Flow boiling heat transfer characteristics on structured surfaces

Ryan Fogarty

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Flow Boiling Heat Transfer
Characteristics on Structured Surfaces

by

Ryan Fogarty

A Thesis Submitted
in
Partial Fulfillment
of the Requirements for the
MASTER OF SCIENCE
in
Mechanical Engineering

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DEPARTMENT OF MECHANICAL ENGINEERING
COLLEGE OF ENGINEERING
ROCHESTER INSTITUTE OF TECHNOLOGY
February 1999
Thesis Title: “Flow Boiling Heat Transfer Characteristics on Structured Surfaces”

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February, 1999

Ryan Fogarty
FORWARD

The last three years has been a trial for me. I have gone through some ups, downs, highs, lows, long days, short nights, achievements, failures, happiness, and sadness. Through this transient part of my life, I have had many certainties. These certainties were constant support, motivation, guidance, and encouragement from people that are important to my life. At this time, I would like to thank the following people for being such an influence during my quest for higher education:

- Dr. Satish Kandlikar – my advisor, mentor, and friend.
- Murat Bulut – my friend forever, even when he goes back to Turkey.
- The crew from 509 – my life long friends, and the people that make sure that I eat.
- The ladies in the mechanical engineering office – they offered me a lot of assistance.
- Jesse McKnight – he laser etched my surfaces.
- The crew from the heat lab – my study partners.
- Tom Locke – helped me with everything that had to be machined.
- Fanny Hillengas – my grandmother.
- The Fogarty family – my motivation and encouragement.
- The Hillengas family – my motivation and encouragement.
- Lynda Fogarty – my wife, sanity and most of all, my best friend.
The fore mentioned people tried to make these years pleasant. For their efforts, I dedicate this thesis to all of them.
The present investigation focuses on studying the performance of three specially designed surfaces in flow boiling heat transfer. The surfaces are prepared by drilling holes of specific sizes using laser ablation drilling. The heat transfer characteristics for three laser drilled enhanced surfaces, and a smooth surface are obtained by collecting experimental data for flow boiling of subcooled water over heater surfaces with different matrix of holes. The matrix of laser etched cavities are as follows:

- Surface 1 - 9.0μm diameter holes spaced at 58.4μm
- Surface 2 - 3.7μm diameter holes spaced at 38.6μm
- Surface 3 - 3.7μm diameter holes spaced at 58.4μm
- Polished Surface - polished with a polishing wheel with 5.0μm solution.

By varying the hole sizes and density in a systematic way, some insight is obtained in the flow boiling heat transfer mechanism on these enhanced surfaces.

The experimental apparatus consisted of a 3 mm x 40 mm flow channel with a circular heater of 9.5 mm mounted in the center of the lower wall. The heater is instrumented with thermocouples to provide a measurement of the surface
temperature and heat flux at the heater surface. The experiments are conducted for two subcooling temperature values, 5 K and 10 K at atmospheric pressure.

It has been observed that the size of the holes plays a major role in the heat transfer characteristics of the surface. An order of magnitude enhancement in performance over a polished surface is obtained with the augmented surfaces tested in this investigation.
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\( \Delta T \) Wall Superheat K
\( T_{fl} \) Inlet Bulk Liquid Temperature \( ^\circ C \)
\( T_g \) Vapor Temperature \( ^\circ C \)
\( T_s \) Surface Temperature \( ^\circ C \)
\( T_{sat} \) Saturation Temperature \( ^\circ C \)
\( T_w \) Wall Temperature \( ^\circ C \)
\( \nu_{lg} \) Difference in Specific Volumes of Saturated Liquid and Vapor m\(^3\)/kg
\( x \) Mass Vapor Quality

**Greek Letters**

\( \phi \) Surface Heat Transfer W/m\(^2\)
\( \zeta \) Dimensionless Parameter Defined by eq. (3.10)
\( \sigma \) Surface Tension N/m
\( \rho_g \) Gas Density Kg/m\(^3\)

**Subscripts**

CRIT Critical
ONB Onset of Boiling
SAT Saturation
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1. INTRODUCTION

Nucleate boiling heat transfer is important in many applications. Nucleate boiling is employed in heating and air conditioning, automotive, nuclear power, pharmaceuticals, oil refineries, and many other systems. Understanding and enhancing boiling characteristics can benefit the equipment designer in reducing the size of the components. Because nucleate boiling involves a phase change, greater heat transfer can be achieved with small temperature differences. To improve this heat transfer further, novel techniques such as sintered surfaces are employed.

Nucleate boiling occurs in pool boiling as well as in flow boiling. Pool boiling can be defined as a process where a heated surface is submerged in a quiescent volume of liquid. A typical example of pool boiling is boiling of water in a teakettle. The bottom surface of the kettle is the origination of the bubbles, which nucleate over preferred sites on the surface. If the liquid is at its saturation temperature, then it is referred to as saturated pool boiling. When the liquid is at a temperature below its saturation temperature, the boiling is referred to as the subcooled boiling.

In flow boiling, liquid flows at some velocity inside a heated tube or over a heated surface. The flow boiling heat transfer consists of two components: nucleate boiling and convective boiling. The heat transfer performance of a heating
surface can be improved by enhancing one or both of these components. Microfins, twisted tapes, and helical inserts mainly improve the convective contribution and are generally employed in evaporators in the high vapor quality region. On the other hand, sintered coating, structured surfaces with specially designed cavities, and foil inserts enhance the nucleate boiling component and are found useful in the partial boiling region (Carey, 1992).

