Evaluation of Jet Impingement, Spray and Microchannel Chip Cooling Options for High Heat Flux Removal

SATISH G. KANDLIKAR and AKHILESH V. BAPAT
Mechanical Engineering Department, Rochester Institute of Technology, Rochester, New York, USA

Thermal management for high heat flux removal from microelectronic chips is gaining critical importance in many earth-based and space-based systems. Heat fluxes greater than 1 MW/m$^2$ (100 W/cm$^2$) have already been realized in high-end server applications, while cooling needs in next generation chips and advanced systems such as high-power electronics and electrical systems, pulsed power weapons systems, solid-state sensors, and phased-array radars are expected to reach 5–10 MW/m$^2$ (500–1000 W/cm$^2$). After evaluating the contributions from different thermal resistances in the chip-to-ambient thermal path, this paper presents a critical review and research recommendations for three prominent contending technologies: jet impingement, spray cooling, and microchannel heat sinks.

INTRODUCTION

Oktay et al. [1] presented an overview of various man-made thermal systems identifying the heat fluxes and associated surface temperatures. The ballistic re-entry encountered one of the highest heat fluxes of 10 MW/m$^2$ (1 kW/cm$^2$), but the surface temperature was also high, about 3500 K. Their diagram is redrawn in Figure 1 to include the current and projected needs in chip-cooling application. It is seen that heat fluxes of the same magnitude as the ballistic re-entry are projected by 2011, but the surface temperature is limited to only 350 K.

A schematic of the thermal resistance elements in a chip thermal path is shown in Figure 2. For the present discussion, heat generation is assumed to be uniform over the surface of the silicon chip (junction level heat transfer is not considered). The thermal resistances encountered are due to conduction in the silicon, conduction through the thermal interface material (TIM), conduction in the base material, and convection and fluid temperature rise in the heat exchanger. A brief discussion on the individual resistances and their impact on the cooling system design are presented next.

It is clear that the thermal resistance of the TIM, the conduction resistance of the substrate, and the convective resistance inside the microchannels are all crucial in determining the maximum heat that can be transferred for a given coolant fluid flow rate and chip surface and fluid inlet temperatures. A separate chip, called thermal chip, may be used for cooling purposes. The thermal chip may incorporate microchannels or other cooling techniques and may be attached to the IC chip with a suitable TIM.

Table 1 shows the temperature drops across the silicon chip (thickness 0.7 mm), TIM, 1-mm thick copper heat spreader base, and heat exchanger. The thermal resistance values used in Table 1 are based on currently available technologies for TIM and heat exchangers. Given an allowable chip surface temperature of 80°C and a coolant design inlet temperature of 35°C, the total available temperature difference is only 45°C. As seen from Table 1, a temperature difference of 67.2°C is needed at a heat flux of 1 MW/m$^2$ (100 W/cm$^2$), rising to 672°C at 10 MW/m$^2$ (1 kW/cm$^2$).

To meet the cooling demand at higher heat fluxes, each thermal resistance listed needs to be reduced significantly. Such reductions may be attainable only through dramatic improvements in the individual technologies.

At 10 MW/m$^2$ (1 kW/cm$^2$), the temperature drop across the silicon substrate itself consumes the entire available temperature potential. Thinning the chip down to 0.2 mm will reduce the temperature drop from 47.6°C to 13.6°C. Structural and
Thermal expansion issues need to be addressed in making a final decision.

Thermal interface materials have been studied extensively in the past decade. A recent report by Samson et al. [2], summarized in Table 2, indicates that the best available TIM in 2004 was a Phase Change Metallic alloy that yielded a minimum thermal resistance of 5 mm² °C/W. This is half the value from Table 1, but the resulting temperature drops across the TIM are still unacceptably large at higher heat fluxes.

Recent advances have been reported on metallic solders with indium and silver. One of the major problems with a solder is void generation and lack of compliance between the two mating interfaces. Colgan et al. [3] report the development of an indium solder to overcome these problems; in the future, a value lower than 3 mm² °C/W is expected. Research is also continuing regarding the mechanical properties of indium solder, to extend its lower operating temperatures to below 0 °C. Another major development in this field is related to the use of carbon nanotubes that are grown on the substrates and connect the two mating interfaces. The high thermal conductivity of carbon nanotubes (3000 W/m °C) coupled with direct growth on the substrate results in a low interface thermal resistance. Xu and Fisher [4] report their work on CNT-copper interfaces and present models to represent various thermal resistances in the composite material. Park and Taya [5] report a new TIM composed of carbon nanotubes, silicon thermal grease, and chloroform. They report a measured value between 2.65 and 3.99 mm² °C/W. A detailed survey of this field is provided by Prasher [6]. With the developments in indium solders and CNT- and nanoparticle-based composites, it is reasonable to expect that future developments in this field will lead to a value of 1 mm² °C/W in the next 3–5 years.

The heat exchanger substrate or heat spreader thickness of 1 mm in copper certainly poses a problem. A reduction in the substrate thickness to 0.2 mm or lower is possible and may be readily implemented. The resulting temperature drop will be only 1.5 °C for a heat flux of 300 W/cm². Direct cooling of silicon chips is a preferred option, but it may not be acceptable due to leakage and manufacturing concerns. Incorporating a thermal chip that has enhancement features is another option.

The heat exchanger system used in chip cooling presents a major challenge, as seen from Table 1. The conventional systems using compact heat exchangers are barely able to meet the cooling requirement of 1 MW/m² (100 W/cm²). It is clear from Table 1 that at 10 MW/m² (1 kW/cm²), major innovations are needed to dramatically reduce the thermal resistance in the heat exchanger. The three competing cooling technologies are reviewed in the following sections.

