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On Contact Line Region Heat Transfer, Bubble Dynamics and Substrate Effects during Boiling

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On Contact Line Region Heat Transfer, Bubble Dynamics and Substrate Effects during Boiling

by

Pruthvik A. Raghupathi

A dissertation submitted in partial fulfilment of the requirements for the degree of Doctor of Philosophy in Engineering

Engineering (PhD Program)
Kate Gleason College of Engineering

Rochester Institute of Technology
Rochester, New York
April, 2018
On Contact Line Region Heat Transfer, Bubble Dynamics and Substrate Effects during Boiling

by

Pruthvik A. Raghupathi

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Abstract

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**Degree:** Doctor of Philosophy  
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**Author:** Pruthvik A. Raghupathi  
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Rapid advancement of electronics used in domestic, commercial and military applications has necessitated the development of thermal management solutions capable of dissipating large amounts of heat in a reliable and efficient manner. Traditional methods of cooling, including air and liquid cooling, require large fluid flow rates and temperature differences to remove high heat fluxes and are therefore unsuited for many advanced applications. Phase change heat transfer, specifically boiling, is capable of dissipating large heat fluxes with low temperature gradients and hence is an attractive technique for cooling high heat flux applications. However, due to the complex interactions between the fluid dynamics, heat transfer, and surface chemistry, the fundamental physics associated with boiling is not completely understood.

The focus of this work is to get a better understanding of the role played by a nucleating bubble in removing the heat from the substrate. The interfacial forces acting on a bubble, contact line motion, and the thermal interaction with the heater surfaces are some of the important considerations which have not been well understood in literature. The work reported in this dissertation is divided into three parts. In the first part, an analytical study of the effect of evaporation momentum force on bubble growth rate and bubble trajectory was conducted. It was shown that the trajectory of a bubble can be controlled by creating an asymmetric temperature field. This understanding was used to develop a bubble diverter that increased the Critical Heat Flux (CHF) over a horizontal tubular surface by 60% and improved the heat transfer coefficient by 75%. In the second part of the work, additional contact line regions were generated using microgrooves. This enhancement technique increased the CHF with water by 46% over a plain copper surface to 187 W/cm². Finally, the effect of the heater properties and surface fouling during boiling was evaluated. This included a study on the effect of thermophysical properties of the heater surface on CHF and an investigation of fouling over a heater surface during boiling of seawater.
Acknowledgement

There are several people whose help and support was critical in completing this dissertation. On top of that list would be my advisor Dr. Satish Kandlikar (better known to those of us in the lab as Dr. K). In addition to his deep knowledge of the field and creativity with which he tackled challenges what made interactions with him enjoyable was the energy and enthusiasm with which he approached each day. The positive work culture that he cultivated in the lab helped in making the lab feel like a second home and made it possible to work late into the night on various projects. I would like to thank my committee members- Dr. Robert Stevens, Dr. Michael Schertzer, and Dr. Jiandi Wan for their feedback and encouragement. I am grateful to the director of the Engineering PhD program- Dr. Edward Hensel for his constant guidance and support.

I am thankful to all the past and present members of the Thermal Analysis, Microfluidics and Fuel Cell Lab for helping me with my work and for their friendship. I would particularly like to thank Rupak, Ankit, Arvind, Alyssa, Isaac, Travis and Aranya for being wonderful lab mates. I would also like to thank Austin, Brittany, Gia, and Jean-Patrick for helping with fabrication and running the tests.

Finally, I would like to thank my dad, mom and sister for being the pillars of strength and support all my life. Your unconditional love and confidence in me really helped me get through the all challenges I faced.
Dedication

To my Mom and Dad
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<td>A</td>
<td>Parameter defined by Mikic et al.[1]</td>
<td>m/s</td>
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<tr>
<td>A*</td>
<td>Hamaker constant</td>
<td>J</td>
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<td>Aₚ</td>
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<td>Bₓ</td>
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<td>Kinematic viscosity</td>
<td>m²/s</td>
</tr>
<tr>
<td>$\Pi$</td>
<td>Disjoining pressure</td>
<td>N/m²</td>
</tr>
<tr>
<td>$\eta$</td>
<td>Evaporative mass flux</td>
<td>kg/m²</td>
</tr>
<tr>
<td>$\rho$</td>
<td>Density</td>
<td>kg/m³</td>
</tr>
<tr>
<td>$\sigma$</td>
<td>Surface tension</td>
<td>N/m</td>
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<tr>
<td>$\hat{\theta}$</td>
<td>Accommodation coefficient</td>
<td></td>
</tr>
<tr>
<td>$\phi$</td>
<td>Angle of inclination of the heater</td>
<td>°</td>
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Chapter 1

1.0 Introduction

Boiling is a phase change process in which vapor bubbles are formed on a heated surface and/or in a superheated liquid. Boiling has been used as a mode of heat transfer for thousands of years and has now become ubiquitous with a wide variety of domestic and industrial applications including electricity generation in power plants, heating and cooling, desalination and dissipation of high heat fluxes from electronic devices among others. Extensive research conducted over the last five decades using state of the art tools for measurement and imaging have helped develop a better understanding of the fundamentals of boiling and in turn have helped develop more efficient heat transfer surfaces. Specifically, research over the past decade has been largely focused on developing boiling surfaces that are able dissipate large amounts under both static and mobile conditions. Even after being the focus of many years of research the physics of boiling heat transfer is yet to be mastered due to the complex interaction between fluid dynamics, heat transfer and surface chemistry. Understanding the effect of bubble dynamics, heat transfer in the contact line region and the effect of surface properties and morphology on boiling and applying that understanding to develop surface enhancements will be the focus of this thesis.

1.1 Background

The two main modes of heat transfer between a solid and a liquid are single phase and two phase heat transfer. Single phase heat transfer is a process where heat from the substrate only results in an increase in the temperature of the liquid. This increase in temperature is dependent on the specific heat of the liquid. The heat transfer coefficient, a measure of the efficiency of the heat transfer process, for single phase heat transfer is limited since the temperature of the liquid at the
solid-liquid boundary increases with time creating a resistance for heat transfer. Under two phase heat transfer, in addition to sensible heating, heat from the substrate evaporates the liquid into vapor. This process is far more efficient than single phase heat transfer since the heat required to evaporate the liquid, known as latent heat, is significantly greater than the specific heat. The temperature rise on the bulk liquid is also limited under two phase heat transfer since liquid begins to evaporate beyond saturation temperature. Boiling defers from evaporation at a preexisting liquid-vapor interface as it involves generation of these interfaces at discrete sites.

Based on the motion of the liquid relative to the heater surface, boiling can be classified as pool boiling and flow boiling. Pool boiling refers to the boiling under natural convection conditions where the bulk liquid is quiescent relative to the heater surface, while flow boiling refers to boiling while the bulk liquid moves over the heater surface. In case of pool boiling the growth and departure of bubbles is the main reason for liquid motion and in flow boiling the presence of an external force also aids in the mixing of the liquid.

1.1.1 Boiling Curve

In pool boiling, the temperature of the heater surface and the heat flux over the heater are directly related, with surface temperature increasing as the heat flux across the surface increases. Broadly, there are two types of boiling systems: a heat flux controlled system and a temperature controlled system. In a heat flux controlled system, the amount of power provided to the heater is regulated and the surface temperature is dependent on the heat flux input to the system. In a temperature controlled system, the heater surface temperature is directly controlled, typically by having a secondary heat exchange process on the other side of the heater surface. The relationship between the heat flux over the surface and the surface temperature was first studied by Nukiyama [5]. A plot of the heat flux (q’’) and the wall superheat (ΔT_w), known as the boiling curve, is used to
identify the various boiling regimes. Figure 1 shows a typical pool boiling curve for a heat flux controlled horizontal plate or wire. Heat flux is the rate of heat transfer per unit area and the wall superheat is the difference between the heater surface temperature and the saturation temperature of the liquid corresponding to the operating temperature. The red arrows indicate the plot traced as the heat flux is increased and the blue arrows indicate the plot traced as the heat flux is reduced. Five different regimes can be identified in boiling and they are

i) Natural convection

ii) Partial nucleate boiling

iii) Fully developed nucleate boiling

iv) Transition boiling

v) Film boiling
As the input heat flux to the surface increases, the first regime of heat transfer is natural convection observed between points ‘o’ and ‘a’ in Figure 1. The low wall superheats in this regime are not sufficient to initiate bubble nucleation and therefore heat transfer to the bulk liquid is through natural convection. As the input heat flux increases, a thermal boundary layer develops over the heater surface with the liquid within the boundary layer being superheated. At point ‘a’, the wall superheat is sufficient for bubble nucleation and the first vapor bubbles begin to grow on the heater surface. This point is the onset of nucleate boiling (ONB). Bubble nucleation on a heated surface was studied by several researchers including Hsu [6], Hsu and Graham [7], and Kandlikar and Spiesman [8]. The wall superheat at which bubble nucleation is observed for a given cavity diameter is given by,

$$\Delta T = \frac{2\sigma T_{sat}}{\rho_v h_{lv} r_c} K_{max}$$

(1)
where $K_{\text{max}}$ is the maximum non-dimensional curvature of the interface. The range of cavity radii where nucleation can occur at a particular wall superheat was initially proposed by Hsu [6] and was later modified by Kandlikar et al. [9] and is given by

\[
\left( r_{c,\text{max}}, r_{c,\text{min}} \right) = \frac{\delta_t \sin \theta}{2.2} \left( \frac{\Delta T_{\text{sat}}}{\Delta T_{\text{sat}} + \Delta T_{\text{sub}}} \right) \left[ 1 \pm \sqrt{\frac{8.8 \sigma T_{\text{sat}} (\Delta T_{\text{sat}} + \Delta T_{\text{sub}})}{\rho_v h_{lv} \delta_t T_{\text{sat}}^2}} \right]
\]

(2)

After the first bubble has nucleated, a large number of cavities on the surface of the heater are activated and boiling is observed throughout the surface of the heater. A temperature drop is sometimes observed at ONB due to the increase in heat transfer coefficient as a result of the sudden increase in the number of sites where nucleation occurs. Between point ‘a’ and point ‘c’, partial nucleate boiling is observed. During partial nucleate boiling, nucleation is observed at distinct locations on the heater surface and there is little interaction between bubbles at different cavities. The density of active nucleation sites and the frequency of bubble formation from nucleation sites increases as the heat flux increases. This results in the slope of the boiling curve increasing significantly beyond point ‘a’.

As the heat flux increases, isolated bubbles begin to coalesce in the vertical direction to form vapor jets or columns. This represents the transition of the boiling regime from partial nucleate boiling to fully developed nucleate boiling and is observed between points ‘c’ and ‘d’. In addition to the bubbles merging in the vertical direction, lateral coalescence of bubbles are observed leading to the formation of mushroom-like bubble structures. During fully developed nucleate boiling, a point of inflection is observed close to point ‘d’ where the heat transfer coefficient is highest.

At point ‘d’, the maximum heat flux possible during nucleate boiling is reached. This point is known as the Critical Heat Flux (CHF). At CHF a thin layer of vapor covers the heater surface and
prevents the liquid from touching the heater surface. The layer of vapor drastically reduces the heat transfer coefficient of the surface and hence the temperature of the heater rises almost instantaneously and the system moves from point ‘d’ to point ‘e’.

**Transition boiling**

After reaching CHF the boiling mechanism instantly changes from fully developed nucleate boiling to film boiling. The change is accompanied by a sudden rise in temperature indicated by the dotted line between ‘d’ and ‘e’ in Figure 1. However, in a temperature controlled system the heat flux drops as the surface temperature increases and the boiling regime is described as transition boiling. During transition boiling an unstable vapor layer is established on the surface that is intermittently removed. Point ‘f’ is the minimum heat flux point (also known as Leidenfrost point). The presence of the vapor layer reduces the efficiency of heat transfer thereby decreasing the heat transfer coefficient as the heat flux increases. After point ‘f’ the vapor film becomes stable and the boiling regime changes to film boiling.

**Film boiling**

Upon formation of a stable film layer the boiling regime changes from transition boiling to film boiling. In the case of a heat flux controlled system, film boiling is observed immediately after CHF is reached without traversing the transition boiling region. Since the surface is completely covered by a layer of vapor and most of the heat is transferred through radiation, the heat transfer coefficient is very low. In a heat flux controlled system, the surface temperature increases until the radiative heat flux becomes equal to the input heat flux. Since most systems cannot handle the high temperatures associated with film boiling they are maintained below CHF in the nucleate boiling regime.
1.1.2 Heat Transfer Mechanisms

One of the areas researchers have focused on is the modes of heat transfer from the heater to the liquid during boiling. Three main modes of heat transfer have been identified during nucleate boiling: i) microconvection, ii) transient conduction, and iii) microlayer evaporation modes of heat transfer have been widely reported and they are:

**Microconvection**: the rapid motion of the liquid-vapor interface during the bubble ebullition creates convective currents adjacent to the bubble base area. Adiabatic solid sphere experiments by Han and Griffith [10] showed that the microconvection currents influence a region that is twice the bubble departure diameter. At higher heat fluxes, the bubble frequency increases and consequently the contribution of microconvection heat transfer also increases. Rohsenow [11] first proposed that convection due to bubble departure was responsible for enhanced heat transfer during boiling. Forced convection correlations were modified by computing the Reynold’s number based on bubble diameter and a relation between the wall super heat and heat flux was proposed and is given by

\[
\frac{C_p \Delta T}{h_{fg}} = C_{sf} \left[ \frac{q''}{\mu_l h_{fg}} \sqrt{\frac{\sigma}{g(\rho_l - \rho_v)}} \right]^{0.33} P_l^{1.7}
\]

where \( C_{sf} \) is a coefficient specific to the liquid-solid combination.

**Transient conduction**: This refers to the transient heat transfer to the bulk liquid that rewets the heater surface upon bubble departure and corresponding removal of the liquid from the thermal boundary layer. Han and Griffith [10] and Mikic and Rohsenow [12] initially studied transient conduction and proposed that transient conduction is observed throughout the influence region of the bubble (twice the bubble departure diameter). Figure 2 shows a schematic of the region where transient conduction is active as proposed by Mikic and Rohsenow [12]. Subsequent studies by
Moghadam and Keiger [13,14] suggested that only the region under the bubble is subjected to transient conduction as the liquid front rewets the surface during bubble departure. Consensus on the region where transient conduction is effective is still missing/elusive.

Figure 2: Transient heat conduction in the influence region as proposed by Mikic and Rohsenow [12].

**Microlayer evaporation:** The microlayer is a thin liquid film that develops between the bubble and the heater surface during bubble growth. The microlayer develops due to the no slip boundary condition at the heater surface. It was initially studied by Moore and Mesler [15], and Cooper and Lloyd [16]. Since the microlayer is just a few microns thick, the resistance to conduction is very low in this region and therefore heat transfer between the heater and the liquid-vapor interface is very high. As a result, the microlayer evaporates rapidly leading to a drop in the temperature under the bubble. The contribution of microlayer heat transfer has been widely debated [13,14,17,18] and depends on the properties of the liquid with the contribution of microlayer evaporation being higher for liquids with large latent heats.

### 1.2 Applications

#### 1.2.1 Electronics Cooling
Thermal management of electronic devices has been a challenge since the inception of the industry. Cooling of certain components in electronic devices such as the processor and power electronics is particularly challenging since the heat fluxes generated are very high and maintaining the device temperature is critical to its functioning. One of the applications where two phase cooling is being widely researched is cooling of high performance computers (supercomputers) and data centers. Figure 3 shows power consumption of CPUs between 2010 and 2020. It is clear that the power density in data center racks has been rapidly increasing. Air cooling is currently the most widely used cooling technique and single phase liquid cooling has been used in certain high performance applications. Due to the relatively low heat transfer coefficients associated with these cooling techniques, large volumetric flow rates will be required to keep up with the ever increasing thermal demand and inefficient cooling of local hotspots. Since two phase cooling can dissipate much higher heat fluxes at significantly lower wall superheats it can be an effective solution to dissipate large amounts of heat from electronic devices.

![Figure 3: CPU power consumption between 2010 and 2020 [19].](image-url)
1.2.2 Desalination

A growing proportion of the water requirements of water-starved large cities is being met by desalination plants. Factors such as population growth, urbanization, industrialization and climate change are expected to accelerate the need for desalinated water [20]. A report by the global water intelligence estimates that the need for desalinated water is expected to increase from 1% currently to 14% by 2025 [21]. Thermal desalination techniques boil or evaporate seawater to produce fresh portable water. In the coming years, integration of desalination plants with renewable power sources are expected to become increasingly popular as the world moves towards cleaner sources of energy [22]. Using the heat generated in solar concentrators, geothermal units and nuclear reactors to operate the thermal desalination units can also potentially produce large amounts of fresh water at very low costs.

1.2.3 Refrigeration and Air Conditioning

Boiling is extensively used in refrigeration and air conditioning for domestic and commercial applications. The refrigeration and air conditioning industry is one of the largest consumers of power in the U.S. responsible for 30% and 7% of the total power consumption in the commercial and manufacturing sectors in the U.S. respectively. In these systems, boiling heat transfer can be seen on tube bundles and microchannel heat exchangers in flooded evaporators and chillers. The working fluid in these systems are typically refrigerants that have low thermal conductivity and latent heat (relative to water) and therefore are poor heat transfer fluids. Since the efficiency of the system is directly dependent on the heat transfer coefficient at the evaporator section, improving the boiling performance can have a direct impact on the size and operational cost of the system.

1.3 Structure of Thesis
Chapter 1: Introduction- An overview of the basic concepts in boiling heat transfer, the main advantages of boiling heat transfer and its application

Chapter 2: Literature Review- A review of the research related to the three aspects of boiling heat transfer studied in this work: i) Bubble growth models, ii) CHF models- Hydrodynamic and thin heater models, and iii) forces acting in the contact line region and the nature of heat transfer in the region.

Chapter 3: Experimental Setup- Description of the three experimental setups used during the study- an open loop and a closed loop setup to test flat surfaces and an open loop setup to test tubular surface. Additionally the test procedure, data acquisition and the uncertainty analysis conducted are also explained.

Chapter 4: Effect of Evaporation Momentum Force on Bubble Dynamics- The effect of evaporation momentum force on bubble growth rate is analytically modeled. Later, the trajectory of a bubble subjected to an asymmetric temperature field is evaluated. High speed visualization is used to validate the analytical predictions. Finally, the findings are applied to develop a surface enhancement on tubular surfaces that increases the CHF and heat transfer coefficient by modifying the trajectory of a nucleating bubble.

Chapter 5: Pool Boiling Enhancement through Contact Line Augmentation- Microgrooved surfaces that are designed to increase the pool boiling performance by increasing the contribution of contact line heat transfer during boiling are discussed. High speed visualization is used to understand the role of the contact line region heat transfer and bubble dynamics over microgrooved surfaces.
Chapter 6: Effect of Surface Properties and Morphology on Pool Boiling- The effect of the thermophysical properties of the heater surface on the CHF over thick heaters is experimentally studied. In the second part of the study, the pool boiling performance with seawater is evaluated and a passive method to reduce crystallization fouling through physical aberration is described.

Chapter 7: Summary and Future Recommendations- Some key contributions and societal impact of the work are outlined in this section. Future research recommendations are also presented.

Chapter 8 and 9: References and Appendix respectively
2.0 Literature Review

2.1 Bubble Growth

The growth rate of a bubble in a superheated liquid or nucleating on a heated surface has been studied by several researchers over the past five decades [1,23–25]. The growth rate of a bubble varies significantly between bubble nucleation and bubble departure and is influenced by the thermophysical properties of the liquid, the operating pressure, the wall/liquid superheat and the gravitational field. Broadly, the growth of a bubble can be divided into the *inertia controlled phase* and the *temperature controlled phase*. Figure 4 shows a generic plot of bubble radius vs time. In the initial stages of bubble growth, a small amount of evaporation at the bubble interface leads to a significant increase in the bubble radius. Additionally, as the bubble radius increases, the vapor pressure within the bubble reduces as the surface tension on the bubble reduces. This leads to a reduction in the saturation temperature and the temperature gradient at the liquid-vapor interface increases. Therefore, in the initial stages the limiting factor for bubble growth is the displacement of the liquid around the bubble. This is the phase where the growth rate of the bubble is highest. Some of the earliest work to predict bubble growth was conducted by Rayleigh [26] while investigating cavitation. The model developed by Rayleigh does not account for the effect of temperature variation on bubble growth and can be used to predict bubble growth rate in the inertia limited growth phase.
As the bubble diameter increases, continued evaporation of the liquid results in a thermal boundary layer developing around the bubble. The growth of the boundary layer results in a reduction in the temperature gradient at the liquid-vapor interface and therefore the bubble growth rate also reduces. This phase of bubble growth can be described as the heat flux limited phase. Theoretical models to predict bubble growth while accounting for a temperature gradient at the liquid-vapor interface have been modeled by Forster and Zuber [23], Plesset and Zwick [2], Scriven [24], Skinner and Bankoff [27] and Mikic et al. [1]. A brief description of some of the most popular bubble growth models are given below.

**Rayleigh-Plesset model:** The Rayleigh-Plesset equation is an ordinary differential equation which describes the growth of a spherical bubble in an infinite body of liquid. It was developed to model liquid cavitation and hence does not include any heat transfer term. Since the initial phase of bubble growth is inertia controlled, the Rayleigh-Plesset equation can be used to model the initial stage
of a bubble. The variation of radial velocity in the liquid domain as a result of bubble growth can be given by

\[ u(r, t) = \frac{R^2}{r^2} \frac{dR}{dt} \] (4)

where \( R \) is the radius of the bubble. Upon expressing the radial velocity in the liquid in terms of the bubble growth rate in the Navier-Stokes equation, the bubble growth rate can be expressed as

\[ \frac{P(R) - P(\infty)}{\rho_l} = R \frac{d^2 R}{dt^2} + \frac{3}{2} \left( \frac{dR}{dt} \right)^2 \] (5)

Eq. 5 is known as Rayleigh’s equation for bubble growth. This was extended by Plesset and Zwick to account for the viscous effects. Expressing \( P(R) \) in terms of the vapor pressure inside the bubble, the Rayleigh-Plesset equation can be written as

\[ \frac{P_v - P_l}{\rho_l} = R \frac{d^2 R}{dt^2} + \frac{3}{2} \left( \frac{dR}{dt} \right)^2 + \frac{4\partial_l}{R} \frac{dR}{dt} + \frac{2\sigma}{\rho_l R} \] (6)

Plesset [28] later modified the equation to account for viscous forces due to liquid displacement and surface tension changes.

