Dependence of Flow Boiling Heat Transfer Coefficient on Location and Vapor Quality in a Microchannel Heat Sink

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ABSTRACT

Experiments were conducted to determine the influence of local vapor quality on local heat transfer coefficient in flow boiling in an array of microchannels. Additionally, the variation of local heat transfer coefficient along the length and width of the microchannel heat sink for given operating conditions was investigated over a range of flow parameters. Each test piece includes a silicon parallel microchannel heat sink with 25 integrated heaters and 25 temperature sensors arranged in a 5x5 grid, allowing for uniform heat dissipation and local temperature measurements. Channel dimensions ranged from 100 $\mu$m to 400 $\mu$m in depth and 100 $\mu$m to 5850 $\mu$m in width; the working fluid for all cases was the perfluorinated dielectric liquid, FC-77. The heat transfer coefficient is found to increase with increasing vapor quality, reach a peak, and then decrease rapidly due to partial dryout on the channel walls. The vapor quality at which the peak is observed shows a strong dependence on mass flux, occurring at lower vapor qualities with increasing mass flux for fixed channel dimensions. Variations in local heat transfer coefficient across the test piece were examined both along the flow direction and in a direction transverse to it; observed trends included variations due to entrance region effects, two-phase transition, non-uniform flow distribution, and channel wall dryout.

INTRODUCTION

Relationship Between Heat Transfer Coefficient and Vapor Quality

Two-phase microchannel heat sinks are capable of achieving high heat transfer rates, with the added advantage of minimizing temperature fluctuations over the heat sink [1]. However, investigations into the relationship between heat transfer coefficient and vapor quality in microchannels have yielded widely varying results. Establishing the relationship between these two variables is critical to the design of two-phase microchannel heat sinks [2]. Trends of variation in the heat transfer coefficient with vapor quality observed in past studies included an M-shaped double peak [3], a single peak [2], continuously decreasing [4], and continuously increasing to very high vapor qualities and then dropping off sharply due to dry-out [5]. All of these studies commonly concluded that the decreases in heat transfer coefficient are due to contact of vapor with the channel walls; the value of the vapor quality at which this decrease occurs depends strongly on the flow regimes present in the test piece.

There is a variety of experimental techniques and approaches that can be utilized in order to obtain the results of interest, which adds to the complexity of investigating relationships between vapor quality and heat transfer coefficient. One approach involves choosing a fixed temperature measurement location and varying the heat input to the fluid upstream of the heat sink or to the bottom of the heat sink in order to adjust the vapor quality at the point of interest. The fixed measurement location chosen is most commonly at the exit of the test piece. Alternatively, multiple measurement locations may be spaced along the flow direction while applying a fixed heat input to the bottom of the heat sink. In this manner, it is possible to simultaneously obtain multiple temperature measurements over a range of vapor qualities. However, even studies that have used similar approaches to obtain the desired range of vapor qualities have still yielded widely different results with respect to the variation in heat transfer coefficient.

Spatial Variation of Heat Transfer Coefficient

The second objective of the present study was to examine the spatial variation of local heat transfer coefficient across the width and length of a microchannel heat sink. The practical implementation of two-phase microchannel technologies demands a more thorough understanding of phase change phenomena in such devices [6]. Several studies in the literature have examined the development of the heat transfer coefficient...
in the streamwise direction; these studies included both numerical [7] and experimental [6] analyses.

A study of the literature reveals an absence of information on the distribution of local heat transfer coefficients transverse to the flow direction in a microchannel heat sink, especially in the presence of flow boiling. In the present study, variation of the heat transfer coefficient across the heat sink, both along the flow direction and in the transverse direction, are investigated. Each test piece includes 25 temperature sensors and heaters fabricated into a silicon microchannel heat sink, enabling the calculation of local heat transfer coefficient, \( h \) (W/m\(^2\)K), and the application of a uniform base heat flux, \( q_b \) (W/m\(^2\)). Additionally, the impact of vapor quality on heat transfer coefficient is examined and compared to findings from past studies.

