FLOW BOILING AND PRESSURE DROP IN PARALLEL FLOW MICROCHANNELS

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ABSTRACT

The use of microchannels for advanced heat transfer applications has quickly become commonplace. They are found in automotive applications, fuel cells, and even electronics cooling. However, there are fundamental issues still unresolved with heat transfer and fluid mechanics and the application of microchannels. Researchers have reported microchannel data using very different hydraulic diameters, sometimes as much as 2 orders of magnitude.

An experimental investigation of the heat transfer, pressure drop, and flow boiling in microchannels is performed. A new channel size classification has been developed based upon the manufacturing techniques as well as the underlying fluid mechanics and heat transfer theory.

Six parallel channels with a hydraulic diameter of 207 micrometers is manufactured and tested. Flow boiling patterns have been observed in the channels. Observations suggest that the conventional flow boiling patterns also occur in microchannels. This suggests that there is no difference in the theory used for conventional channels. Therefore, a microchannel can be modeled in the conventional manner.

Heat fluxes of up to 930 kW/m² have been maintained in the microchannel. The local heat transfer coefficient and quality has been measured. The largest heat transfer coefficient achieved is 192 kW/m²K. In addition, the highest quality achieved is 1.0. Dry-out was also observed during experimentation.

NOMENCLATURE

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INTRODUCTION

The increasing need to dissipate high heat fluxes has once again gained importance in an application such as electronics cooling and MEMS devices. One of the first pioneering works demonstrating the potential of small passages for heat transfer enhancement was performed by Bergles [1]. The first applications of microchannels mainly involved aerospace systems. The microchannels came in the form of compact heat exchangers for managing onboard power systems.

Tuckerman and Pease [2] demonstrated the potential of microchannels in the application of integrated circuitry. Recently, the microchannel heat exchangers are being applied...
to power systems, fuel cells, advanced heat sinks, and several automotive applications. The benefit of reduced channel size and the resulting heat transfer enhancement has been well established.

A wide variety of channel sizes have been investigated in literature. A clear definition of channel size classification is needed for consistency in comparing various results from literature. There is no detailed local flow boiling data available in open literature for a microchannel, which is defined as channels with hydraulic diameter below 200 μm. The present work characterizes the thermal and hydraulic performance of six parallel microchannels with a hydraulic diameter of 207 μm. The slightly larger hydraulic diameter (compared to 200 μm for a microchannel) resulted due to the tolerances associated with manufacturing of these channels. Both single and two-phase flows are investigated.

Two issues that need to be addressed are (i) whether the single-phase and two-phase flow behavior is affected for microchannels due to small passage dimensions, and (ii) what is the effect of dissolved gases on the heat transfer and pressure drop characteristics during flow boiling in microchannels. The effect of dissolved gases was presented in an earlier paper, and the present paper addresses heat transfer and pressure drop characteristics of pure degassed water in microchannels.

LITERATURE REVIEW

As smaller channel sizes were investigated, the terminology for classification has undergone considerable changes in the past decade. In earlier studies, the term microchannel has included hydraulic diameters as large as 3.0 mm. Wambsganss et al. [3] investigated a tube with a hydraulic diameter of 2.92 mm using R113. Ravigururajan et al. [4] used 54 parallel channels that were 1.0 mm deep by 0.27 mm wide. The resulting hydraulic diameter was 0.425 mm. Warrier et al. [5] investigated channels with a hydraulic diameter of 0.750 mm using FC-84.

Several authors have specifically addressed the issue of channel size classification. Kandlikar [6] performed an extensive review of literature and presented it on flow boiling in microchannels. Kandlikar [7] also described many of the fundamental issues with two-phase flow patterns. In a more recent version that includes single phase as well as two-phase flows, Kandlikar and Grande [8] presented a new classification scheme based upon fundamental considerations of fluid flow and channel fabrication. The upper limit for the microchannel classification, 200 μm, is determined based upon conventional fabrication techniques. In addition, the limit for microchannels also considers the continuum, slip flow, and free molecular flow of gases using the corresponding Knudsen number limits. The range between 200 μm and 3 mm is classified as minichannels.

In order to develop accurate predictive techniques for flow boiling in microchannels, a reliable set of local data is needed. The effect of dissolved gases upon the heat transfer has been a recent concern raised by researchers. It has been demonstrated by a number of researchers that the presence of dissolved gases will provide an enhancement in the heat transfer coefficient at low heat fluxes in the low quality range. Kandlikar et al [9] have developed a technique to inexpensively control the concentration of dissolved air in water. They have also demonstrated that once an oxygen concentration of 5.4 ppm is reached the effect of dissolved gases essentially disappears. Therefore, this technique will be employed while generating local data to eliminate any artificial enhancement due to dissolved gases.

