DROP IMPACT IN SPRAY COOLING

by

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Abstract

Spray cooling has enormous potential in addressing high-heat-flux thermal challenges in many cutting-edge technologies. In spray cooling, a flow of coolant drops is emitted from a spray nozzle and impacts a hot surface, which is covered by a flowing film. Heat transfers by convection from the surface to the liquid coolant, and the cooling performance is determined by the fluid dynamics of spray drops impacting the liquid film. The cooling mechanisms involved in spray cooling are still not clear, due to the lack of understanding of the heat transfer and fluid dynamics involved in the drop impact in spray cooling.

The research work puts focus on the drop impact in spray cooling, which is carried out in four major steps. The first step is to study spray impact cooling experimentally with focus on the cooling effects of nozzle positioning parameters including spray height and inclination angle. The positioning parameters are shown to have effects on global and local cooling as the drop impact is affected when changing the spray positioning. The second step is to experimentally and theoretically investigate the fluid dynamics of a single liquid drop impacting a flowing film. The third step is to evaluate heat transfer enhancement during a single drop impact. The work carried out in the second and third steps forms a comprehensive study on the thermal-fluidics of single drop impact on a flowing film. The fourth step moves on to the heat transfer enhancement of drop train impacting flowing films. A drop train is formed when drops are generated in groups, and each group has a consistent number of drops. A drop generator is combined with a special setup to generate drop trains with varied impact frequencies including group frequency (the generation rate of groups) and single frequency (the generation rate of single drops in each group). The results relate the cooling enhancement of continuous drop impact to important impact parameters including drop number, velocity, and impact frequencies.
Preface

All the presented research work was conducted in the Electronic Cooling & Multi-phase Laboratory and the Applied Micro and Nanosystems Facility in the School of Engineering at the University of British Columbia (Okanagan campus) under the supervision of Dr. Sunny Ri Li. Portions of this dissertation have been published in the journals as well as conference proceedings.

Chapter 3 includes the research work from one journal article [J1] and two conference papers [C2, C3]. In journal article [J1], I developed a high-heat-flux spray cooling system in a large heat surface and theoretically analyzed the relationship of local volumetric flux to local cooling performance.

Chapter 4 contains the research work from one journal article [J3]. I carried out the theoretical and experimental investigations for single drop impact on a flowing film.

Chapter 5 involves the research work from one journal article [J2] and two conference proceedings [C1, C4]. I carried out the experimental investigations for single drop impact on a flowing film cooling a hot surface, and analyzed heat transfer mechanism using a theoretical model.

Chapter 6 presents the research work of drop train impact on a flowing film cooling a hot surface. The research outcomes are in preparation for publication as a journal article.

There are some research outcomes during the Ph.D. study that have been published as journal articles [J4, J5] and conference proceedings [C5]. However, they are not included in the dissertation as they are beyond the scope of the Ph.D. study.
Publication List

**Refereed Journal Articles**


**Refereed Conference Proceedings**


[C4] **Gao X**, Kong LJ, Li R, Han JT. Heat transfer during a single drop impacting on a heated
flowing liquid film. *The 1\textsuperscript{st} Pacific Rim Thermal Engineering Conference* 2016, Hawaii, USA. (PRTEC-14414)

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List of Symbols

\( a \)  
jet nozzle radius

\( A \)  
impact area

\( A_0 \)  
perpendicular cross section area

\( A_h \)  
heater area

\( A_n \)  
area of normal impact

\( b \)  
wafer thickness

\( Bi \)  
Biot number

\( c_p \)  
specific heat capacity

\( D_j \)  
hydraulic jump diameter

\( D_0 \)  
drop diameter

\( f_{avg} \)  
average frequency of drop impact

\( f_i \)  
generation frequency of single drop

\( f_b \)  
generation frequency of drop train

\( Fo \)  
Fourier number

\( Fr \)  
Froude number

\( g \)  
gravitational acceleration

\( h \)  
heat transfer coefficient

\( h_f \)  
film thickness

\( h_l \)  
thickness of a thin film layer during drop impact

\( h_{ss} \)  
heat transfer coefficient at steady state

\( h_t \)  
film thickness of drop disc or transient heat transfer coefficient

\( H \)  
spray height

\( H_n \)  
spray height for normal impact

\( H_0 \)  
distance from cross section \( A_0 \) to nozzle

\( k \)  
thermal conductivity of silicon

\( K \)  
threshold parameter of splashing

\( Kc \)  
critical value of \( K \)

\( L \)  
spray impact length

\( n \)  
drop number in single drop train

\( Nu \)  
Nusselt number

\( Pr \)  
Prandtl number

\( Q \)  
volumetric flow rate of spray

\( Q_j \)  
volumetric flow rate of jet

\( Q_t \)  
volumetric flow rate of drop train

\( Q' \)  
film flow rate

\( Q'' \)  
spray flux

\( Q_0'' \)  
spray flux on \( A_0 \)

\( q'' \)  
surface heat flux
<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>$r$</td>
<td>radius on impact area</td>
</tr>
<tr>
<td>$r_c$</td>
<td>drop spreading radius</td>
</tr>
<tr>
<td>$r_i$</td>
<td>initial radius of drop spreading</td>
</tr>
<tr>
<td>$r_0$</td>
<td>radius on cross section $A_0$</td>
</tr>
<tr>
<td>$Re$</td>
<td>Reynolds number</td>
</tr>
<tr>
<td>$Rem$</td>
<td>modified Reynolds number</td>
</tr>
<tr>
<td>$S$</td>
<td>stretching rate</td>
</tr>
<tr>
<td>$t$</td>
<td>time</td>
</tr>
<tr>
<td>$t_i$</td>
<td>initial time for the drop deformation</td>
</tr>
<tr>
<td>$T_l$</td>
<td>water coolant temperature</td>
</tr>
<tr>
<td>$T_s$</td>
<td>surface temperature</td>
</tr>
<tr>
<td>$T_{ss}$</td>
<td>surface temperature at steady state</td>
</tr>
<tr>
<td>$u_i$</td>
<td>mean velocity of drop disc</td>
</tr>
<tr>
<td>$u_t$</td>
<td>velocity of drop disc during the spreading phase</td>
</tr>
<tr>
<td>$U_f$</td>
<td>film velocity</td>
</tr>
<tr>
<td>$U_0$</td>
<td>drop velocity</td>
</tr>
<tr>
<td>$\dot{V}$</td>
<td>film flow rate</td>
</tr>
<tr>
<td>$w$</td>
<td>velocity of the crown sheet in $z$ direction</td>
</tr>
<tr>
<td>$w_0$</td>
<td>velocity of the crown sheet at $z=0$</td>
</tr>
<tr>
<td>$W$</td>
<td>heater width</td>
</tr>
<tr>
<td>$We$</td>
<td>Weber number</td>
</tr>
<tr>
<td>$We_{em}$</td>
<td>modified Weber number</td>
</tr>
<tr>
<td>$x$</td>
<td>$x$ coordinate</td>
</tr>
<tr>
<td>$x_c$</td>
<td>displacement of crown base</td>
</tr>
<tr>
<td>$y$</td>
<td>$y$ coordinate</td>
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<tr>
<td>$z$</td>
<td>$z$ coordinate</td>
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**Greek symbols**

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>$\alpha$</td>
<td>half spray angle or thermal diffusivity or open angle of blocker</td>
</tr>
<tr>
<td>$\beta$</td>
<td>polar angle of spray ray</td>
</tr>
<tr>
<td>$\beta_c$</td>
<td>coefficient in drop spreading</td>
</tr>
<tr>
<td>$\gamma$</td>
<td>duty cycle</td>
</tr>
<tr>
<td>$\delta$</td>
<td>time variable</td>
</tr>
<tr>
<td>$\eta$</td>
<td>cooling enhancement</td>
</tr>
<tr>
<td>$\eta_{max}$</td>
<td>maximum cooling enhancement along the center line</td>
</tr>
<tr>
<td>$\eta_p$</td>
<td>peak cooling enhancement</td>
</tr>
<tr>
<td>$\theta$</td>
<td>inclination angle</td>
</tr>
<tr>
<td>$\lambda$</td>
<td>energy loss factor</td>
</tr>
<tr>
<td>$\lambda_f$</td>
<td>energy loss factor on flowing film</td>
</tr>
<tr>
<td>$\lambda_s$</td>
<td>energy loss factor on stationary film</td>
</tr>
</tbody>
</table>
\( \mu \) dynamic viscosity
\( \nu \) kinematic viscosity
\( \rho \) density
\( \sigma \) surface tension
\( \tau_0 \) time shift in drop spreading
\( \chi \) location along center line
\( \phi \) azimuthal angle on impact area
\( \phi \) azimuthal angle of spray ray
\( \Omega \) solid angle
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Dedication

To my loving wife, Huiru, and loving sons, Jacob and Brandon
Chapter 1 Introduction

1.1 Background

Heat generation takes place in many engineering processes. Heat removal, also referred to as thermal management, is important for maintaining the temperature to meet material and safety constraints. Thermal management becomes difficult for high heat flux, for which heat transfers through small areas. High heat flux can be commonly seen in micro-electronic and power electronic devices which have been experiencing continuous reduction in size and continuous increase in power. Effective cooling solutions are needed for the thermal management of the high-heat-flux devices. Spray cooling is one effective solution, which has the huge potential in handling the high heat fluxes in high-power electronics such as supercomputer, lasers, and radars.

In spray cooling, liquid coolant is emitted from a pressurized nozzle and breaks up into small droplets. The small droplets lands on the cooled surface, where the flow of droplets become a thin liquid film radially flowing on the surface. The cooling is achieved through the convection heat transfer from the cooled surface to the film flow being impacted by continuous flow of droplets.

Spray cooling has several advantages over other cooling techniques. In comparison with air cooling and jet impingement cooling, spray cooling owns a high heat flux removal capacity. Air cooling can handle heat flux ~ 1 W/cm², while jet impingement cooling may manage heat flux up to ~ 100 W/cm². Spray cooling can transfer heat in excess of 100 W/cm² using fluorinerts and more than 1000 W/cm² using water. Due to heat flux removal capacity, spray cooling allows precise temperature control with low fluid inventory. Besides, spray cooling has uniform cooling temperature distribution. This is because the entire spray-cooled area is receiving fresh liquid...
coolant droplets. In contrast for jet impingement cooling the surface temperature continuously increases from the stagnation point.

1.2 Research objectives

The fundamental mechanism in spray cooling is the convection heat transfer from hot surface to impinging spray flow. The cooling performance depends on the flow dynamics of spray impact, involving the interaction of numerous drops with the flowing film cooling hot surface.

Therefore, fundamental experimental and theoretical studies of drop impact and spray impact are conducted on an isothermal surface or a heated surface, with the following objectives:

- To consider effects of spray positioning on cooling performance in a large thin-film heater, including the orifice-surface distance and inclination angle. The global and local cooling performance in association with the nozzle positioning will be analyzed based on the high-resolution temperature record by an infrared camera. Detailed geometrical relations need to derive to track the flow for its cooling performance and spray flux.

- To study impact dynamics after a single drop impacting stationary film and flowing film generated by jet impingement. The spatial and temporal evolution of drop spreading is observed by and further analyzed by the theoretical model. Additionally, instability of crown sheet inducing the formation of secondary droplets is analyzed and the criteria value identifying splashing and non-splashing is proposed.

- To investigate heat transfer mechanism of a single drop impacting a heated surface cooled by flowing film. The factors affecting local heat transfer around impact area are
considered, including the flow rate of flowing film, drop velocity, drop size. The correlation of heat transfer enhancement to the drop and film flow should be obtained.

- To explore heat transfer mechanism of controlled drop train impact on the film-cooled hot surface. The controlled drop train is fulfilled by controlling the continuous drop train generated from the breakup of the liquid jet. The factors affecting local surface temperature and local cooling are considered, including flow rates of flowing film and drop number, impact velocity.

1.3 Structure of this dissertation

Focuses of this dissertation are on the interaction of drop flow with the film flow in spray cooling. Numerous drop impacts form a flowing film on impact surface, and hence drops land on a flowing film rather than dry surface during spray impact. The interaction is affected by the change of drop flow and film flow with varied the nozzle positioning in spray cooling. To further explore impact dynamics of drop flow on film flow, the impacts of single drop or multiple drops are applied with varied drop sizes, drop velocities, and impact frequency, and their effects on local cooling are also analyzed. The fundamental studies of single drop impact and drop train impact explore the interaction of drop and film flows and the relevant cooling performance, which contributes to the understanding of spray cooling performance and provides some methods for cooling enhancement. Based on research objectives the structure of this dissertation is described as follows.

Chapter 1 presents research background, objective and motivation, as well as the structure of this dissertation.

Chapter 2 contains the literature review of spray cooling and drop impact. Firstly, the
studies of spray cooling are summarized from three aspects including spray characterization, phase change, enhanced surface. Second, the studies of single drop impact and drop train impact on varied surface conditions are discussed.

Chapter 3 includes the experimental study of spray cooling. A spray cooling system in a large heat surface is developed to investigate the effects of nozzle positioning on cooling performance. Nozzle positioning affects the interaction of drop flow with film flow, which is fulfilled by changing drop flow and film flow with varied inclined angles. The effects of nozzle positioning on local spray flux have been demonstrated by geometrical model of spray cone, and the relationship of local spray flux to local cooling performance is also included.

Chapter 4 involves the experimental study of single drop impact on a flowing film. To clearly know what happened close to impact surface, the fluid dynamics of single drop impact on flowing film is captured by a high-speed imaging system from the side and bottom of impact surface. After drop impact, the evolution of drop spreading is explained by theoretical models, and the location of splashing occurrence is related to the stretching rate of liquid sheet. Splashing or non-splashing after drop impact depends on the interaction of drop and film flows. This finding contributes to the understanding of bounced spray during spray impact.

Chapter 5 is regarding heat transfer mechanism due to single drop impact on a heated flowing film. The drop spreading process is recorded by a high-speed imaging system, and the relevant surface temperature change is captured by an infrared camera. Drop flow is changed by varied drop velocities and drop sizes, and film flow is adjusted by varied flow rates. Fast drop spreading benefits the local cooling enhancement at the early stage, but slow drop flow turns out to worsen the local cooling. The peak cooling enhancement is correlated to the ratio of drop flow
to film flow.

Chapter 6 presents the experimental study of drop train impact on a film-cooled hot surface. Drop train impact is similar to spray impact at a certain location of impact surface. A drop train is formed when drops are generated in groups, and each group has a consistent number of drops (one or multiple). A drop generator is combined with a special setup to generate drop trains with varied impact frequencies including group frequency (the generation rate of groups) and single frequency (the generation rate of single drops in each group). Due to drop train impact, the surface temperature change and cooling enhancement are investigated for varied drop train and film flow conditions, involving film flow rate, drop number of drop train, impact velocity. The selection mechanism of drop train enables us to determine the most dominant impact parameter affecting local cooling performance.

Chapter 7 summarizes all the research outcomes in this dissertation and proposes further work on this topic.
Chapter 2 Literature Review

Spray cooling can handle high heat flux in the constrictive space of electronic package when comparing to air cooling, pool cooling, and jet cooling. This is because numerous fresh droplets generated by spray nozzle randomly impact the whole cooled surface, and directly transfer the heat of impact surface to the cooling fluid. Drop impact is of critical importance to heat transfer performance of spray cooling. In this chapter, previous research articles on spray impact, single drop impact, drop train impact, as well as their relationship with the cooling performance are summarized.

2.1 Spray cooling review

Due to the high-heat-flux removal capacity of spray cooling, numerous fundamental studies have been conducted theoretically and experimentally, which focus on the key parameters affecting fluid dynamics of spray impact and the relevant heat transfer mechanism. There are three aspects that have been demonstrated to significantly affect spray cooling performance, including spray characterization, phase change and enhanced surface.

2.1.1 Effects of spray characteristics

Since the earliest study on spray cooling by Toda [1, 2], most of the researchers put effort on spray characterization, the relevant heat transfer performance as well as the critical heat flux (CHF) in spray cooling. Once reaching the CHF and coming to the transition boiling (decreasing region in the boiling curve), the efficiency of heat transfer on the heating surface significantly decreases while the electric devices are likely to be overheated. Understanding CHF is of crucial importance to maintain electronics devices within a safe temperature region.
Spray characterization includes drop size, velocity, drop flux, and volumetric flux. In experimental studies, it is difficult to change only one parameter and isolate the remaining parameters. For example, on the cooled surface with a certain impact area the increase of flow rate of coolant spray leads to the increase of volumetric flux, which is also accompanied with the increase of drop velocity. That is reason that the conclusions made on the dominant impact parameter are not consistent in previous studies of spray cooling.

Chen et al. [3] studied effects of three spray parameters of drop size, drop velocity and drop flux on CHF. By adjusting spray nozzles, operating pressures, and spray distance between the nozzle exit and the heater surface, the effect of a spray parameter was studied while the other two parameters were kept constant. It was found that the mean drop velocity is the most dominant parameter affecting CHF followed by the mean droplet flux, while the Sauter mean droplet diameter (D_{32}) has a negligible effect on CHF. In their later study [4], they further demonstrated the CHF varies with N^{1/6} and U^{1/4} (N and U are referred to as drop flux and impact velocity). After determining local spray characteristics and local cooling for the various regimes in the water spray boiling curves, Mudawar and Valentine [5] found the dominant effect of the volumetric spray flux compared to other spray characteristics, and CHF was correlated to the volumetric spray flux and mean droplet diameter. After applying the dielectric coolant of PF-5052 with a lower boiling point of 50 °C, Rybicki and Mudawar [6] concluded that the volumetric flux and Sauter mean droplet diameter are the most significant spray parameters influencing the spray cooling performance. Heat flux increases with increasing volumetric flux for a number of reasons. A larger fluid volumetric flux results in higher liquid velocity over the surface. The impact of the drops onto the film can also interact with the liquid film, thinning the local thermal boundary layer.
Cooling performance can be changed by changing the spray positioning. There are two spray positioning parameters in the study of spray cooling: nozzle-surface distance and inclination angle. Within the small surface area around 1 cm², Mudawar and Estes [7] found that the optimal nozzle-surface distance for the maximum CHF was achieved when the spray footprint was exactly enclosed within the cooling surface.

A few researchers [8-12] focus on the effects of spray inclination on heat transfer performance. Silk et al. [8] studied the effects of spray inclination by comparing the cooling performance under four different inclination angles with the enhanced surface. It was found that heat flux increased with the increase of spray angle for the flat surface, but the heat flux performance was little changed within the experimental uncertainty for spray angle larger than 15°. The increase in heat flux might be attributed to better liquid drainage through the elimination of the stagnation zone. Wang et al. [9] found that inclination of spray nozzle could enhance heat transfer if optimal orifice-surface distance could be found.

However, Visaria and Mudawar [10] indicated that inclination angle had minimal impact on the single-phase or two-phase regions of the boiling curve, and increasing inclination angle would decrease CHF and maximum CHF was always attained with the spray normal to the test surface. Cheng et al. [11] found that the inclination angle would worsen heat transfer when the spray footprint is smaller than the heated surface. When Fu et al. [12] studied the ultra-fast cooling of steel plate using spray cooling technology, they demonstrated that heat transfer would enhance with the inclination angle increasing from 0° to 30° but it would weaken with the inclination angle increasing from 30° to 60°. Rybicki and Mudawar [6] used upward-oriented and downward-oriented spray nozzles to assess their effects on cooling performance. The experimental results showed that the spray orientation has no measurable effects on the global
cooling performance in both single-phase and two-phase regimes. Therefore, the conclusions of these studies on spray inclination are contradictory and the current study has focused on this topic.

In spray cooling, spray drops land on a liquid flowing film rather than a dry surface. The property of the flowing film and its effects on spray cooling performance have been studied by some researchers [13-15]. Pautsch and Shedd [13] used a non-intrusive optical technique to measure the local film thickness generated by sprays. The film thickness was found to remain constant when the heat transfer mechanism was dominated by single-phase convection. Beyond the spray impact area, the dryout phenomena appear even when the CHF is not reached. In the nucleate boiling regime, Horacek et al. [14, 15] measured the dryout area, which was characterized by the three-phase contact line length using a Total Internal Reflectance technique. The wall heat flux was found to correlate very well with the contact line length. The larger contact line length corresponds to the larger wall heat flux. This contact line heat transfer mechanism was summarized by Kim [16] as one of main heat transfer mechanisms in the two-phase regime.

2.1.2 Phase change effects

Toda [1, 2] plotted a heat transfer curve of spray cooling for a wide range of surface temperature and separated it to four regimes, referred as to single phase regime, nucleate boiling regime, transition boiling regime and film boiling regime. In the single phase regime, the heat flux linearly increases with increasing surface temperature difference between the heater surface and coolant, and forced convection and evaporation play dominant roles in this regime. In the nucleate boiling regime, the heat flux increases steeply, and nucleation plays the major role due to a large amount of the latent heat absorbed during evaporation. Once the nucleation sites cover
the whole heated surface, average heat flux will reach a peak value, which is defined as Critical Heat Flux (CHF). During the transition boiling, the heat flux would drop sharply. Liquids absorb much heat from the surface and form the vapor blanket, so the surrounding liquids are hard to get to the heater surface. That is the reason for the sharp decrease of heat flux in this regime. An interesting phenomenon is an increasing trend of heat flux found in the film boiling regime. Massive heat is generated from the heated surface and radiation heat transfer becomes a key heat transfer mechanism between the heated surface and the liquid, so the heat flux tends to increase from the Leidenfrost point. Based on the scope of this study, the focus is put on the literature of spray cooling in single phase regime, nucleation boiling regime.

In the single phase regime, some studies [6, 17, 18] were conducted for correlating the single phase data with cooling performance. Rybicki and Mudawar [6] proposed the following correlation for dielectric PF-5050 spray by using the Sauter mean diameter and volumetric flux as length and velocity scales:

\[
Nu = 4.7 Re^{0.61} Pr^{0.32}
\]  

(2.1)

Karwa et al. [17] developed the heat transfer correlation of full-cone water spray on a constant heat flux heater with varied pressure with an accuracy of ±7.3 %:

\[
Nu = 20.344 Re^{0.659}
\]  

(2.2)

Heieh and Tien [18] used R-134a spray to obtain the correlation for the average Nu in the single phase regime with an accuracy of ±10 %:

\[
Nu = 933 We^{0.36} (d_{32}/d_0)^{0.25} (\Delta T/T_s)^{0.027}
\]  

(2.3)

Compared to the single phase regime of spray cooling, the heat transfer coefficient in the nucleate boiling regime increases even faster. There are two reasons to explain this result proposed by Yang et al. [19]. As surface temperature increases, nucleation boiling occurs, and
the liquid coolant changes to the vapor. During the phase change, a larger amount of heat is absorbed from the heater surface, resulting in a higher heat flux across the heater surface. The other reason is attributed to the contribution of secondary nucleation (firstly proposed by Mesler [20]) and evaporation on the enhancement of the heat flux. When the droplets impinge on a flowing film on the heater surface, they entrain air into the liquid film and form an air layer underneath the droplets. The air layer reaches the liquid-covered surface and finally breaks up into many tiny gas nuclei, which serve as secondary nucleation sites. Hence, the number of secondary nucleation sites is proportional to the droplet flux across the surface, which was proved in Yang’s experiments [21, 22].

Recently, the air layer formed during the drop impact on a solid surface is validated and exactly measured using ultrafast interference imaging technology [23-27]. During the drop impact with a large droplet size, the non-breakup air layer exists for a while, the surface temperature around this air layer is higher than the surrounding temperature, which is proved to worse the local heat transfer [28].

Using water as coolant liquid, Mudawar and Valentine [5] firstly proposed the CHF correlation with respect to the local volumetric flux $Q''$, and Sauter mean diameter (SMD) $d_{32}$.

$$\frac{q_m''}{\rho \varepsilon Q'' h_{fg}} = f\left(\frac{\rho_f}{\rho_g}, \frac{\rho_f Q''^2 d_{32}}{\sigma}, \frac{\rho_f c_{p,f} \Delta T_{sub}}{\rho_g h_{fg}}\right)$$  \hspace{1cm} (2.4)

In the late study of Estes and Mudawar [29], a universal CHF correlation was constructed for spray cooling by using Fluorinerts FC-72 and FC-87 as well as the water.

$$\frac{q_m''}{\rho \varepsilon Q'' h_{fg}} = 2.3\left(\frac{\rho_f}{\rho_g}\right)^{0.3} \left(\frac{\rho_f Q''^2 d_{32}}{\sigma}\right)^{-0.35} \left(1+0.0019\frac{\rho_f c_{p,f} \Delta T_{sub}}{\rho_g h_{fg}}\right)$$  \hspace{1cm} (2.5)
2.1.3 Enhanced surface effects

The early studies of spray cooling mainly have been conducted on flat surfaces [1-7]. A few of them focus on the effects of surface roughness on cooling enhancement [30, 31]. Pais claimed [30] that surface roughness has a major effect on nucleation boiling. With the decrease of surface roughness, the heat flux increases in the region of nucleation boiling. The reason is that the large surface roughness implies a thicker film thickness. The thick film results in later bubble breakup and departure, impeding of vapor escape, increased resistance to heat flux through evaporation on film surface, and dampening of droplet impingement. Bernardin et al. [31] reported that CHF is independent of surface roughness while the Leidenfrost point temperature is sensitive to it. In the last decade, spray cooling on the enhanced surface [32-40] has been attracted huge attention for the huge enhancement of cooling performance. The experimental studies [32-34] applied micro-textured surfaces with surface feature size from 25-480 μm, close to liquid film thickness but larger than average droplet size. Micro-textured surfaces [32-34] showed slight effects on heat transfer enhancement in the flooded region, but greatly enhancing cooling performance in the thin film and partial dry-out regions as compared to the flat surface.

