Single- and Two-Phase Convective Heat Transfer From Smooth and Enhanced Microelectronic Heat Sources in a Rectangular Channel

D. E. Maddox

I. Mudawar

Boiling and Two-Phase Flow Laboratory,
School of Mechanical Engineering,
Purdue University,
West Lafayette, IN 47907

Experiments have been performed to assess the feasibility of cooling microelectronic components by means of single-phase and two-phase forced convection. Tests were conducted using a single heat source flush mounted to one wall of a vertical rectangular channel. An inert fluorocarbon liquid (FC-72) was circulated upward through the channel at velocities up to 4.1 m/s and with subcooling up to 46°C. The simulated microelectronic heat sources tested in this study include a smooth surface and three low-profile microstud surfaces of varying stud height, each having a base area of 12.7×12.7 mm². Correlations were developed for the single-phase convective heat transfer coefficient over the Reynolds number range from 2800 to 1.5×10⁵, where Reynolds number is based on the length of the heater. The results demonstrate that the low thermal resistances required for cooling of microelectronic heat sources may be achieved with single-phase forced convection by using high fluid velocity coupled with surface enhancement. Experiments were also performed to understand better the parametric trends of boiling heat transfer from the simulated microelectronic heat source. It was found that increased velocity and subcooling and the use of microstud surfaces enhance nucleate boiling, increase the critical heat flux, and reduce the magnitude of temperature overshoot upon the inception of nucleation.

Introduction

In recent years continued miniaturization of semiconductor electronics has led to significant increases in the heat dissipation of microelectronic chips. The power density for a single chip is already up to 40 W/cm² and is expected to exceed 100 W/cm² in the next decade. As this trend continues, new technologies must be developed to meet chip cooling demands. One area of interest is cooling by direct immersion in dielectric fluids such as the 3M Fluorinerts. Although the thermal transport properties of these fluids are poor, their effectiveness can be improved considerably by such factors as forced convection, structural enhancement, and boiling.

Single-phase forced convection presents a reliable form of direct immersion cooling, which has been proven capable of meeting requirements in microelectronic cooling. Tuckerman and Pease (1981), for example, attained a heat flux of 790 W/cm² using an enhanced surface consisting of microscopic channels 50 μm wide and 300 μm deep through which water was forced at a rate of about 8.6 ml/s. Based on heat transfer correlations developed for water, they estimated the thermal resistance for the FC-77 Fluorinert to be about 2.3 times greater than that for water. Although these values are above the expected requirements for electronic cooling, a pressure drop of over 214 kPa is required for a single 1 cm² heat source. This problem is a serious drawback when considering a system with a large number of computer chips.

In a more basic study, Incropera et al. (1986) obtained correlations for a single 12.7×12.7 mm² smooth heat source, flush mounted in a rectangular flow channel. The fluids tested were water and FC-77, and experimental data covered the Reynolds number range $1000 < Re_D < 14000$. Their data were correlated by the equation

$$\overline{Nu}_L = 0.13\, Re_D^{0.64}\, Pr^{0.38} \left(\frac{\mu_m}{\mu_w}\right)^{0.25} \tag{1}$$

Samant and Simon (1986) also considered single-phase cooling of a heater in a rectangular channel, but in their case the heater was very small, with a length of only 0.25 mm and a width of 2.0 mm. Because of the short length, the thermal boundary layer was very thin, causing the heat transfer coefficient to be considerably higher than for a typical computer chip. Their data were fitted by the equation

$$\overline{Nu}_H = 0.47\, Re_H^{0.58}\, Pr^{0.50} \tag{2}$$

where both Nusselt number and Reynolds number were based on the height of the flow channel perpendicular to the heater surface.

Additional experiments were performed by Ramadhyani and Incropera (1987) to increase the heat transfer coefficient by means of surface enhancement. Their work included two types of fins: a basic cylindrical pin, 11.2 mm in height and 2.03 mm in diameter; and a finned pin consisting of the basic fin with a series of square fins protruding along its length. These fins increased the area of the heated surface by factors of 8 and 12.8, respectively. The results of the experiments for FC-77 showed that the thermal resistance could be reduced by a factor as high as 20 for the basic pinned surface. Additional reductions in thermal resistance achieved through the use of finned pins were small (≈20 percent).

The results of Ramadhyani and Incropera were attained with moderate levels of pressure drop across the heater surfaces (up to 0.68 kPa). Other methods, such as the microchannel of Tuckerman and Pease, have reached considerably higher cooling rates but with a penalty of very high pressure drop. For

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such high heat fluxes it may be more appropriate to consider cooling by two-phase forced convection.

Although considerable work has been done in the area of pool boiling, there have been few flow boiling studies applicable to electronic cooling. One relevant study was performed by Katto and Kurata (1980). Their experiments involved a submerged jet flowing parallel to a small rectangular heater. Experiments were conducted for water and R-113, and all tests were made using saturated fluid at atmospheric pressure. Three heater surfaces, having the dimensions 10×10 mm^2, 15×10 mm^2, and 20×10 mm^2, were tested. The results were restricted to data and correlations for critical heat flux (CHF). The maximum values of CHF were obtained with the 15×10 mm^2 surface. Typical values for R-113 were 45.4 W/cm^2 at 2.1 m/s and 78.2 W/cm^2 at 6.02 m/s.

One drawback of two-phase systems is the thermal shock associated with a sudden drop in surface temperature that sometimes occurs upon the incipience of boiling. This phenomenon, called hysteresis, was encountered by Samant and Simon (1986) in experiments with flow boiling of FC-72 over the same small heater used in their single-phase experiments. However, the large degree of temperature overshoot in their data may be attributed to two aspects of the small size of their heater. First, the total number of surface cavities with vapor embryos would be small for such a small heater so incipience is more likely to be delayed. Second, the region of nucleation at the point of incipience is likely to be a large fraction of the total surface. Thus the effect of initial nucleation on the overall heat transfer coefficient is considerable. These effects are not likely to be as significant with heat sources as large as a typical computer chip, yet Samant and Simon's results may still be useful in understanding parametric influences on hysteresis. A study by Mudawwar et al. (1987) involving heat transfer to falling films of FC-72 from heat sources similar in size to those used in the present study showed little or no hysteresis for the full range of their operating conditions. Little hysteresis was also found in jet impingement R-113 boiling experiments by Ma and Bergles (1983). The large magnitude of hysteresis associated with pool boiling of low contact angle fluids such as R-113 and FC-72 (Bergles and Chyu, 1982; Marto and Lepere, 1982) suggests that fluid motion tends to reduce hysteresis for larger heat sources.

