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High Flux Heat Removal with Microchannels—A Roadmap of Challenges and Opportunities

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Heat fluxes in IC chips and other electronics equipment have reached the current limits of air-cooling technology. Some of the applications require heat fluxes well beyond the limit of 100 W/cm^2, thus demanding advanced cooling solutions. Liquid cooling technology has also received attention as the advances in single-phase liquid cooling in microchannels have shown considerable promise. The extension of compact heat exchanger technology to microscale applications offers many new possibilities. The liquid cooling technology is expected to reach heat dissipation rates as high as 10 MW/m^2 (1 kW/cm^2) with enhanced microchannels and a junction-to-air temperature difference of 50°C. The challenges facing flow boiling systems are also evaluated. This paper reviews the fundamental technological developments in liquid cooling as well as flow boiling and presents possible solutions in integrating the cooling system with a building's HVAC unit in a large server-type application. The opportunities and challenges are described in an attempt to provide the roadmap of cooling technology for cooling high flux devices in the next decade.

INTRODUCTION

High heat flux removal is a major consideration in the design of a number of systems, such as high-performance computer chips, laser diodes, and future nuclear fusion and fission reactors. Microchannels and minichannels are naturally well suited for this task, as they provide a large heat transfer surface area per unit fluid flow volume. They provide an efficient way to remove heat from a surface but pose challenges in bringing fresh coolant to the heated surface and returning it to the cooling system before the coolant reaches the stringent temperature rise limits. The problem is interlinked with the performance issues of the cooled systems, which dictate a lowering of the surface temperature to increase reliability, speed of the processor units, or other overall system considerations.

Microchannels were first proposed for electronics cooling applications by Tuckerman and Pease [1], who employed the direct circulation of water in microchannels fabricated in silicon chips. The microchannel heat sink was able to dissipate 7.9 MW/m^2 with a maximum substrate temperature to inlet water temperature difference of 71°C. However, the pressure drop was quite large at 200 kPa with plain microchannels and 380 kPa with pin fin-enhanced microchannels. Another major milestone was achieved by Phillips [2], who analyzed the heat transfer and fluid flow processes in microchannels and provided detailed equations for designing microchannel geometries.

The use of enhanced microchannels was suggested by Tuckerman and Pease [1], Phillips [2], Steinke and Kandlikar [3], and Kandlikar and Grande [4], among others. Colgan et al. [5] provided the results of a practical implementation of enhanced microchannels with a strip-fin geometry. Recently, Kandlikar and Upadhye [6] presented a detailed set of equations and the results of an optimization procedure for selecting microchannel flow geometries under a given pressure drop constraint.

The implementation of microchannel heat sinks in a notebook computer was discussed by Pokharna et al. [7]. They considered the operational issues of a microchannel two-phase loop and also discussed the manufacturing and cost issues associated with mass producing such systems.

The developments on incorporating flow boiling in microchannels have been somewhat limited because of the issues

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associated with the stable operation of flow boiling systems and critical heat flux limitations. Another issue that is becoming apparent is the requirement of low operating surface temperatures that cannot be easily accomplished with water as the phase change fluid unless the cooling system is operated at subatmospheric pressures. Experimental data with refrigerants is not yet available to guide a designer in the selection of a proper working fluid to meet the desired high heat transfer coefficient requirement as well.

Although the water cooling of IC chips has received considerable attention in the past, the integration of single-phase microchannel heat exchangers in the overall cooling system has received little attention. The issues related to centralized versus distributed systems and practical options for secondary coolants—air-cooling versus water cooling in conjunction with a cooling tower or refrigeration system—also need to be carefully evaluated.

With the above milestones and issues in mind, the objectives of the present paper are set as follows:

1. identify critical issues associated with high heat flux removal systems
2. review the published literature to develop a suitable design procedure for microchannel heat exchangers
3. identify possible system-level solutions that will enable the dissipation of a higher heat flux under a given set of system constraints
4. identify concerns and challenges that need to be addressed in future single-phase and flow boiling research efforts.

The discussion is largely focused on cooling a cluster of high performance computers. The issues of freezing the primary coolant may not be relevant in these systems as they will be shipped with the primary coolant completely drained from the channel and passages.

OVERALL SYSTEM CONSIDERATIONS

Microchannel heat exchangers employed in high flux cooling applications generally receive heat through conduction from the heat source. The heat source is embedded in a silicon chip in the case of microprocessors or laser diodes and in the nuclear fuel elements in the case of nuclear reactor cooling applications. For the purpose of illustration, a high performance IC chip cooling application is considered here.

Figure 1 shows a schematic of a cooling scheme that is employed in cooling a high-power density processor chip. The actual heat-generating device is cooled with a primary fluid circulated in the primary heat exchanger that is in thermal communication with the device. The primary coolant is cooled in a secondary heat exchanger by a secondary coolant from a secondary cooling system.

