Diameter and Velocity Effects for Cross-Flow Boiling

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The effect of heater diameter on the pool and flow boiling heat transfer of Freon-113 was studied. The heaters were horizontal copper tubes of diameter 1.59-7.94 mm (pool) and 3.18-12.7 mm (flow), heated internally by condensing steam. The test fluid at atmospheric pressure flowed upward, normal to the heaters, at approach velocities of 2.4, 4.0, and 6.0 m/s. For flow boiling, at a fixed velocity and wall temperature, smaller tubes resulted in higher heat flux values, except at low wall temperatures. The maximum heat flow was proportional to the 0.44 power of velocity and inversely proportional to the 0.28 power of diameter. In pool boiling the critical diameter was different for different regimes of boiling and depended on the characteristic bubble size. For tubes smaller than this critical diameter, the heat flux increased as the tube size decreased for a fixed wall temperature. Tubes larger than the critical diameter showed no diameter effect on the heat flux, except for transition boiling. For transition boiling, the heat flux increased as the tube diameter increased beyond the critical diameter.

Nomenclature

C_L = heat capacity of liquid
D = diameter
D^* = dimensionless diameter, Eq. (1)
D_b = nucleate boiling bubble diameter, Eq. (2)
g = gravitational acceleration
Δh_{vap} = heat of vaporization
k_L = thermal conductivity of liquid
Nu_L = Nusselt number, hD/k_L
Pr_L = Prandtl number, $C_L \mu_L / k_L$
q = heat flux
q_B = heat flux based on boil-up
q_c = heat flux based on condenser duty
q_{conv} = convective heat flux
q_F = flow boiling heat flux
q_{max} = maximum heat flux
q_p = pool boiling heat flux
q_{st} = heat flux based on steam condensate
q_{Zuber} = Zuber's expression for maximum heat flux on a flat plate
R_w = heat-transfer resistance of wall plus steam film
Re_L = Reynolds number, $DU\rho_L/\mu_L$
ΔT = metal-to-liquid temperature difference
U = approach velocity
U_{max} = maximum velocity, Eq. (7)
w = back to front boiler dimension
We_L = liquid Weber number, $D\rho_L U^2/\sigma$
We_v = vapor Weber number, $D\rho_v U^2/\sigma$
β = bubble contact angle
λ_c = Taylor wavelength, Eq. (4)
μ_L = liquid viscosity
ρ_L, ρ_v = density of liquid, vapor
σ = surface tension

Introduction

FLOW boiling is a highly efficient means of heat transfer. Most published results are for the case of flow inside tubes. Flow outside tubes is of considerable importance, but it has received less attention. For example, the first determination of the effect of fluid velocity on the entire boiling curve (nucleate, transition, and film boiling) for a boiling liquid flowing normal to a tube was published in 1980 by Yilmaz and Westwater.[1] That paper showed that an increase in fluid velocity always caused an increase in the heat flow.

The study described herein is a continuation of the Yilmaz-Westwater study, but is aimed at examining the effect of heater diameter. No published data describe the heater diameter effect for the entire boiling curve. Studies on isolated regions have been done, with most researchers studying the peak heat flux or film boiling. For the peak heat flux in pool boiling (no forced flow), the effect of diameter has been studied by Kutateladze,[2] Rao and Andrews,[3] and Dhir and Lienhard.[4] They concluded that for large enough heaters, the maximum heat flux is independent of size. But for smaller cylinders, the heat flux increases as the diameter decreases. The various researchers do not agree on the value of the critical diameter. Kutateladeze concluded that the dimensionless diameter is 2, Rao and Andrews gave 3.5, and Dhir and Lienhard gave 2.34. The dimensionless diameter D^* is defined by Eq. (1). It is recognized as being the square root of the Bond number, as

$$D^* = D\left[\frac{g(\rho_L - \rho_v)}{\sigma}\right]^{1/2} \tag{1}$$

The effect of the tube diameter during film boiling in a pool was studied by Bromley[5] and by Breen and Westwater.[6] Both studies showed that, for large enough heaters, the heat flux varies inversely with the heater diameter to the one-fourth power. The latter paper showed that for small heaters, the heat flux increases with heater diameter. The critical tube diameter was shown to be determined by the theoretical wavelength for Taylor instability.

