EXPERIMENTAL STUDY ON THE EFFECT OF STABILIZATION ON FLOW BOILING HEAT TRANSFER IN MICROCHANNELS

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ABSTRACT
The effect of flow instabilities on flow boiling heat transfer in microchannels is investigated using water as the working fluid. The experimental test section has six parallel rectangular microchannels with each having a cross sectional area of $1054 \times 197$ microns. Flow restrictors are introduced at the inlet of each microchannel to stabilize the flow boiling process and avoid the backflow phenomena. The mass flow rate, inlet temperature of water, and the electric current supplied to the resistive cartridge heater are controlled to provide quantitative heat transfer information. The results are compared with the unrestricted flow configuration.

INTRODUCTION
The advancements in microprocessors and other high power electronics have resulted in increased heat dissipation from those devices. In addition, to reduce cost, the functionality of microprocessor per unit area has been increasing. The increase in functionality accompanied by reduction in chip size has caused its thermal management to be challenging. In order to dissipate the increase in heat generation, the size of conventional fin-type heat sinks has to be increased. As a result, the performance of these high heat flux generating electronics is often limited by the available cooling technology and space to accommodate the larger conventional air-cooled heat sinks.

One way to enhance heat transfer from electronics without sacrificing its performance is the use of heat sink with many microchannels and liquid water passing through it. Because of the small size of microchannel heat sink, the performance of a computer system can also be increased by incorporating additional microprocessors at a given space without the issue of over-heated or burned-out chips. The present paper involves cooling of electronic devices using two-phase flow in microchannel heat sink. Two-phase heat transfer has significant advantages over single-phase heat transfer because flow rates are smaller through the use of the latent heat of vaporization, approximately uniform fluid and solid temperatures can be obtained, and it can also be directly coupled with a refrigerant system to provide a lower coolant temperature.

There are, however, complexities associated with vaporization in multiple narrow channel arrays that are not completely understood. The phenomenon characterized by vapor expansion in both the upstream and downstream directions causing flow reversal was observed by Kandlikar et al. [1] and also by Kandlikar and Balasubramanian [2]. Both employed a high-speed digital video camera to observe this behavior in minichannels and microchannels. Similar flow instabilities were observed by Li et al. [3] and Peles [4], among other investigators.

Kandlikar [5] reported that flow instabilities in microchannels are due to rapid bubble expansion and occasional flow reversal, and they cause a major concern in implementing flow boiling in microchannels. Because of this, research on obtaining experimental heat transfer and pressure drop data for flow boiling of water in microchannels is immediately warranted. Focusing on flow instabilities, Kandlikar et al. [6] found that the heat transfer performance in microchannels can be improved by using flow restrictors to stabilize the flow in microchannels, and partially stabilized flow was observed for the first time. Further research is needed to study the effect of different pressure drop elements as flow restrictors. Hence, the objective of the present work is to experimentally investigate the effect of pressure drop elements on microchannel heat transfer performance.
NOMENCLATURE

- $G$ = Mass flux, kg/m$^2$/s
- $q''$ = Heat flux, kW/m$^2$
- $q_{in}$ = Power input to the test section, W
- $q_{loss}$ = Heat loss from the test section, W
- $T$ = Temperature, °C
- $P$ = Pressure, kN/m$^2$
- $\Delta P$ = Pressure drop, kPa
- PDE = pressure drop element
- $I$ = Electrical current, A
- $T_s$ = Surface temperature, °C
- $T_{Amb}$ = Ambient air temperature, °C

Subscript

- in = Inlet

LITERATURE REVIEW

Kandlikar et al. [1] studied flow boiling of water in parallel minichannels using high-speed photographic observation. They found vapor bubble growth occurs in the direction counter to bulk flow, forcing liquid and vapor to flow back into the inlet manifold. The reverse flow phenomenon in small diameter parallel multichannels evaporator was clearly documented.

