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Pool Boiling Heat Transfer from Vertical Heater Array in Liquid Nitrogen

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The heat transfer from an array of discrete sources is expected to differ from the behavior of a single heat source due to the interaction between the flow induced by individual heat sources. This study details the results from experiments conducted to study the pool boiling heat transfer characteristics from a vertical heater array with flush-mounted heat sources. The lower heaters were found to enhance the heat transfer from upper heaters. The bubble pumped convection due to the lower heaters enhanced the preboiling heat transfer coefficient at the upper heater by as much as 700%. The critical heat flux from the upper heaters was also enhanced up to 15%. Correlations are presented for both these effects.

Nomenclature

C_f = heat capacity of the fluid
g = gravitational acceleration
H_c = distance from center of heater surface to the top of the lower heater
$H_{c,1}$ = the smallest value of H_c used in this study, 1.74×10^{-2} m
h = heat transfer coefficient, $q''/(T_w - T_{sat})$
h_{fg} = latent heat of evaporation
k_f = thermal conductivity of liquid
L = vertical length of the heaters
L_c = distance from bottom edge of heater to the center, $L/2$
Nu_c = Nusselt number at center, hL_c/k_f
$Nu_{f,c}$ = forced convection Nusselt number
$Nu_{n,c}$ = natural convection Nusselt number
q'' = heat flux from the heater under study
q''_{lh} = heat flux from lower heater
Ra^*_c = modified Rayleigh number, $C_f\rho^2 g\beta L_c^4 q''/\mu_f k_f^2$
$Re_{L,c}$ = Reynolds number, $\rho_f v_f L_c/\mu_f$
T_{sat} = saturation temperature
T_w = heater surface temperature
v_f = liquid velocity
W = width of the heaters
We_L = Weber number, $\rho_f L v_f^2/\sigma$
β = volume expansion coefficient
μ_b = fluid viscosity at mean film temperature
μ_0 = fluid viscosity at bulk fluid temperature
ρ_f = fluid density
ρ_g = vapor density
σ = surface tension

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Introduction

THE numerous benefits of operating electronics at low temperatures have been pointed out by various researchers: circuits operate faster, semiconductors switch more rapidly, the number of thermally-induced device failures decrease, and the noise-to-signal ratios drop.[1–4] These advantages are usually associated with temperatures between 10–100 K. It is also in this range that thermal and electrical conductivities of common circuit materials, such as copper and silicon, are maximized.[5] Liquid nitrogen (LN_2) is a relatively inexpensive coolant whose boiling point (77.3 K at 1 atm) falls into this temperature range. As decreasing the size of integrated circuits becomes more difficult and more expensive, some observers predict that low-temperature electronics will become a very attractive option. In fact, they believe that operation at liquid nitrogen temperature (LNT) can improve circuit performance more than decreasing their size by a factor of 2.[6] A reliable method that could maintain LNT in circuits would be of vast importance to the electronics industry. Pool boiling in LN_2 that has a reasonable maximum heat flux removal capacity (1.6×10^5 W/m^2 for horizontal surface under 1 atm, calculated using Zuber's model[7]) would be the easiest to apply in LNT electronic cooling.

Although extensive studies have been conducted on pool boiling of LN_2, they have mostly involved conventional pool boiling situations, i.e., single heat transfer surface in a pool of liquid.[8,9] However, the situation in most electronic equipment consists of a series of discrete heat sources that may interfere with the boiling phenomena on each other. Some researchers have speculated that the bubble layer generated by the presence of numerous heat sources on a vertical plate may create less favorable boiling conditions for the upper heaters.[10] However, no comprehensive study regarding the behavior of a vertical heater array in LN_2 pool boiling could be found. The most likely configuration for electronic cooling in LN_2 is probably a vertical array of chips having different heat dissipations on a circuit board immersed in LN_2. In a case like this the chips on the bottom will give rise to a flowfield that may influence the boiling heat transfer from all the chips above. Hence, the upper chips encounter more of a flow boiling condition instead of pool boiling. Although correlations exist for flow boiling from small-size heaters,[11] the flowfield created on an upper chip due to boiling on a lower chip is difficult to estimate. A number of studies dealing with linear heater arrays under flow boiling conditions exist in literature.[12-14] However, the situation in pool boiling is different due to the fact that the flow velocity is dependent on the heat flux at various heaters, whereas, in these flow boiling studies, the flow velocity could be independently varied. Another point of difference is the distinction between a pool of liquid and a limited channel that exists in these flow boiling studies. The flow in channels is largely along the channel direction, whereas in a pool the liquid is free to flow in any direction.