For boiling with forced flow, past research has been concentrated on small ranges of velocities, diameters, and ΔT values. The present work was designed to give a large extension in these ranges and if possible to detect the effect of diameter and velocity on the entire boiling curve. The test system selected was Freon-113 (trichloro-trifluoroethane) flowing upward normal to a horizontal tube. Pertinent prior research with forced flow is discussed later in this paper.

Experimental

A general description of the apparatus and procedure was published previously.[1] Additional details are available.[7]

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The test piece was a 12.7 cm long copper tube polished to a bright lustrous finish before each set of runs. The outside diameter was 12.7-1.59 mm. These heaters were connected to a steam supply system that provided saturated steam at 0.2-9.2 atm. Thermocouples and pressure gages monitored the steam inlet and outlet conditions. The steam condensate was collected and metered to determine the heat-transfer rate. The average copper temperature was measured by a resistance thermometer technique. A 22 A dc passed through the 99.9% pure copper test piece, allowing measurement of its resistance. The desired test piece was housed in a stainless steel boiler with inside dimensions of 39.37 cm high, 12.7 cm wide, and 3.81 cm front to back, sealed by a Teflon gasket. A 20.32 cm square Pyrex window on one face allowed for observation. This boiler was common to both test fluid systems—pool boiling and flow boiling.

Pool Boiling

The pool boiling system (see Fig. 1) was a closed loop constructed of stainless steel, Pyrex, and Teflon. It contained the boiler, connected to an overhead shell-and-tube condenser. Freon-113 was metered and then gravity fed to the boiler via 1.27 cm stainless steel piping. The Freon boiled at its normal boiling point, 47.6°C.

The heat flux was calculated as the average of three independent heat measurements based on condensed steam, Freon-113 boil-up rate, and overhead condenser duty. All boiling curves were traversed starting from the maximum heat flux, down through nucleate boiling, and then to the higher tube wall temperatures. This procedure avoided surface fouling during nucleate boiling. Fouling did occur whenever the tube temperature exceeded 110°C (corresponding to a superheat of about 62°C) and resulted in a very thin, dark coating.

Flow Boiling

The flow boiling system (see Fig. 2) was also constructed of stainless steel, Pyrex, and Teflon. This closed loop consisted of the boiler, connected to a holding tank by 6.1 m of 10.16 cm stainless steel piping. Coils inside the 76.2-cm-diam tank allowed heating or cooling of the test fluid. The Freon-113 was monitored by the four thermocouples located as shown in Fig. 2. The Freon reached the heater with a subcooling of 4-5°C below the 50°C saturation temperature in the flow boiler. The enthalpy of subcooling was less than 2% of the latent heat of evaporation.

The test fluid was circulated by a 7.5 kW centrifugal pump. The velocity was determined by orifice meter measurements. Seven transitions, six below and one above the boiler, transformed the flow from round pipe to the rectangular shape of the boiler. Pitot tube measurements confirmed that the velocity profile was flat as the fluid approached the tube.

For flow boiling, the heat-transfer rate was calculated from the steam condensate rate only. Alternate schemes were not satisfactory, because the pump produced significant heat. Runs were made by starting near the peak heat flux, traversing the nucleate boiling curve, and then traversing the curve in transition to film boiling. This avoided surface fouling for nucleate boiling and at the peak heat flux.

Results and Discussion

Pool Boiling

The pool boiling runs were performed with tubes having outside diameters of 1.59, 2.38, 3.18, 4.76, 6.35, and 7.94 mm. Boiling curves based on the three independent methods for heat flux measurement are shown in Fig. 3. The three measurements at any ΔT agree on average within 15%. The nucleate boiling region is well established in Fig. 3. At the peak heat flux, the three values are 207, 190, and 200 kW/m^2, a difference of 9%. The degree of scatter increased, however, in transition and film boiling because the heat transferred was so low. Here, the worst scatter is 35%. Hence, the effect of heater diameter and velocity on the boiling curve can be identified if greater than 9-35%, depending on the heat flux.