Sometimes holes are etched with a laser beam in creating a number of artificial sites where bubbles are nucleated. This type of surface has not been studied in flow boiling application. The present study focuses on investigating this surface in more depth.
2. OBJECTIVES OF CURRENT WORK

The objective of the current work is to compare the effect of hole size and spacing of laser drilled holes in the partial boiling region. The performance of these surfaces will be compared as a function of Reynolds number and subcooling.
3. Literature Review

3.1 Flow Boiling Background

Consider the progression of the flow regimes in a horizontal tube. The single-phase liquid regime can be considered as a quality of zero and the single-phase vapor regime can be considered as a quality of one. The regimes between single-phase liquid and single-phase vapor possess the quality between zero and one. As the flow distance is increased the quality \( X \) increases. The quality of the flow is the ratio of vapor flow to total flow,

\[
X = \frac{m_v}{m} \tag{3.1}
\]

where,

\[
m = m_v + m_l \tag{3.2}
\]

where,

- \( m_v \) – mass of vapor (kg)
- \( m_l \) – mass of liquid (kg)

Since flow boiling is the focus of this study, a background in convective boiling in tubes and channels will be the topic of this section. There are different types of flow. Figure 3.1 shows the variation of the heat transfer coefficient and flow
regime for internal convective boiling in a horizontal tube, qualitatively. As the specimen is subject to constant heat flux, the flow experiences many different flow regimes. As the subcooled water flow reaches the saturation temperature of water the flow experiences single-phase liquid flow. As the bulk temperature increases above the saturation temperature, the walls begin to form little bubbles. The increasing bulk temperature reaches superheat, and little bubbles start to form.

The bubbles, at this point, do not accumulate on the walls. There is not enough energy to sustain the bubbles and as soon as they appear, they collapse. The collapsing of the bubbles is known as cavitation. The bubble collapses upon itself at an extreme velocity. This is the reason boiling can be loud when it first begins to boil. This phenomenon is referred to as the onset of nucleate boiling or better known as ONB.

The bubbles start to form on the walls of the tube as the bulk temperature increases. The bubbles are able to sustain their existence and collect on the wall. This phenomenon is known as net vapor generation or NVG. The regime between ONB and NVG is considered bubbly flow.

As the bubbles start to collect on the walls of the pipe, they start to combine to make bigger bubbles. The bubbles start to take on an abnormally large size and
Figure 3.1 Qualitative variation of the heat transfer coefficient and flow regime with quality for internal convective boiling in a horizontal tube. Adopted from Carey 1992
are considered slugs. This regime where the flow is dominated by slugs is called plug flow. This regime still has some bubbles, but the number of bubbles diminishes.

The vapor travels along the axis of the pipe. The liquid coats inner lining of the pipe. This regime is known as annular flow. Prior to the onset of partial dryout, transfer of heat across the liquid film becomes more efficient as the film becomes progressively thinner (Carey, 1992). There are minute traces of bubbles, and eventually, the bubbles are totally suppressed in this in this regime. Partial dryout can be observed in the latter region of the annular flow regime. The dryout occurs, because the thin film of water around the wall starts to vaporize.

When all the liquid vaporizes from the wall, it is considered mist flow regime. This regime is considered mist flow, because the pipe contains mist in this region. The mist in the pipe eventually vaporizes, and the pipe experiences the last stage in the flow regime, single-phase vapor flow.

The heat transfer coefficient that is associated with the different flow regimes can be observed in figure 3.1. Notice that the heat transfer coefficient is steadily increasing until the dryout is experienced. There is no medium to carry the heat from the wall, since the flow consists of vapor only.
3.2 Bubble Growth

Boiling is a term that is defined as evaporation of a liquid at a solid-liquid interface. The heat is transferred from a higher temperature or potential to a lower temperature or potential. The higher potential is the wall and the lower potential is the water. Boiling occurs when the wall temperature is greater than the saturation temperature of water. The saturation temperature of water is 100°C at atmospheric temperature, otherwise known as the boiling temperature of water. Newton’s law of cooling can be applied to the transfer of the heat:

\[ q'' = h(T_s - T_{sat}). \]  \hspace{1cm} (3.3)

where,

- \( q'' \) – Heat Flux (W/m²)
- \( h \) – Heat Transfer Coefficient (W/m²K)
- \( T_s \) – Surface Temperature (K)
- \( T_{sat} \) – Saturation Temperature (K)

The nucleation takes place over the cavities on the surface of the solid. The term used for the steady cyclic growth and release of vapor bubbles at an active nucleation site is ebullition cycle (Carey, 1992). Figure 3.2 is a model of the ebullition cycle.
Figure 3.2 Model ebullition cycle. Adopted from Carey, 1992.
The growth of the bubbles in the thermal boundary layer near a superheated surface is complex. The complexity is due to a lack of spherical symmetry and the non-uniformity of the temperature field in the surrounding liquid. There are two regimes of growth. These two regimes are inertia-controlled growth and heat-transfer-controlled growth. The inertia-controlled growth is where the bubble growth is resisted by the inertia of the surrounding liquid. The heat-transfer-controlled bubble growth is where the bubble growth is limited by the ability to transfer heat.

Figure 3.3 is a model of the progression of bubble growth. As a bubble departs from the surface, there is a brief period where the thermal layer goes to zero at the site of the cavity. The surface temperature is greater than the bulk temperature of the liquid. During this period, transient conduction dominates and bubbles do not grow. This period is called the waiting period.

During the early stages of bubble growth the liquid near the wall becomes superheated. The bubble embryo creeps from the nucleation cavity site. Rapid growth of the bubbles can be observed at this stage. Heat-transfer is not the limiting factor at this time. The inertia of the surrounding liquid impedes the growth of the bubble. Because of this phenomenon, this stage is labeled as inertia-controlled growth. The bubbles are growing radially at this point. An evaporation microlayer forms near the wall. This film transfers the heat from
Figure 3.3 The progression of bubble growth.
Adopted from Carey, 1992.
the wall to the interface. The liquid is vaporized at the interface. The film may

evaporate near the cavity where the nucleation begins. This evaporation
elevates the surface temperature significantly. The cyclic dry-outs and rewets
fluctuate the surface temperature.

The liquid near the interface is gradually depleted of its superheat as the bubble
continues to grow. The temperature of the liquid increases as the distance of the
bubble from the interface is increased. The increase comes to a peak, and then
the liquid temperature decreases approaching the ambient temperature. The
heat transfer to the interface limits the bubble growth; hence this phase is called
heat-transfer-controlled bubble growth.