The present study includes results of experiments involving both single-phase and two-phase flow of FC-72 over heat sources with smooth and microstud surfaces. The objectives of the single-phase experiments were to expand the data base of Incropera et al. to a much higher Reynolds number range and to enhance heat transfer using a low-flow blockage microstud surface attachment. The two-phase experiments were performed to develop an understanding of the effects of forced convection, subcooling, and surface enhancement on the boiling curve in an effort to improve heat transfer performance for typical electronic cooling applications. Particular emphasis was placed on reducing hysteresis and increasing CHF. More specifically, it was desired to increase the heat transfer rate in the fully developed nucleate boiling range, especially at a surface temperature of 85°C, which is considered the maximum allowable chip junction temperature for reliable computer operation.

In selecting an appropriate enhanced surface for this study, a low-profile fin enhancement was chosen because of its limited flow blockage and low pressure drop characteristics. Furthermore, for the case where a series of heat sources are mounted in a flow channel, low-profile fins serve to minimize the thickening of the bubble boundary layer formed by nucleate boiling on the upstream heat source. The microstud surface was chosen because of its proven effectiveness in increasing the heat transfer rate in the fully developed nucleate boiling region (Grimley et al., 1987). Choice of spacing and width of the fins was based on the work of Nakayama (1984), and three fin heights were tested to determine the effect of fin height on boiling performance. The maximum fin height of 1.02 mm was limited by machinability and fin strength.

Experimental Apparatus

The flow loop of the experimental system is shown schematically in Fig. 1. The test heater was mounted in one side of a vertically oriented rectangular flow channel with an entrance length of 76 cm. The entrance reservoir upstream from the flow channel contained a nozzle that smoothly converged the flow to the channel dimensions. Also contained in the entrance reservoir was an immersion heater to control fluid temperature at the entrance to the test section. Any vapor formed by this heater was bypassed directly into the upper reservoir. The upper reservoir contained both an immersion heater and a coil-type condenser to aid in control of operating conditions.

Fluid was circulated through the flow loop by means of a magnetically coupled centrifugal pump. At the pump outlet a bypass line branched off from the main line to provide low fluid velocities in the test section while maintaining adequate flow through the pump. The fluid in the main line passed through a flat-plate heat exchanger, a filter, and a turbine flow meter on its way to the test section.

A cross-sectional view of the test heater assembly is shown

Nomenclature

a = microstud height
D = hydraulic diameter of flow channel
G = mass velocity = $\rho_f U$
H = height of flow channel perpendicular to heater surface
k = thermal conductivity of liquid based on channel entrance temperature
L = heater length in the flow direction
$\overline{\mathrm{Nu}}_H$ = average Nusselt number based on channel height perpendicular to heater surface $= qH/(T_w - T_m)k$
$\overline{\mathrm{Nu}}_L$ = average Nusselt number based on heater length $= qL/(T_w - T_m)k$
Pr = Prandtl number of liquid based on channel entrance temperature
q = heat flux
q_m = critical heat flux (CHF)
q_m^* = dimensionless CHF = $(q_m/Gh_{fg})/(\rho_g/\rho_f)^{0.582}$
Re_D = Reynolds number based on channel hydraulic diameter
Re_H = Reynolds number based on channel height perpendicular to heater surface
Re_L = Reynolds number based on heater length
T_{CHF} = mean heater surface temperature at critical heat flux
T_m = liquid temperature at channel entrance
T_{sat} = saturation temperature
T_w = mean temperature of heater surface
ΔT_{sat} = $T_w - T_{\mathrm{sat}}$
ΔT_{sub} = $T_{\mathrm{sat}} - T_m$
U = mean fluid velocity
We = Weber number = $G^2L/(\sigma\rho_f)$
μ_m = dynamic viscosity based on liquid temperature at channel entrance
μ_w = dynamic viscosity of liquid based on heater surface temperature
ρ_f = density of saturated liquid
ρ_g = density of saturated vapor
σ = surface tension

