Experimental and theoretical studies of spray cooling for high power electronics

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EXPERIMENTAL AND THEORETICAL STUDIES OF SPRAY COOLING FOR HIGH POWER ELECTRONICS

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Abstract

Abstract

Spray cooling as a promising thermal management scheme has received much attention from modern industrial and technological applications, such as power electronics, nuclear power generation, high-power lasers and high-power conversion systems. The present work focused on the fundamental investigations of the spray characteristics of spray nozzles in free spray atomisation and in spray cooling. Theoretical models were developed to study the heat transfer in the non-boiling and boiling regimes of spray cooling. A closed loop system was developed to study the applications of using a multi-nozzle array on large area high heat load electronics cooling. Furthermore, the effects of structured surfaces on the spray cooling performance were scrutinised.

Two optical techniques, Phase Doppler Anemometry (PDA) and Particle Image Velocimetry (PIV) were used to characterise the spray structures of pressure swirl nozzles. In free spray atomisation, the spray characteristics were found to be highly dependent on the axial distance. The spray cone produced by the pressure swirl nozzles evolved from a hollow spray cone to a full spray cone with the increase of axial distance. In spray atomisation under spray impingement, the spray cone formation of pressure swirl nozzle is largely dependent on the temperature of the impinged surface. The spray cone of a pressure swirl nozzle expands after impinging on a surface with a relatively high temperature. As a result, the impinging droplet flux near the centre of the spray cone decreases and the spray cone changes from a full spray cone to a hollow spray cone. The heat transfer experiments show that the effects of nozzle-to-surface distance on the heat transfer performance are complex and dependent on surface temperature. The expansion of the spray cone has

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significant effects on the surface temperature non-uniformity and heat transfer coefficient in spray cooling.

A thin film flow model was developed to estimate the thickness of the liquid film formed under spray impingement. On the basis of the thin film flow model, a heat transfer model was developed to study the heat transfer in the non-boiling regime of spray cooling. The modelling results showed that the local film thickness was sensitive to the local droplet flux density and the droplet impingement cooling was the primary heat transfer mechanism in the non-boiling regime of spray cooling. To better understand the heat transfer in the boiling regime of spray cooling, a numerical model based on the experimental spray characteristics was proposed to investigate the dynamics of droplet impingement, bubble boiling as well as their interplay in the spray cooling process. The effects of heat flux, droplet diameter, and droplet impingement frequency on the dynamics of bubble boiling were investigated. The numerical model showed that the fluxes of the collapsed bubbles due to the limited bubble size, as well as the punctured bubbles due to the droplet impingement increased as heat flux increased. A smaller impinging droplet is favourable for bubble boiling due to the more secondary nuclei induced as well as the larger fractions of the bubbles punctured at bigger diameters.

A prototype of a high power closed loop spray cooling system using R134a as the working fluid was constructed to study the feasibility of using a multi-nozzle array on a 6U electronic card cooling. Fifty four pressure swirl nozzles were assembled in an array of 9×6 to cover a 6U card surface. Simple drainage concepts were introduced in the spray chamber design. The experimental results show a promising prospect of using multi-nozzle arrays on large area power electronics cooling. Heat removal of 16 kW is achieved on the 6U card surface by maintaining

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the surface temperature below 26.5°C. High heat transfer coefficient (up to 2.8 W/cm²·K) and high liquid evaporation fraction (up to 0.88) are obtained. Without affecting the surface temperature non-uniformity, the control of chamber pressure is able to maintain the same operating temperature of a device at different heat loads.

Experiments were conducted to study the thermal effects of differently scaled structure surfaces in an R134a spray cooling system. Results show that the arrangement of macro- fins plays a more important role in the cooling performance of macro-structured surfaces rather than a simple increase in the wetted area. Micro-structures improve the heat transfer performance by enhancing the capillary effects and providing more potential nucleation sites on the heated surface. Taking the smooth flat surface as a reference, the micro-structured flat surface achieves a relative heat transfer enhancement of 32% compared to the 36% of the macro-structured surfaces, while the multiscale-structured surfaces which combine the features of micro- and macro- structures gain a heat transfer enhancement of up to 65%. Besides, the macro-structured surfaces prolong the transition period before CHF occurs and shorten the duration that the heated surface remains in film boiling after the occurrence of CHF while powering off the heat source.

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Nomenclature

Nomenclature

A	area, [m ²]
c	velocity of light, [m/s]
C_d	discharge coefficient, [-]
$C_{p,l}$	specific heat of the liquid, [J/kg·K]
$C_{p,v}$	specific heat of the vapour, [J/kg·K]
d_0	orifice diameter, [m]
d_{eq}	equivalent size of the heater surface, m
d_s	heated surface diameter, [m]
D	diameter, [m]
D_{32}	Sauter mean diameter, [m]
f	frequency, [Hz]
g	gravitational acceleration, [m/s ²]
H	nozzle-to-surface distance, [m]
h	heat transfer coefficient, [W/m ² ·K]
$h_{i,l}$	enthalpy of the inlet liquid, [J/kg]
$h_{o,v}$	enthalpy of the outlet vapour, [J/kg]
$h_{o,l}$	enthalpy of the outlet liquid, [J/kg]
h_{fg}	latent heat of vaporization, [J/kg]
h_{film}	liquid film thickness, [m]
m	mass flow rate, [kg/s]
$\dot{m}$	mass flux, [kg/m ² ·s]
m_{eff}	effective mass flow rate, [kg/s]
$m_{i,l}$	inlet liquid mass flow rate, [kg/s]
$m_{o,l}$	outlet liquid mass flow rate, [kg/s]
$m_{o,v}$	outlet vapour mass flow rate, [kg/s]
N	number of droplets, [-]
$\hat{N}$	droplet number flux, [/m ² ·s]
p	pressure, [Pa]

Nomenclature

Q	heat load, [kW]
q''	heat flux, [W/m ²]
q''_{steady}	maximum steady state heat flux, [W/m ²]
R	radius of the heated surface substrate, [m]
r	euclidean distance from the z axis to the point, [m]
r_{bub}	radius of growing bubble, [m]
S_i	momentum source in r direction, [kg/m·s]
T_{boiling}	boiling temperature, [°C]
T_{film}	thin film temperature, [°C]
T_{sat}	saturated temperature of the liquid, [°C]
T_{spray}	spray liquid temperature, [°C]
T_{surf}	surface temperature of the wall, [°C]
T_{ss}	degree of surface superheat, [°C]
u	radial velocity in the thin film, [m/s]
V	axial droplet velocity, [m/s]
$\bar{V}$	number-averaged axial droplet velocity, [m/s]
$\bar{V}$	mass-averaged axial droplet velocity, [m/s]
w	axial velocity in the thin film
Z	axial distance from the PDA measuring plane to nozzle orifice, [m]

Dimensionless parameters

Ja	Jacob number, $Ja = C_{pl} \Delta T_{\text{sat}} / h_{fg}$, [-]
Re	Reynolds number, $Re = \rho_l V h_{\text{film}} / \mu$, [-]
We	Weber number, $We = \rho_l V^2 d_{32} / \sigma$, [-]

Greek

α	thermal diffusivity, [m ² /s]
β	thin film dispersed angle, [°]
ε	liquid evaporation efficiency, [-]
η	heat transfer effectiveness of droplet impingement, [-]
k	thermal conductivity, [W/m·s]

Nomenclature

μ	dynamic viscosity of fluid, [Pa·s]
θ	non-dimensional surface temperature, [-]
ϑ	non-dimensional surface superheat, [-]
ρ	density, [kg/m ³]
σ	surface tension, [N/m]
τ	shear stress, [Pa]
ξ	heat transfer contribution by droplet impingement, [-]
Σ	summation, [-]

Abbreviations

CHF	Critical Heat Flux
PDA	Phase Doppler anemometry
PDPA	Phase Doppler Particle Analyzer
PIV	Particle image velocimetry

Chapter 1

Introduction

1.1 Background

Electronics development is towards designing smaller devices with higher performance speeds and more miniaturised sizes. Such targets can only be achieved at the expense of high thermal dissipation. To date, some of these devices and systems, such as Insulated Gate Bipolar Transistor (IGBTs), MOS controlled Thyristors (MCTs), Laser Diode Arrays (LDAs), Multi-Chip Modules, and other higher-power defense electronics are dissipating more than 1000 W/cm^2 (Silk *et al.*, 2006; Neudeck, 2007). Such massive waste heat should be removed as fast as possible or else the operating temperature of the devices would reach an extremely high level and might cause the devices to fail (Ghodbane and Holman, 1991; Holman and Kendall, 1993; Mudawar, 2001). Due to the inherently limited heat exchange area of electronic devices, high heat flux removal schemes are essentially crucial for their proper functionality and reliability.

Traditional thermal management technologies, such as single-phase and two-phase techniques, have limited heat flux removal capabilities that cannot meet the requirements for power electronics. A comparison on the cooling capabilities of different cooling techniques is summarised by Finch and Ballew (2009) and shown in Fig. 1.1. With the surface area of 1 cm^2 and surface temperature of 30°C higher than the cooling media (air or liquid), the amount of heat rejection is determined by the heat transfer coefficient. For natural air convection, the working principle is that air on the heat source is heated and rises, thereafter the surrounding cooler air moves

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to the hot area where the cooler fluid is then heated and the process continues. Air forced convection removes heat by creating a high speed air flow over the hot surface whereby a higher heat flux is obtained as a result of the increased heat transfer coefficient compared to natural air convection. Liquid forced convection has a better heat removal capability in comparison to the first two techniques due to the higher sensible heat of liquid compared to air. Compared to these single-phase cooling schemes, phase change heat transfer has a much higher cooling capability due to the usage of latent heat. Direct spray cooling outperforms liquid immersion boiling several times. This is primarily due to the fact that the impinging droplets can agitate the thermal boundary layer on the hot surface and break up premature bubbles in the liquid film. It has been reported that the heat transfer mechanisms, such as forced convection, thin film evaporation, nucleate boiling and transient conduction (or three-phase contact line), are all involved in the spray cooling process. Compared to the other high heat flux removal techniques such as microchannels, jet impingement, thermosyphons and heat pipes, spray cooling has been proposed as the most promising approach in rejecting high heat flux levels in future power electronics systems due to its prominent advantages such as high cooling capacity, isothermal surface temperature, minimum liquid inventory and high refrigerant evaporation efficiency (Glassman, 2005; Kim, 2007).

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Fig. 1.1 Comparison of cooling methods (Finch and Ballew, 2009)

Spray cooling works by distributing cold liquid from a spray nozzle in the form of fine and high velocity droplets onto the device that is being cooled. Therefore, the spray characteristics of spray nozzles naturally affect the heat transfer performance of spray cooling. Previous studies have found that droplet velocity has the most significant impact on critical heat flux (CHF) followed by droplet flux, while droplet diameter has negligible effect (Chen *et al.*, 2002; 2004). Chen *et al.* (2002; 2004) have found that a diluter spray with a higher velocity is more effective in heat transfer than a denser spray with a lower velocity. Cheng *et al.* (2010; 2012) have reported that a uniform spray density distribution is favourable to maintain a uniform surface temperature on the heated wall.

In the past decades, most of the studies on spray cooling have been performed with a single commercial nozzle on small heated surface areas ($< 2\text{ cm}^2$) at high surface

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temperatures (Ortiz and Gonzalez, 1999; Visaria and Mudawar, 2008; Panao and Moreira, 2009; Cheng *et al.*, 2010). However, only a few investigations used multi-nozzle arrays to remove high heat flux from large surface areas in closed loop systems (Li *et al.*, 2006; Ahn *et al.*, 2010; Yan *et al.*, 2010). The primary challenge of using multiple nozzles to cover a large surface area for spray cooling is the complexity in system control and liquid delivery (Glassman, 2005). Moreover, the thermal performance of multi-nozzle arrays has been found to deteriorate when compared to that in a single nozzle application due to the interaction between multiple spray cones (Lin and Ponnappan, 2003; Pautsch and Shedd, 2005). Shedd (2007) reported that using multiple nozzles spray cooling technology to cover a larger surface area should overcome the challenges in managing the excessive liquid on the heated surface.

With an increasing demand for higher power electronic systems, the necessity for high heat flux rejection over large surface areas is growing rapidly (Bhunias and Chen, 2010). With this motivation, our approach to realise spray cooling technique in commercial electronic applications is to develop an applicable spray cooling system which will be able to dissipate high heat fluxes and cover the cooling surface area from tens to thousands of square centimetres.

1.2 Objectives and scope

Although spray cooling has many attractive features, the heat transfer mechanisms are not well understood. Studies using a multi-nozzle array to cool a large surface area are still scarce. The main objective of this project is to characterise the spray atomisation of spray nozzles operating at different conditions (e.g., nozzle pressure drops, axial distances) and study the effects of spray characteristics (e.g., droplet

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diameters, velocities, spatial distributions) on the heat transfer performance hence shed some light on the heat transfer mechanisms in spray cooling. Moreover, the thermal effects on the spray cone formation of spray nozzles in spray cooling will be scrutinised as well. These studies will offer an in-depth understanding on the relationship between the spray characteristics and the heat transfer performance. Also, these studies will provide the essential information of spray characteristics for theoretical modelling. This project also aims to fill the gaps between the theoretical studies and practical applications by establishing prototypes of high power, closed loop spray cooling systems using multi-nozzle arrays.

The scope of the project is essentially as follows:

- To explore the spray patterns of pressure swirl nozzles using a Phase Doppler anemometry (PDA) system. This characterises the spatial distributions of droplet diameter, velocity, and volumetric flux under different nozzle pressure drops and axial distances. In addition, the heat transfer experiment using a single spray nozzle to cool a target surface area is conducted to investigate the effects of spray characteristics on the heat transfer performance by varying the nozzle pressure drop and nozzle-to-surface distance.
- To characterise the thermal effects on the spray cone formation of pressure swirl nozzles in spray cooling. Two different optical techniques, PDA and Particle image velocimetry (PIV), are used.
- To develop a semi-analytical model so as to investigate the thin liquid film flow on the impinged surface and the heat transfer in the non-boiling regime of spray cooling.

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- To derive a numerical model to simulate the spray cooling process in the boiling regime based on the experimentally obtained spray characteristics.
- To develop and test a prototype of a high power closed loop spray cooling system using a multi-nozzle array. This study aims to simulate the application of spray cooling on a 6U electronic card (23.3 cm × 16.0 cm) under high heat loads, and thereafter establish practical guide in designing applicable high power spray cooling systems for power electronics cooling.
- To study the effects of structured surfaces on the heat transfer performance in spray cooling. This experiment aims to simulate the application of spray cooling on a small surface area (2.0 cm × 1.0 cm) under high heat flux dissipation.

1.3 Outline of the thesis

Chapter 1 briefly introduces the spray cooling technology. The advantages of spray cooling compared to other conventional thermal management schemes are discussed.

Chapter 2 reviews the phenomena of a single droplet impacting a flat solid surface, as well as the heat transfer mechanisms, parametric effects, and single- and multiple nozzles applications in spray cooling.

In Chapter 3, the spray characteristics of pressure swirl nozzles in free spray, such as droplet diameter, velocity and flux, are investigated using a Dantec PDA system. An open loop heat transfer experiment of the spray nozzles operating at different pressure drops and nozzle-to-surface distances is presented and discussed.

Chapter 4 studies the thermal effects on the spray cone of a pressure swirl nozzle in the spray cooling process. The variations of the spray cone with increasing surface

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temperature are characterised using the PDA and PIV techniques. The effects of the varied spray cone on the heat transfer performance are investigated and discussed.

In Chapter 5, based on the spray characteristics obtained from the PDA measurement, a theoretical model which focuses on the non-boiling regime of spray cooling is derived in terms of the momentum, mass and energy conservations.

In Chapter 6, a phase change heat transfer model developed by combining the aspects of droplet impingement cooling and bubble boiling in the formed liquid film is presented. The dynamics of droplet impingement, bubble boiling as well as their interactions on heat transfer are investigated.

Chapter 7 presents a prototype design of a high power spray cooling system to reject a high heat load from a 6U electronic card dimensions ($23.3 \text{ cm} \times 16.0 \text{ cm}$). Systematic experiments are tested by varying one of the operating parameters but keeping others nearly constant. Effects of the operating parameters on surface temperature, surface temperature non-uniformity, and CHF are investigated and discussed.

Chapter 8 focuses on the heat transfer enhancement of spray cooling by modifying surface properties, such as fabricating micro- structures, macro- structures, and combination of the previous two on a small heat transfer surface ($2.0 \text{ cm} \times 1.0 \text{ cm}$). The heat transfer performance of the structured surfaces is then compared to the flat surface without modification.

In Chapter 9, the thesis is ended with conclusions and future work.

Chapter 2

Literature Review

2.1 Single droplet impingement

From single droplet to spray impingement, there is a controversy in the research community on the extendibility of individual droplet research to spray cooling applications due to the fact that droplet-to-droplet interactions are not considered in the individual droplet impingement study. One could argue that single droplet studies cannot be extrapolated or used to predict spray behaviour since single trains of droplet do not represent the whole spray phenomena. However, without isolating droplet-to-droplet interactions and factors such as droplet collision, droplet spacing, will make the in-depth study of spray cooling almost impossible. In order to gain a better understanding of the fundamental physical mechanisms of spray impingement, research efforts have been focused on single or monodispersed droplets investigation in both hydrodynamic and heat transfer perspectives.

2.1.1 Hydrodynamics of single droplet impingement

The simplest form of hydrodynamics is the impact of a droplet on a dry surface without considering thermal effects. When a single droplet impacts a dry surface, the droplet impingement behaviour depends on many factors, such as the surface topography, impact velocity and droplet size. In most of the cases, the droplet spreads into a thin film immediately when the droplet impacts onto the surface, thereafter the thin film behaves differently at different stages. At the starting stage, the thin film grows into a corolla and propagates radially from the impinging droplet. The end rim that grows at the edge of the corolla is sometimes unstable and develops fingers of liquid. The fingers eventually recede back or break up into small droplets

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due to the Plateau-Rayleigh instability. According to the previous studies, the most important parameter in characterising the phenomena in droplet impingement is the droplet impact energy (Bai and Gosman, 1995; Rioboo *et al.*, 2001). As shown in Fig. 2.1, Rioboo *et al.* (2001) have identified the droplet impingement process into the following stages:(1) stick, (2) spread, (3) splash and (4) rebound, based on the droplet impact energy. In particular, the splashing or disintegration mechanisms of droplet impingement can be sub-divided into other four stages: (1) prompt splash, (2) corona splash, (3) receding breakup and (4) partial rebound.

Fig. 2.1 Summary of droplet behaviour upon impact (Rioboo *et al.*, 2001)

To theoretically understand the geometry generation during droplet impact, Bejan and Gobin (2006) derived a dimensionless group G ($G = (WeRe^{1/2})^{1/5}$) which is the ratio of two lengths: the final radius of the disc that dies viscously divided by the radius of the still inviscid ring that just wrinkles. When $G \leq O(1)$, the splat stops flowing when it is round, and consequently has no future as a shape other than the round disc at rest. When $G \geq O(1)$, the wrinkles occurs on the rim of the splat (ring),

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and consequently splashes into needles (fingers). They optimised the number of needles (fingers) such that the total splash time is minimum.

$$n_{\text{opt}} \sim \left(\frac{\text{WeRe}^{1/2}}{4 \times 10^2} \right)^{2/5} \quad (2-1)$$

$$t_{\text{min}} \sim \frac{5}{4^{4/5}} \left(\frac{10^2}{\text{WeRe}^{1/2}} \right)^{1/5} \quad (2-2)$$

Where, We and Re are the Weber number and Reynolds number of the impact droplet, respectively.

The more complicated case is the impact of a droplet on a liquid film, which is the common case in spray cooling when a liquid layer forms on the hot surface. Roisman *et al.* (2006) studied the impact of individual droplets on a liquid film, and found that the impingement behaviour of droplet not only depends on droplet impact energy (characterised by Webber number We) but also the thickness of the liquid film. They described the behaviours of a droplet impacting on a liquid film by following the categories of droplet impact energy: (1) At $\text{We} < 10$, the droplet deposits and coalesces over the liquid film. (2) At $10 < \text{We} < 300$, a crater is formed on the liquid film at the point where the droplet impacts. (3) At $\text{We} > 300$, a symmetric or asymmetric corona will be generated and splash, whereby the liquid film fluctuates to destruct splashing crown, and subsequently create secondary droplets.

The effect of liquid film thickness on droplet impingement can be distinguished according to the ratio of roughness amplitude to liquid film thickness, as well as the ratio of droplet diameter to liquid film thickness (Tropea and Marengo, 1999). Based on the magnitude of film thickness, four categories of droplet impacting a liquid film

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follows, (1) When the film thickness and surface roughness amplitude have the same order of magnitude, the droplet behaviour depends on surface roughness; (2) When the film thickness is 1.5 times thicker than droplet diameter, the dependence of droplet impingement behaviour on surface roughness becomes weaker; (3) When the film thickness is 1.5 to 4 times thicker than droplet diameter, the droplet impact behaviour is only dependent on the thickness of the liquid film; (4) When the film thickness is more than 4 times thick of droplet diameter (which actually can be considered as a liquid pool), the droplet impact behaviour is totally independent of surface roughness and film thickness.

2.1.2 Heat transfer phenomena

In the literature, efforts have been made to investigate the heat transfer and hydrodynamic mechanisms of a single droplet impacting onto a hot surface (Chandra and Avedisian, 1991; Manzello and Yang, 2002; Jian *et al.*, 2009; Peng, 2010).

Like most boiling systems, Bernardin *et al.* (1997) found that heat transfer for impinging droplets can be categorised into four different heat transfer regimes as identified in pool boiling: (1) Single-phase film evaporation ($T_{\text{surf}} < T_{\text{sat}}$). In this regime, the droplet spreads on the hot surface to transfer heat mainly through conduction and convection with negligible effect of phase change; (2) Nucleate boiling ($T_{\text{sat}} < T_{\text{surf}} < T_{\text{CHF}}$). Heat transfer in this regime is mainly driven by bubble nucleation inside the spreading liquid film during droplet impingement; (3) Transition boiling ($T_{\text{CHF}} < T_{\text{surf}} < T_{\text{Leidenfrost}}$). Under this situation, an unstable insulating vapour layer is formed at the liquid-solid interface where the droplet spreads but most parts of the formed liquid film may not touch on the hot surface.

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Hence, heat transfer decays in this regime until a minimum heat flux is obtained at the Leidenfrost temperature. (4) Film boiling ($T_{\text{surf}} > T_{\text{Leidenfrost}}$). When surface temperature reaches the Leidenfrost point, a stable vapour film is formed on the hot surface, where the droplet may directly rebound before it is able to touch on the hot surface. Therefore, the main mechanisms of heat transfer in this regime are conduction and radiation through the vapour film.

In these four regimes, the liquid-solid contact area is the key parameter for heat transfer efficiency (Pedersen, 1970). In the visualisation experiments, direct contact between the liquid phase and solid phase is achievable in the single-phase and nucleate boiling regimes which are also known as the wetting regime from the hydrodynamic viewpoint. When surface temperature reaches the transition regime, the contact becomes unstable and intermittent. In the film boiling regime, a vapour film forms between the liquid and solid phases to reach a non-wetting regime. Extended research done by Shen *et al.* (2010) also distinguishes the total droplet evaporation process in four stages: (1) impact (liquid temperature increases rapidly), (2) boiling (liquid temperature increases, and results in large oscillations of the fluid near the surface), (3) nearly constant diameter evaporation (liquid temperature becomes constant) and (4) final dry-out period.

Research has also been performed to investigate the effect of droplet impact velocity on the heat transfer performance of droplet impingement. Mehdizadeh and Chandra (2006) photographed different stages of droplet impingement under high impact velocity using a rotating flywheel. The visualisation experiments show that the maximum spreading diameter increases with impact velocity, whereas the amount of superheat needed decreases with impact velocity. At the fixed impact velocity,

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the maximum spreading diameter of droplet increases with surface temperature, which could be due to the reduction in liquid viscosity and surface tension as surface temperature increases.

The boiling phenomena as well as the collision behaviour of droplet impingement are dependent on the materials of solid surface and surface topography (Bernardin *et al.*, 1997; Fujimoto *et al.*, 2010; Negeed *et al.*, 2010; Shen *et al.*, 2010). Bernardin *et al.* (1997) reported that surface texture influences the boiling behaviour of droplet impingement in two major ways: (1) surface roughness creates more nucleation sites at the lower temperature corresponding to a nucleate boiling and the early portion of transition boiling regimes; (2) surface roughness induces violent breakup of the spreading droplet corresponding to the later portion of transition boiling and film boiling regimes. They also found that the droplet lifetime decreases at a lower surface temperature on a rough surface, which is probably due to the enhanced nucleate boiling. Negeed *et al.* (2010) conducted a series experiments of a single droplet impacting on a hot surface. Three kinds of materials (stainless steel SUS304, brass, aluminium) with different degrees of roughness ($R_a = 0.04, 3.0, 10.0 \mu\text{m}$) were used in the experiments. The authors found that both droplet contact time and droplet evaporation time are shorter on the hot surface with higher surface thermal properties, whereas the maximum droplet spreading diameter is bigger. With increasing surface roughness, the droplet contact time decreases. Fujimoto *et al.* (2010) investigated the effect of surface material on boiling behaviour of droplet impingement by fabricating an Inconel alloy flat surface planted with $3 \mu\text{m}$ diameter diamond particles, the boiling phenomena on which was compared to the Inconel flat surface without diamond particles, as shown in Fig. 2.2. They observed that the nucleation bubbles inside the droplet which impacted on the diamond planted

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surface were much more than that of the droplet impacting on the surface without diamond particles. It could imply that the heterogeneous nuclei rather than homogenous nuclei dominate the boiling phenomena. Shen *et al.* (2010) concluded that the structured surfaces enhance nucleate boiling by increasing the spreading diameter of the droplet impacting on the structured surfaces.

Fig. 2.2 Effects of surface conditions on droplet impingement boiling phenomena (Fujimoto *et al.*, 2010)

2.2 Spray cooling

Spray cooling occurs by forcing liquid through a small orifice, causing it to disperse into fine droplets, which then impact onto the surface being cooled. As the droplets continuously impinge on the heated surface, a thin liquid film forms and the convective film flow pushes liquid from the impingement area out toward the periphery. Meanwhile, evaporation occurring at the interface of liquid film takes a

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large amount of heat from the heated surface. When surface superheat is sufficiently high to initiate nucleate boiling on the heated surface, the new incoming droplets interact with the bubbles growing in the liquid film, which cause the bubbles to detach from their nucleation sites prematurely. As a result, the onset of critical heat flux (CHF) is delayed and high heat flux is removed.

In spray cooling, heat transfer is usually classified into two distinct regimes, as shown in Fig. 2.3 (Kim, 2007). In the single-phase regime, the heat transfer curve is typically linear, which indicates a dominated single-phase convection in the heat transfer process. The liquid trapped on the heated surface is always swept away by the new incoming droplets before it is heated up to initiate bubble nucleation at a low surface temperature. In the two-phase regime, the slope of the heat transfer curve increases significantly, which indicates more important phase change heat transfer. Finally, as surface temperature increases continuously, CHF occurs and the Leidenfrost point is reached thereafter.

Fig. 2.3 Typical spray cooling curve (FC-72) (Kim, 2007)

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Kim (2007) reviewed that different types of spray pattern can be produced by specific nozzle types. Hollow cone sprays can be typically generated by forcing liquid into a swirl chamber where the tangential momentum of liquid is led to disintegrate the liquid film outside the nozzle orifice to form fine droplets. Full cone sprays are usually produced by forcing liquid through a stationary vane which helps to create the required turbulence for liquid atomisation. Different spray shapes varied from circular, square, oval and many other possible shapes can be used for different applications. Single-nozzle can be used to cool a small surface with a high heat flux density, whereas multi-nozzle arrays would be designed to cool large surface area with a high heat load.

2.2.1 Heat transfer mechanisms

Although spray cooling technology has been applied extensively in the past decades, the current understanding on the heat transfer mechanisms of spray cooling is still limited due to the complex physical phenomena involved. In the absence of convincing evidence in detailed measurements, all of the reported hypotheses have only highlighted one or two aspects of heat transfer where researchers have focused on. In the literature, the commonly accepted heat transfer mechanisms in spray cooling are illustrated in Fig. 2.4. They are discussed separately as follows.

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Fig. 2.4 Schematic of the heat transfer mechanisms in spray cooling (Tan, 2001)

Droplets induced forced convection

This mechanism is predominantly present in the single-phase regime of spray cooling. When droplets impinge on the heated surface, the induced high velocity above the heated surface will force the convection in the thin liquid film and thereby achieve high heat fluxes. Cabrera *et al.* (2003) reported that a heat flux of 200 W/cm^2 was achieved by using water as working fluid and maintaining the surface temperature under 99°C . In their experiments, no nucleate boiling was present and thus, they concluded that the majority of the heat flux removal should be contributed by the forced convection induced by droplet impingement. In Shedd and Pautsch's (2005) experiments, single-nozzle and multi-nozzle arrays were employed to investigate the heat transfer performance of spray cooling. The single-phase heat transfer dominated or might be responsible for 100% of the heat transfer in their experiments. To strengthen their finding, a non-intrusive optical technique was applied to measure the film thickness under different conditions (Shedd and Pautsch, 2006). They reported that the film thickness measured under a constant heat flux

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condition was almost the same as that in the adiabatic condition thus no phase change occurred. Other researchers mentioned that the high heat flux obtained in the single-phase regime of spray cooling could be owing to the very thin thermal boundary layer in the formed liquid film. When the incoming droplets impinge on the liquid film, the droplets agitate the liquid film and reduce the thermal boundary layer thickness locally to enhance the heat transfer performance (Selvam *et al.*, 2006; Kim, 2007; Shedd, 2007). The description of the possible liquid film flow beneath the impinging droplets is illustrated in Fig. 2.5.

Fig. 2.5 Schematic of the liquid film beneath the impinging spray droplets (Shedd, 2007)

Evaporation from the interface of liquid film

In spray cooling, the thickness of the liquid film formed on the impinged surface is extremely thin ($100 \sim 300 \mu\text{m}$) (Yang *et al.*, 1992; Pautsch *et al.*, 2004; Martinez-Galvan *et al.*, 2009; Zhao *et al.*, 2010), so that the temperature gradient across the liquid film is expected to be very large. Thus, heat transfer through the evaporation of the thin liquid film becomes very important in spray cooling. For instance, when assuming the temperature at the top of a liquid film is a saturation temperature, Kim (2007) estimated that a $1.4 \mu\text{m}$ thick water layer can achieve a heat flux of 1000

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W/cm^2 at a wall superheat of 20°C . Pais and Tilton (1989) suggested that heat transfer associated with spray cooling was largely due to the evaporation of the formed liquid film. Moreover, they suggested that liquid mixing between the impinging droplets and liquid film was deduced to decrease the effective thermal resistance, which in turn enhanced the evaporation at the liquid film interface.

