Flow Boiling in Microgaps for Thermal Management of High Heat Flux Microsystems

In the first part of this paper, a review of fundamental experimental studies on flow boiling in plain and surface enhanced microgaps is presented. In the second part, complimentary to the literature review, new results of subcooled flow boiling of water through a micropin-fin array heat sink with outlet pressure below atmospheric are presented. A 200 μm high microgap device design was tested, with a longitudinal pin pitch of 225 μm, a transverse pitch of 135 μm, and a diameter of 90 μm, respectively. Tested mass fluxes ranged from 1351 to 1784 kg/m²s, and effective heat flux ranged from 198 to 444 W/cm² based on the footprint surface area. The inlet temperature varied from 6 to 12°C, and outlet pressure ranged from 24 to 36 kPa. The two-phase heat transfer coefficient showed a decreasing trend with increasing heat flux. High-speed visualizations of flow patterns revealed a triangular wake after bubble nucleation. Flow oscillations were seen and discussed. [DOI: 10.1115/1.4034317]

1 Introduction

Flow boiling in microscale systems was studied intensively in the past two decades and is still attracting the interest of many researchers [1–5]. Possible improvement in heat transfer over single-phase flow at the same mass flux due to latent heat of vaporization makes it a promising thermal management method for high power dissipation electronics. Moreover, two-phase flow may offer better temperature uniformity than single-phase flow, if dryout and instabilities can be mitigated. A block diagram of a closed-loop two-phase microfluidic cooling system for electronic devices is shown in Fig. 1. The system generally consists of a pump to drive fluid flow, microscale heat sink for heat removal via two-phase flow, heat exchanger to condense the vapor, and reservoir to supply sufficient fluid for circulation. Filters with microscale pore size and pressure regulating valves are also common components in these systems.

Thermodynamic properties of six typical coolants, FC-72, water, HFE-7000, R-123, R-134a, and R-245fa, all of which have been frequently used for flow boiling experiments, are listed in Table 1 [6]. Water is one of the most attractive coolants because of its large-specific heat and latent heat of vaporization, which enable absorption of considerable amount of heat during both sensible heating and boiling. However, saturation temperature of water at atmospheric pressure is 100°C, which may be unacceptably high for continuous operation of complementary metal oxide semiconductor (CMOS) electronic devices. Water is also not as chemically and electrically inert as other coolants in Table 1, but is the most environmental friendly. The liquid to vapor density ratio of water is also the largest among the six coolants, which can result in very large void fraction at relatively low vapor quality. Void fraction is the fraction of the cross-sectional area of the channel that is occupied by the vapor, and quality is the vapor mass fraction. The high density ratio also results in relatively large accelerational pressure drop associated with convective boiling flow in microgeometries.

Dielectric coolants, FC-72 and refrigerants, are of great interest because they are chemically and electrically inert and have low saturation temperature, making them more suitable for electronics cooling. Some refrigerants, such as R-134a, need to operate at high saturation pressures at the temperatures relevant to operating electronics devices, adding extra structural strength requirement to the flow system. Some dielectric coolants contribute to ozone depletion and global warming. Hydrofluoroethers (HFEs) are a better option due to their nearly zero stratospheric ozone depletion and relatively low global warming potential [7]. HFEs (HFE-7000, for instance) also have lower boiling point compared to water.

Reviews of flow boiling in microchannels (single microchannel and parallel microchannels) have been performed by Gariemila and Sobhan [1], Thome [2], and recently by Tibirica and Ribatski [3]. To compliment these works, a review of experimental work on flow boiling in plain and surface enhanced microgaps is presented in Sec. 2. Representative geometries are shown in Fig. 2.

![Fig. 1 Schematic of closed-loop microfluidic cooling system](image-url)
The details of geometry, experimental conditions, and studied parameters in each reviewed paper are summarized in Table 2. Survey of relevant literature revealed that flow boiling of water in microscale system at subatmospheric pressure was rarely studied previously. Lower system pressure reduces saturation temperature to compensate the drawback of high boiling point of water, thus improving its utility for electronics thermal management applications. In Sec. 3, subcooled flow boiling of water in a staggered micropin-finned heat sink using infrared (IR) imaging techniques was studied. The two-phase heat transfer coefficient showed a decreasing trend with increasing heat flux. Reversed flow and instabilities were observed and briefly discussed.

2 Review of Experimental Studies on Flow Boiling in Microgap, and Microchannels/Microgaps With Surface Enhancement

2.1 Plain Microgaps. Lee and Lee [8] and Yang and Fujita [9] studied flow boiling of R-113 in minigap and microgap with gap heights ranging from 2 mm to 0.2 mm. Lee and Lee [8] developed a two-phase heat transfer coefficient correlation for minigap and microgap flows. Their pressure drop data agreed well with a correlation obtained with adiabatic water air two-phase flow in a microgap [40]. Flow patterns such as bubbly, intermittent, wavy, and annular were observed, and gap height was found to impact flow patterns and heat transfer characteristics [9]. As gap height decreased, annular flow was dominant, and intermittent and wavy flow diminished.