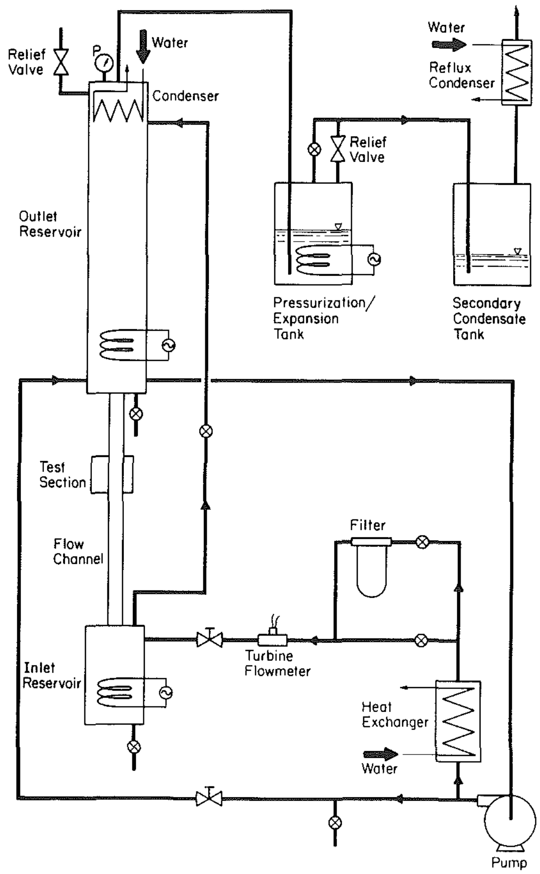

Fig. 1 Schematic diagram of the experimental facility

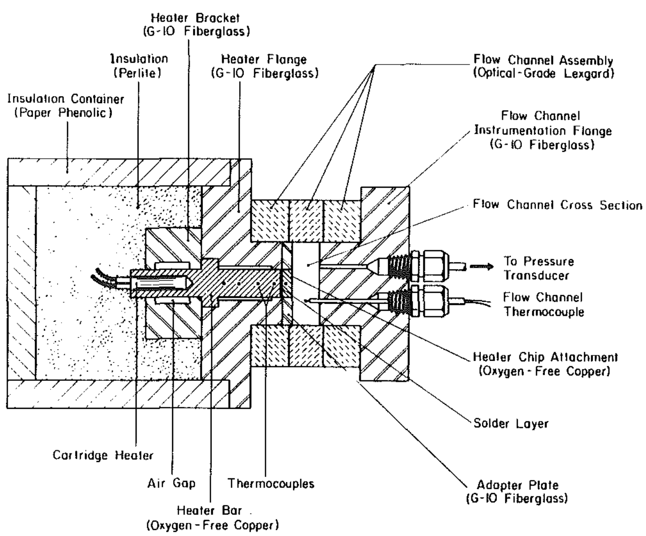

Fig. 2 Cross-sectional diagram of the heater assembly

in Fig. 2. The heater element consisted of a cylindrical cartridge heater embedded in an oxygen-free copper bar, which was mounted in a G-10 fiberglass flange. The $12.7 \times 12.7\ mm^2$ simulated chip was soldered to the bar and surrounded by an adapter plate, which was flush mounted to the wall of the flow channel. The heat flux was measured by a series of four thermocouples located along the length of a section of the copper bar having the same $12.7 \times 12.7\ mm^2$ cross section as the surface attachment. The square-shaped section was insulated from the surrounding fiberglass flange with an air gap partially filled with silicone rubber having a thermal conductivity 0.06 percent

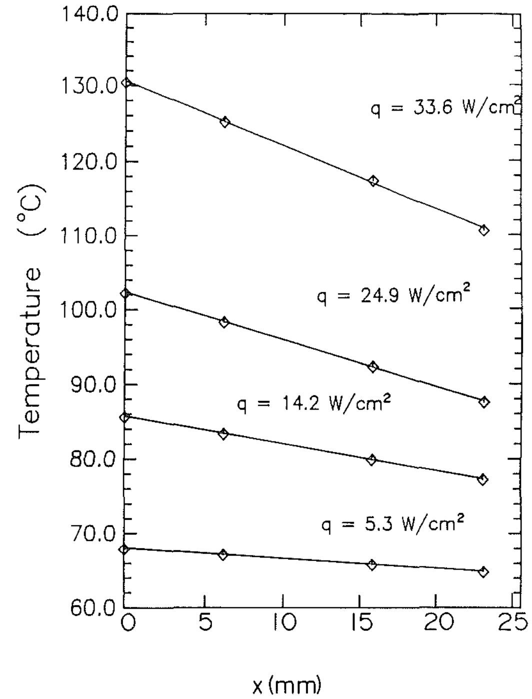

Fig. 3 Sample of measured temperature profiles along the calorimeter bar

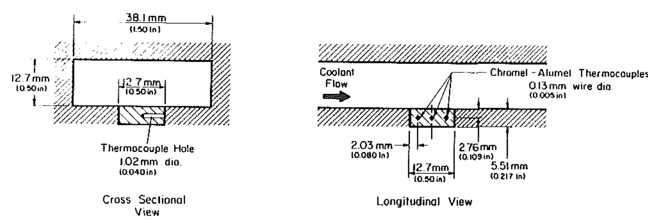

Fig. 4 Location of heat source in the flow channel wall

that of copper. This ensured one-dimensional heat flow along the instrumented portion of the heater. The thermocouples were made from 0.13-mm wires, and were set in the center of the bar. The temperature gradient used for evaluating the heat flux was calculated from a linear least-squares fit to the four temperature measurements. The sample temperature profiles shown in Fig. 3 illustrate the linearity associated with these measurements. A fifth thermocouple embedded in the simulated chip provided a temperature, which was then extrapolated based on conduction resistance between the thermocouple and the boiling surface to obtain the surface temperature. In one variation of the smooth surface chip three thermocouples were embedded in the chip along its length to measure the temperature variation of the heater surface in the flow direction. The total axial temperature differential was limited to 0.1°C except for fluxes close to CHF where the differential was as high as 1.2°C. The thermocouple arrangement is illustrated in Fig. 4, which also shows the location of the surface chip with respect to the flow channel. Voltage signals from the thermocouples and heater were processed and controlled by means of a Compaq 286 microcomputer used with a Keithley System 500 data acquisition and control system.

The heater surface tested in this study included a smooth surface and microstud surfaces of three different fin heights. The construction of the microstud surfaces is illustrated in Fig. 5. The square studs were oriented diagonally with respect to the direction of fluid flow to provide a more streamlined fin arrangement. The finned surfaces were machined by cutting two series of perpendicular grooves diagonally across the sur-

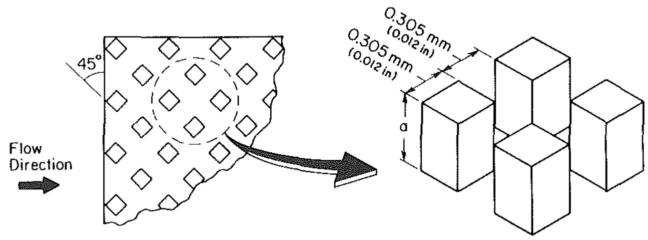

Fig. 5 Schematic of the microstud-enhanced surfaces

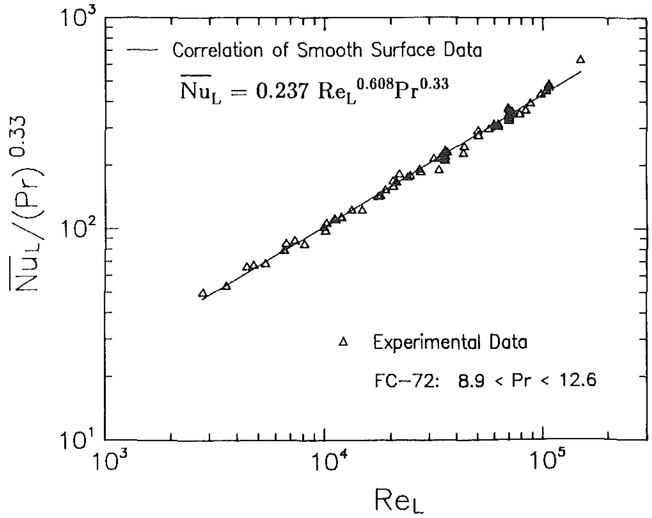

Fig. 6 Single-phase data and correlation for the smooth surface

face. The heights of these fins were 0.25, 0.51, and 1.02 mm. All surfaces were prepared before each experimental run by a vapor blast treatment consisting of a high-pressure stream of air, water, and abrasive particles to ensure uniform surface microstructure.

