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Controlling bubble motion over heated surface through evaporation momentum force to enhance pool boiling heat transfer

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Evaporation momentum force arises due to the difference in liquid and vapor densities at an evaporating interface. The resulting rapid interface motion increases the microconvection heat transfer around a nucleating bubble in pool boiling. Microstructure features are developed on the basis of this hypothesis to control the bubble trajectory for (i) enhancing the heat transfer coefficient, and (ii) creating separate liquid and vapor pathways that result in an increased critical heat flux (CHF). An eightfold higher heat transfer coefficient (629 000 W/m²°C) and two-and-half times higher CHF (3 MW/m²) over a plain copper surface were achieved with water.

Pool boiling is characterized by bubbles nucleating over cavities on a heated surface. The bubbles undergo an ebullition cycle in which nucleation is followed by bubble growth, departure, and a waiting period. It is widely accepted that the interface motion and the ebullition cycle lead to heat transfer contributions from the following three mechanisms:

(i) Transient conduction: This mechanism was identified by Han and Griffith and analyzed by Mikic and Rohsenow. As a bubble departs from a heater surface, it is replaced by the bulk liquid. Heat is transferred by transient conduction to the liquid. It increases with higher bubble frequency, and decreases as the departure bubble diameter decreases. Overall, this contribution increases modestly as the heater surface temperature increases at higher heat fluxes.

(ii) Microconvection: The rapid movement of the bubble interface induces liquid motion. The resulting microconvection heat transfer contribution increases as the interface velocity increases. This is the most important contribution which increases with heat flux.

(iii) Microlayer evaporation: This mechanism results from the evaporation of a thin liquid layer underneath a bubble. It is relatively small and stays relatively constant even as the total heat flux increases.

Heat transfer enhancement in pool boiling is a very active research topic. Among the passive techniques employed, efforts have been focused on fins, increased nucleation site density, artificial re-entrant cavities, combination of pores and tunnels, and hydrophobic and hydrophilic nanostructures to enhance heat transfer and the critical heat flux (CHF). Fins are effective when the heat transfer coefficient is relatively low. The design of pores and tunnels in re-entrant cavities was based on a simple mechanistic description—“liquid is sucked in through the openings, thin film evaporation occurs over the large interior extended surface, and the cycle is repeated.”

High heat fluxes can be achieved with the combination of fins and other enhanced structures, but the wall superheat is rather large for long fins due to conduction resistance. For example, a heat flux of 5 MW/m² leads to a temperature drop of 37.5 °C over 3-mm length in a copper rod with a thermal conductivity of 400 W/m°C.

Recent efforts toward increasing the CHF have focused on supplying liquid to the nucleation cavities. Liquid blockage by the escaping vapor is avoided through the use of nanowires that enable liquid flow by capillary forces. Combining the beneficial effects of hydrophilic surfaces in enhancing the CHF and hydrophobic surfaces in improving the heat transfer coefficient, micro-nano hierarchical surfaces have been developed with hydrophilic patches on a hydrophobic substrate, or a reverse configuration. CHF of 200–250 W/m² was obtained at wall superheats of 20–25 °C.

Figure 1 shows the results of some of the best performing surfaces with water reported in the literature. Mori and Okuyama used a honeycomb structured porous plate. Chen et al. and Yao and Kandlikar used copper nanowires, while Yang et al. employed an open foam cover. Li and Peterson used a sintered wire mesh. Chen et al. demonstrated the ability of nanowires to dissipate high heat fluxes at relatively low wall superheats. Cooke and Kandlikar used open rectangular microchannels to dissipate 244 W/cm² with less than 10 °C wall superheat, resulting in one of the

![FIG. 1. Pool boiling curves for water on enhanced surfaces reported in the literature.](image-url)
highest reported heat transfer coefficients for pool boiling with water of 269 000 W/m²°C until now.

Liter and Kaviany used a modulated porous coating to provide stable columns for vapor outflow through conical microstructures. This resulted in a three-fold increase in CHF for pentane with a reduced wall superheat of 22°C. The apexes of the conical structures brought the liquid to the base by capillary forces, and the vapor escaped from the tapered surfaces. This work clearly demonstrates that understanding the underlying mechanism is essential in overcoming the limits in pool boiling.

In the current design of enhancement features, the evaporation momentum force is utilized to increase the microconvection heat transfer, and provide separate pathways for liquid and vapor flow. Evaporation at the liquid-vapor interface of a bubble results in a force due to the momentum of the vapor phase leaving the interface. This force pulls the interface and the contact line into the liquid and the resulting motion generates microconvection currents. As this force exceeds the retaining force due to surface tension, the bubble base expands in all directions due to symmetry and leads to CHF.

