Liquid Immersion Cooling of a Longitudinal Array of Discrete Heat Sources in Protruding Substrates: II—Forced Convection Boiling

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Forced convection boiling experiments have been performed for an in-line 1 × 10 array of discrete heat sources, flush mounted to protruding substrates located on the bottom wall of a horizontal flow channel. FC-72, a dielectric fluorocarbon liquid, was used as the heat transfer fluid, and the experiments covered a range of flow velocities, degrees of fluid subcooling, and channel heights. The maximum heater-to-heater surface temperature variation was less than 2.5°C and was insensitive to channel height under conditions of fully developed nucleate boiling. Although the fluid velocity influenced the heat flux for partially developed nucleate boiling, its influence was muted for fully developed nucleate boiling. The heat flux associated with a departure from nucleate boiling increased with increasing velocity, subcooling, and channel height; however, the effect of channel height was only significant when all of the upstream heaters were energized.

Introduction

Continued advances in the speed and performance of microelectronic devices will necessitate the development of techniques to cope with the large heat removal requirements imposed by increased circuit concentration at the computer chip level and increased chip densities in multi-chip modules. The development of adequate cooling schemes is rendered particularly challenging by the requirement that chip surface temperatures be maintained below 85°C to ensure reliable operation (Simons, 1983). Indirect liquid cooling schemes have been employed with great success in machines such as the IBM 308X/3090 series (Chu et al., 1982) and the Fujitsu FACOM M-780 (Yamamoto et al., 1987). In the Cray-2 supercomputer (Danielson et al., 1986) thermal management is accomplished under single phase mixed convection conditions by direct immersion of the circuitry in FC-77 (a dielectric fluorocarbon liquid manufactured by 3M Co.). Direct liquid immersion offers much higher convection coefficients than conventional air cooling or indirect liquid cooling with similar pumping power requirements. Boiling heat transfer exhibits even higher heat transfer coefficients than single phase convection and provides an added advantage of dissipating large heat fluxes with only slight variations in the surface temperature of the heated elements.

The objective of the present work is to study forced convection boiling heat transfer from a 1 × 10 linear array of small flat heat sources in a horizontal flow channel, each heat source being flush mounted to a protruding substrate attached to the bottom wall of the channel. This arrangement resembles a possible configuration for an actual electronic package, with the heat sources playing the role of Very Large Scale Integrated (VLSI) chips and the protrusions resembling chip carriers. The dielectric liquid FC-72 (a fluorocarbon manufactured by 3M Co.) was used as the coolant because of its low boiling point (56°C at atmospheric pressure). Average fluid velocities at the heater surface were varied from approximately U_h = 0.1 m/s to 1.0 m/s, corresponding to an approximate channel Reynolds number range of 1000 to 16800. The effects of subcooling and channel height were also studied.

A potential problem with using fluorocarbon liquids, such as FC-72, in cooling electronic packages involves the phenomena of temperature overshoot and hysteresis associated with the onset of nucleate boiling (Bar-Cohen and Simon, 1986). Boiling is induced by residual vapor and non-condensible gases residing in cavities on the component surface (nucleation sites). The highly wetting nature of fluorocarbons hinders the initiation of boiling because only very small vapor pockets can be maintained in surface cavities. A large excess temperature is therefore needed to initiate boiling at these sites. This problem has been described by various investigators including Oktay (1982), who recorded a significant excess temperature before boiling began. After incipience, the temperature dropped by as much as 20°C and the customary nucleate boiling curve was

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followed. In a review paper, Simons (1987) indicated that one of the concerns that led to selection of IBM's Thermal Conduction Module (TCM) over their Liquid Encapsulated Module (LEM) was the significant variability in the incipience temperature that was observed in single chip boiling experiments.

The temperature overshoot and corresponding temperature drop were also observed by Park and Bergles (1986, 1987), who conducted pool boiling experiments using flush mounted foil heaters and foil heaters with uninsulated sides protruding from the substrate by 1.1 mm. The heaters were arranged either singly or in a 1 × 2 array mounted to a vertical wall in a pool of R-113. The boiling curve for the single protruding heater agreed with that for the single flush mounted heater, except in the natural convection regime. For the flush and protruding heat source arrays, incipience occurred first at the top heat source and then at the bottom source. In addition, a smaller temperature overshoot was observed at the top heater than at the bottom heater. After nucleate boiling was established on both heaters, essentially the same boiling curves were obtained with little difference between curves for the flush or protruding heaters.

Samant and Simon (1986) experimentally investigated heat transfer from a single small heated patch (0.25 mm long × 2.0 mm wide) to a subcooled, fully developed turbulent flow. The test patch was located on the floor of a horizontal channel through which FC-72 circulated. The temperature overshoot and boiling hysteresis decreased with increasing subcooling and/or velocity. Maddox and Mudawar (1988) also demonstrated velocity and subcooling effects on hysteresis, as well as a dependence on the period of time during which the chip remained in the non-boiling state before increasing the heat flux to incipience.

Mudawar and Maddox (1989) performed boiling studies using FC-72 to correlate and model critical heat flux (CHF) during subcooled, two-phase forced convection from a discrete heat source flush mounted to one wall of a vertical channel. Increasing the fluid velocity significantly increased heat transfer in the single phase and nucleate boiling regions. The effect of increasing velocity on nucleate boiling diminished at higher heat fluxes, as boiling curves corresponding to different velocities began to converge. Different CHF mechanisms were observed at different velocities, with the higher velocities producing a stronger dependence of CHF on velocity. Fluid subcooling also increased the heat flux necessary for incipience and enhanced CHF. Mudawar and Maddox were able to develop a correlating equation for CHF that is in excellent agreement with experimental data.

