Jet Impingement Heat Transfer from Superheated, Superhydrophobic Surfaces

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Jet Impingement Heat Transfer from Superheated, Superhydrophobic Surfaces

David Jacob Butterfield

A thesis submitted to the faculty of Brigham Young University in partial fulfillment of the requirements for the degree of

Master of Science

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ABSTRACT

Jet Impingement Heat Transfer from Superheated, Superhydrophobic Surfaces

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Master of Science

Liquid jet impingement is a technique ubiquitously used to rapidly remove large amounts of heat from a surface. Several different regions of heat transfer spanning from forced convection to nucleate, transition, and film boiling can occur very near to one other both temporally and spatially in quenching or high wall heat flux scenarios. Heat transfer involving jet impingement has previously shown dependency both on jet characteristics such as flow rate and temperature as well as surface material properties.

Water droplets are known to bead up upon contact with superhydrophobic (SH) surfaces. This is due to reduced surface attraction caused by micro- or nanostructures that, combined with a natively hydrophobic surface chemistry, reduce liquid-solid contact area and attraction, promoting droplet mobility. This remarkable capability possessed by SH surfaces has been studied in depth due to its potential for self-cleaning and shear reduction, but previous research regarding heat transfer on such surfaces shows that it has varying effects on thermal transport.

This thesis investigates the effect that quenching initially hot SH surfaces by water jet impingement has on heat transfer, particularly regarding phase change. Two comparative studies are presented. The first examines differences in transient heat transfer from hydrophilic, hydrophobic, and SH surfaces over a range of initial surface temperatures and with jets of varying Reynolds number ($Re_D$), modified by adjusting flow rate. Comparisons of instantaneous local heat flux from the surfaces are made by performing an energy balance over differential control volumes across the surfaces. General trends show increased heat flux, jet spreading velocity and maximum jet spread radius when $Re_D$ is increased. An increase in initial surface temperature resulted in increased heat flux across all surfaces, but slowed jet spreading. The local heat flux, average heat rate, and total thermal energy transfer from the surface all confirmed that SH surfaces allow significantly less heat to transfer to the jet compared to hydrophilic surfaces, due to the enhanced Leidenfrost condition and reduced liquid-solid contact on SH surfaces which augments thermal resistance.

The second study compares jet impingement heat transfer from SH surfaces of varying microstructures. Similar thermal effects due to modified jet $Re_D$ and initial surface temperature were observed. Modifying geometric pattern from microposts to microholes, altering cavity fraction, and changing feature pitch and width had little impact on heat transfer. However, reducing feature height on the post surfaces facilitated water penetration within the microstructure, slightly enhancing thermal transport.

Keywords: jet impingement, superhydrophobic, quenching, heat transfer, boiling
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NOMENCLATURE

\( a \) Jet radius
\( c_p \) Specific heat of silicon as a function of temperature
\( d \) Microstructure feature width or diameter
\( E \) Thermal energy
\( \hat{E} \) Normalized thermal energy
\( g \) Acceleration due to gravity
\( h \) Microstructure feature height
\( k \) Thermal conductivity
\( Nu \) Nusselt number
\( Q \) Volumetric flow rate
\( \bar{\dot{q}} \) Average heat rate
\( q'' \) Local heat flux
\( r \) Radial distance from stagnation point
\( Re_D \) Jet Reynolds number based on nozzle radius
\( t \) Time
\( T \) Temperature
\( v \) Velocity
\( w \) Microstructure feature pitch

Greek Symbols
\( \delta \) Wafer thickness
\( \varepsilon \) Emissivity
\( \nu \) Kinematic viscosity
\( \rho \) Density of silicon
\( \sigma \) Surface tension of water at room temperature
\( \theta \) Contact angle

Subscripts
\( 0 \) Initial
\( H \) Heater
\( j \) Jet
\( s \) Surface
\( si \) Silicon
\( unc \) Uncertainty
\( w \) Water
CHAPTER 1. INTRODUCTION

Jet impingement is an important part of everyday life, from the textile drying process involved in manufacturing the shirt you’re wearing, to the steel quenching used to form the washing and drying machines that will clean that shirt. From a young age, we recognize almost intrinsically the heat transfer capability of impinging fluid jets when we blow on hot food to cool it off. The amount of heat that can be transferred depends on both the properties of the jet as well as those of the impinged surface. Since water has higher thermal conductivity than air and is nearly as common, it is often used in cooling applications. Water interacts with surfaces differently depending on surface shape, chemistry and temperature. This thesis examines the specific interaction, including heat transfer and hydrodynamic, of a quenching liquid jet impinging a superhydrophobic surface heated past the boiling temperature of water.

Superhydrophobic surfaces, which can be natural or man-made, consist of a combination of micro- or nanoscale roughness on a surface that is natively hydrophobic, or water-repelling. As will be shown later, these modifications can reduce shear in flows across such surfaces due to a slip condition, but can also potentially reduce heat transfer. Superhydrophobic surfaces have potential advantages and disadvantages, but until this research they had not yet been studied in high-temperature jet impingement applications.

The following sections provide first, insight into superhydrophobic surface characteristics, including how surfaces are made to be superhydrophobic, the underlying physics behind how they work, and some practical applications and uses. Second, jet impingement hydrodynamic properties will be presented with standard surfaces as well as showing differences when a slip condition is imposed by the use of superhydrophobic surfaces. Third, previous work regarding heat transfer on superhydrophobic surfaces will be reviewed for different scenarios. Finally, a summary of research regarding jet impingement heat transfer will be given showing how it connects with the current work, followed by an overview of the organization of this thesis.
1.1 Background

1.1.1 Superhydrophobic Surfaces

Surfaces can be characterized by their interaction with water. Surfaces made of a material with low surface energy exhibit weaker attraction forces with water than those with high surface energy [1]. The attraction force between water and a surface can thus be dominated by the cohesive forces within the water molecules themselves. The attraction between water and a hydrophobic (HPo) surface (one with low surface energy) it rests on can result in a water droplet beading up. One measure of the hydrophobicity of a surfaces is the static contact angle formed between a small droplet of water and the surface. HPo surfaces form a static contact angle, \( \theta \), greater than 90° (see Fig. 1.1). Contact angles less than 90° result in a hydrophilic (HPi) surface (left image in Fig. 1.1). In order to neglect gravitational effects and obtain accurate contact angle measurements, the diameter of water droplets must be less than the capillary length, \( L_C \). This is a defined ratio between surface tension and gravitational forces, \( L_C = \sqrt{\sigma / (\rho_w g)} \), where \( \rho_w \) is water density at a given temperature and \( g \) is acceleration due to gravity. Besides the surface energy, which describes the liquid-solid interaction, surface tension, \( \sigma \), can also play a role in the contact angle in general cases. Surface tension is the relationship between the internal cohesive forces and the adhesive forces of water (in the case of HPo or HPi surfaces) to the surrounding air, which is a function of both the liquid and gas as well as temperature.

Fig. 1.1: The wettability of a surface can be shown in part by its static contact angle with a droplet, as can be seen here for different surface types. SH surfaces in non-wetted and wetted states are also shown.

Inclusion of a micro- or nanostructure on a surface can enhance its naturally HPi state by increasing total surface area and consequently the overall attraction of water to the surface. This
is known as a superhydrophilic surface and is characterized by very low contact angles which approach 0°. If the surface is inherently HPo, however, it can actually trap regions of air between the structures where water cannot penetrate, further reducing overall solid-liquid attraction and making it superhydrophobic (SH) as seen in Fig. 1.1. This is known as the Cassie-Baxter state, and results in contact angles of 150° or greater, enhancing droplet mobility as rolling instead of sliding now dominates.

These cavities can also provide an apparent slip condition if liquid is flowing, as there is negligible shear force between the water and air, reducing friction by limiting the contact points between the liquid and the solid. Figure 1.2 is a diagram of the effects of slip in a parallel flow compared to a no-slip surface. The right image of Fig. 1.2 shows an apparent slip velocity at the surface, $u_s$, (an average velocity value, with no-slip at the tops of microstructures and free shear at the air cavity interface), a velocity profile, and what is known as the slip length, or the depth into the surface that would theoretically reach zero velocity based on a linear extrapolation of the velocity profile below the surface. A similar diagram is shown for an apparent thermal jump in the case of a heated surface, which will be described more in depth later.

![Fig. 1.2: SH surfaces cause slip, or an average surface velocity due to low shear at the liquid-gas interfaces in the microstructure. Slip length is the distance into the surface where the velocity would theoretically recover the no-slip condition through a linear extrapolation of the profile at the surface. If the surface is heated, there is a similar effect on the temperature profile due to the conduction through a limited liquid-solid contact area, with a temperature jump defined as a linear extrapolation into the surface where the no-jump boundary condition would be recovered.](image)

Under certain circumstances, water is able to penetrate between the microstructures, which is known as the Wenzel state. A droplet in this state can maintain a high contact angle but have reduced mobility. Often this condition is determined by the Laplace pressure, or the difference in
pressure at the curved interface between the water and the gas in the microstructure gaps. This pressure can be determined using the Young-Laplace equation:

\[ \Delta P = \sigma \left( \frac{1}{R_1} + \frac{1}{R_2} \right) \]  

(1.1)

where \( \Delta P \) is the Laplace pressure, \( \sigma \) is the surface tension, and \( R_1 \) and \( R_2 \) are the principle radii of curvature. If the Laplace pressure is exceeded by the water pressure, water will penetrate the microstructure and enter the Wenzel state.

Surfaces can be made to be SH using a variety of techniques. Commercial spray-coatings such as Rustoleum® NeverWet™ can be applied, which deposits hydrophobic nanoparticles in a random pattern on surfaces that provide both the roughness and chemistry needed. Specific micro- or nanostructure designs may be made in the surfaces using photolithography to specify structure patterning. Another method to make a surface SH is to grow inherently-hydrophobic carbon nanotubes either in random structures or in pre-patterned designs on surfaces. The method employed throughout this work, which will be described in depth in Chapter 2, involves etching a surface to create the desired microstructure, and then coating with Teflon, a well-known non-stick (low surface energy) material. Despite varying differences in the specifics of fabrication processes, the underlying principles are the same for fabricating all man-made SH surfaces: a nano- or microstructure is required along with hydrophobic material properties.

There are several ways to characterize SH surfaces. For patterned surfaces, characterization is often based on cavity fraction, which is the ratio of the actual water-surface contact area (i.e. the tops of the microstructures) to the total projected surface area. Patterned SH surfaces can also be described by their shape. Typical microstructure shapes include posts (also called pillars or columns) and ribs (also known as fins), which are “open” structures that allow for potential gas flow beneath the liquid layer. Other geometries are “closed”, which are called holes here, and prevent vapor flow between microstructure features. Other types, especially nanostructured surfaces, are randomly-ordered and can thus cannot be classified by cavity fraction or an open or closed form, but instead a surface roughness estimate is used.

Examples of SH surfaces are abundant in nature, as shown in Fig. 1.3. One instance is the leaf of the lotus plant, which allows water to easily slide off. Another example is found in penguin
feathers, which provide both drag-reduction and anti-icing characteristics for the birds swimming in frigid water [2]. A third case is insect eyes, which demonstrate anti-fogging properties due to their SH nature [3].

Fig. 1.3: SH surfaces abound in nature, such as on the leaves of the lotus plant, on the feathers of penguins, or on the eyes of some insects. Images are all free stock photos taken from pxhere.com, except the lotus structure, which was taken from Wikimedia Commons with permission from William Thielicke.

1.1.2 Water Jet Impingement Hydrodynamics

Several characterizations of jet impingement exist. Jet shape is one way to distinguish cases, with the most common being planar jets using thin, rectangular nozzles, or axisymmetric jets formed by circular nozzles. Other general categories are air and water jets as they are common cooling fluids. Jet impingement can also be defined by the jet configuration, whether it is submerged into a fluid of the same medium (typical for air jets impinging into air), confined (constrained to a certain geometry for flow across a surface), or a free jet (typically a liquid jet through a gas environment onto a surface with no confinement). The hydrodynamic and heat transfer information presented here will deal almost exclusively with axisymmetric, free water jets, and in general will be designated as merely a “jet” or an “impinging jet” throughout this thesis.
When an axisymmetric water jet impacts a surface, it is forced to expand outward radially. Figure 1.4 shows a cross-section of an impinging jet at steady-state conditions with the following regions of impingement:

I. **Stagnation region**: Region near the impact of the jet, characterized by high pressure, but low velocity.

II. **Thin film region**: Water spreads into a thin film and flows outward parallel to the surface (also known as the parallel-flow region for planar jets or the radial-flow region for axisymmetric jets). To satisfy continuity, this region is very thin while maintaining the initial jet velocity until the boundary layer reaches the upper thin film surface.

III. **Downstream depth region with hydraulic jump**: At a certain critical distance from the stagnation region, the hydrostatic force, momentum, and surface tension balance and form a hydraulic jump to a downstream depth region where velocity slows and the liquid height increases considerably.

![Fig. 1.4: A radial cross-sectional view of an impinging jet, which is divided into the different regions described above: I - Stagnation region, II - Thin film region, III - Downstream depth region with hydraulic jump.](image-url)
A common instance of an impinging jet is turning on the kitchen faucet, where the water spreads out into a thin layer, as shown in Fig. 1.4, and eventually reaches a significantly deeper downstream depth or hydraulic jump. While this is a simple example, the underlying physics are actually quite complex. The first in-depth study performed on the hydrodynamics of impinging liquid jets was by Watson in 1964 [4]. Watson characterized the flow of an impinging jet and produced fairly accurate predictive equations for the radial location of the hydraulic jump. A subsequent investigation was performed by Liu and Lienhard, who researched the flows and eddies produced by impinging jets [5]. They observed smooth transitions at the hydraulic jump for low Weber numbers, but noted several instabilities when the Weber number was increased. Other research includes Lienhard’s contribution on the effect of nozzle type and jet velocity which discussed several nozzle shapes and their impact on developing boundary layers [6]. He also noted the effects of turbulence in the nozzle on inducing jet instabilities, which could lead to splattering and thus poor contact on the surface. Bush et al. added an in-depth analysis on the effects of surface tension to Watson’s model [7, 8], showing that an increase in surface tension leads to a significant reduction in the hydraulic jump radius, mainly for hydraulic jumps of small radius and height.

Investigation into jet impingement on SH surfaces has also been done. Prince et al. performed a combination of theoretical and experimental work, showing that the apparent slip caused by SH surfaces (see Fig. 1.2) decreases the growth rate of the boundary layer, the minimum thin film thickness, and the coefficient of friction [9, 10]. Knowledge of these effects can assist in heat transfer predictions, as the Nusselt number can often be directly correlated to coefficient of friction.

1.1.3 Heat Transfer on SH surfaces

While SH surfaces can provide the advantages of hydrodynamic slip, there is also an apparent thermal jump, which impedes heat transfer as is shown in Fig. 1.2. As water can only contact a fraction of the projected surface area, heat transfer is limited to conduction through those segments and is insulated by the gas-filled gaps elsewhere. This leads to a lower aggregate surface temperature and thus a temperature jump in which the temperature difference between the fluid and the surface is decreased compared to the case of a smooth surface. A temperature jump length is measured as the depth into the surface where a linear extrapolation of the temperature gradient at the liquid-solid interface would reach the temperature of a smooth surface (no temperature jump).
Since heat transfer is dependent on temperature differences, similar to electrical current being dependent on voltage potential, introducing a thermal jump also limits heat transfer. This has been demonstrated for different scenarios related to jet impingement, including convective heat transfer in pool boiling, in microchannel flows, and in droplet impingement. While all contain different physics compared to jet impingement, many of the same underlying heat transfer principles apply and are therefore included here.

**Pool Boiling**

Classical pool boiling knowledge shows that varying regions or conditions of boiling behavior can occur on a surface heated above the liquid saturation point, depending both on the surface temperature as well as the surface heat flux. Figure 1.5 indicates how this behavior changes by altering these parameters. Nucleate boiling occurs initially, which is characterized by vapor bubbles forming on a surface and rising due to buoyancy. Increasing the heated solid surface temperature results in higher heat transfer from the surface to the liquid. At a certain surface temperature, a critical heat flux is reached and vapor is generated not in columns rather than discrete bubbles, with vapor covering a majority of the surface. Typically, the vapor state of a fluid has lower thermal conductivity than the corresponding liquid state, resulting in decreasing heat transfer in the boiling regime known as transition boiling. As surface temperature increases, another critical point is reached in which liquid cannot contact a surface and instead rests on a layer of its own vapor in what is known as the film boiling regime. This critical temperature is called the Leidenfrost point. Typically radiative heat transfer dominates any conduction or convection due to low vapor conductivity when a fluid such as water is boiled. As surface temperature increases in the film boiling regime, so too does the heat transfer, but typically at a slower rate than for nucleate boiling.

Pool boiling heat transfer has been studied previously to determine the effects that micro- and nanostructured SH surfaces can have on boiling condition. Notable work was performed by Searle et al. who showed that transition from nucleate to film boiling can occur at very low superheats when a superhydrophobic surface is used [11]. This Leidenfrost effect was shown to be enhanced with increasing cavity fraction for rib structures (decreasing the Leidenfrost temperature), but had the opposite effect for post structures, which tended to wet between the microstructures for
larger cavity fractions. For some extreme conditions explored (high cavity fraction or large pitch), nucleate boiling was almost completely suppressed. Structure height also influenced heat transfer and boiling condition, as deeper features further delayed film boiling.

Vakarelski et al. also showed nucleate boiling suppression for randomly-structured SH surfaces [12]. Experiments were performed both for quenching superheated spheres, showing temperature progression over time. For the HPo and HPi spheres, there was a clear transition from Leidenfrost state to nucleate boiling, but the SH sphere merely transitioned from film boiling to an apparent sensible heat transfer when liquid eventually contacted the surface. Immersed heaters were also tested, varying the heater input and measuring the corresponding temperature change, with similar results showing complete nucleate boiling suppression.

The implications of this research do not directly correlate to identical effects for jet impingement heat transfer, especially since there is induced fluid motion in jet impingement that can lead to instabilities in the vapor layer during film boiling. However, the principle of a vapor (air) initially filling in the microstructure gaps in SH surfaces is applicable to other high-temperature
heat transfer scenarios. Enhanced Leidenfrost behavior could potentially be seen for SH surfaces as water vapor would need to form over a smaller region in order to suppress nucleate boiling.

**Microchannels**

Due to the small size of microchannels, high heat transfer coefficients can be achieved in forced conduction scenarios. However, there is also a cost in the form of the high pressure required to drive the flow to overcome viscous effects, which dominate momentum at low Reynolds number. Using SH surfaces is an attractive option for reducing the drag force due to the creation of an apparent slip condition. Due to limited liquid-surface contact area in convective microchannel flows over SH surfaces, however, the apparent thermal jump has been shown numerically to reduce heat transfer in microchannels when compared to smooth surfaces [13]. Theoretically, drag reduction due to hydrodynamic slip can be sufficient that the subsequent increase in mass flow rate due to reduced shear can overcome the effects of apparent temperature jump [14]. However, this effect was analyzed using a different working fluid, and the principles might be different for SH surfaces.

Cowley *et al.* studied heated, micro-rib SH microchannels at subcritical temperatures using air-saturated water as the working fluid [15]. As the water heated it became oversaturated with air. As air was already present in the SH microstructures, the air effervesced out of the water onto these preferential nucleation sites. For ribs perpendicular to the flow, this produced even larger air bubbles that negatively impacted both hydrodynamic as well as thermal performance, although there was a negligible effect for surfaces with ribs parallel to the flow. This shows that SH surfaces can have an effect on heat transfer.