Figure 4 shows all the pool boiling curves as a function of heater diameter. Heater diameter does influence the boiling curve. There exists a critical diameter from which, as the tube size decreases, the heat flux increases. But as boiling behavior is different for the three main boiling regions, the diameter effect must be examined for each region.

Nucleate boiling is not as affected by tube size as the other boiling regimes. At tube diameters larger than 3.18 mm, one curve describes all nucleate boiling. Much larger heat fluxes are found as the tube size decreases below 3.18 mm.

The nucleate boiling curve for the two small tubes do not span into the lower wall temperature regions. The major requirement of this work was that the wall temperature be constant along the 12.7 cm tube length. At the low metal-to-liquid temperatures, where the steam was under a vacuum, the pressure drop along the tube length was significant. This pressure drop affected the tube wall temperature. Whenever the inlet and outlet steam temperatures differed by 10°C or more, data collection was halted. Hence, the nucleate boiling curve, for the two smallest heaters, covered a limited range.

The critical diameter can be stated in terms of bubble size. Jakob and Fritz[8] approximated bubble size by the static equilibrium condition.

$$D_b = (6/\pi)^{1/3}(0.0119\beta)[2\sigma/g(\rho_L-\rho_v)]^{1/2} \tag{2}$$

Corty and Foust[9] measured the contact angle β for Freon-113 to be 50 deg. Using this, the nucleate boiling bubble diameter is 1.04 mm. Thus, the critical tube diameter is approximately 3 bubble diameters for nucleate boiling. The peak heat flux is essentially 190 kW/m^2 for all tubes having diameters of 3.18 mm or larger. The value rises to 250

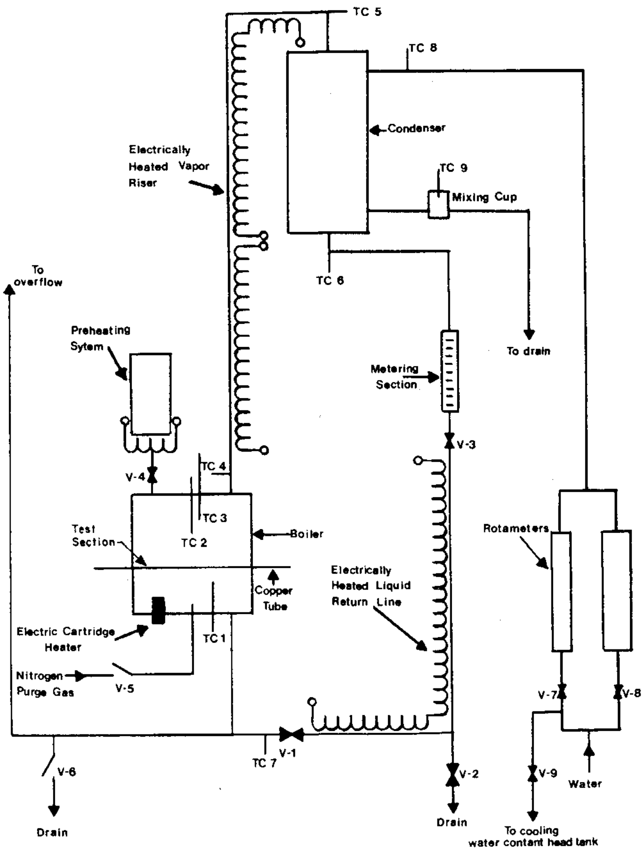

Fig. 1 Pool boiling test system.