Li et al. [3] experimentally studied the flow boiling instability in two parallel microchannels using low mass flux and various heat fluxes. The two microchannels were of triangular cross-section with topside width of 100 microns and hydraulic diameter of 51.7 microns. The spacing between the closest edges of the microchannels was 20 microns. Flow reversal was observed using a high-speed video camera. The flow instabilities were found to cause significant oscillations to the inlet and outlet temperatures, inlet pressure and the pressure drop in the two parallel microchannels configuration. The authors concluded that flow boiling instabilities involving flow reversal should be of great concern for the design of micro evaporator.

Brutin et al. [7] experimentally investigated two-phase flow instabilities in rectangular minichannels with hydraulic diameter of 889 microns. The two-phase flow oscillation phenomenon was visually detected during flow boiling in minichannels. The phenomenon was caused by the formation of vapor slug which pushes the liquid phase back to the entrance of their test section.

Peles [4] mentioned the importance of addressing and studying flow instabilities in microchannels that will be used in flow boiling applications. Hetsroni et al. [8] found that explosive vaporization and significant pressure drop fluctuations occur in high heat flux situations. Flow reversal is observed in some microchannels, and they are caused by expanding bubbles pushing the liquid-vapor interface in both upstream and downstream directions. Steinke and Kandlikar [9] reported flow reversal in microchannels where the vapor interface moves in a direction counter to the bulk fluid flow. Balasubramanian and Kandlikar [10] reported severe flow maldistribution cause by back flow extended into the inlet manifold.

The summary of the above literature review can be found in Table 1. Very few researchers have reported any work on preventing the flow reversal phenomenon in microchannels.

EXPERIMENTAL FACILITY

The experimental setup is shown in Fig. 1 and consists of a water supply loop, microchannel test section, data acquisition system, and high-speed digital video system. The experimental setup is designed to provide degassed water at a constant flow rate and temperature to the test section. For simplicity, Fig. 1 shows only the water supply loop and the test section.

The inputs to the test section are the working fluid (water) and the converted heat energy from the supplied electric current. The outputs from the test section are the heated working fluid and the heat loss from the test section.

![Figure 1 Water supply loop and test section](image-url)
**Table 1** Summary of studies on flow boiling in small channels

<table>
<thead>
<tr>
<th>Author/year</th>
<th>Fluid</th>
<th>Operating conditions</th>
<th>Channel geometry (mm)</th>
<th>Remarks</th>
</tr>
</thead>
<tbody>
<tr>
<td>Kandlikar et al. 2001</td>
<td>Water</td>
<td>( G = 28-155; ) ( q'' = 74.3-133; ) ( T_{in} = 24.2-24.7 )</td>
<td>Square, ( 1 \times 1 ), 6 parallel channels, horizontal</td>
<td>Slug growth occurs in the direction counter to bulk flow, forcing liquid and vapor to flow back into inlet manifold. Reverse flow in small diameter parallel multichannels evaporator is observed for the first time.</td>
</tr>
<tr>
<td>Li et al. 2003</td>
<td>Water</td>
<td>( G = 92.6-117; ) ( q'' = 108-208; ) ( T_{in} = 70-100; ) ( P_{in} = 121-180 )</td>
<td>Triangular, ( 0.100 \times 0.0706 ), 2 parallel channels, horizontal</td>
<td>Flow instabilities is found to cause significant oscillations to the inlet and outlet temperatures, inlet pressure and the pressure drop in the microchannels.</td>
</tr>
<tr>
<td>Brutin et al. 2003</td>
<td>n-Pentane</td>
<td>( G = 240-28800; ) ( q'' = 33-700 )</td>
<td>Rectangular, ( 0.5 \times 4 ), single channel with two lengths of 50 and 200, vertical upward flow</td>
<td>Two-phase flow oscillation phenomena were visually detected. Reverse flow extends into entrance.</td>
</tr>
<tr>
<td>Peles Y. 2003</td>
<td>Water</td>
<td>1-10 g/min; ( q'' = 100-600 )</td>
<td>Triangular, hydraulic diameter (mm) = 0.05-0.2, 16 mm long multiple parallel channels, horizontal</td>
<td>Rapid bubble flow regime is the most commonly observed. Important to address and study flow instabilities in microchannels.</td>
</tr>
<tr>
<td>Hetsroni et al. 2003</td>
<td>Water</td>
<td>( Re = 8-42; ) ( q'' = 100-600 )</td>
<td>Rectangular, hydraulic diameter (mm) = 0.103-0.161, 19-26 parallel channels, horizontal</td>
<td>Flow reversal caused by expanding bubbles pushing the liquid-vapor interface in both upstream and downstream directions.</td>
</tr>
<tr>
<td>Steinke and Kandlikar 2004</td>
<td>Water</td>
<td>( G = 157-1782; ) ( q'' = 5-898 )</td>
<td>Rectangular, hydraulic diameter (mm) = 0.207, 6 parallel channels, horizontal</td>
<td>Reported reverse flow under certain conditions in microchannels.</td>
</tr>
<tr>
<td>Balasubramanian and Kandlikar 2005</td>
<td>Water</td>
<td>( G = 112-120; ) ( q'' = 208-316 )</td>
<td>Rectangular, ( 0.990 \times 0.207 ), 6 parallel channels, horizontal</td>
<td>Severe flow maldistribution caused by back flow extended into the inlet manifold.</td>
</tr>
</tbody>
</table>