**JET IMPINGEMENT COOLING**

In a jet impingement cooling system, high-speed jets issue from nozzles and impinge on the target plate kept at some distance away from the nozzle tip. A very thin boundary layer is formed under the jet, and very high heat transfer coefficients are achieved in this zone. As fluid starts flowing radially outward, the boundary layer thickens, and heat transfer is adversely affected. This is an inherent disadvantage of using a single jet for cooling.

Uniform cooling is not possible with a single jet, and hence multiple jets are employed. The jet streams from adjacent nozzles, however, mix with each other and interact with the exiting fluid. The design of outlet ports becomes an important issue. Figure 3 shows the nozzle plate arrangement of a cooling system with multiple jets.
Table 1  Temperature drop across various thermal resistances

<table>
<thead>
<tr>
<th>Material type</th>
<th>$q = 1 \text{ MW/m}^2 (100 \text{ W/cm}^2)$</th>
<th>$q = 3 \text{ MW/m}^2 (300 \text{ W/cm}^2)$</th>
<th>$Q = 10 \text{ MW/m}^2 (1000 \text{ W/cm}^2)$</th>
</tr>
</thead>
<tbody>
<tr>
<td>Silicon wafer, 0.7 mm</td>
<td>4.7°C</td>
<td>14.3°C</td>
<td>47.6°C</td>
</tr>
<tr>
<td>TIM, 10 mm$^2$ C/W</td>
<td>10.0°C</td>
<td>30.0°C</td>
<td>100.0°C</td>
</tr>
<tr>
<td>Copper, 1 mm</td>
<td>2.5°C</td>
<td>7.5°C</td>
<td>25.0°C</td>
</tr>
<tr>
<td>Convective, $h =$</td>
<td>20,000 W/m$^2$K</td>
<td>50.0°C</td>
<td>100.0°C</td>
</tr>
<tr>
<td>Total $\Delta T$ needed</td>
<td>67.2°C</td>
<td>186.8°C</td>
<td>672.6°C</td>
</tr>
</tbody>
</table>

Jet impingement can be further classified as submerged and free surface jets. Submerged jets have a surrounding liquid through which the jet flows, whereas free surface jets are surrounded by a gas. In general, free surface jets have a lower heat transfer capability, unless evaporation is employed at the interface.

In recent years, a number of investigators have proposed different enhancement techniques to improve the performance of jet impingement cooling, particularly in the field of electronics packaging. Increasing the turbulence of the jet stream, modifying the target surface by coatings, and enhancing the surface structure are some of the commonly employed enhancement techniques. Table 3 outlines recent investigations on jet impingement in chronological order [7–20].

Garimella and Rice [21] investigated the distribution of the heat transfer coefficient for a submerged jet in confined and unconfined spaces for dielectric fluid FC-72. Two peaks were observed, one at the stagnation zone and another at some radial distance from the center line. For smaller diameter jets, both peaks were more pronounced than for larger diameter jets. The secondary peak was associated with probable turbulence and recirculation caused by the confinement due to the upper wall. As the jet diameter increased, the peak became flatter, indicating a broader stagnation zone. Further increase in nozzle diameter caused the heat transfer coefficient to increase further due to the recirculation.

Oh et al. [7] developed liquid jet cooling modules for very high heat fluxes, those encountered in applications such as plasmas, optical beams, and semiconductor laser arrays. At very high heat fluxes, the mechanical failure of the target plate becomes an important issue due to the production of large thermal stresses.

Table 2  Thermal resistances (mm$^2$-C/W) for the best available TIM, adapted and modified from Samson et al. [2]

<table>
<thead>
<tr>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Elastomer</td>
<td>100</td>
<td>70</td>
<td>60</td>
<td>50</td>
<td></td>
</tr>
<tr>
<td>Grease</td>
<td>30</td>
<td>20</td>
<td>20</td>
<td>10</td>
<td></td>
</tr>
<tr>
<td>PCM</td>
<td>—</td>
<td>25</td>
<td>20</td>
<td>10</td>
<td></td>
</tr>
<tr>
<td>PCMA</td>
<td>—</td>
<td>—</td>
<td>10</td>
<td>5</td>
<td></td>
</tr>
<tr>
<td>Gel</td>
<td>—</td>
<td>30</td>
<td>20</td>
<td>8</td>
<td></td>
</tr>
<tr>
<td>Indium solders and</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>2.65-4</td>
</tr>
<tr>
<td>CNT/nanoparticle composites</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

Induced mechanical stress was also evaluated in this study. Dissipations as high as 17 MW/m$^2$ (1.7 kW/cm$^2$) were achieved with single-phase water. The heater area was 1000 mm$^2$, and the surface temperature was greater than 600°C. This dissipation rate was associated with a very high Reynolds number of 200,000 and an inlet pressure of 580 kPa.

Wen and Jang [9] investigated heat transfer between constant heat flux plates and air jets with and without swirl inserts over a Reynolds number range between 500 and 270,000. Figure 4 shows the helical inserts used to give a swirling pattern to the flow.
flow. Symmetric vortices were formed, and their strength decreased as the spacing to jet diameter ratio increased. The vortices were yawed as a result of complex interactions between the centrifugal and buoyancy forces in the swirling streams. The vortices produced after the impingement were recorded. The heat transfer on the impinged surface was highest at the radial location where the jet flow cone impinged the surface. However, a more uniform radial distribution of Nu was obtained without the swirl. An important finding in this study was that the swirl enhanced the heat transfer as a result of air-induced turbulence. The performance enhancement over the plain nozzles was found to be relatively modest, between 9 and 14%.

Fabbi et al. [11] carried out single-phase experiments using water and FC-40 with jet diameters of 50–250 μm. The jet arrays had 61–397 jets and were capable of removing more than 2.5 MW/m² (250 W/cm²) from a 19.5 mm diameter copper surface, which represented the back side of the chip. However, the surface temperatures reached values as high as 350°C.