Shah [43]. Closer agreement with Shah’s correlation at lower quality and with Chen’s prediction at higher quality was found. Alam et al. studied local flow boiling heat transfer and pressure drop characteristics in a silicon microgap heat sink using de-ionized (DI) water [24]. The microgap had an area of 1.27 cm × 1.27 cm, and an array of 5 × 5 heating elements and temperature sensors. Three gap heights were investigated: 180 μm, 285 μm, and 381 μm. The experiments were conducted with three mass fluxes, 420 kg/m²s, 690 kg/m²s, and 970 kg/m²s, and effective heat flux ranging from 0 to 110 W/cm². Confined slug flow/annular flow were observed after onset of nucleate boiling as heat flux increased. At fixed mass flux and heat flux, smaller gap tended to have confined annular flow while larger gap tended to have confined slug flow. Thin liquid film evaporation was the main heat transfer mechanism in confined annular flow, resulting in higher heat transfer coefficients in smaller gap than in larger gap. This also agreed with the work of Kim et al. [18,41]. Pressure drop increased with heat flux in smaller gap, but was independent of heat flux in larger gap. Wall temperatures were almost uniform along flow direction after boiling occurred for all the gap heights.

Alam et al. studied the effects of surface roughness on flow boiling in microgap [29]. The device also had an area of 1.27 cm × 1.27 cm, and an array of 5 × 5 heating elements and temperature sensors. They tested three gap heights of 500 μm, 300 μm, and 200 μm and three surface roughness levels of 0.6 μm, 1.0 μm, and 1.6 μm. Lower wall superheat was sufficient to initiate boiling in microgap with higher surface roughness. Rougher surface also increased nucleation density, wall temperature uniformity, and local two-phase heat transfer coefficients. No significant effect on pressure drop was observed in microgap with different surface roughnesses. However, increased surface roughness showed an adverse effect on pressure instability, and higher amplitude in pressure oscillations was observed. Using the same test setup, Alam et al. compared the ability of minimizing temperature gradient and mitigating hotspot of microgap and microchannel [21,30]. Tested microgap height was from 200 μm to 400 μm, and the 200 μm high microgap was compared with a microchannel with a channel pitch of 200 μm in Ref. [21]. Tested microgap height was 190 μm, microchannel width was 208 μm, and height was 386 μm in Ref. [30]. For uniform heating, at the same mass flux of 690 kg/m²s and heat flux range from 0 to 60 W/cm², microgap cooled device demonstrated a smaller temperature gradient and smaller amplitude of pressure and temperature oscillation than microchannel [21]. Reducing gap heights suppressed flow oscillation as well. When a hotspot was activated, microgap also showed better temperature uniformity than microchannel, and smaller gap height lowered wall temperature compared to higher gap height.

Table 1 Sample flow boiling coolant thermodynamic properties

<table>
<thead>
<tr>
<th></th>
<th>(T_{sat}) (°C) at 1 atm</th>
<th>(C_p) (kJ/kg K)</th>
<th>(h_{sat}) (kJ/kg)</th>
<th>(\rho_1) (kg/m³)</th>
<th>(\rho_s) (kg/m³)</th>
<th>(P_{sat}) (kPa) at 25°C</th>
</tr>
</thead>
<tbody>
<tr>
<td>FC-72</td>
<td>56.3</td>
<td>1.141</td>
<td>83.41</td>
<td>1602</td>
<td>13.28</td>
<td>30.2</td>
</tr>
<tr>
<td>Water</td>
<td>100</td>
<td>4.217</td>
<td>2257</td>
<td>958.4</td>
<td>0.5975</td>
<td>3.2</td>
</tr>
<tr>
<td>HFE-7000</td>
<td>69.5</td>
<td>1.091</td>
<td>132.4</td>
<td>1385</td>
<td>8.337</td>
<td>35.24</td>
</tr>
<tr>
<td>R-123</td>
<td>27.8</td>
<td>1.039</td>
<td>170.6</td>
<td>1457</td>
<td>6.474</td>
<td>91.5</td>
</tr>
<tr>
<td>R-134a</td>
<td>–26.1</td>
<td>1.425</td>
<td>177.8</td>
<td>1207</td>
<td>32.37</td>
<td>665.8</td>
</tr>
<tr>
<td>R-245fa</td>
<td>15.19</td>
<td>1.322</td>
<td>190.3</td>
<td>1339</td>
<td>8.525</td>
<td>147.8</td>
</tr>
</tbody>
</table>

FC-72, water, HFE-7000, and R-123 properties evaluated at \(T_{sat}\) and R-134a and R-245fa properties evaluated at \(P_{sat}\).

