Submerged boiling and jet impingement for cooling high power electronics

Fan, Simiao

2021

Fan, S. (2021). Submerged boiling and jet impingement for cooling high power electronics. Doctoral thesis, Nanyang Technological University, Singapore.
<https://hdl.handle.net/10356/148922>

<https://hdl.handle.net/10356/148922>

<https://doi.org/10.32657/10356/148922>

This work is licensed under a Creative Commons Attribution-NonCommercial 4.0 International License (CC BY-NC 4.0).

Downloaded on 26 Apr 2023 20:17:36 SGT

**NANYANG
TECHNOLOGICAL
UNIVERSITY**

SINGAPORE

**SUBMERGED BOILING AND JET IMPINGEMENT FOR
COOLING HIGH POWER ELECTRONICS**

FAN SIMIAO

**INTERDISCIPLINARY GRADUATE SCHOOL
ENERGY RESEARCH INSTITUTE @ NTU (ERI@N)**

2020

SUBMERGED BOILING AND JET IMPINGEMENT FOR COOLING HIGH POWER ELECTRONICS

FAN SIMIAO

Supervisor: Assoc. Prof. Fei DUAN, School of Mechanical and
Aerospace Engineering

Co-supervisor: Assoc. Prof. LUO Jun, School of Computer Science and
Engineering

Interdisciplinary Graduate School
Energy Research Institute @ NTU (ERI@N)

A thesis submitted to the Nanyang Technological University
in partial fulfilment of the requirement for the degree of
Doctor of Philosophy

2020

Statement of Originality

I hereby certify that the work embodied in this thesis is the result of original research, is free of plagiarised materials, and has not been submitted for a higher degree to any other University or Institution.

09/12/2020

.....

Date

Fan Simiao

.....

Fan Simiao

Supervisor Declaration Statement

I have reviewed the content and presentation style of this thesis and declare it is free of plagiarism and of sufficient grammatical clarity to be examined. To the best of my knowledge, the research and writing are those of the candidate except as acknowledged in the Author Attribution Statement. I confirm that the investigations were conducted in accord with the ethics policies and integrity standards of Nanyang Technological University and that the research data are presented honestly and without prejudice.

09/12/2020

.....

Date

.....

Fei Duan

Authorship Attribution Statement

This thesis contains material from 2 papers published in the following peer-reviewed journals where I was the first author.

Chapter 3 is related to journal paper: **S. Fan**, L. Jiao, K. Wang, F. Duan, Pool boiling heat transfer of saturated water on rough surfaces with the effect of roughening techniques, International Journal of Heat and Mass Transfer 159 (2020) 120054.

The contributions of the co-authors are as follows:

- Prof. Fei Duan supervised me on the overall research direction and edited the manuscript drafts.
- Dr. Jiao Lishi and Dr. Wang Kai designed part of the experiments and gave comments on the manuscript.
- I proposed the idea, performed all the laboratory work, analyzed the experimental data, and prepared the manuscript drafts.

Chapter 4 is related to journal paper: **S. Fan**, W. Tong, F. Duan, Nucleate pool boiling heat transfer enhancement in saturated Novec 7100 using titanium dioxide nanotube arrays, International Communications in Heat and Mass Transfer, 122 (2021) 105166.

The contributions of the co-authors are as follows:

- Prof. Fei Duan provided the initial project direction and edited the manuscript drafts.
- Dr. Tong Wei proposed the idea and gave comments on the manuscript.
- I performed the laboratory work, analyzed the experimental data, and prepared the manuscript drafts.

09/12/2020

.....
Date

Fan Simiao

.....
Fan Simiao

Abstract

The submerged boiling is expected to be a good solution for the ever-increasing cooling demands of electronic devices in the industries including automotive vehicles, aircraft and spacecraft avionics, power inverters, military equipment, mobile devices, supercomputers, and data centers. Although it has been investigated extensively, the development and application of the submerged boiling systems still face some challenging issues: (i) a comprehensive understanding of the heat transfer mechanism is lacked; (ii) results from similar experimental condition are often divergent or even contradictory; (iii) augmentation methods are usually necessary due to poor thermal properties of dielectric fluids; (iv) a universal prediction tool is absent. This thesis contributes to part of the pressing issues mentioned above to further understand the submerged boiling.

To distinguish the large-scale component and the short-scale component in the three-dimensional (3D) surface analysis, arithmetic mean heights for the primary surface and the surface roughness component have been defined as 3D roughness parameters. The effects of the 3D roughness parameters on the pool boiling heat transfer are experimentally investigated on eight testing surfaces fabricated through three different preparation methods. Experimental results show that a rougher surface has higher heat transfer coefficient than a smoother one only within the same surface preparation method, if the surface roughness is characterized by the parameter derived from the primary surface. However, a different trend is observed when the parameter from the roughness component is applied to characterize the surface roughness. The boiling curves are found to shift monotonically to the left as the roughness parameter becomes larger, regardless of the surface preparation methods. A correlation is developed to well predict the heat transfer coefficient as a function of the surface roughness.

Titania nanotube arrays are anodized on titanium plate in an electrochemical system and the potential application of this type of nanotube in two-phase submerged cooling

system is explored. Experimental results show that the titania nanotubes are effective in improving both the cooling performance and the thermal safety margin of a two-phase submerged cooling system for titanium-based devices. However, its applications in devices based on other materials should be cautious unless the potential effect of the thermal interface resistance between the device surface and the nanotube coating can be eliminated. The nanotube structures are recognized to provide faster liquid replenishment hence modify the bubble dynamics and improve the dry spot rewetting capacity.

A submerged jet impingement boiling system with a single circular nozzle is proposed to investigate the steady-state heat transfer characteristics at the stagnation point from single phase convection to partial nucleate boiling. The onset of nucleate boiling is highlighted. The impacts of various parameters including jet Reynolds number, liquid subcooling, and nozzle diameter on the heat transfer characteristics of the onset of nucleate boiling are discussed. Empirical correlations are proposed based on the experimental data to predict the local heat transfer coefficient, heat flux, and wall superheat at the onset of nucleate boiling.

The future research directions are suggested for the submerged boiling heat transfer. Firstly, the departure characteristics of the single bubble are suggested to be investigated under different experimental conditions to explore its dependence on the control parameters hence help further understand the departure mechanism of nucleation bubbles. The second suggestion is to design the elaborate surface structures to separate the liquid and vapor flow pathways. In particular, the 3D printing method can provide more flexibility in designing the surface structures and it will be very interesting to apply the 3D printing in designing separated liquid-vapor flow pathways. Lastly, models for the nucleation site density, bubble departure diameter, and bubble departure frequency will be developed based on the quantitative analysis of the bubble dynamics study in this thesis. The developed models are expected to improve the accuracy of bubble dynamics models hence enhance the reliability of the Rensselaer-Polytechnic Institute (RPI) boiling model in numerical simulations.

Acknowledgments

First and foremost, I would like to express my sincere gratitude towards my supervisor and co-supervisor, Professor Fei Duan and Professor Luo Jun from Nanyang Technological University, for their guidance and continuous support of my Ph.D. study and related research. Their patience, motivation, and immense knowledge have lighted me the way for my academic career as well as my life.

I would also like to convey my highest appreciation to the colleagues in Nanyang Technological University including Dr. Tong Wei, Dr. Qiu Lu, Dr. Wang Kai, Dr. Zhong Xin, Dr. Baris Burak Kanbur, Mr. Wu Chenlong, Mr. Qin Zhen, Mr. Ren Junheng, Mr. Zhou Yi, and Mr. Harikumar Govind, for their continued motivation and assistance along the way. Furthermore, I would like to give my special thanks to Mr. Yeo Boon Chuan and Mr. Ang Koon Tech, who are the technicians at the Energy System Laboratory, for their kind assistance in machine operation and materials preparation.

I would like to thank my friends for their kind support in my work and life. Special thanks to my girlfriend, Ms. Jin Yingzi, for her unselfish support and warm encouragement in my difficult times. Last but not the least, I would like to express my deep love to my parents and all other family members for their unconditional love and support during the past 27 years. They are the reason why I keep pushing, I keep facing all the struggles, pains, and hardships. Their love is what is keeping me get through the tough but rewarding PhD journey.

Table of Contents

Abstract	i
Acknowledgments.....	iii
List of Figures.....	vi
List of Table	ix
List of symbols	x
Chapter 1 Introduction	1
1.1 Background	1
1.2 Motivations.....	5
1.3 Objectives.....	8
1.4 Scopes.....	9
1.5 Thesis organization.....	9
Chapter 2 Literature Review	12
2.1 Working Fluids	13
2.1.1 Traditional coolants	13
2.1.2 Dielectric fluids.....	20
2.1.3 Nanofluids.....	36
2.2 Effects of Surface Characteristics	41
2.2.1 Effects of roughness and contact angle.....	42
2.2.2 Effects of surface size, surface orientation and surface aging	49
2.2.3 Effect of surface structures	54
2.3 Summary	64
Chapter 3 Pool boiling heat transfer of saturated water on rough surfaces with the effect of roughening techniques	66
3.1 Introduction	66
3.2 Experimental setup and procedures.....	70
3.2.1 Experimental setup	70
3.2.2 Surface preparation	72
3.2.3 Experimental procedures	79

3.2.4	Calculations and uncertainty analysis.....	80
3.3	Results and discussion.....	82
3.4	Summary	92
Chapter 4	Nucleate boiling heat transfer enhancement in saturated Novec 7100 using titania nanotube arrays.....	94
4.1	Introduction	94
4.2	Experimental setup and procedures.....	98
4.2.1	Experimental setup	98
4.2.2	Surface preparation	99
4.2.3	Experimental procedures	100
4.2.4	Experimental data and uncertainty analysis.....	101
4.3	Results and discussion.....	102
4.4	Summary	110
Chapter 5	Correlations for heat transfer characteristics of the onset of nucleate boiling in submerged jet impingement	112
5.1	Introduction	112
5.2	Experimental setup and procedures.....	115
5.2.1	Experimental setup.....	115
5.2.2	Experimental procedures	117
5.2.3	Experimental data and uncertainty analysis.....	119
5.3	Results and discussion.....	120
5.3.1	Effects of jet parameters on h_{ONB}	124
5.3.2	Effects of parameters on the ONB point.....	127
5.4	Summary	132
Chapter 6	Conclusions and Future Work	134
6.1	Conclusions	134
6.2	Future work	136
Author's Publications		139
References		141

List of Figures

Fig. 1.1 The boiling curve shows the heat transfer regime (redrawn from Nguyen et al. [5]). The microchannel flow of direct and indirect cooling is illustrated in the inset....	2
Fig. 1.2 Different submerged boiling configurations (redrawn from Molana and Banooni [9], Kim et al. [10], Cardenas and Narayanan [11] and Clark et al. [12]).....	5
Fig. 2.1 Boiling curves in pure water with uncoated and nano-coated heater surfaces under various system pressures (reprinted from kwark et al. [30]).	16
Fig. 2.2 Effect of nozzle diameter at different Re numbers (redrawn from Cardenas and Narayanan [11]).....	29
Fig. 2.3 Boiling curves for various jet velocities at different nozzle-to-surface spacings (reprinted from Shin et al. [15]).	30
Fig. 2.4 Boiling curves and $h-q$ curves at different jet velocities and nozzle-to-surface distances (reprinted from Clark et al. [12]).....	32
Fig. 2.5 Mechanism of particle deposition during boiling process (redrawn from Kwark et al. [93]).	40
Fig. 2.6 Boiling curves for water and FC-77 on a set of surfaces with various surface roughnesses, where EDM stands for Electrical Discharge Machined surface. (redrawn from Jones et al. [102]).	43
Fig. 2.7 SEM images for (a) biphilic surface and (b)superbiphilic surface (reprinted from Betz et al. [109]).....	46
Fig. 2.8 Bubble dynamics on the microporous-coated surface and plain surface under various surface orientations (redrawn from Jun et al. [125]).....	52
Fig. 2.9 Artery-evaporator system with completely separated liquid and vapor flow pathways (redrawn from Liter and Kaviani [148]).	58
Fig. 2.10 (a) Schematics of the porous artery structure. (b) Boiling curves of smooth and porous artery structure surfaces (redrawn from Bai et al. [149]).....	59
Fig. 2.11 (a) Samples of selectively thermal insulated surfaces. (b) Schematic of the separate liquid-vapor flow pathways induced by selectively thermal insulated surfaces (redrawn from Rahman et al. [152]).	61
Fig. 2.12 Heat transfer mechanisms of open microchannels with different porous coating (redrawn from Jaikumar and Kandlikar [158]).	63
Fig. 2.13 Designs of the pool boiling flow: (a) original design; (b) design with a straight guidance tube; (c) design with a trumpet guidance tube. (reprinted from Yuan et al. [163]).....	64

Fig. 3.1 (a) Schematic of the whole apparatus, (b) detailed drawing of the heating block.....	70
Fig. 3.2 Measurement results of the surface topography and surface wettability.....	72
Fig. 3.3 Example of the pre-processing before the roughness parameter evaluation. .	75
Fig. 3.4 Illustration of applying the robust gaussian filter (25 μm cut-off) on the profile and surface, respectively.	76
Fig. 3.5 (a) Measurements of static contact angle at different directions, (b) relation between static contact angle and surface roughness Sa	77
Fig. 3.6 Validation of the present experiments.	83
Fig. 3.7 Results for all the testing surfaces, (a) $q-\Delta T_{sat}$ curves, (b) $h-q$ curves.....	84
Fig. 3.8 Comparison of the experimental results of laser processed 1 with sandpaper polished 80 grit and sandpaper polished 600 grit.	87
Fig. 3.9 Comparison of the visualized results with increasing Sa	87
Fig. 3.10 A simple case in evaluating bubble departure characteristics.....	88
Fig. 3.11 Fitting and parameter development of the $Nu-Ja$ curves.....	91
Fig. 3.12 Evaluation of C in Eq. (3.10).....	91
Fig. 3.13 Comparison results of Eq. (3.14) and experimental data.	92
Fig. 4.1 Schematic diagram of the experimental setup.	99
Fig. 4.2 Tested surfaces with static contact angle measurements.	99
Fig. 4.3 Boiling curves of saturated Novec 7100 and plain copper surface: in present study, previous publications, and Rohsenow correlation.	103
Fig. 4.4 Results of all tested surfaces in saturated Novec 7100, where “Nanotubes on copper substrate (r)” is calculated after reducing a thermal interface resistance of 60 $\text{K}\cdot\text{mm}^2/\text{W}$ from the experimental data of the “Nanotubes on copper substrate”.....	103
Fig. 4.5 $h-q$ curves of all tested surfaces in saturated Novec 7100, where “Nanotubes on copper substrate (r)” is calculated after reducing a thermal interface resistance of 60 Kmm^2/W from the experimental data of the “Nanotubes on copper substrate”. ..	106
Fig. 4.6 Visual observations of different tested surfaces with increasing heat flux...	107
Fig. 4.7 Evaluation of the bubble departure diameter and bubble departure frequency at heat flux of 7.75 W/cm^2 . Please also see the supplementary animation.	109
Fig. 4.8 Schematic diagram of the bubble dynamics on nanotube structures.....	110
Fig. 5.1 Schematic diagram of the flow loop.....	115

Fig. 5.2 Temperature distribution along the thermocouples T_1 to T_3 at different heat fluxes.....	120
Fig. 5.3 A sample case for determining the ONB point.....	122
Fig. 5.4 Experimental results of different liquid subcoolings and jet Reynolds numbers at fixed nozzle diameter and nozzle-to-surface distance.....	123
Fig. 5.5 Experimental results of varying nozzle diameters at constant subcooling and nozzle-to-surface distance for two different Reynolds numbers.	124
Fig. 5.6 Evaluation of the exponent of Reynolds number in Eq. (5.3).....	126
Fig. 5.7 Comparison results of Eq. (5.5) and experimental data.	127
Fig. 5.8 The stagnation heat flux at the ONB point under different Reynolds numbers.	129
Fig. 5.9 The stagnation heat flux at the ONB point under different liquid subcoolings.	129
Fig. 5.10 Relation between the stagnation point heat flux at the ONB point and the nozzle diameter at constant subcooling and nozzle-to-surface distance for two different Reynolds numbers.	131
Fig. 5.11 The ONB heat flux correlated with Reynolds number, liquid subcooling, and nozzle diameter.	132

List of Table

Table 3.1 Summary of the roughness parameter measurements on the testing surfaces.	79
---	----

List of symbols

English symbols

A	area [m ²]
A_p	projective area of the heater surface [m ²]
$C, C_1, C_2, C_3, C_4, C_s$	constants
C_p	specific heat [J/kg·K]
C_{sf}	Rohsenow constant
d	diameter [m]
G	mass velocity [kg/m ² s]
g	gravitational acceleration [m/s ²]
H	nozzle-to-surface distance [m]
h	heat transfer coefficient [W/m ² ·K]
Ja	Jakob number
K	constant
k	thermal conductivity [W/m·K]
L	latent heat [J/kg]
m	mass flow rate [kg/s]

Nu	Nusselt number
Pe	Péclet Number
Pr	Prandlt number
q	heat flux [W/m ²]
r	effective heat transfer area ratio
Ra	arithmetical mean height of roughness component [m]
Re	Reynolds number
Rq	root-mean-square of roughness component [m]
r_c	critical cavity radius [m]
S	modification factor for Kandlikar correlation
Sa	extension of Ra to a surface [m]
Sa_p	extension of Pa to a surface [m]
S_m	mean spacing of surface roughness [m]
T	temperature [K]
t	time [s]
v	velocity [m/s]
W	nozzle width [m]

x streamwise distance from stagnation line [m]

Greek symbols

α thermal diffusivity [m²/s]

ΔT temperature difference [K]

ΔT_{sat} wall superheat [K]

ΔT_{sub} subcooling [K]

Δx distance [m]

γ material influence parameter

λ_c Taylor wavelength [m]

λ_D most dangerous wavelength [m]

λ_H Helmholtz wavelength [m]

μ dynamic viscosity [Pa·s]

v_{fg} latent specific volume change [m³/kg]

ρ density [kg/m³]

σ surface tension [N/m]

θ static contact angle [°]

φ	inclination angle of boiling surface [°]
-----------	--

Subscripts

CHF	critical heat flux
-----	--------------------

c	convective
-----	------------

cri	critical
-------	----------

j	jet
-----	-----

l	liquid
-----	--------

ONB	onset of nucleate boiling
-----	---------------------------

rec	receding
-------	----------

sat	saturation
-------	------------

sub	subcooling
-------	------------

v	vapor
-----	-------

w	wall
-----	------

Superscripts

m_1, m_2	correlation exponents
------------	-----------------------

n correlation exponent

s correlation exponent

Abbreviations

CHF critical heat flux

HTC heat transfer coefficient

NBHT nucleate boiling heat transfer

ONB onset of nucleate boiling

Chapter 1 Introduction

1.1 Background

The high-performance and rapid miniaturization of electronic devices in the industries including power plants, automotive vehicles, electronics, aircraft and spacecraft avionics, power inverters, military equipment, mobile devices, supercomputers, and data centers [1-3] have generated the intense heat flux or heat density. To maintain the devices for working efficiently, we are facing the challenge on effective thermal management. The traditionally applied single phase cooling method based on the forced convection has encountered a threshold and become inadequate for heavily increasing cooling demands. Boiling, relying on the fast bubble induced flow and the accompanied latent heat during liquid-vapor phase change, can remove a large amount of heat with a relatively low device surface temperature.

Boiling heat transfer, as the same as the single phase cooling method, can be divided into direct cooling way and indirect cooling way depending on the contact between the cooling medium and the heating source. As shown in Fig. 1.1, indirect boiling refers to a configuration where the heating sources and coolants are separated by thermal interface materials. Heat is firstly conducted along the thermal interface materials and then transferred to the coolants by boiling heat transfer. The greatest benefit of indirect boiling is good flexibility in coolant selection. However, the thermal resistance of the interface materials restricts the efficiency of indirect boiling. Direct boiling, in contrast,

eliminates the interfacial thermal resistance by immersing the heating surfaces directly into the coolants. Therefore, a higher heat removal capacity can be achieved by direct boiling compared to the indirect one. Moreover, direct boiling is capable of limiting the local temperature rise of area with the concentrated high heat flux, i.e. hot spot [4]. The main limitation of direct boiling is electrical compatibility. To ensure the electrical compatibility, only dielectric coolants are allowed in the direct boiling setups.

Fig. 1.1 The boiling curve shows the heat transfer regime (redrawn from Nguyen et al. [5]). The microchannel flow of direct and indirect cooling is illustrated in the inset.

The desired working range in the boiling curve is between onset of nucleate boiling (ONB) and critical heat flux (CHF), in which, a large heat flux can be removed at a limited wall superheat, as shown in Fig. 1.1. The boiling phenomenon can be effectively categorized into four stages: (i) single phase convection before the ONB, (ii) nucleate boiling between the ONB and the CHF, (iii) transition boiling, and (iv) film boiling. The studies of boiling heat transfer focus on the nucleate boiling and its two confines, namely ONB and CHF. The ONB is a transition point while the single phase convection

ends, and the nucleate boiling begins. The CHF represents the highest heat flux of the nucleate boiling. After which, the heating surface will be covered by a vapor blanket, resulting in a sudden drop of the heat dissipation rate, and finally causing local burnout. Thus, the CHF is crucial in designing a boiling-involved thermal management system. The maximum design heat flux should be below 70% of CHF, as suggested by El-Genk and Ali [6].

The boiling-involved thermal management systems include pool boiling, internal flow boiling, external flow boiling, and the hybrid configurations with their combination [7]. The pool boiling is the simplest scheme among the above mentioned boiling configurations and has significant benefit of so-called passive cooling without requiring pumping power or moving parts in the system. Although it has been extensively investigated since the 1950s, the heat transfer mechanism of pool boiling is still a hot topic in discussion. The flow boiling is characterized by the forced convective flow over a surface induced by the external force, e.g., a pump. It can be categorized as the internal and external configuration. Unlike pool boiling, the internal or external forced convection contributes a significant portion to the total heat transfer at a heated external surface or a heated internal channel in the flow boiling. The jet impingement boiling is the most representative configuration of the external flow boiling. It is particularly suitable for dissipating both transient and steady concentrated heat loads. One major limitation of the jet impingement is the large impact momentum on the heated surface. Thus, the jet impingement is not acceptable for some delicate devices. However, for the electronics which can stand this impact speed, jet impingement is a useful tool due to

its superior heat transfer performance. The jet impingement can be classified into five categories: free surface, submerged, confined, plunging and wall jets [8]. The first three, especially free surface, are widely investigated. The investigations of jet impingement boiling have been referred from the summaries of Wolf et al. [8] and Qiu et al. [7]. Among the various configurations of jet impingement boiling, the submerged type and the confined type have a lot of similarities with the pool boiling: all the configurations have the heated surface submerged in the working fluid. The boiling types can generally be categorized to the submerged boiling configuration which is characterized by the submergence of the heated surfaces in the working fluids.

Fig. 1.2 combines the schematics of different submerged boiling configurations [9] with the correspondingly visual observations of bubble dynamics from previous investigations [10-12]. As shown in Fig. 1.2(b), the submerged jet impingement boiling refers to a configuration where liquid jet discharges into an ambient liquid with the similar properties. Usually, the liquid jet and environmental liquid are the same for a close loop application. The submerged jet impingement boiling shares common features with the pool boiling in Fig. 1.2(a) in both configurations and heat transfer mechanism. It can be regarded as an enhancement method for the pool boiling which adds one/multiple jet(s) to increase two-phase flow in the liquid pool. On the other hand, some researchers treat pool boiling as one specific case of submerged jet impingement boiling where jet outlet velocity (v_j) equals to 0 m/s. Therefore, the influencing parameters in the pool boiling can also affect the submerged jet impingement boiling. Besides, the jet parameters such as jet velocity, jet diameter, nozzle-to-surface distance,

and jet outlet temperature can also significantly influence the heat transfer performance through alerting the forced convection and bubble nucleation. It should be noticed that a particular case exists in the submerged jet impingement boiling where the liquid between nozzle and heating surface is constricted by an upper wall shown Fig. 1.2(c). The specific configuration is named as confined jet impingement [13]. The confined jet impingement is inevitable in the applications when the space is confined or the installation is limited [14, 15]. Generally, the submerged boiling configuration is attractive in various applications due to the outstanding heat transfer performance, spatial claims and/or installation limitations.

Fig. 1.2 Different submerged boiling configurations (redrawn from Molana and Banooni [9], Kim et al. [10], Cardenas and Narayanan [11] and Clark et al. [12]).

1.2 Motivations

So far, a lot of efforts have been made to provide a comprehensive understanding of the flow and heat transfer fundamentals of the submerged boiling. Studies using experimental or numerical methods have been conducted extensively to help build a database for identifying the control parameters in the submerged boiling and

understanding their relation. However, there still exist some challenging issues in the development and application of the submerged boiling systems. In particular, the following issues are waiting to be solved urgently, (i) two-phase flow pattern and heat transfer mechanism, (ii) characteristics of the CHF for the submerged boiling, (iii) universal models or correlations for the submerged boiling heat transfer, and (iv) heat transfer enhancement methods. This thesis is motivated to address part of the pressing issues and help further understand the submerged boiling systems in terms of heat transfer mechanism, enhancement methods, and prediction tools. Specifically, the motivations are listed below:

- The basic heat transfer mechanism of pool boiling is so-called bubble nucleation which is highly dependent on the surface condition. One way to effectively characterize the surface condition is surface roughness. The surface roughness defines the degree of unevenness of the surface. Parameters for the surface roughness of a solid normally include (i) arithmetical mean height roughness value Ra , (ii) root-mean-square roughness value Rq , and (iii) effective area ratio r . Although these roughness parameters have been investigated for several decades, the dependencies of the pool boiling heat transfer on the roughness parameters are still not fully understood. Results from different researchers are often found to be divergent or even contradictory, owing to different techniques used in the surface roughening process. Therefore, the conclusions of the surface roughness effects in pool boiling using the existing roughness parameters are applicable only within the specified surface roughening technique. A universally conclusive description is

lacked. All the evidence indicates that the existing roughness parameters are not accurate enough to characterize the surface roughness in pool boiling. It is of great importance to improve the accuracy of the existing roughness parameters hence help draw a universal conclusion of the surface roughness effects in pool boiling.

- With improvements in nanotechnology, different nano-structured surfaces are developed and their potential applications in pool boiling as an enhancement method are explored. Among the nanostructures, nanotubes attract a lot of attentions since this type of structure usually owns exceptional mechanical and thermal properties, in conjunction with extremely high aspect ratio and chemical inertness. According to the previous investigations, the nanotube structures are effective in delaying CHF, representing a higher thermal safety margin of a two-phase cooling system. However, there are controversial results about improvement/deterioration of the NBHT coefficient for the applications of nanotube coatings in the pool boiling studies. Some researchers believe that the nanotubes could benefit the boiling heat transfer by increasing surface wettability, surface area, bubble nucleation sites, and/or surface capillarity. But the other researchers report that the nanotubes could deteriorate the boiling heat transfer due to decreased surface roughness and/or changed surface-liquid interactions. Obviously, most researchers focus on the nanotube-induced changes in the surface characteristics rather than the nanotube itself. It is an urgent task to identify the isolated and direct influence of the nanotube on pool boiling by controlling the surface characteristics (such as surface area, roughness, and wettability) invariable

during the experiments, thereby providing a new perspective to understand the controversial results about the influence of nanotube coatings on the NBHT coefficient of pool boiling.

- The onset of nucleate boiling (ONB) is a critical concept in submerged boiling, representing the lower limit of the nucleate boiling regime. It has been investigated extensively in the pool boiling studies. However, the works in this topic are insufficient for submerged/confined jet impingement, especially lacking those performed with controlled parameters. Furthermore, a reliable prediction tool for the onset of nucleate boiling of submerged/confined jet impingement is still blank. To deal with these problems, it is necessary to design experiments in a submerged/confined jet impingement boiling system and collect ONB data with controlled parameters.

1.3 Objectives

The overall objective of this project is to explore the application of pool boiling and submerged/confined jet impingement in thermal management of high-heat-flux devices.

Specifically, this work is supposed to achieve the following research objectives:

- To improve the accuracy of the existing roughness parameters and establish a comprehensive understanding of the dependence of pool boiling heat transfer on the surface roughness.
- To identify the direct influence of the nanotube on pool boiling heat transfer and prospect the further applications of the nanotube coatings in the two-phase

submerged cooling systems.

- To build a database of the ONB with controlled parameters in the submerged jet impingement boiling and develop a reliable prediction tool for the heat transfer characteristics at the ONB point.

1.4 Scopes

The scopes of this thesis include:

- Developing a novel pre-processing method for surface roughness measurement and preparing rough copper surfaces with average roughness values in the range of 0.045 μm to 1.35 μm by three different roughening techniques.
- Comparing the nanotube-coated copper and titanium surfaces with the smooth copper and titanium surfaces in terms of the NBHT coefficient, CHF, and bubble nucleation characteristics under invariable surface characteristics.
- Collecting ONB data of the submerged jet impingement with a single circular nozzle under a wide range of jet Reynolds number, jet subcooling, and nozzle diameter, and establishing empirical correlations based on the collected data.

1.5 Thesis organization

This thesis consists of six chapters, which is organized as follows:

Chapter 2 comprehensively reviews the efforts on the submerged boiling.

Investigations in this topic are summarized based on different types of working fluid including traditional coolants, dielectric fluids, and nanofluids. Specifically, the effects of surface characteristics on the heat transfer performance are highlighted. Limitations of the current cooling techniques and future directions of the enhancement methods are discussed.