**EXPERIMENTAL SETUP AND DATA PROCESSING**

**Flow Loop**

Fig. 1 shows the experimental facility used in the present study. The facility is equipped with an expandable reservoir, a pump to circulate the working fluid through the test loop, a pre-heater to control the degree of sub-cooling at the test section inlet, and a heat exchanger after the test section to cool the working fluid before returning to the reservoir. The loop includes flow meters, pressure transducers, and thermocouples at important locations; a high-speed camera is used for visualizing the flow patterns within the microchannels. The pressure at the outlet of the microchannel heat sink is maintained at 1 atm throughout the tests. The setup is equipped with several degassing ports in order to fully degas the FC-77 before each test as it has an affinity for the absorption of atmospheric gasses that result in altered fluid performance, especially during two-phase operation.
Test Section

Each test piece consists of a silicon chip (12.7 mm × 12.7 mm × 650 µm in size) with microchannels sawed into one side. The top cover of the heat sink is transparent, as shown in Fig. 2a, enabling flow visualizations to be obtained during operation. Several test pieces with different channel dimensions are tested with depths ranging from 100 µm to 400 µm, and widths ranging from 100 µm to 5850 µm. The average roughness of the channel side walls is 0.1 µm, while the average roughness of the bottom wall of the channels is in the range of 0.8 to 1.4 µm depending on test piece [8].

On the other side of the silicon chip are 25 diode temperature sensors and resistor heaters arranged in a 5×5 grid as shown in Fig. 2b. The heaters were connected in parallel and a voltage applied in order to generate a uniform base heat flux, \( q_b'' \) (W/m²). There is an inherent variation in heater resistance values due to manufacturing tolerances; however, this led to a maximum variation in base heat flux of only ±5.5% over the heat sink. A constant current was applied to the temperature sensing diodes; the measured voltage drop across the diodes thus corresponded to their temperature. The voltage-temperature correlation for the diodes was obtained via calibration. Fig. 3 shows the reference axes used for determination of position across the heat sink.

Data Processing

In order to obtain local heat transfer coefficients, each of the 25 heated sections is examined individually. The net heat transferred to the fluid from each heater is found using:

\[
\dot{q}_{\text{net}} = \dot{q} - \dot{q}_{\text{loss}}
\]

where \( \dot{q} = V^2 / R \). Heater resistance values show a slight temperature dependence that was accounted for through calibration tests performed on each heater beforehand. The heat losses included in the \( \dot{q}_{\text{loss}} \) term are due to conduction from the heat sink to the exterior surfaces of the test section, as well as convection and radiation to the surroundings. The relationship between the rate of heat loss and the operating temperature was obtained by running the test piece dry at a range of elevated temperatures and measuring the required power inputs. The outcome of this procedure yielded a linear relationship of the form \( \dot{q}_{\text{loss}} = c_1 \times T + c_2 \), indicating that heat loss is primarily by conduction.

In order to obtain the heat transfer coefficient, several heat sink performance parameters are first determined. The fin efficiency, \( \eta_f \), is calculated by assuming an adiabatic tip condition, using \( \eta_f = \tanh(md) / md \), where \( m^2 = 2h / k_s w_f \) [8]. An overall surface efficiency for the microchannel heat sink is then found using:

\[
\eta_s = 1 - \frac{NA_f}{A_w}(1 - \eta_f)
\]

The channel wall temperature, \( T_w \), is found by adjusting the diode temperature reading to account for conduction through the silicon chip using:

\[
T_w = T_d - \frac{q_w''(t-d)}{k_d}
\]

After calculating the overall heat sink surface efficiency and wall temperature the heat transfer coefficient can be found using:

\[
h = \frac{q_w''}{\eta_s(T_w - T_{sat})}
\]

where \( q_w'' \) is the heat flux through the wetted microchannel surfaces. Since heat transfer coefficient appears in calculating the fin efficiency, an iterative approach had to be employed to obtain the correct value for \( h \). The initial value for \( h \) used in the iterations was obtained by assuming 100% efficiency for the heat sink. Using this procedure a local heat transfer coefficient was found at each of the temperature sensor locations. Saturation temperature was adjusted based on variations in pressure. Additionally, during single phase operation the local bulk fluid temperature was used instead to find \( h \). Finally, the local vapor quality for each sensor could be found as:
\[ x = \frac{1}{h_{fg}} \left[ \int_0^x \frac{dq}{m_f} \, dx - c_{p,f} (T_{sat} - T_{inlet}) \right] \]  

(5)

RESULTS AND DISCUSSION

Relationship Between Heat Transfer Coefficient and Vapor Quality

Fig. 5 shows the heat transfer coefficient as a function of vapor quality for two different microchannel dimensions (400 µm deep × 250 µm wide and 400 µm deep × 5850 µm wide); and four different mass fluxes (250, 700, 1150, and 1600 kg/m²s). The results obtained here reveal that the heat transfer coefficient increases with increasing vapor quality until it reaches a maximum value, and then decreased rapidly.