At the beginning of each experimental run the system was deaerated by operating the immersion heaters and the test heater and circulating the fluid through the flow loop. Vapor and air were allowed to exit from the top of the upper reservoir, passing through the pressurization tank and into the secondary condensate tank. The reflux condenser on the condensate tank allowed the air to escape from the system while recondensing the test fluid. When the system was completely deaerated, the outlet of the pressurization tank was closed off. If fluid subcooling was desired, the power to the immersion heaters and the water flow through the heat exchangers were adjusted to decrease the fluid temperature. The system pressure was maintained by adjusting the power to the immersion heater in the pressurization tank.

The procedure for obtaining data in the single-phase experiments was to start at one limit of the velocity range and to adjust the heat flux such that the heater surface temperature was about 10°C below the expected boiling incipience point. When the system reached steady state, the data were recorded, and a new velocity was then chosen. This procedure was repeated until the other limit of the velocity range was reached. In the boiling experiments, the procedure varied depending on the objective of the individual run. In general, the test heater was turned off after dearation for a given nonboiling period, and the power was then increased in small increments. In most cases the period without boiling was less than 2 h. In a few cases, however, it was increased in order to determine its effect on hysteresis. Following each power increment, steady-state conditions were reached after a waiting period of 15 to 30 min. Smaller power increments were added near boiling incipience

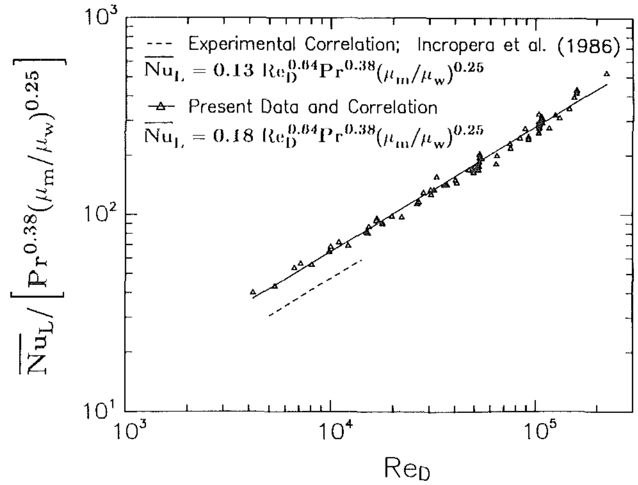

Fig. 7 Comparison of single-phase smooth surface data with the correlation of Incropera et al. (1986)

and CHF in order to obtain accurate measurements at these points.

Results

Single-Phase Studies. Figure 6 shows the single-phase atmospheric pressure FC-72 data for the smooth surface. The data are correlated in the Reynolds number range $2800 < Re_L < 1.5 \times 10^5$ by the equation

$$\overline{Nu}_L = 0.237\ Re_L^{0.608}\ Pr^{0.33} \tag{3}$$

where all properties are based on the mean fluid temperature at the entrance to the flow channel. The Prandtl number exponent was chosen as 1/3, which is typical for turbulent channel flow since the exponent could not be directly deduced from the limited Pr range of the present study. The maximum and mean deviations of data from equation (3) are 13.9 and 3.5 percent, respectively.

The present data were also correlated with respect to Re_D, the Reynolds number based on the channel hydraulic diameter. This parameter is more important than Re_L in electronic cooling applications involving a large array of microelectronic heat sources lined up along the flow channel. A comparison of the smooth surface single-phase results with the correlation of Incropera et al. (1986) is shown in Fig. 7. The viscosity ratio multiplier and Prandtl number exponent given in Fig. 7 are not necessarily recommended for design purposes since they were utilized solely for the purpose of comparison with the correlation of Incropera et al. The slopes of the two correlations are almost identical but the data of the present study lie approximately 37 percent higher than the correlation of Incropera et al. One possible explanation for this difference is channel orientation. Incropera et al. used a horizontal flow configuration with the heater facing upward while, in the present study, a vertical upward flow configuration was used. Density gradients in the vertical configuration may cause the thermal boundary layer to accelerate, thus enhancing heat transfer. Another reason for the difference may be the different treatment of heat losses in the two studies. In the present study, heat flux was calculated from the temperature gradient in the copper bar of the base heater. The temperature profile in this bar was very linear, suggesting that heat loss was very small. A very small heat loss, however, may occur beyond this bar near the chip surface. Nevertheless, the dimensions of this region are similar to those of the heater used by Incropera et al., so the loss should not be greater than the 8 percent predicted for their heater. It should also be noted that their data were corrected for the numerically predicted heat loss, which rep-