During this stage of the bubble growth, the bubble is near its maximum size.
Buoyancy, drag, lift, and/or inertia forces of the surrounding fluid may pull the
bubble away from the surface. As the bubble becomes larger, these forces
become larger. When these forces become large enough, the bubble will
separate from the surface.
3.3 Onset of Subcooled Nucleate Boiling

In flow boiling, the wall temperature has to be greater than the saturation temperature for boiling. The minimum limiting conditions for nucleation are as follows (Collier, 1972):

\[ T_{sat} \leq T_w, T_{sat} \leq \left[ T_f(z) + \frac{\phi}{h_{fo}} \right] \]  \hspace{1cm} (3.4)

where,

- \( T_{sat} \) – Saturation Temperature (K)
- \( T_w \) – Wall Temperature (K)
- \( T_f(z) \) – Bulk Temperature at Axial Position(z) (m)
- \( H_{fo} \) – Heat Transfer Coefficient (W/m\(^2\)K)
- \( \phi \) – Surface Heat Transfer (W/m\(^2\))

and

\[ T_{sat} \leq T_w, T_{sat} \leq \left\{ \phi \left[ \frac{4z}{G C_{pf} D} + \frac{1}{h_{fo}} \right] + T_{fi} \right\} \]  \hspace{1cm} (3.5)

where,

- \( z \) – axial co-ordinate (m)
- \( G \) – Mass Velocity (kg/m\(^2\)s)
- \( C_{pf} \) – Specific Heat of Liquid Phase (J/kgK)
- \( D \) – Pipe Diameter Characteristic Dimension (m)
- \( T_{fi} \) – Inlet Bulk Temperature (K)
These conditions were derived from Colliers derivation for the temperature profile for single-phase liquid heat transfer (Collier, 1972). The substitution of $(\Delta T_{\text{sub}})_i$ for $(T_{\text{sat}} - T_i)$ and $\Delta T_{\text{sub}(z)}$ for $(T_{\text{sat}} - T_{i(z)})$ gives the following relationship:

$$\Delta T_{\text{sub}(z)} \leq \frac{\phi}{h_f}$$  \hspace{1cm} (3.6)

and

$$(\Delta T_{\text{sub}})_i \leq \phi \left[ \frac{4z}{GC_{pf}D} + \frac{1}{h_f} \right]$$  \hspace{1cm} (3.7)

where,

$$(\Delta T_{\text{sub}})_i = \text{Inlet Subcooling (K)}$$

From inequality 3.7, the surface heat flux ($\phi$) can be plotted against inlet subcooling ($(\Delta T_{\text{sub}})_i$). The plot will result in a line with the following slope:

$$\left[ \frac{4z}{GC_{pf}D} + \frac{1}{h_f} \right]^{-1}$$  \hspace{1cm} (3.8)

From figure 3.4, it can be seen that the region below the line is the non-boiling region. Altering the mass velocity, length and diameter of the tube, and heat transfer coefficient can change the slope.
Figure 3.4 Surface heat flux versus inlet subcooling. Adopted from Collier, 1972.
The growth of the bubble can be derived by the temperature profile near the heated wall (Hsu and Ing, 1962). The bubble nuclei of \( r_c \) will grow only if the temperature exceeds the uniform temperature \( T_g \) given by

\[
T_g = \frac{RT_{\text{SAT}}}{i_{fg}} \ln(1 + \zeta) + T_{\text{SAT}}
\]  

(3.9)

where,

- \( T_g \) – Vapor Temperature (K)
- \( R \) – Radius of Bubble Cavity (m)
- \( i_{fg} \) – Latent Heat of Vaporization (J/kg)
- \( M \) – Molecular Weight (kg/kmol)

The parameter \( \zeta \) is defined by

\[
\zeta = \left( \frac{2\sigma}{p_f r_c} \right)
\]  

(3.10)

where,

- \( \sigma \) – Surface Tension (N/m)
- \( p_f \) – Pressure of Liquid Phase (N/m²)
- \( r_c \) – Cavity Mouth Radius (m)

The temperature gradient can be considered linear near a heated wall. The temperature profile can be represented as such:

\[
T_f(y) = T_w - \left( \frac{\phi_y}{k_f} \right)
\]  

(3.11)
where,

\[ y \] - Distance Measured from Boundary (m)

\[ k_l \] - Thermal Conductivity of Liquid Phase (W/mK)

Hsu postulated that the bubble would grow only if the lowest temperature of the bubble surface is greater than \( T_g \). The nucleation of the critical nucleus is considered as

\[ T_f(y) = T_g, \text{ and } \frac{dT_f(y)}{dy} = \frac{dT_g}{dr} \]  \hspace{1cm} (3.12)

With the conditions of eq. (3.12) and the graphical solutions of eq. (3.9) and (3.11), Bergles and Rohsenhow obtained the following equation (Bergles and Rohsenhow, 1963):

\[ (\Delta T_{SAT})_{ONB} = 0.556 \left[ \frac{\phi_{ONB}}{1082P^{1.156}} \right]^{0.463P^{0.034}} \]  \hspace{1cm} (3.13)

where,

\[ (\Delta T_{SAT})_{ONB} \] - Wall Superheat Necessary to Cause Nucleation (K)

\[ P \] - Wetted Perimeter (m)

This equation is valid for water only and in SI units.

Davis and Anderson (Davis and Anderson, 1966) covered an alternative solution. The following is the analytical solution:
\[
\frac{dT_f(y)}{dy} = -\frac{\phi}{k_f}
\]  
(3.14)

\[
\frac{dT_g}{dr} = -\frac{2RT_{SAT}^2 \sigma}{Mi_{fg} P_f r^2 (1 + \zeta)} \left[ 1 - \frac{RT_{SAT}}{Mi_{fg}} \ln(1 + \zeta) \right]^{-2}
\]  
(3.15)

where,

\( r \) – Bubble or Droplet Radius (m)

With the assumption that

\[
\left[ \frac{RT_{SAT}}{Mi_{fg}} \ln(1 + \zeta) \right]
\]

equation (3.15) can be reduced to the following:

\[
\frac{dT_g}{dr} = -\frac{B}{r^2 (1 + \zeta)}
\]
(3.16)

where

\[
B = \left[ \frac{2\sigma T_{SAT} v_{fg}}{i_{fg}} \right]
\]
(3.17)

\( v_{fg} \) – Difference in Specific Volumes of Saturated Liquid and Vapor (m³/kg)