**Plesset and Zwick [2]:** As the bubble grows past the inertia controlled growth phase the transfer of heat from the liquid surrounding the bubble becomes the limiting factor for bubble growth. Plesset and Zwick extended the Rayleigh solution by accounting for the heat diffusion into the bubble. A non-dimensional differential equation was proposed to describe bubble growth in a superheated liquid. An approximate solution for initial phases of bubble growth shows an exponential behavior and is given by

\[ R(t) = R_0 \left( 1 + A_1 e^{A_2 t} \right) \] (7)
As the bubble grows, a thermal boundary layer develops next to the liquid-vapor which results in a reduction in the bubble growth rate. An asymptotic approximation to describe bubble growth in the heat flux limited regime can be given by

\[ R(t) = \frac{2}{\pi^2} \oint \alpha (at)^{\frac{1}{2}} \]

(8)

*Mikic et al. [1]:* Mikic et al. [1] proposed a simple non-dimensional polynomial equation that can be used to predict bubble growth rates in both uniform and non-uniform temperature fields. The model used a force balance approach to predict the growth rate in the inertia controlled phase and incorporated the growth rate expressions proposed by Plesset and Zwick [2] to predict the growth rate in the heat diffusion controlled phase. The growth of non-spherical bubbles like those seen on heated surfaces can also be predicted using the model by varying a shape factor term in the model.

The expression for bubble growth in the inertia controlled phase was obtained by equating the work done by the bubble on the liquid to the total kinetic energy of the liquid. This resulted in the equation:

\[ \frac{dR}{dt} = A^2 \frac{T_v - T_{sat}}{\Delta T} \]

(9)

where \( \Delta T = T_\infty - T_{sat} \) and \( A = \left( \frac{bh_f \rho g \Delta T}{\rho_l T_{sat}} \right)^{\frac{1}{2}} \)

The growth rate of a bubble in the heat diffusion controlled phase developed by Plesset and Zwick [2] was used. The growth rate was rewritten as

\[ \frac{dR}{dt} = \frac{1}{2} \frac{B}{\sqrt{\ell}} \left( \frac{T_\infty - T_v}{\Delta T} \right) \]

(10)
where \( \mathcal{B} = \left( \frac{12}{\pi} \alpha \right)^{\frac{1}{2}} J a \) and \( J a = \frac{\Delta T c p \rho_l}{h_{fg} \rho_v} \)

Combining Eq. 9 and Eq. 10, an equation describing bubble growth in a uniformly heated liquid domain was obtained and is given by,

\[
R^+ = \frac{2}{3} \left( (t^+ + 1)^{\frac{3}{2}} - (t^+)^{\frac{3}{2}} \right)
\]

(11)

\( R^+ \) and \( t^+ \) are non-dimensional terms which describe radius and time respectively and are given by

\[
R^+ = \frac{A R}{B^2} \quad \text{and} \quad t^+ = \frac{A^2 t}{B^2}
\]

The results were extended to non-uniform temperature fields similar to those present in nucleate boiling on a heated surface and is given by

\[
R^+ = (t^+)^{0.5} \left\{ 1 - T^* \left[ \left( 1 + \frac{t_w^+}{t^+} \right)^{0.5} - \left( \frac{t_w^+}{t^+} \right)^{0.5} \right] \right\}
\]

(12)

The model proposed by Mikic et al. accurately predicts bubble growth under various conditions and is widely used. Since the bubble radius and time are expressed in non-dimensional terms, the bubble growth equation can be applied to liquids other than water as well.
2.2 CHF Models

2.2.1 CHF Description

Critical heat flux is the maximum heat flux that can be achieved under fully developed nucleate boiling conditions. Further increase in heater temperature or heat flux results in the transition from nucleate boiling to film boiling. Upon reaching CHF, the temperature of the heater rapidly increases (for heat flux controlled systems) often leading to considerable damage to the heater and sometimes leading to meltdown. The onset of CHF corresponds to the formation of a thin vapor film between the heater surface and the bulk liquid. Since the thermal conductivity of vapor is orders of magnitude lower than the bulk liquid, conduction heat transfer from the heater to the bulk liquid drastically reduces with the formation of vapor layer over the heater surface and most of the heat transfer occurs through radiation. For a plain heater surface, multiple factors that affect the CHF have been identified. These include liquid properties like density, surface tension, viscosity and latent heat; interfacial properties such as contact angle for the heater-liquid pair; system conditions like orientation of the heater surface, gravity and system pressure; and heater properties such as surface roughness, density, specific heat, thermal conductivity and heater thickness.

Surface roughness: The effect of surface roughness on CHF has been widely investigated with multiple works concluding that an increase in surface roughness leads to an increase in CHF. Berenson [29] studied the effect of roughness with n-pentane as the working fluid and showed that surfaces that are mirror finished had a 20% lower CHF than rough surfaces (polished with #60 emery paper). Similarly, experimental works from Ramilson and Lienhard [30] and Haramura [31] have shown that surface roughness can enhance the CHF over a surface up to 35%. However, in a recent study, O’Hanley et al. [32] investigated the effect of surface roughness on CHF for surface
roughness values between 0.01 and 2.62 and found that there is no significant increase in CHF due to an increase in roughness.

Surface wettability: The effect of surface wettability on CHF has been studied by many researchers including Costello and Frea [33], Kirishenko and Cherniakov [34], Maracy and Winterton [35], Liaw and Dhir [36] and O’Hanley et. [32]. Costello and Frea found that CHF increases as the wettability of the heater surface increases. Liaw and Dhir [36] controlled the wettability of the surface by varying the degree of oxidization of the surface. This was believed to be because the removal process of the vapor bubbles from the heater surface becomes more efficient as the surface becomes more wetting. Figure 5 shows the experimental results of CHF as a function of contact angle from Liaw and Dhir [36] and Maracy et al. [35].

Figure 5: Effect of contact angle on CHF. Data from Liaw and Dhir [36] and Maracy et al. [35]. Figure adopted from Kandlikar et al. [9].
Heater size: Shoji et al. [37] studied the effect of heater size on CHF using horizontally oriented stainless steel ribbon heaters in water. It was seen that the CHF increased as the heater width decreased with the CHF on 0.2 mm wide heaters being four times higher than wider heaters. However similar studies on cylindrical heaters showed that CHF increased as the radius of the cylindrical heaters increased. The divergence in the trends has been associated with the different bubble departure mechanisms.

2.2.2 Hydrodynamic CHF Models

The prediction of critical heat flux (CHF) over a heater surface is one of the most widely studied aspects of pool boiling. The models for predicting CHF can be largely classified into hydrodynamic, interfacial and surface effect based. One of the proposed theories for the onset of CHF is the vapor escape instability model. Kutateladze [4] proposed that at high heat fluxes separate liquid and vapor columns are formed and CHF is reached when the two phase flow near the heater surface becomes unstable. It was proposed that the flow becomes unstable when the vapor velocity increases beyond a critical value. Kutateladze performed a dimensional analysis and proposed the following expression to predict the onset of CHF:

\[
\frac{q'_c}{h_f \rho_g^{0.5} \left[ \sigma g (\rho_l - \rho_g) \right]^{1/4}} = K
\]

For pool boiling on a horizontal surface, the value of the constant K used in the above equation was experimentally determined to be 0.16. Zuber [38] extended the work by assuming that the diameter of the vapor columns produced at high heat fluxes is half of the Rayleigh-Taylor wavelength. Further, the maximum velocity of the vapor in the vapor column was assumed to be limited by the Kelvin-Helmholtz instability function. Following this approach, the value of K as used in Eq. 13 was seen to be 0.138-0.157. The model was later simplified and the value of K was
changed to 0.131. Lienhard and Dhir [39] modified the expression for maximum vapor velocity and included the effect of heater geometry to predict the CHF in finite geometries such as vertical plates and cylinders. The modified expression suggested that Zuber’s model under predicted the CHF of a surface by 14%.

Instead of the vertical coalescence model proposed by Kutateladze and Zuber, some researchers proposed an alternate approach and assumed that the bubbles coalesce in the horizontal direction to form vapor mushrooms. Katto and Yokoya [40] proposed that a liquid macrolayer is formed between coalescing vapor bubbles and CHF occurs when the macrolayer dries out before the departure of the coalesced bubble. Haramura and Katto [41] further refined the dryout model by assuming that the thickness of the trapped macrolayer is one-fourth the Kelvin-Helmholtz wavelength. An energy balance approach was used to determine the heat flux required to evaporate the macrolayer before the departure of the coalesced bubble. The final equation arrived at was similar to the one proposed by Kutateladze with a K value of 0.131.

**Figure 6: Forces acting on a bubble growing on a heater surface as considered by Kandlikar [42].**
Kandlikar [42] incorporated the effect of dynamic receding contact angle in the CHF model by using a force balance approach to predict the onset of CHF. Figure 6 shows the forces acting on a nucleating bubble on a heater surface. The forces acting on the bubble parallel to the heater surface were considered and it was proposed that the onset of CHF occurs when the forces that push a bubble outwards overcome the forces that restrict bubble expansion. The bubbles then expand over the heater surface to form a thin vapor blanket. The force responsible for the outward expansion of the bubble was identified as evaporation momentum force. Evaporation momentum force (vapor recoil) is a reactionary force experienced at the liquid-vapor interface due to the sudden expansion of the evaporating mass and the consequent increase in its momentum. The forces that restrict bubble expansion were assumed to be the hydrostatic pressure due to the liquid head and the surface tension acting at the three phase contact line. The departure diameter was assumed to be half of the Taylor instability wavelength. The expression for CHF was given by

\[ q''_c = h_f g \rho_g \left( \frac{1 + \cos \theta}{16} \right) \left[ \frac{2}{\pi} + \frac{\pi}{4} (1 + \cos \theta) \cos \varphi \right]^\frac{1}{2} \times \sigma g (\rho_l - \rho_g)^\frac{1}{4} \]  

(14)

**2.2.3 Thin Heater Studies**

Non-hydrodynamic factors like the thermal properties and thickness of the heater also play a significant role in affecting the CHF of a surface. Multiple works [43–47] have studied the effect of heater thickness and thermal properties, such as the heater material, on CHF over the surface. To study the effect of the thermal properties of the heater substrate on CHF, Tachibana et al. [44] experimentally determined the CHF on thin, vertically oriented ribbon heaters in water. For thin heaters, the CHF was seen to be proportional to the heat capacity, which is the product of heater thickness, density and specific heat of the heater. For a given material, the CHF increased with an increase in the thickness of the heater up to an asymptotic value beyond which no further increase
in CHF was observed. The authors also conducted numerical simulations to show that the heat flux required for the temperature raise is only dependent on the heat capacity of the heater and not the thermal conductivity of the material.

![Figure 7: CHF over thin heaters as a function of thermal activity parameter. Figure adapted from Golobic and Bergles [45].](image)

Golobič and Bergles [45] conducted similar tests with thin vertically oriented heaters in FC-72 at atmospheric pressure. Regression analysis was carried out to determine the effect of properties and thickness of the heater on CHF. The CHF corresponding to a surface was seen to be best predicted using the term $\delta(\rho c_p k)^{0.5}$ which is referred to as the thermal activity parameter. For the obtained dataset, the following expression relating the CHF to the thermal activity of the heater was proposed

$$q'' = q''_{asy} \left( 1 - e^{\left( \frac{\delta H(\rho c_p k)^{0.5}}{2.44}\right)^{0.8498} - \left( \frac{\delta H(\rho c_p k)^{0.5}}{2.44}\right)^{0.0581}} \right)$$

(15)
where $q''_{asy}$ is the asymptotic CHF and was taken to be 135 kW/m$^2$ for the surfaces tested. However, sufficient testing of thick heaters, which are heaters whose thickness is greater than the asymptotic heater thickness, was not conducted to determine the asymptotic CHF for all the materials. Figure 7 shows the CHF obtained for the substrates tested as a function of the thermal activity parameter and the predicted CHF using Eq. 15. The thermal activity parameter resulted in the best fit for the thin heater with the average absolute deviation between the experimental results and the predicted valued being about 8.4%. The authors also evaluated the thickness at which asymptotic CHF is reached. The heater thickness at which 90% of the asymptotic CHF is reached was taken to be the asymptotic heater thickness. Table 1 shows the observed asymptotic heater thickness for the different materials tested.

**Table 1: Asymptotic heater thickness for various heater materials.**

<table>
<thead>
<tr>
<th>Material</th>
<th>Asymptotic Heater Thickness (μm)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Inconel 600</td>
<td>500</td>
</tr>
<tr>
<td>SS 302</td>
<td>550</td>
</tr>
<tr>
<td>Steel</td>
<td>300</td>
</tr>
<tr>
<td>Iron</td>
<td>250</td>
</tr>
<tr>
<td>Nickel</td>
<td>100</td>
</tr>
<tr>
<td>Brass</td>
<td>130</td>
</tr>
<tr>
<td>Aluminum</td>
<td>25</td>
</tr>
<tr>
<td>Copper</td>
<td>15</td>
</tr>
</tbody>
</table>

The transient nature of boiling plays an important role in determining the temperature fluctuations in the heater substrate. Han and Griffith [10] modeled heat transfer during boiling as a transient phenomenon. The liquid over the heater surface is superheated leading to the
growth of a thermal boundary layer over the heater surface. It was proposed that the superheated liquid around the influence region of a bubble is replaced by the cooler bulk liquid during bubble departure. The heater surface temperature reduces due to contact with cooler liquid and a cyclic variation in the local heater temperature corresponding to the bubble ebullition cycle is observed. The local temperature variations get evened out in the heater as the distance from the surface increases and a further increase in the thickness of the heater does not improve the CHF. Watwe and Bar-Cohen [46] numerically modeled heat conduction in thin heaters and studied the variation of the maximum surface temperature for different heater thicknesses and properties. The maximum surface temperature was found to correlate well with the thermal activity parameter when the size of the vapor bubbles on the heater surface were assumed to be small. When the vapor bubbles are large or, in the case of flow boiling, when the frequency of bubble formation and departure is high, the lateral diffusion of heat from the center of the bubble was found to be insignificant and the temperature rise under the hot spot correlated well with the thermal capacity of the heater and the effect of thermal conductivity is negligible. From the studies described above, density, specific heat and thermal conductivity of the heater material have been identified as the properties that could affect the CHF over a thick heater surface. Arik and Bar-Cohen [47] proposed a statistically determined correlation for pool boiling CHF that accounted for the thickness and the thermal properties of the heater, the operating pressure and the degree of liquid sub-cooling.

2.4 Contact Line Region

The contact line or the three-phase contact line is defined as the region where the solid, liquid and the gaseous phases intersect. The contact line region in a bubble is observed at the base of the bubble where the interface of the bubble meets the heater surface. Figure 8 (a) shows the contact line region in a nucleating bubble. Due to the interaction among these phases, the nature of this
region is unlike the rest of the boiling surface and merits a careful examination. While the three phase contact line region macroscopically appears as a sharp boundary between the bulk liquid and the vapor, an enlarged view of the contact line region reveals a more nuanced picture. Based on the size of the meniscus and the nature of the forces acting on the liquid, the contact line region can be divided into four regions – the adsorbed film region, the transition region, the intrinsic meniscus region and the microconvection region. A schematic representation of the regions is shown in Figure 8 (b). The adsorbed film and transition regions together may be described as the thin film region and is characterized by dominant role of the long range intermolecular forces. The region adjacent to the thin film region is the macro region of the contact line and can be further categorized into the intrinsic meniscus region and the microconvection region.

The adsorbed region is a non-evaporating liquid film and hence does not contribute towards the growth of a bubble. Strong attractive forces between the solid substrate and the liquid prevent evaporation and results in a flat liquid-vapor interface with uniform thickness. It is difficult to visually observe this region since the liquid layer is only a few nanometers thick and specialized optical techniques are required to measure the film thickness. The region adjacent to the adsorbed film region where the thickness of the liquid begins to increase is known as the transition region. Due to the increasing separation between the liquid-vapor interface and the heater surface, the attractive forces between the liquid molecules at the interface and the heater surface reduces significantly. Sharp variation in the curvature of the interface causes large capillary forces to be experienced at the interface affecting the liquid flow at the contact line region during evaporation.

The resistance to heat transfer from the contact line region is due to the conduction resistance in the liquid and the resistance to evaporation at the liquid-vapor interface. The transition region experiences large heat fluxes as the resistance offered to heat transfer in the liquid is very low.
given its small thickness and the resistance at the interface is drastically lower than that in the adsorbed region due to reduced disjoining pressure. As the thickness of the liquid film increases, the conduction resistance in the liquid film increases proportionately and thereby heat flux in the bulk region decreases again.

The macro region is characterized by the dominance of forces such as surface tension and inertial force on the liquid. The intrinsic meniscus can be described as the region where the thickness of the liquid is large enough for intermolecular forces to be insignificant but small enough for shear forces to significantly influence fluid flow. The intrinsic meniscus region is also referred to as the ‘microlayer’ in the boiling context. The shear force on the liquid reduces significantly as the thickness of the liquid meniscus increases leading to the development of microconvective currents (microconvection is used here to represent the localized liquid convection adjacent to the contact line due to motion of the interface, and not the increased convection in the region between bubbles). The movement of the contact line during bubble growth and departure induces additional convective effects which enhance the heat transfer in the microconvection region. While the heat flux in the transition region is very high, the size of the region limits its contribution towards net heat transfer during boiling. However, Dhavaleswarapu et al. [48] found that almost 95% of heat is transferred from 30% of the interface close to the contact line. Although further confirmation of this conclusion is warranted, studying heat transfer in the microconvection region is nevertheless critical for understanding heat transfer in the contact line region. The discussion presented in this paper will first cover the forces acting in the adsorbed film and transition regions followed by a review of the nature of heat transfer in the microlayer and microconvection regions and the contribution of the different modes of heat transfer towards boiling.
2.4.1 Forces in the Contact Line Region

Consider a thin film of thickness $\delta$, the free energy per unit area of the film is given by [49]

$$F(\delta) = \gamma_{SL} + \gamma_{LV} + p(\delta)$$

where $\gamma_{SL}$ is the solid-liquid interfacial tension, $\gamma_{LV}$ is the liquid-vapor interfacial tension and $p(\delta)$ represents the surface forces due to long-range intermolecular forces. Researchers have extensively studied the nature of surface forces acting on a thin film between two charged solid surfaces [50–53]. These forces can be further classified into electrostatic, structural, and dispersion structural forces. The origin of the electrostatic component is the osmotic pressure buildup in the thin film due to excessive accumulation of counter ions near a charged surface. Structural forces are repulsive in nature and act on the molecules when the separation is under 3 nm in case of hydrophilic surfaces and 20 nm in case of hydrophobic surfaces. The dispersion component consists of van der Waals interaction between the solid and thin liquid film. van der Waals forces were first studied by Derjaguin [54] and Derjaguin and Landau [55] and later extended by Verwey and Overbeek [56] in what is commonly referred to as the DLVO theory. In the case of non-polar liquids, the dominant intermolecular interaction is the van der Waals force. For the liquid film thickness in the mesoscopic range, $p(\delta)$ can be expressed as
where $A^*$ is the Hamaker constant. In case of non-retarded van der Waals forces, disjoining pressure ($\Pi$) is the derivative of the free energy with respect to the film thickness and is given by

$$\Pi(\delta) = -\frac{dp}{d\delta} = -\frac{A^*}{6\pi\delta^3}, \delta < 30 \, nm \quad (18)$$

For distances greater than 30 nanometers, the Hamaker constant is replaced by dispersion constant $B^*$ and the disjoining pressure is given by

$$\Pi(\delta) = \frac{B^*}{\delta^4}, \delta > 30 \, nm \quad (19)$$

The variation of disjoining pressure for polar liquids was expressed as a logarithmic function by Holm and Goplen [57], and is given by

$$\Pi(\delta) = -\rho_i \bar{R} T_i \ln(\alpha^* \times \delta^\beta^*) \quad (20)$$

where $\alpha^*$ and $\beta^*$ are fluid specific coefficients.

The second type of force experienced by the liquid in the thin film region is called capillary force. Capillary force is a surface force experienced by the molecules at the interface due to asymmetric intermolecular attraction. The magnitude of the capillary pressure is a product of the surface tension of the liquid and the curvature of the interface and is given by

$$P_c = 2\gamma J \quad (21)$$

The capillary pressure experienced by the liquid acts normal to the interface. Experimentally obtained results by Panchamgam et al. [58] showed that the curvature of the interface increases sharply in the transition region before reducing again in the intrinsic meniscus region (Figure 9.
Such variations in the interface curvature result in a non-uniform distribution of capillary pressure at the interface which significantly affects the fluid flow in the region. Since the curvature of the interface is uniform in the macro region, the capillary force experienced in this region does not vary. In case of thin films, the Young Laplace equation is modified to account for the disjoining pressure experienced at the interface. The modified equation, referred to as the augmented Young-Laplace equation [59], is given by,

$$ P = 2\gamma f + \Pi $$

(22)

The third type of force experienced by the liquid is due to the temperature variations at the liquid-vapor interface and is known as thermocapillary convection or the Marangoni effect. The Marangoni effect was first observed by Thomson [60] and Marangoni [61] in the mid-19th century. They noted that the gradient in surface tension as a result of temperature variation results in the liquid experiencing a shear force that drives the liquid away from the hotter region and towards the colder region. Many researchers, e.g. [62–64], have studied the effect of the Marangoni force on liquid spreading and film stability. Due to the varying thickness of the liquid film and non-uniform evaporation in the contact line region, the temperature of the liquid at the liquid-vapor interface also varies considerably (shown in Figure 9 (b)). As the thickness of the liquid film increases from the adsorbed film region to the transition region, the disjoining pressure experienced by the liquid reduces. A sharp increase in the rate of evaporation in the transition region leads to a sudden drop in the temperature of the liquid. This variations in interfacial liquid temperature results in a shear force which affects the nature of liquid transport in this region.

Evaporation momentum force or vapor recoil is the force experienced at the interface due to an increase in the momentum of the evaporating mass at the interface. The difference between the densities of the liquid and the vapor phase causes the velocity of vapor leaving the interface to be
substantially higher than the liquid velocity approaching the interface. The increase in the momentum of the evaporating mass leads to the interface experiencing a reactionary force which pushes the liquid away from the vapor and acts normal to the interface. The resulting evaporation momentum pressure can be expressed as

\[ P_M = \eta^2 (\rho_v^{-1} - \rho_l^{-1}) \]  (23)

where \( \eta \) is the evaporating mass flux at the interface. While the effect of this force in altering the bubble dynamics during boiling has been studied earlier [65,66], its role in the contact line region has not been explored sufficiently.