Fig. 2 (a) Plain microgap and (b) microgap with pin fin surface enhancement
<table>
<thead>
<tr>
<th>Author and Year</th>
<th>Geometry, gap area, gap height, pin dimension (mm)</th>
<th>Fluid</th>
<th>Material</th>
<th>( q^* ) (W/cm²)</th>
<th>( G ) (kg/m²s)</th>
<th>( P ) (kPa)</th>
<th>( T_{in} ) (°C)</th>
<th>( x )</th>
<th>Studied parameters</th>
</tr>
</thead>
<tbody>
<tr>
<td>Lee and Lee [8] 2001</td>
<td>Minigap and microgap, ( 20 \times 20, H = 2, 1, 0.4 )</td>
<td>R-113</td>
<td>Stainless steel</td>
<td>Up to 1.5</td>
<td>50–200</td>
<td>—</td>
<td>—</td>
<td>0.15–0.75</td>
<td>( h_{mf}, \Delta P, h_{mp} ) correlation</td>
</tr>
<tr>
<td>Yang and Fujita [9] 2004</td>
<td>Minigap and microgap, ( 20 \times 100, H = 2, 1, 0.5, 0.2 )</td>
<td>R-113</td>
<td>Copper</td>
<td>0.2–9</td>
<td>20–500</td>
<td>118</td>
<td>—</td>
<td>—</td>
<td>Flow patterns, effects of gap height</td>
</tr>
<tr>
<td>Koşar and Peles [10] 2006</td>
<td>Microgap with circular pin-fin, ( 1.8 \times 10, ) staggered, ( H = 0.243, D = 9.5 )</td>
<td>R123</td>
<td>Silicon</td>
<td>3.5–65.5</td>
<td>351–887</td>
<td>—</td>
<td>—</td>
<td>—</td>
<td>Boiling inception, ( \Delta P ), boiling instabilities</td>
</tr>
<tr>
<td>Koşar and Peles [11] 2007</td>
<td>Microgap with hydrofoil pin-fin, ( 1.8 \times 10, ) staggered, ( H = 0.243 )</td>
<td>R-123</td>
<td>Silicon</td>
<td>19–312</td>
<td>976–2349</td>
<td>—</td>
<td>—</td>
<td>Up to 1</td>
<td>( h_{mp}, \text{CHF}, \text{flow pattern and flow map} )</td>
</tr>
<tr>
<td>Lie et al. [12] 2007</td>
<td>Microgap with square pin-fin, ( 20 \times 150, H = 5, H_f = 0.07, S = 0.2, 0.1 )</td>
<td>FC-72</td>
<td>Silicon</td>
<td>0.1–10</td>
<td>287–431</td>
<td>101</td>
<td>—</td>
<td>—</td>
<td>( h_{sp}, h_{tp}, \text{bubble departure diameter and frequency} )</td>
</tr>
<tr>
<td>Krishnamurthy and Peles [13] 2008</td>
<td>Microgap with circular pin-fin, staggered, ( 1.8 \times 10, H = 0.25, D = 0.1 )</td>
<td>DI water</td>
<td>Silicon</td>
<td>20–350</td>
<td>346–794</td>
<td>101</td>
<td>—</td>
<td>Up to 0.17</td>
<td>( h_{mp}, h_{mp}, \text{correlation, flow regime map} )</td>
</tr>
<tr>
<td>Sheehan et al. [14] 2009</td>
<td>Microgap, ( 30 \times 34, H = 0.26 )</td>
<td>FC-72</td>
<td>Copper</td>
<td>3.2</td>
<td>35</td>
<td>—</td>
<td>—</td>
<td>—</td>
<td>Wall temperature</td>
</tr>
<tr>
<td>Qu and Abel [15] 2009</td>
<td>Microgap with square pin-fin, staggered, ( 10 \times 33.8, H = 0.67, S = 0.2 )</td>
<td>DI water</td>
<td>Copper</td>
<td>23.7–248.5</td>
<td>183–420</td>
<td>103–108</td>
<td>30, 60, 90</td>
<td>Up to 0.27</td>
<td>( h_{mp} ) correlation</td>
</tr>
<tr>
<td>Ma et al. [16] 2009</td>
<td>Plain microgap and microgap with square pin-fin, ( W_{CH} = 30, H = 5, H_f = 0.06, 0.12, S = 0.03 )</td>
<td>FC-72</td>
<td>Silicon</td>
<td>Up to 150</td>
<td>0.5,1,2 m/s</td>
<td>—</td>
<td>( T_{sub} = 15,25,35 )</td>
<td>—</td>
<td>Boiling curve, CHF, effects of pin-fins and pin-fin heights</td>
</tr>
<tr>
<td>Yuan et al. [17] 2009</td>
<td>Plain microgap and microgap with square pin-fin, ( W_{CH} = 30, H = 5, S = 0.05, H_f = 0.06, 0.12 )</td>
<td>FC-72</td>
<td>Silicon</td>
<td>Up to 145</td>
<td>0.5, 1, 2 m/s</td>
<td>101</td>
<td>( T_{sub} = 15,25,35 )</td>
<td>—</td>
<td>Boiling curve, CHF, effects of pin-fins and pin-fin heights</td>
</tr>
<tr>
<td>Kim et al. [18,19] 2008, 2010</td>
<td>Microgap, ( 10 \times 37, H = 0.11, 0.21, 0.5 )</td>
<td>FC-72</td>
<td>Copper</td>
<td>Up to 20</td>
<td>55–1270</td>
<td>—</td>
<td>25</td>
<td>Up to 0.99</td>
<td>Flow regime map, ( h_{mp}, \Delta P, T_{in} ) profile</td>
</tr>
<tr>
<td>Sheehan and Bar-Cohen [20] 2010</td>
<td>Microgap, ( 35 \times 10, H = 0.21 )</td>
<td>FC-72</td>
<td>Copper</td>
<td>10.3–26</td>
<td>195.2</td>
<td>—</td>
<td>—</td>
<td>Up to &gt; 0.9</td>
<td>Wall temperature fluctuations</td>
</tr>
<tr>
<td>Alam et al. [21] 2011</td>
<td>Microgap and microchannel, ( 12.7 \times 12.7, H = 0.2–0.4 )</td>
<td>DI water</td>
<td>Silicon</td>
<td>0–100</td>
<td>400–1000</td>
<td>—</td>
<td>86</td>
<td>—</td>
<td>Wall temperature gradient, comparison of microgap and microchannel</td>
</tr>
<tr>
<td>Morshed et al. [22] 2011</td>
<td>Microgap, ( 5 \times 26, H = 0.36 )</td>
<td>DI water</td>
<td>Copper</td>
<td>0–60</td>
<td>45.9–143.8</td>
<td>104</td>
<td>22–80</td>
<td>—</td>
<td>( h_{mp}, \text{flow instabilities, nanowires on wall surface} )</td>
</tr>
<tr>