**Test Section Design**

Microchannels are fabricated on a copper block. The copper is an Electrolytic Tough Pitch alloy number C11000 which is 99.9 percent copper and 0.04 percent oxygen (by weight). It has a thermal conductivity of 388 W/m·K at 20°C. An optically clear polycarbonate known as Lexan is then placed on top of the copper block to serve as a transparent cover through which flow patterns can be observed. The Lexan cover has a thermal conductivity of 0.19 W/m·K. Figure 2 shows a schematic of the copper block with the Lexan cover. The resistive cartridge heater provides a uniform heat flux to the copper block. The length and width of the copper block are 88.9 mm × 29.6 mm. The thickness of the copper block and the Lexan cover are 19.1 mm and 12.3 mm respectively. The cross section of each of the microchannels measures 1.054 mm × 0.197 mm, and their edge to edge spacing is 0.589 mm. The length of each microchannel is 63.5 mm. There are a total of six microchannels on the copper block. The Lexan cover is being held onto the copper block by ten mounting screws, and the force provided by the screws is enough to seal the microchannels from the ambient environment. The assembly of the test section is shown in Fig. 3. The mounting screws are secured onto a phenolic layer that is placed on the bottom of the copper block. The phenolic plate also acts as an insulating layer on the bottom surface of the copper block. It has the same length and width as the copper block, but its thickness is 12.7 mm. It is a laminate of paper with a thermal conductivity of 0.2 W/m·K. The copper block is cleaned in an ultrasonic bath using water before it is assembled with the Lexan cover and the phenolic plate.

Two layers of six thermocouples each are placed into the sides of the copper block along the length of the
microchannels. The thermocouple layers are 3.18 mm apart from each other, and the top layer is placed at 3.18 mm below the top surface of the microchannels. The thermocouples are inserted into the copper block until it reaches half the width of the copper block. The thermocouple layers are inserted into the copper block from the opposite directions. The thermocouples from both layers are placed at the same locations along the length of the microchannels. The locations, as measured from the inlet of the microchannels and along its length are 6.35 mm, 19.05 mm, 25.40 mm, 38.10 mm, 44.45 mm, and 57.15 mm [11].

To reduce heat transfer in the manifold, the inlet manifold is machined into the Lexan cover so that the working fluid is delivered at the very beginning of the microchannels. When using flow restrictors, each microchannel has a dedicated inlet machined into the Lexan consisting of a 0.127 mm diameter hole. In the case without flow restrictors, the holes are replaced by a slotted opening of the width with the inlet manifold.

Figure 4 shows how each inlet connects to the inlet manifold on the Lexan cover. Water enters through the inlet manifold and is diverted through each restriction hole and into the corresponding microchannel. This 0.127 mm diameter hole has 6.1% of the cross-sectional area of a single microchannel and is 0.7 mm long. It acts as the physical pressure drop element (PDE) that is being studied. Only the pressure drop across the inlet and outlet manifolds is measured because the actual pressure drop in the microchannels could not be easily measured.

