Experimental Investigation of Transition Boiling in Forced Convection

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An experimental investigation was conducted to examine the effects of surface condition and of liquid temperature and velocity on transition boiling from a spherical heater in Freon 113. The experimental data indicated that the transition boiling fluxes as well as the maximum heat flux were very sensitive to the condition of the heated surface. Boiling curves exhibiting a double maxima, which were attributed to instances of liquid–solid contact in the transition regime, were obtained. Oxidation and roughness of the heated surface had a strong impact on the magnitude of the maximum heat flux and on the superheat at which it occurred.

Nomenclature

c_p = specific heat
k = thermal conductivity
q = heat rate
q'' = heat flux
R = sphere radius
r = radial coordinate
T = temperature
t = time
U_∞ = sphere velocity
ΔT = superheat, $T_w - T_{sat}$
θ = sphere angle measured from the forward stagnation point
ρ = density
ϕ = azimuthal angle

Subscripts and Superscripts
b = boiling
c = conduction
i = inner
L = liquid
n = time counter in finite difference formulation
o = outer
s = sphere
sat = saturation
w = wall

Introduction

RELATIVELY few studies have been formed on transition boiling heat transfer in comparison to those on nucleate and film boiling. This is due in part to the difficulties encountered in designing thermal systems to operate in this regime.

Berenson,[1] making use of a copper block that was heated from below by the condensation of high-pressure steam and cooled on top by the boiling of a low-pressure boiling fluid, measured the surface heat fluxes as a function of the temperature difference between the top of the copper block and the saturated fluid. In his experiments, Berenson noted that the peak heat flux in pool boiling was slightly dependent upon the surface condition of the heater. The minimum heat flux and the transition boiling fluxes were exceedingly high for the case where the boiling liquid wetted the heated surface significantly. An examination of the data from Berenson's flat-plate boiling experiments by Witte and Lienhard[2] resulted in their suggestion that a heated surface has two possible heat fluxes at each surface temperature in the transitional regime. It was argued that the increase in the transition heat flux was the result of a jump from the transitional film boiling curve to the transitional nucleate boiling curve, in contrast to Berenson's conclusion that wetting lifted the transition boiling curve. Bui and Dhir[3] studied transition boiling heat transfer from an isothermal vertical surface in a pool of saturated water. Their experimental data revealed that transition boiling was very sensitive to the surface condition of the heater as well as to the history of the process. Two distinct transition boiling curves were observed during transient heating and cooling of clean surfaces. However, the difference between the two curves diminished as the wettability of the surface increased. Confirming Bui's findings, Chowdhury and Winterton[4] concluded that surface wettability did play an important role in transition boiling after systematically studying transition boiling during quenching of metal cylinders. They found that the critical heat flux was strongly affected by contact angle and that the heat flux throughout the transition region improved with better wetting, i.e., lower contact angle. Ramillison and Lienhard[5] recreated Berenson's flat-plate transition boiling experiments with a reduced thermal resistance in the heater, and improved access to those portions of the transitional regime that have a steep negative slope. The resulting data reproduced and clarified certain features of Berenson's work, such as the modest surface finish dependence of the peak heat flux, on what Witte and Lienhard previously identified as the "film-transition boiling" regime.

A review of the limited literature available in transition boiling heat transfer indicates that no work has been done on the combined effect of liquid velocity and temperature, and of surface condition on the transition boiling regime. It is the aim of this article to report on an investigation on the effects of the above-mentioned parameters on forced convection transition boiling of Freon 113 from a spherical heater. Different surface conditions were imposed on a 3.84-cm-outer-diameter (o.d.) copper sphere which after being heated is propelled into a pool of Freon 113. Boiling fluxes were measured at various points on the surface of the sphere and the effects of surface condition and of the liquid temperature and velocity were analyzed.

Experimental Apparatus and Procedures

Figures 1 and 2 show the test model and the facility in which the experiments were conducted. The experimental apparatus

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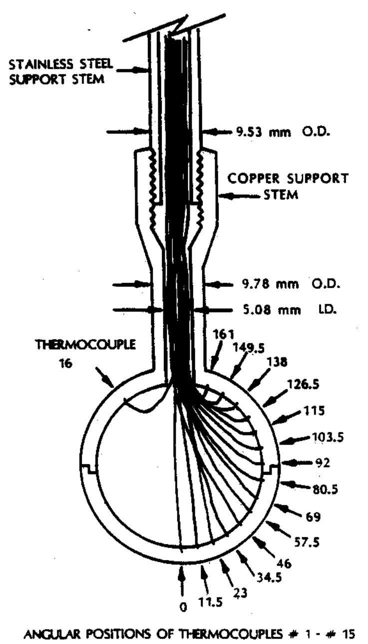


Fig. 1 Test model.