Mini-textured surface [35-37] with feature size above 1 mm would result in heat transfer enhancement for all the enhanced surfaces because the enhanced surfaces considerably extended the heat transfer area. For all the enhanced surfaces, the straight fins had the largest heat flux enhancement followed by the cubic pin fins and the pyramid surface. The latest study by Zhang et al. [38] showed that nanostructured surface had a better cooling performance since the contact angle was smallest on the nanostructured surface as compared to micro-structured surfaces and flat surfaces.
2.2 Single drop impact review

From the literature review of spray cooling, a few researchers focused on fluid dynamics of spray impingement on the impact surface because of inability to capture clean images by the high-speed imaging system, which is commonly used in the research of fluid dynamics. To deeply understand spray impingement mechanism, a lot of researchers simplified spray impingement to drop impingement and studied fluid dynamics as well as its influence on cooling performance. These researchers are summarized and classified according to surface conditions, mainly including dry surface, stationary film as well as flowing film. Figure 2.1 shows single drops impacting three types of surfaces, which is part of the experimental study to be reported in details in Chapter 4.

Figure 2.1 Single water drops with same velocity and diameter ($U_0 = 1.85$ m/s, $D_0 = 3.2$ mm) impact three different types of surfaces: (a) dry surface, (b) stationary water film, (c) flowing water film (The arrow indicates the film flow direction).
2.2.1 Impact on dry surfaces

On the dry surface, as shown in Fig. 2.1(a), the drop spreads, recoils, and eventually forms a sessile drop on the surface. On the stationary film as shown in Fig. 2.1(b), the drop impact produces an axisymmetric rising liquid sheet, which is referred to as a crown. The crown expands radially, and its height changes. On the flowing liquid film as shown by Fig. 2.1(c), a rising liquid sheet is formed where the drop spreading direction is opposite to the film flow, thereby forming a non-axisymmetric crown. The crown sheet looks thinner than the sheet formed on the stationary film. Additionally, small droplets are formed, and the phenomenon is referred to as splash. The splash shown in Fig. 2.1(c) is categorized as a crown splash, as the droplet resulted from the rim breakup at the top of the crown sheet due to capillary instability. It has been observed by Zhang et al. [39] that even smaller droplets originated from the breakup of ejecta sheet and emerging lamella of the film and this type of splash is referred to as micro-droplet splash. These two types of splashes were summarized by Deegan et al. [40].

On dry surface, the process of a liquid drop impact was divided by Rioboo et al. [41] into five successive phases: kinematic, spreading, relaxation, wetting, and equilibrium. Most research work has been focused on spreading and relaxation. In the spreading phase, contact line expands radially until reaching a maximum spread diameter, which is determined by droplet initial diameter, impact velocity, surface tension, viscosity, and wettability of the solid surface [42]. The maximum spread diameter is of critical importance in spreading phase, which has been widely investigated by many researchers [42-49] using experimental and theoretical methods. Clanet et al. [43] found that the maximal spread on a super-hydrophobic surface was significantly dependent on the viscosity of liquid droplets. The maximum spread of an impacting droplet in the capillary regime was found to scale as the Weber number, while the maximum
spread in viscous regime could be plotted as a function of the Reynolds number. Transition to the
viscous regime was found when the impact number $P=\frac{\text{We}}{\text{Re}^{4/5}} \sim 1$. van Dam and Clerc [44]
found a significant difference of maximum spread between substrates with small and large
contact angles, which indicated the influence of surface wettability in the later stage of impact.

Some analytical models [46-49] were proposed to predict impaction process, in particular
for maximum spread. Most models were based on the energy conservation of the droplet. Since
Engel [46] developed the first correlation equation, efforts have been made to improve the
accuracy of theoretical prediction. Chandra and Avedisian [47] developed an empirical
correlation of viscous dissipation, which includes estimated spreading time, simplified
dissipation function, and estimated volume of viscous dissipation. Improved predictive equations
for viscous dissipation was presented by Mao et al. [48] The model after improvement could
provide more accurate prediction of maximum spread for most cases except for low Reynolds
and Weber numbers. It was demonstrated that the theoretical model based on assuming the
droplet shape as a cylindrical disc at the maximum extension does not fit the experiments well in
some cases. Recently, Gao and Li [49] proposed a theoretical model based on the actual dynamic
shape of the drop that could successfully predict the maximum spreading diameter and receding
diameter during the recoiling process.

Some of the researchers [50-55] put efforts on the investigation of splash using varied dry
surfaces. Surface roughness and textures were demonstrated to influence the splash limit [50-53].
For example, Sivakumar et al. [53] found that the texture pattern of the surface can cause splash
to occur during drop impact. Drop impact on a moving dry surface was found to show different
splash and non-splash phenomena as compared to stationary dry surfaces [54]. Previous studies
on splash threshold under different surface conditions are summarized in Table 2.1.
Table 2. Summary of splash thresholds under different surface conditions ($U_s$ denotes the moving speed of dry surface.)

<table>
<thead>
<tr>
<th>Surface conditions</th>
<th>Threshold parameter $K$</th>
<th>Critical value $K_c$</th>
<th>References</th>
</tr>
</thead>
<tbody>
<tr>
<td>Dry surface</td>
<td>$(WeRe^{1/2})^{1/2}$</td>
<td>57.7</td>
<td>Mundo et al. [51]</td>
</tr>
<tr>
<td></td>
<td>$We^{0.5}Re^{-0.391}$</td>
<td>0.8458</td>
<td>Vander Wal et al. [66]</td>
</tr>
<tr>
<td>Moving dry surface</td>
<td>$WeRe^{1/2}(1 - 2.5 \frac{U_s}{U_0}Re^{-1/2})^2$</td>
<td>5700</td>
<td>Bird et al. [54]</td>
</tr>
<tr>
<td>Stationary liquid film</td>
<td>$(WeRe^{1/2})^{0.8}$</td>
<td>2100</td>
<td>Cossali et al.[63], Rioboo et al. [64]</td>
</tr>
<tr>
<td></td>
<td>$We^{0.5}Re^{0.17}$</td>
<td>63</td>
<td>Vander Wal et al.[66]</td>
</tr>
<tr>
<td>Flowing liquid film</td>
<td>$WeRe^{1/2}(1 + \bar{h}_f \bar{U}_f^2)(1 + \bar{h}_f \bar{U}_f)^{1/2}$</td>
<td>3378</td>
<td>Present study</td>
</tr>
</tbody>
</table>

A few researchers [56-59] investigated molten mental drop impact with high velocity greater than 100 m/s. Mehdizadeh et al. [56] studied plasma-sprayed molten molybdenum droplets impacting at high velocity ~140 m/s on a glass substrate at room temperature. The droplets are found to spread into a thin circular film that ruptures and breaks into small fragments. McDonald et al. [57] demonstrated the effects of preheated mental substrate on impact dynamics. Impact on a heated substrate produces splats with less fragmentation, which is attributed to the reduced surface skewness of the preheating substrate. The later study of McDonald et al. [58, 59] further compare the droplets impacting on a smooth glass at room temperature and 400 °C. The spreading diameter of splats on a glass surface at room temperature is almost three times than that at 400 °C. The splashing appears only on the surface held at room temperature since the portion of the splat is not in good contact with the surface. Additionally, the cooling rate on heated surface is significantly larger than that on surface at the room temperature, due to smaller thermal contact resistance [59]. All the findings contribute to practical application of spray coating technology.
2.2.2 Impact on stationary films

On a stationary film, most researchers focused on spread process and splash formation mechanism after single drop impact. The studies by Yarin and Weiss [60] and Roisman and Tropea [61] are significant. Yarin and Weiss [60] developed a quasi-one-dimensional model, which predicts the existence of a kinematic discontinuity in the velocity and film thickness distribution. The discontinuity corresponds to the emergence of an uprising liquid sheet, which is viewed as a crown. Roisman and Tropea [61] generalized Yarin’s theory for the case of arbitrary velocity vectors in the liquid films both inside and outside the crown. The theory was developed for drop impact on both stationary and flowing liquid films. Yarin and Weiss [60] experimentally found the distance of crown rim from the impact center was expressed as a function of the non-dimensional spreading time. However, two empirical parameters existing in their model were not clear. Using a well-designed experimental setup, Cossali et al. [62] observed the impact process from both the side and underneath the transparent substrate. They also investigated the two empirical parameters in Yarin and Weiss model by considering the influence of film thickness and drop impact velocity.

Rayleigh-Taylor [63, 64] instability and Richtmyer-Meshkov [65, 66] instability were commonly used for analyzing the breakup of suddenly accelerated liquid sheet [67, 68]. In the first case the instability occurs when the light fluid accelerates the heavy one, while in the second case the instability appears when an interface between two fluids with different density impulsively accelerates with a shock wave. Dombrowski and Fraser [67] found that the stable liquid sheet is obtained by a liquid of high surface tension, high viscosity, low density, but the liquid sheet breaks up when the velocity is high. Krechetnikov [68] treated the Rayleigh-Taylor and Richtmyer-Meshkov instabilities as a single phenomenon and analyzed the relationship
between these two fundamental instabilities based on operator and boundary perturbation theories. Another study by Roisman et al. [69] discussed the splash mechanism based on bending rim instability, which is caused by the inertia of the liquid entering the rim from the free liquid sheet. It was also demonstrated that the stretching of the liquid sheet normal to rim could significantly influence the rim instability [70]. Krechetnikov and Homsy [71] proposed a Richtmyer-Meshkov instability to describe the instability mechanism of drop splash in association with an impulsive acceleration of the interface. Recently, Zhang et al. [72] showed that the generation of droplets is due to the Rayleigh-Plateau instability of the rim at the top of liquid sheets. The underlying instability mechanism for splash is still an open question from a theoretical perspective.

Drop impact on a stationary film may or may not result in the splash. Finding the threshold condition for splash impact has been the focus of a few experimental studies. Cossali et al. [62] tested drops of various mixtures of water and glycerol impacting a thin liquid film and proposed an empirical parameter for predicting the occurrence of splash impact. In the present work, this parameter is referred to as threshold parameter, denoted by \( K \). The threshold parameter in Cossali et al. [73], expressed as a function of Reynolds number \( Re = U_0 D_0 / \nu \) and Weber number \( = \rho U_0^2 D_0 / \sigma \), is \( (WeRe^{1/2})^{0.8} \). Here \( \nu, \rho, \) and \( \sigma \) are the kinematic viscosity, density, and surface tension of the drop liquid. For thick films, Cossali et al. [73] and Rioboo et al. [74] found a critical value of the threshold parameter, i.e. \( K_c = 2100 \), above which splash impact occurs. Vander Wal et al. [75] used twelve fluids for drop impact on both dry surfaces and liquid films, with a wide range of fluid properties and impact conditions. Based on the experimental observations, the threshold parameter was proposed to be \( We^{0.5} Re^{-0.391} \) for dry surfaces and \( We^{0.5} Re^{0.17} \) for thin liquid films. A further study by Vander Wal et al. [76] demonstrated that
high viscosity promotes splash on dry surfaces, whereas it inhibits splash on thin fluid films. Notably, the surface conditions mainly include dry surfaces and stationary films. It is clear from previous studies that there is not a conclusion on splash threshold general for all surface conditions, as summarized in Table 2.1.

2.2.3 Impact on flowing films

Although many researchers put effort on the study of single drop impact, few of them investigated the interaction between drop spreading flow and flowing film flow. Alghoul et al. [77] presented an experimental investigation of a liquid drop impacting onto horizontal moving liquid films. The impact outcome showed similarities when compared to the impact on stationary film, but transition boundary between non-splashing and splashing was different. In addition, an unsymmetrical crown shape was observed due to the effect of the moving film. Recently, Che et al. [78] demonstrated the inclined falling flow also resulted in the formation of the unsymmetrical crown shape after drop impact. However, none of them did the theoretical analysis for its formation. Until now, the literature of a single drop impact on a thin flowing film (less than 200 μm close to the thickness of spray film) is still limited.

2.2.4 Impacting heated surfaces

In the past few decades, investigation of drop impact on a heated surface [79-87] has received increased attention. Bernardin et al. [79] used two surface temperatures to map the boiling curve of drop impact into the four distinct heat transfer regimes: (1) single-phase liquid cooling, (2) nucleate boiling, (3) transition boiling, (4) film boiling. Two surface temperatures corresponding to critical heat flux and the Leidenfrost point showed little sensitivity to both drop velocity and impact frequency.
In the regime of single-phase liquid cooling, Pasandideh-Fard et al. [80] experimentally and numerically found increasing drop impact velocity would enhance heat flux around the impact area. This was because the raising drop velocity promoted drop spreading, thus increasing the wetted area on the heated substrate. However, increasing drop impact velocity slightly enhanced heat flux at the impact point. Qiao and Chandra [81] demonstrated that before the nucleate boiling addition of the surfactant to water drop would reduce contact angle and increase the surface area wetted by the drop. Adera et al. [82] reported the formation of non-wetting droplets on a super-hydrophilic micro-structured surface by slightly heating the surface above the saturation temperature of the droplet fluid, which was contributed by the increased thermal conductivity and decreased vapor permeability of the structured region.

Recently, Staat et al. [83] explored the behavior of drop impacting superheated surface across different regimes separated by the transition to splashing and the transition to the Leidenfrost state: (1) deposition regime (contact with heated surface and no splashing), (2) contact-splashing regime (contact and splashing), (3) bounce regime (neither contact nor splashing), (4) film-splashing regime (no contact, but splashing). The Leidenfrost transition temperature showed little dependence on the Weber number, but the transition to splashing showed a strong dependence on the surface temperature. Tran et al. [84] showed that in both contact-splashing regime and bounce regime the maximum spreading of a drop impacting heated surfaces follows a universal scaling with the Weber number ($\sim \text{We}^{2/5}$). Celata et al. [85] reported a decrease of the Leidenfrost temperature with increased impact velocities, while other groups [86, 87] found an opposite influence of impact velocities. The influence of drop impact velocities on the Leidenfrost temperature is still not clear with these contradictory results.
2.3 Review on drop train impact

In spray cooling, fresh drops continuously impact heated surface which is covered by the flowing film. The fluid dynamics behind this is the interaction of continuous drop train flow with the flowing film. To investigate the heat transfer of spray cooling from this aspect, a few studies have been conducted on the heat transfer of continuous drop train impinging on hot surfaces [88-98]. Qiu et al. [88-91] found that the surface temperature above the boiling temperature would enhance the spreading rate of the flowing film significantly and affect the splashing angle. Soriano et al. [92] presented an experimental observation of multiple drop train impingement. Impact spacing between multiple drop streams would affect spreading and splashing in impact regimes, and the optimal cooling performance was achieved when the film velocity was not disturbed by adjacent drop streams. The simulation results [93-94] also demonstrated the impact velocity played a dominant role in promoting local heat transfer.

Recently, Zhang et al. [95-98] further demonstrated that both impact spacing and impingement pattern significantly affect local and global cooling performance on the hot surface. In comparison with the circular jet impingement cooling, the drop train impingement achieves a better cooling performance for various impingement patterns. Previous studies of drop train impact mainly focus on heat transfer enhancement on the dry hot surface, but few of them investigate research topic of heat transfer of drop train impacting on flowing film. This topic is significant to the understanding of spray cooling performance. The spray cooling is affected by spray parameters such as impact velocity, drop size, drop number flux, and these parameters are dependent variables in spray impingement. Thus, it is difficult to identify the dominant parameter for spray cooling enhancement.
Chapter 3 Spray Cooling on Thin-film Heater

3.1 Background

Before proceeding to the study of drop impact dynamics, spray cooling on a thin-film heater is investigated by considering effects of spray positioning on global and local cooling performance. As compared to previous spray studies, there are some contributions to the understanding of spray cooling. 1). Both orifice-surface distance and inclination angle are studied. 2). By using an infrared camera, high-resolution local cooling is related to the nozzle positioning. 3). A large heater area with 5.5 cm$^2$ (less than 1 cm$^2$ for previous studies) is used so that the positioning variables can be changed in large ranges. 4). Detailed geometrical relations are derived to track the flow for its cooling performance and flow flux.

3.2 Experimental measurement

The experimental setup used for conducting the tests of spray cooling is schematically shown in Fig. 3.1. It is composed of: 1) a pressurized water supply; 2) a full-cone spray nozzle attached to a positioning system; 3) a heater plate; 4) a high-speed camera for visualizing the spray; 5) an infrared camera for measuring the surface temperature.

![Figure 3.1 Schematic of the experimental setup.](image)
A high-pressure nitrogen cylinder is connected to a pressurized water tank at room temperature. The pressurized water is supplied to a full-cone spray nozzle (TG SS 0.3, Spraying Systems Company, Wheaton, IL, USA), which atomizes the water flow through an orifice with diameter of 0.51 mm. The volumetric flow rate of the spray, denoted by \( Q \), ranges from 2.5 to 6.7 cm\(^3\)/s. For the tested range of flow rate, the spray angle denoted by \( 2\alpha \) ranges from 51° to 60°. The temperature of water denoted by \( T_i \), is measured using a calibrated T-type thermocouple. The nozzle is attached to a positioning system to change the nozzle location and orientation. Parameters associated with the nozzle positioning will be discussed later in this section.

The test surface is made of a silicon wafer (4 inches in diameter) with a thickness of \( b=380 \mu\text{m} \) and thermal conductivity of \( k = 149 \text{ W/m·K} \). The upper surface of the wafer is exposed to the spray, and is called impact surface. At the center of the lower surface, a square area with width \( W = 23.5 \text{ mm} \) is coated with gold. The gold-coated layer serves as a film heater, and its area \( A_h = W \times W \) is called heater area. Between the silicon wafer and the gold layer, there are a dielectric layer and an adhesive layer. All the layers have a total thickness less than 2 \( \mu\text{m} \), which has negligible resistance to heat conduction. The electrical resistance of the film heater is \( \sim 1 \Omega \). A DC power supply (Model 62050P, Chroma, Irvine, CA, USA) is connected to the heater, and the heater power is determined based on the electrical current and voltage drop. The coated area and the electrical connection are designed for achieving uniform current density throughout the gold layer. The average heat flux in the heater area, denoted by \( q'' \), is calculated by dividing the heater power with the heater area, and ranges from 26.2 to 26.8 W/cm\(^2\). Based on the measurement uncertainties of voltage, current and heater area, the overall uncertainty of \( q'' \) is \( \sim \pm 2\% \).
A high speed camera (M310, Vision Research, Wayne, NJ, USA) is used to observe the spray. An infrared (IR) camera (SC660, FLIR, Burlington, ON, Canada) is positioned underneath the silicon wafer to measure the temperature on the lower surface, denoted by $T_s$. To increase the measurement accuracy, the lower surface of the silicon wafer is painted black to achieve a high emissivity of 0.95. The optical resolution of the IR images is $\sim 5.6$ pixels/mm$^2$, which provides detailed temperature distribution.

The heat transfer coefficient in the heater area can be calculated using

$$h = \frac{q^\prime\prime}{T_s - T_l}$$  \hspace{1cm} (3.1)

Here $T_s$ is not the temperature at the solid-fluid interface as it is measured from the lower surface.

To analyze the local cooling and global cooling, the heat transfer coefficient rather than non-dimensional Nusselt number is used. There are two reasons for this. First, the focus of the current study is not on developing local cooling correlation, but is on the distribution pattern of local cooling. Evaluation of local cooling based on local heat transfer coefficient is sufficient to show qualitatively correct patterns. Second, only one fluid, water, was used.

Heat loss includes free air convection and radiation from the lower surface of the silicon wafer and also heat conduction through the electrical connections and the peripheral area (no gold coating) of the wafer. In the present work, the heat transfer coefficient of spray cooling, denoted by $h$, is larger than $1 \text{ W/cm}^2 \cdot \text{K}$ in most of spray-covered area. The heat loss is estimated to be less than 1\% of the total heater power, which is considered negligible. The heat conduction through the thickness of the silicon wafer is neglected as the Biot number $ht/k < 0.1$. Based on the uncertainty of $q^\prime\prime$ and the measurement uncertainties of $T_s$ and $T_l$ ($\Delta T_s \sim \pm 0.5^\circ\text{C}$, and $\Delta T_l \sim \pm 0.1^\circ\text{C}$), the overall uncertainty of $h$ is $\sim \pm 3\%$. 


Figure 3.2 (a) Normal impact of spray ($Q=2.5 \text{ cm}^3/\text{s}$); (b) inclined impact of spray cooling ($Q=5.0 \text{ cm}^3/\text{s}$, $\theta=20^\circ$); (c) Geometrically, the spray impact on a surface is a spray cone intersected by a surface. The 2D geometry is on the central plane of the cone perpendicular to the impacted surface. The positioning of the nozzle is determined by inclination angle $\theta$ and spray distance $H$. $H_n$ is the required spray distance by normal impact to cover a given impact length $L$. The impact area is shown in Fig. 3.3.

The positioning stage for the spray nozzle can change the location of the nozzle vertically and horizontally. It also can adjust the nozzle orientation relative to the impact surface to achieve normal impact (see Fig. 3.2a) or inclined impact (see Fig. 3.2b). Similar to the 2D images shown in Figs. 3.2a and 3.2b, Fig. 3.2c shows a side-view schematic of spray impact.
The nozzle positioning can be specified by the spray height \( H \) and the inclination angle \( \theta \). In the present work, the spray is inclined by rotating the spray nozzle clockwise. The tested ranges of the two positioning variables are \( 5 \ mm \leq H \leq 28 \ mm \) and \( 0^\circ \leq \theta \leq 40^\circ \). In this 2D schematic, the length \( L \) covered by the spray is called impact length. The relation between the spray height, inclination angle, and impact length is expressed by

\[
L = \frac{H \sin \alpha}{\cos \theta} \left[ \frac{1}{\cos(\theta - \alpha)} + \frac{1}{\cos(\theta + \alpha)} \right]
\]

(3.2)

In Fig. 3.2, the viewed plane is the \( z-x \) plane, normal to which is the \( y \) axis. Hence, the impact surface is on the \( x-y \) plane.

The footprint of the spray on the impact surface is called impact area, denoted by \( A \). The impact area is circular for \( \theta = 0^\circ \) and elliptical for \( 0^\circ < \theta < (90^\circ - \alpha) \). For all the tests, the impact area, heater area, and impact surface are concentric, and the center is located at \((x, y) = (0, 0)\). Figure 3.3 shows three impact areas with equal impact length \( L = 20 \ mm \) in the square heater area with \( W = 23.5 \ mm \). The line passing through vertex-1 and vertex-2 is the centerline of the heater area and impact area. For each impact area shown in Fig. 3.3, the spray height \( H \) can be calculated using Eq. (3.2). The geometric equation for plotting the impact area will be derived in Section-3.4.

The spray is considered as a solid circular cone with apex angle \( 2\alpha \). The impact area is the cross section formed by intersecting the spray cone with the impact surface on the \( x-y \) plane. The \( z-x \) plane shown in Fig. 3.2c is the central plane of the spray cone perpendicular to the impact surface. And the centerline shown in Fig. 3.3 is where the central plane intersects the impact surface. As shown in Fig. 3.2, the elliptical cross section is between two parallel cross
sections perpendicular to the axis of the conical body. The smaller cross section, $A_0$, contains vertex-1, and its distance to the nozzle orifice is

$$H_0 = \frac{H \cos \alpha}{\cos(\theta - \alpha)}$$

(3.3)

Figure 3.3 Impact area with constant impact length $L$ formed by the spray inclined at different angles $\theta$. The square 23.5 mm x 23.5 mm represents the heater area. The impact region is concentric with the heater area. The centerline is the intersection line between the impact surface and the central plane of the spray perpendicular to the impact surface.

Another cross section is perpendicular to the axis of the vertical conical body, and is actually the area of normal impact $A_n$ (see Fig. 3.3). Its distance from the orifice, denoted by $H_n$:

$$H_n = \frac{L}{2 \tan \alpha}$$

(3.4)

For normal spray impact to cover a given impact length $L$, Eq. (3.4) is the required spray height. The two cross sections follow $(A_0/A_n) = (H_0/H_n)^2$. Substituting Eq. (3.2) into Eq. (3.4) gives $H_n$ as a function of $H$ and $\theta$, which is
\[ H_n = \frac{H \cos \alpha}{2 \cos \theta} \left[ \frac{1}{\cos(\theta - \alpha)} + \frac{1}{\cos(\theta + \alpha)} \right] \]  

(3.5)

For normal spray impact to cover a given impact length \( L \), Eq. (3.5) is the required spray height, and the circular impact area is identical to the perpendicular cross section \( A_n \).

**3.3 Flow regions on spray-impacted surface**

Spray impacting a solid surface forms a liquid film flowing away from the impact area. If the flowing film is thin and has high velocity, the radially outward flowing film shows an abrupt increase of film thickness as shown in Fig. 3.4a. This is similar to the film flow formed by the impact of a liquid jet on a surface [89]. The phenomenon is referred to as hydraulic jump. Upstream from the hydraulic jump, the gravitational wave speed is lower than the flow velocity. As a result of the hydraulic jump, the flow velocity significantly drops and becomes lower than the gravitational wave speed. In case of normal spray impact as shown in Fig. 3.4a, the hydraulic jump is circular. Inclined spray impact results in non-circular hydraulic jump.