A few studies dealing with vertical arrays in pool boiling were also found. You et al.[15] studied pool boiling heat transfer in room-temperature gas-saturated FC-72 from a 3 × 3 array of 5- × 5-mm heaters. A confining plate was placed in front of the array at a distance of 2.3 mm. The distance between the heaters was 4 mm in both directions. They found that boiling from a lower heater resulted in a decrease in the superheat overshoot (associated with boiling incipience in highly wetting liquids) at the upper heater. They found no consistent trend in the effect of lower heater on the heat transfer coefficients from the upper heaters. Also, they did not observe any significant effect of the lower heater on the critical heat flux from an upper heater. Another study, by Polentini et al.,[16] involved nucleate boiling investigation from a 3 × 3 array (each heater 12.7 × 12.7 mm) in an enclosure containing FC-72 with two cold plates (maintained at 25°C, 20 K below saturation temperature), one opposite the heater array and one on top. They reported a 15% increase in heat transfer coefficient from the middle heaters when the array was vertically orientated. This was attributed to bubble-pumped convection. However, the uppermost row of heaters had a lower heat transfer coefficient. This was attributed to vapor accumulation near the top of the array. Although they did not study the critical heat flux (CHF) conditions in detail, they reported that the CHF generally occurred at one of the heaters in the lowest row.

These studies used subcooled (room temperature) FC-72. In case of subcooled fluids, preheating of the liquid at the lower heaters may in fact cause a lower heat transfer coefficient at the upper heaters (assuming the reference temperature is the bulk fluid temperature). This will not happen in saturated liquids because the preheating of the liquid will not be significant. On the other hand, there would be little bubble condensation in saturated liquids compared to subcooled liquids. Hence, the vapor flow at the upper heaters, when the lower heaters are generating bubbles, may be significantly higher in saturated liquids. The effect of this excess vapor flow on the upper heaters is not clear. In case of cryogens like LN_2, almost all practical situations will involve saturated liquid. Therefore, the behavior of vertical heater arrays in saturated LN_2 has to be studied.

Another point that should be noted is that, except for the study by You et al.,[15] the other studies dealing with heater arrays only consider equal heat flux from each of the heaters. This situation is rare in electronics as different chips dissipate different amounts of heat.

Thus, the purpose of this research effort was to study the boiling heat transfer from heat sources arranged in a vertical array in a pool of LN_2. The experimental setup was designed to enable individual control of each heater in the array. The main points of interest were the influence of the flowfield created by lower heaters on the heat transfer coefficient and critical heat flux of the upper heaters. The following sections provide a description of the experiments conducted, the results, and correlations for the areas of interest.

Experimental Setup and Procedure

This study involved experiments for pool boiling heat transfer from a 3 × 3 vertical array of heaters. Figure 1 shows the heater array; the heaters are mounted flush to the surface of the plate. The bare chip size in electronics range from 5 × 5 mm to a few hundred mm^2. The CHF in pool boiling from liquid nitrogen is around 1.6×10^5 W/m^2. This heat flux is modest, and therefore, many chips operating at LN_2 temperature will require heat spreaders much like those in use for room temperature operation. Hence, the size of the heaters was chosen close to the commonly used chip sizes; each heater in the array was 21.6 × 17.4 mm (the rectangular shape allowed for two height-width combinations). The heaters have a sandwich construction, shown in Fig. 2. An oxygen-free copper heater block with an E-type thermocouple soldered at the center is soldered on to a ceramic substrate that has a thin resistor film deposited on its other face. Thus, the heat is provided by the film resistor and the temperature at the center of the heater is measured by the E-type thermocouple.