From eq. (3.12), eqs. (3.15) and (3.16) can be equated to form the following equation:

\[
r_{CRIT} = -\frac{\sigma}{P_f} \left[ \left( \frac{\sigma}{P_f} \right)^2 + \left( \frac{Bk_f}{\phi} \right) \right]^{\frac{1}{2}}
\]
(3.18)
where,

\[ r_{\text{crit}} \] - critical bubble radius (m)

In order to obtain the equation for the superheat required to initiate nucleate boiling, \( r_{\text{crit}} \) has to be substituted for \( r \) and eq. (3.11) for \( T_g \) in eq. (3.9). The result of the substitution is as follows:

\[
(\Delta T_{\text{SAT}})_{\text{ONB}} = \frac{\left[\left(\frac{RT_{\text{SAT}}^2}{M_i_f}\right)\ln(1 + \zeta_{\text{CRIT}}) / M_i_f\right]}{1 - \left(\frac{RT_{\text{SAT}}}{M_i_f}\right)\ln(1 + \zeta_{\text{CRIT}})} + \frac{\phi_{\text{ONB}} r_{\text{CRIT}}}{k_f}
\]  

(3.19)

where,

\[
\zeta_{\text{CRIT}} = \left(\frac{2\sigma}{p_f r_{\text{CRIT}}}\right)
\]  

(3.20)

Now, if \( \zeta \ll 1 \), eq. (3.16) can be approximated to be

\[
\frac{dT_g}{dr} = -\frac{B}{r^2}
\]  

(3.21)

Under this condition, eq. (3.18) becomes

\[
r_{\text{CRIT}} = \sqrt{\frac{B k_f}{\phi}}
\]  

(3.22)

and eq. (3.19) becomes

\[
(\Delta T_{\text{SAT}})_{\text{ONB}} = \frac{B}{r_{\text{CRIT}}} + \frac{\phi_{\text{ONB}} r_{\text{CRIT}}}{k_f}
\]  

(3.23)
With the appropriate substitutions, the equation for predicting the onset of nucleation can be expressed as

\[
(\Delta T_{\text{SAT}})_{\text{ONB}} = \left[ \frac{8 \sigma \phi_{\text{ONB}} T_{\text{SAT}}}{i_f k_f \rho_g} \right]^{0.5}
\]  

(3.24)

Equation (3.19) and (3.24) are for the prediction for onset of nucleation, but they are limited to velocities up to 17.5 m/s at low pressures.

Figure 3.5 is a curve that predicts the wall superheat when ONB will occur at any given heat flux. This figure was constructed using equation (3.24), which was previously derived. This figure 3.5 correlates very well with figure 5.5. With the combination of the fore mentioned figures, required heat flux and minimum cavity sizes, ONB can be predicted for any given wall superheat. This curve may not be to accurate for low wall superheats, because the curve was created using intervals of 10 kW/m². A higher resolution may be needed for low wall superheats.
Figure 3.5 Prediction of Wall Superheat for ONB.
This study utilizes a pre-existing experimental setup. The apparatus that was used for this study was built in 1996, when Howell wrote “Investigation of Nucleation and Heat Transfer During Subcooled Flow Boiling on Augmented Surfaces” (Howell, 1996). Since this construction, many other students have used the apparatus to study the effects of different surfaces on heat transfer characteristics.

4.1 Experimental Setup

The test set-up employed for this study is shown in Figure 4.1. The water is heated to 95°C in a water circulation bath and flows to a flow meter. After the water passes through a flow meter, it passes through the flow channel where the heater element is located. The interface between the flowing water and the heater surface can be observed using a microscope. Figure 4.2 is an exploded view of the region where data is acquired. A camera and a computer were used to capture the reaction at the interface, via a microscope. The flow in the channel is subjected to atmospheric pressure. After the water passes through the flow channel, it returns back to the water circulation bath, the bulk temperature is maintained at 95°C and circulated again.
Figure 4.1 Experimental Setup

Figure 4.2 Data Acquisition Region
4.2 Experimental Equipment

4.2.1 Water Circulation Bath
The water was heated and circulated by a Brinkman RC-20 system. It is equipped with a heater and circulation pump. The bath is capable of maintaining the water temperature within 0.02 K.

4.2.2 Flow Channel
The flow channel was constructed using 2024-T3 aluminum. It is a 3mm X 40mm rectangular flow channel. The length of the flow channel is approximately 400mm. The polycarbonate window was installed to observe the interaction between the test specimen and the flowing water. A piece of torlon was used as bushing for the test specimen as well as an insulator. In order to maintain atmospheric pressure within the flow channel, the flow channel is equipped with control valves at the inlet and outlet of the test section. A manometer was installed near the nucleation site section of the flow channel to help determine when atmospheric pressure is achieved.

4.2.3 Heater
The heater consists of a structured surface, resistance heater, and a power source. The specimen started out as a 38mm stock 6061-T6 aluminum. After the laser etching was performed on the test specimen, it was machined to the dimensions shown on figure 4.3.
A Watlow circumferencial electrical resistance heater supplied the heat. It was wrapped around the base of the specimen. The orientation of the heater is also shown on figure 4.3

A constant heat flux was provided for this study. An Electronics Measurement Inc. power supply supplied the power to the heater. The power supply is capable of supplying the power in 0.1 V or 0.1 amps increments.

4.2.4 Thermocouples
In order to collect data, E type thermocouples were utilized. The thermocouples were positioned along the length of the 9.5mm rod. Figure 4.3 shows the positions of the thermocouples. One of the thermocouples was placed in the flow channel after the heater surface. This thermocouple was used to determine the bulk temperature.

4.2.5 Thermocouple Scanner
A thermocouple scanner was connected to the thermocouples in order to collect data. A digital Keithley 740 System Scanning Thermometer was the choice of thermocouple scanner. This scanner has the capability to read 10 channels and use multiple types of thermocouples per test. The precision of the scanner is within 0.1°C.
Figure 4.3 Heater with dimensions and thermocouple spacing
4.2.6 Bubble Viewer and Video Capture

Looking at the surface of the heater was essential to determine when ONB occurred. The bubbles are hard to see with the naked eye. In order to compensate for this disadvantage, a Mitutoyo WF microscope was implemented. The microscope is capable of 125X, 250X, and 500X magnification. With these magnifications, it was easy to determine when ONB or NVG occurred.