To date, practical enhancement of heat transfer in microchannel scenarios due to drag-reducing SH surfaces has been unproven. The research described here also has only explored sensible heat transfer, with no phase change, which could drastically alter the heat transfer characteristics. Similar conditions could apply to confined jet impingement, which is different from the free-surface jets explored in this work.
Droplet Impingement

Droplet impingement on SH surfaces has been investigated in depth in regards to both hydrodynamics as well as heat transfer. Research in this area is related to jet impingement, but has different applications as well as distinct thermal and fluid dynamic effects. Figure 1.6 assists in the understanding of droplet impingement physics, particularly on a superheated SH surface. As a droplet impinges on a surface, it forms a lamella, or thin film, as it expands radially outward (see Fig. 1.6b - c). While droplets that impinge on HPi surfaces typically stick and either do not rebound or splatter if their initial velocity is high enough, Clavijo et al. showed that an impinging droplet’s ability to rebound off of a surface (see Fig. 1.6d - f) is increased with taller microstructures and higher surface temperature [16]. This is due to the combination of low surface wettability and high temperature allowing a droplet to contact and dewet the surface. As temperature increases, the impact of a droplet on a hot surface can cause instantaneous boiling, with vapor breaking through the thin liquid film resulting in a spray of smaller droplets in an effect known as thermal atomization (see Fig. 1.6c). Clavijo et al. also investigated this phenomenon of thermal atomization on surfaces of varying wettability, microstructure, and temperature [17]. The results showed similar behavior to pool boiling, with atomization correlating to nucleate boiling. After a temperature region where atomization is at a maximum, this break-up effect is suppressed as the Leidenfrost point is reached and stable film boiling occurs.

Emerson et al. continued this line of research, modifying Weber number to alter how the droplet impacts the surface [18]. For increasing microstructure pitch and decreasing Weber number, atomization was found to decrease while the Leidenfrost point increased. Emerson et al. also investigated the differences between post and hole microstructures of varying height, showing that the post surfaces tended to decrease atomization, and Leidenfrost point was increased with decreasing structure height. Similar trends may be apparent in jet impingement on SH surfaces, but the dynamics differ sufficiently that the trends are not identical.

1.1.4 Jet Impingement Heat Transfer

Scores of studies have been conducted on jet impingement heat transfer because of its impressive ability to cool surfaces rapidly. A brief summary of some of the parameters that have been
explored include the effects of jet velocity, subcooling, size, and distance from the surface as well as surface characteristics. One important parameter is wall temperature or wall heat flux, which primarily governs the method of heat transfer. As explained in Section 1.1.3 on Pool Boiling, with increasing wall temperature a surface can transition from forced convection to nucleate boiling, to transitional and finally film boiling. Each of these has been studied and is included in the following sections for steady-state cases. Transient studies have also been performed, mainly for quenching scenarios, and information regarding that research is also included here.
Convection

Single-phase forced convection heat transfer in jet impingement scenarios has been extensively investigated, often using a metal heating block or thin metal heating film. Liu and Lienhard examined the developing thermal and hydrodynamic boundary layer regions, demonstrating with a combination of theoretical and experimental methods how Nusselt number, \( \text{Nu}_D \), decreases with increasing distance from the stagnation point [19], with a potential increase if there is a transition to turbulence [20]. Evaporative losses were also considered, showing decreasing effect of evaporation on \( \text{Nu}_D \) with increased distance from the stagnation point. Studies all agree that increasing Reynolds number (\( \text{Re}_D \)) increases heat transfer due to both a higher stagnation pressure as well as faster flow rate that augments the effect of advective thermal transport [21, 22]. Nozzle inclination has been reported to have varying effects on the heat transfer coefficient. Decreasing the jet angle from 90° (normal to the surface) has experimentally shown an increase in maximum \( \text{Nu}_D \) for a water jet [23], a decrease in maximum \( \text{Nu}_D \) for an oil jet [24], and velocity-profile dependency on planar jets in a numerical study [25]. Jet length, or the distance from the nozzle to the surface, has been shown to have negligible impact on heat transfer, although it can affect the jet itself by introducing instabilities or modifying the velocity [26].

Convective heat transfer by jet impingement on SH surfaces has also been recently studied by Searle et al. both analytically as well as experimentally [27–29]. Findings indicate that similar behavior applies to SH surfaces in regards to the change in heat transfer coefficient decreasing radially and increasing with \( \text{Re}_D \). Increasing cavity fraction was found to also reduce heat transfer from the surface to the liquid due to an increased apparent temperature jump (see Fig. 1.2).

Nucleate Boiling

When local surface temperatures greatly exceed the saturation temperature of the fluid, nucleate boiling can occur in jet impingement heat transfer. This boiling regime has been heavily investigated due to its potential to transfer thermal energy at high heat fluxes from surfaces. Experimentally, heat flux has been shown to increase in the nucleate boiling regime with increased jet subcooling [30, 31], increased jet velocity [32], increased jet diameter [33] and decreased distance from the stagnation region [34]. Jet length (nozzle-to-surface spacing) seems to have little effect
on heat transfer in this boiling regime [35]. Other studies have shown analytically how the nucleate boiling regime can prevail at surface temperatures much higher than would be expected, delaying film boiling [36].

Modifying surface roughness also impacts nucleate boiling jet impingement heat transfer. Superhydrophilic surfaces can have a significant impact on heat transfer, increasing heat flux by up to 30% compared to a smooth copper surface due to the increased contact area [37]. The nucleate boiling regime is a highly-researched area, and the articles that have been cited here demonstrate how the parameters considered relate to various real-world applications.

**Transition and Film Boiling**

Dramatic increases surface temperature can cause the surface behavior to exceed nucleate boiling conditions and experience transition or film boiling. In film boiling, vapor is generated at a high rate such that the liquid is separated from the surface by a layer of its own vapor, impeding heat transfer due to the lower thermal conductivity of vapor as compared to the liquid phase. While not as common due to the high surface temperature requirement, this condition has still been investigated in steady-state experiments. Similar trends with regard to jet subcooling and velocity as for nucleate boiling (increasing each also increases heat flux) have been shown using combined empirical and numerical approaches [38, 39]. For high subcooling (greater than 55 °C), film boiling can be suppressed beyond wall superheats of 1000 °C, and often the standard transition boiling regime is referred to as a “shoulder” due to its duration over a large range of wall superheats [35]. Bogdanic et al. used a micro-scale optical probe to measure the thickness of a vapor film (nominally 8 µm) and the frequency of film disturbance by liquid contact throughout different boiling regimes [40].

**Transient Studies**

The studies cited thus far have dealt primarily with steady-state conditions. Depending on surface heating conditions, often a variety of heat transfer modes can be observed on a surface being cooled by jet impingement. There is also extensive research that has been performed with transient, quenching situations that are common in different applications. The dynamics of the
jet greatly change, as is shown in Fig. 1.7. This diagram shows that, in a quenching situation, thermal breakup of the thin film into droplets occurs at the thin film front a certain distance from the stagnation point due to the Leidenfrost effect. As the surface cools sufficiently at this location due to latent heat transfer, liquid can contact then contact the surface and the front moves further downstream.

Investigations involving quenching a heated surface often show similar results regarding heat flux due to the effects of jet subcooling and velocity as well as radial position along the surface [41, 42]. Surface material properties play a significant role in heat transfer characteristics. Generally, higher thermal conductivity increases the maximum heat flux for all radial positions, and at any given time the local maximum heat flux across the surface is located where nucleate boiling occurs [43].

Another common measurement in transient experiments is the location where the thin liquid region breaks up throughout time, also known as the thin film front or rewetting front, (see Fig. 1.7) and its corresponding temperature. This thin film spreading has been demonstrated to be influenced by initial surface temperature [44] and surface material properties[45], where spreading

![Diagram of jet impingement quenching with regions of heat transfer: Stagnation, Convective Thin Film, Boiling, Droplet Ejection, and Heat input.](image)
occurs more quickly for lower initial temperatures and lower thermal conductivity (although less heat overall was transferred for materials with lower conductivity). Jet length has a varying effect on thin film spreading, and increased velocity increases the spreading rate [46].

The ability to acquire heat flux data has improved over time. For some early studies, overall heat transfer was determined only by balancing the heat input with the heat removed by an impinging jet [38, 47]. This technique is limited spatially, typically with observations of thermal transport only at the stagnation point, as surface heat flux can vary with distance from impingement. Most of the studies cited here used an array of thermocouples and an inverse heat conduction method to determine local heat flux or Nusselt number. This method provided higher spatial resolution, as additional local surface temperatures could be observed. This technique is still limited both by the contact of the thermocouples to the heater block or heating foil as well as the time constant of the data acquisition device that recorded the temperature data. Higher spatial and temporal resolution can be achieved by employing a thermal camera to measure local temperature [29, 46, 48], which allows for accurate temperature measurements over the entire range of impingement with high resolution and rapid acquisition by capturing infrared emissions. A thermal camera was used in this work, which will be described more in-depth in Chapter 2.

Summary

As shown in prior work, jet impingement heat transfer is influenced by a variety of surface factors, including material properties, heat input, temperature, and wettability. It is also governed by jet properties such as diameter, length, velocity, and temperature. This previous research provides a concrete foundation on the effects of these parameters, and has contributed to the development for the work that will be presented in this thesis. Thus far, no study has been performed for jet impingement on SH surfaces exceeding water saturation temperature. Similar to previous studies involving heat transfer behavior on SH surfaces in microchannels, pool boiling, droplet impingement, and forced convection jet impingement, it was anticipated that heat transfer would be impeded by the air-filled gaps in the microstructure. Since vapor can already flow between microfeatures beneath the liquid, behavior similar to transition or film boiling was expected at high temperatures.
This thesis presents results from work studying the behavior of transient jet impingement heat transfer on SH surfaces initially heated well above the saturation temperature of water. The hydrodynamics of jet spreading and boiling were recorded and quantitative heat transfer values were calculated for scenarios involving varying initial surface temperature and jet Reynolds numbers. Ultimately, the purpose of this study was to compare jet quenching behavior on SH surfaces of varying SH microstructure with that of identical conditions on smooth HPo and HPi surfaces. Agreement is shown with previous studies of SH surface hydrodynamic behavior at high temperatures [12]. The Leidenfrost condition was initiated at lower temperatures and was maintained for longer time periods on SH surfaces than for typical smooth HPi surfaces, resulting in lower heat transfer regardless of SH surface microstructure.

1.2 Thesis Organization

This thesis is divided into several chapters to best convey the contributions of the current work. In Chapter 2, the methodology used to obtain all the results in subsequent chapters will be detailed. The next two chapters include drafts of papers to be submitted for review in peer-reviewed journals. These describe work done comparing SH, HPo, and HPi surfaces in regards to high-temperature jet impingement heat transfer followed by a comparison of SH surfaces of varying geometry. The final section is a discussion of overall findings from this research as well as suggestions for continuing work.
CHAPTER 2. METHODOLOGY

This chapter is organized into four main sections. First, surface production is described for all surface types. Next, the experimental procedure is discussed, including the physical facilities used in experimentation as well as how data was obtained. The third section describes how the results were generated from the acquired data for use in comparing different surfaces and parameters. Finally, there is a section included on uncertainty within this work.

2.1 Test Sample Production

The following section describes how the surfaces used in this research were created. The general process for making a SH surface involves etching a microstructure onto a silicon wafer which then has a HPo coating applied, which in this case was Teflon™, a commercial form of polytetrafluoroethylene (PTFE). HPo surfaces were not etched, but did receive the hydrophobic coating. HPi surfaces were neither etched nor coated. After being processed all surfaces had a heater integrated onto the side opposite of the impingement surface, as will be detailed.

2.1.1 SH Surface Fabrication

Photolithography

The first step in fabricating the SH surfaces used in this work was to pattern the surfaces to create the correct microstructure. The different patterns used in this work are given in Table 2.1, and renderings of each surface are shown in Section 4.4.1. Two general geometries, posts and holes, were investigated by varying height, pitch, shape, and diameter (or width) to explore how these parameters impacted jet impingement heat transfer. Sample SEM images in Fig. 2.1 show representative surface geometries.
Patterning the surfaces was done in a class 10 cleanroom at Brigham Young University. First, oxidized silicon wafers purchased from University Wafer, measuring 100 mm in diameter and 525 \( \mu \)m thick with one side polished to a mirror finish, were spin-cleaned with acetone and isopropyl alcohol (IPA). The wafers were baked in a dehydration oven at 150 °C for 10 min, followed by a spin-coat application of either positive (AZ 3330) or negative (nLOF 2020) photoresist (depending on the pattern) nominally 2 \( \mu \)m thick. Following a 60 s “soft” bake on a hot plate at 90 or 110 °C (depending on the photoresist), wafers were aligned with a patterning mask and exposed to UV light for 10 to 15 s using a Karl Suss Mask Aligner in order to harden (or weaken for positive photoresist) wafer regions that were not covered by the mask. Another mask was also used to create a 6 mm diameter “target” region at the center of each SH surface. This target prevents hydrodynamic wetting due to the high stagnation pressure of the jet overcoming the Laplace pressure that enabled the Cassie-Baxter state on the surface (see section 1.1.1). A “hard” bake with the same parameters as for the soft bake followed the exposures, and then the wafers were developed using AZ300MIF to expose the portions of the wafer that were to be etched.

**Etching and Cleaning**

Following the photolithography process, wafers were anisotropically etched using a Surface Technology Systems (STS) Multiplex Inductively Coupled Plasma (ICP) Reactive Ion Etcher
(RIE). This machine uses fluorine-based gases to form plasma that allows for high etch rates capable of creating deep microfeatures. Features were inspected to ensure proper height, \( h \), as will be described later.

After etching, the wafers were cleaned of the remaining photoresist. This was done by submerging the wafers in a covered dish filled with Nano-strip® at 90 °C for at least 2 hrs in a well-ventilated acid bench area in the cleanroom. In order to ensure all the photoresist was removed, the wafers were then placed in a Planar Etch II machine, which uses an oxygen-based plasma to remove organic contaminants. Finally, the wafers were again dehydration baked at 150 °C for 10 minutes to burn off any undesired residuals.

**Heater Integration**

After etching the SH surfaces, all surfaces (including HPi and HPo wafers) had a silver heater screen-printed on the reverse (non-testing) side using ElectroScience Laboratories 599-E silver-based paste. The screen-print pattern was designed to cover a region with nominal diameter of 50 mm around the center of the wafer using a series of thin traces, providing a total of 0.25 \( \Omega \) of electrical resistance. The width of the traces was intentionally varied to provide a nominally uniform heat flux when power was supplied (see Appendix B.10 of Matthew Searle’s dissertation [29] for details on heater design and application. Further insights are also included in Appendix B.1 here). The wafers were cleaned with compressed air and fired incrementally from 125 to 450 °C for a total of 30 min in an electric furnace in order to solidify the heaters. Figure 2.2 shows the heater pattern on the back side of a wafer after setting.

**Material Deposition and Coating**

After wafer etching (for SH surfaces) and heater integration (SH and HPo surfaces), a chromium layer nominally 100 nm thick was deposited on the surface using a Denton Vacuum Electron-Beam Evaporator to assist in Teflon adhesion. Following this process, the wafers where then coated with a Teflon solution (0.2% Teflon™ Amorphous Fluoropolymer 1601, 99.8% ACROS Organics™ Perfluoro-compound FC-40™), which was spin-coated on at 1,000 rpm for 20 s to form a thin, uniform layer nominally 200 nm thick. The surfaces were placed on hot plates and
heated incrementally between 90 to 330 °C for 40 min, leaving only the Teflon on the surface. This coating rendered the surfaces either HPo or SH.

**Wire Attachment and Surface Painting**

After all surface modifications and heater additions, all wafers were carefully placed face-down on a hot plate. Leads (16-gauge insulated electrical wire) were attached to the bus bars connecting the heater traces using Atom Adhesives AA-DUCT 2979 silver epoxy. The epoxy was cured at 125 °C for one hour on the hot plate and then removed and cooled for the final step. Tips on attaching wires and other fabrication processes are included in Appendix B.1.

In order to accurately measure temperature on the back side of the wafer, each surface was spray-painted with 3 thin coats each of Rust-Oleum® high heat primer (249340) and flat black paint (248903). This provided surfaces with high, known emissivity of 0.97, which improved
reliability in temperature measurements. Variations in the thermal readings from the potential insulating effects of the paint were negligible due to the thinness of the paint coats.

2.1.2 Surface Characterization

SH surfaces were measured using a Zeta-20 3D optical profilometer after etching in order to ensure correct microfeature dimensions. Surfaces with dimensions ±10% out of tolerance were disposed of and replacements were made. The delicate fabrication process resulted in some inherent variability across surfaces, so duplicates were made for each of the parameter sets given in Table 2.1.

Surfaces were also characterized by the static contact angle formed by a small droplet of water on the surface. These measurements were taken by capturing images of droplets on surfaces using a DSLR camera and then using a digital goniometer code implemented in MATLAB to determine the average angle made by a droplet and the surface to within ± 3° accuracy. Contact angles for SH surfaces made for this work were nominally 150°, while HPo and HPi surfaces had contact angles of nominally 125° and 55°, respectively.

Table 2.1: Fabrication surface dimensions, as designated by Fig. 2.1.

<table>
<thead>
<tr>
<th>Holes/Posts (H/P)</th>
<th>H</th>
<th>H</th>
<th>P</th>
<th>P</th>
<th>P</th>
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<tr>
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<td>7.0</td>
<td>7.0</td>
<td>7.0</td>
<td>3.5</td>
</tr>
<tr>
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<td>15</td>
<td>25</td>
<td>15</td>
<td>5</td>
<td>15</td>
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</table>

2.2 Experimental Setup

This section details first the physical equipment used in experiments, including how the surface was positioned as well as how the jet functioned. The method employed to acquire each
data set is then explained. The purpose of these experiments is to compare quenching heat transfer characteristics in jet impingement scenarios at various jet flow rates, given as $Re_D = Q/(\nu a)$ where $Q$ is the jetflow rate, $\nu$ is the kinematic viscosity of water at room temperature, and $a$ is the jet radius. Jet $Re_D$ is set at 6,000, 12,000, or 18,000, and various initial surface temperatures, $T_0 = 200, 280, \text{ or } 320 \degree C$ as well as various surface conditions (micro-geometry and wettability) are explored.

