Enhanced Flow Boiling Over Open Microchannels With Uniform and Tapered Gap Manifolds

Flow boiling in microchannels has been extensively studied in the past decade. Instabilities, low critical heat flux (CHF) values, and low heat transfer coefficients have been identified as the major shortcomings preventing its implementation in practical high heat flux removal systems. A novel open microchannel design with uniform and tapered manifolds (OMM) is presented to provide stable and highly enhanced heat transfer performance. The effects of the gap height and flow rate on the heat transfer performance have been experimentally studied with water. The critical heat fluxes (CHFs) and heat transfer coefficients obtained with the OMM are significantly higher than the values reported by previous researchers for flow boiling with water in microchannels. A record heat flux of 506 W/cm² with a wall superheat of 26.2 °C was obtained for a gap size of 0.127 mm. The CHF was not reached due to heater power limitation in the current design. A maximum effective heat transfer coefficient of 290,000 W/m² °C was obtained at an intermediate heat flux of 319 W/cm² with a gap of 0.254 mm at 225 mL/min. The flow boiling heat transfer was found to be insensitive to flow rates between 40–333 mL/min and gap sizes between 0.127–1.016 mm, indicating the dominance of nucleate boiling. The OMM geometry is promising to provide exceptional performance that is particularly attractive in meeting the challenges of high heat flux removal in electronics cooling applications.

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Keywords: flow boiling, stable boiling, electronics cooling, high heat flux, boiling heat transfer, open microchannels, uniform and tapered manifold

1 Introduction

For the last several decades, air has been the preferred fluid for cooling electronics due to its availability, low cost (cooling fans), and reliable system operation. As chip power densities increased, liquid cooling systems have been introduced. The pioneering work of Tuckerman and Pease [1] revealed the potential for electronics cooling with microchannels. Colgan et al. [2] illustrated how liquid cooling in microchannels can be used to dissipate heat fluxes of approximately 1 kW/cm². However, the temperature variation of the chip surface along the coolant stream and large pumping power required for their system are of concern. Flow boiling in microchannels was expected to provide effective cooling without these concerns; however, the available research indicates that such high heat fluxes are not projected with current microchannel designs.

A number of researchers have investigated the effects of operating conditions on flow boiling heat transfer in microchannels. Qu and Mudawar [3] reported that the heat transfer coefficient for microchannel flow boiling is highly dependent on the mass velocity. Steinke and Kandlikar [4] observed that an increase in vapor quality caused a sharp decrease in the heat transfer coefficient (with a peak value between 100–200 kW/m² °C near the onset of nucleate boiling (ONB) close to x = 0), and significantly lower values of less than 50 kW/m² °C at higher vapor qualities downstream.

Wu and Cheng [5] conducted a series of experiments using trapezoidal microchannels over a range of heat and mass fluxes. They observed three kinds of unstable modes: (1) liquid/two-phase alternating flow at low heat flux and high mass flux, (2) continuous two-phase flow at medium heat and mass fluxes, and (3) liquid/two-phase/vapor alternating flow at high heat flux and low mass flux. Kandlikar [6] studied the flow instabilities due to nucleation in microchannels using high speed visual observations. Hetsroni et al. [7] also reported similar instabilities. Later, Kandlikar et al. [8] and Kuo and Peles [9] employed nucleation cavities and inlet restrictors to initiate nucleation and reduce instabilities. Zhang et al. [10] studied the static Ledinegg instability in horizontal microchannels. The authors concluded that increasing the system pressure and channel diameter, reducing the channel length, and adding inlet flow restrictors stabilized the flow. Mukherjee and Kandlikar [11] numerically studied bubble growth in a microchannel and proposed tapered microchannels in order to reduce instabilities. Lu and Pan [12] obtained stable flow through microchannel heat sinks using diverging parallel microchannels with laser etched artificial nucleation sites. The authors confirmed that significant improvements were observed in suppressing flow reversal, pressure oscillations, and inlet temperature fluctuations.