Chapter 3 investigates the pool boiling heat transfer of saturated water on rough surfaces with the effect of roughening techniques. The accuracy of roughness parameter Sa is improved through a filtration process. The effects of the roughness parameter Sa on the heat transfer coefficient (HTC) and the critical heat flux (CHF) are explored. Meanwhile, the interdependence between the roughening techniques, static contact angle, and surface roughness is discussed. A prediction tool is proposed by correlating the heat transfer coefficient with the roughness parameter Sa .

Chapter 4 explores the potential application of titania nanotube arrays in two-phase submerged cooling systems. Anodization is employed in an electrochemical system to produce the nanotube arrays. The heat transfer performance of nanotube-coated surfaces is investigated in saturated Novec 7100. The boiling phenomenon on nanotube arrays is visualized and its physical mechanism is explained. At last, limitations and further applications of the nanotube arrays in cooling of high-heat-flux devices are discussed.

Chapter 5 examines the steady-state heat transfer characteristics in a submerged jet from single phase convection to partial nucleate boiling. The onset of nucleate boiling (ONB) is highlighted. The heat flux, wall superheat, and heat transfer coefficient at the ONB are investigated and correlated with controlled parameters including jet Reynolds number, liquid subcooling, and nozzle diameter.

Chapter 6 presents the conclusions of the project and provides some recommendations for future work.

Chapter 2 Literature Review*

The submerged boiling is expected to be a solution for effective thermal management in the industries where electronic devices with high heat dissipation rate are employed. It has been investigated extensively, leaving us an immense database. Based on the database, various parameters are identified to have influence on the submerged boiling. The influencing parameters include but are not limited to active mechanisms (electrostatic field, ultrasonic vibrations, etc.), external conditions (pressure, gravity, etc.), fluid properties, hydrodynamic parameters of jets (jet velocity, jet subcooling), geometrical parameters of jets (nozzle dimension, nozzle-to-surface distance, single slot and jet arrays), and surface characteristics (surface roughness and contact angle, surface structures, heater size, surface orientation, surface porosity, capillarity, and surface aging effect) [16-18]. Obviously, the heat transfer process of the submerged boiling is complicated and depends on a lot of control parameters. A comprehensive review of all the parameters in this chapter is unrealistic. Instead, this chapter will focus on the two most important influence factors, i.e., working fluid and surface characteristics. Besides, the effects of pressure, subcooling, hydrodynamic parameters of jets, and geometrical parameters of jets will also be discussed.

* This chapter has been reproduced in its entirety with permission from the following publication:

S. Fan, F. Duan, A review of two-phase submerged boiling in thermal management of electronic cooling, *International Journal of Heat and Mass Transfer*, 150 (2020) 119324.

2.1 Working Fluids

The submerged boiling studies are reviewed based on the type of working fluids applied in the studies. The effects of the influencing parameters including working fluid, pressure, subcooling, hydrodynamic parameters of jets, and geometrical parameters of jets will be discussed here. Traditional coolants such as water, liquid nitrogen, alcohol, and some hydrocarbon compounds are introduced at first, followed by the dielectric fluids. Lastly, a recent research focusing on the nanofluids is discussed and summarized.

2.1.1 Traditional coolants

The study of pool boiling started since 1940s. In the early stage, water, liquid nitrogen, alcohol, and some hydrocarbon compounds are the main target working fluids. Among these traditional coolants, water plays a dominant role due to its superior thermophysical properties on latent heat, non-flammability, and easy storage. However, there are two limitations with water to be used in direct submerged boiling. One problem comes from the direct contact between the electronic devices and working fluids in the submerged boiling. Due to the good electrical compatibility of water, the current leads must be well protected from water by insulating materials. The other problem is the high boiling temperature. Water has a boiling point of around 100 °C under the atmosphere pressure. It means the heated devices must have a surface temperature over 100 °C to initiate boiling or the water must be maintained in a reduced pressure condition. Under the ambient surroundings, the boiling temperature exceeds the most working range of the electronic devices, limiting the application of submerged

boiling configurations. Although water is not an ideal working fluid in the direct cooling system, it is still widely studied in submerged boiling for the indirect two-phase cooling for the electronics. The reason includes superior heat removal capacity, low cost and well-studied thermophysical properties. A lot of correlations are also initially derived from experimental data in water and then extended to other working fluids.

2.1.1.1 Pool boiling

In 1951, Rohsenow [19] derived his correlation based on the experimental data of water, ethyl alcohol and n-pentane. This correlation is still one of the most widely used prediction methods of the nucleate boiling heat transfer (NBHT) coefficient until now.

The Rohsenow's correlation is written as,

$$\frac{c_{p,l}\Delta T_{sat}}{L} = C_{sf} \left[\frac{q}{\mu L} \sqrt{\frac{\sigma}{g(\rho_l - \rho_v)}} \right]^{0.33} Pr_l^s \quad (2.1)$$

where s is usually equal to 1 for water and 1.7 for the other fluids, and C_{sf} is a constant depending on the liquid-surface combination. Eq. (2.1) is actually a correlation arranged in the dimensionless form. The left-hand side is Jakob Number, representing the ratio of sensible heat to latent heat released during the phase-change process. On the right-hand side, q/L is the mass flow rate of vapor per unit area. In the visualized studies of bubble dynamics, the bubble departure diameter is often found proportional to $\sqrt{\frac{\sigma}{g(\rho_l - \rho_v)}}$ [20-22]. Therefore, the second term on the right-hand side of Eq. (2.1) can be regarded as Reynolds Number taking $\frac{q}{L\rho_v}$ as characteristic velocity and bubble departure diameter as characteristic length. Eq. (2.1) can be then converted into the form of dimensionless numbers, $Ja = C_{sf} Re_b^{0.33} Pr_l^s$. The correlation has been proved validations for the different types of traditional coolants on the various flat or modified

surfaces [23-29]. Pioro [27] evaluated the Rohsenow constant C_{sf} of water and ethanol on the four flat substrates with the materials of Copper, Aluminum, Brass, and Stainless steel under the atmosphere and sub-atmosphere conditions. The results were systematically compared with the other studies. Kwark et al. [30] found that Rohsenow's correlation was valid on both the flat surface and the Al_2O_3 nanoparticle coated surface in the pressure range of 20-200 kPa, with the results shown in Fig. 2.1. The boiling curves of both the plain copper surface and nano-coated surface at various pressures fitted well with Rohsenow's correlation, while the Rohsenow constant C_{sf} is equal to 0.0128 for all four prediction curves in Fig. 2.1.

Besides the NBHT coefficient, CHF is another crucial parameter in pool boiling. Zuber is a pioneer who studied the role of bubble dynamics in the CHF [31, 32]. A correlation was derived to predict the CHF using the hydrodynamic instability. He demonstrated that single bubbles became to separated bubble columns near boiling crisis (CHF). The CHF occurred when these bubble columns started interfering with each other. Based on the theory, the mass flow rate of the total vapor column in boiling crisis is given by,

$$m = v_{cri}\rho_v A_v \quad (2.2)$$

where A_v is the area of vapor columns. v_{cri} is the critical velocity of bubble columns, equals to $\sqrt{2\pi\sigma/\rho_v\lambda_H}$, where λ_H is the Helmholtz wavelength. For the saturated pool boiling, the heat is utilized for the liquid-vapor phase change. Therefore, the critical heat flux could be calculated from,

$$q_{CHF} = v_{cri}\rho_v L(A_v/A_p) \quad (2.3)$$

where A_p is the projective area of the heater surface. Zuber related the Helmholtz

wavelength to the Taylor wavelength by $\lambda_H = (\pi/2)\lambda_c$, in which $\lambda_c = 2\pi[\frac{\sigma}{g(\rho_l - \rho_v)}]^{1/2}$, and applied these parameters into Eq. (2.3, the final format is expressed to the widely used Zuber's correlation,

$$q_{CHF} = 0.131\rho_v L [\frac{\sigma g(\rho_l - \rho_v)}{\rho_v^2}]^{1/4} \quad (2.4)$$

Fig. 2.1 Boiling curves in pure water with uncoated and nano-coated heater surfaces under various system pressures (reprinted from kwark et al. [30]).

Liu et al. [33] proved the validation of Zuber's correlation on plain surface in water at the system pressure range of 7.4-101 kPa. The predicted CHF values matched their results with a maximally relative error of 24%. Lee et al. [34] further managed to extend the working range of Zuber's correlation to over 1000 kPa in water.

The experiments were also conducted in the subcooled conditions, which refers to a liquid temperature below its saturation temperature. The contradictory results have been reported regarding the subcooling effect on the NBHT coefficient. For instance, Goel et al. [35] reported that NBHT coefficient was an increasing function of

subcooling in the subcooling range of 5-20 °C, however, Zhang et al. [36] found an inverse trend in which the observed NBHT coefficient deteriorated with an increase of subcooling. Moreover, an independent relationship between the NBHT coefficient and subcooling was reported by the researchers [37-39]. We can see that the contradictory trends are attributed to the combination of two opposite effects. With increased subcooling, the microconvection heat flux and the bubble departure frequency are increased, which improves the NBHT coefficient. Meanwhile, the bubble departure diameter is reduced, which limits the latent heat dissipated by nucleation bubbles. Thus, the improved, decreased, and unaltered NBHT coefficient can be found with increasing subcooling depending on the relative magnitude of the two opposite effects, which is a function of working fluid, surface characteristics and the other experimental conditions. Unlike NBHT coefficient, the CHF was observed to increase almost linearly with subcooling [39-41]. The correlations which estimate the relationship between the subcooling and the CHF have a general form which can be written as,

$$q_{CHF,sub} = q_{CHF,sat}(1 + C\Delta T_{sub}) \quad (2.5)$$

Determining a suitable value of C by fitting the experimental data has been conducted in the previous studies. For example, Kutateladze, one of the pioneers in this topic, performed the study and evaluated the value of C from the experimental data of water, alcohol and isooctane [40],

$$C = 0.065 \left(\frac{\rho_l}{\rho_v} \right)^{0.8} \frac{c_{p,l}}{L} \quad (2.6)$$

Following Zuber's correlation used in predicting CHF on an infinite flat surface at the saturation condition, an additional heat flux was added for the subcooling effect to heat

up the subcooled liquid to saturation. By applying the hydrodynamic instability theory [41], a correlation of subcooling effect on CHF is derived as,

$$C = 5.32 \frac{(k_l \rho_l C_{p,l})^{1/2}}{\rho_v L} \left[\frac{\rho_v^2 g (\rho_l - \rho_v)}{\sigma^3} \right]^{1/8} \quad (2.7)$$

The correlations in the form of Eq. (2.5) are widely applied in predicting pool boiling, however, they must be specialized for the certain fluids and operation range. The difficulties to reach a universal correlation derivation mostly come from the following factors: most correlations were derived by the empirical or semi-empirical methods with lack of the physical basis; since the pool boiling is a complicated heat transfer phenomenon which contains many influencing parameters, it is almost impossible to incorporate all the parameters in a single correlation from the limited series of the experimental data; additionally, unrepeatable experimental design and unknown measurement uncertainties reduce the data reliability between different research groups. A reliable universal prediction method cannot be provided at the stage. More attention can be paid to the theoretical part in the pool boiling studies. The method of computational fluid dynamics deserves more investigations since it is an option of the reliable prediction method. Moreover, the recent developed machine learning can be a tool of data analysis to cover most experimental parameters and help establish reliable models.

2.1.1.2 Submerged and confined jet impingement boiling

Cardenas and Narayanan [13] conducted the submerged jet impingement boiling experiments in both saturated and subcooled water in the pressure range of 17.6-47.8 kPa. They firstly studied the pool boiling case ($Re = 0$) and gradually increased jet

velocity to $Re = 14000$. The jet velocity was found to have positive effect on the heat transfer coefficient (HTC) at the early stage of nucleate boiling. However, when nucleate boiling was completely established, the HTC became insensitive to the jet velocity. The trend was even more obvious in the subcooled water. Consequently, the Rohsenow's correlation could also predict the submerged jet impingement boiling within an acceptable accuracy. On the other hand, a larger Re could extend the nucleate boiling regime, leading to improved CHF.

Zhao et al. [42] observed the same trend as that by Cardenas and Narayanan [13] in a confined jet configuration. Both smooth and structured-porous surfaces were investigated in a single slot. For both surfaces, the NBHT coefficient was independent of jet velocity in the fully developed nucleate boiling regime. The independence was attributed to the dominated role of the NBHT in this regime. Since the jet velocity was considered to influence the heat convection only, its effect on the total HTC became negligible when the NBHT was dominated. Besides the jet velocity, the experiments of subcooling suggested that the NBHT coefficient had an increasing function of fluid subcooling degree. However, a convergence tendency of boiling curves at the different subcooling degrees was observed at high heat flux. It was then indicated that both jet velocity and subcooling had negligible effect on the NBHT coefficient when boiling was fully established. On the other hand, a higher jet velocity or subcooling degree could effectively extend the boiling curves, delay the transition from nucleation boiling to film boiling, and improve CHF accordingly.

Recently, researchers tended to use multiple jets instead of single one. As we know, the

HTC is highest in the stagnation zone and diminishes fast in lateral direction. By using multiple jets, the overall HTC for a heating surface can be enhanced and the surface temperature distributes more uniform [14, 43]. Brún et al. [44] conducted experiments at atmospheric pressure to study confined jet array impingement boiling. Same as the single jet, the NBHT coefficient of jet arrays was also insensitive to jet velocity once boiling was fully established. On the other hand, the ONB was delayed, and the CHF was improved as jet velocity became larger. The similar research was further investigated on a linear micro-groove surface and a radial micro-groove surface [45]. It was found that the radial micro-groove surface achieved the best overall boiling performance and both micro-grooved surfaces provided at least 3-fold maximum HTC over that on the plain copper surface. Besides, the effect of jet velocity on the micro-grooved surfaces was consistent with that on the plain copper surface, that is, the NBHT coefficient was insensitive to jet velocity in the fully developed nucleate boiling regime.

2.1.2 Dielectric fluids

2.1.2.1 Pool boiling

The direct electronic cooling requires the direct contact of the heating devices and coolants, the dielectric properties are essential for working fluids in the submerged boiling configurations [46]. The Freon-type refrigerants were regarded as ideal coolants for the submerged boiling configurations in the past studies, since they have a lot of desired properties such as dielectricity, inert chemical property, stability, low boiling point, non-flammability, etc. However, they are regarded as the main sources to trigger

the global warming effect. The Freon-type refrigerants have gradually been replaced by Fluorocarbon (FC-72, FC-87 and PF-5060) and Novec (Novec 7000, Novec 7100 and Novec 7300) fluids. The kinds of coolants keep the benefits of Freon-type refrigerants while they provide additionally extremely well environmental properties as well. However, compared to water, the dielectric fluids are relatively poor on thermophysical properties, e.g., latent heat. Fluorocarbon and Novec fluids are highly wetting fluids which result in an obvious higher boiling incipient superheat and a high tendency for leakage [47-49]. The onset of nucleate boiling requires generation of nucleation cavities which harbor residual vapor or non-condensable gas. However, the highly wetting fluids would deeply penetrate the cavities, preventing the cavities from trapping residual vapor and thus delaying the boiling incipience. Moreover, in the increasing heat flux experiments of highly wetting fluids, the surface temperature near ONB would become very high and decrease suddenly once boiling is initiated before following the typical nucleate boiling curve. However, the sudden drop does not exist when decreasing the wall heat flux. Therefore, there is normally a hysteresis phenomenon existing in the boiling curves of Fluorocarbon and Novec dielectric fluids, while it is rarely observed in the traditional coolants.

Evaluation of the Rohsenow constant C_{sf} for Freon-type refrigerants (including R-11, R-12, R113 and R-114) has been performed [50-52]. Jabardo et al. [53] investigated the different cylindrical surfaces using R-11, R-123, R-12 and R134a. The roughness was controlled for the substrate surfaces. It was found the Rohsenow constant C_{sf} relied on surface roughness and then a C_{sf} correlation was derived as a function of liquid-surface

combination, dimensionless reduced pressure, and dimensionless surface roughness Ra .

The correlation agreed with the reported experimental data within an acceptable range.

The Fluorocarbon and Novec fluids, produced by 3M company, have been widely investigated in submerged boiling configurations. Schlindwein [54] investigated the pool boiling on a copper disc with FC-72, FC-87 and their binary mixtures. The overall NBHT coefficient of FC-87 was 20% higher than that of FC-72 under the same experimental condition. The binary mixtures of FC-87 and FC-72 were found to deteriorate the overall heat transfer performance as they were compared to the relative pure coolants. The worst heat transfer performance was found as the molar fractions of FC-87 was 50%. The results were also compared with some the pool boiling correlations, while the Rohsenow constant C_{sf} was 0.0065 for FC-72 or 0.0079 for FC-87, respectively. In FC-72, Ho et al. evaluated the C_{sf} of silicon and AlSi10Mg at 0.000146 and 0.0000373, respectively [55, 56]. From the values of C_{sf} , it can be easily observed that the copper surface performs much better than the other plain surfaces in FC-72. The C_{sf} of Novec 7000 and copper combination was evaluated by Bartle et al. [57]. Unlike the traditional coolants and the Freon-type refrigerants, the studies on C_{sf} evaluation of Fluorocarbon and Novec dielectric fluids are very limited. As the data from different researchers have large disparities, it is almost impossible to get a generally accepted value of C_{sf} based on those experimental data. The reason is that the experimental conditions are not identical and C_{sf} is actually a function of many parameters, e.g., surface roughness, contact angle, liquid properties, etc. To derive a generally accepted value of C_{sf} , all the influencing parameters should be identified at

first. Moreover, the other influencing parameters should be controlled strictly the same when studying the effects of each parameter. Once the effects of all the influencing parameters are extracted from C_{sf} , a universal constant will be remained naturally.

The effect of liquid subcooling was studied by some researchers using dielectric fluids as coolant. For example, Watwe et al. [58] investigated the subcooling effect on a plastic pin-grid-array (PPGA) surface with FC-72. The fully developed nucleate boiling curves of different subcooling level (0-65 °C) were found to collapse onto a single line, indicating an irrelevant relationship between the subcooling and the NBHT coefficient. Other researchers also reported the same trend when FC-72 was applied as the working fluid [59-63]. Rainey [59] explained the insensitivity of NBHT coefficient on subcooling by the combined effects of increased natural convection and Marangoni convection, and decreased microconvection and latent heat, as the subcooling degree was increased. However, this irrelevant relationship was valid only when the boiling surface was a smooth one. Different results were reported on modified surfaces. Rainey et al. [60] found that the subcooling could effectively influence the fully developed NBHT coefficient on a microporous, square pin-finned copper surface. Cao et al. [64] also observed an increase function of NBHT coefficient with subcooling on nanoparticle-deposited pin-fin surfaces. Liu et al. [61] investigated micro/nanostructured surfaces processed by femtosecond laser with different processing spacings in pool boiling. Different trends were found on the surfaces with different geometry parameters. For smooth surface, increased subcooling shifted the boiling curves to the left, indicating an improved NBHT coefficient. However, boiling

curves of micro/nanostructured surfaces under different subcooling levels merged into one. They explained that increased subcooling could lead to two opposite effects. On the one hand, subcooling could enhance natural convection hence shifted the boiling curve to the left. On the other hand, subcooled coolants suppressed bubble departure, which deteriorated the NBHT coefficient. For the smooth surface, there were only limited nucleation sites. Consequently, the positive effect of increased subcooling was more significant than the negative effect and the boiling curves was shifted to the left with increased subcooling. As for the micro/nanostructured surfaces, the two opposite effects of increased subcooling were comparable hence the boiling curves under different subcooling levels collapsed onto one. Their experiments further certified that the subcooling effect highly depended on the boiling surface. El-Genk and his group investigated the subcooling effect in another type of dielectric fluid, Novec 7100 [65, 66]. The experiments were conducted on smooth copper surface and porous graphite surface, respectively. On the porous graphite surface, the boiling curves were shifted to the left when fluid subcooling was increased, indicating an improved NBHT coefficient. In contrast, the experimental results on smooth copper surface did not show a clear trend in the subcooling range of 0 to 30 °C. In all the experiments, 30 °C subcooling and the saturation case showed highest and lowest NBHT coefficient, respectively. However, 10 °C subcooling had a better boiling performance compared to 20 °C subcooling. Based on the above discussion, it can be easily found that the subcooling effect on the NBHT coefficient is highly dependent on the liquid-surface combination. It cannot be arbitrarily concluded that subcooling is an improvement method of nucleate

boiling. The liquid-surface combination must be taken into consideration. On the other hand, the CHF in dielectric fluids for pool boiling was observed as well to increase almost linearly with the subcooling degrees [67-69]. Arik et al. [69] further proved this linear relation in the pressure range of 101.3-303.9 kPa. Rainey et al. [60] found that their experimental data followed the relationship of Inoue's correlation. However, the constant in Eq. (2.5), given as follows, were modified by fitting their data,

$$C = C_{Rainey} \left(\frac{\rho_l}{\rho_v} \right)^{3/4} \frac{C_{p,l}}{LPe^{1/4}} \quad (2.8)$$

where

$$Pe = \frac{\sigma^{3/4}}{\alpha \rho_v^{1/2} [g(\rho_l - \rho_v)]^{1/4}} \quad (2.9)$$

and C_{Rainey} equals to 0.7 for plain surface and 1.2 for microporous surface.

Different from subcooling, the effect of pressure on the NBHT coefficients is consistent for the different working fluids. The NBHT coefficient is an increasing function of pressure. Watwe et al. [58] showed that the NBHT coefficient increased with system pressure, however, the surface temperature at a fixed heat flux did not drop as the pressure was increased. The reason was the larger saturation temperature in a higher system pressure. Rainey et al. [60] reported similar result and introduced correlations for both plain and microporous surface to predict the heat flux at a given wall superheat and system pressure.

2.1.2.2 Submerged and confined jet impingement boiling

The effects of jet parameters or liquid properties on the boiling performance also vary with the experimental conditions in the submerged or confined jet impingement boiling.

The experiments of R-113 by Ma and Bergles demonstrated that the boiling incipience was independent of the jet velocity [70]. At the subcooling degree of 18.5 °C, boiling initialed at the wall superheat of 34 °C for various jet velocities ranging from 0 m/s to over 11 m/s, although a case at around 1 m/s having the early boiling incipience was considered the result of the uncontrolled experimental conditions. Zhou et al. [49] conducted single slot submerged jet impingement boiling experiments of R113 and PF-5052. The experiments started from the pool boiling case ($v_j = 0$ m/s) and gradually increased the jet velocity to $v_j = 11.355$ m/s. Their investigations indicated the boiling incipience decreased with an increase of the subcooling degree regardless of the jet parameters. The decreased boiling incipience resulted from the better entrapment of residual vapor due to a larger surface tension induced by increasing the subcooling degree. However, Browne et al. [71] observed the different trend regarding the effect of jet velocity on nucleation incipience by conducting the confined jet impingement experiments of the staggered microjet array in R134a. It was found that the boiling incipience was delayed by an increase in the jet velocity, as explained by the visualization study with boiling initialed at the secondary stagnation region. The increased jet velocity improved the convection heat transfer, reduced the wall superheat, and caused the delayed bubble nucleation.

As mentioned in *Section 2.1.1*, the investigations in submerged water jet have revealed that both jet velocity and subcooling have the negligible effect on the NBHT coefficient when boiling was fully established [13, 42, 72]. However, the experimental results in dielectric fluids are quite different from those in water. Browne et al. [71] found the

improved NBHT coefficient with the increase of subcooling and jet velocity in R134a. The similar trend was reported by Zhang et al. [73], who conducted single slot confined jet impingement experiments in the subcooled FC-72 on both plain silicon chip and micro-pin-finned surfaces. For both smooth surface and micro-pin-finned surfaces, the boiling curves were shifted to the left side as the subcooling or jet velocity was increased. The change in bubble dynamics was considered for accounting for the increased NBHT coefficient. When the fluid subcooling was increased, the nucleation bubbles departed at a smaller diameter or a higher frequency, leading to a larger space for heat convection. Apart from the subcooling and jet velocity, the effect of nozzle-to-surface distance was also investigated. The nozzle-to-surface distance is normally non-dimensionalized by a ratio of nozzle-to-surface distance and nozzle size (H/d or H/W). The experiments of two distances at $H/d = 1$ and $H/d = 2$ found that a smaller nozzle-to-surface distance ($H/d = 1$) could delay the ONB and enhance the NBHT coefficient and CHF. The trend was related to different heat transfer distributions at various nozzle-to-surface distances. For small H/d , the local heat transfer coefficient at the stagnation zone was very high and decreased rapidly at downstream (distance from the stagnation point in lateral direction), while the heat transfer coefficient on the heating surface distributed much more uniformly for larger H/d . The large local heat transfer coefficient at the stagnation point was considered as the main reason of the increased average heat transfer coefficient for the small H/d . It was then confirmed in the subsequent study of Zhang et al. [74].

The jet parameters and liquid properties seem to affect the results notably regarding the

effect of jet velocity on the NBHT coefficient in the dielectric fluids. Cardenas and Narayanan [11] studied the submerged jet impingement boiling with various jet velocities in FC-72 at $H/d_j = 6$. The boiling curves at various jet velocities was found to merge into a single one in the fully developed nucleate boiling regime. As same as their experiments of water [13], they attributed the independent relation between the jet velocity and the NBHT coefficient to the dominant role of bubble nucleation when boiling was fully established. In other words, the bubble nucleation was so powerful in this condition that the forced convection heat transfer induced by the jet could be negligible under $H/d_j = 6$. The effect of nozzle diameter was illustrated in Fig. 2.2. For $Re = 3700$ (Fig. 2.2(a)), the NBHT coefficient was relatively independent of jet diameter while CHF increased slightly with decreasing jet diameter. However, in $Re = 7100$ (Fig. 2.2(b)), both the NBHT coefficient and CHF decreased with a larger jet nozzle. It could be seen that the kinetic energy was reduced for a larger jet at fixed Re . On the other hand, the downward drag force induced by the jet could suppress bubble departure. A larger jet would apply this drag force to a larger impingement area, resulting in deteriorated heat transfer coefficient. For $Re = 3700$, the downward drag force was not as strong as that in $Re = 7100$. Thus, the effect of jet diameter in $Re = 3700$ was not so significant.

Fig. 2.2 Effect of nozzle diameter at different Re numbers (redrawn from Cardenas and Narayanan [11]).

The effect of jet velocity on the NBHT coefficient was also found to depend on nozzle-to-surface distance in the research of Shin et al. [15], who conducted confined jet impingement experiments in 25 °C subcooled PF-5060 at various jet velocities and nozzle-to-surface distances. As shown in Fig. 2.3, the fully developed NBHT coefficient was independent of jet velocity at $H/W = 4.0$. However, when H/W decreased to 1.0 and 0.5, the jet velocity could significantly affect the fully developed NBHT coefficient. In other words, the effect of jet velocity became more significant with the decreased H/W . Besides, they investigated the effect of H/W at the stagnation point and the downstream point ($x/W = 1.5$). The results confirmed with that of Zhang et al. [73, 74]. At the stagnation point, the NBHT coefficient at small H/W was significantly higher than that at larger H/W . The higher NBHT coefficient resulted from the eliminated vapor bubbles removed by strong jet momentum at the stagnation point. On the other hand, the NBHT coefficients at the downstream point ($x/W = 1.5$) were nearly the same at different H/W . The experimental results further proved that the

increased overall NBHT coefficient at the small H/W was only a result of the improved local heat transfer coefficient at the stagnation point. Moreover, CHF was also found to be a function of H/W . To avoid the worst case, a correlation was proposed to predict the nozzle-to-surface distance with the lowest CHF at various mass velocity,

$$\frac{H}{W_{lowest\ CHF}} = 1.4016 \log(G)^2 - 9.4548 \log(G) + 16.493 \quad (2.10)$$

in which the dimension of G was neglected in Ref. [15].

Fig. 2.3 Boiling curves for various jet velocities at different nozzle-to-surface spacings (reprinted from Shin et al. [15]).

The confined jet impingement experiments of Clark et al. [12] using Novec 7100 found

that both jet velocity and nozzle-to-surface distance had negligible influence on the NBHT coefficient at their experimental conditions. The major results are shown in Fig. 2.4, in which Fig. 2.4(a) illustrates the boiling curves at various jet velocities when H/d was fixed to 3, while Figs. 2.4(b) and (c) show the boiling curves of different nozzle-to-surface distances at high jet velocity ($v_j = 9$ m/s) and low jet velocity ($v_j = 1$ m/s), respectively. The jet did enhance the single phase convection, but the boiling curves at the different velocities merged into one when boiling was completely established. The results confirmed the dominant role of bubble nucleation in the confined jet impingement boiling when boiling was fully established. The experiments are also a good supplementation for the work of Cardenas and Narayanan [11, 13], who had similar finding in the different working fluids.

The CHF of the submerged/confined jet impingement boiling was found to consistently increase with subcooling in the dielectric fluids, similar to that in pool boiling. Zhang et al. [73] explained that the improved CHF was a result of the change in bubble dynamics. For a larger subcooling degree, the bubbles departed at a smaller diameter or higher frequency. The smaller bubbles increased the difficulty of bubble layer formation, resulting in delayed CHF. Copeland [75] supported the increased CHF with subcooling in confined multiple jets impingement in the working fluid of FC-72. The results well recalled the correlation of Mudawar and Wadsworth [76], who demonstrated that the CHF increased by around 2% for every increase degree in subcooling.

Fig. 2.4 Boiling curves and h - q curves at different jet velocities and nozzle-to-surface distances (reprinted from Clark et al. [12]).