Figure 2(a) shows a growing bubble on a heater surface. The evaporation momentum forces acting in the x-direction on its two sides are shown. Since the thermal conditions in the liquid surrounding the two sides are the same, the evaporation momentum forces in the x-direction, \( F'_{M_1,x} \) and \( F'_{M_2,x} \), cancel each other, and bubble departure is governed by the forces in the vertical direction. For a given heat flux \( q''_I \) per unit surface area of the interface, the evaporation momentum force per unit length, \( F'_{M_1,x} \) over a bubble height \( H \) is given by

\[
F'_{M_1,x} = \frac{q''_I \times H \times 1}{h_{fg} \times \rho_g} \left( \frac{q''_I}{h_{fg} \times \rho_g} \right)^2 1 \quad \frac{1}{\rho_g} H, \tag{1}
\]

where \( q''_I \) is the heat flux at the interface, \( h_{fg} \) is the latent heat of vaporization, and \( \rho_g \) is the vapor density. A 1-D assumption is employed by considering a unit width of the interface resulting in the projected interface area of \( H \times 1 \) as analyzed by Mikic and Rohsenow. Figure 2(b) shows the two interfaces subjected to different thermal environments in the liquid. Assuming the temperature \( T_1 > T_2 \), the evaporation rate and the heat flux on the right side will be higher, and \( F'_{M_1,x} > F'_{M_2,x} \). This will result in a net force on the bubble in the positive x direction. At microscopic scale, the evaporation momentum force becomes dominant over the gravitational and inertia forces. Since the evaporation momentum force increases in the square proportion of the heat flux, the net force, \( F'_{M_{Net,x}} = F'_{M_1,x} - F'_{M_2,x} \), becomes quite large and dictates the bubble motion at higher heat fluxes.

Figure 2(c) shows a bubble growing in a corner at the fin base. The trajectory will be at 14.1°. Thus, the bubble will follow a path away from the fin surface sweeping over the heated surface as depicted in Fig. 3. The effect of gravitational force is negligible (for water at atmospheric pressure with a bubble diameter of 50 μm, \( F'_{M_{1,x}} \) is estimated to be 21.4 mN/m, while the vertical force due to buoyancy is 2.6 mN/m at a heat flux of 3 MW/m²). Note that the interface heat flux is

FIG. 2. Schematic showing the evaporation momentum forces in the x-direction. (a) opposite interfaces exposed to symmetrical thermal conditions, (b) opposite interfaces exposed to different thermal conditions, (c) bubble growing in a corner at a fin base.

FIG. 3. Enhanced surface design using evaporation momentum force to control bubble trajectory and create separate liquid and vapor pathways.
considerably higher than the surface heat flux as explained by Kandlikar\textsuperscript{16}). Since the bubbles are small, around 10–100 $\mu$m in diameter during the initial growth process, the surface tension forces are dominant and the bubbles may be treated as rigid spheres as seen from the numerical results of the bubble growth process.\textsuperscript{18} The forces acting on the interface thus lead to the motion of the entire bubble.

The induced bubble motion is further utilized to generate preferential liquid pathways. Figure 3 shows a schematic of one such configuration. A small fin structure is created by indenting the heater surface. The fin base makes a sharp corner where nucleation occurs preferentially. The nucleating bubble moves along the fin base as discussed above and causes the bulk liquid to flow over the fins toward the nucleation sites. Providing symmetric indentations on the two sides provides a wider region for liquid flow over the fin. The liquid and vapor flow pathways become more distinct at higher heat fluxes as the bubbles leave at higher velocities along the floor. This facilitates vapor flow without blocking the liquid pathway, so higher CHF can be achieved. The configuration shown in Fig. 3 thus improves both, (i) heat transfer rate due to increased microconvection, and (ii) CHF due to efficient vapor removal from and liquid supply to the nucleation sites.

The profile shown in Fig. 3 was applied on copper test chips for pool boiling tests in a rectangular element shown in Fig. 4. A confocal laser scanning microscope image of the element consisting of a central fin surrounded by channels is shown in Fig. 4(a). A cross-sectional view of the channel profile is shown in Fig. 4(b). This profile is similar to that depicted in Fig. 3 (left half). The depth of the groove was 200 $\mu$m and the corner angle was 60°. This pattern was implemented in the central 10-mm $\times$ 10-mm region of two 20-mm $\times$ 20-mm copper chips. Chip 1 had 8 rectangular elements similar to Fig. 4(a), while chip 2 had 7 elements. An embossing process was used to create these features. First, a tool with mirror impression of the pattern shown in Fig. 4(a) was prepared. It was then used in an embossing attachment to a Bridgeport milling machine. The traverse of the embossing attachment was controlled in small increments to generate the surface features with desired depths. The chips were then cleaned with an air jet to remove the burrs and further cleaned with isopropyl alcohol.

The copper chips were tested to obtain the pool boiling curve in a setup similar to that described by Cooke and Kandlikar.\textsuperscript{13} Experimental uncertainty was 4% in the heat flux and 0.1 °C in the wall superheat. The area augmentation due to the surface features is less than 18%. Further details of the chips and the experimental setup and the chips are given in the supplemental material.\textsuperscript{19} Degassed water at atmospheric pressure was used as the test fluid.

The corners at the base of the fin and the floor have multiple nucleation cavities. Cavities of diameter 5–30 $\mu$m were observed in this region using a laser confocal microscope. According to Hsu’s\textsuperscript{20} criterion over a flat surface, nucleation is initiated over these cavities at wall superheats of 1–2 °C. The corner region is expected to further enhance nucleation due to localized heating of liquid from the fin.