The local heat transfer studies available in literature for minichannels are summarized in Figure 1. It shows that the majority of the available sets are centered around a hydraulic diameter of 2 mm. It is also seen that the low Reynolds number range generally encountered in microchannel application is not covered by the earlier studies. In addition, there are no available data sets for any microchannel flows. There are a few more data sets available in literature, e.g., Mertz, et al. [15], and Kattan et al. [16] for minichannels. However, the detailed data required in the form of x, h, q and G at each point could not be readily available from these papers. Nevertheless, the trends represented in these papers are in line with those observed in the data sets shown in Figure 1 for minichannels. The present work focuses on the heat transfer, pressure drop and flow patterns in microchannels as described in the objectives below.

OBJECTIVES OF THE PRESENT WORK

The objectives of the present study are to investigate the heat transfer and pressure drop performance of water heated in microchannels, and to generate local heat transfer data by varying mass flux, heat flux, and quality along the length of a set of six parallel microchannels. A visualization test section has been constructed to observe the flow patterns occurring in microchannel flow boiling. A long distance microscopic vision system in conjunction with a high-speed camera has been utilized to aid in the detection of the nucleation. A four-side heating test section built for data collection has also been also be utilized. The present work will focused on the upper limit of the microchannel classification. The Reynolds numbers are in the laminar range from 116 to 1318.

EXPERIMENTAL SETUP

The experimental system consists of several sub-systems that include a water loop, data acquisition system, high-speed camera system, and the test sections. A schematic of the system is presented in Figure 2. The test section is located within the water loop. The data acquisition system is not shown and is used to instrument the water loop as well as the test sections.

A high-speed digital CCD camera system is used for the visualization study. The camera is a model number PCI 8000s Olympus CCD camera capable of a maximum frame rate of 8,000 frames per second. Typically, the images are acquired at a frame rate of 1,000 frames per second.

The test section for visualization is a combination of three layers. The top layer is made of Lexan, an optically clear polycarbonate material. The water inlet and outlet plenums are machined into this polycarbonate layer. This is done to eliminate heat transfer in the inlet and outlet manifolds. The second layer is a copper block that contains the microchannels. The copper is an electrolytic tough pitch alloy number C11000. It is comprised of 99.9% copper and 0.04% oxygen (by weight). The thermal conductivity is 388 W/m-K at 20°C. The third piece is a phenolic. It is a laminate of epoxy and paper. It has a very low thermal conductivity and acts as an insulator on the lower surface of the copper plate. It is used to secure the microchannel test section with the help of ten mounting screws. A cartridge heater is used to provide up to 220 Watts of constant input power. The cartridge heater is located 12.5 mm under the microchannel walls.

There are six parallel microchannels machined into the copper substrate. In the visualization test section configuration, the channels are heated from only three sides. The Lexan cover is treated as an adiabatic wall. It is difficult to compare the Nusselt number from this test section to any published value. Therefore, a second heat transfer test section is constructed to have four-sided heating and is used to obtain local data.

Using a microscopic vision system, the channel depth and width are measured at six distinct locations along the channel. These measurements are used to determine the average dimensions of the channels. It is observed that the channels have a slightly trapezoidal cross section, with the top and bottom widths differing by about 15 µm. The average channel dimensions are: 214 µm wide by 200 µm deep and 57.15 mm long. All measured values fall within ±3% of the respective average values. The channels are machined to have an inlet and outlet radius of 2.68 mm, due to the diameter of the milling cutter, with a length of 1.015 mm. The channels have an overall length of 61.468 mm. The first and last channel is located 12.670 mm from each side of the test piece. The channel starts and ends 12.7 mm from each end of the test piece. The channel spacing is 570 µm.

Two thermocouple layers are located in the copper substrate. The A layer is closest to the microchannel surface. The B layer is closest to the cartridge heater. The A layer is 3.175 mm from the microchannel wall. The B layer is 6.35 mm from the microchannel wall. There are 6 thermocouples in each level. The first thermocouple is located 6.35 mm, in the flow direction, from the microchannel inlet. The second, third, fourth, fifth, and sixth thermocouples are located 19, 25.4, 38.1, 44.45, and 57.15 mm from the inlet, respectively. The B layer thermocouples are located at the same distance along the flow length as in the A layer.

The pressure drop for the microchannel was measured between the inlet and exit plenum locations within the Lexan layer. The pressure drop was used to calculate the friction factor for the microchannels, after accounting for the entrance region, area changes, bends, and exit losses. A further discussion will be presented in the Data Analysis Section.

The heat transfer test section is constructed similarly to the visualization test section, but provides heating from all four sides of the channel. Figure 3 shows the heat transfer test
section. It is constructed using four layers. The top layer is a Phenolic resin layer. It acts as an insulating layer to the copper and also supports the test section. The next layer is a top copper layer. This layer acts as a cover for the microchannels. It replaces the Lexan in the visualization test section. The inlet and exit plenums and the pressure taps are located in this layer. A cartridge heater is also located in this layer. The next layer is the same copper layer as in the visualization test section. Only one channel is shown for simplicity. It contains the machined microchannels. The final layer is the same insulating Phenolic layer. A set of securing bolts holds all of the layers together.