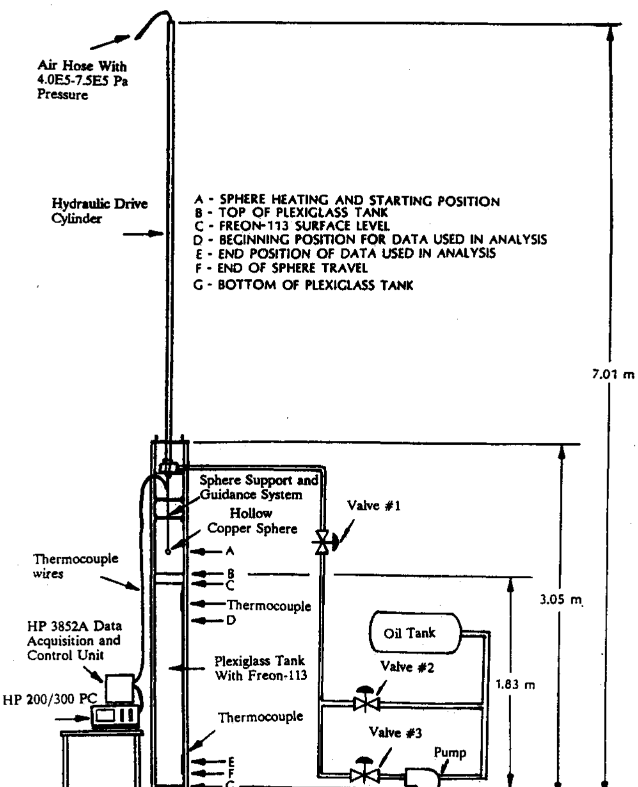


Fig. 2 Test fixture.

was described in detail by Dix and Orozco.[6] The facility was slightly modified to ease the acquisition of experimental data and the recording of photographic and video observations. A brief description of both the test model and the experimental apparatus is given for the benefit of the reader; however, Ref. 6 should be consulted for additional information.

A hollow, 3.84-cm-o.d. copper sphere, with a 0.28-cm average wall thickness was used as the test model. The sphere was instrumented with 16, 30-gauge copper-constantan thermocouples that were mechanically pinned to the inner surface of the sphere and spaced 11.5 deg apart starting at the forward stagnation point. The selection of a hollow sphere as test model was based on the technique used for the installation of the thermocouples. The thickness selected, 0.28 cm, was a compromise between the ability to collect experimental data throughout the whole transition boiling regime and the technique employed to attach the thermocouples. The sphere was heated to a preselected temperature and then propelled into a 108-liter holding tank filled with Freon 113 at speeds up to 1.00 m/s by a 4.00-m-long, double-acting, hydraulic cylinder. The use of a transient experimental technique and of a tank of limited height were also deciding factors in the selection of a thin shell sphere. A sphere with either a large mass or made of poor thermal capacitance materials would take longer to undergo transition boiling. Two video cameras, a Panasonic WVD-5000 and a JVC KY-310V, were used for the video recordings. In addition to the two video cameras, a large format, Crown Graphic Special, view camera with a Polaroid 545 film holder was used to record the sphere–vapor interaction in detail. Temperature measurements were recorded at 16 locations on the inside of the hollow copper sphere and at two different locations inside the holding tank. The data acquisition was accomplished with a Hewlett Packard 3852 computer system. Thermocouples were sampled sequentially at the rate of 150 measurements per second. Sphere velocities were determined with the aid of the video recording system.

The sphere surface was rubbed with sandpaper of various grit sizes to produce different degrees of roughness. Surface roughness measurements were made at different locations on the surface of the sphere with the aid of an electron microscope and averaged to characterize the heated surface. The following is the measured average rms roughness in microns (10^{-6} m) for the different sandpapers used: mirrorlike, 0.100; sandpaper #40, 0.242; sandpaper #24, 0.358. Oxidation of the heated surface was attained by placing the sphere inside a cylindrical, aluminum container, and proceeding to heat the surface of the container by means of a propane torch for approximately 20 min. Once the sphere reached 300°F, it was then immersed in water. This process was repeated several times until there was assurance, based on color change, that a layer of oxide existed on the surface of the sphere. Care was exercised to avoid instances of contact between the flame and the surface of the sphere. Contact angles were measured

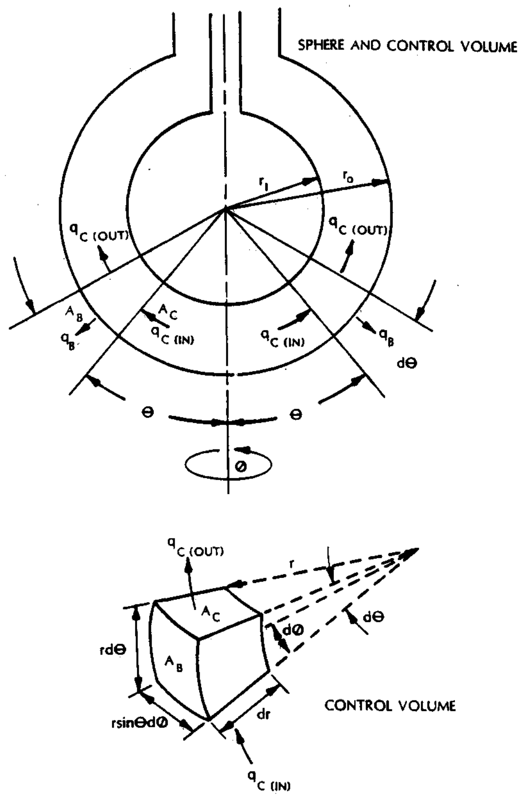


Fig. 3 Control volume for energy balance.

at room temperature. The contact angle for both the clean and the oxidized surfaces were measured using a microscope with a protractor eyepiece. A contact angle of approximately 45 deg was measured for the mirrorlike surface, whereas a contact angle of nearly 18 deg was obtained for the oxidized surface.