As discussed above, there are three flow regions on the surface: impact area, thin-film region, and thick-film region. The heat transfer performance in the three regions varies due to different flow dynamics. In case of normal impact, the three regions can be quantified by the impact length \( L \) and the hydraulic jump diameter \( D_j \) as defined in Fig. 3.4a. According to Eq. (3.5), the diameter of the circular impact area can be calculated using \( L = 2H_n \tan \alpha \). The hydraulic jump diameter \( D_j \) is measured for the normal spray impact with varied flow rates \( Q \) and spray heights \( H_n \), and the measurement in comparison with \( L \) are shown in Fig. 3.4b. The impact length slightly increases with increasing the flow rate due to the change of spray angle. The hydraulic diameter shows significant increase with increasing the flow rate. This indicates
that the higher the spray flow rate the stronger the film flow. Figure 3.4b also shows that $D_j$ can be enlarged by increasing $H_n$ due to the resulted increase of $L$.

**Figure 3.4** (a) Spray impact on a large surface forms three regions: impact area, thin-film region, and thick-film region. From the thin film region to the thick film region is the hydraulic jump. (b) The calculated impact lengths and measured hydraulic jump diameters for varied spray distances and flow rates.

### 3.4 Spray flux: inclined impact versus normal Impact

The impact area is where the spray flow impinges on the surface. Within the impact area, the volume flow rate per unit area is referred to as spray flux, i.e. $Q'' = dQ/dA$. As shown in Fig. 3.3, the overall spray flux must change with the inclination angle $\theta$ due to the change of the
impact area. To characterize the change of spray flux, one way is to compare the spray flux at a fixed \((x, y)\) location on the impact surface for different inclination angles. Here the local spray flux is associated with the location on the impact surface. The other way is to track a specific part of the spray flow and compare its fluxes on the impact surface for different inclination angles. Here the local spray flux is associated with the spray flow. For example, suppose we could repeatedly track a train of spray droplets that fly in the same trajectory and one by one land at the same location on the impact surface. As a result of changing the angle \(\theta\), the train of droplets would land at a different \((x, y)\) location, and the spray flux associated with the train of droplets would also change. In this section we will focus on the change of local spray flux associated with the spray flow as a result of inclination.

In the spray cone, the spray flow can be imagined as rays discharged from the nozzle orifice. In other words, it is assumed that spray droplets are constantly generated and maintain their trajectories until impacting the surface. Each ray can be specified by \((\beta, \varphi)\). As shown in Fig. 3.5, the polar angle \(\beta \in [0, \alpha]\) and the azimuthal angle \(\varphi \in [0, 2\pi]\). The cone forms a solid angle equal to \(2\pi(1 - \cos \alpha)\), within which the volume flow rate of the spray, \(Q\), distributes. We introduce a solid angle \(0 \leq \Omega \leq 2\pi(1 - \cos \alpha)\). Hence, the distribution of the flow rate with respect to solid angle is \(dQ/d\Omega\).

Based on visual observation of the spray and the measured temperature distribution in the impact area with normal spray impact, it is reasonable to assume that the full-cone nozzle generates axisymmetric sprays, i.e. \(\partial(dQ/d\Omega)/\partial \varphi = 0\). Additionally, the images in Fig. 3.2 show that close to the nozzle orifice the spray has fully atomized. Hence, \(dQ/d\Omega\) is independent
of the distance from the orifice. In summary, the flow rate distribution in the spray cone is a function of $\beta$ only, i.e. $dQ/d\Omega = f(\beta)$.

Figure 3.5 (a) A small element of the spray flow that is specified by $d\beta$, and $d\phi$. (b) The perpendicular cross-section $A_0$ that has been shown in Fig. 3.2c. (b) The impact area $A$ that has been shown in Fig. 3.2c.

Figure 3.5 shows part of the spray cone. Consistent with Figs. 3.2c and 3.3, the impact surface is on the $x$-$y$ plane, and the $x$-$z$ plane is the central plane of the spray cone. The perpendicular cross section shown in Fig. 3.5b is the cross section $A_0$ shown in Fig. 3.2c, and the azimuthal angle on the cross section is $\phi$. Hence, we can write

$$d\Omega = \sin \beta d\beta d\phi$$  \hspace{1cm} (3.6)

The flow rate within the solid angle $d\Omega$ is $dQ$. On the cross section $A_0$, as shown by Fig. 3.5b, the radius $r_0$ is given by
A differential area on this cross section is

\[ dA_0 = r_0 d\varphi \left( \frac{dr_0}{d\beta} \right) d\beta \]  

\hspace{1cm} (3.8)

Combining Eqs. (3.6-3.8) gives

\[ \frac{dA_0 \cos \beta}{(r_0 / \sin \beta)^2} = d\Omega \]  

\hspace{1cm} (3.9)

The spray flux on this cross section, \( Q_0'' = \frac{dQ}{dA_0} \), then can be expressed as

\[ Q_0'' = \left( \frac{dQ}{d\Omega} \right) \frac{\cos \beta}{(r_0 / \sin \beta)^2} \]  

\hspace{1cm} (3.10)

Next, we derive the spray flux in the impact area. As shown in Fig. 3.5c, the radius in the impact area is

\[ r = \frac{H \sin \beta}{\cos \theta \sin [\beta + \cos^{-1}(\sin \theta \cos \phi)]} \]  

\hspace{1cm} (3.11)

where the angle \( \phi \) is the azimuthal angle in the impact area. From Fig. 3.5, it can be shown that the azimuthal angles on the perpendicular cross section \( A_0 \) and impact area \( A \) satisfy

\[ \tan \phi = \cos \theta \tan \varphi \]  

\hspace{1cm} (3.12)

Corresponding to \( \varphi = 0 \) and \( \pi, \phi = 0 \) and \( \pi \), which is the centerline of the impact area as defined in Fig. 3.3. The flow specified by \((\beta, \varphi)\) in the spray cone lands on the impact surface at a location \((r, \phi)\), and the two locations are connected by Eqs. (3.11) and (3.12).

With \( \beta = \alpha \), Eq. (3.11) becomes
\[ R = r(\beta = \alpha) = \frac{H \sin \alpha}{\cos \theta \sin \left[ \alpha + \cos^{-1}(\sin \theta \cos \phi) \right]} \]

(3.13)

Eq. (3.13) is the equation used for plotting the impact areas in Fig. 3.3, where the center of the impact area is located at \((x, y) = (0,0)\). In the impact area, the relation between two coordinates systems \((r, \phi)\) and \((x, y)\) is

\[
\begin{align*}
x &= r \cos \phi + \frac{1}{2} \left[ R(\phi = \pi) - R(\phi = 0) \right] \\
y &= r \sin \phi
\end{align*}
\]

(3.14)

In the present work, the location \((x, y)\) determined from the recorded IR images can be converted to \((r, \phi)\) using Eq. (3.14), which can be further converted to \((\beta, \varphi)\) using Eqs. (3.11) and (3.12). The local cooling analysis that will be presented in Sections 3.5.1 and 3.6.1 focuses on the cooling along the centerline of the impact area (see Fig. 3.3). Here we use \(\chi\) to specifically denote the location along the centerline, which is

\[ \chi = x(\phi = 0, \pi) \]

(3.15)

Enclosed by the differential solid angle \(d\Omega\) same as in Eq. (3.9), the differential element of the impact area is

\[ dA = r d\phi dr = r \frac{d\phi}{d\varphi} d\varphi \frac{\partial r}{\partial \beta} d\beta \]

(3.16)

Combined with Eq. (3.6), Eq. (3.16) can be further written as

\[ dA = \frac{r}{\sin \beta} \frac{d\phi}{d\varphi} \frac{\partial r}{\partial \beta} d\Omega \]

(3.17)

The derivative \(\partial r / \partial \beta\) can be obtained from Eq. (3.11). From Eq. (3.12), it can be readily shown that
\[
\frac{d\phi}{d\varphi} = \frac{\cos^2 \theta \cos^2 \phi + \sin^2 \phi}{\cos \theta} = \frac{\cos \theta}{\cos^2 \theta \sin^2 \varphi + \cos^2 \varphi}
\] (3.18)

The flux in the impact area, \(Q'' = dQ/dA\), is

\[
Q'' = \left(\frac{dQ}{d\Omega}\right) \frac{r}{\sin \beta} \frac{d\phi}{d\varphi} \frac{\partial r}{\partial \beta}^{-1}
\] (3.19)

Based on Eqs. (3.10) and (3.19), we can write

\[
\frac{Q''}{Q_0''} = \frac{r_0^2}{r \sin \beta \cos \beta} \frac{d\phi}{d\varphi} \left(\frac{\partial r}{\partial \beta}\right)^{-1}
\] (3.20)

Equation (3.20) is the ratio of spray flux associated with the flow located at \((\beta, \varphi)\) in the spray cone, and the flow has different fluxes on the impact surface and the cross section \(A_0\).

In real applications of spray cooling a given surface, the impact length \(L\) usually is maintained constant for either normal impact or inclined impact. Hence, it is useful to compare the spray flux of inclined impact, \(Q''\), to that of normal impact, \(Q''(\theta = 0)\), when \(L\) remains unchanged. As shown in Fig. 3.2c, the impact area by normal impact, \(A(\theta = 0)\), is identical to the perpendicular cross section \(A_n\). Hence, their spray fluxes are also identical, i.e. \(Q''(\theta = 0) = Q_n'' = dQ/dA_n\). The comparison between the inclined impact and normal impact can be written as

\[
\frac{Q''}{Q''(\theta = 0)} = \frac{Q''}{Q_n''} = \frac{Q''}{Q_0''} \frac{Q_0''}{Q_n''}
\] (3.21)

It can be readily shown that

\[
\frac{Q_0''}{Q_n''} = \left(\frac{H_n}{H_0}\right)^2
\] (3.22)

Putting Eqs. (3.20) and (3.22) into Eq. (3.21) yields
\[
\frac{Q'}{Q^*(\theta = 0)} = \frac{r_0^2}{r \sin \beta \cos \beta} \frac{d \varphi}{d \beta} \left( \frac{\partial r}{\partial \beta} \right)^{-1} \left( \frac{H_n}{H_0} \right)^2
\]  

(3.23)

Substituting Eqs. (3.3), (3.4), (3.7), (3.11), and (3.18) into Eq. (3.23) gives

\[
\frac{Q'}{Q^*(\theta = 0)} = \frac{\cos^2 \alpha \cos \theta \sin^2 \left[ \beta + \cos^{-1} \left( \sin \theta \cos \phi \right) \right]}{4 \cos^4 \beta \left( \cos^2 \theta \cos^2 \phi + \sin^2 \phi \right)} \times
\]

\[
\frac{1}{1 - \tan \beta \cot \left[ \beta + \cos^{-1} \left( \sin \theta \cos \phi \right) \right]} \times
\]

\[
\left[ \frac{1}{\cos(\theta - \alpha)} + \frac{1}{\cos(\theta + \alpha)} \right]^2
\]

(3.24)

Combining Eqs. (3.24) and (3.12) gives \( Q''/Q''(\theta = 0) \) as a function of \( \theta, \beta, \) and \( \varphi \). The ratio of spray flux here shows the effect of inclination \( \theta \) on the spray flux of the flow located at \( (\beta, \varphi) \) in the spray cone.

### 3.5 Normal spray impact

Most spray cooling applications rely on normal spray impact \( (\theta = 0) \), for which the spray height \( H_n \) is the only positioning variable. By increasing or decreasing \( H_n \), the impact area can be increased or reduced. In this section experimental results of normal impact with varied values of \( H_n \) will be analyzed to investigate the local cooling and global cooling of the heater area. The objective here is twofold. The first is to investigate how the cooling performance changes with the spray height. The second is to determine the optimal spray height that provides the most effective cooling for a given flow rate.

#### 3.5.1 Local cooling by normal spray impact

A series of tests are conducted with a constant flow rate \( Q=5.0 \text{ cm}^3/\text{s} \) and spray height ranging from 5.1 to 27.4 mm. The temperature along the centerline of the heater area is used to
calculate the heat transfer coefficient using Eq. (3.1). The heat transfer coefficient along the
centerline, denoted by $h_\chi$, is plotted versus $\chi \in [-W/2, W/2]$ in Fig. 3.6, and $\chi = 0$ is the
concentric center of the heater and impact areas. The 2D symmetric profile of heat transfer
coefficient indicates axisymmetric distribution, and $\chi$ is the radial location for the axisymmetric
distribution.

![Figure 3.6 Local heat transfer coefficient along the center line of the heater area cooled
by normal spray impact with $Q=5.0$ cm$^3$/s.]

From the center outward, the heat transfer coefficient first increases and then decreases.
The distance between the two peaks is roughly equal to the impact length $L$ that can be
calculated using Eq. (3.5). In other words, the maximum local heat transfer coefficient appears at
the edge of the impact area. Hence, within the impact area $h_\chi$ increases radially. Outside the
impact area, i.e. in the thin-film region, $h_\chi$ decreases radially, as the flow in the thin-film region
slows down while flowing radially.
Figure 3.6 also shows that increasing the spray height reduces the cooling in the impact area. This can be explained by Eq. (3.22), which shows that the spray flux of normal impact is inversely proportional to \( H_n^2 \). As the spray height increases, the distribution of the heat transfer coefficient flattens. For \( H_n = 27.4 \) mm, along the entire centerline of the heater area, the spray has covered the heater area, but provides cooling less effective than all the other cases.

In addition to the distribution of local heat transfer coefficient shown in Fig. 3.6, it is also useful to evaluate the local area-averaged cooling. Due to the axisymmetric cooling of normal spray impact and the assumption of uniform heat flux, the area-averaged heat transfer coefficient is defined as

\[
\bar{h}_x = \left[ \frac{2}{\chi^2} \int_0^\chi \frac{1}{h_x} \chi d\chi \right]^{-1}
\]  

(3.25)

Here \( \bar{h}_x \) is the average heat transfer coefficient for the circular area with radius \( \chi \). To calculate \( \bar{h}_x \) based on the data shown in Fig. 3.6, we define and use a discrete form of Eq. (3.25), which is

\[
\bar{h}_x = h_{x_i}
\]

\[
\bar{h}_x = \left[ \frac{2}{\chi^2} \sum_{j=2}^{i-1} \frac{1}{h_{x_j}} \left( \frac{X_{j-1} + X_j}{2} \right) \Delta \chi \right]^{-1} \quad (i \geq 2)
\]  

(3.26)

Here \( \chi_1 = 0 \) and \( \chi_i = (i - 1) \Delta \chi \) are the measurement locations shown in Fig. 3.6. The interval \( \Delta \chi = 0.4 \) mm is determined by the resolution of the IR camera.

Equation (3.26) is used to calculate the area-averaged heat transfer for the test with \( H_n = 5.1 \) mm shown in Fig. 3.6. The local heat transfer coefficient \( h_x \) and average heat transfer coefficient \( \bar{h}_x \) are plotted in Fig. 3.7. Similar to the trend of \( h_x \), \( \bar{h}_x \) first increases and then...
decreases. However, the location for maximum $\bar{h}_x$ is after the location for maximum $h_x$ by a gap distance $\delta$. Therefore, for normal impact, the optimal area for achieving the most effective cooling is larger than the impact area. This implies that to cool a given surface effectively, the spray should cover less than the actual heater area. This is in contradiction to the previous study by Mudawar and Estes [7].

![Graph showing heat transfer coefficients](image)

Figure 3.7 The local heat transfer coefficient $h_x$ and local area-averaged heat transfer coefficient $\bar{h}_x$ of normal spray impact with $H_n = 5.1$ mm and $Q = 5.0$ cm$^3$/s. The radial gap distance, $\delta$, is between the edges of the circular impact area and the circular area with maximum average cooling.

### 3.5.2 Global cooling by normal spray impact

The discussion above indicates that to cool a given area effectively, the impact area should be smaller than the heater area. Since the impact area for normal impact is solely dependent on the spray height, a series of tests are conducted to investigate the relation of global cooling performance to the spray height for varied flow rates. The global cooling performance is evaluated using an average heat transfer coefficient
\[
\bar{h} = \frac{q^*}{\bar{T}_s - T_i}
\]

(3.27)

where \(\bar{T}_s\) is the average of IR image of the heater area (~ 3100 data points of \(T_s\)).

Figure 3.8 shows the average heat transfer coefficient versus the spray height for six different flow rates. Apparently, the higher the flow rate, the higher the heat transfer coefficient. However, this trend diminishes for large spray heights close to \(H_n=28\) mm when the impact area is already larger than the heater area.

![Heat transfer coefficient averaged over the heater area, \(\bar{h}\), versus the spray distance, \(H_n\), for varied flow rates, \(Q\).](image)

For each flow rate, the peak point of the data curve shows the optimal spray distance, \(H_{n,max}\), for achieving the maximum average heat transfer coefficient, \(\bar{h}_{max}\).

For each given flow rate, with increasing the spray height, the global cooling first increases and then decreases. Hence, for each given flow rate, there is maximum cooling, denoted by \(\bar{h}_{max}\), and the corresponding spray height, \(H_{n,max}\), is the optimal spray height for cooling the given surface with the given flow rate. In Fig. 3.8, the optimal points are connected by a dashed line, which shows two trends: 1) \(\bar{h}_{max}\) decreases with decreasing \(Q\); 2) \(H_{max}\) increases with decreasing \(Q\).
To further investigate the second trend, in addition to the six flow rates tested in Fig. 3.8, the optimal spray height $H_{n,\text{max}}$ is determined for another four flow rates. All the data are presented in Fig. 3.9, which shows the optimal heights versus flow rates.

![Figure 3.9 The optimal spray distance of normal impact versus the flow rate for cooling the heater area. In the inset graph, the radial gap distance $\delta$ (defined in Fig. 3.7) shows the difference between the heater area and the optimal impact area. The error bar is the difference between the determined $H_{n,\text{max}}$ and its two neighboring tested spray heights. For all the cases, $L < W$. This indicates that for achieving the optimal cooling, the spray needs to cover less area than the heater area. Although the heater area is square rather than circular, the gap distance between the edges of the impact area and the heater area (see the inset of Fig. 3.9) can be approximated as the gap distance, $\delta$, defined in Fig. 3.7. Hence, the gap distance can be calculated using $\delta = (W - L)/2$. In summary, Fig. 3.9 shows that to achieve the most effective cooling from an increased flow rate, the nozzle needs to be moved closer to the surface ($H_{n,\text{max}}$ decreases) to reduce the impact area ($\delta$ increases).
3.6 Inclined spray impact

For inclined spray impact, there are two positioning variables: \( H \) and \( \theta \). Changing either variable would cause both the impact area \( A \) and impact length \( L \) to change. The focus of our study is on the effect of spray inclination on cooling performance. The experimental study is carried out by changing the angle \( \theta \) and maintaining a constant impact length equal to the width of the heater area, i.e. \( L = W = 23.5 \text{ mm} \). As a result, the spray height \( H \) becomes a dependent variable, and it can be calculated using Eq. (3.2). As shown by Fig. 3.3, increasing \( \theta \) while maintaining \( L \) constant reduces the impact area \( A \). As a result, the overall spray flux, \( Q/A \), increases.

3.6.1 Global cooling by inclined spray impact

Figure 3.10 shows temperature distribution of inclined spray impact with inclination angles \( \theta = 20^\circ, 30^\circ \) and \( 40^\circ \) with \( Q = 5.0 \text{ cm}^3/\text{s} \). Consistent with Figs. 3.2 & 3.5, the inclination of spray is done by rotating the nozzle clockwise. The ellipses in dashed lines show the impact area. For \( \theta = 20^\circ \), the temperature contour is close to being axisymmetric, but the right side is generally cooler than the left side. As \( \theta \) further increases, the cooler region on the right side continues to expand, and the temperature around the center decreases.

The temperature in the heater area is averaged to obtain \( \bar{T}_s \), and the average heat transfer coefficient \( \bar{h} \) is calculated using Eq. (3.27) and plotted in Fig. 3.10d. The effect of inclination on global cooling can be evaluated by comparing with the normal impact (\( \theta = 0^\circ \)), for which \( \bar{h} = 2.35 \text{ W/cm}^2\cdot\text{K} \). The global cooling for \( \theta = 20^\circ \) is \( \bar{h} = 2.22 \text{ W/cm}^2\cdot\text{K} \), showing negative effect of inclination. However, an opposite trend appears for further increased inclination angles \( \theta = 30^\circ \) and \( 40^\circ \), which result in \( \bar{h} = 2.79 \) and \( 3.01 \text{ W/cm}^2\cdot\text{K} \), respectively.
3.6.2 Local cooling by inclined spray impact

To better understand the effect of inclination on global cooling, we look into the local cooling. To simplify the analysis, we focus on the local cooling along the centerline of the impact area. The local heat transfer coefficient along the centerline $h_\chi$ is calculated using Eq. (3.1) and is plotted in Fig. 3.11a for the normal impact ($\theta = 0^\circ$) and inclined impact ($\theta = 20^\circ, 30^\circ, 40^\circ$). For the normal impact, the distribution of $h_\chi$ is symmetric with respect to $\chi = 0$, the center of the impact area. The inclination causes $h_\chi$ to increase on the right side ($\chi > 0$) and decrease on the left side ($\chi < 0$).
Figure 3.11 (a) Local heat transfer coefficient associated with the $\chi$ location along the center line for $\theta = 0^\circ, 20^\circ, 30^\circ, 40^\circ$. The temperature contours have been shown in Fig. 3.10 except for $\theta = 0^\circ$. (b) The change of local cooling due to inclination is quantified by comparing the inclined impact to the normal impact.

Since the temperature is measured at the same $\chi$ locations for all the cases, we can compare the local cooling of inclined impact with that of the normal impact. The comparison, $h_\chi(h_\chi(0^\circ))$, is plotted in Fig. 11b. For $\theta = 20^\circ$, slight enhancement appears only on the right side, whereas the left side shows diminished cooling. For $\theta = 30^\circ$ and $40^\circ$, cooling
enhancement increases and extends to the left side. Nevertheless, cooling diminishment still exists on the left side, the far end from the spray nozzle.

The inclination affects local cooling in two mechanisms. One is related to the velocity direction of spray droplets. The velocity vector can be decomposed into a vertical component perpendicular to the surface and a horizontal component parallel to the surface. The vertical component determines the impact momentum of spray droplets, while the horizontal component affects the velocity of the flow on the surface. Increasing or decreasing either velocity component could enhance or diminish the cooling performance. The inclination must cause one velocity component to increase and the other one to decrease. However, it is difficult to determine the net effect on cooling.

The other mechanism is related to local spray flux. The increase or decrease of local spray flux is expected to enhance or diminish local cooling. The analysis in this section is to relate the change of local spray flux to the change of local cooling. The change of local cooling associated with \( \chi \) location has been shown in Fig. 3.11b. However, the change of local spray flux associated with \( \chi \) location is unknown for the following reason. According to Eq. (3.15), as a result of changing \( \theta \), the spray flow impacting any \( \chi \) location is replaced by new flow associated with a new value of \( \beta \). The spray flow distribution \( dQ/d\Omega = f(\beta) \) is unknown.

The change of spray flux associated with \((\beta, \varphi)\) has been derived in Section-3.4. The flow impacting the centerline of the impact area has \( \beta \in [0, \alpha] \) and \( \varphi = 0 \) and \( \pi \). We will try to obtain the change of local cooling associated with the flow impacting the centerline. First, we convert the local heat transfer coefficient associated with \( \chi, h_\chi \), to the local heat transfer coefficient
associated with $\beta, h_\beta$, by converting the $\chi$ locations in Fig. 3.11a to $\beta$ locations. Here we will derive the equation that relates $\chi$ to $\beta$.

Equation (3.15) relates $\chi$ to $\beta$ for given $H$ and $\theta$, and is composed of $\chi = x(\phi = 0)$ and $\chi = x(\phi = \pi)$ for $0 \leq \beta \leq \alpha$. From now on, if we define $-\alpha \leq \beta \leq \alpha$, Eq. (15) then becomes $\chi = x(\phi = 0)$, which can be fully written as

$$\chi = \frac{1}{2} H \sin \alpha \left[ \frac{1}{\cos(\theta + \alpha)} - \frac{1}{\cos(\theta - \alpha)} \right] + \frac{H \sin \beta}{\cos \theta \cos(\theta - \beta)}$$

(3.28)

Here the angle $-\alpha \leq \beta \leq \alpha$ is on the central plane of the spray cone, which is orthogonal to the impact surface. Combining Eq. (3.2) with Eq. (3.28) gives

$$\chi = \frac{1}{\cos(\theta + \alpha)} - \frac{1}{\cos(\theta - \alpha)} + \frac{2 \sin \beta}{\cos(\theta - \beta)}$$

(3.29)

Equation (3.29) is used to convert the $\chi$ location to $\beta$ location, and the data of heat transfer coefficient in Fig. 3.11a, now denoted by $h_\beta$, are plotted versus $\beta$ in Fig. 3.12. The goal here is to analyze the change of $h_\beta$ due to inclination by comparing the cooling of inclined impact to that of normal impact, i.e. $h_\beta/h_\beta(\theta = 0^\circ)$. However, the ratio cannot be calculated at this moment as in Fig. 3.12 the four cases have different $\beta$ locations. To address this issue, linear interpolation is carried out based on the data shown in Fig. 3.12 to calculate $h_\beta$ for unified $\beta$ locations for all the cases. The interpolated data are presented in Fig. 3.13a, where the data
curves show no visible difference from those in Fig. 3.12. Based on Fig. 3.13a, the ratio $h_\beta / h_\beta(\theta = 0^\circ)$ is calculated and plotted in Fig. 3.13b.

Figure 3.12 The local heat transfer coefficient shown in Fig. 3.11a is plotted versus the $\beta$ location, and $h_\beta$ denotes the local heat transfer coefficient associated with the $\beta$ location.