Heat transfer performance data during flow boiling in microchannels was obtained by several researchers. Qu and Mudawar [13] used a microchannel heat sink containing 21 parallel microchannels 231 µm × 713 µm and obtained a maximum heat flux of 130 W/cm². Kosar et al. [14] used microchannels with a hydraulic diameter of 227 µm and 7.5 µm wide reentrant cavities on the sidewalls. High boiling and Reynolds numbers were reported to increase convective boiling. Hetsroni et al. [15] used triangular microchannels and obtained heat fluxes ranging from 8–33 W/cm² with mass fluxes between 95–340 kg/m² s. Liu and Garimella [16] experimentally investigated flow boiling in microchannels with inlet water temperatures of 67–95 °C, and mass fluxes of 221–1283 kg/m² s. The authors obtained a maximum heat flux of 129 W/cm² and a maximum exit quality of 0.2. Kuo and Peles...
Khanikar et al. [18] experimentally investigated the effect of carbon nanotubes on heat transfer using de-ionized water. They concluded that at low mass fluxes, bubble nucleation was promoted, however, the effect was severely compromised at higher mass fluxes. Bhide et al. [19] used multiwalled carbon nanotubes on a silicon wafer as their enhanced surface. Liquid inlet temperatures of 40 °C and 60 °C and mass fluxes of 18–25 kg/m²s were employed. The flow boiling heat flux was enhanced by 30–80% in the fully developed nucleate boiling regime. Krishnamurthy and Peles [20] used circular micro-pin fins with mass fluxes from 346 to 794 kg/m²s. Vapor slugs and annular flow patterns were observed through high speed flow visualization. The authors concluded that the two-phase heat transfer coefficient was moderately dependent on the mass flux and independent of the heat flux. Morshed et al. [21] experimentally investigated the heat transfer performance of a microchannel with a copper nanowire coating. The authors used a single microchannel of 672 μm hydraulic diameter with de-ionized water at atmospheric pressure. Nanowires were grown using an electrochemical deposition technique on the bottom surface of the microchannel. They obtained a maximum heat flux of 58 W/cm² at a wall superheat of 12 °C, with an inlet fluid temperature of 40 °C and a mass flux of 251 kg/m²s.

It is seen from the available experimental data on flow boiling in microchannels that the maximum reported heat flux is around 130 W/cm² by Qu and Mudawar [13] and Liu and Garimella [16] for a 10 mm × 10 mm chip with wall superheats of around 20–30 °C. This maximum reported heat flux is considerably below the projected cooling requirement of 1 kW/cm² for advanced computer chips [22] and is also lower than the values reported in the literature for some of the enhanced pool boiling surfaces, which are briefly reviewed in the next section.

2 Current State of Enhanced Pool Boiling Heat Transfer

Pool boiling heat transfer is at the core of any flow boiling system. A brief review of the current status of pool boiling is presented first to guide us in setting goals for flow boiling in microchannels. The main advantages of pool boiling for high heat flux removal are a uniform surface temperature and an efficient heat removal system. Recent advances have made this boiling process quite efficient using nanowires, open microchannels, and combinations of different hierarchical surface structures consisting of nanoscale and microscale features. Figure 1 shows a comparison of some of the recent high performing pool boiling structures reported in the literature. Mori and Okuyama [23] used a porous plate, while Chen et al. [24] and Yao et al. [25,26] used copper nanowires. Yang et al. [27] employed an open foam cover and Li and Peterson [28] employed a sintered wire mesh. Cooke and Kandlikar [29,30] used open microchannels that resulted in a heat flux of 244 W/cm² with a heat transfer coefficient of 269,000 W/m²°C. A microstructure design based on utilizing the evaporation momentum force, proposed by Kandlikar [31,32], was able to dissipate a heat flux of over 300 W/cm² with a record heat transfer coefficient of 629,000 W/m²°C. In comparison, microchannel flow boiling systems have a significantly lower performance, as seen from the literature review. In order to justify the additional complexities over pool boiling, it would be reasonable to expect a flow boiling system to provide a stable operation and superior heat transfer coefficient and CHF values.

The objectives of the current study are set to explore the novel open microchannel with manifold (OMM) flow boiling configuration and present the experimental results validating its potential to stabilize the flow and enhance heat transfer performance with water. The goal is to obtain heat fluxes and heat transfer coefficients that are at least comparable to the highest pool boiling systems reported in the literature. In addition, the relevant parameters governing the heat transfer and critical heat flux will be identified.