Significant temperature variation occurred around the surface of the sphere during the boiling process. Thus, the $T - t$ data had to be reduced so as to account for T_W and q''_b variation with angular position. To solve for the sphere boiling flux, an energy balance was conducted on the control volume shown in Fig. 3. Neglecting the effects of radial conduction, and thermal radiation and convection between the air inside the sphere and the sphere's inside surface, the energy balance yields

$$(\rho c_p)_s \frac{\partial T}{\partial t} dV = q_{c(\text{in})} - q_{c(\text{out})} - q_b \tag{1}$$

where

$$q_{c(\text{in})} = k_s r \sin\theta \, d\phi \, dV \frac{\partial T}{\partial \theta} \tag{2}$$

$$q_{c(\text{out})} = q_{c(\text{in})} + \frac{\partial}{\partial \theta}(q_{c(\text{in})}) \, d\theta \tag{3}$$

and

$$q_b = r^2 \sin\theta d\phi \, d\theta q''_b \tag{4}$$

Substituting Eqs. (2–4) into Eq. (1) and completing some algebraic steps yields

$$q''_b = k_s \left(\frac{\partial^2 T_w}{\partial \theta^2} + \cot\theta \frac{\partial T}{\partial \theta} \right) \left(\frac{R_0 + R_i}{R_0 R_i} \right) - (\rho c_p)_s \frac{\partial T_w}{\partial t} (R_0 - R_i) \tag{5}$$

Introducing the temperature measurements in Eq. (5) allows the computation of the boiling fluxes at the various angular position. This is accomplished by converting Eq. (5) to a finite difference equation. Using an explicit formulation that is second-order accurate in θ and first-order accurate in t, Eq. (5) becomes

$$q''_b,\theta = C_1 \left[\frac{T^n_{w,\theta+\Delta\theta} - 2T^n_{w,\theta} + T^n_{w,\theta-\Delta\theta}}{(\Delta\theta)^2} + \cot\theta \frac{T^n_{w,\theta+\Delta\theta} - T^n_{w,\theta-\Delta\theta}}{2\Delta\theta} \right] - C_2 \left[\frac{T^{n+1}_{w,\theta} - T^{n-1}_{w,\theta}}{2\Delta t} \right] \tag{6}$$

where

$$C_1 = k_s \left(\frac{R_0 - R_i}{R_0 R_i} \right) \tag{7}$$

and

$$C_2 = (\rho c_p)_s (R_0 - R_i) \tag{8}$$

Reproducibility between experimental data performed at the same conditions was quite good. The uncertainty in the heat transfer measurements was due to temperature recording errors. A detailed uncertainty analysis indicated that the maximum expected uncertainty of $\pm 10\%$ would occur in the nucleate boiling regime.

Experimental Results

The results of 12 pool boiling and 24 forced convection experiments provided the data for this analysis. The general relationships between time, temperature, heat flux, and angular position are illustrated in Figs. 4 and 5. The tests usually started at the end of the stable film boiling regime, traveled rapidly through the transition and nucleate boiling regimes, and ended up in the natural-convection regime. In film boiling, the sphere temperature dropped almost linearly with time, and the heat transfer rates around the sphere were quite uniform. When the sphere entered the transition boiling regime, the sphere surface temperature dropped quickly as the heat transfer rates increased. Different regions of the sphere would enter the transition regime at different times. The maximum heat flux was reached at the end of transition and at the beginning of nucleate boiling. Finally, in nucleate boiling the heat flux dropped quickly and the sphere entered the natural-convection regime where no bubbles were observed on the surface of the sphere.

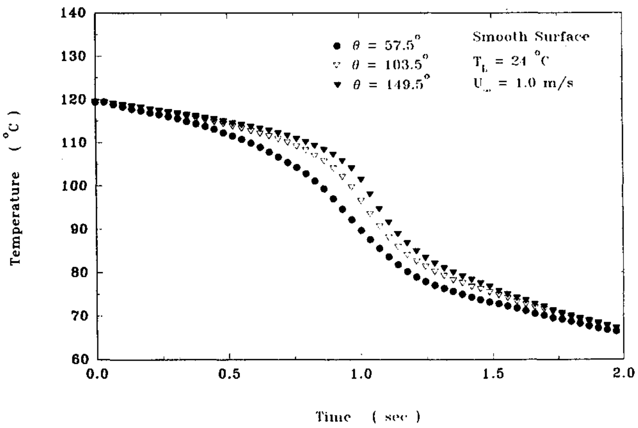


Fig. 4 Temperature versus time trace for test p24v1.