<td>Guo et al. [23] 2011</td>
<td>Microgap with square pin-fin, ( 10 \times 10, S = 0.03, 0.05, H_f = 0.06, 0.12 )</td>
<td>FC-72</td>
<td>Silicon</td>
<td>Up to 150</td>
<td>0.5, 1, 1.5 m/s</td>
<td>—</td>
<td>( T_{sub} = 25, 35 )</td>
<td>—</td>
<td>Effects of subcooling and jet impingement</td>
</tr>
<tr>
<td>Alam et al. [24] 2012</td>
<td>Microgap, ( 12.7 \times 12.7 )</td>
<td>DI water</td>
<td>Silicon</td>
<td>0–110</td>
<td>420, 690, 970</td>
<td>—</td>
<td>86</td>
<td>—</td>
<td>Gap heights, ( h_{mp}, \Delta P )</td>
</tr>
<tr>
<td>Author</td>
<td>Year</td>
<td>Geometry, gap area, gap height, pin dimension (mm)</td>
<td>Fluid</td>
<td>Material</td>
<td>$q^*$ (W/cm$^2$)</td>
<td>$G$ (kg/m$^2$s)</td>
<td>$P$ (kPa)</td>
<td>$T_{in}$ (°C)</td>
<td>$x$</td>
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<td>---------------------</td>
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</tr>
<tr>
<td>Alam et al. [25]</td>
<td>2012</td>
<td>Microgap, 12.7 × 12.7, $H = 0.19$, 0.285, 0.381</td>
<td>DI water</td>
<td>Silicon</td>
<td>0–71</td>
<td>382–905</td>
<td>101</td>
<td>90–91</td>
<td>—</td>
</tr>
<tr>
<td>Morshed et al. [26]</td>
<td>2012</td>
<td>Microgap $H = 372$</td>
<td>DI water</td>
<td>Copper</td>
<td>Up to 60</td>
<td>45.5, 82</td>
<td>101</td>
<td>21</td>
<td>—</td>
</tr>
<tr>
<td>Isaacs et al. [27,28]</td>
<td>2012, 2013</td>
<td>Microgap with staggered circular pin-fin, 10 × 10, $H = 0.02$, 0.3, and 0.5, $R = 0.6$, 1.0 and 1.6 μm</td>
<td>R245fa</td>
<td>Silicon</td>
<td>Up to 40</td>
<td>598–1639</td>
<td>—</td>
<td>$T_{sub} = 10, 13$ Up to 0.1</td>
<td>$h_{sp}, x$, bubble nucleation, two-phase flow pattern</td>
</tr>
<tr>
<td>Alam et al. [29]</td>
<td>2013</td>
<td>Microgap and microchannel, 12.7 × 12.7, $H = 0.19$</td>
<td>DI water</td>
<td>Silicon</td>
<td>0–85</td>
<td>390, 650</td>
<td>101</td>
<td>91</td>
<td>—</td>
</tr>
<tr>
<td>Alam et al. [30]</td>
<td>2013</td>
<td>Microgap with copper mesh, mesh thickness = 0.16, 5.5 × 26, $H = 0.34$</td>
<td>DI water</td>
<td>Copper</td>
<td>Up to 311</td>
<td>Up to 1300</td>
<td>—</td>
<td>—</td>
<td>—</td>
</tr>
<tr>
<td>Dai et al. [31]</td>
<td>2013</td>
<td>Microgap with copper mesh, mesh thickness = 0.16, 5.5 × 26, $H = 0.34$</td>
<td>DI water</td>
<td>Copper</td>
<td>Up to 311</td>
<td>Up to 1300</td>
<td>—</td>
<td>—</td>
<td>—</td>
</tr>
<tr>
<td>Reeser et al. [32]</td>
<td>2014</td>
<td>Microgap with square pin-fin, inline and staggered, 9.6 × 28.8, $H = 0.305$, $S = 0.153$</td>
<td>HFE-7200/DI water</td>
<td>Copper</td>
<td>1–36/10–110 200–600/400–1300</td>
<td>101</td>
<td>70/95</td>
<td>Up to 0.9/up to 0.22</td>
<td>$h_{gp}$, $h_{sp}$, $\Delta P_{gp}$ and their correlations</td>
</tr>
<tr>
<td>David et al. [33]</td>
<td>2014</td>
<td>Microgap with staggered square pin-fin, 10 × 10, $S = 0.35$, $H_f = 1$</td>
<td>R134a</td>
<td>Copper</td>
<td>30–170</td>
<td>230–380</td>
<td>101</td>
<td>&lt;30</td>
<td>0.2–0.75</td>
</tr>
<tr>
<td>Ong et al. [34]</td>
<td>2014</td>
<td>Microgap with circular pin-fin, staggered (45 deg and 27 deg), radial quadrant microgap with circular pin-fin, $H = 0.12$, $D = 0.08$</td>
<td>R1234ze</td>
<td>Silicon</td>
<td>Up to 99.5</td>
<td>Up to 0.00083 kg/s</td>
<td>—</td>
<td>$T_{sub}$ up to 10°C Up to 0.93</td>
<td>$h_{sp}, x, f$, flow instability, flow patterns</td>
</tr>
<tr>
<td>Yang et al. [35,36]</td>
<td>2015</td>
<td>Radial quadrant microgap with circular pin-fin, 20.25 × 20.25, $D = 0.08$</td>
<td>R1234ze</td>
<td>Silicon</td>
<td>Up to 2100</td>
<td>Up to 0.0042 kg/s</td>
<td>590</td>
<td>27.5</td>
<td>Up to 0.72</td>
</tr>
<tr>
<td>Tamanna and Lee [37,38]</td>
<td>2015</td>
<td>Microgap, 12.7 × 12.7, inlet height 0.2, outlet height 0.2, 0.3, 0.46</td>
<td>DI water</td>
<td>Silicon</td>
<td>0–80</td>
<td>400–1000</td>
<td>101</td>
<td>91</td>
<td>—</td>
</tr>
<tr>
<td>Woodcock et al. [39]</td>
<td>2015</td>
<td>Microgap with staggered Piranha Pin Fin, 2.4 × 22.5, $H = 0.2$, $D = 0.15$</td>
<td>HFE-7000</td>
<td>Silicon</td>
<td>Up to 700</td>
<td>1200–7000</td>
<td>140, 280</td>
<td>—</td>
<td>Up to 0.2</td>
</tr>
</tbody>
</table>
Microgaps gave better heat transfer performance at high heat flux due to confined slug/annular flow, which was dominant, and microchannel performed better at low heat flux due to early occurrence of slug/annular flow [30]. At lower mass flux, microgap outperformed microchannel as well.