3 Open Microchannels With Uniform and Tapered Manifolds (OMMs) in Flow Boiling

The two main issues that need to be addressed before implementing microchannels in flow boiling applications are: (1) flow stability, and (2) improved heat transfer and CHF. Flow boiling instability has been identified as a limiting factor in microchannels. Reducing the downstream flow resistance effectively mitigates the backflow associated with such instabilities. Mukherjee and Kandlikar [11,33] suggested several tapered microchannel designs to overcome this problem. Since tapered microchannels are difficult to implement, a new tapered manifold design with open microchannels has been developed in the present study, as shown in Fig. 2. The microchannels have an open area above them with a gap, which provides increased flow area for the fluid. The gap increases downstream, thereby decreasing the slope of the pressure gradient versus the flow rate curve for the flow channel and increasing the flow stability, as illustrated by Zhang et al. [10].

The tapered manifold and open microchannel design also addresses the two factors responsible for low CHF—removal of the vapor from the heat transfer surfaces and the supply of liquid to the nucleation sites. The extra flow area provided by the manifold above the microchannels is helpful in removing the generated vapor without an excessive pressure drop penalty. Since vapor generation increases in the flow direction, a uniform or a tapered manifold provides the extra space downstream for the vapor to flow away from the heat transfer surface. Liquid flow is favored inside the microchannel region due to capillary forces. Finally, the

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**Fig. 1** Comparison of enhanced pool boiling performance reported in the literature

**Fig. 2** Schematic of the open microchannels with tapered manifolds for stable enhanced flow boiling
open microchannel configuration has been shown to be very effective in enhancing pool boiling heat transfer [29,30] and is expected to provide similar enhancement under flow boiling conditions in the present OMM configuration shown in Fig. 2.

The techniques available in the literature to reduce the instabilities can be summarized as artificial nucleation sites, inlet restrictors [8,9] and tapered channels [11,12]. Artificial nucleation sites alone are not able to effectively suppress the instabilities. The main disadvantage of the inlet restrictors and the tapered microchannels is the added pressure drop. The OMM design offers a lower pressure drop due to the availability of the additional flow area. One of the key aspects of this design is that the microchannels promote nucleation and boiling heat transfer, while the open manifold region provides a path for the vapor flow without adversely affecting the liquid flow to the nucleating cavities and the heat transfer regions inside the open microchannels. However, the OMM design may require an additional space above the microchannels and may need careful consideration before implementing it in the 1-D and 3-D electronic chip cooling applications.

3.1 Test Fixture. Flow boiling experiments were performed using the test section configuration shown in Fig. 2. It shows open microchannels with a tapered manifold that provides a larger flow area toward the exit. Figure 3 shows a schematic of the test assembly, which consists of a copper block with eight 200 W cartridge heaters to serve as the main heating unit. Some of the early tests were performed with a heater which could deliver a maximum heat flux of only 300 W/cm². The tip of the square heating block measures 10 mm × 10 mm and has three equally spaced K-type thermocouples that were used to determine the heat flux. A copper test chip contacts the tip of the heating block and a fourth thermocouple in the chip measures its temperature. The chip was supported by a ceramic plate and a manifold block delivers and removes water from the surface of the chip. A silicone gasket between the chip and the manifold block was used to seal the system and provide the manifold gap that can be readily varied by changing the gasket thickness. The gaskets used during testing had thicknesses ranging from 0.127 to 1.524 mm. A Keyence Biromatic microscope with a resolution of 3.4 μm was used for an optical examination of the microchannels. The chip surface temperature was obtained from the measured chip temperature, the heat flux, and the distance L = 1.5 mm between the chip thermocouple and the chip surface.

$$q'' = -k_c u \frac{dT}{dx}$$  \hspace{1cm} (1)

The temperature gradient $dT/dx$ was calculated from a three-point backward difference Taylor series approximation using the heating block temperatures $T_1$, $T_2$, and $T_3$, each at a distance $\Delta x$ of 5 mm apart.

$$\frac{dT}{dx} = \frac{3T_1 - 4T_2 + T_3}{2\Delta x}$$  \hspace{1cm} (2)

Finally, the chip surface temperature was obtained from the measured chip temperature $T_s$, the heat flux, and the distance $L = 1.5 \text{mm}$ between the chip thermocouple and the chip surface.

$$T_s = T_4 - q'' \frac{L}{k_c}$$  \hspace{1cm} (3)

An uncertainty analysis was conducted in a manner similar to that described by Cooke and Kandlikar [29,30]. At the highest flow rate (333 mL/min) the rotameter had an uncertainty of 3% and at the low flow rate the uncertainty was 5%. The individually calibrated thermocouples have an accuracy of 0.1 °C. The resulting uncertainty in the heat flux and surface temperature at high heat fluxes were 4% and 0.24 °C, respectively. All heat fluxes are reported on the basis of the projected area of the boiling surface, which is 100 mm².