The combined effect of the forces acting in the contact line region in terms of the chemical potential per unit volume is given by [67]

\[ \Delta \mu_g = -\Pi_{\text{Disjoining}} - \sigma_{\text{Capillary}} + \frac{\rho_i \Delta h_g}{T} (T_v - T_l) + \sum \frac{\rho_i}{M_w} R \ln \left( \frac{x_i}{x_i^{\text{ref}}} \right) + \rho_i gz - \frac{z_i c_i FE}{C} \]  (24)

where the thermal term is a result of the temperature and pressure jump across the interface under non-isothermal conditions, which are known as the Clapeyron and the Kelvin effect respectively. The electric potential is a result of an externally applied electric field and the concentration term arises from the concentration difference of a species across an interface. Under isothermal conditions, the chemical potential of the vapor is equal to the potential of the liquid at the interface. Upon increasing the temperature of the surface, the vapor pressure at the interface reduces and a temperature jump at the interface is observed. The thickness of the adsorbed layer reduces to a new equilibrium value and the profile of the interface changes. The change in the shape of the
interface leads to an increased flow of liquid towards the contact line region where high evaporative mass flux is observed.

Figure 9: (a) Variation of interfacial curvature under isothermal and non-isothermal conditions, (b) Liquid temperature variation at the solid-liquid and liquid-vapor interface, and (c) Contribution of Marangoni, capillary and disjoining pressure towards fluid flow [58]

2.4.2 Heat Transfer in the Contact Line Region

The enhancement in heat transfer in the contact line region was analyzed by Derjaguin et al. [68] by studying the role of thin film evaporation in capillaries. Schrage [69] presented a relation between the liquid-vapor interfacial condition and the evaporating mass flux across the interface. The proposed expression for interfacial mass flux is given by
where $\tilde{\sigma}$ is the accommodation coefficient, $\tilde{R}$ is the universal gas constant and $M_w$ is the molecular weight of the liquid. The accommodation coefficient is a physical quantity which is used to characterize the behavior of vapor molecules upon impinging a solid or liquid surface. It can be defined as the ratio of the number of molecules that get diffuse-reflected after striking the surface to the total number of molecules striking the surface.

Potash and Wayner [70] modeled a two-dimensional evaporating meniscus to study heat transfer in a two-dimensional extended meniscus. The liquid flow in the meniscus region was approximated to a wedge flow and it was found that the pressure drop due to the change in the profile of the meniscus was sufficient to drive fluid flow towards the contact line for evaporation. The gradient of disjoining pressure aided liquid flow in the lower regions of the meniscus while capillary forces due to variation of the curvature of the interface was driving fluid flow in the upper parts of the meniscus.

The work by Potash and Wayner was extended by Panchamgam et al. [58] who used interferometry to measure the profile of an evaporating meniscus in a vertical constrained vapor bubble setup. The pressure profile in the meniscus was calculated using experimentally obtained film thickness measurements. The calculated pressure value was substituted in the augmented Young-Laplace equation to show that the variation of disjoining pressure affects the motion of the meniscus and the flow of liquid towards the transition region. The contribution of Marangoni stress, capillary pressure and disjoining pressure to the mass flow rate of the liquid is shown in Figure 9 (c). A positive mass flux indicates that the corresponding force drives the liquid towards the contact line.
and a negative mass flux indicates that the liquid is driven away from the contact line. Disjoining pressure was observed to be driving fluid flow in the transition region, where the evaporation rate is highest. As a result of the low liquid temperature in the transition region, Marangoni stress was seen to drive fluid flow towards the transition region from parts of the contact line where the liquid thickness is greater as well as the region after the transition region where the liquid film thickness is lower. Lastly, capillary pressure was generally seen to result in a positive mass flux throughout the thin film and intrinsic meniscus regions.

Wayner et al. [71], in a landmark paper built on the work done by Potash and Wayner [70] and Schrage [69], determined the heat flux in the section between the interline (junction between the adsorbed region and the transition region) and the macro region for wetting non-polar liquids. The equation for mass flux across the interface by Schrage (Eq. 25) was simplified to

\[ m'' = a(T_{lv} - T_v) - b(\Pi + P_c) \] (26)

where the second term in Eq. 26 (a) accounts for the suppression in evaporation due to disjoining pressure and capillary pressure. Wang et al. [72,73] modeled the thin film region of an evaporating meniscus in microchannels using the augmented Young-Laplace model. To evaluate the evaporating mass flux, the authors used Eq. 25, developed by Schrage [69]. This is different from Eq. 26 (a-c) by Wayner et al. [71] which is widely used in literature [71,74–76]. Equation 26 (a-c) was developed from Eq. 10 by using the approximations \( T_{lv} \approx T_v \) and \( P_{v,eq} \approx P_v \). While this is valid at low temperatures, the mass flux is under-predicted at higher superheats, as shown in Figure 10.
Wang et al. [72,73] reported that macroscopic geometric factors such as channel width primarily affects the region beyond the transition region, known as the intrinsic meniscus region. The intrinsic meniscus regions account for over 50% of the net heat transfer through an extended meniscus in a microchannel while the contribution of just the adsorbed layer and transition regions were under 20% of the overall heat transfer through the meniscus.

One of the main assumptions while modeling fluid flow in the contact line region using the lubrication theory is the no-slip condition at the solid-liquid interface. However, depending on the properties of the solid and the liquid, significant slip can be experienced at the solid-liquid interface. The role of slip in the contact line region and its effect on evaporation has been studied by many researchers [77–82]. Slip at the solid-liquid boundary can be expected to occur when a thin depleted layer of a fluid (like a rarefied gas layer) is present at the solid-liquid interface. Biswal et al. [81] developed a model for thin film evaporation in a microchannel with interfacial slip. Accounting for the intermittent slip, the mass transfer rate at any section is given by
\[ \Gamma(x) = -\frac{\delta^3 + 3\beta \delta^2}{3\theta} \frac{d}{dx} (P_l) \]  

(27)

The authors observed that increased slip length resulted in a reduction in the apparent contact angle and increase in length of the adsorbed film. They also concluded that the net evaporative mass flux increases with the increase in the thickness of the rarified gas layer corresponding to larger slip lengths.

As described earlier, the temperature of the liquid at the liquid-vapor interface varies considerably due to non-uniform evaporation rate. The effect of the resultant Marangoni stress on fluid flow and heat transfer has been investigated by many researchers [58, 83–85]. Zhao et al. [85] studied the effect of considering temperature dependent thermophysical properties while modeling thin film evaporation. It was seen that the thickness of the adsorbed film attains a minimum as the wall superheat increases before increasing again at higher wall superheats. This is different from the results obtained when constant thermophysical properties were assumed, where the thickness for the adsorbed film continues to decrease as the wall superheat increases. The minimum adsorbed film thickness causes a disjoining pressure limit on the heat transfer rate in the thin film. The authors concluded that the total heat transfer rate is significantly overestimated at higher heat fluxes when constant property values are assumed, as shown in Figure 11.
Qu and Ma [86] modeled the rate of evaporative heat transfer for different polar liquids. Wee et al. [87] developed a model which combined the effects of thermocapillary effect, slip boundary condition and polarity effects. Figure 12 shows the variation of disjoining pressure for a liquid (water) when the polar properties of the liquid were accounted for (see polar) and when the polar properties were ignored (see non-polar). The disjoining pressure in the thin film region was significantly higher due to the polar effects and this resulted in the elongation of the adsorbed film region and the reduction of evaporative mass flux in the thin film region. Literature pertaining to the effect of liquid polarity is still sparse and factors such as interaction of polar liquids on metal substrates needs to be explored further.
Experimental examination of the adsorbed layer and transition regions has always been a challenging proposition due to the scale of the liquid film in question. The challenges posed by the experimental approach has led to a majority of researchers using analytical and numerical models to examine thin film evaporation. However there has been some significant work that has been carried out using experimental techniques as well. The Vertically Constrained Vapor Bubble (VCVB) is an experimental setup that has been used in multiple studies [58,88–92] to better understand the heat transfer mechanism in the adsorbed film and transition regions. Figure 13 shows a schematic of the VCVB setup. The setup consisted of a quartz cuvette containing an ultrapure liquid, typically pentane, octane or some other non-polar liquid. A vapor bubble is maintained in the cuvette and the liquid forms a meniscus at the edge of the faces of the cuvette. The liquid is heated using a thermoelectric heater that is attached to one end of the quartz cuvette and the other end of the cuvette is either sealed or connected to a liquid reservoir and a vacuum pump to control the amount of liquid present in the cuvette. The profile of the corner meniscus is
measured through interferometry. By varying the heat applied to the liquid through the thermoelectric heater the profile of the meniscus and the rate of evaporation in the meniscus can be controlled and an evaporating meniscus under advancing and receding conditions can be studied.

![Schematic of the vertically constrained vapor bubble setup used to study thin film evaporation (adopted from [93]).](image)

While the above-mentioned works are some examples of the available literature on different factors affecting heat transfer in the adsorbed film and transition regions, many more have been studied. For example, some of these include evaluating the effect of the electrostatic component of disjoining pressure [94], slope and curvature dependent disjoining pressure [95–98], inertial effect [99] in liquid transport to the adsorbed film and transition region, etc.

### 2.4.3 Contact Line Heat Transfer during Boiling

Various mechanisms have been proposed to explain the underlying nature of heat transfer such as transient conduction [12], microconvection [3,100], microlayer evaporation [101,102] and contact line evaporation [103]. Many researchers have attempted to determine the contribution of the
different modes of heat transfer using experimental and numerical techniques. One of the approaches used by researchers to determine the relative contribution of microlayer evaporation and microconvection is to use microheater arrays to determine the amount of heat transferred into an isolated nucleating bubble from the heater surface. Yaddanapudi and Kim [17] and Demirey and Kim [104] used microheater arrays with a resolution of 270 μm and 100 μm respectively to determine the heat flux variation at the heater surface. The local heat flux was obtained by calculating the amount of power required by each of the microheaters in order to maintain a constant surface temperature. It was assumed that all the heat generated by the microheaters under the bubble is associated with microlayer evaporation and found that only about 12% of the total heat transfer was due to microlayer and contact line heat transfer. Alternatively, Myers et al. [105] maintained a constant heat flux from all the microheaters and measured the temperature of each of the microheaters to compute the relative contribution of each of the modes. Using this method, the effect of conduction between the microheaters is also factored in. It was seen that the contribution of microlayer heat transfer accounted for less than 25% of the total heat transfer.

Moghaddam and Kiger [13,14] estimated the contribution of the different modes of heat transfer during nucleate boiling by measuring the local heat flux of the heater using two temperature sensors with a layer of insulating benzocyclobutene sandwiched in between. This approach eliminated the need to determine the amount of heat transferred from the microheaters to the substrate and hence could provide an accurate local heat flux measurement. Temperature sensors, distributed radially around an artificial nucleation site, were used to determine the heater surface temperature as the bubble grew and departed. It was seen that the start of the microlayer evaporation corresponded to a sharp decrease in the heater surface temperature followed by the surface temperature increasing again due to dryout before the surface temperature drops for the
second time due to rewetting of the surface during bubble departure. The authors computed that microlayer evaporation accounted for 28% of the total heat transfer for the smallest wall superheat and its contribution reduced as the surface temperature increased. The authors referred to the rewetting of the dry spot under the bubble as the bubble departs as transient conduction and its contribution was between 32% and 45% with the higher value corresponding to a lower wall superheat. It was seen that there was a spike in the local heat flux near the contact line as the liquid rewets the surface during bubble departure. However the maximum heat flux during rewetting was less than half the maximum heat flux observed due to microlayer evaporation.

In a recent study, Yabuki and Nakabeppu [18] studied heat transfer during boiling using MEMS sensors to map the temperature distribution at the heater surface with water as the working fluid. Resistance Temperature Detectors (RTD) were used to measure the temperature on the heater surface with a spatial resolution of 20 μm and a sampling rate of 50 kHz. It was seen that the contribution of microlayer evaporation was about 40% in the initial stages of bubble growth, about 60% during the middle stages when the microlayer region is significantly larger and negligible as the bubble departs due to dryout of the microlayer. Figure 14 shows the heat flow into a bubble as a function of time and the contribution of microlayer evaporation towards the total heat transferred.
Over the last decade many studies have used high speed infrared cameras to measure the temperature of the heater surface in order to determine the heat transfer mechanics during nucleate boiling [25,106–111]. Recent advances in infrared imaging technology have enabled capturing infrared videos at up to 3000 frames per second. This technique offers a non-intrusive method to determine the temperature of the heater surface. However it requires a thin heater surface so that the temperature can be recorded using the infrared camera from the back end of the heater surface. Wegener and Stephan [106] measured the temperature of a foil type heater surface with a single nucleation site using a high speed infrared camera placed at the bottom of the test setup. The local heat flux under a bubble was computed using a 2D conduction model. As seen by many other researchers, a sharp temperature drop was observed at the three phase contact line region. However, the authors concluded that the contribution of microlayer evaporation was about 50–60% of the total heat transferred into the bubble. It can be seen from these results and the papers reviewed earlier that the exact contribution of microlayer evaporation is still unclear with values
ranging from 20%-60% being attributed to microlayer evaporation and merits a more careful evaluation.

Kunkelmann et al.[109] also used an infrared camera to determine the variation of local heat flux for advancing and receding menisci. It was seen that the local heat flux near the contact line was an order of magnitude greater than the average heat flux. Figure 15 shows the radial variation of heat flux during bubble growth and departure. It can be seen that the heat flux near the contact line is significantly greater than the average heat flux from the rest of the heater surface. In the case of a growing bubble, the heat flux at the contact line remains consistent at about 20kW/m$^2$ whereas when the bubble begins to depart, the heat flux near the contact line increases dramatically with every successive time step. This is because the meniscus moves over the part of the heater surface which had dried out and therefore is at a higher temperature. The section of the heater surface that dries out first is rewet last and therefore the heat flux at the interface continues to grow as the bubble departs. While these results provide valuable insight into the nature of heat transfer in an advancing and receding meniscus during boiling, other results by Moghaddam and Keiger [13,14] do not indicate such a significant difference between the heat transfer in an advancing and receding meniscus. One of the possible reasons for the wide range of values obtained by various studies could be due to the variation of bubble growth time and bubble departure diameters in the different studies. A combination of the bubble growth rate and the footprint of the bubble on the heater surface determines the velocity with which the contact line sweeps the heater surface. Differences in contact line velocity combined with the variation of fluid properties could lead to the contribution of the microlayer to differ between studies. Similarly a possible reason for the discrepancy in the observed heat flux during rewetting can be the varying degrees to which the heater surface heats up due to the formation of the dryspot. Higher the temperature of the heater
surface as the liquid front rewets it, greater will be the heat flux during rewetting. It is therefore clear that in order to determine the nature of heat transfer in the contact line region during boiling, specifically the contribution of microlayer evaporation, a comprehensive study involving multiple fluids and a wide range of heat fluxes and bubble departure frequencies need to be studied. Further research is needed to reconcile the differences in contribution from the microlayer evaporation.

Figure 15: Average heat flux in the radial direction Curves 1-5: growing bubble and curves 6-13: receding bubble (time step-5 ms) [109].

To better understand the variation in the heat transfer characteristics of an advancing and receding meniscus Kandlikar et al. [112] experimentally studied an evaporating meniscus on a moving heated surface. Figure 16 (a) shows the side view of the evaporating meniscus on the rotating heated surface. Due to the motion of the heater surface, the leading edge of the meniscus had an advancing contact angle while the trailing edge of the meniscus had a receding contact angle. It was seen that the receding contact angle was lower than the advancing contact angle at lower heater surface temperatures but at higher surface temperatures the receding and advancing contact angles
were seen to be identical. Complementary numerical work by Mukherjee and Kandlikar [113] showed the convective pattern in the meniscus and the temperature map of the meniscus (Figure 16 (b)). It was postulated that due to the strong evaporation at the receding interface, the evaporation momentum force acting on the interface caused the receding contact angle to increase while the advancing contact angle remains relatively unchanged.

![Figure 16: a) Side view of the liquid meniscus on a rotating heater surface and [112] b) temperature field inside the meniscus [113].](image)

2.5 Research Needs

A review of some of the important works on the contact line region heat transfer is presented and some of the contradictions in the results are highlighted. Despite the large body of work available
on the nature of heat transfer during boiling, the exact mechanism and contribution of each mode still remains unclear. It is therefore important to study the heat transfer mechanism with measurement devices that are more accurate and have better spatial and temporal resolution in order to conclusively determine the role of microlayer evaporation, contact line heat transfer and microconvection during boiling.

Experiments conducted to determine the heat transfer mechanisms during boiling have largely been at lower heat fluxes due to the chaotic nature of boiling and limitations of the measuring devices used to visualize bubble growth and record the temperature of the heater surface. Future research should explore the contribution of each of the modes of heat transfer at higher heat fluxes as they may vary significantly from results currently presented. It is also important to study how the variation of fluid properties affects the contribution of each of the modes of heat transfer. Fluid properties such as thermal conductivity, viscosity and density are known to affect boiling heat transfer. However, an exhaustive study of heat transfer mechanisms for different fluids should be pursued.

A majority of the experimental and analytical works studying contact line heat transfer have been conducted under highly controlled conditions due to the complex nature of the problem. The parameters that were controlled include the kind of liquid used, its purity, the heater surface on which the contact line was established, the velocity with which the contact line moves, etc. In order to replicate conditions encountered in real life heat transfer processes, like boiling, experiments need to be conducted under conditions that are more representative of those seen in real life. This includes studying heat transfer at the contact line that is moving at high velocities like those seen during boiling—especially near CHF, heat transfer for complex fluids like water and ethanol where
the nature of intermolecular interaction between the liquid and the heater surface is significantly different, and the effect of surface roughness on contact line region and thin film evaporation.

Experimental techniques have largely attempted to map the local temperature or heat flux on the heater surface to determine the contribution of the various modes of heat transfer. However the lack of access to the bubble interface is an inherent limitation for most of these methods. The distribution of evaporation mass flux at the bubble interface can therefore only be obtained through numerical simulations. Further research is needed to experimentally determine the mass flux distribution at the bubble interface, especially at higher heat fluxes. This would help validate the numerical models that have been developed and provide insight into the onset of critical heat flux and help develop micro and nanostructured surface enhancements.
Chapter 3

3.0 Experimental Setup

Three different experimental setups were used for conducting pool boiling experiments; an open loop setup that is used to conduct experiments on a flat test section with distilled water as the working fluid, a closed loop setup that is used to conduct experiments with seawater on flat test sections and a test sections used to test boiling performance on tubular test sections.

3.1 Open Loop Setup

3.1.1 Experimental Setup

Figure 17: Schematic of the open loop pool boiling setup.
Figure 17 shows a schematic of the open loop pool boiling experimental setup. The main sub-sections of the setup are i) heater block, ii) test section, and iii) liquid bath. The heater block was fabricated using copper (C101-oxygen free) since it is an excellent conductor of heat. Four 200 W cartridge heaters were inserted at the base of the heater and were powered by a 240 V 10 A power source. The contact area between the heater and the test section was on top of a 40 mm tall 10 mm × 10 mm square pin that extended from the base of the heater. The heater was placed on a ceramic plate to prevent heat loss from the bottom of the heater and ceramic sleeves were used to prevent lateral heat loss. The heater block was placed on top of four compression springs to ensure uniform contact between the heater and the bottom of the test chip.

The test section is placed in a ceramic holder which is used to prevent lateral heat loss from the test section. Access holes are drilled through the ceramic holder to allow for thermocouples to be inserted into the test section. Three thermocouples were press fitted into holes drilled in the test section to measure the surface temperature and heat flux during boiling. The ceramic holder is placed on top of an aluminum plate that is mounted on stainless steel guide rods to allow easy vertical movement. The test section is lowered onto the heater block and a Grafoil ® sheet is used to reduce the interfacial contact resistance between the test section and the heater block.

A square fused quartz corvette is placed over the test surface to contain the water bath. Optically clear Quartz was used to allow for visualization of bubble nucleation and growth over the heater surface during boiling. An additional water reservoir was provided on top of the quartz corvette to ensure that sufficient liquid is present during the boiling tests. A 200 W cartridge heater was immersed in the water reservoir and used as an auxiliary heater to maintain the liquid bath at saturation temperature. The saturation temperature was measured using an Omega K type thermocouple that was immersed in the liquid. A National Instruments cDAQ-9172 chasse coupled
with a NI-9213 temperature measurement card was used for data acquisition. A LabVIEW virtual instrument was used to record the temperature measurements from the thermocouples and display the heat flux and wall superheat in real time.

### 3.1.2 Test Section

![Diagram of Test Section](image)

**Figure 18: Test section a) boiling area and b) thermocouple spacing.**

All the test sections were 17 mm × 17 mm with the central 10 mm × 10 mm being the active boiling area. On the heater side, the central 10 mm × 10 mm section has a 9 mm protrusion. Three 0.76 mm thermocouple holes were machined 3 mm apart to measure the heat flux and surface temperature. K-type Omega thermocouples were used to measure the temperature. The temperature gradient perpendicular to the heater surface was computed using the three point backward Taylors series approximation and is given by

\[
\frac{dT}{dx} = \frac{3T_1 - 4T_2 + T_3}{2\Delta x}
\]  
(28)
where \( T_1, T_2 \) and \( T_3 \) are the top, middle and bottom temperatures and \( \Delta x \) is the spacing between the thermocouples. The heat flux was computed using the Fourier 1-D conduction equation and is given by

\[
q'' = -k \frac{dT}{dx}
\]  

(29)

where \( k \) is the thermal conductivity of the test section. The temperature at the heater surface was obtained by extrapolating the temperature measured by the top thermocouple in the test section using Eq. 29

\[
T_{Wall} = T_1 - q'' \left( \frac{x_1}{k} \right)
\]  

(30)

where \( x_1 \) is the distance between the top thermocouple and the test section and \( T_{Wall} \) is the heater surface temperature.