Nucleate boiling

Nucleate boiling is another important mechanism in spray cooling. Nukiyama (1966) first advanced that the nucleation sites generated in film boiling were much more than that in pool boiling when the wall superheat was the same. Mesler (1977) further found that the film boiling under spray impingement had most nucleation sites compared to pool boiling and normal film boiling when the same superheat was provided. The numerous nucleation sites created in the liquid film of spray cooling could be the reason why spray cooling achieves a much higher heat transfer coefficient than that of pool boiling.

In Fig. 2.4, when the growing bubbles in the thin film burst or collide with incoming droplets, a lot of tiny bubbles would be generated and re-enter the liquid film, and at the same time, the new incoming droplets also entrap several tiny bubbles before they are absorbed into the liquid film. Since the liquid film flow is very thin, these tiny bubbles generated by bubble burst or entrapped by the incoming droplets are very close to the heated surface. Hence, they are easy to work as new nucleation sites in the boiling process (heterogeneous or homogenous nuclei). These tiny bubbles are so-called “secondary nuclei” which enhance the nucleate boiling in spray cooling. As shown in Fig. 2.6, the phenomenon of a droplet impacting a liquid film was photographed by Sigler and Melser (1990).

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Fig. 2.6 Bubble entrainment 10 ms after the impact of one droplet (Sigler and Mesler, 1990)

Yang *et al.* (1996) conducted experiments in spray cooling by using air-assisted spray nozzles. A heat flux of 820 W/cm^2 was obtained when the liquid flow rate was 2 L/h and air pressure was 446 kPa . They attributed the high heat flux to the new tiny bubbles generated through individual droplet striking the formed liquid film. When droplets passed through the liquid film, gas molecules entrapped at the interface of droplets would be released to work as nucleation sites. Meanwhile, droplets puncturing growing bubbles in the liquid film increased the bubbles frequency in the liquid film. To verify this phenomena, Rini *et al.* (2002) used a high speed camera with framing rates up to 8000 fps and shutter speeds of $1/80,000 \text{ s}$ to investigate the dynamic behaviours of droplets and bubbles in the liquid film of spray cooling. FC-72 was the working medium. The nucleation site density in the thin film was observed to be much larger than that in pool boiling for all their experiment cases. For example, the nucleation site density in pool boiling was measured up to 900 /cm^2 at the surface temperature of 68°C , however, in spray cooling it increased to 3500 /cm^2 . The authors also observed that the lifetime of bubbles in the thin liquid film of spray cooling was much shorter than that in pool

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boiling. With increasing impinging droplet number flux, the bubble density in the liquid film increased, which was in accordance with the theory of “secondary nuclei”.

Transient conduction on heated surface

Demiray *et al.* (2004) directly measured the heat transfer performance for nucleating bubbles in pool boiling by using a micro-heater array with 100 μm resolution. In their experiments, FC-72 was the working fluid. The experimental process was recorded with a high-speed digital video camera. The authors discovered that transient conduction was the dominant heat transfer mechanism in pool boiling. In spray cooling, transient conduction was also proposed as an important mechanism for heat transfer.

When vapour bubbles are broken up by the droplet collision or bubble growth, dry heated areas vacated by the broken bubbles are rapidly taken by the colder liquid from top layer of the thin liquid film. At that instant, transient conduction is occurring in spray cooling. Selvam *et al.* (2005; 2006) have conducted a series of numerical studies to simulate this phenomenon by using the level-set method. The phenomenon where a single droplet impinging on the bubble growing in a thin liquid layer was simulated, and the effects including surface tension, gravity, phase change and viscosity were considered. In terms of the simulation results, transient conduction is found to have major influence on the heat transfer performance. Horacek *et al.* (2005) experimentally found that the heat flux removal in spray cooling is directly related to the length of three-phase contact line, which in another way supports the view that transient conduction is an important heat transfer mechanism for spray cooling.

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2.2.2 Parametric effects

Effect of spray characteristics

Chen *et al.* (2002; 2004) conducted a series of experiments to investigate the effects of droplet velocity, size and flux on the heat transfer performance of spray cooling. They studied one parameter by keeping the others constant. In their experiments, droplet velocity was found to most significantly affect heat transfer, followed by droplet flux. They concluded that a diluter spray with a higher droplet velocity would be more effective than a denser spray with lower droplet velocity in achieving high heat flux at a given flow rate. Rini *et al.* (2002) suggested that the droplet induced “secondary nuclei” is the major heat transfer mechanism in spray cooling. Thus, droplet number flux was proposed to be the dominant factor in achieving high heat flux.

With sufficient experimental database, Estes and Mudawar (1995) developed a correlation of CHF in spray cooling depending on the Sauter mean diameter (SMD), volumetric flux and degree of liquid subcooling. In this correlation, the increased volumetric fluxes only caused a slight increase in the slope of the spray cooling curve (heat flux against the wall temperature or superheat), which was similar to that observed by Sehmbe *et al.* (1995) and Yang *et al.* (1996). The authors speculated that the un conspicuous heat transfer enhancement by increasing flow rate was due to the high liquid volumetric flux which suppressed nucleate boiling by forming a thick liquid film on the heated surface. This eventually caused the reduction of the liquid evaporation efficiency. However, CHF was found to significantly increase when increasing volumetric flux, but only slightly increase when increasing the degree of liquid subcooling.

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Mudawar and Estes (1996) investigated the effects of spray coverage area on CHF. Experiments were conducted by varying the coverage area through regulating the nozzle-to-surface distance. FC-72 and FC-87 were the working fluids. It was found that the maximum CHF was achieved when the circular spray footprint was exactly inscribed with the perimeter of heated surface. Namely, the optimal condition to achieve most heat transfer was to make all the droplets effectively impinge on the heated surface. However, in Cheng *et al.*'s experiments (2010), the optimal condition to achieve the highest heat flux was not the case that the heated surface area was exactly fully covered, but a smaller coverage area as compared to the heated surface.

Effect of spray orientation

Spray orientation has been reported to influence heat transfer in spray cooling via assisting the drainage of excess liquid on the heated surface.

Lin and Ponnappan (2004) designed a miniature nozzle array with 48 jet-swirl nozzles to cool a large surface area (19.3 cm^2 , $2.54 \text{ cm} \times 7.6 \text{ cm}$). The heated surface was positioned at two different orientations: horizontal facing downward orientation and vertical facing orientation, as shown in Fig. 2.7. The experimental results showed that the heat transfer coefficient achieved on the horizontal facing downward orientation was slightly higher by 5.3% on average than that on the vertical facing orientation. It was explained that, for this multi-nozzle array application, liquid was much easier to discharge from the horizontal facing downward surface by gravity, such that heat transfer was enhanced by reducing the thickness of liquid film on the heated surface. However, in the case of using a single nozzle to cool a small surface area, Rybicki and Mudawar (2006) found that spray

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orientations such as upward-facing and downward-facing sprays, showed no measureable influence on any of the spray cooling regimes having been examined.

Fig. 2.7 Spray orientation in the experiments of Lin *et al.* (2004)

Schwarzkoef (2004) investigated the effects of spray inclination (spray axis inclined relative to the heated surface normal) on the spray cooling performance. The heated surface was mounted in a horizontal facing downward orientation and working liquid was sprayed upward at 101 kPa by varying the inclined angle from 0° to 60° . The results showed that CHF was kept nearly constant around 63 W/cm^2 with the inclined angle ranging from 0° to 40° . However, when the inclined angle exceeded 40° , CHF decreased very fast. They suggested that the drop-off in cooling capability would be due to the reduction of spray volumetric flux delivered to the heated surface when inclination angle was too large.

Silk *et al.* (2005) also conducted experiments to examine the effects of spray inclination by using a 2×2 multi-nozzle array. The heated surface was facing upward in their experiments. It was reported that the CHF increased by 24% when the spray inclined angle was varied from 0° to 45° . They felt that spray inclination could eliminate the spray stagnation zones under the hollow cone nozzles, and

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meanwhile the liquid accumulation regions between the impingement zones on the heated surface could be prevented as well.

Shedd (2007) identified that the primary factor that limits the cooling performance of full cone spray arrays on a large surface area was the low momentum flow regions generated between the sprays. To prevent this disadvantage, a nozzle plate was designed by using an array of linear sprays to impinge directly on the heated surface at an inclined angle of 45° , as shown in Fig. 2.8. This design achieved a very high heat flux which was 34% larger than the performance of a single spray without spray inclination. The author reported that the enhancement in heat transfer was achieved by eliminating the low momentum zones between sprays, and allowing more uniform impinging coverage.

Fig. 2.8 Schematic of linear spray nozzles (Shedd, 2007)

Effects of surface modification

Heat transfer improvement through modifying surfaces has been known for many years. In pool boiling, heat transfer studies of enhanced surfaces such as structured surfaces, porous surfaces or rough surfaces with the magnitude of macro, submicron

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and micro have been well summarised by Webb and Kim (2005). So far, the studies focusing on enhanced surfaces in spray cooling are still limited.

The earliest work to study the enhanced spray cooling by using micro-scale structured surface was done by Pais *et al.* (1992). A copper surface was artificially modified by using polishing abrasive paper to obtain a roughness ranging from 0.3 to 22 μm . Water was the working fluid in the experiments. At the water flow rate of 1.42 mL/s and air flow rate of 250 mL/s, a maximum heat flux of 1250 W/cm² ($T_{\text{surf}} = 110^\circ\text{C}$) was achieved by a grit surface of 0.3 μm as compared to 700 W/cm² by a grit surface of 22 μm .

Schmbey *et al.* (1992) investigated the spray cooling performance on a diamond laminated surface with a surface roughness of 1.05 μm . Air-assisted nozzle and water were used. Diamond films with areas of 7.1 mm $\times$ 7.1 mm and 11.1 mm $\times$ 11.1 mm were brazed using silver solder to a copper surface. These two diamond surfaces were observed to significantly improve the heat transfer performance as compared to the smooth copper surface.

Kim *et al.* (2004) investigated the spray cooling enhancement using micro-porous coated surfaces. A 500 μm micro-porous layer formed by micron-sized aluminium particles was built on a tested surface. A heat transfer enhancement of 50% was obtained by the coated surface in comparison to the uncoated surface.

Yao and Hsieh (2006) studied the spray cooling on micro-structured silicon surfaces which were pasted on an aluminium heater. The micro-studs sizes varying from 120 to 480 μm were fabricated on a silicon wafer using an inductively coupled plasma deep reactive ion etching system (DRIE). At a water flow rate of 0.46 mL/s, the micro-studs surface (groove width of 120 μm and stud size of 160 μm) achieved a

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maximum heat transfer enhancement of 17% compared to the smooth silicon surface at the surface temperature of 105°C.

Bostanci *et al.* (2009) experimentally investigated the spray cooling enhancement on micro-structured surfaces using RTI's surface modification techniques. Two micro-scaled heated surfaces with indentations or protrusions structures were investigated. It was observed that the indentations surface with roughness of 2 ~2.5 μm enhanced the heat transfer coefficient by 49% over the unstructured surface while the protrusions surface with roughness of 15.0 ~ 16.0 μm achieved 112% heat transfer enhancement.

Silk *et al.* (2006) studied the effects of macro-structured surfaces on enhanced spray cooling using PF-5060 as coolant. Tests were performed at a constant chamber pressure of 41.4 kPa and a flow rate of 3.33 mL/s. The experimented surface with projected area of 2.0 cm^2 was constructed with different geometries: fins, pins and pyramids directly on the heater block, as shown in Fig. 2.9. They found that the straight fin surface had the best heat transfer performance compared to other structured surfaces. Heat flux enhancements of 11% and 75% were obtained by the straight fin surface when the spray nozzles were arranged in a normal direction and in an inclined angle of 45°, respectively.

Fig. 2.9 Structured surfaces in Silk's experiments (Silk *et al.*, 2006)

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Coursey *et al.* (2007) performed the spray cooling experiments on 6 straight fin structured surfaces (projected area: $1.41 \text{ cm} \times 1.41 \text{ cm}$) with the fin height varying from 0.25 to 5 mm. The straight fin surfaces were fabricated by the electric discharge machined (EDM) method on a copper block heater. It was found that the structured surface improved cooling performance significantly compared to the flat surface. The longer fins outperformed shorter fins in the single-phase heat transfer regime. An optimal fin height ranging from 1 to 3 mm was obtained with a maximum heat flux of 124 W/cm^2 at 19°C surface superheat.

2.2.3 Modelling in spray cooling

Detail modelling on spray cooling is not feasible due to the complex processes involved, such as droplet ejection, droplet interactions, droplet impaction, thin film flow and bubble dynamics (Silk *et al.*, 2008). Therefore, most of the efforts addressed to model spray cooling have been focused on deriving empirical correlations. By reducing the complexity of spray cooling process, only limited research has been conducted to numerically model one or two aspects of spray cooling. In the current stage, a complete analytical model to describe spray cooling process is not feasible at all.

Empirical correlations

Empirical correlations for spray cooling are summarised in Table 2.1.

Ghodbane *et al.* (1991) conducted experiments with two types of nozzles to horizontally spray liquid onto a vertically located heated surface. Constant heat flux was applied on the heated surface and Freon-113 was used. The non-dimensional correlation presented in Table 2.1 was developed for full cone circular and square nozzles. In this correlation, Weber number (We) is characterised by the droplets

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impact energy, x is the nozzle-to-surface distance defined as the normal distance from nozzle orifice to heated surface, and ΔT is the temperature difference between spray liquid and heated surface. It indicates that the impact energy of droplets and especially temperature difference are favourable for heat transfer, while the effect of nozzle-to-surface distance is negative. To extend the applicability of the correlation, Holman *et al.* (1993) have continued the study and developed a modified empirical correlation for a wider range of droplet sizes and velocities.

Table 2.1 Empirical correlations of spray cooling

Reference	Correlation
(Ghodbane and Holman, 1991)	$\frac{q'' x}{\mu_l h_{fg}} = 10.55 (We)^{0.6} \left(\frac{c_l \Delta T}{h_{fg}} \right)^{1.46}$
(Holman and Kendall, 1993)	$\frac{q'' x}{\mu_l h_{fg}} = 9.5 (We)^{0.6} \left(\frac{c_l \Delta T}{h_{fg}} \right)^{1.5}$
(Estes and Mudawar, 1995)	$\frac{q''_{m,p}}{\rho_g h_{fg} Q} = 2.3 \left(\frac{\rho_f}{\rho_g} \right)^{0.3} \left(\frac{\rho_f Q^2 d_{32}}{\sigma} \right)^{-0.35} \left(1 + 0.0019 \frac{\rho_f c_{p,f} \Delta T_{sub}}{\rho_g h_{fg}} \right)$
(Visaria and Mudawar, 2008)	$\frac{q''_{m,p}}{\rho_g h_{fg} Q} = 2.3 \left(\frac{\rho_f}{\rho_g} \right)^{0.3} \left(\frac{\rho_f Q^2 d_{32}}{\sigma} \right)^{-0.35} \left(1 + 0.0050 \frac{\rho_f c_{p,f} \Delta T_{sub}}{\rho_g h_{fg}} \right)$
(Ortiz and Gonzalez, 1999)	Smooth surface: $q'' = 133G \left(\frac{\Delta T}{\Delta T_{sub}} \right)^{0.47}$ Rough surface: $q'' = 120G(1 + 0.25(\cos \theta)^{1.75}) \left(\frac{\Delta T}{\Delta T_{sub}} \right)$
(Cabrera and Gonzalez, 2003)	HF: $\frac{q''}{\rho V h_{fg}} = 0.245 (Ja)^{1.038} \left(\frac{\Delta T_{sub}}{\Delta T_{sat}} \right)^{0.491} \left(\frac{\rho \sigma \dot{m}}{\mu^3} \right)^{0.133} \left(\frac{R_l}{D} \right)^{0.0213} \left(\frac{P}{P_0} \right)^{0.291}$ CHF: $\frac{q''}{\dot{m} h_{fg}} = 1.623 (We)^{-0.315} \left(\frac{R_l}{D} \right)^{0.0465}$

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(Shedd and Pautsch, 2005)	Single nozzle:
	$h_{corr,1} = 0.4627\rho_l c_{p,l}Q + 0.01612Q\Delta T_{sat}$
	Four nozzles array: $h_{corr,4} = 0.2284 + 0.2141\rho_l c_{p,l}Q + 0.003812Q\Delta T_{sat}$
$h = C_1 p^{1/2}$	
(Abbasi <i>et al.</i> , 2010)	$C_1 = 830$ for hollow cone nozzle and p ranges from 0 ~ 5,000 Pa
	$C_1 = 850$ for full cone nozzle and p ranges from 0 ~ 5,000 Pa
	$C_1 = 810$ for flat fan nozzle and p ranges from 0 ~ 5,000 Pa
	$C_1 = 710$ for full cone nozzle and p ranges from 0 ~ 20,000 Pa

Estes and Mudawar (1995) correlated volumetric flux, Sauter mean diameter and liquid subcooling to the CHF on small heated surface. Their correlation shows that CHF is strongly dependent on volumetric flux and Sauter mean diameter. Volumetric flux and liquid subcooling are the positive parameters to achieve high CHF, while Sauter mean diameter is a negative parameter. This correlation fits the CHF data for FC-72, FC-87 and water within a mean absolute error of 12.6%. To improve this model, Visaria and Mudawar (2008) further investigated the effect of high subcooling on CHF for full cone spray nozzles. Using the experiment database, they proposed a new CHF correlation which pointed out a more prominent effect of subcooling on CHF. The established new correlation gives a mean absolute error of 16.3% for the entire database.

In the experiments done by Ortiz and Gonzalez (1999), commercial full cone spray nozzles were used to atomise liquid. Distilled water was chosen as the test fluid. Based on their experimental results, they correlated heat flux with the surface superheat, mass flow rate, degree of subcooling and surface inclination angle. The correlations developed for smooth and rough surfaces were found to predict heat

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flux within the mean error of 10% and 18%, respectively. Carbrera and Gonzalez (2003) further developed empirical models to predict heat flux in the boiling regime as well as CHF. The generalised correlations for the dimensionless heat flux (HF) and CHF were developed as a function of dimensionless mass flow rate, wall superheat, degree of subcooling, surface roughness, ambient pressure and Jacob number (Ja). The correlations for heat flux (HF) and CHF were reported to obtain a confidence level greater than 95%, and the differences between the predicted and experimental values were less than 19% and 15%, respectively.

Shedd and Pautsch (2005) experimentally investigated the cooling performance of single- and four-nozzle applications in spray cooling. Two empirical models were proposed to predict heat transfer coefficients for the two studied applications respectively. The correlations reveal that volumetric flux is the dominant parameter in determining the heat transfer coefficient, and the single-phase convective heat transfer contributes most of heat removal. These correlations are able to fit the single- and four-nozzle applications within the average errors of 6.4% and 9.3%, respectively.

Abbasi *et al.* (2010) suggested that the heat transfer rate in spray cooling was primarily relying on the kinetic energy of impinging droplets rather than simply the liquid flow rate as proposed by Shedd and Pautsch (2005). The kinetic energy of impinging droplets was suggested to be transferred to the liquid film in the form of thinning and agitating the local thermal boundary layer, which contributed heat transfer in spray cooling. The authors proposed that the amount of kinetic energy could be related to the local dynamic pressure exerted by the droplets impinging onto the heated surface. Thereby, in their empirical models, local dynamic pressure

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was used to predict local heat transfer coefficients instead of volumetric flux. It was reported that the average error by this model was observed within 25% for all the experimental data obtained from the hollow cone, full cone and flat fan spray nozzles.

Numerical modelling

Numerical modelling is more advantageous in studying the heat transfer mechanisms of spray cooling in fundamental viewpoints. Due to the complex phenomena involved, the numerical models associated with spray cooling in literature have only concentrated on one or two aspects of the full-scale problem. Among the published works, the hydrodynamic or heat transfer behaviour of the thin film is usually of interest as most of the phenomena are taking place there, such as droplet impingement, thin film convection and evaporation, bubble dynamics and nucleate boiling.

Selvam *et al.* (2005; 2006; 2007; 2008; 2009) performed a series of numerical simulations on the dynamic behaviour of one droplet impacting on a thin liquid film to investigate the heat transfer mechanisms in spray cooling. A 2-D numerical model was first developed using the finite difference method to simulate the process of a single droplet impacting on a thin liquid film with a vapour bubble growing inside (Selvam *et al.*, 2006), as shown in Fig. 2.10.

The physical aspects including heat conduction, convection, phase change, surface tension and gravity, were considered in this model. The effects of the positions between bubble and droplet were examined in the computational domain and the optimum positions with respect to heat flux removal capability were identified. Based on the established model, they further investigated the effects of bubble size as well as dynamic behaviour of the liquid-vapour interface on the heat transfer

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performance (Selvam *et al.*, 2006). Thereafter, the effects of hyper-gravity and micro-gravity on the same problem were studied as well (Selvam *et al.*, 2006). In their successive study, an improved 2-D model was performed to investigate the effect of liquid film thickness on heat transfer (Selvam *et al.*, 2007). Besides that, this model was also conducted to study the effect of thermal boundary layer on the dynamics of liquid-vapour interface (Selvam *et al.*, 2009). On the basis of the 2-D models, a more completed 3-D model was developed on the situation where a vapour bubble was growing in a thin film on a heated surface and a droplet is impacting on the thin film (Sarkar and Selvam, 2009). At the same time, this 3-D model was performed for different wall superheats in the absence of vapour bubble in the thin film to compare the effect of two-phase heat transfer and single-phase heat transfer in spray cooling.

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Fig. 2.10 Shape of the liquid and vapour layer at different times during droplet impact in a 2D modelling (Selvam *et al.*, 2006)

Chen *et al.* (2005; 2008) simulated the dynamics of droplets and bubbles in the liquid film formed in spray cooling. The input parameters in the simulation include the

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experimentally obtained bubble growth rate, number of secondary nuclei provided by droplet impingement and bubble puncturing. The interesting aspects including bubble merging, bubble puncturing by the impinging droplets, secondary nucleation, and bubble diameter at puncturing were investigated. The predicted results agree well with their experiments. The model predictions show that an increase in the number of secondary nuclei is not as effective as an increase in the bubble puncturing frequency on enhancing heat transfer. Smaller bubbles with higher bubble density indicate favourable effects on the heat transfer performance.

2.2.4 Challenges in closed loop spray cooling system

In the literature, most of the experimental studies in spray cooling have been associated with free space spray to cover a small cooling area (less than 2 cm²) using a single spray nozzle. However, only a few studies have used multiple nozzles to cool large surface area in closed loop systems. Using a multi-nozzle array to cool a large surface area encounters the liquid management challenges of both supplying liquid to spray nozzles and draining excessive liquid on the heated surface (Glassman, 2005). These challenges will be more crucial when a larger flow rate is demanded to reject higher heat flux in a closed loop (Shedd, 2007). Therefore, future closed loop spray cooling application will require a large and carefully engineered volume in the drainage region of spray chamber so that excessive liquid can be driven more uniformly and smoothly from the heated surface. In addition, a closed loop system embodies more equipment working under high pressure, which makes the system very hard to control and somehow limit the applicability of spray cooling technology in continuous operations (Kandlikar and Bapat, 2007).

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Lin *et al.* (2003; 2004; 2006) developed a spray cooling system by using a magnetic gear pump to circulate a closed loop. In their experiments, the cooling surface area was scaled up from 2 to 19.3 cm² by applying multiple jet-swirl nozzles from 8 to 48 with FC-72 as the working fluid. A schematic of the setup is shown in Fig. 2.11. Experimental results show that both the heat transfer coefficient and CHF decrease in the larger surface area application in comparison to the smaller surface area. The authors suggested that some liquid accumulation zones were occurring between spray cones, which deteriorated the heat transfer performance. During the trial-and-error stage, it was found that vapour present at the suction line of pump led to insufficient liquid filling in the pump and thereby resulted in the unsteady two phase flow at the inlet of spray nozzles. This unsteady two phase flow caused the non-fully liquid charging in the nozzle feeding chamber hence deteriorated the liquid atomisation performance of the jet-swirl nozzles and the heat transfer performance.

Fig. 2.11 Schematic of the test setup with an ejector bypass line in the pump driven closed-loop spray cooling system (Lin *et al.*, 2006)

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Yan *et al.* (2010) developed a closed loop spray cooling system with four gas-assisted nozzles to simulate the spray cooling on a 6U electronic card. As shown in Fig. 2.12, the flow loop was driven by a gas compressor and R-134a was the working medium. The effects of mass flow rate, nozzle inlet pressure and chamber pressure were studied separately. It was reported that this system was able to remove 1 kW from the heated surface without exceeding the mean surface temperature of 25°C. The authors suggested that some liquid accumulation zones might have occurred on the heated surface where the heat transfer performance was deteriorated. Hence, the surface temperature non-uniformity occurred on the heated surface.

Fig. 2.12 Experimental setup of a impingement spray cooling system (Yan *et al.*, 2010)

2.3 Summary

Although numerous efforts have been made to study the heat transfer associated with spray cooling, the heat transfer mechanisms are still unclear. More fundamental studies on the characterisation of spray atomisation, the thermal effects on the

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formation of spray cone, and the theoretical modelling are required to give insightful understanding on the spray cooling process. For instance, spatial distributions (flux, velocity and diameter) of the impinging droplets would provide a better understanding on how the spray characteristics influence the heat transfer performance and surface temperature non-uniformity. The investigation of the thermal effects on the spray atomisation will be helpful in guiding the design of spray cooling applications in engineering. This literature survey also reveals that experimental studies on the usage of multiple nozzles array to cool large surfaces in a closed loop are still lacking. The challenges faced in excessive liquid drainage and system development are waiting to be solved for multiple nozzles spray cooling applications. Hence, the following chapters of this thesis are motivated by the abovementioned gaps of knowledge in the literature of spray cooling.

Chapter 3 Characterisation of spray atomisation and heat transfer of pressure swirl nozzles

Chapter 3

Characterisation of spray atomisation and heat transfer of pressure swirl nozzles

As reviewed in Chapter 2, spray characteristics have remarkable impact on the heat transfer performance in spray cooling. It is important to understand the spray characteristics of spray nozzles before employing them in a spray cooling system. In the literature, pressure swirl nozzles are widely used in modern industries for their good atomisation quality at relatively low operating pressures (Lefebvre, 1989; Bayvel and Orzechowski, 1993). In the past decades, great efforts have been made towards characterising the spray of pressure swirl nozzles operating in free spray with a high pressure drop, but only a few studies have reported the spray characteristics of spray nozzles operating at a low pressure drop. So far, spray characterisations have mostly been studied using optical techniques. Husted *et al.* (2009) used two optical measuring techniques, Particle image velocimetry (PIV) and Phase Doppler Anemometry (PDA), to investigate the spray structures of pressure swirl nozzles. Pavlova *et al.* (2008) used a LaVision PIV system to characterise the flow fields of actively controlled spray flows in spray cooling. The heat transfer enhancement through the active control of spray flow was explained with the flow field obtained by the PIV system. Panao *et al.* (2009) used a PDA system to characterise the droplet dynamics in an intermittent spray cooling situation. The mass fluxes of the impinging droplets and secondary droplets characterised by the PDA system were used to calculate the energy and exergy efficiencies in assessing the intermittent spray cooling performance.

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This chapter investigated the spray characteristics of two pressure swirl nozzles at different axial distances downstream from the nozzle tip in a predefined operating pressure range. The effects of the nozzle pressure drop and axial distance on spray flow structure, droplet size and droplet velocity were studied using a Dantec PDA system. In addition, an open loop spray cooling system was set up to study the heat transfer performance on a relatively large heated surface area (4.91 cm^2) in the non-boiling regime. The surface temperature distribution was analysed in terms of the nozzle spray characteristics. The effects of the parameters including the nozzle-to-surface distance, nozzle pressure drop, and applied heat flux on the heat transfer performance were examined. A general empirical correlation was derived to predict the heat transfer performance in the non-boiling regime.

3.1 Basic principle of PDA technique

The PDA technique is the primary measurement technique used in the laboratory environment and is most suited for detailed quality measurement to understand and improve the performances of atomisers and spray systems (Lefebvre, 1989). The PDA system can be used to measure the spherical droplet diameters, velocities and fluxes or concentrations simultaneously. A typical PDA measurement is illustrated in Fig. 3.1. The underlying principle of PDA is based on light-scattering interferometry. The measurement point which is also known as the “measuring volume” is defined by the intersection of two identically focused laser beams. The measurements are performed on the single-phase particles as they are moving through the measuring volume, and at the same time, a receiver with multiple detectors is located to receive the scattered light from the measuring volume. The optical signals received by the detectors are converted into the so-called ‘Doppler

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bursts' which possess the information about Doppler frequencies and phase shifts.

The Doppler frequencies are linearly proportional to particle velocities, and the phase shifts captured by different detectors is used to calculate particle diameters.

The number of bursts counted by the system in a certain time period is related to the droplet flux.

Fig. 3.1 Basic principle of PDA measurement (Dynamics, 2008)

3.1.1 Velocity measurement

As illustrated in Fig. 3.2, U represents the droplet velocity, and unit vectors e_i and e_s describe the directions of incident and scattered light respectively. In measurement, the applied incident light has the velocity of C and frequency of f_i . From the view point of the receiver, the moving seeding droplet acts as a moving transmitter, where the movement introduces the so-called 'Doppler-shift' in the frequency of the light received by the receiver. According to the Doppler-theory, the frequency of the light reaching the receiver can be calculated as:

$$f_s = f_i \frac{1 - e_i(U / c)}{1 - e_s(U / c)} \quad (3-1)$$

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It is known that the velocity of the seeding droplet U should be much smaller than the velocity of light ($|U/c| \ll 1$). Then, the above expression can be simplified as

$$f_s = f_i \left[1 + \frac{U}{c} (e_s - e_i) \right] = f_i + \frac{f_i}{c} U (e_s - e_i) = f_i + \Delta f \quad (3-2)$$

Thereby, according to the measurements of the Doppler shift Δf , the particle velocity U is determined.

Fig. 3.2 The principle of velocity measurement (First Ten Angstroms, 2002)

3.1.2 Diameter measurement

As shown in Fig. 3.3, the system consists of two photo-detectors at different angular positions receiving light scattered from the surface of a reflecting spherical droplet when the droplet is passing through the measuring volume. In this process, these two detectors receive a Doppler burst of the same frequency but with different phases due to the different angular positions of the detectors, as shown in Fig. 3.4. Based on this phenomenon, the phase difference between the two signals is directly related to the diameter of the spherical particle.