Figure 3.13b shows the effect of inclination on the cooling performance of the flow, which distributes on the central plane of the spray. The trends for the three cases of inclined impact are similar. Generally, from $\beta = \alpha$ to $\beta = -\alpha$, the cooling enhancement first increases and then decreases. The enhancement peaks are within the angular space $\beta > 0^\circ$, and the peak shifts slightly toward the left as $\theta$ increases. The flow close to $\beta = -\alpha$ shows diminished cooling as a result of inclination. The incline impact with $\theta = 40^\circ$ provides the highest cooling enhancement for most of the central plane flow.

In Section-3.4, we have derived the change of local spray flux associated with the flow at $(\beta, \varphi)$ as a function of $\theta$. The comparison of spray flux $Q_\beta'' / Q_\beta''(\theta = 0)$ along the centerline can be evaluated using Eq. (3.24) with input $\phi = 0, \pi$. Similar to the derivation of Eq. (3.28),
\[ \frac{Q''_\beta}{Q'_\beta}(\theta = 0) \] \( \phi = 0, \pi \) can be changed to \( \frac{Q''_\beta}{Q'_\beta}(\theta = 0) \) with \( -\alpha \leq \beta \leq \alpha \), which can be written as

\[ \left[ \frac{Q''_\beta}{Q'_\beta}(\theta = 0) \right]_{\phi = 0} = \left[ \frac{\cos \alpha \cos (\theta - \beta)}{2 \cos^2 \beta} \right]^2 \left[ \frac{1}{\cos(\theta + \alpha)} + \frac{1}{\cos(\theta - \alpha)} \right]^2 \cos \theta \left[ 1 - \tan \beta \tan (\theta - \beta) \right] \]  

(3.30)

Eq. (3.30) is plotted in Fig. 3.14 for the three cases of incline spray impact.

![Figure 3.13](image)

**Figure 3.13** (a) By applying interpolation to Fig. 3.12, \( h_\beta \) is plotted versus the same \( \beta \) locations for all the cases of normal impact and inclined impact. (b) The ratio of local cooling of inclined impact to normal impact calculated from the interpolated data.
Now we have the change of local cooling shown in Fig. 3.13b and change of local spray flux shown in Fig. 3.14, both associated with the flow at $\beta$ location on the central plane of the spray cone. Comparing Figs. 3.14 and 3.13b shows the following similarities between the local spray flux and the local cooling.

![Figure 3.14](image)

Figure 3.14 Corresponding to the local cooling analysis shown in Fig. 3.13, the ratio of local spray flux of inclined impact to normal impact is calculated using Eq. (3.30).

1) The inclination increases the spray flux for most of the central-plane flow, except for the flow close to $\beta = -\alpha$ where the spray flux decreases. The angular zones of $\beta$ for the increase and decrease of spray flux (Fig. 3.14) are similar to the enhancement and diminishment zones of cooling (see Fig. 3.13d).

2) For the three cases, the larger the inclination angle, the more increase of spray flux (Fig. 3.14), and the more cooling enhancement (Fig. 3.13d) showing a significant dependence on $\theta$.

3) Generally, spray flux decreases and cooling enhancement lessens as $\beta$ changes from $\alpha$ to $-\alpha$. However, the flow at $\beta = \alpha$ shows the maximum enhancement of spray flux, while the
maximum cooling enhancement appears at $\beta < \alpha$. The difference could be related to the change of velocity components, which is out of the scope of the present work.

Depending on the location $(\beta, \varphi)$ of the flow, the cooling performance of the flow can be enhanced or diminished by the inclination. One major reason is that the flux of the flow increases or decreases as a result of the inclination. Overall, Figs. 3.13d and 3.14 indicate the correlation between the change of local spray flux and the change of local cooling.

### 3.7 Summary

The spray nozzle can be put closer to or farther from the impact surface to change the spray height, and it can be tilted to change the spray inclination. The spray height and inclination angle are the positioning variables tested in the present work. Spray impact is observed to form three regions on the impact surface: impact area, thin-film region, and thick-film region. For given spray height and inclination angle, the flow landing at a location $(x, y)$ in the impact area is located at $(\beta, \varphi)$ in the spray cone. As a result of changing the nozzle positioning, the flow impacts a different $(x, y)$ location, and its spray flux also changes. The relation between $(x, y)$ and $(\beta, \varphi)$ is derived. Equations are also derived for evaluating the change of spray flux as a function of the nozzle positioning and $(\beta, \varphi)$.

Experimental tests of normal spray impact are conducted to investigate the effect of spray height. It is discovered that the optimal spray height providing the most effective cooling is smaller than the height required for covering the entire heater area. This is because the maximum local cooling is located at the edge of the impact area, and the film flow outside the impact area still provides effective cooling. This is in contradiction to the previous study by Mudawar and Estes [7]. It is found that the optimal height decreases with increasing the flow rate.
The effect of inclination is studied by changing the inclination angle while maintaining the impact length constant and equal to the width of the heater area. The global cooling shows slight diminishment for small inclination angle and enhancement for large inclination angles. Based on the local cooling along the centerline of the impact area, the cooling performance of the flow on the central plane of the spray cone is tracked. The spray flux of the flow on the central plane is analyzed. The enhancement and diminishment of the cooling performance are found to be in general agreement with the increase and decrease of the spray flux.
Chapter 4 Drop Impact on a Flowing Film

4.1 Background

Despite that a number of previous studies were conducted on drop impact, limited research has been done on the drop impact on liquid films, especially on flowing films. The following three aspects need to be addressed: 1) There is lack of experimental validation for the analytical models of drop spreading on liquid films; 2) The understanding of the flow dynamics of the crown sheet in relation to splash is still limited; 3) There is no threshold parameter of splash defined for drop impact on flowing films. The present work aims to address the three issues by theoretically and experimentally investigating drop spreading on stationary and flowing films, stretching of the crown sheet formed from flowing films, and splash mechanism on flowing films.

The main motivation of the present work is to understand the interaction of drops with flowing liquid films, which is common for flow configurations such as spray impingement on solid surfaces. Spray impingement is related to many applications such as painting and spray cooling. Two reasons are proposed for simplifying spray impingement to drop impingement on flowing film. First, the liquid film formed by spray impingement is a radial flow. This is one major reason that, in our experimental study, jet impingement on a solid surface is used to produce radially flowing liquid film. More discussion on the flowing film is in the section of experimental methodology. Second, spray impingement is composed of many drop impact on film flow, which is formed by the numerous drop flow. The fluid dynamics study of drop impacting flowing film enables us to have a better understanding of thermal results of drop impact on the film-cooled hot surface in the following Sections 5 & 6, and further understand spray cooling performance.
4.2 Experimental methodology

The test setup is schematically shown in Fig. 4.1(a). The camera (M310, Vision Research, Wayne, NJ, USA) placed underneath is used when transparent glass substrates are used, providing clear and accurate observation of drop spreading. The other one views impact process from the side and provides impact velocity and sheet dynamics. The cameras are operated with a frame rate 5000 fps (frames per second) and shutter speed 30 μs. Two LED lamps serves as illumination sources, and two diffusers are positioned in front of the lamps to produce uniform lighting.

![Experimental setup diagram](image)

Figure 4.1 (a) Schematic of the experimental setup; (b) A drop impacts a flowing film generated by jet impingement (Details of velocity profile in boundary layer are not shown.); (c) Top view of drop spread on the film with a distance $\chi$ from the jet impingement stagnation (Details of film streamlines affected by the drop impact are not shown.)
Filtered water and glycerin-water solutions with varying concentrations are used as working fluids. The fluid properties are provided in Table 4.1. For each test, the film fluid and the drop fluid are always the same. Drops are generated from stainless steel capillary tubes connected to a syringe pump. The plunger advances very slowly to form a drop at the tip of the tube, which finally detaches from the tube under its own weight. The drop diameter is changed by using tubes with different sizes, and for all the fluids the drop diameter varies from 2.6 to 4.6 mm. The height of the tube orifice above the test surface is adjusted to change the impact velocity, which ranges from 0.63 to 4.2 m/s.

Two groups of tests are conducted in the present work: one group with stationary films, the other group with flowing films. For the first group of tests, a drop lands at the center of a circular stationary liquid film. A transparent glass plate of 60 mm in diameter is placed on the top of a flat circular container with a glass bottom of 100 mm in diameter, as shown in Fig. 4.1(a). The stationary film is generated by adding liquid into the container until liquid submerges the transparent glass. The film thickness is measured by using a height gauge and ranged from 300 to 2000 μm. The second group of tests is carried out on flowing films formed by jet impinging on smooth circular plates (see Fig. 4.1(b)) with 60 mm in diameter. A large jet nozzle with radius $a = 1.91$ mm is used to avoid hydraulic jump on the surface. The volumetric flow rate of the jet $\dot{V}$ ranging from 20 to 38 cm$^3$/s is measured by a flowmeter.

The second group of tests is carried out on flowing films formed by jet impinging on smooth circular plates (see Fig. 4.1(b)) with 60 mm in diameter. To have a smooth steady flow, the liquid jet is produced by supplying fluid from a pressurized tank. A large jet nozzle with radius $a = 1.91$ mm is used to avoid hydraulic jump on the surface. The volumetric flow rate of
the jet \( \dot{V} \) ranging from 20 to 38 cm\(^3\)/s is measured by a flowmeter. Using jet impingement flow as the flowing film is due to the motivation of our study in relation to spray impingement. Another major reason is that it is easy to produce smooth and well-controlled flowing films. However, this methodology also has its own limitations which will be discussed below.

Depending on where the drop impacts the film, the local film thickness and fluid velocity need to be known. For the present work, Watson’s theory [100] for the liquid film of jet impingement was used. The theory has been validated by Craik et al. [101] in good agreement with experimental results. The theoretical prediction for velocity profiles and the film thickness was further validated by Azuma and Hoshino [102-103] using laser-Doppler measurements. Watson’s theory for the liquid film prior to hydraulic jump can be summarized as below. The distance from the stagnation point of the jet impingement is denoted by \( \chi \). The local film thickness is given by

\[
\begin{align*}
    h_f &= \frac{a^2}{2\chi} + 1.7668 \sqrt{\frac{\nu \chi a^2}{\dot{V}}} & \text{for } \chi \leq 0.3155 \left( \frac{Va^2}{\nu} \right)^{\frac{1}{3}} \\
    h_f &= \frac{2\pi^2 \nu (\chi^3 + 0.1823Va^2/\nu)}{3\sqrt{3} \chi \dot{V}} & \text{for } \chi > 0.3155 \left( \frac{Va^2}{\nu} \right)^{\frac{1}{3}}
\end{align*}
\]

(4.1)

The local mean velocity is

\[
U_f = \frac{\dot{V}}{2\pi\chi h_f}
\]

(4.2)

According to Eqs. (4.1) and (4.2), local film thickness and mean velocity can be changed by varying the landing location \( \chi \) and flow rate \( \dot{V} \).

As shown in Fig. 4.1(c), the flow condition of the film is different around the circumference of the drop during impact. Hence, an azimuthal angle \( \phi \in [0^\circ, 360^\circ] \) is defined.
The position $\varphi = 180^\circ$ is where the drop flow is right opposite to the film flow, while the position $\varphi = 0^\circ$ is where the drop flow is in the same direction as the film flow. Figure 4.1(c) also indicates that during impact, the center of the drop flow will move, and the circumferential edge of the drop will move too.

Table 4.1 Properties of the water and glycerin-water solutions at 25°C and atmosphere pressure.

<table>
<thead>
<tr>
<th>Fluid (wt.% glycerin-water)</th>
<th>Dynamic viscosity (mPa·s)</th>
<th>Density (kg/m$^3$)</th>
<th>Surface tension (mN/m)</th>
</tr>
</thead>
<tbody>
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<td>0</td>
<td>1</td>
<td>1000</td>
<td>71.4</td>
</tr>
<tr>
<td>20</td>
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<td>1052.2</td>
<td>70.5</td>
</tr>
<tr>
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<td>3.28</td>
<td>1104.4</td>
<td>68.9</td>
</tr>
<tr>
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<td>5.11</td>
<td>1130.5</td>
<td>68.1</td>
</tr>
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</tr>
<tr>
<td>80</td>
<td>46.47</td>
<td>1208.8</td>
<td>65.7</td>
</tr>
</tbody>
</table>

For the analysis in the present work, the local values of film thickness and velocity at the landing location are used as constants for the impact process. The distance from the stagnation point of the jet impingement is denoted by $\chi$. In our tests, drops are deposited at locations where $\chi/a \gg \sim 1$. According to Watson’s theory [100], film thickness and average velocity are plotted in Figure 4.2 for water with three flow rates. Figure 4.2 shows that the film thickness remains relatively constant for $\chi > \sim 10$ mm. As a result, the mean film velocity decreases almost linearly with the inverse of $\chi$.

As shown by Fig. 2.1(c), the flow condition of the film is different around the circumference of the drop during impact. Hence, an azimuthal angle $\varphi \in [0^\circ, 360^\circ]$ is defined. The position $\varphi = 180^\circ$ is where the drop flow is right opposite to the film flow, while the position $\varphi = 0^\circ$ is where the drop flow is in the same direction as the film flow. The film flow is
in radial direction, whereas the flow is assumed to be unidirectional in our analysis. Since drops are deposited at locations $\chi \gg a$, even at the end of the impact process considered in our analysis, the angle subtended by the spread drop with respect to the jet impingement point is less than 40° (see Fig. 4.1(c)), showing a maximum difference of 20° from unidirectional flow. This could cause discrepancy between the analysis and experiment at locations around $\phi \sim 90°$ and $\phi \sim 270°$. But less effect is expected for the flow dynamics around $\phi \sim 180°$. For simplicity of our analysis, unidirectional flow is assumed. The assumption is justified by the agreement of our analysis with experimental results.

Figure 4.2 The radial distributions of velocity and thickness of flowing film generated by jet impingement with varied flow rates.
4.2 Drop spreading

4.2.1 On stationary liquid films

This section aims to derive an equation that describes the spreading of a single drop after impacting a stationary liquid film. The analysis is based on the model illustrated in Fig. 4.3, which shows two phases of drop impact: deformation phase (Fig. 4.3a to b) followed by spreading phase (Fig. 4.3 b to c). At time \( t = 0 \), a single drop with diameter \( D_0 \) and velocity \( U_0 \) lands on a stationary film with thickness \( h_f \) (see Fig. 4.3a).

![Diagram showing drop impact](image)

Figure 4.3 The impact of a single drop onto a stationary film is composed of two phases. (a) – (b): deformation phase; (b) – (c): spreading phase.

To simplify the complicated impact process, it is assumed that the drop quickly deforms to a disc of radius \( r_i \) and thickness \( h_f \). The time taken for the drop to deform to the disc is \( t_i \). Drop volume conservation requires
where $\tilde{r}_i$ and $\tilde{h}_f$ have been non-dimensionalized by the drop diameter $D_0$.

Starting from $\tilde{r}_i$, the drop disc continues to spread, and the spreading radius is $\tilde{r}_c$. Hence, the initial condition for the spreading process is

$$\tilde{r}_c (\tilde{T} = \tilde{t}) = \tilde{r}_i$$

where the non-dimensional time $\tilde{T} = t U_0 / D_0$. A uniform velocity $u_i$ is assumed for the disc, which, normalized by drop impact velocity $U_0$, is

$$\bar{u}_i = \lambda$$

If the deformation results in energy loss from the drop, $\lambda < 1$, and if there is no energy loss, $\lambda = 1$. Hence, $\lambda$ is also referred to as energy loss factor. The time taken to complete the deformation phase can be roughly estimated by

$$\tilde{T}_i = (\tilde{r}_i - \frac{1}{2}) / \lambda$$

The velocity distribution at $\tilde{T} = \tilde{t}_i$ is shown in Fig. 4.3(b), which is the initial state of the spreading phase. At $\tilde{T} = \tilde{r}_i$ there is a step change of velocity from $\lambda$ to 0. For the incompressible liquid layer, the discontinuity in the velocity distribution produces an outflowing thin liquid sheet along the discontinuity front, which serves as a sink for both mass and momentum. This thin liquid sheet is referred to as a crown sheet, and the disc formed by the drop is the base of the crown. As illustrated in Fig. 4.3(c), the spreading of the drop causes the kinematic discontinuity to propagate, which is viewed as the expansion of the crown. During the spreading phase, the velocity and thickness of the drop disc at $\tilde{r}_c$, denoted by $\bar{u}_t$ and $\bar{h}_t$, are decreasing. However, the
thickness of the liquid film $\tilde{h}_f$ remains constant, and its velocity remains zero. Thus, the mass flow rate into the crown

$$\tilde{h}_i \left( \vec{u}_i - \vec{r}_c^i \right) + \tilde{h}_f \vec{r}_c^i = \tilde{Q}$$

(4.7)

where $\vec{r}_c^i = d\vec{r}_c / d\tilde{r}$. The flow rate $\tilde{Q}$ has been non-dimensionalized by $U_0D_0$. Carried by the flow rate $\tilde{Q}$, flow momentum is also transferred from the drop disc to the crown. To determine the resultant velocity for $\tilde{Q}$, we consider the collision between a moving volume $\tilde{h}_t$ with a velocity $\vec{u}_t$ and a stationary volume $\tilde{h}_f$, which results in one combined volume $(\tilde{h}_t + \tilde{h}_f)$ moving at one single velocity. Hence the resultant velocity is $\tilde{h}_t \vec{u}_t / (\tilde{h}_t + \tilde{h}_f)$. To satisfy momentum conservation, the momentum transfer of the discontinuity is

$$\tilde{h}_i \left( \vec{u}_i - \vec{r}_c^i \right) \vec{u}_i = \tilde{Q} \frac{\tilde{h}_t \vec{u}_t}{\tilde{h}_t + \tilde{h}_f}$$

(4.8)

For which, surface tension and viscous forces are neglected. From Eqs. (4.7) and (4.8), we obtain

$$\vec{r}_c^i = \frac{1}{2} \vec{u}_i$$

(4.9)

The initial velocity distribution ($\vec{u}_i$ for $\tilde{r}_i > \tilde{r} > 0$, and zero for $\tilde{r} > \tilde{r}_i$) can be considered as a single hump problem, for which the solution of $\vec{u}_t$ is given by

$$\vec{u}_t = \left( \frac{2\tilde{u}_t \tilde{r}}{\tilde{r}} \right)^{1/2}$$

(4.10)

Substituting Eq. (4.10) into Eq. (4.9) and then integrating Eq. (4.9), we have

$$\tau_c = \beta_c \sqrt{\tilde{r}} + C$$

(4.11)

where $C$ is a constant to be determined. The coefficient $\beta_c$ is
\[ \beta_c = \left( \frac{2\lambda^2}{3h_f} \right)^{1/4} \]  

(4.12)

where the Eqs. (4.3) and (4.5) have been used. The initial condition for Eq. (4.11) has been given by Eqs. (4.4) and (4.6). The constant \( C \) can be obtained by applying the initial condition to Eq. (4.11). Putting the obtained expression of \( C \) back into Eq. (4.11) gives

\[ \bar{r}_c = \beta_c \sqrt{\bar{t}} + \frac{1}{\sqrt{6h_f}} - \left( \frac{1}{3h_f} - \frac{1}{\sqrt{6h_f}} \right)^{1/2} \]  

(4.11)

In the following discussion, Eq. (4.13) is referred to as the present model. Similar to the constant \( C \) in Eq. (4.11), Roisman and Tropea [61] introduced a time shift \( \bar{t}_0 \) in their theory, which takes the form of

\[ \bar{r}_c = \beta_c \sqrt{\bar{t}} + \bar{t}_0 \]  

(4.14)

The time shift can be determined by applying Eqs. (4.4) and (4.6), which results in

\[ \bar{t}_0 = \frac{1}{2\lambda} \left( 1 - \frac{1}{\sqrt{6h_f}} \right) \]  

(4.15)

It should be noted that the expressions of \( \beta_c \) and \( \bar{t}_0 \) here are different from those in Roisman and Tropea’s original theory, because their initial conditions are different from Eqs. (4.4) through (4.6). Due to the modification, in the following discussion Eqs. (4.14) and (4.15) are referred to as the modified R&T model.

To validate the above analysis for the drop impact on stationary films, a number of experiments have been conducted by varying the film thickness, drop diameter and impact velocity, and working fluid. The time evolution of the crown radius is measured using the images taken by the camera placed underneath. An example is presented in Fig. 4.4 (a) showing the impact of a drop on a stationary film. The inner dark circle is the crown base, and its radius is
taken as the crown radius. The outer circle is the top rim of the rising crown sheet. As observed from most of the tests, the fast expansion of the crown base and the quick rise of the crown sheet took place when $\bar{t} < \sim 3$, which, therefore, is considered as the early stage of spreading.

![Crown radius measurement](image)

Figure 4.4 Crown radius measurement by viewing the impact underneath the impact surface. (a) stationary film ($h_f = 530 \mu m, U_0 = 2.05 m/s, D_0 = 4.35 mm$), (b) flowing film ($\dot{V} = 1.56 L/M, U_0 = 3.54 m/s, D_0 = 4.09 mm$). The liquid used were 40% glycerin-water solution.

The present model and modified R&T model are compared and applied to fit with one test in Fig. 4.5 (a). The method of least squares is used to determine the value of the energy loss factor, which is denoted by $\lambda_s$ for the drop impact on stationary films. The two models provide almost the same fitting with the experimental data, and the values of $\lambda_s$ are slightly different. To show the significance of $\lambda_s$, the present model is also plotted in Fig. 4.5 (a) for $\lambda_s = 1$. It is clear that assuming no change of drop velocity will cause overestimation of drop spreading.
Major causes for $\lambda_s < 1$ include viscous dissipation, transformation of kinetic energy to surface energy, and transfer of kinetic energy from the drop flow to the film fluid. To further investigate the energy loss factor, the present model is applied to all the tests to obtain the values of $\lambda_s$. Figure 4.5(b) shows the fitting of the present model with three tests with the same drop condition but varied thicknesses of the stationary film. Based on the three tests, $\lambda_s$ decreases with increasing the film thickness. Figure 4.5(c) shows the fitting of the present model with another three tests, which have the same film thickness, $h_f$, but use three different working fluids. Therefore, the Reynolds and Weber numbers of the drop have been used to differentiate the tests. Clearly, the value of $\lambda_s$ varies with the fluid.

Major parameters involved in the drop impact on a stationary film include $h_f$, $\mu$, $\rho$, $\sigma$, $D_0$, and $U_0$. The six parameters contain three dimensions. Therefore, $\lambda_s$ is expected to correlate to three non-dimensional parameters, which are $\bar{h}_f$, Re and We. Applying power law, we have the correlation given by

$$
\lambda_s = \frac{0.26}{Re^{-0.05} We^{0.07} \bar{h}_f^{0.34}}
$$

Equation (4.16) and experimental data points are plotted in Fig. 4.6. The correlation indicates that the viscosity reduces $\lambda_s$ while the surface tension increases $\lambda_s$. However, the influences of Re and We are weak as compared to that of $\bar{h}_f$. A small drop impacting a deep pool will lose kinetic energy completely. Equation (4.13) combined with Eq. (4.16) could predict the crown expansion on stationary film at the spreading stage. It should be noted that Eq. (4.16) does not satisfy the constraint $0 < \lambda_s \leq 1$. Therefore, the correlation is not valid for the entire ranges of the non-dimensional parameters.
Figure 4.5 Spreading of single drops impacting stationary films: (a) The present model and the modified R&T model are fitted with one test ($U_0=2.86$ m/s, $D_0=4.29$ mm, 50% glycerin-water solution); (b) Three tests with the same drop condition ($U_0=2.86$ m/s, $D_0=4.29$mm) but varied film thicknesses using 50% glycerin-water solution; (c) Three tests with the same film thickness ($h_f = 750$ μm) with varied drop Reynolds and Weber numbers (■ 40% glycerin-water solution, ○ 50% glycerin-water solution, Δ 80% glycerin-water solution).
The energy loss factor $\lambda_s$ for the drop impact on stationary films. Each symbol represents one test. The curve fitting results in Eq. (4.14).

### 4.2.2 On flowing liquid films

Here we attempt to apply the models derived above to flowing liquid films. Roisman and Tropea [61] derived an analytical solution of $\bar{r}_c$ for drop impacting a flowing liquid film with a uniform thickness $\bar{h}_f$ and a unidirectional velocity $\bar{U}_f$. They assumed the base of crown still remains circular during the deformation phase, and during the spreading phase the base of the crown can be approximated as a growing circle, which moves downstream with the film flow. Neglecting the initial displacement during the deformation phase, the center of the crown base is

$$\bar{x}_c \sim \bar{U}_f (\bar{r} - \bar{r}_f)$$

(4.17)

However, the theory of Roisman and Tropea is not validated by experiments. The drop can maintain circular during deformation only if the film flow has a small effect on the deformation of the drop. This is possible when the flow inertia of drop is larger than that of the film flow, i.e. $\bar{h}_f \bar{U}_f \ll 1$. For all the tests with flowing films in the present work, $0.05 < \bar{h}_f \bar{U}_f < 0.18$. Therefore, for the present work it is expected that the drop spreading on flowing films can be predicted by the analytical models for impact on stationary films.
To validate the above discussion, a number of tests of drop impact on flowing films have been carried out for varied fluids, drop diameters and impact velocities, and film thicknesses and velocities. High-speed images are processed to measure the crown radius $r_c$ and the displacement of the crown base center $x_c$. Figure 4.4 (b) presents a series of images as an example showing the early stage of the spreading, which is the focus of the analysis here. The early stage is when the base still remains relatively circular, which, based on observation, is $\bar{\ell} < \sim 2$.