In a computer chip cooling application or in the cooling of a high-power device such as an array of laser diodes, the primary

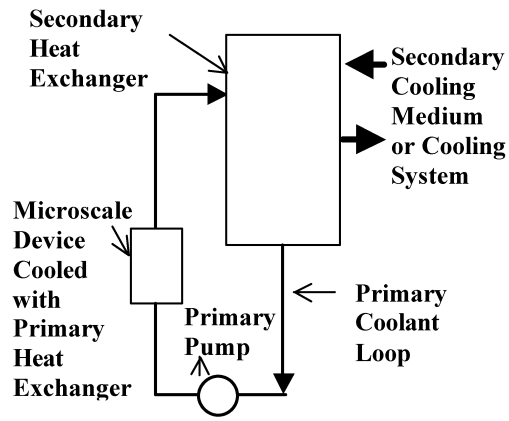


Figure 1 Schematic of a cooling system with primary and secondary heat exchangers and respective coolant loops.

coolant loop may consist of single-phase water or evaporating refrigerant. In such cases, the secondary cooling system is between the water/refrigerant and the room air. With the maximum design air temperatures between 40 and 45°C, the maximum available temperature difference for dissipating heat from a chip maintained at temperatures below 80°C is only 35–40°C. The two major thermal resistances appearing in the heat flow path from the heat-generating junction to the ambient air are addressed in the following sub-sections.

Thermal Resistance between the Junction and the Surface of the Microchannel Heat Exchanger

There is an internal thermal resistance between the junction and the chip or device surface, but it is generally low due to the high thermal conductivity of silicon and short thermal paths. However, junctions producing large amounts of heat present a challenging scenario as the thermal resistance of the silicon thickness may become of the same order as the convective resistance. In such cases, the localized cooling of the IC chip with structured microchannels fabricated on the chip face may be considered.

The thermal interface resistance between a heat sink and the chip is a major contributor to the total resistance in the heat flow path from the junction to ambient air. This is a topic of major research effort in the chip cooling industry, and many proprietary methods are expected to emerge to reduce this resistance to acceptable limits that are constantly being pushed lower. With the current thermal interface materials (TIM), this resistance is approaching as low as 20 mm^2 °C/W in commercial units to 5 mm^2 °C/W in experimental units under development [8], while the convective resistance for water with $h = 30{,}000$ W/m^2 °C and an area enhancement ratio (base to heat transfer surface area ratio) of four results in a convective resistance of 8.3 mm^2 °C/W.

Although etching channels directly onto the back of a functional IC chip or introducing pressurized water or other liquid at the chip level is not looked on favorably by electronics designers, heat dissipation limits will eventually dictate the type of

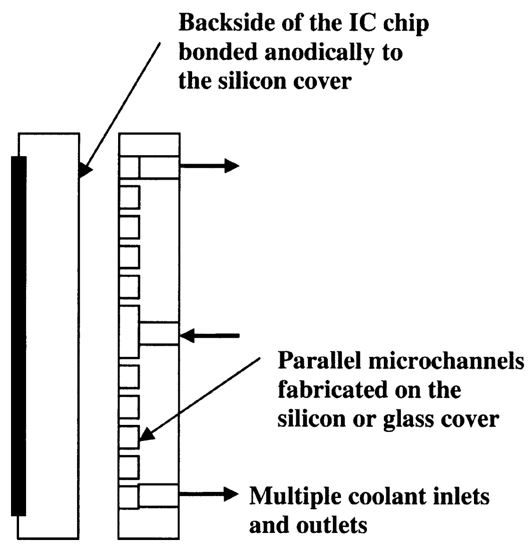


Figure 2 Schematic arrangement of an IC chip cooled with microchannels fabricated on a silicon or glass cover bonded anodically or glued to the backside of the chip (not to scale).

technology that will be introduced in cooling high heat flux devices. The placement of microchannels fabricated directly on the back side of a wafer is investigated by a number of researchers. Alternatively, microchannels can be fabricated in a silicon or glass cover that is anodically bonded or glued to the back side of the chip, as shown in Figure 2. This arrangement eliminates the concerns regarding post-processing of expensive IC chips for microchannel fabrication. Using refrigerants in these passages will also remove some concerns regarding bringing water in close proximity with the IC chips.

A number of issues need to be addressed before the schematic arrangement shown in Figure 2 can be implemented in practical systems, the major ones being the stresses induced in the chip due to bonding/gluing, circulation of pressurized coolant during operation, leakage concerns, and channel blockages due to particulates. Some of these issues will be discussed later in the paper. Other options, such as using a very efficient TIM to bond the silicon chip to a copper cold plate with microchannel flow passages, may also be considered.

Thermal Resistance from the Heat Sink to the Primary Coolant in the Microchannel Heat Exchanger

The thermal resistance in the heat sink arises from three sources: conduction resistance in the heat sink, including the fin effects; convection resistance between the microchannel surfaces and the coolant; and the resistance due to the temperature rise of the cooling fluid in the heat exchanger. As the microchannel passage dimensions become smaller, the heat transfer coefficients become higher for single-phase flow; however, there is an increased pressure drop penalty. The geometrical aspects arising due to fin thickness, channel depth, and aspect ratio also need to be considered in arriving at an optimum channel configuration for a given set of operating conditions.

The primary coolant in the stationary application has somewhat less stringent requirements (compared to desktop or portable computers) as far as the freezing temperatures are concerned. Because the entire system is housed indoors and the charging of the primary coolant is done onsite, water can be used as the primary fluid. With its excellent thermodynamic and thermophysical properties, it is well suited for this application.

Secondary Coolant Loop

The primary loop interfaces with a secondary loop ultimately to transfer the heat to the atmosphere. The secondary coolant loop may consist of a simple air-to-water heat exchanger with single-phase water from the primary loop. The procedure for designing this air-water heat exchanger follows the standard methods available for designing compact heat exchangers (e.g., Shah and Sekulic [9]).