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Fig. 3.3 The light received by two detectors (Dynamics, 2008)

Fig. 3.4 The diameter-phase relation obtained in a system with two detectors (Dynamics, 2008)

The principle is that, phase difference in the Doppler burst caused by the bigger particle should be larger than that by the smaller one. Hence, the droplet diameter is calculated by the phase differences in the measurement. However, as the phase

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difference is a function of modulo 2π , if a droplet size is huge, it could cause the phase difference to go beyond 2π , hence, there will be a problem that the PDA system cannot discriminate the real size of a larger particle and a much smaller particle (threshold limit for particular droplet diameter) from two detectors, such as D3' and D3 as shown in Fig. 3.4. To resolve this problem, an additional detector is integrated in the receiver of the conventional PDA system where the three detectors are asymmetrically positioned. As shown in Fig. 3.5, the detectors 1 and 2 form the larger distant pair which gives the steeper slope of the diameter-phase relationship thereby has the higher resolution but smaller working range Φ_{1-2} ; the detectors 1 and 3 form the smaller distant pair which gives the gentler slope of the diameter-phase relationship thereby corresponds to the larger measurement range Φ_{1-3} but lower resolution. To optimise the measurement, it is possible to achieve the high resolution and large measurement range at the same time by comparing the phase differences between the two detector pairs.

Fig. 3.5 The different slopes of the diameter-phase relation obtained in a system with two pairs of photo-detectors at different separations (Dynamics, 2008)

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3.2 Experimental apparatus

3.2.1 Spray nozzles and flow diagram

In this chapter, two Steinen serial pressure swirl nozzles were selected to investigate the atomisation mechanisms of pressure swirl nozzles in a lower pressure range. These two nozzles have the same internal structure except for the discharge orifice diameter, as shown in Fig. 3.6. In particular, Nozzle A has a smaller discharge orifice of 0.46 mm, while Nozzle B has a bigger discharge orifice of 0.64 mm.

Fig. 3.6 Configuration of the commercial nozzles A and B

To conduct the PDA and heat transfer measurements, an open loop setup was constructed as shown in Fig. 3.7. It is composed of one nitrogen gas tank, one water tank, two filters, and several sensors and valves. The nitrogen gas tank with a precise pressure regulator is used to supply and regulate the pressure in the water tank to pressurise the water to the spray nozzle. A filter installed between the water tank and spray nozzle are used to prevent the impurities in water from clogging the nozzle orifice and changing its spray performance. Besides that, a “Digmesa” digital turbine flow meter (model: FHK LCD G1/4) and “Krystal” pressure transducer (model:

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AST4710) are calibrated and placed at the upstream of the spray nozzle to detect both of the flow rate and corresponding pressure drop across the nozzle.

Fig. 3.7 Schematic of the open loop system

3.2.2 PDA measurement apparatus

A two components Dantec PDA system was used to measure the droplet diameter, droplet velocity along the axial direction. The schematic of the PDA spray characterising system is demonstrated in Fig. 3.8. This system is composed of an Ar-ion laser transmitter, a PDA system, a PDA processor and data acquisition system. The Ar-ion laser transmitter is a two-colour (blue and green), four laser beams fringe-type which allows the simultaneous measurement of two velocity components: radial and axial velocities. In particular, the green laser beams with wavelength of 514.5 nm are used to measure the axial velocity component and the blue laser beams with wavelength of 488 nm are used to measure the radial velocity component.

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Fig. 3.8 Schematic of the experimental setup of PDA measurement

In the experiments, the droplet diameter was measured using the first order refraction scattering model. The scattering angle was fixed at 72° which was optimum for water as the test fluid. The PDA processor combining the functions of counter, buffer interface and coincidence filter was interfaced to the PC with Dantec flow software for data processing. A 3D traverse coordinate system comprising three stepping motors (resolution: $50\ \mu\text{m}/\text{step}$) was used to work with the PDA system. Thus, the movement of the PDA measuring volume can automatically follow up a predefined traverse motion mesh according to the experiments. To minimise the systematic errors, the parallelism of the laser beams was installed under the optimal conditions as recommended by the PDA manufacture. Before each measurement, the experimental setup was checked for the laser beams alignment to intersect at the measuring volume and the receiver optics were adjusted and focused well on the measuring volume.

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During the PDA measurement, the spray nozzle was fixed on a vertical test stand.

The measuring volume of the PDA system was allowed to translate in the x , y and z directions to characterise the spray at different measuring planes downstream from the nozzle orifice with the axial distances Z ranging from 10 to 25 mm. The PDA measurements were conducted by following a predefined measuring grid trajectory that covered the whole spray cone at each plane.

3.2.3 Heat transfer measurement apparatus

In the heat transfer experiments, a heater module was designed, fabricated and stationed under the spray nozzle as shown in Fig. 3.9.

Fig. 3.9 Schematic of the heater module in the experiment

A pure copper block served as the heater body, the top surface of which was a flat circular disc with a diameter of 25 mm. Eight cartridge heaters (capacity: 8×150 W) were used to supply power to the heater block by regulating a variac transformer.

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Before fabricating the heater, the height of the top cylinder of the heater was determined by using the Steady-State Thermal model of ANSYS to ensure one-dimensional isotherms below the test surface (see the simulated temperature contour in Fig. 3.9). Nine K-type thermocouples with wire diameter of 0.3 mm were embedded into the copper block in three distant planes along the one-dimensional top heater cylinder (T_{L1} , T_{L2} and T_{L3}). At each thermocouple plane, three thermocouples were separated by 120° . One of the thermocouples was installed to measure the temperature at the centre of the heated surface (T_{L2}), and the other two thermocouples were installed to measure the temperatures at the radial distance of 6.5 mm off the surface centre (T_{L1} and T_{L3}). If the temperature gradient is known between the neighbouring thermocouple layers, the applied heat flux and the corresponding surface temperature can be determined by the one-dimensional heat conduction. In the experiments, the top part of the copper block was insulated by a Teflon (Polytetrafluoroethylene (PTFE), $k = 0.20 \text{ W/m}\cdot\text{K}$, melting temperature: $260\text{-}280^\circ\text{C}$) sleeve, while the base of the copper block was insulated by the Durablanket (high purity Fiberfrax ceramic fibres, $k = 0.14 \text{ W/m}\cdot\text{K}$, melting temperature: 1760°C). The nozzle-to-surface distance (H) was adjusted by regulating the position of spray nozzle with a precision orthogonal translation stage. At each power level, the heat flux and surface temperature were only recorded when the temperature fluctuation remained within 0.5°C for 15 minutes (steady state).

3.3 Data reduction and uncertainty analysis

3.3.1 Data reduction

The discharge coefficient C_d is used to assess the liquid discharge capability of spray nozzles. It is defined as the ratio of the effective mass flow rate to the theoretical

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maximum mass flow rate through the nozzle at the same pressure drop, and is calculated according to

$$C_d = \frac{m_{eff}}{A\sqrt{2\rho\Delta p}} \quad (3-3)$$

herein, m_{eff} is the effective mass flow rate of the nozzle at the nozzle pressure drop Δp , ρ is the density of liquid, and A is the section area of the nozzle discharge orifice.

To quantitatively specify the spatial droplet number flux distribution in the spray cone, a droplet number flux ($\hat{N}(r)$) is introduced as the ratio of the detected number of droplets ($N(r)$) in a measuring time period (Δt) to the measurement area (A) in the measurement plane,

$$\hat{N}(r) = \frac{N(r)}{A \cdot \Delta t} \quad (3-4)$$

To evaluate the typical droplet size at the measuring plane, the Sauter mean diameter (D_{32}) is chosen as it is popularly used in processes involving heat and mass transfer (Bayvel and Orzechowski, 1993),

$$D_{32} = \sum_{i=1}^N D_i^3 / \sum_{i=1}^N D_i^2 \quad (3-5)$$

where D_i is the diameter of the i th droplet, and N is the total number of droplet samples captured by the PDA system. The local droplet mean axial velocity ($\bar{V}$) is defined as

$$\bar{V} = \sum_{i=1}^N D_i^3 \cdot V_i / \sum_{i=1}^N D_i^3 \quad (3-6)$$

where V_i is the velocity of the i th droplet at the measuring point.

To assess the typical impact energy of the impinging droplets at each measuring plane, a dimensionless number known as the Weber number is defined as

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$$We = \frac{\rho \bar{V}^2 D_{32}}{\sigma} \quad (3-7)$$

where $\bar{V}$ is the mass-averaged axial velocity of the impinging droplets at a measuring plane, ρ is the liquid density, and σ is the surface tension.

In the heat transfer experiments, the surface temperature (T_{surf}) and heat flux (q'') are both calculated using the Fourier's law of heat conduction. The applied heat flux is calculated as

$$q'' = k \frac{\Delta T}{\Delta z} \quad (3-8)$$

where k is the thermal conductivity of the copper block, and ΔT is the temperature difference between the thermocouple planes with the distance of Δz . The average surface temperature $\bar{T}_{surf}$ is calculated as

$$\bar{T}_{surf} = \sum_{i=1}^3 T_{surf,i} / 3 \quad (3-9)$$

where $T_{surf,i}$ is the local surface temperature extrapolated by the i th thermocouple.

The average heat transfer coefficient ($\bar{h}$) was calculated by

$$\bar{h} = \frac{q''}{\bar{T}_{surf} - T_{spray}} \quad (3-10)$$

where T_{spray} is the inlet water temperature and is equal to 27°C.

An empirical model was derived to fit the heat transfer data of nozzles A and B in terms of dimensionless groups. The impinging Reynolds number (Re) is defined based on the nozzle orifice diameter (d_0) and the nozzle pressure drop (Δp) as

$$Re = \frac{\sqrt{2\Delta p \rho} \cdot d_0}{\mu} \quad (3-11)$$

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The Nusselt number (Nu) is defined according to the impinged surface diameter (d_s) and the heat transfer coefficient

$$\text{Nu} = \frac{\bar{h} \cdot d_s}{k_{liq}} \quad (3-12)$$

where k_{liq} is the thermal conductivity of water.

The non-dimensional temperature (θ) is defined as the ratio of the heated surface temperature ($\bar{T}_{surf}$) to the temperature difference between the liquid boiling temperature ($T_{boiling}$) and inlet water temperature (T_{spray})

$$\theta = \frac{\bar{T}_{surf}}{T_{boiling} - T_{spray}} \quad (3-13)$$

The Kelvin scale is applied in Eq. (3-13) for the temperature.

3.3.2 Uncertainty analysis

The uncertainty of measurement is classified into two groups: random uncertainty and systematic uncertainty. Random uncertainty could be treated statistically in experiments, while systematic uncertainty can be minimised by careful experimentations.

Uncertainty of PDA measurement

The uncertainties of the PDA measurements on droplet size and droplet velocity are combinations of the optical configuration of the system, the sampling of the droplets and the data processing. Among them, the accuracy is primarily dependent on the optical configuration of the PDA system (First Ten Angstroms, 2002). To minimise the systematic errors, the parallelism of the laser beams was installed under the optimal conditions which are recommended by the manufacturer (“water droplet in

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air” in (First Ten Angstroms, 2002)). Before each measurement, the experimental setup was checked for laser beams precise alignment to intersect at a crossing volume (measuring volume) and the receiver optics were adjusted and focused well on the crossing volume. At the system level, the accuracies of the droplet size and velocity measurements are suggested to be within $\pm 1\%$ and $\pm 0.5\%$, respectively. A general guide for the PDA measurement is that the uncertainties for droplet size and velocity measurements are 4~5% and 1%, respectively (Winkler and Peters, 2002; Husted *et al.*, 2009).

For the PDA system, the velocity measurement is based on the Doppler Effect and expressed as,

$$V = \frac{\lambda}{2\sin(\theta/2)} f_D \quad (3-14)$$

where λ is the wavelength of the light source, f_D is the frequency shift according to the receiver, and θ is the intersection angle formed by the laser beams. The system error for f_D is $\pm 0.5\%$, and the error of θ is $\pm 0.037^\circ$ (in terms of the error of the spacing of laser beams (25 ± 0.1 mm) and the focal length in the alignment (310 ± 1 mm)), the uncertainty of the droplet velocity was estimated according to Ortiz and Gonzalez (1999),

$$\sigma = \sqrt{\sigma_{f_D}^2 + \left(\frac{\delta_\theta}{\theta}\right)^2} = \sqrt{(0.5\%)^2 + (0.037 / 4.62)^2} = 0.94\% \quad (3-15)$$

In the present study, the uncertainty of the droplet size is estimated from the droplet diameter validation map (experimental data) as shown in Fig. 3.10. The accuracy of the droplet size measurement is determined by the tolerance band indicated by the two dash lines besides the centre continuous line which represents the theoretical

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relationship between the two phase differences for perfectly spherical droplets. It is found that more than 96% of the experimental samples fell into the tolerance band of $\pm 5\%$. Therefore, with sufficient samples obtained, the uncertainty of the droplet size measurement is estimated to be $\pm 5\%$ by deducting the samples beyond the tolerance band ($\pm 5\%$).

Fig. 3.10 Phase validation for a PDA measurement case

As for the droplet mass flux or concentration measurement, the measurement accuracy is too application-specified to allow a single figure to be given due to the fact that the droplet flux or concentration measurement is depending directly on the chosen measuring volume size. In the present study, the droplet mass flow rate was estimated by integrating the droplet mass flux over the area of the measuring plane of the spray cone at the axial distance of Z as following,

$$m = \int_0^R 2\pi r \dot{m}(r) dr \quad (3-16)$$

where $\dot{m}(r)$ is the local mass flux at r , and R is the diameter of the measurement plane of the spray cone at the axial distance of Z . The local mass flux $\dot{m}(r)$ is

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calculated by summing up the mass of the droplets captured by the PDA system at the measuring point in a time period.

$$\dot{m}(r) = \frac{\rho\pi}{6\Delta t} \sum_{i=1}^N \frac{D_i^3}{A} \quad (3-17)$$

Herein, D_i is the diameter of i th droplet passing through the measuring area A , Δt is the measuring time at each location, N is the number of droplets passing through the measuring area and ρ is the density of the fluid. It is noted that in Eq. (3-17) the droplets passing through the measuring area with a negative velocity are subtracted from the mass flux calculation. From Eq. (3-16), the difference of the mass flow rate $\dot{m}$ measured by the PDA system and turbine flow meter is within 15~20% for all the experiments.

Uncertainty of heat transfer measurement

In the heat transfer experiments, all the thermocouples were calibrated with a thermal calibrator (Thermacal Inc., Model 18B). Uncertainties of the calibrated thermocouples in the present study are considered to follow the accuracy of the calibrator which is validated to be $\pm 0.1^\circ\text{C}$ in the range from 20 to 200°C . The error of the distance between the thermocouple planes is less than 0.1 mm. The uncertainty of the surface temperature was therefore calculated according to the method by Ortiz and Gonzalez (1999),

$$\delta_{T_{surf}} = \sqrt{\left(\frac{\Delta T_{1-2} \Delta x_{0-2}}{(\Delta x_{1-2})^2} \delta_x\right)^2 + (\delta_T)^2 + \left(\frac{\Delta T_{1-2}}{\Delta x_{1-2}} \delta_x\right) + \left(\frac{\Delta x_{0-2}}{\Delta x_{1-2}} \delta_T\right)^2} \quad (3-18)$$

From Eq. (3-18), the uncertainty of the extrapolated surface temperature is found to be approximate $\pm 0.64^\circ\text{C}$ for a temperature difference of 10°C between the thermocouple planes T_{L1} and T_{L2} .

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The uncertainties of the heat flux and heat transfer coefficient is calculated based on the error propagation method by Figliola and Beasley (2005),

$$\delta_q = \sqrt{\left(\frac{\partial q}{\partial x} \delta_x\right)^2 + \left(\frac{\partial q}{\partial k} \delta_k\right)^2 + \left(\frac{\partial q}{\partial T} \delta_T\right)^2} \quad (3-19)$$

$$\delta_h = \sqrt{\left(\frac{\partial h}{\partial q} \delta_q\right)^2 + \left(\frac{\partial h}{\partial T} \delta_T\right)^2} \quad (3-20)$$

It is found that the uncertainties of the heat flux and heat transfer coefficient are 8.6% and 8.9%, respectively.

In the present study, the error of the calibrated digital turbine flow meter is less than 0.1 mL/s, and the uncertainty of the calibrated pressure transducer is less than 0.1% of the experimental pressure range.

3.4 Results and discussion

3.4.1 Flow rate and discharge coefficient

Flow rate is the only parameter concerning the quantity rather than quality of the spray atomisation of spray nozzles. If the flow rate is known, the number of nozzles required for a spray cooling system is determined from the total coolant requisite. The discharge coefficient, which is generally a constant value used to predict the flow rate for the nozzle at an arbitrary pressure drop, is obtained experimentally (Bayvel and Orzechowski, 1993).

Figure 3.11 shows the flow rates and discharge coefficients of the studied spray nozzles as a function of nozzle pressure drop. The flow rates of the spray nozzles increase with the increase of nozzle pressure drop. This is reasonable since a higher pressure drop leads to a faster liquid discharge velocity through the discharge orifice

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thereby increases the flow rate. At the same pressure drop, the nozzle with a bigger discharge orifice (Nozzle B) delivers a larger flow rate. This is due to the fact that a bigger discharge orifice has a larger effective discharge area and thus is favourable to discharge more liquid.

Fig. 3.11 Flow rates and discharge coefficients of the spray nozzles

As the nozzle pressure drop increases from 1.5 to 6 bar, the discharge coefficients for the pressure swirl nozzles appear to be independently related to the nozzle pressure drop, which indicates the advantages of using discharge coefficient to predict the flow rate of the spray nozzles operating under different nozzle pressure drops. It is found that the spray nozzles have discharge coefficients which are much smaller than 1.0. This is explained by the internal flow of the spray nozzles as shown in Fig. 3.12. For pressure swirl nozzles, the liquid does not discharge through the whole cross section of its discharge orifice rather the liquid is forced to trace along the annular wall of the discharge orifice due to a swirl effect. At the centre of the swirl chamber and discharge orifice, a vapour core appears to block off the liquid flow, and therefore the effective area for liquid discharge is reduced and a low

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discharge coefficient is the result (Lefebvre, 1989; Bayvel and Orzechowski, 1993).

In the comparison of the spray nozzles, Nozzle B which has the larger discharge orifice obtains the smaller discharge coefficient. This agrees with the results reported by Lefebvre (1989). A summarised correlation $C_d = C(1/d_o)^{0.75}$ shows an inverse relationship between the discharge coefficient (C_d) and discharge orifice diameter (d_o) (Rizk and Lefebvre, 1985).

Fig. 3.12 Internal liquid flow pattern in a pressure swirl nozzle

3.4.2 Spray characterisation

Spray pattern

The spray behaviour of Nozzle A is illustrated in Fig. 3.13. It is observed that the discharged liquid spreads as a high velocity conical film sheet (hollow cone) as soon as it leaves the nozzle's orifice. As the velocity slip between the film sheet and ambient air is pronounced, the film sheet breaks up into a dispersed spray with a dispersion angle of β . The conical sheet disintegration makes the dispersed spray overlap at the centre of the spray cone whereby a curved boundary is formed between the spray cone and the ambient air. Thus, a full cone spray can be achieved when a critical axial distance Z_c is ensured.

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Fig. 3.13 Spray behaviour of Nozzle A: (a) in free spray; (b) in PDA experiment

Figures 3.14 and 3.15 illustrate the spray flow structures of nozzles A and B at the particular measurement planes of the axial distances Z spanning from 10 to 25 mm with $\Delta p = 3$ and 4 bar, respectively. Within the range of axial distances, the swirl nozzles result in two local droplet flux peaks and one local droplet flux valley along the diameter of the spray cone in the droplet flux distribution. As the axial distance increases, the spray cone changes from a hollow cone to a full cone when Z_c is reached, while the magnitude between the peak and valley values decreases and causes a more uniform droplet flux distribution. Similar trends were found in other researchers' works (Lefebvre, 1989; Bayvel and Orzechowski, 1993; Marchione *et al.*, 2007). The spray flow structures presented in Figs. 3.14 and 3.15 reasonably explain the aforementioned phenomenon of spray overlapping in the overlap zone ($Z > Z_c$) as shown in Fig. 3.13.

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Fig. 3.14 Spray flow structure of Nozzle A: (a) $\Delta p = 3$ bar; (b) $\Delta p = 4$ bar

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Fig. 3.15 Spray flow structure of Nozzle B: (a) $\Delta p = 3$ bar; (b) $\Delta p = 4$ bar

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Due to their different discharge orifices, Nozzle A and Nozzle B performed differently in terms of the spray cone. It is noticed that Nozzle A, with its smaller discharge orifice, attained a full cone spray at a relatively shorter axial distance ($Z_c = 15$ mm in Fig. 3.14(a) III and $Z_c = 18$ mm in Fig. 3.14(b) IV at $\Delta p = 3$ and 4 bar, respectively) compared to Nozzle B which has a bigger discharge orifice ($Z_c = 20$ mm in Fig. 3.15(a)V and $Z_c = 23$ mm in Fig. 3.15(b)VI at $\Delta p = 3$ and 4 bar, respectively). This is because Nozzle B produces a larger spray cone angle in the atomisation process which requires a longer axial distance for the dispersed spray to overlap when assuming the same dispersion angle of β . The comparisons between the spatial distributions of droplet flux and velocity in Figs. 3.14 and 3.15 indicate that velocities are higher at the locations where a higher droplet flux is observed. This is reasonable since the droplets at the locations with higher droplet fluxes have higher impinging momentum and therefore are less affected by the drag effect of ambient air (Marchione *et al.*, 2007).

Droplet diameter and velocity

The averaged droplet size D_{32} and velocity are plotted as a function of axial distance in Fig. 3.16. For $Z < Z_c$, the droplet size decreases rapidly with increasing axial distance, while for $Z \geq Z_c$, the droplet size increases more gently. Returning to the flow structures in Figs. 3.14 and 3.15, as the axial distance increases, the droplets are exposed to a larger impinging area where the interaction with ambient air imparts drag that distorts the droplets to cause further break up. Hence, the droplets atomisation improves before the axial distance reaches Z_c . When $Z > Z_c$, the droplets dispersed from the conical film sheet overlap at the spray cone center whereby the droplets collide and coalesce together to generate bigger droplets. As a result, the

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averaged droplet size increases with the axial distance when $Z > Z_c$. It is also observed that the average droplet velocity decreases with increasing axial distance, which agrees with the previous research work (Yang *et al.*, 2003). This is due to stronger drag effect in terms of a longer travel path in the ambient air as well as a larger impinging area that the droplets are exposed to.

Fig. 3.16 Sauter mean diameter, and velocity as a function of axial distance

To evaluate the average droplet impingement energy at different axial distances, the popularly-used Weber number (We) is plotted against the axial distance as shown in Fig. 3.17. It is observed that We decreases monotonically with increasing axial distance. For $Z < Z_c$, We decreases at a higher rate due to the steep decrease in droplet size and velocity as indicated in Fig. 3.16, while the decrease is more gradual when $Z > Z_c$ due to the counterbalance between the variations of droplet size and velocity.

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Fig. 3.17 Droplet Weber number as a function of axial distance

3.4.3 Heat transfer characterisation

Surface temperature distribution

In the following discussions, the nozzle-to-surface distance H in the heat transfer experiments corresponds to the axial distance Z in the spray characterisation studies.

Figure 3.18(a) represents the local surface temperatures extrapolated from the closest thermocouple plane to the heated surface at $\Delta p = 3$ bar. It can be seen that the surface temperature distributions at $q'' < 46$ W/cm² (Fig. 3.18(a) I-II) are solely dependent on the droplet flux distributions (Fig. 3.14(a)). At $10 \leq H \leq 20$ mm, the lower droplet flux and impact velocity at $r = 0$ mm (Fig. 3.14(a) I-V) result in a higher surface temperature at $r = 0$ mm than that at $r = 6.5$ mm (Fig. 3.18(a) I-II). At $H = 25$ mm, the relatively higher droplet flux and droplet impact velocity at $r = 0$ mm (Fig. 3.14(a)VII) result in a lower surface temperature than that at $r = 6.5$ mm (Fig. 3.18(a) I-II). When $q'' > 60$ W/cm², contrary to the results observed at $q'' < 46$ W/cm², the surface temperature at $r = 0$ mm is observed to be higher than that at r

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= 6.5 mm even though $H = 25$ mm (Fig. 3.18(a) III-IV). This could be due to the expanded spray cone when the surface temperature is high, as shown in Fig. 3.19. As the surface temperature increases from 48.4 to 99.2°C, the spray angle of the spray cone increases from 60° to 70.3°. The high surface temperature could accelerate the evaporation of the liquid film formed on the heated surface, which causes a strong uprising vapour flow to push the spray conical film sheet and droplets outward to expand the spray cone. As discussed in Section 3.4.2, the expanded spray cone (with bigger spray angle) is postulated to hinder the overlapping of the dispersed spray at the centre of the spray cone. Hence, the higher droplet flux and impact velocity at $r = 0$ mm may not be attained at $H = 25$ mm when the surface temperature is high enough (e.g., $T_{\text{surf}} > 80^\circ\text{C}$). The same phenomenon is also observed for Nozzle A operating at $\Delta p = 4$ bar as depicted in Figs. 3.14(b) and 3.18(b). The relationship between the distributions of droplet flux and surface temperature strongly indicates that the droplet impingement cooling is the primary heat transfer mechanism in the non-boiling regime of spray cooling. This agrees with the theoretical work in the later Section 5.3.2. It could be inferred that the uniform impinging droplet flux and velocity distributions are favourable for achieving uniform surface temperature on the heated surface.

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Fig. 3.18 Surface temperature distribution of Nozzle A: (a) $\Delta p = 3$ bar; (b) $\Delta p = 4$ bar

Fig. 3.19 Effect of surface temperature on spray cone formation

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Effect of nozzle-to-surface distance (H)

The nozzle-to-surface distance H affects the heat transfer performance due to the change in impact coverage area, impinging droplet flux distribution and droplet impact energy when the spray cone impinges on the heated surface. The heat transfer coefficients as a function of nozzle-to-surface distance for Nozzle A are shown in Fig. 3.20. It is observed that the heat transfer coefficient generally decreases with increasing nozzle-to-surface distance except for certain cases when the nozzle-to-surface distances are low.

Fig. 3.20 Effects of nozzle-to-surface distance on the heat transfer coefficient of Nozzle A: (a) $\Delta p = 3$ bar; (b) $\Delta p = 4$ bar

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In other researchers' studies, the droplet impact energy is reported to be the dominant parameter affecting single- and multiple droplets impingement heat transfer in the wetting and non-wetting regimes (Pedersen, 1970; Abbasi *et al.*, 2010). Higher droplet impact energy assists the spreading of droplets into thin films and hence enhances single droplet impingement cooling. Lower droplet impact energy will deter the droplets from penetrating the liquid or vapour film to contact the heated surface, thereby worsening the heat transfer performance in multiple droplets impingement heat transfer. In the present study, the droplet impact energy is found to decrease with increasing axial distance (corresponding to the nozzle-to-surface distance) as shown in Fig. 3.17. Therefore, a lower heat transfer coefficient at a higher nozzle-to-surface distance could be due to the reduction of droplet impact energy.

At $5 \leq H \leq 10$ mm, the heat transfer coefficient increases with nozzle-to-surface distance for $q'' < 46$ W/cm² when the surface temperature is low. The heat transfer enhancement by increasing nozzle-to-surface distance could be attributed to the increased impact coverage area on the heated surface which overwhelms the disadvantage of the reduced droplet impact energy as discussed above. As the heat flux increases gradually, the gradient of the heat transfer coefficient increment versus nozzle-to-surface distance from 5 to 10 mm becomes gentler, and finally an inverse trend of decreasing heat transfer coefficient is observed at $q'' > 60$ W/cm². Due to the aforementioned monotonic increase in surface temperature with increasing nozzle-to-surface distance in Fig. 3.18, the higher surface temperature at $q'' > 60$ W/cm² could intensify the liquid film evaporation on the heated surface to generate strong uprising vapour. The uprising vapour drags the impinging droplets to reduce their impact energy when the droplets approach the heated surface.

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Consequently, the reduction of droplet impact energy in return counteracts the benefit of the increased impact coverage area at a higher nozzle-to-surface distance and results in a deterioration of heat transfer performance.

From Fig. 3.20, it is also interesting to note that the impinging spray cone has significant effect on the heat transfer performance. As shown in Fig. 3.20(a), when $q'' < 46 \text{ W/cm}^2$, the heat transfer coefficient decreases steeply with the increase of nozzle-to-surface distance in the hollow cone spray regime ($10 < H < 15 \text{ mm}$). Nonetheless, the heat transfer coefficient is insensitive to the nozzle-to-surface distance in the full cone spray regime ($15 < H < 25 \text{ mm}$). It appears that a more uniform full cone spray at a higher nozzle-to-surface distance provides a more uniform liquid flux distribution on the heated surface which could counteract the drawbacks of the reduced droplet impact energy at a higher nozzle-to-surface distance. When $q'' > 60 \text{ W/cm}^2$, the high surface temperature expands the spray cone to deter the formation of the full cone spray, thus causes the heat transfer coefficient to decrease monotonically at a similar rate for the nozzle-to-surface distance from 10 to 25 mm. Similar phenomenon is also observed for Nozzle A operating at $\Delta p = 4 \text{ bar}$ as referring to Fig. 3.14(b) and Fig. 3.20(b). The abrupt drop in the heat transfer coefficient at $H > 20 \text{ mm}$ is explained by the fact that a larger spray angle at a higher pressure drop results in a larger coverage area than the heated surface. The inefficient usage of liquid coolant at $H > 20 \text{ mm}$ lowers the heat transfer performance significantly.

Effect of nozzle pressure drop (Δp)

As shown in Figs. 3.18 and 3.20, the lower surface temperature and higher heat transfer coefficients are obtained for the same applied heat flux as the nozzle

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pressure drop increases from 3 to 4 bar. An increasing nozzle pressure drop increases the coolant volumetric flux, droplet number flux as well as impact velocity. The enhanced heat transfer by increasing pressure drop could be attributed to the following reasons: (a) a higher pressure drop provides more liquid volumetric flux impinging on the heated surface, which is favourable for heat transfer in the non-boiling regime as the sensible heat transfer is dominant. (b) as reported by the previous researchers (Chen *et al.*, 1995; Xie *et al.*, 2012), the thickness of the liquid film on the impinged surface remains nearly constant with increasing pressure drop. Therefore, the liquid film has to flow faster at a higher pressure drop so as to flush away the higher volumetric flow rate, which in turn enhances forced convection in the thin film flow of spray cooling. (c) a higher pressure drop allows more droplets with higher impact velocity to impinge on the heated surface, by which the heat transfer in the non-boiling regime is intensified by agitating the thermal boundary layer in the liquid film flow (Shedd, 2007).

