Study of heat transfer enhancement surfaces generated by additive manufacturing techniques

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**Study of Heat Transfer Enhancement Surfaces Generated
by Additive Manufacturing Techniques**

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**SCHOOL OF MECHANICAL AND AEROSPACE
ENGINEERING
NANYANG TECHNOLOGICAL UNIVERSITY**

January 2019

**Study of Heat Transfer Enhancement Surfaces
Generated by Additive Manufacturing Techniques**

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**SCHOOL OF MECHANICAL AND
AEROSPACE ENGINEERING**

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

January 2019

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.

23 Jan 2019
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Date

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Wong Kin Keong

Supervisor Declaration Statement

I have reviewed the content and presentation style of this thesis and declare that 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.

23 Jan 2019

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Date

Leong

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Prof. Leong Kai Choong

Authorship Attribution Statement

This thesis contains material from one (1) paper published in the following peer-reviewed journal where I was the first author.

Chapter 4 is published as Wong, K.K. and Leong, K.C. Saturated pool boiling enhancement using porous lattice structures produced by Selective Laser Melting, *International Journal of Heat and Mass Transfer*, Vol. 121, pp. 46 - 63, 2018.

The contributions of the co-authors are as follows:

- Prof. Leong Kai Choong provided the initial project direction and edited the manuscript drafts.
- I prepared the manuscript drafts.
- I performed all the laboratory work at the School of Mechanical and Aerospace Engineering, NTU. I also analysed the data.
- The revision to the manuscript was undertaken together by me and Prof. Leong.

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Abstract

Thermal management in electronic devices is becoming increasingly challenging due to the higher heat output from the increased packing density of transistors. Nucleate boiling has been shown to be an effective cooling scheme for high heat flux removal. In the recent years, the use of commercial porous foams has been demonstrated to be highly potential for applications in nucleate boiling applications due to its high surface-to-volume ratio. However, there are limitations in terms of pore design and challenges of random pore distribution and poor inter-connectivity. These issues can be overcome by fabricating metallic engineered porous structures using the Selective Laser Melting (SLM) technique, which is a type of additive manufacturing.

The objectives of this thesis are to investigate the nucleate pool and flow boiling enhancements using engineered three-dimensional (3D) porous substrates. The use of SLM allows highly customised and complex porous substrates to be fabricated with high dimensional accuracy. Heat transfer performances of these substrates were then investigated using a thermosyphon for pool boiling and a two-phase flow facility for flow boiling. Dielectric fluid FC-72 was used as the coolant due to its electrical and chemical compatibility with electronic devices.

For pool boiling, two different designs of porous structures with octet-truss and re-entrant geometries were fabricated. For the octet-truss substrates, effects of unit cell size and substrate height were investigated. The experimental results showed enhanced nucleate boiling heat transfer due to the increased nucleate site density and capillary-assisted suction. For the re-entrant substrates, effects of channel width and configuration were investigated. The design showed potential

to reduce the problem of liquid-vapour counter-flow at high heat fluxes. Using high speed photography, the boiling mechanisms in 3D porous structures were studied and analysed too.

The bubble dynamics were investigated for the re-entrant substrates in the isolated bubble regime. Experimental measurements of the bubble departure diameter, frequency and nucleation site density were obtained for substrates of different channel widths. A semi-empirical model was developed for the bubble departure diameter to further the understanding of nucleate boiling heat transfer mechanism in the re-entrant substrates. Comparison of the model with experimental results showed good agreement.

For flow boiling, engineered 3D substrates with hollow spherical features were fabricated. Three substrates of small, large and gradient unit cells were tested in a two-phase flow facility. For the substrate with gradient unit cells, experiments were conducted with the unit cells varying from small-to-big (Gradient-forward) and big-to-small (Gradient-reverse) with respect to the flow direction. Effects of mass flux and substrate type were investigated. Using high speed photography, it was shown that the heat transfer mechanism was nucleate boiling as opposed to convective boiling dominated. Overall, the Gradient-reverse substrate showed superior performance due to its ability to maintain churn flow and delay dry-out in the channels.

It is hoped that the results of this thesis can be used as guidelines in the design and fabrication of engineered porous 3D structures for high heat flux electronic cooling systems.

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Nomenclature

A	area (m^2)
Bo	Bond number
C_p	specific heat ($\text{J/kg}\cdot\text{K}$)
CHF	Critical Heat Flux (W/cm^2)
C_{sf}	surface-fluid constant
D_B	diameter of bubble departure (m)
D_p	diameter of pore (m)
D_s	diameter of strut (m)
E_d	volumetric energy density (W/m^3)
F	force (N)
f_B	frequency of bubble departure (s^{-1})
g	gravitational acceleration (m/s^2)
G	mass flux ($\text{kg/m}^2\cdot\text{s}$)
h	heat transfer coefficient ($\text{W/m}^2\cdot\text{K}$)
H	laser hatch spacing (m)
h_{fg}	latent heat of vaporisation (J/kg)
I	current (A)
Ja	Jakob number
k	thermal conductivity ($\text{W/m}\cdot\text{K}$)
L	length (m)
$\dot{m}$	mass flow rate (kg/s)
P	pressure (Pa)
PPI	pores per inch (inch^{-1})
Pr	Prandtl number
q	power (W)
q''	heat flux (W/cm^2)
R	thermal resistance (K/W)
R_b	radius of bubble (m)
R_p	radius of pore (m)
Re	Reynolds number
T	temperature (K), or layer thickness (m)

ΔT_{sat}	superheat (K)
V	voltage (V)
V_B	volume of bubble (m ³)
$\dot{V}$	volumetric flow rate (m ³ /s)
x	vapour quality

Greek

α	thermal diffusivity (m ² /s)
ε	porosity
η	relative enhancement
ρ	density (kg/m ³)
σ	surface tension (N/m)
θ	contact angle (°)
μ	dynamic viscosity (kg/m ² ·s)
ν	kinematic viscosity (m ² /s)
ϕ	solid fraction

Subscripts

al	aluminium
ave	average
b	buoyancy
cu	copper
cp	contact pressure
d	drag
ep	epoxy
f	fluid
fb	flow boiling
h	heated surface
i	inertia
l	liquid
L	single-phase liquid contribution
out	outlet
$pool$	pool boiling contribution

<i>sat</i>	saturation
<i>s</i>	surface, or solid
<i>TC</i>	thermal compound
<i>v</i>	vapour
<i>w</i>	wall

Publications

A. Publications Arising from this Thesis

Journal Papers

- [1] Wong, K.K. and Leong, K.C. Saturated pool boiling enhancement using porous lattice structures produced by Selective Laser Melting, *International Journal of Heat and Mass Transfer*, Vol. 121, pp. 46 - 63, 2018.
- [2] Leong, K.C, Ho, J.Y. and Wong, K.K., A critical review of pool and flow boiling heat transfer of dielectric fluids on enhanced surfaces, *Applied Thermal Engineering*, Vol. 112, pp. 999 -1019, 2017.
- [3] Ho, J.Y., Wong, K.K. and Leong, K.C., Saturated pool boiling of FC-72 from enhanced surfaces produced by Selective Laser Melting, *International Journal of Heat and Mass Transfer*, Vol. 99, pp. 107-121, 2016.

Conference Papers

- [1] Wong, K.K., Leong, K.C, Tor, S.B., Heat transfer enhancement of surfaces for pool boiling using additive manufacturing, *Proceedings of the 1st Thermal and Fluid Engineering Summer Conference*, TFESC, August 9-12, 2015, New York City, USA.
- [2] Ho, J.Y., Wong, K.K., Leong, K.C., Yang, C., Nucleate pool boiling from selective laser melted microgrooves/microcavities surfaces with HFE-7000, *Proceedings of the 1st Thermal and Fluid Engineering Summer Conference*, TFESC, August 9-12, 2015, New York City, USA.

- [3] Ho, J.Y., Wong, K.K., Leong, K.C. and Yang, C., Enhanced nucleate pool boiling from microstructured surfaces fabricated by Selective Laser Melting, *Proceedings of the ASME 2016 5th International Conference on Micro/Nanoscale Heat and Mass Transfer*, January 4–6, 2016, Biopolis, Singapore.
- [4] Wong, K.K., Leong, K.C., Pool boiling enhancement of porous structures fabricated by selective laser melting, *Proceedings of the International Heat Transfer Symposium 2016, IHT2016*, Jun 27-29, 2016, The University of Nottingham, UK.
- [5] Leong, K.C., Wong, K.K., Enhanced flow boiling heat transfer using 3D structures fabricated by selective laser melting, *Proceedings of the International Heat Transfer Symposium 2016, IHT2016*, Jun 27-29, 2016, The University of Nottingham, UK.
- [6] Wong, K.K., Leong, K.C., Pool boiling of saturated FC-72 using spherical hollow structures fabricated by Selective Laser Melting, *Proceedings of the 3rd Thermal and Fluids Engineering Conference*, TFEC2018, Mar 4-7, 2018, Fort Lauderdale, U.S.A.

B. Publications Related to this Thesis – Journal and Conference Papers

- [1] Ho, J.Y., Wong, K.K., Leong, K.C. and Wong, T.N., Convective heat transfer performance of airfoil heat sinks fabricated by selective laser melting. *International Journal of Thermal Sciences*, Vol. 114, pp. 213-228, 2017.

- [2] Wong, K.K., Ho, J.Y., Leong, K.C. and Wong, T.N., Fabrication of heat sinks by Selective Laser Melting for convective heat transfer applications. *Virtual and Physical Prototyping*, Vol. 11(3), pp. 159-165, 2016.

- [3] J.Y. Ho, K.K. Wong, K.C. Leong, T.N. Wong and S.B. Tor, Customised design of a water-cooled cold plate fabricated by Selective Laser Melting, *Proceedings of the 3rd International Conference on Progress in Additive Manufacturing, Pro-AM 2018*, May 14-17, 2018, Singapore.

- [4] Wong, K.K., Ho, J.Y., Leong, K.C. and Wong, T.N., Convective heat transfer performance of staggered heat sink arrays fabricated by Selective Laser Melting, *Proceedings of the 2nd International Conference on Progress in Additive Manufacturing, Pro-AM 2016*, May 16-19, 2016, Singapore.

Chapter 1 - Introduction

This chapter provides the background, motivation, objectives and scope of the thesis. The background highlights the current challenges in high heat-flux electronic cooling and illustrates the potential of immersion cooling to solve the challenges of heat removal. The motivation of designing and fabricating novel 3D engineered porous substrates for high-heat flux cooling is then discussed. This can be achieved with the advances in additive manufacturing. Finally, the organisation of the thesis is presented to provide an overview of the content in this thesis.

1.1 Background and motivation

1.1.1 Challenges in high heat flux electronics cooling

Electronic devices and equipment have become an integral part of our current society, with applications ranging from personal entertainment to maintaining international commerce and defense. Thermal management of electronic devices is becoming increasingly challenging due to the exponential increase in transistor densities over the past decades. As projected by Moore's Law [1], the transistor count doubles almost every two years. The ability to pack more transistors into smaller spaces on electronic chips has caused ever-increasing heat flux generation. Bar-Cohen et al. [2] stated that for every 2°C temperature rise, the reliability of a silicon chip decreases by about 10%. Hence, it is of utmost importance to maintain a low surface temperature on the electronic chip despite higher heat flux output.

1.1.2 Development of advanced cooling solutions

Air cooling has been the common method of heat dissipation due to the ease of application. Turbulators were employed on printed circuit cards to enhance heat transfer at a fixed flow rate by disrupting the boundary layers [3]. While forced air convection is commonly used to dissipate the excess heat produced by the chips, it is not viable to use it to keep up with the increasing demand for high heat-flux applications. Single-phase indirect water-cooling systems were later developed to enhance heat transfer. A Coolant Distribution Unit was developed to distribute cooling water to electronic frames and heat transfer was enabled via cold plates and heat exchanger [3].

The magnitude of heat transfer by various modes is shown in Figure 1-1. As discussed by Mudawar [4], forced air convection is suitable to remove 0.1 W/cm^2 of heat flux while maintaining the temperature of electronic chips below industry limit of 85°C . Single-phase forced convection using water can remove heat fluxes of up to 10 W/cm^2 , whereas phase change heat transfer can achieve heat fluxes as high as 100 W/cm^2 by using dielectric coolants.

Figure 1-1 Magnitude of heat transfer from various modes of heat transfer [4]

Direct immersion cooling was developed to accommodate higher heat fluxes coupled with the elimination of most internal thermal resistances [5], as shown

in Figure 1-2. Although water possesses a higher heat transfer coefficient than fluorochemical liquids, it is not compatible with electronic devices for direct immersion cooling due to the problem of electrical short-circuit. On the other hand, fluorochemical liquids, also known as dielectric fluids, are not electrically conductive and proved to be highly suitable as a liquid medium for direct immersion cooling. Some examples of common dielectric fluids are FC-72 and HFC-7000.

Figure 1-2 Direct immersion cooling examples [6, 7]

Systems using phase-change heat transfer in immersion cooling are classified as nucleate pool and flow boiling. Boiling generally occurs when the surface temperature is above the saturated temperature of the fluid. Pool boiling occurs due to buoyancy of the bubbles in the absence of a pump; whereas flow boiling occurs under the action of a pump to circulate the fluid in a channel. When the heat flux increases, the rate of bubble generation increases and tends to cause coalescence near the heated surface. Eventually, a phenomenon known as Critical Heat Flux (CHF) occurs, when the vapour forms a film at the heated surface and impedes fluid replenishment. This usually causes a huge increase in the surface temperature due to the lowered heat transfer rate and may lead to failure in the case of electronic chips. Thus, Mudawar [4] suggested that the CHF constitutes the upper heat flux design limit for any phase change cooling system.

1.1.3 Motivation of research

A comprehensive literature review of the current advances in nucleate boiling enhancement is presented in Chapter 2, while a brief overview is discussed in this section. Researchers have explored various surface modification techniques to enhance the heat transfer in pool and flow boiling. Some examples include the use of micro-fins and porous coatings, which increase the surface area and nucleation site density. More recently, there have been much interest in the use of porous foams for nucleate boiling heat transfer. These porous foams possess high surface-area-to-volume ratio and allows good fluid mixing. However, these commercial foams tend to have limited pore structure designs, random pore size distribution and inter-connectivity issues.

Therefore, the motivation of this thesis is to design and fabricate 3D porous substrates which surpass the current manufacturing limits of commercial foams. This can be achieved by the emerging field of additive manufacturing, which will be discussed in the following section.

1.1.4 Advances in additive manufacturing

Additive Manufacturing (AM), or more commonly known as 3D printing, first emerged in the 1980s under the classification of Rapid Prototyping technologies and have evolved greatly over the past decades [8]. By the definition of American Society for Testing and Materials (ASTM) international standard, F2792-12a [9], the term AM refers to processes that produce a three-dimensional product by adding material successively in a layer-by-layer method. The key advantages of AM processes are their ability to produce parts with complex geometries, elimination of expensive tooling dies and shorter production time [10].

With the advancement in technology, a wider range of material-processing capabilities has been developed. The Selective Laser Melting (SLM) technique has been identified as a potential game-changer in the manufacturing sector by Zhang et al. [11] due to its superior technical features. It allows the processing of numerous metal alloys starting from a powder base. By using aluminium alloy AlSi10Mg, it is possible to fabricate heat transfer substrates which are lightweight and possess high thermal conductivity.

In this thesis, SLM is used as a manufacturing technique to fabricate heat transfer substrates of intricate 3D geometries to improve nucleate boiling heat transfer.

1.2 Objectives and scope

The objectives of this thesis are to investigate the nucleate pool and flow boiling enhancements using engineered 3D porous substrates with FC-72 as the working fluid. The substrates are fabricated using the Selective Laser Melting (SLM) technique, which allows metallic substrates with high degree of design freedom to be built. Fundamental understanding of the boiling mechanisms in the fabricated 3D engineered porous substrates will be studied. The scope of this research includes

1. the design and fabrication of heat transfer substrates for boiling applications using SLM,
2. characterisation of physical properties of heat transfer substrates,
3. comprehensive experimental studies in a thermosyphon for pool boiling and in a two-phase flow facility for flow boiling, and
4. visualisation and theoretical formulation of heat transfer mechanisms in 3D porous substrates for pool and flow boiling.

1.3 Organisation of report

The preceding section of this chapter introduces the background and motivation of this thesis. The objectives and scope have also been presented. In Chapter 2, a thorough literature review is presented on the various techniques to enhance nucleate pool and flow boiling. An overview of additive manufacturing and in-depth review of the SLM technique are also shown. Chapter 3 will then showcase the methodology of SLM and characterisation of the preliminary substrates.

Chapter 4 discusses the results of pool boiling of porous lattice substrates which were fabricated using SLM. The pool boiling facility will be presented here. The effects of structure height and unit cell sizes are elaborated in detail. From the visualisation studies, the heat transfer mechanism is proposed and illustrated.

Chapter 5 focuses on the pool boiling of re-entrant substrates fabricated using SLM. The effects of re-entrant channel sizes and configuration will be elaborated. Preliminary studies of the bubble evolution at various heat fluxes are also presented. Chapter 6 presents the theoretical modelling of bubble dynamics from re-entrant substrates. The modelling of bubble departure diameter will be compared with experimental results.

In Chapter 7, a flow boiling study of hollow spherical substrates fabricated using SLM will be discussed. The effects of substrate and fluid mass fluxes will be elaborated. Visualisations on the flow characteristics under different substrates and conditions are presented to illustrate the heat transfer mechanism occurring within the engineered substrates.

Finally, the conclusions of this thesis will be presented in Chapter 8 and the various findings will be highlighted. Recommendations for future work will be duly discussed.

Chapter 2 - Literature Review

In this chapter, the theories and models of nucleate boiling are introduced first, followed by an extensive review of surface modifications used to enhance nucleate pool and flow boiling. Based on the review, the research gaps are identified, and it is shown how additive manufacturing can be employed to fabricate specially engineered three-dimensional (3D) structures to overcome the current limitations.

2.1 Introduction to nucleate boiling

Nucleate boiling heat transfer is a two-phase phenomenon whereby transition from liquid phase to vapour phase occurs at the solid-liquid boundary. This happens when the surface temperature of the solid is increased above the saturated temperature of the liquid. This can be distinguished from evaporation whereby the phase transition occurs at the liquid-vapour boundary below its saturation temperature. The boiling heat transfer coefficient can be defined by using Newton's law of cooling such that the heat flux is given by

$$q'' = h(T_s - T_{sat}) \quad (2-1)$$

Nucleate boiling can be classified broadly as pool and flow boiling. The main difference is that pool boiling occurs in the absence of an external pump, while flow boiling uses a pump to induce forced convection. In flow boiling, bubbles formed are rapidly swept away by the flow of the bulk fluid from the surface which contrasts with the phenomenon of pool boiling whereby the pool of fluid remains stagnant above the heated surface.

Nucleate pool boiling can be characterised by pool boiling curves. An example of the pool boiling curve of water is shown in Figure 2-1, which describes the heat flux dissipation from a surface in relation to the wall superheats and was obtained by Nukiyama [12] in 1934. The four stages of the pool boiling process, namely natural convection, nucleate boiling, transition boiling and film boiling, are identified based on their distinct features during the boiling process. Nucleate boiling has been the main research focus of previous investigators due to its high heat transfer rate attainable at low surface superheat.

Figure 2-1 Pool boiling curve of saturated water [13]

Figure 2-2 shows the evolution of bubbles in the nucleate pool boiling regime. Before nucleate boiling occurs, heat transfer occurs by natural convection of single-phase liquid and has relatively low heat transfer coefficient. At the onset of nucleation, discrete bubbles can be observed, and they tend to appear at favourable spots due to inherent surface scratches or micro-cavities. At higher heat fluxes, more nucleation sites will be activated, and adjacent bubbles tend to coalesce at departure. When heat flux is further increased, rapid coalescence form mushroom bubbles of sizes can be of magnitudes equivalent to the heated surface. Eventually, the phenomenon of Critical Heat Flux (CHF) occurs when a vapour

film constantly blankets the surface. At this stage, the vapour could not be effectively removed from the surface due to rapid formation and fresh liquid could not replenish effectively. Owing to the poorer thermal conductivity of vapour as compared to liquid, the vapour acts as a thermal barrier and this leads to a sudden temperature overshoot on the heated surface. The occurrence of CHF has been a major focus in nucleate boiling heat transfer applications and it poses typically an upper limit for which the system is safe to operate in.

Figure 2-2 Illustration of different pool boiling regimes [14]

Flow boiling typically allows higher heat transfer rates than that of pool boiling due to the additional effect of forced convection. In a sufficiently long heated tube, the flow regime or pattern varies along the direction of fluid travel and depends heavily on the local vapour quality. An example of different flow patterns in a horizontal tube is shown in Figure 2-3.

At the inlet, single phase liquid is present. With the onset of nucleation, bubbly flow occurs and is characterised by distinct small bubbles in a continuous liquid phase. As the vapour quality increases downstream, formation of bullet-shaped plugs occurs, and they tend to remain on the upper portion of the channel due to effects of buoyancy. Further increase in vapour quality results in slug flow which is characterised by disturbances which causes the plugs to break up. At higher vapour quality, annular flow occurs and is characterised by separation of phases

with liquid film flowing near the walls and a vapour central core. When most of the liquid has vaporised, dry-out can occur and the flow pattern of misty flow can be observed as fine liquid mists in a dominantly gaseous flow. Further heating results in single-phase superheated vapour which has poor heat transfer performance and is typically avoided.

Figure 2-3 Flow boiling patterns in a horizontal tube [15]

Flow boiling can be broadly classified under nucleate boiling dominated or convective dominated mechanisms. Thome and Consolini [16] mentioned that the nucleate boiling mechanism is dominant at low to intermediate vapour qualities, which correspond to the bubbly to slug slow boiling regimes. Convective boiling mechanism, on the other hand, is dominant at high vapour quality for the annular flow regime. However, it is interesting to note that the total heat transfer is due to a combination of both the nucleate boiling and convective mechanisms.

2.2 Nucleate boiling heat transfer models

Despite intensive studies over the decades, the prediction of the boiling heat transfer coefficient is largely based on empirical and semi-empirical methods. This is due to the complex interactions of various heat and mass transfer processes and at vastly different boiling regimes.

The heat transfer performance is closely related to a combination of various factors such as number of active nucleation sites, rate of bubble formations and thermophysical properties of the fluid and solid. The most common correlation for nucleate pool boiling is the one developed by Rohsenow [17] for a smooth surface as shown in Equation (2-2), where C_{sf} and n are constants for different surface-liquid combinations.

$$q'' = \mu_l h_{fg} \left[\frac{g(\rho_l - \rho_v)}{\sigma} \right]^{1/2} \left[\frac{C_{p,l}(T_S - T_{sat})}{C_{sf} h_{fg} \text{Pr}_l^n} \right]^3 \quad (2-2)$$

Rohsenow's correlation does not consider the characteristics of the surface, such as thermal conductivity and surface microstructures, as well as the interactions between the solid, liquid and vapour, such as wettability and adhesion. Pioro et al. [18] mentioned that these parameters affect the heat transfer performance simultaneously and are often interlinked, but there were insufficient data to solve this complex problem. Thus, only separate effects were typically considered.