2.2.1 Experimental Apparatus

Completed surfaces were tested by setting them on a thin stainless cylinder that minimized thermal losses because of the small contact area and relatively low material thermal conductivity. The pipe also allowed line-of-sight access to the bottom of the wafer for the FLIR® SC6100 thermal camera, which recorded impingement progression from below with a resolution of 320 x 256 pixels over a viewing window of approximately 55 x 45 mm. The thermal camera was positioned 0.3 m below the wafer, with an extension tube used with the lens to keep the heater traces in focus and provide a wide viewing angle. The thermal camera was factory-calibrated to function in different temperature ranges. For the experimental conditions used here, 2 calibration regions were required: 20 to 150 °C and 150 to 350 °C. The camera recorded data alternating between these calibrations, each at a frame rate of nominally 200 fps, which was the fastest possible using 2 calibrations with this camera. The wire leads were then attached to a 20-V, 120-A HP 6011A DC power supply. A 15 mm long, stainless steel nozzle 2.55 mm in diameter formed the fully-developed, laminar water jet. The nozzle was connected to a rotameter, which measured the flow rate. The rotameter connected to a large storage tank filled with deionized water, which was pressurized to provide the correct jet $Re_D$. A Photron Fastcam APX RS high-speed camera recorded spreading behavior from above at nominally 45° from the horizontal at a rate of 500 fps over a nominally 70 x 70 mm viewing window at a spatial resolution of 1024 x 1024 pixels. A halogen lamp illuminated the surface from above to provide adequate lighting for the high-speed images. A schematic of the apparatus is shown in Fig. 2.3.
2.2.2 Data Acquisition

To conduct an experiment, a wafer was placed concentrically on top of the pipe and beneath the nozzle. Measurements of ambient conditions (typically 22 to 24 °C temperature and 10 to 25 % relative humidity) were taken to ensure temperatures were recorded accurately. Before testing began on a surface, images were taken with both cameras for spatial calibration purposes. A thermal image with the jet flowing across the surface was taken before any heating to obtain the temperature of the water jet. After the surface was cleaned using compressed air, power was supplied to the heater until the central region of the wafer reached a steady, prescribed value of 200, 280, or 320 °C, after which the water jet was released and the cameras were simultaneously triggered electronically. The thin film typically spread over the region of interest (viewing window) and all boiling behavior ceased within 1 s, after which the jet was stopped and the surface was again cleaned with compressed air, enabling a new test to begin. Water was collected in a container beneath the wafer. Tests were repeated 4 to 5 times per surface for data averaging. Additional details involving data acquisition can be found in Appendix B.
2.3 Data Analysis

Sample images obtained with the thermal camera at two different times during impingement are shown in Fig. 2.4. The dark region near the center of the top figure ($t = 0.02$ s) shows where liquid water is in contact with the surface, as was verified by observing the corresponding, calibrated high-speed images. The second image ($t = 0.05$ s) shows how that liquid-surface contact has spread radially. Wires attached to the power supply are slightly visible on the sides and the resistance heater traces can be seen as the thin, horizontal yellow and red lines.

Fig. 2.4: Thermal (infrared) images of temperature across the back side of a wafer showing two different instants in time of thin film spreading. The alternating red and yellow lines are from the resistance heater, and the darker area shows the progression of the jet across the surface. White radial lines, used for averaging data during analysis, are also shown.
A MATLAB script was used to process the data from the thermal camera and calculate important values (see Appendix A). First, thermal calibration images were converted to usable matrices and compared to a measured region of known dimensions in order to convert pixels to physical distances. The thermal image that recorded the jet temperature on the surface before heating was also analyzed, and the temperature was averaged across the entire viewing region to determine the potential temperature difference. Following these preliminary steps, the actual test data was then read in frame by frame, with the images alternating between the two temperature ranges that were recorded simultaneously. Data was merged between two subsequent images if the surface temperature required the full temperature range of both calibrations.

All tests were aligned temporally by setting the time, \( t = 0 \) s, for the image frame when the center temperature first displayed the effects of jet impact. The center of impingement was determined by using a native MATLAB image processing function, `imfindcircles`, that would find a circle based on the sharp temperature gradient near the edge of the thin film. Averaging the centers of several such circles examined over the duration of the experiment resulted in a highly accurate determination of the stagnation point at the center of impingement. Twenty evenly-distributed, radial lines beginning at the center of impingement were generated digitally (see Fig. 2.4). The temperature values for the pixels at the same radial distance along each of the lines were averaged. The temperature values were averaged again over all tests on the same surface, and then once more averaged over the 2 surfaces used for each set of parameters (see Table 2.1) resulting in a single surface temperature array based on radial position and time, \( T(r, t) \).

To further reduce the effect of uncertainty in measured temperature values, 15 sequential temperature values along the radius, centered around the control volume of interest, were fitted with a 2nd-order polynomial using MATLAB’s native `polyfit` function. The control volume of interest was the center of the 15 values for the inner points (central scheme), while first and last points utilized forward and backward schemes, respectively. Figure 2.5 shows a sample plot of 15 radial temperature measurements and a computed parabolic fit. Appendix A.1 provides details on how this was done. Similarly, smoothing in time was also performed. Temperatures at specific radial locations over 7 sequential data points throughout time were fit with a 2nd-order curve to reduce the impact of uncertainty on temporal changes in \( T(r, t) \). Typical \( R^2 \) values were at least 0.97.
Fig. 2.5: Temperature measurements at 15 radial locations, showing how a curve fit can reduce error in derivative quantities.

An example of how $T(r,t)$ changed throughout time and across the entire surface is shown in Fig. 2.6, and is typical of all surface types and experimental conditions. Initially (at $t = 0$ s), the temperature across the wafer is heated well above the saturation temperature of water (taken to be 95.2 °C at 1411 m above sea level). As time progresses, the wafer cools, beginning in the stagnation region (near $r = 0$ mm) and gradually spreading further in radius as time progresses.

Fig. 2.6: Temperature as a function of time and radial position for a sample SH surface initially heated to 320 °C at the center and cooled by an impinging jet with $Re_D = 12,000$. 
Quantifying the heat flux, or thermal energy transferred per unit time per unit area, from the surface to the jet allows useful comparison of the heat transfer between the various experimental conditions. The averaged temperature data was used to compute local heat flux from the surface throughout time across the entire wafer by solving an energy balance over a local control volume (see Fig. 2.7):

\[ q''(r,t) = \frac{\delta}{r} \frac{\partial}{\partial r} \left( k_{si} r \frac{\partial T}{\partial r} \right) - \delta \rho c_p \frac{\partial T}{\partial t} + \frac{q_H}{\pi r^2_H} \]  

where \( q'' \) represents the local heat flux through the top of the wafer \([W/m^2]\); \( r \) and \( t \) denote radial position \([m]\) and time \([s]\), respectively; \( \delta \) and \( \rho \) are the wafer thickness \([m]\) and density \([kg/m^3]\), respectively; \( k_{si} \) and \( c_p \) are silicon thermal conductivity \([W/m-K]\) and specific heat \([J/kg-K]\) as functions of local temperature, \( T \) \([^\circ C]\); \( q_H \) and \( r_H \) are the heater input power \([W]\) and heater radius \([m]\), respectively.

The first term of Eq. (2.1) represents net conjugate heat transfer into each differential control volume, caused by the temperature gradient in the radial direction. The derivative of the calculated polynomial used to smooth the 15 radial values was taken and then used in calculating the radial temperature gradient, \( \partial T / \partial r \) (see Appendix A.1). The limit of the conjugate term approached infinity as the control volume approached the radial location \( r = 0 \) mm because that term approached infinity as the control volume approached the radial location \( r = 0 \) mm because that term
is divided by the value of the radial position. In order to avoid inappropriate extrapolation of data, the heat flux within the jet radius was set to the same value. At the other boundary, the radial extent of the viewing window (nominally 20 mm), the temperature gradient was still important over the course of testing, because it showed heat transfer to or from outside of the viewing window as the temperature changed within this region. Uncertainty in all terms will be discussed in a later section.

The second term of Eq. (2.1) is the transient term which is based on the temperature change in time. Similarly, the derivative of the fitted polynomial to smooth temperature changes in time represents the temporal temperature gradient, $\partial T / \partial t$. The transient term was dominant in overall heat transfer until boiling activity had ceased within the region of interest. The initial conditions included $T(r, t = 0)$ and $\partial T / \partial t > 0$ as the impinging jet caused a sharp change in temperature between recorded frames.

The final term of Eq. (2.1) is the heater input to the control volume. After testing, this was found to be at minimum one order of magnitude lower than the heat flux removed by the impinging jet due to the other two terms of the equation, and was thus neglected in actual calculations. Other values that were not included here are effects of natural convection and radiation, which were also found to be negligibly small compared to the transient and conjugate components, as will be shown in the uncertainty section.

A MATLAB script was used to calculate the heat flux based off the bottom surface temperature using Eq. (2.1). Fourier’s conduction law ($q'' = -k \partial T / \partial z$) was then used to calculate what the top surface temperature would be, assuming the heat flux was only in the direction normal to the surface, using the wafer thickness and the local thermal conductivity as parameters in the calculation. An average of the top and bottom surface temperatures was then used in Eq. (2.1) again to calculate values with higher accuracy for the local surface heat flux for all times and radial locations. These values were ultimately used in comparing the various surfaces and experimental parameters.

Figure 2.8 shows high-speed images of impingement progression on a SH surface at $t = 0$, 0.05, and 0.1 s after initial jet contact, with corresponding thermal data below. For this case, initial temperature was $T_0 = 320$ °C and flow rate was $Re_D = 12,000$. The patterned jet target on the surface can be seen just below the jet as the reflective circular region. Random discolorations
visible across the surface are natural effects present after Teflon drying. In the second and third images, the liquid thin film region is clearly visible as the darker circular portion surrounding the jet before breaking up into a lighter spray of droplets. The black dashed lines indicate the edge of the thin film in the high-speed images. The corresponding thermal data shows both the local surface temperature (averaged over all surfaces and tests) and the calculated surface heat flux, which has the same value across the diameter of the jet, as explained previously.

A sample plot of heat flux through time across a wafer is given in Fig. 2.9. This data was for a SH case (the same as in Fig. 2.6), and though the behavior shown was typical for all surfaces, the specific values and location of maximum heat flux at later times varied, as will be shown in succeeding chapters. The maximum heat flux peak is shown at the initial time in the stagnation region which, again, was set to be equal for the entire region covered by the jet radius. As the surface cooled locally throughout time and boiling was suppressed, the location of maximum heat flux moved out radially with a decreasing maximum value.
Fig. 2.9: Local heat flux plotted as a function of time and radial position as a sample SH surface was cooled.

With the local heat flux calculated, the approximate location of the thin film region was determined as the location where the heat flux was at a radial maximum, as seen in Fig. 2.4, which corresponds to results previously reported [43, 49]. Inside the thin film, the temperature does not change as much radially or temporally. At the edge of the thin film, however, significantly higher amounts of energy are transferred in a latent form due to the water boiling than the sensible thermal energy transferred within the thin film due to forced convection. Thus, the greatest temperature gradients and consequently highest heat flux occurs near the edge of the thin film. This was confirmed by the user-selected locations of the approximate location of the thin film front on corresponding high-speed images, indicated by the dashed black lines in Fig. 2.8. Figure 2.10 plots the thin film location (in units of jet radii) measured in the high speed images as well as the location of maximum radial heat flux throughout time for a particular set of parameters for HPi, HPo and SH surfaces ($T_0 = 320 \, ^\circ\text{C}, \, \text{Re}_D = 12,000$). This figure shows how similar the values are for both methods across all surface types. Tracking thin film spreading by using the location of maximum heat flux was both simpler and much more accurate than the user-selected locations from the high-speed images, and was therefore used to determine thin film spreading throughout the work presented here. Differences in the two measurements past the target radius for all experimental conditions was typically less than 5% for HPi surfaces, which showed the highest agreement. The
thin film front on the SH surfaces proved the most difficult to track using the high-speed images, and differences between measurements were typically less than 15%.

Fig. 2.10: Instantaneous thin film radius, $r_f$, measured with the high-speed visual camera and location of maximum local heat flux, $r_q$, both normalized by jet radius, $a$, as functions of $t$ and surface type for $T_0 = 320 \degree C$ and $Re_D = 12,000$. Similar behavior was observed for other conditions.

Other general heat transfer values were computed using the calculated heat flux. The total energy transferred up to a certain point in time is calculated based on initial temperature differences between the surface and the jet:

$$E(t) = \int_0^t \int_0^{r_{lim}} 2\pi r \delta c_p (T(r,t) - T_j) dr dt$$  \hspace{1cm} (2.2)

where the $r_{lim}$ indicates the radial limit of the region of interest. The derivative of $E(t)$ with respect to time provided another quantitative means to compare thermal transport: an average heat transfer rate over the surface, $\bar{q}$. This value, as will be shown in later chapters, was fairly constant over the first initial portion of impingement. A line was fit to the first several milliseconds of $E(t)$, and the slope of that line was taken to be the initial $\bar{q}$ (see Appendix A.2).
Due to slight variations in initial heating on surfaces, the thermal energy transfer was normalized by the initial thermal potential energy between the surface and the jet, $E_0$, given by:

$$E_0 = 2\pi \delta \rho \int_0^{r_{lim}} c_p(T_r(r,0) - T_j) r dr$$  \hspace{1cm} (2.3)

This normalization, $\hat{E}(t) = E(t)/E_0$, provided a more equitable comparison between experiments, especially between those with varying initial temperature.

2.4 Uncertainty Analysis

Uncertainty for Eq. (2.1) was found using a propagation of uncertainty method. This was done for all radial positions and values of time by taking the root of the sum of the squares of the uncertainty of each variable (subscript \textit{unc}) multiplied by the derivative of Eq. (2.1) with respect to that variable:

$$q''_{unc} = \left[ \left( \frac{\partial q''}{\partial \delta} \delta_{unc} \right)^2 + \left( \frac{\partial q''}{\partial r} r_{unc} \right)^2 + \left( \frac{\partial q''}{\partial k_{st, unc}} \right)^2 + \left( \frac{\partial q''}{\partial c_p} c_{p, unc} \right)^2 + \left( \frac{\partial q''}{\partial T} T_{unc} \right)^2 + \left( \frac{\partial q''}{\partial t} t_{unc} \right)^2 \right]^{1/2}$$  \hspace{1cm} (2.4)

Values for the uncertainty in each variable is given in Table 2.2. Most are given as a percentage or range of percentages, as will be described for each term below. Overall uncertainty in the maximum heat flux values compared in this work was calculated to be less than nominally ±5% for the surfaces with highest uncertainty (typically SH surfaces at 320 °C initially).

The first squared term of Eq. (2.4) is evaluated using Eq. (2.5) for the derivative term. This was then multiplied by the thickness of the wafer, $\delta$, and by the percent uncertainty reported in Table 2.2. This tabled value was obtained from the root mean sum of squares of a 95% confidence interval (C.I.) value of several measurements in the wafer thickness using a mechanical micrometer and the associated instrumentation error.

$$\frac{\partial q''}{\partial \delta} = \frac{1}{r} \frac{\partial}{\partial r} \left( k_{si} r \frac{\partial T}{\partial r} \right) - c_p \rho \frac{\partial T}{\partial t}$$  \hspace{1cm} (2.5)
Table 2.2: Variable uncertainty values

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Units</th>
<th>Values (±)</th>
<th>Method</th>
</tr>
</thead>
<tbody>
<tr>
<td>Wafer thickness, $\delta$</td>
<td>%</td>
<td>1.0</td>
<td>Measured</td>
</tr>
<tr>
<td>Radial measurement, $r_{\text{meas}}$</td>
<td>%</td>
<td>0.6</td>
<td>Measured</td>
</tr>
<tr>
<td>Radial calibration, $r_{\text{cal}}$</td>
<td>%</td>
<td>1.1</td>
<td>Calibration</td>
</tr>
<tr>
<td>Thermal conductivity, $k_{\text{si}}$</td>
<td>%</td>
<td>1.3 - 4.2</td>
<td>Curve fit</td>
</tr>
<tr>
<td>Specific heat, $c_p$</td>
<td>%</td>
<td>0.04 - 0.38</td>
<td>Curve fit</td>
</tr>
<tr>
<td>Temperature measurement, $T_{\text{meas}}$</td>
<td>%</td>
<td>1.2</td>
<td>95% C.I.</td>
</tr>
<tr>
<td>Temperature acquisition, $T_{\text{instr}}$</td>
<td>%</td>
<td>1</td>
<td>Manufacturer</td>
</tr>
<tr>
<td>Time, $t$</td>
<td>s</td>
<td>0.0025</td>
<td>Framerate</td>
</tr>
</tbody>
</table>

The derivative term associated with the uncertainty in radial position is given by Eq. (2.6). The variable uncertainty here is the local radial position, $r$, multiplied by a root sum of squares of the $r_{\text{meas}}$ and $r_{\text{cal}}$ percentages given in Table 2.2. The $r_{\text{meas}}$ value is itself the root sum of squares of a 95% C.I. for several measurements of the calibration standard (the inner radius of the stainless steel cylinder) using digital calipers, as well as the corresponding instrumentation uncertainty of the calipers. The $r_{\text{cal}}$ value represents the uncertainty due to calibrating the number of pixels to the physical measurement of the calibration standard.

\[
\frac{\partial q''}{\partial r} = \delta \left[ k_{\text{si}} \left( \frac{\partial^3 T}{\partial r^3} + \frac{1}{r} \frac{\partial^2 T}{\partial r^2} - \frac{1}{r^2} \frac{\partial T}{\partial r} \right) + \frac{\partial k_{\text{si}}}{\partial T} \left( 2 \frac{\partial^2 T}{\partial r^2} \frac{\partial T}{\partial r} + \frac{1}{r} \left( \frac{\partial T}{\partial r} \right)^2 \right) + \frac{\partial^2 k_{\text{si}}}{\partial T^2} \left( \frac{\partial T}{\partial r} \right)^3 - c_p \rho \frac{\partial}{\partial r} \left( \frac{\partial T}{\partial t} \right) \right] \tag{2.6}
\]

Derivative quantities for uncertainty in silicon properties are given by Eq. (2.7) for thermal conductivity, and by Eq. (2.8) for silicon specific heat. Uncertainty values are the local interpolated values based on quadratic curve fits to reported data multiplied by the respective percentage ranges in Table 2.2. The percentages are based on minimum and maximum differences between reported data and the curve fits.
The derivative of Eq. 2.4 with respect to temperature is given by Eq. (2.9). Temperature uncertainty is the local temperature value multiplied by the root sum of squared $T_{\text{meas}}$ and $T_{\text{instr}}$ values given in Table 2.2. $T_{\text{meas}}$ is the average of a 95% C.I. of the 20 radial lines averaged over each of 6 tests (see Section 2.3) divided by the average local surface temperature value. The reported value was for a SH surface initially heated to 320 °C, which was the most extreme experimental condition and thus uncertainty should be a maximum for this case. The percentage value reported for $T_{\text{instr}}$ is based on the manufacturer’s instrumentation uncertainty.

\[
\frac{\partial q''}{\partial T} = \delta \left[ \frac{\partial T}{\partial r} \right]^{\frac{2}{3}} \frac{\partial k_{si}}{\partial T} \frac{1}{r} \frac{\partial T}{\partial r} + \frac{\partial k_{si}}{\partial T} \frac{\partial T}{\partial r} + \frac{\partial k_{si}}{\partial T} \frac{\partial T}{\partial r} \frac{1}{\rho c_{p}} \frac{\partial T}{\partial t} \right] \quad (2.9)
\]

The derivative of calculated heat flux with time is given by Eq. (2.10). The uncertainty value used here was half of the difference between time steps.

\[
\frac{\partial q''}{\partial t} = \delta \left[ \frac{\partial T}{\partial r} \left( \frac{3}{r} \frac{\partial T}{\partial r} \frac{\partial T}{\partial t} + \frac{\partial k_{si}}{\partial T} \frac{\partial T}{\partial r} + \frac{1}{\rho c_{p}} \frac{\partial T}{\partial t} \right) \right] + \frac{k_{si}}{r} \frac{\partial^{2} T}{\partial r \partial t} + \frac{k_{si}}{r} \frac{\partial^{3} T}{\partial r^{3} \partial t} - \rho c_{p} \frac{\partial^{2} T}{\partial t^{2}} \quad (2.10)
\]

As mentioned previously, losses due to natural convection and radiation were neglected. Local heat flux from the surface to the jet, as will be shown later, was on the order of MW/m$^2$ near the thin film front, corresponding to similar values reported in previous quenching studies [43, 46, 49]. Initially, the driving potential for heat transfer was also highest due to the elevated temperature. Heat flux due to natural convection prior to impingement was found to be less than 1 kW/m$^2$ using an empirical correlation, assuming heat from the bottom surface is not lost due to the stainless steel cylinder providing a contained, closed cell. Using the standard expression for net radiative heat transfer, the maximum net heat flux to the surroundings would be less than 6 kW/m$^2$. 