Fig. 5. Heat transfer coefficient as a function of vapor quality at sensor #3 for heat sink with microchannel dimensions: (a) 400 µm deep × 250 µm wide, and (b) 400 µm deep × 5850 µm wide.

The vapor quality and heat transfer coefficient are both calculated at a fixed location near the exit of the test piece (sensor #3). A range of vapor qualities was obtained by increasing the heat input to the test piece. Tests were terminated soon after dry-out occurred at the channel walls. The decrease in heat transfer coefficient led to increased heat sink temperatures that could have damaged the test piece.

The vapor quality at which the peak in heat transfer coefficient occurs shows a strong dependence on mass flux, occurring at lower vapor qualities for increasing mass flux as seen in Fig. 5a and Fig. 5b. This trend is in agreement with a study by Madrid et al. [9], and was attributed to the stability of the liquid film in slug and annular flow. Due to higher velocities accompanying the larger mass fluxes, the liquid film grows unstable at lower vapor qualities and is unable to remain attached to the channel walls. This yields churn, or in extreme cases inverted annular, flow regimes resulting in contact between vapor and the channel walls. This partial dryout at the walls drastically lowers the heat transfer coefficient.

Fig. 6 shows four distinct trends in the relationship between heat transfer coefficient and vapor quality found in the literature that differed from the trends observed in the present study. The data in these plots were obtained by digitizing the images in the original studies and plotting in a comparative manner. Table 1 identifies several key operating conditions and experimental parameters in the studies presented.

In contrast with the current study, Bertsch et al. [2] found that the vapor quality at which the peak heat transfer coefficients is observed was independent of mass flux and occurred at roughly the same vapor quality, as seen in Fig. 6a. The cause of this was traced back to the flow regimes present in the study. Due to the experimental setup and flow conditions used in that study, slug flow was the dominant flow regime present, with annular flow occurring only beyond vapor qualities of 0.7 [2]. Fig. 6b shows the results of various other studies [10], [11] compiled by Bar-Cohen and Rahim [3] in which a double peak trend was observed. Saitoh et al. [5] found that \( h \) continuously increased until very high vapor qualities, at which point it drastically dropped as seen in Fig. 6c. Finally, Steinke and Kandlikar [4] observed that there was a continual decrease in \( h \) with increasing \( x \), as shown in Fig. 6d.

All four of these distinct trends differ from those seen in the present study. The studies in the literature had many differences from the present study, and from one another, including channel geometry, working fluid, and operating conditions. All of these factors contribute to differences in the flow regimes present in the channels and the vapor quality at which vapor comes into contact with the heat sink walls.

Trends in heat transfer coefficient as a function of vapor quality were also examined over a range of channel dimensions. Fig. 7 shows \( h \) vs. \( x \) at sensor 3 for a fixed mass flux (250 kg/m²s), channel depth (400 µm) and varying channel widths. At a fixed mass flux, the peak in heat transfer coefficient is seen to occur at lower vapor qualities for a larger channel cross-sectional area. These trends cannot be compared to other studies in the literature as no detailed results related to the effect of cross sectional area on the relationship between \( h \) and \( x \) in microchannel heat sinks are available.
Fig. 6. Results for $h$ vs. $x$ from the literature showing (a) the peak occurring at a fixed $x$ [2], (b) a double peak resulting in an M-shaped curve [3], [10], [11], (c) $h$ increasing until very high vapor qualities, and then dropping rapidly [5], and (d) a continual decrease [4].

Table 1 – Experimental parameters for studies from literature presented in this study. Pressure data, where available, is listed at measurement location(s) unless otherwise noted.

<table>
<thead>
<tr>
<th>Author, Year</th>
<th>Working Fluid</th>
<th>$D_0$ (mm)</th>
<th>$q''$ (kW/m²)</th>
<th>$G$ (kg/m²s)</th>
<th>Inlet Condition</th>
<th>Pressure (kPa)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Steinke and Kandlikar, 2004</td>
<td>water</td>
<td>0.207</td>
<td>184.08-504.47</td>
<td>366</td>
<td>Sub-cooled</td>
<td></td>
</tr>
<tr>
<td>Yang and Fujita, 2004</td>
<td>R113</td>
<td>0.976</td>
<td>20-90</td>
<td>100</td>
<td>Saturated</td>
<td>101-219</td>
</tr>
<tr>
<td>Saitoh, Daiguji, and Hihara</td>
<td>R134a</td>
<td>1.12</td>
<td>13-27</td>
<td>150-300</td>
<td>Saturated</td>
<td>448-488 (Drop across test piece)</td>
</tr>
<tr>
<td>Cortina-Diaz and Schmidt, 2006</td>
<td>n-octane</td>
<td>0.586</td>
<td>11.6–148.3</td>
<td>100</td>
<td>Sub-cooled</td>
<td>101.3</td>
</tr>
<tr>
<td>Bertsch, Groll and Garimella, 2008</td>
<td>HFC-134a</td>
<td>1.089</td>
<td>0-200</td>
<td>20.3-81.0</td>
<td>Sub-cooled and Saturated</td>
<td>550</td>
</tr>
</tbody>
</table>
Spatial Variation of Heat Transfer Coefficient