EXPERIMENTAL PROCEDURE

The experimental procedure for obtaining degassed water is the same degassing procedure as described by Kandlikar et al. [1], and Steinke and Kandlikar [11]. A heat exchanger in conjunction with a coolant bath (see Fig. 1) maintains the temperature of the degassed water delivered to the test section at a set value. The water then passes through a flow meter before entering the test section via the inlet manifold.

Heat loss calibration is performed on the test section after it is well insulated. The calibration is performed without working fluid in the test section. Using the resistive cartridge heater, electric power is supplied to the test section. The test
section is then allowed eight hours to reach steady state. A heat loss calibration chart is constructed by plotting the temperature difference between the microchannel surface and the ambient air ($T_s - T_{Amb}$) versus the corresponding steady state electrical power input, $q_{in}$. Heat losses, $q_{loss}$, were found to be a linear function of the temperature difference between the microchannel surface and the ambient air and generally ranged between 3 to 4 watts for ($T_s - T_{Amb}$) of 40 °C to 50 °C respectively. During the actual experiments, this chart is used to calculate the actual heat carried away by the microchannel array.

Water flow rate and inlet temperature are set and the electrical power is applied to the microchannels. Steady state is achieved when the surface temperature of the microchannels remains constant over a fifteen minute time interval. The flow meter is calibrated and is used to set the flow for the test section. LabView software is used as the data acquisition system and is used to monitor temperatures of all of the thermocouples and pressure transducers.

All of the image sequences are recorded with a high-speed digital camera system once the test section has reached steady state. The camera frame rate is set to 2000 frames per second to capture the details of the rapid two-phase flow interactions and events occurring within each microchannel. Sequences of individual frames are selected to illustrate the boiling characteristics and behavior at the set flow rate and heat flux conditions.

**UNCERTAINTY**

The uncertainty of the experimental data is calculated. The uncertainty in the hydraulic diameter is estimated to be ±1.4%. The accuracies of the digital signals are reported as: Voltage = ±0.05 V, $I$ = ±0.005 Amp, $T$ = ±0.1 °C, $\Delta P$ = ±0.1 kPa. The flow meter has a volumetric flow accuracy of ± 0.0588 cc/min. Heat loss measurements were conducted and a plot of heat loss versus copper block temperature was plotted. The actual heat supplied to the fluid was then calculated by subtracting the heat loss obtained from the experimental heat loss plot. The uncertainty in the heat supplied is estimated to be less than 1%. The thermal uniformity of the test section temperature distribution was verified using temperature measurements and numerical simulation as described in detail in a previous publication by Steinke and Kandlikar [9]. The pressure drop was measured with a pressure transducer with a 1 kHz frequency.

**RESULTS**

The effect of pressure drop elements (PDEs) on flow boiling heat transfer is presented in this section. All tests are conducted with microchannels in the horizontal orientation.

The results from the case without PDEs are compared to those with 6.1% PDEs at the inlet of each channel. The latter case uses manifold which incorporates inlet openings of 127 microns diameter at the inlet to each channel, giving an open area that is 6.1% of the area of a 1054 x 197 microns microchannel. These pressure restrictors are expected to reduce the backflow by forcing an expanding vapor bubble in the downstream direction and not allowing the liquid-vapor mixture to enter the inlet manifold. One example of flow reversal is depicted in Fig. 5. The sequence of frames in Fig. 5 shows an expanding vapor bubble nucleating and moving backward as well and eventually reaching the inlet manifold. Frame (a) shows a vapor bubble nucleating from a nucleation site near the lower corner of the channel. Frame (b) shows the bubble developing into a plug as it begins to push water upstream and downstream. The smaller bubbles seen in the flow upstream also move in the reverse direction rather than slipping around the vapor slug. Finally, Frame (f) shows the vapor reaching the inlet manifold and the channel drying out.

![Figure 5](image-url) Flow reversal without PDEs in manifold. Successive frames from (a) to (f) taken at 1.5 ms time interval illustrating unstable flow in a single channel from a set of six parallel horizontal microchannels. $G$=212.8 kg/m²s, $q''$=367.7 kW/m², $T_s$= 114 °C.