35
Finally, the heater input was typically on the order of $50 \text{ kW/m}^2$ for the highest-temperature case, which is typically less than 5% of the calculated surface heat flux due to jet impingement.
CHAPTER 3. TRANSIENT HEAT TRANSFER OF IMPINGING JETS ON SUPER-HEATED WETTING AND NON-WETTING SURFACES

This chapter is in preparation journal peer-review. The format has been modified to meet the requirements of this thesis.

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3.2 Abstract

Superhydrophobic (SH) surfaces possess desirable anti-fouling properties due to low wettability, but have also been shown to reduce heat transfer to subcooled water in impinging jet scenarios. In this work, superheated silicon substrates with varying wettability (hydrophilic or HPi, hydrophobic or HPo, SH) are quenched by an impinging water jet, where the substrate temperature is above the saturation temperature. Silicon wafers are either oxidized to create HPi surfaces, coated with Teflon to make the surface HPo, or plasma-etched and coated to create the necessary micro-texture for SH conditions. All wafers are integrated with an electric resistance heater and then heated to temperatures of 200 - 320 °C before impingement with an axisymmetric room temperature water jet of varying specified flow rates yielding jet Reynolds numbers between 6,000 - 18,000. High speed visual data is collected, showing how the lamellar liquid contact region, limited by thermal breakup due to boiling, grows radially as the surface cools to temperatures below saturation. This data is correlated to temperature data recorded using a thermal camera from the back side of the wafer. Results of this study confirm previous conjecture that surface wettability can alter maximum heat flux for the described scenario by up to 40%, and can also affect jet thin film spreading by up to 50%. Increasing initial surface temperature decreases thin film spreading
rate on all surfaces, and increases heat transfer on all but the SH surfaces. Reynolds number has little effect on average heat flux, but affects both the spreading rate as well as the maximum radius of the thin film region.

3.3 Introduction

Liquid jet impingement is utilized in many cooling applications due to the highly convective thin film which occurs as the jet spreads on the surface, providing significant heat transfer from a surface [6]. This method of heat transfer finds application in metal processing [42], piston cooling (with oil as the working fluid) [50], gas turbine cooling [51], and emergency nuclear reactor cooling [52]. Other important applications for jet impingement include electronics cooling, rocket launchpad cooling, and rocket nozzle or jet turbine cooling. One common issue in many of these industrial applications is that the ability to transfer heat is reduced when fouling occurs due to water evaporation on the heated surface which can lead to an accumulation of trace chemical residuals [53]. Operations are interrupted in order to clean these surfaces, which further inhibits production or performance.

Fig. 3.1: (a) SEM image of post microstructures on a SH surface. (b) Radial cross-section view of a perpendicular impinging jet on a heated surface. An insert shows the differential control volume used to calculate heat flux from the surface to the water.
Superhydrophobic (SH) surfaces are a potential solution for the issue of fouling. Water-repelling SH surfaces are manufactured by altering surface geometry to provide micro- or nanoscale roughness (as shown in Fig. 3.1a), and by changing the surface chemistry to reduce surface energy, making it inherently hydrophobic. With this combination of surface modification, water only contacts the highest points of the rough surface, and air gaps are left beneath the liquid, reducing the aggregate adhesive force and causing the water to bead up (interior static contact angle greater than 150°). Such low adhesion between the water and surface enables high droplet mobility. Water droplets will readily roll from surfaces, improving anti-fouling properties as water will not remain to evaporate on the surface after impingement.

Prior works addressing the use of SH surfaces in jet impingement cooling have emphasized steady-state hydrodynamics and low temperature heat transfer, showing that increasing hydrophobicity leads to smaller hydraulic breakup of the thin film region [10] and reduced convective heat transfer [27]. Here, the usefulness of micropost-patterned (see Fig. 3.1a) SH surface implementation in transient jet impingement heat transfer situations at superheated wall temperatures is explored.

A schematic illustration of the radial cross-section of an impinging jet on a superheated surface is shown in Fig. 3.1b. After initial impingement, the following dynamics occur (regions shown in Fig. 3.1b): a center region of high stagnation pressure; a liquid lamellar region or thin film where convective heat transfer is dominant; a boiling front where phase-change heat transfer is dominant and thin film spreading is restricted by high temperature differences; and a droplet ejection region where droplets move outward, levitating on a self-generated vapor layer.

Heat transfer properties of impinging jets in single-phase convection [19, 20, 26, 54], nucleate boiling (vapor bubbles generated at the heated surface), and transition and film boiling in steady-state conditions have been investigated. These prior studies show that heat transfer improves drastically as the nucleate boiling condition is reached and then diminishes when transition boiling conditions prevail [35]. Previous authors have experimentally explored several parameters with relation to boiling jet impingement heat transfer, such as increased heat transfer with increased jet subcooling [30, 31, 42] and jet velocity [32, 39]. There is a negligible impact of jet length (nozzle-to-surface spacing) for boiling scenarios [35], and spatial variation in the radial
direction shows decreasing heat flux further from the stagnation point [34]. Others have explored theoretical models based on these parameters, with similar general results [36, 55, 56].

Several studies have also investigated the transient effects of quenching superheated metal surfaces. This has shown a dependence on surface material properties [57], and introduces a scenario where convective, nucleate, transient, and film boiling heat transfer can all occur at various points and times across a surface. Similar results to steady-state cases, in regards to increased subcooling and higher flow rates leading to higher heat transfer, were also observed for transient cases [44–46]. In these transient studies, several experimental techniques have been utilized in order to acquire the time-dependent data necessary for heat transfer calculations, including the use of thermocouples [41, 58, 59] or by balancing the average heat removed with the supplied heat [38]. Both of these techniques have limited spatial resolution of data acquisition, and newer methods such as using high-speed thermal cameras have been shown to improve accuracy [60].

A single paper has been published regarding the impact SH properties exert on the effectiveness of jet impingement heat transfer [27]. In this work, Searle et al. created a model to predict the local Nusselt number for various spatial regions along the surface for sub-critically heated surfaces and found that with increasing surface cavity fraction (ratio of etched microfeature surface area to total surface area), a lower heat transfer rate was observed. Reduced heat transfer occurs because the air gaps formed beneath the impinging liquid and between microstructures act as an insulating layer to heat transfer. However, the work by Searle et al. only dealt with single-phase forced convection, which is not the primary mode of heat transfer at temperatures above saturation.

Research focused on SH surfaces at temperatures above saturation has been explored for droplet impingement scenarios. Clavijo et al. found that the water vapor generated beneath the droplet during impact could escape through the SH microstructures such that there was little to no nucleate boiling on the SH surfaces [61]. Instead, film boiling and Leidenfrost behavior was observed over a broad range of impact conditions [12]. Similar effects were also observed in pool boiling experiments [11]. As surface temperature increases above the fluid saturation temperature in jet impingement quenching scenarios, a boiling region occurs that restricts spreading of the liquid thin film. Due to local rapid phase change, water droplets in this region begins to elevate above the surface due to the local generation of vapor. As there is very little attraction between water and SH surfaces initially, the Leidenfrost effect occurs more readily [15]. However, for
microstructured surfaces that are natively hydrophilic (*superhydrophilic*), Qiu and Liu reported heat fluxes 30% higher due to the lower contact angle and increased contact area [37]. These data confirm wettability and microstructure have a significant effect on jet impingement heat transfer in the superheated regime.

Potential trade-offs remain to be clarified between the desirable anti-fouling properties of SH surfaces and reduced heat transfer effectiveness of film boiling. Here, jet impingement experiments are performed to compare boiling heat transfer during quenching on superheated SH surfaces to hydrophilic (HPi) and smooth hydrophobic (HPo) surfaces (no surface structuring). The jet Reynolds number is defined as $Re_D = Q/(\nu a)$, where $Q$ is the jet volumetric flow rate, $\nu$ is the kinematic viscosity of water and $a$ is the jet radius. Influence of $Re_D$ ($6,000 < Re_D < 18,000$) and initial surface temperature ($200 ^\circ C < T_0 < 320 ^\circ C$) on the thermal transport physics are explored here. Results are presented showing that SH conditions have a significant impact on cooling dynamics during jet impingement if the surface is superheated, which was previously undiscovered. Further, SH surfaces can yield increases in thin film spreading time by up to 170% and decreases in the maximum heat flux heat rate by up to 70%, compared to behavior for a HPi surface.

### 3.4 Methodology

This section describes the experimental apparatus, data acquisition, and data analysis techniques used in this study. The experimental apparatus and methodology used to collect data was similar to that of Searle *et al.* [29] and will be briefly summarized. Important differences of operation due to transient behavior and much higher surface temperatures will be noted.

### 3.4.1 Experimental Apparatus

There are three main components that comprise the experimental apparatus: the surfaces and heating elements, the jet and associated hardware, and the cameras. A schematic of the apparatus is shown in Fig. 3.2.

An oxidized silicon wafer with diameter of 100 mm and thickness $525 \pm 5 \, \mu m$ was used for the standard (HPi) case. Using a digital goniometer, the static contact angle with water was
measured at nominally 55°. The back side of each surface was screen-printed with a silver paste (ESL 599-E) to create an integrated electrical resistance heater. The heater was designed to have a coverage area with nominal diameter of 50 mm, providing 0.25 Ω of resistance through a series of thin traces nominally 0.5 mm thick (see horizontal heated lines in Fig. 3.3). Leads were attached to the heater using a conductive epoxy (Atom Adhesives AA-DUCT 2979) connecting it to a 20V, 120A maximum DC power supply (HP 6011A). A thin layer of flat black coating (Rustoleum® 248903) of known emissivity ($\varepsilon = 0.97$) was spray painted on the entire back of the wafer, covering the heater and allowing for thermal imaging to determine the temperature across the back side of the wafer over time. This paint layer was shown to negligibly affect temperature measurement by comparing temperature on the top surface with a thermocouple.

Fig. 3.2: Schematic of experimental apparatus.

HPo surfaces were created by coating the top surface of an oxidized silicon wafer with a thin layer (100 nm) of chromium via electron-beam evaporation followed by a thin layer (200
nm) of natively hydrophobic DuPont™ Teflon® (commercial brand of polytetrafluoroethylene, PTFE). Teflon was spin-coated onto the surface at 1000 rpm for 20 s, resulting in static water contact angles of 120°, as measured by a goniometer.

SH surfaces were fabricated by first patterning surface features onto the same type of silicon wafer used for HPi and HPo surfaces using standard photolithography methods. A small, 6 mm - diameter circular ”target” was left unpatterned at the center of the wafer to prevent wetting in the jet stagnation region. After patterning, wafers were etched using reactive ion etching (RIE) to create microscale posts with a pitch of 16 µm, diameter of 7 µm, and depth of 25 µm (see Fig. 3.1a). This resulted in a cavity fraction of nominally 85%. Etched silicon wafers were then rendered SH with a Teflon coating and screen-printed with the integrated heater described previously. SH surfaces that were fabricated for this study exhibited a static contact angle of 155 ± 4°.

To provide a fully developed impinging jet, a long (15 cm) stainless steel blunt-tip needle with inner radius \( r_j = 1.275 \) mm was placed nominally 5 cm above the wafer. Upstream the nozzle was connected to a rotameter which was attached to a pressure tank containing deionized water. This allowed for an adjustable jet flow rate and controlled the jet \( \text{Re}_D \). The wafer was placed on a hollow, thin-walled stainless steel cylinder which inhibited conduction from the wafer due to its relatively low thermal conductivity and thin supporting upper rim. The water emptied from the edges of the wafer into a 3D-printed plastic collection container that also supported the entire subsystem.

A thermal camera (FLIR® SC6100) recorded temporally varying temperature data at a frame rate of nominally 200 Hz. The camera resolution was 320 x 256 pixels over a viewing window of approximately 55 x 45 mm. Data was analyzed only over this viewing window as it was insufficient to examine the entire wafer. High-speed images were also taken with a Photron Fastcam APX RS located above the wafer at an angle of about 40° normal to the surface. The high-speed images were acquired at a frame rate of 500 Hz with a spatial resolution of 1024 x 1024 pixels over a viewing area of nominally 70 x 70 mm.

### 3.4.2 Data Acquisition

Test surfaces were electrically heated until the maximum surface temperature reached the prescribed value, after which jet flow was initiated and an electronic trigger to both cameras was
enabled. Values for initial surface temperature and jet Re$_D$ are given in Table 3.1. Synchronized visual and temperature data was then collected. A sample image of thermal data collected from beneath a wafer is shown in Fig. 3.3. The temperature signature of the thin film heater is visible in the hottest regions as thin horizontal lines. The region where liquid is in contact with the surface is apparent by the significantly lower temperature at the center. The edge of this region is nominally where the temperature gradient is highest, both spatially (the highest transition in temperature in Fig. 3.3) as well as temporally, and has been shown to be a good approximation for the thin film spreading front [49]. Since the heater does not cover the entirety of the wafer, there is an initial variation of temperature across the surface, with the highest values near the center, and lower temperatures outside the heater radius.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Units</th>
<th>Values</th>
<th>Uncertainty</th>
</tr>
</thead>
<tbody>
<tr>
<td>Surface Type</td>
<td>–</td>
<td>HPi, HPo, SH</td>
<td>–</td>
</tr>
<tr>
<td>Static Contact Angle</td>
<td>deg, °</td>
<td>55, 123, 155</td>
<td>±4°</td>
</tr>
<tr>
<td>Initial Temperature</td>
<td>°C</td>
<td>200, 280, 320</td>
<td>±3 °C</td>
</tr>
<tr>
<td>Jet Re$_D$</td>
<td>–</td>
<td>6, 12, 18 (x 10$^3$)</td>
<td>±0.5 x 10$^3$</td>
</tr>
</tbody>
</table>

### 3.4.3 Data Analysis

Local heat flux from the wafer to the jet was also calculated as a function of $r$ and $t$ by averaging the radial temperature data over 20 equally-spaced radial lines (shown in Fig. 3.3) and numerically solving the energy balance of Eq. (3.2) for a local control volume as illustrated in Fig. 3.1b:

$$q''(r,t) = \frac{\delta}{r} \frac{\partial}{\partial r} \left( k_{si} r \frac{\partial T}{\partial r} \right) + \frac{q_H}{\pi r_H^2} - \delta \rho c_p \frac{\partial T}{\partial t}$$

where $q''$ is the instantaneous local heat flux from the surface to the fluid as a function of radial position, $r$, and time, $t$. The first term on the right hand side represents conjugate heat transfer through the wafer in the radial direction, where $\delta$ is the wafer thickness and $k_{si}$ is the thermal
The conductivity of silicon as a function of the measured local surface temperature, $T$. The next term consists of the heater input, $q_H$ divided by the heater coverage area, where $r_H$ is the radius of the heater regions. The last term represents transient storage of energy in the wafer, a function of the density, $\rho$, and specific heat, $c_p$, of silicon, as well as $T$ and $t$. After experimentation, it was determined that the heater input was at least two orders of magnitude lower than the other terms and was therefore deemed negligible during the initial quenching process (occurring in less than 1 s). Heat loss during impingement due to natural convection and radiation on the back side of the wafer was also neglected as it was orders of magnitude smaller than the other terms.

The measured $T(r,t)$ was used to calculate $q''(r,t)$ by numerically solving Eq. (3.1). Evaluating the temporal and spatial derivatives of $T(r,t)$ amplifies the measurement error. This was mitigated by using least-squares fits of second-order polynomials to the temperature profile temporally using seven values and radially using 31. After $q''(r,t)$ was calculated, Fourier’s 1D conduction law ($q'' \approx -k_{si}(T_s - T_b)/\delta$) was then used to correct the top surface ($T_s$) temperature using the silicon thermal conductivity ($k_{si}$), local heat flux ($q''$), and wafer thickness ($\delta$). An average of the top and bottom temperatures was then used to recompute the local heat flux from the heated wafer to the water.
The measured temperature data was also examined by computing the rate of total energy transferred from the surface, \( E(t) \), as the thin film spreads over time:

\[
E(t) = \int_0^t \int_0^{r_{lim}} 2\pi r \delta c_p(T(r,t) - T_j) dr dt
\]  

(3.2)

where \( r_{lim} \) is a standard radius over which integration is performed for each case (set to be 20 mm) and \( T_j \) represents the initial temperature of the jet. The derivative of \( E(t) \) throughout time also provided an average heat transfer rate from the surface to the water across the viewable area of the wafer (nominally \( 0 \leq r \leq 20 \) mm). The value of \( E(t) \) computed from Eq. (3.2) was normalized \( (\hat{E}(t)) \) by the total initial thermal energy stored in the wafer over the same radial extent (Eq. (3.2) with \( T(r,t) = T_0(r) \)) in order to compare surfaces with varying initial temperature.

Potential sources of uncertainty that impact calculations include uncertainty in temperature measurements and material properties, as well as error in spatial correlations and calibrations. Performing a root-mean-square propagation of error analysis of the contributing components in Eq. (3.1) resulted in an uncertainty on the maximum heat flux of \( < \pm 5\% \). Table 3.1 presents experimental conditions that were varied in this study and the typical uncertainties in the measured values.

3.5 Results and Discussion

Figure 3.1b illustrates jet impingement quenching of a surface, where the jet rapidly transitions into a radially expanding liquid thin film and concomitantly the surface cools rapidly. This behavior was captured in the high-speed images shown in Fig. 3.4, which will be discussed in detail below. Localized boiling occurs in the vicinity of the thin film edge, and correspondingly the heat flux attains a local maximum at this position (see corresponding data at the bottom of Fig. 3.4). This section of the paper presents qualitative and quantitative results that demonstrate the quenching behavior is altered dramatically when the target surface is SH. Specifically, the results show that the spreading speed of the expanding thin film is notably slower on the SH surface. Further, the maximum instantaneous thermal transport is smaller on the SH surface. As a consequence of these two dynamics, the quenching time for the SH surface is appreciably longer than for the smooth HPi or HPo surfaces.
Fig. 3.4: Images showing impingement at multiple times on the HPi (top), HPo (middle) and SH (bottom) surfaces with $T_0 = 320^\circ\text{C}$ and $Re_D = 12,000$. Temperature and heat flux as functions of time and radial position are also shown for the SH surface, with shaded regions indicating typical uncertainty.
Initial surface temperature and jet $Re_D$ also influence the transient thermal transport between the surface and the water. The effects of these parameters was similar for all surface types, and in general followed trends and orders of magnitude previously examined for HPi surfaces [44–46]. First we consider the hydrodynamics of thin film spreading as a function of initial surface temperature, jet $Re_D$, and surface type, and quantify spreading times for each scenario. Subsequently, the heat transfer rate is quantified from the measured temperature data, following the approach discussed above, for the same scenarios. Finally, qualitative observations regarding wetting of the surface microstructure during the transient process are provided.

### 3.5.1 Hydrodynamics

Images showing jet impingement at $t = 0, 0.05$, and $0.1$ s after the start of impingement are shown in Fig. 3.4 for the SH, HPo, and HPi surfaces at the same initial temperature ($T_0 = T(t = 0, r = 0 \text{ mm}) = 320 \, ^\circ\text{C}$) and for the same jet $Re_D$ (12,000). At $t = 0$ s the unpatterned center target region of the SH surface is clearly visible. Unlike the SH surface, the HPi and HPo wafers had no microscale roughness added and thus retain their original mirror-finish. The light streaks in the background on these surfaces are reflections of the surroundings. Also shown at the bottom of Fig. 3.4 are plots showing the local wafer temperature and heat flux for the SH surface at the same three instants in time.