Later, Wang et al. [13] used two-phase, multiple microjet arrays of diameters 40–76 μm to cool VLSI chips. Although the experiments were performed in an open loop, the study was directed as a potential technology for IC chip cooling. The wall temperature rise was around 100°C.

Kanokjaruvijit and Martinez-botas [14] used an eight-by-eight array of jets on a dimpled surface. The dimples were used as turbulence promoters and tests were carried out at a Reynolds number of 11,500. The dimples acted as tools to thin the boundary layer and enhance the heat transfer over the flat portions between dimples. The effect of cross-flow was also investigated, with air as the working fluid. They found that the dimpled surface did not always enhance the heat transfer, and the cross-flow of the spent air played an important role. For the minimum cross-flow arrangement, impingement phenomena dominate, and the recirculation inside the dimples was not carried away by the flow. In case of maximum cross-flow, the impact of impingement was carried downstream as a result of the channel flow effect, and higher heat transfer coefficients were obtained.

Liu and Qiu [17] studied the CHF in the stagnation zone when using a superhydrophilic surface obtained by spraying TiO₂ and irradiating it with UV rays. It gradually decreased the contact angle with water over time and eventually reached zero. They found that boiling incipience was greatly delayed, but the heat transfer in the nucleate boiling region was enhanced. A superhydrophilic surface caused an increase in the CHF by as much as 50%. These tests were performed at a very high Reynolds number of 40,000, and a subcooling of 80 K and heat fluxes greater than 10 MW/m² (1 kW/cm²) were realized under certain conditions.

Natarajan and Bezama [20] developed a ceramic submerged microjet cooling device that demonstrated a cooling capacity of 2.5 MW/m² (250 W/cm²) with a pressure drop of less than 70 kPa. The low pressure drops are vital in electronics cooling from system reliability considerations. This device had 1,600 jets and 1,681 interstitial returns that required a very complex design of outlet manifolds. Figure 5 gives the plain view of inlet and return jets integrated with the chip. This device was tested with a heated silicon chip. Water was used in the single-phase mode and operated at Reynolds numbers less than 500 to limit the pressure drop. Multilayer ceramic technology is used for making intricate microchannels and interlayer connecting vias inside a multilayer ceramic substrate.

Pavlova and Amitay [19] investigated synthetic jet heat transfer mechanisms. Synthetic jets are produced by applying controlled excitation to the jets by the flapping motion of a piezo disk attached above the orifice. The synthetic jets require less complicated setup than jet impingement. These jets are excited by a piezo-electric sensor at 400 and 1200 Hz. Although the heat fluxes dissipated were quite small, the investigators showed that synthetic jets are more efficient than continuous jets in specific localized cooling applications.

**Summary of Jet Impingement Research and Future Directions**

Jet impingement heat transfer relies on high velocity jets to remove large quantities of heat in the forced convection regime. Under nucleate boiling mode, high flow velocities imply increased bubble departure frequency and higher heat transfer coefficients. Jets offer a good solution for localized cooling. However, for chip cooling, the surface temperature variation needs to be limited to just a few degrees. This necessitates the use of multiple jets. With multiple jets, the interaction of jets becomes crucial. Hence the nozzle arrangement is important. At the microprocessor cooling level, management of the inlet and outlet jets and manifolds becomes an even more critical issue. From Table 3, it is evident that many investigators have focused on the arrangements of different types of nozzles to optimize the cooling based on the surface temperature distribution. Spent fluid management also seems to be linked with the spatial distribution of the jets. If jet impingement is to be considered as a future cooling option for microelectronics purposes, then modeling of the flow distribution will prove to be very useful. Because jet
<table>
<thead>
<tr>
<th>Year, author</th>
<th>Operating conditions</th>
<th>Fluid</th>
<th>q max (W/cm²), (W/m²K)</th>
<th>h (W/m²K)</th>
<th>T S (°C), ΔT sub (°C), A (mm²)</th>
<th>p (kPa), Q (ml/min), V (m/s)</th>
<th>Enhancement technique/major findings</th>
</tr>
</thead>
<tbody>
<tr>
<td>1998, Oh et al. [7]</td>
<td>Single-phase, submerged jet, closed loop</td>
<td>Water</td>
<td>1700, —</td>
<td>600 (rise), —, 1000</td>
<td>580, —, 47</td>
<td>Liquid jet-array cooling modules for very high fluxes</td>
<td></td>
</tr>
<tr>
<td>2001, Zhang et al. [8]</td>
<td>Two-phase, closed loop</td>
<td>Water</td>
<td>45, 2200</td>
<td>80 (rise), —, 100</td>
<td>172 (drop), 3.5, 30</td>
<td>Micromachined single jets on simulated silicon chip</td>
<td></td>
</tr>
<tr>
<td>2003, Bintoro et al. [10]</td>
<td>Two phase, closed loop</td>
<td>Water</td>
<td>177, 72950</td>
<td>95; —, 113</td>
<td>—, 288, —</td>
<td>Practical cooling implementation with triple technologies, Rankin cycle, JI and minichannel heat exchanger</td>
<td></td>
</tr>
<tr>
<td>2004, Silverman and Nagler [12]</td>
<td>Closed loop</td>
<td>Water</td>
<td>1000, —</td>
<td>200, —, 1000</td>
<td>—, 3333, 50</td>
<td>JI cooling for high-energy accelerator targets</td>
<td></td>
</tr>
<tr>
<td>2004, Wang et al. [13]</td>
<td>Two phase, open loop</td>
<td>Water</td>
<td>90, 30000</td>
<td>100 (rise), —, 100</td>
<td>117 (drop), 8, —</td>
<td>Micromachined jets for cooling of VLSI chips</td>
<td></td>
</tr>
<tr>
<td>2005, Kanokjanuvijit and Martinez-Botas [14]</td>
<td>Single phase, open loop</td>
<td>Air</td>
<td>—, —</td>
<td>—, —</td>
<td>—, —</td>
<td>8×8 array jet impinging on staggered array of dimples with different crossflow arrangements</td>
<td></td>
</tr>
<tr>
<td>2006, Sung and Mudawar [16]</td>
<td>Single phase, closed loop</td>
<td>PF-5052</td>
<td>~ 90,—</td>
<td>—, —, 200</td>
<td>70, —, —</td>
<td>Hybrid device of slot JI into a microchannel</td>
<td></td>
</tr>
<tr>
<td>2006, Mozumder et al. [18]</td>
<td>Two phase, closed loop, free jet</td>
<td>Water</td>
<td>750, —</td>
<td>400, 80, 6939</td>
<td>—, —, 15</td>
<td>Investigate parameters controlling maximum heat flux during quenching process</td>
<td></td>
</tr>
<tr>
<td>2006, Natarajan and Bezama [20]</td>
<td>Single phase, closed loop, submerged jet</td>
<td>Water</td>
<td>250, 52000</td>
<td>&lt;85, —, 324</td>
<td>53, —, 1.6</td>
<td>Liquid microjet cooler with 1600 microjets and 1681 interstitial returns using glass ceramic material</td>
<td></td>
</tr>
</tbody>
</table>
impingement largely relies on high velocity flow over a heated surface, structured surfaces are not effective in heat transfer enhancement. Swirl pattern and jet geometry, configuration, and velocity are the parameters that need to be optimized in enhancing the performance of a jet impingement system under given pressure drop constraints.