3.1.3 Test Procedure

Prior to testing the boiling performance of a surface, the wettability and the roughness of each of the test surfaces were measured using a goniometer and a Laser confocal microscope respectively. Distilled water is boiled in a pot for about 20 minutes to degas the water and then added into the setup to check for any leaks in the system. The main and auxiliary heaters are switched on and the test is started when the bulk temperature reaches saturation temperature and the surface temperature is just above saturation temperature. The heat flux to the heater is controlled by varying the voltage applied across the main heaters. The power is increased in steps and the temperature was recorded when the fluctuations in the temperature were less than 0.1°C over 10 mins. The data for the thermocouples were recorded for a period of 10 seconds at a frequency of 2 Hz.
3.2 Closed Loop Setup

3.2.1 Experimental Setup

![Diagram of Closed Loop Test Setup](image)

**Figure 19: Closed loop test setup.**

A closed loop setup was used to test boiling with saline water (seawater) as the working fluid since the concentration of the dissolved salts had to be maintained throughout the test. Figure 19 shows the schematic of the test setup. The working fluid was housed in a cylindrical stainless steel chamber that is 100 mm in diameter. The flanges on the ends of the chamber were attached to the cylindrical chamber using C-clamps. O-rings were used to seal the flanges to the cylindrical chamber to ensure that the working fluid and the vapor generated during testing are contained within the setup. The top flange contained a thermocouple probe, a vacuum port, a pressure gauge and a liquid cooled condensing coil. The thermocouple and the pressure gauge were used to determine the temperature of the liquid and the system pressure respectively. The vacuum port was
used to evacuate the air present within the system in order to degas the working fluid. The system pressure was maintained by varying the temperature and flow rate of the water flowing in the condensing coil. The bottom flange contained an opening for the auxiliary heater and an opening for the test section and the heater assembly. A 120 VDC, 200 W auxiliary heater was used to maintain the liquid bath at saturation temperature during testing. The heater assembly was similar to the one described in Section 3.1 and consisted of four 120 VDC, 200 W cartridge heaters inserted in a copper block as shown in Figure 17. Due to geometric constraints of the setup, the thermocouples used to measure heat flux were placed on the heater block and a single thermocouple was placed on the test section to measure the wall superheat. The test section was placed in the setup inside a Garolite holder. The face of the test section, which is in contact with the heater, extends 8.5 mm from the boiling surface with a hole to measure the test section temperature located 7.5 mm below the boiling surface. The Garolite holder consisted of a lower part on which the test section was placed, an upper plate which is located over the test section, and a vertical Garolite barrier which is attached to the top plate. Garolite was used to fabricate the test section holder since it can withstand temperatures of up to 168°C and is a good insulator with a thermal conductivity of 0.27 W/m·K. The top plate had a 10 mm×10 mm opening over the boiling surface and was attached to the bottom flange using four threaded bolts.

3.2.2 Test Procedure

The system was charged with the test liquid and sealed. The auxiliary heater was switched until liquid temperature reached 90°C and the system pressure was reduced by 15 kPa using a vacuum pump in order to degas the liquid. The gases in the chamber were evacuated repeatedly to remove the dissolved gases from the liquid as the temperature of the liquid bath increased. The system was assumed to be degassed when the system pressure remained constant over an extended period of
1 hour. Once the system was degassed, the main heater was switched on and the power to the auxiliary heater was adjusted to maintain the liquid at saturation temperature.

The power supplied to the main heater was maintained until steady state was attained. The system was assumed to be in steady state when the fluctuations in the thermocouple readings were no greater than 0.1°C over a 10 min interval. The data from the thermocouples was recorded and averaged over a period of 10 seconds before changing the power supplied to the heater for the next data point.

The system was maintained at atmospheric pressure by adjusting the temperature and flow rate of the liquid circulating through the condenser coil. The power supplied to the auxiliary heater was reduced as the heat flux to the test section increased before being completely shut off at higher heat fluxes.

3.3 Tubular Test Section

3.3.1 Experimental Setup

![Figure 20: CAD rendering of test setup.](image)

Figure 20: CAD rendering of test setup.
The third experimental setup used was to test tubular surfaces with water as the working liquid. The setup used was designed and fabricated by Mehta [114]. The tests were conducted at atmospheric pressure and the vapor generated during the tests was vented to the atmosphere. A schematic of the test setup is shown in Figure 20. The setup is 200 mm × 200 mm × 125 mm. The setup consists of a central block, the test section assembly, auxiliary heaters, optically clear windows, and compression plates. The central block was 150 mm × 150 mm × 75 mm and is sandwiched between two 9.5 mm thick, high temperature resistant borosilicate glass. The glass windows were held in place against the central block using aluminium compression plates and M10 fasteners. Silicone gaskets were used on both sides of the glass plate to prevent leaks. A 200 W auxiliary heater was used to maintain the liquid bath at saturation temperature.

### 3.3.2 Test Section

![Figure 21: CAD rendering of test assembly.](image)

The test section was press fitted on the primary heater- a FIREROD ® cartridge heater from Watlow. The heater is rated for 400 W at 120 V. However, due to the high heat transfer coefficient
due to boiling, the heater can be operated at 190 V to deliver a total power of 1000 W. The heater has a diameter of 9.53 mm and a heated length of 19.05 mm. Figure 21 shows an exploded view of the test section assembly. The test section has an inner and outer diameter of 9.53 mm and 15 mm, respectively and is made of copper alloy C101. In order to ensure the contact resistance between the heater and the test section is minimized, a thermal interface material (Omega Therm 201) was used between the heater and the test section. Axial heat loss was minimized by using thermally insulating high temperature ceramic on either side of the test section. The assembly was fastened between aluminium plates and silicone gaskets were used to prevent leaks.

![Figure 21: Exploded view of test section assembly.](image)

**Figure 21: Exploded view of test section assembly.**

Four OMEGA K-type thermocouples were used to measure the temperature around the test section. The thermocouples were 2.5 mm under the heater surface and located at the center of the heater surface, as shown in Figure 22. The heat input into the test section is controlled by varying
the voltage applied across the primary cartridge heater. The heat supplied to the test section is given by

$$q_{in} = V \times I$$  \hspace{1cm} (31)$$

where $V$ is the voltage applied across the heater and $I$ is the current passing through the heater. While most of the heat supplied by the primary heater was conducted to the test surface, a small portion of the heat supplied is dissipated axially across the thermally insulating ceramic spacers.

Unlike boiling over a flat test section where the surface temperature is largely uniform, the surface temperature over a horizontally oriented tubular test section varies due to fundamental differences in bubble dynamics over the tubular surface. The variation in temperature between the top and the bottom thermocouples is negligible at low heat fluxes but at high heat fluxes, the difference between the recorded temperatures ranges between 0.5-2°C. The variation in the temperatures on the sides of the bubble were largely similar. The average of the temperatures measured by the four thermocouples is taken as the temperature within the test section and is given by

$$T_{ave} = \frac{T_1 + T_2 + T_3 + T_4}{4}$$  \hspace{1cm} (32)$$

The surface temperature is computed based on the average temperature recorded by the thermocouples and the radial heat flux determined using Eq. 32. 1-D radial conduction equation is solved to determine the surface temperature and is given by

$$T_W = T_{ave} - \left( q_r \times \frac{\ln(r_2/r_1)}{2\pi kl} \right)$$  \hspace{1cm} (33)$$

where $r_1$ and $r_2$ are the radial distance of the thermocouples and the heater surface, respectively. The bulk temperature is recorded by a K-type thermocouple inserted into the liquid pool.
3.3.3 Test Procedure

Distilled water is degassed before introducing it into the test setup as described in Section 3.1.3. The test procedure is similar to that described in Section 3.1.3. However, due to the high amount of power passing through the heater, the temperature spike resulting from the onset of CHF can overheat the primary heater and damage it. To prevent this, an auxiliary cooling loop similar to that described by Kalani [115] was set up. Pressurized cold water is contained in a reservoir and is connected through a solenoid valve to the test setup. When the temperature spike corresponding to CHF is recorded, the solenoid valve is opened and the cold water is released over the test section. The flow of cold water over the heater surface helps remove the vapor layer from the heater surface and the surface temperature begins to quickly decrease, thus preventing any damage to the test section or the primary heater.

3.4 Uncertainty Analysis

Uncertainty analysis was carried out for all the tests conducted. The two main sources of uncertainty are the bias error and the precision error. The cumulative error due to bias and precision errors is given by

\[ U_x = \sqrt{B_x^2 + P_x^2} \]  \hspace{1cm} (34)

where \( U_x \) is the uncertainty, \( B_x \) is the bias error and \( P_x \) is the precision error. Bias error, also known as systematic error arises due to errors in the temperature measured by the thermocouple. Bias error was seen to be the most significant source of uncertainty. Precision error is a result of variation in the data recorded due to unsteadiness. A list of the parameters whose errors were accounted for in the current work is given in Table 2.
Table 2: Parameters contributing to uncertainty.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Symbol</th>
<th>Units</th>
</tr>
</thead>
<tbody>
<tr>
<td>Temperature</td>
<td>T</td>
<td>°C</td>
</tr>
<tr>
<td>Heater material thermal conductivity</td>
<td>k</td>
<td>W/m°C</td>
</tr>
<tr>
<td>Thermocouple distance</td>
<td>Δx</td>
<td>m</td>
</tr>
<tr>
<td>Bubble diameter</td>
<td>d</td>
<td>m</td>
</tr>
<tr>
<td>Bubble displacement</td>
<td>D</td>
<td>m</td>
</tr>
</tbody>
</table>

To reduce the bias error, the thermocouples were calibrated using a hot-point thermocouple calibrator to reduce the measurement error to under 0.1°C. The precision uncertainty for the temperature measurements is reduced by recording temperature for a period of 10 seconds at a frequency of 5 Hz. The uncertainty in the thermocouple distance is measured by measuring the distance between the thermocouple holes using a laser confocal microscope that can measure distances with an accuracy of 100 nm. The uncertainty in the bubble diameter and displacement is a combination of the bias error (minimum distance the camera is capable of capturing) and precision error. The precision error is significantly higher than the bias error since bubbles do not have perfectly spherical surfaces. Multiple measurements of bubble diameter and displacement were taken for each bubble and the deviation in the measurements were used to compute the precision errors. The uncertainty of derived measurements such as heat flux, surface temperature and heat transfer coefficient were calculated using the propagation of error method which is given by
\[ U_p = \sqrt{\sum_{i=1}^{n} \left( \frac{\partial p}{\partial a_i} u_{ai} \right)^2} \]  

(35)

where \( U_p \) is the uncertainty in the derived parameter \( p \) and \( u_{ai} \) is the uncertainty of the measured parameter \( a_i \).
Chapter 4

4.0 Effect of Evaporation Momentum Force on Bubble Dynamics

The sudden expansion of an evaporating mass during a phase change process like boiling causes the liquid–vapor interface to experience a reactionary force known as evaporation momentum force. Previous publications [42] have illustrated that evaporation momentum force plays a critical role in the onset of CHF and can be used to generate separate liquid-vapor pathways by creating a non-uniform liquid temperature distribution around a bubble [116]. The following work is an analytical investigation of the distribution of evaporation momentum force experienced by a nucleating bubble, its effect on bubble growth rate and its role in determining the bubble departure trajectory. In the first part of the study, the effect of evaporation momentum force on bubble growth under uniform temperature conditions is modeled by modifying the bubble growth equation derived by Mikic et al. [1]. To account for the effect of evaporation momentum force on bubble growth, it is first expressed in terms of bubble growth rate. Subsequently, a first order approximation of the net evaporation momentum force experienced by a bubble subjected to an asymmetric temperature distribution along the heater surface is derived. The effect of non-uniform temperature distribution along the height of the bubble is also accounted for while computing the net force experienced by the bubble. The departure trajectory of the bubble due to the evaporation momentum force is determined and compared with experimental data using high speed imaging. The understanding from this analysis is implemented to improve the pool boiling performance over a tubular surface by strategically placing bubble diverters to prevent bubble agglomeration and thereby increase the heat transfer coefficient and critical heat flux.
4.1 Effect of Evaporation Momentum Force on Bubble Growth

4.1.1 Expression for Evaporation Momentum Pressure

Consider a liquid-vapor interface where the liquid approaches the interface with a velocity $v_l$ and the resultant vapor leaves the interface with a velocity $v_v$ relative to the interface, as shown in Figure 23. By applying a mass balance equation across an infinitesimal area (dA) of the interface, we get

$$\rho_l v_l dA = \rho_v v_v dA$$

(36)

Due to the difference in the liquid and vapor phase densities, the velocity of the vapor leaving the interface is much greater than the velocity of the approaching liquid. This causes an increase in the momentum of the evaporating mass across the interface. The magnitude of evaporation momentum force ($F_M$) experienced at the interface is equal to the rate of change of momentum of the evaporating mass. Expressing mass transfer across the bubble interface in terms of bubble growth rate, the corresponding evaporation momentum pressure ($P_M$) is given by
\[ P_M = \frac{F_M}{\text{Area}} = \frac{4\pi R^2 \dot{R} \rho_v}{4\pi R^2} u_l \left( \frac{u_v}{u_l} - 1 \right) \quad (37) \]

From the mass conservation equation (Eq. 36), the velocity ratio in Eq. 37 can be written in terms of liquid and vapor densities and liquid velocity can be written in terms of bubble growth rate. Equation 37 can therefore be expressed in terms of bubble growth rate as

\[ P_M = \dot{R} \rho_v \left( \frac{\dot{R} \rho_v}{\rho_l} \right) \left( \frac{\rho_l}{\rho_v} - 1 \right) \quad (38) \]

The density of the liquid under reduced pressures \((P/P_{\text{crit}}) \ll 1\) is orders of magnitude higher than the density of the vapor. For example, the density ratio for water boiling at atmospheric pressure \((P/P_{\text{crit}} = 0.0045)\) is 1623. Under such conditions, the term \(\left( \frac{\rho_l}{\rho_v} - 1 \right)\) can be approximated to \(\frac{\rho_l}{\rho_v}\).

This assumption is made in many bubble growth models including Mikic et al. [1]. The expression for evaporation momentum pressure can thus be written as

\[ P_M \approx \dot{R}^2 \rho_v \quad (39) \]

### 4.1.2 Effect of Evaporation Momentum Force on Bubble Growth Rate

To analyze bubble growth in the presence of evaporation momentum force, the bubble growth expression developed by Mikic et al. [1] is modified. Mikic et al. [1] arrived at an expression for bubble growth in the inertia controlled phase by equating the total kinetic energy of the liquid to the work done on the liquid by the bubble. The kinetic energy (KE) of the liquid at any instant is given by

\[ \text{K.E.} = \frac{1}{2} \rho_l \int_R^\infty v_l^2 4\pi r^2 dr \]

\[ \text{K.E.} = 2\pi \rho_l (\dot{R})^2 R^3 \]

63
Mikic et al. [1] computed the work \( W \) done by the bubble by multiplying the displacement of the interface with the force exerted by the bubble on the liquid. To include the effect of evaporation momentum force, the expression for work done is modified to include the additional work done on the liquid due to the evaporation momentum force. The modified expression is given by

\[
W = 4\pi \int_0^R (P_v - P_\infty) R^2 dR + 4\pi \int_0^R (\dot{R}^2 \rho_v) R^2 dR
\]

In the inertia controlled phase, the bubble growth rate is obtained by equating the kinetic energy of the liquid to the work done by the bubble. Therefore equating Eq. 40 and Eq. 41 we get

\[
\dot{R}^2 = b \frac{P_v - P_l}{(\rho_l - b \rho_v)}
\]

where \( b \) is the shape factor, and is equal to \( 2/3 \) for a spherical bubble and \( \pi/7 \) for a bubble on a horizontal surface. The Clausius-Clapeyron relation between pressure and temperature is used to express the pressure difference across the bubble in terms of temperature difference. The above equation can therefore be written as

\[
\dot{R}^2 = b \frac{T_v - T_{sat}}{(\rho_l - b \rho_v) T_{sat}} \rho_v h_f g
\]

Following Mikic et al.’s [1] approach, the above equation can be rewritten as

\[
\dot{R}^2 = A^2 \frac{T_v - T_{sat}}{\Delta T}
\]

where

\[
\Delta T = T_\infty - T_{sat}
\]
\[ A = \left( b \frac{h_f g \rho_v \Delta T}{(\rho_l - b \rho_v) T_{sat}} \right)^{\frac{1}{2}} \]

The term ‘A’ is identical to the term computed by Mikic et al. [1] except for the \(-b \rho_v\) term in the denominator. To predict the bubble growth rate in the phase controlled by heat diffusion, Mikic et al. used the bubble growth equation developed by Plesset and Zwick [2]. Since evaporation momentum force reduces considerably in this phase due to slower bubble growth rate and larger interfacial area, its effect in this phase is neglected. The bubble growth rate in the heat diffusion controlled regime is given by

\[
\dot{R} = \frac{1}{2} \frac{B}{\sqrt{t}} \left( \frac{T_{\infty} - T_v}{\Delta T} \right)
\]

where

\[
B = \left( \frac{12}{\pi \alpha} \right)^{\frac{1}{2}} Ja
\]

\[
Ja = \frac{\Delta T c_l \rho_l}{h_f g \rho_v}
\]

Combining the bubble growth equations given in Eq. 44 and Eq. 45, and integrating with respect to time, a non-dimensional expression for bubble radius was obtained and is given by

\[
R^+ = \frac{2}{3} \left( (t^+ + 1)^{\frac{3}{2}} - (t^+)^{\frac{3}{2}} \right)
\]

where

\[
R^+ = \frac{AR}{B^2}, \quad t^+ = \frac{A^2 t}{B^2}
\]

where \(R^+\) and \(t^+\) are non-dimensional terms which describe bubble radius and time, respectively.
To summarize, the effect of evaporation momentum force on bubble growth rate is determined by modifying the bubble growth expression developed by Mikic et al. [1]. In the inertia controlled phase, the work done by the bubble is modified to include the effect of evaporation momentum force. The additional term in the expression for work done by the bubble results in the term ‘A’ defined by Mikic et al. to be modified. Since the bubble growth rate reduces considerably in the heat diffusion controlled regime, evaporation momentum pressure experienced by the bubble reduces as the bubble continues to grow. The expression for the bubble growth rate in the heat diffusion controlled regime is therefore left unchanged. The final expression for bubble growth rate is identical to the one proposed by Mikic et al. [1] except for a changed term ‘A’.

Using the equations derived above, the bubble growth rate for water boiling at atmospheric pressure at different liquid superheats is computed. The bubble growth rate for liquid superheats ranging from 1°C to 20°C is evaluated with and without the effect of evaporation momentum force. Figure 24 shows the percentage change in the radius of a bubble as a function of time.

![Figure 24: Change in the bubble radius vs time computed using Eq. 46.](image-url)
It was seen that the change in bubble radius due to evaporation momentum force was greater at high wall superheats. This is because the evaporative mass flux at the bubble interface is higher as the wall superheat increases and therefore the evaporation momentum force experienced by the bubble is higher. However the increase in bubble radius due to the evaporation momentum force for all the simulated wall superheats was seen to be well under 1%. It is clear from the results that the magnitude of evaporation momentum force experienced by the bubble is considerably smaller than the force required to influence bubble growth rate in a significant manner. The results will hold true even for the case of a bubble on a heated surface with a non-uniform temperature field.

It will be shown in the following section that the evaporation momentum force experienced by the bubble increases considerably in a non-uniform temperature field like those observed when a bubble nucleates on a heated surface. However, it is expected that the effect of evaporation momentum force on bubble growth rate would remain negligible since the force required to noticeably change bubble growth rate would still be greater than the net evaporation momentum force experienced by the bubble. While the evaporation momentum force experienced by the bubble is not large enough to change the bubble growth rate, it will be shown in the following sections that the force is sufficient to alter the trajectory of the bubble growing in asymmetric temperature conditions.

### 4.2 Effect of Evaporation Momentum Force on Bubble Trajectory

#### 4.2.1 Effect of Thermal Boundary Layer on Evaporation Distribution

By considering a uniform distribution of evaporative mass flux across the bubble interface, the area over which evaporation momentum force is experienced increases. As a result, the magnitude of the force experienced by the bubble reduces. Consider a bubble growing on a horizontal heater surface. Due to heat transfer from the heater surface to the liquid through conduction, a thermal
boundary layer develops with liquid closer to the surface being at a higher temperature and the liquid outside the boundary layer remaining at saturation temperature. The temperature profile of the liquid around the bubble has a significant effect on the distribution of the evaporation rate on the surface of the bubble. Due to higher superheat, the liquid near the base of the bubble evaporates at a faster rate than liquid further away from the heater and therefore the evaporative mass flux at the bubble interface decreases as the distance from the heater surface increases. Since evaporation momentum pressure is proportional to the square of the mass flux, the average mass flux across the interface cannot be used to determine the evaporation momentum force acting on the bubble. In order to accurately predict the magnitude of evaporation momentum force acting on a bubble the local mass flux at the bubble interface has to be considered.

Determining the liquid temperature around the bubble requires knowledge of the thermal boundary layer thickness and the temperature distribution in the boundary layer. The thickness of the thermal boundary layer depends on the wall superheat and the waiting time between two nucleating bubbles. As the bubble grows in the boundary layer, microconvection around the bubble moves hot liquid away from the heater surface and recirculates over the interface. This effect was observed by Mukherjee and Kandlikar [117] while numerically simulating a single nucleating bubble on a heater surface. Figure 25 shows a contour map of the liquid temperature distribution during bubble growth obtained by Mukherjee and Kandlikar [117]. The liquid temperature is shown in non-dimensional terms varying from 0-1, 0 being saturation temperature and 1 being the heater surface temperature. It can be seen that the liquid near the base of the bubble is significantly hotter than the liquid towards the top of the bubble, which is at saturation temperature. It is also seen that the thermal boundary layer thickness away from the bubble is about 0.2 (non-dimensional height) but due to bubble growth, induced microconvection the superheated liquid raises to about
0.5 (non-dimensional height) near the bubble interface. Therefore to determine the evaporative mass flux at the bubble interface, the effective thermal boundary layer thickness, that is the height to which the superheated liquid surrounds the bubble, needs to be determined.

![Figure 25: Contour plot showing temperature distribution in the liquid during bubble growth [117].](image)

During the initial part of bubble growth, the bubble is within the thermal boundary layer and evaporation takes place along the entire height of the bubble, though the rate of evaporation is still significantly higher at the bottom of the bubble. However, as the bubble continues to grow, the height of the bubble exceeds the effective boundary layer and evaporation takes place only in the region within the effective boundary layer. The relation between the height of the bubble and the boundary layer thickness varies throughout the life of the bubble and is dependent on multiple factors. Numerical simulations of pool boiling by Dhir [118] show that the thermal boundary layer is significantly smaller than the bubble height for a large part of the bubble life. In the current
model, as a first order approximation, the effective boundary layer thickness is assumed to be proportional to the bubble height and the temperature distribution within the boundary layer is assumed to be linearly varying with distance.