The thin film region, which forms after water contacts the surface, is clearly seen at $t = 0.05$ and $0.1$ s as the round, glassy region in the center of the image surrounded by a spray of droplets lifting off the surface as the water boils. As mentioned before, a boiling region formed on all superheated wafers that were tested in this study, but the impact on the spreading behavior varied with surface type. Liquid water only contacts the surface in regions where the wafer is cooled below a sustainable film boiling temperature. An enhanced Leidenfrost effect was expected to occur with the SH surfaces due to the non-wetting hydrophobic microstructure, which allows a stable vapor layer to form beneath the liquid between the posts, as has been seen with pool boiling, microchannel flow and droplet impingement research [11, 15, 61]. The images of Fig. 3.4 reveal that the thin film region is smallest for the SH surface compared to the other surfaces at the same time due to this enhanced effect. The spreading rate of the thin film region impacts heat transfer
because it alters how rapidly water contacts and cools a surface (more rapid spreading facilitates more rapid heat transfer).

Fig. 3.5: Non-dimensional thin film radius \( r_f/a \) and location of maximum local heat flux \( r_q/a \) as functions of time and surface type for \( T_0 = 320 \, ^\circ\text{C} \) and \( \text{Re}_D = 12,000 \).

As shown in the images and the \( T(r,t) \) and \( q''(r,t) \) data shown at the bottom of Fig. 3.4, the thin film front location is at approximately the same radius as the measured maximum local heat flux. Figure 3.5 shows the normalized instantaneous radial position of the thin film front \( r_f/a \) determined from the high-speed images as well as the normalized location of maximum heat flux \( r_q/a \) for all three surfaces. The data of Fig. 3.5 correspond to \( T_0 = 320 \, ^\circ\text{C} \) and \( \text{Re}_D = 12,000 \), but all other cases show similar trends. The instantaneous edge of the thin film, selected from the high-speed images, is shown with hollow blue markers (left axis) and the maximum heat flux location is shown with solid red markers (right axis). Locations of the edge of the jet radius and SH target radius are shown with blue and gray shading for reference. Nucleate boiling, which typically demonstrates the highest heat flux, occurs over a finite region, but the peak in heat flux occurs at the center of this nucleate boiling region. The thin film front forms slightly beyond this point radially as transition boiling and film boiling begin to dominate. The maximum heat flux location is thus always slightly smaller than the measured thin film location, consistent with previous findings [43, 62]. However, since the two locations are very close, the location of maximum heat flux is employed in subsequent comparisons presented here.
Fig. 3.6: Time for the jet thin film radius to reach various radii downstream from the stagnation point ($t_n$) as a function of (a) $T_0$ ($\text{Re}_D = 12,000$) as well as (b) jet $\text{Re}_D$ ($T_0 = 280 \, ^\circ\text{C}$).

The thin film spreads most rapidly on the HPi surfaces and thus at the same instant in time the initial interfacial contact between the surface and the thin film is the greatest for this surface.

Fig. 3.7: Maximum thin film spreading for the SH surface at (a) $\text{Re}_D = 6,000$ and (b) $\text{Re}_D = 12,000$, with $T_0 = 280 \, ^\circ\text{C}$.
In contrast, the spreading rate is slowest on the SH surface, due to the enhanced Leidenfrost effect. Figure 3.6 compares the measured time, $t_f$, for the thin film to spread 4, 8, and 12 jet radii from the stagnation point for the three different surfaces. The left panel of the figure shows results at $T_0 = 200, 280, \text{ and } 320 \degree C$ (all at $Re_D = 12,000$), while the right panel shows results at $Re_D = 6,000, 12,000, \text{ and } 18,000$ (all at $T_0 = 280 \degree C$). Shorter data bars indicate shorter time to reach the corresponding normalized location on the wafer, or faster spreading and greater liquid contact. For all cases, the fastest and slowest spreading occurred on the HPi and SH surfaces, respectively. For example, at $T_0 = 200 \degree C$, $t_f$ for 12 jet radii was 63% lower for the HPi case compared to the SH surface. Indeed, for all values of $n$ (4, 8, and 12), $t_f$ was always largest on the SH surface and this behavior was enhanced as $T_0$ increased. This increase in $t_f$ quantitatively supports the idea that SH surfaces generally enhance Leidenfrost conditions, delay thin film spreading and suppress boiling. The net result should be an overall slower rate of surface cooling. Results for the HPo surface show that for $n = 4$ and 8 the HPo and HPi surfaces yield similar $t_f$ values, although they are always slightly greater for the HPo surface. At $n = 12$, however, $t_f$ on the HPo surface showed greater deviation from the HPi data.

The influence of jet $Re_D$ on $t_f$ is illustrated in Fig. 3.6b. In general, increasing the inertia of the subcooled liquid yields faster cooling and spreading, regardless of surface type. As $Re_D$ increases, the $t_f$ decreases, although this effect appears to diminish with increasing $Re_D$.

Note that at $Re_D = 6,000$, the thin film never spreads to 12 radii for any of the surfaces. Further, for the SH surface, the film never even reaches 8 radii. This occurs because the film breaks up into droplets at a point where the jet momentum is balanced by the inward pull of surface tension on the film, which occurs at smaller radial positions as $Re_D$ is decreased. This is visualized in Fig. 3.7, where instantaneous images at times much greater than 0.1 s are shown with $Re_D$ of 6,000 and 12,000. At $Re_D = 12,000$ the maximum spread radius is 2.6 times greater than for the $Re_D = 6,000$ case. In this work the maximum thin film radius was always greatest for the HPi surfaces and smallest for the SH surfaces.

### 3.5.2 Heat Transfer

The temperature and heat flux data shown at the bottom of Fig. 3.4 for the SH surface reveals the following important points. First, the surface cools rapidly at the center of the wafer
and $q''$ is a maximum at $t = 0$ s and $r = 0$ mm (stagnation point). Second, with increasing time the surface cools and approaches the temperature of the impinging jet in the thin film region, where $q''$ also decreases rapidly. The vertical dashed lines shown for $t = 0.05$ and 0.1 s mark the edge of the thin film region, which also corresponds to the boiling front. This is shown in the high-speed images of Fig. 3.4 as the white or lighter region where droplets are ejected from the surface. The local heat flux is at a maximum value (for $t > 0$ s) just inside the boiling or thin film front.

Surface temperature as a function of time and radial position for the SH surface is shown in a 3-D plot in Fig. 3.8a for $Re_D = 12,000$ and $T_0 = 320 \, ^\circ C$. As expected, the stagnation region ($r = 0$ mm) cools much more rapidly compared with outer radial positions. Large temporal and spatial

![Fig. 3.8: Surface temperature ($T(r,t)$) and heat flux ($q''(r,t)$) data at $T_0 = 320 \, ^\circ C$ and $Re_D = 12,000$. (a) Wafer surface temperature as a function of radius and time for the SH surface. (b-d) Heat flux as a function of radius and time for the SH (b), HPo (c), and HPi (d) surfaces.](image-url)
temperature gradients are evident as the surface rapidly cools due to the spreading thin film. The higher surface temperature far from the stagnation point suggests that droplet movement past the surface beyond the edge of the thin film (shown in Fig. 3.4) exerts negligible influence on surface cooling. Figure 3.8b shows the instantaneous local heat flux for the same conditions and parameters as Fig. 3.8a and shows more clearly how the SH wafer cools. Thermal energy is transferred due to spatial and temporal temperature gradients and the peaks in heat flux correspond to the locations in Fig. 3.8a where the temperature gradients are a maximum. Starting with the maximum heat flux at \( r = 0 \) mm and \( t = 0 \) s, the peak heat flux decreases in magnitude with increasing time due to the decreasing difference in surface and jet temperatures. The peak heat flux also decreases with increasing radial location as the thin film spreads and heats up, again lowering the temperature difference.

Instantaneous heat flux for the HPo and HPi surfaces are shown in Figs. 3.8c and 3.8d, respectively, for the same \( \text{Re}_D \) and \( T_0 \). Similar overall trends exist for all surfaces. The initial heat flux is highest for the HPi surface (d) and it cools more rapidly than do the SH (b) or HPo (c) surfaces, as evidenced by the strong curvature in the heat flux peak with increasing time. More quantitative heat flux data will be compared for the three surfaces later in this section.

Figure 3.9 provides the surface temperature and local heat flux as functions of radial position for all surface types at three different times: \( t = 0, 0.05, \) and \( 0.15 \) s. The curves shown represent temporal slices of the 3-D plots shown in Fig. 3.8 and correspond to the same experimental conditions. At \( t = 0 \) s the surface temperature distribution for all three surfaces is similar, however, the peak in local heat flux for the SH case is 29% lower than for the HPi surface and the jet spreads the slowest across it. There exists a negligible radial temperature gradient near the stagnation region at \( t = 0 \) s, although large temporal temperature gradients exist at larger \( r \) that differ significantly for the three surfaces.

Figure 3.9b again illustrates that while the thin films on the HPi and HPo surfaces have spread to nearly the same radial location (\( r = 7 - 8 \) mm) by \( t = 0.05 \) s, on the SH surface the thin film remains confined to \( r \leq 4 \) mm. Consequently, there is a much smaller area that has been cooled by the jet. By \( t = 0.15 \) s (panel c), much of the surface has been cooled to subcritical temperatures, and the peak local heat flux is greatly diminished for all surfaces.
Fig. 3.9: Temperature (left axis) and local heat flux (right axis) as a function of radius at (a) $t = 0$ s, (b) 0.05 s, and (c) 0.15 s for $T_0 = 320 \, ^\circ$C and $Re_D = 12,000$. SH surfaces clearly have the lowest initial heat flux values, and cool the slowest.

To further connect the heat transfer behavior to the thin film hydrodynamics, the total normalized energy transferred from the wafer ($\hat{E}(t)$) was computed using Eq. (3.2). Figure 3.10a provides data illustrating the differences in $\hat{E}(t)$ between surface types based on the extreme $T_0$ values. Initially, significant thermal energy is transferred to the jet, indicated by the steep slope of the $\hat{E}$ vs. $t$ curve. The curve then begins to level off as much of the surface has cooled and
the temperature is no longer changing significantly in time. This is concomitant with a transition from latent to sensible energy transfer as the entire surface cools, boiling ceases, and single-phase convection becomes the dominant mode of heat transfer.

Due to high spatial temperature gradients, $\hat{E}(t)$ can exceed unity at large $t$. Values of $\hat{E}(t)$ in excess of unity can be interpreted as conjugate energy transfer from outside the $r_{lim}$ region. This
is especially true for SH and HPo surfaces, where the thin film spreads more slowly (see Fig. 3.6a). The slope of these energy curves corresponds to the average heat transfer rate (qualitatively, as the energy curves are normalized), and is relatively constant for a significant portion of quenching time. Figure 3.10a shows the highest rate of heat transfer at lower temperatures for all surface types, indicated by the steeper slopes for corresponding lower-temperature cases, which will be quantified later.

Figure 3.10b shows the differences in $\dot{E}(t)$ between the extreme Re$_D$ cases, with Re$_D = 12,000$ data nominally midway between the other Re$_D$ cases. Once again, energy transfer is initially high (steep slopes), which either gradually decreases, for low Re$_D$, or suddenly, for the highest Re$_D$. As was mentioned previously, the lower Re$_D$ values lead to smaller hydraulic breakup radii and thus the thin film does not cover the entire $r_{lim}$ region. This results in the initial stored thermal energy not being completely removed in the time frame explored, shown by the curves not reaching a value of unity.

Qualitatively, surface type also impacts energy transfer. SH surfaces consistently transfer thermal energy to the water the slowest, while HPi surfaces do so most rapidly. For varying $T_0$, HPo data sets lie nearly halfway between those of SH and HPi surfaces. For lower values of Re$_D$, HPo tends to promote behavior more similar to SH surfaces, but for higher values of Re$_D$, the HPo and HPi surfaces have similar energy removal rates.

The derivative of $E(t)$ with respect to time provides an average heat rate throughout time over the entire viewing window ($0 \leq r \leq 20$ mm). In order to better quantify the qualitative results shown in Fig. 3.10, heat rate values were calculated as the slope of a linear curve-fit over the first 0.05 to 0.3 s of $E(t)$ data, depending on the parameters of the particular case. In Fig. 3.11a, the initial average heat rate, $\bar{q}_0$, (in units of W) is plotted as a function of $T_0$ (horizontal axis) for all surface types (marker shape). Figure 3.11b shows a similar comparison as a function of jet Re$_D$. General results are discussed first with regard to $T_0$ and Re$_D$, followed by an analysis of the effect of surface type.

There are competing effects of $T_0$ on heat rate, leading to the results shown in Fig. 3.11a. As $T_0$ increases, there is more initial energy available to be removed, but there is also delayed surface contact due to boiling arresting the thin film and increasing the time for heat removal. This results in a seeming lack of temperature effect on SH and HPo surfaces. However, for the HPi
surface at $T_0 = 200 \, ^\circ C$ there is a high initial heat rate as the surface does not need to cool as much in order for water to contact, increasing the thin film spreading and corresponding heat rate (see Fig. 3.10a).

In contrast to the competing effects of $T_0$, increasing $Re_D$ always increases initial heat rate for all cases, as seen in Fig. 3.11b. This is due to the influence of faster thin film spreading (see Fig. 3.6b), which directly impacts the rate at which heat is transferred from the surface to the wafer.

Surface type has an effect on heat rate as well, as can be seen in Figs. 3.11a and 3.11b. In general, HPi surface heat rates are consistently higher than those of SH surfaces, corresponding to the steeper slopes seen in Figs. 3.10a and 3.10b. Interestingly, the lowest $T_0$ case shows a remarkable difference between HPi surface heat rate and those of the other surfaces (nominally 3 times higher). This supports the idea that increased hydrophobicity preserves Leidenfrost condition even at lower temperatures. HPo surface values range from near those of SH surfaces for low $T_0$ and $Re_D$ to approximately the same as HPi surfaces for the highest $T_0$ and $Re_D$ cases. This transition in behavior could be related to hydrodynamic effects of hydrophobicity (which are similar for SH and
HPo surfaces) competing with thermal effects caused by increased liquid-surface contact (which smooth HPo and HPi surfaces have in common).

The initial average heat rate provides some idea of heat transfer behavior across the wafer. However, local heat flux values also add insight into the thermal transport from the surface to the jet, as was shown in Figs. 3.8 and 3.9. Beyond $r \approx 6$ mm, the maximum local heat flux on any surface decreases uniformly with increasing radius (shown by the peak in Figs. 3.8b - 3.8d). In order to condense the data for simpler comparison and provide another metric of heat transfer comparison between the three surface types, maximum local heat flux values were arithmetically averaged from $r = 6$ to 20 mm. The averaged maximum heat flux values ($\bar{q}_{\text{max}}''$) are given as functions of $T_0$ in Fig. 3.12a, with different markers indicating surface type, similar to Fig. 3.11a. In contrast to the heat rate comparison, there is a definite increase in $\bar{q}_{\text{max}}''$ with increasing $T_0$, showing strong dependence on driving temperature difference. Figure 3.12b shows $\bar{q}_{\text{max}}''$ plotted against $\text{Re}_D$. A general trend of increasing $\bar{q}_{\text{max}}''$ with increasing $\text{Re}_D$ is also seen, regardless of surface type, just as for initial heat rate.

![Figure 3.12](image.png)

**Fig. 3.12**: Maximum local heat flux values averaged over outer radii throughout time, given as a function of (a) $T_0$ ($\text{Re}_D = 12,000$) and (b) jet $\text{Re}_D$ ($T_0 = 280 \, ^\circ\text{C}$), for all surface types.
It is also evident from Figs. 3.12a and 3.12b that \( \bar{q}'_{\text{max}} \) for the HPi surface is consistently higher than for the SH surfaces. The largest difference can be seen at the lowest \( \text{Re}_D \), where \( \bar{q}'_{\text{max}} \) is 263% higher for the HPi surface than the SH surface. The average difference for the other conditions is 1.2 MW/m\(^2\) (50 - 80%) higher on HPi surfaces compared with SH surfaces. In fact, the flow rate for SH surfaces must be triple that of HPi surfaces to reach approximately the same \( \bar{q}'_{\text{max}} \) value.

Values of \( \bar{q}'_{\text{max}} \) vary for HPo surfaces. At \( \text{Re}_D = 6,000 \), the HPo surface \( \bar{q}'_{\text{max}} \) is approximately equal to that for SH surface. However, for the maximum \( \text{Re}_D \) and \( T_0 \) values, \( \bar{q}'_{\text{max}} \) on HPo and HPi surfaces is very similar. This changing dynamic could, again, be due to the competing hydrodynamic and thermal effects. For the case of low \( \text{Re}_D \), a smaller thin film spread is observed and thus thermal effects may be less important than at higher \( \text{Re}_D \), where the increased contact area of a larger thin film region on the HPo surfaces enables them to behave more similarly to the HPi surfaces.

### 3.5.3 Surface Wetting

For completeness, an additional fluid dynamic phenomenon worth noting is the apparent microstructure wetting that was observed on the SH surfaces at elevated surface temperatures. Figure 3.13 shows images of the spreading thin film front on SH surfaces at \( T_0 = 25 \^\circ\text{C} \) (a), 100 \(^\circ\text{C} \) (b), and 320 \(^\circ\text{C} \) (c). All of these images were captured at later times (\( t \approx 1 \text{s} \)), when all boiling activity had ceased. These images suggest that when a thin film forms on an unheated surface (case a) the water at the liquid-surface interface exists in the Cassie-Baxter state, where air fills the cavities between the microstructure features. However, for Fig. 3.13b (\( T_0 = 100 \^\circ\text{C} \)), water has penetrated the cavities between the local microstructures and localized wetting of the surface appears to exist. This is manifest by the localized darker regions on the surface.

This wetted condition, known as the Wenzel state, appears to occur when the wafer is initially at an elevated temperature and the vapor that is generated cannot all escape and flow outwards. Instead, some of the vapor condenses as it moves radially outward where the surface is cooler. Figure 3.13c shows an image of a surface at an initial temperature of 320 \(^\circ\text{C} \). At this temperature the entire thin film region is “dark” and suggests the entire surface has been wetted. Importantly, this wetting did not yield a permanent change to the surfaces, although the water re-
mained trapped in cavities until the surface was completely dried. However, once the water was removed, the surface did not “wet” again when the surface was heated to sub-saturation temperatures. The overall effect on heat transfer of this wetting behavior is unclear and merits further investigation.

Fig. 3.13: Images of SH surface wetting with jet $Re_D = 12,000$ at $t \approx 1$ s. (a) Surface at ambient temperature ($T_0 = 25 \, ^\circ C$) with no visible wetting. (b) Surface with temperature the near center above saturation ($T_0 = 100 \, ^\circ C$), with corresponding wetting in the same region. (c) Entire surface temperature well above saturation temperature ($T_0 = 320 \, ^\circ C$) with wetting over the entire surface.
3.6 Conclusions

Jet impingement transient cooling of superheated smooth HPi ($\theta = 55^\circ$), smooth HPo ($\theta = 123^\circ$) and micropost-patterned SH ($\theta = 155^\circ$) surfaces was investigated. Initial surface temperature was varied from $T_0 = 200$ to $320 \, ^\circ\text{C}$ and jet Reynolds number of $\text{Re}_D = 6,000, 12,000,$ and $18,000$ were explored. Typical jet quenching behavior was observed for all cases, including the rapid formation of a convective liquid thin film bounded by a region of vigorous boiling and localized high heat flux. Conclusions about heat transfer can be drawn based on the three parameters varied: $T_0$, jet $\text{Re}_D$, and surface type.