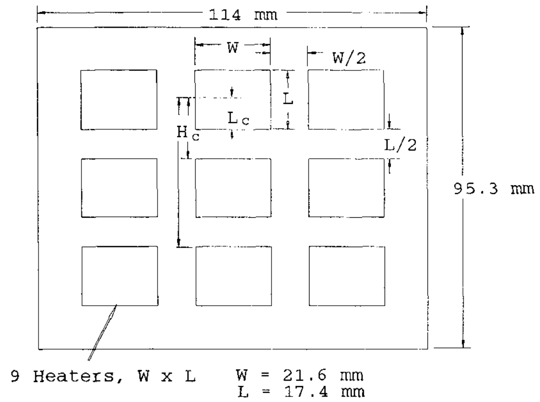

Fig. 1 Heater array.

Nine of these heaters were mounted flush to a thin stainless-steel plate that was bolted on to the heater module casing shown in Fig. 3. The heaters were insulated on the back with closed-cell foam insulation. A low conductivity epoxy provided the seal between the stainless-steel plate and the copper heater blocks. The heater and thermocouple leads pass through an opening in the back of the module, the feed-through hole is thoroughly sealed with closed cell foam to prevent liquid leakage into the module. The power input to each of the heaters can be individually controlled. All data, such as heater temperature and power dissipation, was collected by an IBM PC 386 through a programmed Hewlett Packard 3852A data acquisition/control unit equipped with a 5-1/2 digit voltmeter and a 20-channel relay multiplexer.

Figure 3 shows the details of the experimental setup. Experiments were conducted in a seamless glass (borosilicate) cylinder with an i.d. of 190.5 mm and a height of 254 mm. It contained a pool of LN_2 during the experiment. To prevent heat transfer from the environment, and allow for visualization of the boiling phenomena, this cylinder was placed inside another glass cylinder with an i.d. of 240 mm. During preliminary experiments, it was seen that the cooldown time for the whole system was too long. Hence, a 150-mm-i.d. borosilicate beaker was placed inside the inner chamber, the whole chamber filled with LN_2, and the heater module placed in the beaker. After this adjustment, the liquid pool inside the beaker became stagnant very quickly and this allowed for more efficient operation. The experiments were conducted at atmospheric pressure. The opening on top of the chamber (50 mm in diameter) allowed for liquid refilling, heater, and thermocouple wire feedthrough, and vapor exhaust. The vapor exhaust flow was always enough to prevent any backflow of room air (and moisture) into the chamber. Thus, any ice formation inside the chamber was avoided.

Before collecting each set of data, the heater surfaces were cleaned with dilute hydrochloric acid and deionized water to remove any oxides. The surface roughness of the heaters was measured by a surface profilometer. All the surfaces had an average roughness R_a of around 0.15 μm. Experiments were conducted at varying heat flux distributions. First, a particular heater was selected for study, this was usually one out of the

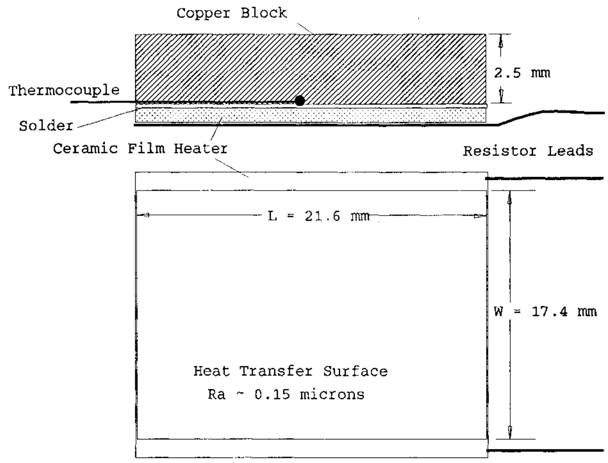

Fig. 2 Heater construction.