Direct air cooling of the primary coolant provides a simple solution but introduces several undesirable features. The low heat transfer coefficients associated with air would require a fan to move the air through the compact heat exchanger. The accompanying noise may present a problem, unless the heat exchanger is located outside the room. This will lead to longer water lines and air duct lengths and increased pumping power requirements. The ambient air temperatures in hot tropical regions could very well reach 45–50°C, significantly reducing the available temperature difference between the junction and the air.

Another option is to use water as the secondary coolant. This option is attractive if a cooling tower is already installed and has enough capacity to handle the thermal loads. The water-to-water heat exchanger between the primary and secondary coolant loops will be more efficient and compact (as compared to air-to-water heat exchangers) due to the higher heat transfer coefficients on both sides of the heat exchanger. The cooling tower provides a lower design temperature for heat dissipation.

Integration with a Refrigeration System

The allowable temperature differences between the chip surface and the inlet temperature of the primary coolant in an air-cooled system may eventually become a limiting factor in spite of the development of highly efficient interface materials. In an effort to reduce this temperature difference, a refrigeration system offers an attractive option for cooling the secondary coolant.

Incorporating a refrigeration system will lower the primary coolant temperature, but the increased cost of adding a refrigeration system needs to be carefully evaluated. Another factor that needs to be considered is the minimum temperature allowable in the system because of the condensation concerns from the environment. Controlling the dew point temperature of the ambient air below the inlet temperature of the primary coolant fluid may prevent this localized condensation. Comparing this with an air-cooled system (no refrigeration), the inlet primary coolant temperature can be lowered from a design temperature of 45 to 50°C to a temperature of around 15°C. This provides an added temperature difference of 25 to 30°C, which offers a significant advantage in dissipating higher heat fluxes.

The cost of the overall system with refrigeration may seem to be high, but in the overall system configuration, the refrigeration system may be directly linked to the building HVAC system where available. Incorporating a water-to-water heat exchanger between the primary coolant loop and the building's chilled water loop will provide multiple benefits over an air-cooled secondary coolant system:

1. The size of a water-to-water heat exchanger will be considerably smaller than an air-to-water heat exchanger.
2. The added cost of the secondary coolant system may be minimal when integrating into the building HVAC system. In both cases, the air-conditioning capacity will be the same, since the building air is used in an air-cooled secondary cooling system. In this case, the design temperature of the air cooled system may also be lowered, since the cooling air from the air-conditioned spaces is used.
3. The noise will be considerably lower in a water-to-water heat exchanger due to the elimination of the fans and blowers in an air-cooled system.
4. Integration of the cooling system with the building HVAC system's chilled water loop is particularly suitable for cooling clusters of computers or servers.

A schematic of the overall cooling system for a server application is shown in Figure 3. A water-to-water heat exchanger is used to cool the primary coolant with the chilled water from the building HVAC system. A separate pump circulates the primary coolant to the microchannel heat exchangers serving each server. Additional controls are needed to ensure the desired mass flow rates through the microchannels. For cooling of the other components, such as the memory units, water-cooled cold plates may be employed. For other smaller and remotely located loads, it may be worthwhile to resort to the air cooling option integrated with the room air if needed.

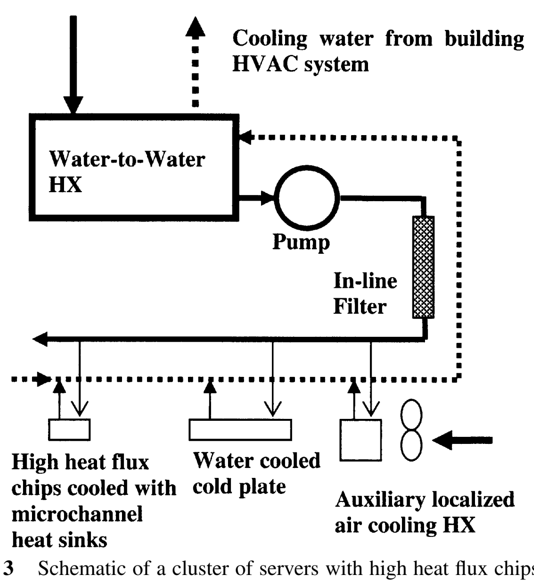


Figure 3 Schematic of a cluster of servers with high heat flux chips cooled with microchannel heat sinks, cold plates, and localized air cooling integrated with a secondary chilled water loop from the building HVAC system.

Additional Considerations

An overall system with either air cooling or water cooling (from a cooling tower or a chiller) will be similar to that shown in Figure 3, with an appropriate heat exchanger between the primary and secondary loops.

The high heat flux chips will be cooled directly with microchannel heat sinks. Additional loads due to other chips and devices can be removed by incorporating a cold plate that is cooled with the primary water loop. The heat generated by additional distributed devices may be removed by using an auxiliary air-to-water heat exchanger and a fan, as shown in Figure 3.

An in-line filter is an essential element in the primary loop as the microchannels are susceptible to channel blockage due to particulates present in the primary water loop. The maximum allowable particulate size depends on the cross-sectional size of the microchannels. In general, the smallest dimension in the microchannel heat sink needs to be considered in selecting the filter. Reducing the particulate filtration size increases the pressure drop across the filter. Although there are no guidelines available, it is suggested that the maximum filtration size be smaller than about one-tenth of the minimum channel dimension. If the particles have a tendency to agglomerate, the filtration sizes may be further reduced. Because the velocities and pressure drops are considerably higher in microchannels than in conventional channels, such relatively large particles are expected to be flushed out from the microchannels. The abrasive effect of the particles also needs to be investigated.