3.4.4 Correlation

Due to the complexity of the spray cooling process, empirical models are developed as a guide for the design of spray cooling systems. In the literature, general parameters of spray characteristics were evaluated to quantify the heat transfer performance in the non-boiling regime of spray cooling, such as the volumetric flux (Shedd and Pautsch, 2005), dynamic pressure by the impinging droplets (Abbasi *et al.*, 2010), Weber and Reynolds numbers of the impinging droplets (Issa and Yao, 2005; Tao *et al.*, 2011). Moreover, the effect of surface temperature on heat transfer cannot be neglected in the non-boiling regime of spray cooling (Wang *et al.*, 2010).

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Unlike these empirical models which only focus on a single flow parameter, the present study shows that multiple flow parameters have significant impact on heat transfer performance. The pressure drop, nozzle-to-surface distance and surface temperature are found to affect the heat transfer performance by introducing different flow rates, droplet flux distributions, and droplet impact energies when droplets approach the heated surface. In an effort to characterise the effects of different flow parameters in spray cooling, the experimental data obtained by nozzles A and B were utilised to generate a general empirical correlation by relating the Nusselt number (Nu) to the impinging Reynolds number (Re), non-dimensional surface temperature (θ) and non-dimensional nozzle-to-surface distance (H/d_s) as shown in Eq. (3-21). In this concise correlation, the impinging Reynolds number and nozzle-to-surface distance are used to include the effects of flow rate, droplet impact energy and droplet flux distribution; the non-dimensional surface temperature θ is combined to take the effect of surface temperature into account.

$$\begin{aligned}
 Nu &= 8.705 \left(\frac{\sqrt{2\Delta p \rho} \cdot d_0}{\mu} \right)^{0.323} \left(\frac{\bar{T}_{surf}}{T_{boiling} - T_{spray}} \right)^{0.8526} \exp(-0.4268 H / d_s) \\
 &= 8.705 Re^{0.323} \theta^{0.8526} \exp(-0.4268 H / d_s) \quad (3-21)
 \end{aligned}$$

The comparisons of the predictions by Eq. (3-21) and experimental data are shown in Fig. 3.21. The flow conditions are: $12600 \leq Re \leq 20250$, $4.2 \leq \theta \leq 5.2$, and $0.2 \leq H/d_s \leq 1.0$. It is found that this empirical correlation provides a good prediction of the experimental data with an average error of 14%. It indicates that both the Reynolds number (Re) and surface temperature (θ) have positive effects on the heat transfer performance, while the nozzle-to-surface distance is expected to have a negative influence. This empirical correlation is also validated with the experimental

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results by Cheng *et al.* (2010) and Tao *et al.* (2011) in the non-boiling regime of spray cooling. The present correlation can predict their experimental results with an average error of 25%.

Fig. 3.21 Comparison between model predictions and experimental data

3.5 Summary

The spray flow structure, droplet Sauter mean diameter, and droplet impingement energy were characterised at different axial distances and nozzle pressure drops by a PDA system. The heat transfer characteristics of the spray nozzles in the non-boiling regime were investigated. A concise empirical correlation for Nusselt number in the non-boiling regime was developed and validated. It was found that the spray cone produced by the pressure swirl nozzles evolved from a hollow cone to a full cone with increasing axial distance. The average droplet size decreases initially with the increase of axial distance but subsequently increases, while the droplet impact energy decreases monotonically. The effect of nozzle-to-surface distance on the heat transfer performance is complex and surface temperature

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dependent. In the full cone spray regime, the effect of nozzle-to-surface distance on the heat transfer performance is not as sensitive as that in the hollow cone spray regime. The developed empirical model indicates that the impinging Reynolds number and surface temperature improve the heat transfer performance, while the nozzle-to-surface distance has a negative effect.

Chapter 4 Thermal effects on the formation of spray cone in spray cooling

Chapter 4

Thermal effects on the formation of spray cone in spray cooling

In Chapter 3, it was observed that the spray cone of pressure swirl nozzle expanded when it impinged on a relatively high temperature surface. Nonetheless, no detailed measurement has been reported regarding this phenomenon in the literature. The expansion of the spray cone in a spray cooling process could affect the thermal performance in terms of the surface temperature non-uniformity and heat transfer efficiency, especially in the phase change regime of spray cooling. In this chapter, the evolution of the spray cone that impinges on a flat surface with different surface temperatures was investigated using two optical measurement techniques. Detailed information of the droplet size, velocity and flux were characterised by the PDA technique as introduced in Chapter 3, while global information of the spray behaviours such as velocity field and spray structure were evaluated using a PIV system.

4.1 Basic principle of the PIV technique

The PIV technique is an optical method of flow visualisation that has been widely used in education and research. The basic principle of PIV measurements is illustrated in Fig. 4.1. In PIV measurements, the property actually measured is the displacement of the seeding particles (or droplets) in the flow field within a known time interval. These seeding particles introduced in the flow field are known as the flow tracers which are used to track the flow field of interest. Therefore, depending on the nature of the flow, the size and density of the particles (or droplets) are of

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vital importance for the particles (or droplets) to truly follow the flow thereby to obtain the accurate velocity field.

Fig. 4.1 Basic principle of PIV measurement

As shown in Fig. 4.1, in order to detect the movement of the seeding particles, an area of the flow field is illuminated by a light sheet. This light sheet, which is generated by a laser and a system of optical components, is not continuous but pulsed to produce a “stroboscopic effect” and freeze the movement of the seeding particles. To detect the illuminated seeding particles, a CCD-camera is positioned at right angle to the area of interest in the light sheet. The pulsing light-sheet and the camera are synchronised so that particle positions at the instant of light pulse number 1 are registered on frame 1 of the camera, and particle positions from pulse number 2 are on frame 2 of the camera.

In data processing, the particle positions appear as light specks on a dark background on each camera frame. As shown in Fig. 4.2, the camera frames are divided into rectangular regions called “interrogation areas” or “interrogation windows”

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(typically from 128×128 , down to 16×16 pixels), and for each of these interrogation areas, the frames from the first and second pulse of the light-sheet are correlated to produce an average particle displacement vector by the means of cross-correlation. Essentially, the cross-correlation function statistically measures the degree of match between the two samples. The location of highest correlation peak in the correlation plane can then be used as a direct measure of the average particle displacement. By doing these for all interrogation areas, a vector map of average particle displacements is produced for the interrogation areas. Eventually, the average particle displacement vectors divided by the known time interval between the two frames are converted into a map of so-called “raw velocity vector”.

Fig. 4.2 PIV measurement using cross-correlation method between two successive frames (Husted *et al.*, 2009)

In most applications, PIV is used to investigate the flow field of a gas flow that is seeded with small particles or droplets (flow tracers) to allow the necessary imaging. However, in the present spray characterisation experiments, no seeding particles are introduced in the flow field, and the velocities measured in this two phase flow (spray to the environment air) are the velocities of the water droplets themselves, but not the air presented in the spray.

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4.2 Experimental apparatus

In this chapter, the selected pressure swirl nozzle for investigation is Nozzle A as discussed in Chapter 3, which performs a full cone spray at a relatively shorter axial distance. The experimental apparatus is described as follows.

4.2.1 Heat transfer measurement apparatus

The heater module is similar to that in Chapter 3 (see Fig. 3.9). The only difference is the locations of the thermocouples being installed. In order to investigate the surface temperature non-uniformity, nine thermocouples in three planes (three thermocouples in each plane) are installed to extrapolate the local surface temperatures at $r = 0$ mm, $r = 6$ mm and $r = 9$ mm, respectively, as shown in Fig. 4.3.

Fig. 4.3 Locations of the thermocouples in the heater block

During the experiments, the spray characterisations of the spray cone impinging on a heated surface were only conducted when the surface temperature reached the steady state status.

4.2.2 PDA measurement apparatus

The PDA setup used to investigate the spray cone impinging on a heated surface is illustrated in Fig. 4.4. Unlike the free spray characterisation in Chapter 3, the

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measuring volume of the PDA system was fixed at 2.5 mm above the heated surface so that the dynamics of the droplets approaching the heated surface can be measured. This is the minimum distance to prevent the intersected laser beams from being blocked by the edge of the heater module. As the spray cone produced by the spray nozzle is approximately axisymmetric, the PDA measurements were only conducted by following a linear measuring grid trajectory which covered the diameter of the spray cone at a predefined axial distance Z ($Z = H - 2.5\text{mm}$).

Fig. 4.4 Schematic of the PDA measurement

During the spray impingement experiments, the PDA system will simultaneously capture the droplets with positive and negative axial velocities at each measuring point. The droplets having positive axial velocity are identified as the impinging droplets while the rest are considered as secondary droplets (generated by rebounding, splashing, and deflection etc.). For data analysis, only the droplet samples with the positive axial velocity were used in the calculation of the representative droplet diameter and velocities of impinging droplets, which, in

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particular, are the Sauter mean diameter (D_{32}), and number-averaged velocity ($\tilde{V}$) and mass-averaged droplet velocity ($\bar{V}$) given by

$$D_{32} = \sum_{i=1}^{N_+} D_i^3 / \sum_{i=1}^{N_+} D_i^2 \quad (4-1)$$

$$\tilde{V} = \sum_{i=1}^{N_+} V_i / N_+ \quad (4-2)$$

$$\bar{V} = \sum_{i=1}^{N_+} \rho D_i^3 \cdot V_i / \sum_{i=1}^{N_+} \rho D_i^3 \quad (4-3)$$

where D_i and V_i are the diameter and velocity of the i th droplet with positive velocity, respectively. N_+ is the number of droplets with positive velocity.

4.2.3 PIV measurement apparatus

Global information of the spray cone formed during spray impingement were acquired using a “LaVision” PIV system, which uses two high-energy (~450 mJ/pulse) Nd: YAG lasers and a 2456×2058 pixels resolution and 16-bit CCD camera. As shown in Fig. 4.5, the laser sheet was aligned to the centre of spray cone, and the CCD camera was mounted perpendicular to the laser light sheet. In the present study, no flow tracers were introduced in the flow field. Hence, the measured velocities in this two phase flow were the velocities of the water droplets, but not the air present in the spray. For each case, a set of 1000 pair images was recorded, and the instantaneous velocity field was computed using a total seven iterations of the algorithm in the PIV software “Davis 8.1.1”. All the velocity vectors were calculated using a cross-correlation technique with a final interrogation windows size of 48×48 pixels with 50% overlap. The velocity fields were validated using a modified median filter to remove erroneous vectors, and each vector field was low-

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pass filtered to remove noise associated with frequencies higher than the sampling frequency of the interrogation.

Fig. 4.5 Schematic of the PIV measurement

4.3 Measurement uncertainties

The uncertainty analysis of the PDA and heat transfer measurements can be found in Chapter 3. In general, the uncertainties of the droplet velocity and droplet diameter are within $\pm 1\%$ and $\pm 5\%$, respectively. Based on the specifications of the PIV system and the experimental setup, the uncertainty of the instantaneous velocity for the image interrogation procedure in this study is approximately 0.1 pixels at 95% confidence level (about 1% of the full range of the velocity measured).

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4.4 Results and discussion

4.4.1 Comparison of PDA and PIV measurement

Before studying the thermal effects on the formation of spray cone in spray cooling, the spray characterisation in free spray was first carried out using the PDA and PIV systems for comparison. The free spray is the spray which is produced by the spray nozzle without any space limitation in the operating environment. Figure 4.6 presents the axial velocity profiles measured at different axial distances below the spray nozzle. The symbols represent the local number-averaged velocities ($\tilde{v}$ in Eq. (4-2)) of the measuring points at a specified axial distance in the PDA measurement. The lines represent the velocity profiles extracted from the PIV flow fields at the same axial distance. These two measuring techniques obtained very similar velocity profiles at the same axial distance below the spray nozzle, which indicates the practicability of using both techniques to evaluate the spray structure of the spray cone in spray atomisation. The spray nozzle produces an approximately axisymmetric spray cone, which is a typical velocity profile for pressure swirl nozzles (Bayvel and Orzechowski, 1993; Xie *et al.*, 2013). As observed from the velocity profiles, two peaks are located at $R_p = 5 \sim 6$ mm from the axis of the spray cone.

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Fig. 4.6 Comparison between PDA and PIV measurements in free spray: $\Delta p = 3$ bar

Figure 4.7 presents the spatial normalised flux distributions of the spray cone measured at different axial distances below the spray nozzle. As compared to the spatial droplet velocity distribution in Fig. 4.6, the peak droplet flux points are at the same radial locations with $R_p = 5 \sim 6$ mm as the peak velocity points occur. Figures 4.6 and 4.7 show that the local droplet velocity and droplet flux are correlated. Namely, a higher local droplet flux corresponds to a higher local droplet velocity in the spatial distributions of droplet flux and droplet velocity. This suggests that both the spatial droplet velocity and droplet flux distributions can be used to describe the spray structure of the spray cone. For brevity, in this chapter the PDA droplet flux distribution and the PIV droplet velocity distribution are used to evaluate the spray cone by taking the advantages of these two techniques.

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Fig. 4.7 Spatial droplet flux distribution by the PDA measurement: $\Delta p = 3$ bar

4.4.2 Effect of surface temperature on the formation of spray cone

In free spray, the radial locations of the peak droplet velocity/flux points in their spatial distributions is determined only by the pressure drop across the spray nozzle (Bayvel and Orzechowski, 1993; Xie *et al.*, 2013), i.e., the radial locations of the peak droplet velocity/flux points are fixed at a constant pressure drop. Therefore, the shift in the radial locations of the peak droplet velocity/flux points in the respective spatial distributions can provide a reference to identify the variation of spray cone in the spray cooling process. For instance, the shift in the radial locations of the peak droplet velocity/flux points along the outward radial direction indicates the expansion of spray cone, and vice versa.

To explore the thermal effects, the droplet flux distribution of the spray cone after interaction with the heated surface was characterised using the PDA technique, as shown in Fig. 4.8. In plotting the graphs, the measuring points where their droplet fluxes are negligible as compared to the droplet flux at the peak point ($< 1\%$) are eliminated in order to indicate an extremely low local droplet flux (or hollow cone

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spray). In free spray, the spray cone produced by pressure swirl nozzles is solely dependent on the pressure drop across the spray nozzle (Bayvel and Orzechowski, 1993). However, during spray impingement, the spray cone formed by the pressure swirl nozzle is found to be influenced by the surface temperature of the impinged surface ($\bar{T}_{surf}$). At $\bar{T}_{surf} = 26.4^\circ\text{C}$ (without heating), the radial locations of the peak droplet flux points remain unchanged at $R_p = 5 \sim 6$ mm for different axial distances below the spray nozzle as shown in Figs. 4.8(a) II to 4.8(d) II. At a relatively higher $\bar{T}_{surf}$, the peak droplet flux points shift outward slightly with increasing $\bar{T}_{surf}$ ($26.4 < \bar{T}_{surf} < 65^\circ\text{C}$), and thereafter shift further away from the axis of the spray cone when $\bar{T}_{surf} > 80^\circ\text{C}$. It is noted that when $\bar{T}_{surf}$ is beyond the boiling temperature ($\bar{T}_{surf} > 100^\circ\text{C}$), the radial locations of the peak droplet flux points shift to $R_p = 9 \sim 10$ mm or even further. This demonstrates a strong thermal effect on the spatial droplet flux distribution of the spray cone formed in spray cooling.

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Fig. 4.8 Evolution of the spray cone development with increasing surface temperature at $\Delta p = 3$ bar: (a) $Z = 17.5$ mm, $H = 20$ mm, (b) $Z = 10.5$ mm, $H = 23$ mm, (c) $Z = 22.5$ mm, $H = 25$ mm, (d) $Z = 25.5$ mm, $H = 28$ mm

Figure 4.9 presents the corresponding images of the formed spray cones in Figs. 4.8(a) and 4.8(d). It is evident that the spray cone expands as $\bar{T}_{surf}$ increases, which agrees with the PDA measurement. The expansion of spray cone can be attributed to the uprising vapour resulting from the liquid evaporation on the heated surface. As shown in Fig. 4.9, more vapour is generated on the heated surface as $\bar{T}_{surf}$ increases, which can be determined from the background light intensities of the captured images. The spray cone is forced to expand due to the additional flow resistance created by the uprising vapour which flows oppositely to the spray cone.

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Fig. 4.9 Visualisation of the spray cones impinging on the heated surface with increasing surface temperature: (a) $H = 20$ mm, (b) $H = 28$ mm

Following the expansion of the spray cone, Fig. 4.8 shows that the droplet flux near the centre of spray cone decreases monotonically. As a consequence, the spray cone evolves from a full cone to a hollow cone. For example, at $H = 28$ mm and $Z = 25.5$ mm, a full cone spray is formed on the heated surface at $\bar{T}_{surf} = 26.4^\circ\text{C}$ as shown in Fig. 4.8(d)II. However, when $\bar{T}_{surf} = 112.1^\circ\text{C}$, a hollow cone spray is formed as shown in Fig. 4.8(d)V. This phenomenon can be explained by the atomisation mechanisms of pressure swirl nozzles as discussed in Section 3.4.2. It is more difficult for the pressure swirl nozzles to maintain or generate a full cone spray when the spray cone angle increases. As illustrated in Figs. 4.8 and 4.9, the expansion of spray cone ultimately increases the spray cone angle as a result.

To further strengthen the finding, the PIV technique was used to provide a global spray structure of the spray cone formed on the heated surface in spray cooling. Figure 4.10 shows the evolution of the velocity vector fields of the spray cones with increasing $\bar{T}_{surf}$.

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Fig. 4.10 Evolution of velocity vector fields of the spray cones impinging on the heated surface with increasing surface temperature ($H = 30$ mm): (a) $\Delta p = 3$ bar, (b) $\Delta p = 4$ bar

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It can be seen that the locations of the peak velocity region (represented by the dash line) shifts outward along the radial direction as $\bar{T}_{surf}$ increases. This reaffirms the PDA results, considering the correlation between the spatial distributions of droplet flux and droplet velocity. As the pressure drop increases from 3 to 4 bar, the shift in the radial locations of the peak velocity regions is relatively smaller. This suggests a reduction of the thermal effects on the spray cone formation at a higher pressure drop. A higher pressure drop increases both the droplet flux and droplet velocity (Xie *et al.*, 2013), which results in a stronger spray in comparison to the uprising vapour. Hence, the uprising vapour has less effect on the spray cone formed at a higher pressure drop. Figure 4.10 also displays that the droplet velocity decreases as $\bar{T}_{surf}$ increases, which will be further discussed in the later sections.

4.4.3 Characteristics of secondary droplets

Since the PDA measurement was fixed at 2.5 mm above the heated surface, the measured secondary droplets with negative velocity are mainly caused by the hydrodynamic impact mechanisms such as droplet rebounding and splashing, as well as the deflection of the small droplets that are elevated by the uprising vapour before impacting on the heated surface. The secondary droplet fraction, defined as the ratio of the number of secondary droplets to the total number of droplets detected at the measuring point (N_{-}/N), is used to estimate the proportion of secondary droplet at each measuring point.

Figure 4.11 shows the spatial distributions of secondary droplet fraction when the spray cone impinges on the heated surface at different $\bar{T}_{surf}$. In contrast to the spatial impinging droplet flux distributions in Fig. 4.8, the secondary droplet fraction

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distribution has an inverse trend wherein a higher impinging droplet flux (see Fig. 4.8) leads to a lower secondary droplet fraction. At the locations with higher impinging droplet fluxes, a stronger impinging droplet flow exists alongside with comparatively weaker uprising vapour. Therefore, the secondary droplet fractions are lower in the regions of higher droplet flux, and vice verses. Moreover, due to the stronger droplet inrush in the regions of higher droplet flux, the secondary droplets are more easily to be redirected back by the impinging droplets (through collision or coalescence mechanisms) at the moment when rebounding or splashing happens.

Fig. 4.11 Spatial distributions of secondary droplet fraction: (a) $H = 23$ mm, $Z = 20.5$ mm; (b) $H = 28$ mm, $Z = 25.5$ mm

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As $\bar{T}_{surf}$ increases, the secondary droplet fraction increases, especially at the centre of the spray cone. This is due to the fact that the enhanced uprising vapour at a higher $\bar{T}_{surf}$ is favourable in elevating the secondary droplets. With an increase of $\bar{T}_{surf}$, the enhanced uprising vapour concentrating towards the centre of the spray cone forces the spray cone to expand. Consequently, the impinging droplet flux at the centre is diluted which resulted in a weaker spray flow (see Fig. 4.8). Hence, the secondary droplet fraction at the centre of spray cone increases more rapidly as $\bar{T}_{surf}$ increases.

Figure 4.12 presents the probability density functions of the secondary droplet diameter and velocity under different spray impingement conditions. It is noted that $\bar{T}_{surf}$ has negligible effect on the probability density functions. The secondary droplet velocity probability density functions reveal that most of the secondary droplets have very low reversed velocities, which was also reported by Panao *et al.* (2009). This infers that the momentum of secondary droplets is relatively small such that the secondary droplets could be easily redirected back to the heated surface by the impinging droplets.

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Fig. 4.12 Probability density functions of secondary droplet diameter and velocity: (a) diameter, (b) velocity

4.4.4 Characteristics of impinging droplets

This section describes the influences of surface temperature on the dynamics of the impinging droplets when they are approaching the heated surface.

Figure 4.13 presents the velocity profiles obtained from the PIV velocity vector fields of the spray cone under various $\bar{T}_{surf}$. These velocity profiles were taken at 3.5 mm above the heated surface ($H=30$ mm and $Z=26.5$ mm).

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Fig. 4.13 Variations of the velocity distributions with increasing surface temperature (PIV measurement)

It is noted that the axial velocity of the impinging droplets decreases monotonically with increasing $\bar{T}_{surf}$. The velocity reduction is more pronounced when $\bar{T}_{surf}$ is close to the boiling temperature. The significant velocity reduction at a higher $\bar{T}_{surf}$ is caused by two aspects. First, the generated uprising vapour is strengthened especially when $\bar{T}_{surf}$ reaches the boiling temperature, by which the impinging droplets are strongly dragged by the uprising vapour. Second, as $\bar{T}_{surf}$ increases, the proportion of secondary droplets has increased. These secondary droplets with

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reversed velocities to the impinging droplets collide with the impinging droplets to decelerate the velocities of the impinging droplets. By increasing the pressure drop from 3 to 4 bar, the velocity reduction is diminished. This again demonstrates that the thermal effects are reduced when the nozzle pressure drop increases.

To exemplify the statistical behaviours of droplet dynamics, Figure 4.14 shows the probability density functions of the impinging droplet diameter and velocity under different $\bar{T}_{surf}$. Unlike the secondary droplets, the probability distribution functions of the impinging droplets are affected by $\bar{T}_{surf}$. In Fig. 4.14(a), at $\bar{T}_{surf} = 26.4^\circ\text{C}$, the diameter with the highest probability density (D_{mp}) increases from $D_{mp} = 120\ \mu\text{m}$ in free spray to $D_{mp} = 160\ \mu\text{m}$ in spray impingement. It is also noted that the probability densities of small droplets decreases, while the probability densities of big droplets increases. The increased probability densities of big droplets in spray impingement are the result of the enhanced coalescing collisions of droplets. As $\bar{T}_{surf}$ increases, the probability densities of big droplets increase concomitantly. This phenomenon could be explained by the proposed spray impingement scenario as depicted in Fig. 4.15. When the impinging droplets approach the heated surface, they will interact with the uprising vapour and secondary droplets. As the small droplets in the spray cone possess low velocities (Santolaya *et al.*, 2010), the small impinging droplets with low momentum can be easily elevated by the uprising vapour before reaching the heated surface. Hence, only the bigger droplets with larger momentum can impact on the heated surface and be measured at the PDA measurement plane (2.5 mm above the heated surface). Moreover, as the secondary droplet fraction increases at a higher $\bar{T}_{surf}$, the coalescing collisions between the impinging and secondary droplets can be further strengthened to generate bigger droplets.

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As shown in Fig. 4.14(b), the velocity with the highest probability density (V_{mp}) decreases with increasing $\bar{T}_{surf}$, which agrees with the PIV measurements that the impinging droplet velocity decreases as $\bar{T}_{surf}$ increases. As expected, the probability densities of the impinging droplets with low velocities are observed to increase significantly with an increase in $\bar{T}_{surf}$.

Fig. 4.14 Probability density functions of impinging droplet diameter and velocity: (a) diameter, (b) velocity

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Fig. 4.15 Droplet behaviours during spray impingement

To characterise the average droplet diameter and velocity of the impinging droplets approaching the heated surface, the Sauter mean diameter (D_{32}) and mass-averaged velocity ($\bar{V}$ in Eq. (4-3)) of the impinging droplets measured at 2.5 mm above the heated surface are summarised in Fig. 4.16. The free spray results (dash lines) are used as baseline to assess the thermal effects. It is found that as $\bar{T}_{surf}$ increases, D_{32} of the impinging droplets increases while $\bar{V}$ of the impinging droplets decreases. This agrees with the previous discussions on the droplet diameter and velocity probability density distributions.

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Fig. 4.16 Typical droplet diameter and velocity with increasing surface temperature: (a) Sauter mean diameter (D_{32}), (b) mass-averaged velocity ($\bar{V}$)

However, as shown in Fig. 4.17, with increasing $\bar{T}_{surf}$, the variations of the impinging droplet diameter (increasing) and velocity (decreasing) together have an un conspicuous effect on the average Weber number ($We = \rho \bar{V}^2 D_{32} / \sigma$) of the impinging droplets. This suggests that the impact energy (We) of impinging droplets is an important criterion for the droplets to impact on the heated surface, i.e., only the droplets with sufficient impact energy can impact on the heated surface at the specified nozzle-to-surface distance. Increasing the droplet impact energy can be accomplished by increasing the droplet size or droplet velocity. Since smaller

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droplets are more conducive to the heat and mass transfer process, the most favourable way to increase the probability of the droplets impacting on the heated surface is to increase the droplet velocity, which is consistent with the findings of Chen *et al.* (2004).

Fig. 4.17 Average Weber number of the impinging droplets with increasing surface temperature

4.4.5 Effect of spray cone formation on spray cooling performance

Section 4.4.2 clearly indicates that the spray cone expands when $\bar{T}_{surf}$ is sufficiently high. In return, the effect of the expanded spray cone on the spray cooling performance is yet to be addressed. This section focuses on a heat transfer experiment aiming to assess the thermally expanded spray cone on two important aspects in spray cooling: surface temperature non-uniformity and heat transfer coefficient. In the experiment, the nozzle-to-surface distance was fixed at $H = 23$ mm by which the spray cone formed in free spray just inscribed the heated surface.

Figure 4.18 presents the surface temperature non-uniformity as a function of $\bar{T}_{surf}$.

The surface temperature non-uniformity is defined as $T_{1@r=0mm} - T_{3@r=9mm}$. As

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shown in Fig. 4.18, the variations of surface temperature non-uniformity with an increase of $\bar{T}_{surf}$ can be divided into two regimes: the low surface temperature regime ($\bar{T}_{surf} \leq 75^\circ\text{C}$) and high surface temperature regime ($\bar{T}_{surf} > 75^\circ\text{C}$).

Fig. 4.18 Surface temperature non-uniformity as a function of mean surface temperature

In the low surface temperature regime, the surface temperature non-uniformity is small due to the low heat load, and $T_{1@r=0mm}$ is lower than $T_{3@r=9mm}$. As $\bar{T}_{surf}$ increases, the difference between $T_{1@r=0mm}$ and $T_{3@r=9mm}$ decreases. Eventually, $T_{1@r=0mm}$ is transited to exceed $T_{3@r=9mm}$ when $\bar{T}_{surf}$ is beyond 75°C which is the beginning of the defined high surface temperature regime. This transition process can be explained by the evolution of spray cone with increasing $\bar{T}_{surf}$ as shown in Fig. 4.8(b)I-III. In the low temperature regime the spray cone gradually expands as $\bar{T}_{surf}$ increases. Correspondingly, the impinging droplet flux decreases at the centre ($r = 0$ mm) and increases at the edge of the heated surface ($r = 9$ mm). As the local surface temperature in the non-boiling regime is solely dependent on the locally

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impinging droplet flux (Abbasi, 2010; Xie *et al.*, 2012; Xie *et al.*, 2013), the transition between $T_{1@r=0mm}$ and $T_{3@r=9mm}$ is due to the variations of their local impinging droplet fluxes with increasing $\bar{T}_{surf}$.

In the high surface temperature regime, $T_{1@r=0mm}$ becomes consistently larger than $T_{3@r=9mm}$. This is also attributed to the variations of their local impinging droplet fluxes with the increase of $\bar{T}_{surf}$ as shown in Figs. 4.8(b)IV - 4.8(b)V. In this regime, the spray cone expands more dramatically, which causes the spray cone to be hollowed and pushes the peak droplet flux region to the edge of the heated surface at $R_p = 9$ mm when $\bar{T}_{surf} = 105.6^\circ\text{C}$ (see Fig. 4.8(b)V). Thus, the heat transfer at the centre of the heated surface becomes less effective compared to that at the edge. When $\bar{T}_{surf}$ approaches the boiling temperature, the surface temperature non-uniformity rises drastically. This is because when $\bar{T}_{surf} \geq 100^\circ\text{C}$, partial “dryout” has been observed at the centre of the heated surface due to the resulted hollow spray cone as shown in Fig. 4.8(b)V.

To further study the effects of spray cone expansion on the spray cooling performance, the heat transfer coefficient as a function of $\bar{T}_{surf}$ in the high surface temperature regime is illustrated in Fig. 4.19. As expected, the heat transfer coefficient increases initially with increasing $\bar{T}_{surf}$, considering that the liquid evaporation becomes more important to the heat transfer when $\bar{T}_{surf}$ is close to the boiling point (Cheng *et al.*, 2010).

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Fig. 4.19 Heat transfer coefficient as a function of surface temperature

However, when $\bar{T}_{surf}$ increases to a few degrees beyond the boiling point (e.g., $\bar{T}_{surf} = 104^{\circ}\text{C}$ at $\Delta p = 3 \text{ bar}$), the heat transfer coefficient starts to decrease. This outcome is attributed to the expanded spray cone when $\bar{T}_{surf}$ is beyond the boiling temperature. There are three consequences of the expanded spray cone on the spray cooling performance. First, by expanding the spray cone, more droplets impinge on the surrounding area or along the circumference of the heated surface while fewer droplets impinge on the centre area. As a result, the sprayed droplet is poorly utilised. Second, with the location of the peak droplet flux region moving to the edge of the heated surface, the fresh impinging liquid is swept off from the heated surface quickly before being sufficiently heated up. Third, partial “dryout” is observed to occur at the centre of the heated surface when $\bar{T}_{surf} \geq 100^{\circ}\text{C}$, which finally deteriorates heat transfer.