Flow boiling heat transfer prediction is further complicated by the combination of nucleate boiling and convective mechanisms. An early correlation was proposed by Chen [19], who proposed a summation of the nucleate boiling (h_{pool}) and single-phase liquid convection (h_L) contributions, as shown in Equation (2-3). The nucleate boiling coefficient was adopted from Forster and Zuber [20] for pool boiling prediction and multiplied by a suppression factor S , whereas the single-phase contribution was adopted from the Dittus-Boelter equation [21] for turbulent flow, as shown in Equation (2-4), and multiplied by a two-phase convection factor F .

$$h_{fb} = S h_{pool} + F h_L \quad (2-3)$$

$$h_L = 0.023(Re_L)^{0.8}(Pr_L)^{0.4}\left(\frac{k_L}{D}\right) \quad (2-4)$$

Gungor and Winterton [22] later proposed Equation (2-5) from a correlation of a large database. They used the same Dittus-Boelter equation for the single-phase convective coefficient multiplied by an enhancement factor E , which is due to the effect of improved heat transfer from disturbances by vapour generation. However for the pool boiling coefficient, the Cooper [23] correlation was used and multiplied by a suppression factor S .

$$h_{fb} = Sh_{pool} + Eh_L \quad (2-5)$$

With an increasing database, researchers can compare and formulate more accurate correlations for flow boiling applications. However, these correlations typically apply only to plain tubes in the horizontal and vertical flow directions and are restricted to limited liquid types and pressure ranges.

Zuber [24] attempted to predict the CHF of a smooth surface by taking into consideration the thermophysical properties of the fluid, as shown in Equation (2-6), where C_2 is a constant due to different surfaces. Zuber derived the value of 0.13 for C_2 from a model of instability in the counter-flow of liquid and vapour normal to a horizontal plate.

$$q''_{CHF} = C_2 h_{fg} \rho_v^{0.5} [g\sigma(\rho_l - \rho_v)]^{0.25} \quad (2-6)$$

For nucleate boiling, bubbles are the main carrier of latent energy from the heated surface. The bubble dynamics play important roles in the heat transfer and the parameters include the bubble departure diameter (D_B) and frequency (f_B). The onset of nucleation is typically from a nucleation site such as an activated cavity which generates a bubble. The bubble grows and departs from the nucleation site

when it attains the size of its departure diameter, which is affected mainly by the force balance on the bubble, such as buoyancy and surface tension forces. The bubble departure frequency is defined as the number of bubbles departing from the nucleation site per second. It is typically correlated to the bubble departure diameter.

Fritz [25] proposed that the bubble departure diameter for a smooth surface can be expressed in terms of the Bond number and fluid contact angle. The Bond number is the ratio of buoyancy force to surface tension force, as shown in Equation (2-7), and the bubble departure diameter as proposed by Fritz is shown in Equation (2-8),

$$Bo = \frac{g(\rho_l - \rho_v)D_B^2}{\sigma} \quad (2-7)$$

$$D_B = 0.0208\theta \left[\frac{\sigma}{g(\rho_l - \rho_v)} \right]^{0.5} \quad (2-8)$$

where θ is the contact angle in degrees. However, the equation does not account for the different heating conditions, such as heat flux and surface superheat.

Cole [26] proposed that the bubble departure diameter for the isolated bubble region of nucleate pool boiling from a smooth surface can be expressed in terms of Jakob number and fluid properties, as shown in Equation (2-9). The Jakob number is the ratio of sensible to latent energy and is defined in Equation (2-10).

$$D_B = 0.04Ja \left[\frac{\sigma}{g(\rho_l - \rho_v)} \right]^{0.5} \quad (2-9)$$

$$Ja = \left(\frac{\rho_l c_{pl}(T_s - T_{sat})}{\rho_v h_{fg}} \right) \quad (2-10)$$

Many researchers have developed correlations for the relationships between the bubble departure diameter and frequency. Jakob and Fritz [25] first proposed that

the product of bubble departure diameter and frequency is a constant value, as shown in Equation (2-11). Zuber [27], on the other hand, correlates the product bubble departure diameter and frequency to the fluid properties, as shown in Equation (2-12). Other researchers fitted the product of fD_B^m where m is an empirical value. For example, Cole [28] and McFadden and Grassmann [29] used $m = 0.5$ as the exponent, as shown in Equations (2-13) and (2-14), respectively. More recently, Kim et al. [30] used FC-72 and water for pool boiling and concluded in Equation (2-15) that when $m = 4.85$, the product can be related to a constant value regardless of heat flux.

$$fD_B = 0.078 \quad (2-11)$$

$$fD_B = 0.59 \left[\frac{\sigma g (\rho_l - \rho_v)}{\rho_l^2} \right]^{0.25} \quad (2-12)$$

$$fD_B^{0.5} = 0.59 \left[\frac{4g(\rho_l - \rho_v)}{3C_d \rho_l} \right]^{0.5} \quad (2-13)$$

where C_d is a bubble drag coefficient.

$$fD_B^{0.5} = 0.56g^{0.5} \quad (2-14)$$

$$fD_B^{4.85} = 7.2 \times 10^{-8} \quad (2-15)$$

2.3 Surface modifications for enhanced nucleate boiling

The nucleation site density is a major factor which affects nucleate boiling heat transfer. Jakob and Fritz [31] investigated the effect of surface finish by using a sandblasted surface. The sandblasted surface was found to increase heat transfer by up to 15% but the performance decayed quickly due to aging effect, which limited the ability to create additional stable nucleation sites. Kurihara and Myers [32] roughened copper surfaces with different grades of emery paper and clearly

established that the increased nucleate boiling coefficients resulted from increased nucleate site densities. The artificially formed nucleation sites allow boiling incipience to occur at a lower superheat as compared to naturally occurring nucleation sites and functioned as stable vapour traps. Griffith and Wallis [33] introduced the reentrant cavity and showed that the mouth diameter determines the superheat needed to initiate boiling and its shape determines its stability once boiling has begun.

2.3.1 Pool boiling enhancement from finned surfaces

Many researchers have studied fins of micrometre to millimetre range for the heat transfer enhancement effects. Wei and Honda [34] investigated boiling heat transfer of silicon chips with micro-pin-fins. The fluid used was FC-72 and the silicon chip has dimensions of 10 mm by 10 mm. Dry etching technique was used to fabricate the micro-pin-fins with thickness of 30 μm or 50 μm and heights varying from 60 μm to 270 μm , as shown in Figure 2-4. A plain silicon chip was used as the benchmark. Their results for saturated boiling showed that all the micro-pin-fin chips achieved significant heat transfer enhancements as compared to a plain chip. While the plain chip reached CHF at about 16 W/cm^2 , the micro-pin-fin chips were able to increase CHF of up to 33 W/cm^2 , which is more than a 100% increase.

Figure 2-4 Scanning electron micrographs of silicon chips with micro-pin-fins
[34]

Yu and Lu [35] performed pool boiling experiments on copper blocks with a base area of 10 mm by 10 mm in saturated FC-72. Finned surfaces with fin thicknesses varying from 0.5 mm to 2.0 mm and heights varying from 0.5 mm to 4.0 mm were compared to a plain copper block. The results showed that substrates with larger surface area, which was achieved by increasing the number of fins and fin height, improved the heat transfer. A maximum CHF of 98.3 W/cm^2 was achieved, which is almost five times that of plain surface. The bubble departure resistance was proposed to be an important process which affected the fluid replenishment path. It was observed that coalescence of bubbles occurred more readily for substrates with fins of small gaps, resulting in longer lift-off times. An example of boiling flow patterns with the smallest gap of 0.5 mm and height of 4 mm was shown in Figure 2-5. It was also observed using high-speed photography that the bubble departure diameter was estimated to be between 0.2 mm and 0.3 mm.

Figure 2-5: Flow patterns of fins with spacing of 0.5 mm, height of 4.0 mm and heat flux at (a) 56.8% of CHF and (b) 81.3% of CHF [35]

Rainey and You [36] conducted similar pool boiling studies on copper blocks with base area of 10 mm by 10 mm in saturated FC-72. Finned surfaces with different heights of 1 mm to 8 mm were tested in comparison to a plain copper surface. It was reported that heat transfer enhancement occurred for fin heights up to 5 mm and further increase in height did not contribute to increasing nucleate boiling as the surface temperature of the upper portion of the fin was too low to sustain boiling. The results showed that substrates with taller fins reached higher CHF. For example, the CHF of substrates with fins of 1 mm and 8 mm heights were 36.1 W/cm^2 and 91.7 W/cm^2 , respectively.

2.3.2 Pool boiling enhancement from porous coatings

Many researchers have studied the heat transfer enhancement mechanisms due to porous surface modifications. O'Neil et al. [37] proposed that the pores formed by inter-connected particles acted as active nucleation sites, such that vapour was generated in the inter-particle space and squeezed out of the open pores. The escaped vapour was then refilled with liquid from adjacent pores that act as liquid

supply channels. Their proposed mechanism for vapour formation and escape is shown in Figure 2-6.

Figure 2-6 Mechanism of vapour formation and escape in porous matrix [37]

Cieśliński [38] produced metallic coatings using various methods of deposition such as electrolytic treatment, plasma spraying and gas-flame spraying on flat surfaces and external surfaces of stainless steel tubes. It was indicated that good metallic contact of the coatings and the substrate base was important for heat transfer. The CHF was determined by the rate which the vapour could escape without totally blocking the liquid fluid flow paths, rather than the surface itself. Webb [39] studied coatings of different parameters such as particle diameter, coating thickness and pore sizes. His results indicated that the maximum heat transfer coefficient was obtained for coating thickness of about three to four times of the mean particle diameter for a highly conductive surface such as copper although it was not so for a lower conductive material such as bronze. The pore size was suggested to produce a more significant effect as compared to particle size.

Chang and You [40] used a mixture of powder and binder to coat copper, aluminium and diamond particles on a square heater. Pool boiling with FC-72 under saturated conditions yielded 100% increase in CHF as compared with a plain surface. The micro-porous surfaces also allowed reduction in boiling

incipience superheat of about 80% to 90%, and 30% enhancement of heat transfer coefficients, as compared to plain surface. The enhancement was attributed to the increased number of active nucleation sites.

Liter and Kaviany [41] developed a modulated porous surface of conical shapes with copper powder of 200 μm for pool boiling tests with pentane under saturated conditions, as shown in Figure 2-7(a). It was suggested that in the porous layers, the liquid supply and vapour escape occurred as a liquid-vapour counter-flow which tended to resist the motions of each other. The liquid and vapour flow resistance increased as the heat flux increased, leading to choking of liquid replenishment and local dry-out, which caused CHF. If the porous coating was too thick, the liquid-vapour counter-flow tended to be high and might cause a viscous-drag induced dry-out, leading to a CHF lower than that of plain surfaces. However, for thin porous coatings, vapour escape resistance was deemed to be of less significance and CHF can be enhanced. The enhancement was believed to be due to the lateral capillary-assist which reduced liquid-vapour counter-flow. Thus, the modulated porous surface was proposed to create alternating regions of low resistance to vapour escape and high capillary-assisted liquid draw, as shown in Figure 2-7(b). Experiments showed that CHF for a plain surface, uniform porous-layer surface and the modulated porous surface were 24.5 W/cm², 48.0 W/cm² and 76.2 W/cm², respectively.

Figure 2-7: (a) SEM image of modulated porous layer, (b) proposed pathway of liquid refilling and vapour escape pathways [41]

Patil and Kandlikar [42] fabricated microporous coatings selectively on fin tops of open micro-channels on a 10 mm by 10 mm copper substrate. Pool boiling was tested with water at saturated conditions. The CHF was enhanced and a significant increase in heat transfer coefficient was achieved as compared to a plain surface and a micro-channel surface with no microporous coatings. From high-speed videography, it was observed that the bubbles nucleated from the top of the fins with microporous surface. It was proposed that the micro-channels acted as passages for water supply from all sides, creating a convective flow, as shown in Figure 2-8. The improved liquid replenishment path thus helped to delay the onset of CHF.

Figure 2-8: Proposed fluid supply pathways in the micro-channel [42]

Jaikumar and Kandlikar [43] extended the study on selectively porous coating on open micro-channel for three configurations, namely, sintered-throughout, sintered-fin-tops and sintered channels. Heat transfer mechanisms were proposed for the different configurations as shown in Figure 2-9. For the sintered-throughout surface, as bubble nucleation was observed to be at both the fin-top and in the channel, it did not produce separate liquid-vapour pathways. The liquid refilling to the bottom of channel was proposed to be due to capillary flow along the porous walls and gravity and chaotic liquid-vapour motion. For the sintered-fin-tops surface, nucleation was observed dominantly on the fin tops, which suggested that the liquid supply in the channels helped to replenish the porous surface on the fin tops. For the sintered-channels surface, nucleation was observed dominantly in the channels, and liquid refilling is proposed to occur from the fin tops to the bottom of channel through capillary and gravity transport.

Figure 2-9: Bubble nucleation and schematic of liquid-vapour pathways for (a-b) sintered-throughout, (c-d) sintered-fin-tops, (e-f) sintered-channels [43]

2.3.3 Flow boiling enhancement from finned surfaces

Ma et al. [44] used the dry etching method to fabricate micro-pin-fins on chip surfaces and flow boiling was carried out with FC-72. They investigated the effects of fluid velocities, subcooling and chip heights. All finned surfaces showed enhanced heat transfer and CHF values were observed to increase with higher fluid velocity and sub-cooling. Wei et al. [45] also used the dry etching method to create micro-fins on silicon chips for flow boiling testing with FC-72. Nucleate boiling was enhanced with larger surface areas on the finned surfaces and mechanisms of micro-convection between the fins were proposed. Guo et al. [46] further used jet impingement in combination with flow boiling for the same silicon substrates and observed increased heat transfer due to turbulence induced by the impingement.

2.3.4 Flow boiling enhancement in microchannels

Microchannels were thoroughly investigated by many researchers due to their potential for increased heat transfer from their high surface-to-volume ratios. Kandlikar and Grande [47] defined a flow passage of hydraulic diameter between 10 and 200 μm as a microchannel due to manufacturing limits of small channels. Kew and Cornwell [48] proposed that the flow boiling patterns in microchannels deviated from macro-sized channels and can be characterised by the confinement number (Co) as shown in Equation (2-16). A confinement number smaller than 0.5 is proposed to be defined as a microchannel.

$$Co = \frac{1}{D} \sqrt{\frac{\sigma}{g(\rho_l - \rho_g)}} \quad (2-16)$$

Kandlikar [49] proposed that surface tension and evaporation momentum forces are dominant in microchannel flow boiling. Due to the confinement effect, the elongated bubble flow pattern was observed to be dominant, as shown in Figure 2-10.

Figure 2-10 Dominant elongated bubble flow pattern in microchannel [49]

Flow instability is a major problem found in microchannel flow boiling. Hetsroni et al. [50] observed flow instability as explosive motions of periodic wetting and dry-outs in a microchannel. This was accompanied by large pressure oscillation

recorded. Wang et al. [51] attributed the instability to sudden bubble expansion in both upstream and downstream directions and verified it using high speed visualisations. To reduce flow instabilities, Koşar et al. [52] introduced flow restrictors at the microchannel inlet, whereas Kandlikar et al. [53] used a combination of artificial nucleation sites and pressure drop elements.

2.4 Porous foams used in nucleate boiling

Recent developments in materials processing technology have led to the emergence of lightweight porous foams for structural and thermal applications. According to Zhao [54], these open-celled foams have high surface-to-volume ratios of $1000 - 3000 \text{ m}^2/\text{m}^3$ with increased heat transfer areas. Fluid flow mixing is enhanced due to the tortuosity of metal foams. The porous foams are typically classified under factors of material type, porosity and pore density. The materials available are aluminium, steel and copper alloys. Recently, graphite foams have been produced as well [55]. The porosity is determined by the volume fraction of void against bulk solid volume. The pore density is typically measured in pores per inch (PPI) by taking the average of the linear line pore density. Figure 2-11 shows a typical porous foam and an illustration of forced convection across a porous foam.

Figure 2-11: (a) A sample of metal foam with 10 PPI. (b) Illustration of forced convection in porous foams [54]

One important factor for heat transfer application is the effective thermal conductivity of the porous foam, which is defined as

$$k_e = \frac{q''}{\Delta T/H} \quad (2-17)$$

where q'' is the uniform heat flux, ΔT is the temperature difference between the top and bottom surfaces and H is the sample thickness.

In a review by Zhao [54], the most commonly employed method to evaluate the effective thermal conductivity is by accounting for the volume fractions of the solid and liquid phases as shown in Equation (2-18),

$$k_e = \varepsilon k_f + (1 - \varepsilon)k_s \quad (2-18)$$

where ε is the porosity of the foam, and k_f and k_s are the fluid and solid thermal conductivities, respectively.

Several researchers have conducted measurements on the effective thermal conductivities of porous foams. Paek et al. [56] showed that the effective thermal conductivity increases as the porosity decreases. However, when the porosity is fixed, no obvious change in effective thermal conductivity is detected by varying the cell size of the foam. Zhao et al. [57] measured the effective thermal conductivity of a steel alloy foam of material FeCrAlY under different temperatures. It was shown that the effective thermal conductivity was increased when the temperature was increased, with the hypothesis that thermal radiation could have contributed dominantly to heat transfer in the range of 500-800 K.

Their results also revealed that the contribution by natural convection was significant.

Due to the differences in material and pore geometry for different foams, theoretical modelling of thermal transport in porous foams is still very limited. Leong and Li [58] used a unit cell model to represent the pore structure of graphite foam in order to obtain the effective thermal conductivity. Their results showed that the effective thermal conductivity decreases with the increase of porosity. However, the fluid thermal conductivity was found to have negligible effect on the effective thermal conductivity. Despite various attempts to determine the effective thermal conductivity of porous foams, the actual geometry of the foam affects the conductive pathways along the solid ligaments [59], thus making it a challenging task to generalise the calculation of effective thermal conductivity.

2.4.1 Pool boiling enhancement from porous foams

Various researchers have conducted pool boiling experiments using porous foams. Xu et al. [60] conducted pool boiling experiments on ultra-light copper foams and studied the bubble characteristics using high-speed photography at 5000 frames per second. The porosities of copper used were 0.88 and 0.95, while the pore densities were 30, 60 and 90 PPI. Acetone was used as the fluid at saturated condition. It was noted that acetone has comparable surface tension as FC-72 at room temperature, but the latent heat of evaporation is significantly higher at 512.94 kJ/kg, as compared to FC-72 at 94.80 kJ/kg. Pool boiling at low heat fluxes showed that miniature bubbles nucleated on the foam ligaments at specific locations. Increase of heat flux caused coalescence of the miniature

bubbles into cage bubbles, which were larger than the pore diameter of the foam, as shown in Figure 2-12. It was postulated that a cage bubble was formed due to the connected web structure of foam ligaments through junctions.

Figure 2-12 Boiling pattern at 20.8 W/cm^2 with frames of 5.0 ms apart [60]

At high heat flux, the cage bubbles were formed very quickly and eventually coalesced to become mushroom-shaped bubbles. This typically caused the entire foam surface to be covered with a vapour layer, which impeded liquid replenishment into the foam structure. A small further increase of heat flux then caused a large uncontrollable temperature rise, leading to CHF. The coalescence of the cage bubbles is shown in Figure 2-13.

Figure 2-13 Boiling patterns at heat flux of 83.3 W/cm² [60]

Xu et al. [61] performed saturated pool boiling of deionised water on copper foams. Porosities of range 0.90-0.98, pore density of range 5-40 PPI and foam thicknesses of range 5-7 mm were used. It was postulated that the heat transfer in pool boiling depended heavily on the competing factors of surface area and bubble escape resistance. It was also observed that bubble pulsation occurred at high heat flux when the entire foam surface was covered by bubbles. The pulsation was divided into four stages, namely suction by previous bubble cluster, fresh liquid supplement, liquid vaporisation and bubble cluster departure, as shown in Figure 2-14. Further increase of heat fluxes caused the pulsation frequency to be increased. The bubble pulsation was necessary to allow liquid to

reach the internal pores when the entire surface was covered with a layer of vapour and it prevented CHF from occurring.

Figure 2-14 Four stages in the period of bubble cluster pulsation. [61]

Xu and Zhao [62] investigated the effect of cutting V-shaped grooves on copper foams for pool boiling experiments. It was shown that the V-shaped grooves reduced the bubble escape resistance and helped to delay CHF as compared to foam without grooves. For foams with smaller pores, the larger contact area between the escaping bubbles and metal struts causes stronger friction on the bubbles, increasing the bubble resistance. However, smaller pores contribute to stronger capillary force of liquid flow and allowed better liquid replenishment. Hence, it was proposed that capillary force and bubble escape resistance are the two opposing forces for pool boiling of foams of small pores and high pore densities.

Xu and Zhao [63] subsequently investigated pool boiling of gradient foams. Two foams of different pore densities were sintered together to form a single foam. By having a foam of higher pore density at the lower part, and a foam of lower

pore density at the upper part, improved nucleation was achieved due to the larger surface area at the base, and smaller bubble escape at the upper portion.

Pool boiling investigations on graphite foams have been investigated by Jin et al. [64] and Pranoto et al. [65]. In both cases, the graphite foams have showed significant nucleate boiling heat transfer enhancement as compared to plain copper surface. The high bulk thermal conductivity of graphite has allowed the foam to maintain a high effective thermal conductivity and allowed good nucleation.

2.4.2 Flow boiling enhancement from porous foams

Studies of flow boiling enhancement from porous foams are scarce. Kim et al. [66] performed flow boiling experiments of FC-72 in a foam-filled channel using copper foams of 10 and 20 PPI. It was reported that the increase of mass flux from $20 \text{ kg/m}^2\cdot\text{s}$ to $48 \text{ kg/m}^2\cdot\text{s}$ enhanced the heat transfer rate by as much as 15%. Further increasing the mass flux, up to $72 \text{ kg/m}^2\cdot\text{s}$ did not enhance the cooling capability. The foam with a larger pore size was shown to have a better heat transfer performance as the vapour could escape more easily at high heat flux.

Lu and Zhao [67] performed numerical and experimental analysis of flow boiling in horizontal metal foam tubes. It was found that a decrease of pore size with increase of pore density (PPI) resulted in significant improvement to overall heat transfer performance due to the increase of surface area and strong flow mixing.

Pranoto and Leong [68] used graphite porous foams with porosities of 61% and 72% and showed that heat transfer coefficients could be enhanced by up to 2.5 and 1.9 times, respectively as compared to a smooth surface. The better

performance of the foam of 61% porosity was due to higher surface-area-to-volume ratio and higher bubble departure frequency.

2.5 Research gaps

From the above literature review, it can be seen that many types of surface modifications such as fins and porous coatings were employed by various researchers to enhance nucleate boiling. However, the geometries available were largely limited by conventional machining methods such as etching and milling.

The use of porous foams in the recent years shows potential to further enhance nucleate boiling heat transfer. However, the use of commercial foams is limited by the pore structure design and problems of random pore size distribution and poor inter-connectivity are prevalent. Due to the complex structures of porous foams, the fundamental understanding of the boiling mechanisms is still not well established.