Alam et al. further studied the effects of microgap heights on two-phase flow regimes, heat transfer coefficient, and pressure drop [25]. They studied microgap heights from 80 μm to 1000 μm. They found that for microgap heights smaller than 500 μm, confined slug flow was the dominant flow pattern at low heat flux, while confined annular flow was the dominant flow pattern at higher heat flux; for microgap heights larger than 700 μm, bubbly flow was dominant at lower heat flux while slug/annular flow was dominant at higher heat flux, which agreed with the findings in Ref. [4]. Thus, they concluded that confinement occurred in microgap heights smaller than 500 μm, and effect of confinement was negligible for microgap heights larger than 700 μm. The microgap of heights from 100 μm to 500 μm among the tested height range presented best performance in terms of maintaining uniform and low wall temperatures and achieving high heat transfer coefficients. Smaller microgap heights assisted to suppress pressure oscillation and wall temperature oscillation as well. Morshed et al. compared two-phase heat transfer in a microgap with bare copper base surface, and copper base surface with electrochemically grown nanowires [22] using DI water. They studied with bare copper base surface, and copper base surface with electrostatic oscillation and wall temperature oscillation as well.

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Reeser et al. recently studied heat transfer and pressure drop characteristics of HFE-7200 and DI water in inline and staggered micropin-fin arrays [32]. The arrays had a 0.96 × 2.88 cm footprint area and square fin width and height of 153 and 305 μm. For HFE-7200 and DI water, the mass flux ranged from 200 to 600 kg/m²s and heat fluxes ranged from 300 to 1300 W/m², and heat fluxes ranged from 10 to 110 W/cm², respectively. They achieved high exit quality up to 0.9 for HFE-7200. Heat transfer coefficients behavior differed significantly for HFE-7200 and DI water due to different material properties of both working fluids. They also found that pressure drop correlation developed by Qu and Abel [15] and heat transfer coefficient correlation developed by Krishnamurthy and Peles [13] showed poor accuracy in prediction for their work and these correlations needed to be modified.

Ong et al. studied flow boiling of R1234ze in a radial hierarchical fluid network [34]. The concept was to introduce fluid inlet at the center of test device, and fluid was then directed radially to outlets located on the edge of test device. They utilized different sizes of orifices at inlet to distribute fluid flow to subsection of the test devices. The radial quadrant microgap with circular pin-fin mitigated the pressure gradients and reduced temperature gradient as well. They also studied microgap with staggered pin-fin (27 deg and 45 deg), but the results were not compared with radial quadrant test device. The observed two-phase flow instabilities and believed two-phase flow instabilities were related to the degree of inlet subcooling. Schultz and coworkers also tested a radial microgap with embedded pin arrays using the same fluid [35,36]. The test device size was 20.25 mm × 20.25 mm and had eight core heaters and 16 hotspot heaters, and they studied the effects of local hotspot. They found that 50% increase in mass flow rate only resulted in 8% in two-phase heat transfer coefficients. Increase in mass flow rate did not necessarily help to mitigate temperature nonuniformity. Using R-134a in an open-loop, David et al. studied effects of transient heat load on two-phase heat transfer coefficients in a microgap with staggered square pin fins [33]. Their results indicated that temperature was maintained near uniform under both steady state and transient heating. Higher heat transfer coefficient was achieved under transient heating than steady-state heating. Heat transfer coefficient varied with vapor quality, and a peak was observed for vapor quality of 0.55.