Using R-113, Strom et al. (1989) performed experiments for a linear array of 10 evenly spaced flush mounted heaters and the same array of heaters protruding 0.8 mm and 1.6 mm into a vertically mounted flow channel. The heat flux was computed on the basis of the total surface area exposed to the flow. Increasing the subcooling and/or flow velocity extended the nucleate boiling range beyond that obtained for pool boiling. At the higher heat flux levels, nucleate boiling was thought to be fully developed and results for different velocities converged to the same boiling curve. This behavior was observed for both the flush and protruding heat sources, even though the nucleate boiling curve was lower for the protruding heat sources than for the flush mounted heat sources.

While the aforementioned study by Strom et al. bears a close resemblance to the present work, an important distinction between the two studies is the channel orientation. The vertical orientation of the channel in their study allowed vapor generated at an upstream heater to wash over downstream heaters, which could have a considerable effect on the boiling characteristics. In the current study, the channel was mounted with a horizontal orientation. This encouraged vapor generated at upstream heaters to rise to the channel ceiling before flowing downstream, thus inhibiting the exposure of downstream heaters to upstream vapor. Also, the protruding heaters used by Strom et al. (1989) and Park and Bergles (1986) had uninsulated sides, which increased the heat transfer surface area. In this study, the heat sources are flush mounted to Lexan protrusions which render the sides nonparticipating in the heat transfer process. Therefore, for each heater, a thermal and bubble boundary layer begins to form within the developing hydrodynamic boundary layer that begins at the leading edge of each protrusion.

Experimental Methods

The flow channel and heater module used in this study were equivalent to those described in the companion paper (Heindel et al., 1992). As shown in Fig. 1 of Heindel et al. (1992), the heater module consisted of ten square Lexan protrusions, each 25.4 mm long and 2 mm thick. The leading edge of the module was installed 600 mm downstream of the channel entrance. The channel width was fixed at 38.1 mm, and two channel heights, H_d (3.58 mm and 6.96 mm), were considered. The protrusions were located in-line, with a fixed separation distance of 6.35 mm, and a square heat source of length 6.35 mm was centered within and flush mounted to each protrusion. As shown in Fig. 2 of Heindel et al. (1992), each heat source consisted of a thickfilm resistive heater mounted to the back surface of a 6.35 mm thick block of oxygen-free copper with a thermal conductivity of ~400 W/mK. Two copper-constantan thermocouples were symmetrically mounted in the streamwise center of each heater with the arithmetic mean used as the average surface temperature (T_{sur}). The surface temperature variation was typically less than 0.5°C, but increased considerably at the highest heat fluxes to almost 1.5°C.

Since the point of onset of nucleate boiling is strongly influenced by surface conditions, such as its roughness and its history prior to boiling (Marto and Lepere, 1982; Anderson and Mudawar, 1988), a uniform surface preparation procedure was employed on all of the heaters prior to the initiation of each test. To begin each experiment, the heater module was removed from the channel and cleaned with ethyl alcohol to remove possible surface contaminants. Each heater surface was then lightly sanded in the streamwise direction with 100 strokes of 600 grit sandpaper. This procedure removed any oxide layer from the heater surface and provided near uniformity in surface conditions. The surface was cleaned again with ethyl alcohol and a soft cloth, and the module was inserted back into the channel. The system was then filled with FC-72. The fluid was degassed within the system using immersion heaters to raise the entire system temperature to near saturated conditions. After degassing, the system was sealed from the atmosphere and maintained at a positive pressure throughout the experiment to prevent the infiltration of ambient air. After the fluid velocity and subcooling were set to the desired experimental conditions and the system reached equilibrium, the individual heaters were powered and boiling was initiated. After maintaining nucleate boiling on all of the heaters for 15 minutes, the power was terminated. Since the time interval between boiling periods is known to affect the incipience proc-

Nomenclature

g = acceleration due to gravity
H_d = total channel height
q'' = heat flux
T_{in} = test section inlet temperature
T_{sat} = inlet saturation temperature
T_{sur} = average surface temperature
ΔT_{sub} = degree of subcooling at the test section inlet, $T_{sat} - T_{in}$
U_h = average longitudinal velocity at heat source location

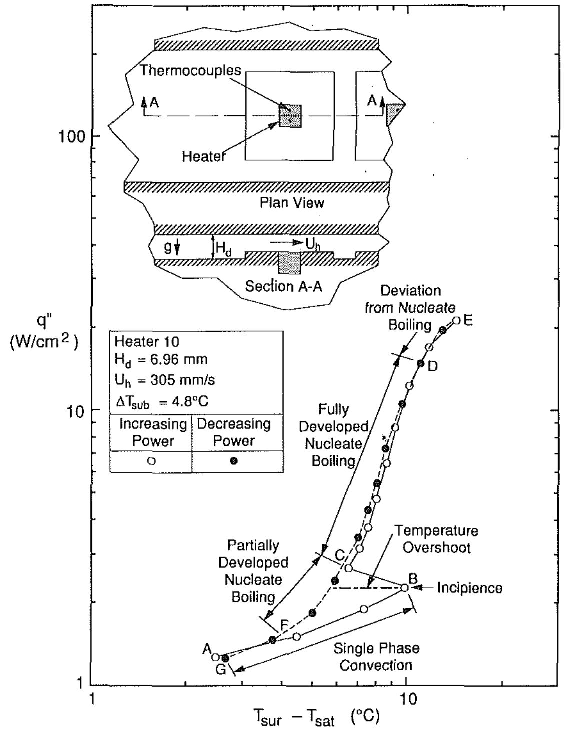

Fig. 1 Typical boiling curve near saturated conditions

ess (Anderson and Mudawar, 1988; Maddox and Mudawar, 1988), the heaters remained in the "power-off" state for 30 minutes before the actual experiment began. This preparation procedure was used to produce equivalent and repeatable test conditions for all of the experiments.