4.2.2 Determining Evaporation Momentum Pressure under Asymmetric Temperature Conditions

To determine the variation of mass flux across the interface, the evaporative mass flux is assumed to be proportional to the local liquid superheat. For boiling taking place in a saturated liquid (zero subcooling), this corresponds to

\[ \eta = c \Delta T \]  (47)

where ‘c’ is the coefficient of proportionality and \( \Delta T \) is the local liquid superheat. The local liquid superheat depends on the temperature of the heater surface and the thickness of the boundary layer. The rate of change of mass of the bubble can be determined by integrating the mass flux across the interface and is given by

\[ \dot{m} = 2 \int_{90}^{\theta_c} c \Delta T \pi R^2 \cos \theta d\theta \]  (48)

The rate of change of mass of the bubble can also be expressed in terms of the bubble growth rate as

\[ \dot{m} = 4\pi R^2 \dot{R} \rho_v \]  (49)

The bubble growth rate (\( \dot{R} \)) can be determined using any bubble growth equation. Equating Eq. 48 and Eq. 49, the value of the proportionality constant (c) can be computed. Having determined the mass flux across the interface, the evaporation momentum pressure on the surface can be found using the expression for evaporation momentum pressure in terms of interfacial mass flux. The horizontal component of evaporation momentum force is computed and is given by
\[ F_M = \int_{-90}^{90} (c\Delta T)^2 (\rho_v^{-1} - \rho_i^{-1}) \pi R^2 \cos^2 \theta d\theta \]  

(50)

For this study the thickness of the effective thermal boundary layer, \( \delta \), is assumed to be a certain fraction of the bubble height \( Y_b \). The analysis is carried out for different \( \delta/Y_b \) ratios and the results are presented in Figure 26. It shows the variation of evaporation momentum pressure along the height of the bubble due to the temperature gradient. The X axis is the ratio of the height at a point on the bubble to the total height of the bubble. The Y axis is the ratio of the local evaporation momentum pressure at the point to the evaporation momentum pressure experienced by a uniformly heated bubble. The cases studied are boundary layers with a linear temperature distribution and an effective boundary layer thickness (\( \delta \)) of 0.70 \( Y_b \), 0.38 \( Y_b \) and 0.19 \( Y_b \). For water boiling on a copper surface (\( \theta_c = 53^\circ \)), a bubble height of 0.38 \( Y_b \) corresponds to the height of the bubble where the width of the bubble is the highest. It can be seen that when \( \delta = 0.19 Y_b \), the evaporation momentum pressure at the base of the bubble is 64 times the pressure experienced by a uniformly heated bubble. However, the evaporation momentum force is only experienced by a small region of the bubble. When the boundary layer thickness is increased to 0.38 \( Y_b \) the maximum pressure experienced by the bubble drops to 16 times the pressure experienced by a uniformly heated bubble and it further drops about 6 times when the boundary layer thickness is 0.7 \( Y_b \).
4.2.3 Determining Bubble Displacement Under Asymmetric Temperature Conditions

To determine the net evaporation momentum force acting on an asymmetrically heated bubble, the forces acting on a bubble subjected to two distinct temperatures is considered. Figure 27 shows a schematic of a bubble subjected to two different heater temperatures- $T_1$ and $T_2$ ($T_2 > T_1$). Along with the variation of liquid temperature on either side of the bubble, the effective boundary layer thickness and the temperature variation within the boundary layer is also considered. To determine the net horizontal force on the bubble, the force acting on each half of the bubble is computed. The

Figure 26: Variation of evaporation momentum pressure along the height of the bubble for different boundary layer thicknesses. PM multiplier is the ratio of the local evaporation momentum pressure to the evaporation momentum pressure experienced by a uniformly heated bubble.

It can be seen from Figure 26 that the temperature variation along the height of the bubble has an important role in deciding the magnitude and distribution of the evaporation momentum pressure experienced by the bubble. The role of evaporation momentum force can be mistakenly considered to be insignificant by taking the area averaged mass flux across the interface instead of considering the local mass flux as seen in the first part related to bubble growth equation.
bubble growth rate is determined for each half independently for the given wall superheat using the expression proposed by Mikic et al. [1].

The distribution of the mass flux is determined using Eqn. 48 after the bubble growth rate for both halves of the bubble is computed. Finally, the force experienced by each half of the bubble is calculated using Eq. 50. The difference between the two forces is taken as the net horizontal force acting on the bubble. An additional force that acts on the bubble that is asymmetrically heated is the thermocapillary or Marangoni force. This force arises due to the surface tension of the liquid being higher at the cooler side of the bubble than the hotter side. However, this is ignored since for the temperature differences that a nucleating bubble might experience, the magnitude of the change in surface tension is relatively small. In case of water at atmospheric conditions, a 5°C change in temperature changes the surface tension by only 2%. Since temperature differences of greater than a few degrees are not expected across a nucleating bubble Marangoni forces are not expected to influence the net horizontal force experienced by the bubble.

The average radius of a bubble subjected to two different wall superheats on the two sides is computed by taking the average of the volumes of the two halves of the bubble. Since the mass of

Figure 27: Schematic of a bubble subjected to an asymmetric temperature distribution and the resultant evaporation momentum force experienced by the bubble, T2>T1.
the bubble is very small, the force required to move the bubble is insignificant. However the bubble experiences a drag force opposing its motion. The expression for drag force on the bubble is

\[ F_D = \frac{1}{2} \rho l v^2 C_D A_b \]  

(51)

Since the drag force experienced by the bubble is orders of magnitude greater than the force required to accelerate the bubble to the required velocity, the drag due to the liquid is assumed to be the only force resisting bubble displacement. The velocity of the bubble is determined by equating the evaporation momentum force acting on the bubble to the drag force experienced by it. Sugioka and Komori [119] numerically computed the lift and drag coefficients for a bubble moving in a liquid. The variation of drag coefficient with respect to Reynolds number for uniform flow is shown in Figure 28.

![Figure 28: Drag coefficient of a bubble vs Reynolds number [119].](image)

Since there is a mutual dependence between the Reynolds Number and the drag coefficient, an iterative process is used to determine the drag coefficient by initially assuming a Reynolds number and varying it until the system is balanced.
4.2.4 Comparison with Experimental Observations

To validate the analytical model, an experiment where the nucleating bubbles can experience asymmetric temperature conditions was setup and the trajectory of nucleating bubbles was observed using a high speed camera. The test section consists of a grooved copper surface with the groove filled with an epoxy. Since the epoxy is a poor conductor of heat (k = 1W/m-K), the temperature of the liquid over the epoxy is lower than the temperature of the liquid above the heater. A bubble nucleating at the edge of the insulating epoxy and the copper surface is expected to move away from the epoxy surface as a result of the difference in the evaporation momentum forces acting on either side of the bubble. Figure 29 shows the schematic of the test section. The test section was developed by machining grooves that are 400 μm deep and 500 μm wide using CNC machining. The grooves were then filled with Poxy Pak™, which is a fast curing, insulating epoxy which can withstand high temperatures. The width of the groove was large enough to affect the temperature of the liquid next to a bubble nucleating at the edge of the groove.

![Schematic of the experimental test section consisting of a grooved copper surface filled with an insulating epoxy.](image)

Figure 29: Schematic of the experimental test section consisting of a grooved copper surface filled with an insulating epoxy.

After the epoxy was cured, the top surface was sanded and then polished to remove any epoxy outside the grooved area. Polishing ensured that the epoxy and the copper were at the same level
and the roughness of the copper was under 0.2 \( \mu \text{m} \) (Ra). By reducing the roughness of the copper surface the number of nucleation sites reduces which helps visualize the trajectory of the bubble and reduces bubble coalescence.

The temperature of the copper surface was measured using three thermocouples that were located under the test section as shown in Figure 29. The temperature gradient between the thermocouples was calculated using Taylor’s backward series approximation. Assuming 1-D conduction, the surface temperature could be estimated using the top thermocouple temperature \( (T_1) \) and the computed temperature gradient.

As the observed bubbles nucleated and grew at the edge of the copper-epoxy surface, one half of the bubble was surrounded by the cooler liquid which was over the epoxy surface and the other half was surrounded by the warmer liquid that was present over the copper surface. The entire bubble was seen to be over the copper surface throughout the life of the bubble, and hence the contact angles on both sides are seen to be the same. High speed images of the bubble being displaced showed a negligible variation in contact angle on either side of the bubble.

The experiments were conducted with the copper surface heated to 106°C. The bubble motion was visualized using a Photron FASTCAM® at 2000 fps. Figures 30 (a)-(e) show an image sequence depicting the horizontal displacement of a bubble as it grows and departs from the heated surface. The region to the right of the vertical dotted line is the epoxy surface and the region to the left of the dotted line is the copper surface. The bubble nucleates on the copper surface close to the edge of the copper-epoxy interface and can be seen 0.5 ms after nucleation in Figure 30 (a). No nucleation was observed on the epoxy surface due to a lack of nucleation sites and because the temperature of the epoxy surface was low. While the bubble nucleated at the edge of the copper-epoxy interface, the base of the bubble is seen to be entirely on the copper surface from bubble
nucleation until bubble departure. 1 ms after nucleation (Figure 30 (b)) the displacement of the bubble is approximately equal to the radius of the bubble. In the next 1.5 ms the bubble continues to move to the left (away from the epoxy surface) while still being attached to the heater surface. The bubble finally departs 2.5 ms after nucleation (Figure 30 (e)) after being displaced by about 140 μm from the nucleation site. Instead of growing at the nucleation site and departing vertically, the bubble was seen to move horizontally towards the copper surface before it departed. The net displacement of the bubble before departure (indicated in Figure 30 (e)) was significantly higher than the radius of the bubble thereby freeing the nucleation site for the next bubble. After departure, the bubble continued to move at an angle due to its inertia until the drag on the bubble prevented it from moving in the horizontal direction.
Figure 30: (a)-(e): Image sequence showing horizontal displacement of the bubble. The bubble is seen after nucleation shown by ‘A’ in the image and departs after 2.5 ms after being displaced by a distance ‘D’.

Accurate physical measurements of the temperature at the epoxy surface would be challenging since a thermocouple placed through the heater would alter surface temperature due to conduction and a thermocouple located over the heater surface in the liquid would be affected by the liquid temperature. Therefore to determine the temperature at the surface of the epoxy layer, a 2D CFD
simulation was conducted. Figure 31 (a) shows the schematic of the test section used for simulation. The lower boundary was subjected to a heat flux of 15 W/cm² and a convective boundary condition with a heat transfer coefficient of 25,000 W/m²K and a free stream temperature of 100°C was imposed on the upper surface. The heat transfer coefficient at the top surface was determined experimentally by measuring the copper surface temperature when a heat flux of 15 W/cm² was applied. Since nucleate boiling was seen at the applied heat flux, the heat transfer coefficient is significantly higher than typical values for natural convection. Figure 31 (b) shows the temperature contour near the top of the test section. The variation of temperature along the top surface is shown in Figure 31 (c). It was seen that when the copper surface temperature was 106°C the temperature over the surface of the epoxy was about 101°C.
Figure 31: (a) Schematic of the 2D model used to simulate surface temperature distribution, (b) temperature contour near the top of the test section and (c) temperature along the heater surface.

The observed bubble displacement was compared with the predicted displacement of the bubble from the model described above using Eqns. (47)-(51). From the CFD simulations, the wall superheats on the two sides of the bubble were taken to be 101°C and 106°C. The drag coefficient experienced by the bubble was determined based on the actual Reynolds number of the bubble and
an average drag coefficient of 0.4 was chosen. For the experimental observations, the displacement was taken to be the distance between the center of the bubble and the location where the bubble nucleated. Four separate bubble trajectories were measured from nucleation until bubble departure. Figure 32 shows the observed and experimentally observed bubble displacement and the predicted bubble trajectory for different boundary layer thicknesses.

![Bubble Displacement Diagram](image)

**Figure 32: Observed and predicted horizontal displacement of the bubble for different effective boundary layer thicknesses ($\delta$).**

It can be seen from Figure 32 that the actual displacement is close to the predicted displacement for $\delta = 0.38 Y_b$ for the first 2 ms and $\delta = 0.7 Y_b$ in the subsequent 1.5 ms. The results show that by accounting for the variation of evaporation along the height of the bubble, the net displacement of the bubble (and therefore the net force) can be predicted reasonably well. It is also clear that the effective boundary layer thickness over the life of the bubble is around half the bubble height. This is consistent with findings from Mukherjee and Kandlikar [117] where the effective boundary layer thickness was seen to be about $0.5 Y_b$. While the magnitude of the errors may vary depending...
on factors such as the liquid temperatures experienced by the bubble, the waiting time between the bubbles, the viscosity of the liquid and the roughness of the surface, the model would still be able to predict the displacement of the bubble reasonably well as shown in Figure 32.

4.2.5 Conclusion

The effect of evaporation momentum force on bubble growth rate and the bubble trajectory was studied analytically and experimentally. The salient conclusions from the works are

- The effect of evaporation momentum force on bubble growth rate was modeled by modifying the bubble growth model developed by Mikic et al. [1]. It was seen that at higher temperatures the effect of evaporation momentum force was larger. However, since evaporation momentum force under the assumption of uniform liquid temperature around the bubble is smaller than other forces that resist bubble growth such as surface tension and inertial forces, its effect on bubble growth rate is insignificant.

- A bubble growing in a thermal boundary layer experiences a non-uniform temperature field with the highest temperature near the base of the bubble. The evaporation momentum force at the interface as a result was seen to vary considerably along the height of the bubble.

- When the bubble is subjected to an asymmetric temperature distribution, a net evaporation momentum force towards the hotter side is observed. The magnitude of the force acting on the bubble depends on the bubble growth rate and the temperature difference across the sides of the bubble. This force acts primarily at the base of the bubble, and was large enough to displace the bubble along the heater surface toward the hotter region.

- If the temperature difference across the bubble is large enough, the displacement of the bubble can be greater than the radius of the bubble thereby freeing the nucleation site for
the next bubble. Additionally, the horizontal displacement of the bubble would create separate liquid-vapor pathways which would improve the boiling performance of the surface.
4.3 Bubble Induced Flow Field Modulation over Tubular Surfaces

Evaporators employing boiling on the outside of tubular surfaces are often used in refrigeration, air-conditioning, and process industries. Enhancement in both critical heat flux (CHF) and heat transfer coefficient (HTC) are desirable to improve the performance characteristics of these systems. An increase in HTC improves the chiller efficiency, while an increase in CHF results in compact evaporators and reduced refrigerant inventory. Boiling enhancements are conventionally achieved through surface modifications such as porous coatings [120,121], flow distribution methods [122–124], microchannels [114,125,126], and commercially available surfaces [127–129]. The enhancement techniques discussed above focus on increasing the HTC at low heat fluxes. However, simultaneous increases in HTC and CHF are desirable in numerous applications and have significant economic benefits.

The current work focuses on increasing both the HTC and CHF through a strategic flow modulation technique in the vicinity of a growing bubble by changing its trajectory and modifying the local two-phase flow structure. From the understanding gained about controlling bubble trajectory by creating asymmetric temperature conditions around a bubble, an enhancement feature is developed that modifies the trajectory of bubbles nucleating on a horizontal tubular surface to increase both the critical heat flux (CHF) and heat transfer coefficient (HTC). The CHF on a plain tube is shown to be triggered by a local dryout at the bottom of the tube due to vapor agglomeration. To mitigate this effect and delay CHF, the nucleating bubble trajectory is modified by incorporating a bubble diverter placed axially at the bottom of the tube. The nucleating bubble at the base of the diverter experiences a tangential evaporation momentum force (EMF) which causes the bubble to grow sideways away from the tube and avoid localized bubble patches that are responsible for CHF initiation.
4.3.1 Onset of CHF Over Plain Tubular Surfaces

As the heat flux increases, the rate of generation of vapor also increases and when the vapor removal rate can no longer match the vapor generation rate, the system reaches a critical heat flux. The onset of CHF is characterized by the formation of a vapor blanket over the heater surface, which is identified by the sudden increase in the surface temperature. In the case of boiling over a horizontal tubular surface, bubble dynamics of a bubble nucleating at the top of the tube is significantly different from a bubble nucleating at the bottom of the tube. Therefore the onset of CHF may not occur simultaneously throughout the heater surface and will be triggered where the vapor removal process is least effective. In order to identify the exact location where CHF is triggered, the circumferential temperature variation on a plain tube is used as guidance. The circumferential temperature variation measured at the top, bottom and the sides of the tube is shown in Figure 33. The experimental setup described in Section 3.3 was used for this study. As evident in the figure, the temperature spike corresponding to the formation of the vapor blanket is first observed at the bottom of the tube. The vapor front is seen to advance over the left side before engulfing the top and right side 2.4 s after the temperature spike observed at the bottom. It is therefore concluded that CHF is initiated at the bottom and the vapor front progressively grows to cover the rest of the tube. The principle reason for initiation of CHF at the bottom of the tube is the inefficient vapor removal. While a bubble nucleating at the top and sides of the tube moves away from the heater due to buoyancy, bubbles nucleating at the bottom of the tube remain on the heater surfaces for extended periods of time which leads to local dryout and CHF.
Figure 33: Temperature variation (5 Hz sampling frequency) along the circumference of the tube during the onset of CHF; temperature spike first observed at the bottom thermocouple.

In order to delay CHF initiation, it is necessary to modulate the vapor flow around the tube surface such that bubbles are rapidly removed from this location and a self-sustained flow field is established. In such an arrangement, referred to as separate liquid-vapor pathways, the bubble is deflected away from the heater surface, thereby permitting liquid to rewet the heater surface. The efficacy of these systems stem from the generation of separate liquid-vapor pathways. One of the configurations that resulted in significant enhancement in both CHF and HTC is the contoured fin geometry. This configuration exploits asymmetric evaporation momentum force (EMF), which is a reactionary force experienced at the bubble interface due to the sudden expansion of the evaporation mass, to control the bubble trajectory. Since the temperature of the fin is always lower than the temperature of the tube surface, a bubble nucleating at the base of the fin will experience
a higher liquid superheat away from the fin. As a result, the evaporation rate and consequently EMF, will be greater on the bubble interface away from the fin. This results in the lateral displacement of the bubble away from the fin as it grows.

### 4.3.2 Effect of Bubble Diverter on Bubble Trajectory

Realizing the potential of generating separate liquid-vapor pathways, an enhancement approach is proposed to deflect the bubbles away from the bottom of the heater surface. The circumferential temperature variation shown in Figure 33 is used as guidance in identifying the location where enhanced bubble removal would be most beneficial. Since the onset of CHF was seen to occur at the bottom of the tube due to inefficient vapor removal, it was hypothesized that placing a fin at that location would lead to a delayed onset of CHF. A 3-mm tall, 0.5-mm thick copper fin, referred to as a bubble diverter, was attached at the bottom of the plain tube. This was achieved by creating an axial groove at the location, and subsequently fitting the bubble diverter using epoxy. The epoxy layer (Loctite® epoxy, K=0.3 W/m-K) between the bubble diverter and the tube reduced the conduction of heat into the diverter. Therefore, direct heat transfer from the bubble diverter to the surrounding fluid was reduced. The height of the diverter is designed to be greater than the bubble height to prevent bubbles from engulfing the diverter. The thickness of the diverter was chosen based on limitations in fabricating the fin diverter. Thinner diverters are desirable as they will cover lesser area of the tube. Analysis of forces acting on the bubble is similar to that described in Section 4.2. The EMF experienced by the bubble is proportional to the square of the interfacial mass flux and is maximum during the initial periods of bubble growth [130]. This force increases with heat flux and is highly effective at higher heat fluxes, which reinforces a continuous rewetting pathway for liquid flow. This configuration helps maintain lower surface temperatures leading to remarkable improvements in HTCs at higher heat fluxes.
Figure 34 shows the ratio of the net EMF to the buoyancy force experienced by a bubble growing at the base of the bubble diverter. The insert image shows the direction in which the evaporation momentum force and buoyancy force acts on a bubble. As the net EMF increases, the lateral displacement of the bubble away from the diverter also increases. It can be seen that the EMF is over two orders of magnitude greater than the buoyancy force in the initial stages of bubble growth. As the bubble size increases, the net EMF decreases and becomes less than the buoyancy force. Therefore, the lateral velocity of the bubble is highest during the initial stages of bubble growth and reduces as the bubble grows due to reduced EMF and increased drag. When this force dominates over the typically observed buoyancy force with plain tubes, an efficient and self-sustained boiling configuration is achieved.

Figure 34: Ratio of EMF to buoyancy force for a bubble nucleating at the base of the diverter as a function of time for water at 5°C, 10°C and 15°C wall superheats; Insert: Direction in which evaporation momentum force and buoyancy acts on a bubble at the base.
Figure 35(a) shows the boiling performance achieved with a plain tube and a tube fitted with the bubble diverter with distilled water at atmospheric pressure. A CHF of 1100 kW/m² at a wall superheat of 16°C was achieved using the tube with the bubble diverter, which translated to an enhancement of ~65% when compared to a plain tube (see supplementary material for pool boiling curve and uncertainty analysis). Additionally, a ~75% increment in HTC was observed when compared to a plain tube at the corresponding CHF values.

**Figure 35**: (a) HTC vs heat flux for plain tube and tube with bubble diverter, and (b) circumferential temperature profile during onset of CHF for a tube with a bubble diverter.
In addition to the pool boiling performance characteristics, the circumferential temperature variation was also gathered as shown in Figure 35(b). The inclusion of the bubble diverter significantly altered the temperature profile when compared to the plain tube (Figure 33). In the case with the bubble diverter, the bottom thermocouple read the lowest temperature when compared to the other circumferential locations during the onset of CHF. This showed that the bubble reinforced liquid circulation significantly improved the wetting in the bottom regions of the tube through EMF directing bubbles away from the tube surface. The CHF is now seen to be initiated at the top surface of the tube. High speed images of the underlying bubble trajectory and rewetting mechanisms are discussed to further supplement the aforementioned analysis.

4.3.3 Visualization

![Figure 36: Bubble displacement captured using high speed imaging at 4000 fps. Bubble nucleation observed in (a) and displacement from the fin is shown in (b-d).](image)
Figure 36 shows the high-speed image sequence obtained at 4000 fps for a surface temperature of 104°C. Figure 36 (a) shows the location of the nucleation site at the intersection of the bubble diverter and the tube surface. Figure 36 (b-d) demonstrates the lateral displacement of the bubble towards the hotter tube surface. This trajectory is expected to be along the resultant force, where the EMF dominates over the buoyancy force in the initial stages of bubble growth, resulting in the sideways motion of the bubble away from the heater surface. The lateral displacement of the bubble is predicted using Eq. 47-51, and is shown in Figure 37. It can be seen that the bubble displacement was in good agreement with the theoretical prediction. This further reinforced that the lateral displacement of the bubble was governed by EMF. Since the force is highest during initial phase of bubble growth (Figure 34), the lateral velocity of the bubble is also highest during this phase and slows down as the size of the bubble increases. This effect is expected to be significantly more pronounced at higher temperatures experienced near CHF.