To overcome the current manufacturing limitations, the author proposed to explore the use of additive manufacturing techniques to fabricate complex and orderly 3D structures for nucleate pool and flow boiling enhancements.

2.6 Additive Manufacturing: Process selection and capabilities

For this section, an overview of the various AM technologies is presented first. Based on the capabilities and suitability, the Selective Laser Melting technique is chosen as the manufacturing method for engineered structures in this thesis.

2.6.1 Overview of AM technologies

Additive manufacturing technologies can be broadly classified as liquid-based or powder-based, as shown in Figure 2-15 and adapted from a review by Wong and Hernandez [69].

Figure 2-15 Overview of AM Technologies [69]

Fused Deposition Modelling (FDM) works by extruding a thin polymer filament using a pair of feed pinch rollers as shown in Figure 2-16. The filament then passes through a heated zone where it melts. The feed pinch rollers sustain the motion using the solid portion and the melted filament is extruded through a nozzle at the print head. The print head moves in a 2D horizontal plane to deposit the material onto a platform, which can be lowered in the vertical direction for 3D parts to be fabricated. The common materials used are polycarbonate and acrylonitrile butadiene styrene (ABS).

Figure 2-16 Schematic of FDM mechanism [70]

Stereolithography (SL) was developed by 3D Systems Inc. and is still a widely used process for rapid prototyping. A platform is submerged in a pool of liquid photosensitive resin at the start of fabrication. An ultraviolet (UV) laser then cures the resin into solid at specific regions in a horizontal plane as shown in Figure 2-17(a). The platform then lowers by a layer and the new layer is bonded to the previous layer. When the part is completed, the residual resin can be drained and recycled for future fabrication.

Polyjet uses inkjet technologies to dispense droplets of photopolymer onto a build platform, as shown in Figure 2-17(b). The unique feature of Polyjet is that the print head can dispense a mixture of two different materials, one being rigid and the other being flexible. The combinations of the two materials in different ratios allow up to nine material blends with unique properties for each pixel. For each layer, the inkjet print head moves in the horizontal plane and a UV bulb cures the dispensed photopolymer immediately after dispensing.

Figure 2-17 (a) Schematic of SLA [69], (b) Schematic of Polyjet [71]

Even though liquid-based processes are versatile for fabrication of many types of materials, they can only be used to manufacture polymer components. This is not useful for heat transfer applications due to the low thermal conductivity of polymer materials. On the other hand, the powder-based processes can be used to manufacture both polymer and metal components, depending on type of process, which are deemed to be more useful for the purpose of this thesis.

Electron Beam Melting (EBM) uses an electron laser beam which is powered by very high voltage, in the range of 30 to 60 kV. The process requires a vacuum environment to reduce dissipation of the electron beam. The schematic of EBM is shown in Figure 2-18(a). During the fabrication process, the electron beam is generated using a tungsten filament heated to above 2000°C. The emitted electrons are then accelerated in an electric field and focused using an electromagnetic coil onto the platform. Electron Beam Melting is mainly used to process metallic powder and the most commonly used material is Ti-6Al-4V.

Selective Laser Sintering (SLS) and Selective Laser Melting (SLM) both use a focused laser beam to melt selected regions on a layer of powder. The general process is shown in Figure 2-18(b). The laser beam deflection system is used to focus the laser beam on a powder bed, thereby melting the powder according to

the pre-defined CAD model. The platform is lowered each layer and a powder deposition system is used to dispense a new layer of powder. However, the two processes differ in the type of laser used and material chosen for processing. Selective Laser Sintering typically uses a carbon dioxide laser with the wavelength of $10.6\ \mu\text{m}$ and is more suitable in the melting of polymer powder. Selective Laser Melting, on the other hand, uses a fibre laser with wavelength of $1.06\ \mu\text{m}$ and is more suitable for melting metallic powder. The wavelength of the laser affects the absorptivity of energy by the powder.

Figure 2-18 (a) Schematic of EBM [72], (b) Schematic of SLS and SLM [73]

For this thesis, metallic materials would be more effective for heat transfer purposes as compared to polymer materials. From the above literature survey, both EBM and SLM are suitable to process metallic materials from a powder base. However, there is a limited range of materials which has been developed for EBM. Selective Laser Melting is favoured as there are more studies to process a wider range of metallic materials, and in particular aluminium alloys which possess high thermal conductivity. Therefore, SLM is selected to be the manufacturing method in this thesis. The advantages, material-processing

capabilities and challenges of using SLM are elaborated in the subsequent sections.

2.6.2 Advantages of using Selective Laser Melting

Parts fabricated using SLM are of high quality, which is characterised by the relative density defined as the ratio of the part density attained with SLM to the theoretical density of the bulk material. A low relative density is typically caused by porosity within the fabricated parts and this can arise due to poor melting and bonding of powder during the manufacturing process. With proper control of the laser parameters, it has been shown that SLM parts produced can be near 100% density [74].

The ability to manufacture complex internal geometries is one major advantage which conventional manufacturing cannot replicate [75]. Another advantage of SLM is that the built part can be near net shape and eliminates the need for a mould or long post-processing methods, as often required in casting. This greatly reduces production cost for small volume of highly customised parts.

Selective Laser Melting is capable of producing parts with superior mechanical properties as compared to those fabricated by casting [76]. This is due to the rapid solidification of the molten materials involved in the SLM process which causes grain size refinement, leading to an improvement in ultimate tensile stress and fracture toughness.

Finally, material wastage is significantly lesser than conventional manufacturing methods. Conventional methods such as drilling and machining are subtractive in nature and the waste materials are usually discarded. However, the unused powder in SLM can be easily sieved and recycled for future production.

2.6.3 Material selection

The SLM process has demonstrated its capability of processing many different metallic alloys. The most common materials are 316L stainless steel, Ti6Al4V titanium alloy and nickel-based alloys [77]. These materials are popular due to their widespread use in aerospace and medical industries. Other materials which have been studied for SLM include aluminium, copper, magnesium and cobalt-chrome alloys.

The key considerations for material selection are the ease of fabrication of high quality parts and high thermal conductivity for heat transfer applications. Aluminium and copper alloys are potential candidates due to their high thermal conductivities as compared to their other counterparts like iron or titanium alloys. However, the formation of copper alloys using SLM is shown to be difficult due to its high reflectivity [78]. Moreover, copper and its alloys oxidise readily under atmospheric oxygen, which leads to problem of long-term storage. The high cost of copper powder is another inhibiting factor. Thus, aluminium alloys are mainly considered for fabrication.

The two most common aluminium alloys used for SLM are AlSi10Mg and Al6061. A summary of the relative densities produced by SLM by various researchers is shown in Table 2-1. It can be observed that there are more available studies on AlSi10Mg and higher relative densities can be obtained as compared to that of Al6061. The superior quality of AlSi10Mg is attributed to the presence of Si which gives the material good weldability, due to the near-eutectic composition of Si and Al. The presence of Mg (0.3 – 0.5 wt. %Mg) allows hardening of the alloy due to the formation of Mg₂Si precipitate [79].

Table 2-1 Summary of relative density produced by SLM

Material	Relative density (%)	Sources
AlSi10Mg	99.5	[80]
	99.8	[81]
	99.4	[79]
	99.0	[82]
	99.2	[83]
	97.7	[84]
Al6061	90.0	[85]
	89.5	[86]

A comparison of the different materials shows that AlSi10Mg is an ideal material for heat transfer due to its high thermal conductivity to density ratio, as shown in Table 2-2. Hence, AlSi10Mg is favoured as the candidate for fabrication of substrates in this thesis.

Table 2-2 Comparison of densities and thermal conductivities of different heat transfer materials [87]

Material	Density, ρ (g/cm ³)	Thermal conductivity, k (W/m·K)	k/ρ ratio
AlSi10Mg	2.68	120 - 170	49 - 60
Cu (pure)	8.92	401	45
Cu alloys	8.30	160 – 340	19 – 41
Zinc	7.14	110	15
Low-alloy steel	7.85	42	5

2.6.4 Process parameters and defects

The SLM fabrication process is complex and requires proper control measures. The key parameters which affect the process are the laser power, laser scan speed and powder layer thickness. For successful fabrication of parts, these parameters need to be carefully selected for each type of powder base. For example, if the laser power is too low, the powder base will not melt sufficiently to form a solid

layer. On the other hand, if the laser power is too high, excessive material evaporation will occur.

For new materials, it is important to obtain the optimal parameters as inappropriate parameters will result in defects such as balling of the molten material during fabrication and residual stresses in built parts [88, 89]. Balling of the molten material occurs when the energy supplied by the laser is insufficient to overcome the surface tension of the molten material, as shown in Figure 2-19(a). This causes poor surface finish and prevents the smooth coating of a new layer of powder, typically leading to job failure. On the other hand, residual stress is caused by high temperature gradients on the powder bed during fabrication. The high laser power causes rapid melting and solidification and contributes to the high temperature gradients. Residual stress can cause defects such as warpage, delamination and cracking as shown in Figure 2-19(b). Both balling and residual stress can be minimised by selecting the correct operating parameters for each type of material.

Figure 2-19 (a) Balling phenomenon at low laser powder [90], (b) Delamination in SLM parts [91]

2.6.5 Applications in Selective Laser Melting

The process of SLM is applied generally in high value-added industries such as the biomedical and aerospace fields. Kruth et al. [92] adopted SLM to fabricate dental implants of complex designs using materials of stainless steel, Ti64 and Co-Cr. Wehmöller et al. [93] worked on customised designs of implants using stainless steel. The fabricated models of lower jaw, lumbar vertebra and tubular bone are shown in Figure 2-20. The key advantages of reduced time and cost for fabrication were achieved using SLM as compared to conventional methods.

Figure 2-20 Fabricated lower jaw, lumbar vertebra and tubular bone [93]

The ability to fabricate cellular lightweight lattice structures has garnered much interest in the field of SLM research. The structures can take up complex geometries and be modified to obtain the desired weight and strength characteristics for the intended application. Various researchers [94-96] have investigated on the strength, shock response and micro-structure analysis of cellular lattice structures. One such cellular lattice structure is shown in Figure 2-21. It can be observed that the orderly lattice can be built to a high degree of accuracy and shows great potential to replace commercial foams.

Figure 2-21 Example of a lightweight lattice structure [94]

Some researchers have attempted to incorporate SLM in the fabrication of heat transfer devices. Wong et al. [85] fabricated five heat sinks using SLM of Al6061 powders. The heat sinks consisted of fins of different geometries and were used for convective heat transfer testing using air as the fluid. They reported that the parts produced were of 90% density and effective thermal conductivity of 70 W/m·K, which deviated from the bulk thermal conductivity of 170 W/m·K. More recently, Wong et al. [97] and Ho et al. [98] have demonstrated the capability of SLM to fabricate heat sinks consisting of fins of airfoil cross-sections using AlSi10Mg. Air convective heat transfer studies have shown that the airfoil fins possessed superior heat transfer as compared to cylindrical fins due to their streamline geometry. An example of the various cylindrical, rectangular rounded and airfoil fins is shown in Figure 2-22.

Figure 2-22 Examples of different fin geometries fabricated using SLM [98]

Neugebauer et al. [87] fabricated a miniature cooler unit using SLM of AlSi10Mg powder. They showcased that the process of additive manufacturing allowed a better design approach to achieve superior cooling capacity with a compact size as delicate and complex geometries were buildable.

2.7 Summary

In this chapter, the theories of nucleate boiling and various models were introduced. A thorough literature review was then presented for nucleate boiling heat transfer enhancements. The key findings and research gaps are summarised as follows:

1. Various surface modification techniques were shown to enhance boiling heat transfer, which include the use of fins, porous coatings, microchannel and porous foams. Porous foams show high potential for boiling enhancement due to its high surface-to-volume ratio. However, studies using porous foams for boiling heat transfer are scarce and were conducted using commercial foams.
2. The main research gaps are the limitation of pore structure design and problems of pore size distribution and poor inter-connectivity in commercial foams. Also, the fundamental understanding of boiling mechanisms in porous foams is limited.

3. Additive manufacturing has the capability to fabricate intricate and complex three-dimensional structures. In particular, SLM can fabricate metallic alloys and is selected as the method for manufacturing specialised engineered 3D structures in this thesis. The various process parameters and applications of SLM are discussed.

Chapter 3 - Selective Laser Melting: Methodology and Characterisation

In this chapter, the process of Selective Laser Melting is described in detail. The entire process of pre-, during and post-processing will be elaborated. The process parameters were selected after comparing with the existing literature and preliminary studies were conducted to ensure parts of high relative density can be fabricated successfully. Finally, the thermal conductivities of preliminary samples were determined using the flash method.

3.1 Powder characterisation

For the research conducted in this thesis, the material used was AlSi10Mg due to the ability to fabricate parts of high quality and its high thermal conductivity as discussed in the previous chapter. Selective Laser Melting requires a powder as its building material. The powder was produced by TLS Technik GmbH & Co., which is a leading manufacturer of metallic powders for industrial use. The composition of the AlSi10Mg powder, as provided by the supplier, is listed in Table 3-1.

Table 3-1 Composition of AlSi10Mg alloy used

Material	Al	Si	Fe	Cu	Mn	Mg	Ni	Zn	Ti
Composition (%)	Bal.	10.4	0.21	0.03	0.004	0.39	0.01	0.002	0.027

The powder distribution size is an important consideration for the SLM process. A large powder size would result in poor resolution and build tolerance. Overly small powder size may result in agglomeration due to the high surface energy

caused by van der Waals forces. This would result in poor flowability of the powder, which is measure of how well the powder can spread on a surface.

For this thesis, powder with a distribution size of 20 μm to 63 μm was used. A Scanning Electron Microscope (SEM) image of the powder at a magnification of 300 times is shown in Figure 3-1. It can be observed that the powder has near spherical shape and falls within the expected range of distribution sizes.

Figure 3-1 SEM image of AlSi10Mg powder at a magnification of 300 times

3.2 Selective Laser Melting machine used

The SLM 250 HL (SLM Solutions GmbH) facility was employed for the fabrication of substrates in the present study, as shown in Figure 3-2(a). It is housed in the Future of Manufacturing Laboratory 1 under the Singapore Centre for 3D Printing (SC3DP) at Nanyang Technological University (NTU), Singapore.

The machine consists of a build chamber on the left and the laser modules with a computer attached to control the operation on the right. The interior of the build chamber is shown in Figure 3-2(b). The centre of the build chamber is a platform in which a substrate baseplate with dimensions of 250 mm by 250 mm is mounted. The platform is controlled by motors and is lowered after each cycle of laser scan. The recoater is seen in the background of Figure 3-2(b). It dispenses

powder for each layer and a rubber blade is attached at the base to smoothen out the powder.

A Gaussian distributed Yb:YAG laser of maximum power of 400 W and laser beam spot size of 80 μm was used for melting the metal powder. The laser is focused onto the build platform from the top of the build chamber using a F-theta lens. During operation, the build chamber was flushed with inert argon gas to attain an oxygen level of less than 0.2% to minimise oxidation and combustion of powder.

Figure 3-2 (a) Photograph of SLM250HL machine, (b) Interior of build chamber

3.3 Pre-processing of SLM

Before the fabrication can occur, the 3D model is first designed in a Computer Aided Design (CAD) modelling software such as SolidWorks. The CAD model is then converted to a Standard Tessellation Language (STL) format, which is a common format used in most AM processes. STL format stores the information of a 3D object as a combination of triangles defined by its three vertices and unit normal in the Cartesian coordinate system. With sufficient number of triangles,

any curved surfaces can be approximated with a high degree of accuracy, as shown in Figure 3-3 with the example of a sphere.

Figure 3-3 STL format with varying degree of accuracy [99]

The STL model is then imported to the Magics (Materialize) software. In Magics software, the STL models can be arranged onto the simulated build platform. The software is then used to slice the 3D models into multiple 2D layers according to the selected layer thickness. For each layer, the laser scanning parameters and paths are designated over the entire surface. This process creates the SLM part file which can then be read by the SLM machine. The laser parameters will be elaborated in the later section. An example of the model arrangement in Magics software and the successfully fabricated models is shown in Figure 3-4.

Figure 3-4 (a) Model arrangement in Magics software, (b) Fabricated models

3.4 During the manufacturing process of SLM

At the start of the fabrication, the powder is first distributed evenly on a substrate base plate. The chamber is sealed and flushing with argon gas is initiated. The laser scanning then melts the powder according to a pre-programmed pathway that is defined in SLM part file. The powder layer is rapidly melted and then solidified to form a thin solid layer. The powder bed is then lowered by one layer and redeposited. Laser scanning subsequently melts and bonds the new layer to the previous layer. This process is repeated until the entire part is formed. The process is shown in Figure 3-5.

Figure 3-5 Process schematic of Selective Laser Melting

3.5 Post-processing of SLM parts

Post-processing begins by removing the baseplate from the build platform. The residual powder is thoroughly removed by sweeping and vacuuming from the baseplate and built parts. The unused residual powder can be recycled by subjecting it to a sieve to remove oversized burnt powder. The baseplate is then mounted onto a wire-cutting machine to separate the built parts from the baseplate, as shown in Figure 3-6. After removal, the built parts are cleaned before it can be used for further analysis and other experimentation.

Figure 3-6 Wire-cutting of SLM parts from baseplate

3.6 Process parameters in Selective Laser Melting

The volumetric Energy Density (E_d) is the most important parameter to characterise the SLM process, as shown in Equation (3-1),

$$E_d = \frac{P}{VHT} \quad (3-1)$$

where P is the laser power, V is the laser scanning speed, H is the hatch spacing and T is the layer thickness. An illustration of the various parameters is shown in Figure 3-7. The volumetric energy density plays an important role to determine the quality of built parts in SLM. Typically, there exists a minimum volumetric energy density required to achieve complete melting of a given material. A combination of high laser power and laser scanning speed is desirable to reduce the overall production time. The hatch spacing, which is the distance between adjacent laser scans, must be smaller than the melt pool of the laser scan to allow complete fusing of each laser scan path to the adjacent scan paths. The layer thickness is to be chosen such that the melt pool can penetrate the previous layer to allow sufficient bonding. A larger layer thickness allows reduced production

time although at the expense of increased surface roughness and dimensional accuracy.

Figure 3-7 Processing parameters for SLM [77]

The process parameters for AlSi10Mg have been investigated by many researchers. A summary of the parameters is shown in Table 3-2. It can be observed that high relative density can be obtained for the range of E_d between 36 and 46 J/mm³.

Table 3-2 Summary of process parameters in SLM

Process parameters					Relative density (%)	Source
P (W)	V (mm/s)	H (mm)	T (mm)	E_d (J/mm ³)		
500	1700	0.15	0.05	39.2	98.5	[80]
200	1400	0.105	0.03	45.4	98.5	[81]
350	1140	0.17	0.05	36.1	98.1	[100]

The scanning pattern describes how the laser scans on different layers of the sample. Some common scanning patterns are the parallel and chessboard scanning patterns, as shown in Figure 3-8. The parallel scanning pattern divides the layer into multiple parallel laser scanning paths, and the spacing between each laser scan depends on the hatch spacing chosen. The chessboard scanning pattern divides the layer into smaller grids. Laser scanning is then carried out on

individual grids. It can be configured to occur from left to right, or in an alternate manner, i.e., scanning all the grids with vertical paths first, then the horizontal paths. The chessboard scanning pattern is typically favoured for samples of larger surface areas to allow more uniform cooling throughout the substrate [101].

Figure 3-8 (a) Parallel scanning pattern, (b) Chessboard scanning pattern

For this thesis, the selected processing parameters are shown in Table 3-3. The laser parameters correspond to those used in Ref. [100] as they have been used for the same machine (SLM250HL), which results in a higher likelihood of successful fabrication. For the chessboard scanning strategy chosen, every consecutive layer is subjected to laser scanning rotation of 79° . This ensures that the fabricated parts are isotropic.

Table 3-3 Processing parameters used

Layer thickness	0.05 mm
Hatch distance	0.17 mm
Laser power	350 W
Laser speed	1150 mm/s
Laser scanning strategy	Chessboard
Laser rotation after each layer	79°

A preliminary study was carried out with fabrication of three cubes of 1 cm³ to as benchmark test for buildability using the selected processing parameters, as shown in Figure 3-9. It can be observed that the parts are well-formed and no obvious defects such as balling and delamination are observed. Further analysis on the relative density measurement is conducted.

Figure 3-9 Fabricated samples for preliminary analysis

3.7 Relative density measurement

The relative density of the fabricated benchmark sample showcases the quality of the printed part and suitability of process parameters. The weight of the sample is measured using a XS Analytical Balance (XS 204, Mettler Toledo), which has an uncertainty of ± 0.1 mg, as shown in Figure 3-10. The device has two separate scales to measure the weight of the sample in air and in a submerged fluid.

Figure 3-10 Weight measurement device (XS204, Mettler Toledo)

The measurement is done according to Archimedes' principle which takes into account the density difference of the weight of the object in air and in a fluid. For this study, each sample is weighed in air and subsequently in distilled water, and the density of sample can be calculated using Equation (3-2). The temperature of the distilled water is measured using a thermometer before the experiment and the density is calibrated in the software for calculation of the density.

$$\rho_{sample} = \rho_{fluid} \left(\frac{m_{air}}{m_{air} - m_{fluid}} \right) \quad (3-2)$$

The bulk density of AlSi10Mg is 2.68 g/cm³. Based on the measurements as shown in Table 3-4, an average relative density of 99.2% was obtained. This confirms the suitability of using the selected process parameters to fabricate AlSi10Mg parts.

Table 3-4 Summary of measured relative density

	Measured density (g/cm ³)	Relative density (%)
Cube #1	2.663	99.4
Cube #2	2.649	98.8
Cube #3	2.661	99.3
Average	2.658	99.2

3.8 Thermal conductivity measurement

Thermal conductivity measurements of the samples fabricated using SLM were carried out using the flash method. The machine used is Discovery Laser Flash (DLF-1200) from TA Instruments as shown in Figure 3-11. The machine is located in the Inorganic Materials Service Laboratory under the School of Materials Science and Engineering in NTU. It allows measurement of a wide range of test temperatures of up to 1200°C.

Figure 3-11 Photograph of laser flash machine (DLF-1200)

During operation, a short pulse of laser is emitted on the front face of a sample of disc-shape. The temperature of the rear face is then measured using an optical sensor with high thermal response. By plotting the temperature-time relationship of the rear face, the sample's thermal diffusivity (α) is determined from the thickness (L) of the sample and the time the thermogram takes to reach half of the maximal temperature increase, as shown in Figure 3-12.

Figure 3-12 (a) Illustration of flash method, (b) Time-temperature history of rear face of sample [102]

Three samples of AlSi10Mg fabricated using SLM and a commercial Al6061 sample were tested using the flash method. Each sample is machined to have a diameter of 12.7 mm and 5 mm thickness. Sandpaper with grit 1200 was used to polish all the samples to mirror-like finish.

During the SLM fabrication process, the test samples experienced large thermal gradients that induce internal stresses built up by the material. Stress relief is a common method to reduce the internal stresses. In the current setup, two of the AlSi10Mg samples underwent a stress relief procedure at 300°C for 2 hours, as recommended by EOS [103].