A higher jet velocity was reported to be an effective way to improve CHF in the submerged and confined jet impingement boiling by a lot of researchers [15, 49, 71]. A higher jet velocity could provide a better liquid supplementation. However, Mudawar

and Wadsworth [76] observed that there existed a transition point dividing the jet velocity into two regimes as they investigated the effect of confined jet velocity in the range of 1-13 m/s by using the FC-72 working fluid. Below 7 m/s, the CHF increased with jet velocity, however, above this transition point, the CHF decreased as the jet velocity increased.

Similar results are reported in the studies of jet array impingement boiling. Copeland [75] investigated the boiling performance of both smooth and pin-finned surfaces in a confined jet array using FC-72 as coolant. The jet velocity showed the same effect on the CHF for both surfaces, that is, CHF was enhanced as the jet velocity became larger. Besides, the jet array showed higher CHF than the single jet and the improvement in CHF was proportional to the number of jets. Meyer et al. [77] systematically studied impingement boiling of an array of three confined rectangular jets in FC-72 and ethanol. The boiling curves for different jet velocities and nozzle sizes tended to converge while increasing jet velocity, nozzle size or inlet subcooling could delay both ONB and CHF. Similar results were reported by Hong et al. [78] and Mira-Hernández et al. [79], who also found that the jet velocity had negligible effect on NBHT coefficient when boiling was fully established. In addition, they found the boiling curves for different nozzle-to-surface distances also converged. As a contrast, Hong et al. [78] reported that liquid subcooling and jet-to-jet spacing could effectively influence the NBHT coefficient in the entire boiling regime (from ONB to CHF). On the other hand, increasing jet velocity and liquid subcooling could extend the nucleate boiling regime hence enlarge CHF while there existed an optimum value to achieve highest CHF for nozzle-to-surface

distance and jet-to-jet spacing. Rau and Garimella [80] conducted confined jet impingement boiling experiments in Novec 7100 with three different jet configurations. H/d was fixed at 4 and the Re ranged from 1920 to 39400. The local surface temperature was measured via IR thermography of a thin-foil heat source backed by a magnesium-fluoride window. They found the overall heat transfer performance increased with the number of nozzles, same as that concluded by Copeland. However, the distribution of surface temperature was found totally different with that in other literature. The jet arrays showed larger nonuniformity in local surface temperature compared to the single jet in their experiments, while surface temperature was found to distribute more uniform in jet arrays than a single jet in other papers [14, 43]. Rau et al. [81] then experimentally compared the confined jet impingement boiling and pressure drop of a single jet ($d_j=3.75$ mm) with that of a 5×5 jet array ($d_j=0.75$ mm) on four different heating surfaces, i.e., flat copper, microporous copper, pin-finned copper and microporous pin-finned copper. The jet array was found to perform better than the single jet on the macroscale enhanced surfaces (pin-finned copper and microporous pin-finned copper). However, on the flat and microporous surfaces, the heat transfer performance of the single jet and jet array were similar. This result is different with their previous finding in [80]. One possible reason is the different experimental designs. In reference [80], the heater source is only a thin foil, which can be considered as non-conjugated condition while the experimental setup in reference [81] is a conjugated one. For pressure drop, it was found that the jet array yielded a slightly higher operating pressure drop than the single jet, for all the four testing surfaces. Generally, it can be concluded based on their

studies that the combination of macroscale enhanced surfaces and jet array could effectively improve heat transfer performance with a slight increase in pressure drop, while it is unworthy to replace single jet by jet arrays in flat surface and microscale enhanced surfaces.

According to the investigations in traditional coolants and dielectric fluids, it can be found that the submerged jet in a pool is able to enhance heat convection. However, the proportion of heat convection among the overall heat transfer is still unidentified. Sometimes, the bubble nucleation is so powerful that the convection heat transfer only accounts for the limited portion in the overall heat transfer, the NBHT coefficient is therefore independent of jet velocity. And the pool boiling correlations can be directly applied in the submerged/confined jet impingement boiling. In the other cases that convection heat transfer cannot be ignored, the NBHT coefficient in the whole heat transfer regime should be a function of jet velocity. In *Section 2.1.1*, it is inferred that an independent relation between the fully developed NBHT coefficient and the jet velocity for water, which indicates that water is the former type having bubble nucleation as the dominant heat transfer mechanism. For the dielectric fluids, on the other hand, contradictory results are reported even in the same fluids, e.g. Zhang et al. [73] and Cardenas and Narayanan [11] in FC-72. Thus, the proportion of convection heat transfer in dielectric fluids must be a function of the dominated experimental parameters, for instance, the dimensionless nozzle-to-surface distance given by Shin et al. [15]. Further experiments of submerged/confined jet impingement boiling should be performed, as the supplemental work to the present available but scattering

experimental data. The guideline of experiment design is to build a database for identifying all influencing parameters and understanding their relation.

2.1.3 Nanofluids

Nanofluids refer to suspensions containing nanoparticles (with size below 100 nm normally) which are uniformly and stably mixed with the base fluid such as water, organic fluids and refrigerants [82]. Nanofluids were initially investigated in pool boiling by Yang and Maa [83] in 1984 and have extensively been studied since 2003. In the last decade, the nanofluids have rapidly become to a hot research focus in pool boiling due to their significant enhancement in the boiling performance. We are to introduce the typical investigations on nanofluids covering metal-oxide, metallic and non-metallic nanoparticle materials, nanoparticle sizes and concentrations using both water and refrigerants as the base fluids.

You et al. [84] investigated Al_2O_3 /water nanofluids with a low concentration range from 0 g/l to 0.05 g/l in pool boiling study. They reported that nanoparticles had negligible effect on the NBHT coefficient. However, the CHF was improved dramatically by around 200% for the nanofluids containing 0.05 g/l nanoparticles compared with that of pure water. Hu et al. [85] investigated the effect of nanoparticle size in nanofluids pool boiling. Different from the results of You et al. [84], the NBHT coefficient was increased by adding nanoparticles into the base fluid in their experiments. They found the NBHT coefficient was improved when the nanoparticle diameter was decreased in the range of 84-120 nm. The maximum NBHT coefficient was improved by around

200% and 40% for 84 nm nanoparticles and 120 nm nanoparticles, respectively. In addition, an optimum volume concentration of 0.25% was suggested based on their experimental data.

Liu et al. [33] researched carbon nanotube (CNT) suspended water in pool boiling. They prepared CNT suspensions with the mass concentration between 0.5 wt.% and 4.0 wt.%, and conducted pool boiling experiments in the system pressure of 7.4 kPa, 20 kPa and 103 kPa. The results revealed that both NBHT coefficient and CHF were improved by adding non-circular CNT particles. A mass concentration of 2.0 wt.% for CNT was found to provide the best overall performance. Additionally, the pressure could not only influence the NBHT coefficient at a certain mass concentration, but also significantly affect the nanoparticle-induced enhancement ratio of the NBHT coefficient. At the sub-atmosphere pressure, the NBHT coefficient enhancement ratio induced by the presence of CNT suspensions was more significant.

Since the ferrous nanoparticles are widely used in nanofluids, the effect of external magnetic field has been investigated. Abdollahi et al. [86] applied the magnetic field on the Fe_3O_4 -water nanofluids for the potentiality of improving pool boiling heat transfer by the external magnetic field. The NBHT coefficient was found to increase by 6% and 13% in the presence of negative magnetic field for pure water and 0.1% volume concentration nanofluids, respectively. However, the presence of a positive magnetic field would deteriorate the boiling heat transfer. Li et al. [87] studied the effect of magnetic field on the pool boiling in the Fe_2O_3 -water nanofluids in the concentration range of 0.005-0.1 g/l, and found the effect of magnetic field was highly depended on

the nanoparticle concentrations. For example, in the N-N magnetic field, the NBHT coefficient increased by 28% in the 0.05 g/l volume concentration nanofluid, while it decreased by 20% when the volume concentration is 0.025 g/l.

Trisaksri and Wongwises [88] suspended the TiO_2 nanoparticles into R141b refrigerant for investigating the effects of nanoparticle concentration and pressure on the pool boiling heat transfer. Experiments were performed at three nanoparticle concentrations of 0.01, 0.03 and 0.05 vol%. While the nanoparticle concentrations were small (0.01 vol% for example), the NBHT coefficient of nanofluids was found to remain almost constant with that of pure R141b in the pressure range of 200-500 kPa. However, the overall NBHT coefficient deteriorated with the further increase in nanoparticle concentration. The increased pressure was found to shift the boiling curves to the left side, indicating an improved overall heat transfer performance. The improved overall heat transfer performance became not so significant at the high nanoparticle concentration (0.05 vol% for example) as that at the low nanoparticle concentrations. The results conformed to the work of Naphon and Thongjing [89], who also conducted pool boiling experiments in the TiO_2 -R141b nanofluids under different pressures of 50, 100 and 150 kPa. The NBHT coefficient in a low nanofluid concentration was as same as that in pure R141b and decreased with the increase in nanoparticle concentration. The investigation is a good agreement of the results of Trisaksri and Wongwises [88].

However, Tang et al. [90] observed a different trend when $\delta\text{-Al}_2\text{O}_3$ was used as nanoparticles in R141b. The conducted pool boiling experiments at three different volume concentrations of 0.001, 0.01, and 0.1 vol% showed the NBHT coefficient was

increased in low concentration nanofluids (0.001 vol% and 0.01 vol%) but deteriorated in high concentration nanofluids (0.1 vol%) with a reference of pure R141b. By adding a surfactant called as the sodium dodecyl benzene sulphonate (SDBS), the deterioration at 0.1 vol% was eliminated. Following the study, Diao et al. [91] researched the nanoparticle concentration of Cu-R141b nanofluids with SDBS surfactant. According to their research, the NBHT coefficient increased with an increasing of the nanoparticle concentration in the range of 0-0.05 vol%. However, a deteriorated CHF was observed by adding Cu nanoparticles into pure R141b. This observation is quite noticeable since the nanofluids are expected to enhance CHF based on the previous investigations.

The heat transfer enhancement mechanism of nanofluids was studied. Kim et al. [92] found the significant enhancement in CHF using different water-based nanofluids with the nanoparticles of alumina, zirconia, and silica in low concentration. The increased surface wettability of nanoparticle deposition was considered for the CHF enhancement. Kwark et al. [93] investigated the mechanism of nanoparticle deposition and showed that the nanoparticle deposition was induced by the bubble microlayer evaporation as illustrated in Fig. 2.5. During the bubble nucleation, the base fluid in the microlayer evaporated and left the nanoparticles to be deposited to the heated surface and formed the porous deposition layer. The CHF increased with an increase in the nanoparticle concentration till an optimum deposition thickness, beyond which, a higher nanoparticle concentration would not improve CHF anymore and deteriorate the heat transfer coefficient due to the increase in thermal resistance of the thick deposition layer. Besides the modification of surface characteristics induced by the nanoparticle

deposition, the change in bubble dynamics is the other widely observed heat transfer enhancement mechanism of nanofluids. Tu et al. [94] observed smaller vapor bubbles in Al_2O_3 -water nanofluids. Xu and Xu [95] found the similar results in the flow boiling. Wang et al. [96] applied pool boiling to investigate the single bubble dynamics in nanofluid containing moderately hydrophilic nanoparticles and showed that the single bubble departure diameter was not influenced by adding the moderately hydrophilic nanoparticles in pure water. However, the bubble departure frequency was significantly increased. Using the same type of nanofluids in the pool boiling experiments, Quan et al. [97] studied multiple bubble dynamics in a much larger surface. The results demonstrated that the size of the multiple bubbles relied on both single bubble departure diameter and bubble coalescence. The moderately hydrophilic nanoparticles could effectively inhibit bubble coalescence; thus, the bubble size was decreased although the single bubble departure diameter remained constant. Generally, the two heat transfer enhancement mechanisms of nanofluids are not incompatible, they normally perform together to improve the boiling performance in the nanofluids.

Fig. 2.5 Mechanism of particle deposition during boiling process (redrawn from Kwark et al. [93]).

Although several researchers reported different trends, the nanofluids could improve

the boiling performance in most literature. The boiling performance of nanofluids depended on base fluid, nanoparticles (materials, sizes, and concentrations), and experimental conditions (pressure and external fields). The modification of surface characteristics induced by nanoparticle deposition and the change in bubble dynamics were the main heat transfer enhancement mechanisms of nanofluids. To our best knowledge, nanofluids have rarely been studied in the submerged or confined jet impingement boiling. There were only reports on the single-phase submerged jet impingement in nanofluids [98, 99], which are in the scope of the present two-phase boiling study. For flow boiling of nanofluids, Lee and Mudawar [100] observed severe nanoparticle deposition and clustering phenomena in the channel flow of nanofluids. Therefore, the clustering effect and pressure drop issue in the jet channel must be solved if the nanofluids are supposed to be applied in submerged jet impingement boiling for the electronic cooling.

2.2 Effects of Surface Characteristics

The methods to enhance the submerged boiling have become a key issue in the related studies. In *Section 2.1*, we introduce the parameters which are able to influence the boiling heat transfer, including pressure, subcooling, nanoparticles, jet velocity, jet diameter, and nozzle-to-surface distance. In this part, we introduce the other important and widely used parameter, surface characteristic. It attracts most attentions in the submerged boiling studies since it is an effectively passive method which improves the boiling performance by altering the nucleation bubble dynamics. The effects of

roughness and contact angle are discussed first, followed by the effects of surface size, surface orientation and surface aging. The effect of surface structures is discussed at last.

2.2.1 Effects of roughness and contact angle

The bubble nucleation in the submerged boiling is dominated by the surface-liquid interaction. Surface roughness and contact angle are two major factors which influence the surface-liquid interaction and thereby the submerged boiling heat transfer performance [101]. However, contact angle can be affected by surface roughness but not reverse. Modification of surface roughness is normally accompanied by a change of contact angle. The studies of surface roughness are always performed in the presence of modified contact angles. Accordingly, these two factors are discussed together below.

Jones et al. [102] studied the roughness effect on a polished aluminum surface in water and FC-77 and found different effects of surface roughness on the different working fluids. As shown in Fig. 2.6, the NBHT coefficient of FC-77 increases with an increase in surface roughness. For water, on the other hand, the roughest surface and smoothest surface have the highest and worst NBHT coefficient, respectively. The surfaces with the intermediate roughness have almost the same heat transfer performance. The disagreeable trends in water and FC-77 could not reach a correlation including both. Thus, a new parameter was suggested to replace Ra for characterizing surface roughness.

Fig. 2.6 Boiling curves for water and FC-77 on a set of surfaces with various surface roughnesses, where EDM stands for Electrical Discharge Machined surface. (redrawn from Jones et al. [102]).

Kim et al. [103] explored the impact of surface roughness on a superhydrophilic surface. The different rough aluminum surfaces were prepared by reacting with hot water to form aluminum oxide superhydrophilic coating. The measured contact angle decreased from 8.5° for $Ra = 0.11 \mu\text{m}$ to 2.7° for $Ra = 2.93 \mu\text{m}$. The NBHT coefficient was found to increase with an increase in the surface roughness because of the nucleation site increase. On the other hand, the CHF initially increased from 150 W/cm^2 for $Ra = 0.11 \mu\text{m}$ to 215 W/cm^2 for $Ra = 0.35 \mu\text{m}$ and then remained constant for further surface roughening. They continued the study of surface roughness on a hydrophobic surface, achieved by coating a thin polytetrafluoroethylene (PTFE) layer on the unidirectional polished copper surface [10]. Different from those on the moderate and superhydrophilic surfaces, the contact angle increased with an increase in surface roughness on the hydrophobic surface. In addition, the pool boiling heat transfer performance showed a different trend. The NBHT coefficient of a rougher surface was

initially higher than that of a smoother one but decreased rapidly to a very low value in the moderate and high heat flux regime. The CHF, on the other hand, decreased with increasing surface roughness. The trend was opposite to the previous studies. It was inferred that the different effect of surface roughness was the result of the early formation of vapor blanket on the hydrophobic surface.

The above mentioned experiments tried different methods to eliminate the disturbance from surface wettability in studying surface roughness effects. However, it is impractical to make sure every single test section has identical contact angle during the experiments. Gong and Cheng [104] numerically investigated the pure effect of surface roughness in pool boiling using water as working fluid. Their simulation work was based on a liquid-vapor phase-change lattice Boltzmann method. The effect of surface roughness was explored on both a hydrophilic surface and a hydrophobic surface. The results showed that rougher surfaces shifted the boiling curves to the left for both hydrophilic and hydrophobic surfaces, indicating an earlier ONB and improved NBHT coefficient. The improvement was a result of increased number of nucleation sites and higher bubble departure frequency. Besides, the improvement of NBHT coefficient on rougher surface was more significant for hydrophobic surface than that for hydrophilic surface.

Different from the surface roughness, the contact angle has been studied independently due to the development of nano-scale surface treatment methods. The surface treatment methods are able to change surface wettability significantly while keeping the surface roughness variation in nanometer scale, which was reported to have negligible effect

on the pool boiling [105]. Generally, the hydrophilic surface with low contact angle would improve CHF, but delay ONB and deteriorate the NBHT coefficient, while the hydrophobic surface has contradictory effect [106-108]. Recently, the attention starts to be shifted to combine the hydrophilic and hydrophobic patterns in one surface and design the heterogeneous wetting or biphilic surface, as shown in Fig. 2.7(a) [109]. Jo et al. [110] fabricated hydrophobic Teflon dots on a hydrophilic SiO₂ substrate to obtain biphilic surface. The biphilic surface was found to improve the NBHT coefficient significantly and keep CHF at a comparable value with that in hydrophilic surface. The improvement of NBHT coefficient was reported as a function of hydrophobic dot distribution. Betz et al. [109] extended the study of biphilic surface to superhydrophilic and superhydrophobic combination as shown in Fig. 2.7(b). It was revealed that the superbiphilic surface could enhance the NBHT coefficient dramatically to 100 kW/m²·K and remain CHF at a relatively high value. In addition to the experiments, numerical works were also performed to investigate the biphilic surface in pool boiling. For example, Gong and Cheng [104] simulated the pool boiling on a biphilic surface with multi-cavities and compared its heat transfer performance with that of a pure hydrophilic surface and that of a pure hydrophobic surface, both having the same surface structure as the biphilic surface. Their simulation work was based on a liquid-vapor phase-change lattice Boltzmann method. In their study, the biphilic surface exhibited the best overall heat transfer performance among the three testing surfaces. Based on the same numerical method, Ma et al. [111] investigated the biphilic heat sinks with micro-pillars. Two different biphilic surfaces were included in their study,

i.e., a bio-inspired one (hydrophobic top with hydrophilic bottom) and a mixed wettability one (hydrophilic top with hydrophobic bottom). A pure hydrophilic heat sink and a pure hydrophobic heat sink were also investigated for comparison. The bio-inspired one was found to have the best overall heat transfer performance among the four testing heat sinks. Besides, separated pathways of liquid and vapor flow were observed on the bio-inspired heat sink at low wall superheat.

Fig. 2.7 SEM images for (a) biphilic surface and (b)superbiphilic surface (reprinted from Betz et al. [109]).

The correlations have been developed to predict the NBHT coefficient at the different surface roughnesses. As mentioned in *Section 2.1.1*, the Rohsenow constant C_{sf} is determined by the surface-liquid interaction. In other words, it should be a function of surface roughness and contact angle. Kim et al. [112] developed a correlation for water which included the parameter Ra in C_{sf} ,

$$\frac{1}{C_{sf}} = 84.3 \left(\frac{Ra}{Ra_0} \right)^{0.109} \quad (2.11)$$

where Ra_0 is the representative surface roughness for typical metal surfaces which equals to $0.4 \mu\text{m}$. Li et al. [105, 113] incorporated the effects of surface roughness, contact angle, and effective heat transfer area ratio r (the ratio of the actual heat transfer

area and its projected area) into the Rohsenow constant C_{sf} by analyzing a semi-analytical model developed by Benjamin and Balakrishnan [114],

$$C_{sf} = 0.013C_s^{-0.33} \quad (2.12)$$

in which

$$C_s = r(1 - \cos\theta)^{0.5} \left[1 + \frac{5.45}{(Ra - 3.5)^2 + 2.61} \right] \gamma_{wl}^{-0.04}, \theta = \max(\theta, 15^\circ) \quad (2.13)$$

where γ_{wl} is the material influencing parameter of heating wall and working fluid, which equals to $\gamma_{wl} = (k_w \rho_w C_{p,w}) / (k_l \rho_l C_{p,l})$. The experiments were then conducted, showing a good agreement with the prediction in Eq. (2.12).

In addition to the NBHT coefficient, the predictions of the contact angle and the surface roughness on CHF have also been investigated. By applying the force balance analysis on a single bubble, Kandlikar [115] proposed a correlation as,

$$q_{CHF} = L\rho_v^{1/2} [\sigma g(\rho_l - \rho_v)]^{1/4} \frac{1 + \cos\theta_{rec}}{16} \left[\frac{2}{\pi} + \frac{\pi}{4} (1 + \cos\theta_{rec}) \cos\phi \right]^{1/2} \quad (2.14)$$

The correlation can well predict saturated pool boiling CHF for water, refrigerants, and cryogenic liquids with surface orientations from 0° (horizontal surface) to 90° (vertical surface). It should be noticed that the contact angle was receding contact angle. This gravely limits the application of Kandlikar's correlation due to the large uncertainty during receding contact angle measurement. Therefore, the receding contact angle has been suggested to be replaced by the static contact angle while a modification factor S is added to fit the experimental data. For instance, Kim et al. [112] found the surface roughness could improve CHF by taking the capillary wicking effect into account. After the capillary force was incorporated in Kandlikar's force balance analysis, a correlation

was then given as,

$$q_{\text{CHF}} = SL\rho_v^{1/2}[\sigma g(\rho_l - \rho_v)]^{1/4} \frac{1+\cos\theta}{16} \left[\frac{2}{\pi} + \frac{\pi}{4}(1 + \cos\theta) + \frac{4K\cos\theta}{1+\cos\theta} \left(\frac{Ra}{S_m} \right) \right]^{1/2} \quad (2.15)$$

where K is a constant and S_m is the mean spacing of surface roughness. The data fitting recommended a value of 0.811 and 87.8 for S and K , respectively. The correlation is valid for horizontal surface with moderate wettability (around 65°) and surface roughness Ra from 0.041 μm to 2.36 μm .

The studies of roughness effect on the submerged and confined jet impingement boiling are still limited. Cardenas and Narayanan [13, 72] conducted boiling experiments in a submerged jet on the surfaces with two distinct roughness of 33 nm and 123 nm. The contact angles were 53° and 66° for the 123 nm and 33 nm roughness surface, respectively. The results showed that the surface roughness had negligible effect on the NBHT coefficient in both partial and fully developed nucleation regime. The trend was same as that in the pool boiling, which also reported that the change of surface roughness in nanoscale had no effect on the NBHT coefficient [105]. However, the CHF was found to be significantly higher on the roughness surface of $Ra = 123$ nm when in comparison with the surface of $Ra = 33$ nm.

Generally, the effect of surface roughness was found to depend on the surface wettability. Different trends were reported on the hydrophilic and hydrophobic surfaces [10, 103, 112]. The effect of contact angle investigated by different researchers was quite consistent. The hydrophilic surface with a low contact angle would improve CHF, but delay ONB and deteriorate the NBHT coefficient, while the hydrophobic surface

had the opposite trend. The heterogeneous wetting or biphilic surface, which had the intermittent hydrophilic and hydrophobic regions in one surface, was developed to combine the benefits of hydrophilic and hydrophobic surfaces. The correlations were developed to incorporate the effects of surface roughness and contact angle. However, the correlations were only applicable under the specified conditions due to the complexity of boiling and the differences in working fluid, the surface preparation method, and the other experimental conditions.

2.2.2 Effects of surface size, surface orientation and surface aging

Since electronics varied in size, it is of great importance to find out the relation between the cooling performance and the surface size in submerged boiling for the application purpose. The effects of surface size on the submerged boiling have been reported in some literature. For example, Kwark et al. [30] investigated heater sizes at $7.5 \times 7.5 \text{ mm}^2$, $10 \times 10 \text{ mm}^2$, $15 \times 15 \text{ mm}^2$, and $20 \times 20 \text{ mm}^2$ on the plain and Al_2O_3 nanoparticle coated surface on the water boiling. The results showed that the NBHT coefficient was not affected by the heater size on the plain surface. For the nano-coated surface, the NBHT coefficient decreased with an increase in the heater size at the low to moderate heat flux conditions but became independent of heater size at the high heat flux case. The results were consistent with those of Park and Bergles [116]. Wang et al. [117] reported a different trend by conducting experiments in subcooled FC-72 on the silicon chips with the areas of $20 \times 20 \text{ mm}^2$ and $10 \times 10 \text{ mm}^2$. The NBHT coefficient of the smaller heater was significantly higher than that of the larger one, regardless of the heat flux. The

results were supported by Baker [118].

CHF is found to decrease as the heater size increases in the most studies. For example, Kwark et al. [30] found the decreased CHF in larger heaters on both plain and nano-coated surfaces. Lu et al. [119] and Mori et al. [120] also found similar result on the silicon wafer and porous copper surface, respectively. It is worth mentioned that CHF was found to increase with the heater size under microgravity in the studies of Wang et al. [117]. The reason of the opposite trends might be a result of the different heat transfer mechanism of various heater sizes under the microgravity condition.

As mentioned in *Section 2.1.1*, Zuber proposed the correlation for predicting CHF on the basis of the hydrodynamic instability theory [32]. Lienhard and Dhir [121] suggested that the Helmholtz wavelength should be equal to the Taylor wavelength,

$$\lambda_H = \lambda_D = \sqrt{3}\lambda_c \quad (2.16)$$

It is known that Zuber's correlation was developed for infinite surface. The heater size should be at least larger than the Helmholtz wavelength (e.g., $\lambda_H = \lambda_D = 2.5$ cm for water). However, some experiments were conducted in a heater size smaller than the Helmholtz wavelength due to the power supply limitation. Lu et al. [119] attempted to validate Zuber's correlation for heater size smaller than 2 cm in water. A factor was used to fit A_v/A_p and reduced the Helmholtz wavelength to the heater size. The modified Zuber's correlation was found to predict CHF on the various heater sizes in an acceptable accuracy.

The surface orientation was reported to have influence on both NBHT coefficient and

CHF in pool boiling. Typically, the surface orientation is characterized by the inclination angle which equals to 0° for upward direction and 180° for downward direction. Kwark et al. [30] explored the impact of surface orientation on both a plain copper surface and the Al_2O_3 nanoparticle coated surface for water pool boiling. For both surfaces, they found that the NBHT coefficient increased with increasing inclination angle at a low heat flux. However, at the high heat flux regime, the boiling curves for different inclination angles gradually merged into a single one. The result was consistent with the observation of Nishikawa et al. [122] and Kim et al. [123]. Moreover, Lienhard [124] explained that this trend of surface orientation effect was a result of the transition from single bubble regime to vapor column regime. In other words, the NBHT coefficient was a function of surface orientation only before single bubbles merged to columns. Jun et al. [39, 125] studied the orientation effect on the sintered copper microporous coated surface in the saturated and subcooled water and compared the results with those on plain copper surface. For the plain copper surface, the trend was similar to that of Kwark et al. [30]. However, on the microporous surface, the inclination angle was found to have negligible effect on the NBHT coefficient in the heat flux regime at both saturated and subcooling conditions. The independent relation was explained by the early formation of vapor columns on the microporous surface. Rainey and You [126] also reported the effect of inclination angle on the plain surface and microporous surfaces. However, it was also suggested that the NBHT coefficient could be reduced as the heat transfer resistance increased as a result of the formation of a large bubble layer for the inclined surface. In the submerged boiling,

bubbles depart from the heating surface due to buoyancy. For inclined surfaces, especially the downward facing surface ($\phi > 90^\circ$), the buoyancy-induced bubble departure is blocked by the heating surface, causing aggregation of bubbles and formation of the large bubble layer (see Fig. 2.8). El-Genk and Suszko [127] and Kim et al. [128] reported that NBHT coefficient decreased with increasing inclination angle in the range of 0-180° as PF-5060 and water were tested. The similar results were also reported by Dadjoo et al. [129], who conducted experiments in both water and SiO₂-water nanofluids with different concentrations in the inclination angle range of 0-90°.

Fig. 2.8 Bubble dynamics on the microporous-coated surface and plain surface under various surface orientations (redrawn from Jun et al. [125]).

On the other side, CHF was found to be a constant or change slightly in the inclination angle range of 0-90°, and then decrease with further increase in inclination angle. An accelerated decline of CHF was stated when the incline angle was greater than 170° [39, 125]. Kim and Suh [130] summarized this transition point was a boundary which

distinguished the near-vertical surface and the downward facing surface. For the decreased CHF between 90° and 180° , the blockage of bubble departure by the heated surface was considered as the main reason.

Joudi and James [131] found that the uncertainty of experimental data in the nucleate pool boiling was a result of “time-in-use” of the heating surface, relative to surface aging. Zhou et al., who studied the effect of surface aging in submerged jet impingement boiling of PF 5020 [49], compared the results at two immersion times at 1 and 10 hours. It was shown that the boiling curves shifted to the left side as the immersion time increased from 1 to 10 hours, indicating an improved h with the surface aging. The observation was supported by Ma and Bergles [70], who conducted the submerged jet impingement experiments in R113. The different trend was reported by Zhao et al. [42] in the water-based confined jet impingement boiling experiments. They found the NBHT coefficient deteriorated with the surface aging. The deterioration was thought as a result of the surface corrosion in high-temperature water. Based on the above discussion, the effect of surface aging is highly related to working fluids.

The effects of surface size, surface orientation and surface aging were discussed. The effects of these surface parameters depend on the boiling configuration, working fluid and other experimental conditions. Most of the investigations regarding the surface effects are performed in pool boiling, while a little part of them is performed in the submerged or confined jet impingement boiling configuration. The factor should be considered in the electronic cooling design.