Figure 3: Heat Transfer Test Section. Test section constructed for data collection consists of: (1) Mounting Phenolic, (2) Top Copper Cover, (3) Top Cartridge Heater, (4) Copper Substrate only one channel shown, (5) Bottom Cartridge Heater, and (6) Insulating Phenolic. Only one channel shown for clarity.

The copper block that contains the microchannels is the same block that was used in the visualization test section. Therefore, the microchannel dimensions and locations are the same. The thermocouple locations are identical to those in the visualization test section.

EXPERIMENTAL PROCEDURE

The working fluid for all experiments is 8.0 MΩ de-ionized water. Three distinct experimental procedures were used. The first procedure involves the acquisition of images using the vision system. The second procedure describes the method to vary the dissolved gas concentration in the water. The third procedure describes the method to acquire the heat transfer and pressure drop data.

The images of flow visualization are acquired after the system has reached steady state. The camera is moved into position over the microchannels. The long-distance microscopic lens is used to get a view of all six channels. The lens could be scanned to detect specific events at specific locations. The flow visualization was used for two different purposes.

First, the vision system was used to determine flow-boiling patterns in the microchannels. For each experimental run, the flow rate was held constant. The input heat flux was varied to gather a complete set over the entire range of heat fluxes at the specified flow rate. Then, the images were captured at three locations along the microchannel; inlet, middle, and exit. The images can be reviewed to determine the present flow patterns. Secondly, the vision system is used to determine the onset of nucleate boiling (ONB). The detection of bubbles beginning to develop and grow is important to determine the effect of dissolved gases. The surface temperature at which nucleation begins for a specific concentration of oxygen is observed using the images from the vision system.

The single cartridge heater is used to deliver the power to the test section. The flow rate is held constant and the power input is varied. The test section remains at steady state for a minimum of ten minutes and then the data is collected. It was established that the test section reading must remain constant for at least 10 to 15 minutes for ascertaining steady state conditions.

The process fluid in this experiment is de-ionized water. The water must be degassed to provide fundamental flow boiling data. A pressure chamber is used to achieve different levels of degassing and therefore different levels of oxygen concentration. The degassing pressure chamber is a commercially available pressure cooker with a five-gallon capacity. Different dead weights are used to obtain different levels of pressure in the chamber. The pressure built up in the pressure cooker is used to drive the flow. A laboratory hot plate is used to provide the heating.

The pressure cooker is filled with the filtered de-ionized water. The appropriate dead weight is added to achieve the desired pressure. For oxygen concentration used in the experiments, the dead weight used is corresponding to 1 atm. The heat is turned on. The pressure cooker is allowed to reach the pressure of 2 atm. Then, the water has a saturation temperature of 121 °C. The dead weight is now removed causing vigorous boiling, which removes excess dissolved air from the water. The dead weight is then reapplied. The pressure returns to 2 atm. The saturation temperature returns to 121 °C. Therefore, no air will precipitate out of the solution until the water encounters a surface temperature higher than 121 °C. The oxygen concentration is measured to be 3.2 ppm. Kandlikar et al. carried out an investigation of the effect of dissolved gas [9]. They concluded that once a degassed level of 5.4 ppm oxygen was achieved, the effect of dissolved gasses is eliminated.

A heat loss experiment was conducted to determine the amount of losses to the surrounding environment. The details are described in the data analysis section. Two cartridge heaters are used to deliver the power. The flow rate is held constant and the power input is varied. The bottom cartridge heater is set to a specified power. The top cartridge heater is adjusted to give the same surface temperatures as in the bottom layer. The test section must remain at steady state for a minimum of twenty minutes and then the data is collected.
DATA ANALYSIS

The analysis of the flow visualization is fairly straightforward. At each test condition, three video images were captured. One video is near the microchannel inlet. The second video is in the middle of the microchannel. The third video is near the exit of the microchannel. The images collected are compared to the accepted flow boiling flow regimes of conventional and minichannels. The possible flow regimes include: nucleation, bubbly flow, slug, annular, annular flow with nucleation, dry-out, and dry-out condition.

Data Collection. The concentration of dissolved oxygen is measured using a dissolved oxygen meter with an accuracy of ± 0.1 ppm. A sample is collected just after the flow meter bank. The sample is collected in a test tube. Care is taken not to expose the sample to air for more than a few minutes. Five readings for each dissolved gas case are taken to determine the oxygen concentration. In addition, the oxygen concentration is checked periodically throughout the experiments.

There are several heat transfer surface areas used in the calculations. It is beneficial to define these areas. The first area is the cross sectional area, \( A_c \), of each microchannel, calculated to be \( 0.0428 \text{ mm}^2 \). The next area is the heat transfer area, \( A_{HT} \). It is the area available for heat transfer. The resulting total areas for the three side and four side heating are 205.74 mm\(^2\) and 274.32 mm\(^2\), respectively.