SPRAY COOLING

Spray cooling is being pursued as a promising technology for electronics cooling applications. Figure 6 shows a schematic representation of a spray cooling system using multiple sprays to cool an array of IC chips.

Spray cooling relies on liquid droplets impinging on the heated surface. When this happens, a very thin liquid film is formed, and through the evaporation of this film, a large amount of heat is transported from the surface. The evaporation of the thin film and secondary nucleation in the film reduces the wall superheat requirement, and hence spray cooling has the potential to remove heat at much lower surface temperatures as compared to jet impingement. When sprays are used with single-phase liquid (without evaporation, liquid film is drained away), droplet impingement induces agitation in the liquid, causing the enhancement.

Although spray cooling is being pursued aggressively, there are still many unknowns due to the complex heat transfer mechanism. Many investigators in recent years have tried to understand the underlying mechanism of heat transfer in spray cooling systems in an effort to enhance their performance. Table 4 gives a brief summary of recent investigations [22–38].

Tilton et al. [22] investigated direct liquid spray cooling of high heat flux electronics. Miniature nozzles were developed that could be incorporated in the packaging. They simulated a high heat flux multi-chip module and used FC 72/87 as working fluids. A heat flux of 144 W/cm² was achieved with a flow rate of 1366 ml/min over a 645 mm² area. High inlet pressures of up to 930 kPa were required. They found that higher mass flow rate resulted in an increased heat transfer rate.

Ortiz and Gonzalez [24] studied the effects of mass flow rate, surface roughness, and degree of subcooling on spray cooling potential. In an open loop, they removed more than 5 MW/m² (500 W/cm²) at 48.5 ml/min over a 122 mm² heat transfer area. An air-assisted spray was used. Air helps in the atomization of droplets and also induces secondary nucleation. Inlet pressures of 861 kPa were required. It was found that heat flux increased with higher mass flow rates as well as the surface roughness; however, higher inlet subcooling yielded a decrease in flux values. Subcooling was found to help in lowering the surface temperature, but it adversely affected the heat transfer coefficient.

Lin and Ponnappan [28] simulated the cooling of high-power laser diodes with a multi-nozzle spray in a confined chamber within a closed loop. Pressure drops of 70–315 kPa were measured across the nozzle. Eight miniature nozzles were used with a swirl insert in each nozzle. Swirl caused wider spray angle and also intensified liquid break-up. They achieved a CHF of more than 500 W/cm² with water as the working fluid at a volumetric flux of 0.0249 m³/m²·s. They were probably the first ones to study the effects of non-condensable gases on spray cooling system performance. It was found that the non-condensable gases adversely affect the overall heat transfer of the closed loop spray cooling systems.

Horacek et al. [33] investigated single nozzle spray cooling with the effects of dissolved gases. The primary effect of non-condensable gases was an increase in saturation temperature and hence the liquid enters in a subcooled state. They obtained a design flow rate of 32 ml/min, with a pressure drop of 371 kPa across the nozzle. The authors specified two types of subcooling: one using a chiller to reduce the temperature of the liquid spray below saturation, called thermal subcooling (TS), and the other by gassing the chamber to increase the saturation temperature, called gas subcooling (GS). It was seen that, in agreement with previous research, TS increased the heat transfer as a result of sensible heating, and there was no change in the temperature at which CHF occurred. The result of more dissolved gases and hence a higher GS was to increase the CHF while at the same time increasing the temperature at which CHF occurs. It was found that in the cases of predominant GS, even with lesser total subcooling (TS + GS), greater heat transfer was observed. The presence of gases causes additional single-phase convection over the heater areas not covered by the droplets and may also contribute to additional evaporation. Authors also suggest that gas may cause bubbles to nucleate within the drops, spreading the liquid over a larger area. The effect of non-condensable gases on condenser performance degradation and the overall system was not investigated.