2.2.3 Effect of surface structures

Modification of the surface with structures is one of the most attracting area for the boiling enhancements, which have been widely applied in the electronic cooling. The surface treatment methods can produce a variety of scales including macroscale, microscale, and nanoscale, as well as hybridscale which combines at least two of these scales. The surface modification configurations include but are not limited to fins/microfins, porous/microporous structures, surface roughening, tunnels and reentrant cavities, nanotube, nanofiber, nanowire, and nanoparticle deposition. These methods of surface modification can improve boiling performance by changing surface roughness and contact angle, increasing effective heat transfer area, modifying bubble departure diameter and frequency, providing more active nucleation sites, and/or improving capillary wicking effect [132-134]. Several review articles regarding this topic can be referred. For example, Honda and Wei [135] reviewed the microscale surface structures developed for improving heat transfer performance in the electronic devices. Kandlikar and his team [136, 137] reviewed the nanoscale surface modification methods and manufacturing techniques of porous surfaces for pool boiling. Mori and Utaka [138] summarized the surface modification methods to enhance the critical heat flux in pool boiling. Liang and Mudawar [48] systematically reviewed the surface modification effects of pool boiling. Besides the experimental methods, pool boiling of modified surfaces was also investigated numerically by Cheng and his group [104, 111, 139]. For submerged/confined jet impingement boiling, modified surfaces with structures were tested under various experimental conditions, for both single jet and jet

arrays [42, 45, 73, 75, 81, 140-143]. These articles can provide us comprehensive understanding of surface modification methods in submerged boiling. In this section, we will emphasize the latest progress of nanostructures in submerged boiling studies. Besides, we would also like to introduce a novel surface structure which improves boiling heat transfer not by the above mentioned mechanisms, but by separating flow pathways of liquid and vapor.

In submerged boiling studies, nanostructures possess the following desired properties: high surface area to volume ratio, smaller volume and mass, and high convective heat transfer coefficient. The nanostructures are developed rapidly in submerged boiling with the improvements in nanotechnology. For example, Jun et al. [144] coated copper-plated electrospun nanofibers on the copper platelets by electroplating. The nanofiber-coated surface and a bare copper surface were compared in pool boiling experiments using ethanol and water as working fluids. Results showed that the heat transfer coefficients of nanofiber-coated surface were 6–8 times higher and 3–5 times higher than those of bare copper surface in the nucleate boiling of ethanol and water, respectively. On the other hand, the two surfaces showed almost coincident CHF which agreed well with the existing theoretical result. Although the nanofiber coating seemed incapable of improving the CHF, it should be noticed that the corresponding wall superheat at CHF for the nanofiber-coated surface was 10 °C lower than that for the smooth surface. Therefore, it could be concluded that the nanofiber coating was able to benefit the entire nucleate boiling regime, from ONB to CHF. Li et al. [145] deposited copper nanorods on a polished copper substrate by oblique-angle deposition in an

electron-beam evaporator with base pressure of 10^{-7} mm Hg. The visual observation revealed that the nanorods deposition could significantly alter the dynamics of bubble nucleation, including smaller bubble diameters, higher bubble release frequencies, and an approximately 30-fold increase in the density of active bubble nucleation sites. As a result, the copper substrate with nanorods deposition had higher NBHT coefficient than the copper substrate without nanorods deposition. Moreover, the wall superheats at the ONB and CHF were also significantly lower for the nanorods-deposited surface. Chen et al. [146] noticed that the unique properties of nanowires could potentially be exploited to promote boiling performance. They fabricated Si and Cu nanowires by electroless etching (EE) technique and electroplating technique, respectively. The surfaces with Si and Cu nanowires coatings were compared with the flat Si surface in pool boiling experiments. It was found that the nanowires-coated surfaces could improve both the NBHT coefficient and the CHF by more than 100% compared to the flat Si surface, irrespective of the material of nanowires. They attributed the improvements to the unique properties of nanowires, such as high nucleate site density, superhydrophilicity, and enhanced capillary pumping effect. Dharmendra et al. [147] coated vertically aligned carbon nanotubes (CNTs) on copper substrate using a double-layered structure deposition method with diamond as the intermediate layer. The diamond intermediate layer was supposed to improve the adhesion between the CNTs and the copper substrate. Both the diamond and the CNTs were coated by the hot filament chemical vapor deposition (HFCVD) technique. The experimental results in pool boiling studies revealed that the boiling curve of the CNT-coated copper surface

shifted leftward compared to the bare copper surface, indicating an enhanced NBHT coefficient and an earlier ONB. Besides, a significant enhancement of 38% in the CHF was also observed on the CNT-coated surface. In addition to the abovementioned techniques, nanoparticle deposition using nanofluids is also commonly applied as the nanofabrication method. The details have been described in Section 2.1.3, therefore it is unnecessary to repeat here.

The concept of separated liquid-vapor flow pathways was proposed by Liter and Kaviani [148]. They systematically investigated the theories and experiments of the pool boiling CHF and suggested that there were two independent mechanisms of CHF: (i) the modified Zuber's hydrodynamic instability limit, (ii) the viscous-drag limit. For the first mechanism, the critical heat flux occurred when the bubble columns started interfering with each other, thus the flow became "unstable". The second limit occurred when the viscous drag within the liquid supply artery surpassed the available capillary-gravity pumping. Each mechanism represented one interruption method of liquid replenishment, i.e., liquid-choking. In real applications, the actual CHF mechanism depended on occurring of liquid-choking mechanism firstly, in other words, the lower of the two liquid-choking limits was predicted to be the critical heat flux for a given system. The different porous-coated surfaces were fabricated for the experiments. The results showed that the hydrodynamic liquid-choking limit was significantly lower than the viscous-drag liquid-choking limit, thus all surfaces were limited by hydrodynamic instability. The experimental data fitted well with the predictions of hydrodynamic liquid-choking limit. For the pool boiling, vapor was found to escape by buoyancy into

the liquid pool thus the liquid supply and vapor escape restricted each other's motion. The liquid-vapor counterflow was regarded as the trigger mechanism of the hydrodynamic liquid-choking. Thus, to remove the hydrodynamic liquid-choking limit, the liquid-vapor counterflow must be eliminated. Following the idea, a system was proposed which completely separated liquid and vapor flow pathways. As shown in Fig. 2.9, the liquid phase is sucked in by the cover layer which also acts as a preventer for vapor escaping. Under the cover layer, there was a porous structure with the escape pathway for nucleation bubbles. The liquid and vapor flow pathways were completely separated and the liquid-vapor counterflow did not exist. Therefore, the hydrodynamic liquid-choking would not occur and the critical heat flux in such a surface was limited by viscous-drag liquid-choking, which could theoretically achieve an order of magnitude enhancement of CHF compared to the plain surface. Regrettably, they only proposed the concept but not really fabricated this surface. The surface fabrication and experiments were finished by the subsequent researchers.

Fig. 2.9 Artery-evaporator system with completely separated liquid and vapor flow pathways (redrawn from Liter and Kaviany [148]).

Bai et al. [149] proposed a porous artery structure to achieve separate liquid and vapor flow pathways. As shown in Fig. 2.10(a), the porous coating would draw liquid in while the vapor escape pathway was provided by rectangular arteries below it. The results of water experiment are illustrated in Fig. 2.10(b). A maximum heat flux of 610 W/cm^2 could be achieved in the porous artery structure surface, which was still far from the actual CHF. The further increase in heat flux was limited by the high temperature of the copper heater. Although CHF was not reached, the achievable heat flux had been almost five times higher than the prediction value of Zuber's correlation and other experimental results in smooth copper surface. The separate flow of liquid and bubble was clearly observed in the visualization study. They then increased the maximum heat flux to 805 W/cm^2 in their follow-up study [150]. However, it was still not the CHF, and a better heater design was required to further approach the actual CHF.

Fig. 2.10 (a) Schematics of the porous artery structure. (b) Boiling curves of smooth and porous artery structure surfaces (redrawn from Bai et al. [149]).

In addition to the porous artery structure, the separate liquid and vapor flow could be also achieved by the other surface structures. For instance, Raghupathi and Kandlikar [151] designed a cooper surface with a groove filled with epoxy, a thermal insulation material with low conductivity, and observed the organized liquid flow and vapor

escape pathways in the selectively thermal insulated surface. Their design utilized evaporation momentum force which was an interfacial force during liquid-vapor phase change induced by sudden expansion of the evaporating liquid. The evaporation momentum force pushed the bubble outwards and had no net effect along the heater surface because it was systematic in all directions. However, the zero net force would be unbalanced as the temperature distribution of the liquid-vapor interface became asymmetric. Thus, the bubble would be pushed to the hotter side. In the design of Raghupathi and Kandlikar, the liquid above the copper surface had a significantly higher temperature compared to the liquid above epoxy. The evaporation momentum force experienced on the copper side was greater than that on the epoxy side. Bubbles generated at the edge of the copper surface and the epoxy would move towards the copper side to create separate liquid-vapor flow pathways on the heating surface. Rahman et al. [152] also observed distinct liquid flow and vapor escape in the selectively thermal insulated surfaces. They fabricated multiple grooves on the copper surface and filled the grooves with epoxy. As shown in Fig. 2.11, the epoxy-filled grooves acted as the liquid replenishing channels while the copper surface acted as pathways for bubble departure. By increasing the groove numbers on the copper base, the NBHT coefficient and CHF could be improved by maximum 400% and 100%, respectively.

Fig. 2.11 (a) Samples of selectively thermal insulated surfaces. (b) Schematic of the separate liquid-vapor flow pathways induced by selectively thermal insulated surfaces (redrawn from Rahman et al. [152]).

Li et al. [153] separated liquid and vapor flow pathways by a modulated porous structure which had a porous base with four porous pillars on the top of the base. Zhang et al. [154] showed the separate liquid and vapor flow in the three-dimensional thin wall grid structures. Additionally, more surface microstructures were designed separate liquid and vapor flow pathways, including contoured fin structure [155], open microchannels [156, 157], open microchannels with selective porous coatings [158-160], and feeder microchannels [161]. Fig. 2.12 illustrates the open microchannels with porous coatings designed by Jaikumar and Kandlikar [158]. The open microchannels with porous coating throughout the surface are shown in Fig. 2.12(a), while the porous layer is selectively sintered on the fin tops and channels as demonstrated in Figs. 2.12(b) and (c). All the three surface structures showed the improved heat transfer performance

in comparison with the plain surface, but the enhancement mechanisms were different. For the sintered-throughout structure, there were no organized liquid supply and vapor escape pathways, the enhancement was a result of increased surface area and active nucleation sites. However, for the two selectively sintered structures, there existed distinct pathways for liquid flow and vapor removal. The sintered porous layer acted as the vapor removal pathway. In the sintered-channels structure, the CHF of 420 W/cm^2 was achieved at a wall superheat of $1.7 \text{ }^\circ\text{C}$ using water as working fluid. Kandlikar [162] further explored the cooling system to remove $500\text{-}700 \text{ W/cm}^2$ heat flux at wall superheat of under $5 \text{ }^\circ\text{C}$ by using water. The liquid supply was proposed and is shown in Fig. 2.12(b) and (c), acting as an impinging jet. It provides us a novel view which is the submerged jet can be regarded as a method which separates the liquid replenishing and vapor escape. To generate a submerged jet, the traditional jet generation system including a nozzle, a pump, and other auxiliary devices, may be not compulsory. An elaborate surface structure can also create submerged jet in pool boiling without the assistance of the jet generation system. For example, Ma et al. [111] observed orderly separation of liquid and vapor flow pathways on a biphilic microchannel surface consisting of hydrophobic fin tops and hydrophilic channels in their simulation study.

Fig. 2.12 Heat transfer mechanisms of open microchannels with different porous coating (redrawn from Jaikumar and Kandlikar [158]).

Recently, Yuan et al. [163] mounted a guidance structure above the heating surface in order to separate the pool boiling flow into upward vapor slug and liquid plug two-phase flow inside the guidance structure and downward replenishing liquid flow in the annulus between the container and the guidance structure. Two different configurations of guidance structure, a straight guidance tube and a trumpet guidance tube, were investigated in saturated water under saturation temperature of 50 °C, 70 °C and 100 °C. A setup without the guidance structure was studied under the same experimental conditions for comparison. The three different types of test section and their flow patterns are illustrated in Fig. 2.13. Due to the promotive vapor bubble removal and liquid replenishment, both guidance tubes achieved higher NBHT coefficient and CHF

compared to that without the guidance structure, especially under low saturation temperature. For example, the CHF of the trumpet guidance tube at saturation temperature of 50 °C was 109 W/cm², which was 3.1 times larger than that of the one without the guidance tube. In general, the guidance structure designed by Yuan et al. [163] provides us a novel idea to achieve separated liquid-vapor flow pathways. In addition, this guidance structure can be applied together with surface structures to further separate the liquid and vapor flow hence amplify the pool boiling enhancement, which make the design even more attractive.

Fig. 2.13 Designs of the pool boiling flow: (a) original design; (b) design with a straight guidance tube; (c) design with a trumpet guidance tube. (reprinted from Yuan et al. [163]).

2.3 Summary

In this chapter, the heat transfer performance of pool boiling, submerged jet impingement boiling and confined jet impingement boiling in various working fluids (water, dielectric fluids and nanofluids) were compared and summarized for the penitential applications in the electronic cooling. The heat transfer mechanism, influence factors, and prediction tools of the various submerged boiling configurations

were comprehensively discussed. The effects of surface characteristics including surface roughness, contact angle, surface size, surface orientation, surface aging, and surface structures with separate liquid-vapor flow pathways were highlighted. According to the reviewed literature, it can be found that considerable progress has been achieved for the research in the submerged boiling systems. Nevertheless, a comprehensive understanding of the heat transfer mechanism and generally accepted correlations are still lacked. In particular, the experimental results for some influence factors are often reported to be divergent or even contradictory, e.g., the effects of surface roughness. Therefore, in the next chapter, we will develop a roughness parameter with higher accuracy through a specific surface filtration process to characterize the surface roughness precisely hence identify the impacts of surface roughness on the pool boiling and establish reliable analysis methods and prediction tools.

Chapter 3 Pool boiling heat transfer of saturated water on rough surfaces with the effect of roughening techniques[†]

3.1 Introduction

As reviewed in *Section 2.2.1*, surface roughness and contact angle are the two factors influencing the submerged boiling heat transfer most. Although the surface roughness has been extensively investigated for several decades, its effect on the submerged boiling is still not fully understood. The roughened surface is usually believed to enhance boiling by providing more nucleation sites and/or larger effective heat transfer area [105, 164, 165]. Therefore, the surface roughness is normally characterized by arithmetic mean roughness Ra or effective heat transfer area ratio r in the previous studies. Chu et al. [133] systematically investigated the pool boiling on surfaces with the effective heat transfer area ratio r ranging from 1.79 to 5.94. Based on their experimental results, the CHF increased monotonously with r due to roughness-amplified capillary pumping. On the other hand, the NBHT coefficient did not show a clear trend with the effective heat transfer area ratio. Moita et al. [166] also found that the surface with larger r could not assure a higher NBHT coefficient. They demonstrated that larger r could indeed increase the liquid-solid contact area, hence

[†] This chapter has been reproduced in its entirety with permission as an academic work from the following publication:

S. Fan, L. Jiao, K. Wang, F. Duan, Pool boiling heat transfer of saturated water on rough surfaces with the effect of roughening techniques, *International Journal of Heat and Mass Transfer*, 159 (2020) 120054.

improving the nucleate boiling. However, sometimes it also caused the early formation of vapor blanket due to the severe interaction mechanism and coalescence effect of the nucleation bubbles. This phenomenon would deteriorate the NBHT coefficient and even overcome the benefits of the increased liquid-solid contact area. Compared to the effective heat transfer area ratio r , the arithmetic mean roughness Ra is used more frequently to characterize the surface roughness. Many researchers visualized increased number of nucleation sites on surfaces with larger Ra , which is believed responsible for the enhancement of the NBHT coefficient [102, 103, 105]. However, some different observations are also reported. Benjamin and Balakrishnan [167] found the number of nucleation sites was not consistently increased with Ra . It decreased at first and then increased, as the Ra became larger. Kozitskii [168] indicated that the NBHT coefficient increased with larger Ra up to a point ($Ra = 0.95 \mu\text{m}$), beyond which no improvement occurred. Similar trend was also reported by Jabardo [169] and El-Genk and Suszko [127].

The major obstacle to fully understand roughness effect on pool boiling comes from the large scattering in the reported data. Generally, these inconsistently experimental data are considered as a result of various surface roughening techniques, roughness measurement uncertainties and different experimental conditions such as working fluid, surface material and surface wettability. As shown in Fig. 2.6, Jones et al. [102] observed totally different roughness effects on the NBHT coefficient in the two working fluids, even though the other experimental conditions were controlled the same. Kim and his group studied the effects of surface roughness on the pool boiling on a

superhydrophilic [103], a moderate-wetted [112] and a hydrophobic surface [10], respectively. On the superhydrophilic and moderate-wetted surfaces, the NBHT coefficient was enhanced as the surface roughness increased, even though several cases exhibited deviation from the overall trend. However, the experiments on the hydrophobic surface showed totally different results. On the hydrophobic surface, the NBHT coefficient of a rougher surface was initially higher than that of a smoother one but decreased rapidly to a very low value in moderate and high heat flux regime, leading to an early occurrence of CHF. They explained that such trend was resulted from the early formation of vapor blanket on the hydrophobic surface, same as that demonstrated by Moita et al. [166].

The surface roughening techniques are essential in studying the roughness effects on pool boiling. Researchers have emphasized that the surface roughness is meaningless without specifying the surface roughening technique. Berenson [170] found the lapped surface had higher HTC than the emery-polished one, although the lapped surface was smoother. Vachon et al. [23] prepared surfaces by unidirectionally polishing and chemically etching. The heat transfer coefficients of surfaces with these two surface roughening techniques were different even though the Rq (root-mean-square roughness) was similar. Other researchers also pointed out the necessity of using a consistent surface roughening technique when investigating the roughness effect in pool boiling [171, 172]. The reason is that surfaces prepared by some roughening techniques have higher tendency to bubble nucleation, regardless of the surface roughness. Besides, the standardized surface roughness parameters, such as Ra and Rq , are statistical average

values which cannot accurately indicate the actual surface texture, especially the number of nucleation cavities. A smooth surface indicated by small Ra or Rq can also have a lot of cavities for nucleation. Many researchers reported the inadequacy of using an individual statistical parameter to characterize the surface roughness for pool boiling [165, 172, 173]. In fact, Berenson believed that a “rougher” surface in pool boiling should be that has more cavities necessary for nucleation, but not the one with a larger value of Ra or Rq . Some researchers developed methods to measure the active cavity size, in order to determine the potential nucleation sites quantitatively [174-176]. However, their methods are based on the simplistic treatment of the surface texture and wetting phenomenon. The accurate quantitative prediction of the nucleation sites is still unrealizable. Researchers still need to characterize the surface roughness with the aid of standardized roughness parameters.

This chapter aims to investigate the effect of surface roughness on copper surfaces. The experiments were conducted in the saturated water at atmospheric pressure. A total of eight testing surfaces were prepared by three different surface roughening techniques, respectively. The different surface roughening techniques could introduce different features of surface texture. These features could be effectively eliminated by applying a novel pre-processing method before surface roughness measurement. The trends in the NBHT coefficient and CHF were discussed, and the bubble dynamics were studied using a high-speed camera in order to help understand the boiling performance. Based on the experimental data, a correlation including the effect of surface roughness was proposed by using the areal roughness parameter Sa .

Fig. 3.1 (a) Schematic of the whole apparatus, (b) detailed drawing of the heating block.

3.2 Experimental setup and procedures

3.2.1 Experimental setup

The schematic diagram of the experimental setup is shown in Fig. 3.1(a). The whole apparatus consists of a boiling chamber, a testing part, a temperature measurement system, and a visualization system. The boiling chamber has a transparent side wall and top cover made of Acrylic in order to perform visualization study. The baseplate of the boiling chamber is made of Teflon for the thermal insulation purpose. Experiments were conducted in this chamber using distilled water under atmospheric pressure. A pressure gauge is connected to the chamber in order to monitor the internal pressure. A copper coil, together with water bath, is performed as a vapor condenser. The condenser can condense the vapor generated in the chamber and maintain the liquid level constant. Four immersion heaters are installed inside the boiling chamber to provide heat to the

liquid. These immersion heaters are connected to a proportional-integral-derivative (PID) controller, which can control the liquid temperature by detecting it with a K-type thermocouple. In this experiment, the liquid temperature is kept at around 99.5 °C. In the testing part, a copper block is surrounded by the thermal insulation material. The testing surface on the top of the copper block has a dimension of 20×20 mm². The copper block is heated by four cartridge heaters with the capability of providing a maximum heat flux of 150 W/cm². Three thermocouples with hot junction diameter of 0.5 mm are aligned 4 mm, 10 mm, and 16 mm from the testing surface in the center of the copper block with holes of 0.6 mm, as shown in Fig. 3.1(b). During the experiments, the bubbles were prior to generate at the edge of the testing surface. Methods of eliminating the edge nucleation sites had been explored. The one shown in Fig. 3.1 is proved to effectively reduce the edge nucleation sites. The gap between the copper and the Teflon holder is first filled with RTV sealant and then covered by a fluidic thermal insulation material 3M Epoxy DP 460. There are still nucleation sites at the edge using this method due to the different thermal expansion rate of the copper, RTV sealant and the epoxy. But this method could effectively reduce the number of the edge nucleation sites to an acceptable range. The temperature measurement system includes four K-type thermocouples, a data acquisition system (Agilent 34972A LXI Data acquisition Unit) and a computer. All the thermocouples were calibrated using a dry block calibrator (Techne Ucal 400+) with an accuracy of 0.2 °C. The visualization system includes a LED light source (Leica CLS150 LED), a high-speed camera (Fastec Imaging HiSpec 2G Mono) and a computer. Bubbles at the boiling surface were captured and recorded

by the visualization system at a frame rate of 2000fps during the experiments.

Fig. 3.2 Measurement results of the surface topography and surface wettability.

3.2.2 Surface preparation

The modified surfaces in this experiment were prepared by two different methods.

Before applying the surface preparation methods, the surface was first randomly polished with 2400 grit sandpaper (Struers Silicon Carbide Grinding Paper) and cleaned

by sonicating in distilled water. The treated surface served as base surface in the experiment. The base surface was then roughened by unidirectional polishing with rougher sandpapers (80, 120, 180, 600, and 1200 grit) or femtosecond laser processing. During the femtosecond laser processing, the laser emitted pulses of 130 fs with linearly polarized light at a central wavelength of approximately 795 nm. The nominal repetition rate is 1 kHz. The total pulse energy was attenuated by a rotating half wave. The mechanical shutter was controlled to release the desired laser on the substrate. The laser beam was focused and controlled by Scanlab galvanometer scanner. The average laser power after the lens was measured using a power meter. Two base surfaces were processed by the femtosecond laser with different scan speeds. In the end, a total of eight samples were prepared, including five unidirectionally polished surfaces, two laser processed surfaces and a base surface as reference. Two important surface characteristics, surface roughness and static contact angle, were measured on the testing surfaces. The surface roughness was measured by a 3D optical surface metrology system, Leica DCM-8. The measurement results are illustrated in Fig. 3.2. It can be found that the unidirectionally polished surfaces have a lot of irregular scratches, while the femtosecond laser processed samples have periodically fluctuating surface texture as shown in Figs. 3.2(g) and (h). Based on the specification standards from ISO (international Organization of Standardization), the raw measured data are defined as surface topography which contains all the surface features, while what we concerned is only part of them, i.e., surface roughness. Therefore, the following preprocessing is applied before analyzing the surface roughness.

Fig. 3.3(a) is the measurement of raw surface. It can be found that some bad points compress the surface dynamic, impeding us from obtaining detailed information on the surface. This can be seen more clearly on the depth distribution of the measured surface (Fig. 3.3(b)). A few numbers of deep holes cover most of the depth range (from 20.3 μm to 50.9 μm), while the main body of the surface (over 99.9% number of point) locates only at 0-20.3 μm . This phenomenon is an inherent drawback of the non-contact instrument, as explained by the manufacturer. A spatial filter operator, with a median denoising filter 3×3 , could eliminate the bad points (Step 1). The surface was then leveled by subtracting the least square plane from the whole surface area (Step 2). The leveled surface (Fig. 3.3(d)) is a rectangular one. A square area was extracted from the leveled surface for better analysis (Step 3). During the measurement, some data points were not measured due to various reasons. These missing points were filled in using a smooth shape calculated from the neighboring points (Step 4). After that, a median smoothing filter 7×7 was applied to the surface to eliminate noise and improve image quality (Step 5). For surface texture, there are two geometrical product specifications published by the International Organization for Standardization (ISO), i.e., ISO 4287 and ISO 25178. The ISO 4287 specifies the terms, definitions and surface texture parameters based on profile method (2D), while the ISO 25178 collects the international standards relating to surface texture based on areal method (3D). The denoised surface after Step 5 can be treated as the S-F surface in ISO 25178, which is a surface obtained after applying an F-operator (a filter to correct tilt and suppress the nominal surface texture characterization). The areal primary parameter Sa_p (corresponds to Pa in ISO

4287) and the effective heat transfer area ratio r could be calculated from this surface.

As shown in Fig. 3.3(g), the S-F surface still consists of long wavelength component (waviness) and short wavelength component (roughness). For pool boiling, the waviness part is obviously too large for nucleation cavities which are only in micro-scale. Therefore, the denoised surface was split into waviness components (Fig. 3.3(h)) and roughness components (Fig. 3.3(i)), by applying an areal robust gaussian filter with a cut-off of 25 μm . The surface roughness parameters were then calculated from the roughness components (Fig. 3.3(i)) according to ISO 25178 standard.

Fig. 3.3 Example of the pre-processing before the roughness parameter evaluation.

Fig. 3.4 Illustration of applying the robust gaussian filter (25 μm cut-off) on the profile and surface, respectively.

In the filtration process, selecting appropriate cut-off value (the limit wavelength between the long-scale component and short-scale component) is critical in order to separate roughness from waviness properly. There is not a generally accepted criterion for determining the cut-off value, especially for the areal filter. The choice of cut-off value is very empirical and highly dependent on the practical application. In the present study, the cut-off value was determined as a result of many attempts, and a cut-off value of 25 μm was finally selected in order to assure that the irrelevant long-scale component was well removed and the short-scale component having similar magnitude with the

nucleation cavities could be remained as much as possible, as shown in Fig. 3.4. In the ISO 25178 standard, both the areal arithmetical mean height calculated from Fig. 3.3(g) and that calculated from Fig. 3.3(i) are denoted as Sa . To distinguish them, the former one was denoted as Sa_p in this study. All the measured results are summarized in Table 3.1. It should be noticed that all the parameters were calculated by averaging the results of three scans in different locations on each surface.

Fig. 3.5 (a) Measurements of static contact angle at different directions, (b) relation between static contact angle and surface roughness Sa .

The static contact angle was measured by an optical contact angle measuring and contour analysis system, DataPhysics OCA 15EC. The volume of the sessile droplet in the measurement was 10 μL . It should be noticed that the static contact angle of the testing surfaces (except the reference sample) is different when the camera is aligned in different directions. Therefore, the measurements were performed in two different directions, i.e., the camera was placed either parallel or normal to the scratches, as shown in Fig. 3.5(a). The measurement results of two directions on each surface are illustrated in Fig. 3.2. The static contact angle is an average value of the measurements in two directions. And for each direction, the measurement was repeated three times at different locations. In general, the static contact angle of a testing surface was calculated by averaging six measurement results which were measured from three different locations and two directions. The results of the contact angle measurement as a function of Sa are shown in Fig. 3.5(b). All the testing surfaces are hydrophobic. However, their hydrophobicity shows significant disparity due to different surface preparation methods. The static contact angle of the laser processed surfaces is much larger than that of the sandpaper polished surfaces. For the polished surfaces, the static contact angle initially increases with surface roughness and remains a constant for further increase in the surface roughness. This trend is different from the results reported in previous literature [10, 177, 178]. The specific surface topography in the experiment can be a possible reason of this trend. As we know, the static contact angle highly depends on the spreading effect of the sessile drop on the testing surfaces. Rougher surface with larger scratches could obstruct droplet from spreading on the surface, especially in the

direction normal to the scratches. Therefore, the static contact angle is increased as the surface becomes rougher. This increase does not occur anymore when the surface roughness Sa reaches $0.57\ \mu\text{m}$. It is speculated that the obstruction effect of the droplet spreading has encountered a threshold value when the surface roughness Sa increases from $0.37\ \mu\text{m}$ to $0.57\ \mu\text{m}$. Further increase in surface roughness will not cause the surface more hydrophobic. On the other hand, for the femtosecond laser processed surfaces, the static contact angles of both surfaces are around 153° . The surface roughness has negligible effect on the static contact angle.

Table 3.1 Summary of the roughness parameter measurements on the testing surfaces.