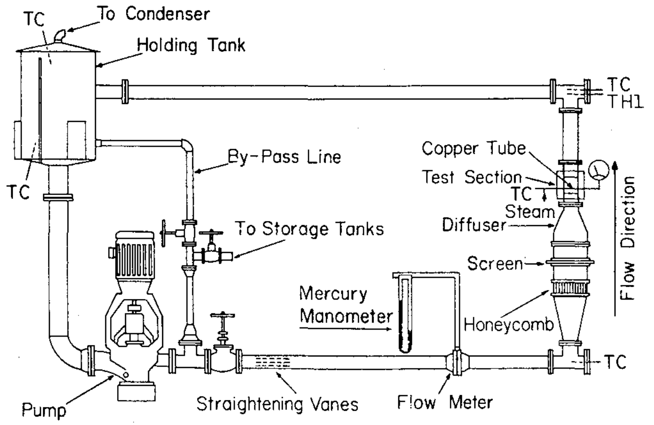

Fig. 2 Flow boiling test system.

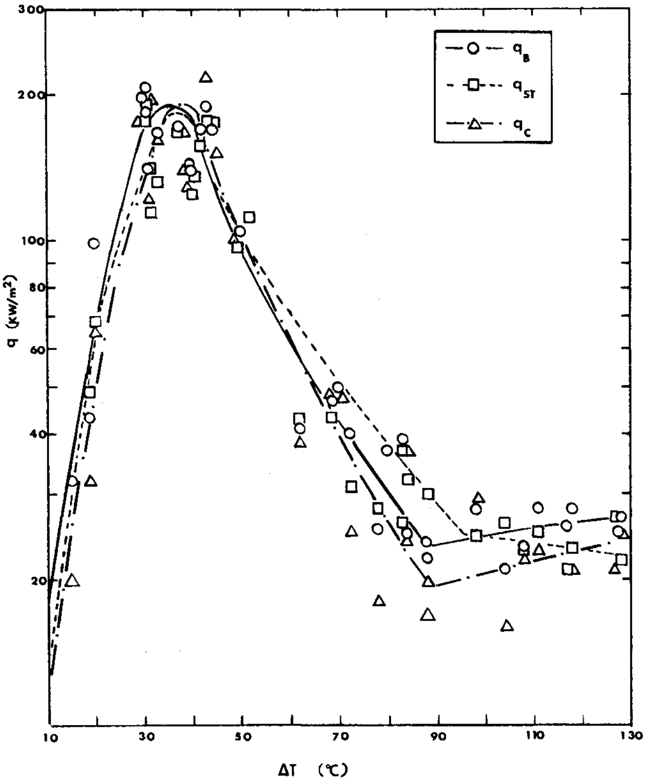

Fig. 3 Pool boiling on 7.94 mm tube with three methods of heat flux measurement.

kW/m² for the 2.38 mm tube and to 385 kW/m² for the 1.59 mm tube.

The observed 190 kW/m² is near the predicted peak value of 207 kW/m² obtained from the Zuber[10] equation for a flat horizontal plate. Lienhard and Dhir[4] extended Zuber's hydrodynamic analysis to cylinders and obtained a limiting ratio $q_{\max}/q_{\text{Zuber}}$ of 0.904. This predicts a peak heat flux of 186 kW/m² at a critical tube diameter of 3.48 mm. The Lienhard-Dhir prediction was that the peak heat flux increases as tube diameter decreases below the critical diameter, in general agreement with the present study.

The transition boiling curves in Fig. 4 show a dependence on tube diameter. The lowest curve is for a diameter of 3.18 mm. As the diameter is either increased or decreased from this value, the heat flux increases at a fixed ΔT. The critical diameter of about 3.18 mm is the same as the critical diameter for nucleate boiling. This lends support to the proposition that some liquid contact followed by bubble nucleation may occur during transition boiling.

