TWO-PHASE HEAT SINKS WITH MICROPOROUS COATING

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ABSTRACT
To minimize flow boiling instabilities in two-phase heat sinks, two different types of microporous coatings were developed and applied on mini- and small-channel heat sinks and tested using degassed R245fa refrigerant. The first coating was epoxy-based and was sprayed on heat sink channels while the second coating was formed by sintering copper particles on heat sink channels. Mini-channel heat sinks had overall dimensions 25.4 mm x 25.4 mm x 6.4 mm and twelve rectangular channels with a hydraulic diameter 1.7 mm and a channel aspect ratio of 2.7. Small-channel heat sinks had the same overall dimensions, but only three rectangular channels with a hydraulic diameter 4.1 mm and channel aspect ratio 0.6. The microporous coatings were found to minimize parallel channel instabilities for mini-channel heat sinks and to reduce the amplitude of heat sink base temperature oscillations from 6 °C to slightly more than 1 °C. No increase in pressure drop or pumping power due to the microporous coating was measured. The mini-channel heat sinks with porous coating had in average 1.5-times higher heat transfer coefficient than uncoated heat sinks. Also, the small-channel heat sinks with the "best" porous coating had in average 2.5-times higher heat transfer coefficient and the critical heat flux was 1.5 to 2-times higher compared with the uncoated heat sinks.

INTRODUCTION
Flow boiling in micro-, mini-, and small-channel heat sinks offers several advantages over flow boiling in large-channel heat sinks in terms of reduced heat sink thermal resistance (i.e. reduced device temperature) and increased heat sink base heat flux. However, flow boiling instabilities and the resulting oscillations in pressure and heat sink base temperature limit the CHF condition (below the predicted value). Such oscillations can potentially cause structural failure not to mention that the device being cooled experiences large temperature swings that can adversely affect performance.
including flow reversal using water in an electrically heated evaporator consisting of six 1 m x 1 mm large parallel channels. In some of the channels of the evaporator, the pressure drop across the channels was found to increase with vapor generation resulting in the decrease of the mass flow rate. As a result, the mass flow rate through other channels increased and led to instabilities. Heatron et al. [4, 5] tested s everal parallel triangular micro-channels with hydraulic diameters from 0.1 to 0.16 mm and number of channels from 17 to 26 using water. They observed that once bubble nucleation begins, the bubble grows rapidly and occupies the entire channel. The rapid bubble growth pushes the liquid-vapor interface on both caps of the vapor slug at the upstream and the downstream ends and leads to a reversed flow. After the bubble reaches the inlet or the outlet plenum, the channel gets rewetted and refilled with liquid. Not all the channels are rewetted and refilled with liquid at the same time resulting in large temperature differences across the heat sink. Qu and Mudawar [11, 12] have visualized compressible volume oscillations in heat sinks. Adiabatic microchannel instabilities for 21 parallel micro-channels with channel width 215 μm and channel depth 821 μm. The compressible flow instability was caused by a relatively large pressure drop at the upstream plenum. Chang and Pan [6] tested a micro-channel heat sink with fifteen 99.4 μm wide and 76.3 μm deep parallel channels using DI water at a working fluid. They observed forward and reverse slug flow or an nular flow in mportant in every channel. Severe pressure oscillations were noticed as well as the vapor flow into the upstream plenum. Lee and Yao [13] conducted tests with DI water and 48 parallel micro-channels that were 0.235 mm wide and 0.71 mm deep. They have visualized long period cyclic fluctuations in heat sink temperature as well as the water inlet temperature. The oscillations in heat sink surface temperature of 15 °C were measured.

Parallel channel instabilities present in two-phase heat sinks may be reduced by eliminating b y one the following means: (1) introducing a relatively large pressure drop at the inlet of each channel; (2) decreasing the superheat required for the onset of nucleate boiling inside the channels; or (3) physically separating the vapor phase from the two-phase mixture in the channels. Kandlikar [10] and Bergles et al. [3] predicted that at b y i ncluding a n r estrictor between each channel, the p arallel channel instabilities can be avoided. Koşar et al. [7], Agostini et al. [14], and Wang et al. [15] tested micro-channel heat sinks with integrated inlet slits and showed that the instabilities were suppressed. The penalty was an increased total pressure drop. In addition, the inlet slits (or r estrictors) are very difficult t o manufacture due to very small size. Furthermore, one r estrictor de sign may not be effective across a wide range of operating conditions. Koşar et al. [16] and Kuo and Peles [17] used reentrant cavities on the micro-channel inner walls to decrease the nucleation superheat and consequently the instabilities. These reentrant cavities were formed on to the micro-channel walls using micromachining techniques. The cavities on microchannel walls were found to successfully delay the parallel channel instabilities and to increase the CHF. Although, the e b enfits of t he cavities were demonstrated, the micromachining method is too expensive for high volume micro-channel heat sinks. David et al. [18] used a hydrophobic membrane to o cally vent the vapor f rom a microchannel. This concept is limited to high surface tension fluids a nd becomes ineffective for many e lectronic cooling fluids that typically have low surface tension (e.g. refrigerants, dielectric fluids). Compressible volume instability can be suppressed by increasing the channel size and decreasing the bubble departure diameter.