Sample	r	$Sa_p\ (\mu\text{m})$	$Sa\ (\mu\text{m})$
Reference sample	1	0.06	0.045
Sandpaper polished 1200 grit	1.02	0.25	0.17
Sandpaper polished 600 grit	1.06	0.53	0.37
Sandpaper polished 180 grit	1.08	1.29	0.57
Sandpaper polished 120 grit	1.13	1.87	0.83
Sandpaper polished 80 grit	1.24	3.48	1.22
Femtosecond laser processed 1	1.15	3.40	0.38
Femtosecond laser processed 2	1.02	0.42	0.15

3.2.3 Experimental procedures

Before the experiment, the water was first heated up by the immersion heaters and boiled for around one hour to release the non-condensable gases. After degassing, the PID controller was set to 99.5 °C and the liquid was maintained at saturation state. Once the internal pressure and liquid temperature got stabilized, a settled voltage was applied to the cartridge heaters. The wall temperature would experience an initial increase and gradually reach steady state. In this experiment, we considered the wall temperature became stable when the variation of the thermocouple measurements in the copper block was less than 0.2 °C within one minute. The temperatures were calculated by averaging 30 measured values in the steady state. Bubble dynamics in the steady state were also recorded by the visualization system. The voltage was then increased to the next predetermined value and the same recording procedure was repeated until the occurrence of a sudden increase in the measured temperature (temperature of the copper block increased over 20 °C in one minute). Once the abrupt increase occurred, the power to the cartridge heaters must be shut down immediately to protect the boiling surface from burnout. The CHF was determined as the heat flux just before the sudden increase in the measured temperature. It is a remarkable fact that all the experiments were conducted once the surfaces had been prepared. The aging effect of surfaces is not considered in present study. Additional studies on the roughness effects of pool boiling as a function of time will be performed in our future work.

3.2.4 Calculations and uncertainty analysis

The heat flux through the surface can be calculated from the measured results of three

pieces of axially aligned thermocouple using a one-dimension heat conduction model,

$$q = -k \frac{\Delta T}{\Delta x} \quad (3.1)$$

where k is the thermal conductivity of the copper, ΔT is the temperature difference and Δx is the corresponding distance. Three different values of heat flux q can be calculated based on different combinations of the thermocouples. The variance of these values is negligible in the nucleate boiling regime. In order to lower the uncertainty of distance, the results based on T_1 and T_3 (as shown in Fig. 3.1(b)) are used in the present study. Therefore, Eq. (3.1) can be written as.

$$q = k \frac{T_1 - T_3}{\Delta x_{13}} \quad (3.2)$$

The wall temperature T_w can be then determined by T_3 and heat flux q ,

$$T_w = T_3 - q \frac{\Delta x_{3w}}{k} \quad (3.3)$$

where Δx_{3w} is the distance between thermocouple T_3 and the top wall of the copper block. The heat transfer coefficient h can be calculated as follows,

$$h = \frac{q}{\Delta T_{sat}} \quad (3.4)$$

in which ΔT_{sat} is wall superheat and can be calculated by:

$$\Delta T_{sat} = T_w - T_{sat} \quad (3.5)$$

where T_{sat} is the liquid saturation temperature.

Uncertainties of the calculated parameters mainly come from temperature measurement error and position error of the thermocouples. The thermocouple with a diameter of 0.5 mm is inserted into an 0.6 mm diameter hole. Therefore, the maximum error for distance between two thermocouples is 0.1 mm. Assuming the conductivity has a

relative error of 1%. Based on the methodology of Kline [179], the uncertainties of heat flux and surface temperature can be obtained by Eqs. (3.6) and (3.7),

$$\frac{\delta q}{q} = \sqrt{\left(\frac{\delta k}{k}\right)^2 + \left(\frac{\delta T_1}{T_1 - T_3}\right)^2 + \left(\frac{\delta T_3}{T_1 - T_3}\right)^2 + \left(\frac{\delta \Delta x_{13}}{\Delta x_{13}}\right)^2} \quad (3.6)$$

$$\delta T_w = \sqrt{\delta T_3^2 + \left(\frac{\Delta x_{3w}}{k} \delta q\right)^2 + \left(\frac{\Delta x_{3w} q}{k^2} \delta k\right)^2 + \left(\frac{q}{k} \delta \Delta x_{3w}\right)^2} \quad (3.7)$$

The estimated maximum uncertainties for q and T_w were 1.4 W/cm² and 0.28 °C, respectively. The contact angle measurement had an accuracy of 0.1° according to the manufacturer. The surface roughness parameter was obtained by averaging five independent measurement data and provided deviation less than 20% refer to the nominal value.

3.3 Results and discussion

To validate the results of the present experiments, the experimental data of the reference surface ($Sa = 0.045 \mu\text{m}$) are compared with previous studies which are performed in similar experimental conditions, i.e., saturated water pool boiling on smooth copper surface at atmosphere pressure. As shown in Fig. 3.6, the present experimental results are almost overlapped with the data from Jun et al. [144] and Yang et al. [180]. However, some discrepancy is observed in the study of other researchers [30, 157, 181]. This is because that the nucleate boiling is a process which depends on a lot of influence factors. The difference in any influence factor could be a possible reason of the discrepancy. Generally, the present experimental data are quite comparable with the previous studies. Besides the previous data, the Rohsenow correlation, Eq. (2.1), is also plotted in Fig. 3.6. For water-copper combination, the Rohsenow constant C_{sf} is usually evaluated to

0.0128, as suggested by Vachon et al. [24]. The data from Kwark et al. and Suriyawong and Wongwises can fit the Rohsenow correlation with $C_{sf}=0.0128$ well [30, 181]. However, this correlation cannot predict other data points in Fig. 3.6. A new Rohsenow correlation with $C_{sf}=0.0107$ is developed based on the present study and the data from Jun et al. and Yang et al. [144, 180]. This new Rohsenow constant $C_{sf}=0.0107$ can well predict the present data on the reference surface and will be applied in the following discussion regarding correlation development.

Fig. 3.6 Validation of the present experiments.

Fig. 3.7 Results for all the testing surfaces, (a) q - ΔT_{sat} curves, (b) h - q curves.

Boiling curves for all the eight testing surfaces are shown in Fig. 3.7. It can be found that when r or Sa_p (corresponds to Pa in ISO 4287) is used to represent surface roughness (see also Table 3.1), there is not a clear trend of the NBHT coefficient with respect to the surface roughness. Based on the experimental results, the same conclusion as that from the previous studies can be drawn, that is, the NBHT coefficient is an increase function of surface roughness only under a consistent surface roughening technique. When two surfaces machined by different techniques are investigated in the pool boiling experiments, the one with smaller r or Sa_p is able to show better heat transfer performance (see sandpaper polished 180 grit and laser processed 1). However, if the waviness components are removed from the surface texture, as shown in Fig. 3.4. The boiling curves shift to the left as the surface roughness parameter Sa (corresponds to Ra in ISO 4287) increases, regardless of the surface preparation method. The trend can be observed more clearly in Fig. 3.7(b). Rougher surface has consistently higher h than the smoother one. Besides, h increases as the heat flux q becomes larger. A lot of researchers emphasized that surface roughness is meaningless without specifying the

surface roughening technique, since they found that a smoother surface prepared by one surface roughening technique shows better heat transfer performance than a rougher one prepared by another surface roughening technique [170-172]. However, a different conclusion can be drawn based on the present study. As we know, nucleation sites in pool boiling are in micro-scale. Some surface roughening techniques which increase the surface roughness parameter by enlarging large-scale surface texture will not provide more potential nucleation sites hence improve the NBHT coefficient, just like the laser processed surfaces in the present study. Only when the same surface roughening technique is used, the one with larger Sa_p must provide more potential nucleation sites. The experimental data can effectively support this point of view. As shown in Fig. 3.8, three groups of data are picked from the boiling curves, i.e., sandpaper polished 80 grit, sandpaper polished 600 grit and femtosecond laser processed 1. The femtosecond laser processed 1 has similar Sa_p with the sandpaper polished 80 grit, while it has almost identical Sa with the sandpaper polished 600 grit after applying the standardized filtration process. It can be found that the femtosecond laser processed 1 surface has overlapped boiling curve with the sandpaper polished 600 grit surface, while the sandpaper polished 80 grit surface shows much better boiling performance. Obviously, the sandpaper polished 80 grit surface is “rougher” than the femtosecond laser processed 1 surface from the perspective of pool boiling, even though they have almost the same Sa_p . It is assumed that the surface texture can be divided into two components, a small-scale component which determines the nucleation cavity and a large-scale component which is irrelevant to the bubble

nucleation. The standardized filtration process shown in Fig. 3.3 aims to eliminate the large-scale surface texture by applying the median denoising filter, the median smoothing filter, the areal gaussian filter with a cut-off of 25 μm and other operations suggested by ISO 3274 and ISO 25178. The filtered roughness parameter Sa can be a better way to characterize the surface roughness in pool boiling. It is reasonable to believe that the result in the present study is closer to the pure effect of the surface roughness by applying such filtered roughness parameter Sa . Besides, the choice of the cut-off value of the areal filter in the present study can be a possible benchmark for the unestablished standard of ISO 25178, at least for surfaces applied in the pool boiling. Different with the surface roughness, the variation of surface wettability for hydrophobic surfaces in the present study seems to have limited effect on the NBHT coefficient. The femtosecond laser processed 1 surface has overlapped boiling curve with the sandpaper polished 600 grit surface, although their contact angles are slightly different. Besides, the laser processed surface 2 ($Sa = 0.15 \mu\text{m}$) and the sandpaper polished 1200 grits ($Sa = 0.17 \mu\text{m}$) also show the same result. On the other hand, the surface roughening technique is able to influence the CHF. In the unidirectionally polished surfaces, the CHF indeed becomes smaller as the surface roughness decreases. However, the smoothest unidirectionally polished one ($Sa = 0.17 \mu\text{m}$) has a lower CHF than the reference surface (randomly polished) and the laser processed surface 2, even though it has a larger surface roughness. Moreover, the CHF values of the two laser processed surfaces are almost identical (around 93.5 W/cm^2), while their roughness parameters Sa_p and Sa are totally different. All the results demonstrate that the CHF is

not a monotonic function of surface roughness. The surface wettability also plays an important role in determining the CHF.

Fig. 3.8 Comparison of the experimental results of laser processed 1 with sandpaper polished 80 grit and sandpaper polished 600 grit.

Fig. 3.9 Comparison of the visualized results with increasing Sa .

The visualization studies of all the testing surfaces under a same heat flux are shown in Fig. 3.9. For a certain testing surface, there is a larger number of nucleation bubbles as q increases. This explains why the NBHT coefficient is an increase function of the heat flux. It can be also found that there always exist more nucleation bubbles on the surfaces with larger Sa at a certain heat flux. This is the qualitative observation of the relation between Sa and the number of nucleation sites. Quantitative description of the potential

nucleation sites can be received by size distribution of cavities on the surfaces, which has been conducted by some researchers [174, 182]. The details will not be discussed here due to limited length. A simple quantitative analysis of the bubble departure characteristics is conducted using the image processing program ImageJ. The results are shown in Fig. 3.10. Three surfaces with totally different surface roughening techniques and surface roughness parameters are included in the analysis. The heat flux is fixed at 4.7 W/cm^2 . The results reveal that bubbles depart at a smaller diameter and higher frequency as the surface becomes rougher. This trend is consistent with that reported in Ref. [103]. The decreased bubble departure diameter and increased bubble departure frequency, together with the increased number of nucleation sites, result in the higher NBHT coefficient of a rougher surface regardless of the surface roughening technique.

Fig. 3.10 A simple case in evaluating bubble departure characteristics.

Eq. (2.1) developed by Rohsenow follows the relation $q \propto \Delta T_{sat}^3$. However, this relation is found invalid for some reported data. The exponent in the $q-\Delta T_{sat}$ relation should be determined based on the experimental conditions. Therefore, Eq. (2.1) is modified as follows,

$$\frac{c_{p,l}\Delta T_{sat}}{L} = C_{sf} \left[\frac{q}{\mu L} \sqrt{\frac{\sigma}{g(\rho_l - \rho_v)}} \right]^{\frac{1}{n}} Pr_l^s \quad (3.8)$$

where n represents the exponent in the $q-\Delta T_{sat}$ relation. In order to perform the data analysis better, Eq. (3.8) is converted into the dimensionless form,

$$\frac{h \sqrt{\frac{\sigma}{g(\rho_l - \rho_v)}}}{k} = C_{sf}^{-n} \left(\frac{c_{p,l}\Delta T_{sat}}{L} \right)^{n-1} Pr^{1-sn} \quad (3.9)$$

As mentioned in *Section 2.1.1.1*, the bubble departure diameter is often found proportional to $\sqrt{\frac{\sigma}{g(\rho_l - \rho_v)}}$ [20-22]. Therefore, the left-hand side of Eq. (3.9) can be regarded as Nusselt number taking the bubble departure diameter as characteristic length. Eq. (3.9) can be then rewritten in the form of dimensionless numbers,

$$Nu = C Ja^{n-1} Pr^{1-sn} \quad (3.10)$$

where

$$Ja = \frac{c_{p,l}\Delta T_{sat}}{L} \quad (3.11)$$

is the Jakob number, Nu is the Nusselt number of nucleation bubbles, and C is a constant depending on the Rohsenow constant C_{sf} and n . In Eq. (3.10), s is still 1 for water and 1.7 for the other fluids. But C and n are dependent on the experimental conditions. In the following analysis, they are correlated with surface roughness Sa based on the experimental data. For convenience, the surface roughness Sa is also nondimensionalized to Sa/Sa_0 , where Sa_0 should be the representative surface

roughness for typical metal surfaces. Following the suggestions of the other researchers in 2D profile (Ra), the representatively areal arithmetical mean height Sa_0 is also equal to $0.4 \mu\text{m}$ [112, 183]. The parameter n in the above equation is first evaluated by performing the best fit from the experimental data. Fig. 3.11 shows the relation between Nu and Ja plotted on a logarithmic scale. The $n-1$ value of a given testing surface can be then determined as the slope of the straight line in the figure. n is then plotted as a function of Sa/Sa_0 in the bottom-right corner of Fig. 3.11 and it is found to increase almost linearly with Sa/Sa_0 ,

$$n = 3.22 + 0.12 \frac{Sa}{Sa_0} \quad (3.12)$$

and the correlation for C as a function of Sa/Sa_0 can be then obtained by fitting the data points in Fig. 3.12.

$$C = 2.74 \times 10^6 [1 + 0.46 (\frac{Sa}{Sa_0})^{3.02}] \quad (3.13)$$

We can summarize the final correlation in the dimensionless form,

$$Nu = C_1 [1 + C_2 (\frac{Sa}{Sa_0})^{3.02}] Ja^{n-1} Pr^{1-sn} \quad (3.14)$$

where

$$C_1 = 2.74 \times 10^6, C_2 = 0.46 \quad (3.15)$$

It should be noticed that Eq. (3.14) is likely to be valid for hydrophobic surfaces having Sa in the range of $0.045 \mu\text{m}$ to $1.35 \mu\text{m}$. Besides, the analysis of Sa in present study is quite specific. It is not recommended correlating other's experimental data with this equation unless the same filtration process is followed in the surface roughness analysis. It should be also noticed that Eq. (3.14) can be only applied to predict the NBHT

coefficient. Before boiling occurs, the heat is mainly removed by natural convection, which is a totally different heat transfer mechanism with nucleate boiling. Using Eq. (3.14) to predict the heat transfer performance at this stage will cause large deviation.

Fig. 3.11 Fitting and parameter development of the Nu - Ja curves.

Fig. 3.12 Evaluation of C in Eq. (3.10).

Fig. 3.13 is a comparison study between the experimental data and the predictions of Eq. (3.14). The predicted values are in good agreement (within 20%) with the experimental data except for the low heat flux region (initial stage of bubble nucleation). In fact, the wall superheat at this region is small, which results in a large relative error of the surface temperature since the absolute error of temperature measurement is a

fixed value. As a result, the uncertainty of h in this region is huge and the result cannot be completely trusted. The inaccurate measurement is the most likely reason for the inconsistency between the experimental data and the predicted values at low heat flux region. Nevertheless, Eq. (3.14) can predict the heat transfer coefficient in the partially and fully developed nucleate boiling regime within an acceptable accuracy for different surface roughnesses.

Fig. 3.13 Comparison results of Eq. (3.14) and experimental data.

3.4 Summary

The effect of surface roughness on the nucleate boiling in a water pool was investigated. Eight testing surfaces were fabricated on plain copper through three preparation methods including random polishing, unidirectional polishing, and femtosecond laser machining. The different surface roughening techniques could introduce different surface features and surface wettability. The surface roughness in the present study was characterized by the areal (three-dimensional, 3D) roughness parameter. By applying a standardized filtration process, the large-scale component of the surface could be

removed, and the areal arithmetical mean height Sa of the testing surfaces was in the range of 0.045 μm to 1.35 μm . The effect of areal arithmetical mean height Sa on the NBHT coefficient was experimentally studied in pool boiling and correlated in the dimensionless form. Bubble dynamics were also investigated to further understand the mechanism of the nucleate boiling.

Chapter 4 Nucleate boiling heat transfer enhancement in saturated Novec 7100 using titania nanotube arrays[‡]

4.1 Introduction

With improvements in nanotechnology, different nano-structured surfaces are developed and their possible application in enhancing pool boiling is explored. The investigated nanostructures include nanofibers, nanorods, nanowires, nanotubes, and nanoparticle depositions [145, 146, 184]. Among the nanostructures, nanotubes attract a lot of attentions since this type of structure usually owns exceptional mechanical and thermal properties, in conjunction with extremely high aspect ratio and chemical inertness. The most known nanotube is carbon nanotube (CNT) which was reported by Iijama in 1991 [185]. Since then, the application of CNTs in pool boiling has been examined by many researchers. For instance, Ujereh et al. [186] developed vertically aligned CNT arrays on Si/Cu bases. Pool boiling experiments were conducted in FC-72 and the effects of density and area coverage of the CNT arrays were explored. The experimental results exposed that the fully CNT-coated surfaces showed significant augmentation in both the NBHT coefficient and the CHF. Sathyamurthi et al. [187] coated vertically aligned multiwalled carbon nanotubes (MWCNTs) with different

[‡] This chapter has been reproduced in its entirety with permission as an academic work from the following publication:

S. Fan, W. Tong, F. Duan, Nucleate pool boiling heat transfer enhancement in saturated Novec 7100 using titanium dioxide nanotube arrays, *International Communications in Heat and Mass Transfer*, 122 (2021) 105166.

coating thicknesses (9 and 25 μm) on silicon wafers, and conducted pool boiling experiments in PF 5060. The MWCNTs coatings were found to effectively improve the NBHT coefficient on the silicon substrate, and the improvement was relatively independent to the coating thickness. Significantly more nucleation bubbles were discovered in the MWCNT-coated surfaces, which was believed responsible for the improvement. The same result was reported by Kumar et al. [188], Ho et al. [55], and Dharmendra et al. [147], who also observed improved NBHT coefficient and CHF for carbon nanotubes and attributed the improvements to the larger number of bubble nucleation sites. On the other hand, there are also some contradictory results reported. For instance, Launay et al. [189] experimentally studied the CNT-coated surfaces in pool boiling with water and PF 5060, respectively. They found that the CNT-coatings could only enhance the NBHT coefficient at low wall superheats compared with the smooth surfaces and attributed the result to the change in surface-fluid interactions. Seo et al. [190] conducted experiments on stainless-steel surfaces with and without single-walled CNTs coating in a water pool. The CHF of the nanotube coated surface was increased up to 55% when compared to the substrate. Increased capillarity of the CNT arrays was believed responsible for the improvement in CHF. However, the NBHT coefficient was found to deteriorate as the bare substrate was coated with nanotubes. The deteriorated NBHT coefficient was speculated to be a result of the decreased surface roughness of the nanotube coating.

Titanium dioxide (TiO_2) is a material with unique surface properties and has been widely used in the applications such as photovoltaics, ceramic coatings, electric devices,

electrochromic devices, earth science, etc. [191]. The titanium dioxide based nanotubes (titania nanotubes) are also utilized by some researchers in the pool boiling experiments. Xu et al. [192] examined the vertically aligned titania nanotube coated surfaces in pool boiling with saturated water, and a smooth titanium surface was also tested for reference purpose. They found that the titania nanotube coating could effectively enhance both the NBHT coefficient and the CHF. The enhancements were attributed to the nanotube-induced change in surface characteristics including increased surface wettability, increased surface area, increased bubble nucleation sites, and increased capillarity. In addition, the deformation of nanotube structures during boiling was also found to benefit the pool boiling heat transfer. Prakash and Ravindran [193] fabricated TiO_2 nanotubes on plain titanium surface using anodization method. The TiO_2 nanotube coated surface was hydrophilic in nature and was treated with stearic acid to acquire hydrophobicity. The plain titanium surface, hydrophilic nanotube surface, and hydrophobic nanotube surface after stearic acid treatment were investigated in pool boiling with water. The effects of the titania nanotubes were dependent on the surface wettability. Comparing to the plain surface, the hydrophilic nanotube surface had enhanced NBHT coefficient and CHF, while the hydrophobic nanotube surface showed opposite effects. Increased bubble nucleation sites, higher bubble departure frequency, smaller bubble departure diameter, and faster rewetting of the dry spots due to capillary wicking caused the enhancements for hydrophilic nanotube surface. For the hydrophobic nanotube surface, the authors believed that the stearic acid treatment would introduce increased surface passivation, which hindered the active surface area

hence deteriorated the heat transfer. Different result was reported by Zhang et al. [194], who also investigated the titania nanotube surfaces in pool boiling. They found that the titania nanotube surfaces had deteriorated NBHT coefficient compared to the plain titanium surface, even the nanotube surfaces were hydrophilic. Furthermore, the hydrophilicity of the titania nanotube coatings was found changeable by controlling the UV light irradiation time during anodic oxidation process. With increasing hydrophilicity of the nanotube surfaces, the NBHT coefficient deteriorated, and the CHF increased.

Based on the reported investigations of pool boiling for carbon nanotubes and titania nanotubes, it can be easily concluded that the nanotube structures are effective in delaying CHF, representing a higher thermal safety margin of the cooling system. However, there are controversial results about improvement/deterioration of the NBHT coefficient for the applications of the nanotube coatings in pool boiling studies. Therefore, in this work, pool boiling experiments are conducted on the plain surfaces and TiO₂ nanotube coated surfaces to clarify the influence of TiO₂ nanotube structures on pool boiling performance, thereby helping prospect the further applications of the TiO₂ nanotubes on the two-phase cooling systems. A highly-wetting liquid, Novec 7100, is applied in the experiments to isolate the effects of surface wettability. The NBHT coefficient and CHF of the tested surfaces are measured and discussed. Visualized studies are conducted to investigate bubble dynamics on different tested surfaces. The present study provides a new perspective to understand the controversial results about the influence of nanotube coatings on the NBHT coefficient of pool

boiling.

4.2 Experimental setup and procedures

4.2.1 Experimental setup

As illustrated in Fig. 4.1, a columnar boiling chamber with front and back window made of Pyrex glass is mounted on the support base. The chamber body is made of stainless steel. A flexible heater is attached to the chamber body to provide heat to the liquid pool. A reflux condenser which is connected to a controllable water bath is performed as the chiller to maintain the two-phase circulation in the chamber. The chamber pressure is monitored by a pressure gauge. Two thermocouples T_5 and T_6 are immersed into the liquid pool to monitor the liquid temperature. For the testing part, a Cu/Ti block is placed into a Teflon holder and the empty part is filled with thermal insulation material. The block is heated by a ceramic plate heater adhered on the bottom side to simulate the heating electronics. Four thermocouples are placed along the center of the block to measure the temperature gradient. The measurements of the thermocouples are reported by a data acquisition system. Calibrations of the thermocouples are completed on a calibrator, Techne Ucal 400+, which is a dry block type with accuracy of 0.2 K. Visual results of the surface boiling were recorded at a frame rate of 2000fps during the experiments. It should be noticed that the high-speed camera is installed perpendicular to the boiling surface, hence it is not illustrated in Fig. 4.1.

Fig. 4.1 Schematic diagram of the experimental setup.

Fig. 4.2 Tested surfaces with static contact angle measurements.

4.2.2 Surface preparation

The saturated pool boiling experiments were conducted in four surfaces, i.e., a plain copper surface, a plain titanium surface, a nanotube surface coated on copper substrate, and a nanotube surface coated on titanium substrate. The schematics of the tested surfaces and their static contact angle measurements are shown in Fig. 4.2. The plain surfaces were polished by silicon carbide grinding paper (Struers, 320 grit). The nanotubes were fabricated by a specific anodization process which has been described by Sun et al. in detail [195-197]. The anodization was conducted on an electrochemical system in ethylene glycol comprising 0.3 wt.% ammonium fluoride and 2 vol%

deionized water. The applied potential was controlled to 60 V by a Keithley 2400 source-meter during the anodization. After the process continued for a period of 30 minutes, nanotube structures were formed on the titanium anode. The same TiO_2 nanotubes were coated on the copper substrate and titanium substrate, respectively. It should be noticed that the TiO_2 nanotubes could be only grown on the titanium material in this study. Therefore, an indium foil was used as interface material to connect the copper substrate and the nanotube coating for surface (III) in Fig. 4.2. Thermal interface resistance should be taken into consideration for this surface. In addition to the materials of the substrate, the two nanotube coated surfaces ((III) and (IV) in Fig. 4.2) are identical.

4.2.3 Experimental procedures

Before running the experiments, liquid Novec 7100 in the chamber was first preheated by the flexible heater and boiled in the chamber then condensed in the reflux condenser to discharge the non-condensable gases. After 1 h degassing, the flexible heater was adjusted to maintain the pool liquid saturated. The block was subsequently heated by the ceramic plate heater to simulate the working electronics. The temperature measurements in the copper/titanium block (T_1 , T_2 , T_3 , T_4) would experience a rapid increase initially and gradually reach a plateau. In this study, we recognized the boiling was stable when the variations of the thermocouple measurements of T_1 , T_2 , T_3 , and T_4 were below 0.2 K in a minute. The steady state temperatures were calculated by averaging 30 consecutive measurements. Visual observation of the boiling surface was also conducted simultaneously. After recording the temperature measurements and the

visual results, the power supply to the ceramic plate heater was increased to the next prearranged value and the same steps were repeated until the occurrence of CHF. In present study, a precipitous increase of more than 10 K in one minute for the temperature measurements of thermocouples T_1 , T_2 , T_3 , and/or T_4 was recognized as the indication of CHF. The experiments should be stopped with the indication, and the CHF value in this study was determined as the heat flux just before the indication. It should be noticed that the CHF value determined in this study is not the exact CHF but only an estimation which is very close to it. However, this is the most reliable value we can obtain since the testing surface cannot reach the exact CHF in order to protect it from burnout.

4.2.4 Experimental data and uncertainty analysis

In this study, the same method was used to calculate q , ΔT_{sat} , and h , as demonstrated in *Section 0*. The only difference is that there are four thermocouples inserted in the block as comparison to three in *Section 0*. As a result, the temperature measurements and location data of thermocouples T_1 and T_4 were used in the calculations while the data of thermocouples T_2 and T_3 were applied to validate the linearity. The heat flux, surface temperature, heat transfer coefficient, and wall superheat could be then calculated based on the same method as Eqs. (3.2)-(3.5).

The uncertainty analysis was performed based on Eqs. (3.6) and (3.7). The maximum uncertainties of q and ΔT_{sat} in this study were 0.9 W/cm² and 0.21 K for copper substrate-based surfaces and 1.3 W/cm² and 0.88 K for titanium substrate-based

surfaces, respectively.

4.3 Results and discussion

To verify the present study could obtain reliable data, the experimental data of plain copper are compared with that also performed with saturated Novec 7100 and plain copper surface in publications [198, 199]. The surface roughness Ra for copper surfaces in Refs. [198] and [199] are $0.78\text{ }\mu\text{m}$ and $0.884\text{ }\mu\text{m}$ respectively, which are comparable with that in present study. Fig. 4.3 compares the present results with that in the publications. It is obvious that the data points for present study and previous studies at low q (below 15 W/cm^2) are almost overlapped. However, the present work shows significantly improved NBHT coefficient in the high heat flux regime when compared to the previous investigations. Aside from the NBHT coefficient, the CHF of the present study is also higher than that in the previous investigations. The small size of boiling surface could be the most possible reason for the improved CHF in this study, while the same explanation was also given by Lienhard and Dhir in their investigation [200]. Of course, there may exist some other reasons for the discrepancy between the present results and that in the previous investigations. Nucleate pool boiling is an extremely complicated heat transfer process which is still not completely understood about its heat transfer mechanism. Any difference in influencing factors, measurement accuracy and/or data processing method could cause the discrepancy in Fig. 4.3. Overall, the present results are comparable with that in the publications, hence the experimental data are considered reliable. Besides, the experimental data are also compared with the Rohsenow correlation which has been listed as Eq. (2.1) in *Section 2.1.1*. The parameter

s is estimated at 1.7 for this study and the Rohsenow constant C_{sf} is developed based on the experimental data to 0.0036. The Rohsenow correlation with $C_{sf} = 0.0036$ could well predict the experimental data below 75% CHF which is approximately the ultimate design heat flux for a boiling-involved thermal management system.

Fig. 4.3 Boiling curves of saturated Novec 7100 and plain copper surface: in present study, previous publications, and Rohsenow correlation.

Fig. 4.4 Results of all tested surfaces in saturated Novec 7100, where “Nanotubes on copper substrate (r)” is calculated after reducing a thermal interface resistance of $60 \text{ K} \cdot \text{mm}^2/\text{W}$ from the experimental data of the “Nanotubes on copper substrate”.