Pressure drop across the filter should be kept as low as possible to reduce the pumping cost. A bank of several filters in parallel is recommended. Online monitoring of the pressure drop across the filter elements may provide information regarding filter blockage and indicate a need for filter element replacement.

MICROCHANNEL DESIGN CONSIDERATIONS FOR SINGLE-PHASE LIQUID COOLING

Microchannel heat exchangers are increasingly being used in high heat flux cooling applications where the other competing technologies are spray cooling and jet impingement cooling. The focus here is on the microchannel heat exchangers.

The basic geometry being considered is shown in Figure 4. The total area being cooled is $W \times L$, with individual microchannel flow passage dimensions of $a \times b$. The wall separating the two channels is of thickness s and acts like a fin. The top cover is bonded, glued, or clamped to provide closed channels for liquid flow.

The channel dimensions a and b, the fin thickness s, and the coolant flow rate are the parameters of interest in designing a microchannel heat exchanger. The maximum allowable

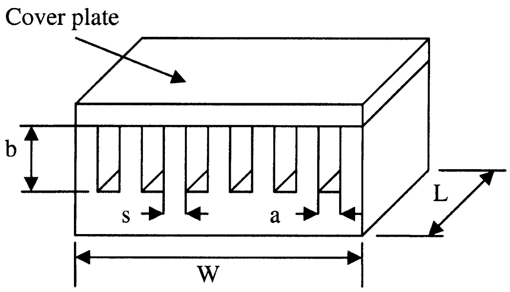


Figure 4 Schematic of a microchannel geometry for high heat flux cooling applications.

temperature of the channel surface, the minimum coolant inlet temperature, and the available pressure drop are the constraints. In addition, there are manufacturing and cost constraints that need to be considered in any practical system design.

A number of investigators have considered the microchannel design problem to arrive at an optimum heat exchanger configuration (e.g., Phillips [2], Knight et al. [10], Ryu et al. [11], Bergles et al. [12], and Upadhye and Kandlikar [13]).

Kandlikar and Upadhye [6] presented an optimization scheme for designing microchannel heat exchangers for cooling silicon chips with water. They considered a 10 mm × 10 mm silicon chip with a channel depth of 300 μm. A maximum chip temperature of 360 K was considered, and a water inlet temperature of 300 K was used. The results were presented as parametric plots.

Figure 5 is a contour plot showing the number of channels vs. the fin spacing ratio (= s/*a*) with solid lines representing fin thickness in μm and dashed-lines representing pressure drop in kPa. Such plots are useful in identifying the design envelope for a given heat flux. For example, considering a minimum fin thickness of 40 μm and a maximum pressure drop limit of 30 kPa,

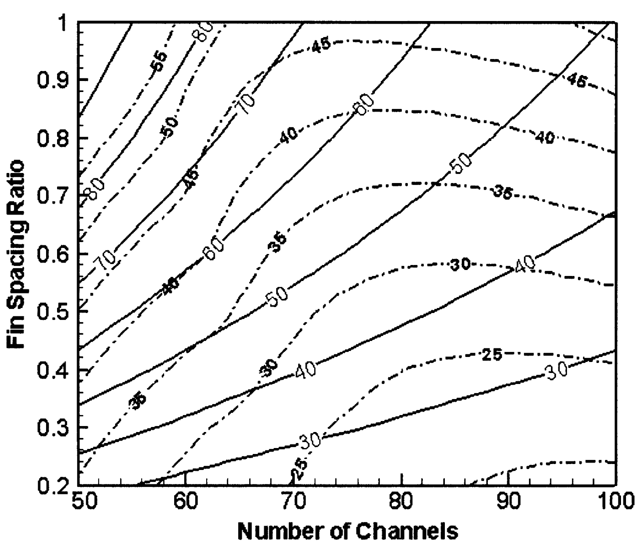


Figure 5 Contour plot of fin spacing ratio $\beta = s/a$ vs. number of channels. Pressure drop (dash-dot lines) and fin thickness in μm (solid lines) are parameters for water flow in plain rectangular microchannels (300 μm deep) on a 10 mm × 10 mm silicon substrate in a single-pass arrangement at a heat flux of 3 MW/m^2 (Kandlikar and Upadhye [6]).

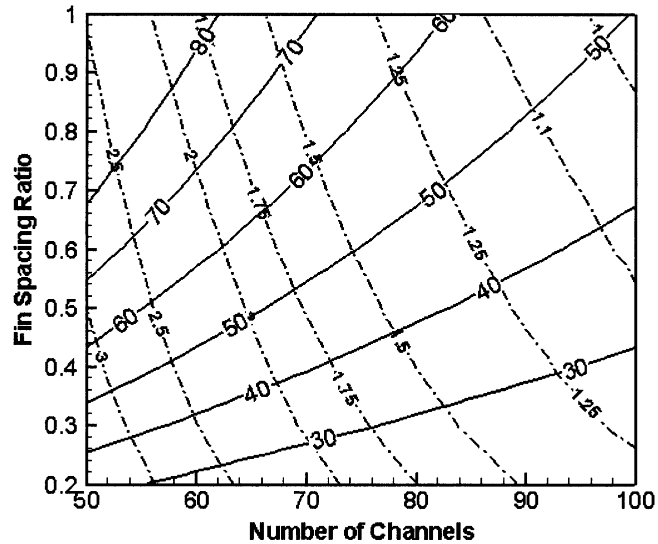


Figure 6 Contour plot of fin spacing ratio β vs. number of channels. Water flow rate in 10^{-3} kg/s (dash-dot lines) and fin thickness in μm (solid lines) are parameters for water flow in plain rectangular microchannels in a single-pass arrangement at a heat flux of 3 MW/m^2 (Kandlikar and Upadhye [6]).

we get the number of channels to be between 68 and 90. Corresponding water flow rates are obtained from Figure 6. Pumping power plots were also presented in a parametric form. Although the plots given by Kandlikar and Upadhye [6] are specific to a 10 mm × 10 mm chip, similar plots can be generated under other design conditions using the equations and procedure presented by them.