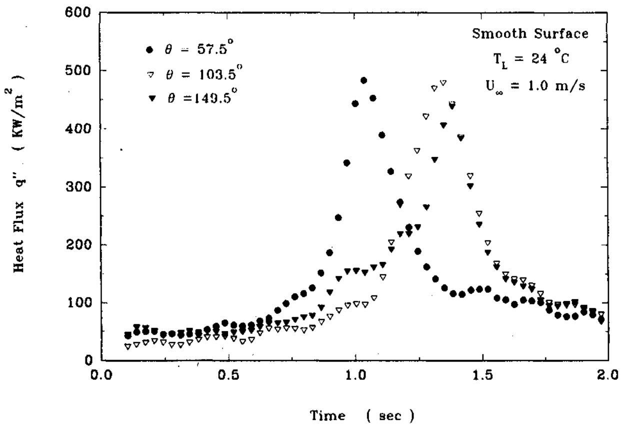


Fig. 5 Time history of heat flux for test p24v1.

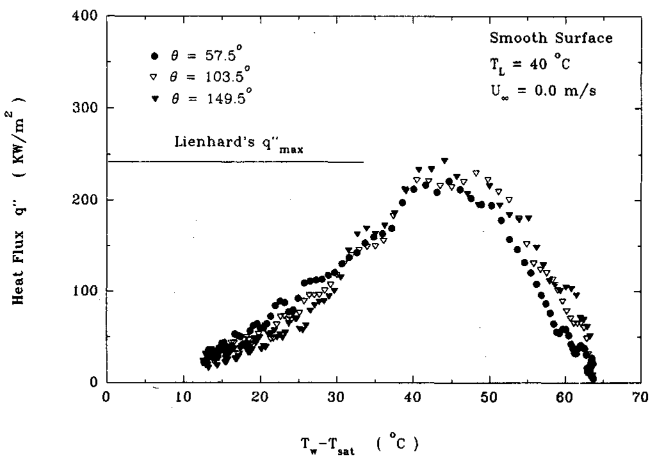


Fig. 6 Heat fluxes in pool boiling near saturation.

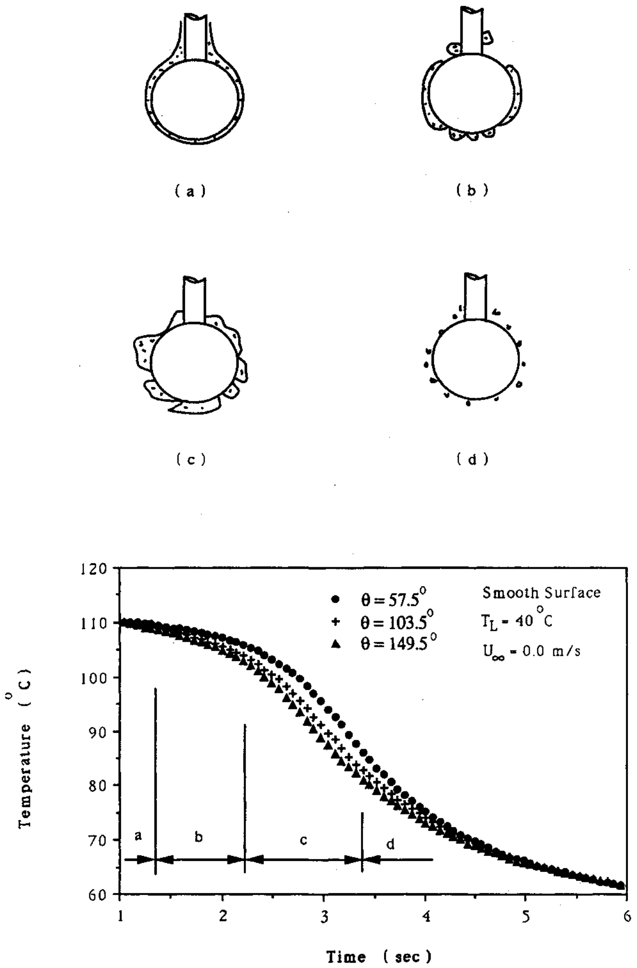


Fig. 7 Various stages of pool boiling.

The comparison between tests was divided into three groups: 1) tests that had different sphere velocities but the same surface condition and liquid temperature; 2) tests that had different liquid temperatures but the same liquid velocity and surface condition; and 3) tests that had different surface conditions but the same liquid temperature and velocity.

Velocity Effect

In order to show the velocity effect, tests P40v3 and P40v1 were run under smooth surface conditions, a 40°C liquid Freon 113 temperature and two liquid velocities, i.e., 0.0 m/s and 1.0 m/s. For the case of zero velocity, the experimental data show good agreement with the Ded and Lienhard's[7] theory for peak heat flux and the boiling fluxes were almost identical for all angular positions. Figure 6 shows the computed pool boiling fluxes at three selected angular locations on the sphere surface. Figure 7 illustrates our observation of the pool boiling process as recorded with the two video cameras. In the film boiling regime, region (a) in the temperature versus time trace, the vapor film was relatively stable but slightly disturbed by regular axisymmetrical waves that originated at the lower stagnation point but moved up around the sphere. In regions (b) and (c), the vapor film started to collapse, first on the upper half of the sphere due to the fin effect of the sphere stem, and next on the sphere lower half. Finally, the system entered the nucleate boiling regime, region (d), where bubbles were seen to originate from the different nucleation sites.