Fig. 4.4 shows all the results of the pool boiling experiments on the testing surfaces. It is clearly apparent to the sight that the two plain surfaces have almost overlapped

boiling curves. As mentioned in the last paragraph, the Rohsenow constant depends on the liquid-surface combination and the surface material often plays a role in determining the Rohsenow constant. However, the result of this work seems to contradict with the abovementioned conclusion. In this study, the Rohsenow constant is relatively independent to the surface material. One possible reason is the well-controlled surface condition in this study. We know that the nucleation of bubbles is the fundamental mechanism of pool boiling. Surface roughness and surface wettability are two factors affecting the bubble nucleation most. In this study, the surface roughnesses for the two plain surfaces are both controlled to around $0.64\text{ }\mu\text{m}$, and the static contact angles for the two plain surfaces are also nearly identical (12.1° for Cu substrate and 14.1° for Ti substrate) as demonstrated in Fig. 4.2. As a result, the heat transfer coefficient shows very limited difference in plain copper surface and in plain titanium surface. Back to the Rohsenow constant, it could be deduced based on the experimental results that the liquid-surface combination is not a concept of the surface material and the liquid type. Therefore, it is incorrect to summarize the Rohsenow constant based on different combinations of surface material and liquid type. The Rohsenow constant C_{sf} should involve the effects of parameters which can represent the affinity and interactions between the surface and the liquid, including but not limited to surface roughness, surface area, surface wettability (contact angle), surface orientation, and surface porosity.

For the nanotube coated surfaces, it can be observed from Fig. 4.4 that the titanium substrate-based nanotube surface shows improvements in both NBHT coefficient and

CHF, compared to the plain surfaces. However, the copper substrate-based nanotube surface has an immensely deteriorated NBHT coefficient compared to the plain surfaces, while the CHF reaches 52.2 W/cm^2 which is the highest among all the tested surfaces. The trends can be observed more clearly in Fig. 4.5 which plots h with q . The deteriorated NBHT coefficient of the nanotube surfaces has also been reported in previous literature and is attributed to decreased surface roughness or change in the surface wettability [189, 190, 193]. However, these two factors cannot explain the results in present study considering the contact angle has been well controlled and the limited decrease in surface roughness cannot result in such significant deterioration in NBHT coefficient. Therefore, it is speculated that the decreased NBHT coefficient is caused by the thermal interface resistance between the copper substrate and the nanotube coating, as explained in Fig. 4.2 already. By reducing a thermal interface resistance of $60 \text{ K}\cdot\text{mm}^2/\text{W}$ from the experimental data of copper substrate-based nanotube surface, the modified boiling curve represented by solid symbols and labeled with (r) in Fig. 4.4 coincides with that of titanium substrate-based nanotube surface. In other words, the boiling curves for two different nanotube surfaces only differ in a constant thermal interface resistance even though they look quite different. The thermal interface resistance of indium foil has been investigated by some researchers and the measurements are between 18 and $71 \text{ K}\cdot\text{mm}^2/\text{W}$ in the literature [201-203]. The estimated value in Fig. 4.4 is within the range in the literature, which effectively supports our speculation. Based on the above analysis, together with the consideration that substrate material should not influence the HTC in the pool boiling, it can be

inferred the experimental data of the titanium substrate-based nanotube surface could represent the real performance of the TiO_2 nanotube arrays in pool boiling. The experimental data of the copper substrate-based nanotube surface, on the other hand, are results of the thermal interface resistance, and its value can be estimated by comparing the boiling curve with that of the titanium substrate-based nanotube surface. In conclusion, the TiO_2 nanotube arrays are effective in improving both the cooling performance and the thermal safety margin of a two-phase cooling system for titanium-based device, by coating the nanotubes directly on the device surface. On the contrary, the nanotube structures are not recommended as surface coatings of two-phase cooling systems for copper-based devices unless the potential effects of the thermal interface resistance could be eliminated.

Fig. 4.5 h - q curves of all tested surfaces in saturated Novec 7100, where “Nanotubes on copper substrate (r)” is calculated after reducing a thermal interface resistance of $60 \text{ Kmm}^2/\text{W}$ from the experimental data of the “Nanotubes on copper substrate”.

The experimental results in Figs. 4.4 and 4.5 can be explained by the visualization study illustrated in Fig. 4.6. In addition, the visual results are animated, and the animation is presented in the supplementary. For each tested surface, it can be found that more sites

are nucleated with increasing heat flux (see columns in Fig. 4.6), which explains the elevated NBHT coefficient h with the heat flux q at the initial stage, as illustrated in Fig. 4.5. However, with increasing heat flux and bubble nucleation sites, the adjacent bubbles start to interfere with each other and merge into a larger one, finally leading to a vapor blanket formed on the boiling surface. The vapor blanket would impede the bubbles' departure from and liquid rewetting to the boiling surface hence significantly deteriorate the heat transfer performance, which is indicated by a downtrend of the NBHT coefficient h with the heat flux q at high heat flux regime in Fig. 4.5. Therefore, the increasing then decreasing trend of the NBHT coefficient h with the heat flux q in Fig. 4.5 can be well explained by the visual observations.

Fig. 4.6 Visual observations of different tested surfaces with increasing heat flux.

Comparing the visual results of different tested surfaces, it can be easily found that the bubble dynamics are significantly different for the plain surface and the nanotube surfaces. It is also worth noting that the two nanotube coated surfaces show similar bubble dynamics under the same heat flux, which effectively supports our perspective in the last paragraph, i.e., thermal interface resistance is the reason causing the deteriorated NBHT coefficient of the copper substrate-based nanotube coated surface and the two nanotube coated surfaces should have identical heat transfer performance if the potential effects of the thermal interface resistance are eliminated. Back to the comparison of the visual results between the plain surface and the nanotube surfaces, it can be found that the nanotube coated surfaces nucleate more bubbles at low heat flux, while the number of bubble nucleation sites is indistinguishable at moderate and high heat flux for all the tested surfaces since the surfaces are entirely covered by the nucleation bubbles. A quantitative analysis of the bubble dynamics at heat flux of 7.75 W/cm^2 is conducted using the image processing program ImageJ and the results are illustrated in Fig. 4.7. The plain surface and the two nanotube surfaces are included in the analysis. It can be found that the single bubble on the nanotube coated surfaces is smaller and departs from the boiling surface at a higher frequency, when compared to that on the plain surface. The bubble departure characteristics of the nanotube coated surfaces are believed a result of the tube-like structures, as shown in Fig. 4.8. Due to capillary wicking effect, the nanotubes could promote liquid replenishment to the boiling surface, leading to faster bubble nucleation and departure hence finally causing the results in the visual observation, i.e., smaller bubble departure diameter and higher

bubble departure frequency. In addition, the faster liquid replenishment of the nanotube coated surfaces could also ensure better rewetting of dry spots and delay the liquid-choking on the boiling surface, hence the CHF is improved. In summary, the faster liquid replenishment of the nanotube structures due to capillary wicking is the key reason for the enhanced heat transfer of the nanotube coated surfaces in this study. It is beneficial to both the NBHT coefficient and the CHF by changing the bubble dynamics and improving dry spots rewetting, respectively.

(a) Plain titanium surface

Average bubble departure diameter: 0.46 mm
Average bubble departure frequency: 112.4 Hz

(b) Nanotubes on Ti substrate

Average bubble departure diameter: 0.37 mm
Average bubble departure frequency: 142.5 Hz

(c) Nanotubes on Cu substrate

Average bubble departure diameter: 0.36 mm
Average bubble departure frequency: 145.6 Hz

Fig. 4.7 Evaluation of the bubble departure diameter and bubble departure frequency at heat flux of 7.75 W/cm^2 . Please also see the supplementary animation.

Fig. 4.8 Schematic diagram of the bubble dynamics on nanotube structures.

4.4 Summary

The pool boiling of the TiO_2 nanotube coated surface was experimental studied in saturated Novec 7100, in order to explore the potential application of this type of nanotube in two-phase electronics cooling. The nanotubes were anodized on titanium plate in an electrochemical system using ethylene glycol electrolyte comprising 0.3 wt.% ammonium fluoride and 2 vol% deionized water. The nanotube coating was applied to a copper substrate and a titanium substrate respectively and studied in pool boiling experiments together with two smooth surfaces for comparison purpose. The surface roughness and static contact angle were controlled almost the same for the testing surfaces. Therefore, the difference in heat transfer performance for the nanotube coated surfaces and the plain surfaces was believed only relate to the nanotube structure. In addition to the heat transfer, bubble dynamics of a plain surface and two nanotube coated surfaces were also visualized to help understand the mechanisms of nucleation heat transfer on different tested surfaces. Up to this chapter, our investigations are still limited to surface characteristics in pool boiling. The impacts of jet impingement on the two-phase submerged boiling systems are not touched. Moreover, the last two chapters focus on the studies of the NBHT coefficient and the CHF. The lower limit of nucleate

boiling regime, onset of nucleate boiling, is ignored. Therefore, we will rivet our attention to the submerged boiling systems with jet impingement in the next chapter.

And the onset of nucleate boiling will be highlighted.

Chapter 5 Correlations for heat transfer characteristics of the onset of nucleate boiling in submerged jet impingement

5.1 Introduction

As we mentioned in *Chapter 2*, jet impingement boiling is particularly attractive for dissipating concentrated heat loads. It can provide improved convection and larger CHF compared to the pool boiling. Besides, the design flexibility makes it very practical in the applications such as steel annealing and thermal management of high-power density electronics [204-206]. The investigations in jet impingement boiling mainly focus on two configurations, i.e., free surface and submerged. The jet impingement boiling is classified as free surface when the liquid jet discharges into a gaseous environment. In contrast, the jet impingement boiling is classified as submerged type when the jet is released into an ambient liquid with similar/same properties. Due to different configurations, the heat transfer characteristics of the two types also show obvious differences. In the free surface jet, droplet breakup and ejection could affect the heat transfer performance deeply, while such a situation would not occur in the submerged jet. The submerged type, on the other hand, shares similar mechanism of heat transfer with pool boiling and is often considered as a potential enhancement method of pool boiling, as demonstrated in *Section 2.1.1.2*. In addition, as introduced in *Section 1.1*, a specific case of the submerged jet is defined as confined type where the space between the nozzle plate and the impingement surface is constricted by an upper wall, forcing

the impinging jets to escape in a small gap. The confined jet impingement is inevitable in the applications when the space is confined, or the installation is limited. Generally, the submerged jet impingement boiling is attractive in various applications due to the outstanding heat transfer performance, spatial claims and/or installation limitations.

The investigations of the boiling heat transfer with submerged/confined jet are performed in both water [42, 72, 207] and dielectric fluids with low boiling point such as R113 [49, 208], R134a [71], FC-72 [73-75], R245fa [143], Novec 7000 [140] and Novec 7100 [12]. The research mainly covers the effects of hydrodynamic parameters of jets (jet velocity, jet subcooling), geometrical parameters of jets (nozzle dimension, nozzle-to-surface distance, single slot, and jet arrays), and impingement surface (surface roughness and contact angle, surface structures, heater size, and surface aging). In addition, reliable prediction tools (models and correlations), pressure drop, flow patterns, and flow instability are also discussed. Wolf et al. [8] comprehensively reviewed the works conducted until 1993 in this field, while the recent developments are summarized by Qiu et al. [7] and Fan and Duan [204]. Moreover, the progress in this topic has been systematically reviewed in *Chapter 2*, there is no need to repeat here. However, it should be realized that the investigations on ONB are very limited for the submerged/confined jet, compared to the studies on nucleate boiling regime and its upper limit CHF. As we know, the heat transfer performance near ONB shows significant difference for that in water and that in dielectric fluids. The nucleation prefers cavities retaining residual vapor or non-condensable gas, but dielectric fluids are highly wetted and will penetrate deeply into the cavities. The penetrability of the

dielectric fluids impedes the cavities from harboring residual vapor. As a result, the ONB is delayed. Moreover, an overshoot of wall superheat near ONB is often reported in the dielectric fluids but it is rarely found in water. Shin et al. [15] observed the overshoot of wall superheat in a confined jet of FC-72 (PF5060). In their study, the ONB heat flux increased with higher jet velocity and/or smaller nozzle-to-surface distance. Their results conformed with that of Brún et al. [44] and Zhang et al. [74], who also found that the ONB was delayed with increased jet velocity in water and FC-72, respectively. In addition to the jet velocity, Meyer et al. [77] concluded that the increases in nozzle size and subcooling could also delay the ONB. Joshi and Dede [143] conducted experiments on four porous coated surfaces with confined R-245fa jet arrays of 5×5 orifices. They found the ONB heat flux was independent of surface structures. Cui et al. [140] observed the different trend in the jet array of Novec 7000. They found that boiling initialed at a reduced wall superheat and higher heat flux on the pin-fin surface, when compared to the smooth one.

In general, the above mentioned literature provides us some basic understanding of the ONB for the submerged/confined jet impingement. However, the investigations in this topic are sporadic and usually involve only one/two control parameter(s) under a given condition. A comprehensive understanding of the ONB on its control parameters for the submerged/confined jet impingement is still lacking. In addition, the identification methods of the ONB are not consistent in the previous works, which causes the results from different research groups almost incomparable. To address these issues, experiments were conducted systematically in subcooled water with a single circular

nozzle under a wide range of jet Reynolds number, jet subcooling, and nozzle diameter. The heat transfer characteristics were investigated from forced convection to nucleate boiling, at the stagnation point of a polished copper surface. A novel identification method of the ONB based on the $h-\Delta T_{sat}$ curves with verification by visual observation of the impinging surface is applied in this study. Empirical correlations were proposed for predicting the heat transfer characteristics of the ONB point in a submerged jet. The present work helps build a database of the ONB with controlled parameters in the submerged/confined jet impingement boiling. To our best knowledge, the systematic investigation of the ONB point in the submerged/confined jet is reported for the first time.

5.2 Experimental setup and procedures

5.2.1 Experimental setup

Fig. 5.1 Schematic diagram of the flow loop.

A schematic diagram of the flow loop in the experiments is displayed in Fig. 5.1 (a).

The distilled water is circulated through the flow loop. A circulator with temperature

control function (Julabo F25) is performed as the reservoir, the pump, and the heat exchanger. The flow rate can be precisely adjusted through a bypass line together with a metering valve. Downstream of the metering valve, a particulate filter (Swagelok SS-6TF-40) is installed to strain the contaminants. The flow rate can be then monitored by an oval gear flow meter (BIOTECH VZS-005-VA). A preheater is placed upstream of the test section to adjust the jet temperature to predetermined value. After impinging on the testing surface, the water returns to the reservoir due to altitude intercept and a closed flow loop can be maintained.

The test section is retrofitted from the design of Fig. 3.1. A single circular nozzle with jet flow path is mounted at the center of the chamber, while the boiling chamber, testing part, temperature measurement system, and visualization system are all maintained. The water exited the preheaters enters the boiling chamber at the top and forms a submerged circular jet at the orifice. After impinging on the surface, the water leaves the boiling chamber through a fluid outlet at the side wall of the chamber to achieve liquid circulation. The copper block is still enclosed by the thermal insulation material to prevent heat loss. And the temperature gradient along the copper block is also measured by three thermocouples (T_1 , T_2 and T_3) placed along the center of the copper block at positions indicated in Fig. 3.1(b). In addition to the three thermocouples inserted inside the copper block, thermocouples are also placed to monitor the pool temperature, the fluid inlet temperature, and jet outlet temperature. The measurements of the thermocouples are reported by a data acquisition system. Calibrations of the thermocouples are completed on a calibrator, Techne Ucal 400+, which is a dry block

type with accuracy of 0.2 K. Visual results of the impinging surface are recorded at a frame rate of 2000fps by the high-speed camera indicated in Fig. 3.1(a).

5.2.2 Experimental procedures

Experiments of a circular jet impingement were conducted from single phase convection to nucleate boiling. A wide range of jet parameters was tested, including jet subcooling (ΔT_{sub} from 20 °C to 50 °C), jet Reynolds number (Re from 3972 to 26899, where $Re = \frac{\rho_l v_j d_j}{\mu_l}$), and jet diameter (d_j from 1.2 mm to 2.4 mm). Besides the three tested parameters, there is another parameter which could effectively influence the submerged boiling heat transfer performance, i.e., nozzle-to-surface distance H/d_j . In this study, each tested parameter was investigated by controlling the other two tested parameters and H/d_j constant. Before the experiment, the impingement surface of the copper block was first polished using Silicon Carbide Grinding Paper (Struers, 2400 grit) and cleaned by sonicating in distilled water. The surface roughness characteristics of the polished surface were measured using the method in *Chapter 3*. The areal arithmetical mean height Sa measured from the surface was around 60 nm. In the boiling heat transfer experiments, there are two major issues which are recognized to disturb the consistency of the experimental results. One is the non-condensable gases, while the other is the surface aging effect. In current study, a priming run was performed for eliminating the effects of non-condensable gases and surface aging. Prior to each experimental test, the PID controller was first set to 99.5 °C (saturation temperature of water) in order to boil the pool liquid. The pump and the metering valve were then

turned on and the flow rate was adjusted to 800 ml/min. Water could be circulated in the flow loop at a relatively high frequency and boiled and condensed in the testing chamber to remove the non-condensable gases. After 1 h degassing, a power input of 300 W was applied to the copper block through the cartridge heaters. Bubbles were nucleated throughout the impinging surface under this experimental condition. After the impinging surface was boiled for another 1 h, the power supply to the cartridge heaters was interrupted and the experimental test could be started. During the experiments, the jet temperature and flow rate were first adjusted to the prearranged values. The PID controller was also set to the same value as the jet temperature. Once the jet flow condition and liquid pool temperature got stabilized, the block was subsequently heated by the cartridge heaters to simulate the working electronics. The temperature measurements of thermocouples T_1 , T_2 and T_3 would experience a rapid increase initially and gradually reach a plateau. In this study, we recognized the heat transfer process was stable when the variations of the thermocouple measurements of T_1 , T_2 , and T_3 were below 0.2 K in a minute. The steady state temperatures were calculated by averaging 30 consecutive measurements after the stable condition was satisfied. The time interval between two consecutive measurements is one second. Visual observation of the impinging surface was also performed simultaneously using the high-speed camera. After recording the temperature measurements and the visual results, the power supply to the cartridge heaters was increased to next prearranged value and the same steps were repeated. In current study, the increase in input heat flux between two consecutive data points near ONB was controlled to around 2 W/cm^2 . The

ONB was determined by slope change on the boiling curves, with the assistance of visual observation of the impinging surface.

5.2.3 Experimental data and uncertainty analysis

The measured data in the experiments were converted to q and T_w using the same method as *Section 0*. Eqs. (3.2) and (3.3) were applied to calculate q and T_w , respectively.

The heat transfer coefficient could be then calculated as follows

$$h = \frac{q}{T_w - T_l} = \frac{q}{\Delta T_{sat} + \Delta T_{sub}} \quad (5.1)$$

Since the local temperature on the heating surface is normally not uniform in submerged jet impingement, the linearity of the temperature gradient along the copper block must be strictly confirmed when q and T_w are calculated by Eqs. (3.2) and (3.3). The temperature distributions along the copper block with their linear regressions at various heat fluxes are illustrated in Fig. 5.2. The R squared values for linear regressions are infinitely close to unity at all the heat fluxes, which effectively supports the assumption of the one-dimensional heat conduction model applied in the calculation.

The major component of the uncertainty in the present study is through the temperature measurements and location data of the thermocouples. The uncertainties of q and T_w can be obtained by Eqs. (3.6) and (3.7) respectively. Besides, the uncertainty of h can be calculated by the same method.

$$\frac{\delta h}{h} = \sqrt{\left(\frac{\delta q}{q}\right)^2 + \left(\frac{\delta T_w}{T_w - T_l}\right)^2 + \left(\frac{\delta T_l}{T_w - T_l}\right)^2} \quad (5.2)$$

The uncertainties of the thermocouple measurements and thermocouple positions in the experiments were 0.2 °C and 0.1 mm, respectively. Other physical quantities such as

liquid thermal conductivity, liquid density, and liquid dynamic viscosity were all estimated to have a relative uncertainty of 1%. According to Eqs. (3.6), (3.7), and (5.2), the maximum estimated uncertainty for q was 1.3 W/cm^2 . T_w and h had maximum uncertainties of $0.26 \text{ }^\circ\text{C}$ and $0.07 \text{ W/cm}^2\cdot\text{K}$, respectively. The flow rate measurement had an uncertainty of 1% given by the manufacturer. The jet temperature and pool temperature had an uncertainty of $0.2 \text{ }^\circ\text{C}$ based on the accuracy of the calibrator. Based on the same method as Eqs. (3.6), (3.7), and (5.2), the highest uncertainties for Re and Nu were given as 658.72 and 2.02, respectively.

Fig. 5.2 Temperature distribution along the thermocouples T₁ to T₃ at different heat fluxes.

5.3 Results and discussion

The onset of nucleation boiling can be generally identified by two methods, i.e., visual observation and slope change in the boiling curves. In this chapter, the ONB point is determined based on the $h-\Delta T_{sat}$ curves, with verification by visual observation of the

impinging surface. As can be seen in Fig. 5.3, h , representing the local heat transfer coefficient of the stagnation point, remains almost constant in the early stage and increases sharply after an inflection point, with the gradual increase in the wall superheat. This inflection point in the $h-\Delta T_{sat}$ curve is the ONB. The ONB point separates the boiling curve into two distinct regimes. Before the ONB point, heat is transferred by forced convection and h is insensitive to the surface temperature. After the ONB point, the heat transfer mode is so-called nucleate boiling and h becomes to an increase function of the surface temperature. The improvement in heat transfer performance after the ONB can be mainly attributed to two reasons, i.e., latent heat of vaporization and the agitating effect of bubbles which enhances the forced convection. The ONB point determined based on the $h-\Delta T_{sat}$ curve can be verified by the visual observation of the impinging surface, as also shown in Fig. 5.3. Although several bubbles could be observed at the edge of the impinging surface before the ONB point, these bubbles were found to consistently attach on the surface edge. Once the first bubble started to depart from the surface (the image indicated by “ONB” in Fig. 5.3), h would experience a sharp increase after that, indicating the rationality of the method used in this chapter for determining the onset of nucleation boiling. The visual observation is displayed more explicitly in the supplementary animation.

Fig. 5.3 A sample case for determining the ONB point.

All the experimental data are plotted to the h - ΔT_{sat} curves and the results are illustrated in Figs. 5.4 and 5.5. The experimental conditions cover a wide range of liquid subcooling (20, 30, 40, 50 K), jet Reynolds number ($3972 \leq Re \leq 26899$), and nozzle diameter (1.2, 1.5, 1.8, 2.4 mm). The effects of these parameters on the ONB will be discussed in the following sections. It should be noticed that the physical quantities in calculating the Reynolds number are evaluated at the average temperature of the impinging surface and the liquid jet, $(T_w + T_l)/2$. Therefore, it is almost impossible to control the Reynolds number constant when studying the effects of subcooling and nozzle diameter, since the liquid properties are changed as the surface temperature T_w increases. The variances in Re due to temperature-induced changes of liquid properties are too limited to alter the trend, but they can exactly influence the detailed heat transfer coefficient. Consequently, the averaged values of Re are used in Figs. 5.4 and 5.5 to

indicate the trend. On the other hand, the precise Re value of each data point is applied in the correlation development, as shown in Fig. 5.6. In addition, all data in the experiments are collected at the stagnation point. Therefore, the physical quantities such as h , q , and ΔT_{sat} in the following sections are all local values at the stagnation point but not averages of the whole impinging surface. The correlations proposed in the following discussion are only valid for the stagnation point and within the experimental conditions, serious errors may arise if they are used outside the ranges of applicability.

Fig. 5.4 Experimental results of different liquid subcoolings and jet Reynolds numbers at fixed nozzle diameter and nozzle-to-surface distance.

Fig. 5.5 Experimental results of varying nozzle diameters at constant subcooling and nozzle-to-surface distance for two different Reynolds numbers.

5.3.1 Effects of jet parameters on h_{ONB}

Since the ONB is the upper limit of the single phase convection regime, it should also share the same heat transfer coefficient with the single phase at this point, i.e., $h_{\text{ONB}} = h_{c,\text{ONB}}$. As illustrated in Fig. 5.4, h_c (HTC in the single phase convection regime) is an increase function of Re while keeps relatively independent with liquid subcooling at fixed nozzle diameter and H/d_j . On the other hand, Fig. 5.5 illustrates that h_c decreases with increasing nozzle diameter under the same subcooling, Re , and H/d_j . The stagnation point single phase heat transfer coefficient of submerged/confined jet impingement has been studied in the previous literature, e.g., Sun et al. [209] and Choo et al. [210] in water jet, Garimella and his group [211, 212] in FC-77 liquid jet, Wang

et al. [213] in mixed molten salt jet, etc. Various correlations for predicting the local heat transfer coefficient had been proposed. Generally, the correlations are in the form of the local Nusselt number as a function of the local Pr , the local Re , and other dimensionless parameters. Unfortunately, the application of these correlations is highly restricted by the applicable range and a universal correlation is still lacked. In this thesis, a correlation is proposed following the same form of that developed by Sun et al. [209]. The local Nusselt number is correlated by a power-law dependence on Re and Pr .

$$Nu = C_3 Re^{m_1} Pr^{m_2} \quad (5.3)$$

in which

$$Nu = \frac{h_c d_j}{k_l} \quad (5.4)$$

Same as the Reynolds number, the physical quantities in calculating Nu should also be evaluated at $(T_w + T_l)/2$. In previous studies, the exponent of Pr , m_2 , was usually equal to 0.33 or 0.4, depending on the working fluid and Pr range. It is found that $m_2 = 0.33$ can well predict our experimental data. The exponent of Re , m_1 , can be then found by performing the best fitting of all the single phase data in Figs. 5.4 and 5.5. The fitting curve is illustrated in Fig. 5.6. And the local Nusselt number for submerged jet impingement in single phase regime can be finally expressed as follows:

$$Nu = 0.215 Re^{0.5} Pr^{0.33} \quad (5.5)$$

The experimental data in Fig. 5.4 are collected at a fixed $H/d_j = 8$, while that in Fig. 5.5 are collected at $H/d_j = 6$. According to Fig. 5.6, the experimental data collected at different H/d_j but constant nozzle diameter ($d_j = 1.8$ mm) almost coincide, indicating this parameter has negligible effect on the local Nu in this study. On the other hand,

most data points are collected at the nozzle diameter of 1.8 mm. The predictions of Eq. (5.5) at nozzle diameter around this value (1.5 mm and 1.8 mm) should have the highest accuracy, while the accuracy of this correlation for nozzle diameter of 1.2 mm and 2.4 mm should be also acceptable. The predicted values of the HTC by Eq. (5.5) are compared with the experimental data and the comparison results are illustrated in Fig. 5.7. There exists an acceptable coincidence between the data and the predictions. The maximum discrepancy falls around 20%. Therefore, h_{ONB} can be obtained by putting the numbers at the ONB point into Eq. (5.5),

$$h_{\text{ONB}} = h_{c,\text{ONB}} = \frac{Nu_{\text{ONB}} k_l}{d_j} \quad (5.6)$$

Fig. 5.6 Evaluation of the exponent of Reynolds number in Eq. (5.3).

Fig. 5.7 Comparison results of Eq. (5.5) and experimental data.

Both Eqs. (5.5) and (5.6) are valid for the parameter ranges of $3972 \leq Re \leq 26899$, $20 \leq \Delta T_{sub} \leq 50$ K, and $1.2 \leq d_j \leq 2.4$ mm. Besides, the dimensionless nozzle-to-surface distance should be restricted to $6 \leq H/d_j \leq 8$.

5.3.2 Effects of parameters on the ONB point

The ONB points for all the experiments are indicated by arrow lines in Figs. 5.4 and 5.5. The colors of the arrow lines are in keeping with their corresponding data points. It can be observed that the wall superheat at the ONB point ($\Delta T_{sat,ONB}$) is relatively insensitive to the jet parameters. This finding is contradictory with that of Li et al. [214] which systematically investigated the stagnation point heat transfer characteristics at the ONB point in the free surface jet. They found the wall superheat at the ONB point was an increase function of both jet Re and liquid subcooling and attributed this trend

to the change of thermal boundary layer thickness. Specifically, the thermal boundary layer would become thinner with increasing jet velocity or liquid subcooling. As a result, higher wall superheat is required to provide a thick enough thermal boundary layer for bubble nucleation. However, this thermal boundary layer theory seems cannot explain the trend in present study. The different trends of this paper and Li et al.'s may come from the differences in the heat transfer mechanism of submerged jet and free surface jet. The submerged jet impingement boiling is often considered as a potential enhancement method of pool boiling since it shares the similar heat transfer mechanism with the latter one. As we know, the bubble nucleation in pool boiling is mainly related to the critical cavity size. The critical cavity radius which can be activated for bubble nucleation was given by Griffith and Wallis [215],

$$r_c = \frac{2\sigma T_{sat} v_{fg}}{L \Delta T_{sat}} \quad (5.7)$$

where σ is the surface tension, v_{fg} is the latent specific volume change, and L is the latent heat of vaporization. Based on Eq. (5.7), the wall superheat at the ONB point ($\Delta T_{sat,ONB}$) mainly depends on the surface condition, i.e., the maximum critical cavity radius $r_{c,max}$. Since the impinging surface was not changed in the experiments, the values of $\Delta T_{sat,ONB}$ at different experimental conditions also show little difference. Of course, there is an inherent defect of the above explanation. The bubble is initially generated at the periphery of the impinging surface in submerged jet impingement boiling while our measurements locate at the stagnation point. Nevertheless, it is reasonable to believe that they can follow the same trend to some extent, which explains the insensitivities of the ONB wall superheat on the jet parameters in present study. In

addition, more attempts should be made in the future study to investigate the local wall superheat at the position where the first bubble nucleates, and its relation to the experimental parameters.

Fig. 5.8 The stagnation heat flux at the ONB point under different Reynolds numbers.

Fig. 5.9 The stagnation heat flux at the ONB point under different liquid subcoolings.