Another area used is the effective area, \( A_{eff} \). It is the footprint area of the microchannels. The effective area is just the total length of the microchannels multiplied by the total width of the microchannels and the spaces between them. The effective area is 273 mm\(^2\). The volumetric flow rate is measured and held constant for each set of data collected. The volumetric flow rates range from 2.52 cc/min to 16.43 cc/min.

The thermocouples in the microchannel copper blocks are used to determine the wall temperatures. The type of boundary condition that occurs inside the microchannels is in question. There are two types of boundary conditions traditionally used in heat transfer. They are the constant heat flux and the constant temperature at the heating wall. The two layers of thermocouples placed in the copper would allow for the calculation of local heat flux, surface temperature, and eventually local heat transfer coefficient. The bottom layer seemed to have a Gaussian profile with the lowest temperature at the inlet and outlet and the highest temperature in the center. The first layer, closest to the wall, seemed to have a straight-line profile with all temperatures being within a degree or so.

Eight nodes of temperature along the flow length are created to align with positions of interest. There are eight specified nodes that go from the inlet to the outlet of the microchannel. A linear curve fit is assumed to get the projected nodes 0 and 7 for the copper. The inlet and outlet nodes 0 and 7 for the water side are measured. The change in temperature between the water and the surface temperatures at each node is determined from the inlet and outlet temperature values.

The heat flux for the test section, \( q_w \), will now be discussed. The input power is varied for each constant volumetric flow rate. The input power, \( q_{in} \), is measured. For the visualization test section the input power is equal to the power of the single cartridge heater. For the heat transfer test section, the input power is equal to the addition of the power from both cartridge heaters.

The test section is well insulated using high temperature insulation. The insulation has baked fibers and can withstand 1260 °C. It is a ceramic mat model RT2300. A heat loss analysis is performed to allow for the energy balanced required to determine the heat transfer coefficient and exit quality. The test section is allowed to reach a steady state temperature. The resulting temperature due to the input power is determined. The ambient temperature is subtracted from the test section temperature to give a differential temperature. The slope of the curve can be used to determine the lost power for a given test section operating temperature. In order to calculate the average heat transfer coefficient, the temperature difference must be known. A log mean temperature difference between the inlet and outlet is used to determine the average heat transfer coefficients.

Theoretical Nusselt Number. The experimental Nusselt number is then determined. A comparison of the Nusselt number for laminar flow is preferred. The Nusselt number comparisons are given for constant surface temperature in Equation (1) and for constant wall heat flux, with circumferentially constant wall temperature and axially constant wall heat flux in Equation (2), Kakac et al [17].

\[
Nu_T = 7.54 \left[ 1 - 2.610\alpha^2 + 4.970\alpha^2 - 5.119\alpha^3 \right] \left( \frac{1}{2} + 2.702\alpha^4 - 0.548\alpha^5 \right) \tag{1}
\]

\[
Nu_H = 8.235 \left( 1 - 2.0421\alpha^3 + 3.0853\alpha^2 - 2.4765\alpha^3 \right) \left( +1.0578\alpha^4 - 0.1861\alpha^5 \right) \tag{2}
\]

where, \( Nu_T \) is the constant surface temperature case, \( Nu_H \) is the constant heat flux case, and \( \alpha \) is the aspect ratio. Based upon the geometry, the value for the constant surface temperature is 2.969 and for constant heat flux is 3.616. These theoretical values of \( Nu \) will be used for comparing the experimental data after applying the property correction factor as described below.

The Nusselt number also has a dependence upon the varying properties due to the temperature gradients. A viscosity correction is used to compensate for this variation. Equation (3) gives the viscosity correction formula.

\[
\frac{Nu}{Nu_{cp}} = \left( \frac{\mu_{w}}{\mu_{w}} \right)^n \tag{3}
\]

where, \( Nu_{cp} \) is the corrected Nusselt number, \( \mu_{w} \) is the viscosity of the fluid at the wall temperature, \( \mu_{w} \) is the viscosity of the fluid at the mean temperature, and \( n \) is equal to -0.14.
**Experimental Heat Transfer Coefficient.** For the single-phase points, the average heat transfer coefficient was determined using the $\Delta T_{\text{LMTD}}$ with the outlet and the inlet. The average heat transfer coefficients can now be compared for the different mass fluxes.

The two-phase data points require more detailed analysis. The average heat transfer coefficient will not be calculated for the two-phase flow. In the two-phase flow, the flow begins as sub-cooled single phase and changes into two-phase flow. A few additional items need to be addressed in dealing with the data analysis of the two-phase flow.

The inlet subcooling must be accounted for. The water coming into the channels is at 22 °C. A certain amount of the power inputted must go towards sensible heating to raise the water up to saturation temperature. The saturation temperature at the local location will vary throughout the channel. The high pressure drops that occur in two-phase flow perpetuate this problem. A linear variation of pressure in the channel is assumed. The pressure variation was determined from the inlet and outlet pressures measurements.