The objective of t his paper is to minimize the parallel channel instabilities and the rapid bubble growth instability by developing heat sinks with microporous coating on channel walls. The microporous coating can reduce the incipient boiling superheat at nucleation vapor bubbles, decrease the bubble departure diameter, and increase the number of channels. The model was used to determine a working fluid with most favorable thermophysical properties. It was a lso critical that t he s elected working fluid had a positive pressure r esistive at ambient temperatures as low as 15°C to ensure that no gases entered the heat sink. Qu and Mudawar [19] a nd the Homogeneous E quilibrium Model was used to determine the two-phase pressure drops across the channels.

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The heat sink overall dimensions were 25.4 mm x 25.4 mm x 6.4 mm. The channel width for the first three heat sinks tested was 1.2 mm in parallel because previous work [20] successfully demonstrated that a microporous coating could be applied to 1 mm wide pin fins, which is approximately the same dimension as the channel size. The model was a lso used to evaluate different heat sinks with different channel depths and number of channels. The d ependence of t he heat sink b ase temperature on the channel depth and the number of channels for R245fa refrigerant are shown in Figure 1.

**NOMENCLATURE**

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
</tr>
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<tbody>
<tr>
<td>wch</td>
<td>channel width (mm)</td>
</tr>
<tr>
<td>dh</td>
<td>channel depth (mm)</td>
</tr>
<tr>
<td>Nch</td>
<td>number of channels</td>
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</tbody>
</table>

**TEST ARTICLES**

A heat sink model was developed to compute the heat transfer coefficient and the pressure drop across the heat sinks for a given heat flux, coolant flow rate and heat sink geometry. The two-phase heat transfer coefficient was predicted using a modified Chen’s correlation [19] and the Homogeneous Equilibrium Model was used to compute the two-phase pressure drops across the channels.

The model was used to determine a working fluid with most favorable thermophysical properties. It was also critical that the selected working fluid had a positive pressure resistance at ambient temperatures as low as 15°C to ensure that no gases entered the heat sink. R245fa refrigerant was selected as the working fluid.

The heat sink overall dimensions were 25.4 mm x 25.4 mm x 6.4 mm. The channel width for the first three heat sinks tested was 1.2 mm in parallel because previous work [20] successfully demonstrated that a microporous coating could be applied to 1 mm wide pin fins, which is approximately the same dimension as the channel size. The model was also used to evaluate different heat sinks with different channel depths and number of channels. The dependence of the base temperature on the channel depth and the number of channels for R245fa refrigerant are shown in Figure 1.
Figure 1: Heat sink base temperature versus channel depth for various heat sink designs

As the channel depth increases, the total cross-sectional area available for flow and the heat sink surface area increase; however, the heat sink surface efficiency decreases as the channel heat transfer coefficient decreases. For a low number of channels (e.g. 5 channels), very deep channels are needed to provide sufficient friction factor. For a large number of channels (e.g. 15 channels), no improvement in performance is achieved by increasing the channel depth as shown in Figure 1, where the heat sink base temperature increases with increasing channel depth for the 12 and 15 channel cases. For reference, the fin thickness for 15 channels is only 0.3 mm resulting in high heat conduction resistance along the fins.

After reviewing various combinations of the number of channels and the channel depth, it was decided to design a heat sink with twelve 3.2 mm deep channels and 0.6 mm wide fins. The heat sink base thickness was decided to be 3.2 mm to ensure good heat sink flexibility and sufficient structural integrity. The binder or glue binds the particles together to form the required heat dissipation surface. Since the particles are solid and the binder is usually very viscous, a volatile carrier liquid is used to allow for the application of the particles and binder, and for the application of the coating material to the surface. The carrier evaporates after the applied coating solidifies, and the binder and particles form a microporous coating on the heat sink surface. The ABM coating was applied to the surface of the heat sink by a spray coating method. The layer thickness was controlled within ±0.3 mm, and the coating was uniform and adhered to the heat sink surface. The coating was obtained by spraying a compressed gas on the heat sink surface.