During each run, the flowrate and the inlet subcooling were held constant while the power to all ten heaters in the test module was increased. The variation in the heater power from heater-to-heater was negligible. Upon reaching steady state, the test section inlet pressure and temperature, pressure drop across the test section, flowrate, and all of the heater temperatures and powers were recorded by the data acquisition unit. The data acquisition software included the facility to display the boiling curve for any heater, as the data were being acquired during the experiment. This facilitated determination of the deviation from nucleate boiling. To determine the effect, if any, of upstream heating on the boiling curve, selected tests were also conducted with only heater 10 energized, while the other heaters remained inactive. Data were collected for velocity and subcooling ranges of U_h = 0.1 m/s to 1.0 m/s and ΔT_{sub} = 5°C to 43°C, respectively, and for channel heights of H_d = 6.96 mm and 3.58 mm. Throughout this discussion, heater 1 refers to the heater located farthest upstream and heater 10 corresponds to the heater located farthest downstream.

Data are presented using boiling curves (Fig. 1). The excess temperature is defined as the difference between the average surface temperature of each heat source (T_{sur}) and the inlet saturation temperature (T_{sat}), which is calculated from the test section inlet pressure. The heat flux, q'', based on the heat source surface area exposed to the fluid (40.3 mm^2), was uncorrected for conduction heat losses. This decision was justified on the basis of the heater/substrate, conduction/convection analysis associated with Fig. 3 of Heindel et al. (1992), which indicates substrate losses of less than 10 percent for nucleate boiling and decreasing losses with increasing excess temperatures. An experimental uncertainty analysis performed using the technique described by Kline and McClintock (1953) revealed random uncertainties in the excess temperature and the heat flux to be less than ±5 and ±2 percent, respectively, with the maximum uncertainties occurring at the highest heat fluxes.

Results

Typical Boiling Curve. All of the data of this study yielded boiling curves characteristic of highly wetting fluids (Oktay, 1982). One such example is shown in Fig. 1, which displays results for heater 10 at a velocity of U_h = 305 mm/s, a subcooling of ΔT_{sub} = 4.8°C, and channel height of H_d = 6.96 mm. At very low heat fluxes, such as at point A, single phase convection is the dominant mode of heat transfer even though the surface temperature is above the saturation temperature (Collier, 1981; Rohsenow, 1985). As the heat flux is increased (solid line), the surface temperature increases substantially above the saturation temperature, but heat transfer remains dominated by single phase convection. Point B represents the last data point before incipience, and point C represents the first point after incipience. A substantial drop in surface temperature is usually observed upon incipience. The *temperature overshoot* is defined as the constant heat flux temperature difference between the maximum temperature recorded under single phase convection conditions and the corresponding theoretical temperature under boiling conditions. From C to D, boiling becomes more vigorous as the heat flux is increased. The data points fall approximately along a straight line on the log-log plot, indicating that $q'' \propto (T_{sur} - T_{sat})^m$, m being a constant. Therefore, the average heat transfer coefficient in this region is approximately constant. This condition is associated with active nucleation sites located over the entire heater surface and is referred to as *fully developed* nucleate boiling. As E is approached, there is a deviation from nucleate boiling and the surface temperature begins to increase nonlinearly due to the reduction in the average heat transfer coefficient caused by excessive vapor generation. To avoid CHF and the corresponding abrupt increase in temperature, the heat flux is decreased (dashed line) once the deviation from the nucleate boiling curve is observed. The fully developed nucleate boiling portion of the curve is repeated from E to C, with the boiling intensity decreasing as the heat flux is decreased. Some of the boiling sites on the surface are eventually deactivated, leading to *partially developed* nucleate boiling (characterized by a few nucleation sites arranged randomly over the entire heater surface) and hysteresis from C to F. Eventually, all of the nucleation sites collapse and the single phase convection limit is again observed at G.

While the same general shape of the boiling curve was observed for all the tests, variations did exist in specific regions for different operating conditions. Most of the variations could be explained in terms of changes in the fluid velocity or degree of subcooling because a higher fluid velocity or larger degree of subcooling requires a larger heat flux to initiate boiling and to sustain fully developed nucleate boiling over the entire heater surface (Mudawar and Maddox, 1989). For example, all of the boiling regions were observed under highly subcooled conditions (ΔT_{sub} > 25°C), but there was a larger partially developed nucleate boiling region and a smaller fully developed nucleate boiling region. Figure 2 displays this variation for U_h = 309 mm/s, ΔT_{sub} = 43.7°C, and H_d = 3.58 mm. Upon incipience in the highly subcooled experiments, there was a transition to partially developed nucleate boiling, in contrast to the transition to fully developed nucleate boiling which occurred when the fluid was near saturated conditions (ΔT_{sub} ≈ 5°C). Therefore, the average heat transfer coefficient associated with partially developed nucleate boiling results from a combination of local nucleate boiling at a few active nucleation sites and single phase convection from regions between bubble patches. As the heat flux is increased, more nucleation sites become active, increasing the average heat transfer coefficient and increasing the slope on the boiling curve. When the