The densities (ρ) of the samples were measured using Archimedes' principle as described in the previous section. The specific heats (C_p) of the AlSi10Mg and Al6061 parts were obtained from the database of EOS [103] and Matweb online database [104], respectively. During each experimental run, all four samples can be mounted together in the machine. For each sample, the measurement for thermal diffusivities (α) was repeated three times and an average was obtained thereafter. After obtaining the thermal diffusivity from the experiment, the thermal conductivity (k) can then be calculated using Equation (3-3).

$$k = \alpha \rho C_p \quad (3-3)$$

The results are summarised in Table 3-5. The thermal conductivity of commercial Al6061 obtained is very close to those obtained from the online database. The thermal conductivity of the as-built AlSi10Mg is the same as that of EOS although the stress relief substrate deviated from the reference. Nevertheless, the currently obtained value of 149 W/m·K is sufficiently high and thus stress relief was also applied all the other substrates used in this thesis.

Table 3-5 Thermal conductivity measurement of AlSi10Mg and Al6061

Material	Density (kg/m ³)	Specific heat (J/kg·K)	Measured thermal diffusivity (cm ² /s)	Calculated thermal conductivity (W/m·K)	Reference thermal conductivity (W/m·K)
Al6061	2700	896	0.700	169	167 [104]
AlSi10Mg (as-built)	2660	910	0.493	119	119 [103]
AlSi10Mg (stress relief)	2660	910	0.615	149	175 [103]

Chapter 4 - Pool Boiling of Porous Lattice Structures Produced by Selective Laser Melting

4.1 Introduction

From the literature review, porous foams are identified as potential candidates for nucleate pool boiling applications due to its high surface-to-volume ratio. However, commercial metallic foams which have been used by other researchers tend to have limited pore geometries and are faced with problems of random pore size distribution and inter-connectivity issues. Hence, SLM has been employed to overcome the difficulties and it shall be shown in this chapter that intricate and complex porous lattice structures with highly controllable repeating unit cells can be fabricated. This would allow the geometrical effects of the porous structures on boiling heat transfer to be controlled more accurately.

To the best of the author's knowledge, there is no prior publication on pool boiling enhancement using porous lattice structures fabricated by SLM or other 3D printing techniques. In addition, reported studies on pool boiling of porous structures using dielectric fluids are scarce. In this study, FC-72 was used as the working fluid at saturated condition under atmospheric pressure. The pool boiling heat transfer performance and delaying of CHF will be studied using porous lattice structures by varying the structure heights and unit cell sizes. Pool boiling mechanisms are also proposed based on a visualisation study of the boiling process for different ranges of heat fluxes.

4.2 Substrate preparation and characterisation

4.2.1 Description of substrates

In this study, the octet-truss geometry was chosen as the repeating unit cell for the porous lattice structure. Figure 4-1 shows a unit cell of the octet-truss lattice. It consists of an octahedral cell in the centre which is depicted in darkened shade, and eight tetrahedral cells at the edges. One example of the tetrahedral cell is shown with nodes labelled as p1 to p4. By repeating the unit cells in a 3D manner, an orderly porous lattice structure can be generated.

Figure 4-1 Unit cell representation of the octet-truss lattice [105]

Using the "Magics" software (Materialize), porous lattice structures of different unit cell sizes of 2 mm, 3 mm and 5 mm were created. The base area of the structure was fixed at 10 mm by 10 mm to be mounted onto the pool boiling facility, which would be described in the next section. For each unit cell size, three different structure heights of 2.5 mm, 5.0 mm and 10.0 mm were created. For each substrate, the porous structure was built on a solid base of 1 mm thickness to allow easy mounting to the pool boiling facility. For example, Figure 4-2 shows a CAD model of a substrate with 5-mm unit cell and 10.0 mm height. A total of 8 unit cells, 2 on each axis, were used to generate the structure.

Figure 4-2 CAD model of porous lattice with 5-mm unit cell and 10 mm height

A total of nine porous lattice structures were fabricated using SLM as the manufacturing process. Examples of the fabricated porous structures at heights of 2.5 and 10.0 mm with different unit cell sizes are shown in Figure 4-3. A plain copper surface that was polished to a mirror-like finish was used as a benchmark for comparison.

Figure 4-3 Fabricated substrates with 2-mm, 3-mm and 5-mm unit cells from
left to right

An abbreviation for naming the substrates is defined for easy reference in the subsequent portion of this thesis. An example of the substrate with the name of

Octet-2.0C-2.5H has a unit cell of 2.0 mm and structure height of 2.5 mm. A summary of the substrate parameters is shown in Table 4-1.

Table 4-1 Summary of porous lattice substrate characteristic

S/n	Name	Unit cell size (mm)	Height of structure (mm)
1	Plain surface	-	-
2	Octet-2.0C-2.5H	2	2.5
3	Octet-2.0C-5.0H	2	5.0
4	Octet-2.0C-10.0H	2	10.0
5	Octet-3.0C-2.5H	3	2.5
6	Octet-3.0C-5.0H	3	5.0
7	Octet-3.0C-10.0H	3	10.0
8	Octet-5.0C-2.5H	5	2.5
9	Octet-5.0C-5.0H	5	5.0
10	Octet-5.0C-10.0H	5	10.0

4.2.2 Light optical imaging of substrates

The characterisation of the lattice structure was performed using a light optical microscope (Olympus SZX7) with its in-built measurement software. It can be observed that the fabricated porous lattice structures are of a very orderly repeating form and the octet-truss geometry is preserved for the different unit cell sizes. The porous lattice also shows good inter-connectivity, which would be important for fluid to permeate throughout the structure. A closer examination of the struts shows the presence of partially sintered powder, which is typical of the SLM process. The partially sintered powders are homogenously distributed, which would act as potential nucleation sites for bubble generation.

Figure 4-4 shows examples of the microscope images for the 2-mm, 3-mm and 5-mm unit cell sizes. The struts were observed to be of cylindrical geometry. The pores were formed by the tetrahedral cells and are of equilateral triangle geometry.

Figure 4-4 Light optical microscope image of lattice of (a) 2-mm, (b) 3-mm unit and (c) 5-mm unit cell

The pore diameter is assumed to be the length of one of the pore edges. A total of 25 to 30 measurements were taken to determine the average pore and strut diameters for each unit cell size. The linear pore density of each unit cell size, in terms of pore per inch (PPI) measurement, was also tabulated for easy comparison with the results from the literature. The surface-area-to-volume ratio was obtained from the “Magics” software by dividing the total curve surface area of the struts to the bulk volume occupied by the lattice structure. Generally, the smaller unit cell size has more struts and hence, a larger surface area for a fixed volume. The surface roughness effect was not considered in this calculation. As the surface roughness is the same across the different unit cell sizes, it would not change the trend of surface-area-to-volume ratio if it were to be considered. The porosity (ϵ) for the different unit cell sizes was fixed at a value of 0.7 when the substrates were generated using the software. A summary of the various measurements is given in Table 4-2.

Table 4-2 Substrate parameters for different unit cell sizes

Parameters	2-mm unit cell	3-mm unit cell	5-mm unit cell
Average pore diameter, D_p (mm)	0.53	1.03	1.72
Average strut diameter, D_s (mm)	0.31	0.48	0.85
Pore density (Pores per inch)	40	30	15
Surface-area-to-volume ratio (m^2/m^3)	2700	2200	1500

4.2.3 Topological mapping of substrates using confocal imaging

Surface roughness measurements were obtained using a PLμ Confocal Image Profiler on the struts of each unit cell size. Five random struts were chosen and 3D topographical mappings were obtained with step intervals of 1 μm . For each topographical mapping, six to eight linear profiles were further selected for linear surface roughness measurements. Figure 4-5 shows a sample of the topological mapping of a strut on the 3-mm unit cell.

Figure 4-5 Topological mapping of a strut on the 3-mm unit cell

Figure 4-6 shows the distribution of surface roughness in terms of the Root-mean-square (RMS) and average roughness (Ra) for each unit cell size. It can be concluded that the surface roughness is similar for the different substrates. This is because the surface roughness is mainly attributed to the partially sintered

powder and rapid solidification process in SLM which is similar across all the substrates.

Figure 4-6 Surface roughness distribution across different unit cell sizes

4.2.4 Wettability measurements of substrates

Surface contact angles were measured using the Attension® Theta machine for a smooth copper surface using deionised water and FC-72. The surface contact angle was determined to be 93° for deionised water and 6° for FC-72 as illustrated in Figure 4-7. This shows that FC-72 is a highly-wetting fluid which is beneficial for fluid replenishment into the porous lattice structure during the pool boiling process.

Figure 4-7 Contact angle measurement on smooth copper surface with (a) deionised water and (b) FC-72

As it is difficult to measure the surface contact angles for the porous lattice structure, the author had adopted the method used by Ji et al. [106] in which the spreading time is used to characterise the surface wettability. A droplet of about 3 μL was gently deposited on the strut of the lattice structure as shown in Figure 4-8. Upon contact with the strut, the FC-72 droplet was quickly absorbed by the porous structure within 48 ms. Measurements of the different unit cell sizes yielded similar results of complete suction below 50 ms.

Figure 4-8 FC-72 droplet suction process for porous lattice structure

4.3 Pool boiling facility and data reduction

4.3.1 Pool boiling facility

A thermosyphon was used to investigate the saturated pool boiling performance of the porous lattice substrates fabricated using SLM. The experimental schematic is shown in Figure 4-9 and an actual photograph in Figure 4-10. The pool boiling chamber with dimensions of 12 cm by 12 cm by 12 cm was constructed with an aluminium body and view glass panels to allow visualisation of the boiling process. The chamber was filled with dielectric fluid FC-72. Two cartridge heaters were submerged in the FC-72 to heat the fluid to its saturated temperature of 56°C, which was measured using two K-type thermocouples

immersed in the fluid. A water-cooled condenser is situated above the fluid to allow the condensed FC-72 to return to the main pool during the boiling process.

Figure 4-9 Schematic of pool boiling facility

Figure 4-10 Photograph of pool boiling facility

A copper block was embedded in the centre of the Teflon block. The test substrate was situated at the centre of the Teflon base, which has an exposed area of 10 mm by 10 mm. It was bonded to the top surface of the copper block using a thermal epoxy cement (Omegabond-200). A cylindrical heater of 6.24 mm

diameter (Watlow Firerod) was embedded in the copper block to act as the heat-generating source, which was powered by a variable transformer, while the current and voltages can be measured accurately using in-built ammeter and voltmeter, respectively. To minimise heat loss to the surroundings, the copper rod was thermally insulated at all sides using Teflon, to allow heat to dissipate mainly through the top exposed surface. Three T-type thermocouples were positioned along the surface of the copper block, as shown in Figure 4-11. The thermocouples T_1 , T_2 and T_3 are at distances of 4 mm, 10 mm and 15 mm, respectively from the substrate top surface. The temperature reading of the bottom-most thermocouple was monitored to prevent overheating of the rod heater.

Figure 4-11 Detailed drawing of placement of thermocouples

The heat flux (q'') into the test substrates was calculated from temperatures of the top two thermocouples using Equation (4-1), where k_{cu} is the thermal conductivity of copper, A is the cross-sectional area of the copper block and L_{1-2} is the distance between Thermocouples 1 and 2. The total heat input to the rod heater was obtained from product of the current and voltage inputs. The deviation

of the electrical heat input and calculated heat transfer rate was attributed to heat loss, which accounted for about 5% of the total heat input.

$$q'' = -k_{cu}A \frac{T_1 - T_2}{L_{1-2}} \quad (4-1)$$

The wall temperature (T_w), which represents the temperature of the substrate top surface, was determined by the gradient from the top-most thermocouple (T_1) using the thermal resistance model under one-dimensional steady-state conduction. The thermal conductivities of copper, epoxy and aluminium are assumed to be 400 W/m·K, 16 W/m·K and 149 W/m·K, respectively. The total resistance across the copper and aluminium substrate is shown in Equation (4-2). Hence, the wall temperature (T_w) can be calculated using Equation (4-3).

$$R_{total} = R_{cu} + R_{ep} + R_{Al} = \frac{L_{cu}}{k_{cu}A} + \frac{L_{ep}}{k_{ep}A} + \frac{L_{Al}}{k_{Al}A} \quad (4-2)$$

$$T_w = T_1 - q''(R_{total}) \quad (4-3)$$

4.3.2 Properties of FC-72

In two-phase cooling systems, the thermophysical properties of the fluid used directly affect the heat transfer performance of the system. To reduce electrical conductivity, the fluid should have a large dielectric strength. FC-72 from 3M Ltd was thus selected as the phase-change coolant. It is a dielectric fluid widely used for electronic cooling applications. FC-72 is also thermally and chemically stable, non-flammable, and practically non-toxic and have low Global Warming Potential. The thermophysical properties of FC-72 are summarised in Table 4-3.

Table 4-3 Thermophysical properties of FC-72 at 1 atm [107]

Properties at saturation temperature (1 atm)	Value
Boiling point (°C)	56
Liquid density, ρ_l (kg/m ³)	1602.2
Vapour density, ρ_v (kg/m ³)	13.21
Liquid dynamic viscosity (kg/m·s)	4.33×10^{-4}
Liquid specific heat, C_p (J/kg·K)	1101
Liquid thermal conductivity, k (W/m·K)	0.054
Latent heat of vaporisation (kJ/kg)	88
Liquid surface tension, σ (N/m)	7.83×10^{-3}
Dielectric strength at 0.1 inch gap (kV)	38

4.3.3 Experimental Procedures and Data Reduction

Experiments were conducted by first filling the aluminium chamber with FC-72 such that the test specimen was at approximately 60 mm beneath the fluid level. Electrical power was supplied to the two cartridge heaters in the bulk fluid to bring the fluid to saturation temperature of 56°C. To allow degassing of the bulk fluid, saturated conditions were maintained for about 1 hour. Subsequently, the electrical power to the rod heater embedded in the copper block was raised at increments of between 1.0 and 5.0 W and the system was allowed to stabilise for 20 to 50 minutes in the natural convection regime and 10 to 20 minutes in the nucleate boiling regime. For every increment, the steady state condition was considered to be reached when the thermocouple reading was maintained within $\pm 0.5^\circ\text{C}$ for more than 2 minutes. The CHF condition was deemed to be reached when there was a sudden huge temperature increase and the bubble formation frequency was observed to decrease significantly. Thereafter, the power supply to the rod heater in the copper block was switched off to prevent overheating of

the rod heater. For each test specimen, the experiment was carried out at least twice under similar conditions to ensure that the results were repeatable.

All thermocouples were calibrated prior to the experiments and the temperatures were acquired under quasi-steady state condition with the Yokogawa MW100 data acquisition unit at the sampling rate of 2 Hz averaged over duration of 1 minute. The saturation temperature of the bulk fluid (T_{sat}) was evaluated under atmospheric condition. The nucleate boiling heat transfer coefficient (h) was calculated from Equation (4-4), where A represents the base area of the test substrates.

$$h = \frac{q}{A(T_w - T_{sat})} = \frac{q''}{(T_w - T_{sat})} \quad (4-4)$$

The uncertainty of the thermocouple measurements was $\pm 0.2^\circ\text{C}$ while the accuracies of current and voltage reading are within $\pm 0.5\%$ of their full scale. Using the method described by Taylor [108], the average uncertainties of the heat transfer coefficient (h) and heat flux (q'') were determined from Equations (4-5) and (4-6). The uncertainty in h was higher in the low heat flux region as compared to higher heat fluxes, with a maximum of $\pm 20\%$. However, the uncertainty was estimated to be below $\pm 3.5\%$ in the nucleate boiling regime above 15 W/cm^2 .

$$\frac{\Delta h}{h} = \sqrt{\left(\frac{\Delta q''}{q''}\right)^2 + \left(\frac{\Delta T_w}{T_w - T_{sat}}\right)^2 + \left(\frac{\Delta T_{sat}}{T_w - T_{sat}}\right)^2} \quad (4-5)$$

$$\frac{\Delta q''}{q''} = \sqrt{\left(\frac{\Delta T_1}{T_1 - T_2}\right)^2 + \left(\frac{\Delta T_2}{T_1 - T_2}\right)^2 + \left(\frac{\Delta L_{1-2}}{L_{1-2}}\right)^2} \quad (4-6)$$

4.4 Results and discussions

4.4.1 Comparison of plain surface and porous structure

The saturated pool boiling curves of the plain surface and Octet-3.0C-2.5H are shown in Figure 4-12(a). The x -axis shows the surface superheat and the y -axis shows the heat flux. The corresponding heat transfer coefficients at different heat fluxes are shown in Figure 4-12(b). For the plain copper surface, boiling incipience superheat was recorded at 12°C at the heat flux of 3 W/cm^2 . However, for Octet-3.0C-2.5H, boiling incipience superheat was recorded at 2.6°C at 1 W/cm^2 . The results of the plain copper surface were compared with that by Yu and Lu [35]. It can be observed that results for both the plain surfaces are in good agreement and showed the validity of the current pool boiling experimental setup. The lower boiling incipience superheat and heat flux is due to the larger surface area and more active nucleation sites for the porous structure. The incipience nucleation is highly sensitive to the nucleation sites available on the surface. For the porous lattice structure, the rougher surface and intersections of struts form cavities which can be suitable for nucleation to occur readily, thus allowing lower boiling incipience superheat as compared to the plain surface.

Figure 4-12 (a) Nucleate pool boiling curves and (b) heat transfer coefficients of plain copper surface and Octet-3.0C-2.5H

The overall heat transfer performances (h_{ave}) of the substrates were calculated by averaging the heat transfer coefficients over the range of heat fluxes in the nucleate boiling regime. By using this computation method, the h_{ave} values of the plain surface and Octet-3.0C-2.5H are $0.48 \text{ W/cm}^2\cdot\text{K}$ and $1.10 \text{ W/cm}^2\cdot\text{K}$, respectively, showing an enhancement of 129% for the porous structure. For the plain surface, the CHF was 17.5 W/cm^2 , whereas the CHF of Octet-3.0C-2.5H was 82 W/cm^2 corresponding to a delay of CHF by 369%.

The macroscopic visualisations of the pool boiling processes of the plain surface and Octet-3.0C-2.5H are depicted in Figure 4-13 and Figure 4-14, respectively. For the plain surface, individual nucleation of bubbles was observed at 5 W/cm^2 in Figure 4-13(a). Coalescence occurred readily at 10 W/cm^2 , forming mushroom-shaped bubbles as shown in Figure 4-13(b). At 16 W/cm^2 , a continuous vapour plume was observed as shown in Figure 4-13(c). CHF was marked by a distinct vapour film or dry-out region forming at the base and corresponding reduction of bubble formation as shown in Figure 4-13(d). The CHF occurred due to hydrodynamic instability as postulated by Zuber [24] and the liquid could not be replenished at the heated surface effectively, leading to a large increase in surface temperature.

For the porous structure Octet-3.0C-2.5H, individual bubble nucleation occurred as shown in Figure 4-14(a) at 7 W/cm^2 . Coalescence of bubbles was delayed significantly as compared to plain surface. In Figure 4-14(b), it can be observed that some coalescence occurred at 25 W/cm^2 . In Figure 4-14(c) at 58 W/cm^2 , the

bubble departure diameter increased, and coalescence occurred rapidly to form a mushroom-shaped bubble encompassing the entire structure. Finally, CHF is deemed to occur at 82 W/cm^2 due to a sudden jump in superheat registered. As shown in Figure 4-14(d), the bubble dynamics of porous structure is vastly different from the plain surface; the porous structure prevented the formation of vapour film and thus significantly delaying the CHF.

Figure 4-13 Pool boiling visualisation of plain copper surface

Figure 4-14 Pool boiling visualisation of Octet-3.0C-2.5H

The enhancement of pool boiling from porous structures can be attributed to increased surface area, higher active nucleation site density and capillary-assisted suction. The solid struts of the porous structure act as conduction media for heat to spread upwards to the higher region of the structure. At higher heat flux, the temperature of the solid struts at the higher region can reach a sufficiently high surface superheat to become active nucleation sites. The process of SLM typically causes partially sintered powder to adhere to the surface of the structure.

The surface roughness caused by the partially sintered powder can produce active nucleation sites for the boiling to occur.

4.4.2 Effect of unit cell size

Figure 4-15 shows the variation of heat transfer coefficient at the different heat fluxes for the same structures at height of 2.5 mm. It was observed that for heat fluxes below 10 W/cm^2 , h is similar across the different structures. However, above 10 W/cm^2 , the larger unit cell sizes performed better. All the porous structures demonstrated delay of CHF as compared to the plain copper surface. The CHF for the 2-mm, 3-mm and 5-mm unit cell sizes were 71, 82 and 96 W/cm^2 respectively, showing an increasing trend with larger unit cell sizes. The results show that for the structure height of 2.5 mm, bubble evacuation resistance is the dominant factor which affects nucleate boiling. The larger pore diameter allows less resistance and more rapid bubble evacuation, thus allowing high rate of heat transfer and delayed CHF. Figure 4-16 shows the boiling patterns of the three substrates with 2.5 mm structure height size at a heat flux of 40 W/cm^2 . The boiling process is vigorous, and the bubbles encompassed the entire substrates. It can be observed that the 5-mm unit cell size exhibits relatively larger bubble size at departure.

Figure 4-15 Heat transfer coefficients of different unit cell sizes at 2.5 mm height

Figure 4-16 Boiling patterns at heat flux of 40 W/cm² for (a) Octet-2.0C-2.5H (b) Octet-3.0C-2.5H and (c) Octet-5.0C-2.5H

Figure 4-17 shows that Octet-3.0C-5.0H performed better than Octet-5.0C-5.0H in the range of 10-70 W/cm², above which there is a rapid deterioration of h . For Octet-3.0C-5.0H within this heat flux range, the combination of higher surface area, higher capillary-assisted liquid suction and sufficiently low bubble evacuation resistance allowed better nucleate boiling performance. However, above 70 W/cm², the high volumetric rate of bubble formation causes rapid coalescence and the dominant factor shifts to bubble evacuation resistance. Thus, the 5-mm unit cell size allowed sustained rapid bubble evacuation due to the

larger pore size and achieved the highest CHF at 107 W/cm^2 . The 2-mm unit cell size was observed to perform the worst throughout entire heat flux range.

Figure 4-17 Heat transfer coefficients of different unit cell sizes at 5.0 mm height

From Figure 4-18, it is observed that Octet-5.0C-10.0H performed better than Octet-3.0C-10.0H for the entire range of heat flux. At the structure height of 10.0 mm, the dominant factor is the bubble evacuation resistance as the bubbles need to traverse a greater height posed by the taller substrate. Thus, a larger pore diameter favours more rapid bubble evacuation and allows higher heat transfer rate. Once again, the 2-mm unit cell performed the worst. Figure 4-19 shows the boiling patterns of the three substrates at heat flux of 40 W/cm^2 . For the 2-mm and 3-mm unit cell sizes, high bubble evacuation resistance causes the bubbles to escape from the sides of substrates preferentially, leaving partial dry-out regions on the upper portion of the structures. However, bubbles could permeate the entire structure for the 5-mm unit cell size and sustain superior nucleate boiling.