3.2 Test Chips. The experiments were performed with two copper chips: one with a plain surface and the other with microchannels in the central 10 mm × 10 mm heated region. The front and back views of the microchannel chip are shown in Fig. 4. The microchannels were computer numerical control machine milled and had a channel width of 217 μm, a fin width of 160 μm, a channel depth of 162 μm, and a length of 10 mm, giving a hydraulic diameter of 185 μm.

The chip and the heater block designs were similar to the design reported by Cooke and Kandlikar [29]. Specifically, a 2 mm wide and 2 mm deep groove was machined on the underside of the chip to reduce the heat losses and the heat spreading effect from the chip.

![Fig. 4 Schematic of the 3 mm thick copper microchannel chip with a 217 μm wide, 162 μm deep, and 160 μm fin width in the central 10 mm × 10 mm region (left image), and a 2 mm wide × 2 mm deep groove on the underside (right image)]
effect from the test section were minimized due to the gasket and edges of the test chip. The heat losses and the heat spreading in order to inhibit contact between the working fluid and the outer channel features.

The square opening in the gasket over the chip region with microflux due to heat losses and heat spreading effects were estimated the gasket was aligned over the 10 mm microchannel region. A 10 mm region are 0.305, 0.476, 0.809, 1.442, and 2.035 mm, respectively.

4 Results

4.1 Effect of Tapered Manifold. The effect of the manifold taper on flow stability was evaluated by comparing the inlet pressure fluctuations at steady state conditions for both uniform and tapered manifold blocks, with a plain chip and an entry spacing of $S = 0.127$ mm, as shown in Fig. 5. For each heat flux, the pressure drop readings taken over a 20 sec period at 5 Hz frequency are shown. The uniform and tapered manifold tests were run with a flow rate of $V = 80$ mL/min. Uniform manifold shows pressure fluctuations between 80–160 kPa as seen in Fig. 5 for heat fluxes in the range of 50–250 W/cm². Tapered manifold in the same heat flux range shows pressure fluctuations below 20 kPa. This indicates that the tapered manifold mitigates the flow boiling instability experienced with the uniform manifold.

The improvement in stability is also demonstrated in video footage of the bubble formation for each test. Figure 6 shows a sequence of three high speed images 8 ms apart, with a uniform manifold block and a plain test chip at $V = 103$ mL/min and $S = 1.524$ mm. A nucleating bubble forms on the right edge of the frame and then expands both upstream and downstream, thereby inducing a back flow and pressure fluctuations in the test section. The image sequence shown in Fig. 7 corresponds to a tapered manifold block with a plain test chip at $V = 74$ mL/min and $S = 1.524$ mm. In this case, the nucleating bubble forms and detaches from the chip surface without expanding against the flow direction. This results in a reduction in pressure fluctuations and improved flow boiling stability.

4.2 Comparison of the Tapered Manifold and the Uniform Manifold. Figure 8 illustrates the comparison of the results for the tapered manifold and the uniform manifold on the heat transfer performance. The tests were conducted with a microchannel chip with $S = 0.127$ mm and $V = 225$ and 40 mL/min. The depth of the tapered manifold increased by 0.180 mm over the length of the microchannel region. At $V = 40$ mL/min, the tapered manifold achieved 315 W/cm² at $\Delta T_{\text{sat}} = 17.2 \, ^\circ C$, while the uniform manifold reached 308 W/cm² at $\Delta T_{\text{sat}} = 20.5 \, ^\circ C$. Both manifolds did not reach the CHF limits.

The tapered manifold seems promising, as seen from its improved performance in Fig. 8. This is believed to be due to the flow stabilization effect of the tapered manifold. By mitigating the back pressure caused by the expansion of large bubbles on the chip surface, the working fluid has an uninterrupted flow passage and a more continuous contact with the heated region. Reducing
the manifold depth may provide additional heat transfer performance enhancement by increasing flow velocity and by further preventing back flow and bubble expansion.