![Figure 37: Lateral displacement of bubble interface away from the bubble diverter.](image-url)
4.3.4 Applicability to Other Fluids

As discussed earlier, the forces that determine the trajectory of a bubble nucleating at the bottom of a tube are (i) EMF, which acts tangential to the heater surface and (ii) buoyancy that acts against gravity. A non-dimensional number, $K_3$, is introduced which is the ratio of EMF to buoyancy expressed in terms of the relevant property groups. This number can be beneficial in predicting the viability of the proposed technique for different working fluids.

$$K_3 = \frac{\left(\frac{q''}{h_{lw}}\right)^2 \frac{D}{\rho_v} \frac{\sigma}{(\rho_l - \rho_v)gD^2}}{\rho_v h_{lw}^2 (\sigma g (\rho_l - \rho_v))^2}$$

where $q''$ is the heat flux, $D$ is the bubble diameter, $h_{lw}$ is the latent heat of vaporization, $\sigma$ is the surface tension, and $g$ is acceleration due to gravity. The terms for EMF and buoyancy are identical to those used in Kandlikar [116] and the expression for bubble diameter is obtained using Fritz’s equation [131]. A higher value of $K_3$ indicates that EMF experienced by the bubble will be higher and therefore displacement of the bubble away from the nucleation site during bubble growth will be amplified. It can be seen from the above expression that $K_3$ is high for liquids with low latent heat (such as refrigerants) and at high heat fluxes. The value of $K_3$ for some common liquids corresponding to their plain surface CHF [132–134] is given in Table 3. All fluids except FC-87 are expected to result in similar or better performance improvement over their respective plain tube CHF and HTC values.

<table>
<thead>
<tr>
<th>Liquid</th>
<th>CHF (W/cm$^2$)</th>
<th>$K_3$</th>
</tr>
</thead>
<tbody>
<tr>
<td>Ethanol</td>
<td>50</td>
<td>0.0041</td>
</tr>
<tr>
<td>PP1</td>
<td>15.6</td>
<td>0.0044</td>
</tr>
</tbody>
</table>

Table 3: $K_3$ values for various fluids corresponding to reported CHF values [132–134].
4.3.5 Conclusion

- The circumferential temperature variation was used as a guide for placement of the bubble diverter. It was seen that the CHF is initiated at the bottom of a tube due to bubble agglomeration and the dryout front propagates upwards.

- A single bubble diverter is attached to the bottom of the tube surface to create an asymmetric temperature field around a bubble nucleating at the base of the diverter. The asymmetric temperature field influences the bubble trajectory and guides it away from the bottom of the tube. This effect is amplified at higher heat fluxes, thereby creating a continuous self-sustained boiling configuration.

- The pool boiling performance over a tubular surface with the bubble diverter increased the heat transfer coefficient by 75% and the critical heat flux by 50%.

- High speed visualization was used to measure the displacement of the bubble and the experimental values were seen to be in good agreement with the theoretical predictions.

- A non-dimensional number was introduced to provide guidance to develop future strategies based on the concept of separate liquid-vapor pathways. The non-dimensional number indicates that this mechanism can be extended to refrigerants, except for FC-87 (due to lower CHF), thereby realizing its benefits for adaptation in industrial applications. The concept of delaying CHF and increasing HTC through selective and strategic placement of...
enhancement features has been demonstrated here. Extension of this study to other regions of the tube, based on the liquid-vapor interaction can be beneficial in the quest to increase the boiling limits of HTC and CHF.
Chapter 5

5.0 Pool Boiling Enhancement through Contact Line Augmentation

The microstructured surfaces have been of much interest to researchers in recent years since they are able to significantly increase the CHF while having very small feature sizes. Based on the configuration of the microstructured surfaces, three main enhancement mechanisms have been proposed- i) roughness based model [135], ii) wicking based model [136] and iii) enhanced microlayer evaporation model [137,138]. The roughness model proposed by Chu et al. [135] states that roughness due to microstructures augments the capillary forces pinning the contact line of the bubbles and is responsible for the increase in CHF. Rahman et al. [136] proposed that liquid wicking to the dry spot under the bubble was responsible for the enhancement in CHF and a linear relationship between wicking rate and CHF was identified. Zou and Maroo [137,138] developed nano and micro ridged surfaces on a silicon substrate and proposed that fragmentation of the microlayer caused it to evaporate faster and resulted in higher CHF. A new heat transfer mechanism is proposed that increases the CHF by augmenting the effective contact line length under the base of a bubble.

5.1 Hypothesis

Since heat flux in the contact line region is significantly higher than the bulk heat flux [71,139], increasing the contact line length is an attractive approach to increase the CHF over a surface. By introducing additional contact line regions under the base of a bubble, it is anticipated that the evaporation rate would increase from these regions resulting in enhanced CHF. In the current work, additional contact line regions are generated using microgrooves.
Figure 38 (a) shows a schematic of bubble growth over a microgrooved surface and the generation of liquid menisci leading to the formation of additional contact line regions. Microgrooved surfaces are shallow parallel grooves that are under 100 μm deep. There are two sources contributing to the increase in the effective contact line length of the bubble. The first source is due to the longer bubble base perimeter due to the surface roughness effect [135] (wetted area/projected area). However, this contribution is small since the roughness factor of tested microgrooved surfaces is low (R=1.01-1.32) as compared to a range of 1.79-5.94 studied by Chu et al. [135]. The second source is the newly formed contact line regions under the bubble base. Since the depth of the shallow microgrooves is much smaller than the typical size of a nucleating bubble, the interface of a bubble can traverse over multiple grooves during bubble growth without getting pinned. As the bubble interface moves over a microgroove, a liquid meniscus will be left behind on both sides of the microgroove. The contact line regions thus created as a result of these liquid menisci lead to strong evaporation under the base of the bubble, thereby enhancing the CHF. The additional contact line regions introduced by the microgrooves do not contribute towards the force balance at the bubble interface and therefore do not alter the CHF mechanism. Therefore similar to a plain surface, CHF is believed to be triggered when the evaporation momentum force at the bubble interface overcomes the surface tension force experienced by the bubble [42].

Multiple experimental works using interferometry and other techniques have shown that the thickness of the microlayer is about 1 μm for water [138,140–142]. Numerical investigation of heat transfer in the contact line region by Zhou et al. [143] found that most of the evaporation in the contact line region takes place before the liquid film thickness becomes 2 μm thick. Zou et al. [138] used pillar heights of up to 5 μm to contain the microlayer. An additional liquid reservoir is however needed to replenish liquid that is evaporated in the contact line region throughout the
bubble growth cycle. Assuming a heat flux of 200 W/cm² over the meniscus region, the rate of evaporation of the liquid is estimated to be about 1 μm/ms. Therefore, a meniscus that is 10-20 μm tall is expected to have sufficient liquid to sustain contact line evaporation throughout the bubble cycle.
Figure 38 (a) Schematic of bubble growth over a microgrooved surface and generation of contact line at the base of the bubble and (b) bubble pinning observed on 100 micron deep microgrooved surface.

5.2 Experimental Results

In the current work, pool boiling experiments were conducted on 10 mm ×10 mm copper substrates with microgrooves that were 10, 20, and 100 μm deep and 100-500 μm wide. The width of the fins between two microgrooves was maintained at 25 μm for all the surfaces tested. The surface roughness was measured at different locations on the machined surfaces and was found to be between 2-3 μm for all chips and this is not expected to contribute towards any CHF enhancement [32,144]. The test sections were fabricated using CNC machining and water was used as the working fluid. Laser confocal images were captured for each of the surfaces tested to check the actual groove width and depth. No significant surface asperities were observed from the confocal images. The confocal images of some of the test sections are shown in Figure 39.

![Confocal images](image)

**Figure 39**: Confocal image of 400 μm wide 20 μm deep microgrooves and (b) 300 μm wide and 20 μm deep microgroove.

The CHF value for each of the surfaces tested is given in Table 4. The maximum uncertainty in the critical heat flux was 5.2 %. A pool boiling test was first conducted over a plain surface and a
CHF of 128 W/cm² was obtained at a wall superheat of 19°C. This agrees well with the established CHF value for water boiling over a smooth copper surface [30].

Table 4: CHF (wetted area) and wall superheat recorded for each of the surfaces tested and the predicted CHF using the roughness model.

<table>
<thead>
<tr>
<th>Nominal Channel Width (µm)</th>
<th>Nominal Channel Depth (µm)</th>
<th>CHF (Wetted Area) (W/cm²)</th>
<th>Wall superheat (°C)</th>
<th>Predicted CHF: Roughness model (W/cm²)</th>
</tr>
</thead>
<tbody>
<tr>
<td>100</td>
<td>10</td>
<td>121</td>
<td>20.4</td>
<td>108</td>
</tr>
<tr>
<td>200</td>
<td>10</td>
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<td>18.5</td>
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</tr>
<tr>
<td>300</td>
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<td>17.0</td>
<td>109</td>
</tr>
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<td>400</td>
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<td>153</td>
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<td>110</td>
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<td>500</td>
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<td>81</td>
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<td>61</td>
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<tr>
<td>300</td>
<td>100</td>
<td>108</td>
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<tr>
<td>500</td>
<td>100</td>
<td>124</td>
<td>20</td>
<td>91</td>
</tr>
</tbody>
</table>

The pool boiling performance for some of the tested surfaces is shown in Figure 40. The first number in the legend indicates the groove width and the second number indicates the groove depth. The uncertainty bars are given for one of the curves to avoid crowding of the plot. The uncertainty for the rest of the curves is similar to the one shown. The maximum CHF was observed for the 300
μm wide 20 μm deep grooved surface. The corresponding wall superheat and heat transfer coefficient were 16.9°C and 109 kW/cm²·K respectively.

![Pool boiling curves for some of the surfaces tested.](image)

**Figure 40:** Pool boiling curves for some of the surfaces tested.

The CHF data was fitted with existing models to predict CHF over structured surfaces, including the roughness [135], and the wicking rate [136] based models which under predicted the values by over 30%. Since the roughness factor for microgrooved surfaces is quite small (R=1.01 to 1.32), the CHF model by Chu et al. [135] is used to show that the increase in the surface tension force experienced at the bubble interface is marginal. The additional contact line length generated in the microgrooves does not affect the net surface tension force experienced by the bubble but contributes significantly to the enhancement in CHF. Therefore a roughness factor-based model to predict CHF under predicts the experimentally obtained CHF. With regard to the wicking-based models, most of the microgrooved surfaces have a high width-to-depth ratio and therefore are not good wicking surfaces. Therefore the evaporation of the liquid retained in the meniscus as the
bubble grows over the microgroove walls is believed to be responsible for the higher CHF and not the wicking of the liquid.

To understand the relation between the microgroove geometry and the CHF, the wetted area CHF was plotted against the groove width and is shown in Figure 41. Two distinct trends were observed in the variation of CHF with varying groove widths. In the case of shallow grooves (10, 20 μm deep) the CHF initially increased with an increase in groove width and reached a maximum CHF corresponding to a groove width of 300 μm. The CHF was seen to reduce with further increase in groove width. The maximum CHF based on the projected area was 187 W/cm$^2$ and was obtained with 300 μm wide, 20 μm deep grooves. The maximum CHF (wetted area) was 168 W/cm$^2$ seen with 300 μm wide, 10 μm deep grooves. Since heat transfer at the contact line is very efficient, the increase in CHF was coupled with an increase in the heat transfer coefficient to 109 kW/m$^2$-K.

In the case of the 100 μm deep grooves, the CHF (wetted area) was seen to consistently increase with an increase in groove width with the maximum CHF (wetted area) of 124 W/cm$^2$, which is less than the CHF over a plain surface. The nature of CHF variation over the microgrooved surfaces shows that two different heat transfer mechanisms are active and are influenced by the groove depth.
Figure 41: CHF (wetted area) vs groove width for the grooves tested.

5.3 Visualization

Figure 42 (a-f): Bubble growing over multiple grooves on a 200 μm wide, 20 μm deep grooves.
To identify the mechanism responsible for the variation of CHF for the different surfaces tested, bubble dynamics during boiling were visualized using a high speed camera at 4000 fps. The images were captured with the camera oriented both along and perpendicular to the groove length. The bubble diameters were seen to be similar along both directions with no noticeable asymmetry in the bubble shape. The geometry of the microgrooved surface significantly influenced bubble dynamics and bubble base diameter during departure. Firstly, the ability of a bubble to grow over multiple grooves was directly influenced by the depth of the grooves. Figure 42 shows bubble growth over a 200 μm wide, 20 μm deep microgrooved surface at a heat flux of 20 W/cm². It was seen that the bubble nucleates within a groove and as the size of the bubble increases, it grows over multiple grooves generating additional contact line regions on both sides of the fins under the bubble base. On the other hand, bubble growth over the deeper microgrooves (100 μm) is constrained within a single groove due to the pinning effect in the taller groove walls. This can be seen in Figure 43 which shows boiling over a 200 μm wide, 100 μm deep microgrooved surface. A bubble is seen to nucleate within the microgroove and is pinned by the microgroove walls before departing. Therefore, an increase in CHF (based on the projected area) is primarily due to the increase in the surface area. However, since the growth of the bubbles in the grooves is restricted due to pinning, the boiling process is not as efficient and the CHF (wetted area) is lower than that of a plain surface. This is consistent with results published by Cooke and Kandlikar [145] where microchannels with narrow channel widths resulted in lower CHF (wetted area) compared to a plain surface. The groove width was also an important consideration which affected bubble dynamics for shallow grooves (10 and 20 μm) as well. The bubble base area was smaller in the case of the narrow grooves and increased as the groove width increased with the maximum bubble base area observed in the case of 300 μm wide grooves. While shallow grooves affect the bubble
base diameter, they were not pinned within the microgroove where they nucleated. The reason for the variation in the base diameter can be due to fluid dynamics or heat transfer and merits further investigation. Further increase in groove width resulted in a reduction in the bubble diameter.

Figure 43, and (e-g) bubble constrained within a groove on a 200 μm wide 100 μm deep microgrooved surface (yellow dotted lines indicate the location of the groove walls).

5.4 Role of Effective Contact Line Length

To determine the role of the additional contact line regions on the CHF, the effective contact line length corresponding to a bubble nucleating over each of the surfaces was computed. The bubble diameter and bubble base area were measured for five distinct bubbles. All bubble diameters were recorded at a heat flux of $27 \pm 3$ W/cm$^2$. At higher heat fluxes, it was difficult to obtain clear views of individually growing bubbles. It is expected that the bubble growth will follow a similar trend at higher heat fluxes since a similar trend in contact line length vs CHF was observed at lower heat fluxes as well. The effective contact line length for a nucleating bubble is determined by measuring
the bubble base diameter from the high speed videos. Figure 44 shows a schematic of a bubble growing over a microgrooved surface.

Figure 44: Schematic of a bubble growing over a microgrooved surface and the additional contact line regions generated.

The total contact line length is the sum of the contact line length at the circumference of the bubble and the additional contact lines generated along the groove walls, and is a function of the bubble base area and the number of groove walls present per unit area. The additional contact line under the bubble is the product of the bubble base area and the total microgroove length per unit area and is given by

$$\text{CL length} = r^* \pi D_b + 2L \times \frac{\pi D_b^2}{4}$$  

where $r^*$ is the roughness of the surface, $D_b$ is the bubble departure diameter and $L$ is the microgroove length per unit area. The first term in the above equation corresponds to the perimeter of the bubble and the second term corresponds to the contact line regions generated due to the formation of liquid menisci along the groove walls. Figure 45 shows the variation of CHF (wetted area) as a function of the effective contact line length. The effective contact line length for narrow
grooves was relatively low despite having a large number of grooves per unit area because the bubble base area reduced for these cases. As the groove width increases, a large increase in the bubble size was observed resulting in an increase in the number of microgrooves under the bubble base. However, due to the reduction in the number of grooves per unit area, the effective contact line length for bubbles growing over wide grooves also reduced.

The CHF (wetted area) was seen to be linearly related (Figure 45) to the effective contact line length in shallow grooves with the highest CHF seen in case of the bubbles with the largest contact line length. Therefore the CHF enhancement over the shallow microgrooved surfaces can be attributed to the generation of additional contact line regions which act as reservoirs supplying liquid to an evaporating microlayer. In the deep microgrooves (100 µm), the bubbles are pinned within the grooves and no additional contact line regions are generated. Therefore the contact line augmentation mechanism to enhance CHF is not applicable, and area enhancement and liquid recirculation similar to that observed in conventional microchannels [145] are believed to be responsible for the CHF enhancement. This explains the different trend seen in the data points corresponding to the deep grooves shown in Figure 45.
Figure 45: CHF (wetted area) vs contact line length.

The variation of contact line length as a function of groove width is shown in Figure 46. It can be seen that the variation of contact line length closely matches the variation in the CHF (wetted area) with the longest effective contact line length corresponding to the highest CHF (wetted area).

Figure 46: Contact line length vs microgroove width.
5.5 Effect on Heat Transfer Coefficient

As the contact line length increases, the efficiency of heat transfer also increases. Figure 47 shows the variation of heat transfer coefficient as a function of contact line length for 100 to 300 μm groove widths for heat fluxes of 25 W/cm², 75 W/cm² and at CHF. It can be seen that in each case, the heat transfer coefficient increases as the contact line length increases. This indicates that the boiling process is indeed more efficient as the contact line length increases. It was also seen that the dependence of the heat transfer coefficient on contact line length increases (slope of the HTC vs contact line length plot becomes steeper) as the heat flux increases with the relation between the two being most significant at CHF. This is consistent with other studies which shows that the heat transfer in the contact line region increases as the wall superheat increases. This goes to show that the role of evaporation in the contact line region becomes more dominant as the wall superheat increases.

![Figure 47: Heat transfer coefficient vs contact line length.](image-url)
5.6 Conclusion

An experimental study of pool boiling over a microgrooved surfaces was conducted. The geometry relied on augmented contact line regions to improve the boiling performance over the heater surface. The following conclusions can be made from the study:

- The additional contact line regions generated under the base of the bubble as it grows over microgrooved surfaces was hypotheses to improve the critical heat flux and heat transfer coefficient. The depth of the microgrooves (10-100 μm) were designed such that they were small enough for bubbles to grow over the groove walls and large enough to contain enough liquid in the meniscus to sustain evaporation in the contact line region throughout the bubble cycle.
- The maximum CHF (projected area) of 187 W/cm$^2$ was recorded over the 20 μm deep, 300 μm wide grooves. The maximum CHF (wetted area) was 168 W/cm$^2$ seen with 300 μm wide, 10 μm deep grooves. This is the highest CHF (wetted area) for microstructured surfaces reported in literature.
- Shallow grooves resulted in higher CHF (wetted area) compared to deep microgrooves. In shallow grooves, the CHF (wetted area) increases as the groove width increases up to 300 μm groove width and reduces as groove width further increases. For deep grooves, the CHF linearly increased as the groove width increased with the maximum CHF (wetted area) of 124 W/cm$^2$.
- High speed visualization of bubble growth dynamics over microgrooved surfaces showed that bubbles grow over multiple grooves in shallow grooves and are pinned between the groove walls in the deep grooves.
• A linear relationship was found between the effective contact line length (calculated based on the high speed videos) and the CHF (wetted area) for the shallow grooves with the highest CHF corresponding to the largest effective contact line. The lower CHF over narrow microgrooved surfaces is due to the lower bubble base area over these surfaces.

• The effective contact line length is also directly related to the heat transfer coefficient with higher contact line length corresponding to higher heat transfer coefficient. This variation was more significant as the heat flux increased.

• The findings of this work could be used to design heat transfer surfaces that significantly enhance the contact line region contribution and CHF by placing shallow enhancement features on the surface of the heater.
Chapter 6

6.0 Effect of Surface Properties and Morphology on Pool Boiling

6.1 Effect of Thermophysical Properties of Heater Substrate on CHF

Understanding the importance of the properties of the boiling liquid, the heater surface and the interaction between them is crucial in understanding boiling heat transfer. The prediction of critical heat flux (CHF) over a heater surface is one of the most widely studied aspects of pool boiling. Over the years, several models have been proposed to explain the onset of CHF and predict the variation of CHF over a range of parameters. The hydrodynamic models incorporate the effects of the properties of the liquid and the interfacial models account for the effects of the wettability of the boiling liquid over the heater surface. While the surface effect models consider the effect of the thermophysical properties of the heater surface on CHF, a majority of studies focus on boiling over thin heaters (discussed in Section 2.2). These studies have shown that the CHF corresponding to a heater surface is dependent on the thickness of the heater, and the density, specific heat and thermal conductivity of the substrate material. The CHF reaches an asymptotic value as the thickness of the heater increases. However the role of the thermal properties of the heater material in determining the CHF over thick heater surfaces has not been explored. Thick heaters can be defined as heaters whose thickness is greater than the asymptotic heater thickness corresponding to the heater material and any further increase in the heater thickness does not influence the CHF over the surface. In the current work, the variation of the CHF corresponding to thick heaters as a function of material properties is studied.

To evaluate the effect of the thermophysical properties of the heater material on CHF, pool boiling tests were conducted with distilled water over six different heater materials that are horizontally
oriented at atmospheric pressures. The materials tested were aluminum (6061), brass, copper, carbon steel, Monel 400 and silver. In addition to these tests, CHF data available in literature for silicon is used. The materials tested have a wide range of thermal conductivities ranging from 40-430 W/m-K as well as a wide range of thermal mass \((\rho c_p)\) ranging from \(1.65 \times 10^3\) to \(3.85 \times 10^3\) kJ/m\(^3\)-K. The obtained CHF is mapped against various property groups to obtain the parameter that can best predict the variation of CHF. Finally, Kandlikar’s CHF model [42] is modified to account for the effect of thermal properties of the heater on CHF and a conjugate model that accounts for both the liquid and heater side effects is proposed.

6.1.1 Experimental Test Section

The experimental setup and the test section geometry is described in Section 3.1. The geometry of the silver test section was different to the general configuration due to material and manufacturing considerations. The silver test section was 1.5 mm thick and had a single thermocouple hole to measure the surface temperature. The heat flux was measured by placing three thermocouples, spaced 5 mm apart on the primary heater block.

To ensure that the CHF corresponding to the heaters tested is independent of the thickness of the heater, the asymptotic heater thicknesses for all the materials tested works are evaluated. Multiple works have found that the asymptotic heater thickness is a function of the thermal activity parameter and can be expressed as Eq. 54 [45,46,146]. Different values for the constant ‘A’ used in Eq. 54 have been proposed previously. The constant \(A=5\) Ws\(^{0.5}/(mK)\) is taken to determine the asymptotic heater thickness as proposed by Carvalho and Bergles [146]. Table 5 shows the thickness of the heater surfaces used as well as the predicted asymptotic heater thickness.

\[
\delta_{asy} = \frac{A}{(\rho c_p k)^{0.5}}
\]  

(54)
One parameter that is known to be a factor in determining the CHF of a surface is the microgeometry of the surface [147,148]. Roughness of the surface has been noted to be an important factor in influencing the CHF of a surface. In order to ensure that the roughness in not a factor in the current work a common machining and polishing technique was used for all the surfaces tested.