Local variations in heat transfer coefficient were examined along the length and width of the microchannel heat sink. The test piece was subject to both single- and two-phase operating conditions, with fluid flow in the positive $X$ direction. For streamwise measurements, data were gathered from sensors 3, 8, 13, 18 and 23 (Fig. 2b); for measurements normal to the flow, measurements were taken at sensors 11, 12, 13, 14 and 15. The coordinate axes used for reference measurements are shown in Fig. 3.

Fig. 8 shows the spatial variations in $h$ for a fixed channel size (depth 400 µm × width 250 µm) operating at two different mass fluxes. The variations are presented in the two directions mentioned previously: along the flow direction ($X$) and orthogonal to it ($Y$). For each case several values of base heat flux are considered.

Several key features may be identified when examining the variations in heat transfer coefficient normal to the flow direction ($Y$-direction). During single-phase conditions the heat transfer coefficient is relatively constant; however, as heat flux
is increased, boiling causes variations in $h$ in the transverse direction. These variations were due to the non-uniform incipience of boiling across the chip, as well as uneven flow distribution due to manifold design.

In Fig. 8a, for $q_{b}'' = 110.18$ kW/m$^2$, two different values for $h$ are observed on two transverse sides of the heat sink, with the lower value being close to the single-phase cases. This demonstrates that at a fixed location downstream from the inlet, boiling has started in the channels along one side of the heat sink, resulting in an increase in the heat transfer coefficient, while single-phase liquid is flowing in the rest of the channels. Uneven incipience of boiling in different microchannels was confirmed through visualizations as well, as shown in Fig. 9.

Additionally, during two-phase operation there was a maximum of 12.9% variation in $h$ transverse to the flow direction. For the cases presented in Fig. 8 the test piece was operating at $Re = 360$ for $G = 700$ kg/m$^s$, and $Re = 820$ for $G = 1600$ kg/m$^s$. Jones et al. [12] demonstrated that at $Re = 102$ there was up to 30% more flow in the center channels than the channels near the sides of a microchannel heat sink, this maldistribution of flow was due to the design of the inlet and exit manifolds.

For a sufficiently high heat input, dry-out occurs at the channel walls, yielding drastically lower heat transfer coefficients; this can be seen in Fig. 8a where the heat transfer coefficient for $q_{b}'' = 646.68$ kW/m$^2$ is less than even that of $q_{b}'' = 273.57$ kW/m$^2$. Trends in the variation of heat transfer coefficient transverse to the flow direction in microchannel heat sinks are not readily available in the literature for comparison.

As seen in Fig. 8c ($q_{b}'' = 64.96, 82.26$ kW/m$^2$) and Fig. 8d ($q_{b}'' = 144.96$ kW/m$^2$) during single-phase operation, the heat transfer coefficient in the streamwise direction started at a maximum at the test section inlet and steadily decreased. This can be attributed to the development of thermal and momentum boundary layers within the channels, as well as variations in fluid properties as the fluid temperature changes. These findings are consistent with the results of a past numerical study by Li et al. [7] and an experimental study by Kuznetsov and Shamirzaev [6] investigating streamwise heat transfer coefficient variation in microchannel heat sinks. A sudden increase in the heat transfer coefficient for a small increase in heat input can be observed when boiling begins in the test piece. This is noticed in Fig. 8c between the two single-phase cases ($q_{b}'' = 64.96, 82.26$ kW/m$^2$) that lie on top of each other, and the two-phase case with $q_{b}'' = 110.18$ W/m$^2$. Additionally, in Fig. 8d, it is evident that for $q_{b}'' = 179.69$ kW/m$^2$, the transition from single-phase to two-phase flow occurs near $X = 5$mm where the heat transfer coefficient suddenly increases. Even during two-phase operation it is apparent that the heat sink has an entrance region where boundary layers are developing, as indicated by the fact that the heat transfer coefficient is highest near the inlet and decreases in the streamwise direction. In the present study fluid entering the heat sink was always sub-cooled. While Kuznetsov and Shamirzaev [6] obtained similar results using sub-cooled fluid, they also considered a two-phase inlet condition and found that the heat transfer coefficient remained relatively constant along the heat sink and did not exhibit entrance-region trends. Finally, at the highest heat input values, the test piece experiences dry-out at the channel walls near the exit of the heat sink, resulting in a dramatic drop in the heat transfer coefficient towards the exit of the channels, as seen in Fig. 8c for $q_{b}'' = 646.68$ kW/m$^2$.