The key requirements for the coating were uniform thickness, good adhesion, and good stability. Two different approaches for the fabrication of the microporous coating were investigated. The first approach used an ABM (Aluminum powder, B=Brushable ceramic epoxy, M=Methyl ethyl ketone) coating while the second approach used a microporous coating by sintering of carrier particles on channel walls. The ABM coating consists of particles, binder, and carrier. The particles provide the structure material to create the required re-entrant cavities. The binder or glue binds the particles together and to a heat dissipation surface. Since the particles are solid and the binder is usually very viscous, a volatile carrier liquid is used to allow for the application of the particles and binder, and for the application of the coating material to the surface. The ABM coating was applied to the surface of the heat sink by a spray coating method. The layer thickness was controlled within ±0.3 mm, and the coating was uniform and adhered to the heat sink surface. The coating was obtained by spraying a compressed gas on the heat sink surface.

Table 1: Heat sink parameters and microporous coating types

<table>
<thead>
<tr>
<th>Heat Sink Test Sample</th>
<th>wch (mm)</th>
<th>dch (mm)</th>
<th>Nch</th>
<th>Microporous Coating</th>
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<tr>
<td>1</td>
<td>1.2</td>
<td>3.2</td>
<td>12</td>
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<td>3</td>
<td>1.2</td>
<td>3.2</td>
<td>12</td>
<td>Sintered coating 1</td>
</tr>
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<td>3.2</td>
<td>3</td>
<td>none</td>
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<td>5</td>
<td>5.8</td>
<td>3.2</td>
<td>3</td>
<td>Sintered coating 1</td>
</tr>
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<td>6</td>
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<td>3.2</td>
<td>3</td>
<td>Sintered coating 2</td>
</tr>
<tr>
<td>7</td>
<td>5.8</td>
<td>3.2</td>
<td>3</td>
<td>Sintered coating 3</td>
</tr>
<tr>
<td>8</td>
<td>5.8</td>
<td>3.2</td>
<td>3</td>
<td>ABM coating</td>
</tr>
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to the surface and can be expected to be stable through multi-year service. A photograph of the ABM coating applied to a mini-channel heat sink is shown in Figure 2.

Figure 2: ABM microporous coating on a copper heat sink fin; (a) 200-times magnification, (b) 1540-times magnification

The thickness of the coating was approximately 45 μm. The average pore size of the coating ranges approximately from 2 to 10 μm.

The Sintered Coating was produced by bonding copper metal particles to a heat sink wall by solid state sintering. Sintering forms a strong metallurgical bond among the particles with very good adhesion to the base substrate. Three different variations of the coating were tested. Sintered coating 1 had approximately 5 to 15 μm large pores and was 35 μm thick; Sintered coating 2 had approximately 10 to 30 μm large pores and was 70 μm thick, and Sintered coating 3 had approximately 10 to 30 μm large pores and was 500 μm thick. A photograph of Sintered coating 1 on the heat sink fin is shown in Figure 3.

Figure 3: Sintered coating 1 on a copper heat sink fin; (a) 200-times magnification, (b) 1000-times magnification

The average coating thickness was around 35 μm. As shown, the average pore size ranges approximately from 5 to 15 μm and provides a high density of nucleation sites for bubble nucleation.

TEST BED AND TEST SECTION

A two-phase pumped loop test bed for testing heat sinks with and without the porous coating was designed and built. A layout of the test loop is shown in Figure 4.

The test bed is composed of a test section, a condenser with a cooling bath, a DC pump (Fluid-O-Tech MG213XPB17), a single phase flow filter or filter 1 (440 μm) and a flow meter 1 (McMillan 1 04-8E, ± 0.05 LPM), a two-phase flow filter or filter 2 (15 μm) and a flowmeter 2 (McMillan 1 04-5E, ± 0.005 LPM) in parallel to the single phase flowmeter and the filter, a pre-heater with a PID controlled power supply, and a two-phase reservoir to regulate the loop pressure (by adjusting the temperature of the reservoir). A charging port was placed before the pump and used to evacuate and charge the loop. The loop was also...
instrumented with several T-type thermocouple probes (Omega TJC48-CPSS-062G-2) that were inserted into the refrigerant lines. All the loop components were well insulated using silicon foam insulation.