Results from this study show that for increasing $T_0$ the spreading of the jet thin film is impeded on all surface types. This is due to rapid vaporization creating a barrier to liquid-surface contact and yielding high localized heat transfer. However, the increased temperature also increases the driving potential for heat transfer, resulting in higher local heat fluxes. Increased driving temperature difference and impeded thin film spreading result in competing influences on average heat transfer rate over the surface.

Increasing jet $\text{Re}_D$ adds fluid momentum that can overcome the effects of thermal impedance, leading to faster and farther thin film spreading. This was reflected in the numerical heat transfer comparisons of heat flux and heat rate, which both increased an average of 200% for all surface types when jet $\text{Re}_D$ was increased from 6,000 to 18,000.

Finally, surface type was shown to have a large impact on jet impingement heat transfer. The thin film spread more slowly on the SH surfaces compared to HPi and HPo cases due to an enhanced Leidenfrost effect, which also impeded heat transfer. Quantitative comparisons between SH and HPi surfaces showed the former had up to 70% lower values for both averaged maximum heat flux and initial heat rate.
CHAPTER 4. EFFECT OF SUPERHYDROPHOBIC SURFACE MICROSTRUCTURE ON TRANSIENT JET IMPINGEMENT COOLING

This chapter is in preparation journal peer-review. The format has been modified to meet the requirements of this thesis.

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4.2 Abstract

Water jet impingement is an effective method of rapid cooling, but heat transfer is highly dependent on surface condition and material properties. Superhydrophobic (SH) surfaces are useful for a variety of applications, but have not been studied in depth with regard to jet impingement heat transfer at high surface temperatures. Here, thin silicon wafers are ion-etched to form different micropatterns (posts or holes) and coated with Teflon to render them SH. On the reverse side, a screen-printed resistance heater is applied. Surfaces are heated to between 200 to 320 °C, and then impinged on by an axisymmetric, room-temperature water jet with Re_D = 6,000 to 18,000. Temperature is measured beneath the surfaces with a thermal camera and local surface heat flux from the surface to the jet is calculated based on an energy balance. High-speed images from above capture jet spreading and boiling behavior. Heat transfer is shown to be highly dependent on jet Re_D, but not on initial surface temperature. Results also show that varying microstructure by feature shape, width or diameter, height, and pitch (distance between features) overall had little effect on heat transfer. Shorter microstructures tended to promote slightly higher heat transfer, where a decrease in post height from 25 to 5 µm can increase local heat flux up to 91% for low Re_D cases.
4.3 Introduction

Jet impingement is known to be an effective method of rapidly cooling hot objects through convective heat transfer. If the jet is liquid and the surface is sufficiently hot, boiling can occur and enhance heat transfer further through latent energy transfer. Both the fluid jet properties and the properties of the material being cooled significantly impact heat transfer. For example, since water is a better conductor than air, it is generally used in applications such as metal quenching and nuclear-reactor emergency cooling [42, 57]. Material conductivity [43] and wettability [27–29, 37] play a significant role in heat transfer in this type of scenario.

Liquid jet impingement heat transfer generally follows the traditional pool boiling relationship between surface temperature and heat flux. Initially, increasing surface temperature increases heat flux to a water jet as the nucleate boiling regime is reached, creating a mixture of liquid and vapor flow. However, as temperature continues to increase the liquid water becomes wholly vapor and no longer contacts the surface. This is known as the Leidenfrost point, or minimum film boiling, and here heat transfer is much lower due to significantly lower thermal conductivity of vapor compared with liquid water.

A high-speed image of a jet impinging on a superhydrophobic surface and corresponding diagram of the behavior is shown in Fig. 4.1. In water jet quenching scenarios, different regions of an impinging jet cool the surface by the different modes of heat transfer depending on surface condition and temperature. The jet shown in Fig. 4.1 has radius $a$, flow rate $Q$, and viscosity $\nu$ and spreads out radially in a thin film when the local temperature is sufficiently low to allow liquid contact. A stagnation region forms directly under the jet, characterized by high pressure and high heat flux [49]. The thin film front occurs where liquid-surface contact is reduced due to nucleate and film boiling, as is indicated by the dashed line in Fig. 4.1. As shown in this case, if the surface is not cooled quickly enough for water to contact, droplets are ejected at the thin film front and hover over the surface in a Leidenfrost state.

Since the highest surface cooling occurs when liquid is in contact with the surface, heat transfer can be quantified by identifying this region. It is defined by the thin film front spreading, or how quickly the Leidenfrost state is overcome and liquid contacts the surface, and by measuring local heat flux. Much of boiling and spreading behavior is indicated by the initial surface temperature [44]. In addition, more rapid spreading, leading to increased heat flux, has been observed due
Fig. 4.1: A photograph from above of transient heat transfer on a superheated SH surface is shown, with a side view radial cross-section beneath for clarification (thin film height is exaggerated).

to higher jet flow rate, while nozzle-to-surface length has varying effects [46]. Heat flux has also been shown to decrease with increasing distance from the stagnation region as well as decreased jet subcooling [41, 42].

Material effects, including local roughness and surface energy, affect the Leidenfrost point and thus the amount of heat that can be transferred from a surface. Surface energy directly alters the liquid-solid contact angle of a water droplet on a surface by changing the balance of cohesive forces between water molecules and attractive forces to the surface. When surface energy is reduced, water can bead up on a surface which is the case for hydrophobic surfaces, which have contact
angles greater than 90°. A hydrophobic surface can be made SH, or have a contact angle greater than nominally 150°, by altering the surface geometry to include micro- or nano-scale roughness, such as the micro posts and micro holes shown in Fig. 4.2. These SH surfaces are characterized by structure height, $h$, width or diameter, $d$, and pitch (distance between features), $w$. Cavity fraction, or the ratio of the top surface contact area of the structures to the total projected area of the surface, can also be used to describe SH surfaces. In addition, the microstructure can have different designs and be “open” like the round posts in Fig. 4.2 or “closed” like the square holes.

Water does not wet the SH microstructure if the Laplace pressure is not exceeded, known as the Cassie-Baxter state. Instead, air-filled gaps provide a lower aggregate surface energy and a local slip condition for water flowing above them. However, this reduction in liquid-surface contact also inhibits heat transfer from the surface as thermal energy is only conducted through the microstructures to the liquid, which has been demonstrated in droplet impingement, [61] microchannel flows [15], and pool boiling [11]. Furthermore, Leidenfrost effects can be enhanced on SH surfaces because of the minimal surface contact [12].

![Fig. 4.2: (a) SEM image of open design microstructure pattern in the form of round posts, with diameter ($d$), height ($h$), and pitch ($w$) labeled. (b) SEM image of closed design microstructure pattern in the form of square, hole microstructures with similar dimensional labeling.](image)

Research regarding jet impingement on heated SH surfaces specifically is limited to a few theoretical and experimental studies at subcritical temperatures [27–29] and one at supercritical temperatures (see Chapter 3). Subcritical work showed that increasing slip (experimentally in-
creased by increasing pitch and decreasing diameter of SH micropost surfaces) led to a reduction in contact area, which resulted in lower convective heat transfer. For superheated SH surfaces, significantly slower cooling time due to lower heat fluxes was found when compared to smooth hydrophilic (HPi) silicon surfaces, and to smooth, Teflon-coated hydrophobic (HPo) silicon surfaces (Chapter 3). To the authors’ knowledge, it is the only publication regarding superheated SH surfaces being cooled by jet impingement.

Other related work explored droplet impingement on superheated SH surfaces with varying microstructure [18]. Thermal atomization was quantified in this work, which is an indicator of nucleate boiling behavior (atomization is suppressed in the film boiling regime). This work showed that shorter microposts (decreasing $h$ in Fig. 4.2) tended to decrease the Leidenfrost point due to intermittent surface wetting, or water penetration within the microstructure. Increasing the pitch ($w$) between microposts decreased atomization due to increased area for vapor escape beneath the droplet, and hole surfaces tended to trap vapor in the microstructures, increasing atomization for some experimental conditions.

Here, superheated SH surfaces are cooled by an impinging liquid jet with an emphasis on varying surface microscale geometry including pitch, height, and type (holes vs posts) on oxidized silicon wafers. A room-temperature deionized water jet of radius, $a$, and kinematic viscosity, $\nu$, impinged on the surfaces at various jet Reynolds numbers (6,000, 12,000, or 18,000), defined as $\text{Re}_D = \frac{Q}{(\nu a)}$ and modified by altering flow rate, $Q$. Surface temperature was also varied between tests (200, 280, or 320 °C) by use of an integrated resistance heater on the back side of the wafers. Temperature changes over the course of impingement were used to calculate local heat flux, average heat rate, and total thermal energy transfer from the surfaces to the impinging jet to explore heat transfer differences between experimental conditions. Results of these tests reveal a strong dependence on jet $\text{Re}_D$, but very little variation with $T_0$ and surface type.

4.4 Methodology

4.4.1 Experimental Apparatus

To quantify the heat transfer from SH surfaces to an impinging jet, a similar experimental procedure from a previous work was followed (see Chapter 3). Polished, oxidized silicon wafers
525 \( \mu \text{m} \) thick with 100 mm diameter were used to fabricate the test surfaces. The polished side of the wafers were first patterned using standard photolithography methods, then etched by a reactive ion etching process to create the necessary microstructure. Each surface also had a 6 mm diameter smooth (unpatterned) “target” at the center where the jet impinges, as can be seen in the photo in Fig. 4.1, to reduce the possibility of hydrodynamic wetting in the microstructure due to the high stagnation pressure of the jet. Following the etching process, the surfaces had a chromium layer 100 nm thick added by electron-beam physical deposition for improved coating adhesion. A 200 nm thick layer of natively hydrophobic DuPont\textsuperscript{TM} Teflon\textsuperscript{®} (commercial version of polytetrafluoroethylene) was then applied via spin-coating.

Several types of SH surfaces were fabricated as summarized in Table 4.1, with references to surface parameters shown in Fig. 4.2. Renderings of all surfaces made are shown in Fig. 4.3. In general, the cases given in Table 4.1 can be divided into two categories. Cases 1 - 4 compare hole and post structures of equal cavity fraction for two low cavity fraction values. Cases 5 - 8 compare round post structures of equal cavity fraction but varying height or pitch and diameter. Therefore, the impact of varying cavity fraction, height, or pitch and diameter on heat transfer can be quantified.

Table 4.1: Surface fabrication parameters, as denoted in Fig. 4.2.

<table>
<thead>
<tr>
<th>Case Number</th>
<th>1</th>
<th>2</th>
<th>3</th>
<th>4</th>
<th>5</th>
<th>6</th>
<th>7</th>
<th>8</th>
</tr>
</thead>
<tbody>
<tr>
<td>Holes/Posts (H/P)</td>
<td>H</td>
<td>H</td>
<td>P</td>
<td>P</td>
<td>P</td>
<td>P</td>
<td>P</td>
<td>P</td>
</tr>
<tr>
<td>Round/Square (R/S)</td>
<td>R</td>
<td>S</td>
<td>R</td>
<td>S</td>
<td>R</td>
<td>R</td>
<td>R</td>
<td>R</td>
</tr>
<tr>
<td>Cavity Fraction (%)</td>
<td>50</td>
<td>75</td>
<td>50</td>
<td>75</td>
<td>85</td>
<td>85</td>
<td>85</td>
<td>85</td>
</tr>
<tr>
<td>Height (( \mu \text{m} )), ( h )</td>
<td>15</td>
<td>15</td>
<td>15</td>
<td>15</td>
<td>25</td>
<td>15</td>
<td>5</td>
<td>15</td>
</tr>
<tr>
<td>Pitch (( \mu \text{m} )), ( w )</td>
<td>24</td>
<td>24</td>
<td>24</td>
<td>40</td>
<td>16</td>
<td>16</td>
<td>16</td>
<td>8</td>
</tr>
<tr>
<td>Width/Diam. (( \mu \text{m} )), ( d )</td>
<td>19.0</td>
<td>20.4</td>
<td>19.0</td>
<td>20.0</td>
<td>7.0</td>
<td>7.0</td>
<td>7.0</td>
<td>3.5</td>
</tr>
</tbody>
</table>

Heaters were integrated on the back (unpolished) side of the wafers by a screen-printing process using a silver paste (ESL 599-E) deposited in a series of thin traces that form a pattern with
a nominal heated region diameter of 50 mm and overall resistance of 0.25 Ω. Following screen printing, the heaters and back sides of the wafers were spray-painted with a flat black coating (Rustoleum® 248903) of known emissivity ($\varepsilon = 0.97$) to aid thermal imaging. This coating was thin and did not impact heat transfer. Heater buses connected the thin traces, and were then connected via wire leads epoxied in place (Atom Adhesives AA-DUCT 2979) to a 20-V, 120-A DC power supply (HP 6011A). Using this process, surface temperatures up to 320 °C could be reached with uniform heat flux across the heater area.

During testing, surfaces were supported by a custom mounting apparatus (see Fig. 4.4). The wafers, oriented with the SH surface facing up, were placed upon a stainless steel cylinder with thin walls that inhibited significant conduction. A FLIR® SC6100 thermal imaging camera was mounted beneath the hollow region of the cylinder to allow visual access to the bottom of the wafer mounted above it; thermal images were recorded at nominally 200 Hz with a resolution of 320 by 256 pixels, corresponding to a nominally 55 x 45 mm viewing window. High-speed images of the experiments were obtained with a Photron Fastcam APX RS located above the wafer, at a
slight angle from vertical, which recorded images of approximately 70 x 70 mm at a rate of 500 Hz and a spatial resolution of 1024 by 1024 pixels.

Deionized water was supplied to the surfaces through a long (15 cm) stainless steel nozzle of inner radius $r_j = 1.275$ mm situated 5 cm above the wafer, connected to a pressurized tank. The tank pressure was regulated to alter flow rate, which was measured with a manual rotameter.

### 4.4.2 Experimental Procedure

Continuously water jet impingement was recorded by the time-synchronized high-speed and thermal cameras until the impinging water had reached steady state, defined when no further growth of the thin film region (see Fig. 4.1) was observed and phase change ceased, which was typically less than 1 s. Experiments were conducted while varying one of three parameters: surface temperature (200, 280, or 320 °C); jet Re$_D$ (6,000, 12,000, or 18,000); or surface type (round or square microposts or microholes). All experiments varying surface temperature were conducted with a constant Re$_D$ of 12,000, while all experiments varying Re$_D$ had a constant initial surface temperature of $T_0 = 280$ °C. A minimum of 5 repetitions of each scenario was performed. Duplicate wafers were made for most surface types to reduce uncertainty due to fabri-
cation imperfections. Observations of the surfaces during testing suggest no significant change in the hydrophobicity of the surface over the course of testing.

4.4.3 Analysis

The process used to obtain the local heat flux ($q''$) is identical to previous work (see Chapter 3) and is reviewed briefly here. Twenty radial lines over which surface temperature data was averaged are shown on a single thermal image during an impingement test in Fig. 4.5b. The averaged values of local temperature as a function of radius, $r$ and time, $t$ were used to calculate $q''$ from the wafer to the water using an energy balance around a control volume (Fig. 4.5a):

$$ q''(r,t) = \delta \frac{\partial}{\partial r} \left( k_{si} r \frac{\partial T}{\partial r} \right) - \delta \rho c_p \frac{\partial T}{\partial t} $$

(4.1)

where $\delta$, $k_{si}$, $\rho$, and $c_p$ are the silicon wafer’s thickness, thermal conductivity, density, and specific heat, respectively, as functions of the local wafer temperature, $T$. Thus, $q''$ at all times and spatial locations can be found by using Eq. (4.1) based off the measured $T(r,t)$. To reduce error amplification due to derivatives of finite points, temperature values were curve-fit quadratically in $r$ and $t$ over a number of data points and derivatives of those polynomial fits were used to compute $q''$. The calculated local heat flux was also averaged over the viewable area of the wafer (up to nominally $r = 20$ mm) to develop an average heat rate, and then integrated again over time to estimate the total energy transferred to the water as a function of time.

Similarly, the total energy transferred from the wafer to the water throughout time, $E(t)$, can be calculated based on the difference between the $T(r,t)$ and initial jet temperature:

$$ E(t) = \int_{0}^{r_{lim}} \int_{0}^{t} 2\pi r \delta \rho c_p (T(r,t) - T_j) dr dt $$

(4.2)

where $r_{lim}$ represents the radial extent of the viewable area of the wafer and the jet temperature is given as $T_j$. Calculated $E(t)$ values were normalized (denoted symbolically by a hat) by the initial temperature difference (setting $T(r,t) = T_0$ in Eq. (4.2)) in order to account for any discrepancies in heating. We will also utilize the instantaneous derivative of $E(t)$, which is equal to the average heat transfer rate and is another useful criterion for comparing thermal transport between surfaces.
Uncertainty in temperature and spatial measurements as well as temporal resolution and material properties was accounted for using a propagation of error method with Eq. (4.1). This resulted in typical uncertainty of less than 10% in local heat flux. This calculation neglects the heater component as well as potential losses due to natural convection or radiation. An estimate of the heat flux using this equation confirmed that each of these additional terms was negligible compared to the heat flux removed by the impinging jet, which is on the order of heat flux values reported in similar studies [43, 46, 49].

4.5 Results

In this study, $T_0$, jet $\text{Re}_D$ and SH surface patterning were varied to explore the effects each has on jet impingement heat transfer using three comparisons: time for the thin liquid film to spread to different radial locations ($t_n$), average initial heat rate from the surface to the water ($\bar{q}_0$), and maximum local heat flux values ($q''_{\text{max}}$). As will be shown throughout this section, decreasing $T_0$ and increasing $\text{Re}_D$ in general have similar effects of increasing jet impingement heat transfer. Except for decreasing height, varying microstructure was observed to have little impact on thermal transport.
Fig. 4.6: Case 5 surface cooling through time with $T_0 = 320 \, ^\circ C$ and $Re_D = 12,000$. High-speed images are shown in top panel at the three indicated times. Values of $T(r,t)$ and $q''(r,t)$ with shaded uncertainty are shown below corresponding spatially and temporally with the high-speed images.

First, spreading dynamics will be discussed, including the impact of thermal effects that inhibit thin film spreading. Figure 4.6 shows three high-speed images in time as an impinging jet ($Re_D = 12,000$) contacts a heated ($T_0 = 320 \, ^\circ C$) case 5 surface and accelerates radially outward into a thin liquid film. In all images the target region is the dark, reflective circle at the jet stagnation region. The first image is from just before impingement. Subsequent images show how the thin film spreads across the surface at later times, visible as the darker gray region near the center of the wafer and bounded by the lighter spray of droplets being ejected. Corresponding thermal results are shown beneath the high-speed images, including measured $T(r,t)$ and calculated $q''(r,t)$ values with associated uncertainty shaded. It can be seen by the black, vertical dashed lines showing the outside edge of the thin film region that it corresponds to the location of maximum heat flux for later times. This point of calculated maximum local heat flux was used to define the thin film spread through time.
Fig. 4.7: Thin film radius as a function of time for all hole surfaces and posts of corresponding cavity fraction (cases 1 - 4) for all $T_0$ (with $Re_D = 12,000$) and $Re_D$ (with $T_0 = 280 \, ^\circ C$) values tested.