The use of a single-pass flow of water through the microchannel heat exchanger is not desirable for two reasons: the pressure drop is higher for the longer lengths, and larger mass flow rates are needed to keep the temperature rise of the water through the heat exchanger below the specified limit. The increased water flow requirement further adds to the pressure drop penalty. To overcome these problems, a split flow arrangement, shown in Figure 7, was employed by Kandlikar and Upadhye [6].

The split flow arrangement shown in Figure 7 can be further extended to provide multiple inlets and outlets as originally recommended by Tuckerman and Pease [1]. Reducing the flow length through the channel passages results in the lowering of the

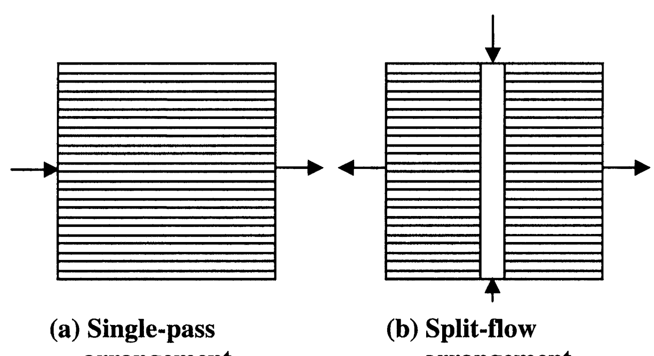


Figure 7 Schematic of single-pass and split flow arrangements showing fluid flow through microchannels (Kandlikar and Upadhye [6]).

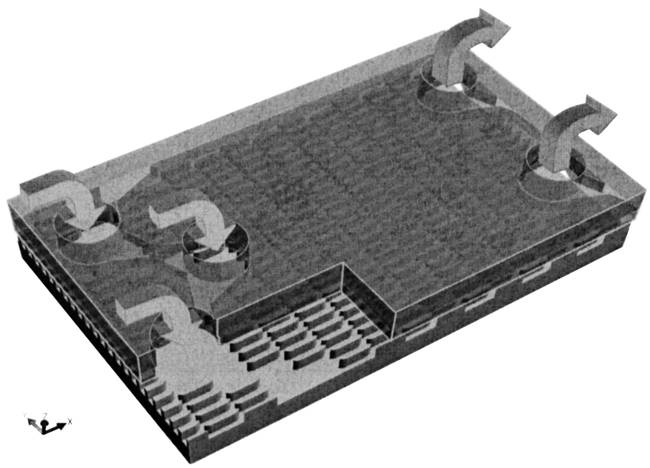


Figure 8 3-D view of the microchannel heat exchanger studied by Colgan et al. [5].

pressure drop. Another advantage of the split-flow arrangement is seen in the increased heat transfer coefficient near the multiple channel entrance regions due to entrance effects (thermally developing flow).

An increased heat transfer coefficient is very desirable for accommodating higher heat fluxes. Kandlikar and Grande [4] showed the need for enhanced channels, and Steinke and Kandlikar [3] presented various channel configurations that would provide higher heat transfer coefficients. Colgan et al. [5] studied the offset strip-fin arrangement in microchannels and obtained significantly higher heat transfer coefficients with a single-phase flow of water (see Figure 8).

The work by Colgan et al. [5] represents a major paradigm shift in the practical implementation of microchannel heat exchangers for high heat flux dissipation. The Nusselt numbers obtained by Colgan et al. [5] were as high as 25, with staggered fin arrangements at a pitch of 100 μm and fin lengths of 210 and 250 μm. The resulting heat transfer coefficients were as high as 130,000 W/m^2 °C. Although the associated friction factors were considerably higher, the multiple inlet and outlet headers kept the flow lengths to only 3 mm. The resulting pressure drop was less than 35 kPa.

The investigations described in this section point to the tremendous possibilities with single-phase water cooling. The effective utilization of advanced microfabrication technologies, combined with an understanding of the basic heat transfer and fluid flow phenomena, will be able to meet the challenges of dissipating heat loads in excess of 5 to 10 MW/m^2 (0.5 to 1 kW/cm^2). Further opportunities exist to improve the enhanced microchannel designs and the header arrangements to extend this limit even further.

MICROCHANNEL DESIGN CONSIDERATIONS FOR COOLING WITH FLOW BOILING

Flow boiling is another attractive option for high heat flux cooling. Its major advantages are:

1. the heat transfer coefficients in conventional flow boiling systems (>3 mm hydraulic diameter) are very high compared to the corresponding single-phase values
2. the mass flow rates are reduced because of the use of latent heat in carrying the heat away rather than just the sensible heat of the coolant (being limited by the available temperature rise in the coolant),
3. the heat removal process does not raise the temperature of the coolant as in the single-phase case, where the available temperature difference between the channel surface and the cooling fluid decreases along the flow length.