Upon increasing the sphere velocity to 1.0 m/s, higher boiling fluxes were obtained on the heated surface. The flowfield created a bubbly boundary layer on the surface of the sphere which underwent separation at a short distance past the equator of the sphere. The size and length of the vapor wake formed on the back of the sphere was found to be dependent on liquid temperature. Figure 8 shows the wake patterns observed for two different liquid temperatures, 24°C and 40°C, for the same conditions of liquid velocity and sphere temperature. This large accumulation of vapor gave the transition boiling regime in the wake region an oscillatory character with

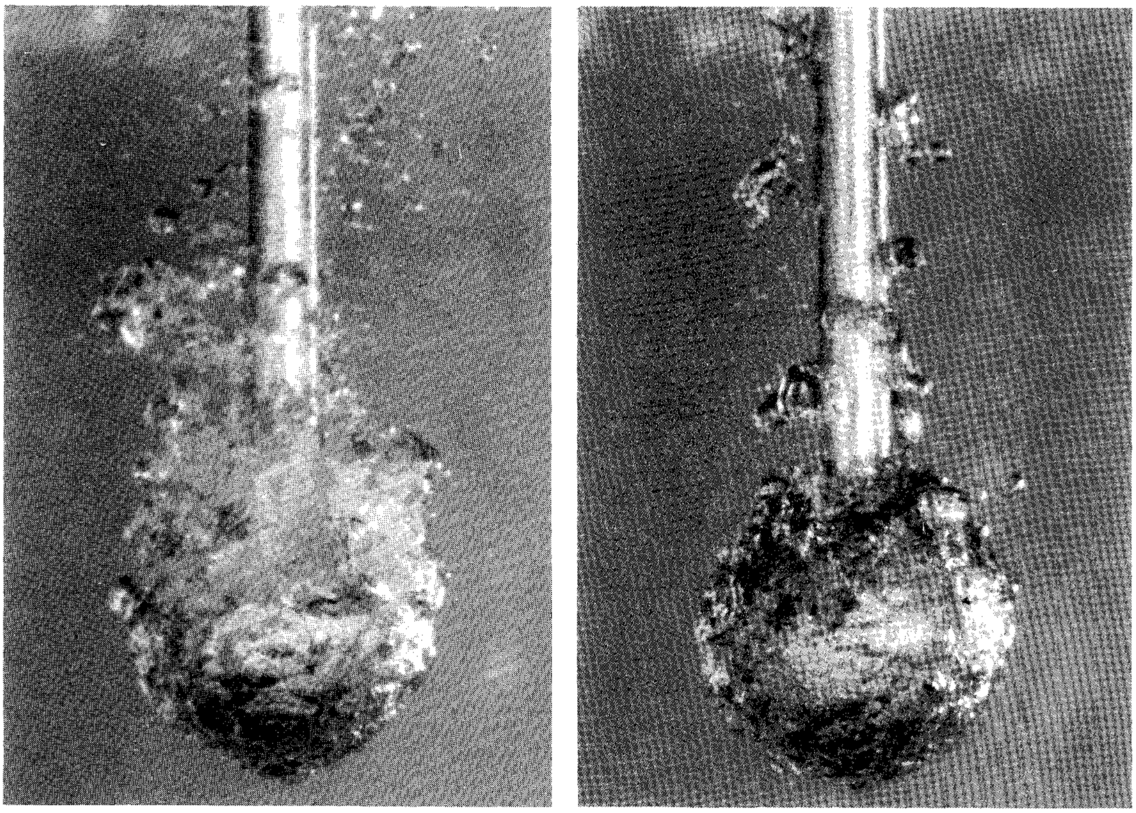


Fig. 8 Still photograph of transition boiling wakes observed for U_∞ = 1.0 m/s, T_w = 105°C and a) T_L = 24°C; b) T_L = 40°C.

enhanced frequency of liquid–solid contact. Figure 9 shows that not only the transitional boiling fluxes at the 149.5-deg angular location were much lower than those recorded at the 57.5-deg and 103.5-deg angular locations but also that the boiling flux versus superheat trace displayed a double maxima in this region. We attributed this dual maxima to the oscillatory nature of the transitional regime which was characterized by intermittent wetting of the heated surface by the liquid Freon 113.

Liquid Temperature Effect

Tests P40v1, P30v1, and P50v1 were conducted with the sphere having a mirrorlike surface finish for three different Freon 113 liquid temperatures (24°C, 36°C, and 40°C) and a sphere velocity of 1.0 m/s. As expected, lowering the liquid temperature yielded higher fluxes in all regimes. Figure 10 shows the boiling fluxes for a Freon 113 liquid temperature of 24°C. The back of the sphere continued to yield lower boiling fluxes and to exhibit a dual maxima on its q''_b versus ΔT trace. However, the data indicated that this dual maxima disappeared as one further lowered the liquid temperature. The video recordings of the boiling process revealed that near saturation, a large vapor wake surrounded the rear of the sphere as it entered the transitional regime. The size of the wake, as well as the thickness of the vapor layer in the film boiling regime, decreased as the degree of liquid subcooling was increased, which resulted in a shorter lived transitional regime and the disappearance of the dual maxima heat flux at the 149.5-deg location.