Tamanna et al. investigated the effect of expanding the microgap height on flow boiling heat transfer and pressure drop characteristics [37,38]. The microgap was formed with silicon base and polycarbonate cover with the inlet height of 200 μm for all the gaps, and the outlet height increases from 200 μm to 300 μm and to 460 μm. A delay of partial dryout was observed in the 200–460 μm microgap at a heat flux of 79 W/cm², compared to the straight 200–200 μm microgap at a heat flux of 61 W/cm². The expanding microgap with outlet height of 300 μm gives the smallest pressure drop, by providing room for the vapor expansion without excessive flow acceleration, and best wall temperature uniformity of all the three tested heights. Further expansion of outlet height 460 μm increased pressure drop due to unstable boiling and vapor acceleration. The fluctuations in temperature caused by unstable boiling in microgap were found to be independent of fluid quality and heat fluxes [20].

Woodcock et al. developed a piranha pin fin (PPF) structure in a microgap [39]. They investigated flow boiling of HFE-7000 in PPF-enhanced microchannel and achieved heat flux as high as 700 W/cm². Tested mass fluxes ranged from 1200 kg/m²s to 7000 kg/m²s. A staggered array PPF, each of diameter 150 μm and with a 300 μm long tail, was used. The PPFs had open mouths on leading edges, with PPFs wall thickness of 30 μm, and fluid flow could come inside the PPFs and be extracted from the bottom fluid passage of each PPF. This way heat transfer was significantly enhanced in single-phase and two-phase conditions.

In summary, the review on flow boiling in plain microgap and microgap with pin fin surface enhancement revealed the ability of this promising strategy as thermal management method for high heat flux removal. However, fundamental research is still needed to understand the physics of boiling in microgap and especially in microgap with pin fin surface enhancement. The dependences of boiling mode, two-phase heat transfer coefficient and pressure drop on inlet temperature, heat flux, and mass flux are still unclear and require further study.

3 Experimental Study of Water Flow Boiling in Micropin-Fin Heat Sink at Reduced Pressure

3.1 Experimental Setup and Procedure. Here, we describe new flow boiling experiments performed with DI water for pin fin enhanced microgaps. The experiments were performed in a closed loop flow shown schematically in Fig. 3, consisting of gear pump, filter, flow meter, preheater, test section, heat exchanger, and fluid reservoir. The gear pump (Cole-Parmer EW-07002-27) could deliver volumetric flow rates from 4.2 mL/min to 420 mL/min, with maximum differential pressure of 40 psi. The filter (Swagelok inline particulate filter B-4F-7) has a pore size of 7 μm. The flow meter (McMillan S-114) is used to measure volumetric flow rate of fluid in the range of 50–500 mL/min. The nickel ribbon resistance wire preheater is wrapped on the outer surface of the section of tubing upstream of test section to elevate fluid temperature at test sample inlet close to saturation condition. Two-phase flow from the exit of test section is condensed in the heat exchanger (LYTRON LL520G12), which is cooled by a thermostatic bath circulator (LabCompanion RW-1025 G). The capacity of stainless steel fluid reservoir (Swagelok 316L-HDF4-300) is 300 mL. Pressure and temperature are measured at multiple locations in the loop, as marked in Fig. 3. Pressure transducers (Omega PX219-300 G-5V) and T-type thermocouples (Omega HTQSS-116 G-12) are utilized for these measurements.

Figure 4 illustrates the schematic of the test section. The staggered micropin-fin heat sink was microfabricated from silicon and was anodically bonded to Pyrex to form a microgap. The transparency of Pyrex enables flow visualization. The device is sandwiched between the printed circuit board (PCB) and package, which are affixed with screws. The device is electrically connected to the PCB by wire bonding. The package is designed to connect the device to the flow loop and to enable local pressure and temperature measurements. Thermocouples are inserted into inlet and outlet fluid plenums inside package, so inlet and outlet fluid temperatures can be measured at locations that are only 8 mm away from the device. O-rings are used to seal between the device and the package.

A schematic of the test device is shown in Fig. 5. The staggered pin fins cover an area of 1 cm × 1 cm in the center of device. There is one row of pins both upstream and downstream of the staggered pin-fin arrays to redistribute fluid flow. Pressure taps are placed between the staggered pin-fin arrays and inlet and outlet flow redistribution pins, as seen in Fig. 5(a). Figure 5(b) shows the back side of the device. Four platinum resistance heaters are deposited on the back side, also covering an area of 1 cm × 1 cm directly underneath the pin-fin arrays. The heaters also work as resistance temperature detectors (RTDs) due to the near linear dependence of platinum resistance on temperature. The dimensions of tested device are shown in Fig. 6. The pin height is 200 μm, which is the same as the gap height. The overall device size is 28 mm × 13.5 mm, and the total thickness is 500 μm.