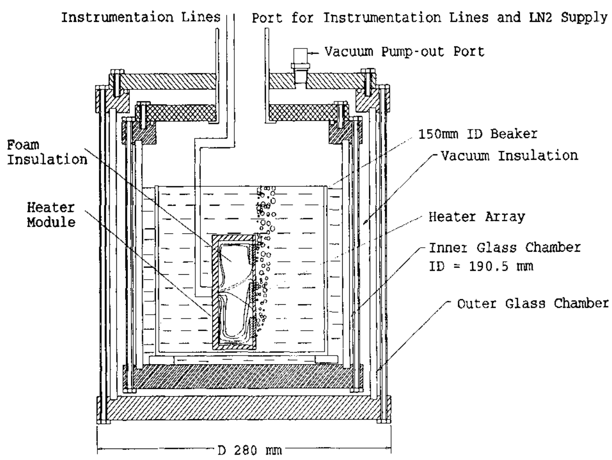

Fig. 3 Experimental setup.

uppermost heaters in the array. The inner glass chamber was filled with LN_2. After the entire setup cooled down to LNT, one or both of the heaters below the selected heater were turned on and maintained at a constant heat flux. After the temperatures of the heaters reached steady state, the heat flux to the designated upper heater was increased. The heat flux was increased again when the heater temperature restabilized. The process was repeated until the variable heater reached CHF. This was indicated by a rapid increase in heater temperature. The power to the heater was shut off immediately on observing this temperature excursion. The boiling data for variable heaters with the other heat sources at 0 W/m^2 were also taken for comparison. After each trial, the pool of liquid nitrogen was replenished to the original level. The decrease in liquid level during the experiment was never more than 2–3 cm.

Since the heater module and the heaters were rectangular, experiments were conducted for both vertical orientations to investigate the influence of interheater distance and heater dimensions. The effects of different environmental conditions, such as varying bubble generation from surrounding heaters, were thus determined. In addition, the reproducibility of the data was determined by repeating some of the experiments. It was found that the heat transfer curve and the CHF were reproducible within the experimental uncertainty limits mentioned in the next section. The only data that was not reproducible was the temperature overshoot (typical of highly wetting liquids) prior to onset of nucleate boiling when only one heater was operating. In addition, the h and CHF data from different heaters under study was comparable when they were operated individually.

Uncertainty Analysis

The heat flux was estimated by measuring the power input to the heater and subtracting the estimated heat loss. The power input was measured by the product of dc voltage (error: $< \pm 0.01\%$) and the current (error: $< \pm 2\%$) across the heater/resistor. The maximum overall uncertainty in power measurement is therefore on the order of $\pm 2\%$. The heat loss from each heater was estimated by insulating the heat transfer surface and determining the $T - T_{\rm sat}$ vs power input curve (while the module is immersed in LN_2). This loss ranged from 300–400 W/m^{2}K (calculated by dividing the power loss by the superheat and the heater area) for different heaters. The error in loss measurement does not cause a significant error in heat flux measurement because of the relatively small value of the loss. Thus, the accuracy of the heat flux estimate depends on the accuracy in power measurement and the accuracy of the heater area measurement ($\pm 0.7\%$). Taking all these into account, the uncertainty in heat flux was less than $\pm 3\%$.

The E-type thermocouples used in the setup have a manufacturer-specified uncertainty of ± 1.7 K at LN_2 temperature. However, since all the thermocouples were made with the same batch of wire, all the thermocouples read within ± 0.2 K when immersed in liquid nitrogen. Therefore, for the purpose of calculating superheat, this is a better representation of the error in temperature.

The temperature T_w was extrapolated using the heater thermocouple reading and the heat flux (based on one-dimensional conduction model). Based on the uncertainties in thermal conductivity, heat flux, and heater thickness, the extrapolation uncertainty is about $\pm 5\%$. However, the calculated temperature drop along the thickness of the heater never exceeds 1 K. This gives the maximum uncertainty in extrapolation as ± 0.05 K. The maximum uncertainty in superheat $(T_w - T_{\rm sat})$ then comes to ± 0.45 K.

In the CHF region, the power increments to the heater were decreased to 2.5×10^3 W/m^2. Thus, the uncertainty in CHF is equal to this value plus the uncertainty in heat flux estimate. This gives a total uncertainty of less than 5% of the CHF.