Surface Condition Effect—Roughness Effect

Tests were designed to determine the effect of surface condition on transition boiling in both pool boiling and forced convection situations. The condition of the sphere surface was changed from that of a mirrorlike finish to a rough finish (0.24 μ) and to a rougher finish (0.35 μ) while maintaining the Freon 113 temperature at 40°C. A comparison of Figs. 11a and 11b with Fig. 6 indicates that upon roughening the surface finish of the sphere, the pool boiling fluxes increased very slightly in value and continued to be surface orientation independent. The pool boiling data also revealed that the maximum heat fluxes occurred at a lower degree of superheat than those obtained for a mirrorlike surface finish. A drop in 10°C and 22°C of superheat in the occurrence of the maximum heat flux was observed for the 24- and the 35-μ surface finishes, respectively. Shoji and co-workers,[8] in their experimental investigation on the effect of liquid subcooling and surface roughness on film/transition pool boiling, observed that surface roughness caused the boiling curve to deviate from that of the mirror finish surface. They found that surface roughness strongly affected the nucleate boiling regime and that the magnitude of the minimum and maximum heat fluxes as well as the superheat at which they occurred were affected by the condition of the heated surface. If we equate the roughening of the heated surface with the creation of additional nucleation sites, it would then be valid to assume that one could attain the maximum heat flux at a lower wall temperature due to the now larger number of nucleation sites. It would also be fair to state that in a quenching process surface condition dictates the occurrence of liquid–solid contact in the film boiling regime. Therefore, for a heated surface with a finite amount of stored energy and undergoing a quenching process, liquid–solid contact in the film boiling regime and its accompanying increase in energy transfer should force the heated surface to enter the transitional boiling regime at a higher temperature and the occurrence of the minimum heat

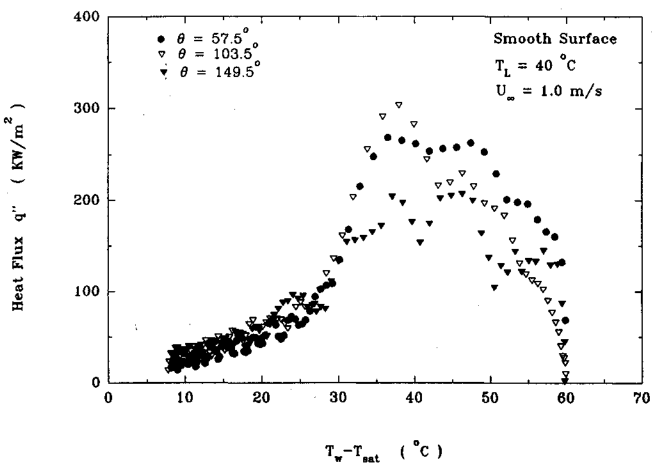


Fig. 9 Heat fluxes near saturation in forced convection.

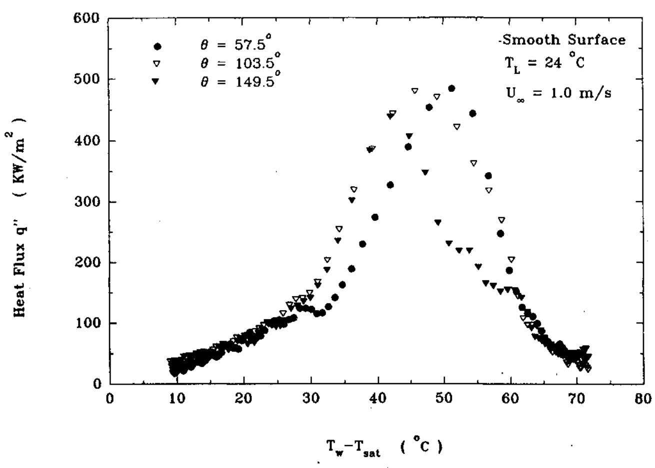
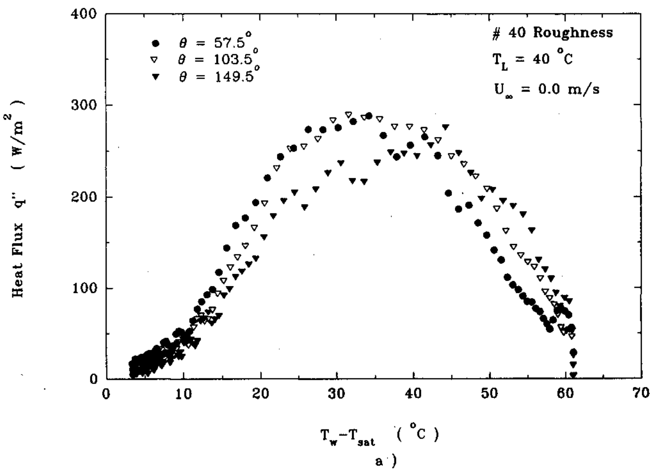


Fig. 10 Boiling fluxes in forced convection for T_L = 24°C.