Bash et al. [29] demonstrated how thermal inkjet technology can be effectively utilized to cool a heat source with non-uniform power density. They achieved CHF of 2.7 MW/m² (270 W/cm²). The advantage cited by the authors is that better spatial flow variation can be achieved to match the local cooling requirement. With the thermal inkjet printing technology, an array of nozzles is assembled with individual operable resistive heaters, and these nozzles can be arranged and fired independently. Also, fast response in flow rate changes is possible due to high

Figure 6 Illustration of spray cooling application to chip cooling, the Laboratory of Physical Sciences, University of Maryland.
resistor cycling frequency. At low volumetric flux, the spray is affected by the vapor rising from the heater surface because of the lower spray momentum. However, sustained operation with higher volumetric fluxes was achieved at higher flow rates.

Silk et al. [31, 39] investigated spray cooling on enhanced structures. Copper surfaces were enhanced with either straight copper channels or cubic pin fins and pyramid structures. For straight-finned surfaces, they found that enhancement decreased with increasing fin length. The finned surface promoted nucleation at lower wall temperatures than flat surfaces. The effect was more pronounced under two-phase conditions. The second investigation with cubic fins and pyramids also studied the effects of degassing. The highest heat flux was obtained for gassy conditions of stator surface array cooling for simulated chip module cooling.

### Table 4 Recent experimental studies on spray cooling

<table>
<thead>
<tr>
<th>Year, author</th>
<th>System details</th>
<th>Fluid</th>
<th>q max (W/cm²), h (W/m²°C)</th>
<th>T_s (°C), ΔT_{sub} (°C), A(mm²)</th>
<th>p (kPa), Q (ml/min), V (m/s)</th>
<th>Enhancement technique/principal investigation</th>
</tr>
</thead>
<tbody>
<tr>
<td>1992, Tilton et al.</td>
<td>Two phase, closed loop</td>
<td>FC-87, FC-72</td>
<td>144, 2200</td>
<td>~55, 5–25, 645</td>
<td>930 (inlet), 1366, —</td>
<td>Simulated chip module cooling</td>
</tr>
<tr>
<td>1996, Yang et al.</td>
<td>Air-assisted</td>
<td>Water</td>
<td>&gt;800–20000</td>
<td>~150, 121</td>
<td>446 (inlet), 33/17400</td>
<td>Effect of liquid and secondary gas flow on boiling investigated</td>
</tr>
<tr>
<td>1999, Ortiz and</td>
<td>Air-assisted, two</td>
<td>Water</td>
<td>~500, —</td>
<td>120, 76, 122</td>
<td>517–861 (drop), 24–48.5, —</td>
<td>Effects of subcooling, impact angle, and surface roughness</td>
</tr>
<tr>
<td>Gonzalez [24]</td>
<td>phase, open loop</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>2002, Rini et al.</td>
<td>Two phase, closed loop</td>
<td>FC-72</td>
<td>80, —</td>
<td>68, —, 100</td>
<td>275 (inlet), —, —</td>
<td>Study of bubble behavior in saturated spray cooling of FC-72</td>
</tr>
<tr>
<td>2002, Xia [26]</td>
<td>Two phase</td>
<td>Water</td>
<td>920, —</td>
<td>40 (rise), —, 100</td>
<td>240 (drop), —, 1–2</td>
<td>Piezo-electric vibrators to atomize the droplets. Micro nozzle arrangement.</td>
</tr>
<tr>
<td>2003, Cabrera and</td>
<td>Two phase, air-assisted, open</td>
<td>Water</td>
<td>~500, —</td>
<td>500, 25–78, 38</td>
<td>180, 34–60, 9–15</td>
<td>Correlation for heat flux in nucleate boiling regime</td>
</tr>
<tr>
<td>Gonzalez [27]</td>
<td>loop</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>2003, Lin and</td>
<td>Two phase, closed loop</td>
<td>FC-87, FC-72,</td>
<td>500/490 (water/methanol), 97800/64500 (water/methanol)</td>
<td>~120, 3–10, 200</td>
<td>69–310 (drop), —, —</td>
<td>8 miniature nozzles with swirl inserts used for closed loop, high heat flux cooling</td>
</tr>
<tr>
<td>Ponnappan [28]</td>
<td></td>
<td>methanol, water</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>2003, Bash et al.</td>
<td>Two phase, open loop</td>
<td>Water</td>
<td>270, —</td>
<td>—, —, 281.25</td>
<td>—, 4.56, —</td>
<td>Use of thermal inkjet printer technology for microprocessor cooling</td>
</tr>
<tr>
<td>2004, Kim et al.</td>
<td>Two phase, air-assisted</td>
<td>Water</td>
<td>2.1, 500</td>
<td>—, —, 2500</td>
<td>48 (inlet), 2.4, —</td>
<td>Microporous coating of ABM on the heater surface</td>
</tr>
<tr>
<td>2004, Silk et al.</td>
<td>Degased/gassed chamber, two</td>
<td>PF-5060</td>
<td>156, 23956 (gassed chamber)</td>
<td>86.7, —, 200</td>
<td>379 (drop), 200, —</td>
<td>2×2 nozzle array in conjunction with structured surface: cubic pin fins, pyramids, and straight fins</td>
</tr>
<tr>
<td>2004, Chen et al.</td>
<td>Two phase, closed loop</td>
<td>Water</td>
<td>708.1, —</td>
<td>132, —, 100</td>
<td>551 (inlet), —, —</td>
<td>Investigation of effect of efficiency of liquid usage on CHF</td>
</tr>
<tr>
<td>2005, Horacek et al.</td>
<td>Two phase, closed loop</td>
<td>FC-72</td>
<td>80, —</td>
<td>90, 39.6, 49</td>
<td>—, —, —</td>
<td>Effect of dissolved gases on single nozzle spray cooling</td>
</tr>
<tr>
<td>2005, Coursey et al.</td>
<td>Two phase, closed loop</td>
<td>PF-5060</td>
<td>131, —</td>
<td>80, —, 198.8</td>
<td>365, 96, —</td>
<td>Copper heat sinks with straight fins of varying fin lengths</td>
</tr>
<tr>
<td>2005, Fabbri et al.</td>
<td>Single phase, closed loop,</td>
<td>Water</td>
<td>93, 15,000</td>
<td>—, —, 43.4</td>
<td>843 (drop), 81.56, —</td>
<td>Comparative study of microjets and spray nozzles</td>
</tr>
<tr>
<td>2005, Pautsch and</td>
<td>Two phase, closed loop,</td>
<td>FC-72</td>
<td>77.8, ~14,000</td>
<td>—, —, 103–301 (inlet), 23.4–289, —</td>
<td></td>
<td>Detail parametric study of nozzle array pattern effects</td>
</tr>
<tr>
<td>Shedd [36,37]</td>
<td>degassed</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>2006, Sleiti and</td>
<td>Single phase, closed loop</td>
<td>Polyalphaolein</td>
<td>—, 1100</td>
<td>170 (rise), —, 22528</td>
<td>758 (inlet), —, —</td>
<td>Triangular and rectangular array cooling for simulated conditions of stator surface</td>
</tr>
<tr>
<td>Kapat [15]</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>2006, Hsieh and</td>
<td>Two phase, open loop</td>
<td>Water</td>
<td>60, 4800</td>
<td>—, 77–82, 635.04</td>
<td>507 (inlet), —, 31.9</td>
<td>Micro structured silicon surface at low mass fluxes</td>
</tr>
<tr>
<td>Yao [38]</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