The heat transfer coefficient h is the ratio of the heat flux and the superheat. The error in h is therefore dependent on the superheat and the uncertainty in heat flux measurement. The maximum relative error in h can then be written as: $0.03 + 0.45/(T_w - T_{\rm sat})$.

Results and Discussion

Preliminary results revealed two important differences between the boiling curves of heaters in an array and solitary heaters. First, for the upper heaters in an array, the critical heat flux increased as the heat fluxes of the lower heaters increased. Secondly, the coefficient of heat transfer h, during the preboiling regime increased dramatically as the heat flux to the lower heaters rose. The heaters not directly below a heater were not seen to have any significant effect on heat transfer characteristics. The heater just below the one under study was always the most influential. When any top heater in the array was studied with both of the lower heaters operating at a particular heat flux, the heat transfer characteristics were not significantly different from those with the lowest heater off. Hence, the following discussion involves only the effect of one heater operating below the heater under study. The experimental setup thus allowed for four interheater distances and two heater lengths using both vertical orientations of the array.

Figure 4 shows the pool boiling characteristics of a heater with and without a lower heater operating at a fixed heat flux. A little bit of temperature overshoot, typical of highly wetting fluids like LN_2, is seen for the solitary heater case. As seen from the figure, the preboiling heat transfer coefficient was as much as 700% higher when the lower heater was in operation. Similarly, the CHF was also enhanced from 1.44 to

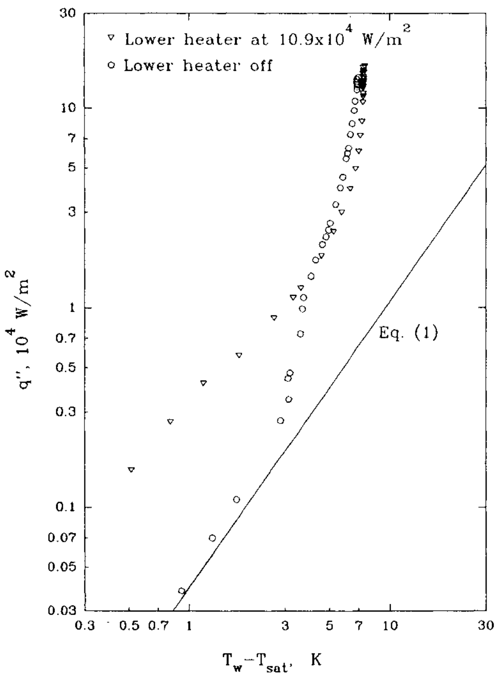

Fig. 4 Heat transfer characteristics of upper heater.

1.64×10^5 W/m^2. The nucleate boiling portion of the curves are within experimental uncertainty limits. Hence, this study concentrated on characterizing the enhancement in the preboiling regime and the CHF.

The influence of the lower heater on the heat transfer coefficient prior to the onset of nucleate boiling is very pronounced. This is due to the intense convection caused by the bubble flow from the lower heater. Figure 5 shows the effect of the lower heater heat flux q''_{lh}, on the preboiling heat transfer coefficient of the upper heater. As the heat flux to the lower heater increases, the value of h increases very rapidly, however, past 2.5×10^4 W/m^2, increasing heat flux from the lower heater causes small increase in the heat transfer coefficient. Also, the enhancement in h decreases as the distance between the upper and lower heater is increased.

The heat transfer in the preboiling regime is due to mixed convection, the bubble-flow-induced convection is assisted by natural convection. Hence, the heat transfer correlation for this regime has to incorporate both of these contributions:

$$Nu_{n,c} = \alpha Ra_c^{*\delta}$$

where

$$\alpha = 0.906\left[1 + \frac{0.0111}{(W/W_\infty)^{3.965}}\right]^{0.2745}$$

$$\delta = 0.184\left[1 + \frac{2.64 \times 10^{-5}}{(W/W_\infty)^{9.248}}\right]^{-0.0362} \tag{1}$$

$$W_\infty = 70 \text{ mm}$$

Numerous correlations for natural convection from a finite vertical plate exist.[5-17] The best match was found to be the correlation of Park and Bergles,[17] shown here as Eq. (1). Although Eq. (1) was proposed for small vertical heaters in R-113 (Freon-113), and the Ra_c^* in the present study exceeded the range of this correlation, it had a reasonable agreement with the LN_2 data, as shown in Fig. 4.