**Experimental Pressure Drop.** The measured pressure drop is corrected for entrance and exit losses to determine the pressure drop in the microchannels only. The pressures are measured in the inlet and outlet plenums. There are losses due to the contraction from to the changing of area from the inlet plenum into the microchannels, the microchannel length, and the expansion from the change in area of the microchannels to the exit plenum.

The entrance contraction loss can be found knowing the area ratio, $AR$. The plenum area, $A_1$, is 23.39 mm$^2$. The total microchannel inlet area, $A_2$, is 0.24 mm$^2$. The AR for the contraction is 0.010. The minor loss coefficient, $K_e$, is 0.48 from Fox and McDonald [18]. The exit expansion is found in the same manner. The microchannel exit area, $A_e$, is 0.24 mm$^2$. The exit plenum area, $A_3$, is 23.39 mm$^2$. The AR for the expansion is 0.010. The minor loss coefficient, $K_{es}$ is 0.8.

The pressure drop due to the channel alone is the measured pressure drop minus the pressure drops due to the expansion and the contraction. The pressure drop due to the microchannel is used to determine the friction factor. The Fanning friction factor is widely accepted for use with heat transfer and will be reported here. The correlation for a square channel, along with other geometries, can be found in Kakac et al. [17] and is given in equation (4).

$$f \frac{Re}{Re_c} = 24 \left(1 - 1.3553\alpha^4 + 1.9467\alpha^2 - 1.7012\alpha + 0.9564\alpha^{-4} - 0.2537\alpha^{-6}\right)$$  \hspace{1cm} (4)

The aspect ratio for the microchannels is 0.935. For the microchannel geometry, the $fRe$ number is 14.25. Therefore, the predicted friction factor is found using Equation (5).

$$f = \frac{14.25}{Re}$$  \hspace{1cm} (5)

The friction factor is inversely dependent upon the Reynolds number. In turn, it is dependent upon the density and the viscosity. The friction factor is sensitive to changes in density and viscosity with a change in temperature. The property changes that occur near the wall for diabatic flows can result in fluctuations in the friction factor. One commonly used method for laminar flows is to correct for the viscosity dependent properties. Equation (6) is the equation for the viscosity correction.

$$\frac{f}{f_{cp}} = \left(\frac{\mu_w}{\mu_m}\right)^m$$  \hspace{1cm} (6)

where, $f_{cp}$ is the corrected property friction factor, $\mu_w$ is the viscosity of the fluid at the wall temperature, $\mu_m$ is the viscosity of the fluid at the mean temperature, and $m$ is equal to 0.58. This equation corrects for the variation in the viscosity from the temperature gradient from the wall to the bulk fluid temperature.

**UNCERTAINTY**

The uncertainty, $U$, for the experimental results will be discussed in this section. $U$ is determined by evaluating $B$ and $P$; where, $B$ is the bias error and $P$ is the precision error. The bias error is the error that causes the values to be different from the true value. The precision error is the portion of error that is generated by random values.

The accuracy of the measurement instruments must be known first. The pressure transducer has an accuracy of ±0.69 kPa. The temperature reading has an accuracy of ±0.1 °C. The power supply used to provide input power has accuracies for the voltage of ±0.05 V and for the current ±0.005 amps. The flow meter has a volumetric flow accuracy of ±0.0588 cc per minute. Finally, the dissolved oxygen meter has a reading accuracy of ±0.05 ppm. The resulting uncertainties are calculated as: heat transfer coefficient is 8.61%, friction factor is 7.19%, and friction factor is 4.80%.

**RESULTS**

**High-Speed Visualization.** The vision system is used to collect videos of the flow boiling in the 207 μm hydraulic diameter channels. The following flow patterns are observed in the microchannels; nucleate boiling, bubbly flow, slug flow, annular flow, annular flow with nucleation in the thin film, churn, and dry-out. Each flow regime will be shown and described in this section. Changes in image capture rate, ambient light, and flow regime require adjustments to exposure time and lighting that affect the appearance of the channel surroundings. The flow direction for all of the following figures is from left to right.

The first flow regime detected is nucleate boiling. Figure 4 shows nucleate boiling in a single channel. The successive frames are 1 ms apart. A bubble is formed in the channel in
The next flow regime discovered is bubbly flow in Figure 5. The successive frames are 4 ms apart and only four of the six channels are shown. Several different sized bubbles are seen moving through multiple channels. The bubbles range in size from 15 µm to 193 µm. The average liquid velocity is determined from the mass flux and the flow area and is found to be 0.160 m/s, at the inlet. The velocity of the bubbles is measured to be in the range of 0.184 m/s to 1.289 m/s and varies with size. The smallest and largest bubbles move the slowest, around 0.184 m/s. The smallest bubbles, diameter of 15 to 60 µm, are just reaching or not much larger than the departure diameter and stay very close to the wall. The wall still influences their behavior. The largest bubbles, diameter of 125 to 200 µm, are large enough to have a large cross sectional void fraction and occupy most of the channel. Therefore, they are largely influenced by the friction and drag of the wall. The bubbles of medium size, diameter of 60 to 200 µm, have the largest velocities, around 1.28 m/s. The bubble has sufficient vapor and has moved into the middle of the flow channel. Therefore, it has little restriction and wall interference.