Figure 4: A layout of the test bed for testing heat sinks

Several factors were considered during the design of the test section. The test section had to ensure a simple installation of the heat sink and to provide temperature measurements of the heat sink base as well as temperature and pressure measurements of the refrigerant at the inlet and outlet of the heat sink. Furthermore, the test section had to provide visual access to the boiling process inside the heat sink. A cross-section view of the test section with a heat sink is provided in Figure 5. The test section housing was fabricated from high temperature resistant PolyEther-Ether-Ketone (PEEK) plastics. The heat sink was inserted into the PEEK housing. A silicon O-ring was inserted into a groove formed in the PEEK to seal the heat sink from the ambient. Borosilicate glass was placed atop the heat sink and compressed against the PEEK housing. A copper block with two 500 W cartridge heaters was used to simulate the heat source.

To obtain an accurate measurement of the amount of heat that was conducted into the heat sink, three holes were drilled into the pedestal of the heater with a depth of 7.87 mm and a distance from each other of 3.18±0.05 mm. Three T-type thermocouples were inserted into the holes in the channels. A second O-ring was used to seal the gap between the borosilicate glass and the PEEK housing. A spring-loaded bolt with a swivel pad was used to uniformly press the heater against the heat sink. Shin-Etsu G-751 thermal interface material was applied on the interface between the heat sink and the heater block to minimize thermal interface resistance. Four thermocouples with a solid ceramic insulation were made and inserted through the heater block and pressed against the heat sink base. The thermocouples were located 7.6 mm from the edge of the heat sink and 0.5 mm underneath the channels. It is believed that the thermocouples were close enough to the heat sink channels so that variations in refrigerant flow could be registered. A thermocouple (Omega TJC48-CPSS-062G-2) and an absolute pressure transducer (Omega PX481AD-100-G5V, ±0.1 PSI) were placed at the inlet to the heat sink and another thermocouple and absolute pressure transducer were placed at the outlet of the heat sink. In addition, a differential pressure transducer (Omega P X2300-25DI, ±0.06 PSI) was connected between the inlet and the outlet of the heat sink. The heater block was well insulated using an amorphous silica blanket with thermal conductivity 0.02 W/m K.

Figure 5: Test section with the heat sink (cross-section view)

All the heat sinks were fabricated from C1100 copper by CNC milling. Six of the heat sinks were coated with microporous coating and two remained uncoated as indicated in Table 1. A photograph of two of the heat sinks is shown in Figure 6.

Before proceeding with the test, the test loop was evacuated to <0.13Pa (<10⁻³ torr) and charged with degassed R245fa refrigerant. The level of dissolved gas in the refrigerant was <5 ppb. No additional degassing of the refrigerant was required. During the operation, the heat from the heater block vaporized a portion of the refrigerant. A two-phase refrigerant was condensed and subcooled in the condenser. Slightly subcooled liquid entered the test section which contained the heat sink. Temperatures, pressures, and flow rates were recorded at a sampling rate of 1 Hz.
RESULTS AND DISCUSSION

Before proceeding with two-phase heat sink tests, a series of single phase heat sink tests were completed to verify the operation of the test bed. The heat applied to the heat sink was calculated from measured current and voltage (the accuracy of the measurement is estimated to be ±3%). The heat load calculated from voltage and current was in average 5% higher than the heat load calculated from the heat conduction through the heater pedestal. The difference was due to heat losses through the insulation. The amount of heat absorbed by the refrigerant and the amount of heat removed at the condenser were calculated to be less than 3% different from the amount of heat applied to the heat sink. These results confirmed that the test loop was well insulated and that the accuracy of the heat load calculated from the conduction through the pedestal was better than 3%. After verifying the conservation of energy with the single phase flow, two-phase tests at various conditions were then performed.

After starting the two-phase tests with heat sink 1, significant compressible volume instabilities were observed [11, 12, and 13]. To eliminate this type of instability, the valve at the inlets to the test section was slightly closed. This additional flow resistance at the inlets to the test section was sufficient to suppress the compressible volume instability without measurable increase in the pumping power.

After eliminating the instabilities caused by the two-phase reservoir upstream of the test section, tests were performed with different heat sinks. The heat sinks were tested at various heat fluxes and a constant flow rate. An example of the refrigerant temperatures at different heat fluxes is given in Figure 7.