For a surface under mixed convection conditions, Churchill[18] correlated the mean Nusselt number at center of the surface using the equation

$$Nu_c = (Nu_{n,c}^3 + Nu_{f,c}^3)^{1/3} \tag{2}$$

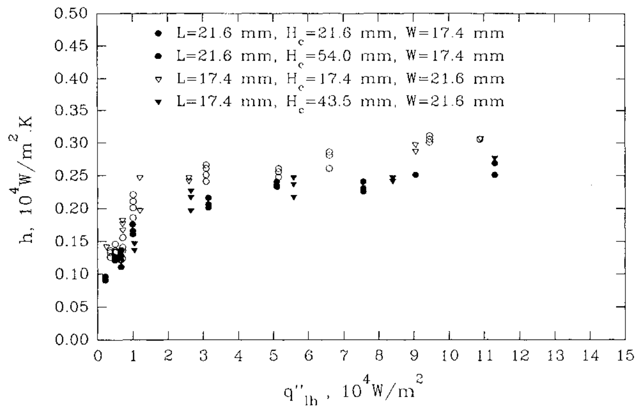

Fig. 5 Effect of lower heater on *h*.

A similar approach was used in the present study. The natural convection Nusselt number was estimated using Eq. (1), however, the forced convection Nusselt number poses a problem. The flow velocity due to the bubbles is difficult to obtain analytically. Hence, an empirical approach had to be used. Incropera et al.[19] proposed a general correlation [Eq. (3)], for small-flush heaters under forced convection. The value of C and p vary depending on heater geometry and fluid:

$$Nu_{f,c} = C Re_{L_c}^{p} Pr^{0.38}(\mu_0/\mu_h)^{0.11} \tag{3}$$

This equation [Eq. (3)] requires a knowledge of the Reynolds number. One way to obtain this information is through the measurement of fluid velocity due to the bubble flow. This, however, was not possible because the presence of large number of bubbles interferes with optical measurement techniques. Another way of obtaining the fluid velocity is by relating it to the vapor velocity.

The fluid velocity due to the bubble pumped flow can be assumed to be function of the vapor velocity and some fluid properties. With this assumption, we can write the fluid velocity over the heater as Eq. (4). The first term in Eq. (4) represents the vapor velocity from the

$$v_f = C_1\left(\frac{q''_{lh}}{\rho_g h_{fg}}\right)\left(\frac{H_c}{H_{c,1}}\right)^n \tag{4}$$

where

$$H_{c,1} = 1.74 \times 10^{-2} \text{ m}$$

lower heater. The second term, the nondimensional distance between the center of the upper heater and the edge of the lower heater, can be termed as dissipation factor. This represents the effect on vapor velocity as the bubble source is moved lower. This effect takes place due to the lateral spread of bubble flow as it moves up in a liquid. The main parameter that influences the lateral spreading of the bubble flow is the vertical distance from the bubble source H_c. This effect is similar to the boundary-layer growth along a plate. Here, the distance between the lower bubble source and the center of the upper heater H_c is nondimensionalized by dividing by the lowest value of H_c used in this study, termed as $H_{c,1}$ (which is equal to 1.74×10^{-2} m).