Unlike the wall superheat, the heat flux at the ONB point (q_{ONB}) is related to the jet parameters. q_{ONB} is plotted with jet Reynolds number and liquid subcooling in Figs. 5.8 and 5.9, respectively. It can be easily found that both the jet Reynolds number and liquid subcooling are in positive relation with q_{ONB} . Besides, their effects on q_{ONB} are relatively independent. q_{ONB} and Re can be fitted into the same power function relation regardless of the liquid subcooling as shown in Fig. 5.8, and the same result can be also found in Fig. 5.9 except that q_{ONB} and ΔT_{sub} are in a linear dependence. On the other hand, the nozzle diameter is found to have negative influence on q_{ONB} . The results are illustrated in Fig. 5.10. Obviously, the two curves at different Reynolds numbers can be still fitted by the same power function relation. In summary, increasing the jet Reynolds number and liquid subcooling, or decreasing the nozzle diameter could broaden the single phase convective regime and delay the ONB in terms of heat flux. The heat transfer characteristics at the ONB point in the submerged jet had also been investigated by previous researchers [71, 74, 77]. Unfortunately, they only reported the trend but not gave a persuasive explanation of their results. In this chapter, an attempt is made to offer the reason for the observed trend. Based on Eq. (5.1), the stagnation point heat flux at the ONB point can be expressed by, $q_{\text{ONB}} = h_{\text{ONB}} (\Delta T_{\text{sat,ONB}} + \Delta T_{\text{sub}})$. Since $\Delta T_{\text{sat,ONB}}$ is relatively independent of the jet parameters (as demonstrated in the previous paragraph), the increased jet Reynolds number and/or decreased nozzle diameter could enlarge q_{ONB} by improving h_{ONB} while the liquid subcooling can be explicitly found in the above expression.

Fig. 5.10 Relation between the stagnation point heat flux at the ONB point and the nozzle diameter at constant subcooling and nozzle-to-surface distance for two different Reynolds numbers.

Since the effects of the Re , ΔT_{sub} , and d_j on q_{ONB} are relatively independent, the final expression of q_{ONB} can be written as,

$$q_{ONB} = C_4 Re^{0.493} (1 + 0.16 \Delta T_{sub}) d_j^{-0.75} \quad (5.8)$$

where C_4 is a constant. The constant C_4 is evaluated to 4.61 by performing the least-squares method on the experimental data, as shown in Fig. 5.11. The predictions and experimental data are in agreement within 15%. It should be noticed that Eq. (5.8) does not include the dimensionality and the values of the physical quantities should be obtained in SI units, especially the nozzle diameter is in m but not in mm. The stagnation point wall superheat at the ONB point can be predicted by combining Eqs. (5.6) and (5.8),

$$\Delta T_{sat,ONB} = 21.44 Re^{-0.007} (1 + 0.16 \Delta T_{sub}) Pr^{-0.33} d_j^{0.25} k_l^{-1} - \Delta T_{sub} \quad (5.9)$$

The applicable range of Eqs. (5.8) and (5.9) is still within the experimental conditions, i.e., $3972 \leq Re \leq 26899$, $20 \leq \Delta T_{sub} \leq 50$ K, and $1.2 \leq d_j \leq 2.4$ mm. In addition, the dimensionless nozzle-to-surface distance should be restricted to $6 \leq H/d_j \leq 8$.

Fig. 5.11 The ONB heat flux correlated with Reynolds number, liquid subcooling, and nozzle diameter.

5.4 Summary

The heat transfer performance of a submerged water jet was experimentally investigated on the stagnation zone of a polished copper surface. The effects of jet Reynolds number, liquid subcooling, and nozzle diameter on the ONB were systematically studied. The ONB could separate the submerged jet impingement boiling into two distinct regimes with different heat transfer mechanisms. Before the ONB point, the single phase forced convection played a dominant role and the local heat transfer coefficient h was insensitive to the surface temperature. After the ONB point, h increased sharply with the surface temperature. The ONB point was determined

based on the $h-\Delta T_{sat}$ curves, with verification by visual observation of the impinging surface. Correlations for predicting the heat transfer characteristics of the ONB point were proposed.

Chapter 6 Conclusions and Future Work

6.1 Conclusions

The works in this thesis fulfilled the main objectives of the project: developing a more accurate parameter to characterize surface roughness and investigating the relation between the developed parameter and the submerged boiling heat transfer performance; exploring the further applications of titania nanotube arrays on the two-phase cooling system; studying the onset of nucleate boiling in a submerged jet with controlled parameters and establishing reliable prediction tools. All the efforts were contributed to further understand the heat transfer mechanism of submerged boiling and improve its performance in the applications.

In Chapter 3, a filtration process was applied to effectively remove the large-scale surface texture from the surface and improve the accuracy of the areal surface roughness parameter, S_a . The improved roughness parameter S_a was found to characterize the surface roughness in pool boiling properly. The boiling curves were shifted to the left as the parameter S_a became larger, regardless of the surface roughening technique. On the other hand, the CHF was affected by both the surface roughness and the surface roughening technique. It was not a monotonic function of surface roughness without specifying the surface roughening technique. The surface wettability also played an important role in determining the CHF. Bubble dynamics were visualized to explore the mechanism of the boiling heat transfer with surface roughness. The rough surfaces enhanced the nucleate boiling by providing more

nucleation sites, decreasing the bubble departure diameter, and increasing the bubble departure frequency. In the end, the heat transfer coefficient was correlated with the parameter Sa in the form of dimensionless numbers. The validation results showed that the correlation was able to well predict the heat transfer coefficient at surface roughness Sa in the range of 0.045-1.35 μm , especially in the fully developed nucleate boiling regime.

In Chapter 4, the NBHT coefficient and CHF of copper and titanium surfaces with and without the titania nanotube arrays were experimentally investigated in a dielectric fluid, Novec 7100. Both nanotube-coated surfaces showed enhanced CHF of around 40% compared to the plain surfaces. On the other hand, the nanotubes coated on titanium substrate could effectively enhance the heat transfer coefficient, while the nanotubes coated on copper substrate showed opposite effect. Based on the coincident boiling curves after reducing a thermal interface resistance of 60 $\text{K}\cdot\text{mm}^2/\text{W}$, together with the similar bubble departure characteristics of the two nanotube coated surfaces in visual observation, it was concluded that the experimental data of the titanium substrate based nanotube surface could represent the heat transfer performance of the titania nanotubes. According to the visual observation, the improvements of nanotube surfaces in heat transfer coefficient and CHF could be attributed to the modified bubble dynamics and better rewetting of the dry spots respectively, which were both related to the faster liquid replenishment due to capillary wicking of the tube-like structure. The titania nanotube arrays showed a high potential for improving both the cooling performance and the thermal safety margin of a two-phase cooling system for titanium-based device,

by coating the nanotube arrays directly on the device surface. On the contrary, this type of nanotubes was not recommended as a surface coating of the two-phase cooling system for devices based on other materials unless the potential effects of the thermal interface resistance could be eliminated.

In Chapter 5, the heat transfer characteristics of the ONB were experimentally investigated at the stagnation point under a wide range of subcooling (ΔT_{sub} from 20 °C to 50 °C), jet Reynolds number (Re from 4000 to 33000), and jet diameter (d_j from 1.2 mm to 2.4 mm). The results indicated that the single phase heat transfer coefficient at the stagnation point was an increase function of the jet Reynolds number, a decrease function of the nozzle diameter, and remained relatively independent with the liquid subcooling. For the wall superheat of the ONB point, all the investigated parameters showed limited influence on it. An explanation was provided based on the critical cavity radius which was widely applied in the bubble nucleation of pool boiling. Unlike the wall superheat of the ONB point, the jet Reynolds number and liquid subcooling were found in positive relation with the heat flux of the ONB point, while the nozzle diameter had opposite effect. Moreover, their effects on the ONB heat flux were relatively independent. Correlations for predicting the ONB heat flux, the ONB wall superheat, and the ONB heat transfer coefficient at the stagnation point were proposed based on the experimental data. The correlations were able to well predict the present data and could be applied under the ranges of the investigated parameters.

6.2 Future work

In this thesis, the research works provide new experimental data and research methods in the study of submerged boiling heat transfer regarding surface roughness, nanoscale structures, and onset of nucleate boiling for submerged jet. Besides the issues investigated in this thesis, the following topics are worth to be explored in the future.

The major obstacle in investigating the submerged boiling still comes from the incomprehensive understanding of the heat transfer mechanism, e.g., the dependence of nucleation bubbles on the control parameters such as wall superheat, fluid properties, jet parameters, and surface characteristics. In studies of nucleation bubbles, the bubble departure diameter and bubble departure frequency are usually random and almost incalculable due to the coalescence of isolated bubbles. To address this issue, submerged boiling experiments can be designed and conducted on an extremely smooth surface with an artificial cavity/fin in scale of several micrometers. The design of experiments is expected to generate a single bubble on the artificial cavity/fin. Therefore, the departure characteristics of the single bubble can be investigated under different experimental conditions to explore its dependence on the control parameters hence help further understand the heat transfer mechanism of submerged boiling.

Surfaces can be designed elaborately to provide separated liquid-vapor flow pathways hence create submerged jets in pool boiling without the assistance of the traditional jet generation system. It combines the benefits of pool boiling and submerged jet, giving us the possibility of designing a system with heat removal capacity in submerged jet impingement boiling level but keeping the system away from pumping power and moving parts. The separated liquid-vapor flow pathways are normally achieved by

fabricating artery structures to prevent bubble escape or creating some area on the surface with extremely high/low priority for bubble departure. Methods include selectively thermal insulation, selective porous coating, open microchannels, feeder microchannels, etc. Compared to the traditional ways, 3D printing, or additive manufacturing, provides us more flexibility in designing the elaborate surface structures. Therefore, it would be interesting to apply the 3D printing in designing the separated liquid-vapor flow pathways.

The accuracy and applicability of correlations as prediction tool are restricted by the applicable ranges. Numerical simulation, taking the physical mechanism of heat transfer into consideration, has higher potential for accurately and universally predicting the submerged boiling heat transfer. The Rensselaer-Polytechnic Institute (RPI) boiling model which partitions the total heat flux into convection, quenching, and evaporation, has been developed and applied in simulating the submerged boiling heat transfer. Its reliability highly depends on the accuracies of sub-models for nucleation site density, bubble departure diameter, and bubble departure frequency. In this thesis, bubble dynamics are investigated qualitatively through the high-speed camera. But the quantitative analysis is lacked. Therefore, the visual observation of bubble dynamics in Chapters 3 and 4 would be studied quantitatively and the quantitative results would be applied in developing models for the nucleation site density, bubble departure diameter, and bubble departure frequency. The developed models will be incorporated to RPI boiling model and its validation will be performed by comparing the predicted values with the experimental data.

Author's Publications

Journal

[1] **S. Fan**, F. Duan, A review of two-phase submerged boiling in thermal management of electronic cooling, *International Journal of Heat and Mass Transfer*, 150 (2020) 119324.

[2] **S. Fan**, L. Jiao, K. Wang, F. Duan, Pool boiling heat transfer of saturated water on rough surfaces with the effect of roughening techniques, *International Journal of Heat and Mass Transfer*, 159 (2020) 120054.

[3] **S. Fan**, W. Tong, F. Duan, Nucleate pool boiling heat transfer enhancement in saturated Novec 7100 using titanium dioxide nanotube arrays, *International Communications in Heat and Mass Transfer*, 122 (2021) 105166.

[4] B.B. Kanbur, C. Wu, **S. Fan**, W. Tong, F. Duan, Two-phase liquid-immersion data center cooling system: Experimental performance and thermoeconomic analysis, *International Journal of Refrigeration*, 118 (2020) 290-301.

[5] B.B. Kanbur, C. Wu, **S. Fan**, F. Duan, System-level experimental investigations of the direct immersion cooling data center units with thermodynamic and thermoeconomic assessments, *Energy*, 217 (2021) 119373.

Conference

[1] W. Tong, **S. Fan**, F. Duan, Pool boiling heat transfer in dielectric fluids for electronic cooling: in *International Conference on Thermal Engineering 2017*.

[2] **S. Fan**, W. Tong, J. Luo, F. Duan, Nucleate pool boiling heat transfer on a titania nanotube-coated surface: in International Heat Transfer Conference Digital Library 2018.

References

- [1] I. Mudawar, Assessment of high-heat-flux thermal management schemes, *IEEE Transactions on Components and Packaging Technologies*, 24(2) (2001) 122-141.
- [2] D. Lee, J.S. Lim, N. Lee, H.H. Cho, Enhanced thermal uniformity and stability in pool boiling heat transfer using ultrasonic actuation, *International Communications in Heat and Mass Transfer*, 106 (2019) 22-30.
- [3] B.B. Kanbur, C. Wu, S. Fan, W. Tong, F. Duan, Two-phase liquid-immersion data center cooling system: experimental performance and thermoeconomic analysis, *International Journal of Refrigeration*, 118 (2020) 290-301.
- [4] A. Bar-Cohen, M. Arik, M. Ohadi, Direct liquid cooling of high flux micro and nano electronic components, *Proceedings of the IEEE*, 94(8) (2006) 1549-1570.
- [5] T.B. Nguyen, D. Liu, M.I. Kayes, B. Wang, N. Rashin, P.W. Leu, T. Tran, Critical heat flux enhancement in pool boiling through increased rewetting on nanopillar array surfaces, *Scientific Reports*, 8(1) (2018) 1-9.
- [6] M.S. El-Genk, A.F. Ali, Saturation boiling critical heat flux of PF-5060 dielectric liquid on microporous copper surfaces, *Journal of Heat Transfer*, 137(4) (2015) 041501.
- [7] L. Qiu, S. Dubey, F.H. Choo, F. Duan, Recent developments of jet impingement nucleate boiling, *International Journal of Heat and Mass Transfer*, 89 (2015) 42-58.

- [8] D.H. Wolf, F.P. Incropera, R. Viskanta, Jet impingement boiling, in: *Advances in Heat Transfer*, Elsevier, 1993, pp. 1-132.
- [9] M. Molana, S. Banooni, Investigation of heat transfer processes involved liquid impingement jets: a review, *Brazilian Journal of Chemical Engineering*, 30(3) (2013) 413-435.
- [10] J.S. Kim, A. Girard, S. Jun, J. Lee, S.M. You, Effect of surface roughness on pool boiling heat transfer of water on hydrophobic surfaces, *International Journal of Heat and Mass Transfer*, 118 (2018) 802-811.
- [11] R. Cardenas, V. Narayanan, Heat transfer characteristics of submerged jet impingement boiling of saturated FC-72, *International Journal of Heat and Mass Transfer*, 55(15-16) (2012) 4217-4231.
- [12] M.D. Clark, J.A. Weibel, S.V. Garimella, Identification of nucleate boiling as the dominant heat transfer mechanism during confined two-phase jet impingement, *International Journal of Heat and Mass Transfer*, 128 (2019) 1095-1101.
- [13] R. Cardenas, V. Narayanan, Submerged jet impingement boiling of water under subatmospheric conditions, *Journal of Heat Transfer*, 134(2) (2012) 020909.
- [14] S.J. Wu, C.H. Shin, K.M. Kim, H.H. Cho, Single-phase convection and boiling heat transfer: Confined single and array-circular impinging jets, *International Journal of Multiphase Flow*, 33(12) (2007) 1271-1283.

- [15] C.H. Shin, K.M. Kim, S.H. Lim, H.H. Cho, Influences of nozzle-plate spacing on boiling heat transfer of confined planar dielectric liquid impinging jet, *International Journal of Heat and Mass Transfer*, 52(23-24) (2009) 5293-5301.
- [16] R.P. Rioux, E.C. Nolan, C.H. Li, A systematic study of pool boiling heat transfer on structured porous surfaces: From nanoscale through microscale to macroscale, *AIP Advances*, 4(11) (2014) 117133.
- [17] H.S. Jo, M.W. Kim, K. Kim, S. An, Y.I. Kim, S.C. James, J. Choi, S.S. Yoon, Effects of capillarity on pool boiling using nano-textured surfaces through electrosprayed BiVO₄ nano-pillars, *Chemical Engineering Science*, 171 (2017) 360-367.
- [18] M. Khooshehchin, A. Mohammadidoust, S. Ghotbinasab, An optimization study on heat transfer of pool boiling exposed ultrasonic waves and particles addition, *International Communications in Heat and Mass Transfer*, 114 (2020) 104558.
- [19] W.M. Rohsenow, A method of correlating heat transfer data for surface boiling of liquids, *Transactions of ASME*, 74 (1952) 969-975.
- [20] R. Cole, Bubble frequencies and departure volumes at subatmospheric pressures, *AIChE Journal*, 13(4) (1967) 779-783.
- [21] J. Kim, M.H. Kim, On the departure behaviors of bubble at nucleate pool boiling, *International Journal of Multiphase Flow*, 32(10-11) (2006) 1269-1286.

- [22] Y. Nam, E. Aktinol, V.K. Dhir, Y.S. Ju, Single bubble dynamics on a superhydrophilic surface with artificial nucleation sites, *International Journal of Heat and Mass Transfer*, 54(7-8) (2011) 1572-1577.
- [23] R.I. Vachon, G.E. Tanger, D.L. Davis, G.H. Nix, Pool boiling on polished and chemically etched stainless-steel surfaces, *Journal of Heat Transfer*, 90 (1968) 231-238.
- [24] R.I. Vachon, G.H. Nix, G.E. Tanger, Evaluation of constants for the Rohsenow pool-boiling correlation, *Journal of Heat Transfer*, 90 (1968) 239-246.
- [25] P.K. Tewari, R.K. Verma, M.P.S. Ramani, S.P. Mahajan, Effect of surface roughness & polymeric additive on nucleate pool boiling at subatmospheric pressures, *International Communications in Heat and Mass Transfer*, 13(5) (1986) 503-514.
- [26] S.P. Liaw, V.K. Dhir, Void fraction measurements during saturated pool boiling of water on partially wetted vertical surfaces, *Journal of Heat Transfer*, 111 (1989) 731-738.
- [27] I.L. Pioro, Experimental evaluation of constants for the Rohsenow pool boiling correlation, *International Journal of Heat and Mass Transfer*, 42(11) (1999) 2003-2013.
- [28] B.J. Jones, S.V. Garimella, Effects of surface roughness on the pool boiling of water, in: *ASME/JSME 2007 Thermal Engineering Heat Transfer Summer Conference collocated with the ASME 2007 InterPACK Conference*, 2007, pp. 219-227.

- [29] C. Qi, Y. He, Y. Hu, B. Jiang, T. Luan, Y. Ding, Experimental study on boiling heat transfer of α -Al₂O₃-water nanofluid, *Nanoscience and Nanotechnology Letters*, 5(8) (2013) 895-901.
- [30] S.M. Kwark, M. Amaya, R. Kumar, G. Moreno, S.M. You, Effects of pressure, orientation, and heater size on pool boiling of water with nanocoated heaters, *International Journal of Heat and Mass Transfer*, 53(23-24) (2010) 5199-5208.
- [31] H.K. Forster, N. Zuber, Dynamics of vapor bubbles and boiling heat transfer, *AIChE Journal*, 1(4) (1955) 531-535.
- [32] N. Zuber, Hydrodynamic aspects of boiling heat transfer, United States Atomic Energy Commission, Technical Information Service, 1959.
- [33] Z.H. Liu, X.F. Yang, J.G. Xiong, Boiling characteristics of carbon nanotube suspensions under sub-atmospheric pressures, *International Journal of Thermal Sciences*, 49(7) (2010) 1156-1164.
- [34] J.H. Lee, T. Lee, Y.H. Jeong, The effect of pressure on the critical heat flux in water-based nanofluids containing Al₂O₃ and Fe₃O₄ nanoparticles, *International Journal of Heat and Mass Transfer*, 61 (2013) 432-438.
- [35] P. Goel, A.K. Nayak, P.P. Kulkarni, J.B. Joshi, Experimental study on bubble departure characteristics in subcooled nucleate pool boiling, *International Journal of Multiphase Flow*, 89 (2017) 163-176.

- [36] S. Zhang, Y. Tang, J. Zeng, W. Yuan, J. Chen, C. Chen, Pool boiling heat transfer enhancement by porous interconnected microchannel nets at different liquid subcooling, *Applied Thermal Engineering*, 93 (2016) 1135-1144.
- [37] L. Lee, B.N. Singh, The influence of subcooling on nucleate pool boiling heat transfer, *Letters Heat Mass Transfer*, 2 (1975) 315-323.
- [38] D. Deng, W. Wan, J. Feng, Q. Huang, Y. Qin, Y. Xie, Comparative experimental study on pool boiling performance of porous coating and solid structures with reentrant channels, *Applied Thermal Engineering*, 107 (2016) 420-430.
- [39] S. Jun, J. Kim, S.M. You, H.Y. Kim, Effect of subcooling on pool boiling of water from sintered copper microporous coating at different orientations, *Science and Technology of Nuclear Installations*, 2018 (2018) 8623985.
- [40] S.S. Kutateladze, L.L. Schneiderman, Experimental study of influence of temperature of liquid on change in the rate of boiling, *USAEC Report, AECtr*, 3405 (1953) 95-100.
- [41] N. Zuber, The hydrodynamic crisis in pool boiling of saturated and subcooled liquids, *International Developments in Heat Transfer, ASME*, 27 (1961) 230-236.
- [42] Z. Zhao, Y. Peles, M.K. Jensen, Water jet impingement boiling from structured-porous surfaces, *International Journal of Heat and Mass Transfer*, 63 (2013) 445-453.

- [43] M. Monde, T. Inoue, Critical heat flux in saturated forced convective boiling on a heated disk with multiple impinging jets, *Journal of Heat Transfer*, 113 (1991) 722-727.
- [44] C. de Brún, R. Jenkins, T.L. Lupton, R. Lupoi, R. Kempers, A.J. Robinson, Confined jet array impingement boiling, *Experimental Thermal and Fluid Science*, 86 (2017) 224-234.
- [45] R. Jenkins, R. Lupoi, R. Kempers, A.J. Robinson, Heat transfer performance of boiling jet array impingement on micro-grooved surfaces, *Experimental Thermal and Fluid Science*, 80 (2017) 293-304.
- [46] B.B. Kanbur, C. Wu, S. Fan, F. Duan, System-level experimental investigations of the direct immersion cooling data center units with thermodynamic and thermoeconomic assessments, *Energy*, 217 (2021) 119373.
- [47] K.C. Leong, J.Y. Ho, K.K. Wong, A critical review of pool and flow boiling heat transfer of dielectric fluids on enhanced surfaces, *Applied Thermal Engineering*, 112 (2017) 999-1019.
- [48] G. Liang, I. Mudawar, Review of pool boiling enhancement by surface modification, *International Journal of Heat and Mass Transfer*, 128 (2019) 892-933.
- [49] D.W. Zhou, C.F. Ma, J. Yu, Boiling hysteresis of impinging circular submerged jets with highly wetting liquids, *International Journal of Heat and Fluid Flow*, 25(1) (2004) 81-90.

- [50] H.J. Sauer, R.A. Medrow, A.M. Sinnarwalla, Effects of surface condition on nucleate boiling of refrigerant-11, ASHRAE Transactions, 81 (1975) 274-281.
- [51] G.T. Kartsounes, A study of the effect of surface treatment on pool boiling heat transfer in refrigerant 12, ASHRAE Transactions, 18 (1975) 320-329.
- [52] J.M.S. Jabardo, C.L. Silva, Modeling of the nucleate boiling of refrigerant-oil mixtures, in: XVIIIth International Congress of Refrigeration, 1991, pp. 514-518.
- [53] J.M. Jabardo, E. Silva, G. Ribatski, S.F. de Barros, Evaluation of the Rohsenow correlation through experimental pool boiling of halocarbon refrigerants on cylindrical surfaces, Journal of the Brazilian Society of Mechanical Sciences and Engineering, 26(2) (2004) 218-230.
- [54] A.R. Schlindwein, F.O. Martin, M. Misale, J.C. Passos, Nucleate boiling of FC-87/FC-72 zeotropic mixtures on a horizontal copper disc, Heat and Mass Transfer, 45(7) (2009) 937-944.
- [55] J.Y. Ho, K.C. Leong, C. Yang, Saturated pool boiling from carbon nanotube coated surfaces at different orientations, International Journal of Heat and Mass Transfer, 79 (2014) 893-904.
- [56] J.Y. Ho, K.K. Wong, K.C. Leong, Saturated pool boiling of FC-72 from enhanced surfaces produced by selective laser melting, International Journal of Heat and Mass Transfer, 99 (2016) 107-121.

- [57] R.S. Bartle, K. Menon, E. Walsh, Pool boiling of resin-impregnated motor windings geometry, *Applied Thermal Engineering*, 130 (2018) 854-864.
- [58] A.A. Watwe, A. Bar-Cohen, A. McNeil, Combined pressure and subcooling effects on pool boiling from a PPGA chip package, in: *InterSociety Conference on Thermal Phenomena in Electronic Systems, I-THERM V*, IEEE, 1996, pp. 284-291.
- [59] K.N. Rainey, S.M. You, S. Lee, Effect of pressure, subcooling, and dissolved gas on pool boiling heat transfer from microporous surfaces in FC-72, *Journal of Heat Transfer*, 125(1) (2003) 75-83.
- [60] K.N. Rainey, S.M. You, S. Lee, Effect of pressure, subcooling, and dissolved gas on pool boiling heat transfer from microporous, square pin-finned surfaces in FC-72, *International Journal of Heat and Mass Transfer*, 46(1) (2003) 23-35.
- [61] B. Liu, J. Liu, J. Zhou, B. Yuan, Y. Zhang, J. Wei, W. Wang, Experimental study of subcooled boiling pool heat transfer and its “hook back” phenomenon on micro/nanostructured surfaces, *International Communications in Heat and Mass Transfer*, 100 (2019) 73-82.
- [62] J. Kim, J.F. Benton, D. Wisniewski, Pool boiling heat transfer on small heaters: effect of gravity and subcooling, *International Journal of Heat and Mass Transfer*, 45(19) (2002) 3919-3932.

- [63] C.D. Henry, J. Kim, A study of the effects of heater size, subcooling, and gravity level on pool boiling heat transfer, *International Journal of Heat and Fluid Flow*, 25(2) (2004) 262-273.
- [64] Z. Cao, B. Liu, C. Preger, Z. Wu, Y. Zhang, X. Wang, M.E. Messing, K. Deppert, J. Wei, B. Sundén, Pool boiling heat transfer of FC-72 on pin-fin silicon surfaces with nanoparticle deposition, *International Journal of Heat and Mass Transfer*, 126 (2018) 1019-1033.
- [65] M. El-Genk, Combined effects of subcooling and surface orientation on pool boiling of HFE-7100 from a simulated electronic chip, *Experimental Heat Transfer*, 16(4) (2003) 281-301.
- [66] M.S. El-Genk, J.L. Parker, Enhanced boiling of HFE-7100 dielectric liquid on porous graphite, *Energy Conversion and Management*, 46(15-16) (2005) 2455-2481.
- [67] J.J. Wei, H. Honda, Effects of fin geometry on boiling heat transfer from silicon chips with micro-pin-fins immersed in FC-72, *International Journal of Heat and Mass Transfer*, 46(21) (2003) 4059-4070.
- [68] Y. Zhang, J. Zhou, W. Zhou, B. Qi, J. Wei, CHF correlation of boiling in FC-72 with micro-pin-fins for electronics cooling, *Applied Thermal Engineering*, 138 (2018) 494-500.

- [69] M. Arik, A. Bar-Cohen, S.M. You, Enhancement of pool boiling critical heat flux in dielectric liquids by microporous coatings, *International Journal of Heat and Mass Transfer*, 50(5-6) (2007) 997-1009.
- [70] C.F. Ma, A.E. Bergles, Jet impingement nucleate boiling, *International Journal of Heat and Mass Transfer*, 29(8) (1986) 1095-1101.
- [71] E.A. Browne, G.J. Michna, M.K. Jensen, Y. Peles, Microjet array single-phase and flow boiling heat transfer with R134a, *International Journal of Heat and Mass Transfer*, 53(23-24) (2010) 5027-5034.
- [72] R. Cardenas, V. Narayanan, Critical heat flux in submerged jet impingement boiling of water under subatmospheric conditions, *Journal of Heat Transfer*, 134(8) (2012) 081502.
- [73] Y. Zhang, J. Wei, X. Kong, L. Guo, Confined submerged jet impingement boiling of subcooled FC-72 over micro-pin-finned surfaces, *Heat Transfer Engineering*, 37(3-4) (2016) 269-278.
- [74] Y. Zhang, B. Liu, J. Wei, B. Sundén, Z. Wu, Heat transfer correlations for jet impingement boiling over micro-pin-finned surface, *International Journal of Heat and Mass Transfer*, 126 (2018) 401-413.
- [75] D. Copeland, Single-phase and boiling cooling of small pin fin arrays by multiple nozzle jet impingement, *Journal of Electronic Packaging*, 118(1) (1996) 21-26.

- [76] I. Mudawar, D.C. Wadsworth, Critical heat flux from a simulated chip to a confined rectangular impinging jet of dielectric liquid, *International Journal of Heat and Mass Transfer*, 34(6) (1991) 1465-1479.
- [77] M.T. Meyer, I. Mudawar, C.E. Boyack, C.A. Hale, Single-phase and two-phase cooling with an array of rectangular jets, *International Journal of Heat and Mass Transfer*, 49(1) (2006) 17-29.
- [78] F.J. Hong, C.Y. Zhang, W. He, P. Cheng, G. Chen, Confined jet array impingement boiling of subcooled aqueous ethylene glycol solution, *International Communications in Heat and Mass Transfer*, 56 (2014) 165-173.
- [79] C. Mira-Hernández, M.D. Clark, J.A. Weibel, S.V. Garimella, Development and validation of a semi-empirical model for two-phase heat transfer from arrays of impinging jets, *International Journal of Heat and Mass Transfer*, 124 (2018) 782-793.
- [80] M.J. Rau, S.V. Garimella, Local two-phase heat transfer from arrays of confined and submerged impinging jets, *International Journal of Heat and Mass Transfer*, 67 (2013) 487-498.
- [81] M.J. Rau, S.V. Garimella, E.M. Dede, S.N. Joshi, Boiling heat transfer from an array of round jets with hybrid surface enhancements, *Journal of Heat Transfer*, 137(7) (2015).