4.3 Effect of Manifold Depth With the Tapered Manifold. The effect of the manifold depth was studied for the tapered manifold block with \( S = 0.254 \) and \( 0.127 \) mm, at both \( V = 40 \) and \( 225 \) mL/min. The results are shown in Fig. 9. At a heat flux of approximately 270 W/cm\(^2\) and a flow rate of 40 mL/min, the thinner gasket had a \( \Delta T_{\text{sat}} = 15.5 \) °C, while the thicker gasket had a \( \Delta T_{\text{sat}} = 37.1 \) °C. When a flow rate of 225 mL/min is applied, the performance difference is evident, but less dramatic; the same heat flux of 270 W/cm\(^2\) shows a \( \Delta T_{\text{sat}} \) difference of 10.4 °C.

The performance with a 0.127 mm gap is better than that with a 0.254 mm gap for both flow rates. In addition, the performance with 40 mL/min is better than that with 225 mL/min for both gaps. Thus, the smaller gap with the lower flow rate is seen to perform better than the other configurations with the tapered manifold.

4.4 Performance Enhancement With Microchannels With the Uniform Manifold. The uniform manifold configuration forms the baseline for the OMM design. It is investigated in more detail in the following sections. The heat dissipation performance was evaluated for the plain and microchannel test chips using a uniform manifold with a spacing of \( S = 0.254 \) mm. Tests were conducted at two flow rates of \( V = 225 \) and \( 333 \) mL/min. The results are shown in Fig. 10. The microchannel chip performed significantly better than the plain chip for both flow rates. The plain chip dissipated 100 W/cm\(^2\) between \( \Delta T_{\text{sat}} = 20 \) and 25 °C and the CHF was reached at this heat flux. The microchannel chip dissipated over 255 W/cm\(^2\) at \( \Delta T_{\text{sat}} = 14.1 \) °C for \( V = 225 \) mL/min without reaching the CHF. As the flow rate was increased to 333 mL/min, the performance slightly deteriorated and the test was conducted only up to 140 W/cm\(^2\). By comparison, at a \( \Delta T_{\text{sat}} = 14.1 \) °C and \( V = 225 \) mL/min, the plain test chip dissipated...
only 54.8 W/cm². Boiling curves for the plain test chip were unaffected by changes in the flow rate.

The difference in the performance between the plain and microchannel test chips is partly the result of the differences in the chip surface structure and boiling behavior. When nucleate boiling occurs on a plain chip, the area surrounding a nucleation site is obstructed by the vapor bubble and the effective wetted area of the chip is reduced. This is shown in Fig. 11, where portions of the plain chip under the nucleating bubble have dried out, as seen from the lighter color in the outlined region. The microchannel test chip avoids this problem by preventing large sections of the surface to be covered by a nucleating bubble. The land regions of the microchannels under a bubble are dry (indicated by the light color), but the microchannel cavity coloration is unchanged, indicating that the channels are still wet. The outline of the bubble surrounding the dry area is also visible over the microchannels. The microchannel chip also has an inherent surface augmentation factor of 1.8 and, therefore, has 80% more area than the plain chip. The heat flux improvement beyond the area augmentation factor is due to the enhanced boiling mechanism in the present OMM configuration.

4.5 Effect of Manifold Depth With the Uniform Manifold. Five different gasket thicknesses (0.127, 0.254, 0.508, 1.016, and 1.524 mm) were used to study the effect of the manifold depth on the flow boiling performance. The boiling data shown in Fig. 12 was obtained with a constant flow rate of 333 mL/min, the microchannel test chip, and the uniform manifold block. A maximum heat flux of 360 W/cm² at ΔT_{sat} = 28.4 °C was achieved using a manifold depth of 0.127 mm. The boiling curves for all depths were similar, except for the 1.524 mm gasket. The configurations with depths of 0.254, 0.508, and 1.016 mm were not tested in the high heat flux region due to the heater limitation in the earlier version of the test setup. When the manifold depth was increased to 1.524 mm, the heat transfer performance dropped significantly to a maximum heat flux of only 122 W/cm² at ΔT_{sat} = 22.0 °C.

The boiling data in Fig. 12 suggests that there is a critical manifold depth, while the flow rate also has a small effect, as seen from Fig. 10. The performance increases with a decrease in the manifold depth from 1.524; however, the performance with a depth of 0.127 mm seems to be slightly lower than the intermediate three depths.