Although a two-phase system enjoys the above advantages over a single-phase system, recent advances have addressed many of the shortcomings of the single-phase systems. In particular, the enhancement features have increased the heat transfer coefficients to above 100,000 W/m$^2-$°C with a single-phase flow of water in channels with minimum passage widths of around 100 μm.

High heat flux cooling systems with flow boiling have lagged behind the single-phase liquid cooled systems because of some of the operational challenges that still remain to be resolved, such as:

1. The need for low pressure water or a suitable refrigerant to match the saturation temperature requirement for electronics cooling.
2. Unstable operation due to rapid bubble expansion and occasional flow reversal.
3. Low critical heat flux levels due to instability, the unavailability of critical heat flux data over a wide operating range, and a lack of fundamental understanding of the flow boiling phenomenon in microchannel passages.

Low Pressure Water or Refrigerant in Flow Boiling Systems

The challenges listed above are being addressed by researchers worldwide. The need to match the saturation temperature was addressed by Pokharna et al. [7], who proposed a low-pressure water system to reduce the saturation temperature to the desired level. The low-pressure water system is advantageous in that as the saturation temperature decreases, the latent heat of vaporization increases. However, the required vacuum in the primary loop is undesirable as air may leak into the system. The pressure drops are also expected to rise drastically because of the increase in the vapor specific volume at low pressures. Research on obtaining experimental heat transfer and pressure drop data for flow boiling of water in microchannels is immediately warranted.

The use of refrigerants in flow boiling systems is also being considered. Although the positive system pressure requirement is easily met with the proper selection of a refrigerant, the (fluorinated) refrigerants in general have a low heat transfer coefficient. Their performance suffers further from a low latent heat of vaporization, which leads to a high refrigerant circulation rate in the primary loop.

As a refrigerant, ammonia has highly desirable heat transfer properties, but its high saturation pressure at the desired operating temperatures (around 15 to 30°C), toxicity and corrosive nature make it impractical for electronics cooling applications.

The desirable characteristics of an ideal refrigerant in a flow boiling system may be listed as follows:

1. saturation pressure slightly above the atmospheric pressure at operating temperatures
2. high latent heat of vaporization
3. good heat transfer- and pressure drop-related properties (high liquid thermal conductivity, low liquid viscosity, low hysteresis for onset of nucleate boiling)
4. high dielectric constant, if applied directly on the chip
5. compatible with silicon (for direct chip cooling), copper (for heat sink applications), and other system components
6. low leakage rates through pump seals
7. chemical stability under system operating conditions
8. low cost
9. safe for human and material exposure under accidental leakages

A concerted effort to develop such a refrigerant or refrigerant mixture may lead to an ideal refrigerant. Research efforts in the chemical industry are therefore needed that are similar to their efforts on developing environmentally friendly refrigerants for air conditioning systems in the past decade. However, such efforts are unlikely to happen until the single-phase cooling option has been fully exploited.

Instabilities during Flow Boiling

Flow boiling instabilities pose a major concern in microchannels. Instabilities during flow boiling in conventional diameter tubes have been studied extensively in literature. A recent paper by Bergles and Kandlikar [14] summarizes the nature of these instabilities. These instabilities are present in both microchannels and minichannels, and an additional phenomenon has been identified that is unique to flow boiling in narrow channels. As a subcooled liquid enters the channels, it undergoes single-phase heating with a very high heat transfer coefficient. At some point, nucleation begins on the channel walls. The nucleation criterion developed by Hsu and Graham [15], Bergles and Rohsenow [16], Sato and Matsumura [17], and Davis and Anderson [18] were investigated by Kandlikar et al. [19] for flow boiling of water in a minichannel. A modified criterion was proposed to account for the contact angle and flow around bubbles in the wall region. As the liquid flows around a bubble of radius r_b and nucleates over a cavity of radius r_c, a stagnation region was identified to occur at a height of $1.1r_b$ from the channel wall. The temperature at this location was taken as the temperature at the top of the nucleating bubble. The resulting nucleation criterion is expressed as follows:

$$\{r_{c,\min}, r_{c,\max}\} = \frac{\delta_t \sin\theta_r}{2.2}\left(\frac{\Delta T_{Sat}}{\Delta T_{Sat} + \Delta T_{Sub}}\right) \times\left[1 \mp \sqrt{1 - \frac{8.8\sigma T_{Sat}(\Delta T_{Sat} + \Delta T_{Sub})}{\rho_V h_{LV}\delta_t \Delta T_{Sat}^2}}\right] \tag{1}$$

where $r_{c,\min}$ and $r_{c,\max}$ make up the range of radii of nucleating cavities, δ_t is the thickness of the thermal boundary layer ($=k/h$), k is the thermal conductivity of liquid, h is the heat transfer coefficient in the single-phase flow prior to nucleation, σ is the surface tension, ρ_V is the vapor density, h_{LV} is the latent heat of vaporization, ΔT_{Sat} and ΔT_{sub} are the wall superheat and liquid subcooling at the inception location, and θ_r is the receding contact angle.

The original equation reported in Kandlikar et al. [19] had a typographical error (an incorrect constant value of 9.2 was reported in place of the correct value of 8.8 as given in Eq. (1)). However, all the calculations and plots reported in their work were performed with the correct value of 8.8.

Figure 9 shows a comparison of different models with the data obtained by Kandlikar et al. [19] for flow boiling of water in a 3 mm × 40 mm aluminum ($\theta_r = 40°$) rectangular channel. The lines from different models represent the minimum wall superheat needed to nucleate a cavity of given radius. The data points with higher values of superheat indicate that the wall superheat is larger than the minimum required to nucleate a cavity of a given radius. If a data point falls below a line, the corresponding model is predicting a higher wall superheat than the experimental value.