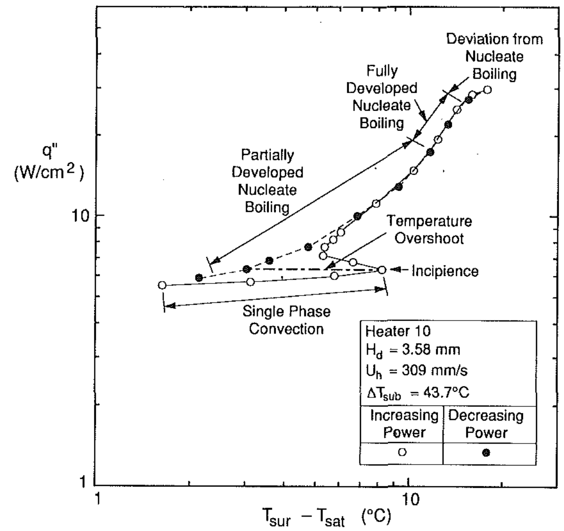

Fig. 2 Typical highly subcooled boiling curve

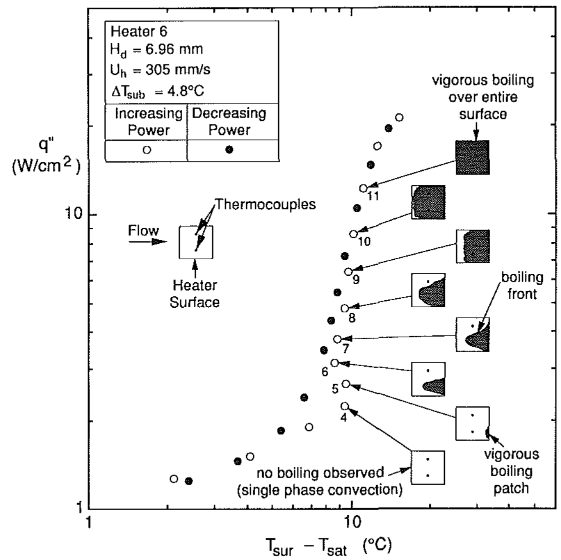

Fig. 3 Typical incipience pattern for the boiling patch incipience mode

heat flux is large enough to maintain nucleation sites over the entire heater surface, fully developed nucleate boiling is reached and the average heat transfer coefficient remains constant.

Another variation in the boiling curve can be attributed to the diversity in which the boiling front progresses over the heater surface. Visual observations of the heater surface corresponding to the data of Fig. 1 indicated that incipience was manifested by boiling engulfing the entire heater surface in a violent vapor burst accompanied by a direct shift to the nucleate boiling curve. Contrast this with Fig. 3 where visual observations of the surface, presented as artist's renderings on the figure, reveal that incipience occurred at a small downstream patch (data point 5) in this particular test for heater 6. As the heat flux was increased, the boiling patch encompassed more surface area with the boiling front progressing upstream until the entire surface was undergoing nucleate boiling (data point 11). At intermediate data points, the excess temperature and boiling surface area reached equilibrium values for each heat flux increment and did not change until the heat flux was altered. Due to the fact that only a portion of the surface was boiling, the surface temperature was slightly higher than that which occurs when the entire surface is in the nucleate boiling region (solid symbols). The two modes can be distinguished on subsequent boiling curves by either an abrupt change from

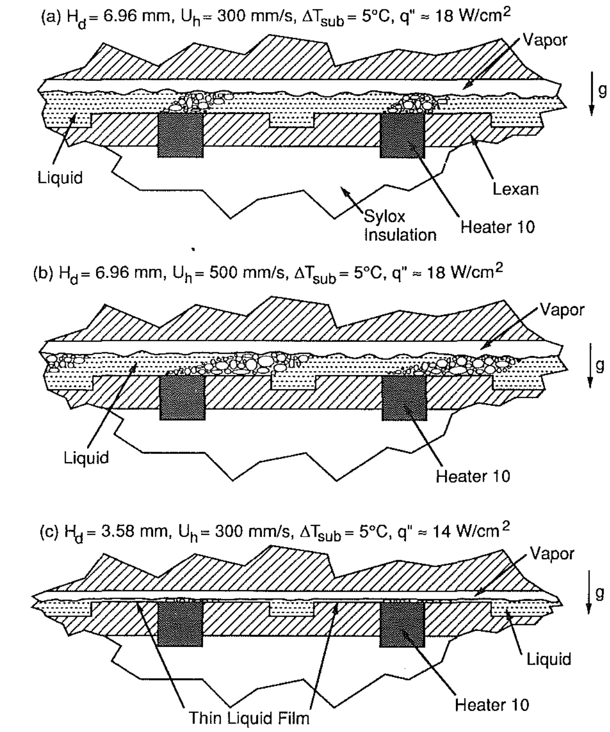

Fig. 4 Schematic of the effect channel reduction has on vapor accumulation

single phase convection to nucleate boiling (vapor burst), or a meandering path of data points between the single phase convection region and the developed nucleate boiling region (boiling patch).