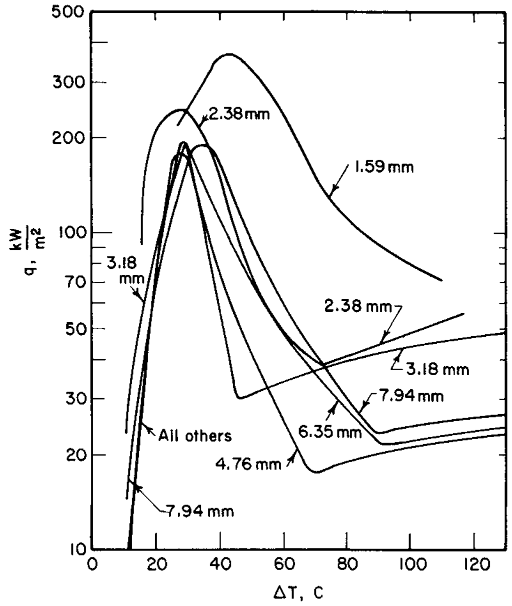

Fig. 4 Pool boiling curves for different size heaters.

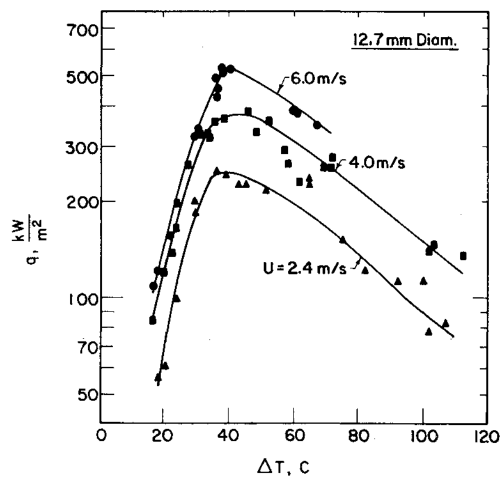

Fig. 5 Velocity effect for 12.7-mm-diam heater.

Although the transition region can be traversed by use of steam-heated tubes, stability is not always guaranteed, Kovalev[11] and Stephan[12] showed that one requirement for stability at any ΔT is

$$\frac{1}{R_w} > -\frac{dq}{dT} \tag{3}$$

where R_w is the resistance of the metal wall plus the resistance of the steam film, and dq/dT is the slope of the boiling curve. For Fig. 3, it was not possible to obtain data for ΔT in the range of 52-60°C, presumably because Eq. (3) was violated. All the other data points for transition boiling were for stable conditions. Stephan[12] noted that stability is most difficult at the inflection point in the transition boiling curve. This seemed to be true for the runs in this study, for example, in Fig. 3 between 52 and 60°C.

Film boiling in a pool is a function of tube size as is evident in Fig. 4. The Bromley[5] equation has the heat flux q proportional to $D^{-1/4}$. Breen and Westwater[6] state that this functionality holds only when the tube diameter is in the range $\lambda_c/D = 0.8$-8. The Taylor instability critical wavelength

λ_c is given by

$$\lambda_c = 2\pi \left[\frac{\sigma}{g(\rho_L - \rho_v)} \right]^{1/2} \tag{4}$$

and its value is 6.29 mm for Freon-113. Thus Bromley's equation should be valid for $D = 0.79$-7.9 mm which includes all diameters in Fig. 4. In Fig. 4, the smallest diameter gave the greatest heat flux as expected, and the next three larger tubes line up in the expected order. However, the three largest tubes are not truly separable by heat flux, because their curves are within the data scatter (such as demonstrated for film boiling on the 7.94 mm tube in Fig. 3). Note that the variables which fix λ_c in Eq. (4) also occur in Eqs. (1) and (2), thus λ_c, D_b, and D^* are closely interrelated.

Flow Boiling

Flow boiling is not a simple extension of pool boiling. Observations show that the addition of velocity disrupts the bubble size distribution. In pool boiling, the heat removal has been explained by bubble dynamics. In flow boiling, convection is important and can dominate the bubble dynamics.