Figure 4-18 Heat transfer coefficients of different unit cell sizes at 10.0 mm height

Figure 4-19 Boiling pattern at heat flux of 40 W/cm^2 for (a) Octet-2.0C-10.0H (b) Octet-3.0C-10.0H and (c) Octet-5.0C-10.0H

4.4.3 Effect of height of structure

Figure 4-20 shows the pool boiling heat transfer performance of different structure heights at fixed 2.0-mm unit cell size. The results show that Octet-2.0C-10.H performed the best, followed by Octet-2.0C-2.5H and Octet-2.0C-5.0H. The substrate of 2.5-mm height performed better than that of 5.0-mm height due likely to less bubble evacuation resistance. However, further increase of height to 10.0 mm allowed more surface area for nucleation, which surpassed the effect caused by higher bubble evacuation resistance.

Figure 4-20 Heat transfer coefficients of different heights for 2-mm unit cell

The boiling patterns are shown in Figure 4-21. It can be observed that there are non-bubble regions for the 5-mm structure height which cause it to perform sub-optimally. Despite having some local non-bubble regions for the 10-mm structure height, the overall nucleate boiling surface is significantly higher than the other two substrates, thus allowing more vigorous nucleation and leads to the highest nucleate boiling performance. The non-bubble regions are marked by the blue circles.

Figure 4-21 Boiling patterns at heat flux of 40 W/cm^2 for (a) Octet-2.0C-2.5H
(b) Octet-2.0C-5.0H and (c) Octet-2.0C-10.0H

Figure 4-22 shows the pool boiling heat transfer performance of different structure heights at fixed 3.0-mm unit cell size. The substrate with 2.5-mm height performed worse than the 5.0 and 10.0-mm heights. The heat transfer performance increases with increasing height from 2.5 mm to 5.0 mm due likely

to the larger surface area for nucleation. However, further increase of height to 10.0 mm resulted in a worse performance compared to the structure of 5.0 mm height, due likely to the additional bubble evacuation resistance imposed. It can also be observed that the 5.0 mm height performed significantly better than the other two substrates above 15 W/cm^2 . Thus, for the 3.0-mm unit cell size, a good balance of surface area and bubble evacuation resistance is achieved at the height of 5.0 mm.

Figure 4-22 Heat transfer coefficients of different heights for 3-mm unit cell

Figure 4-23 shows the pool boiling heat transfer performance of different structure heights at fixed 5-mm unit cell size. The substrate with 2.5-mm height performed worse than the 5.0 and 10.0-mm height. The 10.0-mm height structure performed better than the 5.0-mm height structure at heat flux below 50 W/cm^2 , however the performance deteriorated to that of 2.5-mm height structure beyond 50 W/cm^2 . The result shows that bubble resistance increases rapidly above 50 W/cm^2 , which negates the effect of increased surface area for the 10.0-mm height structure. Thus, for the 5-mm unit cell size, a good balance of surface area and bubble evacuation resistance is achieved at the height of 5.0 mm.

Figure 4-23 Heat transfer coefficients of different height for 5-mm unit cell

4.4.4 Summary of nucleate boiling performance and CHF

A summary of the nucleate pool boiling performances of all substrates is shown in Table 4-4. The enhancements of the average heat transfer coefficient and delay of CHF are benchmarked against both the plain surface and Octet-2.0C-5.0H structure. The nucleate boiling enhancement is attributed to the combined effects of increased surface area and capillary-assisted liquid suction but inhibited by form drag caused by bubble evacuation through the porous lattice structure. The competing effects are coupled together, and the contributions of each factor are difficult to distinguish. For the same structure height, the smaller unit cell structure has a larger surface area and larger capillary-assisted liquid suction, albeit at the expense of larger bubble evacuation resistance. For the same unit cell size, a larger height favours a larger surface area for nucleation especially at higher heat fluxes but imposes a longer distance for bubbles to evacuate the structure.

Based on the results, the highest average heat transfer coefficient is achieved for Octet-3.0C-5.0H. Generally, the 2.0-mm unit cell size performed the worst. This shows that the bubble evacuation resistance plays a more dominant role as

compared to the surface area and capillary-assisted liquid suction, especially at high heat flux levels when the volumetric vapour generation rate is very high. The highest CHF is achieved by Octet-5.0C-5.0H at 107 W/cm². Generally, a higher CHF is achieved for larger unit cell sizes and intermediate structure height of 5.0 mm.

Table 4-4 Summary of nucleate pool boiling performances of all substrates

Name	h_{ave}	$\frac{h_{ave}}{h_{plain}}$	$\frac{h_{ave}}{h_{Octet-2.0C-2.5H}}$	CHF	$\frac{CHF}{CHF_{plain}}$	$\frac{CHF}{CHF_{Octet-2.0C-2.5H}}$
		h_{plain}	$h_{Octet-2.0C-2.5H}$		CHF_{plain}	$CHF_{Octet-2.0C-2.5H}$
Plain surface	0.48	-	-	17.5	-	-
Octet-2.0C-2.5H	0.84	1.75	1.00	71	4.06	1.00
Octet-2.0C-5.0H	0.78	1.63	0.93	81	4.63	1.14
Octet-2.0C-10.0H	0.96	2.00	1.14	84	4.80	1.18
Octet-3.0C-2.5H	1.1	2.29	1.31	82	4.69	1.15
Octet-3.0C-5.0H	1.35	2.81	1.61	95	5.43	1.34
Octet-3.0C-10.0H	1.2	2.50	1.43	93	5.31	1.31
Octet-5.0C-2.5H	1.01	2.10	1.20	96	5.49	1.35
Octet-5.0C-5.0H	1.24	2.58	1.48	107	6.11	1.51
Octet-5.0C-10.0H	1.23	2.56	1.46	90	5.14	1.27

4.4.5 Comparison with results from the literature

The results of the current work are compared with those of other investigators using FC-72 as the working fluid as shown in Figure 4-24. The comparison focuses on other surface modification techniques as there very few studies of pool boiling from porous foams with FC-72 as the working fluid. Compared to the current substrate Octet-3.0C-5.0H, Rainey and You [36] obtained similar heat transfer performance with a finned surface of 2-mm height although the CHF was observed to occur at a lower heat flux of about 50 W/cm². Wu et al. [109] used TiO₂-coated nanoparticles on copper and achieved similar heat transfer performance but a low CHF of about 20 W/cm². Kim et al. [110] placed free particles of 0.6 to 0.8 μm on a plain copper surface which yielded significantly

lower heat transfer performance. Sarangi et al. [111] deposited sintered copper particles of about 200 μm on a copper surface and obtained high heat transfer coefficients, but observed a lower CHF of about 10 W/cm^2 due to insufficient fluid replenishment. An et al. [112] used supersonically-sprayed reduced graphene oxide on a copper substrate which achieved moderate heat transfer performance and CHF of about 15 W/cm^2 . Recently, Ho et al. [113] fabricated a micro-fin substrate of 400 μm height and achieved CHF of about 45 W/cm^2 . As seen from the literature, surface modification techniques can enhance the heat transfer coefficients but were often limited in their capability to delay the CHF, which limits the heat flux range which the surfaces can operate in. The current designs of porous lattice structures thus have the potential to enhance the heat transfer coefficient and delay CHF effectively.

Figure 4-24 Comparison with other investigators using FC-72 as working fluid

The comparison with published results of similar porous foam structures but with different fluids is shown in Figure 4-25. Fluid properties play an important role in the heat transfer behaviour and bubble dynamics for pool boiling. One of the

more important properties is the latent heat of vaporisation (h_{fg}). FC-72 has an h_{fg} value of 88 kJ/kg, while the values for acetone and water are 513 kJ/kg and 2257 kJ/kg, respectively. A higher h_{fg} allows a lower volumetric rate of bubble generation for the same heat removal which typically reduces coalescence and delays the CHF. Xu and Zhao [62] investigated the pool boiling from copper foams of 80 PPI porosity with water as the coolant and achieved high heat transfer coefficients and CHF of about 95 W/cm². Xu et al. [60] used copper foams of 30 PPI with acetone as the coolant and achieved higher heat transfer coefficients but a similar CHF of about 95 W/cm² as that of Octet-3.0C-5.0H. Ji et al. [114] used a 3D porous surface of about 2 mm thickness and acetone as the fluid, which achieved high CHF of 130 W/cm².

Figure 4-25 Comparison with porous foam or 3D porous materials using other fluids

4.5 Heat transfer mechanisms

In this section, the heat transfer mechanisms of the different heat flux levels in the nucleate boiling regime are proposed based on observations of the boiling process. In general, the porous structures allow better nucleation as compared to the plain surface due to the larger surface area and capillary-assisted liquid suction which allow better liquid replenishment. However, the solid struts of the porous structure impose form drag on the evacuating bubbles, causing bubble evacuation resistance. The bubble evacuation resistance due to form drag is elevated for higher heat flux, higher structure height and smaller unit cell size. The heat transfer mechanisms are classified under the low, mid, high and very-high heat flux levels. Due to the 3D geometry imposed by the porous structures, there are numerous pathways for the liquid and vapour to flow. Depending on the type of structure, the boiling mechanism changes from stable to unstable form at high heat flux due to major liquid-vapour counter-flow.

4.5.1 Low heat flux level

In the low heat flux level, typically below 15 W/cm^2 , nucleation generally occurs at the base of the porous structures. This is because the temperature is not high enough at the higher region of the porous structure to initiate nucleation. Bubbles generated are typically small and do not coalesce. The bubbles rise to the top of the porous structure by buoyancy and liquid replenishment can only occur from the sides of the structure, forming an orderly convection pathway, as shown in Figure 4-26. In this heat flux level, there is minimal liquid-vapour counter-flow.

Figure 4-26 Boiling mechanisms at low heat flux level

4.5.2 Mid heat flux level

In the mid heat flux level, typically between 15 and 40 W/cm², nucleation occurs readily throughout the porous structure as the temperature at the upper region of the structure becomes sufficiently high to activate the nucleation sites. The coalescence of bubbles can be readily observed. The bubbles typically coalesce within the porous structure and become bigger than the pore diameter before departure as shown in Figure 4-27. This phenomenon was also observed by Xu et al. [60], and the term “cage-shaped” bubbles was used to describe it. The formation can be attributed to increased drag force of the bubbles within the porous structure and surface tension at the structure interface, thus delaying the bubble departure frequency.

For the plain surface, CHF occurs at 17.5 W/cm² due to hydrodynamic choking when the liquid cannot replenish effectively to the heated surface. However, due to increased surface area and capillary-assisted suction into the porous structure, nucleate boiling regime can be maintained. However, the formation of cage-shaped bubbles slightly impedes effective liquid replenishment causing some liquid-vapour counter-flow.

Figure 4-27 Boiling mechanisms at mid heat flux level

4.5.3 High heat flux level

The high heat flux level is typically between 40 and 80 W/cm². As the heat flux continues to increase, the temperature of the entire porous structure also increases and allows vigorous nucleation to take place throughout the entire structure. The high rate of bubble formation and increased drag force cause rapid coalescence within the porous structure and often lead to formation of mushroom-shaped bubbles encompassing the entire porous structure. At this juncture, two separate modes of stable and unstable patterns can be observed.

The stable boiling pattern is observed for porous structures of the larger pore diameter, i.e., 5-mm unit cell size. Due to the larger pore diameter, there is smaller drag force acting on the bubbles within the porous structure, allowing quicker evacuation of bubbles which occurs mainly from the top of the structure. This allows the liquid-vapour counter-flow to be lowered as liquid replenishment can still readily occur from the sides, facilitating a good convection pathway as illustrated in Figure 4-28(a).

However, for porous structures of smaller pore diameter, i.e., 2-mm and 3-mm unit cell sizes, an unstable boiling pattern is observed. The larger drag force caused by the smaller pore diameters, however, makes it difficult for the bubbles

to evacuate upwards effectively through the porous structure, causing the bubbles to evacuate from the sides instead, as illustrated in Figure 4-28(b). This causes a large liquid-vapour counter-flow as the orderly liquid replenishment is impeded, leading to a lower heat transfer coefficient and causing CHF to occur earlier than where the boiling pattern is stable.

Figure 4-28 Boiling mechanism at high heat flux level for (a) stable boiling pattern and (b) unstable boiling pattern

Figure 4-29 shows the experimental visualisation of various porous structures with a height of 10.0 mm at heat flux of 70 W/cm^2 . For Octet-2.0C-10.0H and Octet-3.0C-10.0H, vigorous nucleation occurs nearer to the base of the structures, whereas dry-spots were observed at the upper portion of the structure. This shows the unstable boiling process for such porous structures, which reduces the ability for fresh liquid to replenish to the base of the structure. However, it can be observed for Octet-5.0C-10.0H that bubbles can evacuate effectively through the upper portion of the structure. The fresh liquid can then replenish more effectively to the base, forming an orderly convection pathway, and thus, allowing a stable boiling process.

Figure 4-29 Boiling patterns at heat flux of 70 W/cm² for (a) Octet-2.0C-10.0H
(b) Octet-3.0C-10.0H and (c) Octet-5.0C-10.0H

4.5.4 Very high heat flux level

The very high heat flux level typically occurs above 80 W/cm². For the stable boiling pattern, further increase in heat flux causes higher rate of bubble formation. At this stage, an interesting phenomenon of bubble pulsation occurs. The different stages of the bubble pulsation are illustrated in Fig. 25 for the substrate of Octet-3.0C-2.5H at heat flux of 80 W/cm². Figure 4-30(a) shows the vaporisation of liquid and formation of mushroom-shaped bubbles which encompass the entire substrate. Figure 4-30(b) shows that the bubbles start to rise due to buoyancy. In Figure 4-30(c), necking is observed between the rising bubbles and substrate itself, and the cycle is completed in Figure 4-30(d) when the bubbles depart, leaving behind a thin film surrounding the substrate. During the pulsation process, fresh liquid can replenish the porous structure during the period when the mushroom-shaped bubbles depart, thus allowing sustained nucleate boiling at very high heat flux and delaying the hydrodynamic choking effect which causes CHF. This phenomenon was also observed by Xu et al. [61] who reported that the pulsation frequency increased as the heat flux was increased. However, there is a limit to which the pulsation could keep up to

replenish the fresh liquid at further increased heat flux, eventually causing CHF to be reached and an unstable increase in surface temperature.

Figure 4-30 Boiling patterns at heat flux of 80 W/cm^2 for Octet-3.0C-2.5H for different stages of pulsation

4.6 Summary

In this chapter, pool boiling of porous structures was investigated with saturated FC-72 under atmospheric condition. Nine porous structures of the octet-truss unit cell geometry were fabricated using the SLM technique. The porous structures were varied in terms of unit cell sizes of 2.0 mm, 3.0 mm and 5.0 mm, and structure heights of 2.5 mm, 5.0 mm and 10.0 mm. The following conclusions are derived from this study:

1. The porous structure enhanced nucleate boiling heat transfer coefficient and delayed CHF effectively as compared to the plain surface as benchmark. The enhancement is attributed to the increased surface area, increased nucleation site density and capillary-assisted suction of the porous structure. The porous structure allows sustained liquid replenishment which delays the hydrodynamic choking and CHF significantly.
2. The effect of unit cell size is studied for the different substrates of the same structure height. The larger unit cell sizes of 3 mm and 5 mm are better for

enhancing nucleate pool boiling as the dominant factor is bubble evacuation resistance, even though the smaller unit cell size of 2 mm has a larger surface area and capillary-assisted liquid suction. The best performing substrate is Octet-3.0C-5.0H and it has an average nucleate boiling heat transfer coefficient of $1.35 \text{ W/cm}^2\cdot\text{K}$, which is 2.81 times that of the plain surface at $0.48 \text{ W/cm}^2\cdot\text{K}$.

3. The effect of structure height is studied for the different substrates of the same unit cell size. A higher structure height has more surface area for nucleation although it imposes more bubble evacuation resistance. In general, the 5-mm height structure possesses a good balance between the two factors and allows higher nucleate boiling heat transfer coefficient. The highest CHF is achieved by Octet-5.0C-5.0H at 107 W/cm^2 , which is 6.11 times that of the plain surface at 17.5 W/cm^2 .
4. The heat transfer mechanisms are proposed for the different heat flux levels in the nucleate boiling regime of the porous structures based on visual observations. The boiling patterns are classified as low, mid, high and very-high heat flux levels. At low and mid heat flux levels, the boiling pattern change from discrete bubble departure to coalescence. At high heat flux level, two separate modes of stable and unstable boiling patterns are observed. For the stable boiling pattern, there are distinct bubble departure and liquid replenishment pathways which facilitates good convection flow. However, for the unstable boiling pattern, there is major liquid-vapour counter-flow, which disrupts the orderly liquid replenishment pathway. At the very-high heat flux level, bubble pulsation is observed, which helps to sustain fresh liquid intake and high rate of vapour formation.

Chapter 5 - Pool Boiling of Re-entrant Substrates

5.1 Introduction

In the previous chapter, pool boiling enhancements have been achieved by using porous lattice structures of varying heights and pore densities. SLM has been proven to successfully fabricate the lattice substrates with intricate and complex internal features accurately. However, visual observations showed that vapour evacuation was impeded by the struts of the lattice structure at high heat fluxes and led to deterioration of heat transfer performance. Thus, in this chapter, a new design concept using re-entrant 3D substrates was proposed to provide good separation of fluid and vapour flow paths and to minimise counter-flow.

For this chapter, the effects of channel width and configuration were investigated. FC-72 was used as the working fluid at saturated condition under atmospheric pressure. In addition, high speed visualisation was employed to analyse the heat transfer mechanisms occurring in the re-entrant substrates.

5.2 Substrate preparation and characterisation

5.2.1 Description of substrates

In this study, the re-entrant substrates consist of vertical and horizontal channels extruded within a solid block of 10 mm (length) by 10 mm (width) by 5 mm (height). They were designed in the SolidWorks software. The design concept is that during the pool boiling process, the vertical channels allow quick vapour evacuations due to buoyancy effect, while the horizontal channels at the base of the substrate allow liquid replenishment to sustain the boiling. A convection

cycle can thus be set up to allow separate pathways for liquid-vapour movements, while maximizing the internal surface area for nucleate boiling process to take place. One example of the re-entrant substrate with the liquid-vapour pathways is shown in Figure 5-1.

Figure 5-1 CAD model of re-entrant substrate

To study the effect of channel width, substrates of channel widths of 0.8 mm, 1.0 mm and 1.3 mm were designed, as shown in Figure 5-2. For each individual substrate, the widths of vertical and horizontal channels were kept identical. Generally, the substrate with smaller channel width can accommodate more channels and vice versa. The total internal surface areas were maintained to be almost similar while varying the number of channels available for each channel width. The parameters of the substrates are presented in Table 5-1.

Figure 5-2 Re-entrant substrates of different channel widths

In addition, to study the effect of area density, each substrate of a fixed channel width was designed with a dense and sparse arrangement. As the bulk volume of the substrate was kept constant, a dense arrangement would have a smaller pitch and accommodate more channels than the sparse one. The differences in number of channels for the dense and sparse arrangement are also summarised in Table 5-1.

Finally, a third set of design consists of specialised internal channels where the vertical channels gradually spread outwards. The purpose is to direct the vapour evacuation pathway outwards radially and make use of the side surfaces for bubble evacuation, yet maintain high internal surface area by maximising the number of channels. These substrates are labelled as the Spread substrates and consists of channel widths of 0.8 mm, 1.0 mm and 1.3 mm. The internal features of these substrates are shown in Figure 5-3.

Figure 5-3 Illustration of Spread substrates with different channel widths

Therefore, a total of nine re-entrant substrates were designed. For easy reference in the subsequent sections, the substrates were labelled based on the channel width and configuration. For example, the substrate with the name of Vert-0.8C-dense has vertical channels with channel width of 0.8 mm and dense arrangement, while Spread-0.8C has channels of 0.8 mm and spreading configuration. A plain surface is used as a benchmark. The internal surface areas and porosities of each substrate were evaluated too. A summary of the substrate parameters is shown in Table 5-1.

Table 5-1 Summary of re-entrant substrate characteristics

S/n	Name	Channel width (mm)	Channel arrangement	Internal surface area (mm ²)	Porosity
1	Plain surface	-	-	100	-
2	Vert-0.8C-dense	0.8	8 x 8	1019	0.483
3	Vert-1.0C-dense	1.0	7 x 7	984	0.574
4	Vert-1.3C-dense	1.3	6 x 6	893	0.698
5	Vert-0.8C-sparse	0.8	6 x 6	684	0.310
6	Vert-1.0C-sparse	1.0	5 x 5	600	0.350
7	Vert-1.3C-sparse	1.3	5 x 5	508	0.468
8	Spread-0.8C	0.8	8 x 8	968	0.466
9	Spread-1.0C	1.0	7 x 7	920	0.550
10	Spread-1.3C	1.3	6 x 6	816	0.665

5.2.2 Fabrication of substrates

The substrates were fabricated using SLM, which was elaborated in Chapter 3. To briefly summarise the process, the CAD files were converted to STL format and imported into the “Magics” software to input the laser parameters for the SLM 250 HL machine. After fabrication, the substrates were vacuumed to remove internal residual powder and then separated from the baseplate using wire-cutting method. The fabricated substrates are shown in Figure 5-4.

(a) Left to right: Vert-0.8C-dense, Vert-1.0C-dense, Vert-1.3C-dense

(b) Left to right: Vert-0.8C-sparse, Vert-1.0C-sparse, Vert-1.3C-sparse

(c) Left to right: Spread-0.8C, Spread-1.0C, Spread-1.3C

Figure 5-4 Images of fabricated re-entrant substrates

To ascertain the proper formation of internal channels, especially for the Spread substrates which contain the highest complexity, some sectioned samples were

produced. For illustration, the Spread-1.0C substrate and its different sectioned profiles are shown in Figure 5-5. The figure shows a comparison of the CAD drawings of the substrate sectioned at different regions and the actual fabricated samples. It can be observed that all the internal channels were built successfully.

Figure 5-5 Comparison of fabricated samples with design for Spread-1.0C

5.2.3 Light optical imaging of substrates

Measurements of the re-entrant substrates was performed using a light optical microscope (Olympus SZX7) with its in-built measurement software. Figure 5-6 shows the top and side views of the vertical substrates with dense arrangement and different channel widths. It can be observed that the channels were well-formed and orderly. From the side views, some excess sintered powder can be observed due to the overhang feature when during the fabrication process. However, there was no blockage of the horizontal channels.

Figure 5-6 Light optical imaging of top and side views of re-entrant substrates with dense arrangement

A summary of the dimensions measured using the optical microscope for the substrates is shown in Table 5-2. The average measurements were obtained from about 20 readings for each vertical and horizontal channel widths. Comparisons with the design values show small deviations, with the highest deviation at 6.53%, thus confirming that the dimensional accuracy of the fabricated parts.

Table 5-2 Summary of measurements of re-entrant substrates

Substrate	Width of vertical channel (μm)		Deviation (%)	Width of horizontal channel (μm)		Deviation (%)
	Design	Measured		Design	Measured	
Vert-0.8C-dense	800	829.6	3.70	800	812.2	1.52
Vert-1.0C-dense	1000	998.5	-0.15	1000	1065.3	6.53
Vert-1.3C-dense	1300	1352.6	4.04	1300	1283.7	-1.25

5.3 Pool boiling facility and data reduction

The pool boiling facility used for this study was the same as that described in Section 4.3. However, one minor change was that a clear polycarbonate chamber was fabricated to replace the previous aluminium chamber for better viewing purpose. The chamber maintained the same dimensions of 12 cm by 12 cm by 12 cm. The new polycarbonate chamber is shown in Figure 5-7.