The manifold depth also has an important role on the pressure drop that must be considered in a test setup. Figure 13 shows a plot of the pressure drop versus the heat flux for both 0.127 and 0.508 mm manifold depths using the uniform manifold V = 225 mL/min and the microchannel chip. At the highest heat flux of 250 W/cm², the 0.127 mm manifold had a pressure drop of
Performance in terms of heat transfer coefficient currently exceeds the OMM configurations investigated in this paper. Cooke and Kandlikar [29,30] achieved a heat flux of 250 W/cm² at \( \Delta T_{sat} = 10°C \) using the best performing microchannel chip tested in pool boiling, while in the initial tests reported here, a maximum heat flux of 506 W/cm² was obtained with \( \Delta T_{sat} = 26.2°C \). It may be noted that the CHF was not reached during any of the tests due to the temperature limitation of the heater and only one microchannel configuration was tested to validate the concept. Thus, the CHF was significantly enhanced in the OMM design.

Figure 17 shows a comparison between the heat transfer coefficients for pool boiling and flow boiling. The three tests which reached the maximum heat flux of above 500 W/cm² are included in this plot. The three cases yield very similar heat transfer coefficients. The pool boiling setup with open microchannels yielded a maximum heat transfer coefficient of 270 kW/m²°C at a heat flux of 250 W/cm² [30]. While pool boiling with a novel microstructure design by Kandlikar [30,32] yielded a heat transfer coefficient of 629 kW/m²°C. The current OMM flow boiling setup attained a maximum heat transfer coefficient of 193 kW/m²°C at a heat flux of 506 W/cm². Pool boiling outperformed flow boiling in terms of the heat transfer coefficient at all heat fluxes. These results indicate that flow boiling in the OMM does not improve the heat transfer efficiency in the nucleate boiling mode within the

**4.6 Effect of Flow Rate With the Uniform Manifold.** Three different tests were conducted with the uniform manifold in order to investigate the effect of the flow rate and manifold depth on the flow boiling performance. The first test had a set manifold depth of 0.127 mm with five different flow rates (\( V = 333, 225, 152, 80, \) and 40 mL/min) and the microchannel chip. The highest heat flux obtained for this manifold depth was 506 W/cm² at \( \Delta T_{sat} = 26.2°C \) with a flow rate of 152 mL/min, as seen in Fig. 14. Tests using flow rates of 80 and 40 mL/min performed very similarly with no more than one or two degrees of wall superheat separating each test at any given heat flux. Higher flow rates of 333 and 225 mL/min showed reduced performance, since they typically had a \( \Delta T_{sat} \) that was 5–10°C higher than the other tests for any given heat flux. The CHF was not reached in any of the tests.

The second set of tests used a manifold depth of 0.254 mm, the uniform manifold, and the microchannel chip. The results are shown in Fig. 15. A maximum heat flux of 258 W/cm² at \( \Delta T_{sat} = 14.6°C \) was obtained with a flow rate of 152 mL/min for this configuration. The boiling data for tests with \( V = 225 \) and 152 mL/min have very comparable profiles, while the data for \( V = 333 \) mL/min has a slightly lower heat flux for a given wall superheat in the low heat flux region. Higher heat fluxes could not be obtained since these tests were performed with an earlier version of the test setup with a lower power rating.

The final flow rate tests used a manifold depth of 1.524 mm, the uniform manifold, and the microchannel chip (see Fig. 16). The flow rate of 333 mL/min produced a maximum heat flux of 127 W/cm² and the boiling performance for each test was comparable in profile and performance. Only the lowest flow rate of 152 mL/min showed a slightly better performance. The performance with the manifold depth of 1.524 mm was substantially lower than those with the thinner manifold spacing.

Figures 14 through 16 illustrate that the flow rate has a minimal effect on the heat dissipation performance in comparison to the manifold depth. The boiling performance was consistent across a wide range of flow rates for each depth, however, manifold depths of 1.016 mm and thinner dramatically outperformed the largest depth of 1.524 mm throughout the range of heat fluxes tested.

**4.7 Comparison to Pool Boiling.** The best pool boiling performance in terms of heat transfer coefficient currently exceeds the performance in the nucleate boiling mode within the OMM.
microchannels and that its utility primarily lies in enhancing the CHF through efficient vapor removal and liquid supply to the nucleation sites.

It is seen that the OMM design exceeds the pool boiling performance of the advanced open microchannel design in terms of the CHF. Further improvements are expected as the manifold gap, taper, and microchannel geometry undergo refinements.