The width of the heaters would be important if the edge effects were significant. From visual observations, the maximum lateral bubble-plume spreading was approximately 1 mm (both parallel to and perpendicular to the heater surface). Thus, the spreading parallel to the heater surface is very small compared to the total width of the bubble layer (equal to the width of the heater: 17.4 or 21.6 mm). Hence, the effect of changing the heater width is very small. The spreading per-

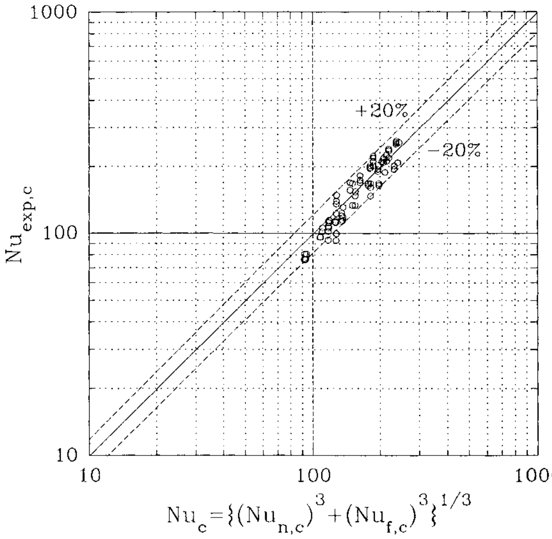

Fig. 6 Preboiling *h* correlation.

pendicular to the surface is more important because the bubble-layer thickness in that direction was on the order of a few millimeters.

The constant C_1 and exponent n in Eq. (4) have to be determined empirically. Equations (3) and (4) can be combined and written as Eq. (5). The constants C_1 and C are combined into

$$Nu_{f,c} = C_2 \left[\frac{\rho_f(q''_{lh}/\rho_g h_{fg})(H_c/H_{c,1})^n L}{\mu_f}\right]^p Pr^{0.38} \tag{5}$$

C_2 and the last term in Eq. (3) is dropped as it approaches 1.0 for saturated liquid. A best fit correlation to the data was obtained. The best fit values of C_2, n, and p were: $C_2 = 21.50$, $n = -0.62$, and $p = 0.24$. Combining Eqs. (1), (2), and (5) we can write the overall Nusselt number at the midpoint of the heater surface as Eq. (6).

Figure 6 shows the comparison of Eq. (6) with the experimental data. As seen from the figure, most of the data falls within ±20% of the correlation. Although this is a significant spread, it can be explained by the fact that the experimental error in the preboiling regime can be fairly large due to the very low values of $T_w - T_{sat}$. The uncertainty in temperature difference

$$Nu_c^3 = (\alpha Ra_c^{*\delta})^3 + \left\{24.3\left[\frac{\rho_f(q''_{lh}/\rho_g h_{fg})(H_c/H_{c,1})^{-0.71}L}{\mu_f}\right]^{0.21} Pr^{0.38}\right\}^3 \tag{6}$$

was about 0.45 K; at a surface superheat of 2 K, this translates into a 23% error in $T_w - T_{sat}$ and a 26% error in h. Therefore, the correlation shows a good fit to the data considering the experimental error.

The other point of interest in this study was the CHF region. As mentioned earlier, the flow caused by the lower heater elevates the CHF from the upper heater. Figure 7 shows the effect of lower heater heat flux on the CHF, here CHF_a/CHF_0 represents the ratio of CHF with the lower heater on to the CHF for the solitary heater. As shown in the figure, the CHF shows an increase as the heat flux from the lower heater is increased. When the heat flux from the lower heater rises higher than about 3.0×10^4 W/m^2, increasing the heat flux of the lower heater causes lesser increase in CHF. Overall, the maximum enhancement due to the presence of the lower heater was around 15%. This enhancement also reduced as

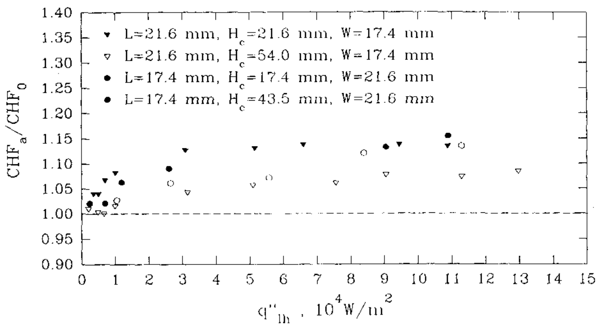

Fig. 7 Effect of lower heater on CHF.