Figure 4: Nucleation in flow. Flow is from left to right, single channel shown, Δt = 1 ms. Bubble diameter: (a) 67 µm and (e) 160 µm. For: \( G = 115 \text{ kg/m}^2\text{s}, q_{in}'' = 41.7 \text{ kW/m}^2, x = 0.002 \).

Figure 5: Bubbly Flow. Flow is from left to right, four channels shown, Δt = 4 ms. Bubble diameters: 15 µm to 193 µm. Bubble velocities: 0.184 m/s to 1.2889 m/s. For: \( G = 115 \text{ kg/m}^2\text{s}, V_f = 0.160 \text{ m/s}, q_{in}'' = 41.7 \text{ kW/m}^2, x = 0.002 \).

Figure 6: Counter flow. Flow is from left to right, single channel shown, Δt = 8 ms. Counter flow interface velocity: 0.197 m/s. For: \( G = 467 \text{ kg/m}^2\text{s}, V_f = 0.480 \text{ m/s}, q_{in}'' = 140.1 \text{ kW/m}^2, x = 0.001 \).

One flow boiling phenomenon found in parallel flow channels is counter flow. It was first discovered by Kandlikar et al. [19] in minichannels. Counter flow means that the flow of vapor moves in the opposite or counter to the flow direction. In Figure 6a, a bubble is formed as in Figure 6b. Figure 6b shows the slug expanding toward the channel exit like a normal slug. The right side liquid vapor interface of the slug begins to
move toward the channel entrance, counter to flow. Finally, the right side interface begins to move toward the exit in Figure 6g. The right side interface has a velocity of 0.197 m/s. The explanation of this occurrence lies within the communication between the parallel channels. The flow and pressure in the other channels compensates and allows for the high pressure of slug generation to dissipate through the other channels.

The most surprising flow regime observed in the microchannels is churn flow, Figure 7. The successive frames are 2 ms apart. In Figure 7a, an annular flow is established. At the right hand side, some bubbles are formed in the thin film. Figure 7b shows a wavy interface of vapor and liquid. The liquid is on the top side of the channel and the vapor is on the bottom side. The wavy liquid vapor interface is observed along almost the whole length of the shown channel section. Figure 7c shows that the liquid-vapor interface has moved downstream and the channel is refilled with liquid.

Another flow boiling condition is the dry-out condition. Figure 8 shows the captured dry-out event. Once again, flow is from left to right and the figure caption contains the flow details. Figure 8a shows the channel in a dry out condition. The local surface temperatures are rising, as there is no film on the surface. In Figure 8b, an annular slug comes into view. The annular slug has a head of liquid as a front cap. An advancing contact angle is seen in the frame. The slug and cap move forward a little distance. In Figure 8c, the contact angle changes and shifts from an advancing to a receding contact angle. This signifies the onset of the dry-out condition because the interface begins to move backward toward the inlet, in Figure 8g. The interface is now wavy because the light is reflecting differently in Figure 8g through 8j. In Figure 8k, the upstream vapor has joined with the vapor in the annulus. Figure 7m and 7n show the thin film interface retreating toward the inlet as the channel begins to dry-out. Finally, Figure 8p shows complete dry-out in the channel.

Figure 7: Churn Flow. Flow is from left to right, single channel shown, $\Delta t$: (a) 0 ms, (b) 2 ms, (c) 4 ms. For: $G = 467$ kg/m$^2$s, $V_f = 0.480$ m/s, $q_{\text{in}}'' = 150.1$ kW/m$^2$, $x = 0.002$.

Figure 8: Dry-out. Flow is from left to right, single channel shown, $\Delta t = 4$ ms For: $G = 375$ kg/m$^2$s, $q_{\text{in}}'' = 632$ kW/m$^2$, $x = 0.631$.

Figure 9: Dry-out in Schematic Form. Representation of flow occurring in Figure 8. (a) channel dry-out, (b) annular slug with liquid head, (c) contact angle shift, (d) vapor penetration at dry-out occurrence, (e) thinning film as it returns to dry-out.

To aid in the discussion of the dry-out condition, another figure has been made to represent the key steps in what is occurring in Figure 8. Figure 9 is a schematic figure of the flow boiling dry-out. Figure 9a begins with a channel in the dry-out regime. An annular slug with liquid head is introduced into the channel in Figure 9b. The interline, the liquid-vapor-surface contact line, is in an advancing contact position. Dry-out begins to occur in Figure 9c and the interline shifts to a receding contact orientation. The dry-out is occurring because...
of the rapid evaporation of the liquid in the contact line region. This rapid movement causes a force imbalance that causes the reaction force and interface movement in the opposing direction, Kandlikar and Steinke [20, 21]. Finally, Figure 9e shows the channel returning to a post dry-out regime.