Before testing, the heaters were calibrated in an oven to obtain the resistance–temperature curve for each heater. The resistances measured, single-phase heat transfer tests were performed. The
power required to increase water temperature from inlet to outlet was calculated and subtracted from the total power supplied to the heaters to estimate heat loss. Heat loss is estimated from single-phase measurements by

\[ Q_{\text{loss}} = \frac{P_{\text{total}} - nC_p(T_{out} - T_{in})}{m} \]  

(1)

where \( n \) is the mass flow rate of water. The average heat loss was 11% of the total supplied power. These heat loss estimates were used to calculate effective heat fluxes for the two-phase data.
Thus, local two-phase heat transfer coefficient comes in the device at a significant subcooling condition, and

\[ q''_{\text{eff}} = k_s \frac{T_h - T_w}{t} \]  

(11)

where \( t \) is the distance from the pin fin base to heaters, and \( T_h \) is the heater temperature.

The exit quality was calculated from

\[ x = \left[ \frac{q''_{\text{sat}}A_h - mC_p(T_{\text{sat}} - T_{\text{in}})}{h_{\text{vap}}} \right] / \eta_f \]  

(12)

where \( T_{\text{sat}} \) and \( h_{\text{vap}} \) are the water saturation temperature and latent heat of vaporization, both of which are evaluated at device exit pressure.

The estimated measurement uncertainties in absolute scale are listed in Table 3.

### 3.3 Results and Discussion

The operating conditions for each test are presented in Table 4. The inlet temperature varied slightly due to the fluctuation in temperature at the condenser heat exchanger, which affected the reservoir fluid temperature. The fluctuation in condenser temperature was caused by the limited ability of the thermostatic bath circulator to control bath coolant temperature. The fluctuation in condenser temperature also caused saturation pressure at condenser to fluctuate and also the device outlet pressure. Device outlet pressure also increased as fluid quality increased, because increasing quality would increase pressure drop across the tubing from device exit to condenser. For the same mass flux, the boiling area started at the rear portion of the microgap, close to exit and eventually moved forward into the pin fin array. Two-phase heat transfer coefficients for conditions where boiling was in the pin fin array are shown in Fig. 7(a), and similar trends were reported earlier [15]. Other works found two-phase heat transfer coefficient slightly increased with heat fluxes [13,32], and all these works used different device designs and different experimental conditions, such as inlet temperatures and mass fluxes. The decreasing trend of two-phase heat transfer coefficient becomes less dependent on heat flux as heat flux increases.

Two-phase heat transfer coefficient also showed dependence on mass flux due to different degrees of mixing at different mass fluxes. This dependence on mass flux was also observed by other researchers [13,15,32]. The wall superheat and estimated exit quality increase almost linearly with increasing heat flux as shown in Figs. 7(b) and 7(d). At heat flux of 444 W/cm², wall superheat was as high as 60°C. The high wall superheat was because of large degree of inlet subcooling. The pressure drop increases with increasing heat flux, more rapidly at higher heat fluxes as seen in Fig. 7(c), because increase in pressure drop due to vapor phase was more prominent than decrease in pressure drop in single-phase region due to decrease in viscosity.

Flow instabilities were observed at high heat flux. The wall temperatures and pressure drop, for the mass flux 1351 kg/m²s at 267 W/cm², fluctuated over a large range, as shown in Fig. 8. This happened when vapor pressure increased enough to overcome inlet pressure and induced reverse flow. Liquid was pushed backward and not enough liquid flow occurred in the heat sink, causing dramatic increase in temperature. As liquid accumulated at inlet, and pressure was sufficient to overcome pressure drop across the pin-fin arrays, the liquid was forced into the microheat sink again, and the temperature decreased. Two-phase flow instabilities in microchannels have been studied by many other researchers, and

### Table 3 Measurement and data reduction uncertainty

| Pressure drop | ± 1.5% |
| Temperature of water | ± 0.5°C |
| Temperature of wall | ± 1.4°C |
| Heat flux | ± 1% |
| Flow rate | ± 0.8% |
| Heat transfer coefficient | ± 15% |
| Quality | ± 1.9% |

Mass flux is defined as

\[ G = \frac{m}{A_{c,\text{min}}} \]  

(2)

\( A_{c,\text{min}} \) is the minimum cross section area of the microgap, and thus

\[ A_{c,\text{min}} = \left( \frac{W_{ch} - W_{ch}}{S_T} \right) H_{ch} \]  

(3)

The average net heat flux is calculated accounting for heat losses

\[ q''_{\text{eff}} = \frac{P_{\text{total}} - Q_{\text{loss}}}{A_h} \]  

(4)

where \( A_h \) is the total heated area and equals to 1 cm². Water comes in the device at a significant subcooling condition, and boiling only occurs in the finned area close to microgap exit. Thus, local two-phase heat transfer coefficient \( h_{\text{lp}} \) was evaluated in a unit cell area containing a single pin using local \( T_w \) from

\[ q''_{\text{eff}}A_{uc} = h_{\text{lp}}(A_{uc} - A_r)(T_w - T_{\text{sat}}) + h_{\text{lp}}\eta_f A_r(T_w - T_{\text{sat}}) \]  