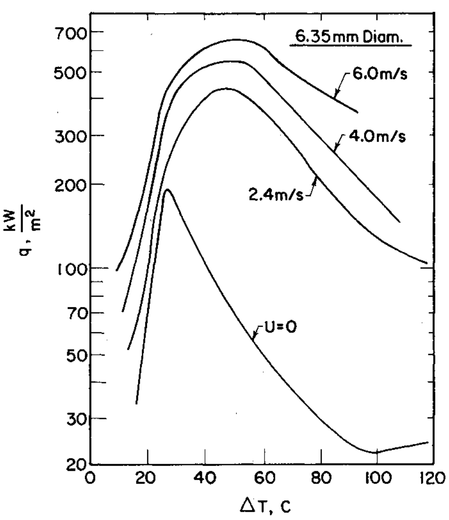

Fig. 6 Velocity effect for 6.35-mm-diam heater.

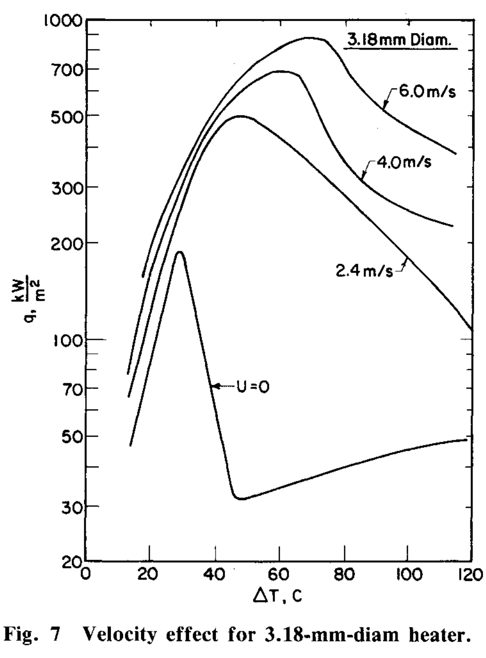

Fig. 7 Velocity effect for 3.18-mm-diam heater.

Three test pieces were used with diameters of 12.7, 6.35, and 3.18 mm, corresponding to $12.7 > D^* > 3.18$. Figures 5-7 show the velocity effect on the boiling curve. Data points are shown in Fig. 5 to demonstrate the accuracy, but are omitted from the other graphs for clarity. Similar to the Yilmaz and Westwater[1] results, there is no overlap or intersection of the curves in Figs. 5-7. Increasing the Freon-113 velocity at a constant steam temperature always resulted in a higher heat flux and a lower wall temperature. At each wall temperature, the boiling rate increased as the fluid velocity increased. As the velocity increased, the peak heat flux required higher wall-to-fluid temperatures. The latter effect is much more evident on the 3.18 mm tube.

The boiling curves cover a wide range of nucleate boiling and transition boiling, but film boiling was not obtained under flow conditions. At higher wall temperatures, film boiling will undoubtedly occur, but the required steam pressure was beyond that available in the present laboratory.

Rohsenow[13] proposed that flow nucleate boiling can be predicted from the pool boiling heat flux by adding a convective heat flux. This correlation

$$q_F = q_p + q_{\text{conv}} \tag{5}$$

is shown in Fig. 8 with the convective heat flux calculated from Fand and Keswani's[14] correlation,

$$Nu_L = (0.255 + 0.699 Re_L^{0.5}) Pr_L^{0.29} \tag{6}$$

Figure 8 shows that Rohsenow's method gives a first-order estimate. Its predicted heat fluxes are consistently somewhat too great. The method cannot be extended beyond the critical ΔT for nucleate pool boiling. During flow, nucleate boiling does extend to higher ΔT values. Hence, this additive approximation is useful in flow boiling at low ΔT values only.

Figures 9-11 show the diameter effect on the flow boiling curve. All three sizes of heaters are plotted at constant superficial velocity U, also called the approach velocity.

Some researchers[15,16] used the unblocked velocity $U_{\max}$ where

$$U_{\max} = \left(\frac{4w}{4w - D} \right) U \tag{7}$$

and w is the front to back channel dimension (3.81 cm).