As shown, the refrigerant temperatures at the inlets and outlets of the test section remained steady during the test. This was achieved by controlling the temperature of the two-phase reservoir. As the heat load increased and more vapor was produced, the excess liquid was displaced into the reservoir. The condenser outlet temperature also increased as the heat flux increased. The reason for this can be understood as follows: as more heat was applied to the heat sink, more vapor was generated and a larger portion of the refrigerant surface was then used to condense the vapor. Thus, the condenser area available for subcooling the liquid was reduced. The inlet temperature was subcooled in average for 2.9 °C. The average refrigerant temperature at the outlet of the heat sink was 51.6 °C and the average refrigerant pressure at the outlet of the heat sink was measured to be 362.7 kPa. The saturation temperature for R245fa at 362.7 kPa is exactly 51.6 °C (NIST REFPROP 8.0). The average condenser outlet temperature was 26.8 °C meaning that the condenser was cooled down to 26.8 °C. The pre-heater reduced the refrigerant temperature by 2.9 °C. Subcooling the refrigerant less than 2-3 °C is very difficult to achieve since the pre-heater setting with subcooling less than 2 °C easily results in boiling of the refrigerant within the pre-heater.

Heat sink 1 was tested at a flowrate of 0.3 LPM and at base heat fluxes that ranged from 0 to 214 W/cm² (and back to 0 W/cm²). The base heat flux was defined as the heat flux into the heat sink base while the wall heat flux was defined as the heat flux into the refrigerant (from the heated wall). For discrete base heat fluxes within this range, the refrigerant pressures at the inlets and the outlet were recorded at a sampling rate of 1 Hz and are presented in Figure 7.
The oscillations in pressure are clearly present; however, they significantly affect the heat sink base temperature only at heat fluxes above ~150 W/cm² as shown in Figure 9. Thermocouples TC1 to TC4 were located underneath the heat sink 7.6 mm from the edge of the heat sink and 0.5 mm underneath the channels. (In Figure 9, the thermocouple TC4 readings overlap the rest of the thermocouple readings at 214 W/cm².)

After testing heat sink 1, heat sinks 2 and 3, with different types of microporous coatings were also tested. Heat sinks 2 and 3 were tested at exactly the same test conditions as heat sink 1. For heat sink 3, the results for the refrigerant pressures at the inlet and outlet are presented in Figure 10 and the heat sink base temperatures in Figure 11.

For heat sink 3, the amplitude of the pressure oscillations is smaller than that for heat sink 1 and the maximum amplitude of the oscillations in the heat sink base temperature is only 1.4 °C (compared with 5.8 °C for the uncoated sample). Similar results were obtained for heat sink 2. Heat sinks with porous coating on the channel walls therefore result in more stable operation and have smaller temperature oscillations. Cooling an electronic device with a maximum heat sink temperature oscillation of ~1 °C is much more desirable. For completeness, some discrepancies were noted in the temperatures measured at different locations on the heat sink, and the test was therefore repeated several times and it was found that the differences were due to the thermocouple contact resistances. Therefore, it
Heat transfer performances of heat sinks 1 to 3 were compared in terms of heat sink thermal resistance and pressure drop through the heat sink. Four steady-state tests were conducted at base heat transfer of 50 W/cm² and single-phase flow. The average temperature drop through the heat sink was 0.425 °C-cm²/W. For the heat sink 2 at a base heat flux of 50 W/cm², if the heat sink base temperature is even smaller than it appears in Figure 11. For reference, the model predicts that the temperature of the heat sink base varies only 1 °C at 225 W/cm².

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A comparison of the thermal resistances of the three heat sinks clearly shows that the heat sinks with the porous coating result in a lower thermal resistance at all heat fluxes. The largest difference occurs at the lower heat fluxes. Ammerman and You [23] observed that boiling surfaces with microporous coating have a higher number of nucleation sites and higher initial vaporization. Therefore, they have observed using FC-72 and a microporous coating that as initial vaporization increased within the microporous coating, the emulsion bubble grew rapidly and the heat sink thermal resistance decreased with increasing heat flux up to ~100 W/cm². To better understand this behavior, the model was used to predict the heat sink thermal resistance. Also no differences were observed in the amplitude of heat sink base temperature. For example, the average temperature drop through the heat sink was 0.425 °C-cm²/W. For the heat sink 2 at a base heat flux of 50 W/cm², if the heat sink base temperature is even smaller than it appears in Figure 11. For reference, the model predicts that the temperature of the heat sink base varies only 1 °C at 225 W/cm².