The test surface was polished to ensure artifacts of the machining process such as burrs, pits or grooves do not alter the observed CHF. In order to ensure that the microgeometry of the surface does not vary between the different substrates tested, the surface roughness of all the substrates is maintained at 1.1 μm (±0.3 μm) which represented the average arithmetic surface roughness (Ra). This range is low enough to ensure no observable variation in CHF due to surface features is observed. Table 5 shows the thermal properties of the different substrates tested and their respective roughness.
Table 5: Thermal properties and roughness of the substrates tested.

<table>
<thead>
<tr>
<th>Material</th>
<th>Density [kg/m³]</th>
<th>Specific Heat [J/(kg K)]</th>
<th>Thermal Conductivity [W/(m K)]</th>
<th>Asymptotic Heater Thickness (mm)</th>
<th>Heater Thickness (mm)</th>
<th>Arithmetic Roughness (µm)</th>
<th>Receding Contact Angle (°)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Aluminum (6061)</td>
<td>2,700</td>
<td>896</td>
<td>170</td>
<td>0.247</td>
<td>9</td>
<td>0.85</td>
<td>20</td>
</tr>
<tr>
<td>Brass</td>
<td>8,500</td>
<td>380</td>
<td>115</td>
<td>0.260</td>
<td>9</td>
<td>1.44</td>
<td>35</td>
</tr>
<tr>
<td>Carbon steel</td>
<td>7,870</td>
<td>486</td>
<td>52</td>
<td>0.355</td>
<td>9</td>
<td>1.21</td>
<td>26</td>
</tr>
<tr>
<td>Copper (101)</td>
<td>8,933</td>
<td>385</td>
<td>391</td>
<td>0.136</td>
<td>9</td>
<td>0.93</td>
<td>40</td>
</tr>
<tr>
<td>Monel 400</td>
<td>8,800</td>
<td>427</td>
<td>41*</td>
<td>0.393</td>
<td>9</td>
<td>1.12</td>
<td>27</td>
</tr>
<tr>
<td>Silver</td>
<td>10,500</td>
<td>235</td>
<td>429</td>
<td>0.154</td>
<td>1.5</td>
<td>1.4</td>
<td>21</td>
</tr>
<tr>
<td>Silicon [137]</td>
<td>2,330</td>
<td>712</td>
<td>148</td>
<td>0.319</td>
<td>0.5</td>
<td>0.15</td>
<td>25</td>
</tr>
</tbody>
</table>

* experimentally determined
6.1.2 Pool Boiling Results

Figure 48: Pool boiling curves for the surfaces tested. The red arrows indicate the last measured data point before CHF was observed.

The pool boiling curves for the various surfaces tested are shown in Figure 48. The heat flux was incremented in steps of 10 W/cm² in the initial part of the boiling curve and by less than 3 W/cm² when the heat flux was near CHF. The number of markers on the boiling curve in Figure 48 was reduced to prevent overcrowding of the plots. The highest CHF of 158 W/cm² was seen for the carbon steel substrate while lowest CHF was recorded in case of silver at 120 W/cm². The wall superheat also varied considerably with the highest wall superheat observed in case of Monel 400 at 35°C while the lowest recorded wall superheat before the onset of CHF was seen in silver at 19°C. The CHF over three of the surfaces - copper, silver and aluminum was tested again to check the repeatability of the obtained CHF. The CHF obtained was within 4 W/cm² of the results reported in Figure 48 which is less than the uncertainty of the CHF reported. In addition to the surfaces tested, the data of CHF over a plain silicon surface reported by Zou and Maroo [137] was also considered. While the roughness of a plain silicon surface is lower than the roughness of the other surfaces tested, a 0.6 μm (Ra) difference in roughness is not expected to significantly affect
the CHF of the surface. They reported a CHF 89 W/cm\(^2\) at a wall superheat of 26.7°C for the plain silicon surface. It can be seen that there is significant variation in the values of CHF as well as the temperature of the heater surface at CHF. The results reported here highlight the need to study the variation in CHF as a result of the thermal properties of the heater. Previous works studying the role of the thermal properties of the heater on CHF have either not explored the variation of asymptotic CHF for the different materials tested or have assumed that the asymptotic CHF is independent of the thermal properties of the heater [45,47,146].

6.1.3 Comparison with Hydrodynamic Models

Before attributing the variation in CHF to the thermal properties of the substrate, the effect of wettability/contact angle variation on CHF needs to be factored in. The receding angles corresponding to each of the surfaces tested is given in Table 5. Kandlikar’s model [42] is used to predict the effect of a receding contact angle on CHF. Figure 49 shows a comparison between the measured CHF for each of the surfaces tested and the predicted CHF corresponding to the contact angle for the respective surfaces. While the model was able to predict the CHF for three of the materials tested within an error margin of 10%, the model over predicted the CHF for silicon, silver, and aluminum substrates by 65%, 25% and 20% respectively. This variation in CHF was taken to be as a result of the material properties of the heater surface and, further analysis in this study was directed towards improving the accuracy of CHF prediction by incorporating the effects of heater material properties.
To understand the role of the material properties of the heater, the effect of the thermophysical properties of the heater needs to be isolated from the hydrodynamic and wettability effects. The CHF model by Kandlikar [42] is one of the most widely used models and accounts for the hydrodynamic and wettability effects. The experimental CHF is therefore normalized against the predicted CHF obtained using the receding contact angle corresponding to the substrates. Silicon and carbon steel exhibited the lowest and highest normalized CHF values, respectively. The variation of the normalized CHF was evaluated against a range of parameters such as $k/\rho c_p$, $\sqrt{k\rho c_p}$, $\rho c_p$ and $\sqrt{\rho c_p}$. Figure 50 (a-c) shows the normalized CHF as a function of thermal conductivity, thermal diffusivity and $\sqrt{\rho c_p}$ for each of the surfaces tested. Thermal conductivity was not a significant factor in determining the CHF of a substrate. It was seen that thermal diffusivity did not correlate well with the normalized CHF. The normalized CHF vs $\sqrt{\rho c_p}$ resulted
in the best fit with an $R^2$ value for a linear fit of 0.92. Therefore, there is a linear dependence between CHF and the parameter $\sqrt{\rho c_p}$.

Figure 50: Variation of normalized CHF as a function of a) thermal conductivity, b) thermal diffusivity, and c) thermal mass.

It is therefore clear that in addition to the hydrodynamic effects that have been widely studied by previous models, the effect of thermophysical properties of the heater surface also needs to be factored in to predict CHF. In order to account for the effect of the thermal properties of the substrate while predicting CHF, Kandlikar’s model was modified by adding a surface property
correction factor. The correction factor that is introduced has to be non-dimensional in order to balance the equation dimensionally. The correction factor introduced was the ratio of the thermal mass of the substrate to the thermal mass of a reference material. For the current work, the reference material was chosen as copper since the CHF of copper has been extensively reported in literature and is largely in agreement with the predicted CHF using Kandlikar’s model. Equation 55 (a) and (b) gives the proposed equation to predict the CHF. Regression analysis was carried out to determine the value of the exponent for the correction factor in Eq. 55 (a). A value of 0.5 resulted in the least average error for most of the surfaces considered in this study.

\[ q''_c = S^{0.5} \times h_{fg} \rho_g^2 \left( \frac{1}{16} \cos \theta \right) \left[ \frac{2}{\pi} + \frac{\pi}{4} (1 + \cos \theta) \right]^{1/2} \times \left[ \sigma g (\rho_l - \rho_g) \right]^{1/4} \]  

\[(55 \text{a})\]

where

\[ S = \frac{(\rho C_p)_{\text{Substrate}}}{(\rho C_p)_{\text{Cu}}} \]  

\[(55 \text{b})\]

For boiling over a copper surface, the correction factor is equal to one and the predicted CHF will remain unchanged. When the heater surface is made of materials with low thermal mass such as silicon and aluminum, the corrected CHF will be lower than the CHF predicted using Kandlikar’s model (Eq. 14) and when the heater surface is made of materials with high thermal mass such as Monel and steel, the corrected CHF prediction will be greater than the initial prediction. The experimentally obtained CHF and the predicted CHF using Eq. 55 is shown in Figure 51. It can be seen that the modified model accurately predicts the CHF for all the materials with an error of less than 15%. The correction was most significant for materials with low thermal masses, like silicon and silver, where the original model considerably over predicted the CHF.
6.1.4 Effect of Heater Properties on Wall Superheat

Another aspect that was also evaluated was the variation of wall superheat at CHF as a function of the thermophysical properties of the heater. Figure 52 shows the variation of wall superheat as a function of thermal mass, thermal conductivity and thermal diffusivity. Contrary to the trend seen in the variation of CHF, the role of thermal conductivity is prominent in the variation of wall superheat at CHF and the thermal mass of the heater material does not influence the wall superheat at CHF. The highest wall superheat was observed for the Monel 400 which has a high thermal mass but a low thermal conductivity while the lowest wall superheat was measured over the silver surface that has a high thermal conductivity and low wall superheat. This is consistent with findings by other works [148] that propose that for smooth heaters the heat transfer coefficient increases as the thermal conductivity of the heater material increases. This can be attributed to the ability of the heater to dissipate the heat from the dryout region under the bubble to the area around
the bubble that is still in contact with the bulk liquid. As a bubble grows over the heater surface, a dry spot is formed under the bubble. Due to the poor thermal conductivity of the vapor covering the dry spot, the local temperature of the heater under the dry spot begins to increase. For materials with high thermal conductivity, this increase in temperature is less significant while for materials with low thermal conductivity, the temperature increase is larger. This results in higher walls superheats and lower heat transfer coefficients for materials with low thermal conductivity.

Figure 52: Variation of wall superheat as a function of a) thermal mass, b) thermal conductivity, and c) thermal diffusivity.
6.1.5 Discussion

As discussed in previous sections, the role of thermal properties of the heater in determining CHF has been studied by many researchers and various models have been proposed to relate the heater properties to CHF. A majority of the works studying material properties have used thin heaters since the effect of the heater thickness and material properties are closely related. Sufficient data on the variation of asymptotic CHF for different heater materials is not available in literature. Therefore, to analyze and compare the results of the current study with findings from previous works, the thin heater data from previous works was extrapolated and the trends observed in the thin heater region are assumed to be valid for thick heaters as well.

Two aspects of the results from literature that are particularly important with respect to the current work are the thickness of the heater at which asymptotic CHF is reached and the relation between the thermal properties of the heater and CHF for the specific surface. Many researchers including Golobič and Bergles [45], and Carvalho and Bergles [146] have expressed the thickness of the heater at which asymptotic CHF is reached as a function of the parameter \((\rho c_p k)^{-0.5}\) of the heater material. Golobič and Bergles considered the asymptotic heater thickness to be the thickness when 90\% of the assumed maximum CHF was reached. However, the values presented are believed to be a good approximation of the actual asymptotic heater thicknesses. Closer analysis of the data presented by Golobič and Bergles [45] shows that while the asymptotic heater thickness increase as the parameter \((\rho c_p k)^{-0.5}\) increases, it is actually \(k^{-0.5}\) that is the parameter that affects the asymptotic heater thickness. The parameter \((\rho c_p)^{-0.5}\) independently does not show a clear correlation with asymptotic heater thickness. Figure 53 shows the variation of asymptotic heater thickness as a function of \((\rho c_p k)^{0.5}\) and \(k^{0.5}\). While there is a linear relationship between the
asymptotic heater thickness and \((\rho C_p k)^{0.5}\) the effect of \(k^{-0.5}\) is the operational part which showed a linear variation with an \(R^2\) value of 0.93 for the linear fit. The asymptotic heater thickness can therefore be taken to be proportional to \(k^{-0.5}\) where \(k\) is the thermal conductivity of the heater material.

Figure 53: Asymptotic heater thickness as a function of a) thermal activity parameter \((\rho C_p k)^{-0.5}\) and b) \(k^{-0.5}\) [45].
Many correlations have been proposed relating the thermal properties of the heater to CHF. The most widely used correlation involves relating the thermal activity parameter ($\delta\sqrt{\rho c_p k}$) to CHF. Experimental and numerical results from literature have shown that the thermal activity parameter is a fairly accurate indicator of the CHF over the heater surface. The expression proposed by Golobič and Bergles [45] relating the thermal activity parameter to CHF is given in Eq. 15. Arik and Bar-Cohen [47] proposed an alternate expression which yielded similar CHF values as the expression proposed by Golobič and Bergles. Broadly, the CHF is seen to vary linearly with the thermal activity parameter before the asymptotic CHF is reached. Therefore, the CHF of the surface can be taken to be proportional to the thermal activity parameter and can be expressed as

$$\frac{q''_{CHF}}{\delta(\rho c_p k)^{0.5}} = constant$$  \hspace{1cm} (56)

Equation 56 can be extended to predict the asymptotic CHF by considering the heater thickness to be the asymptotic heater thickness. Since the asymptotic heater thickness has been shown to be proportional to $k^{-0.5}$, the expression for the asymptotic CHF can be given by

$$\frac{q''_{CHF,asy}}{\delta_{asy}(\rho c_p k)^{0.5}} = constant \hspace{1cm} (57 \text{ a})$$

$$\frac{q''_{CHF,asy}}{(\rho c_p)^{0.5}} = constant \hspace{1cm} (57 \text{ b})$$

Therefore, by extending the trends that have been reported in literature and appropriately choosing the asymptotic heater thickness, it can be seen that that the proposed $\sqrt{\rho c_p}$ term to account for the variation in the thick heater CHF is indeed consistent with the trends reported in literature.
6.1.6 Conclusion

The CHF of seven different heaters at atmospheric pressure with water as the working fluid were compared. The following conclusions were drawn from the work:

- The CHF over thick heater surfaces varies based on the thermal properties of the heater material. The silicon heater was seen to have the lowest CHF of 89 W/cm$^2$ and the highest CHF was seen for the carbon steel substrate at 158 W/cm$^2$.
- The variation of CHF is seen to be dependent on the thermal mass of the heater material and is independent of the thermal conductivity of the heater material. The CHF corresponding to thick heaters was seen to be proportional to $\sqrt{\rho c_p}$.
- The thermal conductivity of the heater material is seen to affect the wall superheat at CHF. The higher the thermal conductivity, the lower the wall superheat during boiling. Silver, with a thermal conductivity of 429 W/m$^2$K resulted in a wall superheat of 19°C at CHF while Monel with a thermal conductivity of 41 W/m$^2$K has the highest wall superheat at 35°C.
- A non-dimensional correction factor was introduced in Kandlkar’s CHF model to account for the variation of the thermal properties of the heater. The correction factor is the ratio of the $\sqrt{\rho c_p}$ of the substrate with respect to copper. The modified expression was able to predict the CHF of the surfaces tested within 15% error.
6.2 Characterization of Pool Boiling Of Seawater and Regulation of Crystallization Fouling by Physical Aberration

As the availability of fresh water is becoming scarce, desalination of seawater is becoming increasingly important to meet the fresh water (portable water) requirements of the world. Thermal distillation continues to be one of the most important and widely used methods of desalination currently used. Scale formation, corrosion of the heater surface and the subsequent degradation of the heat exchanger is one of the biggest challenges in thermal desalination. In the current work, the pool boiling characteristics of artificial seawater at atmospheric pressure are examined. Characteristics of boiling such as heat transfer coefficient, critical heat flux and bubble growth rates are compared with the corresponding values for distilled water. The pool boiling curve is compared with the Rohsenow correlation. Subsequently, a passive method for preventing scale build-up by adding stainless steel beads into the liquid is explored. The process employed uses agitation of the stainless steel beads covering the heat transfer surface to break the fouling layer. The agitation of the liquid and the rising vapor bubbles causes the beads to be lifted away from the heater surface and drop back on to the surface due to their weight. The scales formed on the heater surface are mechanically removed when the beads impact the scales. The reduction of the scales on the heater surface is expected to improve the boiling performance over extended periods.

6.2.1 Background on Fouling

One of the principle challenges faced with respect to boiling of seawater is the fouling of the heater surface due to the precipitation of the dissolved salts, the low pH of seawater and presence of organic matter in water. Fouling due to the presence of seawater can be broadly classified into biological fouling, corrosion fouling and crystallization fouling [149]. Fouling of heat transfer surfaces results in reduced efficiency of heat transfer, in terms of higher wall superheat
requirements, thereby leading to significant economic losses. Corrosive fouling was one of the biggest challenges faced by the desalination plants that were built in the 1960s and this challenge was overcome due to the development of corrosion resistant metals, typically stainless steel and titanium [150,151]. Biofouling has been studied extensively and its effect can be reduced by pre-treating the water before passing through the distillation chambers [152–154]. From a heat transfer perspective, crystallization fouling is the most challenging to overcome and can result in large drops in heat transfer rates. While the solubility of most of the salts increases as the temperature of the liquid increases, the solubility of some salts such as calcium sulfate and calcium carbonate reduces [155]. This causes the salts to precipitate as the liquid near the heater surface superheats during boiling. Since the precipitated salts are poor conductors of heat, the build-up of the precipitate leads to a reduction in the heat transferred from the heater surface. While it is possible to limit the precipitation of calcium carbonate by maintaining an acidic pH, limiting the precipitation of calcium sulfate is not possible using this technique [156]. Due to the build-up of the scales on the heater surface, the resistance to heat transfer to the liquid increases and therefore the required wall superheat for a given heat flux also increases. Some of the techniques that are currently used to reduce the fouling on the heat transfer surfaces include using materials with low surface energies and introducing surface modifications. However these techniques are expensive to implement and challenges with reproducibility of the results have persisted [157,158]. Therefore, a method to eliminate or reduce the effect of precipitation and crystallization fouling is required to ensure that the heat transfer performance of the system remains high during operation of the desalination unit.
6.2.2 Test Setup

The Garolite holder consisted of a lower part on which the test section was placed, an upper plate which is located over the test section, and a vertical Garolite barrier which is attached to the top plate. Garolite was used to fabricate the test section holder since it can withstand temperatures of up to 168°C and is a good insulator with a thermal conductivity of 0.27 W/m-K. The top plate had a 10 mm × 10 mm opening over the boiling surface and was attached to the bottom flange using four threaded bolts/screws. The Garolite barrier was 30 mm tall with a 10 mm × 10 mm through hole in the center. The purpose of the Garolite barrier was to restrict the beads within the boiling region which would otherwise be pushed away from the boiling region to the adjacent horizontal surface of the test section due to the agitation of the liquid.

The stainless steel beads used in the experiment were spherical in shape with a diameter of 2 mm. Each of the beads weighed 0.032 g and was 8.1 times the density of water at saturation conditions. Stainless steel was chosen as a material of the beads as it is highly resistant to corrosion and has a high hardness index. The high hardness ensures that the beads are capable of chipping away the scales deposited on the heater surface upon impact. Preliminary tests conducted showed that covering the entire heater surface with the beads restricted liquid supply over the heater surface and thereby reduced the boiling performance. In order to ensure adequate supply of the liquid to the heater surface and efficient venting of the vapor generated, the number of beads added to the test section is limited such that only 25% of the boiling area is covered by a single layer of the beads.

6.2.3 Artificial Seawater

The concentration of the dissolved salts in seawater varies depending on the location from which the seawater sample is taken, the temperature and the pH of the water. It is therefore not possible
to conduct controlled pool boiling experiments using naturally available seawater. For the current work, experiments are conducted using standard synthetic seawater (ASTM D1141-98 (2013)) that is commercially available (produced by Lake Products Company LLC.). The composition of synthetic seawater is reported for all the dissolved salts whose concentration is greater than 0.0004%. Using artificial seawater ensures that the seawater tested is stable and the concentration of the dissolved salts are known and consistent. The total mass of the salts in 1 liter of water is 33.5g. This corresponds to a mass fraction of 0.034. The composition of the artificial seawater used in the experiment is given in Table 6.

**Table 6: Composition of synthetic seawater.**

<table>
<thead>
<tr>
<th>Salt</th>
<th>Concentration (g/L)</th>
</tr>
</thead>
<tbody>
<tr>
<td>NaCl</td>
<td>24.53</td>
</tr>
<tr>
<td>MgCl&lt;sub&gt;2&lt;/sub&gt;</td>
<td>5.20</td>
</tr>
<tr>
<td>Na&lt;sub&gt;2&lt;/sub&gt;SO&lt;sub&gt;4&lt;/sub&gt;</td>
<td>4.09</td>
</tr>
<tr>
<td>CaCl&lt;sub&gt;2&lt;/sub&gt;</td>
<td>1.16</td>
</tr>
<tr>
<td>NaHCO&lt;sub&gt;3&lt;/sub&gt;</td>
<td>0.201</td>
</tr>
<tr>
<td>KBr</td>
<td>0.101</td>
</tr>
<tr>
<td>H&lt;sub&gt;3&lt;/sub&gt;BO&lt;sub&gt;3&lt;/sub&gt;</td>
<td>0.027</td>
</tr>
<tr>
<td>SrCl&lt;sub&gt;2&lt;/sub&gt;</td>
<td>0.025</td>
</tr>
<tr>
<td>NaF</td>
<td>0.003</td>
</tr>
</tbody>
</table>
6.2.4 Pool Boiling Results

The experiments conducted can be classified into two groups. The initial set of experiments were conducted to characterize pool boiling of seawater with and without the beads. The second group of experiments conducted were to determine the pool boiling performance over extended periods of time at a constant heat flux.

![Pool boiling curve for seawater and distilled water.](image)

**Figure 54: Pool boiling curve for seawater and distilled water.**

Tewari et al. [159] studied the pool boiling characteristics of a sodium chloride solution at atmospheric and sub atmospheric conditions. The pool boiling performance was determined for extremely low heat fluxes between 1.5 W/cm\(^2\) to 9 W/cm\(^2\). The pool boiling performance of the sodium chloride solution was compared with distilled water and it was seen that for a given wall superheat, the heat flux increased as the concentration of sodium chloride increased at atmospheric pressures whereas the heat flux reduced as the salt concentration increased for lower pressures. However the range of heat fluxes tested were limited and data relating to fully developed nucleate
boiling regime or critical heat flux was not obtained. Figure 54 shows the pool boiling curves for distilled water and seawater over a plain, horizontal copper surface. The uncertainties associated with the readings are shown in the figure. The uncertainty bars are not shown in the subsequent plots to prevent overcrowding. The boiling performance of distilled water is used as a baseline to evaluate the pool boiling performance of seawater. For distilled water, a CHF of 129 W/cm$^2$ at a wall superheat of 19°C was observed. The resulting heat transfer coefficient was seen to be 67.8 kW/m$^2$-K. 