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4.5 Summary

This chapter experimentally investigated the thermal effects on the formation of spray cone in spray cooling. The spray cone expands under impinging on a heated surface with high surface temperatures, which consequently influences the spatial distributions of the droplet flux and velocity, as well as the probability density distributions of the diameter and velocity of the impinging droplets and secondary droplets. The heat transfer experiments reveal that the thermally expanded spray cone has significant effects on the surface temperature non-uniformity and heat transfer coefficient in spray cooling.

Chapter 5

Modelling of the thin liquid film flow and heat transfer under spray impingement

5.1 Introduction

When the droplets dispersing from a spray nozzle impinge on a heated surface, a thin liquid film is formed, and a large amount of heat is transported from the heated surface through droplet impingement, thin film convection, evaporation and nucleate boiling etc. Most of the previous studies suggests that the thin film flow under spray impingement has dominant effect on heat transfer performance (Horacek *et al.*, 2005; Shedd and Pautsch, 2005; Shedd and Pautsch, 2006; Yan *et al.*, 2010; Martinez *et al.*, 2011). However, due to the extreme conditions under spray impingement, limited experimental studies were conducted to measure the thickness of the formed thin film. Tilton (1989) firstly estimated the film thickness by using a needle mounted on a traversing measuring scope to measure the film thickness after the hydraulic jump of a single-nozzle spray. Based on the continuity requirements for a single phase liquid film flow, the film thickness under the spray cone was estimated around 120-350 μm before the hydraulic jump with an uncertainty of 10%. Thereafter, Chen *et al.* (1995) experimentally investigated the film thickness produced by a pressure swirl nozzle using a conductive needle probe with a measurement uncertainty about 25%. It was reported that the spray velocity and mass flow rate distribution were the main factors that affect the film thickness under spray impingement. Pautsch and Shedd (2006) applied a non-intrusive optical measurement technique to measure the film thickness produced by single- and four-

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nozzle sprays adiabatically and diabatically. They found that the variation of the film thickness was not measurable by adding a heat load under the single-nozzle spray.

Although the film thickness could be measured by some techniques, the experimental setup is rather complex and the study on thin film velocity measurement is still lacking (Pautsch, 2007). Therefore, a mathematical model to predict the flow pattern of this thin film flow will allow a better understanding of the heat transfer mechanisms in spray cooling. Researchers reported that a low Reynolds number flow ($Re < 1510$) has a strong damping factor which could ensure a laminar flow even under a strong disturbance (Schlichting and Gersten, 2000). When a laminar velocity profile was assumed in the liquid film flow under spray impingement, Chen *et al.* (1995) modelled the thickness of the thin film flow produced by the pressure swirl nozzles. They indicated that the Reynolds number calculated by the experimental film thickness was much smaller than the critical value. Hence, the assumption of a laminar flow in this thin film flow could be reasonable due to its extremely thin film thickness with micrometre magnitude. On the basis of the mass and energy conservations, Yao *et al.* (2006) derived a theoretical model on the film thickness in spray cooling. It was suggested that the film thickness was in direct proportion to the mass flow rate and decreased with the increase of droplet velocity. But no experimental result was compared with the model. In this chapter, a mathematical thin film flow model was first derived from the spray characteristics of spray nozzles to estimate the film thickness. Then, a semi-empirical heat transfer model was developed to predict the heat transfer performance in the non-boiling regime of spray cooling.

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5.2 Mathematical model

Unlike the circular or planar jet impingement (Waston, 1964; Chen *et al.*, 2005), the interface of the liquid film under spray impingement is not an absolute free surface, where there are continuous mass and momentum (droplets) entering, as shown in Fig. 5.1. It is clear that the momentum of the impinging droplets is the only driving force for this thin film flow as there are no other external forces on a horizontal impinged wall. From the Phase Doppler Particle Analyzer measurements (PDPA) (Winkler and Peters, 2002), the frequency of the impinging droplet was observed to be in the thousands, which is too high to consider the effect of individual droplets impacting on the liquid film in mathematical modelling. However, during steady-state impingement, the local film thickness of this liquid film flow was found to be stable (Shedd and Pautsch, 2006; Martinez *et al.*, 2011), viz., at steady state, the high frequency of droplet impingement could generate a continuous and even spray impingement on the liquid film to form a stable liquid film interface. In consideration of this, the averaged summed-up droplet momentum is suggested to be a better choice to include the effect of droplet impingement in the present mathematical model.

Fig. 5.1 Schematic of the thin liquid film flow under spray impingement

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5.2.1 Assumptions

As shown in Fig. 5.1, the thin film flow under spray impingement is assumed to be laminar and balanced by the local droplet momentum source S and the viscous force on the impinged wall. The droplet momentum source S is composed of two components, which are explained in the following assumptions:

1. When the droplets impinge on the liquid film, all the droplets are totally absorbed by the liquid film, and there is no rebound, splash or evaporation occurring so as to satisfy the mass conservation.
2. Abbasi *et al.* (2010) reported that the momentum of the impinging droplets could be converted into a local pressure in the thin film flow in spray cooling. In this study the axial momentum of the droplets is assumed to be converted to the local pressure in the thin film when they are merged into the liquid film,

$$p = \dot{m}V \quad (5-1)$$

where $\dot{m}$ is the local droplet mass flux, and V is the axial component of the droplets velocity. At the same time, the radial momentum of the droplets is assumed to be totally absorbed by the liquid film and the liquid is driven to flow out from the impingement zone. Herein, the radial momentum source S_r is introduced as

$$S_r = \dot{m}V \quad (5-2)$$

3. There is no velocity slip at the boundary of the impinged wall.

5.2.2 Thin liquid film flow model

For the liquid film flow under spray impingement, there are generally two characteristic length scales: the characteristic film thickness (h_{film}) and the characteristic substrate length (R). As the thickness of the thin film is much smaller

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than the substrate length ($h_{film}/R \sim 0$), this thin liquid film flow could be simplified according to the lubrication approximation method. The momentum equations for the thin film flow in axisymmetric cylindrical coordinates are presented as follows:

$$u \frac{\partial u}{\partial r} + w \frac{\partial u}{\partial z} = -\frac{1}{\rho} \frac{\partial p}{\partial r} + \nu \left[\frac{\partial^2 u}{\partial r^2} + \frac{1}{r} \frac{\partial u}{\partial r} - \frac{u}{r^2} + \frac{\partial^2 u}{\partial z^2} \right] + g_r \quad (5-3)$$

$$u \frac{\partial w}{\partial r} + w \frac{\partial w}{\partial z} = -\frac{1}{\rho} \frac{\partial p}{\partial z} + \nu \left(\frac{\partial^2 w}{\partial r^2} + \frac{1}{r} \frac{\partial w}{\partial r} + \frac{\partial^2 w}{\partial z^2} \right) + g_z \quad (5-4)$$

Let U be the scale of u , then by the continuity, the scale of w must be Uh_{film}/R in order not to violate mass conservation. Leaving the pressure scale of P , the normalised variables are introduced and denoted by the primes as follows:

$$r = Rr', \quad z = h_{film}z', \quad u = Uu', \quad w = U \frac{h_{film}}{R} w', \quad p = Pp'$$

The magnitudes of the scale variables are usually stated as follows (Chen *et al.*, 1995; Shedd and Pautsch, 2006; Abbasi *et al.*, 2010; Martinez *et al.*, 2011):

$$R \sim 10^{-2}, \quad h_{film} \sim 10^{-4}, \quad U \sim 10^{-1}, \quad P \sim 10^3, \quad \nu \sim 10^{-6}$$

With these normalised variables, the momentum equations are transformed to

$$\text{Re} \frac{h_{film}}{R} \left(u' \frac{\partial u'}{\partial r'} + w' \frac{\partial u'}{\partial z'} \right) = -\frac{Ph_{film}^2}{\rho \nu R U} \frac{\partial p'}{\partial r'} + \left[\frac{h_{film}^2}{R^2} \left(\frac{\partial^2 u'}{\partial r'^2} + \frac{1}{r'} \frac{\partial u'}{\partial r'} - \frac{u'}{r'^2} \right) + \frac{\partial^2 u'}{\partial z'^2} \right] + \frac{g_r h_{film}^2}{\nu U} \quad (5-5)$$

$$\text{Re} \frac{h_{film}^3}{R^3} \left(u' \frac{\partial w'}{\partial r'} + w' \frac{\partial w'}{\partial z'} \right) = -\frac{Ph_{film}^2}{\rho \nu R U} \frac{\partial p'}{\partial z'} + \frac{h_{film}^2}{R^2} \left[\frac{h_f^2}{R^2} \left(\frac{\partial^2 w'}{\partial r'^2} + \frac{1}{r'} \frac{\partial w'}{\partial r'} \right) + \frac{\partial^2 w'}{\partial z'^2} \right] - \frac{g_z h_{film}^3}{\nu U R} \quad (5-6)$$

On the left hand side of the two equations above, $\text{Re} h_{film}/R$ is much less than 1 in terms of the scale variables so that the inertial terms could be neglected. On the right

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hand side, $Ph_{film}^2 / \rho \nu R U$ has the value of nearly 1. Then, omitting the terms of the order h_{film}/R and the even smaller ones, the momentum equations is finally simplified into Eq. (5-7). The body force term g_r is kept as it will be used to represent the radial momentum source from the impinging droplets in the following analysis.

$$-\frac{\partial p}{\partial r} + \mu \frac{\partial^2 u}{\partial z^2} + g_r = 0 \quad (5-7)$$

As discussed previously, the thin film flow under droplets impingement has a relatively stable interface when it reaches steady state. From this point of view, the multiple impinging droplets being the only driving force for the thin film flow could be addressed as the internal momentum source or body force when they are absorbed into the liquid film, such that the interface of this thin liquid film flow could be treated as a free surface as in jet impingement (Waston, 1964; Chen *et al.*, 2005). According to the second assumption, the momentum source induced by the impinging droplets could be described by two components in the momentum equations. The axial momentum of the impinging droplets results in a local pressure gradient along the radius $-\partial p / \partial r$ as observed in (Abbasi *et al.*, 2010), and the radial momentum of the impinging droplets can be treated like a body force (e.g., the acceleration in r direction, $g_r = \partial S_r / \partial r$ in Eq. (5-7)). Hence, this momentum equation represents the balance between the viscous force and the impinging droplets momentum source.

In the cylindrical coordinates, as shown in Fig. 5.1 the impinging droplets at the location where the nozzle-to-surface distance is H and radius is r have a velocity of V , the axial component of the droplets velocity could be calculated as $VH / \sqrt{r^2 + H^2}$.

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Considering the second assumption, the expression for the local pressure in Eq. (5-7) is given by

$$p = \dot{m} \cdot V \cos \alpha = \dot{m} \cdot V \frac{H}{\sqrt{r^2 + H^2}} \quad (5-8)$$

Likewise, the radial momentum source induced by the impinging droplets is expressed as

$$S_r = \dot{m} \cdot V \sin \alpha = \dot{m} \cdot V \frac{r}{\sqrt{r^2 + H^2}} \quad (5-9)$$

With the non-slip velocity boundary condition at the wall surface and a free surface boundary condition at the thin film interface in the third assumption,

$$u_{z=0} = 0 \quad (5-10)$$

$$\left. \frac{du}{dz} \right|_{z=h_{film}(r)} = 0 \quad (5-11)$$

The velocity expression at the local radius r could be derived from integrating Eq. (5-7),

$$u(r, z) = \frac{1}{\mu} \left(\frac{dS_r}{dr} - \frac{dp}{dr} \right) [h_{film}(r)z - z^2] \quad (5-12)$$

where $h_{film}(r)$ is the local film thickness at the radius r .

According to the first assumption, the mass conservation at the circumference with radius of r is satisfied as

$$m(r) = \int_0^{h_{film}(r)} \rho 2\pi r u(r, z) dz \quad (5-13)$$

where $m(r)$ is the impinged mass flow rate in the circular area with radius of r . Consequently, the local film thickness is obtained from combining Eqs. (5-12) and (5-13).

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$$h_{film}(r) = [m(r) / (\frac{\rho\pi r}{3\mu} (\frac{dS_r}{dr} - \frac{dp}{dr}))]^{1/3} \quad (5-14)$$

In the analysis, this model is derived to approximate the thin liquid film thickness under spray impingement, and the droplet velocity distribution, droplet mass flux distributions and the corresponding nozzle-to-surface distance are required.

5.2.3 Heat transfer model

Pautsch and Shedd (2005) experimentally demonstrated that the single-phase heat transfer is dominant or even responsible for 100% of the heat transfer removal for single-nozzle spray cooling. They applied a non-intrusive optical technique to measure the liquid film thickness under spray impingement with and without adding a heat load, and found that adding a heat load still obtained a stable liquid film thickness and the applied heat load did not affect the film thickness under the single-nozzle spray (Shedd and Pautsch, 2006). Hence, in the non-boiling regime of spray cooling, it can be assumed that the liquid film flow under droplets impingement could be modelled with a stable free interface when it reaches steady state and the thin liquid film flow model is still applicable. Therefore, the element analysis is performed for the thin liquid film flow on the heated surface as shown in Fig. 5.2. Each control element satisfies the local mass and energy conservations in the model analysis.

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Fig. 5.2 Element analysis of the thin liquid film flow in the heat transfer model

In Fig. 5.2, m_{i-1} and $T_{film,i-1}$ are the upstream mass flow rate incoming to the element control volume and its corresponding temperature, respectively. m_i and $T_{film,i}$ are the mass flow rate and mean temperature of the control element, respectively. Finally, $m_{drop,i}$ and T_{drop} are the incoming droplets mass flow rate and the initial droplet temperature, respectively. q''_i is the local heat flux generated from the heated surface. Assuming negligible heat loss from the heated surface, and consider the thin film to the ambient air, the energy conservation for the control element is satisfied as

$$c_p m_{i-1} T_{film,i-1} + q''_i A_i + c_p m_{drop,i} T_{drop} = c_p m_i T_{film,i} \quad (5-15)$$

In the non-boiling regime, the heat transferred from the hot surface to the thin film is attributed to two cooling mechanisms: the droplet impingement cooling and thin film convection.

$$q''_i A_i = q_{drop,i} + q_{film,i} \quad (5-16)$$

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where $q_{drop,i}$ is the rate of heat removed by the droplet impingement process, and $q_{film,i}$ is the heat flux removed by the thin film convection. These two mechanisms are discussed separately below.

(a) Droplet impingement cooling

Issa and Yao (2005) summarised other researchers' work and proposed two correlations for the water droplet impingement heat transfer effectiveness η which is related to the droplets impact energy (i.e., Weber number). The effectiveness parameter η is defined as the ratio of the actual heat transfer rate induced by the droplets to the heat transfer limit of the droplet.

$$\eta = \frac{q_{di}}{m[\Delta h_{fg} + c_{p,l}(T_{sat} - T_l) + c_{p,v}(T_{surf} - T_{sat})]} \quad (5-17)$$

where q_{di} is the rate of heat removed by the impinging droplet, m is the droplet mass flow rate and T_{surf} is the surface temperature.

The droplet wetting heat transfer correlation is used to predict the droplet heat transfer effectiveness in the situations where the droplets can continuously or semi-continuously achieve direct contact with the heated surface ($\eta = 9.844 \times 10^{-2} We^{0.1428}$) (Issa and Yao, 2005). The nonwetting or film boiling heat transfer correlation is applied for the droplets which only make a very short period contact with the heated surface before a vapour film quickly forms between the droplets and the heated surface ($\eta = 3.771 \times 10^{-3} We^{0.691} e^{-9.079 \times 10^{-4} We}$) (Issa and Yao, 2005). In the present study, the wetting heat transfer correlation is utilised as the impinging droplets can reach the heated surface easily through the thin film. Therefore, the heat removed by the droplets impingement could be represented as

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$$q_{drop,i} = \eta_i m_{drop,i} [c_p (T_{surf,i} - T_{drop})] \quad (5-18)$$

(b) Thin film convection

The rate of heat transferred by the thin film convection in the heat transfer model could be addressed as

$$q_{film,i} = \frac{k_l}{h_{film,i}} Nu_i \cdot A_i (T_{surf,i} - T_{film,i}) \quad (5-19)$$

According to Zhao *et al.* (2010), the Nusselt number (Nu_i) in this thin film flow is suggested to be similar as in the laminar boundary layer flow over a flat plate (Incropera, 2011), which is expressed as

$$Nu_i = 0.322 Re_{film,i}^{1/2} Pr_{film}^{1/3} \quad (5-20)$$

Herein, the local Reynolds number ($Re_{film,i}$) is obtained according to the film thickness and velocity predicted by the thin film flow model. Combining Eqs. (5-15), (5-16), (5-18) and (5-19), the surface temperature of the heated wall is derived as

$$T_{surf,i} = \frac{q''_i A_i}{\eta_i m_{drop,i} c_p + \frac{\lambda_l}{h_{film,i}} Nu_i A_i} + \frac{\eta_i m_{drop,i} c_p T_{drop} + \frac{\lambda_l}{h_{film,i}} Nu_i A_i T_{film,i}}{\eta_i m_{drop,i} c_p + \frac{\lambda_l}{h_{film,i}} Nu_i A_i} \quad (5-21)$$

5.3 Results and discussion

Comparisons of the local film thickness (Eq. (5-14)) and the surface temperature (Eq. (5-21)) were made between the proposed model and the published experimental results with water as the test fluid.

5.3.1 Results of thin liquid film flow model

The experimental data for comparisons are from the research work done by Chen *et al.* (1995). In their experiments, the droplet velocity, diameter and mass flux

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distributions of a pressure swirl nozzle were measured by a PDPA system. The thickness of the liquid film flow was measured by a conductive needle probe under steady state. In the thin film flow modelling, the characteristic velocities applied were 18.8 m/s, 21.2 m/s and 23.4 m/s corresponding to the nozzle pressure drops of 276 kPa, 345 kPa and 414 kPa, respectively. The reported droplet mass flux distribution along the radius is illustrated in Fig. 5.3, which is represented by the fitting polynomial in Eq. (5-22).

$$\dot{m}(r)=29.182-1.062\times10^4r+1.912\times10^7r^2-3.305\times10^9r^3 \tag{5-22}$$

Fig. 5.3 Mass flux distribution of a pressure swirl nozzle (Chen *et al.*, 1995)

Due to the relatively hollow cone spray, Eqs. (5-12) and (5-14) may not be suitable at the small radius region so that the calculation from the model starts from $r = 0.8$ mm. Comparisons between the model predictions and experimental results are demonstrated in Figs. 5.4 and 5.5. It is observed that the predicted film thickness by the thin film flow model is very close to the experimental results to be within the error of the experimental measurement.

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Fig. 5.4 Comparison of local film thickness in predictions and experiments

Fig. 5.5 Comparison of film thickness under different nozzle inlet pressure

In Fig. 5.4, the model prediction is compared with the experimental results at the nozzle pressure drop of 276 kPa. It is observed that the liquid film thickness is sensitive to the droplets mass flux distribution in Fig. 5.3. The film thickness is smaller at the location under a higher flux spray than that at the location under a lower flux spray. It is believed that the resulted driving force under the high flux spray region is stronger in the liquid film to accelerate the thin film flow and consequently reduce its film thickness. Near the centre of the impinged zone, the

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low flux spray results in a lower pressure in the liquid film as measured (Abbasi *et al.*, 2010). The positive pressure gradient along the radius could result in a reduced film velocity according to Eq. (5-12), and finally, a thick liquid film is built up to increase the flow section area in order to assist the liquid outflow. This phenomenon is also reflected in the experimental results as shown in Fig. 5.4.

Instinctively, it is expected that the film thickness would increase with nozzle pressure drop due to the larger flow rate. However, the model prediction shows that the film thickness decreases with the increase of nozzle pressure drop as shown in Fig. 5.4. It is reasonable that the impact energy of impinging droplets increases as well as the nozzle pressure drop and flow rate. The counterbalance between them leads to a slight reduction of film thickness at the higher nozzle pressure drop. In Fig. 5.5, a favourable comparison is also observed between the model predictions and experimental results at different nozzle pressure drops. As the nozzle pressure drop increases, the predicted film thickness decreases slightly, while the experimental results show a more or less constant film thickness as the nozzle pressure drop ranges from 276 to 414 kPa.

5.3.2 Results of heat transfer model

Comparison between the heat transfer model and experimental results (Cheng *et al.*, 2010; Zhao *et al.*, 2010) is illustrated in Fig. 5.6. The experimental spray characteristics of a pressure swirl nozzle including the spatial distributions of droplet flux, velocity, and diameter are obtained from Cheng *et al.* (2010). The flow conditions are set as $q''_i = 108 \text{ W/cm}^2$, and $T_{drop} = 28^\circ\text{C}$ with reference to Cheng *et al.* (2010) and Zhao *et al.* (2010). It is observed that the heat transfer model is able to predict the surface temperature reasonably well. As seen in Figs. 5.3 and 5.6, for

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the two investigated nozzle-to-surface distances, a higher surface temperature is observed at the centre and edge of spray cone where the impinging droplet flux is low, while a lower surface temperature is obtained in the region $1 < r < 3.5$ mm where the impinging droplet flux is high. This infers that the droplet impingement cooling could be the most important heat transfer mechanism in the non-boiling regime of spray cooling.

Fig. 5.6 Comparison of surface temperature in modelling and experiments

In an effort to investigate the heat transfer contributions by droplet impingement and thin film convection quantitatively, the proportions of the heat removed by these two mechanisms were evaluated from the heat transfer model. In Eq. (5-23), the local heat transfer contribution by droplet impingement is defined as

$$\xi_{drop,i} = \frac{q_{drop,i}}{q''_i A_i} \times 100\% \tag{5-23}$$

and the local heat transfer contribution by thin film convection is defined as

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$$\xi_{film,i} = 1 - \xi_{drop,i} \tag{5-24}$$

As shown in Fig. 5.7, the heat transfer contribution by the droplets impingement achieves the maximum value at the centre ($r = 0$), then decreases drastically ($0 \text{ mm} < r < 1 \text{ mm}$) to a median value ($1 \text{ mm} < r < 3.5 \text{ mm}$), and finally reduces to the minimum value at the edge ($3.5 \text{ mm} < r < 6 \text{ mm}$). This could be due to: (1) the lower droplet mass flux at the centre ($r = 0$) which forms a thicker liquid film with smaller velocity so that the heat transfer contributed by the thin film convection is much smaller compared to the droplet impingement; (2) as the radius increases, the higher droplet flux ($1 \text{ mm} < r < 3.5 \text{ mm}$) tends to reduce the film thickness and speed up the thin film flow, which intensifies the thin film convection; (3) near the edge of spray cone, the lower droplet mass flux makes the droplet impingement heat transfer contribution even smaller, and the thin film convection becomes more important ($3.5 \text{ mm} < r < 6 \text{ mm}$).

Fig. 5.7 Heat transfer contributions in the heat transfer model analysis

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For further validation of this model, the predicted mean surface temperatures at different nozzle-to-surface distances are also compared with the experimental results as shown in Fig. 5.8. The square symbols represent the experimental results obtained by Cheng *et al.* (2010), and the solid and dash lines are the predicted mean surface temperature and the heat transfer contribution by droplet impingement, respectively. It is observed that the mean surface temperature increases with an increase of nozzle-to-surface distance in both the experimental data and model predictions, and the heat transfer contribution by droplet impingement is the primary heat transfer mechanism ($> 60\%$) which decreases as the nozzle-to-surface distance increases. This phenomenon could be resulted from the reduced impact energy of the droplets when the nozzle-to-surface distance increases.

Fig. 5.8 Comparison of mean surface temperature in modelling and experiments

At the lower nozzle-to-surface distance ($H < 7$ mm), the model prediction is much closer to the experimental results, while at the higher nozzle-to-surface distance ($H \geq 7$ mm) the deviation between the model prediction and experimental results

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becomes less satisfactory. This could be due to the high surface temperature which causes significant evaporation at the thin film interface. As discussed in Chapter 4, the uprising vapour drags the impinging droplets to either reduce the impact energy or even blow away the droplets before it contacted with the heated surface during the experiment. Therefore, the reduced droplet impact energy or total mass flow rate would cost the heat transfer by the insufficient use of the impinging droplets. However, in the model analysis, the spray characteristics applied are obtained from the free spray atomisation, and the drag effect at the high surface temperature is not considered. As a result, the predicted surface temperature 25% lower than that in the experiments.

5.4 Summary

In this chapter, a thin liquid film flow model has been derived to investigate the thin film flow under spray impingement. The comparison of the liquid film thickness between the proposed model and the measured data (Chen *et al.*, 1995) shows good agreement. Also, a heat transfer model was proposed to predict the heat transfer performance in the non-boiling regime of spray cooling. It is found that the heat transfer model gives reasonable agreement with the measured surface temperature at various nozzle-to-surface distances. The model prediction suggests that droplet impingement is the primary heat transfer mechanism in the non-boiling regime of spray cooling.

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Chapter 6

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6.1 Introduction

A majority of experimental studies have been conducted to understand the heat transfer mechanisms of spray cooling. Shedd and Pautsch (2005; 2006) reported that the droplet induced single phase convection dominated the heat transfer process in spray cooling. Abbasi *et al.* (2010) correlated the local heat transfer coefficient to the droplet exerted local dynamic pressure. A higher local impinging droplet flux resulted in a higher local dynamic pressure, which led to a higher local heat transfer coefficient. They suggested that the impinging droplets induced heat transfer played the dominant role in the non-boiling regime of spray cooling. Xie *et al.* (2013) found that the local surface temperature is sensitive to the spatial droplet flux distribution. A higher local impinging droplet flux resulted in a lower local surface temperature and hence, a higher local heat transfer coefficient. In the boiling regime, Yang *et al.* (1996) suggested that the high heat flux achieved by spray cooling was due to the tiny bubbles entrapped by the impinging droplets when they were striking on the liquid film. These tiny bubbles worked as nucleation sites (secondary nuclei) in the liquid film to enhance bubble boiling in spray cooling. Rini *et al.* (2002) experimentally observed that the nucleation sites in spray cooling was much more than that in pool boiling at the same surface superheat. They attributed the increased nucleation sites in spray cooling to the secondary nuclei induced by spray droplets.

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Chen *et al.* (2002; 2004) systematically investigated the influences of spray characteristics (such as droplet flux, velocity and diameter) on the CHF of spray cooling by varying one parameter but keeping others constant. However, without in-depth investigation on the spray characteristic effects on the liquid film flow where the heat and mass transfer takes place, it is still unclear about the respective influence of each spray characteristics. Such in-depth experimental investigation on the liquid film flow is too difficult to conduct due to the complex phenomena involved. Recently, most of the fundamental studies on the liquid film flow in spray cooling are through computational simulation by simplifying the spray cooling process. In the simplest way, Selvam *et al.* (2005; 2009) derived a numerical model to simulate the process of a single droplet impinging on a thin liquid film with and without a growing bubble inside. Chen *et al.* (2008) simulated the interaction of the impinging droplets and growing bubbles in the liquid film under a uniform droplet flux impingement. The relationships between droplet flux and bubble size, droplet flux and bubble puncture rate, along with their effects on the heat transfer performance were investigated. Zhao *et al.* (2010) developed a numerical model based on spray characteristics to investigate the heat transfer rates due to different heat transfer mechanisms.

However, all these simulation models either consume a lot of computation time, or only consider one or two aspects of spray cooling. Few studies have proposed to simulate the spray cooling process in a more general way. In this chapter, a dynamic spray cooling model which includes simulating the dynamics of droplet impingement, bubble boiling, and their interactions is presented. Using the experimental spray characteristics as boundary conditions, the model predictions are compared with the experimental results. The parametric effects including the droplet

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diameter, droplet impinging flux, and bubble dynamics are investigated, and the heat transfer rates due to different heat transfer mechanisms are discussed.

6.2 Simulation model

As shown in Fig. 6.1, the spray cooling process mainly includes two parts: (1) the liquid atomisation process, which is to generate droplets with different velocities and diameters, and spatially distributed droplet fluxes, and (2) the heat transfer process, which includes the heat transfer by droplet impingement as well as bubble boiling in the liquid film formed on the heated surface. Therefore, the presented simulation model focuses on these two parts which are discussed in the following sections in detail.

Fig. 6.1 Schematic of the spray cooling process (Sarkar and Selvam, 2009)

6.2.1 Spray characteristics model

The spray characteristics obtained in Chapter 4 show that the spray droplet diameter and velocity have unique probability density distributions at different axial distances and nozzle pressure drops. Moreover, the spatial droplet flux distribution is also unique for the spray nozzle operating at different conditions. To simulate the spray

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cooling process generally and accurately, it is necessary to import the experimental spray characteristics as boundary conditions in the simulation. In the proposed model, three aspects of the droplet hydrodynamics have been considered: (1) probability density distribution of droplet diameter, (2) probability density distribution of droplet velocity, and (3) spatial distribution of droplet flux.

To simulate the droplet dynamics, the Monte Carlo simulation proposed by Kreitzer and Kuhlman (2010) is adopted. The concept of this approach is to generate a group of random numbers which satisfy the specified probability density distributions of the spray characteristics in experiment. Practically, an inverse transformation method is performed to find the desired probability density functions to represent the experimental droplet diameter and velocity distributions. The desired probability density functions are then performed in the numerical code to randomly assign the values of diameter and velocity for each impinging droplet. The “rand” function in MATLAB is used to perform the inverse transform method to find the desired probability density functions of the experimental spray characteristics. The comparison between the experiment and simulation will be discussed in the later section.

Spatial droplet flux distribution has significant effects on the heat transfer performance of spray cooling, especially on the surface temperature distribution (Abbasi *et al.*, 2010; Xie *et al.*, 2012; Xie *et al.*, 2013). Therefore, it is essential to include the effects of spatial droplet flux distribution in the simulation model for spray nozzles operating at different working conditions. In PDA measurement, the number of droplets passing through the measuring volume of the PDA system in a certain time period is analysed. Using the “arriving time” for the first and last

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droplets passing through the measuring volume, a “mean arriving time interval” for two consecutive droplets passing through the measuring location can be estimated. Hence, the experimental droplet flux distribution can be achieved in the simulation by using the “mean arriving time interval”. A smaller “mean arriving time interval” corresponds to a higher local droplet flux in the simulation, as shown in Fig. 6.2.