S. G. KANDLIKAR AND A. V. BAPAT 917
Amon et al. [40] discussed a MEMS-based droplet impingement embedded cooling device to remove 1 MW/m² (100 W/cm²) from the IC chips. The project, called EDIFICE, is designed with micronozzles that incorporate swirl nozzles and micro-injectors for jet breakup. Also included in EDIFICE are enhanced silicon surfaces for better heat transfer, and electrostatic microvalves for on-demand control of dielectric coolant flow rates. Dielectric fluid HFE-7200 was used because it is expected to give better atomization, which is crucial in this case as the operating space is very small. The cooling surface area was 25.4 × 25.4 mm². Irregularly shaped nozzles with hydraulic diameters of 150 μm were used with swirl inserts and injection pressures of 30–105 kPa. Tests carried out on a prototype removed 45 W/cm² of heat at 33.3 g/cm²-min.

Hsieh and Yao [38] studied the evaporation of water spray on microstructures at low mass fluxes. The authors found that the low Bond number of the microstructures is primarily responsible for the heat transfer enhancement. The three grooved surfaces, etched using DRIE technique on silicon, are illustrated in Figure 7.

The grooves on the heater surface help to retain the liquid film as well as spread the liquid to make it thinner and improve the evaporative heat transfer. The effect of dissolved nitrogen gas was found to have a negligible effect on heat transfer. It was found that the liquid film on a microtextured silicon surface holds the original contact line (after the drop impingement) during most of the evaporation period. That is, the surface tension-induced contraction is balanced by the generated capillary forces.

Hsieh and Yao [38] observed four stages for liquid film distribution, as shown in Figure 8. Heat transfer was found to remain unaffected by surface structure in the flooded and dryout regimes, as the surfaces have liquid and vapor coverings, respectively. However, in the thin film and partial dryout regions, the micro-textures enhanced the heat transfer by spreading the liquid film over a larger surface area. This is a result of capillary forces and surface wetting for a longer time period. It was observed that heat transfer deteriorated after the liquid film breakup and was related to the Bond number. Smaller Bond numbers produced better performance. Using water as the test fluid, heat transfer around 50 W/cm² was achieved, with more than 80°C of subcooling. Heat transfer coefficients of 4800 W/m²°C were reported.

**Summary of Spray Cooling Research and Future Directions**

Unlike jet impingement, spray cooling offers a much more uniform cooling and is a more attractive option for electronics.
cooling. Spray cooling mainly relies on evaporative heat transfer. Secondary nucleation, which is caused by the entrainment of air in the thin film, also enhances the heat transfer. Sehmbey et al. [41] gave a detailed explanation for spray cooling mechanisms. As evaporation is the key to heat transfer, capillary forces might play an important role in spray cooling by maintaining a layer of liquid on the heater surface. It is not a coincidence that in Table 4, many investigators can be seen trying different surface structures. The grooves in the surface help to retain the thin layer of fluid, and hence more heat is transferred. However, gas-assisted cooling may not be very conducive to electronics applications. Another way of atomizing droplets is to use pressure-assist nozzles. However, the high inlet pressures required to produce microdroplets remain a concern for the application of spray cooling in the thermal management of electronic systems.

MICROCHANNEL COOLING

Single-Phase Flow

Microchannels are identified as channels with a minimum channel dimension in the range of 10–200 μm [42]. The small hydraulic diameter leads to a high heat transfer coefficient even with laminar flow. At the same time, it leads to a very large pressure gradient for the circulating fluid. Microchannels can be used with single-phase flow of a liquid or with two-phase flow under flow boiling conditions.

Microchannels provide a high heat transfer coefficient due to the small passage dimensions employed. In addition, the area enhancement resulting from channels with high aspect ratio is also important. For example, utilizing a microchannel heated from three sides, 100 μm wide, 450 μm deep, with a wall thickness of 50 μm, yields a hydraulic diameter of 163.6 μm and an area enhancement factor of 6.67. This area enhancement is crucial in achieving a projected surface area-based heat transfer coefficient on the order of 500,000 W/m² °C in meeting the high heat flux challenges of 1 kW/cm² or higher for chip cooling application.