Boiling of seawater over a plain surface resulted in a CHF of 196 W/cm$^2$ at a wall superheat of about 29°C. This corresponds to a 52% increase in both the CHF and the wall superheat compared to distilled water. The average heat transfer coefficient for boiling seawater over a plain surface was 67.6 kW/m$^2$K. It is clear that the increase in CHF also corresponded to a proportional increase in the wall superheat.

6.2.5 Comparison with Rohsenow’s Correlation

Rohsenow [11] developed a model to predict the relation between the heat transferred during boiling and the wall superheat. The prediction was based on the agitation of the liquid as a result of bubble departure and the increase in the heat transfer coefficient due to the liquid circulation. The expression for heat transferred during pool boiling developed by Rohesnow is given by

$$\frac{C_p \Delta T}{h_{fg}} = C_{sf} \left[ \frac{q'''}{\mu_l h_{fg}} \sqrt{\left( \frac{\sigma}{g (\rho_l - \rho_v)} \right)} \right]^{0.33} Pr_l^{1.7}$$

(58)

where $C_{sf}$ is a coefficient specific to the liquid-solid combination. When considering boiling at a constant pressure, the thermal properties of the liquid can be assumed to remain constant. Equation 58 can therefore be written as
\[ \Delta T = \gamma q^\beta \]  

(59)

where \( \beta \) is 0.33 and \( \gamma \) is dependent on the thermal properties of the liquid and \( C_{sf} \). Trendlines were plotted to determine the values of \( \beta \) for boiling with seawater and distilled water. In the case of pool boiling of distilled water, the value of \( \beta \) was found to be 0.34 with a R\(^2\) value of 0.99 and in the case of boiling of seawater, \( \beta \) was seen to be 0.52 with the corresponding R\(^2\) value of 0.90. The value of \( \beta \) obtained for distilled water is close to the value of \( \beta \) proposed by Rohsenow and this suggested that the heat transfer coefficient for boiling of distilled water can be accurately predicted by Rohsenow’s model. However, the value of \( \beta \) obtained for boiling of seawater suggests that boiling, particularly at higher wall superheats, is not as efficient as the model proposed by Rohsenow. Extending the trendlines for boiling of distilled water, the wall superheat at a heat flux of 200 W/cm\(^2\) is predicted to be 23°C, which is significantly lower than the actual wall superheat experienced with seawater.

6.2.6 Visualization

High speed imaging of bubble growth was done using a Photron FASTCAM® at 2000 fps. Figure 55 shows the bubble growth rate for boiling on a plain surface using distilled water and seawater. The images in both the cases were captured at a heat flux of 13 W/cm\(^2\). The bubble growth rate as well as the bubble departure diameter were significantly higher in the case of distilled water compared to seawater. This is consistent with results of boiling in a binary system [160] where the bubble growth rate is affected by a liquid concentration gradient at the interface. Due to evaporation of water at the liquid–vapor interface of a bubble, the concentration of the dissolved salts around the interface increases. This causes the effective boiling point at the interface to increase. Therefore a larger temperature gradient is required in the boundary layer. As a result, the
bubble growth rate during boiling of seawater is significantly lower than that of distilled water at the same heat flux.

Apart from the small variation in the thermal properties of seawater compared to distilled water, boiling of seawater is predicted to be different from boiling of distilled water in two ways. As the seawater is heated and nucleate boiling occurs at the heater surface, precipitation of the dissolved salts in the seawater begins and the salts begin to deposit over the heater surface. The deposited salt layer, being uneven and filled with cavities, creates additional sites for bubble nucleation. The increase in the number of nucleation sites results in better boiling performance and higher values of CHF. However as the thickness of the deposited scales increases, the resistance to heat transfer from the heater surface to the liquid increases. Therefore in order to maintain the heat flux through the heater surface, the temperature of the heater surface under the deposited scales also increases. This results in an increase in the wall superheat. The net result of an increase in the nucleation sites and the increased resistance to heat transfer due to the formation of the scales is an increase in the CHF as well as the wall superheat.
6.2.6 Extended Boiling and Fouling Mitigation

When boiling of seawater is carried out over extended periods of time, the build-up of the scales can significantly reduce the boiling efficiency. To study the effect of scale build-up during boiling of seawater, boiling tests were carried out at high heat fluxes over extended periods of time and the variation of wall superheat over time was recorded. Figure 56 shows the variation of wall superheat over time as the heat flux is maintained at a constant level. In the case of boiling of seawater without beads, a constant heat flux of about 125 W/cm² was maintained and the wall superheat was seen to steadily increase over time. The wall superheat was seen to rise from 27°C initially to about 34°C towards the end of the test. This increase is consistent with the expected result due to the build-up of the scales. In order to reduce the build-up of scales on the surface of the heater, stainless steel beads are added to the liquid.

Figure 55: Bubble growth rate for distilled water and seawater at a heat flux of 13 W/cm².
Figure 56: Variation of wall superheat when the heat flux is maintained over 125 W/cm\(^2\) for a sustained period of time for seawater a) without beads and b) with beads.

When beads were added to the liquid bath, the wall superheat was seen to be significantly lower than without the beads. Additionally, the wall superheat was seen to reduce over time. The wall superheat at an average heat flux of 135 W/cm\(^2\) was initially 16.6 °C and was seen to drop to 12°C
after 840 minutes. The lowest wall superheat of 8°C was observed at 630 minutes. This indicated that the boiling of seawater resulted in the formation of additional nucleation sites for the bubbles while the presence of the beads ensured that the thickness of the scales on the heater surface remained small and didn’t significantly increase the thermal resistance to heat transfer. The variation in the wall superheat in the presence of beads shows that the bouncing beads periodically removed the scales from the heater surface. As the thickness of the scales increased, the thermal resistance to heat transfer increased and the wall superheat was seen to increase. After a certain thickness is reached, the scales appear to break away from the heater surface due to the bouncing beads - causing the wall superheat to drop.

The addition of beads therefore effectively handles the problem of scale build-up on a heater surface without the need for expensive chemicals or physical treatment. Since the thickness of the scales is regulated during active operation of the heater, the system would not have to be shut down to remove the built-up scales, thereby improving operational efficiency. This configuration however would not work when the test section is oriented vertically since the beads strike the test section due to gravity pulling it down. A parametric study regarding the material, shape and size of the beads as well as the number of beads added to the liquid can be conducted to determine the optimum configuration.
Figure 57: Laser confocal images of the heater surface a) before boiling, b) after a single pool boiling test c) after an extended boiling test without beads, and d) after an extended boiling test with beads.

To study the effect of scale formation on the roughness of the heater surface, the test surface was characterised using a laser confocal microscope. Figure 57 shows the laser confocal images of the test section before boiling and the scales formed on the test section after conducting the pool boiling and extended boiling tests. The roughness values of each of the surfaces is given in Table 7. Figures 57 (a) and 57 (b) show the laser intensity image of the heater surface before and after a single pool boiling test was conducted. The vertical grains on the heater surface are a result of the machining process employed to fabricate the test section. It can be seen that the roughness of the test section increased from 0.22 µm to 1.07 µm due to the deposition of the scales. The higher
roughness leads to an increase in the number of nucleation sites thereby enhancing the heat transfer performance. Figures 57 (c) and 57 (d) show the test section after the extended boiling tests were conducted without and with beads respectively. It can be seen that the presence of the beads limited the growth of the scales on the heater surface thereby leading to significantly lower wall superheats at a given heat flux.

<table>
<thead>
<tr>
<th>Test</th>
<th>Surface Roughness (Ra)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Plain surface(before boiling)</td>
<td>0.22</td>
</tr>
<tr>
<td>After pool boiling test</td>
<td>1.07</td>
</tr>
<tr>
<td>After extended boiling without beads</td>
<td>3.13</td>
</tr>
<tr>
<td>After extended boiling with beads</td>
<td>3.13</td>
</tr>
</tbody>
</table>

6.2.7 Conclusion

This work investigated the pool boiling performance of seawater on a plain horizontal surface, and a passive method to reduce crystallization fouling has been proposed. The salient conclusions from this work are given below:

- Boiling of seawater resulted in a CHF of 196 W/cm² at a wall superheat of 29°C. The enhancement in CHF compared to distilled water is expected to be due to the increase in the nucleation site density as a result of the scale formation. However, they also lead to an additional thermal resistance to heat transfer causing the wall superheat to increase.
- The results obtained were compared with the predicted boiling performance by Rohsenow’s correlation [11]. The value of $\beta$ in Rohesnow’s correlation was seen to be 0.52. This is considerably different from the value proposed by Rohsenow of 0.33.
Boiling of seawater over extended periods resulted in an increase in the wall superheat. The wall superheat for a heat flux of 125 W/cm$^2$ was seen to increase from 27°C to 34°C over 840 minutes of boiling. The wall superheat with the addition of beads was initially 16.6°C and reduced to 12°C after 840 minutes. This showed that the addition of beads reduced the effect of scale formation while increasing the heat transfer performance. The addition of beads can therefore be a cost effective strategy to handle the formation of scales on the heater surface without having to resort to more expensive chemical or physical treatments.
Chapter 7

7.0 Key Contribution and Future Recommendations

7.1 Contribution

Boiling heat transfer has a wide range of applications including electronics cooling, refrigeration and air conditioning, and power generation. Rapid advancements in the packaging of electronics has led to an ever increasing thermal load on the cooling systems. Traditional methods of thermal management including air and liquid cooling are not well suited to dissipate high heat fluxes and effectively cool hotspots. A fundamental understanding of the boiling phenomenon including understanding bubble dynamics, heat transfer mechanism and effect of heater surface properties and morphology can help design better two phase heat transfer systems. The first part of the current work examines the effect of evaporation momentum force on bubble growth rate and bubble trajectory, the second part describes an enhancement technique that utilizes contact line augmentation to increase the boiling performance and the last part of this dissertation is an investigation of the effect of thermophysical properties of the heater surface on CHF on boiling of seawater and the effect of scale formation on boiling heat transfer.

i. The effect of evaporation momentum force on bubble growth rate was analytically modeled by modifying the expression for bubble growth rate originally proposed by Mikic et al. [1]. It was seen that the effect of evaporation momentum force on bubble growth rate is insignificant. The trajectory of a bubble growing under asymmetric temperature conditions due to unbalanced evaporation momentum force was also modeled. The expression for the net evaporation momentum force experienced by the bubble was developed by incorporating the effects of variable evaporation rates due to the presence of a thermal
boundary layer and the asymmetric variation of liquid superheat around the bubble. The experimental results for the lateral trajectory of a bubble growing under asymmetric temperature conditions closely matched the predicted path.

ii. The insight from the work on modeling evaporation momentum force on bubble trajectory was used to develop strategically located surface enhancements on tubular surfaces. The bottom of a tube was identified as the location where CHF is triggered by analyzing the circumferential temperature profile around a plain tube. A bubble diverter was placed at the bottom of the tube to prevent bubble agglomeration and delay the onset of CHF. The addition of the diverter resulted in increasing the CHF by 60% and the HTC by 75%.

iii. A critical review of the heat transfer in the contact line region was conducted and the research needs were highlighted. A novel surface enhancement consisting of microgrooves 10-100 μm deep was developed that increases the pool boiling performance over a surface by augmenting the heat transfer in the contact line region. A maximum CHF of 187 W/cm$^2$ (projected area) and 167 W/cm$^2$ (wetted area) was recorded over the shallow grooves; this is among the highest CHF (wetted area) reported for microstructured surfaces in literature. High speed visualization was used to understand the relationship between the effective contact line length and the CHF over the surface. Increase in contact line length also corresponded to an increase in the heat transfer coefficient over the surfaces tested.

iv. The effect of thermophysical properties on CHF over a thick heater was experimentally evaluated. It was seen that the thermal mass of the heater influences the critical heat flux while the thermal conductivity of the heater has an effect on the wall superheat over the heater surface. A correction factor which is given by the ratio of the thermal mass of the heater material to the thermal mass of a reference material (copper) was included in the
CHF correlation developed by Kandlikar [42] to account for the thermophysical properties of the heater substrate. The corrected model was able to predict the CHF over all the surfaces tested with an error of <15%.

v. The pool boiling performance of seawater over a horizontally oriented, plain surface at atmospheric pressure was characterized. Boiling of seawater resulted in a CHF of 196 W/cm² at a wall superheat of 29°C. The increase in CHF was attributed to the increase in the number of nucleation sites due to scale formation over the heater surface. A passive method to reduce scale buildup during extended periods of boiling using stainless steel beads was tested. After 840 minutes of boiling at 125 W/cm² the wall superheat without the beads increased from 27°C to 34°C while it reduced from 16.6°C to 12°C with the beads.

vi. The findings from the work were published in 7 journal articles and 8 international conference proceedings.
7.2 Future Recommendations

The work discussed above studied a method to control the bubble dynamics during boiling to generate separate liquid-vapor pathways, increase the pool boiling performance over a heater surface by augmenting the contact line heat transfer and evaluated the effect of thermophysical properties of the heater surface and crystallization on CHF and pool boiling performance, respectively. Some possible directions for future research are presented below:

- **Applying the bubble displacement model to design surface enhancements**: Similar to the approach followed in developing the bubble diverters to enhance heat transfer over tubular surfaces, the analytical model developed to predict bubble displacement can be used to design surface enhancements that create separate liquid-vapor pathways.

- **Conduct a fundamental study of heat transfer in the contact line region**: A fundamental evaluation of heat transfer mechanism in the contact line region and the effect of a dynamic contact line on heat transfer can be conducted. A majority of the prior work on heat transfer in the contact line region has been conducted on quasi-static contact lines or during boiling. Studying heat transfer in a dynamic contact line where the contact line velocity can be accurately controlled can potentially provide substantial insight into the mechanism of heat transfer in the contact line region.

- **Couple microgrooves with other enhancement techniques**: It has been shown in the current work that the pool boiling performance can be significantly enhanced using microgrooved surfaces that are 10-20 μm tall. Since the feature size of these enhancements are small compared to traditional enhancement structures such as microchannels (feature sizes up to 500 μm), it is possible to design surfaces that have microgrooves fabricated over other
enhancement structures. By coupling two different enhancement techniques with complimentary enhancement mechanisms the boiling performance can be increased further.

- **Predict effect of thermophysical properties of the heater surface numerically**: The effect of thermophysical properties of the heater surface on boiling can be further investigated by numerically or analytically modeling the transient heat conduction within the heater surface. This can be done by either considering a single bubble and studying the temperature variation under the bubble surface or by performing a complete numerical simulation of boiling over a heater surface and considering the interaction between bubbles from adjacent nucleation sites. One challenge that can be foreseen is determining the relationship between the heater surface temperature and the CHF since the wall superheat at CHF also varies as a function of the material properties of the heater.

- **Parametric study on reducing fouling through physical aberration**: Using the energy of the rising vapor bubbles to reduce fouling on a heater surface through physical aberration has been shown to be an effective technique. An exhaustive study of the effect of the packing density of the beads, size of the beads and the shape and hardness of the beads on boiling performance and foiling mitigation can also be evaluated.
8.0 References


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9.0 Appendix

9.1 Uncertainty Derivation

The uncertainty for all relevant experimental data described in Section 3.0 is shown below

9.1.1 Heat Flux Uncertainty

The heat flux is determined using the temperature measurement by three thermocouples and is given by

\[ q'' = -k_{Cu} \left( \frac{3T_1 - 4T_2 + T_3}{2\Delta x} \right) \] (1)

The error propagation due to each of the terms above is shown below

\[ \frac{U_{q''}}{q''} = \sqrt{\left( \frac{\partial q''}{\partial k_{Cu}} U_{k_{Cu}} \right)^2 + \left( \frac{\partial q''}{\partial \Delta x} U_{\Delta x} \right)^2 + \left( \frac{\partial q''}{\partial T_1} U_{T_1} \right)^2 + \left( \frac{\partial q''}{\partial T_2} U_{T_2} \right)^2 + \left( \frac{\partial q''}{\partial T_3} U_{T_3} \right)^2} \] (2)

To simplify the terms in the subsequent derivation a variable \( \alpha \) is defined as

\[ \alpha = 3T_1 - 4T_2 + T_3 \] (3)

The sensitivity coefficients for each of the terms in Eq. 2 is given below

\[ \frac{\partial q''}{\partial k_{Cu}} = -\frac{\alpha}{2\Delta x} = \frac{q''}{k_{Cu}} \] (4)

\[ \frac{\partial q''}{\partial \Delta x} = -k_{Cu} \frac{\alpha}{2\Delta x^2} = -\frac{q''}{\Delta x} \] (5)

\[ \frac{\partial q''}{\partial T_2} = -k_{Cu} \frac{4}{2\Delta x} = -\frac{4q''}{\alpha} \] (6)
\[
\frac{\partial q''}{\partial T_3} = -\frac{k_{Cu}}{2\Delta x} = -\frac{q''}{\alpha}
\]  

(7)

Substituting the individual sensitivity coefficients into Eq 2 the following expression is obtained

\[
\frac{U_{q''}}{q''} = \sqrt{\left(\frac{q''}{k_{Cu}U_{k_{Cu}}}\right)^2 + \left(-\frac{q''}{\Delta d}U_{\Delta d}\right)^2 + \left(\frac{3q''}{\alpha}U_{T_1}\right)^2 + \left(-\frac{4q''}{\alpha}U_{T_2}\right)^2 + \left(\frac{q''}{\alpha}U_{T_3}\right)^2}
\]

(8)

Upon simplification of Eq. 8, the final expression for uncertainty of heat flux is obtained and is given by

\[
\frac{U_{q''}}{q''} = \sqrt{\left(\frac{U_{k_{Cu}}}{k_{Cu}^2}\right)^2 + \left(\frac{U_{\Delta d}}{\Delta d}\right)^2 + \left(\frac{9U_{T_1}}{\alpha^2}\right)^2 + \left(\frac{16U_{T_2}}{\alpha^2}\right)^2 + \left(\frac{U_{T_3}}{\alpha^2}\right)^2}
\]

(9)

9.1.2 Chip surface temperature derivation

The chip surface temperature is determined by extrapolating the top thermocouple temperature based on the temperature gradient measured between the thermocouples. The uncertainty in surface temperature is given by

\[
\frac{U_{T_S}}{T_S} = \sqrt{\left(\frac{\partial T_S}{\partial T_1}U_{T_1}\right)^2 + \left(\frac{\partial T_S}{\partial q''}U_{q''}\right)^2 + \left(\frac{\partial T_S}{\partial \Delta d}U_{\Delta d}\right)^2 + \left(\frac{\partial T_S}{\partial k_{Cu}}U_{k_{Cu}}\right)^2}
\]

(10)

The sensitivity coefficients for each of the terms in Eq. 10 is given below

\[
\frac{\partial T_S}{\partial T_1} = 1 = \frac{T_S}{T_S}
\]

(11)
\[
\frac{\partial T_s}{\partial q''} = -\left( \frac{d_1}{k_{Cu}} \right) = -\left( \frac{d_1}{k_{Cu}} \right) \left( \frac{T_s}{T_s} \right) 
\]  

(12)

\[
\frac{\partial T_s}{\partial d_1} = -\frac{q''}{k_{Cu}} = -\frac{q''}{k_{Cu}} \left( \frac{T_s}{T_s} \right) 
\]  

(13)

\[
\frac{\partial T_s}{\partial k_{Cu}} = -q''d_1 = -q''d_1 \left( \frac{T_s}{T_s} \right) 
\]  

(14)

Substituting the sensitivity coefficients in Eq. 11-15 in Eq. 10, the following expression is obtained

\[
\frac{U_{T_s}}{T_s} = \sqrt{\left( \frac{T_s}{T_s} \frac{U_{T_1}}{T_s} \right)^2 + \left( -\left( \frac{d_1}{k_{Cu}} \right) \left( \frac{T_s}{T_s} \right) U_q'' \right)^2 + \left( -\frac{q''}{k_{Cu}} \left( \frac{T_s}{T_s} \right) U_{d_1} \right)^2 + \left( -q''d_1 \left( \frac{T_s}{T_s} \right) U_{k_{Cu}} \right)^2} 
\]  

(15)

Upon simplification of Eq. 15, the final expression for uncertainty in chip temperature is obtained and is given by

\[
\frac{U_{T_s}}{T_s} = \sqrt{\frac{U_{T_1}^2}{T_s^2} + \frac{U_{q''d_1}^2}{k_{Cu}^2 T_s^2} + \frac{U_{q''}^2 q''^2}{k_{Cu}^2 T_s^2} + \frac{U_{k_{Cu}q''d_1}^2}{k_{Cu}^2 T_s^2}} 
\]  

(16)

9.1.3 Heat transfer coefficient uncertainty

The heat transfer coefficient is obtained by dividing the heat flux by the wall superheat (computed using the chip temperature). Therefore the uncertainty in the heat transfer coefficient comprises of the uncertainties in heat flux and wall superheat and is given by

\[
\frac{U_h}{h} = \sqrt{\left( \frac{\partial h}{\partial q''} U_{q''} \right)^2 + \left( \frac{\partial h}{\partial T_s} U_{T_s} \right)^2 + \left( \frac{\partial h}{\partial T_{sat}} U_{T_{sat}} \right)^2} 
\]  

(17)
The sensitivity coefficients for each of the terms in Eq. 17 are given below

\[
\frac{\partial h}{\partial q'} = \frac{1}{T_s - T_{sat}} = \frac{h}{q''}
\]

(18)

\[
\frac{\partial h}{\partial T_s} = -\frac{q}{(T_s - T_{sat})^2} = -\frac{h}{(T_s - T_{sat})}
\]

(19)

\[
\frac{\partial h}{\partial T_{sat}} = \frac{q}{(T_s - T_{sat})^2} = \frac{h}{(T_s - T_{sat})}
\]

(20)

Substituting the sensitivity coefficients described in Eq. 18-20 in Eq. 17, the uncertainty in heat transfer coefficient can be expressed as

\[
\frac{U_h}{h} = \sqrt{\left(\frac{h}{q''}U_{q''}\right)^2 + \left(-\frac{h}{(T_s - T_{sat})}U_{T_s}\right)^2 + \left(\frac{h}{(T_s - T_{sat})}U_{T_{sat}}\right)^2}
\]

(21)

Upon simplification of Eq. 21, the final expression for uncertainty in heat transfer coefficient is obtained and is given by

\[
\frac{U_h}{h} = \sqrt{\frac{U_{q''}^2}{q''^2} + \frac{U_{T_s}^2}{\Delta T_{sat}^2} + \frac{U_{T_{sat}}^2}{\Delta T_{sat}^2}}
\]

(22)
9.2 Publication list

9.2.1 Journal Papers


9.2.2 Conference Papers