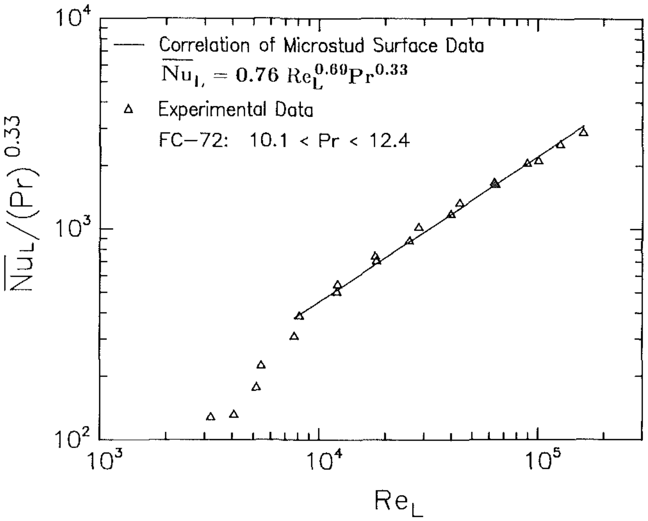

Fig. 8 Single-phase data and correlation for the microstud surface

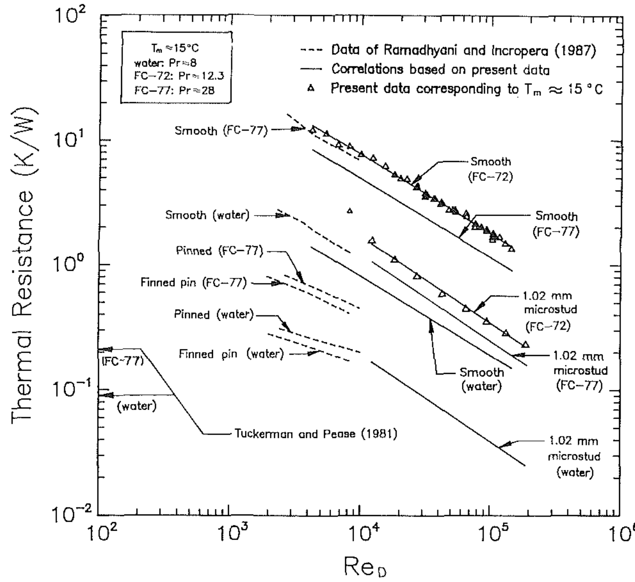

Fig. 9 Single-phase thermal resistance versus Reynolds number for smooth and microstud surfaces of the present study along with the data of Ramadhyani and Incropera (1987)

resents an upper limit, rather than the actual value, of heat loss. Another argument against the influence of heat loss in the present study is the fact that, although the data were taken over a wide range of heat flux (1.4 to 29.7 W/cm^2), they followed a well-established correlation as shown in Fig. 7. On a percentage basis, heat loss should decrease with increasing heat transfer coefficient. The fact that the slope of the present correlations was almost equal to the slope of the correlation by Incropera et al. suggests that heat loss in the present study does not account for the differences between the two correlations. Thus it can be concluded that the heat flux calculations of the present study satisfactorily accounted for heat loss.

Experimental data were also obtained in the present study using a compact heater of shallower construction (see details of heater design in Mudawwar et al., 1987), which consisted of a resistive wire sandwiched between two thin plates of a thermally conducting ceramic material and clamped against a copper plate, which supplied the heat to the chip attachment. This heater was designed to reduce heat loss by bringing the heating element closer to the wetted surface. The heat loss was numerically estimated to be less than 6 percent of the supplied electrical energy for the conditions of the present study. The compact heater was utilized primarily in high critical heat flux experiments where the calorimeter bar overheated beyond the maximum allowable temperature of the fiberglass flange. It was found that the electrical flux measured with the compact heater was about 4 percent lower than the heat flux measured by the calorimeter bar via linear curve fits of thermocouple readings. This fact is further evidence of the accuracy of the flux measurements using the calorimeter bar.

Figure 8 shows the single-phase data and correlation for the 1.02-mm microstud surface. At the lower end of the Reynolds number range the Nusselt number falls off rapidly, suggesting a change in the flow regime. Figure 9 shows a comparison of thermal resistances for the smooth and microstud surfaces of the present study along with the water and FC-77 data of Ramadhyani and Incropera. At the low end of the Reynolds number range, the thermal resistance for the microstud surface is reduced by a factor of 3.1 compared to the smooth surface. The reduction is much larger at higher velocities, with values of 4.3 at $Re_D = 12{,}000$, 5.4 at $Re_D = 1.5\times10^5$, and 6.2 at $Re_L = 1.5\times10^5$. Although this reduction is not as great as that obtained by the pin fins of Ramadhyani and Incropera, the use of higher velocity can offset the difference. For example, at $Re_D = 1.9\times10^5$ the thermal resistance of the microstud surface was 0.23 K/W, whereas for the most enhanced surface of Ramadhyani and Incropera, the minimum thermal resistance for FC-77 was about 0.42 K/W at $Re_D = 8100$. This conclusion may sound trivial, yet its practical implications are very important. Although microfin surfaces require higher coolant flow rates compared to the heavily finned surfaces of Ramadhyani and Incropera, they offer the advantages of minimal flow blockage and potentially lower pressure drop, allowing a large number of microelectronic heat sources to be mounted in series along the same flow channel.

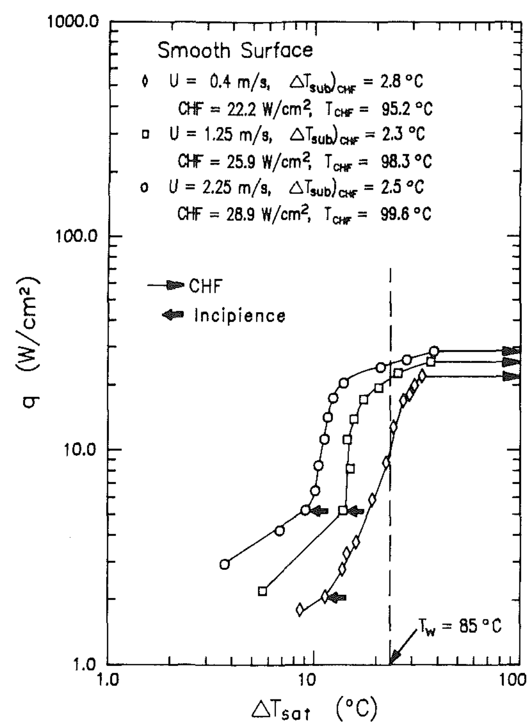

Fig. 10 Effect of velocity on the boiling curve for the smooth surface

The thermal resistance values can also be used to calculate the maximum values of heat dissipation for each surface, based on a typical surface-to-fluid temperature difference of 40°C (for FC-72) and a surface area of 12.7×12.7 mm^2. At $Re_D = 1.9\times10^5$, a thermal resistance of 1.3 K/W for the smooth