Figure 12: Heat sink thermal resistance versus base heat flux for heat sinks 1 to 3

For all heat sinks, the thermal resistance decreases with increasing heat flux up to ~100 W/cm². To better understand this behavior, the model was used to predict the heat sink thermal resistance. For example, the average temperature drop through the heat sink was 0.425 °C-cm²/W. For the same heat sink tested at the same heat flux, the thermal resistance was reduced to 0.104 °C-cm²/W as the flow regime changed from a single-phase flow to a two-phase flow. For example, at a base heat flux of 50 W/cm², if the heat sink base temperature is to be maintained at 50 °C, a single phase loop will require a coolant at 29 °C while a two-phase loop could use a coolant with a temperature of 45°C. In addition, the flow rate for the two-phase flow is 5-times lower than for the single-phase flow, which translates into a significant reduction in the pumping power, which directly depends on the flowrate.

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base heat flux of 115 W/cm², the thermal resistance was 0.141 °C·cm²/W at an inlet subcooling of 12 °C, while the thermal resistance was reduced to 0.093 °C·cm²/W for an inlet subcooling of 3 °C. It is expected that by having a saturated liquid at the inlet, the heat sink with no subcooling, the thermal resistance will be further reduced. By increasing the amount of subcooling from 3 to 12 °C, the heat sink wall temperature did not however decrease. This is because a single phase heat transfer coefficient is several times lower than that associated with the uncoated heat sink (heat sink 1). Heat sink 5 had the highest thermal resistance which was almost twice that of the heat sink base temperature oscillations at subcooling 8 °C and 12 °C was 1.1 °C while it was slightly increased to 1.4 °C as the inlet subcooling decreased to 3 °C.

Heat sinks 4 to 8 were tested at two different flow rates: 0.3 LPM and 0.6 LPM, and at different heat fluxes. Heat sinks 4 to 8 reached CHF at lower base heat flux than heat sinks 1 to 3. At 0.3 LPM (the same flow rate used to test heat sinks 1 to 3), heat sinks 4, 5, and 6 reached the CHF at a wall heat flux of ~23 W/cm² while heat sinks 7 and 8 reached the CHF at wall heat flux ~40 W/cm². For reference, heat sinks 1 to 3 reached CHF at wall heat flux of ~23 W/cm². Since heat sinks 1 to 3 had a total wetted area that was 2.5-times higher than heat sinks 4 to 6, they were able to handle 2.5-times higher base heat fluxes (by only considering uncoated heat sinks and neglecting the increase of the wetted area due to the porous coating). For heat sinks 4 to 8, the oscillations in temperature were similar to oscillations in pressure for heat sinks 3 shown in Figure 10, while the oscillations in temperature were reduced to ±0.25 °C. This was expected since the channel hydraulic diameter was larger for heat sinks 1 to 3 and the bubbles were not radially restricted by the channel walls.

The summary of the test results at 0.3 LPM for heat sinks 4 to 8 are presented in Figure 13. The thermal resistance decreases with increasing heat flux for all of the heat sinks, similar to the results obtained with heat sinks 1 to 3 reported above. There is however an even more significant improvement in terms of the reduction in the heat sink thermal resistance (or increasing heat sink base temperature oscillations in pressure) for the heat sinks with larger base heat fluxes (by only considering uncoated heat sinks and neglecting the increase of the wetted area due to the porous coating). For heat sinks 4 to 8, the oscillations in temperature were similar to oscillations in pressure for heat sinks 3 as shown in Figure 10, while the oscillations in temperature were reduced to ±0.25 °C. This was expected since the channel hydraulic diameter was larger for heat sinks 1 to 3 and the bubbles were not radially restricted by the channel walls.

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through the heat sink for any of the tested heat sinks.

- Mini-channel heat sinks with porous coating (heat sink 2 and d 3) resulted in average 1.5-times higher heat flux than the uncoated microchannel heat sink (heat sink 1).
- Small-channel heat sinks with porous coating (heat sinks 5 to 8) had an average 2.5~2.7-times higher heat transfer coefficient and 1.5~2-times higher critical heat flux than the uncoated heat sink (heat sink 4).
- Heat sinks with large channels and optimized porous coating (heat sink 7 and 8) can remove heat at the same or lower heat sink thermal resistance and are more stable (improved temperature uniformity of ±0.5°C) than the heat sinks with almost four times smaller channels without the porous coating (heat sink 1).

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