5 Discussion and Future Work

The OMM design was able to enhance the CHF beyond 500 W/m² with a wall superheat of only 26.2 °C. The CHF was not reached in any of the OMM configurations tested. Initial testing was limited to heat fluxes below 250–350 W/cm² since such high performance was not anticipated. Even the new heater design delivering 500 W/cm² heat flux over a 10 mm x 10 mm area was exceeded by the OMM design without reaching the CHF. This is obviously an area for future experimental work in establishing the CHF limits of different OMM configurations.

This study shows that the manifold depth, manifold taper, and flow rate influence the heat transfer performance. The effect of the flow rate seems to be relatively small over the ranges of the heat and mass fluxes tested. The inlet subcooling is also an important parameter, which was not studied in this investigation since the tests were carried out with a near-saturation inlet with a subcooling of 2–5 °C.

The work presented here confirms that the OMM design is able to provide high CHF and heat transfer coefficients during flow boiling. It opens up a new field where a number of parameters, such as microchannel geometry, manifold gap depth, and taper configuration, could be varied and optimized to achieve even higher values of heat fluxes and heat transfer coefficients. Other techniques recently employed in pool boiling enhancement, such as nanowires, nanofluids, and hydrophobic/hydrophilic hierarchical surface patterns, are expected to provide further performance enhancements. Experiments are also planned to study the effect of a higher vapor quality on the heat transfer performance.

6 Conclusions

A novel flow boiling configuration with open microchannels and a uniform or tapered manifold (OMM) is presented. The experiments conducted indicate that this configuration is capable of achieving very high heat fluxes with high heat transfer coefficients. The following specific conclusions are drawn from this study:

1. A tapered manifold profile reduces the back flow caused by the bubble nucleation and stabilizes flow boiling when compared to a uniform manifold at low flow rates. The pressure fluctuations exhibited during flow boiling with a uniform manifold of spacing 1.524 mm are mitigated by the use of a tapered manifold.
2. The open microchannel surface shows a dramatic improvement in the heat transfer performance over a plain surface. High-speed video footage shows that liquid flow in the microchannels prevents the nucleating bubbles from drying out large areas of the heated surface.
3. The manifold depth has a significant effect on the heat transfer performance. A maximum heat flux of 506 W/cm² at a wall superheat of 26.2 °C was achieved using a uniform manifold with a spacing of 0.127 mm, a microchannel test chip, and a volumetric flow rate of 152 mL/min. The highest heat transfer coefficient obtained during this test was 193,000 W/m²°C with a 0.254 mm manifold depth. Manifold depths of 0.254 mm to 1.016 mm had similar boiling curves, which reached a maximum heat flux of 160 W/cm² at a wall superheat of 14.1 °C. The largest manifold depth of 1.524 mm had the worst overall performance.
4. Changes in the volumetric flow rate for a given manifold depth have only a small impact on the heat transfer performance. Varying the flow rates for manifold depths of 0.127, 0.254, and 1.524 mm produced similar boiling curves for each manifold depth.
5. The flow boiling heat transfer coefficient in the OMM configuration is currently exceeded by the most recent pool boiling performance reported in the literature with open microchannels [29,30] and novel microstructures [31,32].
6. The CHF was not reached in any of the OMM testing. The maximum heat flux reached was 506 W/cm². This indicates the tremendous potential offered by the OMM configuration.
7. Future work is planned in order to conduct an in-depth parametric study to evaluate and extend the performance limits of the OMM.

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Nomenclature

\[ h = \text{heat transfer coefficient, } \text{W/m}^2\text{°C} \]
\[ k_{Cu} = \text{thermal conductivity of copper block, } \text{W/m°C} \]
\[ L = \text{length from the thermocouple to the chip surface, mm} \]
\[ P = \text{pressure, Pa} \]
\[ q' = \text{applied heat flux, } \text{W/cm}^2 \]
\[ S = \text{manifold depth (gap), mm} \]
\[ T = \text{temperature, °C} \]
\[ V = \text{volumetric flow rate, } \text{mL/min} \]
\[ x = \text{vapor quality} \]

Greek Symbols

\[ \Delta T_{\text{sat}} = \text{wall superheat, °C} \]
\[ \Delta x = \text{distance between thermocouples, mm} \]

Superscript

\[ s = \text{surface} \]