![Fig. 9. The location of incipience of boiling (solid red lines), and fully developed boiling (dashed green lines) in different channels in a heat sink.](image)

**CONCLUSIONS**

The dependence of local heat transfer coefficient on location and vapor quality in a silicon microchannel heat sink has been investigated experimentally using FC-77 as the working fluid and compared to trends found in past studies.

There have been many studies investigating the relationship between heat transfer coefficient and vapor quality with varying results. The main issue identified in the present study, as well as that of Madrid et al. [9], was that there is a stronger relationship between $h$ and the flow regime present within the test piece than vapor quality. Ultimately, the maxima were associated with the point just before vapor came into contact with the channel walls. In the present study, a single peak was observed, and this peak showed a strong dependence on mass flux, occurring at lower vapor qualities with higher mass flux. For a fixed mass flux it was observed that the peak also occurs at lower vapor qualities for increasing channel cross sectional area.

When examining the variation of heat transfer coefficient transverse to the direction of the flow it was observed that at a fixed downstream location boiling can occur in some channels while not in others. When this occurred the heat transfer coefficient differed by up to 36.6% across the test piece. Due to manifold design there was a slight maldistribution of flow...
among the channels as made evident by the 12.9% maximum variation in heat transfer coefficient transverse to the flow direction during two-phase operation. These fluctuations can be minimized with more careful manifold design at the inlet and exit of the heat sink.

In agreement with past numerical and experimental studies, it was observed that the heat transfer coefficient was at a maximum value near the entrance of the channels. For single-phase flow, the heat transfer coefficient was at a maximum at the inlet of the channels, decreased through the entrance region, and approached a fully developed value. In some cases, there was a large jump in $h$ at a location downstream from the inlet where boiling began. When the entire length of the channels was operating under two-phase flow, a distinct entrance region with a higher heat transfer coefficient was still present. Ultimately at high heat fluxes the heat transfer coefficient rapidly decreased towards the exit of the heat sink due to dry-out at the channel walls.

NOMENCLATURE

- $A_b$: Heat sink base area, $m^2$
- $A_f$: Wetted area of a fin, $m^2$
- $A_w$: Total wetted area of microchannels, $m^2$
- $c_1, c_2$: Constants in heat loss and temperature relation
- $d$: Microchannel depth, $m$
- $D_h$: Hydraulic Diameter, $mm$
- $G$: Mass flux, $kg/m^2s$
- $h$: Heat transfer coefficient, $W/m^2K$
- $h_{fg}$: Latent heat of vaporization for FC-77, $J/kg$
- $k_d$: Thermal conductivity of silicon, $W/mK$
- $L$: Heat sink width, microchannel length, $m$
- $m$: Intermediate variable in fin efficiency calculation, $m^{-1}$
- $m_f$: Mass flow rate of fluid, $kg/s$
- $N$: Number of microchannels in a heat sink
- $q$: Heat dissipated by heater resistors, $W$
- $q_b$: Base heat flux, $W/m^2$
- $q_{loss}$: Heat loss, $W$
- $q_{net}$: Heat transferred to fluid, $W$
- $q_w$: Wall heat flux, $W/m^2$
- $R$: Heater resistance, $\Omega$
- $Re$: Reynolds number
- $t$: Heat sink thickness, $m$
- $T_d$: Diode temperature, $C$
- $T_{inlet}$: Fluid temperature at heat sink inlet, $C$
- $T_{sat}$: Fluid saturation temperature, $C$
- $T_w$: Wall temperature, $C$
- $V$: Voltage applied to heaters, $V$
- $w$: Microchannel width, $m$
- $w_f$: Fin width, $m$
- $x$: Vapor quality
- $X$: Position in $X$ direction, $m$
- $Y$: Position in $Y$ direction, $m$

Greek

- $\eta_f$: Efficiency of a fin in heat sink
- $\eta_o$: Overall microchannel heat sink efficiency

REFERENCES