(5)

where \( A_{uc} \) is the base area of a unit cell, \( A_r \) is the surface area of a single pin fin, and \( A_r \) is the cross section area of a single pin fin. Thus,

\[ A_r = S_lS_L \]  

(6)

\[ A_f = P_f H_f \]  

(7)

\[ A_c = \frac{\pi D^2}{4} \]  

(8)

where \( P_f = \pi D \) is the pin perimeter, and \( H_f \) is the pin height. Assuming that fin tip is insulated, fin efficiency \( \eta_f \) can be calculated using

\[ \eta_f = \frac{\tan h(mH_f)}{mH_f} \]  

(9)

where

\[ m = \sqrt{\frac{h_{\text{lp}}P_f}{k_sA_c}} \]  

(10)

where \( k_s \) is the solid material thermal conductivity, and \( T_{\text{in}} \) is the pin fin base temperature. Assuming one-dimensional conduction in heat sink base

### Table 4 Performed tests parameters

<table>
<thead>
<tr>
<th>Test no.</th>
<th>( T_{\text{in}} ) (°C)</th>
<th>( G ) (kg/m²s)</th>
<th>( \dot{q}_{\text{sat}} ) (W/cm²)</th>
<th>( P_{\text{sat}} ) (kPa)</th>
<th>( P_{\text{out}} ) (kPa)</th>
<th>( T_{\text{sat}} ) at ( P_{\text{in}} ) (°C)</th>
<th>( T_{\text{sat}} ) at ( P_{\text{out}} ) (°C)</th>
<th>( T_{\text{cond}} ) (°C)</th>
<th>( P_{\text{sat}} ) at ( T_{\text{cond}} ) (kPa)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>7–10</td>
<td>1784</td>
<td>237–444</td>
<td>129–157</td>
<td>22–31</td>
<td>106.9–112.8</td>
<td>62.2–68.3</td>
<td>0–5</td>
<td>0.6–0.9</td>
</tr>
<tr>
<td>2</td>
<td>6–7</td>
<td>1637</td>
<td>251–391</td>
<td>111–130</td>
<td>26–36</td>
<td>102.6–107.1</td>
<td>65.9–73.4</td>
<td>0–3</td>
<td>0.6–0.8</td>
</tr>
<tr>
<td>3</td>
<td>7–9</td>
<td>1351</td>
<td>198–267</td>
<td>88–94</td>
<td>24–27</td>
<td>96.1–97.9</td>
<td>64.1–66.7</td>
<td>0–5</td>
<td>0.6–0.9</td>
</tr>
</tbody>
</table>
Fig. 7  (a) Two-phase heat transfer coefficient versus effective heat flux, (b) boiling curve, (c) pressure drop across staggered pin-fin arrays versus effective heat flux, and (d) exit quality versus effective heat flux.

Fig. 8  (a) Temperature oscillations at $G = 1351$ kg/m$^2$s and $q_{in} = 267$ W/cm$^2$ and (b) pressure drop at $G = 1351$ kg/m$^2$s and $q_{in} = 267$ W/cm$^2$. 
pressure restrictor at inlet was recommended to stabilize two-phase flow [45–48].

High-speed images taken at a frame rate of 4200 fps at multiple values of heat flux for test 1 are shown in Fig. 9. Boiling initiation started in the vicinity of the outlet inside the device and moved toward inlet, as supplied heat flux increased. The dotted area in Fig. 10 indicates two-phase area. The majority of finned area was in single phase due to inlet subcooling. Once vapor bubbles formed, they expanded rapidly into a triangular-shaped vapor wake for all the heat fluxes. This agrees with observation reported in Refs. [27,28]. Due to the substantially subcooled inlet condition, vapor phase only existed in a few rows of pins close to exit before reverse flow and oscillations occurred.

3.4 Conclusions. In this study, subcooled flow boiling of water across staggered circular micropin-fin array heat sink was investigated. The main conclusions from the new experiments are as follows:

(1) Two-phase heat transfer coefficient decreased with increasing heat flux, and the dependence became less pronounced at higher heat fluxes. The highest heat transfer coefficient achieved was 48 kW/m² K, and the lowest was 25 kW/m² K.

(2) Wall superheat and quality increased almost linearly with heat flux. The wall superheat was as high as 60°C at effective heat flux of 450 W/cm².

(3) The pressure drop ranged from 60 kPa to 130 kPa for the tested mass flux and heat flux ranges and increased rapidly at higher heat flux due to accelerational vapor flow contributions.

(4) Instead of traditional two-phase flow patterns such as bubbly flow and slug flow, triangular-shaped vapor wakes were observed after bubble nucleation sites.

(5) Reverse flow would cause pressure and temperature to fluctuate dramatically and degrade the effectiveness of cooling significantly if no provisions made for suppressing instabilities using flow restrictors or other means.

For future work, flow restrictors at the inlet are recommended to prevent reverse flow. Elevated inlet temperatures could be used to reduce wall superheat and to achieve higher vapor quality. Effects of nonuniform heating should be studied since it will mimic a real microelectronic device better due to local hot spots.

Acknowledgment

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Nomenclature

\[ A_c = \text{single pin fin cross section area, m}^2 \]
\[ A_{c,\text{min}} = \text{minimum channel cross section area, m}^2 \]


References