The nondimensional displacement of the base center $\bar{x}_c = x_c/D_0$ is plotted in Fig. 4.7 in the form of $\bar{x}_c/\bar{U}_f$ versus $\bar{t}$. For the three tests, $\bar{x}_c/\bar{U}_f \sim \bar{t}$, which indicates the base center is moving downstream at a speed close to the film velocity during the early stage of spreading. Figure 4.8 shows the growth of the crown radius for three tests with varied jet flow rates of 50% glycerin-water solution, i.e. varied film thickness and velocity.

![Figure 4.7](image)

Figure 4.7 Displacement of the center of the crown base during spreading phase. Total of six tests of the drop impact on flowing films are shown here (three for each working fluid).
In Fig. 4.8 (a), the present model and modified R&T model for the stationary liquid film are employed to fit with one test using the method of least squares. Here the energy loss factor for the drop impact on flowing films is denoted by $\lambda_f$. The agreement between the models, particularly the present model, with the experimental observations is satisfactory, indicating that the solution of spread radius for stationary films are also applicable for flowing films with $\bar{h}_f \bar{U}_f \ll 1$. Figure 4.8 (b) shows the fitting of the present model with the three tests with the same drop condition but varied film conditions. The fitting results in different values of $\lambda_f$ for the three tests.

![Graph](image)

Figure 4.8 Spreading of single drops impacting flowing films ($U_0=3.58$ m/s, $D_0=4.16$ mm, 50% glycerin-water solution): (a) The present model and the modified R&T model are fitted with one test; (b) Three tests (the same as the three tests shown in Fig. 4.7 with the same fluid) with varied flow rates (i.e. varied film thicknesses and velocities).
The present model is applied to fit with all the flowing-film tests. Here we attempt to correlate the energy loss factor to major non-dimensional parameters. As compared to the drop impact on stationary films, film flow velocity $U_f$ is another parameter affecting the drop impact on flowing films. This makes the number of parameters to be seven, and the number of dimensions is still three. Hence, $\lambda_f$ needs to be correlated to four non-dimensional parameters. In addition to Re, We, and $\bar{h}_f$, $\bar{U}_f = U_f/U_0$ is included. Carrying out power-law fitting, we have

$$\lambda_f = \frac{0.53}{\text{Re}^{-0.02} \text{We}^{0.03} \left(\bar{U}_f\right)^{-0.26} \left(\bar{h}_f\right)^{0.12}}$$

Equation (4.18) and the results of $\lambda_f$ are presented in Fig. 4.9. Comparing Eqs. (4.16) and (4.18) shows that the relations of $\lambda_f$ to Re and We are similar to those of $\lambda_s$. However, $\lambda_f$ shows significant dependencies on the velocity ratio, $\bar{U}_f$, and the size ratio, $\bar{h}_f$.

![Figure 4.9](image)

Figure 4.9 The energy loss factor $\lambda_f$ for the drop impact on flowing films. Each symbol represents one test. The curve fitting results in Equation (4.18).
Similar to the discussion of Eq. (4.16), for drop impact on flowing film, the transfer of kinetic energy from the drop flow to the film flow is the major cause of energy loss for the drop during the deformation phase. Equation (4.13) combined with Eq. (4.18) could predict the crown expansion on flowing films at the early spreading stage. Similar to the correlation for $\lambda_s$, the correlation for $\lambda_f$ does not satisfy the constraint $0 < \lambda_f \leq 1$. Therefore, the correlation is not valid for the entire ranges of the non-dimensional parameters.

### 4.3 Stretching rate of crown sheet

Figure 4.10 shows two single drops impacting a flowing liquid film. The impact of the drop with higher inertia ($D_0 = 3.6 \text{ mm}, U_0 = 2.17 \text{ m/s}$) produces a larger and thinner crown sheet, and small droplets are generated from the top rim of the crown. For the drop with lower inertia ($D_0 = 3.2 \text{ mm}, U_0 = 1.06 \text{ m/s}$), the formation of crown is less significant, and there is no droplet generation. For both tests, the crown sheet mainly arises where the drop spreading is opposite to the film flow (around $\varphi = 180^\circ$). This is also the location of droplet generation.

The small droplets are generated from the rim of the crown sheet, which is due to the instability of the sheet rim. Roisman et al. [70] indicated the stretching of the crown sheet in a direction normal to the sheet rim is one of the major factors affecting the rim instability. In this section we will analyze the stretching of crown sheet. As shown in Fig. 4.3, $z$ coordinate is normal to the liquid film. The free surface of the flowing film is located at $z = 0$, and the rim of crown sheet is located at $z = l(t)$. To simplify our analysis, we will only consider the $z$ component of stretching, and will focus on the local stretching at the crown sheet rim, which is

$$
\bar{S} = \frac{\partial \bar{w}}{\partial z} \bigg|_{z=l(t)} \tag{4.19}
$$
Here $\tilde{S}$ is non-dimensional stretching rate that has been normalized by $U_0/D_0$. Non-dimensionalized by $U_0$, the velocity $\tilde{w}$ is the velocity of the crown sheet in z direction. Both $\tilde{z}$ and $\tilde{l}$ have been non-dimensionalized by $D_0$.

Figure 4.10 Impact of single water drops on flowing water films: (a) $\tilde{h}_f = 0.058$, $\tilde{U}_f = 0.479$; (b) $\tilde{h}_f = 0.065$, $\tilde{U}_f = 0.981$. Arrows indicate the flow direction of the film flow.

Roisman and Tropea [61] obtained a solution for the local fluid velocity of the crown sheet at $z = 0$. If we assume that the local thickness of crown sheet is approximately equal to the thickness of the flowing liquid film, the velocity component can be written as

$$
\tilde{w}_0(\varphi, \tilde{s}) = \frac{\sqrt{(\beta_c \cos \varphi - \tilde{U}_f \sqrt{\tilde{l} + \tilde{r}_0})^2 + (\beta_c \sin \varphi)^2}}{2 \sqrt{\tilde{s} + \tilde{r}_0}}
$$

(4.20)
The analysis in the previous section has shown the modified R&T model works for drop impacting flowing liquid film. Therefore, the same modification will be applied to Eq. (4.20) by using Eqs. (4.6), (4.12), and (4.15) as the expressions of $\overline{t}_i$, $\beta_i$, $\overline{t}_o$, respectively. We have replaced $\overline{t}$ with another time variable $\overline{\delta}$. Here $\overline{\delta}$ is associated with any fluid particle in the crown sheet, and is the time instant at which the fluid particle is ejected from the film surface. It is clear that $\overline{\delta} \geq \overline{t}_i$.

Assuming the rising liquid film as 1-D inviscid flow with constant pressure, the momentum equation reduces to

$$\frac{\partial \overline{w}}{\partial \overline{t}} + \overline{w} \frac{\partial \overline{w}}{\partial \overline{\delta}} = -Fr^{-1}$$ (4.21)

where $Fr = U_0^2 / g D_0$ is Froude number, and $g$ is the gravitational acceleration. If we track a fluid particle that appears at $\overline{z} = 0$ at time $\overline{t} = \overline{\delta}$ with initial velocity given by Eq. (4.20), Eq. (4.21) can be changed to

$$\frac{D\overline{w}}{D(\overline{t} - \overline{\delta})} = -Fr^{-1}$$ (4.22)

Integrating Eq. (4.22) gives

$$\overline{w} = \overline{w}_0 - Fr^{-1}(\overline{t} - \overline{\delta})$$ (4.23)

It can be shown that the location of the particle at any time $\overline{t} > \overline{\delta}$ is

$$\overline{z}(\overline{\delta}, \overline{t}) = \overline{w}_0(\overline{t} - \overline{\delta}) - \frac{1}{2} Fr^{-1}(\overline{t} - \overline{\delta})^2$$ (4.24)

If the fluid particle being tracked is one in the top rim of the crown sheet, $\overline{\delta} = \overline{t}_i$. The location of the particle is the height of the sheet, which is

$$\overline{z}(\overline{t}) = \overline{w}_0(\overline{t} - \overline{t}_i) - \frac{1}{2} Fr^{-1}(\overline{t} - \overline{t}_i)^2$$ (4.25)
The non-dimensional stretching rate is rewritten as

\[
\bar{S} = \left. \frac{\partial \bar{w}}{\partial \bar{\delta}} \frac{\partial \bar{\delta}}{\partial \bar{z}} \right|_{\bar{z} = \bar{t}} \tag{4.26}
\]

Firstly, we obtain the first differential on the right hand side of Eq. (4.26). Equation (4.23) can be considered as temporal (\(\bar{t}\)) and spatial distribution (\(\bar{\delta}\)) of velocity, as the location of each fluid particle depends on when it departs from the flowing film. Hence, from Eq. (4.23) we can get

\[
\frac{\partial \bar{w}}{\partial \bar{\delta}} = \frac{\partial \bar{w}_0}{\partial (\bar{\delta} + \bar{r}_0)} + Fr^{-1} \tag{4.27}
\]

where \(\partial \bar{t} / \partial \bar{\delta} = 0\) has been used. Substituting Eq. (4.20) into Eq. (4.27) gives

\[
\left. \frac{\partial \bar{w}}{\partial \bar{\delta}} \right|_{\bar{z} = \bar{t}} = -B + Fr^{-1} \tag{4.28}
\]

The first term on the right hand side \(B\), is

\[
B = \frac{\bar{w}_0 (\bar{\delta} = \bar{t})}{\bar{t}_i + \bar{r}_0} = \frac{\sqrt{\left(\beta_i \cos \varphi - \bar{U}_f \sqrt{\bar{t}_i + \bar{r}_0}\right)^2 + \left(\beta_i \sin \varphi\right)^2}}{4(\bar{t}_i + \bar{r}_0)^{3/2}} \tag{4.29}
\]

As indicated by its expression, \(B\) is a characteristic acceleration due to the flow inertia of the crown sheet. Now we obtain the second differential on the right hand side of Eq. (4.26). Equation (4.24) shows the location of all the fluid particles with varied \(\bar{\delta}\) at a given instant \(\bar{t}\). Hence, we can write

\[
\frac{\partial \bar{z}}{\partial \bar{\delta}} = -\frac{\partial \bar{z}}{\partial (\bar{t} - \bar{\delta})} \tag{4.30}
\]

From Eq. (4.24), we can get

\[
\frac{\partial \bar{z}}{\partial (\bar{t} - \bar{\delta})} = \bar{u}_0 - \frac{d \bar{u}_0}{d(\bar{\delta} + \bar{r}_0)}(\bar{t} - \bar{\delta}) - Fr^{-1}(\bar{t} - \bar{\delta}) \tag{4.31}
\]
where a relation \( \partial \bar{w}_0 / \partial (\bar{t} - \bar{\delta}) = - \partial \bar{w}_0 / \partial (\bar{\delta} + \bar{\tau}_0) \) has been used. Putting Eq. (4.20) into Eq. (4.31) and combining with Eq. (4.30), we have

\[
\frac{\partial \bar{\delta}}{\partial \bar{z}} \bigg|_{\bar{t} = \bar{t}_i} = -\frac{1}{2B(\bar{t}_i + \bar{\tau}_0) + B(\bar{t} - \bar{\delta}) - Fr^{-1}(\bar{t} - \bar{t}_i)} \tag{4.32}
\]

Substituting Eq. (4.28) and Eq. (4.32) into Eq. (4.26) gives

\[
\bar{S} = \left[ 2 \frac{BFr}{BFr - 1} (\bar{t}_i + \bar{\tau}_0) + (\bar{t} - \bar{t}_i) \right]^{-1} \tag{4.33}
\]

Since \( B \) is a function of the azimuthal angle \( \varphi \), Eq. (4.33) gives the local stretching rate in the top end of the crown sheet as a function of time and azimuthal position. Equation (4.33) is plotted in Fig. 4.11 for a test of drop impacting a flowing film with \( \bar{U}_f = 1.623 \) and \( \bar{h}_f = 0.061 \). The value of \( \lambda \) has been determined to be 0.79 based the circle fitting of its spreading radius \( \bar{r}_c \).

Three major trends can be observed. First, the stretching rate decreases over time. Second, the largest stretching rate appears at \( \varphi = 180^\circ \) where the crown spreading is right opposite to the flowing liquid film. Third, the stretching rate shows low sensitivity to the azimuthal angle \( \varphi \), as the stretching changes slightly over a large portion of the circumference and decreases only close to \( \varphi = 0^\circ \). The second and third trends qualitatively agree with the experimental observations shown in Fig. 2.1 (c) and Fig. 4.10 (a). The maximum stretching rate exists at \( \varphi = 180^\circ \). At this azimuthal position,

\[
B(\varphi = 180^\circ) = \frac{\beta_c + \bar{U}_f \sqrt{\bar{t}_i + \bar{\tau}_0}}{4(\bar{t}_i + \bar{\tau}_0)^{3/2}} \tag{4.34}
\]

Then the maximum stretching rate \( \bar{S}(\varphi = 180^\circ) \) can be obtained by substituting Eq. (4.34) into Eq. (4.33). For stationary film, \( \bar{U}_f = 0 \), Eq. (4.29) reduces to
\[ B_s = \frac{\beta_s}{4(\bar{r} + \tau_0)^{3/2}} \]  

(4.35)

where \( B_s \) represents the characteristic acceleration of crown flow formed by a drop impacting a stationary film. We consider \( B_s Fr \gg 1 \), which means the initial acceleration of the crown sheet is much larger than gravitational acceleration. The stretching rate can be simplified as

\[
\bar{S} = \left[ \bar{r} + \bar{r}_t + 2\tau_0 \right]^{-1}
\]

(4.36)

Figure 4.11 Local stretching rate at the top end of the crown sheet formed by a drop impacting a flow film at varied azimuthal positions \( \varphi \) (\( \bar{U}_f = 1.623, \bar{h}_f = 0.061 \)).

### 4.4 Splashing and non-splashing impact on flowing film

Figure 4.10 has shown two types of phenomena as a result of drop impacting a flowing liquid film. One is splash impact, for which small droplets are generated from the top rim of the rising crown sheet. The other one is non-splash impact, for which no small droplets are produced. On dry surfaces and wetted surfaces, a few previous studies [51, 54, 69, 73, 104] have used \( WeRe^{1/2} \) as threshold parameter for predicting the occurrence of two phenomena. This
expression of threshold parameter can be qualitatively explained by the following. A flow rate \( U_0D_0 \) is forced through a thin liquid layer with thickness \( h_l \). The dynamic pressure of the flow in the layer is \( \sim \rho(U_0D_0/h_l)^2 \), and the static pressure is \( \sim \sigma/h_l \). When these two pressures of drop impact on a dry surface are comparable, the spreading liquid rim is destabilized and deflected upwards, finally resulting in the splashing [52, 55]. Using the value of viscous boundary as a scale of the film thickness was firstly proposed by Yarin and Weiss [60] in the study of drop train impact. Roisman et al. [69] further indicated that at high Reynolds number, the thickness of the layer can scale as the thickness of the viscous boundary layer, i.e. \( h_l \sim D_0Re^{-1/2} \). Whether it is splash impact or non-splash impact depends on the competition between the dynamic pressure and static pressure, and the ratio of the two pressures is \( WeRe^{1/2} \).

Figure 4.10 also shows that splash or non-splash impact simply depends on the local flow dynamics at \( \varphi = 180^\circ \), where the stretching rate of the crown sheet is the highest. Hence, predicting the occurrence of splash needs to evaluate the interaction between the drop flow and film flow only at \( \varphi = 180^\circ \). Similar to the analysis in previous studies as discussed above, we define

\[
K = We_m Re_m^{1/2}
\] (4.37)

Here the value of the threshold parameter \( K \) is a measure of local interaction of drop and film flows at \( \varphi = 180^\circ \). A modified Weber number \( We_m \) and a modified Reynolds number \( Re_m \) are introduced. The two modified non-dimensional numbers are defined as

\[
We_m = \frac{\rho}{\sigma} \left( D_0U_0^2 + h_j U_j^2 \right)
\] (4.38)

\[
Re_m = \frac{\rho}{\mu} \left( D_0U_0 + h_j U_j \right)
\] (4.39)
Substituting Eqs. (4.38) and (4.39) into Eq. (4.37) gives

\[ K = \frac{WeRe^{1/2}}{2} \left( 1 + \frac{\bar{h}_f \bar{U}_f^2}{(1 + \bar{h}_f \bar{U}_f)^{1/2}} \right) \]

(4.40)

A simple explanation of Eq. (4.40) can be given by the following. The value of \( K \) indicates whether or not the flow inertia can overcome surface tension and viscous effects, which in turn determines the occurrence of splash. A number of tests of the drop impact on flowing films are conducted with varied drop sizes, impact velocities, film thicknesses, film velocities, and fluids. Based on the experimental observations, the tests are categorized into two groups: splash and non-splash. For each test, the value of \( K \) is calculated using Eq. (4.40). It is found that splash group generally has large values of \( K \), while the non-splash group has low values of \( K \). The average of all the \( K \) values in this range is considered as a critical value \( K_c \). For the present work, it is found that \( K_c = 3378 \). All the tests of the drop impact on flowing films are shown in Fig. 4.12 by plotting \( WeRe^{1/2} \) versus \( (1 + \bar{h}_f \bar{U}_f^2)(1 + \bar{h}_f \bar{U}_f)^{1/2} \).

Figure 4.12 Log-log plot for all the tests of the drop impact on flowing films that have been conducted for the present work. Solid star symbols represent splash impacts, and hollow circle symbols represent non-splash impacts. The solid line is Equation (4.41) for which \( K_c = 3378 \).
Figure 4.12 shows two regions separated by a line given by

\[ \text{WeRe}^{1/2} = K_c \left(1 + \frac{\overline{H}}{\overline{U}_f^2}\right)^{-1/2} \left(1 + \frac{\overline{H}}{\overline{U}_f}\right)^{1/2} \]  

(4.41)

The line intercepts the vertical axis at \( K_c = 3378 \). There is a splash region above the line where \( K > K_c \), while the non-splash region is below the line where \( K < K_c \).

4.5 Summary

The impact of drops on flowing liquid films is theoretically and experimentally studied. The focus of the work is put on drop spreading, stretching of the crown sheet in rising direction, and prediction of splash impact. The study starts with the drop impact on stationary films. The impact process is divided into two phases: a fast deformation phase followed by a spreading phase. Two models are developed for predicting the base radius of the crown during spreading phase: a new model, and a modified model based on Roisman and Tropea’s approach [61]. Satisfactory agreement is shown between the two models and our experimental observations of the drop impact on stationary films. Based on the assumption that on flowing film the drop remains circular while spreading, the two models could also be applied to flowing films.

It shows that the non-dimensional velocity of the drop flow right after deformation, i.e. the initial velocity of the spreading phase, is less than unity, which indicates the loss of kinetic energy during the deformation phase. The non-dimensional initial velocity is referred to as energy loss factor. Correlations of the energy loss factor are developed for the drop impact on both stationary films and flowing films. The correlations in combination with the analytical models can be used to predict the expansion of the crown base on stationary films and flowing films.
In previous studies, the stretching of the crown sheet is considered as a major factor affecting the instability of the crown sheet. The modifications introduced in the spreading model are combined with Roisman and Tropea’s [61] solution of rising velocity from the kinematic discontinuity. Based on the modified solution, the local stretching rate at the top of the crown sheet in the rising direction is derived. The highest stretching rate is at the azimuthal location where the drop spread flow is right opposite to the film flow. The threshold parameter for drops impacting flowing films is defined as a function of modified Weber and Reynolds numbers, and the two modified numbers take into account both drop and film flows. The threshold value for the occurrence of splash impact on flowing films is determined based on our experimental observations.
Chapter 5 Drop Impact on a Flowing Film Cooling a Hot Surface

5.1 Background

The fluid dynamics involved in the drop impacting a flowing film is complicated. Figure 5.1 shows a water drop impacting a radially flowing water film generated by a vertical jet impinging on a transparent surface. High speed images are taken from the side and under the surface. Figure 5.1a shows the interaction of the drop and film produces a crown-like rising liquid sheet. For cooling applications, the rising liquid sheet can be considered as the local loss of coolant, which does not contribute to local convection heat transfer on the surface. From Fig. 5.1b, the drop impact generates a spreading area, where local flow spreads downstream and upstream. The spreading area deforms and moves over time. Figure 5.1 clearly shows a transient development of the flow as a result of the drop impact. The flow development must cause transient change of the local convection heat transfer, which will eventually affect cooling performance. Unfortunately, there is very limited study on this topic.

This transient heat transfer process is experimentally investigated using an IR camera (SC660, FLIR, Burlington, ON, Canada) to record the surface temperature ($T_s$) of the wafer’s underneath. The following four aspects are studied: 1) The three steps of heat transfer process are analyzed based on recorded two steps of surface temperature change. The reason for their difference is also addressed. 2) Enhancement of convection is studied for varied drop and film flow conditions. 3) An enhancement factor based on the change of heat transfer coefficient rather than the temperature change is introduced to evaluate the enhancement of convection. The curve of maximum enhancement $\eta_{max}$ is used to investigate local enhancement around impact area. 4) The peak value of $\eta_{max}$ are used to investigate the enhancement effects of film flow rate, drop
diameter, and drop impact velocity. Peak enhancement is correlated to the ratio of drop impact to film flow.

![Figure 5.1 A water drop impacting a water film radially flowing on a transparent substrate: (a) high-speed images taken from the side; (b) high-speed images taken under the substrate.](image)

The present work aims to obtain the cooling results due to drop impact on the film-cooled hot surface, which is observed by the measured surface temperature change and evaluated by the peak enhancement factor. The correlation of the cooling enhancement to the interaction of drop and film flow can provide the direct evidence for explaining spray cooling performance induced by the change of spray parameters, such as impact velocity, drop size.
5.2 Experimental methodology

Tests are conducted using the experimental setup shown schematically in Fig. 5.2. It is composed of five components: 1) a substrate with uniform surface heat flux; 2) a circular water jet which impinges on the substrate to generate radially flowing film; 3) a drop generator to generate water drops impacting the film flow; 4) a HS (high speed) camera for recording flow dynamics; 5) an infrared camera for recording temperature distribution and change.

Figure 5.2 Schematic of test setup for studying a liquid drop impacting a radially flowing film generated by jet impingement on a silicon wafer. The lower surface of the wafer is coated with a thin gold film as electrical heater. Only half of the wafer is shown.

The substrate is a silicon wafer with a diameter of 76 mm and a thickness $b=380$ µm. As shown in Fig. 5.2, the upper surface of the wafer is exposed to the drop impact, while the underneath is coated with a gold layer, which serves as an electrical heater. The detained process of fabrication of thin-film heater is described in Appendix A. The gold layer is painted black which has high radiative emissivity calibrated to be 0.95. The electrical resistance of the gold layer is $\sim 1$ $\Omega$. The heater power is determined based on the electrical current read from the DC
power supply (Model 62050P, Chroma, Irvine, CA, USA) and the voltage measured across the gold-coated area.

Heat loss includes the free air convection and radiation from the coated underneath and conduction through the wafer edge and electrical connections. For the free convection, a heat transfer coefficient of $10 \text{ W/m}^2 \cdot \text{K}$ and an ambient air temperature of 20°C are used. A large surrounding with temperature 20°C is used for calculating the radiation. Due to the high heat transfer coefficient of the film flow, the heat loss accounts for less than 0.3% of the total heater power. The coated area and the electrical connection are designed such that uniform electrical current throughout the gold layer. Hence, a uniform heat flux ($q''$) is calculated using the heater power divided by the coated area, which ranges from 34.1 W/cm² to 34.6 W/cm² for all the conducted tests.

Water at room temperature is used as the working fluid for both jet impingement and drop impact, and its temperature, denoted by $T_l$, is measured using a calibrated T-type thermocouple. Water jet from a large jet nozzle with radius $a = 1.91 \text{ mm}$ impinges on the center of the wafer. The volumetric flow rates of the jet, denoted by $Q$, ranges from 20 to 60 cm³/s. There is no hydraulic jump on the wafer surface. Depending on the jet flow rate $Q$ and the landing location $X$ (the distance from the stagnation point of jet impingement to the drop impact point), the local film flow at the landing location can be characterized by

$$Q' = \frac{Q}{2\pi X} \tag{5.1}$$

Here $Q'$ is referred to as the film flow rate, which is the product of local film thickness and mean velocity.
Water drops are generated from stainless steel capillary tubes connected to a syringe pump. The plunger advanced very slowly to form a drop at the tip of the tube, which finally detached from the tube under its own weight. Drop diameter, denoted by $D_0$, ranges from 2.2 to 4.3 mm by using different tube sizes. The height of the tube orifice above the wafer surface is adjusted to change drop impact velocity, denoted by $U_0$, which ranges from 0.17 to 4.5 m/s determined from the high speed video.

A high-speed camera (M310, Vision Research, Wayne, NJ, USA) is used to capture the impact dynamics of water drops on flowing films. To capture the change of the surface temperature ($T_s$), an infrared (IR) camera (SC660, FLIR, Burlington, ON, Canada) is positioned underneath the wafer and operated at 240 fps. The high emissivity of the black paint on the lower surface of the wafer is to increase the measurement accuracy of the IR camera.

The resolution of the IR images is ~3.3 pixels/mm$^2$. Hence, the IR camera provides a detailed temperature distribution on the lower surface of the wafer. We use the measured temperature to study the convection cooling on the upper surface of the wafer. Two concerns need to be addressed. One is regarding the lag between the observation from the underneath and the actual development on the other side. The other concern is regarding the difference between the measured underneath temperature and the temperature of the spray-cooled surface.

To address the first concern, we consider a Fourier number defined as

$$F_o = \frac{\alpha D_0}{U_0 b^2}$$

(5.2)

where $\alpha$ is the thermal diffusivity of the silicon wafer, $b$ is the thickness of the silicon wafer. Here $F_o$ compares the drop impact time scale to the thermal diffusion time scale through the
wafer thickness. For the tested ranges $D_0$ and $U_0$, $Fo$ ranges roughly from 1 to 10. As will be shown later, the entire thermal process related to the drop impact is actually much longer than $D_0/U_0$. So the lag due to heat transfer across the wafer thickness is insignificant as compared to the entire time period.