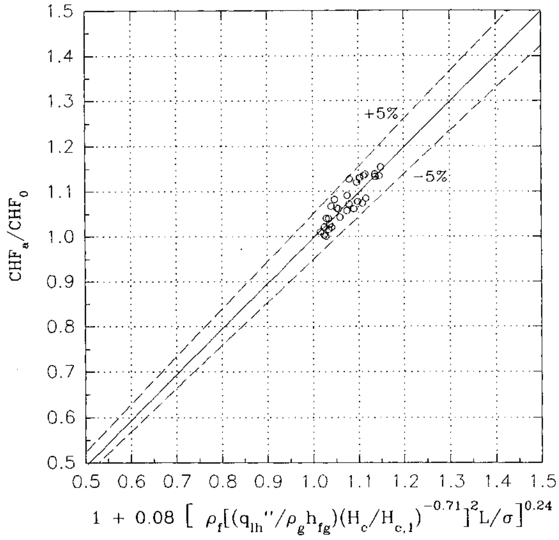

Fig. 8 CHF correlation.

the distance between the lower and the upper heater increased. The values for critical heat flux from a solitary heater were found to correspond almost exactly with the asymptotic value from Lienhard and Dhir's correlation,[20] Eq. (7) (note that the heater dimensions in this study exceeded the very small heater limit of $L' < 6.0$).

The CHF in flow boiling conditions generally correlates well to the Weber number of the flow.[21] Hence, we can write the CHF ratio as Eq. (8). Using a similar approach to that used in writing the Reynolds number, the Weber number can be written as Eq. (9). Then by combining Eqs. (8) and (9), and finding the best fit to the data, we can write the CHF ratio

$$CHF_0 = 0.9 CHF_{Zuber}$$

for

$$L' = \left[L \Big/ \sqrt{\frac{\sigma}{g(\rho_f - \rho_g)}}\right] > 6.0 \tag{7}$$

where

$$CHF_{Zuber} = 0.131 \rho_g h_{fg} \left[\frac{\sigma(\rho_f - \rho_g)g}{\rho_g^2}\right]^{1/4}$$

$$CHF_a/CHF_0 = 1 + C_2 We_L^r \tag{8}$$

$$We_L = \left\{\frac{\rho_f[C_1(q''_{lh}/\rho_g h_{fg})(H_c/H_{c,1})^{-0.71}]^2 L}{\sigma}\right\} \tag{9}$$

as Eq. (10). Figure 8 shows the comparison of this correlation with the data. As shown in the figure, the data fits the correlation to within ±5%:

$$\frac{\mathrm{CHF}_a}{\mathrm{CHF}_0} = 1 + 0.08 \left\{ \frac{\rho_l[(q''_{\mathrm{lh}}/\rho_g h_{fg})(H_c/H_{c,1})^{-0.71}]^2 L}{\sigma} \right\}^{0.24} \tag{10}$$

Conclusions

This study found that the presence of lower heaters increases both the critical heat flux and the preboiling heat transfer coefficient from an upper heater. Only the closest operating lower heater was found to have significant effect on the heat transfer characteristics of the upper heater. CHF was enhanced by as much as 15%, and h by as much as 700% when a lower heater was operating at some fixed heat flux. This enhancement is attributed to the forced convection caused by the vapor generated from the lower heater. The amount of enhancement generally decreased as the distance between the lower heater and upper heater increased. A correlation based on mixed convection was found for the enhancement of heat transfer coefficient. This correlation gives the overall Nusselt number in terms of the natural convection Nusselt number and a forced convection Nusselt number derived from the vapor flux from lower heater. Another correlation relating the enhancement in CHF to the Weber number was also obtained. Both of the above-mentioned correlations fit the data reasonably well.

Based on this study, it can be seen that in saturated pool boiling, the heat transfer from upper heaters is not degraded due to the vapor flow from the lower heaters. In fact, the effect is beneficial to the upper heaters. However, when a limited space exists in front of the heater array (as in a channel), this effect could indeed be detrimental due to insufficient space for vapor-escape/liquid-replenishment. There would be an optimum spacing in that case, beyond which the channel gap should not be reduced. This parameter is of great interest in electronic cooling and should be studied in more detail.

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