The mode of incipience could not be correlated with fluid velocity, subcooling, channel height, or heater location, and both modes were observed on the same heater for different experiments performed at identical test conditions. Even when a particular mode of incipience was repeated on a heater for the same test conditions, the excess temperature required for incipience was not necessarily repeated. Similar incipience patterns for saturated pool boiling were reported by Danielson et al. (1987) and You et al. (1990). The incipience point for various perfluorinated liquids (Danielson et al., 1987) and R-113 (You et al., 1990) differed widely from test to test, while the nucleate boiling and single phase convection curves were almost identical. Tong et al. (1990) and You et al. (1990) concluded that, due to the good wetting characteristics and the poor reproducibility of the dynamic and static contact angles from case to case in highly-wetting liquids, incipience is a random process, best described by statistical means.

Heater-to-Heater Variations. With 10 inline heaters, vapor produced upstream could influence the heat transfer characteristics of downstream heaters. When H_d = 6.96 mm, a thin vapor layer is formed on the channel ceiling, with vapor being added in the downstream direction, causing the vapor layer to widen and thicken slightly. However, the vapor layer did not grow to such an extent that a continuous wall of vapor formed over the entire cross-section of the channel. This is schematically shown in Fig. 4(*a*) (from visual observations) for a heat flux of approximately 18 W/cm^2, a nominal velocity of U_h = 300 mm/s, and a nominal subcooling of ΔT_{sub} = 5°C. A vapor plume rises from the heater surface and merges into the vapor layer, but does not affect the downstream heater. If the nominal velocity is increased to U_h = 500 mm/s (Fig. 4(*b*)), the vapor plume remains attached to the protruding substrate over a longer distance, but detaches at the trailing edge and rises to the vapor layer, preventing any influence over the down-

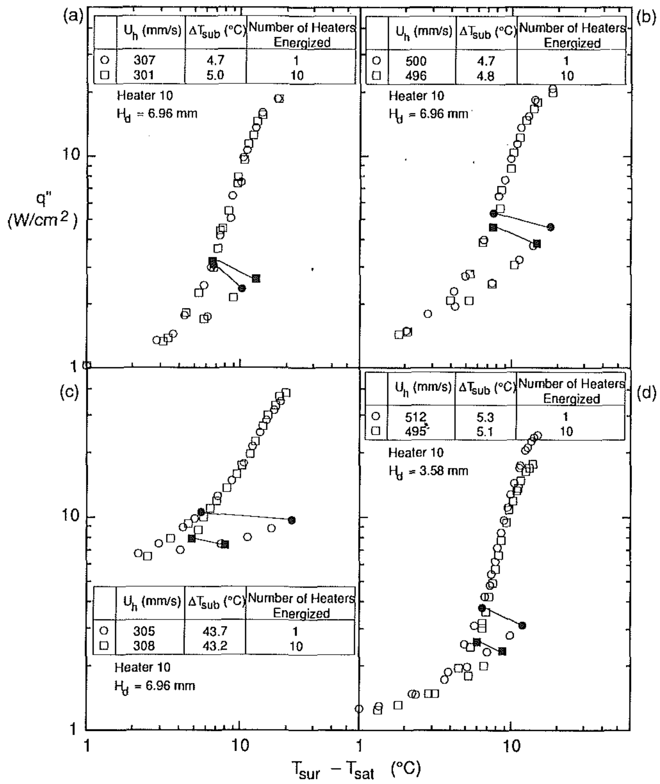

Fig. 5 Boiling curves for heater 10 with and without energized upstream heaters

stream heaters. When the channel height is reduced to H_d = 3.58 mm (Fig. 4(c)), the vapor layer thickness at high heat fluxes is of the same order as the clearance between the heater surface and the channel ceiling (1.58 mm). Hence, the possibility of a vapor layer blanketing downstream heaters, thereby preventing liquid access to the heater surface, is enhanced.

To verify the foregoing effect, selected tests were conducted with only heater 10 energized, and the results were compared with those obtained from subsequent tests for which all of the heaters were energized. With H_d = 6.96 mm, a nominal velocity of U_h = 300 mm/s, and a nominal subcooling of ΔT_{sub} = 5°C (Fig. 5(*a*)), results for heater 10 are virtually identical, irrespective of whether the upstream heaters are powered (square symbols) or inactive (circular symbols). The solid symbols connected by a line represent the last point before incipience and the first point after incipience. Although incipience appears to be delayed with upstream heating in Fig. 5(*a*), other results show the opposite trend (Fig. 5(*b*, *c*, *d*), and differences are attributed to the random nature of incipience. Apart from differences related to incipience for H_d = 6.96 mm, boiling curves are independent of whether or not there is upstream heating, irrespective of velocity or subcooling. For H_d = 3.58 mm (Fig. 5(*d*)), upstream heating did influence the boiling curve to the extent that the heat flux associated with a deviation from nucleate boiling was approximately 30 percent less than that corresponding to no upstream heating. At high heat fluxes and with all of the heaters energized, the vapor layer begins to impede liquid access to the heater surface, thereby increasing the surface temperature and advancing the deviation from nucleate boiling.

Figure 6(*a*) shows the boiling curves for heaters 1, 5, and 10 at a velocity of U_h = 305 mm/s, a subcooling of ΔT_{sub} = 5.2°C, and a channel height of H_d = 6.96 mm. The results indicate that the boiling curve is approximately independent of heater position at this channel height. Variations of the heater-to-heater surface temperature in the single phase convection region are discussed by Heindel et al. (1992). The maximum and minimum differences in surface temperatures between all heaters at each heat flux in the nucleate boiling region are 2.2°C and 0.9°C, respectively. This result has an important bearing on electronic packaging. If the heaters were undergoing nucleate boiling at different heat fluxes in the range from 4 to 15 W/cm^2, their surface temperatures would vary only slightly from approximately 63°C to 68°C. Therefore, if this cooling scheme were implemented in an actual multi-chip electronic module, large variations in the power dissipation from chip to chip could be accommodated while maintaining nearly uniform surface temperatures that are well below the 85°C maximum.