A Hitachi KP-C501 solid state camera was mounted on top of the microscope. The camera was attached to the computer where images could be captured and saved. The computer was installed with Image-Pro Plus™ software, which is an image capture and analysis software package.

4.2.7 Flowmeter

This study observed heat transfer characteristics as a function of the flow rate. The procedure section explains in more detail about the different flow rates. An Omega FL-1503A rotameter was used to set the flow at different flow rates. The flowmeter is capable of measuring flow up to 2.53 GPM. The precision of the flowmeter is 0.0253 GPM at maximum flow.
4.3 Experimental Procedure

4.3.1 Preparation of the Test Specimen

This work is a study of the boiling enhancement with the laser-etched surfaces. A set of three 38mm stock aluminum pieces was used. They were machined at the R.I.T. machine shop. The surface was polished with the polishing wheel, using 5.0μm size particles. Then, laser-etching on the surfaces was performed by a vendor. The matrix of the laser-etching is shown on table 4.1.

<table>
<thead>
<tr>
<th>Surface Type</th>
<th>Spacing (μm)</th>
<th>Hole Diameter (μm)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Surface 1</td>
<td>58.4</td>
<td>9.0</td>
</tr>
<tr>
<td>Surface 2</td>
<td>38.6</td>
<td>3.7</td>
</tr>
<tr>
<td>Surface 3</td>
<td>58.4</td>
<td>3.7</td>
</tr>
<tr>
<td>Polished</td>
<td>N/A</td>
<td>N/A</td>
</tr>
</tbody>
</table>

Table 4.1 Matrix of laser-etched surfaces

The specimen was returned with the different cavity sizes and spacings. They were taken to the machine shop and turned down using a lathe to obtain the appropriate dimensions. The holes for the thermocouples were drilled using a mill. The holes were drilled 4.75 mm deep. Figure 4.3 shows the dimensions and the spacing for the thermocouples. In order to fit the specimen into the torlon bushing without breaking the thermal couple wire, a channel was milled using a 5/32” end mill running from T2 to T3. The thermal couple wire fit into the channel. The length of the channel varied with each specimen, but half the distance from T2 to T3 was used as a rule of thumb.
Next, E type thermocouples were made using a thermal couple welder. The thermal couple wire was cut in five foot sections before making the thermal couple connections. Long lengths were required in order to route away from any source of heat.

Once the thermocouple connections were made, the thermocouples were ready to be placed in the holes, T2 – T5. It was determined that two part epoxy was adequate to keep the thermocouples in the holes. The thermocouple was held in place while the epoxy was carefully placed in the holes. The holes were filled with the epoxy so that it was level with the rod. Extra epoxy was avoided, because it would interfere with the bushing when placing the specimen in the apparatus.

The specimen was placed in the flow channel. The clearance between the torlon insulating bushing and the specimen was very tight. It was critical that the thermocouple remained in the holes as it was forced in the bushing. The channel that was machined into the rod was very useful for this process. Room temperature vulcanizing caulkling was smeared on and around the rod under T3. The rod was then forced into the bushing. It was helpful if the specimen was twisted as it was forced into the tight spot. The specimen was placed so that the surface was flush with the floor of the flow channel.
The thermocouple was then wired to the thermocouple scanner. Some of the channels were not working well. Only the channels that worked were employed. The scanner was set for E type thermocouples and to read in °C. This concludes the thermocouple preparation.

4.3.2 Data Acquisition

Though the amount of time needed for actual acquisition is extremely small, the test required a significant amount of time. It is estimated that one run of the test requires at least eight hours of total time. The data was read every thirty minutes.

The first part of data collection is to warm up the bath water. The bath was set at 95°C. The water should not be boiling in the water bath. Once the water circulation was set, the flow rate was set at 10% of the maximum flow (2.53 GPM). The experimental setup was left alone for three hours to let the system reach steady state.

Once the setup reached steady state, it was ready to actually collect data. At this time the power supply was set at twenty volts. It took around thirty minutes to reclaim steady state. At steady state, the thermocouples were read and the temperatures at T2 – T5, time, volts, and amperes were recorded. Power was increased at five-volt increments until fifty volts was reached. During the data collection, the microscope was utilized to determine when NVG occurred.
The data acquisition part of the procedure was repeated for the 30% flow. As for the entire procedure, it was implemented for the other specimen. The procedure was repeated four times, one for every specimen.
4.4 Data Reduction

The raw data was a matrix of temperatures, and is shown in table 4.2.

<table>
<thead>
<tr>
<th>Volts</th>
<th>Amps</th>
<th>T2 °C</th>
<th>T3 °C</th>
<th>T4 °C</th>
<th>T5 °C</th>
<th>T6 °C</th>
<th>Time</th>
</tr>
</thead>
<tbody>
<tr>
<td>20</td>
<td>0.29</td>
<td>90.3</td>
<td>91.5</td>
<td>92.8</td>
<td>94.0</td>
<td>84.2</td>
<td>18:30</td>
</tr>
<tr>
<td>25</td>
<td>0.36</td>
<td>96.2</td>
<td>98.8</td>
<td>102.0</td>
<td>105.4</td>
<td>84.4</td>
<td>18:50</td>
</tr>
<tr>
<td>30</td>
<td>0.43</td>
<td>99.7</td>
<td>104.2</td>
<td>109.5</td>
<td>114.9</td>
<td>84.2</td>
<td>19:10</td>
</tr>
<tr>
<td>35</td>
<td>0.50</td>
<td>105.1</td>
<td>112.0</td>
<td>120.5</td>
<td>129.0</td>
<td>84.3</td>
<td>19:30</td>
</tr>
<tr>
<td>40</td>
<td>0.56</td>
<td>109.2</td>
<td>119.4</td>
<td>132.3</td>
<td>147.2</td>
<td>85.3</td>
<td>19:50</td>
</tr>
<tr>
<td>45</td>
<td>0.63</td>
<td>111.2</td>
<td>124.3</td>
<td>140.9</td>
<td>159.0</td>
<td>84.8</td>
<td>20:10</td>
</tr>
<tr>
<td>50</td>
<td>0.70</td>
<td>113.6</td>
<td>130.3</td>
<td>151.4</td>
<td>174.5</td>
<td>84.7</td>
<td>20:30</td>
</tr>
<tr>
<td>40</td>
<td>0.56</td>
<td>108.2</td>
<td>118.2</td>
<td>130.6</td>
<td>144.7</td>
<td>84.3</td>
<td>20:50</td>
</tr>
<tr>
<td>30</td>
<td>0.42</td>
<td>100.1</td>
<td>105.3</td>
<td>111.5</td>
<td>118.1</td>
<td>84.2</td>
<td>21:10</td>
</tr>
<tr>
<td>20</td>
<td>0.28</td>
<td>90.7</td>
<td>92.2</td>
<td>93.8</td>
<td>95.6</td>
<td>84.1</td>
<td>21:30</td>
</tr>
</tbody>
</table>

Table 4.2 Sample of raw data.