The time for the thin film to spread to $n$ radii ($t_n$) is shown in Fig. 4.7 for cases 1 - 4 (hole and post surfaces of equal cavity fraction). In the case of the lowest $Re_D$ (6,000), the thin film radius never reaches 12 jet radii downstream due to low relative momentum. As $T_0$ increases, $t_n$ increases on all surfaces and for all $n$. At elevated temperature, the Leidenfrost condition prevails for longer before the surface can cool sufficiently (via film boiling) to allow liquid-surface contact. Increasing $T_0$ from 200 to 320 $^\circ C$ increases the average $t_{12}$ by 78 - 135%, depending on the SH surface case.

Increasing jet $Re_D$ generally results in a slight decrease in $t_n$ for each case due to increased relative momentum in the initial flow. The change in $t_n$ is nominally negligible for $n = 4$ and 8. However, the reduction in $t_n$ to $n = 12$ jet radii downstream can be as high as a 33% decrease for cases 1 and 4 when $Re_D$ is increased from 12,000 to 18,000.

Values of $t_n$ for all cases show similar trends for $T_0$ and $Re_D$, in addition to being nearly equal. Thus, for these microstructure designs, altering cavity fraction from 50 to 75% has a negli-
gible effect on $t_n$. Similarly, altering microstructure design from holes to posts also has a negligible effect on $t_n$.

At least partial wetting of the microstructures on all surfaces was observed during testing. As the unpatterned target center prevented wetting due to high jet stagnation pressure (Liu and Lienhard showed that local pressure recovers the atmospheric condition at approximately $n = 1.6$ jet radii downstream of the stagnation point [63]), this behavior was most likely due to rapid vaporization of water at the thin film front. The vapor could then flow into the microstructures. A reduction in surface tension at high temperatures, combined with subcooled water passing above the gaps in the microstructure, may cause the trapped vapor to condense and remain pinned within these gaps. Post structures allow vapor flow between microstructure features, which may prevent some wetting and thus lead to slightly slower spreading compared to closed hole-patterned surfaces. This behavior could explain the slight deviation from typical behavior shown for case 1 at $T_0 = 320 \, ^\circ\text{C} \, (\text{Re}_D = 12,000)$ and at $\text{Re}_D = 6,000 \, (T_0 = 280 \, ^\circ\text{C})$, where $t_{12}$ is shorter than for the other cases. Case 1 has the lowest cavity fraction and thus the highest liquid-solid contact area, which could enable slightly faster spreading for some experimental conditions. However, while $t_n$ decreases on the case 1 surface for high $T_0$ and low $\text{Re}_D$, this is not the case for the post structure of same cavity fraction (case 3), which may be due to the open structure allowing vapor escape and leading to slower spreading. Jet impingement experiments were performed on room temperature surfaces in the current study, and showed no hydrodynamic wetting of microstructures due to high pressure (prevented by the unpatterned target at the jet center, where pressure was the highest). It is thus likely that some effect of vapor generation near the thin film front with limited escape leads to intermittent surface wetting, impacting thin film spreading.

Values of $t_n$ for cases 5 - 8 are shown in Fig. 4.8. All of these post-patterned SH surfaces have the same cavity fraction of 85% where cases 5 - 7 have identical pitch and diameter, but decrease in post height from 25 to 5 $\mu\text{m}$ for each subsequent case in 10 $\mu\text{m}$ increments. Case 8 has smaller diameter posts and a smaller pitch than the others, and the same height as case 6. Overall trends here with regard to $T_0$ and $\text{Re}_D$ are similar to the previous comparison between posts and holes; $t_n$ increases with decreasing $T_0$ and increasing $\text{Re}_D$.

Another trend is seen here as well with the thin film front spreading more rapidly as post height decreases (moving from case 5 to case 7), which is evident for $n = 4$ and 8 in Fig. 4.8. For
Fig. 4.8: Thin film radius as a function of time for cases 5 - 8 (round post surfaces of equal cavity fraction) for all $T_0$ and $Re_D$ investigated (same parameters as Fig. 4.7). This shows comparisons between post diameter and pitch as well as height.

$n = 12$ the trend is not as clear. Since at these later times the thin film is nearing its maximum spread radius. At $Re_D = 18,000$ the thin film reaches $n = 12$ jet radii on the surface with the tallest posts (case 5) 39% later than the shortest post structure (case 7), which has an 80% reduction in post height comparatively.

This trend of faster spreading with decreasing post height may be attributed to enhanced wetting. The shorter posts have less space in the microstructure gaps for water vapor to escape, and it may condense and interrupt film boiling, allowing for faster spreading. A similar effect was observed in droplet impingement on SH surfaces where thermal atomization was suppressed on SH surfaces with taller post microstructures [18].

Comparing cases 6 and 8 demonstrates the effect of post pitch and diameter, as cavity fraction and post height are held constant. In general, trends of faster spreading with increased pitch and diameter are minimal in comparison to post height. Further exploration is required in order to ascertain if there is an effect of microstructure spacing.
Fig. 4.9: Total thermal energy transferred from the surface to the water throughout time, normalized by the initial potential difference ($E_0$) for (a) cases 1 - 4 and (b) cases 5 - 8. Results are shown for Re$_D$ = 18,000 and $T_0$ = 280 °C. Previously-obtained values for smooth HPi and HPo surfaces are also included (see Chapter 3).

Thin film spreading directly correlates with heat transfer in jet impingement quenching as the thin film is the only region where liquid-surface contact occurs and heat transfer is significant. This can be seen by exploring thermal energy transfer through time, $E(t)$, which is calculated using Eq. (4.2), and normalized by the initial difference in local $T_0$ and jet temperature as explained in Section 4.4.3. $\dot{E}(t)$ for cases 1 - 4 is shown in Fig. 4.9a and for cases 5 - 8 in Fig. 4.9b. The conditions for these plots are $T_0$ = 280 °C and Re$_D$ = 18,000. The curves all show rapid initial thermal energy transfer, indicated by the steep gradient during the first 0.2 s of impingement, at which point the thin film reaches the edge of the viewing window (nominally 0 ≤ $r$ ≤ 20 mm). As the dominant mechanism of heat transfer transitions from latent to sensible heat transfer, thermal transport slows considerably and the slope of $\dot{E}(t)$ levels off.

Figure 4.9 shows that all SH surfaces transfer heat significantly more slowly than either of the smooth cases (data from Chapter 3). However, there is some spread in the rate of energy transfer
between SH cases examined in this study, which can be quantified by estimating the temporal derivative of $E(t)$ using a linear curve fit to the initial moments (from $t = 0$ to nominally 0.15 - 0.2 s) for each experiment. This provides an initial average heat transfer rate, $\bar{q}_0$, useful for comparing thermal transport for the investigated parameters.

Values of $\bar{q}_0$ for all surfaces are shown in Fig. 4.10a as a function of $T_0$, and in Fig. 4.10c as a function of $Re_D$. These values were normalized by $\bar{q}_0$ for smooth HPi surfaces from Chapter 3, which are given in Table 4.2. The normalized $\bar{q}_0$ values are shown in Fig. 4.10b and 4.10d as functions of $T_0$ and $Re_D$, respectively. Figures 4.10b and 4.10d also include the normalized $\bar{q}_0$ values for a smooth HPo surface for comparison. All clusters of data were acquired at the standard surface temperatures and jet $Re_D$, but are staggered in the plot for clarity. As can be seen in Fig. 4.10a, the data is nominally $\bar{q}_0 = 800$ W regardless of $T_0$ and surface type. The lack of dependence on $T_0$ may be explained by a balance of mechanisms. The total amount of energy available to be transferred (based on thermal energy initially stored in the wafer) increases with $T_0$. However, higher $T_0$ also limits the liquid water’s capacity to contact the surface due to the Leidenfrost effect (see Figs. 4.7 and 4.8). These competing effects of increased initial energy and slowed thin film spreading mitigate the impact of $T_0$ on $\bar{q}_0$ for the cases observed here.

Table 4.2: Values of $\bar{q}_0$ for HPi surfaces at different $T_0$ and $Re_D$ from Chapter 3, used in normalization.

<table>
<thead>
<tr>
<th>$Re_D$</th>
<th>6</th>
<th>12</th>
<th>18</th>
<th>12</th>
<th>12</th>
</tr>
</thead>
<tbody>
<tr>
<td>$T_0$ [°C]</td>
<td>280</td>
<td>280</td>
<td>280</td>
<td>200</td>
<td>320</td>
</tr>
<tr>
<td>$\bar{q}_0$ [W]</td>
<td>749.1</td>
<td>1075.2</td>
<td>1807.7</td>
<td>3351.3</td>
<td>1243.1</td>
</tr>
</tbody>
</table>

When normalized, however, there is a significant dependence on $T_0$, as shown in the plot in Fig. 4.10b. This is due to the dynamics occurring on the smooth HPi surface, which are different for low $T_0$ as the Leidenfrost point has not been reached and thus film boiling is suppressed. At $T_0 = 200$ °C, for example, values of $\bar{q}_0$ measured on the SH surfaces are 20 - 30% of those for the HPi surface, while at higher $T_0$ the heat transfer performance on all SH surfaces is in the range of 50 - 80% of the HPi value.
Fig. 4.10: Initial average heat rate for all cases, given as a function of $T_0$ in (a) and (b) and as a function of $Re_D$ in (c) and (d). Plots in (b) and (d) are normalized by the values for smooth HPi surfaces for the same experimental conditions, as obtained in Chapter 3, and include the ratios for smooth HPo surfaces from the same paper. Hollow and solid markers represent hole and post surfaces, respectively. Downward, side-facing, and upward triangles represent shortest, medium-sized, and tallest post surfaces, respectively.
Figure 4.10c indicates a strong dependence of $\bar{q}_0$ on jet $Re_D$ for all SH surfaces. As $Re_D$ is increased from 6,000 to 18,000, $\bar{q}_0$ increases between 153% and 326% depending on SH surface case. This is the same trend as for the smooth HPi surface (see Chapter 3) and shows that irrespective of the level of surface hydrophobicity, as $Re_D$ increases the enhanced relative momentum causes faster initial spreading, leading to higher heat transfer.

Values of $\bar{q}_0$ normalized for each surface by the corresponding smooth HPi case are compared against $Re_D$ in Fig. 4.10d. SH surface values range from 30 - 60% of the corresponding HPi values for low $Re_D$ and up to nominally 80% for moderate $Re_D$. For completeness, it is important to note the departure of the smooth HPo case from near SH surface behavior as $Re_D$ increases. At high $Re_D$, $\bar{q}_0$ on the smooth HPo surface approaches that of the HPi case, while the values for the SH cases fall near 50 - 70% of the smooth HPi surface. The normalized plot also indicates that dependency on $Re_D$ is different for all surface types. While Fig. 4.10c confirmed that there is a direct correlation of $\bar{q}_0$ with $Re_D$, Fig. 4.10d shows that this dependency is weaker for SH surfaces compared to the relationship between $\bar{q}_0$ and $Re_D$ for the smooth HPo and HPi cases.

Trends in relation to surface design are difficult to extract using this spatially-averaged measurement. Generally speaking, the difference between hole and post surfaces (open and solid markers) is minimal. The largest differences between surfaces seem to be for high $T_0$ or high $Re_D$, but again there is no consistent trend. The only case-consistent trend is associated with case 7, the case of the shortest posts, which typically has slightly higher $\bar{q}_0$ than most other surfaces, likely due to intermittent wetting.

While $\bar{q}_0$ comparisons showed average behavior across the surfaces, heat flux was also used to quantify heat transfer locally between cases and experimental parameters, as was shown in Fig. 4.6. The maximum heat flux ($q''_{max}$) in each case is at the stagnation point during initial jet contact (see the bottom left plot of Fig. 4.6, at $t = 0$ s). There were no distinguishable trends in $q''_{max}$ for cases 1 - 4, just as there were none seen for spreading time or heat transfer rate, so values for those cases are not included here. Readers interested in these values are directed to the Table 4.3 of the Appendix. Cases 5 - 8, however, are included here and shown as a function of $T_0$ in Fig. 4.11a and as a function of $Re_D$ in 4.11b, similar to the previous plots of $\bar{q}_0$ in Figs. 4.10a and 4.10c.

A trend of increasing $q''_{max}$ with increasing $T_0$ is clear in Fig. 4.11a. This direct relationship between $q''_{max}$ and $T_0$ is due to the increased temperature difference between the surface and the
water that drives thermal transport. Since $q''_{max}$ is a local measurement and not an average value, as is $\bar{q}_0$, the effect of thin film spreading is not as influential. There is also an increase in the spread of the data as $T_0$ increases, showing that the effects of microstructure (which will be discussed below) impact $q''_{max}$ more at higher $T_0$.

It can be seen in Fig. 4.11b that the value of $q''_{max}$ increases nearly linearly with Re$_D$. Increased Re$_D$ here implies higher stagnation pressure, which leads to increased heat transfer at the stagnation point. This agrees with behavior observed for surfaces that are not SH as well [46].

In regards to SH surface design, there seems to be a significant effect of post height on $q''_{max}$. As post height decreases, $q''_{max}$ increases. Case 7 (shortest posts) has between 2.2 - 91.4% higher heat flux than case 5 (tallest posts), depending on the experimental parameters. There are clear microstructure effects that impact heat transfer even at the time of initial jet contact on the smooth target region of the SH surfaces despite no direct contact with the microfeatures. Examining other parameters, case 6 has 6.8 - 39.9% higher $q''_{max}$ than case 8, which has the same height but smaller microstructure diameter and pitch. This trend is much smaller than for the differences due to post height, and would thus require further exploration to confirm any dependency.
Fig. 4.12: Wetting behavior on all surfaces during impingement with jet $Re_D = 18,000$ and $T_0 = 280 \, ^\circ C$. Images (a) - (d) corresponding to cases 1 - 4, and (e) - (h) to cases 5 - 8, respectively. The darkened regions show the areas of wetting, which in some cases is nearly at the thin film front.

Another interesting observation from this study is in regards to apparent surface wetting, which changes slightly based on surface type. Figure 4.12 shows all surfaces at $t = 0.1 \, s$ under an
impinging jet of $\text{Re}_D = 18,000$ and $T_0 = 280 \, ^\circ\text{C}$. Surface wetting is distinguishable in particular in image (f), where the dark region around the target is distinct from the dry pattern around the outer edge of the image. It is also clear from image (f) that the wetted region is broken up by jagged edges where the microstructure has not yet wetted, before the thermal breakup into droplets. Typically, it is the transition in darkness that indicates where a surface is wetted, where nucleate boiling is occurring, and where the thin film breaks up. The effect of this wetting pattern directly on heat transfer is unclear from the current data. However, it is interesting to note behaviors similar to what has been shown for micropatterned superhydrophilic surfaces, as was observed by Dressaire et al., particularly for image (d) [64]. This case is a square post patterned surface and results in more of a square thin film spreading across the surface, which show that the post shape leads to preferential flow direction.

Often, excessive water pressure or surface defects can lead to wetting in microstructures. For the surfaces explored here, the target at the center prevents hydrodynamic wetting. This wetting behavior is only observed at high temperature, showing that a likely cause is due to rapid phase change near the thin film front. The water vaporizes due to the excessive local surface temperature and can then penetrate the microstructure and flow radially outward. Then due to the rapid jet spreading, liquid water can pass over the vapor and cool it sufficiently for condensation, resulting in a wetted condition known as the Wenzel state. No permanent effects on surface condition were observed once the surfaces were heated such that the water in the microstructure was able to evaporate and leave the surface.

4.6 Conclusions

In this work, SH surfaces comprised of post or hole microstructures with varying height, pitch, width or diameter, and cavity fraction were used in a jet impingement heat transfer study. The water jet, at room temperature, varied in $\text{Re}_D$ from 6,000 - 12,000. Initial surface temperature, $T_0$, varied from 200 - 320 °C. Both hydrodynamic observations and computed heat transfer values were examined.

Surfaces were first compared against each other to determined which allowed for fastest liquid contact and spreading across the surface, which increases thermal transport by allowing more rapid convection and high latent heat transfer in the form of boiling. Decreasing $T_0$ and
increasing jet $\text{Re}_D$ resulted in faster thin film spreading for all surfaces. For the surfaces examined here, holes and posts of equal cavity fraction did not have an effect on thin film spreading for low cavity fractions (50 - 75%). However, decreasing post height decreased thin film spreading time, specifically by up to 39% for the thin film to reach 12 jet radii downstream when the height was reduced from 25 to 5 $\mu$m.

Initial average heat transfer rate from the surface to the water showed a strong increase with increasing jet $\text{Re}_D$, but no quantifiable dependency on SH surface design or $T_0$. Using data from past research confirmed that all SH surfaces tested here had lower values of initial average heat rate compared smooth HPi surfaces, regardless of other parameters, by 20 - 80%.

The third method of comparing calculated heat transfer was using the maximum heat flux. Maximum heat flux increases with increasing $T_0$ and increasing jet $\text{Re}_D$ on all SH surfaces examined here. A trend of higher maximum heat flux of up to 91% with decreasing post height by 80% (from 25 to 5 $\mu$m) was observed, holding other parameters and geometry constant. The same was true of decreased maximum heat flux with decreased post diameter and pitch by up to 40%. Again, overall comparison with previous work shows that all SH surfaces tested here promote slower thermal transport than do smooth HPi and HPo surfaces.

4.7 Appendix

Maximum local heat flux data for cases 1-4 is included in Table 4.3 below for reference.
Table 4.3: Maximum heat flux (in MW/m²) for cases 1-4 based on experimental conditions.

<table>
<thead>
<tr>
<th>Case Number</th>
<th>1</th>
<th>2</th>
<th>3</th>
<th>4</th>
</tr>
</thead>
<tbody>
<tr>
<td>( \text{Re}_D = 6,000, \ T_0 = 280 \ ^\circ\text{C} )</td>
<td>6.2</td>
<td>7.8</td>
<td>8.6</td>
<td>6.7</td>
</tr>
<tr>
<td>( \text{Re}_D = 12,000, \ T_0 = 280 \ ^\circ\text{C} )</td>
<td>8.5</td>
<td>10.4</td>
<td>10.8</td>
<td>8.4</td>
</tr>
<tr>
<td>( \text{Re}_D = 18,000, \ T_0 = 280 \ ^\circ\text{C} )</td>
<td>9.1</td>
<td>10.2</td>
<td>10.0</td>
<td>9.2</td>
</tr>
<tr>
<td>( \text{Re}_D = 12,000, \ T_0 = 200 \ ^\circ\text{C} )</td>
<td>3.8</td>
<td>5.6</td>
<td>7.1</td>
<td>5.3</td>
</tr>
<tr>
<td>( \text{Re}_D = 12,000, \ T_0 = 320 \ ^\circ\text{C} )</td>
<td>8.1</td>
<td>10.6</td>
<td>12.1</td>
<td>7.2</td>
</tr>
</tbody>
</table>
CHAPTER 5. CONCLUSIONS

Jet impingement has been shown to be an effective method of rapidly removing heat from objects at elevated temperatures. Surfaces that are SH have shown promise in potential self-cleaning and drag reduction applications, but can also impede heat transfer. This thesis includes the methods and results of experiments performed on initially superheated surfaces quenched by an impinging water jet. Parameters explored here include initial temperature, jet Re_D, and surface type. Specifically, HPi, HPo, and SH surfaces with varying microstructure geometry were tested with the resulting maximum local heat flux, average heat rate, total thermal energy transfer and thin film spread rate compared among the situations to define differences. A summary of important conclusions from this work are included in the sections that follow, detailing the impact of these experiments. A section on potential future extensions for this research is also included.