Figure 5-7 Photograph of polycarbonate pool boiling chamber

A high-speed camera (Photron FASTCAM 1024 PCI) was used to capture the boiling process. The camera was positioned to capture the frontal profile of the substrates. Illumination was provided by a main lighting source with high-power lamps and two auxiliary LED lightings. The set-up is shown in Figure 5-8. High-speed videos were obtained at different heat fluxes for each substrate. For each video, capturing frequency of either 1000 Hz or 2000 Hz was used and 600 images were obtained for analysis.

Figure 5-8 High-speed camera set-up for pool boiling

The experimental procedures and data reduction are the same as those presented in Section 4.3. Saturated FC-72 was also used as the fluid for this study, which has a boiling point of 56°C at atmospheric conditions. The other key properties can be found in Table 4-3.

5.4 Results and discussions

5.4.1 Effect of channel width

The heat transfer performance of the re-entrant substrates with dense configuration is shown in Figure 5-9. From Figure 5-9(a), all the re-entrant substrates showed significant nucleate boiling heat transfer enhancements as shown by the left-ward shift of the boiling curves. For the plain surface, CHF was reached at about 18 W/cm², which was comparable to the previous result in Chapter 4 at 17.5 W/cm². Figure 5-9 (b) shows a better relative performance for the different re-entrant substrates by comparing the heat transfer coefficient (h) for various heat fluxes.

For the re-entrant substrates, h generally increases as heat flux increases, reaching a peak, and decline thereafter. The increase in h initially can be explained by increased nucleation sites at higher surface superheat. However, upon reach high heat fluxes, high volumetric generation rate of vapour caused bubble evacuation resistance. Eventually, some liquid-vapour counter-flow was observed at high heat fluxes, which greatly impeded the convection cycle and thus reduces h . To better understand the boiling phenomenon, visualisation images obtained from the high-speed camera were analysed.

(a)

(b)

Figure 5-9 (a) Nucleate pool boiling curves and (b) heat transfer coefficients of vertical substrates with dense configuration

Figure 5-10 shows the boiling pattern of the plain surface at various heat fluxes. Like the visualisations presented in Chapter 4, the boiling patterns evolved from discrete bubbles (Figure 5-10-a) to coalescence (Figure 5-10-b) and mushroom-shaped (Figure 5-10-c) bubbles as the heat flux increases. Finally, at CHF, an obvious vapour film could be observed in Figure 5-10(d). This resulted in rapid rise of temperature of the heated surface and thus the experiment was terminated.

Figure 5-10 Pool boiling visualisations of the plain surface

Figure 5-11 shows the boiling patterns of the re-entrant substrates with dense configuration at various heat fluxes. Generally, at a low heat flux of 10 W/cm²,

orderly and discrete bubbles emerged from the top surface of the substrates. For Vert-1.3C-dense, there is significantly less active nucleation sites than those of 0.8-mm and 1.0-mm channel widths, thus resulting in lower h as shown in Figure 5-9(b) previously. At 20 W/cm^2 , the nucleation density increases and some coalescence at the top surface of the substrates could be observed. At 40 W/cm^2 , due to higher vapour evacuation resistance, some bubbles could be seen spilling from the base of the Vert-0.8C-dense and Vert-1.0C-dense substrates, however no spillage was observed for Vert-1.3C-dense. The spillage of bubbles increases at higher heat flux and eventually causes some liquid-vapour counter-flow. This led to the decrease in h for Vert-0.8C-dense and Vert-1.0C-dense, whereas the larger channel width of Vert-1.3C-dense was able to sustain liquid replenishment and maintained relatively higher h .

Figure 5-11 Pool boiling visualisations of re-entrant substrates with dense configuration

The variation of heat transfer coefficient with heat flux for the re-entrant substrates with sparse arrangement is shown in Figure 5-12. Similar to the substrates with dense arrangement, h increases as heat flux increases initially before reaching a peak. While Vert-0.8C-sparse and Vert-1.0C-sparse show sharp deterioration of h at high heat flux, Vert-1.3C-sparse shows relatively stable performance.

Figure 5-12 Heat transfer coefficients of vertical substrates with sparse configuration

The boiling patterns of the re-entrant substrates with sparse arrangement is shown in Figure 5-13. Like the substrates with dense arrangement, spillage of bubbles from the base is observed at heat fluxes above 40 W/cm^2 for Vert-0.8C-sparse and Vert-1.0C-sparse. As the total number of inlet horizontal channels for the sparse arrangement is less than the dense arrangement, each channel needs to handle higher volumetric liquid intake to sustain the boiling process at same heat flux. Thus, the issue of spillage seems to worsen, and relatively larger

occurrences of spillage can be observed from 40 W/cm^2 onwards. In the case of Vert-1.3C-sparse, some spillage can be observed at 60 W/cm^2 too, but it did not seem to be severe enough to cause major liquid-vapour counter-flow as compared to that of Vert-0.8C-sparse and Vert-1.0C-sparse.

Figure 5-13 Pool boiling visualisations of re-entrant substrates with sparse configuration

5.4.2 Effect of substrate configuration

The heat transfer performance of the re-entrant substrates at fixed channel width of 0.8 mm is shown in Figure 5-14. From the results, the dense configuration performed the best, due to the highest internal surface area, followed by spread and sparse arrangement. This shows that for a fixed volume, the total surface area has a strong effect on the heat transfer performance as it affects the nucleation site density directly. A higher nucleation site density allows better removal of latent heat by the bubbles and correspondingly decrease the substrate surface temperature.

Figure 5-14 Heat transfer coefficients of substrates with 0.8 mm channel width

Figure 5-15 shows the boiling pattern of the re-entrant substrates with channel width of 0.8 mm. Since the macroscopic images at 10 W/cm^2 were similar across the substrates, the focus would be on the images at higher heat fluxes. For the Spread substrate, some bubbles be observed to evacuate from the side channels at the upper portion of substrate from 20 W/cm^2 onwards, thus relieving some vapour evacuation resistance. At 40 W/cm^2 , there is visibly more spillage for both the sparse and spread substrates as compared to that of dense configuration.

Figure 5-15 Pool boiling visualisations of re-entrant substrates with fixed channel width of 0.8 mm

Figure 5-16 shows the heat transfer performance of substrates with 1.0 mm channel width. The dense configuration performed the best and the trend is similar to that of 0.8 mm width as discussed previously. From the visualisations in Figure 5-17, Vert-1.0C-dense substrate could be observed to minimise spillage at 40 and 60 W/cm² as compared to Vert-1.0C-sparse and Spread-1.0C. A larger internal surface area is thus favoured for the nucleate boiling process in the re-entrant substrates. Comparing between the vertical and spread design, even though the internal surface areas are similar, the vertical pathway seems to be the most effective way to minimise the bubble evacuation resistance due to the buoyancy effect.

Figure 5-16 Heat transfer coefficients of substrates with 1.0 mm channel width

Figure 5-17 Pool boiling visualisations of re-entrant substrates with fixed channel width of 1.0 mm

Figure 5-18 shows the heat transfer performance of substrates with 1.3 mm channel width. Once again, the dense arrangement performed the best. However, for all the substrates of 1.3 mm channel width, h seems to plateau instead of deteriorating at high heat fluxes. From the visualisations in Figure 5-19, all the substrates with 1.3 mm channel width show minimal spillage and thus encounter less liquid-vapour counter-flow at high heat fluxes. The larger channel width allows sustained liquid replenishment and also reduces bubble evacuation resistance. For Spread-1.3C, the utilisation of the upper side channels seems to be better as compared to the those of smaller widths.

Figure 5-18 Heat transfer coefficients of substrates with 1.3 mm channel width

Figure 5-19 Pool boiling visualisations of re-entrant substrates with fixed channel width of 1.3 mm

5.4.3 Summary of results and discussions

The overall heat transfer performances (h_{ave}) of the substrates were calculated by averaging the heat transfer coefficients over the range of heat fluxes in the nucleate boiling regime. The results are summarised in Table 5-3. By using this computation method, the highest h_{ave} value was obtained for Vert-1.0C-dense at $1.44 \text{ W/cm}^2\cdot\text{K}$, which is 3.07 times that of the plain surface. Using the worst-performing re-entrant substrate as the second baseline, which is Vert-1.3C-sparse, the Vert-1.0C-dense substrates shows up to 31% enhancement. The relative performances of the different substrates are tabulated as well.

Table 5-3 Summary of results for re-entrant substrates

S/n	Name	h_{ave} (W/cm ² .K)	$\frac{h_{ave}}{h_{plain}}$	$\frac{h_{ave}}{h_{vert-1.3-sparse}}$
1	Plain surface	0.47	-	-
2	Vert-0.8C-dense	1.26	2.67	1.14
3	Vert-1.0C-dense	1.44	3.07	1.31
4	Vert-1.3C-dense	1.28	2.72	1.16
5	Vert-0.8C-sparse	1.11	2.35	1.00
6	Vert-1.0C-sparse	1.12	2.39	1.02
7	Vert-1.3C-sparse	1.10	2.34	1.00
8	Spread-0.8C	1.20	2.56	1.09
9	Spread-1.0C	1.27	2.70	1.16
10	Spread-1.3C	1.28	2.72	1.16

Based on the findings, the effects of channel widths and configurations are significant on the heat transfer performance. For a fixed channel width, the dense configuration performed better than the sparse and spread configuration. This is because the dense configuration possesses the largest internal surface area for nucleation and the vertical channels are more favourable for bubble evacuation as compared to the spread channels. For a fixed configuration, the 1.0-mm channel width performed the best. Even though the substrates with 0.8-mm channel width possess higher internal surface area, bubble evacuation resistance becomes dominant at high heat fluxes and causes spillage, which in turn led to significant liquid-vapour counter-flow.

5.5 Visualisations and analysis

In this section, further analysis is conducted on the visualisations to better understand the boiling phenomenon in the re-entrant substrates.

5.5.1 Analysis of Spread substrates

The design concept of the Spread substrates was to allow better utilisation of the substrate to distribute the bubble evacuation pathways evenly throughout the upper portions of the substrate. While there were some successes, typically problems of spillage and formation of vapour film still plagued the Spread substrates at high heat fluxes.

The transient behaviours of the Spread substrates at different heat fluxes are analysed for better clarity. Figure 5-20 shows the transient behaviours at 20 W/cm² at 5-ms intervals. For all Spread substrates, the upper side channels show bubbles emerging and thus demonstrated that the concept is feasible. However, the frequency of the bubbles emerging from the upper side channels seem to be lower than those from the top channels. This is likely due to a longer pathway experienced by the vapour by moving through the upper side channels as compared to the top, thus reducing its effectiveness.

Figure 5-20 Transient behaviours of Spread substrates at 20 W/cm²

Figure 5-21 shows the transient behaviours at 40 W/cm² at 5-ms intervals. Despite having similar number of channels as compared to the substrates with dense configuration, much spillage can be observed for the Spread.0.8C and Spread-1.0C. This shows that the design did not positively distribute the vapour as intended at moderate heat flux. However, with a large channel width, Spread-1.3C seems to minimise spillage quite well, which is evident for the substrates with vertical channels too.

Figure 5-21 Transient behaviours of Spread substrates at 40 W/cm^2

Overall, the design concept of the Spread substrates shows potential for passive distribution of the vapour in pool boiling. However, due to the current volume constraint, the scope for variation and complexity in the design is limited.

5.5.2 Classification of boiling patterns for re-entrant substrates

Based on the observations of the re-entrant substrates, the boiling patterns can be classified into different categories. The classification and description of each category are as such:

Category	Name	Description
1	Isolated bubbles	Typically occurs in low heat flux, where discrete bubbles are observed to evacuate from the top surface of the internal channels.
2	Coalescence	As heat flux increases, boiling became more vigorous. Some coalescence occurs at the top surface of substrate as the departing bubble diameters are larger than the channel pitch, however no spillage of bubbles from base.
3	Minor spillage	Due to the increased hydrodynamic resistance in the internal channels, some bubbles spill from base of substrates. However, liquid-vapour counter-flow was minimal.
4	Major spillage	As heat flux and associated vaporisation rate increases, the vertical channels could not effectively evacuate the bubbles from the top and major spillage of bubbles occur at the substrate base. Some liquid-vapour counter-flow is expected at this stage.
5	Counter-flow	Massive spillage of bubbles from base of substrates due to high volumetric vapour generation at high heat flux and inability to evacuate from top of substrate. Major liquid-vapour counter-flow and leads to poor heat transfer performance.

From the visualisation study, the classification is mapped out in Figure 5-22. Discrete bubble category typically occurs below 20 W/cm^2 . For the same configuration, it can be observed that the substrate with larger channel widths can delay spillage and occurrence of counter-flow. For the substrates with dense configuration, the larger internal surface areas greater number of channels allow liquid replenishment and bubble evacuation effectively, thus preventing counter-flow from occurring.

Figure 5-22 Classification of boiling pattern for re-entrant substrates

5.6 Summary

Pool boiling experiments were conducted using saturated FC-72 as the fluid at atmospheric condition. Re-entrant substrates were designed with the intention to separate the liquid-vapour pathways. A total of nine re-entrant substrates with channel widths of 0.8, 1.0 and 1.3 mm and different configurations of dense, sparse and spread were fabricated using SLM. The heat transfer performance was

studied and high-speed visualisations were obtained to understand the boiling mechanism. The following conclusions can be drawn for this chapter:

1. For a fixed channel width, the dense configuration performed better than the sparse and spread configuration. This is because the dense configuration possesses the largest internal surface area for nucleation and the vertical channels are more favourable for bubble evacuation as compared to the spread channels.
2. For a fixed configuration, the 1.0-mm channel width performed the best. Even though the substrates with 0.8-mm channel width possess higher internal surface area, bubble evacuation resistance becomes dominant at high heat fluxes and causes spillage, which in turn led to significant liquid-vapour counter-flow.
3. The Spread substrates show potential to distribute the vapour pathways evenly on the upper surface of the substrate. However, the longer diagonal pathways seem to pose higher vapour resistance than the vertical ones and receive lower bubble frequency.
4. The boiling patterns are classified into five regimes, namely isolated bubble, coalescence, minor spillage, major spillage and counter-flow. To avoid major spillage and counter-flow at high heat fluxes, high density of channels and large channel widths are recommended.

Chapter 6 - Bubble Dynamics

Modelling of Re-entrant Substrates

6.1 Introduction

In nucleate pool boiling, bubbles act as energy carriers to remove heat from the heated surface. The study of bubble dynamics is thus important for the understanding of boiling mechanisms. The bubble departure diameter is the most important parameter of bubble dynamics which has been investigated by many researchers in the past decades. However, due to the non-linear growth of bubbles and interactions of many accompanying factors, it is often challenging to characterise the bubble dynamics for a given surface and conditions. Many researchers have relied on high-speed visualisations and obtained correlations to predict the bubble dynamics, as discussed in Chapter 2.

In this chapter, the author aims to study the bubble dynamics of the re-entrant substrates as presented in Chapter 5. The bubble departure diameter will be analysed for the re-entrant substrates of different channel widths. Subsequently, a force balance analysis is developed to predict the bubble departure diameter using semi-empirical method. Finally, the model will be compared with experimental results.

6.2 Description of bubble regime for analysis

The bubble dynamics will be studied for the isolated bubble regime for the re-entrant substrates. In the isolated bubble regime, the individual bubble departure diameter and frequency could be measured with high accuracy. At higher heat fluxes when the boiling become more vigorous, lateral coalescence of bubbles

makes it difficult to accurately measure and predict the bubble departure diameter.

For the re-entrant substrates, the isolated bubble regime occurs when the bubbles are observed to depart from the internal channels at the substrate surface without lateral coalescence. The regime can be sub-divided into two categories, namely onset of nucleation and coalescence within internal channels.

At the onset of nucleation, small bubbles typically nucleate at the base of the re-entrant channels. These bubbles rise in the vertical channels by buoyancy effect and depart the substrate with the bubble sizes typically smaller than the channel width or pore size.

As more nucleation sites are activated, the bubbles tend to coalesce within the vertical channels before departing from the substrate. As the channels become filled with vapour, effect of surface tension becomes more important and the bubbles tend to be held back by the pore at the top of vertical channel. The bubble at the pore then grows due to the continuous supply of vapour from the internal channels. Eventually, the buoyancy force becomes more dominant and the bubble is lifted off with a departure diameter larger than the channel width or pore diameter.

To illustrate the two different sub-categories in the isolated bubble regime, Figure 6-1(a) shows the Vert-1.3C-dense substrate at onset of nucleation. It can be observed that small discrete bubbles nucleate at the base of the substrate and the departing bubbles from the top are typically smaller than the channel width. However, in Figure 6-1(b) at higher heat flux, more nucleation sites are activated, and rapid coalescence takes place in the substrate when the vapour is rising up

through the vertical channels. This causes the bubble to adhere to the channel walls and linger at the pore surface before departing to the bulk fluid. The bubble departure diameters thus become larger than the channel width.

Figure 6-1 Illustration of sub-categories of isolated bubble regime for Vert-1.3C-dense substrate

The process of bubble adherence affects the heat transfer of the re-entrant substrates. However, the mechanism for bubble adherence and growth at the pore entrance is not well-understood. It is hypothesised that the channel width and heat flux have strong effects on the bubble dynamics. Thus, in this chapter, measurements were analysed for substrates in the isolated bubble regime and a force analysis on the departing bubble based semi-empirical model is developed to better understand the phenomenon. The analysis would be focused on re-entrant substrates with dense configuration. For simplicity, the substrates will be identified as their channel width or pore diameter of 0.8, 1.0 and 1.3-mm.

6.3 Experimental measurements of bubble dynamics

6.3.1 Measurement of bubble departure diameters

As described in Chapter 5, a high-speed camera (Photron FASTCAM 1024 PCI) was used to capture the boiling process for the re-entrant substrates. The camera was positioned to capture the frontal profile of the substrates. For each video, capturing frequency of either 1000 Hz or 2000 Hz was used and 600 images were obtained for analysis. Each image is comprised of 512 by 512 pixels.

The measurements of the bubble departure diameter were conducted using the in-built measurement function in the high-speed camera software (Photron FASTCAM Viewer). For each video, using the bulk dimension (10 mm) of the substrate as reference, the conversion factor from pixels-to-length was determined. Subsequently, each image could be zoomed in and the sizes of the individual bubbles could be determined based on the conversion factor. An example of the measurement process is shown in Figure 6-2.

Figure 6-2 Measurement process of bubble diameters

To determine the bubble departure diameter, the images were reviewed to locate the start of a bubble cycle. An example of a complete bubble cycle is shown in Figure 6-3. The cycle begins as the preceding bubble just leaves the pore. Usually a waiting time (t_w) exists before the formation of a new bubble. After the appearance of a new bubble, the time it takes for the bubble to grow is known as the growth time (t_g). Finally, the bubble departure diameter is determined at the

frame which the bubble just departs from the substrate surface, as shown in Figure 6-3(f).

Figure 6-3 Illustration of a complete bubble cycle

To ensure sharpness of the images, only the bubbles appearing from first row of pores closest to the camera were measured for the bubble departure diameters. For each pore, between 10 to 15 bubble departure diameters were measured, and for the entire substrate, at least 30 bubble departure diameters were measured in total. This ensures a large sample size to obtain the mean departure diameter. The same method was used to obtain the mean bubble departure diameter for the different substrates at various heat fluxes in the isolated bubble regime.

It is observed that there is a variation of bubble departure diameter for each substrate at a fixed heat flux, and they were found to approximate a normal distribution. The distributions of bubble departure diameters for the substrate with 1.0-mm pore diameter at 7 and 10 W/cm² are shown in Figure 6-4.

Figure 6-4 Distribution of bubble departure diameters for 1.0-mm pore diameter

Figure 6-5 shows the bubble departure diameters for the different pore diameters at various heat fluxes. The trend shows that for a fixed substrate, the bubble departure diameter increases with heat flux. Across the substrates, the larger pore diameter corresponds to a larger bubble departure diameter. The trends are agreeable with Ramaswamy et al. [115] where high-speed visualisation was conducted on structured substrates with pores of different sizes.

Figure 6-5 Bubble departure diameters at various heat fluxes

6.3.2 Bubble frequency

The bubble frequency was evaluated by the inverse of time taken to complete a bubble cycle. As mentioned previously, a bubble cycle consists of the waiting

time (t_w) and growth time (t_g). The average bubble cycle was measured by taking the time taken for 10 consecutive bubble formation and departure at a pore. At least three pores were used in the measurement for each substrate at a given heat flux to obtain the overall mean bubble frequency.

The bubble frequency for the different pore sizes is shown in Figure 6-6. For each substrate, the bubble frequency increases initially with heat flux and maintained at a relatively constant level subsequently. The frequency of 1.3-mm pore size is significantly higher than those of smaller pores. At 0.8-mm and 1.0-mm pore sizes, the frequencies appear to be similar.

Figure 6-6 Bubble frequency at different heat fluxes

6.3.3 Nucleation site density

The number of activated nucleation sites on the substrate was determined by direct observation. As the pores at the top of the substrate were arranged in a grid pattern, it is possible to estimate the percentage of nucleation by observing from the front, side and orthogonal views.

The percentage of active sites at different heat fluxes for the different substrates is shown in Figure 6-7. For all substrates, the percentages of active sites increase with heat flux due to the corresponding increase in surface superheat. For the 0.8-mm and 1.0-mm pore, almost all the sites were activated at low heat fluxes. However, for 1.3-mm pore, the percentage of active sites increases gradually at low heat flux and only reaches 100% at 10 W/cm². The result shows that a smaller pore size is more favourable for bubble nucleation.

Figure 6-7 Percentage of active sites at various heat fluxes

6.4 Modelling of bubble departure diameter

6.4.1 Proposed physical model

The physical model of the re-entrant substrate is shown in Figure 6-8. In Figure 6-8(a), the liquid intake occurs at the horizontal channel at the base of substrate. Vaporisation occurs at the base and internal vertical walls. When nucleation is sufficiently vigorous, coalescence occurs within the vertical channels and the bubble adhere to the pore surface prior to departure. Figure 6-8(b) shows a close-up model of a single channel. Due to the supply of vapour from the internal

channel, the bubble has a diameter (D_B) which is larger than the pore diameter (D_p). The bubble growth and departure are modelled subsequently.

Figure 6-8 Physical model of re-entrant substrate

A bubble growth and departure model for an individual pore is proposed in Figure 6-9. A bubble is first initiated with the diameter equivalent to the pore size in Figure 6-9(a). Subsequently, vapour is continuously fed to the bubble by the evaporation in the channel and causes the bubble to grow in Figure 6-9(b) and (c). Finally, the bubble departure diameter is determined when the bubble just depart from the pore as shown in Figure 6-9(d).

Figure 6-9 Bubble growth model for an individual pore

6.4.2 Assumptions:

For the proposed model, the following assumptions are made:

1. The liquid temperature is uniform at saturated temperature under atmospheric pressure of 1 bar.
2. The bubble maintains a spherical shape during the growth and departure period.
3. The channels are completely filled with vapour.
4. The vaporisation rate is constant for a given heat flux and the channel provides a steady flow of vapour to allow bubble growth.
5. The pore geometry is assumed to be circular instead of square to simplify the analysis.