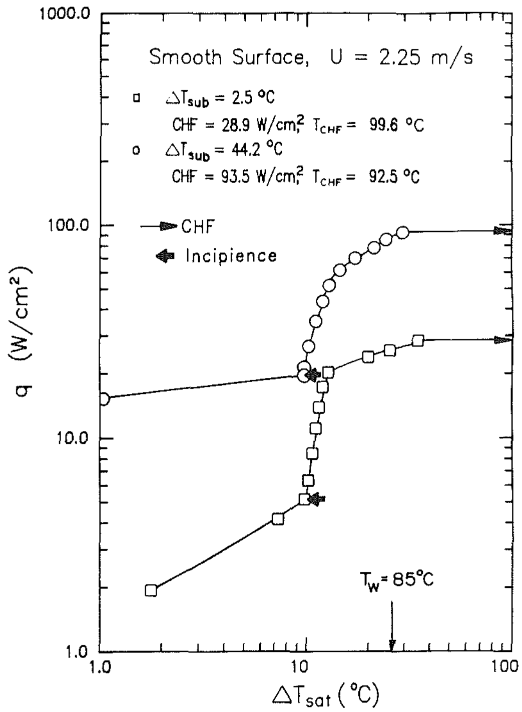

Fig. 11 Effect of subcooling on the boiling curve for the smooth surface

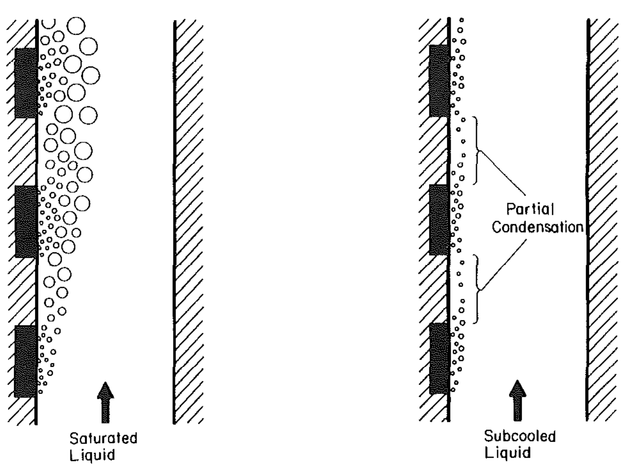

Fig. 12 Development of bubble boundary layers for the cases of high and zero subcooling

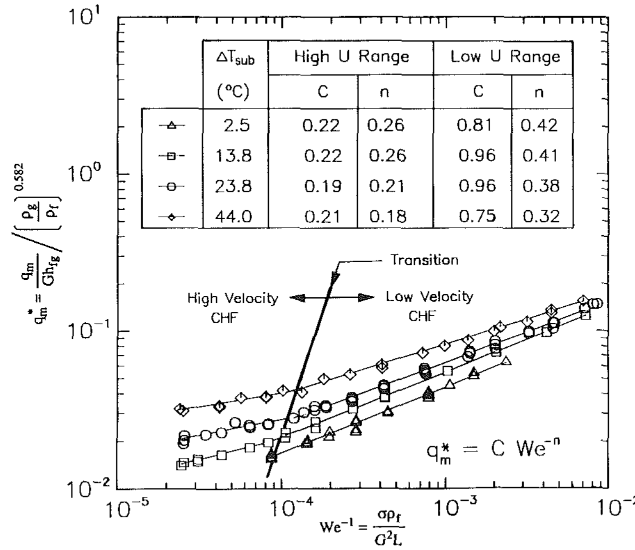

Fig. 13 Correlations of the critical heat flux for the smooth surface with velocity and subcooling

surface under these conditions corresponds to a heat flux of 19 W/cm². At the same Reynolds number for the microstud surface, a thermal resistance of 0.23 K/W corresponds to a heat flux of 108 W/cm². However, single-phase heat transfer rates calculated directly from equation (3) or Fig. 9 should be limited to heat flux levels below those required to trigger nucleation of bubbles on the surface. This limitation is of paramount importance in some applications where operation in the two-phase mode may be undesirable.

Two-Phase Studies. Figure 10 shows the effect of velocity on the cooling performance of the smooth surface. The higher velocities resulted in significant enhancement in the single-phase and nucleate boiling regions. At a surface temperature of 85°C, for example, an increase in velocity from 0.4 m/s to 2.25 m/s increased the heat flux by a factor of 2.6. At fluxes near CHF, however, the higher velocities showed significant reduction in the heat transfer coefficient, and the curves began to converge. The enhancement of CHF due to an increase in velocity from 0.4 m/s to 2.25 m/s was 28 percent.

With fluid subcooling, the enhancement of the heat transfer coefficient was considerably higher at fluxes close to CHF. This is shown in Fig. 11, which includes boiling curves for the smooth surface with 2.5°C and 44.2°C subcooling. The heat flux for the case with 44.2°C subcooling was increased above that for saturated boiling by a factor of 3 throughout the higher heat flux region. The value of CHF with 44.2°C subcooling was 93.5 W/cm² and the heat flux at a surface temperature of 85°C was 75 W/cm². This enhanced heat transfer performance can be explained by the lower temperature of liquid leaving the bulk region toward the heater surface during boiling.

Another advantage of subcooling is the fact that the low bulk fluid temperature causes vapor bubbles to recondense both as the bubbles are formed and after they leave the surface. Flow visualization has revealed that subcooling significantly reduces both bubble departure diameter and the thickness of the bubble boundary layer. This is illustrated schematically in Fig. 12, which shows the development of bubble boundary layers for cases of high and low subcooling. With high subcooling the bubbles are so small and recondense so quickly, they are barely visible. These facts have important implications for electronic cooling since highly subcooled flow would allow an array of heat sources to be mounted along the length of a flow channel without the danger of compromising the cooling performance of downstream heat sources.

CHF data were taken over a velocity range of 0.2 to 4.1 m/s and subcooling up to 46°C. Figure 13 shows CHF increasing with both velocity and subcooling. The data reveal a transition from a lower slope at lower velocity to a steeper slope at high velocity. This trend indicates a marked change in the CHF mechanism with increased velocity. Flow visualization revealed that for the lower velocity range, CHF was caused by dryout following the formation of a single continuous blanket, which covered the entire heated surface. On the other hand, CHF in the high velocity range was accompanied by dryout over several smaller discrete portions of the surface. Figure 13 also shows CHF correlations with the inverse Weber number for each of the four levels of subcooling. The exponent of the vapor-to-liquid density ratio term was chosen as the average of the values from the studies by Katto and Kurata (1980) and Yagov and Puzin (1984) for channel flow because the range of density ratio in the present study was fairly constant.