**Single- and Two-Phase Heat Transfer.** The single and two-phase flows are investigated for microchannels in laminar flow. The water has been degassed to a level of 3.2 ppm for all of these experiments. Based upon a previous dissolved gas investigation by Kandlikar et al. [9], this level is sufficient to remove any effect of dissolved gases from the heat transfer data. The result is a true set of data for pure water boiling in microchannels. The test section used for this set of experiments is the four side heated heat transfer test section.

Several mass fluxes have been studied, 157 kg/m$^2$s, 366 kg/m$^2$s, 671 kg/m$^2$s, 1022 kg/m$^2$s, and 1782 kg/m$^2$s. The resulting Reynolds numbers, calculated at the saturation temperature, are 116, 270, 496, 756, and 1318, respectively. The flows in this investigation are all considered to be laminar.

The Nusselt number was calculated for the single-phase flow. The Nusselt number should be constant for the single-phase flow. A shifting the Nusselt number upwards with higher heat fluxes is seen. There is a direct relationship between mass flux and heat transfer coefficient in the single-phase regime. The predicted Nusselt number is used to determine the predicted heat transfer coefficients. The systematic variation in Nusselt number is seen in the present work as well as other data sets available in literature. However, previous researchers have failed to explain this behavior.

In the present work, the hydraulic and thermal entrance lengths are considered to have an effect upon the heat transfer. The Nusselt number is a function of the inverse Graetz number, Gz$^{-1}$, in the developing region. The experimental value takes a mean value of that entire quantity instead of the actual fully developed Nusselt number. Therefore, a technique has been developed to account for the thermal entrance length effect upon the Nusselt number. The inverse Graetz number, Gz, is determined for the cases and used to determine the non-dimensionalized x location. The mean Nusselt number is determined from the work of Chandrupatla and Sastri. The values for their experiments are tabulated in Kakac et al. [17]. The entrance length and remaining lengths are determined to be $l_1$ and $l_2$, respectively. Equation 7 shows the formula to calculate the average Nusselt number in the microchannel.

$$\overline{Nu} = \frac{l_1 N_{um} + l_2 N_{ud}}{l_1 + l_2} \quad (7)$$

where, $N_{um}$ is the mean Nusselt number from the table, $N_{ud}$ is the fully developed Nusselt number from theory. Using this formula provides good agreement between the experimental and theoretical values. The experimental Nusselt number has been normalized with the average theoretical Nusselt number to see the resulting error. Figure 10 shows the normalized Nusselt number and the error is only about 25%. Figure 11 shows the experimental Nusselt number versus the average theoretical Nusselt number, with respect to constant surface temperature and constant heat flux boundary conditions.

![Figure 10: Normalized Nusselt Number vs. Heat Flux. Solid points represent constant surface temperature boundary condition. Hollow points represent constant heat flux boundary condition. For: G = 671 kg/m$^2$s and 1022 kg/m$^2$s; 5.02 kW/m$^2$ < $q''$ < 117.25 kW/m$^2$, x < 0.](image)

The two-phase flow results for the different mass fluxes will now be presented. The superheat is used to plot the heat flux.
for the different mass fluxes. The superheat is the temperature difference between the average surface temperature and the local saturation temperature. Figure 12 shows the resulting plot and the behavior is as expected. As the heat flux is increased, the superheat is increased.

![Figure 12: Heat Flux vs. Superheat. For: G = 157 kg/m²s, 366 kg/m²s, 671 kg/m²s, 1022 kg/m²s, and 1782 kg/m²s; 5.02 kW/m² < q” < 898.08 kW/m²; 0 < x < 1.0. As mentioned earlier, the local heat transfer coefficients and local qualities are measured for the two-phase flow. Figure 13 shows the local heat transfer coefficients verses local quality for a mass flux of 157 kg/m²s. The Reynolds number for this case is 116. As the local quality is increased, the heat transfer coefficient decreases.

![Figure 13: Local Heat Transfer Coefficients vs. Local Quality. For: G = 157 kg/m²s; Re = 116; 55.91 kW/m²K < q” < 309.95 kW/m²K; 0 < x < 1.]

The other local heat transfer coefficients verses local quality for mass fluxes of 366, 671, 1022, and 1782 kg/m²s are seen in Figures 14, 15, 16, and 17, respectively. The Reynolds numbers are 270, 496, 756, and 1318. All of these figures are grouped together because the all exhibit similar trends. The same decreasing trend is seen in the heat transfer coefficient. The trend is consistent with a nucleate boiling dominance flow.