The solid line representing the Kandlikar et al. [19] model is able to cover the data points quite well (all points lie very close to or above this line). Similar observations were made for different inlet subcooling and wall temperature conditions.

The local wall superheat at the onset of nucleate boiling (assuming that cavities of all the size ranges needed for nucleation

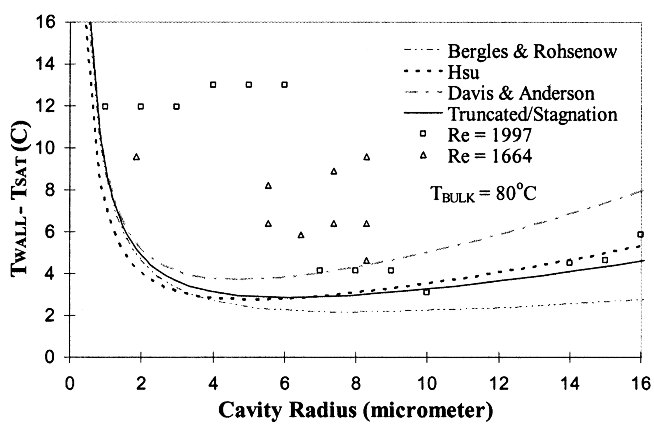


Figure 9 Comparison of different nucleation criteria against the experimental data taken with water at 1 atm pressure in a 3 × 40 mm channel, $\theta_r = 40°$ (Kandlikar et al. [19], Eq. (1)).

at various operating conditions are present) is given by:

$$\Delta T_{Sat,ONB} = \sqrt{8.8\sigma T_{Sat} q''/(\rho_V h_{LV} k_L)} \quad (2)$$

At the onset of nucleate boiling the bulk liquid may be subcooled, saturated, or even superheated. The local subcooling at this location is given by:

$$\Delta T_{Sub,ONB} = \frac{q''}{h} - \Delta T_{Sat,ONB} \quad (3)$$

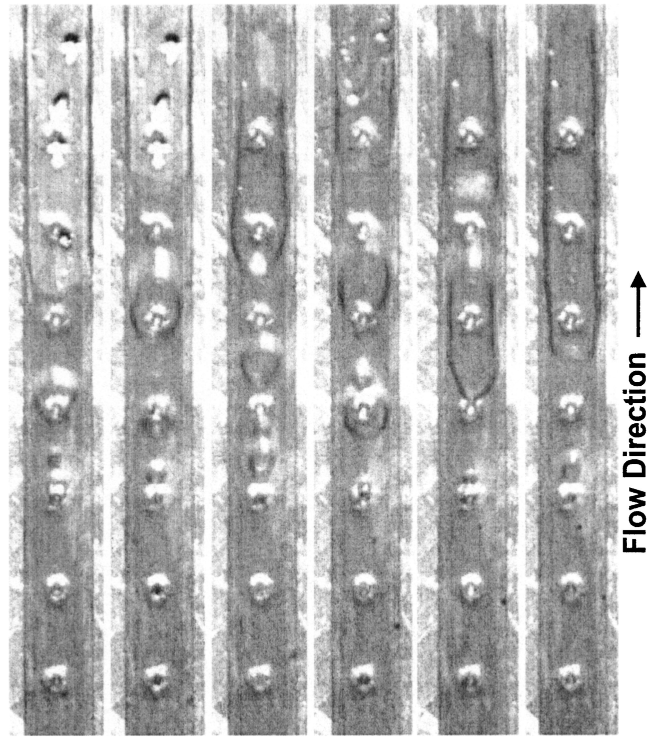
When a bubble nucleates in a saturated or superheated bulk liquid, the bubble growth rate is extremely high due to the sudden release of the liquid superheat at the liquid–vapor interface. The experimental observations by Balasubramanian and Kandlikar [20] and Kandlikar and Balasubramanian [21] indicate that this growth rate yields interface velocities as high as 3.5 m/s. Such high values were confirmed by Mukherjee and Kandlikar [22] through numerical simulation of a growing vapor bubble in a microchannel. As a result of such rapid expansion, the vapor bubbles occupy the entire channel width and push liquid in both the upstream and downstream directions. This leads to flow instabilities when a compressible vapor volume is present in the inlet supply line. Additionally, the presence of alternative flow paths in parallel channels interconnected through an inlet manifold also leads to a reverse flow condition.

Kandlikar et al. [23] introduced pressure drop elements at the inlet of each microchannel. These pressure drop elements introduce a flow resistance opposing the reverse flow in the channels. Additionally, artificial nucleation cavities with diameters between 5 and 30 μm as calculated from the nucleation criterion for the range of heat flux and flow conditions tested were drilled in the channels.

The resulting flow boiling process is shown in Figure 10. Nucleation was initiated early in the channel before the bulk liquid reached a high degree of wall superheat. In addition to the nucleation cavities, inlet pressure restrictors were fabricated at the entrance to each channel. Figure 10 shows the images of a channel taken at 0.83 ms time intervals. Nucleation is seen to initiate over the cavities, but the rapid expansion seen in plain channels is replaced with gradual growth and flow of the bubbles in the channel. There was no evidence of any reversed flow, and the pressure drop across the channels was also quite steady. Kandlikar et al. [23] showed that the pressure drop elements and artificial nucleation sites used independently still lead to reverse flow, while incorporating both of them together provides the stabilized flow.