The results for the microstud surfaces, shown in Fig. 14, revealed more complicated nucleate boiling characteristics. This was probably due to the existence of different boiling regimes

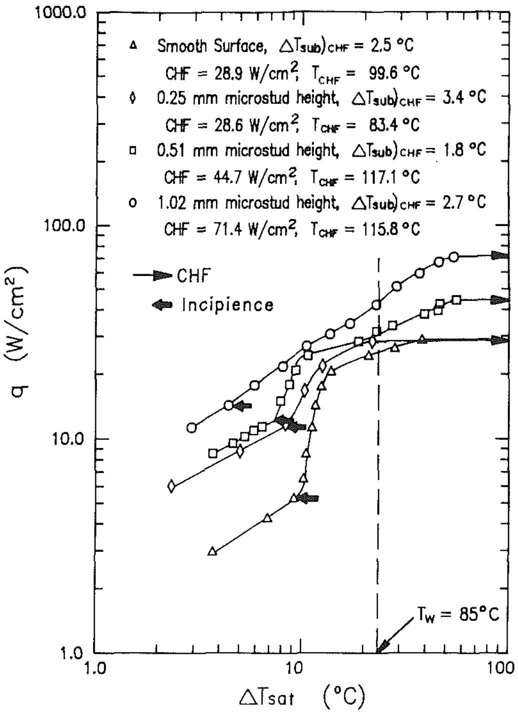

Fig. 14 Effect of fin height on the boiling curve for microstud surfaces

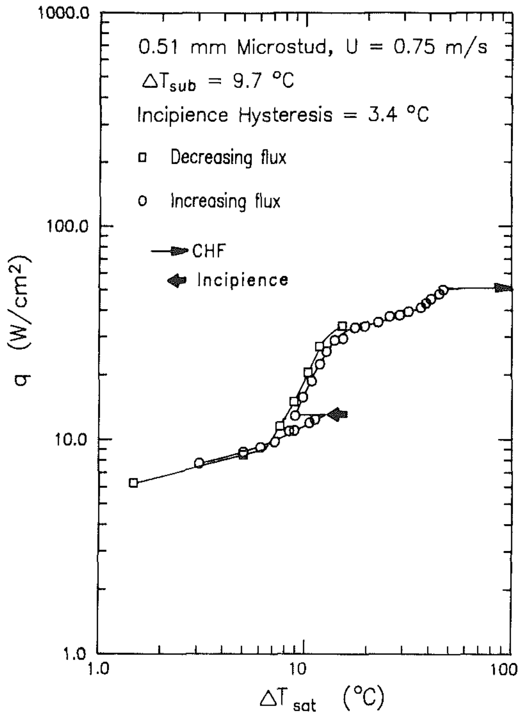

Fig. 15 Hysteresis in the boiling curve of the 0.51-mm microstud surface

between the base surface and the tips of the fins. All fin heights provided substantial enhancement in the single-phase region. The degree of enhancement then decreased in the nucleate boiling region. As the curves progressed toward the point of departure from nucleate boiling, the curves for the smooth surface and the surface with the shortest microstud height both broke abruptly to CHF, while surfaces with longer microstuds extended the boiling region to higher fluxes. The explanation for this behavior is that the tips of the longer fins remained in the nucleate boiling regime even after the base surface had departed from nucleate boiling. The 0.51 and 1.02-mm microstud heights showed enhancement of CHF over the smooth surface by factors of 1.5 and 2.5, respectively. The highest CHF value of 262 W/cm^2 was obtained with the 1.02-mm microstud surface at a velocity of 4.1 m/s and 46°C subcooling.

Hysteresis. It is very difficult to correlate the phenomenon of hysteresis because of the large number of variables that influence the onset of nucleation from wall cavities. Small deviations in such factors as size and distribution of surface cavities, fluid purity, and the history of vapor embryos at the boiling surface all have substantial influences on the amount of temperature overshoot associated with the onset of nucleation. Therefore efforts were made to determine qualitatively the effects of certain parameters on hysteresis and to obtain worst case values for overshoot.

One factor that has been found to have a substantial influence on hysteresis is the period of time during which the chip remains in the nonboiling state both before and during the increase of heat flux toward incipience. It is hypothesized that during this waiting period vapor embryos within the surface cavities shrink in size, requiring higher heat flux for nucleation. For every case in which hysteresis exceeded 2°C, the nonboiling period was greater than one hour. Aside from this fact there was no apparent correlation between the waiting period and the magnitude of hysteresis.

Another factor that has been considered is the presence of dissolved air in the fluid. If air makes its way into a surface cavity, it would create artificial embryos, resulting in premature boiling and reduced temperature overshoot. Two experimental runs with large amounts of air in the system yielded hysteresis values of 2°C. Thus, while the presence of air may reduce hysteresis, it does not eliminate it entirely.

The worst case of hysteresis obtained in the present study suggests some trends of hysteresis associated with the effects of enhancement. A temperature overshoot of 7.5°C occurred in an experiment with the smooth surface at a low velocity of 0.75 m/s and almost zero subcooling. However, extensive testing with the microstud surface at similar conditions resulted in a maximum temperature overshoot of 4°C. Thus, it can be concluded that the presence of microstuds generally reduces hysteresis. One explanation for this behavior is that the studs enhance single-phase heat transfer considerably compared to the lower nucleate boiling range. Thus the percentage increase in the heat transfer coefficient at incipience is less, and the resulting temperature drop is smaller than for an unenhanced surface. Another possible explanation is related to the hydrodynamics of fluid flow around a stud. Higher velocities tend to increase cavitation downstream of the studs, which may trigger boiling at a relatively lower wall superheat as Fig. 14 clearly indicates.

Another explanation for the relatively high degree of overshoot at low velocity and zero subcooling is the effect of fluid flow on the propagation of the nucleation front with increased heat flux. Boiling tends to start at the downstream edge of the heater because the fluid superheat is greatest at this point. Fluid drag forces at high velocities tend to push the bubbles downstream away from nonboiling portions of the heater, inhibiting the spread of nucleation. Since only a small portion of the heater experiences boiling at incipience, the overall heat transfer coefficient for the surface increases only slightly, resulting in a small surface temperature drop. Higher fluid velocities also increase the single-phase heat transfer coefficient prior to boiling. Thus, the nucleate boiling contribution to the overall heat transfer coefficient following the onset of nucleation becomes less significant with increased velocity. Fluid subcooling produces a similar effect by decreasing the bubble size. The smaller bubbles are less likely to spread boiling to neighboring nucleation sites, and propagation of the boiling front is inhibited.

These velocity and subcooling effects are in general agreement with the results of Samant and Simon as reported by Bar-Cohen and Simon (1986). Their data showed a very strong relation between velocity and temperature overshoot, and a weaker but still distinct relationship for subcooling.