The heat flux was calculated first. The following equation was used to obtain the heat fluxes between each of the nodes:

\[ q'' = \frac{k}{x} \Delta T \]  

(4.1)

The distance (x) can be obtained from figure 4.3. Thermal conductivity of the aluminum can be found in a table of properties. The ΔT is simply the difference of temperature of the two nodes. The temperature of the surface was extrapolated using the heat flux of the other nodes. Linear extrapolation was used to determine the surface temperature.
The surface temperature was obtained by rearranging eq. (4.1). After the rearranging the equation the following equation was used to obtain the surface temperature:

\[ T_s = T_2 - \frac{q''_s}{k} \times \frac{\Delta T}{k} \] (4.2)

The heat transfer coefficient was obtained by manipulating the following equation:

\[ q''_s = h\Delta T \] (4.3)

After manipulating eq. (4.2) so that the heat transfer coefficient can be obtained, it becomes

\[ h = \frac{q''_s}{\Delta T} \] (4.4)

where \( \Delta T \) is the difference between the \( T_{\text{surface}} \) and \( T_{\text{bulk}} \).

The wall superheat is the difference between the wall temperature and the saturation temperature of water. The equation for wall superheat can be expressed as follows:

\[ \Delta T = T_w - T_{\text{sat}} \] (4.5)

where ,

\( \Delta T \) – Wall Superheat (K)

The subcooled temperature is the difference between the saturation temperature of water and the temperature of the bulk water temperature. The bulk water
temperature is always below the saturation temperature for this case. The equation for subcooled temperature can be expressed as follows:

\[ \Delta T_{\text{sub}} = T_{\text{sat}} - T_w \]  \hspace{1cm} (4.6)

where,

- \( T_w \) – Wall Temperature (K)
- \( T_{\text{sat}} \) – Saturation Temperature (K)
- \( \Delta T_{\text{sub}} \) – Subcooled temperature (K)

The Reynolds number (Re) is a non-dimensional parameter encountered in fluid dynamics. When Re \( \leq 2300 \) in a pipe, the pipe flow is considered laminar, and pipe flow is considered turbulent for larger values. The Reynolds number is calculated using the following equation:

\[ \text{Re} = \frac{\rho v d_h}{\mu} \]  \hspace{1cm} (4.7)

where,

- \( \rho \) – Density of the fluid (kg/m\(^3\))
- \( v \) – the flow velocity (m/s)
- \( d_h \) – Hydraulic diameter (m)
- \( \mu \) - Viscosity of the fluid (Ns/m\(^2\))

Because the flow channel is rectangular, hydraulic diameter \( (d_h) \) will be substituted for \( d \). The equation for \( d_h \) is as follows:

\[ d_h = \frac{4A_c}{P_w} \]  \hspace{1cm} (A.2)
where,

\[ A_c - \text{Cross sectional area (m}^2) \]

\[ P_w - \text{Wetted perimeter (m)} \]

The following is a table of the Reynolds Numbers for the corresponding flow and bulk temperature:

<table>
<thead>
<tr>
<th>Bulk Water Temperature</th>
<th>10% Flow</th>
<th>30% Flow</th>
</tr>
</thead>
<tbody>
<tr>
<td>80°C</td>
<td>2055</td>
<td>6165</td>
</tr>
<tr>
<td>90°C</td>
<td>2291</td>
<td>6873</td>
</tr>
<tr>
<td>95°C</td>
<td>2418</td>
<td>7254</td>
</tr>
</tbody>
</table>

Table 4.3 Matrix of Reynolds Numbers

All data reduction was performed using Excel spreadsheet.
5. EXPERIMENTAL RESULTS AND DISCUSSION

5.1 Results and Discussion

The experiments were conducted at two flow rates and subcooled temperatures resulting in the Reynolds number range between 2055 and 7254. The corresponding flow rates and subcooled temperatures are 0.253 and 0.759 GPM, and 5 and 10K, respectively. The results fall into three comparisons. The comparisons are heat transfer characteristics as a function of different surfaces, subcooled temperatures and Reynolds number.

Figure 5.2 shows the data in 5.1 replotted as heat transfer coefficient versus wall superheat. The comparisons in literature compare heat transfer coefficient rather than heat flux. This work will compare the heat transfer coefficients.
Figure 5.1 Heat Flux at 5K Subcooling with Re=2418
Re=2418, Subcooled=5K

Wall Superheat, $\Delta T$ (K)

- Surface 1
- Surface 2
- Surface 3
- Polished

Expon. (Surface 1)
Expon. (Surface 2)
Expon. (Surface 3)
Expon. (Polished)

Figure 5.2 Heat Transfer Coefficient at Subcooling at 5K with Re=2418 and Corresponding Trendlines
5.1.1 Surface Comparison

Figure 5.2 shows the heat transfer coefficient versus wall superheat plot for the four surfaces at subcooling at 5K and Reynolds number of 2418. The curves merge into a single line at low wall superheat values. Here the heat transfer is by single phase mechanism only as the wall temperature is below the value required to cause nucleation. As the wall temperature increases, the performance of all three enhances surfaces is better than the polished surface. Performance for Surface 2 and 3 are close to each other, while Surface 1 performs much better than both. Surface 2 does outperform Surface 3 by a small amount.