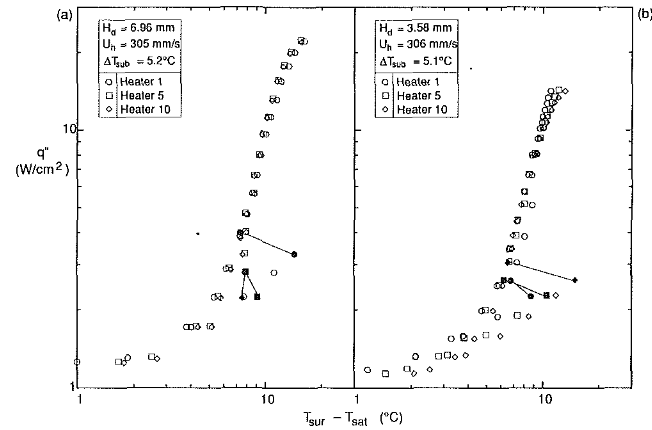

Fig. 6 Heater-to-heater boiling curve variations at a nominal velocity and subcooling of 300 mm/s and 5°C, respectively

In Fig. 6(*b*), results for heaters 1, 5, and 10 are again displayed for the same nominal velocity and subcooling, but the channel height has been reduced to H_d = 3.58 mm. The most significant difference occurs at high heat fluxes ($\gtrsim$ 10 W/cm^2), where downstream heaters consistently have a higher surface temperature and heater 10 is the first heater to deviate from nucleate boiling. This trend is due to the vapor layer caused by upstream vapor generation, which inhibits liquid access to downstream heaters. Therefore, for this channel height, variations from heater-to-heater are detected, but only at high heat fluxes where deviations from nucleate boiling are observed.

To determine heater-to-heater incipience trends, comparisons incorporating the entire data base were made between each heater and its immediate upstream neighbor. The data base consisted of 11 tests, with 9 comparisons for each test, yielding a total of 99 comparisons. No discernable incipience trend with heater position was observed, with 49 percent of the upstream neighbors showing a lower temperature overshoot, 43 percent showing a higher temperature overshoot, and 8 percent showing a virtually identical overshoot. The variations in incipience shown in Fig. 6 are therefore attributed to the random nature of nucleation.

Velocity Effects. The effect of the fluid velocity on the boiling curve was studied for both channel heights at nominal subcooling values of ΔT_{sub} = 5°C and 43°C. In pool boiling on a horizontal surface, vapor ascends from the heat source. However, with superposition of a horizontal fluid flow, motion of the vapor is influenced by both the imposed horizontal velocity component and the buoyancy-driven vertical velocity component. An increase in the fluid velocity would therefore be expected to influence the boiling curve by causing vapor generated at upstream heaters to pass in close proximity to the surface of downstream heaters. However, as shown in Fig. 4, this was not the case and there was no influence on the fully developed nucleate boiling curve for either channel.

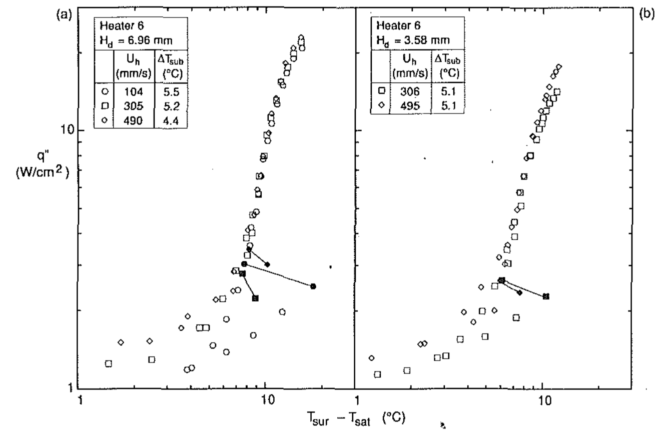

Fig. 7 Effect of velocity on a representative boiling curve at a nominal subcooling of 5°C

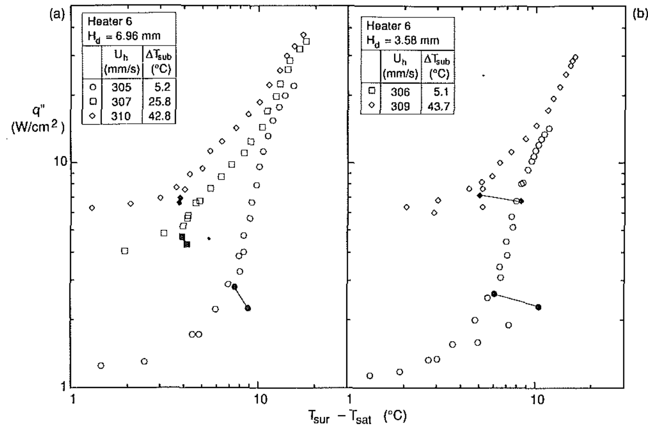

Fig. 9 Effect of subcooling on a representative boiling curve at a nominal velocity of 300 mm/s