To address the temperature measurement, we evaluate the Biot number defined as

$$Bi = \frac{hb}{k}$$  \hspace{1cm} (5.3)

where $h$ is the heat transfer coefficient. In the present work, $h \sim 1 \text{ W/cm}^2 \cdot \text{K}$, which gives $Bi \sim 0.03$. Hence, the temperature measured on the lower side is the same as the upper side at the same location.

During the drop impact, heat transfer coefficient can be calculated based on the measured temperature. One way to that is to solve one-dimensional heat conduction equation using two boundary conditions at the lower side of the wafer: 1) measured surface temperature; 2) constant and uniform local temperature gradient based on the assumption of uniform heat flux. However, since $Bi < 0.1$, we assume uniform temperature through the wafer thickness and write an energy balance as

$$q'' = b \rho c_p \frac{dT_s}{dt} + h_i (T_s - T_i)$$ \hspace{1cm} (5.4)

Here heat loss has been neglected. Here $h_i$ is transient heat transfer coefficient. The heat transfer coefficient is further written as

$$h_i = \left( q'' - b \rho c_p \frac{dT_s}{dt} \right) \frac{1}{T_s - T_i}$$ \hspace{1cm} (5.5)
The IR camera records surface temperature at a constant frame rate. For any location and any transient point, the term $dT_s/dt$ can be approximately calculated using the local temperature change between two consecutive images multiplied by the frame rate. Without drop impact, the cooling is at steady state, and Eq. (5.5) reduces to

$$h_{ss} = \frac{q''}{T_{ss} - T_l}$$

(5.6)

Here $h_{ss}$ and $T_{ss}$ represent the heat transfer coefficient and surface temperature at steady state, respectively. It should be noted that $T_l$ in Eqs. (5.5) and (5.6) is the fluid temperature of the drop and jet. However, the free surface temperature of the local flow during the impact process and at the steady state could be higher than $T_l$.

In the present work, the surface temperature ($T_s$, and $T_{ss}$) is measured by the IR camera, which has a measurement error $\pm 0.5^\circ$C associated with the accuracy of the IR camera and the uncertainty of the emissivity calibration. The water temperature ($T_l$) is measured using a calibrated T-type thermocouple which has a measurement error $\pm 0.1^\circ$C.

The heat flux $q''$ is calculated based on the heater power and heater area ($A$), and the heater power is calculated based on the measured voltage ($V$) and current ($I$). The uncertainty of the heat flux can be obtained from

$$\frac{|\delta q''|}{q''} = \sqrt{\left(\frac{\delta U}{U}\right)^2 + \left(\frac{\delta I}{I}\right)^2 + \left(\frac{\delta A}{A}\right)^2}$$

(5.7)

In the present work, $|\delta q''/q''|\sim 2\%$.

The uncertainty of the steady-state heat transfer coefficients ($h_{ss}$) is evaluated using
\[
\frac{\delta h_{ss}}{h_{ss}} = \sqrt{\left(\frac{\delta q''}{q''}\right)^2 + \left(\frac{\delta T_{ss}}{T_{ss} - T_i}\right)^2 + \left(\frac{\delta T_i}{T_{ss} - T_i}\right)^2}
\] 

(5.8)

In the present work, \(\frac{\delta h_{ss}}{h_{ss}}\) ranges from ~2.5\% to ~3\%. Replacing \(T_{ss}\) with \(T_s\), Eq. (5.8) can also be used to approximately evaluate the uncertainty of \(h_t\). For each test, the lowest value \(T_s\) is chosen for the uncertainty analysis. As a result, \(\frac{\delta h_t}{h_t}\) ranges from ~3\% to ~5\%.

### 5.3 Transit heat transfer after drop impact

The influence of drop impact on heat transfer should be investigated by determining the difference between the steady cooling of the film flow and the transient cooling during the drop impact. The steady-state cooling of the film flow is shown in Fig. 5.3 when the drop is about to land on the film. The temperature increases from the center toward the edge, showing a concentric distribution. The center is the stagnation of the jet impingement and is used as the origin point of the coordinates system. The location where the drop lands on the film flow is the impact point. In Fig. 5.3, the distance between the impact point and the stagnation point \(X=16\) mm.

At the early stage of the drop impact such as \(t=1.6\) and \(4.8\) ms shown in Fig. 5.3a, the drop impact results in a rising crown sheet which expands and breaks up over time. At the same time, local temperature starts decreasing due to the change of the local flow. To highlight the temperature change, we can subtract the transient thermal images from the steady-state image, and the resulted images show the temperature change \(\Delta T_s = T_s - T_{ss}\). The contour of the temperature difference is plotted in Fig. 5.3b, which roughly is concentric with a cold spot in the center. Figure 5.3c shows the distribution of the temperature difference along the center line, which is the line crossing the jet stagnation point and the drop impact point.
Figure 5.3 (a) A drop \((U_0=3.24 \text{ m/s}, D_0=3.2 \text{ mm})\) impacting flowing film \((Q'=2.49 \text{ cm}^2/\text{s})\) at \(X=16 \text{ mm}\); (b) contour of surface temperature \(T_{ss}\) at the steady state prior to drop impact (b1); contours of temperature change \(\Delta T_s\) (b2, b3); (c) temperature change along the center line.

The temperature difference profile shown in Fig. 5.3c is like a valley. At \(t=1.6 \text{ ms}\), a small jag appears at the valley bottom and disappears in the following images. This is also visible in the temperature change contour shown in Fig. 5.3b. This phenomenon has been observed consistently in many tests at the very early stage of impact. A possible reason could be a thin layer of air trapped between the drop and the flowing film [25, 27, 105]. However, it is still reasonable to assume perfect contact between falling drop and flowing film since the air layer
disappears at the early stage of impact ~5.8 ms (no temperature jag at the valley bottom in Fig. 5.3c).

Figure 5.3a also shows that the center of the spreading area moves downstream from the impact point, which is consistent with the observations in Fig. 5.1. As a result, the center of the temperature change contour moves downstream from the impact point (comparing t= 1.6 ms and 5.8 ms in Fig. 5.3b), which can also be seen from the valley bottom of temperature change profile in Fig. 3c. This is due to the interaction of the drop and the film flow during the drop impact, which results in a higher film flow downstream of the impact point. This can be easily determined by comparing the spreading extents downstream and upstream of the impact point [49, 61, 78].

The drop impact on the film flow breaks the steady-state cooling. As a result, the temperature within the spreading area changes with time, as shown by Fig. 5.3. The transient process can be characterized by monitoring the local temperature at the impact point of the drop. The temperature is plotted in Fig. 5.4, where t=0 ms is when the drop lands on the film flow. The transient process is composed of two stages. In the first stage, the temperature quickly decreases, showing quick response to the drop impact. The time taken for this stage, denoted by $\Delta t_1$, is referred to as response time. From Fig. 5.1, it is reasonable to believe that in the first stage the cooling is dominated by the impacting and spreading flows driven by the drop impact. In the second stage, after reaching the lowest point, the temperature starts rising toward the steady state. The time taken for this stage, denoted by $\Delta t_2$, is referred to as recovery time. During the second stage, the film flow is taking over while the spreading flow is diminishing. For the test shown in Fig. 5.4, $\Delta t_1 \sim 40$ ms, while $\Delta t_2 \sim 160$ ms, both are much longer than the time scale $D_0/U_0 = 1$ ms.
Figure 5.3 has shown that the temperature change valley deepens from $t=1.6$ to 5.8 ms. To have more information about the evolution of temperature change in the entire transient process, the temperature change along the center line is plotted for a longer time period from $t=13$ to 196 ms, which is much longer than the drop impact time scale $D_0/U_0 \sim 1$ ms. It can be seen that the valley continues to deepen until $t=38$ ms, and then comes back until it almost disappears at $t=196$ ms. This is consistent with the response and recovery stages observed in Fig. 5.4. It also shows that the valley bottom moves in the direction of the film flow.

![Temperature Change Graph](image)

Figure 5.4 The local temperature measured at the impact point ($Q'=2.49$ cm$^2$/s, $U_0=3.5$m/s, $D_0=3.5$mm). The drop impact starts at $t=0$ ms. The first stage of the drop impact is $\Delta t_1$, and the second stage is $\Delta t_2$.

Without the drop impact, the heat transfer coefficient of the steady-state cooling is the highest in the center and decreases toward the edge. This trend can be seen in Fig. 5.4b, where the heat transfer coefficient prior to the drop impact is calculated using Eq. (5.6). When a drop impacts, the interaction of the drop and the film flow significantly affects the local heat transfer coefficient around the impact area, which can be calculated using Eq. (5.5). The transient term in Eq. (5.5) is approximately calculated using the temperature change from the previous thermal
image multiplied by the frame rate of the IR camera. The heat transfer coefficient along the center line is plotted for a few transient points in comparison with the heat transfer coefficients at the steady state. To better show the change of heat transfer coefficient, the difference between the local steady state and transient heat transfer coefficients, $\Delta h = h_t - h_{ss}$, is plotted as an inset in Fig. 5.5b.

Figure 5.5 (a) The temperature change along the center line after the drop impact ($Q'=2.49$ cm$^3$/s, $U_0=3.5$ m/s, $D_0=3.5$ mm). (b) The heat transfer coefficient along the center line including the steady state ($t=0$ ms) $h_{ss}$ and transient state $h_t$. The inset plot shows $\Delta h = h_t - h_{ss}$. (c) Cooling enhancement calculated using Eq. (5.9). The top curve is the maximum enhancement curve $\eta_{max}$, and its peak is the peak enhancement $\eta_p$. 
A few interesting points can be observed by comparing Figs. 5.5a and 5.5b. At the early stage, along the center line $T_s$ decreases and $h_t$ increases. The temperature $T_s$ reaches the maximum decrease at $t=38$ ms, while $h_t$ reaches its maximum increase at $t=21$ ms. In other words, although $T_s$ continues to decrease from $t = 21$ to 38 ms, the heat transfer coefficient already starts decreasing as $h_{t=21\,ms} > h_{t=38\,ms}$. This is because the temperature decreases at a faster rate during the early stage of drop impact, i.e. $-(dT_s/dt)_{t=21\,ms} > -(dT_s/dt)_{t=38\,ms}$.

Second, after reaching the maximum at $t = 21$ ms, the heat transfer coefficient continues to decrease, and passed the steady-state curve at $t=63$ ms (the curve for $t=63$ ms is not shown). At $t=75$ ms when the temperature change curve is still rising toward the steady state, the heat transfer coefficient curve reduces below the steady state, showing $\Delta h < 0$. At $t=125$ ms, the heat transfer coefficient has almost recovered to the steady state, while the temperature is still on the way to the steady state.

The discussion above can be summarized as follows. The change of the temperature shows two steps: decrease and recovery. The change of the heat transfer coefficient shows three steps: increase, decrease, and recovery. The differences between the trends for $T_s$ and $h_t$ is due to the existence of the transient term in Eq. (5.5). In other words, if the cooling target had zero thermal mass, the change of $T_s$ would also be three steps in directions opposite to the change of $h_t$. The three-step change of heat transfer coefficient shown in Fig. 5.5b can be explained from both hydrodynamic and thermal perspectives. From the hydrodynamic perspective, the drop impact results in a fast spreading flow which slows down before being replaced by the film flow. Particularly, at some location the flow eventually switches from flowing upstream to flowing downstream. From the thermal perspective, the arrival of the drop breaks the local steady-state thermal boundary layer, and a new thermal boundary layer develops in the spreading flow.
Before the takeover by the film flow, it could be possible that the thermal boundary layer encompasses the spreading flow thickness, and consequently the free surface temperature is higher than $T_l$.

The change of heat transfer coefficient represents the change of the convection, which can be quantified by introducing an enhancement factor defined as

$$\eta = \left( h_t - h_{ss} \right) / h_{ss}$$  \hspace{1cm} (5.9)

For the enhancement analysis, instead of Eq. (5.9) based on the heat transfer coefficient, one might choose to use a temperature ratio given by $\Delta T_s / (T_{ss} - T_l)$. However, from Eqs. (5.5) and (5.6), the convective heat flux during the drop impact is different from that at the steady state. Therefore, the temperature ratio shows the enhancement of cooling performance, while the ratio of heat transfer coefficient evaluates the capability of the convective heat transfer.

Based on the uncertainties of $h_t$ and $h_{ss}$ discussed in Section 5.3, the uncertainty of the enhancement $|\delta \eta/\eta|$ ranges from ~4% to ~6%. Eq. (5.9) is plotted in Fig. 5.5c, which clearly shows the enhancement develops over time. At $t=21$ ms, except a small region around X~25 mm, most of the impact region has reached local maximum enhancement, denoted by $\eta_{max}$. Hence, for the drop impact shown in Fig. 5.5, the curve at $t=21$ ms shows the distribution of local maximum enhancement, and therefore is referred to as the maximum enhancement curve. The maximum enhancement occurs in the first stage of the drop impact when the flow driven by the drop impact is dominant. On the $\eta_{max}$ curve, there is a peak enhancement $\eta_p = ~55\%$ which occurs at a location close to the impact point at $t=21$ ms. At $t=75$ ms, there is ~10% decrease of convection close to the impact point. This occurs when the film flow is taking over from the
spreading flow. The trends of the enhancement factor shown in Fig. 5.5c can be explained by the discussion above from both hydrodynamic and thermal perspectives.

The observed temperature change during the drop impact is the result of the change of convection heat transfer. The change of convection heat transfer is caused by the change of fluid dynamics, which is driven by the interaction of the drop and film flows. In the following, we will evaluate the effects of film flow rate $Q'$, drop diameter $D_0$, and impact velocity $U_0$.

### 5.4 Changing film flow rate

A series of tests are conducted to investigate the effect of the film flow when the drop parameters are maintained constant ($U_0=3.5\text{m/s}$, $D_0= 3.5\text{mm}$). The film flow rate $Q'$ is varied from 2.49 to 6.63 cm$^2$/s by changing the jet flow rate $Q$ while the drop landing location is maintained constant at $\chi =16 \text{ mm}$. The maximum enhancement curves of a few tests are presented in Fig. 5.6a, which shows the local maximum enhancement $\eta_{max}$ along the center line. Generally, the curves with lower film flow rates encompass the curves with higher film flow rates. This can be clearly observed from the two curves with the lowest and highest film flow rates ($Q' = 2.49$ and 6.63 cm$^2$/s). Hence, the film flow rate has two major effects. First, drop impact tends to cause significant local enhancement if the film flow rate is low. Second, the enhancement tends to occur in a large area if the film flow rate is low.

A close look at Fig. 5.6a also shows that the location of the peak enhancement $\eta_p$ shifts downstream of the impact point. Interestingly, the higher the film flow rate, the less shift from the impact point. For example the peak location almost coincides with the impact point for $Q' = 6.63 \text{ cm}^2/\text{s}$. When the film flow rate is high, the drop impact can cause appreciable effect only at the early stage of the impact. When the film flow rate is low, the flow driven by the drop
impact has more time to develop, and at the same time the center of the spreading area is moving downstream. Figure 6a also shows that $\eta_{max}$ decreases with increasing $Q'$. To clearly show the trend, $\eta_p$ is plotted versus $Q'$ in Fig. 5.6b. For the film flow rate increasing from 2.49 to 6.63 cm²/s, the peak enhancement decreases from ~55% to ~10%.

Figure 5.6 (a) The maximum enhancement along the centerline caused by the drop impact ($U_0=3.5$ m/s, $D_0=3.5$ mm) on films with varied flow rates. (b) The peak enhancement versus film flow rate.

Changing the film flow while keeping the drop flow constant also affects the time duration of the impact process. The temperature at the impact point is recorded for all the tests with
constant drop conditions ($U_0=3.5\text{m/s}$, $D_0=3.5\text{mm}$), and the same pattern as shown by Fig. 5.4 is observed for all the tests. Figure 5.7 shows that the recovery time $\Delta t_2$ decreases with increasing $Q'$. This is understandable as a strong film flow is expected to quickly remove the drop impact flow and recover the flow in the spreading area to the steady state. In contrast, the response time $\Delta t_1$ is almost independent of $Q'$, as the fluid dynamics in the early stage is dominated by the drop impact. Since $\Delta t_1 + \Delta t_2$ is the time duration of the entire transient thermal process, Fig. 5.7 also shows that the transient process shortens as $Q'$ increases.

![Figure 5.7 The response time and recovery time versus the film flow rate under constant drop conditions ($U_0=3.5\text{m/s}$, $D_0=3.5\text{mm}$).](image)

### 5.5 Changing drop velocity

Tests are conducted to investigate the effects of drop impact velocity $U_0$ while the film flow rate ($Q'=2.49\text{ cm}^2/\text{s}$) is maintained constant. Three groups of tests are carried out for three drop diameters: $D_0=2.6$, 3.5 and 4.3 mm. The impact velocity is observed to affect the spreading and splashing. Figure 5.8 shows high speed video images of three drops impacting the flowing film, which have different velocities but the same drop diameter $D_0=3.5$ mm. At any transient
point from $t=1$ to 7 ms, a higher impact velocity always shows a larger spreading area. This indicates that a higher impact velocity results in a faster spreading flow, which is expected to enhance the local convection. Figure 5.8 also shows that a higher impact velocity results in higher rising sheet. Similar phenomenon has been reported by Cossali et al. [106] that the maximum crown height is related to the Weber number of drop impact: the larger drop velocity, the higher crown height. The rising sheet can be considered as local loss of coolant, as the ejected fluid does not contribute to local cooling. It is reasonable to believe the rising sheet has negative effect on local cooling.

![Figure 5.8](image)

Figure 5.8 Drop spreading and splashing at different drop velocities: (a) $U_0=1.67$ m/s, (b) $U_0=3.50$ m/s, (c) $U_0=4.38$ m/s. The drop size ($D_0=3.5$ mm) and flowing film conditions ($Q'=2.49$ cm$^2$/s) are the same for all the three tests.

The maximum enhancement curve, $\eta_{max}$, is plotted in Fig. 5.9a for five tests with $D_0=3.5$ mm and different impact velocities. Generally, Fig. 5.9a shows a higher impact velocity results
in a larger area to be thermally affected, both upstream and downstream of the impact point. Consistent to the observation discussed above, the location the peak enhancement shifts away from the impact point. For the two low impact velocities, $U_0=0.16$ and 1.13 m/s, the peak enhancement is closer to the impact point. The possible reason is that if the drop flow is relative weak and the film flow is dominant, the maximum enhancement can occur only at the early stage of the impact when the center of the impact region is still very close to the impact point.

Figure 5.9 (a) The maximum enhancement with varied drop impact velocities from 0.16 m/s to 4.38 m/s and constant drop diameter ($D_0=3.5$ mm) and film flow rate ($Q'=2.49$ cm$^2$/s). (b) The peak enhancement of drop impact ($Q'=2.49$ cm$^2$/s) versus the drop impact velocity.
The most interesting observation from Fig. 5.9a is regarding the position of the $\eta_{max}$ curve for the highest velocity $U_0=4.38$ m/s. Rather than being the top, the curve is between the $\eta_{max}$ curves for $U_0=1.13$ m/s and 3.37 m/s. This indicates that the maximum enhancement does not follow a monotonic trend with the impact velocity. In other words, faster impact does not necessarily cause higher enhancement.

To further investigate the trend of enhancement with drop impact velocity, the peak enhancement $\eta_p$ is plotted versus the impact velocity $U_0$ in Fig. 5.9b for all the conducted tests. Each data curve in Fig. 5.9b represents a constant drop diameter. From each data curve, the following can be observed. The peak enhancement first increases with increasing the impact velocity for a relatively large portion of the tested range of $U_0$ (see region-I in Fig. 5.9b). Further increasing $U_0$ causes $\eta_p$ to decrease (see region-II in Fig. 5.9b). Then $\eta_p$ becomes relatively stable and shows low dependency on $U_0$ (see region-III in Fig. 5.9b). In region-I, the convection is enhanced as the drop impact velocity increases. In region-II, increasing $U_0$ could significantly increases rising sheet and splashing causes more loss of coolant, and as a result the enhancement is negatively affected. Figure 5.9b also shows the effect of the drop diameter, which will be discussed in the following section.

5.6 Changing drop diameter

The focus of this section is on the drop diameter $D_0$, which is changed from 2.2 to 4.3 mm. Three groups of tests are conducted with constant film flow ($Q'=2.49$ cm$^2$/s) and three different impact velocities $U_0=1.39$, 3.01, and 3.50 m/s. Figure 5.10 shows three drops with $D_0=2.2$, 3.2, and 4.3 mm and the same velocity $U_0=3.01$ m/s impacting the film flow. A larger drop results in larger spreading area and higher rising sheet. The shift of the spreading center from the impact point is clear for each case.
In Fig. 5.11a, the maximum enhancement curve is plotted for the tests with \( U_0 = 3.01 \text{ m/s} \). Increasing \( D_0 \) extends the enhanced area both upstream and downstream of the impact point. The peak enhancement takes place downstream of the impact point. The larger the drop size, the more shift from the impact point. The effect of the drop diameter is significant. The peak enhancement \( \eta_p \sim 35\% \) for \( D_0 = 2.2 \text{ mm} \), while \( \eta_p \sim 60\% \) for \( D_0 = 4.3 \text{ mm} \).

![Figure 5.10](image)

Figure 5.10 The drop impact \( (U_0 = 3.01 \text{ m/s}, Q' = 2.49 \text{ cm}^2/\text{s}) \) with varied drop diameters: (a) \( D_0 = 2.2 \text{ mm} \); (b) \( D_0 = 3.2 \text{ mm} \); (c) \( D_0 = 4.3 \text{ mm} \).

The peak enhancement \( \eta_p \) is plotted versus the drop diameter \( D_0 \) in Fig. 5.11b for all the tests conducted in this section. Each data curve represents a constant impact velocity. For each case, the peak enhancement increases with increasing the drop size, showing a monotonic trend within the tested range of \( D_0 \). Although Fig. 5.10 shows that increasing \( D_0 \) also promotes the rising sheet and splashing similar to the effect of increasing \( U_0 \), it should be noted that a larger drop increases the amount flow added to the impact area. This could be the reason that \( \eta_p \) is not observed to decrease with increasing \( D_0 \).
Figure 5.11 (a) The maximum enhancement of drop impact ($U_0=3.01$ m/s,) with varied drop diameters; (b) the peak enhancement of drop impact ($Q' = 2.49$ cm$^2$/s) versus the drop diameter.

5.7 Discussion on peak enhancement

Three important parameters including the film flow rate $Q'$, the drop impact velocity $U_0$, and the drop diameter $D_0$ have been experimentally tested. Their effects on the maximum enhancement along the center line has been shown in Figs. 5.6a, 5.9a, and 5.11a. Each test has a value of peak enhancement, $\eta_p$, which has been plotted versus the three parameters in Figs. 5.6b, 5.9b, and 5.11b, respectively. Figure 5.6b shows that $\eta_p$ decreases with increasing $Q'$. Figure
5.9b shows that for a large portion of the tested range of $U_0$, $\eta_p$ increases with increasing $U_0$. Figure 5.11b shows that $\eta_p$ increases with increasing $D_0$. Therefore, if we exclude the regions-II and III in Fig. 5.9b, the three parameters can be grouped into $U_0D_0/Q'$, and we expect an increasing trend of $\eta_p$ with respect to $U_0D_0/Q'$. It should be noted that $U_0D_0$ and $Q'$ are the two-dimensional flow rates for the drop and the film flow, respectively.

All the tests (except the tests in regions-II and III in Fig. 5.9b) are presented in Fig. 5.12, which shows $\eta_p$ versus $U_0D_0/Q'$ in a log-log plot. The data points show a general trend that the peak enhancement increases with increasing the ratio of the drop flow rate and film flow rate. Since the least mean square linear fitting has a slope of 1/2, the relationship can be expressed as $\eta_p \propto (U_0D_0/Q')^{1/2}$. It should be noted that the grouped parameter $U_0D_0/Q'$ can also be replaced with the ratio of two Reynolds numbers: one for the drop, and the other one for the film flow.

![Graph showing $\eta_p$ versus $U_0D_0/Q'$](image)

Figure 5.12 The peak enhancement versus the ratio of drop flow rate to the film flow rate plotted in a log-log scale. The tests shown in regions-II & III in Fig. 5.9b are excluded.
5.8 Summary

An experimental study is presented to analyze the heat transfer involved in the impact of single water drops on a water film flow cooling a hot surface. The surface is made of a thin silicon wafer with a thin film heater coated on one side while the other side is cooled for the water flow. IR camera is used to capture temperature with high spatiotemporal resolutions. Focus is put on the change of convection heat transfer in relation to the observation of flow dynamics during the drop impact. Based on the temperature measurement, the drop impact causes the surface temperature to first decrease and then return to the steady state, showing a thermal response process followed by a recovery process. It is found that the recovery time decreases with increasing the film flow rate, while the response time is negligibly affected by the film flow.

Different from the surface temperature change, the transient heat transfer coefficient ($h_t$) presents three steps compared to the steady state: the increase step, the decrease step, and the recovery step. After drop impact, $h_t$ continuously increases until reaching a maximum value, indicating enhancement of convection, and then decreases toward and eventually below the steady state. In the recovery step $h_t$ increases toward and returns to the steady state. To evaluate the change of convection, an enhancement factor ($\eta$) based on the change of heat transfer coefficient, is introduced. In the increase step, when most of the impact area reaches its local maximum enhancement ($\eta_{max}$), the distribution of $\eta_{max}$ along the centerline of the impact area is referred as the maximum enhancement curve, and the peak value of the curve is referred to as the peak enhancement ($\eta_p$).