![Figure 14: Local Heat Transfer Coefficients vs. Local Quality. For: G = 366 kg/m²s; Re = 270; 118.68 kW/m² < q” < 504.47 kW/m²; 0 < x < 1.]

![Figure 15: Local Heat Transfer Coefficients vs. Local Quality. For: G = 671 kg/m²s; Re = 496; 216.82 kW/m² < q” < 635.72 kW/m²; 0 < x < 1.]

The other local heat transfer coefficients verses local quality for mass fluxes of 366, 671, 1022, and 1782 kg/m²s are seen in Figures 14, 15, 16, and 17, respectively. The Reynolds numbers are 270, 496, 756, and 1318. All of these figures are grouped together because the all exhibit similar trends. The same decreasing trend is seen in the heat transfer coefficient. The trend is consistent with a nucleate boiling dominance flow.
One explanation could be that the decreased channel size has reduced the available space for bulk flow to develop.

Figure 16: Local Heat Transfer Coefficients vs. Local Quality. For: $G = 1022 \text{ kg/m}^2\text{s}; \text{Re} = 756$; $348.27 \text{ kW/m}^2 < q'' < 699.64 \text{ kW/m}^2$; $0 < x < 1$.

Figure 17: Local Heat Transfer Coefficients vs. Local Quality. For: $G = 1782 \text{ kg/m}^2\text{s}; \text{Re} = 1318$; $609.94 \text{ kW/m}^2 < q'' < 898.08 \text{ kW/m}^2$; $0 < x < 1$.

**Single- and Two-Phase Pressure Drop.** The adiabatic friction factor for laminar flow is first determined experimentally to provide validity of the test section and measurement techniques. The adiabatic friction factor experiments are conducted for both the visualization and heat transfer test sections. This was conducted before any diabatic experiments began to ensure that the channels behaved in the proper manner. The adiabatic friction factor for the heat transfer experiments is shown in Figure 18. There is also very good agreement with the predicted friction factor and is also within 10% of the predicted values.

Figure 18: Adiabatic Friction Factor vs. Reynolds Number for Heat Transfer Experiments. For: $161 \text{ kg/m}^2\text{s} < G < 1047 \text{ kg/m}^2\text{s}; q'' = 0.0 \text{ W/m}^2$.

Figure 19: Corrected Diabatic Friction Factor vs. Reynolds Number for Heat Transfer Experiments. For: $G = 157, 366, 671, 1022, \text{ and } 1782 \text{ kg/m}^2\text{s}; \text{Re} = 116, 270, 496, 756, \text{ and } 1318$; $5 \text{ kW/m}^2 < q'' < 898 \text{ kW/m}^2$.

The pressure drop was measured during all of the heat transfer experiments as well. The measured pressure was taken across the length of the microchannel as the fluid entered and
The diabatic friction factor for laminar flow of water in microchannels is accurately described by the established relationship for large (conventional) diameter channels. The underlying fluid mechanics still applies.

CONCLUSIONS

An experimental investigation is conducted to study the single and two-phase flow in microchannels with a 207-micrometer hydraulic diameter during laminar flow. The following conclusions are drawn from the present study.

- The following flow patterns are observed in the microchannels; nucleate boiling, bubbly flow, slug flow, annular flow, annular flow with nucleation in the thin film, churn, and dry-out. All of these regimes have been identified in conventional and minichannels.
- A flow reversal has been observed under certain conditions in microchannels. The vapor interface moves in a direction counter to the bulk fluid flow. This is also seen in minichannels. This phenomenon will also be observed in conventional sized channels in parallel configuration for certain combination of pressure drop, heat flux and mass flux conditions. Under this condition, the vapor interface moves in a direction counter to the bulk fluid flow.
- Another flow boiling phenomenon detected is the local dry-out phenomenon. The interface at which the dry-out occurs has been visually inspected and documented. The contact angles of the interline exhibit similar behavior to that found in pool boiling as noted by Kandlikar and Steinke [20].
- The single and two-phase flow heat transfer in the microchannel has been investigated. The fluid has been properly degassed and the validity of the data is sound. As expected, the heat transfer in the microchannels is improved over a conventional and minichannel. The Nusselt numbers for the experimental data strongly agree with those predicted using conventional theory. A heat flux of 930 kW/m² has been achieved. Heat transfer coefficients as high as 192 kW/m²K have been achieved along with qualities of 1.0. The application of two-phase microchannels could provide a viable alternative to future electronics cooling.
- The present work constitutes one of the first sets of local heat transfer coefficients and quality for microchannels. The trends that have been observed for the low Reynolds number and laminar Reynolds number cases are consistent with a nucleate boiling dominant flow.
- The adiabatic single-phase friction factor for laminar flow of water in microchannels is accurately described by the established relationship for large (conventional) diameter channels.
- The diabatic friction factors for a single-phase flow can be accurately predicted using the property ratio correction factors used for large diameter (conventional) channels. The underlying fluid mechanics still applies.

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REFERENCES