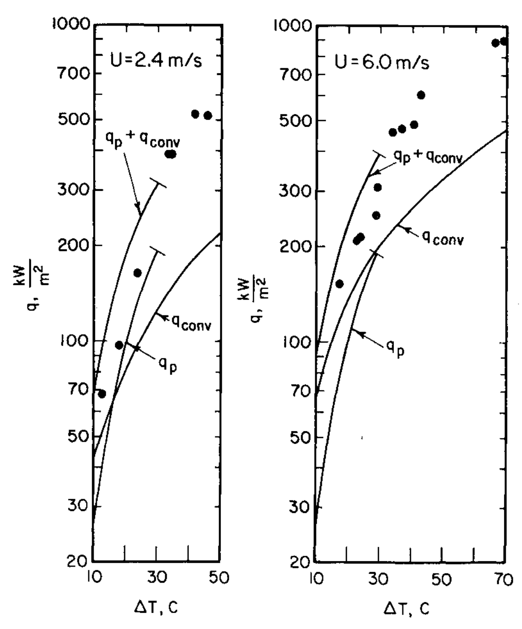

Fig. 8 Nucleate boiling flow data compared to Rohsenow's additive method.

In the nucleate boiling region, the curves are weak functions of diameter. At the same velocity and tube temperature, a large-diameter heater gives a lower heat flux. This difference, however, is difficult to state quantitatively because of overlap of data scatter.

The maximum heat flux is obviously diameter dependent. Higher heat fluxes occur on smaller tubes at the same velocity. This is consistent with other researchers.[17-19] Figure 12 shows that the Weber number,

$$We_v = D\rho_v U^2/\sigma \tag{8}$$

may be used to obtain a least-square correlation with diameter and velocity. Hence,

$$\frac{q_{\max}}{\rho_v \Delta h_{\text{vap}} U} = 0.363\, We_v^{-0.28} \tag{9}$$

based on the data gathered in this study, using the approach velocity.

Figure 12 also shows the Weber number correlation based on maximum velocity of Eq. (7) rather than the approach velocity. The two simple correlations in Fig. 12, based on the two definitions of velocity, seem equally satisfactory. Yilmaz and Westwater[1] correlated their data by a similar equation

$$\frac{q_{\max}}{\rho_v \Delta h_{\text{vap}} U} = 0.369\, We_v^{-0.26} \tag{10}$$

using the approach velocity.

An alternate way of correlating the peak heat flux with diameter and velocity is an equation of Hasan et al.[20] The form shown here corrects a misprint in their paper,

$$\frac{\pi q_{\max}}{\Delta h_{\text{vap}} \rho_L U} = 0.000919 \left[1 + \frac{16.3}{(We_L)^{1/3}} \right] \tag{11}$$

This equation is graphed in Fig. 13. The equation disagrees with the data by 19% on average and by 30% at worst. Equation (9) disagrees with the data by 10.5% on average and by 38% at worst. Equation (11) has the advantage that it is semitheoretical, whereas, Eq. (9) is completely empirical.

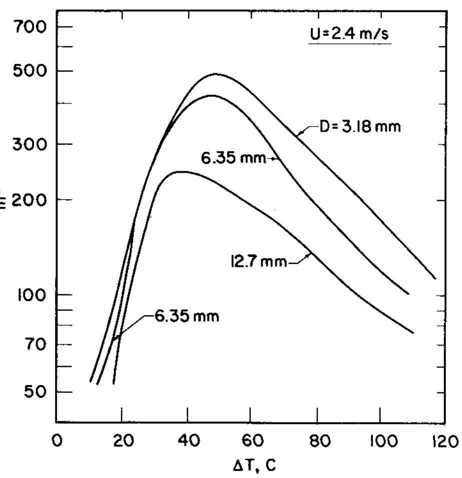

Fig. 9 Heater diameter effect for 2.4 m/s approach velocity.

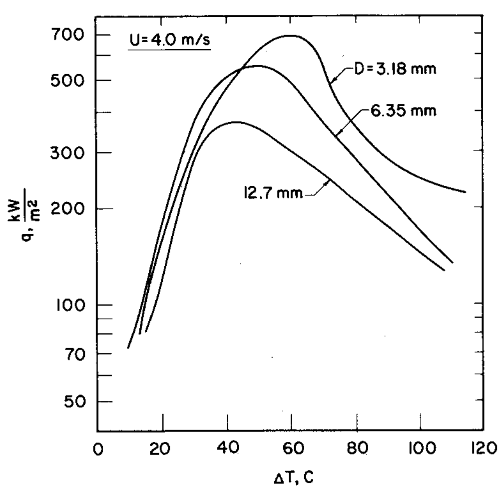

Fig. 10 Heater diameter effect for 4.0 m/s approach velocity.