5.1 General Effects of Hydrophobicity on High-Temperature Jet Impingement Heat Transfer

Chapter 3 of this thesis focused on the differences between smooth HPi, smooth HPo, and micropatterned SH surfaces of varying initial temperature (T_0 = 200, 280, or 320 °C) being quenched by impinging jet of variable Reynolds number (Re_D = 6,000, 12,000, or 18,000). In general, increasing T_0 while holding Re_D increases averaged maximum local heat flux, q'^{\prime\prime}_{max}, across all surfaces. This is due to a higher driving temperature difference increasing the rate at which thermal energy is transferred from the surface to the water on a per unit area basis. This effect was opposed, however, by the inability for liquid water to contact the surface as temperature increases, increasing thin film spreading across the surface. These competing effects influence the average heat rate, \bar{q}, which showed no clear T_0 dependence.

Increasing Re_D while holding T_0 constant increases heat transfer on all surfaces, regardless of the level of hydrophobicity, shown by all metrics compared. Increasing the momentum of the
flow overcomes some of the thermal effects of high temperature that impede thin film spreading of impeding jets, resulting in faster thin film spreading. This in turn also increases $\bar{q}$ and $q_{max}'$ as both are measures of thermal energy transferred on a per unit time basis.

By all measures, SH surfaces transfer heat at the lowest rates. At low $Re_D$, HPo surfaces had similar thermal performance to SH surfaces, while for higher $Re_D$ (12,000 and 18,000) they more closely matched the heat transfer characteristics of HPi surfaces. In regards to thin film spreading, when measuring the spreading time to smaller radial locations the HPi and HPo surfaces were typically very similar, with increased divergence with increased distance from the stagnation point. HPi surfaces always spread the quickest and typically had the highest heat transfer over the range of $T_0$ and $Re_D$ examined in this work.

As an application example, according to the results of this work, in order to match roughly the same thermal transport performance when switching from a HPi or HPo surface to a SH one, jet flow rate would need to be increased by around 50% (increasing $Re_D$ from 12,000 to 18,000).

5.2 Effect of Varying Microstructure on High-Temperature Jet Impingement Heat Transfer

Knowing that SH surfaces generally have lower thermal transport properties than smooth HPi and HPo surfaces brings up the question, do all SH surfaces demonstrate the same trend? Chapter 4 of this thesis examined similar experiments for transient jet impingement heat transfer scenarios on SH surfaces with varying microstructure. Feature height ($h = 5, 15, \text{ or } 25 \mu m$), pitch ($w = 8 \text{ or } 16 \mu m$), cavity fraction (50, 75, or 85%), and shape (round or square) were all varied.

General trends with initial surface temperature were similar to those shown for all surface types in Chapter 3. Increasing surface temperature tended to decrease spreading time, increase maximum heat flux values, and have little effect on average heat rate, for similar reasons as have been previously described.

Increasing jet $Re_D$ also impacted all SH surfaces tested. Doing so decreased spread time and increased both maximum heat flux as well as average heat rate. This behavior was expected based on the reported results of previous work that showed similar dependence on jet $Re_D$. Increased $Re_D$ in this study again meant higher momentum, which increased both the stagnation pressure as well as the ability for the thin film to spread across the surface.
The novel conclusions for this work are in regards to surface microstructure geometry. For the holes and posts surfaces tested at relatively low (50 - 75%) cavity fractions, no significant differences were observed with spreading time. Average heat rate and maximum heat flux also showed no clear trends as to higher values for varying cavity fraction or feature pattern (holes or posts).

When comparing surfaces with round posts of equal cavity fraction (85%) but varying height, it was found that decreasing post height tended to decrease thin film spread time. It also tended to increase maximum heat flux values (up to a 90% increase between the 5 and 25 µm surfaces), but an effect on average heat rate was difficult to determine. It was hypothesized that an increase in post height allowed for additional vapor to escape rather than condense within the microstructure and partially wet the surface.

In summary, initial surface temperature, jet Re₅, and surface type do affect jet impingement quenching and heat transfer. To a certain extent, altering SH surface microstructure can also affect jet impingement behavior.

5.3 Suggestions for Further Work

The potential for research within the field of jet impingement heat transfer is extensive. Specifically, there are several investigations regarding surfaces with apparent slip (see 1.1.1) that can be explored. Several parameters have been adjusted in previous work that could be examined with SH surfaces, including varying Re₅ by altering nozzle diameter, increasing the jet temperature, or examining the effects of angled impingement, which could be useful in space-constrained applications. Impingement in other directions, such as upward or horizontal, and arrays of jets impinging on SH surfaces would also be interesting to investigate for other specific applications.

Another simple extension is to utilize a surface material or coating that can withstand higher temperatures than were available using PTFE in the present study. Higher initial temperatures would lead to increased delays in wetting and more obvious film boiling, which has not been examined up to this point. The present study was also limited by the means of heat addition. The integrated heaters provided a convenient, rapid way to bring surfaces to superheated initial temperatures, but supplied insufficient heat to conduct steady-state experiments that could maintain
the surface at a constant surface temperature (see Appendix B). This could demonstrate further differences between SH, HPo, and HPi surfaces.

Nanostructured and hierarchical (micro- and nanostructured) SH surfaces have been shown to promote dropwise condensation and avoid wetting due to pressure. The SH surfaces in this study were influenced by rapid vaporization and subsequent condensation within the microstructures, causing surfaces to enter the Wenzel state. Use of nanostructures could be investigated to potentially avoid this wetting and show what effects a Cassie-Baxter state would have on jet impingement heat transfer. Varying surface geometry further, such as with higher cavity fractions, would also likely provide interesting results. Ultimately, it would be most beneficial to perform similar jet impingement work on superomniphobic surfaces, which repel all liquids and would thus have more universal applicability. This type of extension would apply to utilizing dielectric fluids in electronics thermal management, or for machinery cooled by oil and not limited to situations where water is the working fluid.
REFERENCES


APPENDIX A. CODE FOR HEAT TRANSFER COMPUTATION

Included here are parts of the MATLAB script that deal with calculating components that are not as straightforward as plugging values into the described equations. First, Section A.1 describes the smoothing process involved in obtaining the radial and temporal temperature gradients, \( \partial T / \partial r \), \( \partial T / \partial t \), and \( \partial (k_s \mu r(\partial T/\partial r)) / \partial r \). A section of code is then included in Section A.2 for taking the derivative of energy transfer in order to get an average heat transfer rate.

A.1 Temperature Derivatives

This section takes the radial position (\texttt{mmRadialPosition}) and time value (\texttt{time}) of each data point, fitting the averaged temperature data (\texttt{meanProfile2}) to a 2nd-order polynomial to obtain the temporal temperature gradient (\texttt{dTdt}). The process is similar for radial derivatives, where first \( \partial T / \partial r \) (\texttt{dTdr}) is calculated as indicated. Thermal conductivity (\texttt{ks}) is calculated as a function of local temperature, followed by computation of the second derivative, \( \partial (k_s \mu r(\partial T / \partial r)) / \partial r \) (\texttt{dkrdTdrdr}). All of these values are used to calculate \( q''(r,t) \) in Eq. (2.1).

```matlab
%% Calculate dT/dt
tempT = zeros(7, length(mmRadialPosition));
dTdt = zeros(length(time)-3, length(mmRadialPosition));
for j = 1:length(mmRadialPosition)
    for i = 1:3
        tempt = time(1:7);
        tempT(:,j) = meanProfile2(1:7,j);
        p = polyfit(tempt, tempT(:,j), 2);
        dp = polyder(p);
        dTdt(i,j) = polyval(dp, time(i));
    end
```

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for i = 4:length(time)-3
    tempt = time(i-3:i+3);
    tempT(:,j) = meanProfile2(i-3:i+3,j);
    p = polyfit(tempt,tempT(:,j),2);
    dp = polyder(p);
    dTdt(i,j) = polyval(dp,time(i));
end
for i = length(time)-2:length(time)
    tempt = time(end-6:end);
    tempT(:,j) = meanProfile2(end-6:end,j);
    p = polyfit(tempt,tempT(:,j),2);
    dp = polyder(p);
    dTdt(i,j) = polyval(dp,time(i));
end
end

%% Calculate dT/dr, d(k*r*(dT/dr))/dr
mRadialPosition = mmRadialPosition./1000;
n = floor(length(meanProfile2(1,:))/5.5);
smoothcount = 15;
Temp = zeros(length(meanProfile2(:,1)),length(meanProfile2
(1,:)));
dTdr = zeros(length(meanProfile2(:,1)),length(meanProfile2
(1,:))-1);
tempT = zeros(length(meanProfile2(1,:)),smoothcount);
for i = 1:length(meanProfile2(:,1)) % step through time
    Temp(i,:) = meanProfile2(i,:)
    for j = 1:(smoothcount-1)/2 % first pts are forward difference
        temprad = mRadialPosition(1:smoothcount);
tempT(i,:) = Temp(i,1:smoothcount);
p = polyfit(temprad,tempT(i,:),2);
dp = polyder(p);
dTdr(i,j) = polyval(dp,mRadialPosition(j));
end
for j = (smoothcount+1)/2:length(mRadialPosition)-(smoothcount-1)/2 % other pts are central difference
    temprad = mRadialPosition(j-(smoothcount-1)/2:j+(smoothcount-1)/2);
    tempT(i,:) = Temp(i,j-(smoothcount-1)/2:j+(smoothcount-1)/2);
    p = polyfit(temprad,tempT(i,:),2);
    dp = polyder(p);
    dTdr(i,j) = polyval(dp,mRadialPosition(j));
end
for j = length(mRadialPosition)-(smoothcount-1)/2+1:length(mRadialPosition) % last pts are backward difference
    temprad = mRadialPosition(end-smoothcount+1:end);
    tempT(i,:) = Temp(i,end-smoothcount+1:end);
    p = polyfit(temprad,tempT(i,:),2);
    dp = polyder(p);
    dTdr(i,j) = polyval(dp,mRadialPosition(j));
end
end

krdTdr = dTdt;
dkrdTdrdr = krdTdr;
for i = 1:length(dTdt(:,1))
    for j = 1:length(dTdt(1,:))
\[ ks = 8.835 \times 10^{-4} \left( \text{meanProfile2}(i,j) + 273 \right)^2 - 1.0719 \left( \text{meanProfile2}(i,j) + 273 \right) + 388.0812; \% \text{based off curve fit} \]
\[ \text{krdTdr}(i,j) = ks \times \text{mRadialPosition}(j) \times \text{dTdr}(i,j); \]

end

end

for \( i = 1: \text{length(krdTdr(:,1))} \) \% step through time
\[ \text{Temp}(i,:) = \text{krdTdr}(i,:); \]

for \( j = 1:(\text{smoothcount}-1)/2 \) \% first 3 pts are forward difference
\[ \text{temprad} = \text{mRadialPosition}(1: \text{smoothcount}); \]
\[ \text{tempT}(i,:) = \text{Temp}(i,1: \text{smoothcount}); \]
\[ p = \text{polyfit}(\text{temprad}, \text{tempT}(i,:),2); \]
\[ dp = \text{polyder}(p); \]
\[ d\text{krdTdrdr}(i,j) = \text{polyval}(dp, \text{mRadialPosition}(j)); \]
end

for \( j = (\text{smoothcount}+1)/2: \text{length(mRadialPosition)}-(\text{smoothcount}-1)/2 \) \% other pts are central difference
\[ \text{temprad} = \text{mRadialPosition}(j-(\text{smoothcount}-1)/2:j+(\text{smoothcount}-1)/2); \]
\[ \text{tempT}(i,:) = \text{Temp}(i,j-(\text{smoothcount}-1)/2:j+(\text{smoothcount}-1)/2); \]
\[ p = \text{polyfit}(\text{temprad}, \text{tempT}(i,:),2); \]
\[ dp = \text{polyder}(p); \]
\[ d\text{krdTdrdr}(i,j) = \text{polyval}(dp, \text{mRadialPosition}(j)); \]
end

for \( j = \text{length(mRadialPosition)}-(\text{smoothcount}-1)/2+1: \text{length(mRadialPosition)} \) \% last pts are backward difference

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temprad = mRadialPosition(end-smoothcount+1:end);
tempT(i,:) = Temp(i,end-smoothcount+1:end);
p = polyfit(temprad,tempT(i,:),2);
\[ dp = \text{polyder}(p); \]
dkrdTdrdr(i,j) = polyval(dp,mRadialPosition(j));
end
end

A.2 Energy Transfer Derivative

This section of code takes the calculated energy transferred data (E temp1), where a user input selects the first linear portion of data up to the knee (see Section 3.5.2). The data within this range is curve fit to a line, and the resulting initial average heat rate (Q) is the slope of that line.

\[
\text{plot(time(1:length(E temp1)),E temp1);}
\]
\[
\text{title('Select beginning and end of region of interest:')}\]
\[
[xE,yE] = \text{ginput(2);}
\]
\[
\text{range = find(time>xE(1) \& time<yE(2));}
\]
\[
\text{trange = time(range(1):range(end));}
\]
\[
\text{E range = E temp1(range(1):range(end));}
\]
\[
\text{q1 = polyfit(trange,E range,1);}
\]
\[
Q = q1(1)
\]
APPENDIX B. LESSONS LEARNED

B.1 Surface Production Tips

General details for several of the processes of surface fabrication can be found in Appendix B of Matthew Searle’s dissertation [29]. Using those described processes worked well in general, with a few modifications that were found to be helpful for producing good results, detailed as follows.

B.1.1 SH Surface Microfabrication

A few comments regarding fabricating surfaces in the Brigham Young University clean-room are provided here. In general, the STS etching machine is not consistent in its etch rate and must be carefully monitored. This may be partially due to the initial passes removing softened photoresist, but a careful balance is required to etch a sufficient amount with each pass, but not so much to overshoot on the desired depth.

In Chapter 2 surfaces with multiple cavity fractions were created. One observation worth noting is regarding the challenge in fabricating surfaces with very high cavity fraction (nominally 96%). The particular post-patterned surface that was desired had a pitch of 16 µm and a diameter of 3.5 µm, with a designed height of 15 µm. However, for some reason none of the 4 silicon wafers that were patterned with this design would etch properly. Instead of near-cylinders, the surface would result in tiny, fragile cones that were unusable. Again, the cause of this is uncertain, as another surface had the same diameter post structures but did not result in this behavior.

B.1.2 Creating Integrated Heaters

In addition to the steps provided in Matthew Searle’s dissertation Appendix B.10.2, additional insights provided by Cole Thatcher, who performed most of this work, include the following.
First, the use of Ferro® (formerly ElectroScience Laboratories) 9913 was discontinued in favor of 599-E Ferro, which cures as much lower and safer temperatures (from 850 to 450 °C), with comparable heater performance. Special care was taken to avoid exposure to air, as the material would degrade and become defective rapidly with increased age and exposure to air. A space heater was often used in the fume hood where heaters were fabricated, especially in cold conditions. This helped the heater paste be more malleable and improved surface adhesion.

Using a smaller amount of this material on the squeegee in screen-printing (0.5 x 0.5 cm area rather than 1 x 1 cm) also gave better results. It was noted that the plastic squeegee should bend significantly in order to apply sufficient pressure to allow well-defined heater leads. Performing only one pass with the squeegee to apply the paste and only one pass to remove excess provided the best definition, using only one side of the squeegee (the other side should be clean). This was also done parallel and not perpendicularly to heater lines. During the heater drying and firing process, it was best to clean the heater area using compressed air to remove dust and particles that would burn at the high temperatures.

If multiple heaters were fabricated at one time, the side of the silk screen that contacted the wafer was cleaned. This was done by wiping the head of several Q-tips along the heater lines using a rotating motion so as not to accumulate and smear excess paste in other places along the screen surface.

### B.1.3 Attaching Wire Leads to Heaters

Additional pointers to what is included in Appendix B.11 of Matthew Searle’s dissertation are given here. Often it was useful using the apparatus in this research to remove 4 cm of insulation from the lead wires. There was little risk of exposure due to the cylinder exposure, and the high heater temperatures tended to melt and burn the wire insulation used for this study. Bending the wires further from the tip (2 cm rather than 0.6 cm) allowed for higher contact area between wires and the heater bus bars. Covering all the heater wires except the bus bars with tape prevented undesired application of epoxy between heater lines (remove the tape before curing).

Epoxy application required laying a thin layer along the bus bar the length of the bent portion of the wire, as well as pre-coating the wires using a small spatula. Good connection was
ensured by using the wooden end of a Q-tip to push the past around the wire so as to not leave any air gaps.

B.1.4 Spray-Painting Surfaces

It was noted that applying too much of the Rust-Oleum® primer and paint had an adverse affect on the heat transfer measurements. Accordingly, paint application was performed using coats that were applied swiftly and lightly, holding the cans 20 cm from the surface with 2 quick passes per coat. One coat seemed to work fine with both surface adhesion and with obtaining accurate results. The quickest check is to note if the heater leads are visible to both the naked eye as well as in the IR camera measurements after applying the paint coating.

B.2 Best Practices for Data Acquisition

There were a few important discoveries over the course of over 1,000 collected data sets that were found to be useful in acquiring the best data, as are included in the following sections.

B.2.1 Thermal Camera Settings

The thermal camera used in this study (described in Chapter 2) was factory calibrated over different, limited temperature ranges. In order to view data over the full range of temperatures in this study (nominally 20 to 320 °C), two such calibrations were necessary. This limited the frame rate at which the camera could be used at the highest spatial resolution. The spatial resolution was altered such that the entire desired viewing area was visible at both the temporal and spatial resolutions previously specified (see Chapter 2). The camera acquired the two calibrations as sequential image frames and the data was aggregated in the MATLAB code (see Appendix A).

B.2.2 Jet Initialization

For the current work, the flow rate was pre-set by opening the rotameter fully and adjusting the pressure in the tank until the desired flow rate was attained, as measured by the rotameter, and then the rotameter would be closed. To begin a test, the rotameter was again opened fully to
allow water to flow at the pre-defined flow rate. Occasionally, a small “pre-jet” would form upon initiating the jet, which would impact the surface before the jet itself and significantly alter the heat transfer results. In order to avoid this, it was important to open the rotameter just enough that a small meniscus would be visible at the nozzle exit, which would help prevent these initial pre-jets and mitigate this undesirable effect on heat transfer.

B.3 Steady-State Investigations

Originally, it was preferable to conduct steady-state experiments in order to reduce the complications and uncertainty involved in a transient study. However, multiple concerns described in the succeeding paragraphs prevented such work from being performed at the present.

B.3.1 Heater Block Technique

Several studies utilize a conductive metal block in order to track jet impingement heat transfer [34, 43, 45, 65] Often some heating element is used and an array of thermocouples was employed using an inverse heat conduction calculation to obtain the surface heat flux. Initially this was the method employed here, using an aluminum block with cartridge heaters and thermocouples as shown in Figure B.1.

The main complication that was encountered was the high contact resistance between a wafer and the heating block, which resulted in irregular cooling across the surface of the block. Both increasing contact pressure and using thermally conductive paste did not solve the issue adequately. In addition, the jet removed far more heat than was able to be supplied by the cartridge heaters for steady-state experiments at supercritical surface temperatures.

B.3.2 Steady-State with Integrated Heaters

The integrated heater method used by Searle [29] was investigated at higher temperatures. One issue was the lack of sufficient power to heat the surface to supercritical temperatures. Increased power to the heaters required thicker wires, which reduced flexibility. A potential solution was to increase the jet temperature, which would reduce the temperature difference. This was attempted, heating the water to 80 °C before impinging on a surface already at nominally 280 °C.
After a short amount of time, the wafer cracked in half for uncertain causes. The combination of pressure and temperature also caused some of the plastic tubes to deform and balloon out, so using heated water would require different tubing. After these incidents, a cooling study with unsteady conditions was pursued.