Fig. 6.2 The spatial droplet flux distribution and “mean arriving time interval” distribution

6.2.2 Heat transfer model

When the surface becomes superheated, the heat transfer process of spray cooling can be considered as film flow boiling which combines the single phase convection and nucleate boiling. Bejan (2004) reported that the flow boiling heat transfer can be approximated by combining the heat removal by the single phase convection and nucleate boiling occurring on the heated surface. Hence, the heat removed in spray cooling could be reasonably approximated by combining two closely interrelated parts: (1) forced convection, which is primarily induced by the impinging droplets (Issa and Yao, 2005; Abbasi *et al.*, 2010; Xie *et al.*, 2012), and (2) nucleate boiling, which is due to the bubbles growing and rupturing in the liquid film (Rini *et al.*,

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2002). The superposition of the heat fluxes is represented in Eq. (6-1). Herein, q'' is the heat flux applied on the heater surface, q''_{di} is the heat flux removed by the forced convection due to the droplet impingement, and q''_{nb} is the heat flux removed by the bubble boiling.

$$q'' = q''_{di} + q''_{nb} \quad (6-1)$$

Droplet impingement heat transfer (q''_{di})

To estimate the droplet impingement heat transfer rate, the heat transfer effectiveness η (Issa and Yao, 2005) for a single droplet impinging on a heated surface is employed as discussed in Section 5.2.3. The effectiveness parameter η is defined as the ratio of the actual heat transfer induced by the droplet to the heat transfer limit of the droplet, as shown in Eq. (6-2). Herein, q_{di} is the amount of heat removed by the droplet impingement, $\dot{m}$ is the droplet mass flow rate and T_{surf} is the surface temperature.

$$\eta = \frac{q_{di}}{\dot{m}[\Delta h_{fg} + c_{p,l}(T_{sat} - T_l) + c_{p,v}(T_{surf} - T_{sat})]} \quad (6-2)$$

Particularly, a wetting heat transfer correlation ($\eta = 9.844 \times 10^{-2} We^{0.3428}$) for droplet impingement cooling is derived to predict the droplet impingement heat transfer effectiveness in the situations where droplets can directly contact with the heated surface (Issa and Yao, 2005). The focus of the present study is to simulate the film flow boiling under droplet impingement. Hence, the wetting heat transfer correlation for droplet impingement is adopted in the numerical model to estimate the forced convection heat transfer rate.

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Bubble boiling (q''_{nb})

In the visualisation experiments of pool boiling and spray cooling, previous works (Rini *et al.*, 2001; Tan, 2001; Chen *et al.*, 2008) show that the bubble growing and disappearing behaviours are similar in both pool boiling and spray cooling. In spray cooling, there are two major distinct processes by which nucleation sites appear: fixed nucleation sites and secondary nuclei. The fixed nucleation sites are due to surface roughness which acts as cavities on the heated surface to trap vapour and serve as the nucleus during the boiling process. The secondary nuclei are introduced by two phenomena in spray cooling: (1) the impinging droplets carry some tiny bubbles at the interface of the droplets which enter the liquid film to form nucleation sites (Yang *et al.*, 1996; Rini *et al.*, 2002), and (2) when bubbles burst or are punctured by the impinging droplets, some tiny bubbles will be generated in the liquid film to form new secondary nuclei for bubble boiling (Bergman and Mesler, 1981; Gopalen and Mesler, 1990). Since the impinging droplet flux is significantly large and the broken bubbles induce new secondary nuclei in spray cooling, the secondary nuclei are deemed to overwhelm the effect of the fixed nucleation sites on the heated surface (Chen *et al.*, 2008). Hence, the proposed model mainly focuses on the bubble boiling due to the secondary nuclei.

In an attempt to simulate the bubble boiling heat transfer rate in spray cooling, the bubble size versus the bubble growing time is needed. Carey (2008) stated in his review that the growing bubble radius for the heat-transfer-controlled growth bubbles is proportional to the square root of the bubble growing time ($t^{1/2}$). Recently, Rini *et al.* (2001; 2008) experimentally validated that the relationships between the bubble radius and bubble growing time in both pool boiling and spray cooling are

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similar to the heat-transfer-controlled growth bubble model ($t^{1/2}$). Carey (2008) reported that the heat-transfer-controlled growth bubbles are more likely to occur in the liquid with high latent heat of vaporisation and moderate contact angle at low wall superheat. As water is the working fluid in the present study, the heat-transfer-controlled bubble growth model by Carey (2008) is employed to approximate the bubble radius at different bubble growing times, as shown in Eq. (6-3).

$$r_{bub} = 0.470 Ja Pr^{-1/6} (\alpha_l t)^{1/2} \quad (6-3)$$

where, r_{bub} is the bubble diameter, Ja is the Jacob number, and α_l is the thermal diffusivity of liquid.

To account for the interactions between the impinging droplets and growing bubbles, three cases viz.: (a) bubble collapsing, (b) bubble puncturing, and (c) bubble merging, are evaluated according to the criteria stated below. The schematics for these cases are depicted in Figs. 6.3(a) to 6.3(c). Besides, a “snapshot” of the impinging droplets and growing bubbles on the heated surface in a simulation case is illustrated in Fig. 6.3(d).

(a) Bubble collapsing due to the limited size

A growing bubble has a limited bubble diameter (D_m) which is similar to the magnitude of the liquid film thickness (h_{film}) (Tan, 2001). It is assumed that the growing bubble will collapse when the bubble is growing beyond the liquid film thickness, as the bubble is easy to break up when it is exposed to the free interface of the liquid film. Once the bubble collapses, the volume of the bubble is used to calculate the bubble boiling heat transfer rate.

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Fig. 6.3 Schematics of interactions between impinging droplet and growing bubble in the simulation. (a) bubble collapsing; (b) bubble puncturing; (c) bubble merging; (d) a “snapshot” of the impinging droplets and growing bubbles in a time step ($\Delta t_D = 0.01$ ms, $q'' = 88$ W/cm², $D_{32} = 182$ μ m)

(b) Bubble puncturing

When a droplet impinges onto a location where a bubble is growing, the bubble may be punctured or continuously growing depending on the sizes of the growing bubble and impinging droplet. The criterion to judge the possibility of bubble puncturing is that, when $R_b + D_d > L_m$, the bubble is said to be punctured by the impinging droplet, otherwise the bubble will continue to grow until it is punctured by the consecutive

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impinging droplet or collapsed due to D_m . In this criterion, R_b is the radius of the growing bubble in the numerical grid cell with a dimension of L_m , and D_d is the diameter of the incoming droplet impacting on the grid cell. The determination of the magnitude of L_m will be discussed in the later section 6.2.3. Once the bubble is punctured, the released vapour volume of the punctured bubble is calculated to estimate the bubble boiling heat transfer rate. Meanwhile, a secondary nucleus is introduced by the impinging droplet to the impinged location.

(c) Bubble merging

When a droplet impinges on a location where a small bubble is growing, the existing bubble may not be punctured and the incoming droplet will bring one more secondary nucleus into the liquid film to activate a new bubble at the impinged location. As the bubbles are growing with time, the bubbles will merge to form a new bigger bubble when $R_1 + R_2 > L_m$. In this criterion, R_1 is the radius of the original growing bubble and R_2 is radius of the new bubble induced by the impinging droplet. The new bubble which combines the volume of the merged bubbles will continuously grow until it collapses due to D_m or is punctured by the consecutive impinging droplet.

6.2.3 Simulation procedure

A dynamic model with discrete time steps was developed. The simulation was based on the experimental droplet diameter and velocity probability density distributions, as well as the spatial droplet flux distribution which is a function of the local “mean arriving time interval” of two consecutive impinging droplets (see Fig. 6.2). They were evaluated and initialised as the input parameters in the simulation. The thickness of the liquid film on the heated surface was estimated from the analytical

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model presented in Chapter 5. In order to simulate the dynamics of the droplet impingement, bubble boiling, and the heat transfer process, two simulation time steps: (a) small time step (Δt_D), (b) large time step (Δt_H), were used in the numerical model.

The small time step Δt_D is to simulate the real-time dynamics of droplet impingement, bubble boiling, and their interactions in the liquid film. In performing the simulation, the code will scan if the location had been impinged by a droplet or if a growing bubble had collapsed or had been punctured at each Δt_D . If the incidences occur, the code will record the corresponding heat transfer rates due to the droplet impingement or bubble boiling. In such a short period of Δt_D , the droplets are randomly impinging on the heated surface as shown in Fig. 6.3(d), and hence, the effects of the spatial droplet flux distribution on heat transfer cannot be properly reflected (e.g., surface temperature distribution). Therefore, a large time step Δt_H is required to include the effect of spatial droplet flux distribution. By performing the simulation continuously in N small time steps ($N \cdot \Delta t_D$), the heat transfer rates due to the droplet impingement and bubble boiling in a large time period ($\Delta t_H = N \cdot \Delta t_D$) are calculated. This large time period Δt_H is then used to iterate the local surface temperatures according to their energy conservations. Once the local surface temperatures are updated, the new local surface temperatures are utilised in the next cycle of Δt_H to calculate the instantaneous heat transfer rate per Δt_D . The whole process will be repeated until the convergence condition is reached.

The flow chart of the simulation is shown in Fig.6.4.

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Fig. 6.4 Flow chart of the simulation model

It is said that the convergence exists for the numerical model if the bubble density on the surface becomes dynamically unchangeable (i.e., the surface temperature is dynamically unchangeable) and the energy balance is satisfied ($q'' = q''_{di} + q''_{nb}$). As shown in Fig. 6.5, the proposed simulation model finally reaches the same steady state condition when the simulations are iterated from two different initial surface temperatures (low and high initial surface temperatures).

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Fig. 6.5 Simulation converged at two different initial surface temperatures for the case: $q'' = 100 \text{ W/cm}^2$, $D_{32} = 136 \text{ }\mu\text{m}$. (a) number of nucleation sites; (b) iterated surface temperature

In this study, Δt_D was specified as 0.01 ms. It will take more than 100 time steps of Δt_D to simulate two consecutive droplets at the peak droplet flux location, and hence the process of bubble growing, bubble puncturing, merging and collapsing were simulated in real-time. The large time step was chosen as $\Delta t_H = 0.2 \text{ s}$ ($2 \times 10^4 \Delta t_D$) so that the effects of the droplet diameter, velocity statistical distributions, and the spatial droplet flux distribution can be sufficiently reflected on the surface temperature distribution.

Using the PDA measurement data in Chapter 3, the film thickness and velocity estimated from the analytical model in Chapter 5 are 200 μm and 0.16 m/s, respectively. According to Tan (2001), the limited bubble diameter (D_m) existing in the liquid film is assumed to be 200 μm . In consideration of the bubble dynamics in the liquid film, the grid cell size L_m was chosen to be 200 μm in the simulation. Assuming that bubbles move with the same speed as the liquid film, at least 1.25 ms would be needed for the bubbles move to the neighbouring cell. However, for the

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bubble growing to the maximum size of 200 μm , only 0.69 ms would be needed when the surface superheat is 10°C. This means that the bubble collapses due to D_m before it can move to the neighbouring cell, and the corresponding heat transfer rate by the bubble can only contribute to the grid cell which holds this bubble. Considering the discrete time step of 0.01 ms, 69 discrete time steps are needed to reach D_m at a surface superheat of 10°C, which means that there are up to 69 discrete bubble sizes existing in the liquid film on the heated surface.

6.3 Comparison of spray characteristics

The experimental data for comparisons are from the PDA measurements in Chapter 3. The simulated conditions are: Nozzle A at $\Delta p = 3.0$ bar, $\dot{m} = 1.8$ g/s and $H = 23$ mm. The comparisons of the experimental and simulated results for the droplet diameter and velocity distributions are shown in Figs. 6.6(a) - 6.6(b). Figure 6.6(c) shows the comparison of the experimental and simulated data for the normalised spatial droplet flux distribution which is defined as the ratio of the local droplet flux to the peak droplet flux.

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Fig. 6.6 Comparison of the spray characteristics in experiment and simulation: (a) droplet diameter distribution; (b) droplet velocity distribution; (c) spatial droplet flux distribution; (d) random number of the impinging droplet per Δt_D

In Figs. 6.6(a) - 6.6(b), the experimental droplet diameter and velocity distributions are based on 136800 impinging droplets obtained in the PDA measurement, and the simulated droplet diameter and velocity distributions are evaluated from 10^6 impinging droplets in the simulation. It is observed that the simulated results of the impinging droplets compare favourably with the experimental results, which indicates a great advantage of using the proposed model to simulate the spray cooling process. As shown in Fig. 6.6(c), the spatial droplet flux distribution is well controlled by the local time intervals in the simulation model. Figure 6.6(d) illustrates the random number of impinging droplets per time step Δt_D in the simulation. At each Δt_D , the number of droplets impinging on the heated surface is

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random from 10 to 50, but is overall averaged at 32 in order to satisfy the flow rate in the experiments.

6.4 Results and discussion

6.4.1 Validation and comparison

The bubble boiling heat transfer model was first validated using the experimental results of Rini *et al.* (2002). In their experiments, the impinging droplets were saturated, the number of nucleation sites on a uniform temperature surface was captured using a high speed camera, and the corresponding heat flux was measured. In the simulation, a uniform droplet flux distribution of 44 impinging droplets per time step ($\Delta t_D = 1 \times 10^{-5} s$) was applied to match the experimental conditions, and the experimental heat flux was applied as the boundary condition. The surface temperature was predicted and the corresponding number of nucleation sites on the heated surface was counted. Table 6.1 shows the comparisons between experiments and simulations. It is observed that the simulated surface temperatures are slightly lower than the experimental results, but overall the predicted surface temperatures agree favourably with the experiments in both value and trend.

Table 6.1 Comparison between experiment and simulation

Study cases	Heat flux (W/cm ²)	Droplet flux (1/cm ² ·s)	Nucleation sites	Surface Temperature(°C)
Experimental results (Rini <i>et al.</i> , 2002)	20.0	4,443,000	2400	63.0
	25.0	4,443,000	2900	65.5
	32.0	4,443,000	3500	67.0
Simulation results	20.0	4,440,000	3200	60.2
	25.0	4,440,000	3400	63.9
	32.0	4,440,000	3700	66.4

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The simulation model was further validated with the experimental surface temperature distribution and heat transfer coefficients in Chapter 4. Three local surface temperatures measured at $r = 0$ mm, $r = 6$ mm and $r = 9$ mm, were used to compare with the predicted surface temperature map.

Since the spray cone is characterised to be approximately axisymmetric, only a quarter of the heated surface is performed in the simulation. Figure 6.7 presents the one-dimensional temperature distribution and a two-dimensional temperature contours. In comparison to Fig. 6.6(c), the predicted surface temperature distribution is inversely related to the spatial droplet flux distribution. The surface temperature is lower in the region under a higher droplet flux impingement and vice versa. Since the spray cone is assumed to be axisymmetric, the surface temperature is expected to be the same for a given radius r^* at different locations. However, the simulation demonstrates surface temperature fluctuating at $r = r^*$. This is because the droplets impinging on the locations with $r = r^*$ have random diameters and velocities (see Figs. 6.6(a) and 6.6(b)) even though the droplet fluxes are the same. Besides, the bubble dynamics at the locations with $r = r^*$ are different due to their unique interactions between bubbles and impinging droplets in the random simulation process. Nonetheless, there is a clear trend of surface temperature versus radial distance which agrees with the experimental results. Especially, a favourable comparison is observed in the region under a higher droplet flux impingement at $r = 6.0$ mm. This is because the heat transfer in the region under a high droplet flux impingement is dominated by the droplet impingement cooling (Xie *et al.*, 2012). In the regions with lower impinging droplet fluxes, the predicted surface temperatures are higher as compared to the experimental results. This is probably due to the non-inclusion of film flow convection in the proposed model.

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Fig. 6.7 Comparison of the surface temperature distribution in experiment and simulation: $q'' = 88 \text{ W/cm}^2$, $D_{32} = 182 \text{ }\mu\text{m}$

Further, comparisons for the heat transfer coefficient at different heat fluxes are shown in Fig. 6.8. It is noted that the simulation agrees well with the experiment when the surface temperature is below and near the boiling point ($\bar{T}_{surf} \leq 105^\circ\text{C}$). However, more discrepancies occur when $\bar{T}_{surf} > 105^\circ\text{C}$. This could be explained by the differences between the spray conditions in the simulations and experiments. As observed in Chapter 4, the spray cone was thermally expanded when $\bar{T}_{surf} > 100^\circ\text{C}$. As a result, the impinging droplet flux at the centre of the heated surface decreases, and a hollow spray cone forms on the heated surface. The expanded spray cone causes a decrease in the heat transfer coefficient due to the wastage of spray droplets and partial “dryout” at the centre of the heated surface. However, in the simulation, the input spatial droplet flux distributions are based on the free spray condition. Besides, in the experiments, the droplet velocity was found to decrease with increasing $\bar{T}_{surf}$, which could also cause the heat transfer coefficient to decrease when $\bar{T}_{surf}$ reaches a certain level, e.g., $\bar{T}_{surf} > 105^\circ\text{C}$.

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Fig. 6.8 Comparison of heat transfer coefficients in the experiments and simulations

6.4.2 Parametric study of heat flux

To quantitatively investigate the effects of the different heat transfer mechanisms, the heat transfer rates due to the droplet impingement and bubbles boiling were evaluated for different heat fluxes. As shown in Fig. 6.9(a), the droplet impingement heat transfer rate (n_{di}) decreases as the heat flux increases, while the bubble boiling heat transfer rate (n_{nb}) increases. At $q'' = 70 \text{ W/cm}^2$, more than 60% of the heat is removed through the droplet impingement. As q'' increases up to 150 W/cm^2 , the heat transfer rate contributed by bubble boiling rises to 75%. In the aspect of the local heat transfer contribution, as expected, the heat transfer mechanisms are heavily dependent on the spatial droplet flux distributions which can be seen by Figs. 6.6(c) and 6.9(b).

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Fig. 6.9 Contributions of the heat transfer mechanisms in the simulations

To better understand the effects of heat flux on the bubble dynamics in spray cooling, the bubble characteristics were evaluated. The spatial distributions of bubble collapsing flux due to the limited bubble diameter (D_m) under different heat fluxes are illustrated in Fig. 6.10 (the contour is the number of collapsing bubbles / Δt_H). It is noted that as heat flux (or $\bar{T}_{surf}$) increases, the number of collapsing bubbles increases. This is because a higher heat flux (or $\bar{T}_{surf}$) speeds up the growth of the bubbles, which enhances the probability of bubbles growing to D_m before being

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punctured by the impinging droplets. The spatial distribution of the bubble collapsing flux also shows that the local bubble collapsing flux is dependent on the local impinging droplet flux. In the region under a higher droplet flux impingement, fewer bubbles can grow to D_m because of a higher impingement frequency and a lower surface temperature as shown in Figs. 6.6(c) and 6.7. The situation of the bubbles growing on the heated surface can be referred to Fig. 6.3(d). With these observations, the simulation model can qualitatively predict the location where “dryout” will first occur.

Fig. 6.10 Spatial flux distributions of the bubble collapsing flux in a period of Δt_H : (a) $q'' = 80 \text{ W/cm}^2$; (b) $q'' = 150 \text{ W/cm}^2$

Pais *et al.* (1992) suggested that “dryout” in spray cooling occurred as a result of the combination of neighbouring bubbles on the heated surface which formed large dry vapour stems and spots on the heated surface. Bubbles being able to approach each other are easier to merge and form large dry spots on the heated surface. These dry spots exist momentarily if they are rewetted immediately by the impinging droplets or by the liquid film flow. Otherwise, partial “dryout” forms. As shown in Fig. 6.10, more bubbles are growing to larger sizes in the centre and outside regions than that

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at the region under a higher impinging droplet flux. The bigger neighbouring bubbles have shorter distance between each other so that the probability of mergence is enhanced and dry spots initially form on the centre and outside regions of the heated surface. The generated dry spots on the centre cannot be rewetted immediately by the impinging droplets due to the lower droplet impingement frequency. Besides, they also cannot be rewetted by the liquid film flow since the formed liquid film naturally flows from the centre to outside of the heated surface. Therefore, partial “dryout” incipiently occurs on the centre of the heated surface in the experiments.

Furthermore, to investigate the bubble characteristics, the bubble puncturing fluxes (N_p) and the punctured bubble diameter fractions at different heat fluxes are illustrated in Fig. 6.11. It is demonstrated that the bubble puncturing flux suddenly increases from 26642 to 39760 when the q'' increases from 80 to 100 W/cm², but increases progressively when q'' increases from 100 to 150 W/cm². This is because at $q'' = 80$ W/cm², part of the surface is still not superheated and hence, no bubbles will form on this area and no bubbles will be punctured. However, when $q'' \geq 100$ W/cm², the whole surface becomes superheated, and all the impinging droplets have the possibility to puncture the bubbles growing on the heated surface. When q'' increases gradually from 100 to 150 W/cm², the increased surface temperature reduces the life time of the bubbles, causing more bubbles to be generated in a period of Δt_H . Hence, the impinging droplets have a higher possibility to puncture the bubbles in the liquid film when the heat flux increases.

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Fig. 6.11 Distributions of the punctured bubble diameters at different heat fluxes

The punctured bubble diameter fractions in Fig. 6.11 show another interesting phenomenon that the fractions of the bubbles being punctured at bigger diameters increases progressively with the increase of heat flux. This suggests that bubbles can grow to bigger volumes before they are punctured by the impinging droplets at a higher heat flux.

6.4.3 Parametric study of droplet diameter

By fixing the applied heat flux ($q'' = 120 \text{ W/cm}^2$) and droplet impinging velocity, the effects of droplet diameter on spray cooling performance were investigated. Four droplet diameters with their corresponding droplet impinging fluxes to maintain the same flow rate were studied. The surface temperature contours for these four studied cases are presented in Fig. 6.12. As the same spatial droplet flux distribution is applied in the simulations (shown in Fig. 6.6(c)), the trends of the surface temperature distributions are similar for all the studied cases.

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Fig. 6.12 Effect of impinging droplet diameter at $q'' = 120 \text{ W/cm}^2$. (a) $D_{32} = 87 \text{ }\mu\text{m}$; (b) $D_{32} = 107 \text{ }\mu\text{m}$; (c) $D_{32} = 136 \text{ }\mu\text{m}$; (d) $D_{32} = 182 \text{ }\mu\text{m}$

It is noted that as the droplet diameter increases from $D_{32} = 87 \text{ }\mu\text{m}$ to $D_{32} = 182 \text{ }\mu\text{m}$, the surface temperature non-uniformity defined as $T_{surf,max} - T_{surf,min}$ increases from 4.5 to 9.0°C. By varying the impinging droplet diameter, the surface temperature varies more significantly in the region under a higher droplet flux impingement, but varies less in the regions under a lower droplet flux impingement. This phenomenon is attributed to the different heat transfer contributions in the spray cooling process. In the simulation model, the droplet impingement heat transfer rate is largely dependent on the impact energy (characterised by $We = \rho v^2 D / \sigma$). With the same droplet impinging velocity, increasing the droplet diameter increases the droplet

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impact energy, which results in a higher heat transfer rate by the droplet impingement. In the region under a higher impinging droplet flux, the heat transfer is more influenced by the droplet impingement cooling, so that the local surface temperature is more dependent on the droplet diameters. In the region under a lower impinging droplet flux, the heat transfer is mainly contributed by the bubble boiling whereby the local surface temperature is less affected by the droplet diameters.

Table 6.2 shows the summarised bubble characteristics and bubble boiling heat transfer ratio (n_{nb}) for the droplet diameters being investigated. N_d is the impinging droplet flux (number of droplets impinging on the heated surface per Δt_H) and N_p is the bubble puncturing flux (number of bubbles punctured by the droplets per Δt_H). As the droplet diameter decreases from $D_{32} = 182 \mu\text{m}$ to $D_{32} = 87 \mu\text{m}$, the bubble puncturing rate increases from 43740 to 191976 per Δt_H , which indicates an increment ratio of 4.39 ($n_p = N_{p@D_{32}=87\mu\text{m}} / N_{p@D_{32}=182\mu\text{m}}$). This is because the smaller droplets having a higher droplet impingement frequency, which naturally increases the possibility of the bubbles being punctured. However, it is found that the increment ratio of the impinging droplet flux ($n_d = N_{d@D_{32}=87\mu\text{m}} / N_{d@D_{32}=182\mu\text{m}}$) cannot achieve the same value as the increment ratio of the bubble puncturing flux (n_p), but is rather consistently smaller ($n_p < n_d$). This is because a smaller droplet can only puncture a bubble after growing to a sufficiently larger size. Thus, the probability of bubbles puncturing per droplet is lower by the smaller droplets than by the bigger droplets. The mean punctured bubble diameters ($\bar{D}_p$) indicated in Table 6.2 further supports this argument. The mean punctured bubble diameter is

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146.3 μm when $D_{32} = 182 \mu\text{m}$ for the impinging droplets, and 172.9 μm when $D_{32} = 87 \mu\text{m}$.

Table 6.2 also shows that the bubble boiling heat transfer rate increases from 0.59 to 0.67 when the impinging droplet diameter decreases from $D_{32} = 182 \mu\text{m}$ to $D_{32} = 87 \mu\text{m}$. As the mean surface temperature does not vary significantly with the impinging droplet diameter, the increased bubble boiling heat transfer rate for the small droplets impingement is attributed to the increased secondary nuclei from the higher impinging droplet flux. On the other hand, the bubbles can grow to the bigger sizes before being punctured by the smaller impinging droplets, which enhances the bubble boiling as well. Figure 6.13 presents the distributions of the punctured bubble diameter fractions. It is clear that the fractions of the punctured bubbles with $D_p > 160 \mu\text{m}$ is much larger at $D_{32} = 87 \mu\text{m}$ compared to that at $D_{32} = 182 \mu\text{m}$.

Table 6.2 Effects of droplet diameter on the bubble characteristics and bubble boiling heat transfer

D_{32} (μm)	N_d ($/\Delta t_H$)	N_p ($/\Delta t_H$)	$\bar{D}_p$ (μm)	n_d	n_p	n_{nb}
87	38.4×10^5	191976	172.9	6.0	4.39	0.67
107	25.6×10^5	142819	162.1	4.0	3.26	0.65
136	12.8×10^5	80516	153.6	2.0	1.84	0.61
182	6.4×10^5	43740	146.3	1.0	1.0	0.59

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Fig. 6.13 Distributions of the punctured bubble diameters at different impinging droplet diameters

6.5 Summary

In this chapter, a spray cooling model simulating the dynamics of spray characteristics, droplet impingement, and bubble boiling is presented. The spray characteristics obtained in Chapter 4 are used to validate the spray characteristics submodel, and boundary conditions in the heat transfer submodel. The spray characteristics submodel is based on a Monte Carlo simulation to match the droplet diameter, velocity and spatial droplet flux distributions under actual spray conditions. The heat transfer submodel considers the spray cooling process as film flow boiling, which combines the processes of droplet impingement induced forced convection and liquid film bubble boiling. Parametric effects on the heat transfer performance and bubble dynamics are investigated. Reasonable agreement is obtained between the experiments and model predictions.

Chapter 7

Prototype of a high power closed loop spray cooling system

7.1 Introduction

In chapters 3 and 4, the atomisation mechanisms of pressure swirl nozzles were studied, and the thermal effects on the formation of spray cone were characterised. The results obtained from chapters 3 and 4 are important for designing spray cooling systems which use pressure swirl nozzles to spray cool electronics devices requiring low and uniform operating temperature. This chapter proposed a prototype of a high power closed loop spray cooling system which used the pressure swirl nozzle characterised in chapters 3 and 4 to simulate the spray cooling over a 6U electronic card (23.3 cm × 16.0 cm). An array of multiple nozzles were used to cover the 6U card surface area. Parametric effects on the surface temperature, surface temperature non-uniformity, and CHF were investigated. The experimental results show promising prospects of using multi-nozzle arrays on large area power electronics cooling.

7.2 Design of a closed loop high power spray cooling system

The proposed high power closed loop system has been developed based on a “Dorin” condensing unit (Cooling capacity is ranging from 10.1 to 43.8 kW by varying the compressor speed from 20 to 75 Hz) using R134a as the working refrigerant. Figure 7.1 shows the schematic of the overall experimental setup, which consists mainly of a condensing unit (including a variable speed compressor and an air-cooled condenser), a liquid-vapour separator, a closed loop spray chamber, a heat exchange

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accumulator, and other necessities. To ensure that the system is in proper working conditions, several sight-glasses were installed to track the nature of flow in the closed loop.

Fig. 7.1 Schematic of the high power spray cooling system

The working principle of the closed loop system is as follows. The single-phase liquid from the separator is supplied to the spray nozzles, which atomise the liquid into fine and high velocity droplets in the spray chamber. Most of the droplets are vaporised after impinging on the heated surface, and the unevaporated liquid flows directly back to the heat-exchange accumulator. There, the liquid drained from the spray chamber continues to evaporate due to the integrated heat exchange coils. Hence, the entire refrigerant is transported back to the compressor in the form of low pressure vapour. After undergoing compression in the compressor, the low pressure vapour becomes the high temperature and high pressure vapour. Following which,

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the compressed vapour is converted into liquid refrigerant after being cooled by the air-cooled condenser. The converted liquid has a relatively higher temperature and is thereafter channelled to the heat exchange accumulator to assist the liquid evaporation in the accumulator. Eventually, the converted liquid is further cooled down before going into the separator for the next cooling cycle. In this system, the primary function of the heat-exchange accumulator is to prevent the excess liquid from flooding in the accumulator and being carried over to the compressor directly. The schematic of the heat-exchange accumulator is depicted in Fig. 7.2.

Fig. 7.2 Schematic of the heat exchange accumulator

7.3 Design of the closed loop spray chamber

The closed loop spray chamber aims to establish proper conditions for the multi-nozzle array spray cooling to take place over a large surface area. As shown in Fig. 7.3, the closed loop spray chamber is assembled vertically with a liquid feeding chamber, a nozzle plate, a spray chamber, a heater block and Teflon insulators. Hence, the liquid on the heated surface is drained by gravity.

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Fig. 7.3 Schematic of the closed loop spray chamber

The heater block is a rectangular copper block with overall dimensions of 25 cm $\times$ 18 cm $\times$ 8 cm. After being assembled with the spray chamber, the exposed surface area (cooling area) of the heater block in the spray chamber is 23.3 cm $\times$ 16.0 cm which is the exact size of a 6U electronic card. Twelve cartridge heaters are inserted into the copper block in two planes from two sides, as shown in Fig. 7.3(a). Each cartridge heater is in cylindrical shape with 1.91 cm in diameter and 15.2 cm in length, and is capable of delivering 2.4 kW of heat. These cartridges are connected

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in parallel and the supplied power is regulated by a 3-phase variac transformer. Before fabricating the heater block, a 3D steady-state thermal simulation in ANSYS was performed to decide the locations of cartridges in the copper block. The simulated temperature contours are shown in Fig. 7.3(a) ($q = 16.0$ kW in the simulation). This simulation aims to ensure that: (1) under the designed heat loads, the maximum temperature in the heater block will not exceed the working temperature of the Teflon insulators, and that (2) isotherms below the heated surface are one-dimensional so that an easy determination of T_{surf} is fabricate3d. The thickness of the Teflon insulator was determined by the same steady-state thermal simulation that the copper block and insulators are coupled and the boundary condition at the outside walls of the insulators is set as a natural convective heat transfer coefficient ($h = 25$ W/m²·K (Rathore and Kapuno, 2011)). By assuming the environment temperature of 26°C, the predicted heat loss under the 2-cm-thick Teflon insulators is less than 1% when the heat load is varying from 2 to 20 kW, as shown in the Appendix.