Figure 5 shows the pool boiling curve for a plain chip and the two tested chips. The heat flux (based on the projected active area of 100 mm$^2$) is plotted as a function of wall superheat for a plain copper chip, and the two chips with enhancement features. The plain copper chip reached a CHF of 1.20 MW/m$^2$ at wall superheat of 17 °C. Chip 1 reached a heat flux of 3 MW/m$^2$ at wall superheat of 4.8 °C. Chip 2 dissipated 2.5 MW/m$^2$ at a wall superheat of 5 °C. It reached a higher heat flux of 2.8 MW/m$^2$, but the wall temperature increased to 6.2 °C before CHF was attained.

![FIG. 5. Saturated pool boiling curves for a plain copper surface, and chips 1 and 2 with water at atmospheric pressure.](image-url)
The heat transfer mechanisms can be better explained by looking at the heat transfer coefficient plots shown in Fig. 6. The heat transfer coefficient is seen to increase with heat flux. Chip 1 reached the highest recorded value of heat transfer coefficient of 629 000 W/m$^2$·C at a heat flux of 300 W/m$^2$. Since the area enhancement is only 18%, the 12.5 fold increase in heat transfer coefficient is believed to be mainly caused by separate liquid-vapor pathways and the enhancement in the heat transfer mechanisms.

Since the area augmentation is only 18%, while the heat flux enhancement is over 2.5 fold, it may be concluded that the area augmentation is not the driving mechanism. The enhancement in heat transfer is a result of the higher micro-convection and transient conduction heat transfer rates caused by the evaporation momentum force. To verify this hypothesis, high speed visualization is performed to investigate the bubble motion in such configurations.

A single fin of 1-mm height and 2-mm thickness was machined on a copper substrate. Nucleation cavities of 5–30 μm were made by indenting a sharpened needle in the corner. The surface was then tested in a pool boiling apparatus. The images of nucleating bubbles were recorded with a Keyence high-speed camera with microscope attachment. Figure 7 shows the video sequences obtained at 1000 frames per second. Each frame is obtained at an exposure of 1/2000 s.

Fig. 7(a) shows the location of the fin on the copper substrate with the nucleation cavity identified as “A.” This geometry simulated the bubble nucleation and growth at the base of the internal fin shown in Figs. 3 and 4(b). The substrate is heated to a wall superheat of 4°C. The camera is aimed into the corner looking down at about 20° angle. A bubble is seen to nucleate at the base of the fin in frame (b) at 0 ms. The bubble grows rapidly in frame (b) and the interface is blurred even at an exposure setting of 1/2000 s. In frame (c), the bubble grows to the right. In frames (d) and (e), the bubble departs along the substrate in the horizontal direction. The interface was not pinned on any surfaces or edges on the substrate or fin. It has been observed that as the surface temperature increased, the bubble diameter decreased and the bubble activity became too fast for obtaining clear video images. At higher heat fluxes, the rapid bubble departure following the substrate trajectory provides the liquid and vapor pathways as depicted in Fig. 4. It is hypothesized that the bubble motion seen in Fig. 4 is the result of the differences in the evaporation momentum forces on the two sides, and this motion is responsible for the bubble trajectory and the resulting heat transfer enhancement in the test device. This notion is in agreement with the visual observations and numerical simulations made on an evaporating interface.21,22 Further research on the effect of evaporation momentum force on interface motion and heat transfer is warranted.

The pool boiling performance of the chips may be further improved by optimizing the shape of the floor in Fig. 2 to reduce the flow resistance, and through additional nucleation sites at the corners. Nucleation may not be desirable on other parts of the channel surfaces as it will disrupt the flow streams established by the bubbles. Some of the variables which govern the performance are (i) height and thickness of the fin affecting the fin temperature profile, and (ii) channel width and shape, affecting the flow resistance of the bubble and contribution from the microconvection mechanism.

Heat transfer can be also be enhanced by introducing micro-fins, such as pin-fins or offset strip-fins (of dimensions as small as 10 μm) along the liquid and vapor flow paths. However, this may have an adverse effect on the flow resistance. Future development involves incorporation of

![FIG. 6. Heat transfer coefficient comparison during saturated pool boiling of water over a plain surface, and two enhanced surface chips.](image-url)
nanostructures on different regions of the surface. This will improve the liquid flow to the nucleation sites and keep the surface wetted and aid in further enhancing the CHF.

Evaporation momentum force becomes the dominant force on a bubble interface at high heat fluxes during pool boiling. It is hypothesized in this work that this force can be utilized to enhance heat transfer and critical heat flux by directing the interface motion over the heated surface and creating separate liquid-vapor pathways. A surface structure designed on the basis of this model resulted in significant heat transfer coefficient and CHF enhancements. A record heat transfer coefficient of 629 000 W/m²°C at a CHF of 3 MW/m² has been attained with water using this enhancement technique on a copper surface.

19See supplementary material at http://dx.doi.org/10.1063/1.4791682 for test section description.