- [82] S.U.S. Choi, Enhancing thermal conductivity of fluids with nanoparticles, *Developments and Applications of Non-Newtonian Flows*, ASME FED 231/MD 66 (1995) 99-105.
- [83] Y.M. Yang, J.R. Maa, Boiling of suspension of solid particles in water, *International Journal of Heat and Mass Transfer*, 27(1) (1984) 145-147.
- [84] S.M. You, J.H. Kim, K.H. Kim, Effect of nanoparticles on critical heat flux of water in pool boiling heat transfer, *Applied Physics Letters*, 83(16) (2003) 3374-3376.
- [85] Y. Hu, H. Li, Y. He, Z. Liu, Y. Zhao, Effect of nanoparticle size and concentration on boiling performance of SiO₂ nanofluid, *International Journal of Heat and Mass Transfer*, 107 (2017) 820-828.
- [86] A. Abdollahi, M.R. Salimpour, N. Etesami, Experimental analysis of magnetic field effect on the pool boiling heat transfer of a ferrofluid, *Applied Thermal Engineering*, 111 (2017) 1101-1110.
- [87] S.Y. Li, W.T. Ji, C.Y. Zhao, H. Zhang, W.Q. Tao, Effects of magnetic field on the pool boiling heat transfer of water-based α -Fe₂O₃ and γ -Fe₂O₃ nanofluids, *International Journal of Heat and Mass Transfer*, 128 (2019) 762-772.
- [88] V. Trisaksri, S. Wongwises, Nucleate pool boiling heat transfer of TiO₂-R141b nanofluids, *International Journal of Heat and Mass Transfer*, 52(5) (2009) 1582-1588.

- [89] P. Naphon, C. Thongjirakul, Pool boiling heat transfer characteristics of refrigerant-nanoparticle mixtures, *International Communications in Heat and Mass Transfer*, 52 (2014) 84-89.
- [90] X. Tang, Y.H. Zhao, Y.h. Diao, Experimental investigation of the nucleate pool boiling heat transfer characteristics of δ -Al₂O₃-R141b nanofluids on a horizontal plate, *Experimental Thermal and Fluid Science*, 52 (2014) 88-96.
- [91] Y.H. Diao, C.Z. Li, Y.H. Zhao, Y. Liu, S. Wang, Experimental investigation on the pool boiling characteristics and critical heat flux of Cu-R141b nanorefrigerant under atmospheric pressure, *International Journal of Heat and Mass Transfer*, 89 (2015) 110-115.
- [92] S.J. Kim, I.C. Bang, J. Buongiorno, L.W. Hu, Surface wettability change during pool boiling of nanofluids and its effect on critical heat flux, *International Journal of Heat and Mass Transfer*, 50(19) (2007) 4105-4116.
- [93] S.M. Kwark, R. Kumar, G. Moreno, J. Yoo, S.M. You, Pool boiling characteristics of low concentration nanofluids, *International Journal of Heat and Mass Transfer*, 53(5) (2010) 972-981.
- [94] J.P. Tu, N. Dinh, T. Theofanous, An experimental study of nanofluid boiling heat transfer, in: *Proceedings of 6th International Symposium on Heat Transfer*, Beijing, China, 2004.

- [95] L. Xu, J. Xu, Nanofluid stabilizes and enhances convective boiling heat transfer in a single microchannel, *International Journal of Heat and Mass Transfer*, 55(21) (2012) 5673-5686.
- [96] D. Wang, X. Quan, C. Liu, P. Cheng, An experimental investigation on periodic single bubble growth and departure from a small heater submerged in a nanofluid containing moderately hydrophilic nanoparticles, *International Communications in Heat and Mass Transfer*, 95 (2018) 1-8.
- [97] X. Quan, D. Wang, P. Cheng, An experimental investigation on wettability effects of nanoparticles in pool boiling of a nanofluid, *International Journal of Heat and Mass Transfer*, 108 (2017) 32-40.
- [98] Q. Li, Y. Xuan, F. Yu, Experimental investigation of submerged single jet impingement using Cu–water nanofluid, *Applied Thermal Engineering*, 36 (2012) 426-433.
- [99] L. Nakharintr, P. Naphon, Magnetic field effect on the enhancement of nanofluids heat transfer of a confined jet impingement in mini-channel heat sink, *International Journal of Heat and Mass Transfer*, 110 (2017) 753-759.
- [100] J. Lee, I. Mudawar, Assessment of the effectiveness of nanofluids for single-phase and two-phase heat transfer in micro-channels, *International Journal of Heat and Mass Transfer*, 50(3) (2007) 452-463.

- [101] L. Dong, X. Quan, P. Cheng, An analysis of surface-microstructures effects on heterogeneous nucleation in pool boiling, *International Journal of Heat and Mass Transfer*, 55(15) (2012) 4376-4384.
- [102] B.J. Jones, J.P. McHale, S.V. Garimella, The influence of surface roughness on nucleate pool boiling heat transfer, *Journal of Heat Transfer*, 131(12) (2009).
- [103] J. Kim, S. Jun, J. Lee, J. Godinez, S.M. You, Effect of surface roughness on pool boiling heat transfer of water on a superhydrophilic aluminum surface, *Journal of Heat Transfer*, 139(10) (2017) 101501.
- [104] S. Gong, P. Cheng, Two-dimensional mesoscale simulations of saturated pool boiling from rough surfaces. Part II: Bubble interactions above multi-cavities, *International Journal of Heat and Mass Transfer*, 100 (2016) 938-948.
- [105] Y.Y. Li, Z.H. Liu, B.C. Zheng, Experimental study on the saturated pool boiling heat transfer on nano-scale modification surface, *International Journal of Heat and Mass Transfer*, 84 (2015) 550-561.
- [106] H.S. Ahn, C. Lee, H. Kim, H. Jo, S. Kang, J. Kim, J. Shin, M.H. Kim, Pool boiling CHF enhancement by micro/nanoscale modification of zircaloy-4 surface, *Nuclear Engineering and Design*, 240(10) (2010) 3350-3360.
- [107] S. Gong, P. Cheng, Lattice Boltzmann simulations for surface wettability effects in saturated pool boiling heat transfer, *International Journal of Heat and Mass Transfer*, 85 (2015) 635-646.

- [108] X. Ma, P. Cheng, 3D simulations of pool boiling above smooth horizontal heated surfaces by a phase-change lattice Boltzmann method, *International Journal of Heat and Mass Transfer*, 131 (2019) 1095-1108.
- [109] A.R. Betz, J. Jenkins, C.J. Kim, D. Attinger, Boiling heat transfer on superhydrophilic, superhydrophobic, and superbiphilic surfaces, *International Journal of Heat and Mass Transfer*, 57(2) (2013) 733-741.
- [110] H. Jo, H.S. Ahn, S. Kang, M.H. Kim, A study of nucleate boiling heat transfer on hydrophilic, hydrophobic and heterogeneous wetting surfaces, *International Journal of Heat and Mass Transfer*, 54(25) (2011) 5643-5652.
- [111] X. Ma, P. Cheng, X. Quan, Simulations of saturated boiling heat transfer on bio-inspired two-phase heat sinks by a phase-change lattice Boltzmann method, *International Journal of Heat and Mass Transfer*, 127 (2018) 1013-1024.
- [112] J. Kim, S. Jun, R. Laksnarain, S.M. You, Effect of surface roughness on pool boiling heat transfer at a heated surface having moderate wettability, *International Journal of Heat and Mass Transfer*, 101 (2016) 992-1002.
- [113] Y.Y. Li, Y.J. Chen, Z.H. Liu, A uniform correlation for predicting pool boiling heat transfer on plane surface with surface characteristics effect, *International Journal of Heat and Mass Transfer*, 77 (2014) 809-817.

- [114] R.J. Benjamin, A.R. Balakrishnan, Nucleate pool boiling heat transfer of pure liquids at low to moderate heat fluxes, *International Journal of Heat and Mass Transfer*, 39(12) (1996) 2495-2504.
- [115] S.G. Kandlikar, A theoretical model to predict pool boiling CHF incorporating effects of contact angle and orientation, *Journal of Heat Transfer*, 123(6) (2001) 1071-1079.
- [116] K.A. Park, A.E. Bergles, Effects of size of simulated microelectronic chips on boiling and critical heat flux, *Journal of Heat Transfer*, 110(3) (1988) 728-734.
- [117] X. Wang, Y. Zhang, B. Qi, J. Zhao, J. Wei, Experimental study of the heater size effect on subcooled pool boiling heat transfer of FC-72 in microgravity, *Experimental Thermal and Fluid Science*, 76 (2016) 275-286.
- [118] E. Baker, Liquid cooling of microelectronic devices by free and forced convection, *Microelectronics Reliability*, 11(2) (1972) 213-222.
- [119] M.C. Lu, C.H. Huang, C.T. Huang, Y.C. Chen, A modified hydrodynamic model for pool boiling CHF considering the effects of heater size and nucleation site density, *International Journal of Thermal Sciences*, 91 (2015) 133-141.
- [120] S. Mori, S.M. Aznam, K. Okuyama, Enhancement of the critical heat flux in saturated pool boiling of water by nanoparticle-coating and a honeycomb porous plate, *International Journal of Heat and Mass Transfer*, 80 (2015) 1-6.

- [121] J.H. Lienhard, V.K. Dhir, Extended hydrodynamic theory of the peak and minimum pool boiling heat fluxes, NASA Report No, CR2270, 1973.
- [122] N. Kaneyasu, F. Yasunobu, U. Satoru, O. Haruhiko, Effect of surface configuration on nucleate boiling heat transfer, International Journal of Heat and Mass Transfer, 27(9) (1984) 1559-1571.
- [123] J.H. Kim, K.H. Kim, S.M. You, Pool boiling heat transfer in saturated nanofluids, in: ASME International Mechanical Engineering Congress and Exposition, American Society of Mechanical Engineers, 2004, pp. 621-628.
- [124] J.H. Lienhard, On the Two Regimes of Nucleate Boiling, Journal of Heat Transfer, 107(1) (1985) 262-264.
- [125] S. Jun, J. Kim, S.M. You, H.Y. Kim, Effect of heater orientation on pool boiling heat transfer from sintered copper microporous coating in saturated water, International Journal of Heat and Mass Transfer, 103 (2016) 277-284.
- [126] K.N. Rainey, S.M. You, Effects of heater size and orientation on pool boiling heat transfer from microporous coated surfaces, International Journal of Heat and Mass Transfer, 44(14) (2001) 2589-2599.
- [127] M.S. El-Genk, A. Suszko, Saturation nucleate boiling and correlations for PF-5060 dielectric liquid on inclined rough copper surfaces, Journal of Heat Transfer, 136(8) (2014).

- [128] T. Kim, J.M. Kim, J.H. Kim, S.C. Park, H.S. Ahn, Orientation effects on bubble dynamics and nucleate pool boiling heat transfer of graphene-modified surface, *International Journal of Heat and Mass Transfer*, 108 (2017) 1393-1405.
- [129] M. Dadjoo, N. Etesami, M.N. Esfahany, Influence of orientation and roughness of heater surface on critical heat flux and pool boiling heat transfer coefficient of nanofluid, *Applied Thermal Engineering*, 124 (2017) 353-361.
- [130] Y.H. Kim, K.Y. Suh, One-dimensional critical heat flux concerning surface orientation and gap size effects, *Nuclear Engineering and Design*, 226(3) (2003) 277-292.
- [131] K.A. Joudi, D.D. James, Surface contamination, rejuvenation, and the reproducibility of results in nucleate pool boiling, *Journal of Heat Transfer*, 103(3) (1981) 453-458.
- [132] K.N. Rainey, S.M. You, Pool boiling heat transfer from plain and microporous, square pin-finned surfaces in saturated FC-72, *Journal of Heat Transfer*, 122(3) (2000) 509-516.
- [133] K.H. Chu, R. Enright, E.N. Wang, Structured surfaces for enhanced pool boiling heat transfer, *Applied Physics Letters*, 100(24) (2012) 241603.
- [134] L. Dong, X. Quan, P. Cheng, An experimental investigation of enhanced pool boiling heat transfer from surfaces with micro/nano-structures, *International Journal of Heat and Mass Transfer*, 71 (2014) 189-196.

- [135] H. Honda, J.J. Wei, Enhanced boiling heat transfer from electronic components by use of surface microstructures, *Experimental Thermal and Fluid Science*, 28(2) (2004) 159-169.
- [136] Y.W. Lu, S.G. Kandlikar, Nanoscale surface modification techniques for pool boiling enhancement—A critical review and future directions, *Heat Transfer Engineering*, 32(10) (2011) 827-842.
- [137] C.M. Patil, S.G. Kandlikar, Review of the manufacturing techniques for porous surfaces used in enhanced pool boiling, *Heat Transfer Engineering*, 35(10) (2014) 887-902.
- [138] S. Mori, Y. Utaka, Critical heat flux enhancement by surface modification in a saturated pool boiling: A review, *International Journal of Heat and Mass Transfer*, 108 (2017) 2534-2557.
- [139] X. Ma, P. Cheng, Dry spot dynamics and wet area fractions in pool boiling on micro-pillar and micro-cavity hydrophilic heaters: A 3D lattice Boltzmann phase-change study, *International Journal of Heat and Mass Transfer*, 141 (2019) 407-418.
- [140] F.L. Cui, F.J. Hong, P. Cheng, Comparison of normal and distributed jet array impingement boiling of HFE-7000 on smooth and pin-fin surfaces, *International Journal of Heat and Mass Transfer*, 126 (2018) 1287-1298.

- [141] F. Cui, F. Hong, C. Li, P. Cheng, Two-phase flow instability in distributed jet array impingement boiling on pin-fin structured surface and its affecting factors, *International Journal of Heat and Mass Transfer*, 143 (2019) 118495.
- [142] M.J. Rau, S.V. Garimella, Confined jet impingement with boiling on a variety of enhanced surfaces, *Journal of Heat Transfer*, 136(10) (2014) 101503.
- [143] S.N. Joshi, E.M. Dede, Two-phase jet impingement cooling for high heat flux wide band-gap devices using multi-scale porous surfaces, *Applied Thermal Engineering*, 110 (2017) 10-17.
- [144] S. Jun, S. Sinha-Ray, A.L. Yarin, Pool boiling on nano-textured surfaces, *International Journal of Heat and Mass Transfer*, 62 (2013) 99-111.
- [145] C. Li, Z. Wang, P.I. Wang, Y. Peles, N. Koratkar, G.P. Peterson, Nanostructured copper interfaces for enhanced boiling, *Small*, 4(8) (2008) 1084-1088.
- [146] R. Chen, M.C. Lu, V. Srinivasan, Z. Wang, H.H. Cho, A. Majumdar, Nanowires for enhanced boiling heat transfer, *Nano Letters*, 9(2) (2009) 548-553.
- [147] M. Dharmendra, S. Suresh, C.S. Sujith Kumar, Q. Yang, Pool boiling heat transfer enhancement using vertically aligned carbon nanotube coatings on a copper substrate, *Applied Thermal Engineering*, 99 (2016) 61-71.

- [148] S.G. Liter, M. Kaviany, Pool-boiling CHF enhancement by modulated porous-layer coating: theory and experiment, *International Journal of Heat and Mass Transfer*, 44(22) (2001) 4287-4311.
- [149] L. Bai, L. Zhang, G. Lin, G.P. Peterson, Pool boiling with high heat flux enabled by a porous artery structure, *Applied Physics Letters*, 108(23) (2016) 233901.
- [150] K. Zhang, L. Bai, G. Lin, H. Jin, D. Wen, Experimental study on pool boiling in a porous artery structure, *Applied Thermal Engineering*, 149 (2019) 377-384.
- [151] P.A. Raghupathi, S.G. Kandlikar, Bubble growth and departure trajectory under asymmetric temperature conditions, *International Journal of Heat and Mass Transfer*, 95 (2016) 824-832.
- [152] M.M. Rahman, J. Pollack, M. McCarthy, Increasing boiling heat transfer using low conductivity materials, *Scientific Reports*, 5(1) (2015) 13145.
- [153] C.H. Li, T. Li, P. Hodgins, C.N. Hunter, A.A. Voevodin, J.G. Jones, G.P. Peterson, Comparison study of liquid replenishing impacts on critical heat flux and heat transfer coefficient of nucleate pool boiling on multiscale modulated porous structures, *International Journal of Heat and Mass Transfer*, 54(15) (2011) 3146-3155.
- [154] C. Zhang, L. Zhang, H. Xu, P. Li, B. Qian, Performance of pool boiling with 3D grid structure manufactured by selective laser melting technique, *International Journal of Heat and Mass Transfer*, 128 (2019) 570-580.

- [155] S.G. Kandlikar, Controlling bubble motion over heated surface through evaporation momentum force to enhance pool boiling heat transfer, *Applied Physics Letters*, 102(5) (2013) 051611.
- [156] D. Cooke, S.G. Kandlikar, Pool boiling heat transfer and bubble dynamics over plain and enhanced microchannels, *Journal of Heat Transfer*, 133(5) (2011) 052902.
- [157] D. Cooke, S.G. Kandlikar, Effect of open microchannel geometry on pool boiling enhancement, *International Journal of Heat and Mass Transfer*, 55(4) (2012) 1004-1013.
- [158] A. Jaikumar, S.G. Kandlikar, Ultra-high pool boiling performance and effect of channel width with selectively coated open microchannels, *International Journal of Heat and Mass Transfer*, 95 (2016) 795-805.
- [159] C.M. Patil, S.G. Kandlikar, Pool boiling enhancement through microporous coatings selectively electrodeposited on fin tops of open microchannels, *International Journal of Heat and Mass Transfer*, 79 (2014) 816-828.
- [160] A. Jaikumar, S.G. Kandlikar, Enhanced pool boiling for electronics cooling using porous fin tops on open microchannels with FC-87, *Applied Thermal Engineering*, 91 (2015) 426-433.
- [161] A. Jaikumar, S.G. Kandlikar, Pool boiling enhancement through bubble induced convective liquid flow in feeder microchannels, *Applied Physics Letters*, 108(4) (2016) 041604.

- [162] S.G. Kandlikar, Enhanced macroconvection mechanism with separate liquid–vapor pathways to improve pool boiling performance, *Journal of Heat Transfer*, 139(5) (2017) 051501.
- [163] L. Yuan, F. Hong, P. Cheng, Pool boiling enhancement through a guidance structure mounted above heating surface, *International Journal of Heat and Mass Transfer*, 139 (2019) 751-763.
- [164] H.M. Kurihara, J.E. Myers, The effects of superheat and surface roughness on boiling coefficients, *AIChE Journal*, 6(1) (1960) 83-91.
- [165] A. Luke, Pool boiling heat transfer from horizontal tubes with different surface roughness, *International Journal of Refrigeration*, 20(8) (1997) 561-574.
- [166] A.S. Moita, E. Teodori, A.L.N. Moreira, Influence of surface topography in the boiling mechanisms, *International Journal of Heat and Fluid Flow*, 52 (2015) 50-63.
- [167] R.J. Benjamin, A.R. Balakrishnan, Nucleation site density in pool boiling of saturated pure liquids: Effect of surface microroughness and surface and liquid physical properties, *Experimental Thermal and Fluid Science*, 15(1) (1997) 32-42.
- [168] V.I. Kozitskii, Heat transfer coefficients for boiling of n-butane on surfaces of various roughness, *Chemical and Petroleum Engineering*, 8(1) (1972) 23-24.
- [169] J.M. Jabardo, An overview of surface roughness effects on nucleate boiling heat transfer, *The Open Conservation Biology Journal*, 2(1) (2010) 24–34.

- [170] P.J. Berenson, Experiments on pool-boiling heat transfer, *International Journal of Heat and Mass Transfer*, 5(10) (1962) 985-999.
- [171] S.K.R. Chowdhury, R.H.S. Winterton, Surface effects in pool boiling, *International Journal of Heat and Mass Transfer*, 28(10) (1985) 1881-1889.
- [172] K. Ferjančič, I. Golobič, Surface effects on pool boiling CHF, *Experimental Thermal and Fluid Science*, 25(7) (2002) 565-571.
- [173] H. O'Hanley, C. Coyle, J. Buongiorno, T. McKrell, L.W. Hu, M. Rubner, R. Cohen, Separate effects of surface roughness, wettability, and porosity on the boiling critical heat flux, *Applied Physics Letters*, 103(2) (2013) 024102.
- [174] Y. Qi, J.F. Klausner, R. Mei, Role of surface structure in heterogeneous nucleation, *International Journal of Heat and Mass Transfer*, 47(14) (2004) 3097-3107.
- [175] A. Luke, Active and potential bubble nucleation sites on different structured heated surfaces, *Chemical Engineering Research and Design*, 82(4) (2004) 462-470.
- [176] A. Luke, Preparation and analysis of different roughness structures for evaporator tubes, *Heat and Mass Transfer*, 45(7) (2009) 909-917.
- [177] K.T. Hong, H. Imadojemu, R.L. Webb, Effects of oxidation and surface roughness on contact angle, *Experimental Thermal and Fluid Science*, 8(4) (1994) 279-285.
- [178] H. Nakae, R. Inui, Y. Hirata, H. Saito, Effects of surface roughness on wettability, *Acta Materialia*, 46(7) (1998) 2313-2318.

- [179] S.J. Kline, The purposes of uncertainty analysis, *Journal of Fluids Engineering*, 107 (1985) 153-160.
- [180] Y. Yang, X. Ji, J. Xu, Pool boiling heat transfer on copper foam covers with water as working fluid, *International Journal of Thermal Sciences*, 49(7) (2010) 1227-1237.
- [181] A. Suriyawong, S. Wongwises, Nucleate pool boiling heat transfer characteristics of TiO₂-water nanofluids at very low concentrations, *Experimental Thermal and Fluid Science*, 34(8) (2010) 992-999.
- [182] A. Luke, Preparation, measurement and analysis of the microstructure of evaporator surfaces, *International Journal of Thermal Sciences*, 45(3) (2006) 237-256.
- [183] D. Gorenflo, D. Kenning, H₂ Pool Boiling, in, Springer, Berlin Heidelberg, 2010, pp. 757-792.
- [184] H. Sakashita, CHF and near-wall boiling behaviors in pool boiling of water on a heating surface coated with nanoparticles, *International Journal of Heat and Mass Transfer*, 55(23) (2012) 7312-7320.
- [185] S. Iijima, Helical microtubules of graphitic carbon, *Nature*, 354(6348) (1991) 56-58.
- [186] S. Ujereh, T. Fisher, I. Mudawar, Effects of carbon nanotube arrays on nucleate pool boiling, *International Journal of Heat and Mass Transfer*, 50(19) (2007) 4023-4038.

- [187] V. Sathyamurthi, H.S. Ahn, D. Banerjee, S.C. Lau, Subcooled pool boiling experiments on horizontal heaters coated with carbon nanotubes, *Journal of Heat Transfer*, 131(7) (2009) 071501.
- [188] G.U. Kumar, K. Soni, S. Suresh, K. Ghosh, M. Thansekhar, P.D. Babu, Modified surfaces using seamless graphene/carbon nanotubes based nanostructures for enhancing pool boiling heat transfer, *Experimental Thermal and Fluid Science*, 96 (2018) 493-506.
- [189] S. Launay, A.G. Fedorov, Y. Joshi, A. Cao, P.M. Ajayan, Hybrid micro-nano structured thermal interfaces for pool boiling heat transfer enhancement, *Microelectronics Journal*, 37(11) (2006) 1158-1164.
- [190] G.H. Seo, H. Hwang, J. Yoon, T. Yeo, H.H. Son, U. Jeong, G. Jeun, W. Choi, S.J. Kim, Enhanced critical heat flux with single-walled carbon nanotubes bonded on metal surfaces, *Experimental Thermal and Fluid Science*, 60 (2015) 138-147.
- [191] U. Diebold, Structure and properties of TiO₂ surfaces: a brief review, *Applied Physics A*, 76(5) (2003) 681-687.
- [192] J. Xu, M. Yang, J. Xu, X. Ji, Vertically oriented TiO₂ nanotube arrays with different anodization times for enhanced boiling heat transfer, *Science China Technological Sciences*, 55(8) (2012) 2184-2190.
- [193] J. Prakash Chakrapani Gunarasan, P. Ravindran, Significance of chemical engineering in surface wettability tuning and its boiling hydrodynamics: A boiling heat transfer study, *Industrial & Engineering Chemistry Research*, 59(10) (2020) 4210-4218.

- [194] B.J. Zhang, R. Ganguly, K.J. Kim, C.Y. Lee, Control of pool boiling heat transfer through photo-induced wettability change of titania nanotube arrayed surface, *International Communications in Heat and Mass Transfer*, 81 (2017) 124-130.
- [195] L. Sun, S. Zhang, X.W. Sun, X. Wang, Y. Cai, Double-sided anodic titania nanotube arrays: A lopsided growth process, *Langmuir*, 26(23) (2010) 18424-18429.
- [196] L. Sun, X. Wang, M. Li, S. Zhang, Q. Wang, Anodic titania nanotubes grown on titanium tubular electrodes, *Langmuir*, 30(10) (2014) 2835-2841.
- [197] L. Sun, S. Zhang, X. Wang, X.W. Sun, D.Y. Ong, A.K. Ko Kyaw, A novel parallel configuration of dye-sensitized solar cells with double-sided anodic nanotube arrays, *Energy & Environmental Science*, 4(6) (2011) 2240-2248.
- [198] S.J. Thiagarajan, R. Yang, C. King, S. Narumanchi, Bubble dynamics and nucleate pool boiling heat transfer on microporous copper surfaces, *International Journal of Heat and Mass Transfer*, 89 (2015) 1297-1315.
- [199] P.F. Alvariño, M.L.S. Simón, M. dos Santos Guzella, J.M.A. Paz, J.M.S. Jabardo, L.C. Gómez, Experimental investigation of the CHF of HFE-7100 under pool boiling conditions on differently roughened surfaces, *International Journal of Heat and Mass Transfer*, 139 (2019) 269-279.
- [200] J.H. Lienhard, V.K. Dhir, Hydrodynamic prediction of peak pool-boiling heat fluxes from finite bodies, *Journal of Heat Transfer*, 95(2) (1973) 152-158.

- [201] J. Xu, T.S. Fisher, Enhancement of thermal interface materials with carbon nanotube arrays, *International Journal of Heat and Mass Transfer*, 49(9) (2006) 1658-1666.
- [202] N. Na, K. Hasegawa, X. Zhou, M. Nihei, S. Noda, Denser and taller carbon nanotube arrays on Cu foils useable as thermal interface materials, *Japanese Journal of Applied Physics*, 54(9) (2015) 095102.
- [203] W.X. Chu, P.H. Tseng, C.C. Wang, Utilization of low-melting temperature alloy with confined seal for reducing thermal contact resistance, *Applied Thermal Engineering*, 163 (2019) 114438.
- [204] S. Fan, F. Duan, A review of two-phase submerged boiling in thermal management of electronic cooling, *International Journal of Heat and Mass Transfer*, 150 (2020) 119324.
- [205] D.E. Hall, F.P. Incropera, R. Viskanta, Jet impingement boiling from a circular free-surface jet during quenching: part 2—two-phase jet, *Journal of Heat Transfer*, 123(5) (2001) 911-917.
- [206] S. Bolduc, L.I. Kiss, Sensitivity study of the influence of the water boiling parameters on aluminum semi-continuous DC casting, *International Journal of Thermal Sciences*, 151 (2020) 106276.

- [207] C. Bartoli, P. Di Marco, S. Faggiani, Heat transfer and flow pattern at a cylinder impinged by a slot jet during incipient and nucleate boiling, *Experimental Thermal and Fluid Science*, 15(2) (1997) 101-108.
- [208] D.W. Zhou, C.F. Ma, Local jet impingement boiling heat transfer with R113, *Heat and Mass Transfer*, 40(6) (2004) 539-549.
- [209] H. Sun, C.F. Ma, W. Nakayama, Local characteristics of convective heat transfer from simulated microelectronic chips to impinging submerged round water jets, *Journal of Electronic Packaging*, 115 (1993) 71-77.
- [210] K. Choo, B.K. Friedrich, A.W. Glaspell, K.A. Schilling, The influence of nozzle-to-plate spacing on heat transfer and fluid flow of submerged jet impingement, *International Journal of Heat and Mass Transfer*, 97 (2016) 66-69.
- [211] S.V. Garimella, R.A. Rice, Confined and submerged liquid jet impingement heat transfer, *Journal of Heat Transfer*, 117(4) (1995) 871-877.
- [212] S.V. Garimella, B. Nenaydykh, Nozzle-geometry effects in liquid jet impingement heat transfer, *International Journal of Heat and Mass Transfer*, 39(14) (1996) 2915-2923.
- [213] B. Wang, Y. Chen, J. Cai, J. Zhao, C. Ma, Experimental investigation of circular submerged jet impingement heat transfer with mixed molten salt, *Experimental Thermal and Fluid Science*, 98 (2018) 30-37.

- [214] Y.Y. Li, Y.J. Chen, Z.H. Liu, Correlations for boiling heat transfer characteristics of high-velocity circular jet impingement on the nano-characteristic stagnation zone, *International Journal of Heat and Mass Transfer*, 72 (2014) 177-185.
- [215] P. Griffith, J.D. Wallis, The role of surface conditions in nucleate boiling, *Chemical Engineering Progress Symposium Series*, 56 (1960) 49-63.