = 6.8$ m/s, $\Delta T_w = 90$ °C, $\theta = 90$ deg.

Equations (4) and (5) can be substituted into Eq. (6), and since η is entirely arbitrary, the coefficient of η must vanish. Thus the equation for the wave velocity is found as

$$c = \frac{\rho_v V\coth(n\delta) + 2\rho_l V\coth(n\zeta)}{\rho_v\coth(n\delta) + \rho_l\coth(n\zeta)} + \left\{\frac{\sigma n}{\rho_v\coth(n\delta) + \rho_l\coth(n\zeta)} - \left[\frac{\rho_v\rho_l V^2\coth(n\delta)\coth(n\zeta)}{\rho_v\coth(n\delta) + \rho_l\coth(n\zeta)^2}\right]\right\}^{1/2} \tag{7}$$

which consists of the mass mean velocity and the velocities caused by surface tension and inertia shown in the brackets. The quantities inside the brackets determine the nature of the wave velocity. If inertia overrides surface tension, the wave velocity becomes complex instead of real. Under such conditions, the interfacial wave is amplified according to the growth rate nc_i, where $c = c_r + ic_i$.

By treating the liquid layer as very deep compared to the vapor film (a legitimate assumption), the coth $(n\zeta)$ term approaches unity in Eq. (7). Let the quantity coth $(n\delta)$ be represented by s. The condition of neutral stability is then obtained by setting the bracket of Eq. (7) to zero, as

$$\rho_v\rho_l V^2 s = \sigma n(\rho_v s + \rho_l) \tag{8}$$

and the critical wave number is

$$n_{cr} = \frac{\rho_v\rho_l V^2 s}{\sigma(\rho_v s + \rho_l)} \tag{9}$$

The most dangerous wavelength occurs when the wave is the fastest growing, $\lambda_d = 2\pi/n_d$. The n_d is found using the condition $d(nc_i)/dn = 0$ to maximize nc_i. Figure 5 shows the effect of liquid subcooling on the stability. Physically, the vapor film thickness decreases as the subcooling increases. Thus the analysis shows that thinner films are more susceptible to instability than thick ones. This agrees with experimental observations that always show the tendency for instability to set as ΔT_B is increased.

The Relationship of Observations to Stability Analyses

The Kelvin-Helmholtz analysis predicts that the growth rate of an interfacial wave increases with liquid subcooling and velocity. Thus increased velocity and subcooling cause a breakdown of the film at angles closer to the 90-deg point. If we interpret the separation angle as an indicator of vapor film instability, then the movement toward 90 deg as subcooling and velocity increases supports the Kelvin-Helmholtz analysis. Orozco and Witte[7] reported that the instability of the vapor film for boiling of R-11 around a sphere was a wave-like breakdown of the vapor film where the vapor film separated to form a wake. This is further support of the Kelvin-Helmholtz mechanism of vapor film instability.

Liquid-solid contact data reported by Chang[3] showed that for the velocities in our experiments the liquid-vapor interface separating the liquid wake from the vapor wake probably experiences a Taylor-like instability. That is, the liquid above the vapor wake can fall through the interface and touch the heater surface. Thus film boiling heat transfer analyses for upflowing liquids over cylinders should include this type of analysis of the vapor wake.

Acknowledgments

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Heat Transfer Across Aluminum/Stainless-Steel Surfaces in Periodic Contact

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Introduction

DURING the past forty years, the primary emphasis in the study of heat transfer across contacting surfaces has been focused on steady-state contacts. In the area of dissimilar surfaces across the contact, most of the research activity has been devoted to the problem of thermal rectification—the phenomenon of the dependence of the thermal contact conductance h_c on heat flux direction. Studies of dissimilar steady-state contacts involving aluminum/stainless steel have been reported by several investigators, as summarized by Dodd and Moses.[1]

A wide variety of engineering problems are also of interest that involve intermittent loading conditions across contacting surfaces, including aircraft braking systems, thermal control of electronic components, and a variety of manufacturing processes. Although the problem of periodically contacting dissimilar metals is not addressed in the literature, the problem of periodically contacting similar metallic interfaces has received some attention. Since the temperature during any cycle is always a function of time for these problems, a true steady-state condition is never attained. However, as a reference condition, the quasisteady state will be defined as the condition where the temperature distribution for cycle n is the same as that for cycle $(n+1)$ (and succeeding cycles).

The problem of the quasisteady-state heat transfer across two surfaces coming into regular, periodic contact has been examined analytically for perfect thermal contact[2,3] and with the effects of the thermal contact conductance at the contact interface.[4,5] Experimental results for periodic contacts in similar metal surfaces are given by Howard[6] and Moses and Johnson.[7,8]

The objective of this Note is to extend the study of heat transfer across periodically contacting surfaces to include observations on the effects of dissimilar metals across the contact interface. The inclusion of these effects on the behavior of the thermal contact conductance and the temperature distributions across such interfaces should provide insight into these parameters for a wider range of material characteristics than is currently addressed.

Experimental Apparatus

The mathematical model of the experimental problem is the same as that considered by Vick and Ozisik[5] for finding $T(x,t)$ for one-dimensional heat transfer through two specimens with thermal conductivity k, and thermal diffusivity α, each of length L, heated at $x=0$ and cooled at $x=2L$, with the contact interface at $x=L$. The experimental apparatus consists of two test cylinders—each held at one end in a thermal reservoir—the supporting frame, and the equipment required to bring the test specimens uniformly into and out of contact. Details of the apparatus are provided by Moses and Johnson.[7,8] The test specimens are nominally flat and free of coatings or surface oxidation. Surface profilometer measurements show the CLA roughness to be 0.71 μm for the aluminum (alloy 2024) specimen and 2.54 μm for the stainless steel (ANSI 303).

Procedure and Data Acquisition

Since the purpose of the experimental program is to examine the influence of periodic contact on the heat transfer across the contact interface, both apparent interface pressure and mean interface temperature are controlled to reduce the number of parameters influencing the thermal contact conductance. After allowing the test specimens to come to a steady-state condition while separated, the air-pressure regulator is set to provide an applied load at the contact interface of 85 kPa and the experiment is activated. The mean interface temperature for the test specimens is 48.5°C. Heat transfer during the experimental series is directed from the aluminum specimen to the stainless-steel specimen.

Experimental measurements are made in two phases. First, measurements are conducted to determine the length of time for the problem to come to a nondimensionalized quasisteady-state condition. Temperature measurements are made at different intervals for each experiment in this phase. Sampling intervals of 3, 6, 12, and 24 s are used for contact/separation intervals of 15, 30, 60, and 120 s, respectively. Once the number of cycles required to attain the quasisteady state has been determined, the experiments are rerun to collect data over one cycle in the quasisteady state. During this second phase, data records over all contact/separation ranges are made at 3-s intervals.

Of the two quantities of interest, the temperature distribution is available from the experimental output. Computation of the thermal contact conductance is accomplished by the use of a parameter estimation technique for solution of the inverse conduction heat-transfer problem.[9] Using the method of Kline and McClintock, the overall uncertainty for the experiments is computed to be ±20.6%.

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