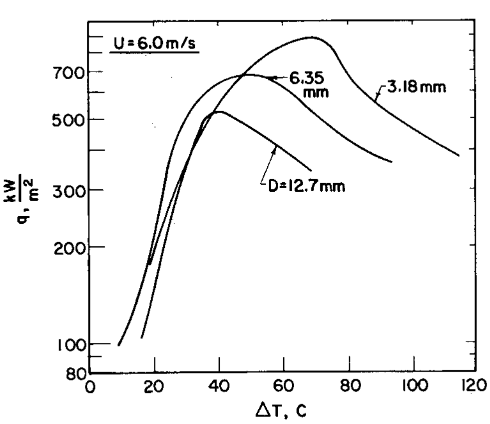

Fig. 11 Heater diameter effect for 6.0 m/s approach velocity.

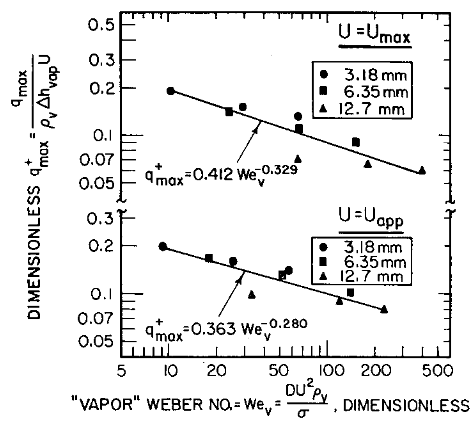

Fig. 12 Simple correlation for peak heat flux using vapor density.

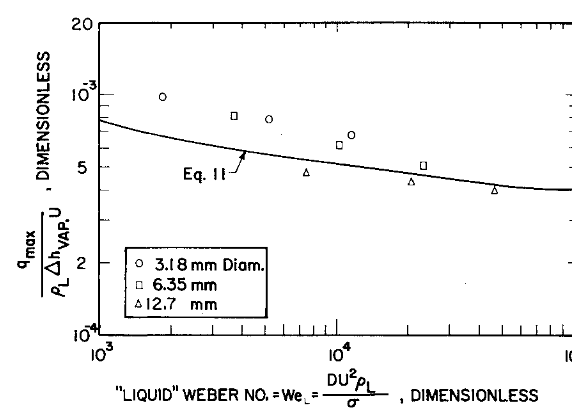

Fig. 13 Hasan et al.[20] correlation for peak flux using liquid density.

Note that Eqs. (9) and (10) and Fig. 12 use vapor density and are therefore highly sensitive to pressure. On the other hand, Eq. (11) and Fig. 13 use the liquid density and are less sensitive to pressure. Future tests with pressure as a variable are needed to demonstrate which density should be used.

Conclusions

1) For flow boiling on a horizontal tube, the heat flux is sensitive to heater size. The dependence is determined by the regime of boiling and the liquid velocity.

2) Nucleate boiling during flow is a weak function of tube size, except in the region of the peak heat flux.

3) The peak heat flux during flow boiling increases approximately as the 0.44 power of the velocity and decreases approximately as the 0.28 power of the tube diameter.

4) Rohsenow's additive method for nucleate flow boiling heat flux is an adequate first-order approximation for metal-to-liquid temperatures below the maximum ΔT for pool nucleate boiling.

5) For pool boiling with no forced flow a critical tube diameter exists, which is equal to approximately 3 bubble diameters for nucleate boiling and transition boiling. The boiling behavior for larger tubes is insensitive to tube size. For smaller tubes, the heat flux increases as the tube diameter decreases.

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