It is shown that the relation of $\eta_p$ with drop velocity does not follow a monotonic trend. With increasing drop velocity, the peak enhancement first increases, then decreases, and
eventually becomes relatively stable, showing low dependency on \( U_0 \). However, the peak enhancement increases with the increase of drop size in the current experimental range, since a larger drop increases the amount flow added to the impact area. Combining the effects of the drop flow and film flow, the parameters can be grouped into the ratio of the drop flow rate and film flow rate as \( U_0 D_0 / Q' \). The least mean square linear fitting shows \( \eta_p \propto (U_0 D_0 / Q')^{1/2} \).
Chapter 6 Drop Train Impact on a Flowing Film Cooling a Hot Surface

6.1 Background

Heat transfer of single drop impact on cooling-film has been studied in Chapter 5. In spray cooling spray impingement is composed of many continuous drops impacting on flowing film at the same sites of the hot surface. To further explore the cooling mechanism of spray cooling, heat transfer process of drop train impact on film-cooled surface is experimentally investigated using an IR camera to record the surface temperature ($T_s$) of the wafer’s underneath. By using a drop generator, continuous drop train is firstly generated by the breakup of liquid jet, while the controlled drop train with varied drop number is applied by an aluminum-plate sector blocker (with open angle $\alpha$) positioning underneath the drop generator. Drop train can pass through the open sector $\alpha$ while stop by the solid plate of the blocker. During each cycle of blocker rotation, there is one group of drop train passing through the blocker, and the generation frequency of drop train $f_b$ is the same as the rotation frequency of the blocker. Hence the drop number of one group of drop train is controlled by not only the open angle $\alpha$, but also the drop generation frequency $f_s$ and the drop train generation frequency $f_b$.

The main contributions of this work are addressed: 1) The controlled drop train is fulfilled by controlling the open angle $\alpha$, the drop generation frequency $f_s$ and the drop train generation frequency $f_b$. 2) The heat transfer process due to drop train impact is analyzed based on recorded surface temperature change. 3) Cooling enhancement is investigated for varied drop train and film flow conditions. 4) An enhancement factor based on the change of heat transfer coefficient is introduced to evaluate the enhancement of convection. The peak value of $\eta_{max}$ are used to
investigate the enhancement effects of film flow rate, drop number, and drop train impact velocity.

6.2 Experiments

6.2.1 Experimental setup

Experimental setup for the tests of drop train impact on a flowing film cooling a hot surface is shown schematically in Fig. 6.1. It is mainly composed of six subsystems: 1) an impact substrate with uniform surface heat flux; 2) a circular water jet impinging on the substrate to generate radially flowing film; 3) a drop generator to generate water drop train impacting the film flow; 4) a circular sector aluminum plate serving as drop train blocker; 5) a HS (high speed) camera for visualizing impact dynamics; 6) an infrared camera for measuring surface temperature distribution.

![Experimental setup diagram]

Figure 6.1 Experimental setup of drop train impact on a flowing film cooling a hot surface.

Similar to setup of single drop impact, the upper surface of the wafer is covered by the flowing film, while the underneath is coated with a gold layer with the thin thickness of 40 nm,
as shown in Fig. 6.1. This gold-coated rectangular thin film serves as an electrical heater and its area is ~20 cm². The electrical resistance of the thin film heater is tested to be ~2 Ω. The good uniformity of sputter-coated area and the well-designed electrical connection allow uniform electrical current throughout the gold layer. Hence, a uniform heat flux \( q'' \) is calculated using the effective heater power divided by the coated area, ranging from 14.2 W/cm² to 25.1 W/cm² for all the experimental tests. The heat loss accounts for less than 0.3% of the total heater power due to high heat transfer coefficient of the film flow and most of the heat removal from the upper surface. In association with the measured temperature, heat transfer coefficient can be calculated based on Eqs. (5.4) and (5.5). Water is used as the coolant fluid for both jet impingement and drop train impact. Its temperature, denoted by \( T_l \), maintains at room temperature of 21°C.

Water drop train is generated from a monodisperse drop generator (MTG-01-G3, FMP Technology GMBH, Erlangen, Germany) and its volumetric flow rate \( Q_t \) ranges from 20 ml/Min to 150 ml/Min. The landing point maintains at \( x=15 \) mm for all the conducted tests. At the bottom of the drop generator, there is a platinum/iridium pinhole plate (with the nozzle diameter \( D_{hole} \) of 500 μm) with fine laser-drilled outlet to emerge the fluid jet. The liquid jet breaks up into continuous drop train at the downstream due to the jet interface instability (the wavelength of deformation of the jet surface is larger than the circumference of the jet). To induce the instability, the high-frequency excitation in the form of sine wave, supplied by a function generator (TCE7404, Toellner, Herdecke, Germany), is inputted into the piezo-quartz oscillation module inside the generator. The generation frequency of coolant drop \( (f_s) \) is the same as the excitation frequency of the generator. Hence, based on volume conservation the drop diameter can be calculated by
\[ D_0 = \left( \frac{6Q}{\pi f_s} \right)^{1/3} \]  

Single drop diameter ranges from 0.8 to 1 mm by using different generation frequencies. The change of drop velocity \( U_0 \) is also in association with the volumetric flow rate of drop train, which is tested from 3.7 to 17.1 m/s.

### 6.2.2 Control mechanism of drop train

In spray cooling sparse or dense spray have been demonstrated to result in different cooling performance. The cooling mechanism behind this can be quantitatively attributed to the impact frequency and momentum of drop stream on the film flow. Studying these factors as independent variables is a too complicated work in spray impact. Hence, the selection mechanism of drop train is designed for independent drop property, which is performed by a plate blocker as shown in Fig. 6.1.

The circular sector aluminum plates (the thickness of 0.5 mm and the diameter of 100 mm) with varied open angles, \( \alpha \), serve as the drop train blocker, which is manipulated by an electric blocker controller. In the tests, open angle \( \alpha \) changes from 0° to 360°. \( \alpha=0^\circ \) represents closed blocker without drops through the blocker, while \( \alpha=360^\circ \) indicates open blocker so that all the drops could pass through the blocker. When \( 0^\circ < \alpha < 360^\circ \), the drop can pass through the open section (\( \alpha \)) and be blocked by the solid section of the plate (360°-\( \alpha \)), as shown in Fig. 6.2. After one complete rotating circle, there is one group of drop train through the blocker. The generation frequency of drop train, as the same as the rotational frequency, is defined as group frequency (\( f_b \)). The speed of the blocker ranges from 300 to 1850 RPM, corresponding to the rotational frequency from 5 to 30.8 Hz.
Figure 6.2 Selection of the drop number in single drop train by controller with \( f_s = 1000 \) Hz, \( \alpha = 30^\circ \), and changing the generation frequency of drop train \( f_b \): (a) 9.9 Hz, (b) 18.3 Hz, (c) 25.0 Hz, (d) 30.1 Hz.

Figure 6.3 shows two time periods for drop generation: with drop generation (on), without drop generation (off), which is similar to periodical signal in an electrical device. The fraction of one period in which drop is generated is defined as duty cycle, denoted by \( \gamma \). The duty cycle of drop generation, \( \gamma \), is calculated by

\[
\gamma = \frac{\alpha}{2\pi}
\]  

(6.2)

The average frequency of drop generation in one period is expressed as

\[
f_{avg} = f_s \gamma
\]  

(6.3)

which is dependent of \( f_s \) and \( \alpha \), but independent of \( f_b \). For one complete rotating circle \( \Delta t = 1/f_b \), the drop number in single drop train through the blocker, denoted by \( n \), is calculated by

\[
n = \frac{f_{avg}}{f_b}
\]  

(6.4)

which is in relation to \( f_s \), \( f_b \) and \( \alpha \). The controlled drop train is fulfilled by adjusting independent parameters: single frequency \( f_s \), group frequency \( f_b \), and open angle of the blocker \( \alpha \).
Specifically to the drop number of drop train, we amply two parameters to quantify drop flow landing on the surface: $f_{avg}$ and $n$. Based on Eq. (6.3), $f_{avg}$ is dependent of $f_s$ and $\alpha$, controlling the drop number during one period time. Since $n = f_{avg} / f_b$, the drop number in single drop train $n$, is not only dependent of $f_s$ and $\alpha$, but also $f_b$. Adjusting $f_b$ would not change the average frequency $f_{avg}$ of coolant drops passing through the blocker, but change the drop number $n$ in single drop train. In Eq. (6.4) the integer part of $n$ is used for the calculation prediction of drop number, which shows good agreement with experimental tests in Fig. 6.2. For example, by inputting $f_s=1000 \text{ Hz}$, $f_b=9.9 \text{ Hz}$ and $\alpha=30^\circ$ into Eq. (6.4), $n=8$ agrees well with the experimental results in Fig. 6.2 (a). Changing $\alpha$ affects both $f_{avg}$ and $n$, and their effects on convection cooling are addressed in Section 6.3.3.

For producing the same drop size by breakup of capillary jet in the Rayleigh regime, the jet instability criterion requires the wavelength of jet surface deformation larger than the jet circumference [107-109] that limits the frequency range: $0.3 U_0 / \pi D_{hole} < f_s < 0.9 U_0 / \pi D_{hole}$. The experimental tests show that there is an optimal $f_s$ for steady breakup at a certain drop velocity $U_0$. That means changing $f_s$ would change the drop velocity $U_0$ and vice versa. For all the conducted tests for drop number of drop train, $f_s = 5000 \text{ Hz}$ and $Q_t=1.67 \text{ cm}^3/\text{s}$ so that the drop train velocity and single drop size would maintain constant.

Controlling the drop velocity independent of the drop flux is too complicated in spray impact. However, controlled drop train impact allows us varying drop velocity independent of drop flux or drop number. Tests are carried out to investigate the effects of impact velocity $U_0$. As shown in Table 6.1, the impact velocity is changed by varying $f_s$ and $Q_t$, while single drop diameter $D_0$ is maintained at ~0.86 mm for all the conducted tests based on Eq. (6.1). To
maintain $f_{avg}$ and $n$ constant, based on Eqs. (6.3) and (6.4) the product of $f_s \cdot \alpha$ is the same for all the tests while $f_b=12$ Hz. Effects of impact velocity $U_0$ are emphasized in the section 6.3.4.

Table 6.1 Drop train parameters for controlling impact velocities independent of drop number.

<table>
<thead>
<tr>
<th>No.</th>
<th>$Q_t (cm^3/s)$</th>
<th>$f_s$ (Hz)</th>
<th>$f_b$ (Hz)</th>
<th>$\alpha$ (°)</th>
<th>$f_{avg}$ (Hz)</th>
<th>$n$</th>
<th>$U_0$ (m/s)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>3.33</td>
<td>10000</td>
<td>12</td>
<td>20</td>
<td>556</td>
<td>46</td>
<td>17.1</td>
</tr>
<tr>
<td>2</td>
<td>2.22</td>
<td>6667</td>
<td>12</td>
<td>30</td>
<td>556</td>
<td>46</td>
<td>11.3</td>
</tr>
<tr>
<td>3</td>
<td>1.67</td>
<td>5000</td>
<td>12</td>
<td>40</td>
<td>556</td>
<td>46</td>
<td>8.5</td>
</tr>
<tr>
<td>4</td>
<td>1.33</td>
<td>4000</td>
<td>12</td>
<td>50</td>
<td>556</td>
<td>46</td>
<td>6.8</td>
</tr>
<tr>
<td>5</td>
<td>1.12</td>
<td>3333</td>
<td>12</td>
<td>60</td>
<td>556</td>
<td>46</td>
<td>5.7</td>
</tr>
<tr>
<td>6</td>
<td>0.95</td>
<td>2857</td>
<td>12</td>
<td>70</td>
<td>556</td>
<td>46</td>
<td>4.9</td>
</tr>
<tr>
<td>7</td>
<td>0.83</td>
<td>2500</td>
<td>12</td>
<td>80</td>
<td>556</td>
<td>46</td>
<td>4.2</td>
</tr>
<tr>
<td>8</td>
<td>0.73</td>
<td>2222</td>
<td>12</td>
<td>90</td>
<td>556</td>
<td>46</td>
<td>3.8</td>
</tr>
</tbody>
</table>

Figure 6.3 Schematic of drop train selection over time that is fulfilled by controlling single frequency $f_s$ and group frequency $f_b$, and the open angle $\alpha$ of the blocker.
Besides the temperature difference, another concern is regarding the lag between the observation from the underneath and the actual development on the upper surface, so that we consider two Fourier number defined as

\[
Fo_s = \frac{\alpha}{b^2 f_s} 
\]

\[
Fo_b = \frac{\alpha}{b^2 f_b} 
\]

where \(\alpha\) is the thermal diffusivity of the silicon wafer. Here \(Fo_s\) compares the single drop impact time scale to the thermal diffusion time scale through the wafer thickness of \(b\). For the tested ranges of \(f_s\), \(Fo_s \ll 1\). The lag due to thermal diffusion across the wafer is significant and the temperature change between two consecutive drops could not be detected by the IR camera. \(Fo_b\) compares the drop train impact time scale to the thermal diffusion time scale and \(Fo_b \gg 1\). The entire thermal process related to the drop train impact is much shorter than \(1/f_b\), so that the lag due to heat transfer through the wafer is insignificant as compared to the entire time period of the drop train impact.

### 6.3 Results and discussion

#### 6.3.1 Comparison between circular jet cooling with drop train cooling

In jet cooling the high density of heat removal with the jet impingement zone produces elevated spatial temperature gradient and the dry-out area tends to appear at the downstream [110]. In spray cooling numerous of small drops that individually impinges different sites of the heated surface contribute to the uniform surface temperature distribution on the spray-covered area and enhance the contact of coolant drop with the surface, thus resulting in the delay of CHF in the boiling regime [111]. In the studies of the jet cooling and spray cooling there are still two aspects that are not emphasized.
In comparison with jet cooling, the advantage of cooling enhancement by spray cooling is attributed to the fresh coolant drop landing on distinct locations over the whole heated surface. Like jet impingement cooling, in single drop train cooling drop train is produced by the breakup of the continuous jet fluid and lands on the same location. One aspect is about whether heat transfer is enhanced or reduced by continuous drop train impact on the same location of the impact surface as compared to jet cooling. Moreover, in spray cooling the interaction of coolant flow with flowing film significantly affects the local cooling. Hence, we concern the interaction of jet flow and drop train flow with flowing film as well as its effects on local cooling.

To address these two aspects, comparison of circular jet impingement and continuous drop train impact on flowing film is conducted mainly on their effect on local cooling, as shown in Fig. 6.4. The location where the drop train and jet flow lands on the flowing film is the impact point. For all the conducted tests, the distance between the impact point and the stagnation point \( x = 15 \) mm. The film flow rate maintains at \( Q' = 2.8 \) cm\(^2\)/s while the flow rate of jet and drop train \( Q_t = 100 \) ml/M. During jet impingement and drop train impact, the IR camera records surface temperate distribution over the heated surface up to 1000 frames, and surface temperature almost maintains at the steady state even for drop train impact, since the temperature change could be not detected in such a short impact time interval of \( 1/f_s \sim 0.2 \) ms. Each error bar represents standard derivation of surface temperature within 1000 frames of the IR image record.

At the impact point \( x = 15 \) mm, surface temperatures with circular jet impact and drop train impact are compared at varied heat flux ranging from 16.5 to 25.1 W/cm\(^2\), as plotted in Fig. 6.4 (a). Both the circular jet cooling and drop train cooling present the same change trend: the surface temperature goes up with increasing the surface heat flux. The trend line of the jet cooling is below that of the drop train cooling. That is, its surface temperature is higher than that
of the drop train cooling, up to 0.7 °C at $q''=22.1$ W/cm$^2$. The better cooling performance is obtained by the drop train impact at the impact point, and the same conclusion was made in comparison of drop train cooling and jet cooling on the dry heated surface [95].

![Figure 6.4 Comparison of surface temperature change for continuous drop train impact ($f_s=4000$ Hz) and circular jet impact under the same working condition ($Q'=2.83$ cm$^2$/s): (a) at the impact point, (b) around the impact area with the diameter of 10 mm.]

Two reasons may contribute to this phenomenon. Jet flow and drop train are generated by the identical nozzle of $D_{nozzle}=500$ μm but the flow characterizes have been changed during the jet breakup into the drop train. One significant difference is jet diameter and drop train diameter. Jet diameter almost maintains as the same as the nozzle diameter of 500 μm while drop train diameter $D_0$ (see Fig. 6.3) is increased to 860 μm based on Eq. (6.1). A large drop diameter has been demonstrated to enhance the local heat transfer with single drop impacting flowing film [28]. Another key factor is the increase of the contact surface area of the drop train with the flowing film due to the jet breakup. These two reasons lead to the enhanced interaction of drop train with the flowing film as well as a more effective convective heat transfer mechanism.

The similar phenomenon is found around the impact area within the diameter of 10 mm in Fig. 6.4 (b). The average surface temperature around the impact area is calculated and the
temperature difference between two cooling methods becomes even larger around the impact area, up to 1 °C at $q''=22.1$ W/cm$^2$. This is because in drop train cooling the larger drop size would induce more drop-covered area on the heated surface and further transfer more heat to the coolant flow.

### 6.3.2 Heat transfer during drop train impact

To study heat transfer process during the drop train impact, the surface temperature change should be determined by comparing to the steady state. Figure 6.5 plots transient surface temperature change at the impact point prior to and during the drop train impact (see blue line), and the relevant heat transfer coefficient (see red line), which is calculated based on Eq. (5.4). Prior to drop train impact film flow cools the heated surface, and the surface temperature maintains at the steady state, $t=0$ s is when the drop train just lands on the film flow in Fig. 6.5.

![Figure 6.5 Transient surface temperature and heat transfer coefficient recorded at impact point prior to the drop train impact and during the impact ($Q'=2.83$ cm$^2$/s, $f_s=3000$ Hz, $f_b=14$ Hz, $\alpha=30^\circ$).](image)

During the impact, there are two stages: response stage and recovery stage. In the first response stage, the surface temperature quickly reaches the lowest point, showing quick response
to the drop train impact. The surface temperature at the lowest point is the response temperature, denoted by $T_b$ (see Fig. 6.5). In the second recovery stage, after reaching the lowest point, the temperature starts rising and reaches the peak point. The temperature when reaching the peak point is the recovery temperature $T_p$. However, for the test shown in Fig. 6.5, $T_p$ cannot recover to the original temperature at the steady state. This is because the time taken for the recovery stage is too short for the temperature recovery to the steady state before the next drop train comes. For $Q' = 2.83 \text{ cm}^2/\text{s}$ the recovery time was found to be $\sim 200$ ms for the temperature recovery in single drop impact [28] while only $\sim 50$ ms less than $1/f_b$ in the current test. Hence, there is a temperature difference between the recovery temperature and the steady-state temperature when the recovery time is shorter than $\sim 200$ ms and the corresponding $f_b > 5$ Hz.

By comparing the change process of $T_s$ and $h_t$, there are a few observations. The transient heat transfer coefficient, $h_t$, at the impact point is firstly going up and then going down, which is opposite to the change of $T_s$. During one cycle of drop train impact ($83 \text{ ms} < t < 157 \text{ ms}$), $T_s$ decreases, reaching the maximum decrease at $t = 104 \text{ ms}$ (see dash line in Fig. 6.5). Meanwhile, $h_t$ increases and reaches its maximum increase at $t = 91 \text{ ms}$. Hence, $h_t$ is earlier than $T_s$ to reach its maximum change, $\Delta t \sim 13 \text{ ms}$, which is attributed to a faster rate of temperature decrease, i.e. $-(dT_s/dt)_{t=91 \text{ ms}} > -(dT_s/dt)_{t=104 \text{ ms}}$. After reaching the maximum enhancement, $h_t$ continues to decrease even below the steady-state curve, showing $\Delta h < 0$ at 124 ms. Then, the heat transfer coefficient has almost recovered to the steady state, while the temperature still increases from the lowest point. The difference of change process for $T_s$ and $h_t$ is due to the existence of the transient term in Eq. (5.4).
The effect of drop train impact on heat transfer is further investigated by evaluating the difference of steady-state cooling and the transient cooling around the impact area of drop train. Before the drop train impacts on the film, the steady-state cooling of the film flow maintains as shown in Fig. 6.6. From the center toward the edge the temperature increases (see Fig. 6.6 b1), and the center is the stagnation point of the jet impingement serving as the origin point of the coordinates system. The line through the jet stagnation point and the impact point is defined as the center line (see dash line in Fig. 6.6 b1).

At the early stage of the drop train impact (see t= 1ms and 5.3ms shown in Fig. 6.6a), the impact results in a rising crown sheet which expands and breaks up over time. At the same time, the change of the local flow leads to local temperature decreasing toward the lowest point $T_b$, and then the temperature returns to the recovery temperature $T_p$. The temperature change cycle continuously repeats from the recovery state to the response state. To address the temperature change in one cycle, we subtract the transient temperature from the recovery temperature, and obtain the temperature change $\Delta T_s = T_s - T_p$. The contours of the temperature difference $\Delta T_s$ are plotted in Fig. 6 (b2) and (b3), showing a concentric distribution with a cold spot at the center.

The heat transfer coefficient of the steady-state cooling $h_{ss}$ is highest at the stagnation point and decreases toward the downstream, which is calculated based on Eq. (5.5) and shown in Fig. 6 (c1). The drop train impact would strengthen the mixing of the drop and the film flow and decrease the surface temperature. The transient heat transfer coefficient $h_t$ around the impact area can be calculated by Eq. (5.4). To better show drop train effect, the difference between the local steady-state and transient heat transfer coefficients, $\Delta h = h_t - h_{ss}$, is plotted in Fig. 6.6 (c2) and (c3). The heat transfer coefficient quickly increases around the impact point at $t=1$ ms. From
t=1 to 5.3 ms the peak value of the enhancement goes up to ~ 1.5 W/cm²K while the surface area of cooling enhancement continues to enlarge.

The early stage of the change of surface temperature and heat transfer coefficient is shown in Fig. 6.6. To monitor the entire transient process, the temperature change along the center line is plotted for a longer time period up to $1/f_b \sim 80$ ms in Fig. 6.7 (a). The profile of the temperature change is like a valley and the valley continues to deepen until $t=22$ ms. Then temperature comes back towards the recovery temperature and almost reaches at $t=80$ ms. The temperature change is consistent with the response and recovery stages at the impact point observed in Fig. 6.5. Fig. 6.7 (a) also shows that the valley bottom of temperature change moves towards the downstream of the impact point, as a result of the move of crown base in Fig. 6.6 (a).

Figure 6.6 (a) Phenomena of a drop train ($n=21, f_b=5000$ Hz) impacting on flowing film ($Q'=2.83$ cm²/s) at $x=15$ mm; (b1) contour of surface temperature at the steady state, (b2, b3) contour of surface temperature difference $\Delta T_s$; (c1) contour of heat transfer coefficient at the steady state, (c2, c3) contour of heat transfer enhancement around the impact area.

Due to the temperature change, the heat transfer coefficient changes over time. An enhancement factor is introduced for quantifying the change of the convection heat transfer by
\[ \eta = \left( h_t - h_{ss} \right) / h_{ss} \]  

(6.7)

Based on Eq. (6.7) the development of cooling enhancement is plotted over time in Fig. 6.7 (b). At \( t=9 \) ms, most of the impact region has reached local maximum enhancement, denoted by \( \eta_{max} \). The curve at \( t=9 \) ms is referred to as the maximum enhancement curve, showing the distribution of local maximum enhancement. The peak enhancement, denoted by \( \eta_p \) in the \( \eta_{max} \) curve, is up to \( \eta_p \approx 175\% \) and occurs at a location close to the impact point. At \( t=33 \) ms, there is \( \approx 20\% \) decrease of convection, when the local flow changes from flowing upstream to downstream.

Figure 6.7 (a) The surface temperature change \( \Delta T_s \) along the center line over time (\( n=21, f_s=5000 \) Hz, \( Q'=2.83 \) cm\(^2\)/s); (b) heat transfer enhancement along the center line at the steady state and transient state.
Drop train impact affects local flow dynamics in association with the interaction of drop flow with film flow. Meanwhile, their interaction works for the change of convection mechanism, which is presented by the local surface temperature change. To evaluate the enhancement of convection heat transfer, the surface temperature change should be analyzed under different flow conditions, involving the drop number of controlled drop train, the drop velocity, as well as flowing film. These three aspects are emphasized below.

6.3.3 Effects of drop number

Only changing $f_b$ would not affect $f_{avg}$, but change $n$. By increasing $f_b$ the total drop number landing on the flowing film at the unit time maintains constant while the drop number of single drop train ($n$) is decreased based on Eq. (6.4). Its effect on local cooling is evaluated by monitoring the surface temperature change at the impact point $x=15$mm, as shown in Fig. 6.8(a). For all the conducted tests $f_s =5000$ Hz and $\alpha=20^\circ$ gives $f_{avg}=278$ Hz. Varying group frequency $f_b$ from 11 Hz to 25 Hz leads to the change of drop number $n$ from 11 to 24.