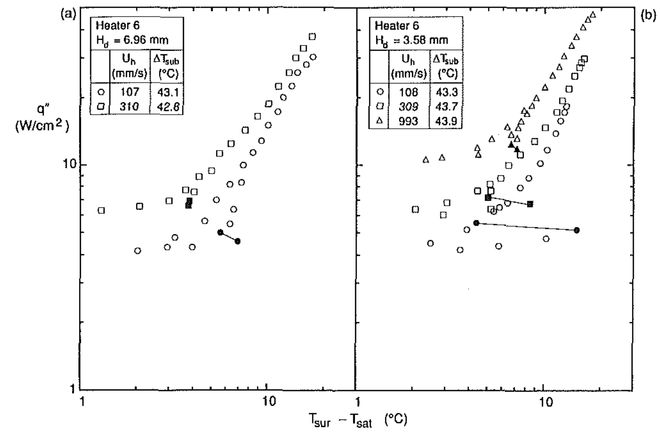

Fig. 8 Effect of velocity on a representative boiling curve at a nominal subcooling of 43°C

Figure 7(*a*) shows the effect of three different velocities, at nearly identical degrees of subcooling, on a representative boiling curve (heater 6) for H_d = 6.96 mm. As expected, in the single phase convection region, the heat flux at a given excess temperature increases with velocity due to an increase in the convection heat transfer coefficient. Incipience is significantly delayed for the 104 mm/s data, as the corresponding excess temperature exceeds that required for the larger velocities by more than 10°C. Although this behavior was not observed on all heaters, the general trend was one of a reduction in the temperature overshoot with increasing velocity (72 percent of the heaters displayed a reduced temperature overshoot with increasing velocity). Departures from this trend are attributed to the strong dependence of nucleation on surface conditions and past surface history (Tong et al., 1990; You et al., 1990).

In the fully developed nucleate boiling regime, increasing the velocity up to 490 mm/s did not produce a significant change in the boiling curve. This is consistent with the results of Strom et al. (1989), who reported essentially no velocity effect for heater 10 when undergoing fully developed nucleate boiling. Also, no velocity effects have been observed for subcooled flow boiling in round tubes (Collier, 1981). The heat flux which marks a deviation from nucleate boiling increases with increasing velocity, and although these variations are small, they were observed for all heaters. Increasing the velocity by 3 and 5 times only increased the heat flux associated with the deviation from nucleate boiling by 5 and 10 percent, respectively. When the channel height is reduced to H_d = 3.58 mm (Fig. 7(*b*)), similar trends are observed.

The effect of velocity on the boiling curve is magnified by increasing the level of subcooling. Figure 8 displays the data for a typical heater (no. 6). There is a large difference in the single phase convection region, and upon incipience, partially developed nucleate boiling conditions are reached if the initial subcooling of the fluid is high. For these conditions heat transfer is still influenced by single-phase convection, since a substantial quantity of energy is required to maintain fully developed nucleate boiling. In the partially developed nucleate boiling region, increasing the velocity can substantially increase the heat flux while holding the excess temperature constant. Conclusions are much less definitive for fully developed nucleate boiling, since results for the lower velocities deviate from nucleate boiling before those for the higher velocities reach fully developed nucleate boiling. Finally, variations in the heat flux at the deviation from nucleate boiling are much more pronounced for different velocities under highly subcooled conditions.

In summary, if the heat sources are undergoing nucleate boiling in a fluid near saturated conditions ($\Delta T_{sub} \approx$ 5°C), varying the velocity between 100 mm/s and 500 mm/s has little effect on the surface temperature for a given heat flux. If the fluid is highly subcooled, however, partially developed nucleate boiling persists over a wider range of heat fluxes and increasing the velocity can significantly reduce the surface temperature for a prescribed heat flux.

Subcooling Effects. The effect of subcooling on the boiling curve was studied for both channel heights by conducting experiments at three different degrees of subcooling for H_d = 6.96 mm and two different degrees of subcooling for H_d = 3.58 mm. All tests were conducted at a nominal velocity of U_h = 300 mm/s. As the degree of subcooling increases, the fluid is better able to absorb energy dissipated by the heater. A direct result of this effect is the condensation of generated vapor over a shorter distance downstream of the heater.

Figure 9(*a*) shows the effect of subcooling on a representative boiling curve (heater 6) for H_d = 6.96 mm. The single phase convection region persists at large heat fluxes for the highly subcooled case (~6 W/cm² for ΔT_{sub} = 43°C) because significant energy is required to bring fluid to the saturation temperature before phase change occurs. Upon incipience, partially developed nucleate boiling exists for the 25.8°C and 42.8°C subcooling conditions. In contrast, for ΔT_{sub} = 5.2°C, there is a transition from forced convection to almost complete fully developed nucleate boiling. The temperature overshoot associated with incipience was shown to decrease as the amount of subcooling increased, with 64 percent of the heaters displaying this trend. Deviations from this behavior are again attributed to factors discussed by Tong et al. (1990) and You et al. (1990).