The large pressure gradient experienced in the small passages of microchannels can be effectively reduced by reducing the flow length. This is accomplished by designing the header with multiple inlets and outlets.

Table 5 shows some of the recent publications on both single-phase flow and flow boiling studies in microchannels published in the last two years. An earlier survey reported in [42] provides details of the work done until 2004.

A comprehensive survey of single-phase cooling technology with microchannels was provided by Kandlikar and Grande [43]. The main identifiable issues were high pressure gradients and fouling concerns. The high pressure gradients can be overcome by reducing the flow length. Multiple inlet/outlet headers will be needed at shorter distances along the flow length. This leads to heat transfer enhancement in the entry region.

Although the heat transfer coefficients in microchannels are very high, still higher heat transfer coefficients are needed to meet the heat flux challenges of 5–10 MW/m² (500–1000 W/cm²). Significant heat transfer enhancement can be achieved by employing the offset strip-fin geometry that is successfully used in compact heat exchangers. The enhancement potential of this technique is truly outstanding as the entry region lengths could be limited to only a few hundred micrometers.

Colgan et al. [3] employed the offset strip-fin geometry with a fin length of 500 μm and a channel width between 50 and 200 μm. The heat transfer coefficient for this arrangement was shown to be on the order of 200,000 W/m² °C [3, 44]. Further enhancement is possible through the effective usage of the offset strip-fin configuration. Figure 9 is a rendition of a section of the microchannel cooler geometry employed by Colgan et al. [3].

Kosar and Peles [45] studied the effect of 243 μm pin fins staggered in a rectangular channel with an effective hydraulic diameter of 99.5 μm. Their results indicated that the performance of this geometry was predictable using the conventional compact heat exchanger correlations for similar geometries above a Reynolds number of 50. Below Re = 50, some variations were noted, but it is not clear whether these are due to higher uncertainties at lower ranges or a specific phenomenon.

Header design remains an important consideration in a microchannel cooling system. The ability to provide and remove the cooling liquid at short distances along the microchannel flow length poses design challenges for manifolds. The objective is to provide uniform flow with a low pressure drop penalty in the distributor.

Flow Boiling in Microchannels

The high heat transfer coefficient resulting from the small hydraulic diameter in microchannels, combined with the inherently more efficient process of flow boiling, has long been expected to provide significant performance enhancement. The early experimental work indicated unstable flow behavior in microchannel
Table 5  Recent investigations after 2004 on microchannels

<table>
<thead>
<tr>
<th>Year, author</th>
<th>System flow conditions</th>
<th>Fluid</th>
<th>q max (W/cm²), h (W/cm²K)</th>
<th>TS (°C), ΔTSub (°C), Dh (μm)</th>
<th>Δp (kPa), Re</th>
<th>Enhancement technique/principal investigation</th>
</tr>
</thead>
<tbody>
<tr>
<td>2005, Lee et al. [50]</td>
<td>Rectangular microchannel, single phase</td>
<td>Water</td>
<td>—, 23,000</td>
<td>—, —, 318–903</td>
<td>—, 300–3500</td>
<td>Investigate validity of large scale correlations for predicting thermal behavior at microscale</td>
</tr>
<tr>
<td>2005, Colgan et al. [3]</td>
<td>Offset strip fins, single phase</td>
<td>Water</td>
<td>300, 130000</td>
<td>—, —, ~100</td>
<td>35, 26–282</td>
<td>Practical implementation of single phase silicon microchannel on to a high power chip</td>
</tr>
<tr>
<td>2005, Lee and Mudawar [51, 52]</td>
<td>Rectangular microchannel, two phase</td>
<td>R134a</td>
<td>93.8, 50000</td>
<td>—, —, —</td>
<td>660 (inlet), —</td>
<td>High heat flux micro channel cooling for refrigeration cooling applications</td>
</tr>
<tr>
<td>2005, Ghiu and Joshi [53]</td>
<td>Enhanced structured microchannel, two phase</td>
<td>PF 5060</td>
<td>45, —</td>
<td>—, —, —</td>
<td>—, —</td>
<td>Boiling performance from enhanced single layered structure</td>
</tr>
<tr>
<td>2006, Lee and Mudawar [54]</td>
<td>Rectangular microchannel, two phase</td>
<td>R134a</td>
<td>100, 40000</td>
<td>55, —, 200–500</td>
<td>~120</td>
<td>Microchannel evaporator for maintaining low surface temperatures</td>
</tr>
<tr>
<td>2006, Kosar et al. [49]</td>
<td>Rectangular microchannel, two phase</td>
<td>Water</td>
<td>614, —</td>
<td>—, —, 227</td>
<td>—, —</td>
<td>Suppression of flow boiling oscillations through inlet restrictors</td>
</tr>
</tbody>
</table>

flow boiling [49–51]. However, these issues seem to be now resolved [44, 52].

Table 5 summarizes some of the recent work published on microchannel flow boiling [3, 44, 45, 49–55]. It is seen that there is an increased focus on obtaining the heat transfer characteristics as well as CHF limits. Also, further work on understanding the fundamental mechanisms and developing new models is needed [56, 57]. This is an open area where more experimental data are needed. The application of enhanced geometries is also desirable, and some work in that direction has already begun in a number of research labs all over the world.

Critical heat flux is another important consideration. The experimental data available so far cover only the lower range of mass fluxes. Additional experimental work on CHF is needed with plain as well as enhanced microchannels.