The single phase heat transfer coefficient values for the four surfaces at low superheat (prior to onset of boiling) are nearly equal at around 4000 W/m²K. An approximate value for onset of nucleate boiling can be obtained by looking at figure 5.2. For the plain surface, ONB is seen to occur around 12K, while for surfaces 2 and 3, it occurs at around 5-6K. For surface 1, ONB occurs earlier, at around 3-4K. The steep vertical trend in the data for water and surface 1 indicates the approach to the fully developed boiling.
5.1.2 Subcooled Comparison

Figure 5.3 shows the heat transfer coefficient versus wall superheat for the enhanced surfaces at a subcooling of 10K and Reynolds number of 2291. The superiority of the performance of Surface 1 is observed in figure 5.3. The difference in the subcooling does not effect the order of performance of the enhanced surfaces. Again, the order of performance is Surface 1, 2, then 3.

The subcooled did separate the performance between Surface 2 and 3. There is a distinct increase in performance in Surface 2 over Surface 3. This distinction was not observed in figure 5.2. Surface 2 had higher performance, because there were more nucleation sites than Surface 3.

By observing the heat transfer coefficients at 7.5K Wall Superheat in figure 5.2 and 5.3, the increased performance of all three surfaces in figure 5.3 over figure 5.2 can be observed. For high values of subcooling, bubble growth was slow due to the rapid condensation occurring on top of the bubble (Howell, 1996).
Figure 5.3 Heat Transfer Coefficient at Subcooling at 10K with Re=2291 and Corresponding Trendlines
5.1.3 Reynolds Number Comparison

Figure 5.4 shows the heat transfer coefficient versus wall superheat plot for the four surfaces at a subcooling of 5K and Reynolds number of 7254. By comparing figure 5.4 and 5.2, the enhancement in heat transfer coefficient due to forced convection can be observed in figure 5.4. The single-phase heat transfer coefficient is around 5000 W/m²K for Reynolds number of 7254. There is an increase of 1000 W/m²K in performance over the single phase heat transfer coefficient with Reynolds number of 2418.

A larger convection heat transfer is associated with a higher Reynolds number. Figure 5.4 shows that the heat transfer coefficients are tighter than in figure 5.2. The convection heat transfer is much more influential in figure 5.4 than the nucleation heat transfer. As expected, the plots are converging to a fully developed boiling curve.

Like figure 5.2, the performance of Surfaces 2 and 3 are nearly identical, while the performance of Surface 1 is superior to all the other surfaces. The performance of all three enhances surfaces resulted in better heat transfer characteristics than the polished surface.

The curves merge into a single line at low superheat values. Among all three figures the merging curves at low superheat values is consistent. This phenomenon is due to the absence of nucleation at low superheat values. The
heat transfer is strictly contributed by convection. Once the curve experiences ONB, the separation in performance can be observed. This phenomenon can be observed in figures 5.2-5.4.
Figure 5.4 Heat Transfer Coefficient at Subcooling at 5K with Re=7254 and Corresponding Trendlines
5.1.4 Nucleation Criterion

To see the effect of nucleation characteristics, the nucleation criterion given by Hsu and Graham (1961) and written in the following form is utilized.

\[ r_{\text{max}}, r_{\text{min}} = \frac{\delta_t}{2} \left[ \frac{\Delta T_{\text{sat}}}{\Delta T_{\text{sat}} + \Delta T_{\text{sub}}} \pm \sqrt{\left( \frac{\Delta T_{\text{sat}}}{\Delta T_{\text{sat}} + \Delta T_{\text{sub}}} \right)^2 - \frac{8\sigma T_{\text{sat}} v_{l_0}}{i_{l_0} \delta_t (\Delta T_{\text{sat}} + \Delta T_{\text{sub}})}} \right] \] (5.1)

\( r_{\text{max}}, r_{\text{min}} \) are the maximum and minimum cavity radii which satisfy the nucleation criterion, and \( \delta_t \) is the thermal boundary layer thickness which is assumed to be \( k/h \). The results are plotted in figure 5.5. The minimum and maximum cavity diameters are plotted against the wall superheat. For a given wall superheat, the region between the two curves represents the range of active cavity diameters satisfying the nucleation criterion. Two separate plots are shown in figure 5.5 for the two flow rate cases, both at 5K subcooling. The two plots lie close to each other.

The nucleation characteristics for the heater surfaces can be compared using the plot shown in figure 5.5. For Surface 1, the cavity diameter is 9.0 \( \mu m \), which corresponds to a wall superheat of 4K. For Surfaces 2 and 3, the corresponding wall superheat for nucleation corresponding to 3.7 \( \mu m \) is about 7K. These values compare very well with the observed wall superheat at ONB for theses in figures 5.2 – 5.4 as discussed in the preceding paragraphs. Clearly, having the cavities in the correct size range is seen to improve the nucleate boiling heat transfer by
Figure 5.5 Active Cavity Sizes for Experimental Conditions

- $h_L = 3900$, $Re = 2418$, Subcooled = 5K
- $h_L = 5000$, $Re = 7254$, Subcooled = 5K
providing more active sites. For the conditions of this test, Surface 1 seems quite close to the maximum cavity diameter. However, at higher wall temperatures smaller cavities will also be activated, and proper “mix” of cavities providing good nucleation behavior at low wall superheats, and high heat transfer (approaching fully developed boiling at high heat fluxes) is desirable.

5.1.5 Trendline Equations

Table 5.1 shows the test conditions along with the trendline for heat transfer coefficient. Exponential trend resulted with the highest $R^2$ values. All the $R^2$ values are greater than 0.95 except the trendline for polished surface with 5K subcooling at Reynolds number of 2418. The value for the fore mentioned surface is 0.63. The trendline is not a good fit to the actual curve.