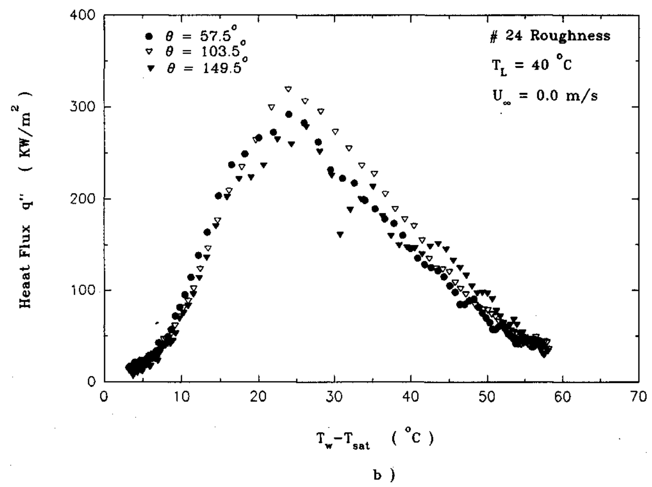


Fig. 11 Effect of surface roughness in pool boiling. a) 0.24 μ; b) 0.35 μ.

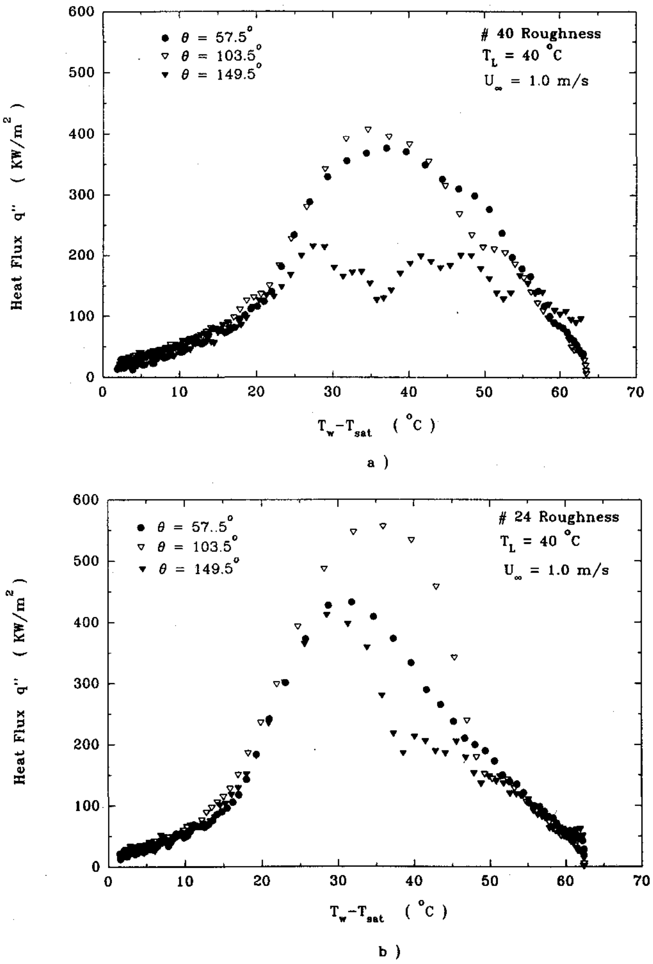


Fig. 12 Effect of roughness in forced convection. a) 0.24 μ; b) 0.35 μ.

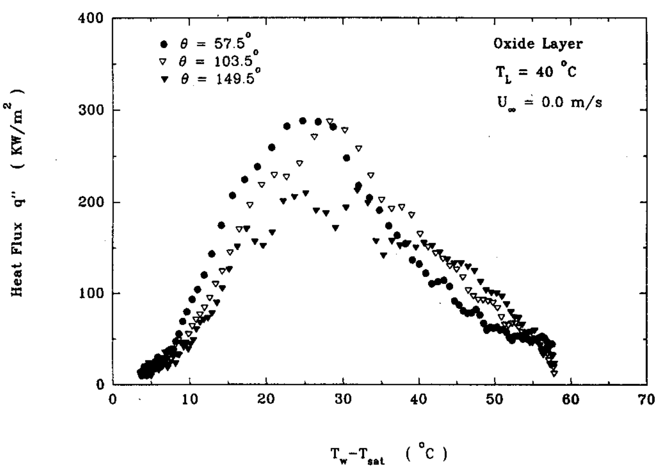


Fig. 13 Effect of oxidation on pool boiling fluxes.

flux at a higher superheat. Figures 12a and 12b illustrate the effect of surface roughness in forced convection transition boiling. The maximum heat flux once more occurred at a lower superheat than that of a smooth surface; however, contrary to the pool boiling results, surface roughness greatly affected the magnitude of the maximum heat flux in forced convection. A careful examination of Figs. 12a and 12b also indicates that the increase in the magnitude of the boiling fluxes could be attributed to the disappearance of the double maxima in the system boiling curve in the rear of the sphere. The first maxima has been explained by Shoji et al.[8] as high release of energy due to the localized collapse and re-establishment of the vapor film while the heated surface is in the film boiling regime. The suppression of this first maxima thus make more energy available to the boiling surface as it enters the nucleate boiling regime.