From a geometrical analysis for a spherical bubble growing from a pore, as proposed by Chien and Webb [116], the contact angle of the bubble at the channel top can be related to the bubble (R_B) and channel radii (R_p). In this case, the contact angle can be evaluated Equation (6-1) and the schematic is shown in Figure 6-10.

$$\theta = \alpha = \sin^{-1}\left(\frac{R_p}{R_B}\right) \quad (6-1)$$

Figure 6-10 Schematic of geometry of bubble

6.4.3 Force balance analysis

The bubble departure diameter can be modelled using a force balance analysis. During the bubble growth process at a pore, the bubble is subjected to various forces. Positive forces will pull the bubble away from the pore, while negative forces will cause the bubble to be held on to the surface. At initial stages of bubble growth, the total force acting on the bubble is negative and thus the bubble adheres to the mouth of the pore. The bubble departs from the pore when the total force acting on the bubble becomes positive.

For this model, five types of forces are acting a bubble growing at the pore surface. These forces are:

1. Buoyancy force (F_b)
2. Contact pressure force (F_{cp})
3. Surface tension force (F_{st})
4. Drag force (F_d)

5. Inertia force (F_i)

The schematic of the forces acting on the bubble is shown in Figure 6-11.

Figure 6-11 Forces acting on a bubble at the pore surface

Based on the force analysis, the bubble will depart from the pore when the resultant force becomes positive. The resultant force can be evaluated in Equation (6-2).

$$F_b + F_{cp} = F_{st} + F_d + F_i \quad (6-2)$$

6.4.3.1 Buoyancy force

The buoyancy force is a positive force due to the density difference between the vapour and liquid. For the case of a spherical bubble, it equal to the weight of the fluid displaced and can be calculated in Equation (6-3). The volume of a spherical bubble can be evaluated in Equation (6-4) and thus the buoyancy force is shown in Equation (6-5).

$$F_b = V_B(\rho_l - \rho_v)g \quad (6-3)$$

$$V_B = \frac{1}{6}\pi D_B^3 \quad (6-4)$$

$$F_b = \frac{1}{6}\pi D_B^3(\rho_l - \rho_v)g \quad (6-5)$$

6.4.3.2 Surface tension force

The surface tension is a negative force acting on the liquid-solid interface along the perimeter of the pore. The surface tension can be evaluated using Equation (6-6),

$$F_{st} = \pi D_p \sigma \sin \theta \quad (6-6)$$

where θ is the contact angle which is determined in Equation (6-1) previously.

6.4.3.3 Drag force:

As the bubble expands, the surrounding liquid will resist the motion. The drag force is a negative force and can be modelled as a spherical body moving in an inviscid medium, using Equation (6-7),

$$F_d = C_d \rho_l \pi R_B^2 \left(\frac{u_B^2}{2} \right) \quad (6-7)$$

where u_B is the bubble velocity, which can be related to the growth rate of bubble in Equation (6-8) as proposed by Zeng et al. [117] and illustrated in Figure 6-12.

$$u_B = 2 \frac{dR_B}{dt} \quad (6-8)$$

Figure 6-12 Schematic for growth rate of bubble

The drag coefficient (C_d) can be determined by the experimental correlation proposed by Keshock and Siegel [118] as

$$C_d = \frac{45}{\text{Re}} \quad (6-9)$$

where Re is the bubble Reynolds number and can be evaluated in Equations (6-10) and (6-11)

$$\text{Re} = \frac{2\rho_l R_B u_B}{\mu_l} \quad (6-10)$$

$$\text{Re} = \frac{4\rho_l R_B}{\mu_l} \left(\frac{dR_B}{dt} \right) \quad (6-11)$$

Combining Equations (6-7) to (6-11), the drag force can be determined using Equation (6-12).

$$F_d = \frac{45}{2} \mu_L \pi R_B \frac{dR_B}{dt} \quad (6-12)$$

In general, the growth rate of bubble ($\frac{dR_B}{dt}$) follows a power law as shown in Equation (6-13),

$$R_B(t) = Kt^m \quad (6-13)$$

where K and m are determined empirically as mentioned by Zeng et al. [117]. However, in this case, the bubble growth is assumed to be due to the vapour supplied from the vertical channel. Thus, knowledge of the vapour volumetric flow rate ($\dot{V}$) can be used to evaluate the bubble growth rate. For a single pore, the volumetric flow rate ($\dot{V}$) can be evaluated using Equation (6-14).

$$\dot{V} = \frac{q}{n\rho_v h_{fg}} \quad (6-14)$$

where q is the power supplied and n is the number of active sites, which can be obtained from experimental observations as mentioned previously in Section 6.3.3.

For a spherical bubble, the relationship between the radius and volume is

$$V_B = \frac{4}{3}\pi R_B^3 \quad (6-15)$$

and the rate of change of radius is

$$\frac{dR_B}{dt} = \frac{\dot{V}}{4\pi R_B^2} \quad (6-16)$$

Therefore, Equation (6-16) can be substituted into Equation (6-12) and the drag force can be evaluated using Equation (6-17).

$$F_d = \frac{45}{8}\mu_L \frac{\dot{V}}{R_B} \quad (6-17)$$

6.4.3.4 Contact pressure force

The contact pressure force is due to the pressure difference across the inside and outside surfaces of the bubble at the reference point over the contact area as

mentioned by Klausner et al. [119]. It is a positive force which pushes against the base of the pore and can be evaluated using Equation (6-18).

$$F_{cp} = \frac{\pi D_p^2}{4} \left(\frac{2\sigma}{R_c} \right) \quad (6-18)$$

where R_c is the radius of curvature at the base of bubble. For this case, the radius of curvature is related to the bubble diameter as

$$R_c = \frac{D_B}{2} \quad (6-19)$$

Hence, the contact pressure force can be evaluated in Equation (6-20).

$$F_{cp} = \pi D_p^2 \sigma D_B \quad (6-20)$$

6.4.3.5 Bubble inertia force

The bubble possesses an inertia force which is a negative force which resists motion. For a bubble growing at a pore surface, Voit et al. [120] proposed the inertia force to be

$$F_i = 1.3 \rho_v \left(\frac{\dot{V}}{D_B} \right)^2 \quad (6-21)$$

where the factor of 1.3 was determined from fitting of experimental data.

6.4.3.6 Computation and results

Iterations were carried out using the equations modelled in this chapter for the bubble departure diameters. The magnitudes of the various forces were evaluated at different assumed bubble diameters and the net force is plotted. An example of the relative magnitude of forces for a 1-mm pore diameter at heat flux of 6 W/cm² is shown in Figure 6-13. It can be observed that at a small bubble

diameter, the negative forces of surface tension and drag dominate and thus the bubble adheres to the pore. As the bubble grows, the buoyancy force increases, and together with the contact pressure force, they can overcome the negative forces. The inertia force is, however, rather negligible in this model. Overall, the net force can be seen to be increasing with expected bubble diameter. The bubble departure diameter is determined to be the value when the net force just reaches zero.

Figure 6-13 Relative magnitude of forces for 1-mm pore diameter at 6 W/cm²

The process of iteration is repeated for substrate of different pore sizes and at various heat fluxes to obtain the bubble departure diameters. The predicted bubble diameters, D_B , at various heat fluxes are shown in Figure 6-14. For a fixed substrate, D_B increases at higher heat flux. The larger pore diameter also corresponded to larger D_B .

Figure 6-14 Predicted bubble departure diameters at various heat fluxes

6.5 Comparison of model with experiment

The comparison of the bubble departure diameters predicted by the model and the experimental measurements is shown in Figure 6-15. The results show good agreement for the trends and range of bubble departure diameters. It can be observed that deviation of model from experiment is higher at the low heat flux range. However, at moderate heat fluxes, the model is able to predict the bubble diameter quite well.

Figure 6-15 Comparison of predicted and experimental values for bubble departure diameters

Figure 6-16 shows the relative deviation between the predicted and experimental values for bubble departure diameters. The relative deviation is calculated using Equation (6-22). The absolute deviation in predicting the bubble departure diameter is within a range of -6.8% to +24.3%. It can be observed that most of the data falls within $\pm 20\%$ of the prediction.

$$r.d. = \frac{1}{N} \sum_{k=1}^N \frac{|D_{B,pred,k} - D_{B,exp,k}|}{D_{B,exp,k}} \times 100 \quad (6-22)$$

Figure 6-16 Relative deviation of model

6.6 Summary

A semi-empirical model is developed to predict the bubble departure diameter from the re-entrant substrate. The model assumes a spherical bubble growing and departing from the pore surface of different diameters of 0.8, 1.0 and 1.3 mm. Experimental measurements were obtained with high-speed visualisations for comparison. The following conclusions can be drawn from this chapter:

1. For experimental measurements, the bubble departure diameter increases with heat flux and pore sizes. The bubble diameters also follow a normal distribution for each pore size at a given heat flux.
2. The model uses a semi-empirical approach to determine the forces acting on a spherical bubble at the pore surface. The bubble growth is modelled to be due to vapour supplied from the vertical channels. The dominant negative forces are surface tension and drag, while the positive forces are buoyancy and contact pressure. The bubble inertia force was found to be negligible.
3. Comparison of model and experimental measurements shows good agreement for the bubble departure diameter, with most of the values falling within $\pm 20\%$ of the prediction.

Chapter 7 - Flow Boiling Through Three-Dimensional Hollow Spherical Substrates

7.1 Introduction

In this chapter, flow boiling experiments were conducted using FC-72 as the fluid to investigate the heat transfer enhancement using specialized three-dimensional (3D) structures. The 3D substrates were fabricated using SLM and they possess repeating cells of hollow spherical features to promote nucleation and reduce coalescence of bubbles.

The flow boiling heat transfer process will be studied in a two-phase closed-loop rig. Effects of substrate type and fluid mass flux were investigated. High-speed visualisation was employed to understand the flow pattern evolution within the substrates.

7.2 Substrate preparation and characterisation

7.2.1 Description of 3D hollow spherical substrates

Porous substrates with engineered hollow structures were designed and fabricated in this study. Each substrate consists of repeating unit cells with an internal spherical void and six connecting cylindrical through-holes. Each unit cell is fixed as a cube of 5-mm length on all sides. The concept of this design is to increase the surface area and promote nucleation within the spherical void, whereas the cylindrical through-holes act as connecting channels for liquid and

vapour to move downstream. The constriction caused by the cylindrical through-holes is likely to reduce coalescence of the bubbles too.

A total of three substrates were designed in SolidWorks. Two of the substrates were populated with uniform unit cells but with different porous parameters and were labeled as Sphere-small and Sphere-large substrates. Sphere-small has a connecting hole of 2-mm diameter and inner spherical void of 2-mm radius, whereas Sphere-large has a connecting hole of 2.8-mm diameter and inner spherical void of 2.8-mm radius, as shown in Figure 7-1(a-b). A third substrate, labeled as Gradient, consists of unit cells with increasing spherical void radii varying from 2-mm to 2.8-mm in a length-wise manner. A length-wise cutout of the Gradient substrate is shown in Figure 7-1(c) for illustration purpose. For each substrate, the unit cells were arranged to form geometries of 80 mm (length) by 60 mm (width) by 10 mm (height), which correspond to a total of 16 by 12 by 2 unit cells. The fabricated substrates are shown in the next section. The parameters of the substrates are summarised in Table 7-1. The porosity is defined as the ratio of void to actual bulk volume occupied by the substrates.

(c) Length-wise cutout of gradient unit cells

Figure 7-1 Illustration of unit cells of different porous substrates

Table 7-1 Summary of substrate characteristics

S/N	Name	Cylinder diameter, a (mm)	Hollow sphere radius, b (mm)	Average porosity (%)
1	Sphere-small	2	2	36
2	Sphere-big	2.8	2.8	72
3	Gradient	2 – 2.8	2 – 2.8	54

7.2.2 Substrate characterisation

The substrates were fabricated using SLM, which was elaborated in Chapter 3. To briefly summarise the process, the CAD files were converted to STL format and imported into the “Magics” software to input the laser parameters for the SLM 250 HL machine. After fabrication, the substrates were vacuumed to remove internal residual powder and then separated from the baseplate using wire-cutting method.

The fabricated substrates are shown in Figure 7-2. The internal hollow features were observed to be formed in good order and accuracy. The bulk external dimensions were measured using a Vernier caliper and were found to be accurate within ± 0.1 mm. Further measurements were obtained using a light optical microscope (Olympus SZX7). The diameters of 20 random external cylindrical holes were measured for each Sphere-small and Sphere-large substrate. The average values when compared to the design showed a small deviation of ± 0.1 mm. Thus, it has been demonstrated that SLM is able to produce three-

dimensional substrates with complex and intricate features that is otherwise difficult to achieve using conventional subtractive methods.

Figure 7-2 Substrates fabricated using SLM for (a) Sphere-small, (b) Sphere-large, (c) Gradient and (d) Top views of all substrates

7.3 Flow boiling facility and data reduction

7.3.1 Experimental flow boiling facility

A two-phase closed-loop experimental flow boiling facility was developed to study the nucleate flow boiling performance of the substrates as shown in Figure 7-3. The facility consists of five main components, namely, the fluid reservoir, gear micro-pump, preheater section, evaporator test section and condenser. A funnel was located at the reservoir to introduce the fluid before the start of experiment. Two cartridge heaters were installed in the reservoir to warm up the fluid. A gear micro-pump (Cole-Parmer R-73011-18) was used to circulate the fluid in the system. It can be varied accurately between 0.2 to 5.0 L/min. A flow

meter (McMillan S-112) was installed after the micro-pump to measure the volumetric flow rate. A preheater section was installed before the evaporator to allow fine control of temperature of the fluid before it enters the test section. The test substrates were embedded inside the evaporator for heat transfer testing and a heat simulator was located under the evaporator to control the heat input. The details of evaporator would be elaborated in the next section. The condenser was a plate-type heat exchanger through which cold water of 20°C was circulated by a chiller to condense the liquid-vapour mixture before it was returned to the reservoir to complete the cycle. “Swagelok” stainless steel tubings with outer diameter of 3/8 inch (9.5 mm) were used to connect the different components and insulation was provided around the tubing to minimise heat losses.

The temperatures of the fluid were monitored using four type-K thermocouples at various locations. The first thermocouple was installed in the reservoir, and subsequent ones in the flow channel before the pre-heater section, and before and after the evaporator section, as shown in Figure 7-3. The pressures were monitored at various locations. A pressure gage was installed at the reservoir. In the flow channel, a pair of absolute pressure transducers (Cole-Parmer EW-68074-08) and a differential pressure sensor (Omega PX409-005DWUV) were installed before and after the evaporator test section.

Figure 7-3 Schematic of experimental flow boiling facility

7.3.2 Evaporator test section

The evaporator section consists of a rectangular flow channel with dimensions of 160 mm (length) by 60 mm (width) by 12 mm (height), as shown in Figure 7-4. An expanding and a contracting vane before and after the rectangular flow channel allow smooth convergence and divergence of the flow. The flow channel is made of an aluminium base and a polycarbonate top cover to allow observation of the boiling process during the experiments. The polycarbonate top cover is secured to the base and sides with 24 screws and gaskets and no leakage was observed during the experiments.

The heat simulator is attached below the rectangular flow channel. It contains a copper block to provide a heating surface of 80 mm by 60 mm with a thickness of 10 mm. Eight cartridge rod heaters were embedded in the copper block and a power transformer allowed the power to the rod heaters to be varied accurately up to 3 kW. The copper block is then encompassed by a 10-mm thick Teflon layer to provide insulation. A thin layer of thermal grease “Omega-therm-201”

was applied between the copper block and the aluminium base of rectangular channel to ensure good thermal contact.

Figure 7-4 Illustration of evaporator section

As mentioned previously, the test substrates were fabricated to the dimensions of 80 mm by 60 mm by 10 mm. During each experiment, a substrate was fitted securely in the evaporator section. A silicone rubber was embedded at the top of the substrate to minimise fluid bypass. A thin layer of thermal compound (Artic MX-4) was applied between the test substrate and aluminium base to reduce contact resistance. Surface temperatures of the substrate were measured using six type-K thermocouples which were inserted through drilled holes in the aluminium base. They were located 3 mm beneath the substrate surface and distributed evenly throughout the surface. The locations of the thermocouples are shown in Figure 7-5 which illustrates the detailed schematic of the evaporator with the test substrate embedded.

Figure 7-5 Detailed schematic of evaporator section and location of thermocouples

7.3.3 Properties of fluid – FC-72

FC-72 from 3M Ltd was selected as the phase-change fluid due to its large dielectric strength and electrical non-conductivity. It has a boiling point of 56°C under atmospheric condition. The other key properties can be found in Table 4-3 too.

7.3.4 Experimental procedure and data reduction

Before the commencement of each experiment, heaters in the reservoir were used to boil the fluid vigorously for 30 min and the degassing valve was opened to purge any non-condensable gas. After the degassing procedure, the micro-pump was switched on to allow the fluid to circulate through the entire facility. The pre-heater section was then tuned to heat the fluid to 56°C and careful

management was taken to avoid boiling before it enters the evaporator section. Only after the temperature of the fluid has stabilized was the power supplied to the heat simulator at the test section at increasing intervals of between 50 W and 150 W. For each interval, it takes about 15 to 20 min for the system to reach quasi-steady conditions, whereby the temperatures recorded do not fluctuate more than 0.2°C within a 5-min interval. The temperatures and pressures were recorded using a MX100 data acquisition system. For each set of readings at steady-state, time-averaging values were recorded within 1 min with sampling rate of 2 Hz and total of 120 readings to minimise data uncertainty.

The total electrical power input was calculated by the product of current (I) and voltage (V) provided to the heat simulator. The net power input to the fluid was calculated by the difference of electrical power input and heat loss, as shown in Equation (7-1).

$$q = IV - q_{loss} \quad (7-1)$$

In order to estimate the heat loss across the evaporator, a series of tests was conducted in the single-phase heat transfer regime. The net power input can be calculated by the sensible heat gain by the fluid using the temperature difference between the inlet (T_{in}) and outlet (T_{out}) of the evaporator section, as shown in Equation (7-2),

$$q = \dot{m}C_p(T_{out} - T_{in}) \quad (7-2)$$

where $\dot{m}$ denotes the mass flow rate and c_p denotes the specific heat of fluid.

Combining Equations (7-1) and (7-2), the rate of heat loss can be determined using Equation. (7-3),

$$q_{loss} = IV - \dot{m}C_p(T_{out} - T_{in}) \quad (7-3)$$

The single-phase tests showed that the heat loss was between 5% and 10% depending on the coolant flow rates. The mean values of the heat loss were then used for calculation of net heat gain by the substrates in the two-phase regime. The net heat flux input to the substrates is determined using Equation (7-4),

$$q'' = \frac{q}{A_b} \quad (7-4)$$

where A_b is the heated base area of 80 mm by 60 mm.

For each set of readings at steady-state, the temperature obtained at the aluminium base was calculated based on the average of the six thermocouple readings, as shown in Equation (7-5). Subsequently, the average wall temperature at the substrate was obtained based on one-dimensional heat conduction from the aluminium base as shown in Equation (7-6). The thermal conductivities of aluminium (k_{Al}) and the MX-4 thermal compound (k_{TC}) were taken to be 220 W/m·K and 8.5 W/m·K, respectively, while the thicknesses of the aluminium (L_{Al}) and MX-4 thermal compound (L_{TC}) used were 3.0 mm and 0.1 mm, respectively.

$$T_{ave} = \frac{1}{6} \sum_6^1 T_i \quad (7-5)$$

$$T_w = T_{ave} - q'' \left(\frac{L_{TC}}{k_{TC}} + \frac{L_{Al}}{k_{Al}} \right) \quad (7-6)$$

The heat transfer performance of each substrates was characterised by the average heat transfer coefficient which is shown in Equation (7-7).

$$h_{ave} = \frac{q''}{(T_w - T_{sat})} \quad (7-7)$$

During the boiling process, the pressure within the evaporator section tends to increase. The pressure was measured using the average of the readings from two absolute pressure transducers before and after the evaporator section. The saturated fluid temperature was then determined using Equation (7-8) as provided by the manufacturer to obtain a more accurate prediction of heat transfer coefficient.

$$T_{sat}(^{\circ}C) = \frac{1562}{9.729 - \log_{10}(P_{sat})} - 273 \quad (7-8)$$

For each experiment, the fluid mass flux was controlled by regulating the volumetric flow rate supplied by the micro-pump. The mass flux is shown in Equation (7-9),

$$G = \frac{\rho \dot{V}}{A_f} \quad (7-9)$$

where A_f is the substrate frontal cross-sectional area of 60 mm by 10 mm, ρ is the fluid density and $\dot{V}$ is the volumetric flow rate as measured by the flow sensor.

Finally, the local vapour quality can be calculated using energy balance in Equation (7-10).

$$x = \frac{1}{h_{fg}} \left[\frac{q}{\dot{m}} \frac{L_i}{L} - C_p(T_{sat} - T_{in}) \right] \quad (7-10)$$

where h_{fg} is the latent heat of vaporisation, L_i is the distance from the leading edge of substrate to the location of interest and L is the entire length of substrate of 80 mm.

The measurement uncertainty for the flow meter is 1% FS (0.02 L/min). The uncertainties of the absolute pressure transducer and differential pressure sensor are 0.13 % FS (22 Pa) and 0.08% FS (28 Pa), respectively. The uncertainties of

type-K thermocouples and power supply yields are ± 0.3 °C and 1% FS (10 W), respectively. The main uncertainties are contributed by the heat flux and thermocouple readings, while the uncertainties from the pressure sensors are negligible. Using the method described by Taylor [108], the uncertainty of heat transfer coefficient (h_{ave}) was determined from Equation (7-11). The uncertainty in h_{ave} was determined to be higher in the low heat flux region with a maximum of $\pm 15\%$ and decreased rapidly to $\pm 5\%$ at heat fluxes above 8 W/cm^2 .

$$\frac{\Delta h}{h} = \sqrt{\left(\frac{\Delta q''}{q''}\right)^2 + \left(\frac{\Delta T_w}{T_w - T_{sat}}\right)^2 + \left(\frac{\Delta T_{sat}}{T_w - T_{sat}}\right)^2} \quad (7-11)$$

7.3.5 Design of experiment

The empty channel, which was labelled as Plain surface, was used as a benchmark in the experiment. For the Gradient substrate, it can be mounted in the evaporator with the gradient structure varying from small-to-big (forward) or big-to-small (reverse) with respect to the flow direction. Thus, it has been labelled as Gradient-forward and Gradient-reverse substrates for simplicity. Together with the Sphere-small, Sphere-large and plain surface, a total five types of substrates have been tested. For each substrate, experiments for three mass fluxes of 20, 30 and $40 \text{ kg/m}^2\cdot\text{s}$ were conducted to study the effect of mass flux. The experiment for each condition was conducted twice to ensure repeatability. The summary of the design of experiment is shown in Table 7-2. The experiments were conducted for heat fluxes between 2.0 and 20.0 W/cm^2 .