This table is a good summary of the results. From the table, it can be observed that the combination of greater Reynolds number and subcooling results in better single phase heat transfer characteristics.
<table>
<thead>
<tr>
<th>Surface Type</th>
<th>Re</th>
<th>Subcooled (K)</th>
<th>Trendline Equation</th>
<th>$R^2$</th>
</tr>
</thead>
<tbody>
<tr>
<td>Subcooled</td>
<td>2418</td>
<td>5</td>
<td>$y=3.3454e^{0.1244x}$</td>
<td>0.9638</td>
</tr>
<tr>
<td>Subcooled</td>
<td>2418</td>
<td>5</td>
<td>$y=3.0864e^{0.0996x}$</td>
<td>0.9844</td>
</tr>
<tr>
<td>Subcooled</td>
<td>2418</td>
<td>5</td>
<td>$y=3.3199e^{0.0833x}$</td>
<td>0.9525</td>
</tr>
<tr>
<td>Polished</td>
<td>2418</td>
<td>5</td>
<td>$y=3.8931e^{0.63x}$</td>
<td>0.6313</td>
</tr>
<tr>
<td>Polished</td>
<td>7254</td>
<td>5</td>
<td>$y=5.6997e^{0.6997x}$</td>
<td>0.9913</td>
</tr>
<tr>
<td>Polished</td>
<td>7254</td>
<td>5</td>
<td>$y=5.2223e^{0.0751x}$</td>
<td>0.9864</td>
</tr>
<tr>
<td>Polished</td>
<td>7254</td>
<td>5</td>
<td>$y=4.7734e^{0.0802x}$</td>
<td>0.9928</td>
</tr>
<tr>
<td>Polished</td>
<td>7254</td>
<td>5</td>
<td>$y=5.341e^{0.0579x}$</td>
<td>0.9774</td>
</tr>
<tr>
<td>Polished</td>
<td>7254</td>
<td>10</td>
<td>$y=3.6298e^{0.1309x}$</td>
<td>0.9765</td>
</tr>
<tr>
<td>Polished</td>
<td>7254</td>
<td>10</td>
<td>$y=4.1189e^{0.0937x}$</td>
<td>0.9836</td>
</tr>
<tr>
<td>Polished</td>
<td>7254</td>
<td>10</td>
<td>$y=3.5159e^{0.0804x}$</td>
<td>0.9929</td>
</tr>
<tr>
<td>Polished</td>
<td>6873</td>
<td>10</td>
<td>$y=7.6594e^{0.0966x}$</td>
<td>0.9874</td>
</tr>
<tr>
<td>Polished</td>
<td>6873</td>
<td>10</td>
<td>$y=6.547e^{0.0967x}$</td>
<td>0.9977</td>
</tr>
<tr>
<td>Polished</td>
<td>6873</td>
<td>10</td>
<td>$y=5.9753e^{0.0441x}$</td>
<td>0.9966</td>
</tr>
</tbody>
</table>

Table 5.1 Trendline Equations
6. EXPERIMENTAL ERROR

The error analysis is based on the Journal of Heat Transfer “Policy on Reporting Uncertainties in Experimental Measurements and Results” (Kline, 1985). Each represented result should include the precision limit, bias limit and uncertainty. The precision limit is the estimate of the lack of repeatability caused by random error and unsteadiness. The bias limit is the estimate of the magnitude of the fixed, constant error. The uncertainty is the nominal result in which the experimenter is 95% confident. The uncertainty calculation is as follows:

\[ U = \left( B^2 + P^2 \right)^{1/2} \]  \hspace{1cm} (6.1)

where,

- \( U \) is the uncertainty
- \( P \) is the precision limit
- \( B \) is the bias limit.

Table 6.1 is the table of bias limits and precision limits. These values were employed to obtain the uncertainty associated with the experimental results.

Figure 6.2 is the graphical representation of the uncertainty analysis. The uncertainty is much higher near the single-phase region. This can be attributed to the small temperature difference values. The region where boiling occurs the uncertainty is in the range of 3-10 percent. This value is important, because it is the region of interest for this investigation.
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<table>
<thead>
<tr>
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</thead>
<tbody>
<tr>
<td><strong>Bias Limit</strong></td>
<td></td>
</tr>
<tr>
<td>Temperature measurement</td>
<td>±0.1K</td>
</tr>
<tr>
<td>Saturation temperature-</td>
<td>±0.2K</td>
</tr>
<tr>
<td>Thermocouple location-</td>
<td>±0.25mm</td>
</tr>
<tr>
<td>Flow velocity measurement</td>
<td>±0.006 m/s</td>
</tr>
<tr>
<td>Water temperature measurement</td>
<td>±0.1K</td>
</tr>
</tbody>
</table>

*Precision limits derived from 50 samples*

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<tbody>
<tr>
<td>Heater thermocouples-</td>
<td>±0.2K</td>
</tr>
<tr>
<td>Water temperature-</td>
<td>±0.025K</td>
</tr>
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</table>
Figure 6.1 Uncertainty associated with the experimental results
The flow boiling performance of three surfaces with different hole sizes and patterns generated using laser drilling was obtained and compared with a plain surface. Subcooled water at atmospheric pressure was used as the test fluid. The setup allowed localized measurement of heat transfer under flow boiling conditions. From the results of this investigation, following conclusions were drawn:

- All the specially etched surfaces (Surface 1 - 9.0μm diameter holes spaced at 58.4μm, Surface 2 - 3.7μm diameter holes spaced at 38.6μm, Surface 3 - 3.7μm diameter holes spaced at 58.4μm) outperformed the polished surface by at least 50%.

- The results for ONB for the three enhanced surfaces are found to be in agreement with the nucleation criterion given by Hsu and Graham (1961). The experimentally observed wall superheat at ONB for the surfaces is quite close to the predicted values from the nucleation criterion.

- The surface with the best heat transfer characteristics is Surface 1. Figure 5.2 shows a 200% increase in the heat transfer coefficient for approximately 9°C superheat. This surface had the largest cavity size among the surfaces tested. This is in line with observations made from nucleation criteria shown figure 5.2.
• The flow with the higher subcooled water has better heat transfer characteristics. At approximately 7°C superheat, the heat transfer coefficient for 10K subcooling is 13% higher than 5K subcooling.

• Reynolds number is a great influence on heat transfer in the range of wall superheats tested in this investigation. At approximately 8°C, the heat transfer coefficient for Re=7254 is 33% higher than Re=2418.

• All data is believed to be in the partial boiling region.

• There is potential for laser-etched surfaces for future study.

• Proper selection of hole sizes and spacing is recommended for each fluid under a set of operation conditions.
8. References


APPENDIX A

DRAWING OF THE SURFACE

*(not drawn to scale)
APPENDIX B
PICTURES OF THE SURFACES AT 20 X MAGNIFICATION

Surface 1
Surface 3

50μm
APPENDIX C

PICTURES OF THE SURFACES AT 50 X MAGNIFICATION

Surface 1
Surface 2

50µm
Surface 3

50 µm