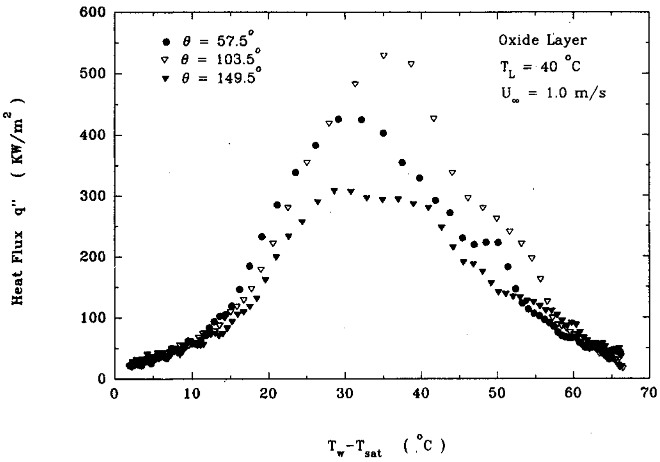


Fig. 14 Effect of oxidation on forced convection fluxes.

Oxidation Effect

The effect of surface oxidation on transition boiling was investigated in both pool and forced convection boiling. Static values of contact angles were measured before and after selected experimental runs. Small surface variations in contact angles were observed after completion of the selected flow boiling experiments. Contact angle values increased in magnitude by 1–2 deg in the frontal region of the sphere where the highest boiling fluxes were registered. A comparison between Figs. 6 and 13 indicates that surface oxidation had strong effect on the pool boiling heat fluxes in the nucleate and transition boiling regimes. Surface oxidation caused the occurrence of the maximum heat flux at a lower superheat and created a dependence of boiling fluxes on surface orientation. The effect of surface oxidation became more dramatic in forced convection. An examination of Figs. 8 and 14 reveals that a sharp increase in the magnitude of the maximum heat flux occurred for all three angular locations considered. The dependence of the boiling flux magnitude on surface orientation also became more pronounced for the oxidized surface. These findings agree well with the results of Maracy and Winterton.[9] They found that the heat flux through the transition regime improved with better wetting, i.e., lower contact angle. The oxidized surface used in this investigation had a reduction by a factor of two in contact angle from that of a clean surface.

Conclusions

The results of this investigation can be summarized as follows:

1) An increase in either liquid subcooling or liquid velocity yields higher fluxes for all boiling regimes. The existence of a double maxima on a system boiling curve, which is induced by the flowfield, disappears as one increases the degree of liquid subcooling.

2) The roughening of the heated surface causes higher heat fluxes by reducing the occurrence of multiple "humps" on the boiling curve, therefore maximizing the heat dissipation efficiency of the heated surface.

3) Oxidation causes higher fluxes in both pool and forced convection boiling. A dramatic shift in the occurrence of the maximum heat flux to a lower surface temperature and a disappearance of the multiple "humps" were observed.

An examination of all the tests results allows us to state that lower liquid temperature, higher liquid velocity, surface roughness, and surface oxidation have similar effects on the system boiling curve. They enhance the ability of the heated surface to dissipate energy.

Acknowledgment

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References

[1]Berenson, P. J., "Experiments on Pool Boiling Heat Transfer," *International Journal of Heat Mass Transfer*, Vol. 5, 1962, pp. 985–999.

[2]Witte, L. C., and Lienhard, J. H., "On the Existence of Two Transition Boiling Curves," *International Journal of Heat Mass Transfer*, Vol. 25, 1982, pp. 756–763.

[3]Bui, T. D., and Dhir, V. K., "Transition Boiling Heat Transfer on a Vertical Surface," *Journal of Heat Transfer, Transactions of the American Society of Mechanical Engineers*, Vol. 107, 1985, pp. 756–763.

[4]Chowdhury, S. K., and Winterton, R. H. S., "Surface Effect in Pool Boiling," *International Journal of Heat Mass Transfer*, Vol. 28, 1985, pp. 1881–1889.

[5]Ramillison, J. M., and Lienhard, J. H., "Transition Boiling Heat Transfer and Film Transition Regime," *Journal of Heat Transfer, Transactions of the American Society of Mechanical Engineers*, Vol. 109, 1987, pp. 746–752.

[6]Dix, D., and Orozco, J., "An Experimental Study of Nucleate Boiling Heat Transfer from a Sphere," *Journal of Heat Transfer, Transactions of the American Society of Mechanical Engineers*, Vol. 112, 1990, pp. 258–263.

[7]Ded, J. S., and Lienhard, J. H., "The Peak Heat Pool Boiling Heat Flux from a Sphere," *AIChE Journal*, Vol. 18, No. 2, 1972, pp. 373–377.

[8]Shoji, M., Witte, L. C., and Sankaran, S., "The Effect of Liquid Subcooling and Surface Roughness on Film/Transition Boiling," *Proceedings of the 1988 National Heat Transfer Conference*, HTD—Vol. 96, pp. 667–673.

[9]Maracy, M., and Winterton, R. H. S., "Hysteresis and Contact Angle Effects in Transition Pool Boiling of Water," *International Journal of Heat Mass Transfer*, Vol. 31, 1988, pp. 1443–1449.