The data for an experimental run in which hysteresis occurred are shown in Fig. 15. This run was made for the 0.51-mm microstud surface with a fluid velocity of 0.75 m/s and 9.7°C subcooling. The procedure for the run was to start at a high rate of boiling and progress down the boiling curve into the single-phase region. Next, the system was run for two hours without boiling before progressing up the curve first toward incipience and finally to CHF. The intention of this procedure was to measure both the incipience hysteresis and an overall hysteresis in the boiling curve. The incipience hysteresis in this case was 3.4°C, while the curve hysteresis was much smaller.

Summary

Studies have been performed based on experiments in single-phase and two-phase forced convection cooling of a simulated microelectronic heat source. Key results are as follows:

1 New single-phase correlations were developed over the Reynolds number range $2800 < \mathrm{Re}_L < 1.5\times10^5$ ($4200 < \mathrm{Re}_D < 2.25\times10^5$) for a smooth surface, and $7700 < \mathrm{Re}_L < 1.6\times10^5$ ($1.16\times10^4 < \mathrm{Re}_D < 2.4\times10^5$) for a microstud surface. For a heater-to-fluid temperature difference of 40°C and a Reynolds number $\mathrm{Re}_D = 1.9\times10^5$, it was determined that the smooth surface could dissipate 19 W/cm^2, and the microstud surface 108 W/cm^2. The authors postulate that use of the low pressure drop microstud surface with high fluid velocity may be preferred over the use of higher pressure drop heavily finned surfaces at lower velocities.

2 Increasing fluid velocity resulted in significant enhancement in the single-phase and nucleate boiling regions of the boiling curve, but severe degradation in the nucleate boiling performance occurred prior to CHF with higher velocities and CHF enhancement was considerably smaller.

3 Fluid subcooling substantially increased cooling performance near CHF. At 44.2°C subcooling, the value of CHF was 93.5 W/cm^2, showing an increase by a factor of 3.2 over the case of near-saturated boiling.

4 For multichip cooling applications, subcooling offers the advantage of reduced bubble boundary layer thickness, thus making it more feasible to line heat sources up along the flow direction in a channel.

5 Increasing stud height showed significant enhancement throughout the boiling curve. The 1.02 mm microstud showed CHF values as high as 260 W/cm^2, presumably because nucleate boiling continued at the tips of the fins even after a departure from normal nucleate boiling had occurred near the base surface. Although trends in the data suggest that longer studs should improve performance further, such a surface would be difficult to manufacture. However, it is possible that for a different stud width and spacing, a longer stud may still improve performance.

6 It was found that the degree of hysteresis at boiling incipience was influenced by a number of factors. First, significant hysteresis occurred only after the heater surface remained in a nonboiling state prior to incipience for at least one hour. Also, increases in the single-phase heat transfer coefficient due to increased velocity and surface enhancement tend to reduce hysteresis because the sudden increase in heat transfer rate at boiling incipience is less. The microstuds may further reduce hysteresis as a result of cavitation downstream individual studs. Furthermore, flow visualization indicated that velocity reduces temperature drop at the incipience by inhibiting propagation of the bubble front in the upstream direction. These points agree with the fact that the worst case of temperature overshoot in the present study, which was 7.5°C, occurred with the smooth surface at a low velocity and low subcooling.

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References

Bar-Cohen, A., and Simon, T. W., 1986, "Wall Superheat Excursions in the Boiling Incipience of Dielectric Fluids," *Heat Transfer in Electronic Equipment*, ASME HTD-Vol. 57, pp. 83–94.

Bergles, A. E., and Chyu, M. C., 1982, "Characteristics of Nucleate Pool Boiling From Porous Metallic Coatings," ASME JOURNAL OF HEAT TRANSFER, Vol. 104, pp. 279–285.

Grimley, T. A., Mudawwar, I., and Incropera, F. P., 1987, "Enhancement of Boiling Heat Transfer in Falling Films," *Proceedings of the 1987 ASME/JSME Thermal Engineering Joint Conference*, Vol. 3, pp. 411–418.

Incropera, F. P., Kerby, J. S., Moffatt, D. F., and Ramadhyani, S., 1986, "Convection Heat Transfer From Discrete Heat Sources in a Rectangular Channel," *Int. J. of Heat and Mass Transfer*, Vol. 29, pp. 1051–1058.

Katto, Y., and Kurata, C., 1980, "Critical Heat Flux of Saturated Convective Boiling on Uniformly Heated Plates," *Int. J. of Multiphase Flow*, Vol. 6, pp. 575–582.

Ma, C. F., and Bergles, A. E., 1983, "Boiling Jet Impingement Cooling of Simulated Microelectronic Chips," in: *Heat Transfer in Electronic Equipment*, ASME HTD-Vol. 28, pp. 5–12.

Marto, P. J., and Lepere, V. J., 1982, "Pool Boiling Heat Transfer From Enhanced Surfaces to Dielectric Fluids," ASME JOURNAL OF HEAT TRANSFER, Vol. 104, pp. 292–299.

Mudawwar, I., Incropera, T. A., and Incropera, F. P., 1987, "Boiling Heat Transfer and Critical Heat Flux in Liquid Films Falling on Vertically-Mounted Heat Sources," *Int. J. of Heat and Mass Transfer*, Vol. 30, pp. 2083–2095.

Nakayama, N., Nakajima, T., and Hirasawa, S., 1984, "Heat Sink Studs Having Enhanced Boiling Surfaces for Cooling of Microelectronic Components," ASME Paper No. 84-WA/HT-89.

Ramadhyani, S., and Incropera, F. P., 1987, "Forced Convection Cooling of Discrete Heat Sources With and Without Surface Enhancement," *Proceedings of the International Symposium on Cooling Technology for Electronic Equipment*, Honolulu, HI, pp. 249–264.

Samant, K. R., and Simon, T. W., 1986, "Heat Transfer From a Small, High-Heat-Flux Patch to a Subcooled Turbulent Flow," *Proceedings of the AIAA/ASME Heat Transfer and Thermophysics Conf.*, Boston, MA.

Tuckerman, D. B., and Pease, R. F., 1981, "Ultrahigh Thermal Conductance Microstructures for Cooling Integrated Circuits," *Proceedings of the 32nd Electronics Components Conference*, pp. 145–149.

Yagov, V. V., and Puzin, V. A., 1984, "Critical Heat Fluxes in Forced-Convection Boiling of Refrigerant-12 Under Conditions of Local Heat Sources," *Heat Transfer—Soviet Research*, Vol. 16, pp. 47–55.