![Figure 6.8](image)

Figure 6.8 (a) Transit surface temperature change at impact point with varied generation frequency of drop train $f_b$, leading to the change of corresponding drop number $n$. For all the conducted tests $f_{avg}=278$ Hz. (b) The change of average surface temperature $T_s$ and $\Delta T_c$ with varied $n$. 
Figure 6.8 (a) shows that with drop train impact the surface temperature at the impact point first going down, and once reaching the response temperature $T_b$, then returning towards the recovery temperature $T_p$. Along a center line between $T_b$ and $T_p$ (see dash line in Fig. 6.8 a), the temperature periodically fluctuates and its fluctuation period is $\sim 1/f_b$. A larger drop number $n$ brings the lower $T_b$ and the higher $T_p$, showing a larger temperature fluctuation in one cycle (denoted by $\Delta T_c = T_p - T_b$ in the inset in Fig. 6.8 b). Interestingly, the center line of temperature fluctuation is almost the same for varied $n$, which corresponds to an average surface temperature $\bar{T}_s$ over time. By varying $n$, $\bar{T}_s$ maintains at $\sim 40.4$ °C, as plotted in Fig. 6.8 (b). In the current range of tested experiments, the larger $n$ results in the larger the temperature fluctuation $\Delta T_c$ as plotted in the inset in Fig. 6.8 (b). $\Delta T_c$ is $\sim 1.9$ °C for $n=11$, while $\Delta T_c$ is up to 4.8 °C for $n=24$. The conclusion is made based on experimental observation that with the constant frequency of drop impact $f_{avg}$ changing $n$ would not affect the average temperature $\bar{T}_s$ at the impact point, but significantly change the temperature fluctuation $\Delta T_c$.

Another question proposed is whether changing $n$ would affect the local convection heat transfer. To address this concern, we analyze the change of local surface temperature by varying $n$ during the drop train impact. Along the center line between the stagnation point and the impact point, Fig. 6.9 (a) plots the curves of maximum surface temperature change $\Delta T_{max}$ for the tests with $f_{avg}=278$ Hz. The $\Delta T_{max}$ curves with the larger $n$ encompass the curves with the smaller $n$. Hence, the increase of $n$ not only extends the cooling area towards both upstream and downstream of the impact point, but also enhances convection heat transfer around the cooling enhancement area. Different from the peak surface temperature change appearing downstream of the impact point of single drop impact [28], the peak point of $\Delta T_{max}$
locates nearby the impact point of the drop train impact. This is because as compared to the film flow $Q' \sim 2.8 \text{ cm}^2/\text{s}$ the tested drop train flow has high momentum $U_0D_0$ greater than ~72 cm$^2$/s, which dominates the local flow dynamics and cooling enhancement at the impact point.

In Fig. 6.9 (b) the peak enhancement $\eta_p$ is plotted versus the drop number $n$. The drop number in single drop train shows significant effect: with increasing $n$ the peak enhancement continuously increases, showing a monotonic trend within the current tested range of $n$. The peak enhancement $\eta_p$ is 75% for $n=11$, while $\eta_p$ is up to 212% for $n=24$. The significant cooling enhancement is attributed to the increase of the amount of drop train flow to mix with the film flow cooling the hot surface.

Figure 6.9 (a) the maximum enhancement along the centerline with varied drop number $n$ from 11 to 24, (b) the peak enhancement versus $n$. 
Increasing $\alpha$ would increase both $f_{avg}$ and $n$. To study their effects on local cooling, experimental tests are conducted increasing $\alpha$ from $10^\circ$ to $180^\circ$ while $f_s$ =5000 Hz and $f_b$ =14 Hz. Besides, the special condition of $\alpha$ =360$^\circ$ is studied, which indicates open blocker and $f_b$ =0 Hz so that all the drops pass through the blocker with $f_{avg} = f_s$. Thus, when $\alpha$ increases from $10^\circ$ to $360^\circ$, $f_{avg}$ increases from 139 Hz to 5000 Hz, which is calculated based on Eq. (6.3). Fig. 6.10 (a) plots the transient surface temperature change at the impact point. The larger $f_{avg}$ produce the lower $T_b$ and the larger temperature fluctuation $\Delta T_c$. As plotted in Fig. 6.10 (b), average temperature $\bar{T}_s$ shows a monotonic decreasing trend with the increase of $f_{avg}$. $\bar{T}_s$ ~41.3 $^\circ$C for $f_{avg}$=139 Hz, while $\bar{T}_s$ ~29.3 $^\circ$C for $f_{avg}$=2480 Hz. Only changing $n$ would change $\Delta T_c$ rather than $\bar{T}_s$, which has been demonstrated in Fig. 6.8. Therefore, the change of $f_{avg}$ by varying $\alpha$ works for the change of the average temperature $\bar{T}_s$ at the impact point.

![Figure 6.10](image)

Figure 6.10 (a) Transit surface temperature change at impact point with varied blocker angles $\alpha$ leading to the change of $f_{avg}$, (b) the change of average surface temperature with varied $f_{avg}$.

The peak enhancement is further analyzed by varying $\alpha$ and $f_b$. $\alpha$ changes from $10^\circ$ to $180^\circ$ while $f_b$ ranges from 5 Hz to 28 Hz. Based on Eq. (6.5) with $f_s$ =5000 Hz the drop
number \( n \) can be adjusted from 5 to 500. In Fig. 6.11 experimental data is added for studying the relationship of \( \eta_p \) with \( n \). There are two stages of the trend line. The first stage shows an increasing trend: the larger the drop number \( n \), the higher the peak enhancement \( \eta_p \). \( \eta_p \) is 75 \% for \( n=9 \) while \( \eta_p \) is up to 1130 \% for \( n=389 \). Once reaching the maximum value the trend line almost becomes flat in the second stage. Further increasing \( n \) would not increase \( \eta_p \), and the maximum value of the controlled drop train is almost the same as that of the continuous drop train impact (see the red square symbol in Fig. 6.11). Therefore, \( \eta_p \) shows dependency on \( n \), involving the influence of three parameters: single frequency \( f_s \), group frequency \( f_b \), and open angle of the blocker \( \alpha \).

![Graph showing peak enhancement versus drop number.](image)

**Figure 6.11** The peak enhancement versus the drop number \( n \). The flowing film conditions \( (Q' = 2.83 \text{ cm}^2/\text{s}) \) are the same for all the conducted tests.

### 6.3.4 Effects of drop train velocity

Figure 6.12 shows drop spreading and splashing by high speed images with three drop trains impacting the flowing film. Three drop trains have three different velocities from 4.2 m/s to 17.1 m/s while \( n=46 \). The impact velocity is observed to affect the spreading and splashing.
At the early stage of drop train impact ~4.5 ms, a faster impact velocity seems to cause a quick spreading flow on liquid film, showing a large spreading area, which is expected to enhance the local convection. Additionally, the higher impact velocity shortens the impact time of drop train. At t=4.5 ms, drop trains continuously impact the liquid film at $U_0=4.2$ m/s and 8.5 m/s (see Figs. 6.12 a and b), but no drop train above the impact surface at $U_0=17.1$ m/s (see Figs. 6.12 c). Figure 6.12 also presents that a higher impact velocity results in higher rising sheet and stronger splashing. The rising sheet and splashing may have negative effect on local cooling, as the ejected fluid is considered as local loss of coolant.

To investigate the cooling effect of impact velocity, the transient surface temperature changes for three tests are plotted at the impact point in Fig. 6.13 (a) while average surface temperature $\bar{T}_s$ for more tests are shown in Fig. 6.13 (b). With drop train impact the surface
temperature at the impact point quickly drops down to the response temperature $T_b$, and then returns towards the recovery temperature $T_p$. By increasing impact velocity, the recovery temperature $T_p$ slightly increases but the response temperature $T_b$ significantly increases, as marked by solid symbols. The times taken to reach $T_b$ are different for varied impact velocities, and the lower impact velocity takes more time than the higher impact, which is consistent to the observation of impact time discussed in Fig. 6.12. Figure 6.13 (b) plots average surface temperature $\bar{T}_s$ versus impact velocity. With the increase of impact velocity $\bar{T}_s$ firstly decreases and then increases, and the critical point appears at $U_o=5.7$ m/s.

Figure 6.13 (a) Transient surface temperature change at impact point with varied impact velocities, (b) the change of average surface temperature, (c) the peak enhancement versus impact velocity. The flowing film conditions ($Q'=2.83$ cm$^2$/s) are the same for all the conducted tests.
To further investigate the trend of peak enhancement, $\eta_p$ is plotted versus the impact velocity $U_0$ in Fig. 6.13c. The peak enhancement first increases with increasing the impact velocity from 4.2 m/s to 5.7 m/s, and the higher impact velocity enhances the convection around impact area in this region. Further increasing velocity results in the decrease of $\eta_p$, which is attributed to loss of coolant due to the rising sheet and splashing. This indicates that faster impact does not necessarily cause higher enhancement. Hence, as compared to a monotonic increasing trend of $\eta_p$ with drop number in Fig. 6.11, impact velocity shows less significance on enhancement, and increasing drop number is more likely to obtain better local cooling.

### 6.3.5 Changing film flow rate

As shown in Fig. 6.14, the effects of the film flow are studied by varying film rate $Q'$ from 2.83 to 4.95 cm$^2$/s while the equivalent size of drop train and impact velocity maintain constant ($U_0$=8.5m/s, $n$=42). The landing location of drop train is maintained constant at $x$=15 mm. The maximum enhancement curves of two tests are compared in Fig. 6.14a, showing the local maximum enhancement $\eta_{max}$ along the center line.

Two major effects of the film flow rate are found. First, the location of the peak enhancement $\eta_p$ shifts downstream of the impact point and the displacement from the impact point is denoted by $\Delta x$ (see Fig. 6.14 a). The change trend of $\Delta x$ is presented with varying film flow rate in the inset of Fig. 6.14a. By increasing $Q'$ from 2.83 to 4.42 cm$^2$/s, the higher film flow rate, the larger displacement from the impact point, and the trend line becomes flat from 4.42 to 4.95 cm$^2$/s. For instance, $\Delta x$~0.33 mm at $Q'$=2.83 cm$^2$/s while $\Delta x$~2.33 mm at $Q'$=4.42 and 4.95 cm$^2$/s. This is because when the film flow rate is high, the flow driven by the drop train impact has less time to develop, and quickly move downstream of impact point [112-113].
Second, the higher film flow rate causes the lower peak enhancement of the $\eta_{max}$ curve. To show the trend in detail, $\eta_p$ is plotted versus $Q'$ in Fig. 6.14b. The peak enhancement $\eta_p$ decreases with increasing $Q'$. This can be clearly observed by comparing the lowest with highest film flow rates ($Q'$=2.83 cm$^2$/s and 4.95 cm$^2$/s). For the film flow rate increasing from 2.83 to 4.95 cm$^2$/s, the peak enhancement decreases from ~380% to ~254%.

Figure 6.12 The maximum enhancement along the center line caused by the drop train impact ($n=42, f_b=13$ Hz, $\alpha=40^{\circ}, U_0=8.5$ m/s) on films with varied flow rates $Q'$. (b) The peak enhancement versus film flow rate.
6.4 Summary

Heat transfer of controlled drop train impact on the film-cooled hot surface is studied experimentally. The focus is put on the change of convection heat transfer in relation to the working condition of the drop train impact as well as film flow. An infrared camera with high spatiotemporal resolutions is used to record temperature change.

The temperature measurement shows two stages during the drop train impact: response stage and recovery stage. In the first response stage, the surface temperature quickly reaches the lowest point, which is referred as to the response temperature \( T_b \). In the second recovery stage, the temperature starts rising and reaches the highest point, which referred as to the recovery temperature \( T_p \). The tests show \( T_p \) cannot recover to the original temperature at the steady state when the recovery time is short. The transient heat transfer coefficient \( (h_t) \) presents three steps compared to the steady state: the increase step, the decrease step, and the recovery step. After drop impact, \( h_t \) continuously increases until reaching a maximum value, and then decreases toward and eventually below the steady state. An enhancement factor \( (\eta) \) is introduced to evaluate the cooling enhancement and the peak value of the curve is referred to as the peak enhancement \( (\eta_p) \).

The controlled drop train is fulfilled by adjusting independent parameters: single frequency \( f_s \), group frequency \( f_b \), and open angle of the blocker \( \alpha \). Specifically to the drop number, we apply two parameters to quantify drop flow landing on the surface: \( f_{avg} \) (average drop generation frequency) and \( n \) (drop number in single drop train). The tests show with the constant \( f_{avg} \) changing \( n \) would not affect the average temperature \( \bar{T}_s \) at the impact point while the change of \( f_{avg} \) by varying \( \alpha \) works for the change of the average temperature \( \bar{T}_s \). \( \eta_p \) shows dependency
on \( n \), involving the influence of three parameters: single frequency \( f_s \), group frequency \( f_b \) and open angle of the blocker \( \alpha \). As compared to a monotonic increasing trend of \( \eta_p \) with drop number, faster impact velocity impact does not necessarily cause higher enhancement. Additionally, the higher film flow rate causes the lower peak enhancement.
Chapter 7 Conclusion and future work

7.1 Conclusion

In spray cooling, the spray impingement is actually the impact of spray droplet on flow film. The interaction of drop flow with film flow dominates the local cooling around the impact area. Chapter 3 reveals that changing spray positioning affects both global cooling and local cooling. One of the major reasons is that changing spray positioning results in the change of impact on flowing film for all spray droplets. Chapter 4 & 5 contribute to the fundamental understanding of the flow dynamics and heat transfer associated with single drop impacting flowing film. Chapter 6 contributes to the understanding of the thermal-fluidics involved in the continuous drop impact on flowing film.

In Chapter 3, the spray height and inclination angle are the positioning variables tested in the present work. Spray impact is observed to form three regions on the impact surface: impact area, thin-film region, and thick-film region. Equations are also derived for evaluating the change of spray flux as a function of the nozzle positioning and \((\beta, \varphi)\). Experimental tests show that the optimal spray height providing the most effective cooling is smaller than the height required for covering the entire heater area. This is because the maximum local cooling is located at the edge of the impact area, and the film flow outside the impact area still provides effective cooling. The global cooling shows slight diminishment for small inclination angle and enhancement for large inclination angles. By analyzing the local cooling and the spray flux of the flow along the centerline of the impact area, the enhancement and diminishment of the cooling performance are found to be in general agreement with the increase and decrease of the spray flux.
Next, the focus is put on the drop impact on the film-covered surface, for having a fundamental understanding of drop impact dynamics with film flow in spray cooling. Therefore, there are three topics addressed for drop impact study, and relative conclusions are made below.

In Chapter 4, the fluid dynamics of drop impact on flowing liquid films is theoretically and experimentally studied. The focus of the work is put on drop spreading, stretching of the crown sheet in rising direction, and prediction of splash impact. The theoretical models are developed for predicting the base radius of the crown on stationary films and flowing films during spreading phase. Additionally, based on the modified solution, the local stretching rate at the top of the crown sheet in the rising direction is derived. The highest stretching rate is at the azimuthal location where the drop spread flow is right opposite to the film flow. Finally, the threshold parameter for drops impacting flowing films is defined as a function of modified Weber and Reynolds numbers, and the two modified numbers consider both drop and film flows. The threshold value for the occurrence of splash impact on flowing films is determined based on our experimental observations.

In Chapter 5, heat transfer involved in single water drop impact on a water film flow cooling a hot surface is studied experimentally. Based on the temperature measurement, the drop impact causes the surface temperature to first decrease and then return to the steady state, showing a thermal response process followed by a recovery process. However, the transient heat transfer coefficient \((h_t)\) presents three steps compared to the steady state: the increase step, the decrease step, and the recovery step. To evaluate the change of convection, an enhancement factor \((\eta)\) based on the change of heat transfer coefficient, is introduced. It is shown that the relation of the peak enhancement \((\eta_p)\) with drop velocity does not follow a monotonic trend. With increasing drop velocity, the peak enhancement first increases, then decreases, and
eventually becomes relatively stable, showing low dependency on $U_0$. However, the peak enhancement increases with the increase of drop size in the current experimental range, since a larger drop increases the amount flow added to the impact area. Combining the effects of the drop flow and film flow, the parameters can be grouped into the ratio of the drop flow rate and film flow rate as $U_0D_0/Q'$. The least mean square linear fitting shows $\eta_p \propto (U_0D_0/Q')^{1/2}$.

In Chapter 6, heat transfer of controlled drop train impact on the film-cooled hot surface is studied experimentally. The controlled drop train is fulfilled by adjusting independent parameters: single frequency $f_s$, group frequency $f_b$, and open angle of the blocker $\alpha$. Specifically to the drop number, we apply two parameters to quantify drop flow landing on the surface: $f_{avg}$ (average drop generation frequency) and $n$ (drop number in single drop train). The tests show with the constant $f_{avg}$ changing $n$ would not affect the average temperature $\bar{T}_s$ at the impact point while the change of $f_{avg}$ by varying $\alpha$ works for the change of the average temperature $\bar{T}_s$. $\eta_p$ shows dependency on $n$, involving the influence of three parameters: single frequency $f_s$, group frequency $f_b$ and open angle of the blocker $\alpha$. As compared to a monotonic increasing trend of $\eta_p$ with drop number, faster impact velocity impact does not necessarily cause higher enhancement. Additionally, the higher film flow rate causes the lower peak enhancement.

Generally, the contribution of the study is filling the gap between spray characterization and cooling performance. The relationship between local cooling performance and local spray flux has been demonstrated at varied nozzle positioning. The average frequency of drop impact is found to be the dominant parameter affecting cooling performance in study of drop train impact. These two conclusions provide an available way to enhance spray cooling performance: increasing volumetric flux in spray-covered area. The secondary spray is found during spray
impact while the splashing may occur during drop impact. The bouncing liquid sheet or small droplets away from impact surface could not contribute to the local cooling. That is the reason that increasing drop impact velocity does not necessarily enhance the local cooling performance when the local coolant loss is significant. The splashing threshold proposed in the study of single drop impact tells us how to avoid the coolant loss from the impact surface: decrease the impact velocity, increase surface tension or viscosity of the coolant. The fundamental studies of drop impact provide methods of cooling enhancement in spray cooling.

The limitation of the current study includes two aspects. The research results of drop impact cannot directly apply to the practical application of spray cooling technology, since the study of drop impact does not consider the interaction of multiple groups of single drop impact or drop train impact on different sites of impact surface. For example, the formation of the uprising liquid sheet between two drop spreading flows would worsen the local cooling where the spreading flows become weak. Another limitation is regarding the flowing film. The flowing film in spray cooling is generated by numerous drops which impact on solid surface, while it is formed by the jet impingement in the current study. This also may result in different cooling performance between drop impact and spray impact.

7.2 Further work

The future work will focus on the cooling of multiple drop trains impacting on a hot surface. There are three reasons: 1) Impact of multiple drop trains can better simulate the drop impact in spray cooling. The multiple drop train can form a flowing film by themselves to cool the hot surface. This enables the cooling technology of multiple drop trains directly using for the practical application. 2) Cooling by multiple drop trains could be used as a novel spray cooling.
The property of drop train is predicted with the stable breakup of the liquid jet, which contributes to the predictable cooling performance on the hot surface and to avoiding the occurrence of the dry-out area in spray cooling application. 3) Impact of multiple drop trains has a huge potential in the high-heat-flux removal application. The generation frequency of single drop is up to 50 KHz using the high frequency drop generator. The higher drop generation frequency, the more coolant drops added to cool the hot surface, and the higher cooling enhancement. Further study can produce predictable and controlled spray cooling.
Bibliography


Appendices

Appendix A Microfabrication Processes of Thin-film Heater

Appendix A describes the main process of microfabrication of thin-film heater used in the study of spray cooling in Chapter 3, heat transfer of single drop impact in Chapter 5, and heat transfer of drop train impact in Chapter 6. The sputtering coating and UV photolithography processes for the fabrication of thin-film heater are given.

Sputter Coating Process:

The thin-film heater contains three layers including Si$_3$N$_4$ layer serving as the dielectric layer, Cr layer as the adhesive layer, and Au as the heating layer. The silicon wafer is sputter-coated firstly with Si$_3$N$_4$ layer followed by Cr layer and Au layer. The detailed parameters of sputter coating are shown below.

<table>
<thead>
<tr>
<th>Material</th>
<th>Si$_3$N$_4$</th>
</tr>
</thead>
<tbody>
<tr>
<td>Sputter system</td>
<td>Armstrong Engineering Corp. Nexdep Deposition System</td>
</tr>
<tr>
<td>Power Source</td>
<td>DC (Direct Current)</td>
</tr>
<tr>
<td>Process gas</td>
<td>Argon</td>
</tr>
<tr>
<td>Process gas pressure</td>
<td>3.0 mTorr</td>
</tr>
<tr>
<td>Gas flow rate</td>
<td>8.06 sccm</td>
</tr>
<tr>
<td>Deposition rate</td>
<td>1 Å per second</td>
</tr>
<tr>
<td>Final thickness</td>
<td>50 nm</td>
</tr>
<tr>
<td>Deposition time</td>
<td>Approximately 10 minutes</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Metal</th>
<th>Cr</th>
</tr>
</thead>
<tbody>
<tr>
<td>Sputter system</td>
<td>Armstrong Engineering Corp. Nexdep Deposition System</td>
</tr>
</tbody>
</table>
Power Source: AC (Alternating Current)

Process gas: Argon
Process gas pressure: 3.0 mTorr
Gas flow rate: 8.06 sccm
Deposition rate: 1 Å per second
Final thickness: 50 nm
Deposition time: Approximately 10 minutes

Metal: Au
Sputter system: Armstrong Engineering Corp. Nexdep Deposition System
Power Source: AC (Alternating Current)
Process gas: Argon
Process gas pressure: 3.0 mTorr
Gas flow rate: 8.06 sccm
Deposition rate: 1 Å per second
Final thickness: 50 nm
Deposition time: Approximately 10 minutes

**UV Photolithographic Process:**

Material: 50/50/50 nm Si₃N₄/Cr/Au
Photoresist: MICROPOSIT® S1813 Positive
Spinning Coating: Operate by Laurell Technol. Corp. WS-650 Spin Processor
  First step: Spread at 250 rpm for 30 seconds;
  Second step: Spread at 500 rpm for 30 seconds;
Third step: Spread at 2000 rpm for 60 seconds (~2 μm thickness)

Soft Baking: Operate by Fisher Isotemp 500 Economy Vacuum Lab Oven

Bake at 110 °C for 10 minutes

UV Exposure: 45 seconds at 11.5W/cm² (OAI Model 200 Mask aligner)

Development: Develop for approximately 60 seconds (MICROPOSIT®MF-319)

Hard Baking: Operate by Fisher Isotemp 500 Economy Vacuum Lab Oven

Bake at 120°C for 60 minutes

Etching: Dip in Au etchant for 58 seconds (Transene Gold Etchant TYPE TFA)

Dip in Cr etchant for 15 seconds (Transene Chromium Etchant TYPE 1020)

Removal: Dip in photoresist remover for 10 seconds (MICROPOSIT®Remover 1165)
Appendix B Matlab Code for Transient Surface Temperature Change

This Matlab code is made for obtaining transient surface temperature change along center line (see an example in Fig. 6.7).

```matlab
clc
clear all
close all
gap = 1/240;
threshold_plot = -6; % Set up maximum surface temperature change
PathName = uigetdir('Select the dir'); % Input the data from the target folder I
PathName2 = uigetdir('Select the dir'); % Save the data in the target folder II
FileList = dir(PathName);
NumFileList = length(FileList) - 2;
reference = csvread([PathName '\ FileList(3).name]);
[length width] = size(reference);

% matrix (length * width * NumFileList-1): all initial values are zeros
grid = zeros(length, width, NumFileList-1);
time = gap:gaps*(NumFileList-1);
time = time*1000;
% read csv file
```
for i = 4:NumFileList+2
    captured = csvread([PathName \ FileList(i).name]);
    substrated = reference - captured;
    grid(:,:,i-3) = substrated;
end

% derive data of surface temperature change
max = max(max(max(grid)));
[x, y, z] = ind2sub(size(grid),find(grid == max));
xlswrite([PathName2 \ 'tempdiff_minus.xls'], reshape(-grid(x,:,:),width,(NumFileList-1)));

% derive data surface temperature along center line
for i = 4:NumFileList+2
    captured = csvread([PathName \ FileList(i).name]);
    temp_max(:,1) = reference(x,:);
    temp_max(:,i-2) = captured(x,:);
end
xlswrite([PathName2 \ 'temp.xls'], temp_max);

% derive single plot
for i = 1:(NumFileList-1)
    figure
    linex=plot(-grid(x,:));
xlabel('$\text{\it x} \ (m)$', 'Fontname', 'Times New Roman', 'Fontsize', 16);
ylabel('$\Delta T_{s \_1} \ (^0C)$', 'Fontname', 'Times New Roman', 'Fontsize', 16);
legend(['t = ' num2str(i*(1/240))], 'Location', 'Northwest', 'FontSize',18);
axis([0 45 threshold_plot 0]);

% replace pixels by unit mm in xlabel
set(gca, 'xtick', [0 7.5 15 22.5 30 37.5 45], 'xticklabel', {'0', '5', '10', '15', '20', '25', '30'});
set(linex, 'Linewidth', 4);
saveas(gca,[PathName2 'FileList(i+2).name '.png]);
end

% ribbon plot
figure
ribbon(reshape(-grid(x,:,:),width,(NumFileList-1)));xlabel('Time (1/240 s)','Fontname', 'Times New Roman', 'Fontsize', 14);
ylabel('$\text{\it x} \ (mm)$', 'Fontname', 'Times New Roman', 'Fontsize', 16);
set(gca, 'ytick', [0 7.5 15 22.5 30 37.5 45], 'yticklabel', {'0', '5', '10', '15', '20', '25', '30'});
set(gca, 'ztick', [0 7.5 15 22.5 30 37.5 45], 'zticklabel', {'0', '5', '10', '15', '20', '25', '30'});
set(gca, 'zlabel', '$\Delta T_{s \_1}(^0C)$', 'Fontname', 'Times New Roman', 'Fontsize', 14);