As shown in Fig. 7.3(b), there are 37 temperature sensing points (red spots) designed in the heater block 0.4 cm below the heated surface to estimate the local surface temperatures. The applied thermocouples are K-type sheathed thermocouples with the outer diameter of 0.1 cm.

The spray chamber with the internal volume of 23.3 cm × 16.0 cm × 3.73 cm was attached to the copper surface using an O-ring as the sealing. Two outlets were fabricated at the top and bottom walls of the spray chamber to drain the vapour and liquid, respectively. In order to prevent the buildup of liquid and vapour at the corners of the spray chamber (which are farther away from the outlets), two long

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slots ($18.5 \text{ cm} \times 2.3 \text{ cm} \times 1.2 \text{ cm}$) were fabricated at the top and bottom walls to serve as reservoir buffers to drain the liquid and vapour on the heated surface more uniformly (instead of converging all the liquid and vapour to the relatively small orifices of the outlets). The nozzle plate, which comprises 54 full cone pressure swirl nozzles (Nozzle A in Chapter 3) with an array of 9×6 , is installed between the feeding and spray chambers for liquid atomisation and uniform liquid distribution on the heated surface. Meanwhile, the nozzle plate ensures that most of the heated surface will be directly impacted by the spray droplets. As shown in Fig. 7.3(b), these spray nozzles are spaced out such that the spray overlapping between the spray nozzles is prevented. In such an arrangement, 76.8% of the heated surface is allowed to be impinged by the spray cones directly. The feeding chamber attached to the nozzle plate acts as a liquid reservoir which supplies liquid refrigerant uniformly to all the spray nozzles.

7.4 Experimental procedure

During the experiment, the refrigeration system was first activated before the cartridge heaters were switched on. When fully-filled liquid was observed from the sight-glass installed upstream of the closed loop spray chamber, the supply of heat begins. To test the spray cooling performance under different heat loads, the power input to the heater block was gradually increased until the CHF was reached. At each power level, the readings of all sensors (temperature and pressure) were monitored using two “Agilent” data acquisition systems (Model: 34972A) which were incorporated to the PCs. When a steady state case was reached, the experimental data for the power level were saved and a new power input was regulated to attain a

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new steady state. The data recorded in each steady state were averaged for data analysis.

The experimental liquid flow rate, nozzle inlet pressure, and chamber pressure were adjusted by regulating the compressor frequency and the manual valves installed in the system. In particular, the liquid flow rate was measured by a “Brooks” rotameter (Model: GT1000), rated from 28-260 mL/s, with an accuracy of 2%. The nozzle inlet pressure and the spray chamber pressure were measured using two STS absolute pressure transducers (Model: 232-XX-13-01-47-0-1-U), rated from 0-10 bar, with accuracy of 0.5%.

7.5 Data reduction and uncertainties

According to the ANSYS simulation, the heat loss to environment is negligible.

Thus, the heat load applied to the heated surface is calculated by

$$q = V \cdot I \quad (7-1)$$

where V is the voltage and I is the current in the electric circuit of cartridge heaters.

Therefore, the heat flux applied on the heated surface is calculated as

$$q'' = \frac{q}{A} \quad (7-2)$$

where A indicates the heated surface area. According to the linear temperature profile below the heated surface, the local surface temperatures can be determined as

$$T_{surf,i} = T_{thermo,i} - \frac{q''}{k} \Delta x \quad (7-3)$$

where $T_{thermo,i}$ is the local temperature reading of the i th thermocouple, k is the thermal conductivity of the heater block, and Δx is the distance from the

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thermocouple plane to the heated surface. The mean surface temperature is thereby calculated as the arithmetic averaged value of the local surface temperatures,

$$\bar{T}_{surf} = \frac{1}{N} \sum_{i=1}^N T_{surf,i} \quad (7-4)$$

where N is the number of local surface temperatures. The average heat transfer coefficient $\bar{h}$ is thereby calculated as

$$\bar{h} = \frac{q''}{\bar{T}_{surf} - T_{sat}} \quad (7-5)$$

where T_{sat} is the saturation temperature corresponding to the chamber pressure.

The surface temperature non-uniformity is defined as the difference of the maximum local surface temperature minus the minimum local surface temperature on the heated surface,

$$\Delta T_{non-uniformity} = T_{surf_max} - T_{surf_min} \quad (7-6)$$

In order to study the liquid usage efficiency in the experiments, evaporation fraction (ε) which represents the proportion of liquid evaporated in the spray and heat transfer process with respect to the total liquid flow rate is evaluated. Considering the spray chamber as a control volume, the evaporation fraction is defined according to the energy and mass conservations in the spray chamber,

$$m_{i,l} h_{i,l} + q = m_{o,l} h_{o,l} + m_{o,v} h_{o,v} \quad (7-7)$$

$$m_{i,l} = m_{o,l} + m_{o,v} \quad (7-8)$$

$$m_{o,v} = m_{i,l} \cdot \varepsilon, \quad m_{o,l} = m_{i,l} \cdot (1 - \varepsilon) \quad (7-9)$$

$$\varepsilon = \frac{m_{i,l} (h_{i,l} - h_{o,l})}{m_{i,l} (h_{o,v} - h_{o,l})} + \frac{q}{m_{i,l} (h_{o,v} - h_{o,l})} \quad (7-10)$$

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where, $m_{i,l}$ and $h_{i,l}$ are the inlet liquid mass and enthalpy, respectively, $m_{o,l}$ and $h_{o,l}$ are the outlet liquid mass and enthalpy, respectively, $m_{o,v}$ and $h_{o,v}$ are the outlet vapour mass and enthalpy, respectively. In Eq. (7-10), the first term on the right hand side is the evaporation rate due to the nozzle expansion, and the second term is the evaporation rate attributed by the applied heat load.

The uncertainty of the electrical power through the power analyzer (Carlo Gavazzi, Type WM14-96) is $\pm 0.5\%$ for voltage and current measurements. The accuracy of the calibrated thermocouples is a function of the thermo-calibrator, which has the accuracy within $\pm 0.1^\circ\text{C}$ in the experimental temperature range. The heat loss to environment is less than 1.0% based on the ANSYS steady state thermal simulation. According to the uncertainty analysis in Chapter 3, the uncertainty of the heat transfer coefficient is less than $\pm 3.0\%$.

7.6 Results and discussion

7.6.1 Open loop visualisation

An open loop visualisation experiment was set up to explore the impingement pattern of the multi-nozzle array spray cones impinging on a flat surface. An acrylic sheet with a thickness of 0.5 cm was fabricated to replace the heater block so that the footprints of the multi-nozzle array spray cones could be observed on the other side of the acrylic sheet. As shown in Fig. 7.4(a), the inside and outside views of the multi-nozzle array spray cones are presented. It is observed that the footprints of the 54 spray cones are approximately the same, which suggests a uniform liquid feeding to all the spray nozzles. For each spray cone, the footprint shape is not circular but rather like a square. This is due to the spray-to-spray interactions between the

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neighbouring spray cones, as well as the gravitational effect. Note also that, between the neighbouring rows and columns of the spray cones, there are horizontal and vertical liquid bridges formed to assist the liquid drainage. Figure 7.4(b) schematically depicts the flow pattern of a segment of the liquid flow on the impinging surface. From the perspective of the spray-to-spray interactions, the horizontal and vertical liquid bridges are formed due to the collision of the surrounding spray cones and gravity. With the aid of gravity and the specified liquid drainage design, no liquid accumulation has been observed in the spray chamber.

Fig. 7.4 Visualisation of multi-nozzle array spray cones impinging on a flat surface: $\Delta p = 3$ bar

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7.6.2 Effects of nozzle pressure drop

By maintaining the chamber pressure, the mean surface temperature and average heat transfer coefficient as a function of nozzle pressure drop are illustrated in Figs. 7.5 and 7.6, respectively. Two different heat loads (q) were tested at their respective chamber pressures. The nozzle pressure drop was varied from 2.4 to 4.2 bar, resulting in the flow rate ranging from 78 to 99 mL/s, respectively.

Fig. 7.5 Mean surface temperature and flow rate as a function of nozzle pressure drop

Fig. 7.6 Heat transfer coefficient and flow rate as a function of nozzle pressure drop

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At $q = 8.5$ kW, it is observed that the mean surface temperature decreases from 19.7 to 16.3°C as the nozzle pressure drop increases from 2.4 to 3.7 bar (see Fig. 7.5), and correspondingly the heat transfer coefficient increases from 1.61 to 2.11 W/cm²·K (see Fig. 7.6). At $q = 12.6$ kW, the same observation is found. As the nozzle pressure drop is increased from 3.7 to 4.2 bar, the mean surface temperature is reduced from 24.7 to 22.2°C with the corresponding heat transfer coefficient increasing from 2.38 to 2.81 W/cm²·K. This demonstrates the positive effects of nozzle pressure drop on the heat transfer effectiveness of spray cooling.

The enhanced heat transfer performance by increasing nozzle pressure drop is due to the improved spray characteristics. Increasing nozzle pressure drop consequently increases the flow rate, droplet flux and outfits the droplets with higher impact energy (Xie *et al.*, 2013). First, increasing flow rate had been reported to improve the spray cooling performance by intensifying the convective flow and sufficiently wetting the heated surface (Chow, 1997; Hsieh *et al.*, 2004; Yan *et al.*, 2013). Second, an increase in droplet flux caused more secondary nuclei which enhance the heat transfer in the boiling regime of spray cooling (Rini *et al.*, 2002). Last, the higher impact energy of droplets would assist droplets to impact the heated surface, which at first enhances the heat transfer effectiveness of droplet impingement cooling (Issa and Yao, 2005). Besides, the droplets with higher impact energy strongly agitate the thermal boundary layer of the formed liquid film on the heated surface, which enhances the heat transfer performance (Shedd, 2007).

7.6.3 Effects of chamber pressure

By regulating the control valves at the outlets of spray chamber, the chamber pressure can be adjusted over a wide range. Meanwhile, the flow rate can be

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maintained in a narrow range by controlling the expansion valve and compressor running speed in the closed loop. Figures 7.7 and 7.8 present the effects of chamber pressure on the mean surface temperature and heat transfer coefficient, respectively. Four different heat loads were studied by maintaining the flow rate ranging from 94 to 98 mL/s.

Fig. 7.7 Effect of chamber pressure on the mean surface temperature

Fig. 7.8 Effect of chamber pressure on the heat transfer coefficient

Figure 7.7 shows that the mean surface temperatures increase almost linearly with increasing chamber pressure. Meanwhile, the chamber saturation temperature

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increases linearly with the chamber pressure at the similar slope of the surface temperature curve. This implies that the chamber pressure affects mainly the mean surface temperature by varying the saturation temperature inside the spray chamber. By regulating the chamber pressure, the same surface temperature can be achieved even if the applied heat load varies in a wide range. For instance, as shown in Fig. 7.7, the same mean surface temperature of $\bar{T}_{surf} = 22^\circ\text{C}$ is obtained when the heat load increases from 10.3 to 14.2 kW (with 37.9% heat load increment). The result reveals that controlling the chamber pressure is an effective way to maintain the same working temperature of an electronic device when it works under different heat loads.

Figure 7.8 demonstrates that increasing chamber pressure and thus the saturation temperature of the refrigerant, is appreciably conducive to the heat transfer performance in spray cooling. An increase in chamber pressure results in an increase of heat transfer coefficient, which agrees with the findings by Lin *et al.* (2004) and Yan *et al.* (2010) that the heat transfer at a higher saturation temperature is more effective than that at a lower saturation temperature. The improved heat transfer performance is probably due to the increased active nucleation site density as a result of increasing chamber pressure. Rainey *et al.* (2003) found that increasing chamber pressure resulted in a wider range of cavity radius that can be activated at a given wall superheat in pool boiling.

7.6.4 Critical heat flux (CHF)

A series of CHF experiments were conducted by slowly increasing the heat load until a sudden temperature jump (related to CHF) was observed. As shown in Fig. 7.9, the typical transition process from the last steady state point to the incidence of

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CHF is illustrated. Three distinct regimes can be divided in the transition process: the steady state regime, transition regime, and film boiling regime. In the steady state regime where $q = 16.1 \text{ kW}$, the thermocouple readings are stable and constant. As the heat load increases slightly to 16.5 kW , the thermocouple readings shoot up rapidly from 30 to 75°C in 10 minutes which is so-called the transition regime. Thereafter, a sudden temperature jumps occurs to reach the film boiling regime, and at that instant, heater is powered off immediately. The heater surface remains in film boiling until the stored heat in the heater block is not enough to sustain a high surface temperature for film boiling. As film boiling disappears, the normal spray cooling scenario returns and the surface temperature decreases rapidly.

Fig. 7.9 Transition process from steady state to CHF

Figure 7.10 shows the variations of mean surface temperature and evaporation fraction with increasing heat load during the CHF test. As expected, the mean surface temperature as well as evaporation fraction monotonically increases with increasing heat load. When $q < 15 \text{ kW}$, the mean surface temperature increases gently with the increase of heat load, thereafter increases more rapidly until CHF

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occurs. As can be seen from Fig. 7.10, although the mean surface temperature obtained from the last steady state point is very low ($\bar{T}_{surf} = 26.5^{\circ}\text{C}$ at $q = 16.1\text{ kW}$), CHF occurs when the heat load is slightly increased to 16.5 kW (see Fig. 7.9). This is attributed to the insufficient liquid supply on the heated surface. For example, assuming the steady state working conditions at $q = 16.5\text{ kW}$ are similar to that at $q = 16.1\text{ kW}$, the estimated liquid evaporation fraction at $q = 16.5\text{ kW}$ reaches 0.91 . Such evaporation fraction is too high to maintain a steady state heat transfer situation thereby leads to the incidence of CHF.

Fig. 7.10 Mean surface temperature as a function of heat load during CHF test

Figure 7.11 demonstrates the corresponding heat transfer coefficient and heat flux with increasing heat load. It reveals that when $q < 15\text{ kW}$ the heat transfer coefficient increases with increasing heat load and subsequently decreases when $q > 15\text{ kW}$. This agrees with the results obtained by Bostanci *et al.* (2012) which reported that the heat transfer coefficient decreased before the incidence of CHF. When $q > 15\text{ kW}$, the evaporation fraction is too high ($\epsilon > 0.85$), which makes a continuous liquid film spreading over the heated surface to be questionable. On the contrary, it is more

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likely that at some locations on the heated surface, the sprayed liquid is completely vaporised to create some local “dryout”. These “dryout” locations resulted in some hot spots on the heated surface, and hence caused a rapid increase in the mean surface temperature and a decrease in the heat transfer coefficient before CHF occurred.

Fig. 7.11 Heat transfer coefficient, heat flux as a function of heat load during CHF test

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Table 7.1 lists the CHF tests under different flow rates. It can be seen that for all the CHF tests, the evaporation fraction obtained at the last steady state point is almost at the same level ($\varepsilon = 0.88$). When the applied heat load increases slightly, CHF occurs. This implies that the evaporation fraction of 0.88 could be the critical value to separate the heat transfer situation from steady state operation to the incidence of CHF. Therefore, it is believed that increasing flow rate is able to delay the incidence of CHF by maintaining the evaporation efficiency below the critical value.

Table 7.1 Evaporation fractions during the CHF tests

CHF tests	Case 1	Case 2	Case 3	Case 4	Case 5	Case 6	error
Flow rate (mL/s)	88	90	91	93	94	97	$\pm 2.0\%$
Last steady state load q (kW)	15.5	15.6	15.8	15.7	16.1	16.0	$\pm 0.5\%$
Last steady state temperature $\bar{T}_{surf}$ ($^{\circ}\text{C}$)	25.3	25.1	25.3	25.6	26.5	25.6	± 0.2
heat load q at CHF (kW)	15.8	16.0	16.1	16.1	16.5	16.7	$\pm 0.5\%$
Evaporation fraction (ε) at last steady state point	0.89	0.88	0.88	0.88	0.89	0.88	± 0.018

7.6.5 Surface temperature non-uniformity

Another important criterion to evaluate the reliability of using multi-nozzle array spray cooling technique on large area electronics cooling is the surface temperature non-uniformity. As shown in Fig. 7.12, the surface temperature non-uniformity as a function of heat load is demonstrated. It indicates that the surface temperature non-uniformity is sensitive to the applied heat load, especially when the flow rate is lower. This is due to the higher liquid consumption at a higher heat load. A higher evaporation fraction is easier to cause local “dryout” on the heated surface, which consequently results in some hot spots with high local surface temperatures. This

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phenomenon is more evident when the applied heat load is near to the incidence of CHF as shown in Fig. 7.12. When the flow rate ranges from 91 to 93 mL/s, the surface temperature non-uniformity suddenly increases from 5.9 to 9.1°C as the heat load increases from 14.2 to 15.7 kW ($0.79 \leq \varepsilon \leq 0.88$). The same phenomenon is also observed at the flow rate from 94 to 98 mL/s when the heat load increases from 14.4 to 16 kW ($0.80 \leq \varepsilon \leq 0.89$). The other possible reason that accounts for a larger surface temperature non-uniformity at a higher heat load is the effect of uprising vapour generated from the heated surface. As found in Chapter 4, the vapour uprising from the heated surface influences the spray cone formation, which consequently changes the liquid distribution on the heated surface and depraved the surface temperature uniformity. Figure 7.12 also manifests that increasing liquid flow rate is beneficial to maintain a uniform surface temperature (e.g., $97 < \text{flow rate} < 100 \text{ mL/s}$). This is because a higher flow rate creates a more stable liquid film on the heated surface, hence a more uniform surface temperature distribution on the heated surface.

Fig. 7.12 Effects of heat load and flow rate on the temperature non-uniformity

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Although the chamber pressure has a distinguished influence on the mean surface temperature, Figure 7.13 shows no detectable effects of chamber pressure on the surface temperature non-uniformity. This is reasonable since chamber pressure only influences the mean surface temperature by changing the saturation temperature inside the spray chamber.

Fig. 7.13 Effect of chamber pressure on the surface temperature non-uniformity

Furthermore, to characterise the surface temperature distribution in detail, the heated surface has been divided into five regions (from A to E), as shown in Fig. 7.14.

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Fig. 7.14 Temperature regions divided on a 6U card area

The mean surface temperatures of these five regions are illustrated in Fig. 7.15. It is noted that region B has the highest surface temperature, followed by regions A and C, while regions D and E have the lowest surface temperature. Region B being at the highest surface temperature could be due to the fact that it is situated at the place directly facing to the liquid and vapour outlets at the bottom and top of the spray chamber. As a result, some of the sprayed liquid at this region could drift to the outlets before impacting on the heated surface. It is suspected that the sprayed liquid directed to region B may not be fully utilised before being drained away to regions D and E (near to the outlets). Therefore, region B has the highest surface temperature while regions D and E receiving more refrigerant have the lowest surface temperature. In addition, the lower surface temperature obtained at regions D and E could be due to their relatively lower local chamber pressure as well. As shown in

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Fig. 7.14(a), regions D and E are closest to the outlets hence experience a lowest local chamber pressure as compared to regions A, B and C (noted that vapour generated in the spray chamber flows from regions A, B and C to regions D and E). A lower chamber pressure results in a lower saturation temperature which causes a lower mean surface temperature.

Fig. 7.15 Mean surface temperature at different regions on the 6U card area

To characterise the effects of spray-to-spray interactions on local surface temperatures, two sub regions (see regions I and II in Fig. 7.14) in regions D and E are investigated. The proposed liquid flow pattern in these sub regions is depicted in Fig. 7.4(b). Regions I and II are identical to each other, except region I is near to the vapour outlet (top) and region II is near to the liquid and vapour outlet (bottom). Six thermocouples were used to measure the local surface temperature in each region. Particularly, the surface temperatures of the interested locations were measured as shown in Figs. 7.14(b) and (c), such as the stagnation points of spray cones (I-1, I-2, II-1, II-2), the converging points of four spray cones (I-3, I-4, I-6, II-3, II-4, II-6), and the converging points of two spray cones (I-5, II-5). The local surface

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temperatures under different heat loads for regions I and II are illustrated in Figs. 7.16 and 7.17, respectively. It is noted that at $q \leq 11.8$ kW, the local surface temperatures at I-1 to I-4 have no distinct discrepancies between each other in region I, as well as II-1 to II-4 in region II. While, the surface temperatures at I-5 to I-6 are higher for all the heat loads in region I, as well as that at II-5 to II-6 in region II. The results imply that rather than the spray-to-spray interactions, the distance of a location relative to the outlets has more significant effect on the local surface temperature. For instance, at $q = 16.1$ kW, the local surface temperature is highest at I-6 and II-6 which are the farthest points to the outlets, followed by I-5 and II-5 (second farthest), then I-4 and II-4 (third farthest) as shown in Figs. 7.16 and 7.17.

Fig. 7.16 Local surface temperatures at different measuring points: region I

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Fig. 7.17 Local surface temperatures at different measuring points: region II

7.7 Design considerations of a spray cooling system

The present study shows that designing an effective closed loop spray cooling system must include all aspects of the total system, not just the spraying equipment. Various items need to be considered, and potential problems need to be addressed before a successful spray cooling design can be achieved. Some important design-considerations raised in designing the present closed loop system are listed below:

1. **Pressurised system or Non-Pressurised system.** In system level, it is very important to decide whether the system is pressurised or non-pressurised. Pressurised system has a few advantages, although can lead to high cost and complexity. By maintaining a positive pressure inside the system, external contaminants tend to be kept out from the system, and a more accurately control of the component operating temperature can be achieved (evaporation temperature affects the component operating temperature). On the contrary, Non-Pressurised system allows air or gas to infiltrate, which makes the precise system operating temperature control be impossible.

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2. **Nozzle selection.** Different types of nozzles have different spray patterns which affect the thermal performance of spray cooling. To suit a special application, a good nozzle is of great importance to achieve the required thermal performance (i.e., heat flux, surface temperature, and surface temperature uniformity). The selected nozzle should provide sufficient flow rate, good liquid atomization quality at the designed pressure difference, and suitable coverage area on the component surface. Moreover, the working principles of nozzles heavily affects the complexity of the spray cooling system. For package-volume-confined system, the applied nozzles would be considered custom or proprietary.
3. **Power equipment selection.** The power equipment is chosen according to the fluid and the nozzle used in the designed system. For example, FC fluids can be circulated using pumps, while refrigerant coolants prefer compressors. The overall liquid control in the closed loop system is heavily dependent on the combination of the power equipment and spray nozzle. Therefore, the power equipment should supply adequate pressure and fluid flow for the spray nozzles to generate good atomization and spray pattern for spray cooling, while also keeping electrical power use and noise to a minimum.
4. **Orientation of heat source.** In a gravity-dependent system, the orientation of heat source is better to be placed vertically (parallel to gravity), hence gravity can help draw fluid down into a reservoir for collection and redistribution and the liquid drainage system becomes simpler. In a gravity-independent system, the heat source can be positioned in any orientation. However, a proper liquid drainage system is complicated.

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5. **Fluid selection.** The selection of the cooling fluid should consider the following issues: safety, material compatibility, operational parameters, operating life, environment release, heat transfer requirement, surface temperature requirement, heat flux requirement, and reliability.

7.8 Summary

A prototype of a high power, multi-nozzle array spray cooling system was built and tested. Experimental results show a promising prospect of using multi-nozzle arrays on large area power electronics cooling. Increasing nozzle pressure drop or flow rate was found to enhance the heat transfer coefficient and give better surface temperature uniformity, while, increasing heat load worsens surface temperature uniformity. The results show that chamber pressure significantly influences the mean surface temperature by managing the saturation temperature in the spray chamber. Hence, controlling the chamber pressure can keep a device working at the same temperature when the applied heat load varies largely. In the present study, the spray-to-spray interactions show insignificant effects on the local surface temperatures, but rather the distance from a location relative to the drainage outlets.

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Chapter 8

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8.1 Introduction

Although efforts have been made to enhance spray cooling by modifying surface properties, it is unclear on the actual mechanisms associated with the enhanced heat transfer performance. This chapter investigated the heat transfer performance of differently scaled structure surfaces, such as micro-, macro-, and multiscale-structured surfaces. In addition, experiments were conducted to track the transient process from the steady state to the incidence and thereafter of CHF, which shed some light on the behaviours of the impinging droplets and growing bubbles during the CHF and film boiling in spray cooling.

8.2 Experimental apparatus

8.2.1 Experimental setup

Figure 8.1 illustrates the schematic of the closed loop in this experiment. Its working principle is similar to that of the high power spray cooling system in Chapter 7. However, the entire system is scaled down due to the limited heat load on a small heated surface in the experiments. The main components in the closed loop include a Dorin open-type condensing unit (cooling capacity: 1 kW, variable speed compressor), a liquid-vapour separator, a closed loop spray chamber and an accumulator. R134a is the working refrigerant. During the experiments, the system allows the control of the mass flow rate, pressure drop across the nozzle plate and

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chamber pressure by regulating the compressor frequency, expansion valve (metering valve) and control valves 1 and 2 (needle valves).

Fig. 8.1 Schematic of the experimental setup

8.2.2 Heater design

Figure 8.2 depicts the closed loop spray chamber in the experiments. It mainly comprises the liquid atomisation module and heater module. The liquid atomisation module includes a liquid feeding chamber, a nozzle plate and a spray chamber. The features of the nozzle plate will be described in Section 8.2.3.

The heater module includes a heater block and insulators. The heater block includes two parts: a heater base and an enhanced surface adaptor, which were made of copper blocks. The heater base is heated with 10 heater cartridges (10×120 W). The enhanced surface adaptor is the replaceable part. On the top surface of which, the designed surface structures were directly fabricated. With the aid of an ANSYS simulation, the height of the enhanced surface adaptor was determined to achieve

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one-dimensional heat conduction below the test surface, as shown in Fig. 8.2. Four K-type thermocouples with bead diameter of 0.5 mm were inserted into the adaptor in two planes to extrapolate the surface temperature and heat flux. To fix the enhanced surface adaptor on the heater base, Arctic Silver® 5 thermal grease with 99% silver particles was pasted in the interfaces between the heater base and enhanced surface adaptor to reduce the contact thermal resistance. The entire heater block was insulated with Teflon blocks to prevent heat loss.

Fig. 8.2 Schematic of the closed loop spray chamber

In operating the experiments, the closed loop spray chamber is vertically orientated. The excess liquid flows downward to the bottom of the spray chamber to be drained by gravity. The vapour generated on the heated surface is uprising and is discharged from the vapour outlet at the top of the spray chamber.

8.2.3 Nozzle plate

The nozzle plate employed in the setup was assembled with six in-house jet-swirl nozzles in an array of 3×2 , as shown in Fig. 8.3. The nozzle plate features a housing

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plate and jet-swirl inserts. The housing plate has been fabricated to integrate a swirl chamber and a discharge orifice for each jet-swirl insert to form a jet-swirl nozzle. The discharge orifice of the jet-swirl nozzle has a diameter of 0.3 mm. The jet-swirl insert was fabricated with three swirl grooves and one axial groove with the diameter of 0.3 mm. The spray performance of the jet-swirl nozzles was characterised using the PDA system as described in Chapter 3. Each jet-swirl nozzle performs a full cone spray with a spray cone angle around 35°. For a nozzle pressure drop from 3 to 5 bar, the atomised droplets have the droplet diameter (D_{32}) from 189 ($\Delta p = 3$ bar) to 172 μm ($\Delta p = 5$ bar) and droplet velocity from 18.1 to 24.6 m/s correspondingly. In the experiments, the nozzle-to-surface distance was fixed at 8.8 mm which made the spray cones just inscribe the heated surface ($2.0\text{ cm} \times 1.0\text{ cm}$).

Fig. 8.3 Assembly of nozzle plate in the experiment

8.2.4 Tested surfaces

Three categories of enhanced surfaces were investigated: micro-, macro- and multiscale-structured surfaces. Table 8.1 shows the detailed descriptions and surface characteristics of the tested surfaces. Area enhancement is defined as the ratio of the total surface area to the projected surface area. The schematics including the

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scanning electron microscope (SEM) images and solid models of macro- structures are shown in Fig. 8.4. The smooth flat surface as shown in Fig. 8.4(a) was well machine-finished to have a roughness of 0.5 μm . The micro-structured surface as shown in Fig. 8.4(b) was fabricated using wire-cut EDM technique to create a roughness of $R_a = 3.0 \mu\text{m}$ which was reported to significantly enhance the heat transfer in R134a pool boiling (Jabardo *et al.*, 2009). For the micro-structured surface, there are numerous re-entrant cavities (black spots in Fig. 8.4(b)) randomly distributed on the surface. The macro-structured surfaces are directly fabricated on the enhanced surface adaptor with fin or pin structures as shown in Fig. 8.4(c). The multiscale-structured surfaces combine the features of micro- and macro-structures surfaces by integrating the same micro- structures on macro- structures.

Table 8.1 Description of surfaces characteristics in the experiments

Heater ID	Area enhancement	Surface condition	Fabrication technique
Smooth-Flat	1.00	Smooth-, flat, ($R_a \approx 0.5 \mu\text{m}$)	Milling
Micro-Flat	1.00	Micro-structured, flat ($R_a \approx 3.00 \mu\text{m}$)	Wire-cut EDM
Ma-Straight.f-H	2.00	Macro- straight fin, horizontally placed ($R_a \approx 0.5 \mu\text{m}$)	Milling
Ma-Straight.f-V	2.00	Macro- straight fin, vertically placed ($R_a \approx 0.5 \mu\text{m}$)	Milling
Ma-Square.p	2.00	Macro- square pin ($R_a \approx 0.5 \mu\text{m}$)	Milling
MI-Straight.f-V	2.00	Multiscale- straight fin, vertically placed ($R_a \approx 3.00 \mu\text{m}$)	Wire-cut EDM
MI-Square.p	2.00	Multiscale- square pin ($R_a \approx 3.00 \mu\text{m}$)	Wire-cut EDM
MI-Triangle.f-V	1.75	Multiscale-triangular fin, vertically placed ($R_a \approx 3.00 \mu\text{m}$)	Wire-cut EDM

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Fig. 8.4 Schematics of the tested surfaces: (a) SEM image of the smooth-flat surface, (b) SEM image of the micro- structures, (c) solid model of the macro-structured surfaces

8.3 Experimental procedure, data reduction and uncertainties

To study the thermal effects of enhanced surfaces, a series of experiments were conducted at a fixed pressure drop of 4.8 ± 0.2 bar with the corresponding flow rate of 8.2 ± 0.2 g/s. The saturation temperatures in the spray chamber were maintained at $10 \pm 1.5^\circ\text{C}$. The input heat fluxes were adjusted by a variac transformer from an AC power source. All the sensors were connected to an “Agilent” data acquisition system (Model: 34972A) which was incorporated with a PC to record the

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experimental data. Experimental data recorded for each steady state case were averaged for data analysis.