It is conceivable that the IC chip may be cooled directly using any of these cooling techniques. Microchannels may be especially suited for this application, as the fluid can be easily contained within the cooling passages. The fabrication issues involved, however, with integrating large-scale material removal processes in microchannel production with the sub-micrometer scale IC circuits may prove to be difficult to resolve from a yield consideration. Electronics engineers may be reluctant to subject the expensive finished IC chips to further processing. If a non-dielectric fluid such as water is used for cooling the IC directly, the danger of leakage and accompanying catastrophic failure may pose an unacceptable risk. However, these options need to be carefully reevaluated in light of the future high heat flux dissipation requirements.

**COMPARISON OF COOLING OPTIONS**

**Thermohydraulic Performance Comparison**

Jet impingement technology relies mainly on the high velocity jet to provide a high heat transfer rate. A large number of jets are needed to provide a uniform heat transfer rate over a surface. Current performance needs to be further improved for meeting the 10 MW/m² (1 kW/cm²) target. The enhancement methods that are being experimented with JI are swirl and jet turbulence. The enhancement potential of these techniques is seen to be modest as per the above review.

Spray cooling and microchannels both offer high heat transfer performance, which can be further increased by incorporating specific methods. In the case of spray cooling, structured surfaces are seen to be extremely promising. Further research is expected to identify specific configurations that provide significant improvements. However, the pressures required in the spray systems are very high. Unless effective spray patterns (droplet size and distribution) can be obtained at relatively modest pressures (less than one atmosphere), it is rather difficult to implement spray technology in electronics cooling applications. This is a major challenge that needs to be addressed by the research community.
Microchannels in single-phase mode provide a very high level of performance by using enhanced surfaces such as offset-stripe fins. Multiple inlet/outlet headers effectively reduce the pressure drop requirement to less than 100 kPa. The optimization of microchannel geometry is expected to provide an effective solution to the high flux cooling requirement, with heat transfer coefficients in excess of 500,000 W/m²·°C.

Microchannels with flow boiling are expected to provide even better performance than the single-phase mode, although this is yet to be proven. A major obstacle is the critical heat flux limitation; working fluid poses another challenge. Water systems are able to provide very high level of performance, but need to be operating under vacuum conditions [58]. Refrigerant performance is generally a factor of 5–10 lower than water systems, both in heat transfer coefficient and CHF limits. This is an open area where considerable research efforts are needed.

**Direct Cooling of IC Chips with Boiling Refrigerants**

An option that has long been overlooked is providing the refrigerant under flow boiling mode directly over the IC chips. Such a technique, using a microgap above the IC chip, was illustrated by Kandlikar [42]. The advantages of this configuration are the elimination of the thermal resistances of the silicon layer, TIM, and heat spreader substrate. Although the heat transfer coefficients of such a device may be below a microchannel device, the effective performance will be comparable when the overall thermal resistances are taken into account. The concern regarding damage to the ICs could be easily addressed by employing developments in inkjet printer technology such as protecting the heater surface with a titanium oxide layer. Further research is needed to establish the heat transfer coefficient and CHF limits for this configuration. Once thermal performance is verified, the system needs to be evaluated from a packaging perspective.

**Structural and Leakage Considerations**

Packaging issues are often overlooked in evaluating cooling options. The pressures required by spray cooling systems are unacceptably high. System reliability becomes a major issue when high pressures—often exceeding several atmospheres—are employed. Another issue that needs to be considered is the leakage concern. Providing a large unsupported area for accommodating a spray or a jet system will cause large deflections into the silicon chips. Confining the working fluid in this space has been a major challenge. The direct cooling option with a microgap will also face the same problems, although the pressures may not be as high as in jet or spray systems. In this regard, microchannels provide an effective solution, as the multiple channel walls provide more contacting points to add to the strength of the unit.

Leakproof operation has been demonstrated in such coolers [3]. The structural redesign of the spray cooling option may overcome some of these problems by providing multiple intermediate support structures.

**CONCLUDING REMARKS**

A critical review of jet impingement and spray and microchannel cooling options is presented in this paper, with a special emphasis on developments in this field over the last five years. The thermohydraulic performance comparison indicates that the single-phase microchannel option is most viable at this time, and additional research on enhanced surfaces and flow boiling systems is needed to make them more effective. Further research in spray systems is expected to make them an attractive alternative as well. However, pressures in spray systems need to be reduced in order to address reliability issues. Structural and leakage considerations again make the microchannels most attractive at the current time. The direct cooling of IC chips with microgaps provides an interesting alternative, but significant research efforts are needed to make it a viable option. Further research is needed in all these systems from a packaging perspective.

**NOMENCLATURE**

- $A$: surface area, mm²
- $h$: heat transfer coefficient, W/m²·°C
- $Nu$: Nusselt number
- $p$: pressure, kPa
- $q^\prime\prime$: heat flux, W/cm²
- $Q$: flow rate, ml/min
- $Re$: Reynolds number
- $T_s$: surface temperature, °C
- $\Delta T_{sub}$: subcooling degree, °C
- $V$: flow velocity, m/s

**REFERENCES**

heat transfer engineering vol. 28 no. 11 2007


Satish Kandlikar is the Gleason Professor of Mechanical Engineering at RIT. He received his Ph.D. from the Indian Institute of Technology in Bombay in 1975 and has been a faculty member there before coming to RIT in 1983. His research is mainly focused in the area of flow boiling, microchannels, roughness effects, and water management in fuel cells. He has published more than 175 journal and conference papers. He is a fellow member of ASME and has been the organizer of the international conferences on microchannels and minichannels sponsored by ASME.

Akhilesh Bapat is a M.S. student at the Rochester Institute of Technology in New York. He completed his Bachelor’s degree in mechanical engineering from Government College of Engineering, Pune, India, in 2004, after which he worked for a year in Thermax India Ltd as a graduate engineer. He joined the Thermal Analysis Lab at RIT in 2005 and is currently working on single- and two-phase chip cooling.