Table 7-2 Summary of design of experiment

Parameters	Values
Substrate name	Plain surface
	Sphere-small
	Sphere-large
	Gradient-forward
	Gradient-reverse
Mass flux	20 kg/m ² ·s
	30 kg/m ² ·s
	40 kg/m ² ·s

The flow conditions can also be represented using the Reynolds number. The Reynolds number can be evaluated using Equation (7-12),

$$Re = \frac{GD}{\mu} \quad (7-12)$$

where D is defined as the diameter of the spherical hollow cell and μ is the liquid dynamic viscosity. For Sphere-small and Sphere-large substrates, Re varies from about 9 to 18 and 13 to 26, respectively for the various mass fluxes used in this study.

7.4 Results and discussion

7.4.1 Effect of mass flux

The effect of mass flux on the heat transfer coefficients for Sphere-small and Sphere-large substrates are shown in Figure 7-6. Both figures show a common trend of the heat transfer coefficient increasing with heat flux up to a peak value before decreasing. For both figures, it can be observed that at low heat fluxes, the curves for all the mass fluxes collapse to a single curve. This indicates that the heat transfer coefficient is independent of mass flux within this range and suggests that nucleate boiling is the dominant heat transfer mechanism. At higher heat fluxes, the heat transfer performance starts to diverge for different mass

fluxes, indicating the higher contribution of convective heat transfer mechanism. However, the transition appears to occur at different heat fluxes for the two substrates, which correspond to about 12 and 16 W/cm² for Sphere-small and Sphere-large, respectively.

(a) Sphere-small

(b) Sphere large

Figure 7-6 Effect of mass flux on heat transfer coefficients for (a) Sphere-small and (b) Sphere-large substrates.

7.4.2 Effect of outlet vapour quality

In Figure 7-7, the effect of outlet vapour quality on heat transfer coefficient is shown for Sphere-small and Sphere-large substrates. For both substrates, the heat

transfer coefficient curve is observed to be parabolic and experiences a peak value before declining at high outlet vapour quality. At low mass flux of 20 kg/m²·s, the outlet vapour quality almost approaches the dry-out vapour quality of $x = 1.0$. Typically, the heat transfer performance rapidly deteriorates near dry-out due to transition to misty flow. Comparing both substrates, the deterioration for Sphere-large substrate at high vapour quality is more pronounced as observed from the gradient of the curves. From both figures, the peak values of heat transfer coefficient occur in the mid-range of exit vapour qualities where $0.3 < x < 0.6$. Therefore, for practical consideration, the mass flux should match the heat flux level where the corresponding vapour quality would be in that range for optimal heat transfer performance. However, it is noteworthy that the optimal range varies for different substrates.

Figure 7-7 Effect of outlet vapour quality on heat transfer coefficients for (a) Sphere-small and (b) Sphere-large substrates.

7.4.3 Effect of substrate type

Figure 7-8 shows the effect of substrate type on the heat transfer performance at mass fluxes of 20 and 40 kg/m²·s. Compared to the benchmark Plain surface, all hollow porous substrates showed significant heat transfer enhancements. The enhancements could be attributed to the internal structure of the spherical hollow cells, which provides larger surface areas and more favourable nucleation sites as compared to the Plain surface. Also, as the liquid-vapour mixture flow within the spherical hollow cells, the alternate expanding and contracting cross-sectional is likely to intensify the flow mixing. The phenomenon of flow mixing will be discussed in the subsequent section on the visualisation study.

For both figures, the Gradient-reverse substrate demonstrates superior heat transfer coefficients as compared to other substrates across the entire heat flux range. Also, while the other substrates experience decreasing heat transfer coefficients at higher heat fluxes, the Gradient-reverse substrate shows a delay in

the deterioration to a higher range of heat flux. For example, in Figure 7-8(a) for mass flux of 20 kg/m²·s, the Gradient-reverse substrate only show deterioration in the heat transfer coefficient above 18 W/cm², whereas the other substrates tend to show deterioration from 12 W/cm² onwards.

The superior performance of the Gradient-reverse substrate could be attributed to its unique structure of reducing cross-sectional areas along the flow direction. This is likely to alter the flow pattern and sustain bubbly and churn flow patterns by channelling the liquid-vapour mixture into smaller chambers as it flows downstream. It would also sustain high nucleate boiling performance even at high heat flux and associated high vapour quality. The phenomenon is examined subsequently in the visualisation study in the next section.

Figure 7-8 Effect of substrate type at mass fluxes of (a) 20 kg/m²·s and (b) 40 kg/m²·s.

7.4.4 Pressure drop analysis

Figure 7-9(a) shows the pressure drop curves of the different substrates at adiabatic condition. The data were collected for each substrate prior to heat input at the evaporator section. The pressure drops were obtained at various mass fluxes for comparison. In general, the pressure drop increases at higher mass flux due to increasing single-phase frictional losses. Logically, the Plain surface shows the lowest pressure drop due to the lack of obstruction in the flow. For the other substrates, the pressure drops correspond well to the porosities of the substrates. The Sphere-small substrate has the highest pressure drop due to the smallest cross-sectional flow area. For both gradient substrates, they presented very similar pressure drops despite the direction of variation in the unit cell sizes.

Figure 7-9 shows the pressure drop at various heat fluxes during the actual experimental runs. As compared to Figure 7-9(a) at adiabatic condition, the magnitude of the two-phase pressure drop for each substrate is noted to be several

times higher. Under conditions of flow boiling, the formation of vapour causes the two-phase fluid to accelerate and contribute to the accelerative pressure drop. In Figure 7-9(b), the pressure drop generally increases at higher heat fluxes due to the corresponding higher outlet vapour quality. Among the structured substrates, although the Sphere-large and Sphere-small substrates retained the lowest and highest pressure drop, respectively, it is interesting to note that the pressure drops of the gradient substrates deviate from each other. The Gradient-reverse substrate consistently shows a higher pressure drop than that of the Gradient-forward substrate. Due to the gradually expanding channel size in Gradient-forward, it is likely to relieve the accelerative pressure drop contribution.

(b) flow boiling condition

Figure 7-9 Pressure drop characteristics at (a) adiabatic condition and (b) flow boiling condition

The higher heat transfer performance achieved by the porous substrates comes at the penalty of higher pressure drop, which translates to the need for higher pumping power. To relate the heat transfer enhancement to the increased pressure drop experienced by the various substrates as compared to the plain surface, the relative enhancement ratio (η) is proposed in Equation (7-13).

$$\eta = \frac{(h/h_{ref})}{(\Delta P/\Delta P_{ref})} \quad (7-13)$$

where the subscript *ref* refers to the reference values of h or ΔP at the same mass flux.

Figure 7-10 shows the relative enhancement ratio of each substrate for the various mass fluxes. In general, for individual substrate, the higher mass flux corresponds to a higher relative enhancement ratio, except for Sphere-large where the mid mass flux performed the best. If pumping power is an important concern, Sphere-large is favoured. Otherwise, Gradient-reverse showed the highest absolute heat

transfer coefficient as discussed in the previous section and has decently high relative enhancement ratio as shown in Figure 7-10.

Figure 7-10 Relative enhancement ratios of porous substrates at different mass fluxes

7.4.5 Comparison to literature

Due to the diverse types of fluid, mass fluxes and configurations of setup used by other researchers in flow boiling experiments, it is challenging to draw out meaningful comparisons. The author has selected works by Madani et al. [121] and Kim et al. [66] for comparisons due to the similarity in the range of heat and mass fluxes tested.

Madani et al. [121] soldered metallic copper foam of 36 PPI into a tube for their flow boiling investigation. The fluid used was *n*-pentane with a boiling point of 36°C which is lower than that of FC-72. The latent heat of vaporization of *n*-pentane is 357 kJ/kg, which is about 4 times of the value of 88 kJ/kg for FC-72 of. It is shown in Figure 7-11 that an increase of mass flux from 36 to 57 kg/m²·s causes a corresponding increase of *h*. However, at high heat fluxes, both the heat

transfer coefficients decrease. According to Madani et al., the decrease could be attributed to the coalescence of bubbles and the resistance of the metallic foam to bubble movement.

Kim et al. [66] investigated flow boiling of metallic foam using FC-72. In their case, subcooled boiling was performed. For the mass flux of $20 \text{ kg/m}^2\cdot\text{s}$, nucleate boiling was initiated at lower heat flux range. However, for the case of mass flux of $40 \text{ kg/m}^2\cdot\text{s}$, the heat transfer coefficients increase at increasing heat flux due to transition from subcooled to nucleate boiling. The peak heat transfer coefficient increased by about 10% by increasing mass flux from 20 to 48 $\text{kg/m}^2\cdot\text{s}$.

Figure 7-11 Comparison of results with literature

7.5 Flow pattern analysis

For this section, visualisations were conducted using a high-speed camera (Photron FASTCAM 1024 PCI) for the Sphere-large and Sphere-small substrates. To better observe the flow pattern within the substrates, the edge of each substrate was milled or about 1-mm depth to reveal more of the internal

features without major disruption of the flow structure. High-speed videos were obtained at different heat fluxes and two locations of each substrate, which corresponds to $L = 30$ mm and $L = 60$ mm downstream of the leading edge. For each video, capturing frequencies of between 2000 Hz and 6000 Hz and 600 images were obtained for analysis.

7.5.1 General flow patterns

The presence of spherical hollow unit cells used in this study caused the flow patterns to deviate from that of plain tubes. The general flow patterns of Sphere-large substrate at various heat fluxes are shown in Figure 7-12. Each figure focuses on the flow pattern within two adjacent unit cells. At low heat flux and correspondingly low vapour quality, bubbly flow was observed in Figure 7-12(a). As the heat flux and vapour quality increased, the bubbles coalesced and increased in size. The flow also became more turbulent and is classified as churn flow in Figure 7-12(b). Flow mixing continued to increase at increasing vapour quality and the flow was classified as wavy-churn in Figure 7-12(c). Finally, when vapour quality was sufficiently high, partial dry-out was observed and was characterised by lower bubble count in a dominantly vapour medium, as shown in Figure 7-12(d).

Figure 7-12 Bubble regime at different heat fluxes at mid-section for Sphere-large ($G = 40 \text{ kg/m}^2\cdot\text{s}$, $L = 60 \text{ mm}$)

The general flow patterns of the Sphere-small substrate are shown in Figure 7-13. At low vapour quality as in Figure 7-13(a), the flow pattern was visibly more vigorous than that of Sphere-large substrate at the same vapour quality and thus is classified as bubbly-churn flow. It is likely that the smaller cross-sectional area in the Sphere-small substrate caused the bubbles to be more confined and resulted in more fluid mixing. Subsequently, the flow intensified at higher vapour qualities and evolved to churn and wavy-churn as shown in Figure 7-13(b) and (c), respectively. It is interesting to note that the relative bubble sizes were smaller than that of Sphere-large substrates, although the flow patterns were similar. At high vapour quality, however, minimal dry-out was observed and the wavy-churn flow pattern was sustained as shown in Figure 7-13(d). However, the bubble sizes were observed to increase generally at higher vapour quality due to coalescence.

Figure 7-13 Bubble regimes for different heat fluxes at mid-section for Sphere-large ($G = 40 \text{ kg/m}^2\cdot\text{s}$, $L = 60 \text{ mm}$)

For both the Sphere-small and Sphere-large substrates, no annular flow was observed even at high vapour qualities, as opposed to observations in the plain tubes. The reason for the lack of annular flow pattern is likely attributed to the intense flow mixing caused by the hollow spherical geometry, which limited the formation of an orderly liquid film at the substrate walls as is often observed in annular flow for plain tubes. To better understand the different flow pattern evolution, more analyses are provided in the subsequent section.

7.5.2 Transient behaviour of flow patterns at different vapour qualities

Further analyses were conducted on the various flow patterns which occurred in the Sphere-small and Sphere-large substrates. Figure 7-14 shows the transient behaviour of the flow pattern in the Sphere-small substrate at low heat flux and vapour quality, which corresponds to bubbly-churn flow. The images were taken at 5-ms intervals. The figure on the left shows the actual flow pattern, while the corresponding figure on the right shows the schematic for better clarity. For all images, the flow direction is from the left to the right.

At the beginning of the flow in Figure 7-14(a), a plug bubble was observed to emerge from the left and to move downstream to the right. As the plug bubble approached the constriction point connecting two adjacent spherical unit cells, it underwent elongation and broke up into two smaller bubbles in Figure 7-14(b). Upon emerging from the constriction point, the frontal bubble began to reform into a more spherical geometry in Figure 7-14(c). The next wave of bubbles underwent similar process of elongation, breaking up and squeezing through constriction points in Figure 7-14(d). The presence of cylindrical connecting channels as constriction points between the spherical hollow cells helped to reduce coalescence of bubbles in the Sphere-small substrate.

a) 0 ms			Initial plug emerging from left side.
b) 5 ms			Plug bubble breaking up into two smaller bubbles. Frontal bubble squeezing through constriction.
c) 10 ms			Frontal bubble began to reform into a more spherical geometry
d) 15 ms			Second wave of bubbles separating when nearing constriction.

Figure 7-14 Transient behaviour of bubbly-churn flow pattern in Sphere-small substrate ($q'' = 8 \text{ W/cm}^2$, $G = 30 \text{ kg/m}^2\cdot\text{s}$, $L = 30 \text{ mm}$)

Figure 7-15 shows the transient behaviour of the flow pattern in the Sphere-large substrate at low heat flux and vapour quality. In Figure 7-15(a), a large bubble was observed to approach the constriction point. In Figure 7-15(b), the frontal portion of the bubble managed to squeeze through the constriction point and

necking occurred which resulted in break-out into two bubbles. However, as the cross-sectional area of the constriction is much larger than that of the Sphere-small substrate, the bubbles did not elongate as much to squeeze through. In fact, it appears that there was mass transfer from the preceding bubble to the frontal bubble in Figure 7-15(c). Finally, in Figure 7-15(d), the bubbles coalesced and re-form into a larger bubble after passing through the constriction. Therefore, the larger spherical hollow void and wider constriction point allowed the bubbles to coalesce more easily in the Sphere-large substrate.

a) 0 ms			Large bubble approaching constriction.
b) 5 ms			Necking occurred and partial break-out into two bubbles when squeezing through constriction.
c) 10 ms			Transfer of mass towards downstream bubble.
d) 15 ms			Coalescence of two bubbles back into original large bubble after passing through constriction.

Figure 7-15 Transient behaviour of bubbly flow pattern in Sphere-large

substrate ($q'' = 8 \text{ W/cm}^2$, $G = 30 \text{ kg/m}^2\cdot\text{s}$, $L = 30 \text{ mm}$)

Figure 7-16 shows the transient behaviour of flow pattern in the Sphere-small and Sphere-large substrates in the mid heat flux and vapour quality level. Churn flow was present for both substrates, which was characterised by the formation of rapidly coalescing bubbles and chaotic movements of bubbles. In this flow pattern, the heat transfer process is highly nucleate boiling dominated and

corresponds to the high heat transfer coefficients as discussed in Section 7.4.2. It is apparent that the relative sizes of bubbles for Sphere-large are larger than that of Sphere-small, which suggests that the geometry of the constriction channel plays an important role on the coalescence of bubbles in this regime.

Figure 7-16 Churn flow inside Sphere-small and Sphere-large substrates ($q'' = 8$ W/cm², $G = 30$ kg/m²·s, $L = 30$ mm)

Figure 7-17 shows the transient flow patterns of Sphere-large and Sphere-small substrates at high vapour quality. For both substrates, some dry-out was observed but the dry-out was more prevalent for Sphere-large substrate. Although the flow was likely to transit to convective heat transfer mechanism at high vapour quality, it appears that the contribution of nucleate boiling mechanism was crucial throughout the entire range of vapour quality due to the lack of annular flow formation. Therefore, the dry-out flow condition is not favourable for the current

study, which corresponds to the rapidly declining heat transfer coefficients of Sphere-large substrates as discussed in Section 7.4.2.

Figure 7-17 Different flow patterns inside Sphere-small and Sphere-large substrates ($q'' = 16 \text{ W/cm}^2$, $G = 40 \text{ kg/m}^2\cdot\text{s}$, $L = 60 \text{ mm}$)

From the analyses of the different flow patterns, it is apparent that the smaller cross-sectional area is more favourable for maintaining high nucleate boiling heat transfer mechanism due to less bubble coalescence and delaying of dry-out condition. Although not shown in the current visualisation study, it is likely that the Gradient-reverse substrate showed superior heat transfer performance due to the ability to break up the bubbles as they pass through constriction channels downstream and maintain churn-wavy flow even at high vapour quality.

7.5.3 Effect of mass fluxes

The flow patterns of both the Sphere-small and Sphere-large substrates at mid heat flux of 8 W/cm^2 and various mass fluxes are shown in Figure 7-18. For both

the substrates, it can be observed that bubbly flow pattern with some stratification was dominant at a low mass flux of $20 \text{ kg/m}^2\cdot\text{s}$. Stratification also tends to occur in plain tube at low mass flux when the flow is not turbulent enough to cause much mixing. Generally, stratification should be avoided due to lower flow mixing and the inability to utilize the entire substrate for heat transfer. At higher mass fluxes, churn and churn-wavy flows become dominant for both the substrates.

Figure 7-18 Flow patterns of Sphere-large and Sphere-small substrates at various mass fluxes ($q'' = 8 \text{ W/cm}^2$, $L = 30 \text{ mm}$)

7.6 Summary

In this study, flow boiling heat transfer experiments were conducted using FC-72 as the working fluid in a closed-loop facility. Specialised porous substrates were fabricated using SLM and a plain surface was used as benchmark. Visualisations were obtained using a high-speed camera to observe the evolution of flow patterns within the substrates. The following conclusions are derived from this study:

1. All porous substrates showed enhanced flow boiling heat transfer performance as compared to the plain surface. The spherical hollow voids

- provided favourable nucleation sites and flow mixing is enhanced in the substrates due to the alternating constriction channels and spherical voids.
2. For the fixed substrate type, heat transfer coefficient increases as the exit vapour quality increases and peaks at $0.3 < x < 0.6$. Further increase in heat flux and correspondingly exit vapour quality caused the heat transfer coefficient to deteriorate, especially when approaching dry-out condition.
 3. Visualisations showed that bubbly and churn flow dominated across a wide range of heat flux and outlet vapour quality. Annular flow was not observed even at high vapour quality due to the intense flow mixing which prevented the formation of orderly liquid film at the surface. Nucleate boiling is likely to be the dominant heat transfer mechanism as opposed to the convective mechanism.
 4. The Gradient-reverse substrate showed superior heat transfer performance over the entire range of heat flux tested as compared to all other porous substrates. The ability to channel the two-phase mixture into decreasing cross-sectional area as flow moves downstream in the substrate is likely to maintain a churn flow and delay dry-out condition, which helped to maintain high nucleate boiling heat transfer.

Chapter 8 - Conclusions and Future Work

In this thesis, pool and flow boiling experiments were conducted using 3D porous structures fabricated by the Selective Laser Melting technique. A thermosyphon and a two-phase flow facility were used for experimental testing of pool and flow boiling performance, respectively, with the fluid FC-72 used under saturated conditions. The conclusions are as follows:

1. It has been demonstrated that novel 3D substrates could be designed and fabricated using the Selective Laser Melting technique for nucleate boiling applications. These substrates possess intricate and complex internal features and were built to good dimensional accuracy. It would be otherwise difficult to manufacture these 3D substrates using conventional techniques.
2. Pool boiling experiments were conducted for substrates with octet-truss geometry for different substrate heights and unit cell sizes. Nucleate pool boiling heat transfer enhancement was obtained and it was attributed to the increased surface area and capillary-assisted liquid replenishment. However, at high heat fluxes, vapour evacuation was impeded by small unit cell size and large substrate height. Octet-3C-5H showed superior performance with a good balance of high surface area and sufficiently low vapour evacuation resistance. It has an average nucleate boiling heat transfer coefficient of $1.35 \text{ W/cm}^2\cdot\text{K}$, which is 2.81 times that of the plain surface.
3. For the pool boiling experiments of re-entrant substrates, different channel widths and configurations were investigated. For a fixed channel width, the dense configuration performed the best due to the highest internal surface for

nucleation. For a fixed configuration, the substrate with smaller channel width performed better at low heat flux due to higher internal surface area. However, at high heat fluxes, the vapour evacuation resistance becomes dominant and the phenomenon of spillage of bubbles at the base occur, which led to liquid-vapour counter-flow. Thus, Vert-1.0C-dense showed the optimal performance overall due to its high surface area and ability to delay liquid-vapour counter-flow at high heat fluxes.

4. A semi-empirical model of the bubble departure diameter was developed for the re-entrant substrates in the isolated bubble regime. A force analysis was applied to the bubble growing at a pore at the top of substrate. The dominant forces were buoyancy, surface tension, contact pressure and drag. Inertia force was found to be negligible. The bubble departure diameter was determined when the net force on the bubble just become positive. The bubble departure diameters were predicted for substrates of different pore diameter and heat flux. Comparison with high speed visualisations of experimental study showed good agreement with most of the values falling within $\pm 20\%$ of the prediction.
5. Flow boiling investigation was conducted for substrates with hollow spherical features. Heat transfer enhancements were achieved as the spherical hollow voids provided favourable nucleation sites and flow mixing was enhanced in the substrates due to the alternating constriction channels and spherical voids. Visualisations showed that bubbly and churn flow dominated across a wide range of heat flux and outlet vapour quality. Annular flow was not observed even at high vapour quality due to the intense flow mixing which prevented the formation of orderly liquid film at the surface. The

Gradient-reverse substrate showed superior heat transfer performance due to its ability to maintain churn flow and delay the dry-out condition.

8.1 Recommendations for future work

The recommendations for future work are outlined as follows:

1. The use of porous foams for pool boiling has demonstrated the ability to enhance heat transfer coefficient and delay CHF. A small unit cell size is favourable to increase internal surface area but possess high vapour evacuation resistance due to the small pore diameters. Using additive manufacturing, further work can be done to fabricate porous structures with a gradient unit cell design. Such design can have a smaller unit cell at the base and incremental larger unit cell as the structure increases in height. It is likely to maintain high internal surface area and relieve the vapour evacuation resistance simultaneously with such a design concept.
2. The design of re-entrant substrates shows potential to reduce the effects of liquid-vapour counter-flow. For the current design, the dimension of the horizontal entry channel at the base were made to match that of the vertical channel. However, for small channel widths, spillage of bubbles at the base is prominent at high heat fluxes. The problem can be mitigated by enlarging the horizontal entry channels at the base to facilitate liquid intake.
3. The bubble dynamics of the re-entrant substrates were studied for the isolated bubble regime. As the heat flux increases, coalescence is prevalent at the top surface of the substrates and the phenomenon persists for a large portion of the nucleate boiling process. The coalescence

phenomenon can be studied and modelled to improve the understanding of boiling mechanism, leading to better possible designs.

4. The heat transfer mechanisms can be investigated when the porous structures have sub-micron pore sizes. The surface area and capillary suction would be greatly increased; however, the fluid and bubble resistance would be significantly higher too.
5. The influence of different working fluids can be investigated as the current fluid used, FC-72, has a low thermal conductivity and latent heat of vaporisation. The use of non-dielectric fluid could be considered if insulation can be achieved between the fluid and electronic chips, such as using a thin gold layer.

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