For flat surfaces, the average heat transfer coefficient is calculated using,

$$\bar{h} = q'' / (T_{surf} - T_{sat}) \quad (8-1)$$

For macro-structured surfaces (except the MI-Triangular.f-V surface which has the fins with irregularly cross-sectional area), the average heat transfer coefficient is determined from an iterative solution of an expression for total heat transfer rate from the finned or pinned surface (Hansen and Webb, 1993) as follows:

$$q'' A_p = \bar{h} A_b (T_{surf} - T_{sat}) + \sum_{i=1}^n M_i \frac{\sinh(\chi_i H_{fin}) + (\bar{h} / \chi_i k_s) \cosh(\chi_i H_{fin})}{\cosh(\chi_i H_{fin}) + (\bar{h} / \chi_i k_s) \sinh(\chi_i H_{fin})} \quad (8-2)$$

where

$$M_i = [\bar{h} P_i k_s A_{c,i}]^{1/2} (T_{surf} - T_{sat}) \quad (8-3)$$

$$\chi_i = [\bar{h} P_i / k_s A_{c,i}]^{1/2} \quad (8-4)$$

In Eqs. (8-2)-(8-4), A_p is the projected area of the heated surface. A_b is the base surface area. n is the total number of fins or pins on the heated surface. H_{fin} is the height of a fin or pin. P is the perimeter and A_c is the cross-sectional area of a fin or pin. The heat transfer coefficient is then reduced to the Nusselt number (Nu) which is defined as,

$$Nu = \bar{h} \cdot d_{eq} / k_{liq} \quad (8-5)$$

where d_{eq} is the equivalent size of the heated surface which is defined as the ratio of the total surface area of the heated surface to its perimeter. k_{liq} is the thermal conductivity of the liquid refrigerant. Note that the average Nusselt number here is

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not an index of the total heat transfer rate from the fin/base surface but rather a measure of average heat transfer coefficient. Besides, a non-dimensional surface superheat is defined as

$$\mathcal{G} = (T_{surf} - T_{sat}) / T_{sat} \quad (8-6)$$

According to the uncertainty analysis, the uncertainties of the heat flux and heat transfer coefficient (Nu) are less than 5.8 % and 6.1% respectively, when the heat flux is larger than 50 W/cm².

8.4 Results and discussion

Due to the complex process involved in spray cooling, the mechanisms by which heat is removed are still not well understood even for smooth surfaces. When the liquid flow rate is too high, the sprayed liquid is swept away quickly before being heated up to the nucleate boiling regime. Researchers therefore suggest that the sensible heat removal is the primary heat transfer mechanisms in spray cooling (Shedd and Pautsch, 2005; Xie *et al.*, 2012). Another proposed mechanism is that the spray produces a thin liquid film on the surface in which both evaporation and nucleate boiling occur. Thinner liquid film and more nucleation sites are reported to result in better heat transfer (Pais *et al.*, 1992; Rini *et al.*, 2002). The previous research show that the addition of enhanced structures improves the single-phase heat transfer by increasing the total wetted surface area and retaining liquid for a longer time (Webb and Kim, 2005; Silk *et al.*, 2006). Moreover, the increased wetted area enhances the heat transfer performance by providing more available fixed nucleation sites in pool boiling (Webb and Kim, 2005; Jabardo *et al.*, 2009). In the present study, the phase change heat transfer occurs in all the cases as liquid is saturated after expansion through the spray nozzles. As shown in Fig. 8.5, the regime

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of most interest in this study is from *a* to *b* in the boiling curve (White, 1988), which is of practical interest in electronics cooling. The transient process which is just before and after the incidence of CHF is also investigated. In the following sections, all the presented heat flux data are calculated based on the projected area (2.0 cm × 1.0 cm).

Fig. 8.5 The experimental regime in the boiling curve (White, 1988)

8.4.1 Macro-structured surfaces

As shown in Fig. 8.4, all the macro-structured surfaces are characterised by the way the fins or pins are arranged on the heated surface. Different arrangements of fins cater to different liquid flow patterns on the heated surface which are expected to influence the heat transfer performance. For the Ma-Straight.f-H surface which has straight fins arranged in parallel with the horizontal plane, the fins form horizontal grooves to distribute liquid evenly to the regions where are less or no droplets directly impinged (e.g., corners of the surface). However, the liquid in the grooves is constricted by the fins to form a very thick liquid film in the grooves. Therefore, the droplets need to penetrate the thick liquid film before they can impinge on the

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heated surface. In contrast, the Ma-Straight.f-V surface has the straight fins perpendicular to the horizontal plane. The grooves formed between the fins are in parallel with gravity so that liquid flow is accelerated towards the outlet at the bottom of spray chamber. Hence, a thinner liquid film flow with a higher velocity is conceivable on the Ma-Straight.f-V. Nonetheless, the atomised droplets in the spray cones may be shaded by the fins from impinging or spreading to the regions having a shortage of liquid (e.g., corners of the surface). The Ma-Square.p surface combines the features of the Ma-Straight.f-H and Ma-Straight.f-V surfaces by fabricating regularly arranged square pins on the heated surface. Hence, the liquid on the heated surface is able to spread and flow. But the film flow velocity is suggested to be lower as compared to that on the Ma-Straight.f-V surface since there are two directional flow cross-sections for the liquid (Silk *et al.*, 2006).

Figure 8.6 illustrates the heat fluxes as a function of surface superheat (ΔT_{ss}) for the macro-structured surfaces.

Fig. 8.6 Heat flux as a function of surface superheat for the macro-structured surfaces

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The results clearly show that the Ma-Straight.f-V and Ma-Square.p surfaces achieve the better heat transfer performance as compared to the smooth flat surface. However, it is surprised that even though the wetted surface area is doubled, the Ma-Straight.f-H surface has equal or even worse heat transfer performance as compared to the smooth flat surface. This indicates that rather than a simple increase of the wetted surface area, the arrangement of fins which creates different liquid flow patterns on the heated surface has important effect on the heat transfer performance. The positive effect of increasing wetted surface area can be offset by the thick and slow liquid film flow which is caused by improper fin arrangement on the heated surface.

Figure 8.6 indicates that the heat fluxes obtained by the Ma-Straight.f-V or Ma-Square.p surfaces do not scale up as two times as compared to the total surface area. This is attributed to the temperature gradient from the tip to the base of fins or pins as reported by Webb and Kim (2005) and Silk *et al.* (2006). The temperature at the tip of fins or pins is lower than that at the base. Hence, the heat transfer efficiency at the tip part is poorer. At $\Delta T_{ss} = 35^\circ\text{C}$, the heat flux enhanced by the Ma-Straight.f-V and Ma-Square.p surfaces is 36%, which is much more than that reported by Silk *et al.* (2006) (11% enhancement with the similar fin structures). One possible reason for the further improved heat transfer performance is the accelerated liquid flow on the heated surface with the aid of gravity as shown in Fig. 8.4(c).

8.4.2 Micro- and multiscale-structured surfaces

Heat fluxes as a function of surface superheat (ΔT_{ss}) for the micro- and multiscale-structured surfaces are demonstrated in Fig. 8.7. It is evident that both micro- and multiscale-structured surfaces outperform the smooth flat surface. Especially, the Micro-Flat surface (area enhancement ratio = 1.0) has achieved a competitive heat

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flux enhancement (~32%) as that by the macro-structured surfaces (area enhancement ratio = 2.0). The significant heat transfer improvement for the Micro-Flat surface is probably due to the pronounced capillary effect between the micro-structures which assists in retaining and spreading the thin liquid film on the heated surface uniformly (Yao and Hsieh, 2006; Bostanci *et al.*, 2009). As a result, the thin film evaporation is enhanced. Moreover, the numerous re-entrant cavities on the Micro-Flat surface (see Fig. 8.4(b)) have the potential to enhance nucleate boiling as well.

Fig. 8.7 Heat flux as a function of surface superheat for the micro- and multiscale-structured surfaces

By combining the features of micro- and macro-structured surfaces, Figure 8.7 shows that the multiscale-structured surfaces achieve a higher heat flux compared to the macro-structured surfaces which have the same surface geometries. Using the Smooth-Flat surface as the baseline, the MI-Straight.f-V, MI-Square.p and MI-Triangular.f-V surfaces obtain the similar heat flux enhancement up to 65% at $\Delta T_{ss} = 35^\circ\text{C}$. The further improved heat transfer by the multiscale-structured

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surfaces is attributed to the effects caused by the integration of micro-structures. Note also that, the MI-Triangular.f-V surface which has the area enhancement of 1.75 gains the similar heat transfer performance as that obtained by the MI-Straight.f-V and MI-Square.p surfaces which have the area enhancement of 2.0. It is again convinced that the increase of wetted surface area is not a decisive factor in enhancing the heat transfer in spray cooling. The comparison between the Micro-Flat and multiscale-structured surfaces further supports the argument that proper fin arrangement on the heated surface enhances the heat transfer performance.

8.4.3 Heat transfer mechanisms

Figure 8.8 illustrates Nu as a function of non-dimensional surface superheat. In the range of low surface superheat, nucleate boiling is enhanced with increasing surface superheat and hence, the heat transfer coefficient and Nu increase gradually. When the surface temperature reaches the range of high surface superheat (e.g., $\theta = 2.6$), some unstable bubble barriers form on the heated surface to exacerbate the heat transfer performance. As a result, the heat transfer coefficient decreases after achieving the maximum value, and Nu finally descends with increasing surface superheat. This phenomenon agrees with the typical boiling curve as shown in Fig. 8.5.

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Fig. 8.8 Nu as a function of non-dimensional surface superheat

Figure 8.8 shows the comparisons of thermal performance between the geometrically similar surfaces. The surface pairs for comparison are: (a) Smooth-Flat and Micro-Flat; (b) Ma-Straight.f-V and MI-Straight.f-V; (c) Macro-Square.p and MI-square.p. The results reveal that the presence of micro-structures on a geometrically similar surface enhances heat transfer by enhancing the evaporation and nucleate boiling hence achieves a higher Nu . Besides, the comparisons between Ma-Straight.f-V and Macro-Square.p surfaces, and MI-Straight.f-V and MI-square.p surfaces show that the macro- geometry does not affect Nu much. But the arrangement of straight fins on the heated surface influences Nu clearly. Ma-Straight.f-H gives lower Nu values due to a thicker liquid film formed in the grooves, which suppresses the liquid evaporation and nucleate boiling on the heated surface. In contrast, the Ma-Straight.f-V surface which generates a thinner liquid film over the heated surface is conducive to the liquid evaporation as well as nucleate boiling. Hence, the Ma-Straight.f-V achieves higher Nu values.

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To further investigate the heat transfer mechanisms, another important parameter, evaporation fraction (ϵ) was evaluated. As shown in Fig. 8.9, evaporation fraction increases monotonously with the increase of surface superheat. This is expected since increasing surface temperature not only intensifies nucleate boiling but also intensifies liquid evaporation. A higher evaporation fraction is observed for the surface with micro- structures than the one having the same surface geometry but without micro- structures. This indicates that the employment of micro- structures is positive for phase change heat transfer in the spray cooling process. The macro- structured surfaces (except the Ma-Straight.f-H surface) having higher evaporation fractions than that of the smooth flat surface is due to the faster and thinner liquid film flow on the heated surface. Apart from that, the macro-structured surfaces give larger liquid-vapour interfaces due to the liquid film meniscus formed between the fins or pins (Xie *et al.*, 2011), which are favourable for liquid evaporation.

Fig. 8.9 Evaporation fraction as a function of non-dimensional surface superheat

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8.4.4 Critical heat flux (CHF)

The transient process from the steady state to the incidence of CHF and thereafter were tracked by the thermocouple readings, system pressures and surface heat fluxes, as shown in Fig. 8.10. Three surfaces: Smooth-Flat surface, Ma-Straight.f-V surface, and Ml-Triangular.f-V surface, were studied.

Fig. 8.10 Transient process before and after the incidence of CHF. (a) dynamics of temperature T_1 and system pressures during the transient process; (b) dynamic heat flux dissipated from the heater surface during the transient process

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The incidence of CHF can be observed when power input is slightly increased above the maximum steady state heat flux (q''_{steady}) obtained in the experiments. The maximum steady state heat fluxes for the Smooth-Flat surface, Ma-Straight.f-V surface, and Ml- Triangular.f-V surface are 164.2 W/cm², 235.3 W/cm² and 270.7 W/cm² respectively, and the corresponding surface temperatures are 61°C, 61°C and 56°C.

Figure 8.10(a) illustrates the readings of T_1 temperature, nozzle inlet pressure, and chamber pressure as a function of time in the typical transient process of each surface. The transient process can be divided into three regimes: transition regime, film boiling regime, and re-impingement regime. Repeated experiments were conducted to verify that such phenomena indeed happen within a narrow heat flux range of ± 15 W/cm². When the power input was increased above q''_{steady} , the temperature gradually increased over time before a sudden temperature jump to reach film boiling. As suggested by Pais *et al.* (1992), the incidence of CHF in spray cooling is due to the combination of neighbouring bubbles on the heated surface which finally form large vapour films with dry vapour stems and hot spots. The closer the nucleation sites get together, the easier the neighbouring bubbles combine. However, Lin and Ponnappan (2003) suggested that CHF is caused by the inability of the impinging droplets to reach the heated surface due to the counter-current vapour arising from the heated surface. If CHF is attributed to the combination of neighbouring bubbles, the multiscale-structured surfaces which have numerous re-entrant cavities on the heated surface has the highest possibility to reach CHF first. Hence, the explanation proposed by Pais *et al.* (1992) agrees with the present experiments that Ml- Triangular.f-V reaches CHF at a lower surface temperature

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($T_{surf} = 56^{\circ}\text{C}$). In the following paragraphs, both hypotheses are used to describe the typical transient process of each surface.

As shown in Fig. 8.10(a), when $q'' > q''_{steady}$, T_1 starts to increase and the transient process begins. When a sudden temperature jump occurs, the heat source was turned off immediately to prevent the damage of heater module while the spray continues. Figure 8.10(a) shows that the transition period ($\Delta t_{transition}$) from steady state to the temperature jump depends on the geometry of the heated surface. The Smooth-Flat surface reaches film boiling most quickly ($\Delta t_{transition} = 5$ minutes), followed by the Ma-Straight.f-V ($\Delta t_{transition} = 8$ minutes) and MI-Triangular.f-V surfaces ($\Delta t_{transition} = 12$ minutes). To explain this phenomenon, the schematic of droplet behaviours is shown in Fig. 8.11.

Fig. 8.11 Droplet and bubble behaviours in the transient process: (a) Smooth-Flat surface, droplets take the longest distance (8.8 mm) to impinge on the heat surface and growing bubbles are easy to merge; (b) Ma-Straight.f-V surface (fin height = 1 mm), droplets take the shortest distance (7.8 mm) to impinge on the heat surface (top surface) and growing bubbles are separated by the fins; (c) MI-Triangular.f-V surface (fin height = 0.87 mm), droplets take mediate distance (7.93 mm) to impinge on the heater surface (top surface) and growing bubbles are separated by the fins

For the Smooth-Flat surface, as the surface temperature increases, the bubbles on the heated surface grow rapidly. Eventually, more bubbles combine together to

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generate a thick vapour film which prevents the droplets from impinging on the heated surface (see Fig. 8.11(a)), thus resulting in a shorter $\Delta t_{transition}$ before CHF occurs. For the Ma-Straight.f-V surface, as the base surface has a higher surface temperature, the bubbles are deemed to incipiently form vapour films or stems in the grooves (Xie *et al.*, 2011). The straight fins configuration prevents the bubbles in the neighbouring grooves from combining. In addition, the top surfaces of the straight fins are available for droplets to impinge even though the grooves are isolated with vapour films (see Fig. 8.11(b)). CHF does not occur until the surface temperature is sufficiently high and the entire heated surface (including the straight fins) is totally blanketed by a vapour film. As a result, $\Delta t_{transition}$ of Ma-Straight.f-V surface is prolonged. This is also the reason for the longer $\Delta t_{transition}$ of MI-Triangular.f-V surface as compared to the smooth flat surface.

Figure 8.10(b) shows the corresponding heat flux as a function of time. The maximum heat flux (q''_{CHF}) occurs slightly earlier before the occurrence of film boiling, which displays a very short transition boiling regime (see Fig. 8.5) in spray cooling. During film boiling, the heated surface is totally covered by a vapour film to result in a minimum and constant heat flux (q''_{film}). It is worth to note that the duration of the heated surface remaining in film boiling ($\Delta t_{film.boiling}$) depends on the geometry of the heated surface. The Ma-Straight.f-V experiences the shortest $\Delta t_{film.boiling}$ ($\Delta t_{film.boiling} = 4$ minutes), followed by the MI-Triangular.f-V surface ($\Delta t_{film.boiling} = 5$ minutes) and Smooth-Flat surface ($\Delta t_{film.boiling} = 7$ minutes). The surfaces with macro-fins having larger surface area are favourable to heat conduction and radiation across the vapour film to the impinging droplets, thus

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resulting in a higher q''_{film} (Ma-Straight.f-V: $q''_{film} = 52.3 \text{ W/cm}^2$, MI-Triangular.f-V: $q''_{film} = 46.7 \text{ W/cm}^2$, Smooth-Flat: $q''_{film} = 25.3 \text{ W/cm}^2$). The energy stored in the heater with macro- fins is dissipated more rapidly, so that the surface temperature decreases faster during the film boiling regime. Moreover, as the surfaces with macro- fins shorten the travel distance of the impinging droplets to arrive at the heated surface (e.g., the tip surfaces of fins as shown in Fig. 8.11), the droplets can easily re-impinge on the tip surfaces of macro fins. When the surface temperature decreases to attain the droplet re-impingement regime, the heat transfer becomes intensified. As a result, a sudden increase in the surface heat flux and a sudden decrease in temperature are observed in Fig. 8.10.

The dynamics of the measured nozzle inlet pressure and chamber pressure in Fig. 8.10(a) support the aforementioned descriptions on the film boiling regime. It was observed that the nozzle inlet and chamber outlet pressures remained constant before the incidence of film boiling. When film boiling occurred, the chamber pressure dropped abruptly, and the chamber inlet pressure decreased gradually. This is because during film boiling, the impinging droplets could not penetrate the vapour film before they were drifted out from the spray chamber. Therefore, less vapour was generated in the spray chamber. However, as the running speed of the compressor was kept constant, the amount of generated vapour was insufficient to match the compressor speed so that the chamber pressure reduced abruptly as film boiling occurred. When the droplets re-impinged on the heated surface, film boiling disappeared. An enormous amount of vapour was generated in the spray chamber due to the intensified heat transfer, and as a result, the chamber pressure rose abruptly.

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The investigation shows that, apart from the enhanced heat transfer performance, the macro-structured surface can provide a safer operation for a spray cooling system as it can prolong the transition period ($\Delta t_{transition}$) before CHF occurs and reduce the duration ($\Delta t_{filmboiling}$) when the heated surface remains at a high temperature (film boiling) after the heater is powered off.

8.5 Summary

In this chapter, the enhanced surfaces including micro-, macro- and multiscale-structured shapes along with a referenced smooth flat surface were tested in a closed loop system. The experimental results indicate that the fin arrangement played an important role in the cooling performance of macro-structured surfaces. Fabricating the surfaces with micro-structures improved liquid evaporation and nucleate boiling by enhancing the capillary effects and providing more potential nucleation sites on the heated surface. Using the smooth flat surface as a reference, the flat micro-structured surface produced a relative heat transfer enhancement of 32% compared to the 36% of the macro-structured surfaces, while the multiscale-structured surfaces achieve a heat transfer enhancement of up to 65%. In addition to improving the heat transfer performance, macro-structured surfaces provide other advantages such as prolonging the transition process before CHF occurs and reducing the time duration when the heated surface remains in the film boiling regime after powering off the heat source.

Chapter 9

Conclusions and Future Work

9.1 Conclusions

In this thesis, the spray patterns of the pressure swirl nozzles operating under different conditions were characterised using the PDA and PIV techniques. The effects of spray characteristics on the spray cooling performance were investigated. Two theoretical models were derived based on the experimental spray characteristics. A prototype of a high power multiple nozzles spray cooling system was established and tested. Heat transfer mechanisms associated with the enhanced heat transfer performance of structured surfaces in spray cooling were experimentally scrutinised. The main conclusions can be drawn as follows.

In free spray atomisation, a “Dantec” PDA system was used to characterise the spray characteristics of two pressure swirl nozzles. The spray cone produced by the pressure swirl nozzles evolves from a hollow spray cone to a full spray cone with increasing axial distance. The spatial droplet flux distribution plays an important role in determining the local surface temperature in the non-boiling regime of spray cooling. The effects of nozzle-to-surface distance on the heat transfer performance are complex and surface temperature dependent. A concise empirical model has been derived based on the impinging Reynolds number, non-dimensional nozzle-to-surface distance and non-dimensional surface temperature. The model shows positive effects of the impinging Reynolds number and surface temperature on the heat transfer performance, while negative effects of nozzle-to-surface distance.

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With the aid of the PDA and PIV techniques, significant thermal effects were observed on the spray cone formation in spray cooling. The spray cone of a pressure swirl nozzle was found to expand after impinging on a heated surface with relatively high temperatures. The spray cone expands more drastically at a higher surface temperature. As a result, the impinging droplet flux near the centre of spray cone decreases and the spray cone changes from a full spray cone to a hollow spray cone. Increasing the surface temperature leads to an increase in the impinging droplet diameter but a decrease in the impinging droplet velocity, and hence has little effects on the impinging droplet impact energy (We). The expansion of spray cone has significant effects on the surface temperature non-uniformity and heat transfer coefficient in spray cooling.

An analytical thin film flow submodel was first developed to investigate the thin film flow under spray impingement. Then, a heat transfer submodel was constructed based on the thin film flow model to study the heat transfer in the non-boiling regime of spray cooling. This work made the first effort in spray cooling that the film thickness and heat transfer performance were reasonably estimated in an analytical way. Although the modelling was greatly dependent on the detailed spray characteristics, such as the droplet diameter/velocity, and spatial flux distributions for the thin film flow submodel, and the heat transfer empirical correlations for the heat transfer submodel, the attempt made in the present study could stimulate the future modelling work on spray cooling.

A dynamic spray cooling model was developed to investigate the dynamics of droplet impingement, bubble boiling as well as their interactions in the boiling regime of spray cooling. Experimental spray characteristics were applied as the

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boundary conditions in the simulation to predict the surface temperature distribution and heat transfer coefficient. Reasonable comparisons are achieved between the experiments and simulations. This model provided a real-time simulation on the spray cooling process, by which the more insightful understanding on the spray cooling process was possible. The model shows that the fluxes of the collapsed bubbles due to the limited bubble size (D_m), as well as the punctured bubbles due to the droplet impingement, increase as heat flux increases. By fixing the heat flux, flow rate, and droplet velocity, a smaller impinging droplet diameter is favourable for bubble boiling due to the more secondary nuclei induced by a higher impinging droplet flux as well as the larger fractions of the bubbles punctured at bigger diameters.

A prototype of a high power closed loop spray cooling system was constructed to study the feasibility of using a multi-nozzle array on a 6U electronic card cooling. Simple drainage concepts were introduced in the spray chamber design. An open loop visualisation experiment reveals that a uniform liquid distribution by the multi-nozzle array is achieved on the impinged surface without flooding the spray chamber. Heat transfer experiments demonstrate a promising prospect of using multi-nozzle arrays on the large area power electronics cooling. A heat load of 16 kW was removed from the 6U card surface by maintaining the mean surface temperature below 26.5°C. A heat transfer coefficient of up to 2.8 W/cm²·K is obtained, and the liquid evaporation fraction of up to 0.88 (critical value) was achieved before CHF occurs. It was found that increasing nozzle pressure drop or flow rate enhances the heat transfer coefficient and gives better surface temperature uniformity. The chamber pressure significantly influenced the mean surface temperature, but had no observable effect on the surface temperature uniformity. The control of chamber

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pressure is able to maintain the same surface temperature of the device under different heat loads. The present study shows that the spray-to-spray interactions have little effects on local surface temperatures but rather the distance from a location on the heated surface to the drainage outlets.

Experiments were conducted to study the thermal effects of structured surfaces in spray cooling. The structured surfaces fabricated with micro-, macro- and multiscale-structures were tested in a closed loop R134a system. Fins arrangement rather than a simple increase in wetted area has been found to play an important role in the heat transfer performance of macro-structured surfaces. Micro- structures fabricated on the heated surface improves the heat transfer performance by enhancing the surface capillary effects and providing more potential nucleation sites. Multiscale-structured surfaces which combine the micro- and macro- structures enhance the heat transfer performance even further. Taking the smooth flat surface as a reference, the micro-structured flat surface achieves a relative heat transfer enhancement of 32% compared to the 36% of the macro-structured surfaces, while the multiscale-structured surfaces achieve a heat transfer enhancement of up to 65%. Moreover, macro-structured surfaces were found to prolong the transition period before CHF occurs and to shorten the time period that the heated surface remains in film boiling after powering off the heat source.

9.2 Future work

In the light of the obtained experience and conclusions from the current project, some additional topics are recommended as follows for future research in order to achieve better understanding of the heat transfer mechanisms of spray cooling and design potential applications in engineering.

Chapter 9 Conclusions and Future work

1. In view of the significant thermal effects on the spray cone formation of pressure swirl nozzles in spray cooling, it will be interesting to investigate the thermal effects on the spray cone formation of other types of spray nozzles, which have different atomisation mechanisms compared to pressure swirl nozzle, such as jet nozzles, and jet-swirl nozzles. This experiment can be carried out using the experimental facility as shown in Figs. 4.4 and 4.5
2. For more comprehensive model validations, further experimental studies are needed to measure the film flow pattern under spray impingement, such as spatial distributions of the film thickness (Fig. 9.1) and velocity. The heat transfer and spray characterisation of other types of spray nozzles (e.g., jet nozzle, jet-swirl nozzle) will test the applicability of the presented models further.

Fig. 9.1 Schematic of liquid film thickness measurement (Gong *et al.*, 2010)

3. The cooling capacity of the proposed high power spray cooling system is limited by the maximum flow rate that can be delivered by the current multi-nozzle array. Further experiments are needed to investigate the cooling limit

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from a 6U electronic card by using nozzles which have higher flow rate capacities. This experiment can be carried out using the facility as shown in Fig. 7.1.

4. The numerical simulations of single or monodispersed droplets impinging on a heated surface with and without a saturated liquid film will be significant to the fundamental understanding on the spray cooling process. By doing so, the effects of the impinging angle of droplet on the heat transfer performance can be investigated in detail. This numerical model can be developed using ANSYS FLUENT CFD software.

Fig. 9.2 Schematic of the proposed simulation

Publications Arising from this Thesis

Journal papers

Xie, J. L., R. Zhao, F. Duan and T. N. Wong, Thin liquid film flow and heat transfer under spray impingement, *Applied Thermal Engineering*, Vol. 48, pp. 342-348 (2012)

Xie, J. L., Z. W. Gan, F. Duan, T. N. Wong, S. C. M. Yu and R. Zhao, Characterization of spray atomization and heat transfer of pressure swirl nozzles. *International Journal of Thermal Sciences*, Vol. 68, pp. 94-102 (2013)

Tan, Y. B., **J. L. Xie**, F. Duan, T. N. Wong, K. C. Toh, K. F. Choo, P. K. Chan and Y. S. Chua, Multi-nozzle spray cooling for high heat flux applications in a closed loop system, *Applied Thermal Engineering*, Vol. 54, pp. 372-379 (2013)

Xie, J. L., Y. B. Tan, F. Duan, K. Ranjith, T. N. Wong, K. C. Toh, K. F. Choo and P. K. Chan, Study of heat transfer enhancement for structured surfaces in spray cooling, *Applied Thermal Engineering*, Vol. 59, pp. 464-472 (2013)

Xie J. L., Z. W. Gan, T. N. Wong, F. Duan, S. C. M. Yu and Y. H. Wu, Thermal effects on a pressure swirl nozzle in spray cooling, *International Journal of Heat and Mass Transfer*, Vol. 73, pp. 130-140 (2014)

Conference paper

Xie, J. L., R. Zhao, F. Duan and T. N. Wong, Characterization of spray atomization and heat transfer for a swirl nozzle, *International Conference on Renewable Energies and Power Quality*, 2012, Santiago de Compostela, Spain.

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Appendix: Sample analysis of heat loss through the insulation material

Appendix: Sample analysis of heat loss through the insulation material

A 3D thermal analysis was performed to estimate the heat loss from the heater block using ANSYS 12.1. Fig. A1 shows the physical model of the heater block. The dimension of the rectangular copper block is 25 cm × 18 cm × 8 cm. After being assembled with the spray chamber, the sprayed surface area is 23.3 cm × 16 cm. Twelve cartridge heaters are inserted into the copper block from two sides, and each heater is capable of delivering 2.4 kW heat. The Teflon insulators with thickness of 2 cm are used to insulate the copper block.

Fig. A1 Physical model of the heater simulation

The boundary conditions in the simulations are shown in Fig. A2. Heat power input to the heater block was set at the applied heat load. A uniform surface temperature was set on the sprayed surface at 25°C. The boundary condition at the outside surfaces of the insulators was set as a natural convection heat transfer coefficient ($h = 25 \text{ W/cm}^2 \cdot \text{K}$ (Rathore and Kapuno, 2011)).

Appendix: Sample analysis of heat loss through the insulation material

Fig. A2 Boundary conditions of the thermal analysis

The simulation results without showing the insulators are demonstrated in Figs. A3 and A4. The thermal analysis shows that the maximum temperature inside the heater block is 112°C at the heat load of 20 kW (assuming the surface temperature is 25°C), which is much lower than the melting temperatures of the copper block and the insulation material. Moreover, a linear temperature distribution was observed below the heated surface, which gave the advantage to calculate the surface temperature in the experiments. Using this design, a uniform heat flux distribution can be achieved on the sprayed surface as shown in Fig. A4.

Fig. A3 Temperature distribution of the heater block at $q = 20\text{ kW}$

Appendix: Sample analysis of heat loss through the insulation material

Fig. A4 Heat flux distribution of the heater block at $q = 20 \text{ kW}$

Table A1 summarizes the heat loss of the applied cases with different heat loads. It shows that the heat loss to the environment is negligible as compared to the heat input to the heat block.

Table A1 Summary of heat loss to the environment under different conditions

	Heat input (W)	Heat loss (W)	Percentage of loss (%)
Case 1	6000	25.8	0.43
Case 2	8500	37.0	0.44
Case 3	10000	44.2	0.44
Case 4	14000	62.6	0.45
Case 5	16000	71.8	0.45
Case 6	20000	90.2	0.45