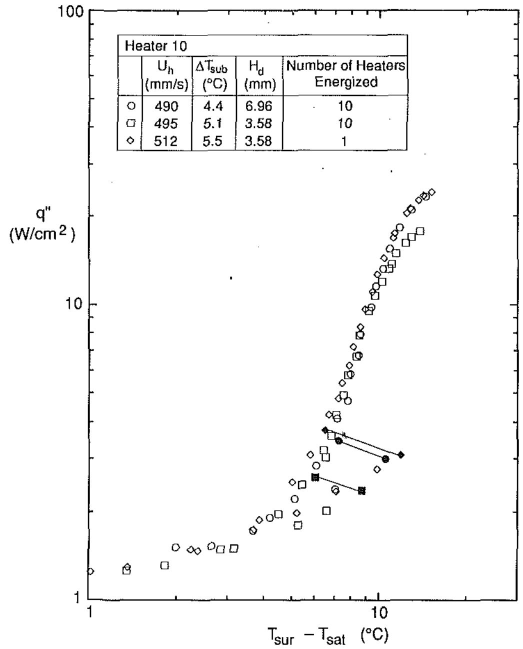

Fig. 10 Effect of channel height on the boiling curve for heater 10 at a nominal velocity (measured at the heat source) and subcooling of 500 mm/s and 5°C, respectively

When heat dissipation rates are high enough for the highly subcooled conditions to maintain nucleation sites over the entire heater surface, fully developed nucleate boiling is reached. Although the 25.8°C and 42.8°C subcooled boiling curves for heater 6 nearly coincide in the fully developed nucleate boiling region, suggesting that the heat flux may be independent of subcooling, this trend was not observed for all heaters. The 5.2°C subcooled data do not follow this trend because the heat flux associated with a deviation from nucleate boiling is much lower (approximately 40 percent) than the heat flux associated with the establishment of fully developed nucleate boiling at the higher levels of subcooling. When the subcooling is increased from $\Delta T_{sub} = 5.2°C$ to $\Delta T_{sub} = 25.6°C$, the heat flux associated with the deviation from nucleate boiling increases by 57 percent. However, the difference in heat flux associated with the deviation from nucleate boiling between the 25.8°C and 42.8°C subcooled data is very small, indicating a possible limit to the degree to which subcooling can extend the nucleate boiling regime. However, Mudawar and Maddox (1989) have shown that, if the heat flux is increased to the limit of CHF, increasing the amount of subcooling would increase the heat flux associated with this limit. For the channel height of $H_d = 3.58$ mm (Fig. 9(*b*)), the effects of subcooling are similar to those for $H_d = 6.96$ mm.

In summary, an increase in subcooling from 5.2°C to 25.8°C clearly affects the boiling curve by increasing the heat flux associated with every region of the curve. When the subcooling increases from 25.8°C to 42.8°C, however, these effects diminish for heat fluxes greater than 20 W/cm^2.

Channel Height Effects. Channel height effects were studied by comparing boiling curves obtained from experiments conducted in two channels for which $H_d = 6.96$ mm and 3.58 mm. The reduction in channel height reduced the flow passage above the heater by over 65 percent, thereby affecting vapor removal from the heater surface (refer to the discussion associated with Fig. 4). At nearly saturated conditions ($\Delta T_{sub} \approx 5°C$) for $H_d = 6.96$ mm, a thin vapor layer formed on the channel ceiling but did not interact with downstream heaters. In contrast, for $H_d = 3.58$ mm, vapor generated by upstream heaters was observed to interact with downstream heaters. At the higher heat fluxes ($\gtrsim 10$ W/cm^2), vapor from upstream heaters washed over downstream heaters, and visual observations suggested the formation of vapor slugs spanning from the protrusion surface to the channel ceiling. Such a flow would be detrimental to an electronic packaging scheme, since blanketing of an electronic component by a vapor layer would hinder heat transfer, thereby promoting large temperature variations and possible dryout.

Figure 10 shows the effect of channel height for heater 10 when the nominal velocity and subcooling are 500 mm/s and 5°C, respectively. Slight differences between the curves for the two channel heights under single phase conditions are due to effects discussed by Heindel et al. (1992). Although channel height has little effect on the fully developed nucleate boiling regime, it does affect the deviation from nucleate boiling. Increasing the channel height from 3.58 mm to 6.96 mm increased the heat flux associated with a deviation from nucleate boiling by 30 percent. This effect is only present, however, when all of the heaters are active because the increased channel height allows ascending vapor generated at upstream heaters to be swept away from downstream heaters. When upstream vapor generation is eliminated in the smaller channel by energizing only heater 10 (diamond symbols), the channel height no longer affects the deviation from nucleate boiling and nearly identical boiling curves are obtained. Therefore, if a packaging engineer desires to use the smaller channel to minimize space, the component heat flux should be maintained below 10 W/cm^2, which still allows for the possibility of transient power fluctuations.

Conclusions

This study has shown that a horizontal linear array of 10 discrete heat sources can be effectively cooled by forced convection boiling. Surface temperatures can be maintained below the desired maximum of 85°C, while maintaining only small variations in temperature over the entire array. Increasing the fluid velocity was shown to decrease the temperature overshoot, increase the heat flux associated with partially developed nucleate boiling, and increase the deviation from nucleate boiling; however, in fully developed nucleate boiling, the velocity did not have a substantial influence on the boiling curve. Subcooling decreased the temperature overshoot and allowed the fluid to absorb more energy dissipated from the heater surface, increasing the heat flux associated with every region of the boiling curve. Reducing the channel height restricted the escape of vapor from the surfaces, causing downstream heaters to be influenced by upstream vapor generation and a reduction in the heat flux associated with the deviation from nucleate boiling.

The preferred operating condition for an electronic cooling scheme of this nature would correspond to fully developed nucleate boiling. This condition would allow for large heat flux variations but would still provide for nearly uniform surface temperatures. However, because a precursor to this operating condition is boiling incipience and the attendant temperature drop, there is an inherent reluctance to implement the scheme in a chip cooling application (Simons, 1987). To overcome this barrier, a method must be developed to significantly attenuate (or eliminate) the temperature overshoot, thus allowing for a smooth transition from single phase convection to fully developed nucleate boiling and a stable operating condition.

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