The results are reported for mass fluxes of 362.9 kg/m²s and 144.4 kg/m²s. Each of these mass fluxes includes results from the cases with the 6.1% PDEs and without PDEs. Using a constant mass flux of 362.9 kg/m²s, Figs. 6–9 show the surface temperatures of the microchannels for heat fluxes of 335.9 kW/m², 352.9 kW/m², 370.2 kW/m² and 388.2 kW/m². As stated in the experimental facility section, the surface temperatures shown in all the figures are measured and calculated at the locations where measured from the inlet of the microchannels are 6.35 mm, 19.05 mm, 25.40 mm, 38.10 mm, 44.45 mm, and 57.15 mm. As can be seen from the figures, the
heat transfer performance is improved by using the header with the 6.1% PDEs.

**Figure 6** Microchannel surface temperatures with and without the 6.1% PDEs in manifold. $G = 362.9\ \text{kg/m}^2\text{s}$, $q'' = 335.9\ \text{kW/m}^2$.

**Figure 7** Microchannel surface temperatures with and without the 6.1% PDEs in manifold. $G = 362.9\ \text{kg/m}^2\text{s}$, $q'' = 352.9\ \text{kW/m}^2$.

**Figure 8** Microchannel surface temperatures with and without the 6.1% PDEs in manifold. $G = 362.9\ \text{kg/m}^2\text{s}$, $q'' = 370.2\ \text{kW/m}^2$.

In the case using mass flux of 144.4 kg/m²s, the heat transfer performance is reduced when using the header with 6.1% PDEs. Figures 10–13 show the surface temperatures of the microchannels for heat fluxes of 151.9 kW/m², 171.5 kW/m², 188.0 kW/m² and 206.9 kW/m². The surface temperatures shown in the figures are measured and calculated at the same locations as in the previous case with a constant mass flux of 362.9 kg/m²s.

**Figure 9** Microchannel surface temperatures with and without the 6.1% PDEs in manifold. $G = 362.9\ \text{kg/m}^2\text{s}$, $q'' = 388.2\ \text{kW/m}^2$.

**Figure 10** Microchannel surface temperatures with and without the 6.1% PDEs in manifold. $G = 144.4\ \text{kg/m}^2\text{s}$, $q'' = 151.9\ \text{kW/m}^2$. 

The inlet pressures of various heat fluxes for both mass fluxes of 362.9 kg/m²s and 144.4 kg/m²s are shown in Fig. 14. Note that without PDEs, the inlet pressures for both the mass fluxes are very similar. In the case of 6.1% PDEs, the inlet pressure is higher when using mass flux of 362.9 kg/m²s than the case of 144.4 kg/m²s.

Using the mass flux of 362.9 kg/m²s together with 6.1% PDEs at the inlet of each channel has resulted in higher heat transfer performance, and the backflow is reduced by the higher inlet pressure of 190 kPa and above. The more stable flow has resulted in higher heat transfer performance. The 6.1% PDEs did not improve heat transfer performance for lower mass flowrates.

CONCLUSIONS

1. The present study focused on the effect of flow restrictors on the heat transfer performance during flow boiling in microchannels. When using flow restrictors, each microchannel has a dedicated inlet hole machined into the inlet manifold. The flow restrictors act as the physical pressure drop element (PDE) that is being studied. The results from the case without PDEs are compared to those with 6.1% PDEs.
2. The heat transfer result for mass fluxes of 362.9 kg/m²s and 144.4 kg/m²s were reported.
3. In the case of 362.9 kg/m²s, the heat transfer performance is improved by using the header with the 6.1% PDEs.
4. In the case of 144.4 kg/m²s, the heat transfer performance is reduced when using the header with 6.1% PDEs.
5. In the case of 6.1% PDEs, the inlet pressure is higher when using mass flux of 362.9 kg/m²s than the case of 144.4 kg/m²s. Using the mass flux of 362.9 kg/m²s together with 6.1% PDEs at the inlet of each channel has resulted in higher inlet pressure, and the backflow is reduced by the higher inlet pressure of 190 kPa and above. The more stable flow has resulted in higher heat transfer performance.
6. The 6.1% PDEs did not improve heat transfer performance for lower mass flowrates.
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