The effect of pressure drop elements was numerically investigated by Mukherjee and Kandlikar [24]. They introduced inlet area reductions to provide an inlet-to-outlet flow area ratio of 25 percent. Their numerical results are shown in Figure 11.

Flow stabilization seems to be possible through the introduction of artificial nucleation sites and inlet pressure drop elements. Further research in this area is needed for obtaining reliable experimental data under a wide range of flow boiling and critical heat flux conditions.



(a) (b) (c) (d) (e) (f)

Figure 10 Stabilized flow with large fabricated nucleation sites. Successive frames from (a) to (f) taken at 0.83 ms time intervals illustrate stabilized flow in a single channel from a set of six parallel vertical microchannels (water, G = 120 kg/m^{2}s; q″ = 308 kW/m^2; (Kandlikar et al. [23]).

SUMMARY OF OPPORTUNITIES AND CHALLENGES

High heat flux cooling of electronics components and devices with single-phase liquid flow and flow boiling in microchannels and minichannels is reviewed. At the systems level, the

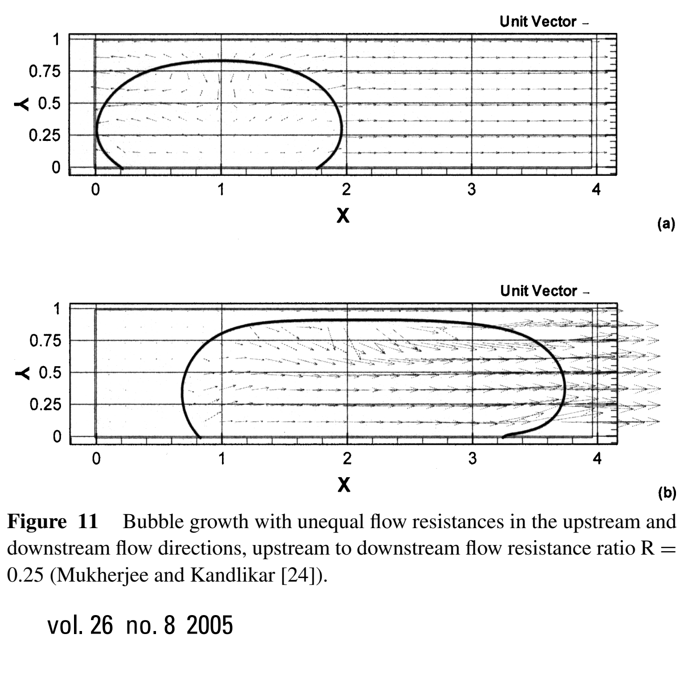


Figure 11 Bubble growth with unequal flow resistances in the upstream and downstream flow directions, upstream to downstream flow resistance ratio R = 0.25 (Mukherjee and Kandlikar [24]).

operational scenarios available with narrow channels are discussed with some possible solutions addressing the high heat flux cooling requirements of electronics systems and the integration of these systems with a building HVAC system. The design and operational issues of single-phase microchannel cooling for a high end server application are discussed. The opportunities available in developing new system design concepts to integrate additional loads from other low power components into the building cooling systems have been identified.

With the current single-phase enhanced microchannel technology, a heat dissipation rate of as high as 10 MW/m^2 (1 kW/cm^2) seems possible. Further opportunities exist to improve the performance of the enhanced microchannels to yield a lower pressure drop through enhanced microchannel designs and reduction of the flow lengths by improved header designs.

Current research needs in single-phase cooling are identified as:

1. the design of novel single-phase microchannel configurations to provide high thermal performance with a low pressure drop
2. cost-effective manufacturing
3. fouling studies
4. effective system integration

In flow boiling systems, more research is needed to provide a fundamental understanding of the flow boiling phenomena and develop reliable and cost-effective stabilization techniques for the safe operation of these systems. Recent research work is presented on the two methods for overcoming flow boiling instabilities: the introduction of artificial nucleation sites of appropriate radii on the heated channel surface, and the location of pressure drop elements at the entrance to the microchannels. These concepts are also applicable to minichannels in various other applications, including automotive, aircraft, and stationary evaporators in diverse applications.

Although flow boiling seems to be an attractive option, the operational instability and the resulting low critical heat flux levels have prevented their implementation in practical devices. In the near term, single-phase cooling holds more immediate promise due to its reliable operation and ability to handle heat fluxes.

NOMENCLATURE

a	channel width, m
b	channel height, m
h	heat transfer coefficient, W/m^{2}K
h_{LV}	latent heat of vaporization, J/kgK
k	thermal conductivity, W/mK
L	chip length being cooled, m
r_b	bubble radius, m
r_c	cavity radius, m
$r_{c,\min}$ and $r_{c,\max}$	minimum and maximum radii of nucleating cavities, m
s	fin width, m
W	chip width being cooled, m

Greek Symbols

β	fin spacing ratio, $= a/b$
δ_t	thickness of the thermal boundary layer, $= k/h$, m
ρ_V	vapor density, kg/m^3
σ	surface tension, N/m
ΔT_{Sat}	wall superheat, K
ΔT_{sub}	liquid subcooling, K
θ_r	receding contact angle, degrees

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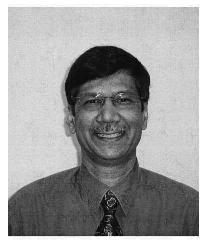

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